CODES AND STANDARDS ENHANCEMENT INITIATIVE (CASE)

Working Draft Measure Information Template – Refrigerated Warehouse

2013 California Building Energy Efficiency Standards

California Utilities Statewide Codes and Standards Team, March 2011

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1. Purpose

This document is a report of proposed changes to the Mandatory Requirements for Refrigerated Warehouses, Section 126 of the 2008 California Building Energy Efficiency Standards ("the 2008 Standards"). Refrigerated warehouses are extremely energy intensive and are fertile ground for additional energy savings and demand reductions.

Systems used to condition refrigerated warehouses are specialized equipment and are quite different from equipment used to condition spaces intended for human occupancy. Refrigerated facility indoor design conditions can range from -40°F freezers to moderate +50°F temperature coolers; outside air ventilation is low or non-existent. Refrigeration systems in large warehouses typically use ammonia rather than more conventional halocarbon refrigerants, and evaporators (essentially fan coils) are suspended or otherwise mounted in the cooler or freezer and coupled to multiple compressors and condensers. Systems for refrigerated warehouses are typically custom designs rather than packaged. Product freezing and cooling processes with high load intensity often share the same refrigeration plant as the refrigerated warehouse spaces. These process spaces, as well as the associated refrigeration plant and various types of food processing equipment that may be coupled with refrigeration systems serving refrigerated warehouses, may be exempt from the 2008 Standards.

A number of questions arose during utility-sponsored educational efforts regarding the 2008 Standards. As part of this 2013 Codes and Standards Enhancement (CASE) initiative, we looked at ways to simplify the 2008 Standards and better align with changes in industry practice. The following measures from the 2008 Standards were analyzed:

- freezer floor and roof insulation
- screw compressor part load performance
- allowing air-cooled condensers for ammonia systems
- removing the evaporator fan control exception for single compressor systems

We also conducted analysis on new measures for the 2013 Standards:

- condenser specific efficiency
- infiltration barriers
- acceptance tests

The analysis included research of refrigerated warehouse energy efficiency, data collection from the Savings By Design (SBD) utility new construction program and equipment manufacturers, interviews with contractors and designers, detailed energy modeling, and economic analysis. Based on the results of these activities, we propose a set of changes to the 2008 Standards for the 2013 Standards.

2. Overview

a. Measure Title	Refrigerated Warehouses
b. Description	The proposed changes to Title 24 apply to Section 126 – Mandatory Requirements for Refrigerated Warehouses. The proposed changes are as follows: ▪ Condenser Specific Efficiency – impose a maximum fan power per unit of capacity on refrigerant condensers utilized on refrigerated warehouse facilities. ▪ Infiltration Barriers – require devices at door openings to refrigerated spaces to minimize air infiltration, resulting in a reduction in refrigeration system load. ▪ Freezer Roof Insulation – require a higher minimum R-value for freezer ceilings than 2008 requirements, which are below industry-standard practice and ASHRAE recommendations. ▪ Freezer Floor Insulation - reduce the current minimum R-value requirement for freezer floors. Industry reports indicate that the current requirement is higher than necessary, and is not easily constructed with current floor insulation thickness increments available on the market. ▪ Screw Compressor Part-Load Performance – simplify the current single-point, part-load performance exception to the variable-speed, capacity-control requirement by implementing an application-based requirement for variable speed capacity control. ▪ Evaporator Fan Control for Single Compressor Systems – eliminate the current evaporator fan speed control exception for evaporators served by a single compressor without variable capacity capability. The new measure requires controls to reduce fan speed or stage fans off when the compressor is not operating. ▪ Allow Air-Cooled Ammonia Condensers – remove the current requirement to use only evaporative-cooled condensers with ammonia systems, and concurrently establish a minimum specific efficiency mandate for air-cooled ammonia condensers. ▪ Acceptance Test – require performance of an acceptance test to ensure that refrigerated warehouse control measures comply with Section 126 of Title 24.
c. Type of Change	The proposed changes to the code constitute mandatory code requirements.
d. Energy Benefits	Values in the summary table below are weighted for different refrigerated warehouse building prototypes. The measures presented below are the measures that have energy savings reported in 2013 statewide energy savings. The measures not included below, i.e., acceptance tests, floor insulation, infiltration barriers, air-cooled ammonia systems and compressor VFD measures, do not have 2013 statewide energy savings benefits. Analysis on incremental savings for these measures is presented in Section 4. For a description of prototype buildings and weighting, refer to Section 3 and Section 4 below.

CTZ03 Oakland	Energy Savings (kWh)	Energy Savings (kWh/ft^2)	Demand Savings (kW)	Demand Savings (W/ft^2)	TDV Savings (MMbtu)	TDV Savings (kBTU/ft^2)
Condenser Specific Efficiency – Outdoor Evaporative-Cooled	8,305	0.09	4.3	0.047	213	2.3
Condenser Specific Efficiency – Outdoor Air-Cooled	1,556	0.06	5.2	0.201	73	2.8
Condenser Specific Efficiency – Indoor Evaporative-Cooled	356	0.01	0.1	0.004	9	0.33
Freezer Roof Insulation	4,367	0.12	0.7	0.019	146	3.9
Evaporator Fan Control for Single Cycling-Compressor Systems	168,907	6.50	7.0	0.270	3,424	132
CTZ05 Santa Maria	Energy Savings (kWh)	Energy Savings (kWh/ft^2)	Demand Savings (kW)	Demand Savings (W/ft^2)	TDV Savings (MMbtu)	TDV Savings (kBTU/ft^2)
Condenser Specific Efficiency – Outdoor Evaporative-Cooled	7,909	0.09	3.3	0.035	197	2.1
Condenser Specific Efficiency – Outdoor Air-Cooled	1,443	0.06	5.2	0.201	49	1.9
Condenser Specific Efficiency – Indoor Evaporative-Cooled	350	0.01	0.1	0.003	8	0.32
Freezer Roof Insulation	3,854	0.11	1.7	0.047	130	3.5
Evaporator Fan Control for Single Cycling-Compressor Systems	168,102	6.5	6.6	0.255	3,419	132
CTZ07 San Diego	Energy Savings (kWh)	Energy Savings (kWh/ft^2)	Demand Savings (kW)	Demand Savings (W/ft^2)	TDV Savings (MMbtu)	TDV Savings (kBTU/ft^2)
Condenser Specific Efficiency – Outdoor Evaporative-Cooled	8,548	0.09	3.4	0.037	209	2.3
Condenser Specific Efficiency – Outdoor Air-Cooled	1,779	0.07	5.2	0.201	64	2.5
Condenser Specific Efficiency – Indoor Evaporative-Cooled	387	0.01	0.1	0.003	9	0.35
Freezer Roof Insulation	5,338	0.15	0.7	0.022	138	3.8
Evaporator Fan Control for Single Cycling-Compressor Systems	171,977	6.6	3.0	0.114	3,536	136
CTZ10 Riverside	Energy Savings (kWh)	Energy Savings (kWh/ft^2)	Demand Savings (kW)	Demand Savings (W/ft^2)	TDV Savings (MMbtu)	TDV Savings (kBTU/ft^2)
Condenser Specific Efficiency – Outdoor Evaporative-Cooled	8,850	0.10	5.9	0.064	232	2.5
Condenser Specific Efficiency – Outdoor Air-Cooled	5,594	0.22	6.8	0.263	269	10.4
Condenser Specific Efficiency – Indoor Evaporative-Cooled	378	0.01	0.1	0.004	9	0.35
Freezer Roof Insulation	5,819	0.16	0.9	0.027	170	4.6
Evaporator Fan Control for Single Cycling-Compressor Systems	172,892	6.6	12	0.467	3,494	134
CTZ 12 Sacramento	Energy Savings (kWh)	Energy Savings (kWh/ft^2)	Demand Savings (kW)	Demand Savings (W/ft^2)	TDV Savings (MMbtu)	TDV Savings (kBTU/ft^2)
Condenser Specific Efficiency – Outdoor Evaporative-Cooled	9,337	0.10	6.3	0.068	235	2.6
Condenser Specific Efficiency – Outdoor Air-Cooled	4,658	0.18	6.8	0.261	246	9.4
Condenser Specific Efficiency – Indoor Evaporative-Cooled	338	0.01	0.1	0.004	8	0.31
Freezer Roof Insulation	5,654	0.16	1.1	0.036	160	4.4
Evaporator Fan Control for Single Cycling-Compressor Systems	168,098	6.5	3.7	0.143	3,446	133

CTZ13 Fresno	Energy Savings (kWh)	Energy Savings (kWh/ft^2)	Demand Savings (kW)	Demand Savings (W/ft^2)	TDV Savings (MMbtu)	TDV Savings (kBTU/ft^2)
Condenser Specific Efficiency – Outdoor Evaporative-Cooled	9,612	0.10	4.9	0.053	245	2.7
Condenser Specific Efficiency – Outdoor Air-Cooled	7,680	0.30	6.8	0.261	316	12.1
Condenser Specific Efficiency – Indoor Evaporative-Cooled	344	0.01	0.1	0.004	8	0.3
Freezer Roof Insulation	6,772	0.18	1.0	0.032	177	4.8
Evaporator Fan Control for Single Cycling-Compressor Systems	169,862	6.5	4.0	0.153	3,468	133
CTZ14 Palmdale	Energy Savings (kWh)	Energy Savings (kWh/ft^2)	Demand Savings (kW)	Demand Savings (W/ft^2)	TDV Savings (MMbtu)	TDV Savings (kBTU/ft^2)
Condenser Specific Efficiency – Outdoor Evaporative-Cooled	9,441	0.10	7.3	0.079	273	3.0
Condenser Specific Efficiency – Outdoor Air-Cooled	7,915	0.30	6.8	0.263	322	12.4
Condenser Specific Efficiency – Indoor Evaporative-Cooled	350	0.01	0.1	0.004	9	0.3
Freezer Roof Insulation	5,114	0.14	1.2	0.036	172	4.7
Evaporator Fan Control for Single Cycling-Compressor Systems	170,423	6.6	8.2	0.317	3,430	132

e. Non-Energy Benefits	The change to allow ammonia rather than a Hydro-Fluoro-Carbon (HFC) refrigerant to be used in conjunction with air-cooled condensers would result in various benefits in addition to energy savings, including lower capital costs (for large systems), lower maintenance costs, reduced HFC emissions and longer system life.
f. Environmental Impact	The proposed refrigerated warehouse measures will have relatively small statewide changes in materials, water consumption and water quality. The condenser specific efficiency measure may be achieved in some instances with larger condenser surface, in other instances with more efficient motors or improved technology. In the case of halocarbon condensers, a rapidly emerging technology (micro-channel condenser surface) provides higher specific efficiency while potentially reducing material costs, weight and refrigerant charge. The changes to roof insulation and floor insulation could slightly increase roof insulation, although industry practice is generally already higher than the 2008 code requirement, as reflected during stakeholder meetings. The small adjustment in floor insulation to allow consistency with available size increments will reduce freezer floor insulation material; the net change of roof insulation material and floor insulation material is expected to be small. The change to allow the use of ammonia air-cooled condensers will likely result in a small number of systems being designed with ammonia that otherwise would have required HFC refrigerants. In these cases, eliminating the HFC charge and leakage will reduce the greenhouse gas impact of direct HFC emissions. The CASE study also determined that unlike evaporative-cooled ammonia systems, air-cooled ammonia systems eliminate water consumption and chemical use in addition to saving energy.

g. Technology Measures	**Measure Availability:** ▪ Condenser Specific Efficiency: establishing a minimum mandated efficiency will eliminate the lowest-performing models from the market. The proposed requirements can easily be met by larger condensers, normally manufactured with single circuits; however, there are few small, multi-circuit, evaporative-cooled halocarbon condensers that meet the proposed efficiency levels. A size threshold was established to allow the use of outdoor evaporative-cooled, forced-draft, centrifugal condensers which are common in small sizes and multi-circuit designs. Along with the sizing requirement for condensers in the existing Section 126 code, this measure provides additional encouragement for ASHRAE, AHRI and/or CTI to improve condenser test and rating standards and to certify condenser ratings. More information is available in Appendix D. ▪ Evaporator Fan Control for Single Compressor Systems: manufacturers of low- and medium-profile evaporator coils are responding to market demand for fan speed control. Two manufacturers already offer two-speed fan control technology as an option (one offers the technology for no additional cost), while a third offers fan-staging control as an option. Aftermarket two-speed fan controllers are also available. More information is available in Appendix D. **Useful Life, Persistence, and Maintenance:** The effective useful life (EUL) of a condenser is not affected by specific efficiency and would be 15 years, the same as other refrigeration and HVAC equipment. There is no persistence issue with condenser specific efficiency; the savings remain through the EUL. The EUL of the evaporator fan control is through the life of the evaporator, also 15 years. Persistence of savings can be as little as a few years as operators may disable controls or change settings. Persistence can be improved by initial commissioning and through maintenance and/or periodic re-commissioning.
h. Performance Verification of the Proposed Measure	Commissioning and acceptance testing of refrigeration plant control systems, field verification of minimum equipment requirements, and factory verification of condenser performance are performance verification options applicable to this effort. A mandatory acceptance test is a proposed measure in this report. The acceptance test is to verify operation of condenser fan controls, screw compressor VFD controls, evaporator fan controls and under-floor electric-resistance heating system controls.

i. Cost Effectiveness

The following table summarizes the cost-effectiveness of the measures proposed in this report. The Energy Commission Life Cycle Costing Methodology posted on the 2013 Standards website was used to evaluate the cost-effectiveness of each measure. Insulation measures utilized 30-year Time Dependent Valuation (TDV) multipliers and all other measures utilized 15-year multipliers. Cost to maintain measure performance over the EUL was included in the evaporator fan control measure cost.

CTZ03 Oakland	Measure Life (Years)	Measure Cost ($)	Measure Cost ($/ft^2)	Maintenance Cost ($)	Maintenance Cost ($/ft^2)	TDV Cost Savings ($)	TDV Cost Savings ($/ft^2)	LCC ($)	LCC ($/ft^2)
Condenser Specific Efficiency – Outdoor Evaporative-Cooled	15	$5,812	$0.06	$0	$0	$18,912	$0.21	($13,100)	($0.14)
Condenser Specific Efficiency – Outdoor Air-Cooled	15	$3,735	$0.14	$0	$0	$6,461	$0.25	($2,726)	($0.10)
Condenser Specific Efficiency – Indoor Evaporative-Cooled	15	$279	$0.01	$0	$0	$765	$0.03	($486)	($0.02)
Freezer Roof Insulation	30	$10,764	$0.29	$0	$0	$22,544	$0.61	($11,779)	($0.29)
Evaporator Fan Control for Single Cycling-Compressor Systems	15	$30,016	$1.15	$11,251	$0.43	$304,716	$12	($263,449)	($10)
CTZ05 Santa Maria	Measure Life (Years)	Measure Cost ($)	Measure Cost ($/ft^2)	Maintenance Cost ($)	Maintenance Cost ($/ft^2)	TDV Cost Savings ($)	TDV Cost Savings ($/ft^2)	LCC ($)	LCC ($/ft^2)
Condenser Specific Efficiency – Outdoor Evaporative-Cooled	15	$5,812	$0.06	$0	$0	$17,523	$0.19	($11,712)	($0.13)
Condenser Specific Efficiency – Outdoor Air-Cooled	15	$3,735	$0.14	$0	$0	$4,396	$0.17	($661)	($0.03)
Condenser Specific Efficiency – Indoor Evaporative-Cooled	15	$279	$0.01	$0	$0	$748	$0.03	($468)	($0.02)
Freezer Roof Insulation	30	$10,764	$0.29	$0	$0	$19,947	$0.54	($9,183)	($0.23)
Evaporator Fan Control for Single Cycling-Compressor Systems	15	$30,016	$1.15	$11,251	$0.43	$304,303	$12	($263,037)	($11)
CTZ07 San Diego	Measure Life (Years)	Measure Cost ($)	Measure Cost ($/ft^2)	Maintenance Cost ($)	Maintenance Cost ($/ft^2)	TDV Cost Savings ($)	TDV Cost Savings ($/ft^2)	LCC ($)	LCC ($/ft^2)
Condenser Specific Efficiency – Outdoor Evaporative-Cooled	15	$5,812	$0.06	$0	$0	$18,556	$0.20	($12,744)	($0.14)
Condenser Specific Efficiency – Outdoor Air-Cooled	15	$3,735	$0.14	$0	$0	$5,669	$0.22	($1,934)	($0.07)
Condenser Specific Efficiency – Indoor Evaporative-Cooled	15	$279	$0.01	$0	$0	$810	$0.03	($531)	($0.02)
Freezer Roof Insulation	30	$10,764	$0.29	$0	$0	$21,269	$0.58	($10,505)	($0.26)
Evaporator Fan Control for Single Cycling-Compressor Systems	15	$30,016	$1.15	$11,251	$0.43	$314,723	$12	($273,456)	($11)
CTZ10 Riverside	Measure Life (Years)	Measure Cost ($)	Measure Cost ($/ft^2)	Maintenance Cost ($)	Maintenance Cost ($/ft^2)	TDV Cost Savings ($)	TDV Cost Savings ($/ft^2)	LCC ($)	LCC ($/ft^2)
Condenser Specific Efficiency – Outdoor Evaporative-Cooled	15	$6,178	$0.07	$0	$0	$20,612	$0.22	($14,434)	($0.16)
Condenser Specific Efficiency – Outdoor Air-Cooled	15	$5,291	$0.20	$0	$0	$23,967	$0.92	($18,676)	($0.72)
Condenser Specific Efficiency – Indoor Evaporative-Cooled	15	$281	$0.01	$0	$0	$801	$0.03	($520)	($0.02)
Freezer Roof Insulation	30	$10,764	$0.29	$0	$0	$26,238	$0.71	($15,473)	($0.39)
Evaporator Fan Control for Single Cycling-Compressor Systems	15	$30,016	$1.15	$11,251	$0.43	$310,973	$12	($269,706)	($11)
CTZ 12 Sacramento	Measure Life (Years)	Measure Cost ($)	Measure Cost ($/ft^2)	Maintenance Cost ($)	Maintenance Cost ($/ft^2)	TDV Cost Savings ($)	TDV Cost Savings ($/ft^2)	LCC ($)	LCC ($/ft^2)
Condenser Specific Efficiency – Outdoor Evaporative-Cooled	15	$6,358	$0.07	$0	$0	$20,932	$0.23	($14,574)	($0.16)
Condenser Specific Efficiency – Outdoor Air-Cooled	15	$5,380	$0.21	$0	$0	$21,867	$0.84	($16,487)	($0.63)
Condenser Specific Efficiency – Indoor Evaporative-Cooled	15	$288	$0.01	$0	$0	$721	$0.03	($433)	($0.02)
Freezer Roof Insulation	30	$10,764	$0.29	$0	$0	$24,619	$0.68	($13,854)	($0.35)
Evaporator Fan Control for Single Cycling-Compressor Systems	15	$30,016	$1.15	$11,251	$0.43	$306,675	$12	($265,408)	($10)

CTZ13 Fresno	Measure Life (Years)	Measure Cost ($)	Measure Cost ($/ft^2)	Maintenance Cost ($)	Maintenance Cost ($/ft^2)	TDV Cost Savings ($)	TDV Cost Savings ($/ft^2)	LCC ($)	LCC ($/ft^2)
Condenser Specific Efficiency – Outdoor Evaporative-Cooled	15	$6,358	$0.07	$0	$0	$21,822	$0.24	($15,464)	($0.17)
Condenser Specific Efficiency – Outdoor Air-Cooled	15	$5,380	$0.21	$0	$0	$28,096	$1.08	($22,716)	($0.87)
Condenser Specific Efficiency – Indoor Evaporative-Cooled	15	$288	$0.01	$0	$0	$730	$0.03	($442)	($0.02)
Freezer Roof Insulation	30	$10,764	$0.29	$0	$0	$27,299	$0.75	($16,534)	($0.41)
Evaporator Fan Control for Single Cycling-Compressor Systems	15	$30,016	$1.15	$11,251	$0.43	$308,621	$12	($267,354)	($10)
CTZ14 Palmdale	Measure Life (Years)	Measure Cost ($)	Measure Cost ($/ft^2)	Maintenance Cost ($)	Maintenance Cost ($/ft^2)	TDV Cost Savings ($)	TDV Cost Savings ($/ft^2)	LCC ($)	LCC ($/ft^2)
Condenser Specific Efficiency – Outdoor Evaporative-Cooled	15	$6,178	$0.07	$0	$0	$24,261	$0.26	($18,083)	($0.20)
Condenser Specific Efficiency – Outdoor Air-Cooled	15	$5,291	$0.20	$0	$0	$28,675	$1.10	($23,384)	($0.90)
Condenser Specific Efficiency – Indoor Evaporative-Cooled	15	$281	$0.01	$0	$0	$801	$0.03	($520)	($0.02)
Freezer Roof Insulation	30	$10,764	$0.29	$0	$0	$26,542	$0.72	($15,777)	($0.39)
Evaporator Fan Control for Single Cycling-Compressor Systems	15	$30,016	$1.15	$11,251	$0.43	$305,279	$12	($264,012)	($10)

j. Analysis Tools	None – mandatory measures.
k. Relationship to Other Measures	If measures have strong interactions, the economics of the measures can be affected. This may require incremental comparison (with the economics being affected by the order of incremental analysis) or evaluation of measures in various combinations. The savings interaction between the measures in this study is relatively small and therefore more complex analysis procedures were not necessary, particularly considering the attractive benefit-cost (BC) ratio of most measures. The condenser specific efficiency analysis utilized the existing 2008 standard with floating head pressure and required condenser sizing as the reference baseline. If a facility were to float head pressure lower than the minimum required by the standard (as some do), the energy savings of the proposed improved condenser specific efficiency would increase slightly. The decrease in heat gain resulting from improved roof insulation would decrease compressor heat of rejection and have a small impact on specific efficiency, but this would be a small fraction of a percentage.

3. Methodology

This section provides a description of the methodology used to evaluate the various refrigerated warehouse measures under consideration of the 2013 code change cycle. Topics in this section include:

- Refrigerated Warehouse Prototype Definitions
- Simulation and Cost Effectiveness Methodology
- Stakeholder Meeting Process

3.1 Refrigerated Warehouse Prototype Definitions

Prototype refrigerated warehouse models were developed to estimate the cost effectiveness of the proposed changes to the 2008 Title 24 refrigerated warehouse standards addressed in this report. Both small and large warehouse prototypes were developed. The small warehouse models utilized halocarbon refrigeration systems consisting of reciprocating compressors and air-cooled condensers (with the exception of Prototype Warehouse #5, which shared a shell configuration with Prototype Warehouse #3, but utilized condensing units instead of a built-up refrigeration system). The large refrigerated warehouse models utilized ammonia refrigeration systems with screw compressors and evaporative-cooled condensers. All of the warehouses were single story. A description of the refrigerated warehouse prototypes used in this analysis is shown in Appendix B. Figure 1 summarizes the warehouse prototypes used in this analysis.

Prototype Warehouse	Occupancy Type (Residential, Retail, Office, etc)	Area (S.F.)
1	Large Refrigerated Warehouse with Refrigerated Shipping Dock	92,000
2	Large Refrigerated Warehouse with Dry Storage Area	100,000
3	Small Refrigerated Warehouse with Refrigerated Shipping Dock	26,000
4	Small Refrigerated Warehouse with Dry Storage Area	30,000
5	Small Refrigerated Warehouse with Refrigerated Shipping Dock (Condensing Units)	26,000

Figure 1: Prototype warehouse summary

Figure 2 below shows a breakdown of space utilization for each prototype.

Prototype	Area per Space Type				
	35°F Cooler (S.F.)	-10°F Freezer (S.F.)	40°F Dock (S.F.)	Unconditioned Dry Storage (S.F.)	Total (S.F.)
1	40,000	40,000	12,000	0	92,000
2	40,000	40,000	0	20,000	100,000
3	10,000	10,000	6,000	0	26,000
4	10,000	10,000	0	10,000	30,000
5	10,000	10,000	6,000	0	26,000

Figure 2: Summary of space utilization for each prototype warehouse

3.2 Simulation and Cost Effectiveness Methodology

The energy usage for each measure in each prototype warehouse was evaluated using DOE-2.2R energy simulation software. The DOE2 version used (2.2R) is a sophisticated component-based energy simulation program that can accurately model the interaction between the building envelope, lighting systems, and refrigeration systems. The DOE-2.2R version is specifically designed to include refrigeration systems, using refrigerant properties, mass flow and component models to accurately describe refrigeration system operation and controls system effects.

Measures under consideration for the 2013 code change cycle were evaluated in seven different climate zones:

- CTZ03 – Oakland
- CTZ05 – Santa Maria
- CTZ07 – San Diego (Lindbergh)
- CTZ10 – Riverside
- CTZ12 – Sacramento (Sacramento Executive Airport)
- CTZ13 – Fresno
- CTZ15 – Palm Springs

Climate zones were selected to cover the variety of California climates where the majority of refrigerated warehouses are located.

The cost-effectiveness of the proposed measures was calculated using the Life Cycle Costing (LCC) Methodology prepared by the California Energy Commission (CEC). Measure costs are equal to the material costs, freight cost, sales taxes, labor costs, and tool rental costs associated with installing and commissioning the equipment or material embodied by the measure, minus the same costs associated with the equipment or material embodied by the base case. Measure costs also include the Present Value of maintenance costs and acceptance test costs, when applicable. A negative value of LCC represents high savings relative to cost. Measure costs are described in Appendix D.

The net present value of the energy savings was quantified using the Time Dependent Valuation (TDV) methodology.[1] Energy costs differ depending on the time of the day, week, and year that the energy is consumed. TDV assigns an energy cost to each hour of the year in order to capture the actual cost of energy to users, the utility systems, and society. TDV multipliers are statistically correlated to the weather files used in the simulation, the energy market, estimated escalation rates, and other factors. A unique set of TDV energy values was used for each weather file.

The base case assumptions concerning load, facility operations and other factors were held constant, with the only changes being those specific equipment changes or control strategies embodied in each measure. Some measures required adjustments to the base case in order to properly evaluate the energy savings. These “baseline” adjustments are described in Section 4 where applicable.

This report also recommends code changes without accompanying simulation analysis. These changes are either code clarifications requested by the industry, corrections that align the code with the intent of the 2008 analysis, or changes based on the consensus of industry stakeholders and the CEC. These code changes are described in Section 4 (“Code language additions not analyzed”).

[1] TDV methodology, Version 2

3.3 Acceptance Test

Acceptance test energy savings were assumed to be captured in the 2008 CASE analysis for refrigerated warehouses as the measures being tested in the acceptance test were evaluated in 2008 CASE as if the equipment was working properly. The 2008 CASE measures that are addressed in the 2013 CASE acceptance test are evaluated for cost effectiveness by adding the Present Value (PV) of acceptance test costs to the incremental measure cost from 2008 CASE and subtracting the 2008 TDV measure cost to get the LCC. This analysis ensures that the measures evaluated in 2008 that require an acceptance test are still cost effective once the costs of the acceptance tests are added. Assumptions for labor costs were gathered by survey and by protocol field tests. Survey results are presented in Appendix D.

3.4 Stakeholder Meeting Process

As part of the CASE study development process, a series of stakeholders meetings were conducted to present CASE study findings to, and solicit comments from, industry stakeholders affected by the potential changes to the Title 24 code for refrigerated warehouses. Stakeholders included refrigeration equipment manufacturers and distribution representatives; refrigerated warehouse and system designers; refrigeration system control manufacturers, representatives, installers and operators; refrigerated warehouse owners; utility reps; code officials; members of affiliated organizations (e.g., ASHRAE, AHRI); and staff from the CEC.

Three stakeholder meetings were held. The first two meetings presented outlines of the proposed analysis methodology and proposed measures. At the third meeting, cost effectiveness of proposed measures and proposed requirements was presented. Background on current code requirements and the code revision process was provided at all three stakeholder meetings.

In addition, stakeholders were contacted during ASHRAE meetings, by phone, at field tests of the acceptance test protocol, and at Title 24 Refrigerated Warehouse training classes for 2008 code.

The stakeholder meeting minutes are posted at www.h-m-g.com/T24/RefrigeratedWH/refrigeratedwh.htm.

4. Analysis and Results

Section 4 presents the measure descriptions and incremental analysis results. There were two objectives of the analysis: to determine which requirements are cost effective over the life of the facility; and to determine which requirements can be achieved with currently available technology or technology that can reasonably be expected to be available in the marketplace by the time the 2013 standard takes effect. Each specific measure was analyzed individually and in accordance with the methodology outlined in Section 3.

4.1 Statewide Energy Savings

The total energy and energy cost savings potential for condenser specific efficiency and freezer roof insulation are 0.23 kWh/ft^2 and 0.87 TDV $/ ft^2. Applying these unit estimates to the statewide estimate of refrigerated warehouse new construction of approximately 1.58 million square feet per year resulted in an overall statewide energy savings of 5.8 GWh and 22 million $ over 15 years. The energy and energy cost savings potential for evaporator speed controls on single compressor suction groups are 6.5 kWh/ft^2 and 12 TDV $/ ft^2. Applying these unit estimates to 7.8 percent of the statewide new construction estimate for 15 years resulted in a statewide energy savings of 10.7 GWh and 20 million $ over 15 years. The savings from these three measures resulted in a statewide savings as shown in Figure 3. There were no expected impacts on natural gas savings.

Total Electric Energy Savings (GWh)	Total TDV Savings ($)
16.5	**42,000,000**

Figure 3: Statewide energy and energy cost savings

4.2 Freezer Roof and Floor Insulation

A re-evaluation of the 2008 Title 24 insulation requirements for the freezer floor and roof was performed. Prototype Warehouses #2 and 4 were used to evaluate this measure. Measure cost information can be found in Appendix C.

Roof Insulation

For the roof insulation analysis, incremental insulation thicknesses were simulated in order to establish a regression of prototype building energy versus roof insulation rated R-value. Insulation was simulated with conductivity values at 40°F mean temperature in an effort to simulate the insulation performance at applied conditions (which are typically different for refrigerated warehouses than the 75°F mean temperature conditions applicable to the published (rated) R-values). Figure 4 summarizes the assumed material properties for the evaluated insulation types.

Material Type	Conductivity	Density	Specific Heat
Polyurethane panels, prefabricated urethane cam-lock panels, and polyisocyanurate over deck insulation	0.0098 Btu/(hr-ft^2-°F) at 40°F mean temperature, 0.0110 Btu/(hr-ft^2-°F) at 75°F mean temperature	1.50 lb/ft^3	0.38 Btu/(lb-°F)
Expanded polystyrene panels	0.0200 Btu/(hr-ft^2-°F) at 40°F mean temperature, 0.0216 Btu/(hr-ft^2-°F) at	1.80 lb/ft^3	0.29 Btu/(lb-°F)

	75°F mean temperature		

Figure 4: Insulation material assumptions

Figure 5 lists the freezer roof insulation thicknesses simulated as part of this analysis.

Simulation run	Insulation type	Insulation thickness	R-value at 75°F mean temperature (standard rating conditions)	R-value at 40°F mean temperature (simulated conditions)
1	Polyurethane panels, prefabricated urethane cam-lock panels, and polyisocyanurate over deck insulation	4”	30	34
2		4.5”	34	38
3		5”	38	43
4		5.5”	42	47
5		6”	45	51
6		6.5”	49	55
7	Expanded polystyrene panels	7”	27	29
8		8”	31	33
9		9”	35	38
10		10”	39	42
11		11”	42	46
12		12”	46	50

Figure 5: Simulated freezer roof insulation thicknesses

4.2.2 Roof Insulation Analysis Results by Climate Zone

The freezer roof measure was evaluated using seven climate zones. Figure 6 summarizes the analysis results for R-40 roof insulation, the proposed value, as compared to the current R-36 code requirement.

Prototype Warehouse		Annual Energy Savings		TDV Cost Savings		Incremental Cost ($)	Benefit/Cost Ratio
		kWh	kWh/SF	$	$/SF		
CTZ03 Oakland							
Large	Polyurethane	4,403	0.11	$22,837	$0.57	$11,649	2.0
	Expanded Polystyrene	4,955	0.12	$25,932	$0.65	$13,419	1.9
	Pre-Fab Cam-Lock	4,403	0.11	$22,837	$0.57	$31,908	0.7
	Over deck	4,403	0.11	$22,837	$0.57	$9,152	2.5
Small	Polyurethane	1,790	0.18	$7,007	$0.70	$2,912	2.4
	Expanded Polystyrene	1,894	0.19	$7,638	$0.76	$3,355	2.3
	Pre-Fab Cam-Lock	1,790	0.18	$7,007	$0.70	$7,977	0.9
	Over deck	1,790	0.18	$7,007	$0.70	$2,288	3.1
CTZ05 Santa Maria							
Large	Polyurethane	3,653	0.09	$18,340	$0.46	$11,649	1.6
	Expanded Polystyrene	4,784	0.12	$26,471	$0.66	$13,419	2.0
	Pre-Fab Cam-Lock	3,653	0.09	$18,340	$0.46	$31,908	0.6
	Over deck	3,653	0.09	$18,340	$0.46	$9,152	2.0
Small	Polyurethane	1,787	0.18	$6,991	$0.70	$2,912	2.4
	Expanded Polystyrene	1,876	0.19	$7,546	$0.76	$3,355	2.3
	Pre-Fab Cam-Lock	1,787	0.18	$6,991	$0.70	$7,977	0.9
	Over deck	1,787	0.18	$6,991	$0.70	$2,288	3.1
CTZ07 San Diego							
Large	Polyurethane	5,594	0.14	$21,820	$0.55	$11,649	1.9

	Expanded Polystyrene	5,681	0.14	$23,607	$0.59	$13,419	1.8
	Pre-Fab Cam-Lock	5,594	0.14	$21,820	$0.55	$31,908	0.7
	Over deck	5,594	0.14	$21,820	$0.55	$9,152	2.4
Small	Polyurethane	2,012	0.20	$7,777	$0.78	$2,912	2.7
	Expanded Polystyrene	2,118	0.21	$8,408	$0.84	$3,355	2.5
	Pre-Fab Cam-Lock	2,012	0.20	$7,777	$0.78	$7,977	1.0
	Over deck	2,012	0.20	$7,777	$0.78	$2,288	3.4
CTZ10 Riverside							
Large	Polyurethane	6,090	0.15	$26,686	$0.67	$11,649	2.3
	Expanded Polystyrene	6,186	0.16	$29,643	$0.74	$13,419	2.2
	Pre-Fab Cam-Lock	6,090	0.15	$26,686	$0.67	$31,908	0.8
	Over deck	6,090	0.15	$26,686	$0.67	$9,152	2.9
Small	Polyurethane	2,280	0.23	$9,316	$0.93	$2,912	3.2
	Expanded Polystyrene	2,402	0.24	$10,210	$1.02	$3,355	3.0
	Pre-Fab Cam-Lock	2,280	0.23	$9,316	$0.93	$7,977	1.2
	Over deck	2,280	0.23	$9,316	$0.93	$2,288	4.1
CTZ 12 Sacramento							
Large	Polyurethane	6,029	0.15	$25,886	$0.65	$11,649	2.2
	Expanded Polystyrene	5,767	0.14	$25,901	$0.65	$13,419	1.9
	Pre-Fab Cam-Lock	6,029	0.15	$25,886	$0.65	$31,908	0.8
	Over deck	6,029	0.15	$25,886	$0.65	$9,152	2.8
Small	Polyurethane	2,280	0.23	$9,547	$0.96	$2,912	3.3
	Expanded Polystyrene	2,422	0.24	$10,471	$1.05	$3,355	3.1
	Pre-Fab Cam-Lock	2,280	0.23	$9,547	$0.96	$7,977	1.2
	Over deck	2,280	0.23	$9,547	$0.96	$2,288	4.2
CTZ13 Fresno							
Large	Polyurethane	6,950	0.17	$26,933	$0.67	$11,649	2.3
	Expanded Polystyrene	7,508	0.19	$32,369	$0.81	$13,419	2.4
	Pre-Fab Cam-Lock	6,950	0.17	$26,933	$0.67	$31,908	0.8
	Over deck	6,950	0.17	$26,933	$0.67	$9,152	2.9
Small	Polyurethane	2,531	0.25	$10,271	$1.03	$2,912	3.5
	Expanded Polystyrene	2,646	0.27	$11,103	$1.11	$3,355	3.3
	Pre-Fab Cam-Lock	2,531	0.25	$10,271	$1.03	$7,977	1.3
	Over deck	2,531	0.25	$10,271	$1.03	$2,288	4.5
CTZ14 Palmdale							
Large	Polyurethane	5,046	0.13	$26,024	$0.65	$11,649	2.2
	Expanded Polystyrene	5,975	0.15	$31,999	$0.80	$13,419	2.4
	Pre-Fab Cam-Lock	5,046	0.13	$26,024	$0.65	$31,908	0.8
	Over deck	5,046	0.13	$26,024	$0.65	$9,152	2.8
Small	Polyurethane	2,268	0.23	$9,178	$0.92	$2,912	3.2
	Expanded Polystyrene	2,408	0.24	$10,040	$1.00	$3,355	3.0
	Pre-Fab Cam-Lock	2,268	0.23	$9,178	$0.92	$7,977	1.2
	Over deck	2,268	0.23	$9,178	$0.92	$2,288	4.0

Figure 6: Freezer roof insulation analysis results

When compared with R-36, results showed that R-40 insulation was cost-justified for all evaluated climate zones, and for all insulation types except for pre-fabricated buildings with cam-lock urethane panels more common on smaller boxes (i.e., <3,000ft^2). The poor benefit/cost ratio for these panels is attributed to their high incremental cost. Because cam-lock type panels are an elective design choice and not a necessary construction method for any particular refrigerated warehouse applications, there is no exception for application(s) with lower insulation value(s). In general, stakeholders (mostly larger cold storage contractors) commented that common practice was to use R-40 or greater insulation in most freezer applications and also noted that R-40 was still lower than ASHRAE recommendations.

The ASHRAE recommendation for -10°F to -20°F holding freezer roof insulation is R-45 to R-50.[2] This is a consensus recommendation for a year-round facility (with a standard efficiency refrigeration plant) and was not determined through energy analysis and cost effectiveness calculations. The refrigeration plant efficiency that results from the 2008 refrigerated warehouse standards is far higher than past efficiencies, which reduces the cost-effective insulation thickness. Also, the rating basis for the ASHRAE R-values can be assumed to be the commercial rating at applied temperatures (40°F) which is (higher than the rating basis for Title 24 standards (measured at 70°F) and probably without reduction for aging. With these considerations, the proposed value of R-40 can be considered approximately consistent with the recommended values in ASHRAE.

The proposed code change is to increase the minimum roof insulation from R-36 to R-40.

Floor Insulation

Analysis was performed to evaluate the cost-effectiveness of mandating less insulation than the 2008 freezer floor insulation code requirement. The purpose of the analysis was to align the standard with an R-value that matches available insulation thickness. Extruded polystyrene is most commonly available in 2” increments but can be purchased in 1” increments. Accordingly, the R-36 requirement in the 2008 code can not be constructed with the typical or optionally available thicknesses.

For the floor insulation analysis, incremental insulation thicknesses were simulated in order to establish a regression of prototype building energy versus freezer floor insulation R-value. This analysis simulated extruded polystyrene, the sole insulation method found for freezer floor insulation which has a thermal resistance of R-5.0 per inch of thickness at the 75°F mean temperature rating conditions, and R-5.4 per inch at 40°F mean temperature (the assumed condition in the simulation). Figure 7 lists the freezer floor insulation thicknesses simulated as part of this analysis.

Simulation run	Insulation type	Insulation thickness	R-value at 75°F mean temperature (standard rating conditions)	R-value at 40°F mean temperature (simulated conditions)
1	Extruded Polystyrene	3”	15	16
2		4”	20	22
3		5”	25	27
4		6”	30	32
5		7”	35	38
6		8”	40	43

Figure 7: Simulated freezer floor insulation thicknesses

[2] ASHRAE Refrigeration Handbook, 2010. p. 23.13, Table 2.

4.2.3 Floor Insulation Analysis Results by Climate Zone

The freezer floor measure was simulated in three climate zones with an assumed soil temperature of 48°F year-round, overriding the soil temperature specified in the weather file to account for freezer under-floor heating. Figure 8 summarizes the analysis results for R-35 compared to R-30 floor insulation, and Figure 9 summarizes the results for R-40 compared to R-35 floor insulation.

Prototype Warehouse	Energy Savings		TDV Cost Savings		Incremental First Cost	Benefit/Cost Ratio
	kWh	kWh/SF	$	$/SF		
CTZ12 - Sacramento Executive						
Small	3,630	0.36	$13,424	$1.34	$4,850	2.8
Large	8,622	0.22	$35,709	$0.89	$19,400	1.8
CTZ10 – Riverside						
Small	3,481	0.35	$12,493	$1.25	$4,850	2.6
Large	8,860	0.22	$37,709	$0.94	$19,400	1.9
CTZ05 - Santa Maria						
Small	3,160	0.32	$11,054	$1.11	$4,850	2.3
Large	7,019	0.18	$35,946	$0.90	$19,400	1.9

Figure 8: R-35 compared to R-30 freezer floor insulation analysis results

Prototype Warehouse	Energy Savings		TDV Cost Savings		Incremental Cost	Benefit/Cost Ratio
	kWh	kWh/SF	$	$/SF		
CTZ12 - Sacramento Executive						
Small	2,600	0.26	$9,613	$0.96	$4,850	2.0
Large	5,754	0.14	$24,506	$0.61	$19,400	1.3
CTZ10 - Riverside						
Small	2,567	0.26	$9,112	$0.91	$4,850	1.9
Large	5,731	0.14	$21,033	$0.53	$19,400	1.1
CTZ05 - Santa Maria						
Small	2,399	0.24	$8,320	$0.83	$4,850	1.7
Large	4,360	0.11	$16,161	$0.40	$19,400	0.8

Figure 9: R-40 compared to R-35 freezer floor insulation analysis results

Analysis shows that R-35 was easily cost-effective compared with R-30 for both warehouse prototypes in all simulated climate zones, while R-40 was less cost cost-effective than R-35 overall and was close to or below a 1.0 BC ratio for the large warehouse prototype. Stakeholders felt strongly that R-35 was sufficient and cost-effective, and noted other considerations including the fact that many boxes were built as "convertible" freezers that could operate as coolers or freezers, which would affect the average cost-effectiveness.

The proposed code change is to reduce the minimum freezer floor insulation from R-36 to R-35, with an exception if underfloor heat is provided through heat exchange with the refrigeration system in a manner that produces productive cooling (such as with a mechanical subcooler). In such a case, the minimum freezer floor insulation minimum requirement is R-20.

4.3 Evaporator Fan Control for Single Cycling-Compressor Systems

Evaporator fan control was evaluated for single-compressor refrigeration systems without variable capacity capability. The 2008 code includes an exception to fan speed controls for "evaporators served by a single compressor without unloading capability." For this measure, a separate base case (Prototype Warehouse #5) was developed based on the small Prototype Warehouse #3 but utilized single-compressor condensing units. Figure 10 summarizes the base case assumptions for this measure.

Design City	**Santa Maria (CTZ05)**	**Riverside (CTZ10)**	**Sacramento (CTZ12)**
Building envelope, lighting, schedules, and design refrigeration loads	Same as Prototype Warehouse #3		
Design ambient temperature	90°F	106°F	104°F
Design SST	Cooler System: -22°F Freezer System: 23°F Dock System: 28°F	Cooler System: -22°F Freezer System: 23°F Dock System: 28°F	Cooler System: -22°F Freezer System: 23°F Dock System: 28°F
Condensing unit catalog capacity at design conditions	Cooler: 192.0 MBH Freezer: 85.1 MBH Dock: 209.9 MBH	Cooler: 160.7 MBH Freezer: 68.6 MBH Dock: 181.5 MBH	Cooler: 163.9 MBH Freezer: 72.2 MBH Dock: 185.0 MBH
Compressor nominal hp	All systems: 15 HP	All systems: 15 HP	All systems: 15 HP
Number of Required Condensing Units	Cooler: 2 Freezer: 5 Dock: 2	Cooler: 3 Freezer: 7 Dock: 3	Cooler: 3 Freezer: 7 Dock: 3

Figure 10: Base case assumptions for evaporator fan speed control measure

Figure 11 summarizes the simulated runs for the evaluation of this measure. Two control methods were considered based on discussion with evaporator coil manufacturers and other vendors: the two-speed control and variable speed control. The two-speed control method can be accomplished at almost no cost on small low-profile evaporators. Although these evaporators are smaller than needed in most refrigerated warehouses, their capabilities may soon be available in larger single-phase electronically commutated (EC, also called brushless DC or BLDC) motors. For larger three-phase motors, external controllers are currently in the market or variable speed control can be used in a two-speed configuration. Two methods are available for variable speed control: a variable speed drive or an evaporator with EC motors. This analysis assumed a separate variable speed drive and output filter for each evaporator and a controller with associated control capability on each condensing unit.

One evaporator manufacturer has offered the ability to stage fans or "cycle off" some of the fans in each evaporator coil. While this method has not been frequently utilized and may not be attractive to all users due to concern for frost patterns and other issues, it is a feasible alternative.

Run 1	Base Case
Run 2	2-speed Fan, 90% low speed
Run 3	2-speed Fan, 80% low speed
Run 4	2-speed Fan, 70% low speed
Run 5	2-speed Fan, 60% low speed
Run 6	2-speed Fan, 50% low speed
Run 7	Fan staging – cycle 1 of 2 fans
Run 8	Fan staging – cycle 3 of 4 fans
Run 9	Fan staging – cycle 2 of 3 fans

Figure 11: Simulation summary for evaporator fan control measure

4.3.1 Evaporator Speed Control Analysis Results by Climate Zone

Figure 12 summarizes the results for the evaporator speed control measure.

	Energy Savings		TDV Cost Savings		Incremental Cost	15-Year Maintenance Cost	Benefit/Cost Ratio
	kWh	kWh/SF	$	$/SF			
CTZ03 - Oakland							
2-Speed Fan, 90% Low Speed	87,170	8.7	$153,608	$15	$37,867	$15,124	2.9
2-Speed Fan, 80% Low Speed	134,412	13.4	$241,003	$24	$37,867	$15,124	4.5
2-Speed Fan, 70% Low Speed	172,325	17.2	$311,106	$31	$37,867	$15,124	5.9
2-Speed Fan, 60% Low Speed	201,963	20.2	$365,759	$37	$37,867	$15,124	6.9
2-Speed Fan, 50% Low Speed	224,335	22.4	$407,027	$41	$37,867	$15,124	7.7
Fan Staging - Cycle 1 of 2 Fans	145,311	14.5	$261,250	$26	$8,664	$4,795	19
Fan Staging - Cycle 3 of 4 Fans	202,373	20.2	$366,515	$37	$8,664	$4,795	27
Fan Staging - Cycle 2 of 3 Fans	183,366	18.3	$331,460	$33	$8,664	$4,795	25
CTZ05 – Santa Maria							
2-Speed Fan, 90% Low Speed	86,575	8.7	$153,653	$15	$37,867	$15,124	2.9
2-Speed Fan, 80% Low Speed	133,737	13.4	$240,986	$24	$37,867	$15,124	4.5
2-Speed Fan, 70% Low Speed	171,496	17.1	$310,617	$31	$37,867	$15,124	5.9
2-Speed Fan, 60% Low Speed	201,064	20.1	$365,118	$37	$37,867	$15,124	6.9
2-Speed Fan, 50% Low Speed	223,401	22.3	$406,297	$41	$37,867	$15,124	7.7
Fan Staging - Cycle 1 of 2 Fans	144,563	14.5	$260,965	$26	$8,664	$4,795	19
Fan Staging - Cycle 3 of 4 Fans	201,474	20.1	$365,875	$37	$8,664	$4,795	27
Fan Staging - Cycle 2 of 3 Fans	182,509	18.3	$330,917	$33	$8,664	$4,795	25
CTZ07 – San Diego							
2-Speed Fan, 90% Low Speed	89,015	8.9	$159,206	$16	$37,867	$15,124	3.0
2-Speed Fan, 80% Low Speed	136,968	13.7	$249,200	$25	$37,867	$15,124	4.7
2-Speed Fan, 70% Low Speed	175,451	17.5	$321,278	$32	$37,867	$15,124	6.1
2-Speed Fan, 60% Low Speed	205,529	20.6	$377,560	$38	$37,867	$15,124	7.1
2-Speed Fan, 50% Low Speed	228,223	22.8	$420,020	$42	$37,867	$15,124	7.9
Fan Staging - Cycle 1 of 2 Fans	148,020	14.8	$269,918	$27	$8,664	$4,795	20
Fan Staging - Cycle 3 of 4 Fans	205,946	20.6	$378,334	$38	$8,664	$4,795	28
Fan Staging - Cycle 2 of 3 Fans	186,660	18.7	$342,264	$34	$8,664	$4,795	25
CTZ10 - Riverside							
2-Speed Fan, 90% Low Speed	88,022	8.8	$153,555	$15	$37,867	$15,124	2.9
2-Speed Fan, 80% Low Speed	137,139	13.7	$244,910	$24	$37,867	$15,124	4.6
2-Speed Fan, 70% Low Speed	176,417	17.6	$317,576	$32	$37,867	$15,124	6.0
2-Speed Fan, 60% Low Speed	207,198	20.7	$374,463	$37	$37,867	$15,124	7.1
2-Speed Fan, 50% Low Speed	230,459	23.0	$417,475	$42	$37,867	$15,124	7.9
Fan Staging - Cycle 1 of 2 Fans	148,397	14.8	$265,780	$27	$8,664	$4,795	19.8
Fan Staging - Cycle 3 of 4 Fans	207,625	20.8	$375,255	$38	$8,664	$4,795	28
Fan Staging - Cycle 2 of 3 Fans	187,881	18.8	$338,766	$34	$8,664	$4,795	25
CTZ12 - Sacramento							
2-Speed Fan, 90% Low Speed	85,188	8.5	$151,757	$15	$37,867	$15,124	2.9
2-Speed Fan, 80% Low Speed	132,950	13.3	$240,843	$24	$37,867	$15,124	4.5
2-Speed Fan, 70% Low Speed	171,635	17.2	$313,447	$31	$37,867	$15,124	5.9
2-Speed Fan, 60% Low Speed	201,680	20.2	$369,470	$37	$37,867	$15,124	7.0
2-Speed Fan, 50% Low Speed	224,339	22.4	$411,664	$41	$37,867	$15,124	7.8
Fan Staging - Cycle 1 of 2 Fans	144,067	14.4	$261,642	$26	$8,664	$4,795	19
Fan Staging - Cycle 3 of 4 Fans	202,096	20.2	$370,244	$37	$8,664	$4,795	28
Fan Staging - Cycle 2 of 3 Fans	182,831	18.3	$334,334	$33	$8,664	$4,795	25
CTZ13 - Fresno							
2-Speed Fan, 90% Low Speed	85,734	8.6	$151,784	$15	$37,867	$15,124	2.9

2-Speed Fan, 80% Low Speed	134,233	13.4	$242,071	$24	$37,867	$15,124	4.6
2-Speed Fan, 70% Low Speed	173,438	17.3	$315,494	$32	$37,867	$15,124	5.9
2-Speed Fan, 60% Low Speed	203,914	20.4	$372,087	$37	$37,867	$15,124	7.0
2-Speed Fan, 50% Low Speed	226,906	22.7	$414,778	$41	$37,867	$15,124	7.8
Fan Staging - Cycle 1 of 2 Fans	145,547	14.6	$263,315	$26	$8,664	$4,795	20
Fan Staging - Cycle 3 of 4 Fans	204,336	20.4	$372,870	$37	$8,664	$4,795	28
Fan Staging - Cycle 2 of 3 Fans	184,791	18.5	$336,568	$34	$8,664	$4,795	25
CTZ14 - Palmdale							
2-Speed Fan, 90% Low Speed	85,389	8.5	$148,464	$15	$37,867	$15,124	2.8
2-Speed Fan, 80% Low Speed	134,261	13.4	$238,520	$24	$37,867	$15,124	4.5
2-Speed Fan, 70% Low Speed	173,872	17.4	$311,569	$31	$37,867	$15,124	5.9
2-Speed Fan, 60% Low Speed	204,982	20.5	$369,061	$37	$37,867	$15,124	7.0
2-Speed Fan, 50% Low Speed	228,428	22.8	$412,402	$41	$37,867	$15,124	7.8
Fan Staging - Cycle 1 of 2 Fans	145,579	14.6	$259,372	$26	$8,664	$4,795	19.3
Fan Staging - Cycle 3 of 4 Fans	205,413	20.5	$369,862	$37	$8,664	$4,795	27
Fan Staging - Cycle 2 of 3 Fans	185,462	18.5	$332,981	$33	$8,664	$4,795	25

Figure 12: Statewide savings results for evaporator fan control measure

This measure is cost-effective under all conditions and assumptions.

The proposed code addition requires all evaporator fans served by a suction group with a single compressor without variable capacity to utilize evaporator fan controls capable of reducing airflow by at least 40 percent whenever the compressor is not operating. The controls may provide periodic full-speed operation for additional air circulation that does not exceed 25 percent of the time the compressor is not running.

4.4 Condenser Specific Efficiency

The cost-effectiveness of establishing a minimum condenser specific efficiency was analyzed. Specific efficiency is the ratio of full-load condenser capacity at standardized conditions divided by the consumed condenser electric power. Figure 13 describes the condenser types and the corresponding warehouse prototypes that were evaluated for this measure.

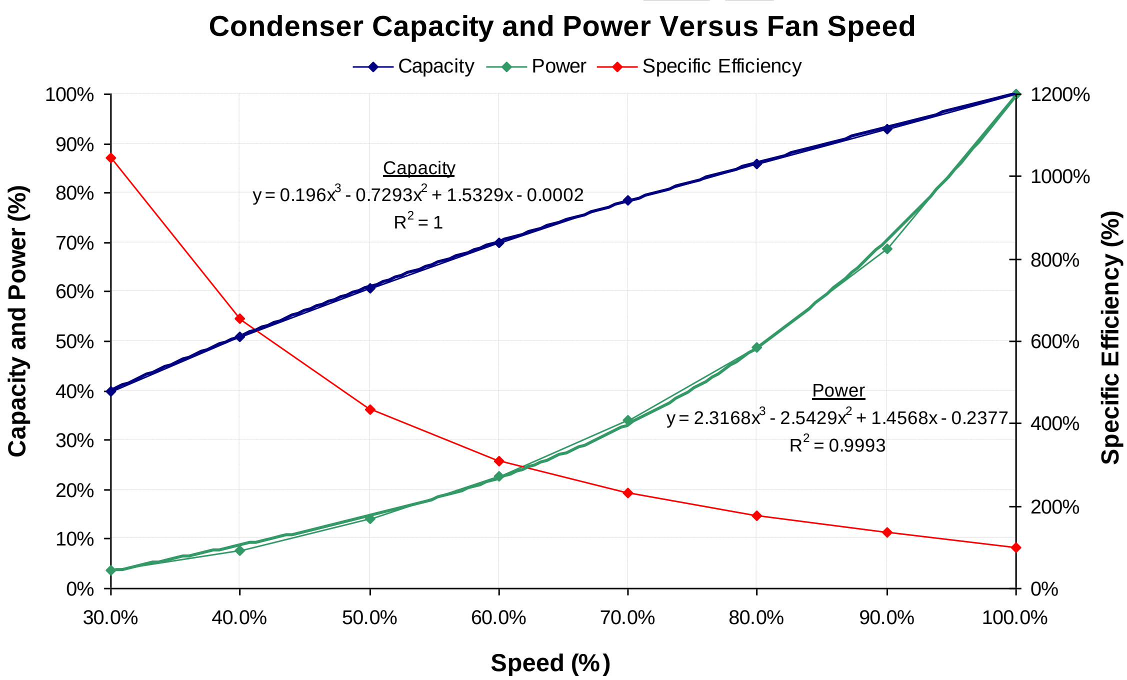
Condenser Category	Exemplifying Condenser Description	Prototype Warehouse
Outdoor Evaporative-Cooled	Forced-draft axial-fan evaporative-cooled ammonia condenser	Large with Refrigerated Dock (#1)
Indoor Evaporative-Cooled	Forced-draft centrifugal-fan halocarbon evaporative condenser	Small with Refrigerated Dock (#3)
Outdoor Air-Cooled	Axial-fan air-cooled halocarbon condenser	

Figure 13: Description of prototype warehouses for condenser specific efficiency measure

A direct correlation between cost and specific efficiency could not be determined from manufacturer catalog information, as manufacturing cost is not proportionately reflected in model-by-model sale prices for these units. An alternative method was employed to establish the minimum cost-effective condenser specific efficiency. This method is more consistent with how manufacturers could easily comply with an efficiency standard when redesigning products. In general, specific efficiency is improved by reducing the fan power for a given condenser.

Condenser fan power reduces by approximately the "third-power" of fan speed reduction whereas condenser capacity is roughly linear (or better than linear) with reduction in fan speed. Manufacturers stated that both air-cooled and evaporative-cooled condensers generally have flexibility in fan design and speed, and thus motor power. In particular, the maximum speed for air-cooled condensers using variable speed EC motors can easily be reprogrammed at the factory, making specific efficiency essentially a "settable" parameter.

The air-cooled condenser data provided by one manufacturer, shown in Figure 14, illustrates the sensitivity of specific efficiency to fan speed. The left axis shows heat rejection capacity and fan power; the right axis shows specific efficiency.



Speed:	100%	90%	80%	70%	60%	50%	40%	30%
Capacity:	100%	93%	86%	78%	70%	61%	51%	40%
Power:	100%	69%	49%	34%	23%	14%	8%	4%
Specific Efficiency:	100%	135%	177%	231%	308%	434%	655%	1,044%

Figure 14: Graph of condenser capacity and power versus speed

Figure 14 shows that the relationship between fan speed and condenser capacity is nearly linear, while fan power is subject to the fan affinity laws which state that fan power exhibits a "third power" relationship with fan speed. Consequently, specific efficiency increases exponentially at reduced fan speed. Without substantial product line changes, manufacturers could utilize this relationship by reducing or limiting the full-load fan speed and motor power of any non-compliant condenser to a speed which achieves the required efficiency, thus still being able to market the condenser with a revised capacity listing.

In many instances, improvements could also be made with higher efficiency motors, fan blades or fan venturis. These improvements are the most likely path for certain air-cooled halocarbon condensers which utilize inefficient motors. The methodology described above is considered the most conservative with respect to measure cost, and also an approach that could be adopted without major product line changes or "tooling" difficulty for smaller manufacturers.

A comparable method was employed to calculate measure cost for this analysis. This method utilized a correlation between end-user cost and full-speed condenser Total Heat of Rejection (THR) capacity (Figure 15 shows an example for axial-fan evaporative-cooled ammonia condensers). The correlation was used to calculate the cost of incrementally oversizing the condenser, then limiting the maximum condenser fan speed to match the capacity of the original condenser size with a consequent increase in condenser specific efficiency. The table in Figure 16 demonstrates this concept with a starting full-speed capacity of 8,537 MBH and a starting specific efficiency of 325 Btuh/Watt.

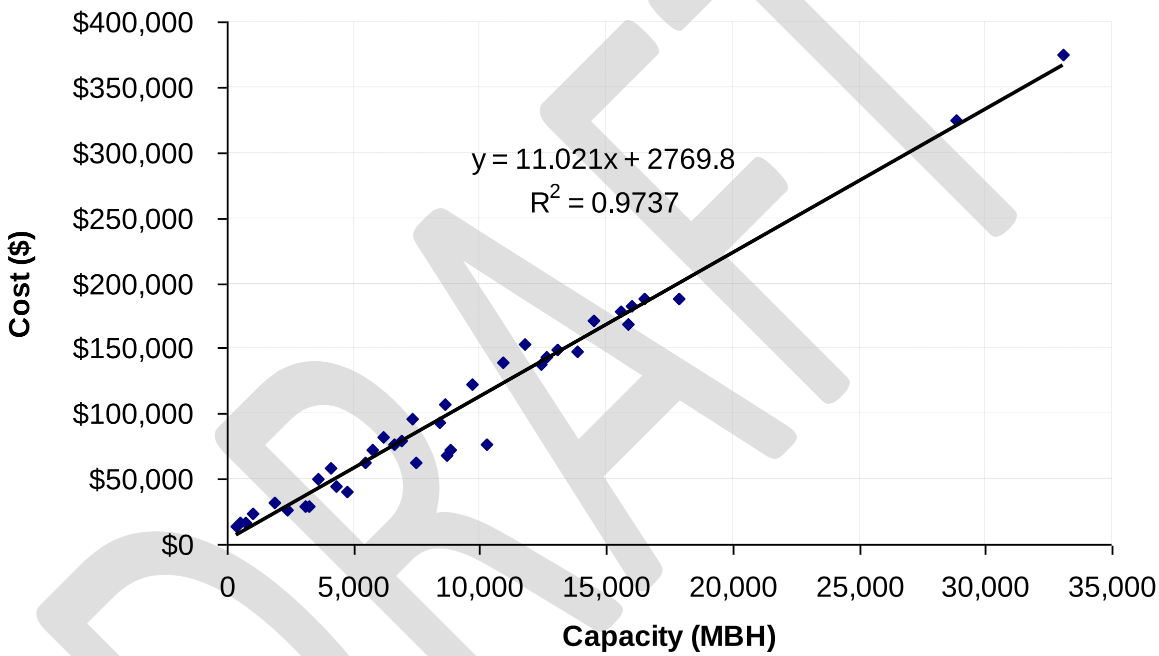


Figure 15: Condenser cost versus capacity at specific-efficiency rating conditions

Percent incremental increase in condenser size	Capacity of larger condenser at 100% speed (MBH)	Power of larger condenser at 100% speed at original specific efficiency (kW)	Required percent capacity of oversized condenser to match original capacity	Maximum speed of new condenser to match original capacity	Power at reduced maximum speed (kW)	New Specific Efficiency (Btuh/Watt)
0%	8,537	26.27	100.0%	100.0%	26.3	325
1%	8,622	26.53	99.0%	98.6%	25.1	340
2%	8,708	26.79	98.0%	97.1%	24.2	353
3%	8,793	27.06	97.1%	95.7%	23.2	367
4%	8,878	27.32	96.2%	94.4%	22.4	382
5%	8,964	27.58	95.2%	93.0%	21.6	396

Figure 16: Example of incrementally increasing condenser size and resultant specific efficiency

A DOE2.2R simulation was used to calculate prototype building energy use and TDV energy cost with varying condenser specific efficiency (condenser fan power was adjusted, with all other parameters held constant). Figure 17 shows the simulation results for the large prototype warehouse with an evaporative-cooled condenser.

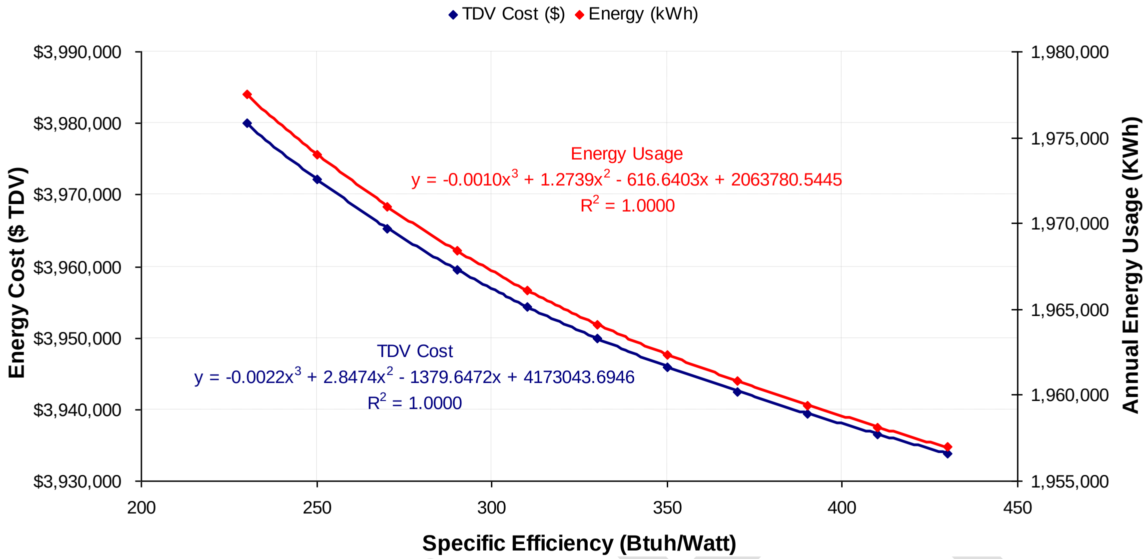


Figure 17: Example of building energy use and TDV energy cost versus specific efficiency

The simulation results, condenser costs, and incremental oversize analysis were combined to determine the most cost-effective condenser specific efficiency (defined as the efficiency at which further incrementally increasing the condenser size is not cost-effective).

For this measure, the prototype warehouses were simulated with a 70°F minimum condensing temperature, an ambient-following control strategy and variable speed control of all condenser fans. DOE-2.2R simulation keywords explicitly applied the subject control strategy.

The assumed specific-efficiency rating basis was 95°F ambient drybulb temperature and 105°F saturated condensing temperature for air-cooled condensers, and 70°F ambient wetbulb temperature and 100°F saturated condensing temperature for evaporative condensers.

4.4.1 Incremental Analysis Results

For each evaluated condenser type, the condenser specific efficiency was incrementally increased until the cost-effectiveness of subsequent incremental improvements was no longer justified (based on LCC methodology). The final specific efficiency increment became the proposed specific efficiency. Analysis to identify the proposed specific efficiency levels utilized climate zones CTZ05 (Santa Maria), CTZ10 (Riverside), and CTZ12 (Sacramento), with the results then applied to a total of seven climate zones. Figure 18 summarizes the results from the preliminary analysis.

The base case specific efficiency for statewide savings analysis listed in Figure 18 was obtained from Savings By Design (SBD) new construction projects. This efficiency is the average of condensers installed on new refrigerated warehouse projects in California between 2006 and 2010 (i.e., the average of condensers which were below the cost-effective specific efficiency). The SBD data for the statewide analysis base case is included in Appendix F.

Condenser Type	Cost-effective minimum specific efficiency (Btuh/Watt)	Basis of comparison for incremental analysis (Btuh/Watt)	Base Case specific efficiency for statewide analysis (Btuh/Watt)
Outdoor air-cooled with ammonia refrigerant	75	65	NA[3]
Outdoor air-cooled w/ halocarbon refrigerant	65	55	53
Outdoor evaporative-cooled	350	325	265
Indoor evaporative-cooled	160	140	155

Figure 18: Preliminary condenser specific efficiency results

4.4.2 Condenser Specific Efficiency Analysis Results by Climate Zone

Figure 19 summarizes the simulation results for the condenser specific efficiency measure simulated in seven climate zones.

	Annual Energy Savings (kWh)		TDV Cost Savings ($)		Measure Cost ($)	Benefit/ Cost Ratio
	Total	Per SF	Total	Per SF		
CTZ03 Oakland						
Outdoor NH_3 evaporative-cooled	8,305	0.09	$18,912	$0.21	$5,812	3.25
Indoor HFC evaporative-cooled	356	0.01	$765	$0.03	$279	2.74
Outdoor HFC air-cooled	1,556	0.06	$6,461	$0.25	$2,531	2.55
Outdoor HFC air-cooled BLDC motors	1,556	0.06	$6,461	$0.25	$4,939	1.31
CTZ05 Santa Maria						
Outdoor NH_3 evaporative-cooled	7,909	0.09	$17,523	$0.19	$5,812	3.02
Indoor HFC evaporative-cooled	350	0.01	$748	$0.03	$279	2.68
Outdoor HFC air-cooled	1,443	0.06	$4,396	$0.17	$2,531	1.74
Outdoor HFC air-cooled BLDC motors	1,443	0.06	$4,396	$0.17	$4,939	0.89
CTZ07 San Diego						
Outdoor NH_3 evaporative-cooled	8,548	0.09	$18,556	$0.20	$5,812	3.19
Indoor HFC evaporative-cooled	387	0.02	$810	$0.03	$279	2.90
Outdoor HFC air-cooled	1,779	0.07	$5,669	$0.22	$2,531	2.24
Outdoor HFC air-cooled BLDC motors	1,779	0.07	$5,669	$0.22	$4,939	1.15
CTZ10 Riverside						
Outdoor NH_3 evaporative-cooled	8,850	0.10	$20,612	$0.224	$6,178	3.34
Indoor HFC evaporative-cooled	378	0.02	$801	$0.031	$281	2.85
Outdoor HFC air-cooled	5,594	0.22	$23,967	$0.922	$4,110	5.83
Outdoor HFC air-cooled BLDC motors	5,594	0.22	$23,967	$0.922	$6,472	3.70
CTZ12 Sacramento						
Outdoor NH_3 evaporative-cooled	9,337	0.10	$20,932	$0.23	$6,358	3.29
Indoor HFC evaporative-cooled	338	0.01	$721	$0.03	$288	2.50
Outdoor HFC air-cooled	4,658	0.18	$21,867	$0.84	$4,179	5.23
Outdoor HFC air-cooled BLDC motors	4,658	0.18	$21,867	$0.84	$6,581	3.32
CTZ13 Fresno						

[3] Based on the Savings By Design new construction experience, there are few, if any, air-cooled ammonia condensers used on refrigerated warehouse in California. There are, however, two manufacturers with applicable product lines who have sold large ammonia air-cooled condensers in other areas inside and outside the US. A minimum specific efficiency requirement for air-cooled ammonia condensers was developed using the performance and costs of equipment offered by these two manufacturers.

Outdoor NH_3 evaporative-cooled	9,612	0.10	$21,822	$0.24	$6,358	3.43
Indoor HFC evaporative-cooled	344	0.01	$730	$0.03	$288	2.53
Outdoor HFC air-cooled	7,680	0.30	$28,096	$1.08	$4,179	6.72
Outdoor HFC air-cooled BLDC motors	7,680	0.30	$28,096	$1.08	$6,581	4.27
CTZ14 Palmdale						
Outdoor NH_3 evaporative-cooled	9,441	0.10	$24,261	$0.26	$6,178	3.93
Indoor HFC evaporative-cooled	350	0.01	$801	$0.03	$281	2.85
Outdoor HFC air-cooled	7,915	0.30	$28,675	$1.10	$4,110	6.98
Outdoor HFC air-cooled BLDC motors	7,915	0.30	$28,675	$1.10	$6,472	4.43

Figure 19: Analysis results by climate zone for condenser specific efficiency measure

Each climate zone analysis considered condensers with "standard" induction motors as well as outdoor HFC air-cooled condensers equipped with brushless DC (BLDC) motors. Nearly all air-cooled HFC condenser manufacturers offer condensers with BLDC fan motors; these motors are more expensive but have the advantage of being inherently variable-speed with the application of a control signal, thus eliminating the need for a variable speed drive. As noted previously, the maximum speed (and therefore the specific efficiency) for these condensers is effectively a factory-settable parameter.

One climate zone had a BC ratio of less than 1.0 for air-cooled BLDC condensers. For all other cases and climate zones, results showed that the proposed specific efficiency levels were cost-effective using the standard LCC methodology. Because BLDC motors are an elective design choice when purchasing condensers, it was decided that one cost-prohibitive climate zone, would not justify establishing climate-specific exceptions to the standard.

An important observation is that several manufacturers have recently introduced new air-cooled condensers using "micro-channel" heat exchanger surfaces. This is a major technology change that is currently evolving. Initial information indicates these condensers will have higher specific efficiencies than the current condenser designs, particularly higher than the condensers using EC motors with standard condenser surface which were generally found to have the lowest specific efficiency of all air-cooled condensers. Assuming micro-channel condensers become dominant in the market, the proposed condenser efficiency will potentially be met quite easily and at lower cost than the assumptions in this study.

This measure did not include evaluation of specific efficiency of closed-loop evaporative-cooled fluid coolers, air-cooled fluid coolers/dry-coolers, or open cooling towers. These designs would utilize an interconnecting water loop and water-cooled condensers. A review of Savings By Design projects indicates that these design choices are uncommon in California for new refrigerated warehouses. Typically, only special circumstances dictate the use of closed-loop fluid cooler systems, which are generally more expensive due to the additional heat exchanger and pumping equipment. Establishing a specific efficiency requirement for air-cooled and evaporative-cooled condensers without addressing fluid cooler systems, particularly at the proposed levels, is not expected to result in a change to fluid-coolers. In the future, particularly for smaller systems (i.e., those commonly using HFC) the need to reduce refrigerant leakage may result in more systems with fluid coolers. A performance method of compliance is potentially a more suitable method to address fluid coolers (as well as other design variables), and may be considered in a future code update.

The proposed standard includes two categories for evaporative condensers, distinguished by condenser size and location. Larger outdoor condensers have a higher minimum efficiency standard since these applications are always served with axial fan condensers and the systems are large enough that they do not employ multiple circuit condensers.

A lower minimum efficiency standard was established for outdoor condensers that are smaller (i.e., less than 8,000 MBH capacity) or condensers that are located indoors. The latter category allows for the fact that centrifugal-fan type condensers may need to be used in outdoor locations for some systems since cost-effective axial fan condenser selections may not be available in small sizes and are not generally available with multiple circuits; also, smaller facilities may be located in noise-sensitive areas requiring centrifugal fan condensers. In the event that an axial fan condenser can be used, the potential efficiency is higher.

Use of two efficiency requirements for the smaller condenser category based on type of condensers (i.e., axial or centrifugal fan type) was considered, but it was determined that this would increase complexity of the standard with little potential benefit. In fact, it could cause designers to choose less efficient centrifugal-fan condensers over axial condensers which are generally all more efficient, even without setting a code standard for small axial-fan condensers.

4.5 Screw Compressor Part-Load Performance

The 2008 Title 24 screw compressor part-load performance exemption to variable-speed control was examined to determine if the code can be simplified by changing to an application-based requirement. The current code requires variable-speed control on any suction group consisting of one screw compressor whose part-load power is greater than 60 percent of full-load power at 50 percent capacity. This requirement has proven to be controversial within the industry since many compressors and applications fall very close to this value, and because the subject compressor capacity ratings are not necessarily based on a published rating standard and no manufacturer ratings are certified. The industry has further questioned the wisdom of an exacting part-load performance value when the full-load performance is not specified (i.e., a particular compressor might have a better part load ratio but a lower full-load efficiency) and noted that the accuracy of part-load performance factors in software are often generalized and in any event much less stringent than the advertised full load performance. It should also be noted that field measurements of refrigerant mass flows are essentially non-existent; no body of field test data or independent lab data exists on industrial refrigeration compressor mass flow at part load.

Prototype Warehouse #1 was used to evaluate this measure, with base case alterations to the suction groups to facilitate evaluation of three applications: a single-compressor low-temperature suction group, a single-compressor medium-temperature suction group, and a single-compressor low-temperature booster suction group that discharges into a medium-temperature suction group. In each case, the compressors were assumed to be controlled with a fixed SST set point control strategy with a 1°F throttling range. For the booster evaluation, the intercooler was simulated in DOE-2.2R using a combination of subcooler and intercooler code words to account for both the saturated liquid supplied to the low-temperature evaporator coils, and the desuperheating requirements in the intercooler to cool booster gas flow to the high-stage compressors.

DOE-2.2R uses part-load performance curves to calculate compressor power and capacity at part-load conditions. For this analysis, part-load curves were updated to reflect the blended-average of major manufacturer screw compressor part-load performances, based on published ratings (from manufacturer product selection software). Slide valve capacity control was assumed in the base case, while variable-speed capacity control was used in the proposed case. For the variable-speed case, the part-load profiles captured the compressors' performance down to the manufacturer-specified minimum speed before unloading via slide valve. The variable-speed case includes 4 percent variable speed drive losses, with 2 percent assumed to be fixed and 2 percent assumed to be variable with drive output power.

Figure 20 shows the simulated part-load performance curves for both the base case and the proposed case.

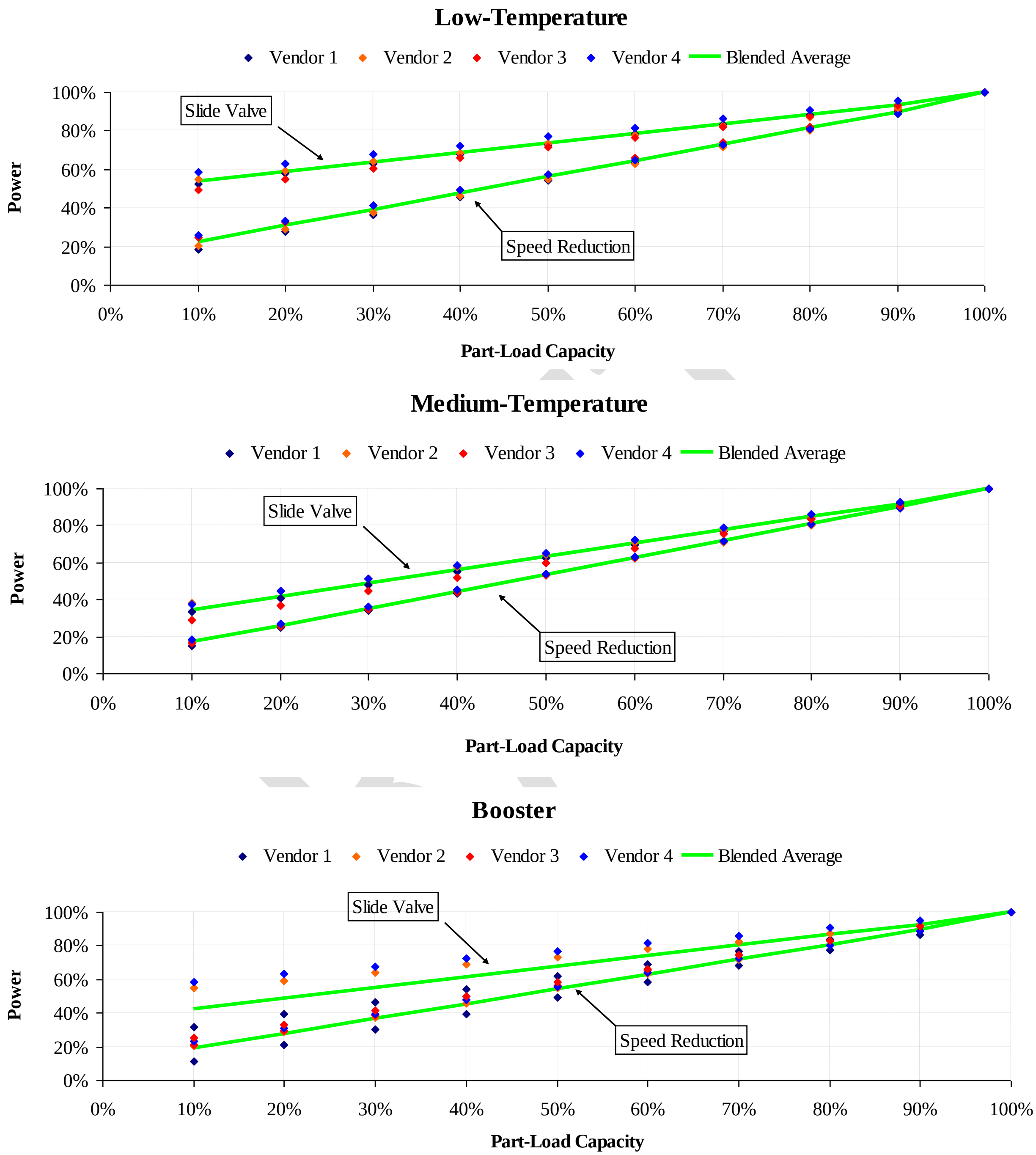


Figure 20: Part-load performance curves for slide valve and variable-speed control

The refrigeration load calculations and compressor selections were consistent with common industry practice and included typical safety factors. The observed part-load fraction in this simulation was consistent with observations made in actual installations (which are few due to the small number of these particular systems having one compressor per suction group).

4.5.1 Compressor Variable Speed Control Analysis Results by Climate Zone

Figure 21 summarizes the results for the analysis described above.

Compressor VFD Results	Energy Savings (kWh)	Energy Savings (kWh/ft2)	TDV Cost Savings ($)	TDV Cost Savings /SF ($)	Measure Cost ($)	Measure Cost ($/ft2)	Benefit/ Cost Ratio	LCC ($)	LCC ($/ft2)
CTZ03 Oakland									
LT System	300,672	3.27	$544,500	$5.92	$45,545	$0.50	12.0	-$498,955	-$5.42
MT System	106,965	1.16	$193,826	$2.11	$39,032	$0.42	5.0	-$154,794	-$1.68
Booster System	70,621	0.77	$130,914	$1.42	$32,318	$0.35	4.1	-$98,596	-$1.07
CTZ05 Santa Maria									
LT System	299,518	3.26	$544,260	$5.92	$45,545	$0.50	11.9	-$498,715	-$5.42
MT System	106,896	1.16	$197,359	$2.15	$39,032	$0.42	5.1	-$158,327	-$1.72
Booster System	70,010	0.76	$128,974	$1.40	$32,318	$0.35	4.0	-$96,656	-$1.05
CTZ07 San Diego									
LT System	303,381	3.30	$555,669	$6.04	$45,545	$0.50	12.2	-$510,124	-$5.54
MT System	116,026	1.26	$212,613	$2.31	$39,032	$0.42	5.4	-$173,581	-$1.89
Booster System	70,569	0.77	$132,027	$1.44	$32,318	$0.35	4.1	-$99,709	-$1.08
CTZ10 Riverside									
LT System	382,815	4.16	$695,848	$7.56	$45,545	$0.50	15.3	-$650,303	-$7.07
MT System	125,169	1.36	$230,377	$2.50	$39,032	$0.42	5.9	-$191,345	-$2.08
Booster System	67,365	0.73	$124,266	$1.35	$32,318	$0.35	3.8	-$91,948	-$1.00
CTZ 12 Sacramento									
LT System	396,834	4.31	$741,539	$8.06	$45,545	$0.50	16.3	-$695,994	-$7.57
MT System	128,130	1.39	$244,581	$2.66	$39,032	$0.42	6.3	-$205,549	-$2.23
Booster System	77,800	0.85	$142,600	$1.55	$32,318	$0.35	4.4	-$110,281	-$1.20
CTZ13 Fresno									
LT System	403,101	4.38	$750,029	$8.15	$45,545	$0.50	16.5	-$704,484	-$7.66
MT System	131,392	1.43	$247,224	$2.69	$39,032	$0.42	6.3	-$208,192	-$2.26
Booster System	77,776	0.85	$142,457	$1.55	$32,318	$0.35	4.4	-$110,139	-$1.20
CTZ14 Palmdale									
LT System	372,909	4.05	$680,247	$7.39	$45,545	$0.50	14.9	-$634,702	-$6.90
MT System	119,228	1.30	$218,051	$2.37	$39,032	$0.42	5.6	-$179,019	-$1.95
Booster System	67,019	0.73	$122,486	$1.33	$32,318	$0.35	3.8	-$90,168	-$0.98

Figure 21: Savings analysis results for screw compressor variable speed measure

4.5.2 Compressor Size Sensitivity Analysis

Since the subject compressor in this analysis was running continuously at part load, the average part-load ratio was a direct result of how closely the compressor could be sized to the peak load requirement. The above analysis used typical load calculations and selection practice. A smaller compressor would change the savings, potentially changing the conclusions.

To address this variability, a sensitivity analysis was performed for this measure to quantify the effect of a relatively smaller compressor by incrementally reducing the compressor size until the compressor's highest-loaded hour of the year equaled 100 percent loading (i.e., the smallest compressor with perfect understanding of the maximum cooling load). Figure 22 shows the results of the sensitivity analysis.

	Application	Loading (%)			Energy Savings (kWh)	Energy Savings (kWh/ft^2)	TDV Cost Savings ($)	TDV Cost Savings ($\$/ft^2$)	Measure Cost ($)	Benefit/ Cost Ratio
		Min.	Max.	Avg.						
CTZ12 Sacramento	LT	30%	97%	62%	102,484	3.9	$164,902	$6.3	$32,239	5.1
	MT	23%	100%	54%	39,132	1.5	$60,135	$2.3	$27,956	2.2
	Booster	33%	98%	67%	17,530	0.19	$27,224	$0.30	$24,244	1.1
CTZ05 Santa Maria	LT	37%	101%	76%	42,546	1.6	$57,723	$2.2	$27,956	2.1
	MT	23%	100%	51%	43,383	0.47	$69,889	$0.76	$27,956	2.5
	Booster	37%	101%	75%	9,798	0.11	$13,252	$0.14	$23,673	0.6
CTZ10 Riverside	LT	34%	100%	69%	73,496	2.8	$106,298	$4.1	$32,239	3.3
	MT	20%	100%	46%	59,602	0.65	$92,459	$1.0	$29,383	3.2
	Booster	36%	101%	74%	10,509	0.11	$13,679	$0.15	$24,244	0.6

Figure 22: Sensitivity analysis of screw compressor variable-speed measure

The sizing sensitivity analysis shows that variable-speed capacity control was cost-justified for single compressors on low temperature and medium temperature suction groups even if the compressor was sized exactly to the peak load. However, the booster application would not be cost-effective. Moreover, booster compressors have been observed in the field to be operating at somewhat higher average load fractions, although this may be coincidental.

These results support an application-based requirement for variable speed, which will potentially require cost-effective variable speed control in certain applications that previously may have met the requirements of the exception. The number of installations that utilize one screw compressors on a suction group is thought to be quite small, with only a handful identified over several years of Savings By Design new construction projects.

The proposed code change is to require variable-speed capacity control for open-drive screw compressors that are applied as one compressor on a suction group with design saturated suction temperature below 28°F and that discharge to the condenser pressure.

4.6 Infiltration Barriers

Infiltration barriers on passageways between internal spaces were evaluated for the purpose of establishing a minimum infiltration barrier requirement. There are no previous requirements for this measure in the 2008 Title 24 code. In practice, infiltration barriers are selected based on a variety of criteria, including opening height (e.g., 14 ft. fork trucks for tall racking) or width, frequency of doorway passages, hours and nature of facility operations, product type vs. suitability of door closures opening, and other factors.

No statewide savings or cost-effectiveness calculations were performed for this measure due to the extremely variable nature of door operations and the practical infiltration barriers that could be appropriate for each passageway.

Both small and large warehouse prototypes were used to evaluate infiltration barriers between the -0°F freezer and both the refrigerated loading dock (Prototype Warehouses #1 and #3) and the partially conditioned warehouse (Prototype Warehouses #2 and #4). Barriers were also simulated between the 35°F cooler and the partially conditioned warehouse. Certain barrier types were simulated with varying traffic levels, reflecting the wide variation of traffic that can occur in different facilities. Figure 23 shows the range of infiltration barriers evaluated for this measure.

Run	Infiltration barrier type	Assumed traffic rate (passages/hour)	Door open-time per passage (seconds)	Resultant percent open time
1	Wide-open door (base case)	NA	NA	100%
2	Wide-open door with strip curtains	NA	NA	100%
3	Standard-speed automatic-closing door	5	13	2%
4		80	13	29%
5		160	13	58%
6	High-speed automatic-closing door	5	10	1%
7		80	10	22%
8		160	10	44%
9	High-speed automatic-closing door with strip curtains	5	10	1%
10		80	10	22%
11		160	10	44%
12	High-speed automatic-closing door with air curtains	5	10	1%
13		80	10	22%
14		160	10	44%
15	Wide-open door with air curtains	NA	NA	100%

Figure 23: Summary of simulation runs for infiltration barrier measure

Figure 24 summarizes the assumptions made for strip curtain and air curtain effectiveness and fan power.

Assumed strip curtain effectiveness[4]	85% (vs. wide-open door,)
Assumed air curtain effectiveness[5]	75% (vs. wide-open door)
Air curtain fan power assumptions	(3) 1/2 HP fans per air curtain

Figure 24: Summary of assumptions for infiltration barrier measure

DOE-2.2R analysis software is capable of modeling both the latent and sensible heat exchange associated with density-driven convection through door openings between two spaces. The convection model is based on the equations presented in the “2006 ASHRAE Handbook – Refrigeration,” pp. 13.4-13.6. Weather is not a factor in this analysis, since only the exchange of air between refrigerated spaces is being considered. Therefore, the infiltration barrier measure was evaluated in only one climate zone, CTZ12 (Sacramento Executive Airport), which is a reasonable average for annual system efficiency. Also, the extremely high cost effectiveness of infiltration barriers as compared to a wide-open door does not warrant analysis in multiple climate zones.

Figure 25 shows the analysis results of simulating wide-open doors and incremental cost-effectiveness results for manually-operated doors and strip curtains. The analysis was performed to establish the cost-effectiveness of a minimally-compliant infiltration barrier and does not imply that the wide-open door scenario reflects a typical practice base case.

[4] ASHRAE Refrigeration Handbook, 2010. p.23.13
[5] Ibid

		Electric Energy Savings (kWh)	TDV Energy Savings (MMBtu)	TDV Cost Savings ($)	Measure Cost ($)	Benefit/Cost Ratio
Small Warehouse	*35°F Cooler to Partially-Conditioned Warehouse*					
	Manual Door	156,285	3,599	$320,255	$2,695	119
	Strip Curtains	144,571	3,274	$291,331	$954	305
	-10°F Freezer to Partially-Conditioned Warehouse					
	Manual Door	546,389	12,907	$1,148,717	$2,695	426
	Strip Curtains	461,458	10,745	$956,262	$954	1002
	-10°F Freezer to 40°F Refrigerated Dock					
	Manual Door	155,982	4,046	$360,063	$2,695	134
	Strip Curtains	127,529	3,336	$296,902	$954	311
Large Warehouse	*35°F Cooler to Partially-Conditioned Warehouse*					
	Manual Door	237,942	5,752	$511,892	$5,390	95
	Strip Curtains	218,929	5,179	$460,941	$1,908	242
	-10°F Freezer to Partially-Conditioned Warehouse					
	Manual Door	934,551	21,149	$1,882,175	$5,390	349
	Strip Curtains	680,209	14,886	$1,324,780	$1,908	694
	-10°F Freezer to 40°F Refrigerated Dock					
	Manual Door	426,298	10,056	$894,925	$5,390	166
	Strip Curtains	357,041	7,639	$679,855	$1,908	356

Figure 25: Manual door and strip curtain savings and cost-effectiveness analysis results

Figure 25 compares manual doors or strip curtains to wide-open doors. Both are obviously cost-effective on the basis of energy savings. Again, this analysis was performed to establish a minimally-compliant infiltration barrier, and is not intended to imply that wide-open doors are common in refrigerated warehouses or that wide-open doors are an appropriate basis of savings calculations for infiltration barriers. The cooling system often will not maintain design temperatures with wide-open doors; refrigeration loads normally assume some form of infiltration barrier.

Figure 26 compares energy use for a variety of infiltration barrier options to strip curtains. Some options are evaluated at multiple traffic levels and are not directly comparable. Also, as noted previously, the savings are for comparison to ASHRAE strip curtain effectiveness which is not adjusted for variations in traffic.

Infiltration Barrier Type	Passages Per Hour	Energy Savings Relative to Strip Curtains (kWh): Large Warehouse, Freezer to 40°F Dock	Large Warehouse, Cooler (Opens to Conditioned Warehouse)	Large Warehouse, Freezer (Opens to Conditioned Warehouse)	Small Warehouse, Freezer to 40°F Dock	Small Warehouse, Cooler (Opens to Conditioned Warehouse)	Small Warehouse, Freezer (Opens to Conditioned Warehouse)
Standard-speed automatic-closing door	5	79,664	25,870	300,596	38,571	15,978	120,684
	80	(186,921)	(25,561)	(214,304)	(37,529)	(17,462)	(158,679)
	160	(291,023)	(94,860)	(557,882)	(92,556)	(75,696)	(387,206)
High-speed automatic-closing door	5	80,533	26,847	305,543	39,906	16,557	125,800
	80	(101,533)	(13,232)	(140,247)	(19,515)	(8,475)	(74,300)
	160	(254,211)	(54,417)	(437,929)	(72,744)	(46,167)	(324,053)
High-speed automatic-closing door with strip curtains	5	82,698	29,530	319,738	43,686	18,321	141,042
	80	75,600	22,633	281,116	33,710	13,911	103,005
	160	61,959	15,518	230,973	23,371	9,576	70,274
High-speed automatic-closing door with air curtains	5	81,970	28,720	316,874	42,831	17,841	138,149
	80	56,341	10,506	225,804	20,325	6,946	67,600
	160	11,426	(7,700)	42,621	(1,717)	(3,599)	8,631
Wide-open door with air curtains	NA	(223,449)	(52,520)	(251,038)	(55,039)	(31,754)	(184,726)

Figure 26: Energy savings for various infiltration barriers relative to strip curtains

As stated above, infiltration barrier types are selected based on a multitude of criteria—including but not limited to energy efficiency. The results in Figure 26 are not all directly comparable; results are presented here for informative purposes only. Since stakeholders generally agree that a strip curtain is already the minimum infiltration barrier and because cost analysis shows that it is least capital cost, strip curtains are recommended as the minimum for the standard, with exceptions allowing other methods which may be superior or necessary in that strip curtains are not feasible in certain applications for a variety of reasons.

The proposed code addition requires that freezers opening to a higher temperature space and coolers opening to a non-refrigerated space must have strip curtains installed, with several exceptions including automatically-closing swing doors; automatically-operated horizontal or vertical doors (meaning the door must close under its own power, but operation may be enabled manually such as with a pull-cord); and use of an air curtain designed by its manufacturer for use on the subject door application and operating temperature.

4.7 Allow Air-Cooled Ammonia Condensers

Air-cooled ammonia condensers on refrigerated warehouses were prohibited in the 2008 standards. The use of ammonia was considered to be synonymous with large systems; it was assumed and undoubtedly met with stakeholder agreement that evaporative-cooled condensing was automatically more efficient than air-cooled condensing. The requirement was reportedly a way of saying that all large refrigeration systems should be evaporative cooled. Of course, not all large refrigerated warehouse systems are ammonia systems. There are locations where ammonia is not desirable or feasible, and the new construction project may be an expansion of an existing non-ammonia system. The current code has no requirements on refrigerant type and no requirements on the method of condensing, except in this situation.

There are systems that must be air-cooled due to cost and/or availability of water, or because water in some areas is very difficult to treat and results in rapid condenser fouling and frequent need for replacement. The result of the existing requirement would be to force owners to utilize an HFC refrigerant with an air-cooled system. This would be counter-productive in that ammonia is generally more efficient than HFC refrigerants, even in systems operating at high condensing temperatures.

There are few air-cooled ammonia systems in California (none are known through the new construction program). Air-cooled ammonia condensers have a special design item since ammonia is incompatible with the copper tubing used for halocarbon refrigerants. Recently, at least two U.S. manufacturers have begun to offer air-cooled ammonia condensers and one large company has built a grocery distribution center with air-cooled ammonia condensers.

Figure 27 below illustrates the comparative efficiency of ammonia versus HFC refrigerants by showing the performance of an example 300 hp screw compressor in kW per ton refrigeration (TR).

	NH_3			R-507		
Condition (SST/SCT)	**TR**	**HP**	**kW/TR**	**TR**	**HP**	**kW/TR**
-20°F /100°F	76.8	193.8	1.982	70.6	233.7	2.60
20°F /100°F	203.3	253.7	0.980	179.3	282.8	1.24

Figure 27: Typical screw compressor performance with ammonia and HFC refrigerant

The subject compressor is more efficient with ammonia than with R-507 in both a low-temperature and a medium-temperature application, confirming the benefit of allowing the use of ammonia if the design includes air-cooled condensing.

The proposed change to the existing code language removes the requirement that ammonia systems utilize only evaporative-cooled condensers.

4.7.1 Informational Air-Cooled vs. Evaporative-Cooled Study

Air-cooled condensing is highly intriguing to the refrigeration industry. Water conservation is very important in California and is becoming a high priority for many refrigerated warehouse and food plant operators, even affecting the viability of new and existing operations.

As an additional information study concurrent with the CASE work, a comparison of air-cooled and evaporative-cooled condensing on an ammonia system for the large warehouse prototype was conducted in several climate zones. The analysis considered total operating cost using typical time-of-use electric rates, water costs and water-treatment costs, along with the additional capital costs for air-cooled condensers rather than evaporative-cooled condensing. The study found that air-cooled condensing can be financially attractive in cool climates. Results are presented in Appendix G.

4.8 Acceptance Tests

Energy savings associated with acceptance tests were assumed to be captured in the 2008 CASE analysis for refrigerated warehouses since the measures were evaluated as commissioned. The cost effectiveness of the evaporator and condenser acceptance tests were evaluated by adding the cost of the acceptance test to the 2008 measure cost and conducting BC ratio and LCC calculations using the 2008 TDV values. Since this CASE study examined the screw compressor variable speed measure, the cost of the screw compressor acceptance test was added to the measure incremental cost and LCC analysis was completed using 2013 TDV values. Assumptions for labor hours were gathered by survey input and field testing of the protocol. Survey results are presented in Appendix D.

Cost assumptions for the acceptance test labor are as follows:

- Eight hours per person to conduct the acceptance test at one site (one technician and one engineer required, 16 hours total).
- Eight hours per test, with two completions of the acceptance test to ensure all systems are operational (16 hours total).
- Ten minutes per evaporator, for two people, in addition to the 8 hours already specified, to complete the evaporator acceptance test for 15 evaporators for the small warehouse and 50 evaporators for the large warehouse (conservative cost assumption).
- Two hours per person to coordinate the site visit for two people (four hours total).
- Two hours each way for two people, two trips (16 hours of travel time total).
- Six hours of paperwork.
- $150/hr labor cost (average of one engineer and one technician).
- Cost per site for acceptance test, not including calibration = $13,900 for small warehouse and $15,700 for large warehouse (numbers rounded to nearest 100).

Cost assumptions for the biannual instrument standard calibration costs brought to PV:

- All instruments calibrated the first year (2013) and every two years thereafter consist of two temperature instruments, two pressure instruments, and one humidity instrument.
- Temperature calibration costs = $120/sensor.
- Pressure calibration costs = $100/sensor.
- Humidity calibration costs = $400/sensor.
- Four hours to pack and ship, and to track calibration due dates.
- Present value = cost * $(1/1+d)^{\wedge n}$, d= 0.03, n = year number
- PV of calibration cost is $4700.

Total Cost of acceptance test per site = $18,600 for small warehouse and $20,400 for large warehouse (numbers rounded to nearest $100).

Cost for condenser acceptance test and compressor acceptance test is assumed to be one third of total cost (excluding extra cost for evaporators) ~ $4,500. This is cost added to the incremental cost of the measure.

Cost for evaporator acceptance test in small warehouses ~ $5,200 and cost for evaporator acceptance test in large warehouses ~ $7000.

Results of the cost analysis of acceptance tests in shown in Figure 28. Results are the average of all climate zones and of small and large warehouses.

Acceptance Test Cost Results	TDV Cost Savings ($/ft²)	Measure Cost ($/ft²)	Benefit/ Cost	LCC ($/ft²)
evaporator fan control	$6.16	$0.45	14	-$5.72
air-cooled condenser control	$7.17	$1.62	4.4	-$5.55
evaporative -cooled condenser control	$5.82	$0.52	4.4	-$1.79
compressor speed control	$4.72	$0.46	10	-$4.26

Figure 28: Acceptance test cost analysis results

4.9 Code Language Changes Not Analyzed

A number of changes were made to the code language that was not based on analysis of cost-effectiveness. These code changes constitute clarifications to the intent of the language, close possible loopholes in the 2008 language, or allow for design innovation that would otherwise be prohibited. The changes were vetted during the industry stakeholder meeting process with no disagreement.

1. The code language stating:

> A refrigerated warehouse with total cold storage and frozen storage area of 3,000 square feet or larger shall meet the requirements of this section

Was changed to:

> Enclosed spaces greater than 3,000 square feet with operating temperatures less than 55°F shall satisfy subsections (a), (b), (c), (f) and (g) of Section 126. Refrigeration systems (compressors and condensers) serving a total of 3,000 square feet or more of cold storage space, even if individual spaces served by the system are all less than 3,000 square feet, shall satisfy subsections (d), (e) and (g) of Section 126.
>
> An enclosed space with an area less than 3,000 square feet with an operating temperature less than 55°F shall meet the space requirements of the Appliance Efficiency Regulations for walk-in refrigerators or freezers (California Code of Regulations, Title 20, Sections 1601 through 1608).

This clarification was made to include refrigeration systems that do not serve any individual space greater than 3,000 square feet, but serve a sum total of more than 3,000 square feet of refrigerated space, which were otherwise exempted.

2. Cold storage was changed to cooler and minimum temperature changed from 32°F to 28°F. Frozen storage was changed to freezer and maximum temp was changed from 32°F to 28°F.

The change to “cooler” and “freezer” designation is more consistent with industry terminology, where “cold storage” is often understood to mean freezer temperatures. The change from 32°F to 28°F to distinguish between coolers and freezers was made because the design temperature for storage of meat, fish and deli products is very frequently below 32°F, but is rarely less than 28°F. Refrigerated warehouses rarely have spaces designed between 28°F and 5°F. The proposed standards for freezer insulation would not be cost effective at the higher temperatures, so this change is necessary for the proposed freezer insulation change as well as improving the application of the existing code requirements.

3. EXCEPTIONS 1 and 2 to Section 126 are taken from the compliance manual, and were added to the code language for clarification.

4. EXCEPTIONS 1 and 2 to Section 126 (a) were added to clarify the intended building type for

the mandates described in Section 126. Refrigeration systems that serve process loads are subject to different design criteria, and should therefore be exempted from Section 126 requirements. This exemption is described in the 2008 compliance manual.

5. Table 126-A has new freezer floor criteria of R-20 for floors that have all underslab heating provided by productive cooling. If the heat transfer through the floor insulation is replaced in a manner that creates productive refrigeration, there is no net load on the system. This fact does not mean that no insulation is either feasible or prudent. Allowing the designer to select less insulation (as low as R-20) provides sufficient cost-savings to encourage systems that achieve floor heating and concurrent productive cooling.

6. Section 126(c) 2 verbiage was changed from "…speed shall be controlled in response to space *conditions*" to "…speed shall be controlled in response to space *temperature or humidity*." The change clarifies the intended control parameter for air unit (evaporator coil) fan speed control strategy.

7. EXCEPTION to Section 126(d) 2 was amended to include a horsepower threshold for exemption, rather than simply stating that "unitary" systems are exempted from this requirement. The 2008 code intended to exempt small packaged condensing units (consisting of a compressor and a condenser in one package), but the code could be interpreted to include much larger systems that happen to be mounted to a common chassis (and are thus "unitized"), that should be required to comply with the code. Exception was amended to exempt chillers.

8. Section 126(d)5 was added to explicitly mandate ambient-following controls for evaporative-cooled condensers, the intent of the 2008 code analysis but technically exempted by the 2008 code. The subsequent exemption was also added, providing a path for implementation of control strategies that are better than ambient-following.

9. Section 126(d) 7 was added concurrent with the condenser specific efficiency minimum requirements. The minimum requirements could be met with a condenser with very close fin spacing, but the condenser would quickly foul with dirt and contaminants. This change was discussed in the stakeholder meetings with no disagreement. Condensers with a micro-channel exchange surface are subsequently exempted from this requirement, because the micro-channel surface is not as susceptible to permanent fouling in the same manner as that in traditional tube-and-fin condensers with tight fin spacing.

10. The compressor minimum nominal horsepower criteria in Section 126(e)2 was changed to specifically state open-drive, rather than give a size with the intent to exempt semi-hermetic compressors which can cycle rapidly to modulate capacity and not benefit from speed control.

11. Variable Vi control (the ability to automatically vary the compressor volume ratio) was added in Section 126(e) 3 to align the code with the 2008 Compliance Manual. Market research was conducted, and findings indicate that the most prominent screw compressor manufacturers already offer variable Vi control as a standard feature.

5. Recommended Code Language

Section 5 presents the proposed code language changes to Title 24, section 126 and Refrigerated Warehouse Acceptance Test addition to Non-residential Appendix NA7. New proposed language is underlined and proposed deletions to 2008 code are in strikeouts.

5.1 Title 24 Draft Code Language

SECTION 101 – DEFINITIONS

BUBBLE POINT is refrigerant liquid saturation temperature at a specified pressure.

CONDENSER SPECIFIC EFFICIENCY is the Total Heat of Rejection (THR) capacity divided by the fan input electric power at 100 percent fan speed (including spray pump electric input power for evaporative condensers).

COOLER is space greater than or equal to 28°F but less than 55°F.

CONTROLLED ATMOSPHERE describes a cooler designed to be airtight and maintained at reduced oxygen levels for the purpose of reducing respiration of perishable product in long term storage.

DEW POINT is refrigerant vapor saturation temperature at a specified pressure.

FREEZER is space designed to maintain less than 28°F and space designed to be convertible between cooler and freezer operation.

FLUID COOLER is a fan-powered heat rejection device that includes a water circuit connected by a closed circulation loop to a water-cooled refrigerant condenser, and may be either evaporative-cooled or air-cooled.

MICRO-CHANNEL CONDENSER is an air-cooled condenser for refrigeration systems which utilizes multiple small parallel gas flow passages in a flat configuration with unitized fin surface between the gas passages, rather than round tubes arranged at a right angle to separate plate fins.

SATURATED CONDENSING TEMPERATURE (CONDENSING TEMPERATURE). For single component and azeotropic refrigerants, the saturation temperature corresponding to the refrigerant pressure at the condenser entrance. For zeotropic refrigerants, the arithmetic average of the Dew Point and Bubble Point temperatures corresponding to the refrigerant pressure at the condenser entrance.

~~**STORAGE, COLD,** is a storage area within a refrigerated warehouse where space temperatures are maintained at or above 32° F.~~

~~**STORAGE, FROZEN** is a storage area within a refrigerated warehouse where the space temperatures are maintained below 32° F.~~

TOTAL HEAT OF REJECTION (THR) is the heat rejected by refrigeration system compressors at design conditions, consisting of the design cooling capacity plus the heat of compression added by the compressors.

SECTION 126 – MANDATORY REQUIREMENTS FOR REFRIGERATED WAREHOUSES

~~A refrigerated warehouse with total cold storage and frozen storage area of 3,000 square feet or larger shall meet the requirements of this section.~~

Enclosed spaces greater than 3,000 square feet with operating temperatures less than 55°F shall satisfy subsections (a), (b), (c), (f) and (g) of Section 126. Refrigeration systems (compressors and condensers) serving a total of 3,000 square feet or more of cold storage space, even if individual spaces served by the system are all less than 3,000 square feet, shall satisfy subsections (d), (e) and (g) of Section 126.

An enclosed space with an area less than 3,000 square feet with an operating temperature less than 55°F shall meet the space requirements of the Appliance Efficiency Regulations for walk-in refrigerators or freezers (California Code of Regulations, Title 20, Sections 1601 through 1608).

~~**EXCEPTION 1 to Section 126:** A refrigerated space less than 3,000 square feet shall meet the Appliance Efficiency Regulations for walk-in refrigerators or freezers.~~

EXCEPTION 1 ~~2~~ to Section 126: Areas within refrigerated warehouses that are designed solely for the purpose of quick chilling or freezing of products with design cooling capacities of greater than 240 Btu/hr-ft^2 (2 tons per 100 ft^2).

EXCEPTION 2 to Section 126: Compressors and condensers on a refrigeration system, defined by a common refrigerant charge, whose design refrigeration cooling load from quick chilling or freezing of products (areas with design cooling capacities of greater than 240 Btu/hr-ft2) is more than 20 percent of the total design refrigeration system cooling load.

(a) **Insulation Requirements.** Exterior surfaces of refrigerated warehouses shall be insulated at least to the R-values in Table 126-A.

TABLE 126-A REFRIGERATED WAREHOUSE INSULATION

Space	**Surface**	**Minimum R-Value (°F Hr ft^2/Btu)**
Freezers ~~Frozen Storage~~	Roof/Ceiling	~~R-36~~ R-40
	Wall	R-36
	Floor	~~R-36~~ R-35
	Floor with all heating from productive refrigeration capacity*	R-20
Coolers ~~Cold Storage~~	Roof/Ceiling	R-28
	Wall	R-28
*If all underslab heating is provided by a heat exchanger that provides refrigerant subcooling or other means that result in productive refrigeration capacity on the associated refrigerated system.		

(b) **Underslab heating.** Electric resistance heat shall not be used for the purposes of underslab heating.

EXCEPTION to Section 126 (b): Underslab heating systems controlled such that the electric resistance heat is thermostatically controlled and disabled during the summer on-peak period defined by the local electric utility.

(c) **Evaporators.** Fan-powered evaporators used in coolers and freezers shall conform to the following:

1. Single phase fan motors less than 1 hp and less than 460 Volts shall be electronically commutated motors.

2. Evaporator fans served either by a suction group with multiple compressors, or by a single compressor with variable capacity capability shall be variable speed and the speed shall be controlled in response to space temperature or humidity. ~~conditions~~.

EXCEPTION to Section 126 (c) 2: ~~Evaporators served by a single compressor without unloading capability.~~ Coolers within refrigerated warehouses that maintain a Controlled Atmosphere long term storage for which a licensed engineer has certified that the types of products stored will require constant operation at 100 percent of the design airflow.

3. Evaporator fans served by a single compressor that does not have variable capacity shall utilize controls to reduce airflow by at least 40 percent for at least 75 percent of the time when the compressor is not running.

(d) **Condensers.** Fan-powered condensers shall conform to the following:

~~1. Condensers for systems utilizing ammonia shall be evaporatively cooled.~~

1. Design saturated condensing temperatures for evaporative-cooled condensers ~~including but not limited to~~ and water-cooled condensers served by fluid coolers or cooling towers shall be less than or equal to:
   A. ~~T~~the design wetbulb temperature plus 20°F in locations where the design wetbulb temperature is less than or equal to 76°F,
   B. ~~T~~the design wetbulb temperature plus 19°F in locations where the design wetbulb temperature is between 76°F and 78°F, or
   C. ~~T~~the design wetbulb temperature plus 18°F in locations were the design wetbulb temperature is greater than or equal to 78°F.

2. Design saturated condensing temperatures for air-cooled condensers ~~under design conditions~~ shall be less than or equal to the design drybulb temperature plus 10°F for systems serving ~~frozen storage~~ freezers and shall be less than or equal to the design drybulb temperature plus 15°F for systems serving coolers ~~cold~~ ~~storage~~.

   EXCEPTION to Section 126 (d) 2: ~~Unitary~~ Condensing units and chillers with a total compressor horsepower less than 100 HP.

3. All condenser fans for evaporative-cooled condensers or fans on cooling towers or fluid coolers shall be continuously variable speed, and the condensing temperature

control system shall control the speed of all ~~condenser~~ fans serving a common condenser ~~loop~~ high side in unison. The minimum condensing temperature set point shall be less than or equal to 70°F.

4. All condenser fans for air-cooled condensers shall be continuously variable speed and the condensing temperature or pressure control system shall control the speed of all condenser fans serving a common condenser high side in unison. The minimum condensing temperature set point shall be less than or equal to 70°F~~, or reset in response to ambient drybulb temperature or refrigeration system load~~.

5. Condensing temperature reset. The condensing temperature set point of systems served by air-cooled condensers shall be reset in response to ambient drybulb temperature. The condensing temperature set point of systems served by evaporative-cooled condensers or water-cooled condensers (via cooling towers or fluid coolers) shall be reset in response to ambient wetbulb temperatures.

   EXCEPTION to Section 126 (d) 5: Condensing temperature control strategies approved by the Executive Director that have been demonstrated to provide at least equal energy savings.

~~6. All single phase condenser fan motors less than 1 hp and less than 460 V shall be either permanent split capacitor or electronically commutated motors~~.

6. Fan-powered condensers shall meet the condenser efficiency requirements listed in Table 126-B. Condenser efficiency is defined as the Total Heat of Rejection (THR) capacity divided by all electrical input power including fan power at 100 percent fan speed, and power of spray pumps for evaporative condensers.

TABLE 126-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	
Outdoor Air-Cooled	Ammonia	75 Btuh/Watt	105°F Saturated Condensing Temperature (SCT), 95°F Outdoor Drybulb Temperature
	Halocarbon	65 Btuh/Watt	
Indoor Air-Cooled	All	Exempt	

7. Air-cooled condensers shall have a fin density no greater than 10 fins per inch.

EXCEPTION to Section 126 (d) 7: Micro-channel condensers.

(e) **Compressors**. Compressor systems utilized in refrigerated warehouses shall conform to the following:

1. Compressors shall be designed to operate at a minimum condensing temperature of 70°F or less.

2. ~~The compressor speed of~~ An open-drive screw compressor with a design saturated suction temperature (SST) of 28°F or lower that discharges to the system condenser pressure ~~greater than 50 hp~~ shall ~~be controllable~~ control compressor speed in response to the refrigeration load ~~or the input power to the compressor shall be controlled to be less than or equal to 60 percent of full load input power when operated at 50 percent of full refrigeration capacity~~.

   EXCEPTION 1 to Section 126 (e) 2: Refrigeration plants with more than one dedicated compressor per suction group.

3. Screw compressors with nominal electric motor power greater than 150 hp shall include the ability to automatically vary the compressor volume ratio (Vi) in response to operating pressures.

(f) **Infiltration Barriers.** Passageways between freezers and higher-temperature spaces, and passageways between coolers and non-refrigerated spaces, shall have an infiltration barrier consisting of strip curtains, an automatically-closing door, or an air curtain designed by its manufacturer for use in the passageway and temperature for which it is applied.

EXCEPTION 1 to Section 126 (f): Openings with less than 16 ft^2 of opening size.

EXCEPTION 2 to Section 126 (f): Dock doorways for trailers.

(g) **Refrigeration System Acceptance.** Before an occupancy permit is granted for a new refrigerated warehouse, or before a new refrigeration system serving a refrigerated warehouse is operated for normal use, the following equipment and systems shall be certified as meeting the Acceptance Requirements for Code Compliance, as specified by the Reference Nonresidential Appendix NA7. A Certificate of Acceptance shall be submitted to the enforcement agency that certifies that the equipment and systems meet the acceptance requirements:
   1. Electric resistance underslab heating systems shall be tested in accordance with NA 7.9.1.
   2. Evaporators fan motor controls shall be tested in accordance with NA 7.9.2.
   3. Evaporative condensers shall be tested in accordance with NA 7.9.3.1.
   4. Air-cooled condensers shall be tested in accordance with NA 7.9.3.2.
   5. Variable speed compressors shall be tested in accordance with NA 7.9.4.

5.2 Acceptance Test Language

All language is new. For ease of reading, new text is not underlined here.

NA7.9 Refrigeration Systems Acceptance Test

The measurement devices used to verify the refrigerated warehouse refrigeration system instruments will be calibrated once every two years using a NIST traceable reference. The calibrated instruments are called the standard in section NA7.9. The temperature standard is to be calibrated to +/- 0.7°F between -30°F and 200°F. The pressure standard is to be calibrated to +/- 2.5 psi between 0 and 500 psig. The relative humidity (RH) standard is to be calibrated to +/- 1% between 5% and 90% RH.

NA7.9.1 Electric Resistance Underslab Heating System

NA7.9.1.1 Construction Inspection

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

Verify that summer on-peak period is programmed into all underslab heater controls to meet the requirements of Section 126(b).

NA7.9.1.2 Functional Testing

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

- The underslab electric resistance heater is off.

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

- 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:

- The underslab electric resistance heater is off.

Step 4: Restore system to correct schedule and control set points.

NA7.9.2 Evaporators and Evaporator Fan Motor Variable Speed Control

NA7.9.2.1 Construction Inspection

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

- All refrigerated space temperature sensors used for control are verified to read accurately (or provide an appropriate offset) using a temperature standard.
- All refrigerated space humidity sensors used for control are verified to read accurately (or provide an appropriate offset) using a humidity standard.
- All refrigerated space temperature and humidity sensors are verified to be mounted in a location away from direct evaporator discharge air draft.
- Verify that all fans motors are operational and rotating in the correct direction.
- Verify that fan speed control is operational and connected to evaporator fan motors.
- Verify that all speed controls are in “auto” mode.

NA7.9.2.2 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 set point in 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 set point in 1 degree or 1% RH increments below any control dead band range until:

- Evaporator fan controls modulate to increase fan motor speed.
- Evaporator fan motor speed increases in response to controls.

Verify and document the above.

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

- Evaporator fan controls modulate to decrease fan motor speed.
- Evaporator fan motor speed decreases in response to controls.
- Minimum fan motor control speed (rpm or percent of full speed).

Step 4: Restore control system to correct control set points.

NA7.9.3 Condensers and Condenser Fan Motor Variable Speed Control

NA7.9.3.1 Evaporative Condensers and Condenser Fan Motor Variable Speed Control

NA7.9.3.1.1 Construction Inspection

Prior to functional testing, document the following:

- Verify the minimum condensing temperature control set point is at or below 70°F.
- Verify the master system controller saturated condensing temperature input is the temperature equivalent reading of the condenser pressure sensor.
- Verify all drain leg pressure regulator valves are set below the minimum condensing temperature/pressure set point.
- Verify all receiver pressurization valves, such as the outlet pressure regulator (OPR), are set lower than the drain leg pressure regulator valve setting.
- Verify all condenser inlet and outlet pressure sensors read accurately (or provide an appropriate offset) using a pressure standard.
- Verify all ambient dry bulb temperature sensors used by controller read accurately (or provide an appropriate offset) using a temperature standard.
- Verify all relative humidity sensor used by controller read accurately (or provide an appropriate offset) using RH standard.
- Verify all temperature sensors used by the controller are mounted in a location that is not exposed to direct sunlight.
- 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.)
- Verify that all fan motors are operational and rotating in the correct direction.
- 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.
- Verify that all speed controls are in “auto” mode.

NA7.9.3.1.2 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.

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

Step 2:

- 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.
- Calculate and document the temperature difference (TD), defined as the difference between the wet bulb temperature and the refrigeration system saturated condensing temperature (SCT).
- Document current head pressure control set point.

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

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

- Fan motor speed decreases.
- All condenser fan motors serving common condenser loop decrease speed in unison in response to controller output.
- Minimum fan motor control speed (rpm or percent of full speed).

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 set point in 1 degree (or 3 psi) increments until the condenser fan control modulates to increase fan motor speed. Verify and document the following:

- Fan motor speed increases.
- 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 set point. Using the control system, change the minimum condensing temperature set point to a value greater than the current operating condensing temperature. Verify and document the following:

- Condenser fan controls modulate to decrease capacity.
- All condenser fans serving common condenser loop modulate in unison.
- 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 set point 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 set point to the value documented in Step #6.

NA7.9.3.2 Air-Cooled Condensers and Condenser Fan Motor Variable Speed Control
Refrigerated warehouses with air-cooled condensers shall perform the inspections in 7.9.3.2.

NA7.9.3.2.1 Construction Inspection

Prior to functional testing, document the following:

- Verify that the minimum condensing temperature control set point is at or below 70°F.
- Verify that the master system controller saturated condensing temperature input is the temperature equivalent reading of the condenser pressure sensor.
- Verify all drain leg pressure regulator valves are set below the minimum condensing temperature/pressure set point.
- Verify all receiver pressurization valves, such as the outlet pressure regulator (OPR), are set lower than the drain leg pressure regulator valve setting.
- Verify all condenser inlet and outlet pressure sensors read accurately (or provide an appropriate offset) using a pressure standard.
- Verify all ambient dry bulb temperature sensors used by controller read accurately (or provide an appropriate offset) using temperature standard.
- Verify all temperature sensors used by the controller are mounted in a location that is not exposed to direct sunlight.
- 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.)
- Verify that all fan motors are operational and rotating in the correct direction.
- 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.
- Verify that all speed controls are in “auto” mode.

NA7.9.3.1.1 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.

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

Step 2:

- Document current outdoor ambient air dry bulb temperature and refrigeration system condensing temperature/condensing pressure readings from the control system.
- 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 set point.

Step 3: Using the desired condenser fan motor cycling or head pressure control strategy, program into the control system a set point equal to the reading or calculation obtained in Step 2.

This will be referred to as the "test set point." Allow 5 minutes for condenser fan speed to normalize.

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

- Fan motor speed decreases.
- All condenser fan motors serving common condenser loop decrease speed in unison in response to controller output.
- Minimum fan motor control speed (rpm or percent of full speed).

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 set point in 1 degree (or 3 psi) increments until the condenser fan control modulates to increase fan motor speed. Verify and document the following:

- Fan motor speed increases.
- All condenser fan motors serving common condenser loop increase speed in unison in response to controller output.

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

- Condenser fan controls modulate to decrease capacity.
- All condenser fans serving common condenser loop modulate in unison.
- 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 set point 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 set point to the value documented in Step #6.

NA7.9.4 Variable Speed Screw Compressors

Refrigerated warehouses with variable-speed screw compressors shall perform the inspections in 7.9.4.

NA7.9.4.1 Construction Inspection

Prior to functional testing, document the following:

- Verify all single open-drive screw compressors dedicated to a suction group have variable speed control.
- Verify all compressor suction and discharge pressure sensors read accurately (or provide an appropriate offset) using a standard.
- Verify all input or control temperature sensors used by controller read accurately (or provide an appropriate offset) using temperature standard.
- 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.)
- Verify that all compressor speed controls are operational and connected to compressor motors.

- Verify that all speed controls are in “auto” mode.
- Verify that compressor panel control readings for “RPMs”, “% speed”, “kW”, and “amps” match the readings from the PLC or other control systems.
- Verify that compressor nameplate data is correctly entered into the PLC or other control system.

NA7.9.4.2 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 set points, etc.) as may be required to perform the Functional Testing.

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 set point. Program into the control system a target set point 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 set point”.

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

- Compressor speed decreases.
- Compressor speed continues to decrease to minimum speed.
- 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 set point in 1 psi increments until the compressor controller modulates to increase compressor speed. Verify and document the following:

- Any slide valve or other unloading means first goes to 100 percent before compressor speed increases from minimum.
- Compressor begins to increase speed.
- Compressor speed continues to increase to 100 percent.

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

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

6. Appendix A: Load Calculations and Equipment Selection

6.1 Load Calculations

Equipment sizing for the prototype warehouses were established according to load calculations for each refrigerated space. Loads included envelope transmission loads, exterior and inter-zonal air infiltration, forklift and pallet-lift traffic, employee traffic, air unit (evaporator coil) fan motor heat gain, evaporator coil defrost heat gain, heat gain from the lighting systems, and product respiration and pull-down load. A 1.15 safety factor was used in the equipment selection process.

The refrigeration systems for each of the prototype warehouses were sized using design climate data.[6] For calculating statewide savings, three system sizes were developed to typify standard design practice in the California climate zones that have the majority of refrigerated warehouses in the state. Figure 29 describes the three designs and lists the climate zones where the designs were simulated.

Design	Climate Type	Design City	Design (0.1%) DBT/WBT	Simulated in Climate Zones
1	Mild Temperature, Coastal	Santa Maria	90°F/67°F	CTZ03 – Oakland CTZ05 – Santa Maria CTZ07 – San Diego (Lindbergh)
2	Medium-Temperature, Central Valley	Sacramento	104°F/74°F	CTZ12 – Sacramento Executive Airport CTZ13 – Fresno
3	Hot Temperature, Inland Empire	Riverside	106°F/75°F	CTZ10 – Riverside CTZ14 – Palmdale

Figure 29: Description of three design climate zones

Figure 30 through Figure 37 represent example load calculation worksheets used to size refrigeration equipment in the prototype warehouses.

[6] Design climate data from the 2008 Joint Appendices.

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BOX:	Cooler			INSULATION TYPE:	Polyisocyanurate			
TEMP (°F):	35				THICKNESS (in.)	"R/INCH"	"R"	"U"
LENGTH (ft.):	200	AREA (S.F.):	40,000	CEILING:	5.283	5.3	28	0.0357
WIDTH (ft.):	200	VOLUME (ft^3):	1,200,000	FLOOR:	8.000	0.66	5.28	0.1894
HEIGTH (ft.):	30			WALLS:	5.283	5.3	28	0.0357
				INTER-ZONAL WALL:	6.792	5.3	36	0.0278

*** TRANSMISSION LOADS***

	DIMENTION 1 (ft.)	DIMENTION 2 (ft.)	"U"	OUTSIDE WALL T (°F)	BOX T (°F)	LOAD (Btuh)	% OF TOTAL
CEILING	200	200	0.0357	134	35	141,429	13.9%
WALL 1	200	30	0.0357	40	35	1,071	0.1%
WALL 2	200	30	0.0278	-10	35	(7,500)	-0.7%
WALL 3	200	30	0.0357	104	35	14,786	1.5%
WALL 4	200	30	0.0357	104	35	14,786	1.5%
PERIMETER	400	0.440	0.1894	85	35	1,668	0.2%
					TOTAL TRANSMISSION LOAD:	166,239	16.3%

*** INTERNAL LOADS***

	QUANTITY	LOAD EA (Btuh)			LOAD (Btuh)	% OF TOTAL
PEOPLE	26.67	1,450			38,667	3.8%
FORKLIFTS	13.04	20,000			260,870	25.6%
PALLET LIFTS	4.35	10,000			43,478	4.3%
	Design Capacity (Btuh)	p. Eff. (Btuh/Watt	WATTS EA	FACTOR		
COILS	1,277,384	34.0	37,570	3.413	128,227	12.6%
	WATTS/S.F.	FACTOR				
LIGHTS	0.70	3.413			95,564	9.4%
OTHER EQUIP IN SPACE	0	3.413			0	0.0%
				TOTAL INTERNAL LOADS:	566,805	55.7%

*** INFILTRATION LOAD***
ASHRAE DOOR USAGE METHOD

Pyschrometric Information

Temperature of refrigerated air (Tr) =	35 F	'emperature of infiltration air (Ti) =	40 F
Relative humidity in refrigerated area (RHr) =	90 %	e humidity of infiltration air (Rhi) =	90 %
Density of refrigerated air (Dr) =	0.080 lb/cu.ft	Density of infiltration air (Di) =	0.0787 lb/cu.ft
Enthalpy of refrigerated air (Hr) =	13 Btu/lb	Enthalpy of infiltration air (Hi) =	14.50 Btu/lb

Dock to Cooler Door Information :-

Number of doors =	2	Door Dimensions =	10x10
Area of each door (S.F.) =	100	Total doorway area (A) =	200 S.F.

Refrig. load for fully developed air flow

Density factor (Fm) =	1.829
Infiltration load for fully developed air flow (q) =	74,277 Btuh

Dock to freezer door operation information

# of doorway passages per door per day (P) =	360 (15 per hour)	Door open-close time (Tp) =	20 secs
Time door simply stands open (To) =	0 min	Hours of operation (T) =	24 Hours
Door-way open time factor (Dt) =	0.083		
Doorway flow factor (Df) =	0.800	Effectiveness (E) =	0.00

	LOAD (Btuh)	% OF TOTAL
INFILTRATION LOAD (Dock to Cooler) :	4,952	0.5%

*** PRODUCT LOAD***

LBS/PERIOD: 400,000 PERIOD (HRS): 24

PULLDOWN:	TEMP IN	TEMP OUT	h SENS	h LATENT	BTU'S
ABOVE FREEZING	50	40	0.65		2,600,000
LATENT				0	0
BELOW FREEZING	0	0	0		0

RESPIRATION:	TONS	RESP RATE		LOAD (Btuh)	% OF TOTAL
	750	5,500	PULLDOWN PER HR:	108,333	10.6%
			RESPIRATION PER HR:	171,875	16.9%
			TOTAL PRODUCT LOAD:	280,208	27.5%

		LOAD (Btuh)	% OF TOTAL	Tons	SF/Ton
	INSTANTANEOUS LOAD TOTAL (Btuh):	1,018,204	100.0%	84.9	471
SAFETY FACTOR: 1.15	LOAD WITH SAFETY FACTOR (Btuh):	1,170,935	115.0%	97.6	410
COIL OP. HOURS: 22	COIL DESIGN LOAD (Btuh):	1,277,384	125.5%	106.4	376

Figure 30: Load calculations, 35°F cooler space (Prototype Warehouses #1 and 2)

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Prototype Warehouse 1 and 2

BOX:	**Freezer**			INSULATION TYPE:	Polyisocyanurate		
TEMP (° F):	-10				THICKNESS (in.)	"R/INCH"	"R"	"U"
LENGTH (ft.):	200	AREA (S.F.):	40,000	CEILING:	6.792	5.3	36	0.0278
WIDTH (ft.):	200	VOLUME (ft^3):	1,200,000	FLOOR:	6.792	5.3	36	0.0278
HEIGTH (ft.):	30			WALLS:	6.792	5.3	36	0.0278
				INTER-ZONAL WALL:	6.792	5.3	36	0.0278

*** TRANSMISSION LOADS***

	DIMENTION 1 (ft.)	DIMENTION 2 (ft.)	"U"	OUTSIDE WALL T (° F)	BOX T (° F)	LOAD (Btuh)	% OF TOTAL
CEILING	200	200	0.0278	134	-10	160,000	14.5%
WALL 1	200	30	0.0278	40	-10	8,333	0.8%
WALL 2	200	30	0.0278	104	-10	19,000	1.7%
WALL 3	200	30	0.0278	104	-10	19,000	1.7%
WALL 4	200	30	0.0278	35	-10	7,500	0.7%
FLOOR	200	200	0.0278	70	-10	88,889	8.1%
					TOTAL TRANSMISSION LOAD:	302,722	27.4%

*** INTERNAL LOADS***

	QUANTITY	LOAD EA (Btuh)			LOAD (Btuh)	% OF TOTAL
PEOPLE	26.67	1,450			38,667	3.5%
FORKLIFTS	13.04	20,000			260,870	23.6%
PALLET LIFTS	4.35	10,000			43,478	3.9%
	Design Capacity (Btuh)	p. Eff. (Btuh/Watt	WATTS EA	FACTOR		
COILS	1,385,001	34.0	40,735	3.413	139,030	12.6%
	WATTS/S.F.	FACTOR				
LIGHTS	0.70	3.413			95,564	8.7%
OTHER EQUIP IN SPACE	0	3.413			0	0.0%
				TOTAL INTERNAL LOADS:	577,608	52.3%

*** INFILTRATION LOAD***

ASHRAE DOOR USAGE METHOD

Pyschrometric Information

Temperature of refrigerated air (Tr) =	-10 F	emperature of infiltration air (Ti) =	40 F
Relative humidity in refrigerated area (RHr) =	100 %	e humidity of infiltration air (Rhi) =	90 %
Density of refrigerated air (Dr) =	0.088 lb/cu.ft	Density of infiltration air (Di) =	0.0787 lb/cu.ft
Enthalpy of refrigerated air (Hr) =	0.01 Btu/lb	Enthalpy of infiltration air (Hi) =	14.50 Btu/lb

Dock to Freezer Door Information :-

Number of doors =	2	Door Dimensions =	10x10
Area of each door (S.F.) =	100	Total doorway area (A) =	200 S.F.

Refrig. load for fully developed air flow

Density factor (Fm) =	1.758
Infiltration load for fully developed air flow (q) =	2,297,023 Btuh

Dock to freezer door operation information

# of doorway passages per door per day (P) =	360 (15 per hour)	Door open-close time (Tp) =	20 secs
Time door simply stands open (To) =	0 min	Hours of operation (T) =	24 Hours
Door-way open time factor (Dt) =	0.083		
Doorway flow factor (Df) =	0.800	Effectiveness(E) =	0.00

	LOAD (Btuh)	% OF TOTAL
INFILTRATION LOAD (Dock to Freezer) :	153,135	13.9%

*** PRODUCT LOAD***

LBS/PERIOD: 400,000 PERIOD (HRS): 24

PULLDOWN:	TEMP IN	TEMP OUT	h SENS	h LATENT	BTU'S
ABOVE FREEZING	0	0	0		0
LATENT				0	0
BELOW FREEZING	-5	-10	0.5		1,000,000

RESPIRATION:	TONS	RESP RATE		LOAD (Btuh)	% OF TOTAL
	0	0	PULLDOWN PER HR:	41,667	3.8%
			RESPIRATION PER HR:	0	0.0%
			TOTAL PRODUCT LOAD:	41,667	3.8%

*** DEFROST LOAD***

LOAD (Btuh)	% OF TOTAL
28,854	2.6%

			LOAD (Btuh)	% OF TOTAL	Tons	SF/Ton
		INSTANTANEOUS LOAD TOTAL:	1,103,986	100.0%	92.0	435
SAFETY FACTOR:	1.15	LOAD WITH SAFETY FACTOR:	1,269,584	115.0%	105.8	378
COIL OP. HOURS:	22	COIL DESIGN LOAD:	1,385,001	125.5%	115.4	347

Figure 31: Load calculations, -10°F freezer space (Prototype Warehouses #1 and 2)

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Prototype Warehouse 1

DATE: 12/31/2009

BOX:	Dock			INSULATION TYPE:	Polyisocyanurate			
TEMP (°F):	40				THICKNESS (in.)	"R/INCH"	"R"	"U"
LENGTH (ft.):	400	AREA (S.F.):	12,000	CEILING:	5.283	5.3	28	0.0357
WIDTH (ft.):	30	VOLUME (ft^3):	360,000	FLOOR:	8.000	0.66	5.28	0.1894
HEIGTH (ft.):	30			WALLS:	5.283	5.3	28	0.0357
				INTER-ZONAL WALL:	4.906	5.3	26	0.0385

*** TRANSMISSION LOADS***

	DIMENTION 1 (ft.)	DIMENTION 2 (ft.)	"U"	OUTSIDE WALL T (°F)	BOX T (°F)	LOAD (Btuh)	% OF TOTAL
CEILING	400	30	0.0357	134	40	40,286	4.6%
WALL 1	400	30	0.0357	104	40	27,429	3.1%
WALL 2	30	30	0.0357	104	40	2,057	0.2%
WALL 3	200	30	0.0357	-10	40	(10,714)	-1.2%
WALL 4	200	30	0.0357	35	40	(1,071)	-0.1%
WALL 5	30	30	0.0357	104	40	2,057	0.2%
PERIMETER	460	0.667	0.1894	85	40	2,614	0.3%
					TOTAL TRANSMISSION LOAD:	62,656	7.1%

*** INTERNAL LOADS***

	QUANTITY	LOAD EA (Btuh)			LOAD (Btuh)	% OF TOTAL
PEOPLE	8.00	1,450			11,600	1.3%
FORKLIFTS	3.91	20,000			78,261	8.9%
PALLET LIFTS	1.30	10,000			13,043	1.5%
	Design Capacity (Btuh)	p. Eff. (Btuh/Watt	WATTS EA	FACTOR		
COILS	1,107,846	34.0	32,584	3.413	111,208	12.6%
	WATTS/S.F.	FACTOR				
LIGHTS	0.70	3.413			28,669	3.2%
OTHER EQUIP IN SPACE	0	3.413			0	0.0%
				TOTAL INTERNAL LOADS:	242,782	27.5%

*** INFILTRATION LOAD***

Pyschrometric Information

Temperature of refrigerated air (Tr) =	40 F		emperature of infiltration air (Ti) =	104 F
Relative humidity in refrigerated area (RHr) =	90 %		e humidity of infiltration air (Rhi) =	50 %
Density of refrigerated air (Dr) =	0.0787 lb/cu.ft		Density of infiltration air (Di) =	0.0678 lb/cu.ft
Enthalpy of refrigerated air (Hr) =	14.50 Btu/lb		Enthalpy of infiltration air (Hi) =	50.00 Btu/lb
Number of dock doors:	20 doors			
Assumed infiltration per door:	200 CFM			
Total infiltration:	4,000 CFM			

	LOAD (Btuh)	% OF TOTAL
INFILTRATION LOAD :	577,627	65.4%

*** PRODUCT LOAD***

LBS/PERIOD: 0 PERIOD (HRS): 24

PULLDOWN:	TEMP IN	TEMP OUT	h SENS	h LATENT	BTU'S
ABOVE FREEZING	75	40	0.65		0
LATENT				0	0
BELOW FREEZING	0	0	0		0

RESPIRATION:	TONS	RESP RATE		LOAD (Btuh)	% OF TOTAL
	0	0	PULLDOWN PER HR:	0	0.0%
			RESPIRATION PER HR:	0	0.0%
			TOTAL PRODUCT LOAD:	0	0.0%

			LOAD (Btuh)	% OF TOTAL	Tons	SF/Ton
		INSTANTANEOUS LOAD TOTAL:	883,065	100.0%	73.6	163
SAFETY FACTOR:	1.15	LOAD WITH SAFETY FACTOR:	1,015,525	115.0%	84.6	142
COIL OP. HOURS:	22	COIL DESIGN LOAD:	1,107,846	125.5%	92.3	130

Figure 32: Load calculations, 40°F dock space (Prototype Warehouse #1)

VaCom Technologies DATE: 12/31/2009

Prototype Warehouse 2

BOX:	Dry Warehouse			INSULATION TYPE:	Polyisocyanurate			
TEMP (° F):	85				THICKNESS (in.)	"R/INCH"	"R"	"U"
LENGTH (ft.):	400	AREA (S.F.):	20,000	CEILING:	3.585	5.3	19	0.0526
WIDTH (ft.):	50	VOLUME (ft^3):	600,000	FLOOR:	8.000	0.66	5.28	0.1894
HEIGTH (ft.):	30			WALLS:	2.453	5.3	13	0.0769
				INTER-ZONAL WALL:	4.906	5.3	26	0.0385

*** TRANSMISSION LOADS***

	DIMENTION 1 (ft.)	DIMENTION 2 (ft.)	"U"	OUTSIDE WALL T (° F)	BOX T (° F)	LOAD (Btuh)	% OF TOTAL
CEILING	400	50	0.0526	134	85	51,579	9.2%
WALL 1	400	30	0.0769	104	85	17,538	3.1%
WALL 2	50	30	0.0769	104	85	2,192	0.4%
WALL 3	200	30	0.0278	-10	85	(15,833)	-2.8%
WALL 4	200	30	0.0357	35	85	(10,714)	-1.9%
WALL 5	50	30	0.0769	104	85	2,192	0.3%
PERIMETER	500	0.667	0.1894	85	85	0	0.0%
					TOTAL TRANSMISSION LOAD:	46,954	8.4%

*** INTERNAL LOADS***

	QUANTITY	LOAD EA (Btuh)		LOAD (Btuh)	% OF TOTAL
PEOPLE	13.33	1,450		19,333	3.5%
FORKLIFTS	6.52	20,000		130,435	23.3%
PALLET LIFTS	2.17	10,000		21,739	3.9%
	WATTS/S.F.	FACTOR			
LIGHTS	0.70	3.413		47,782	8.5%
OTHER EQUIP IN SPACE	0	3.413		0	0.0%
			TOTAL INTERNAL LOADS:	219,289	39.2%

*** INFILTRATION LOAD***

Pyschrometric Information :-

Temperature of refrigerated air (Tr) =	85	F	Temperature of infiltration air (Ti) =	104	F
Relative humidity in refrigerated area (RHr) =	40	%	e humidity of infiltration air (Rhi) =	50	%
Density of refrigerated air (Dr) =	0.0787	lb/cu.ft	Density of infiltration air (Di) =	0.0678	lb/cu.ft
Enthalpy of refrigerated air (Hr) =	32.00	Btu/lb	Enthalpy of infiltration air (Hi) =	50.00	Btu/lb
Number of dock doors:	20	doors			
Assumed infiltration per door:	200	CFM			
Total infiltration:	4,000	CFM			

	LOAD (Btuh)	% OF TOTAL
INFILTRATION LOAD :	292,881	52.4%

			LOAD (Btuh)	% OF TOTAL	Tons	SF/Ton
		INSTANTANEOUS LOAD TOTAL:	559,125	100.0%	46.6	429
SAFETY FACTOR:	1.15	LOAD WITH SAFETY FACTOR:	642,994	115.0%	53.6	373
COIL OP. HOURS:	24	COIL DESIGN LOAD:	642,994	115.0%	53.6	373

Figure 33: Load calculations, 85°F dry storage space (Prototype Warehouse #2)

VaCom Technologies DATE: 12/31/2009

Prototype Warehouse 3 and 4

BOX:	**Cooler**			INSULATION TYPE:	Polyisocyanurate			
TEMP (° F):	35				THICKNESS (in.)	"R/INCH"	"R"	"U"
LENGTH (ft.):	100	AREA (S.F.):	10,000	CEILING:	5.283	5.3	28	0.0357
WIDTH (ft.):	100	VOLUME (ft^3):	300,000	FLOOR:	8.000	0.66	5.28	0.1894
HEIGTH (ft.):	30			WALLS:	5.283	5.3	28	0.0357
				INTER-ZONAL WALL:	6.792	5.3	36	0.0278

*** TRANSMISSION LOADS***

	DIMENTION 1 (ft.)	DIMENTION 2 (ft.)	"U"	OUTSIDE WALL T (° F)	BOX T (° F)	LOAD (Btuh)	% OF TOTAL
CEILING	100	100	0.0357	134	35	35,357	12.8%
WALL 1	100	30	0.0357	40	35	536	0.2%
WALL 2	100	30	0.0278	-10	35	(3,750)	-1.4%
WALL 3	100	30	0.0357	104	35	7,393	2.7%
WALL 4	100	30	0.0357	104	35	7,393	2.7%
PERIMETER	200	0.440	0.1894	85	35	834	0.3%
					TOTAL TRANSMISSION LOAD:	47,762	17.2%

*** INTERNAL LOADS***

	QUANTITY	LOAD EA (Btuh)			LOAD (Btuh)	% OF TOTAL
PEOPLE	6.67	1,450			9,667	3.5%
FORKLIFTS	3.85	20,000			76,923	27.7%
PALLET LIFTS	1.15	10,000			11,538	4.2%
	Design Capacity (Btuh)	p. Eff. (Btuh/Watt	WATTS EA	FACTOR		
COILS	347,787	34.0	10,229	3.413	34,912	12.6%
	WATTS/S.F.	FACTOR				
LIGHTS	0.70	3.413			23,891	8.6%
OTHER EQUIP IN SPACE	0	3.413			0	0.0%
				TOTAL INTERNAL LOADS:	156,931	56.6%

*** INFILTRATION LOAD***

ASHRAE DOOR USAGE METHOD

Pyschrometric Information

Temperature of refrigerated air (Tr) =	35 F	Temperature of infiltration air (Ti) =	40 F
Relative humidity in refrigerated area (RHr) =	90 %	e humidity of infiltration air (Rhi) =	90 %
Density of refrigerated air (Dr) =	0.080 lb/cu.ft	Density of infiltration air (Di) =	0.0787 lb/cu.ft
Enthalpy of refrigerated air (Hr) =	13 Btu/lb	Enthalpy of infiltration air (Hi) =	14.50 Btu/lb

Dock to Cooler Door Information :-

Number of doors =	1	Door Dimensions =	10x10
Area of each door (S.F.) =	100	Total doorway area (A) =	100 S.F.

Refrig. load for fully developed air flow

Density factor (Fm) =	1.829
Infiltration load for fully developed air flow (q) =	37,138 Btuh

Dock to freezer door operation information

# of doorway passages per door per day (P) =	360 (15 per hour)	Door open-close time (Tp) =	20 secs
Time door simply stands open (To) =	0 min	Hours of operation (T) =	24 Hours
Door-way open time factor (Dt) =	0.083		
Doorway flow factor (Df) =	0.800	Effectiveness (E) =	0.00

	LOAD (Btuh)	% OF TOTAL
INFILTRATION LOAD (Dock to Cooler) :	2,476	0.9%

*** PRODUCT LOAD***

LBS/PERIOD: 100,000 PERIOD (HRS): 24

PULLDOWN:	TEMP IN	TEMP OUT	h SENS	h LATENT	BTU'S
ABOVE FREEZING	50	40	0.65		650,000
LATENT				0	0
BELOW FREEZING	0	0	0		0

RESPIRATION:	TONS	RESP RATE		LOAD (Btuh)	% OF TOTAL
	187.5	5,500	PULLDOWN PER HR:	27,083	9.8%
			RESPIRATION PER HR:	42,969	15.5%
			TOTAL PRODUCT LOAD:	70,052	25.3%

			LOAD (Btuh)	% OF TOTAL	Tons	SF/Ton
		INSTANTANEOUS LOAD TOTAL (Btuh):	277,221	100.0%	23.1	433
SAFETY FACTOR:	1.15	LOAD WITH SAFETY FACTOR (Btuh):	318,804	115.0%	26.6	376
COIL OP. HOURS:	22	COIL DESIGN LOAD (Btuh):	347,787	125.5%	29.0	345

Figure 34: Load calculations, 35°F cooler space (Prototype Warehouses #3 and 4)

VaCom Technologies
Prototype Warehouse 3 and 4

DATE: 12/31/2009

BOX:	**Freezer**			INSULATION TYPE:	Polyisocyanurate			
TEMP (° F):	-10				THICKNESS (in.)	"R/INCH"	"R"	"U"
LENGTH (ft.):	100	AREA (S.F.):	10,000	CEILING:	6.792	5.3	36	0.0278
WIDTH (ft.):	100	VOLUME (ft^3):	300,000	FLOOR:	6.792	5.3	36	0.0278
HEIGTH (ft.):	30			WALLS:	6.792	5.3	36	0.0278
				INTER-ZONAL WALL:	6.792	5.3	36	0.0278

TRANSMISSION LOADS

	DIMENTION 1 (ft.)	DIMENTION 2 (ft.)	"U"	OUTSIDE WALL T (° F)	BOX T (° F)	LOAD (Btuh)	% OF TOTAL
CEILING	100	100	0.0278	134	-10	40,000	11.4%
WALL 1	100	30	0.0278	40	-10	4,167	1.2%
WALL 2	100	30	0.0278	104	-10	9,500	2.7%
WALL 3	100	30	0.0278	104	-10	9,500	2.7%
WALL 4	100	30	0.0278	35	-10	3,750	1.1%
FLOOR	100	100	0.0278	70	-10	22,222	6.3%
					TOTAL TRANSMISSION LOAD:	89,139	25.4%

INTERNAL LOADS

	QUANTITY	LOAD EA (Btuh)			LOAD (Btuh)	% OF TOTAL
PEOPLE	6.67	1,450			9,667	2.7%
FORKLIFTS	3.85	20,000			76,923	21.9%
PALLET LIFTS	1.15	10,000			11,538	3.3%
	Design Capacity (Btuh)	p. Eff. (Btuh/Watt	WATTS EA	FACTOR		
COILS	441,113	34.0	12,974	3.413	44,280	12.6%
	WATTS/S.F.	FACTOR				
LIGHTS	0.70	3.413			23,891	6.8%
OTHER EQUIP IN SPACE	0	3.413			0	0.0%
				TOTAL INTERNAL LOADS:	166,299	47.3%

INFILTRATION LOAD
ASHRAE DOOR USAGE METHOD

Pyschrometric Information

Temperature of refrigerated air (Tr) =	-10 F	emperature of infiltration air (Ti) =	40 F
Relative humidity in refrigerated area (RHr) =	100 %	e humidity of infiltration air (Rhi) =	90 %
Density of refrigerated air (Dr) =	0.088 lb/cu.ft	Density of infiltration air (Di) =	0.0787 lb/cu.ft
Enthalpy of refrigerated air (Hr) =	0.01 Btu/lb	Enthalpy of infiltration air (Hi) =	14.50 Btu/lb

Dock to Freezer Door Information :-

Number of doors =	1	Door Dimensions =	10x10
Area of each door (S.F.) =	100	Total doorway area (A) =	100 S.F.

Refrig. load for fully developed air flow

Density factor (Fm) =	1.758
Infiltration load for fully developed air flow (q) =	1,148,511 Btuh

Dock to freezer door operation information

# of doorway passages per door per day (P) =	360 (15 per hour)	Door open-close time (Tp) =	20 secs
Time door simply stands open (To) =	0 min	Hours of operation (T) =	24 Hours
Door-way open time factor (Dt) =	0.083		
Doorway flow factor (Df) =	0.800	Effectiveness(E) =	0.00

	LOAD (Btuh)	% OF TOTAL
INFILTRATION LOAD (Dock to Freezer) :	76,567	21.8%

PRODUCT LOAD

LBS/PERIOD: 100,000 PERIOD (HRS): 24

PULLDOWN:	TEMP IN	TEMP OUT	h SENS	h LATENT	BTU'S
ABOVE FREEZING	0	0	0		0
LATENT				0	0
BELOW FREEZING	-5	-10	0.5		250,000

RESPIRATION:	TONS	RESP RATE		LOAD (Btuh)	% OF TOTAL
	0	0	PULLDOWN PER HR:	10,417	3.0%
			RESPIRATION PER HR:	0	0.0%
			TOTAL PRODUCT LOAD:	10,417	3.0%

DEFROST LOAD

LOAD (Btuh)	% OF TOTAL
9,190	2.6%

			LOAD (Btuh)	% OF TOTAL	Tons	SF/Ton
		INSTANTANEOUS LOAD TOTAL:	351,612	100.0%	29.3	341
SAFETY FACTOR:	1.15	LOAD WITH SAFETY FACTOR:	404,354	115.0%	33.7	297
COIL OP. HOURS:	22	COIL DESIGN LOAD:	441,113	125.5%	36.8	272

Figure 35: Load calculations, -10°F freezer space (Prototype Warehouses #1 and 2)

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Prototype Warehouse 3

BOX:	Dock			INSULATION TYPE:	Polyisocyanurate		
TEMP (°F):	40				THICKNESS (in.)	"R/INCH"	"R"	"U"
LENGTH (ft.):	200	AREA (S.F.):	6,000	CEILING:	5.283	5.3	28	0.0357
WIDTH (ft.):	30	VOLUME (ft^3):	180,000	FLOOR:	8.000	0.66	5.28	0.1894
HEIGTH (ft.):	30			WALLS:	5.283	5.3	28	0.0357
				INTER-ZONAL WALL:	4.906	5.3	26	0.0385

*** TRANSMISSION LOADS***

	DIMENTION 1 (ft.)	DIMENTION 2 (ft.)	"U"	OUTSIDE WALL T (°F)	BOX T (°F)	LOAD (Btuh)	% OF TOTAL
CEILING	200	30	0.0357	134	40	20,143	6.3%
WALL 1	200	30	0.0357	104	40	13,714	4.3%
WALL 2	30	30	0.0357	104	40	2,057	0.6%
WALL 3	100	30	0.0278	-10	40	(4,167)	-1.3%
WALL 4	100	30	0.0357	35	40	(536)	-0.2%
WALL 5	30	30	0.0357	104	40	2,057	0.6%
PERIMETER	260	0.667	0.1894	85	40	1,477	0.5%
					TOTAL TRANSMISSION LOAD:	34,746	10.8%

*** INTERNAL LOADS***

	QUANTITY	LOAD EA (Btuh)			LOAD (Btuh)	% OF TOTAL
PEOPLE	4.00	1,450			5,800	1.8%
FORKLIFTS	2.31	20,000			46,154	14.3%
PALLET LIFTS	0.69	10,000			6,923	2.2%
	Design Capacity (Btuh)	p. Eff. (Btuh/Watt	WATTS EA	FACTOR		
COILS	403,672	34.0	11,873	3.413	40,522	12.6%
	WATTS/S.F.	FACTOR				
LIGHTS	0.70	3.413			14,335	4.5%
OTHER EQUIP IN SPACE	0	3.413			0	0.0%
				TOTAL INTERNAL LOADS:	113,733	35.3%

*** INFILTRATION LOAD***

Pyschrometric Information

Temperature of refrigerated air (Tr) =	40 F	emperature of infiltration air (Ti) =	104 F
Relative humidity in refrigerated area (RHr) =	90 %	e humidity of infiltration air (Rhi) =	50 %
Density of refrigerated air (Dr) =	0.0787 lb/cu.ft	Density of infiltration air (Di) =	0.0678 lb/cu.ft
Enthalpy of refrigerated air (Hr) =	14.50 Btu/lb	Enthalpy of infiltration air (Hi) =	50.00 Btu/lb

Number of dock doors:	6 doors
Assumed infiltration per door:	200 CFM
Total infiltration:	1,200 CFM

	LOAD (Btuh)	% OF TOTAL
INFILTRATION LOAD :	173,288	53.9%

*** PRODUCT LOAD***

LBS/PERIOD: 0 PERIOD (HRS): 24

PULLDOWN:	TEMP IN	TEMP OUT	h SENS	h LATENT	BTU'S
ABOVE FREEZING	75	40	0.65		0
LATENT				0	0
BELOW FREEZING	0	0	0		0

RESPIRATION:	TONS	RESP RATE		LOAD (Btuh)	% OF TOTAL
	0	0	PULLDOWN PER HR:	0	0.0%
			RESPIRATION PER HR:	0	0.0%
			TOTAL PRODUCT LOAD:	0	0.0%

			LOAD (Btuh)	% OF TOTAL	Tons	SF/Ton
		INSTANTANEOUS LOAD TOTAL:	321,768	100.0%	26.8	224
SAFETY FACTOR:	1.15	LOAD WITH SAFETY FACTOR:	370,033	115.0%	30.8	195
COIL OP. HOURS:	22	COIL DESIGN LOAD:	403,672	125.5%	33.6	178

Figure 36: Load calculations, 40°F dock space (Prototype Warehouse #3)

VaCom Technologies DATE: 12/31/2009
Prototype Warehouse 4

BOX:	**Dry Warehouse**			INSULATION TYPE:	Polyisocyanurate		
TEMP (° F):	85				THICKNESS (in.)	"R/INCH"	"R"	"U"
LENGTH (ft.):	200	AREA (S.F.):	10,000	CEILING:	3.585	5.3	19	0.0526
WIDTH (ft.):	50	VOLUME (ft^3):	300,000	FLOOR:	8.000	0.66	5.28	0.1894
HEIGTH (ft.):	30			WALLS:	2.453	5.3	13	0.0769

*** TRANSMISSION LOADS***

	DIMENTION 1 (ft.)	DIMENTION 2 (ft.)	"U"	OUTSIDE WALL T (° F)	BOX T (° F)	LOAD (Btuh)	% OF TOTAL
CEILING	200	50	0.0526	134	85	25,789	6.0%
WALL 1	200	30	0.0769	104	85	8,769	2.0%
WALL 2	50	30	0.0769	104	85	2,192	0.5%
WALL 3	100	30	0.0278	-10	85	(7,917)	-1.8%
WALL 4	100	30	0.0357	35	85	(5,357)	-1.2%
WALL 5	50	30	0.0769	104	85	2,192	0.4%
PERIMETER	300	0.667	0.1894	85	85	0	0.0%
					TOTAL TRANSMISSION LOAD:	25,670	6.0%

*** INTERNAL LOADS***

	QUANTITY	LOAD EA (Btuh)	LOAD (Btuh)	% OF TOTAL
PEOPLE	6.67	1,450	9,667	2.3%
FORKLIFTS	3.33	20,000	66,667	15.5%
PALLET LIFTS	1.00	10,000	10,000	2.3%
	WATTS/S.F.	FACTOR		
LIGHTS	0.70	3.413	23,891	5.6%
OTHER EQUIP IN SPACE	0	3.413	0	0.0%
		TOTAL INTERNAL LOADS:	110,224	25.7%

*** INFILTRATION LOAD***

Pyschrometric Information :-

Temperature of refrigerated air (Tr) =	85 F	Temperature of infiltration air (Ti) =	104 F
Relative humidity in refrigerated area (RHr) =	40 %	e humidity of infiltration air (Rhi) =	50 %
Density of refrigerated air (Dr) =	0.0787 lb/cu.ft	Density of infiltration air (Di) =	0.0678 lb/cu.ft
Enthalpy of refrigerated air (Hr) =	32.00 Btu/lb	Enthalpy of infiltration air (Hi) =	50.00 Btu/lb

Number of dock doors:	20 doors
Assumed infiltration per door:	200 CFM
Total infiltration:	4,000 CFM

	LOAD (Btuh)	% OF TOTAL
INFILTRATION LOAD :	292,881	68.3%

			LOAD (Btuh)	% OF TOTAL	Tons	SF/Ton
		INSTANTANEOUS LOAD TOTAL:	428,775	100.0%	35.7	280
SAFETY FACTOR:	1.15	LOAD WITH SAFETY FACTOR:	493,091	115.0%	41.1	243
COIL OP. HOURS:	24	COIL DESIGN LOAD:	493,091	115.0%	41.1	243

Figure 37: Load calculations, 85°F dry storage space (Prototype Warehouse #4)

6.2 Equipment Selection

The following figures summarize the compressor selection criteria and the selected compressor performance for the prototype warehouses. Prototype Warehouses #1 and 2 are summarized in Figure 38, while Prototype Warehouses #3 and 4 are summarized in Figure 39.

	LT Suction Group	**MT Suction Group (Cooler and Dock)**	**MT Suction Group (Cooler Only)**
Prototype warehouse:	1 and 2	1	2
Design Criteria			
Refrigerant:	Ammonia (R-717)	Ammonia (R-717)	Ammonia (R-717)
Evaporator design capacity:	1,385,001 Btuh	2,385,229 Btuh	1,316,898 Btuh
Number of compressors:	2	2	2
Design space temperature:	-10°F	35°F	35 °F
Design TD (SET - space temp):	10°F	10°F	10 °F
Estimated suction line pressure losses:	3°F	3°F	3 °F
Design WBT:	73°F	73°F	73 °F
Condenser design TD:	23°F	23°F	23 °F
Assumed compressor run-time	100%	100%	1
Compressor design capacity:	692,500 Btuh	1,192,615 Btuh	658,449 Btuh

	57.7 TR	99.4 TR	54.9 TR
Compressor design mass flow:	1,524 lb/hr	2,553 lb/hr	1,409.7 lb/hr
Compressor design SST:	-23°F SST	22°F SST	22°F SST
Compressor design SCT:	96°F SCT	96°F SCT	96°F SCT
Selected Compressor Performance			
Mass flow at design conditions:	1,651 lb/hr	2,831 lb/hr	1,667.3 lb/hr
Capacity at design conditions:	62.5 TR	110.2 TR	64.9 TR
	750,000 Btuh	1,322,400 Btuh	778,800 Btuh
Power at design conditions:	164.8 HP	134.1 HP	80.7 HP
	131.4 kW	106.9 kW	65.2 kW
Drive motor nameplate HP:	175 HP	150 HP	100 HP
Assumed motor nameplate efficiency:	93.6%	93.6%	92.4%

Figure 38: Prototype Warehouse #1 and 2 compressor selection

	LT System	**MT System (Cooler and Dock)**	**MT System (Cooler Only)**
Prototype warehouse:	3 and 4	3	4
Design Criteria			
Refrigerant:	R-404A	R-404A	R-404A
Evaporator design capacity:	441,113 Btuh	751,459 Btuh	367,484 Btuh
Number of compressors:	8	4	2
Design space temperature:	-10 °F	35 °F	35 °F
Evap design TD (SET - space temp):	10 °F	10 °F	10 °F
Estimated suction line pressure losses:	3 °F	3 °F	3 °F
Design DBT:	104 °F	104 °F	104 °F
Condenser design TD:	10 °F	15 °F	15 °F
Assumed compressor run-time:	1	1	1
Selected Compressor Performance			
Compressor design capacity:	55,139 Btuh	187,865 Btuh	183,742 Btuh
	4.6 TR	15.7 TR	15.3 TR
Compressor design mass flow:	1,598.2 lb/hr	4,905.1 lb/hr	4,797.4 lb/hr
Compressor design SST:	-23 °F SST	22 °F SST	22 °F SST
Compressor design SCT:	114 °F SCT	119 °F SCT	119 °F SCT
Flow rate at -23°F SST, 114°F SCT:	1,930 lb/hr	5,550 lb/hr	5,550 lb/hr
Capacity at -23°F SST, 114°F SCT:	66,585 Btuh	212,565 Btuh	212,565 Btuh
Power at -23°F SST, 114°F SCT:	22.1 kW	32.2 kW	32.2 kW

Figure 39: Prototype Warehouse #3 and 4 compressor selection

7. Appendix B: Base Case Prototype Descriptions

7.1 Base Case Facility Description

The base case design is the starting point from which energy efficient design alternatives were considered. The base case is defined using 2008 Title 24 standards. Figure 40 shows the base case design assumptions.

	Large Prototype Warehouse		Small Prototype Warehouse	
	With Refrigerated Dock (Prototype Warehouse #1)	***With Dry Storage Area (Prototype Warehouse #2)***	***With Refrigerated Dock (Prototype Warehouse #3)***	***With Dry Storage Area (Prototype Warehouse #4)***
Envelope Description				
Hours of Operation	9 AM to 1 AM, 7 days/week			
Freezer Area	40,000 S.F.		10,000 S.F.	
Cooler Area	40,000 S.F.		10,000 S.F.	
Refrigerated Dock Area	12,000 S.F.	*N/A*	6,000 S.F.	*N/A*
Conditioned Dry Storage Area	*N/A*	20,000 S.F.	*N/A*	10,000 S.F.
Total Facility Area	92,000 S.F.	100,000 S.F.	26,000 S.F.	30,000 S.F.
Ceiling Height	30 Ft.			
Temperature Set points	Freezer: -10°F Cooler: 35°F Dock: 40°F	Freezer: -10°F Cooler: 35°F Dry Storage Cooling: 85°F Dry Storage Heating: 70°F	Freezer: -10°F Cooler: 35°F Dock: 40°F	Freezer: -10°F Cooler: 35°F Dry Storage Cooling: 85°F Dry Storage Heating: 70°F
Lighting Type	Fluorescent lighting, non-ventilated reflectors			
Lighting Power	All areas: 0.70 Watts/S.F.			
Roof Construction	Built-up roof, polyurethane insulation. Inside film resistance: 0.90 Hr-S.F.-°F/Btu. Absorptance: 0.45 (Cool Roof)	Built-up roof, polyurethane insulation (refrigerated spaces), fiberglass batt insulation (dry storage area). Inside film resistance: 0.90 Hr-S.F.-°F/Btu. Absorptance: 0.45 (Cool Roof)	Built-up roof, polyurethane insulation. Inside film resistance: 0.90 Hr-S.F.-°F/Btu. Absorptance: 0.45 (Cool Roof)	Built-up roof, polyurethane insulation (refrigerated spaces), fiberglass batt insulation (dry storage area). Inside film resistance: 0.90 Hr-S.F.-°F/Btu. Absorptance: 0.45 (Cool Roof)
Roof Insulation Thickness/R-value at 75°F mean temperature	Freezer: 4.23” (R-36) Cooler/Dock: 3.29” (R-28)	Freezer: 4.23” (R-36) Cooler: 3.29” (R-28) Partially-Conditioned Warehouse: 6.13” (R-19)	Freezer: 4.23” (R-36) Cooler/Dock: 3.29” (R-28)	Freezer: 4.23” (R-36) Cooler: 3.29” (R-28) Partially-Conditioned Warehouse: 6.13” (R-19)
Wall Construction	All spaces: 8” hollow CMU construction, polyurethane insulation	8” hollow CMU construction, polyurethane insulation (refrigerated spaces), fiberglass batt insulation (dry storage area)	All spaces: 8” hollow CMU construction, polyurethane insulation	8” hollow CMU construction, polyurethane insulation (refrigerated spaces), fiberglass batt insulation (dry storage area)
Wall Insulation Thickness/R-	Freezer: 4.23” (R-36)	Freezer: 4.23” (R-36)	Freezer: 4.23” (R-36)	Freezer: 4.23” (R-36)

value at 75°F mean temperature	Cooler/Dock: 3.29” (R-28)	Cooler: 3.29” (R-28) Dry storage area: 4.19” (R-13)	Cooler/Dock: 3.29” (R-28)	Cooler: 3.29” (R-28) Dry storage area: 4.19” (R-13)
Floor Construction	Concrete slab (R-36 extruded polystyrene insulation below slab in Freezer area, no under-floor insulation in Cooler or Dock/Dry Storage Areas)			
Inter-Zonal Doors	(2) 10’ x 10’ doors between cooler and dock. (2) 10’ x 10’ doors between freezer and dock. Doors are automatic bi-parting break-away warehouse doors	(2) 10’ x 10’ doors between cooler and dock. (2) 10’ x 10’ doors between freezer and dock. Doors are automatic bi-parting break-away warehouse doors	(1) 10’ x 10’ doors between cooler and dock. (1) 10’ x 10’ doors between freezer and dock. Doors are automatic bi-parting break-away warehouse doors	(1) 10’ x 10’ doors between cooler and dock. (1) 10’ x 10’ doors between freezer and dock. Doors are automatic bi-parting break-away warehouse doors
Door Opening Frequency/Duration	Doors assumed open 15 times per hour (once every 4 minutes). 12 second total stand-open time (including opening, passage and closing time). ASHRAE density-driven methodology used to calculate air exchange			
Load Information				
Outside-Air Infiltration	4,000 CFM into refrigerated dock or dry storage area (assumed 20 dock doors, 200 CFM outside air per door, subject to hourly weather conditions and production schedule)		1,200 CFM into refrigerated dock or dry storage area (assumed 6 dock doors, 200 CFM outside air per door, subject to hourly weather conditions and production schedule)	
Product Pull-down	Freezer: 5°F (-5°F to -10°F) Cooler: 10°F (45°F to 35°F)			
Average Product Specific Heat	Freezer: 0.50 Btu/lb-°F Cooler: 0.65 Btu/lb-°F			
Product Throughput	Freezer: 400,000 lb/day Cooler: 400,000 lb/day		Freezer: 100,000 lb/day Cooler: 100,000 lb/day	
Product Pull down Load	Freezer: 41.7 MBH Cooler: 108.3 MBH		Freezer: 10.4 MBH Cooler: 27.1 MBH	
Respiring Product Load	Freezer: none Cooler: 171.9 MBH (750 tons of respiring product @ 5,500 Btuh/ton respiration rate)		Freezer: none Cooler: 43.0 MBH (187.5 tons of respiring product @ 5,500 Btuh/ton respiration rate)	
Occupancy	Assumed 1,500 S.F. per person. Heat gain from occupants assumed to be 580 Btuh sensible, 870 Btuh latent. Occupancy subject to production schedules			
Forklifts and Pallet Lifts	30 forklifts plus 10 pallet lifts distributed evenly throughout facility. Assumed 20 MBH/forklift, 10 MBH/pallet lift		10 forklifts plus 3 pallet lifts distributed evenly throughout facility. Assumed 20 MBH/forklift, 10 MBH/pallet lift	
Refrigeration System Information				

Refrigerant	R-717 (Ammonia)		R-404A	
System Configuration	Single-stage built-up central system, two suction groups (Low Temperature, Medium Temperature), two equal-sized screw compressors with thermosyphon oil cooling per suction group, evaporative condenser.		Two systems: Low Temperature (LT) and Medium Temperature (MT). Each system consists of parallel racks of semi-hermetic reciprocating compressors, served by air-cooled condensers	
Compressor Information				
SST Control Strategy	Fixed SST set point with 1°F throttling range LT Suction Group: -23°F SST MT Suction Group: 22°F SST		Fixed SST set point with 1°F throttling range LT System: -23°F SST MT System: 22°F SST	
Compressor Capacity Control	Slide Valve		None (cycling capacity control)	
Condenser Information				
Condenser Type	Evaporative-Cooled		Air-Cooled	
Number of Condensers	1		MT System: 1, LT System: 2	
Fan Quantity	1		LT System: 6 each MT System: 10	
Condenser Specific Efficiency	330 Btuh/Watt at 100°F SCT, 70°F WBT		53 Btuh/Watt at 10°F TD	
SCT Control	Floating head pressure to 70°F minimum SCT, variable set point (wetbulb or drybulb following) control strategy, variable speed fan control with all fans controlled in unison down to a minimum speed of 10-15% before cycling fans. 69°F backflood set point. 1°F throttling range.			
Air Unit (Evaporator Coil) Information				
Evaporator Feed Type	Flooded		Direct Expansion	
Design Saturated Evaporating Temperature (SET)	Freezer: 25°F Cooler: -20°F Dock: 30°F 10°F design TD in all spaces	Freezer: 25°F Cooler: -20°F 10°F design TD in all spaces	Freezer: 25°F Cooler: -20°F Dock: 30°F 10°F design TD in all spaces	Freezer: 25°F Cooler: -20°F 10°F design TD in all spaces
Air Unit Specific Efficiency	34 Btuh/Watt at 10°F TD			
Defrost Method	Freezer: Hot Gas Cooler/Dock: Off-Cycle			
Defrost Frequency/Duration	All units: (2) defrosts/day, 30 minutes/defrost			
Air Unit Fan Operation	All units: fans run continuously (except during defrost). Fan speed controlled according to space temperature (entering coil air temperature). 70% minimum speed. Fans forced to 100% speed for two non-consecutive hours/day in simulation to reflect real-world variations in fan speed.			

Figure 40: Base case facility description

8. Appendix C: Measure Cost

Cost calculators for the measures evaluated in this report are presented below.

8.1 Freezer Roof Measure Cost

Four different insulating materials were evaluated for the freezer roof insulation measure: polyurethane panels, expanded polystyrene panels, urethane cam-lock panels and polyisocyanurate overdeck insulation. The general cost calculation method for this measure was to first produce a polynomial regression of end-user cost per square-foot versus insulation thickness (or R-value), then produce a polynomial regression curve of prototype warehouse energy usage versus insulation thickness (or simulated R-value). Simultaneous analysis of the two regressions permitted calculation of cost-effectiveness for incremental increases in insulation thickness. Figure 41, Figure 42, and Figure 43 show regression analysis for urethane cam-lock panels, polyurethane panels, expanded polystyrene panels, and polyisocyanurate overdeck insulation.

Prefabricated Cam-Lock Building Costs

Prefab Building Dimensions

Length:	199.42	ft
Width:	199.42	ft
Height:	30.17	ft

Total Wall Area:	24,063	SF
Total Roof Area:	39,767	SF
Total Insulation Area:	63,830	SF

Insulation Thickness (in):	6	5	4	
Quoted Price:	$ 545,027	$ 473,758	$ 426,246	Notes:
Cost/SF:	$ 8.54	$ 7.42	$ 6.68	
Cost/SF/inch thickness:	$ 1.42	$ 1.48	$ 1.67	

Costs are for panels and cam-lock mechanisms ONLY--costs do not include building structure. Costs are from the factory to a reseller and do not include sales tax.

Shipping Costs

Shipping Distance (NC to CA) (miles):	2967	
Cost/mile/truck:	$1.57	Number provided by prefabricated building manufacturer
Truck capacity:	20,000	lbs (number provided by prefabricated building manufacturer)

Total insulated panel weight (lbs):	360,424	336,395	300,353	Notes:
Weight/SF:	5.65	5.27	4.71	
# of trucks to ship:	19	17	16	
Shipping cost:	$88,506	$79,189	$74,531	
Shipping cost/SF:	$1.39	$1.24	$1.17	
Shipping cost/SF/inch thickness:	$0.23	$0.25	$0.29	

Shipping cost and methodology from prefabricated building manufacturer

End-User Costs

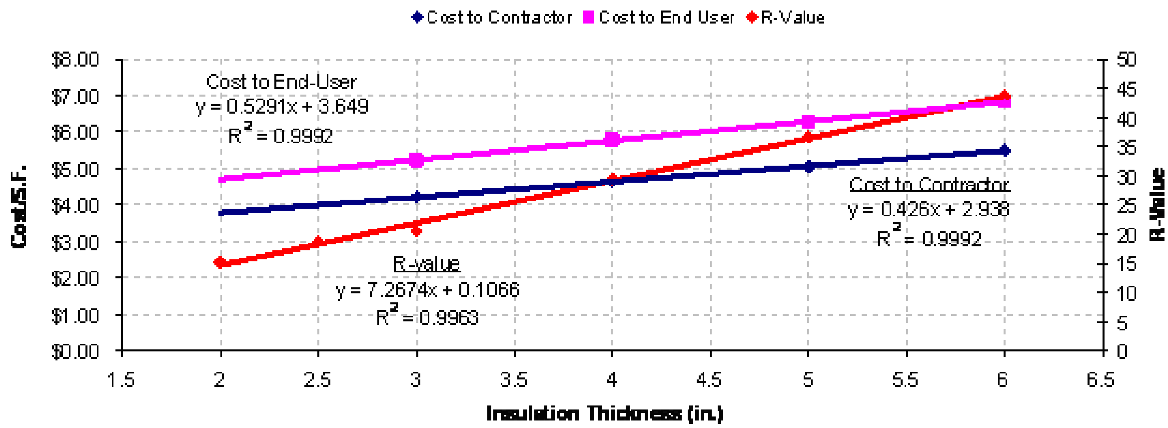
Reseller Mark-Up:	20%	(est)
Contractor Mark-up:	20%	(est)

Thickness (inch)	6	5	4
Total Cost	$873,344	$761,401	$688,325
Cost/SF	$13.68	$11.93	$10.78
Cost/SF/inch thickness	$2.28	$2.39	$2.70

Figure 41: Cost calculation worksheet for prefabricated urethane cam-lock panels

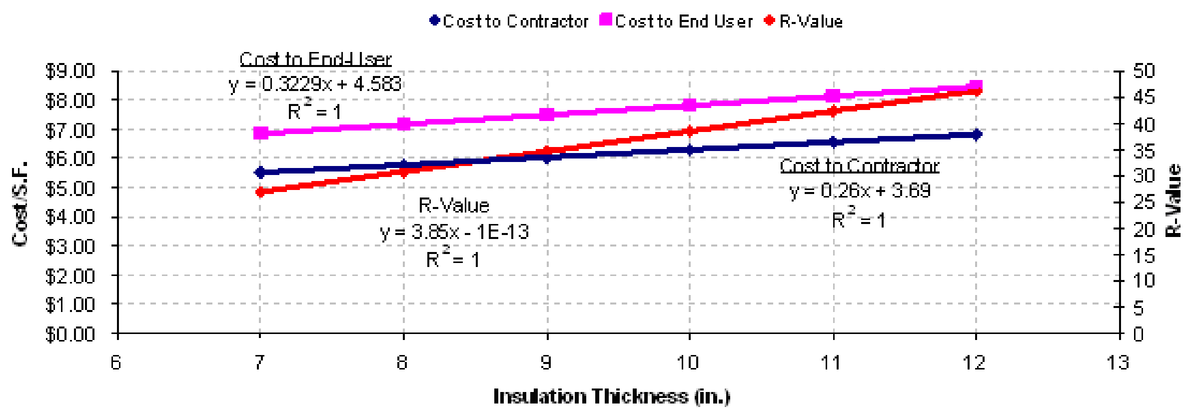
Polyurethane

Thickness	2	2.5	3	4	5	6
R-Value	15.1	18.7	20.5	29.4	36.6	43.7
Cost to contractor ($/S.F)			$4.21	$4.66	$5.05	$5.50
Cost to end-user ($/S.F)			$5.23	$5.79	$6.27	$6.83



Expanded Polystyrene

Thickness	7	8	9	10	11	12
R-Value	26.95	30.8	34.65	38.5	42.35	46.2
Cost to Contractor ($/S.F)	$5.51	$5.77	$6.03	$6.29	$6.55	$6.81
Cost to end-user ($/S.F.)	$6.84	$7.17	$7.49	$7.81	$8.14	$8.46

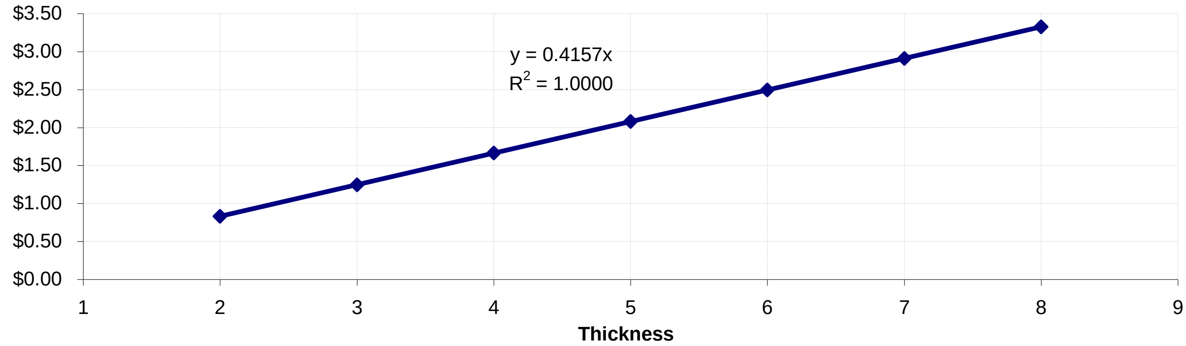


Notes:
R-value based on 75°F mean temperature, based on ASTM test method.
Costs to contractors do not include taxes, contractor mark-ups, structural supports, or accessories.
End-user costs are installed costs, including taxes, contractor mark-ups, structural supports, etc.
Costs to contractors were based on interviews with both contractors and insulation manufacturers.
Contractor indicated that there is economy of scale with installed costs--large projects can be up to 30% cheaper per S.F. for insulation, especially if multiple contractors are bidding on project. Assumed 8% sales tax and 15% contractor mark-up.

Figure 42: Cost calculation worksheet for urethane and expanded polystyrene panels

Overdeck Insulation

Thickness	2	3	4	5	6	7	8
Cost from mfr to first-time buyer	$0.64	$0.96	$1.28	$1.60	$1.92	$2.24	$2.56
Cost to end-user ($/S.F)	$0.83	$1.25	$1.66	$2.08	$2.49	$2.91	$3.33



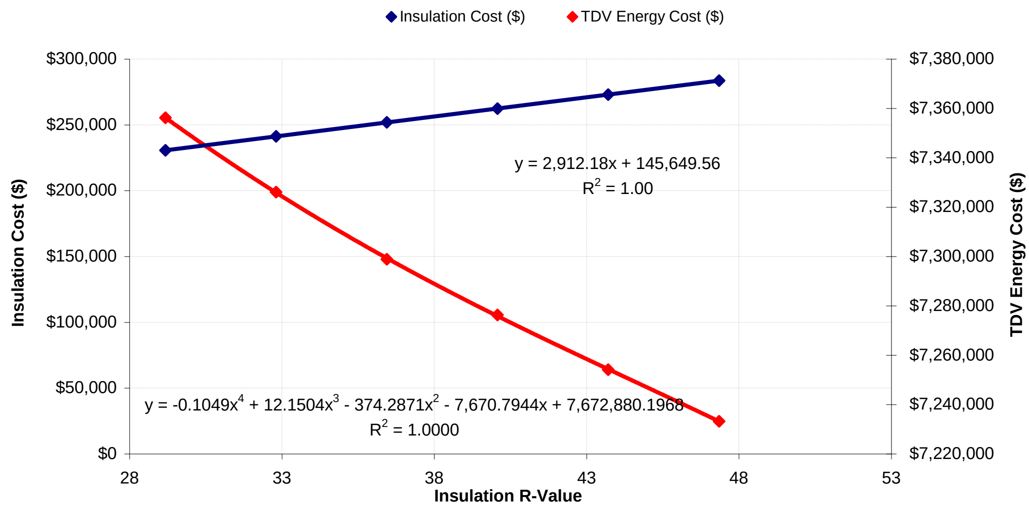
Notes

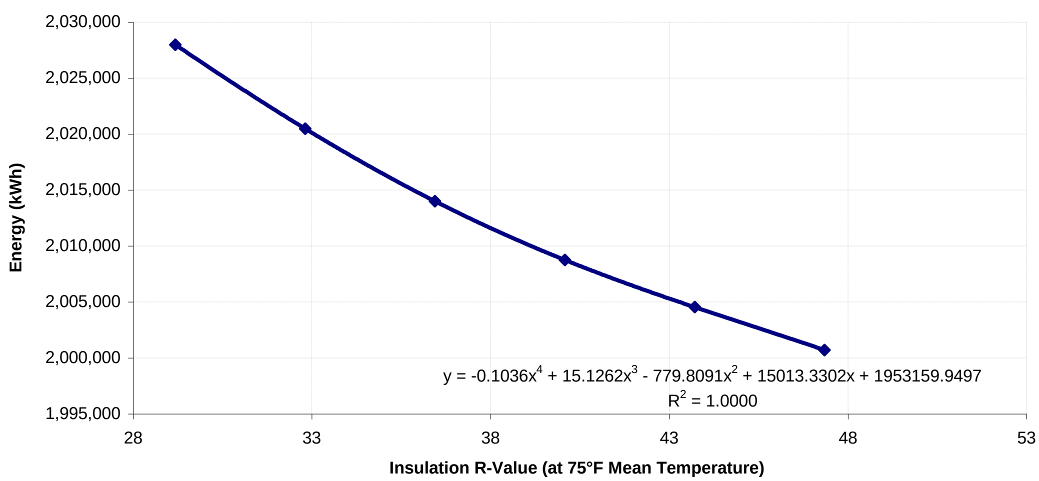
R/inch is about 6.25 based on a test method called LTTR (long-term thermal resistance), in which they take a wafer of iso board and prematurely age it, then test the resistance. Cost per SF for a 1" increment is about $0.30-$0.32 from the manufacturer to the first-time buyer (usually a distributor) for a larger warehouse project. Distributor mark-up is about 5-7% if the board is direct-shipped from the factory to the job site, and about 20% if the board has to be warehoused by the distributor. Usually the product is direct-shipped. Distributor costs can vary by about 10% or so based on the number of shipments required.

Figure 43: Cost calculation worksheet for polyisocyanurate overdeck insulation

Figure 44 shows an example of the simultaneous analysis method of calculating the BC ratio for incremental increases insulation thickness. The example shown is for the large warehouse (Prototype Warehouse #2) with polyurethane panel insulation.

Insulation Thickness (in.)	R-Value at 75°F MTD	kWh	TDV Electricity (Mbtu)	Total TDV Cost ($)	Insulation Cost ($/S.F.)	Total Insulation Cost ($.)
4	29.18	2,027,983	47,771	$ 7,356,186	$5.765	$230,616
4.5	32.81	2,020,480	47,575	$ 7,326,097	$6.030	$241,198
5	36.44	2,013,995	47,398	$ 7,298,887	$6.295	$251,780
5.5	40.08	2,008,749	47,251	$ 7,276,250	$6.559	$262,362
6	43.71	2,004,535	47,108	$ 7,254,091	$6.824	$272,944
6.5	47.34	2,000,707	46,972	$ 7,233,225	$7.088	$283,526





Coefficients from above graphs

	c0	c1	c2	c3	c4
Insulation Cost:	145,649.56	2,912.18			
Building Energy:	1953159.9497	15013.3302	-779.8091	15.1262	-0.1036
TDV Cost:	7672880.1968	-7670.7944	-374.2871	12.1504	-0.1049

	Base Case	Proposed	**Difference**	Difference /SF
R-Value	36	40	**4**	
Energy Usage (kWh)	2,014,727	2,008,859	**5,868**	**0.147**
TDV Utility Cost ($)	$7,302,353	$7,276,271	**$26,082**	**$0.652**
Insulation Cost ($)	$250,488	$262,137	**$11,649**	**$0.291**
		Benefit/Cost Ratio:	**2.239**	

Figure 44: Example simultaneous analysis of cost regression and building energy use regression.

Freezer Floor Measure Cost

Expanded polystyrene was the only freezer floor insulation measure evaluated. The general method for calculating cost for this measure was the same as the roof insulation measure: produce a polynomial regression of end-user cost per square-foot versus insulation thickness (or R-value), then simultaneously analyze the regression with a polynomial regression curve of prototype warehouse energy usage versus insulation thickness (or R-value). Figure 45 shows regression analysis worksheet for expanded polystyrene.

Extruded Polystyrene (Floor)

Thickness	3	4	5	6	7	8
R-Value	15	20	25	30	35	40
Cost to end-user ($/S.F)	$1.46	$1.94	$2.43	$2.91	$3.40	$3.88

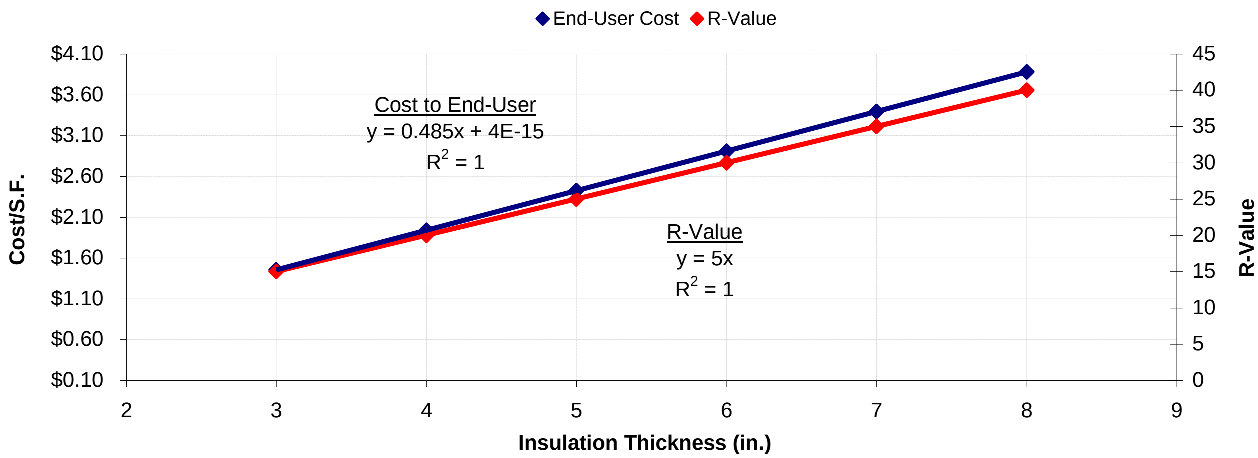


Figure 45: Cost regression analysis for expanded polystyrene floor insulation

Extruded polystyrene R-value is R-5.0/inch at 75oF mean temperature and R-5.4/inch at 40oF per ASTM C518. Prices were estimated by a refrigeration subcontractor and include installation labor but not vapor barriers or adhesives. Contractor indicated that cost will vary based on floor size. Contractor mark-up was included in quoted price.

Costs according to compressive strength:
30 psi: $0.44 per S.F
40 psi: $0.53 per S.F
60 psi: $0.64 per S.F

The prices used for analysis are based on the average cost of 30 psi and 40 psi panels which are the most common insulation board compressive strengths used in refrigerated warehouses.

8.2 Evaporator Fan Control for Single Cycling-Compressor Systems

Two methods of fan control were analyzed for the evaporator fan control measure: fan speed control and fan staging control. Fan speed control was assumed to be the most expensive option, so the costs associated with installing variable-speed drives were used in the cost-effectiveness analysis. Figure 46-Figure 49 summarize the first-costs and maintenance cost assumptions for fan speed control and for fan staging control, respectively.

VFD Drive Option (considered most expensive option)

Use microcontroller , which would typically also
be used for temperature and defrost control
Some coolers would be electric defrost, assumed more expensive controller
with defrost load capacity.
Variable speed requires additional technical support and related costs
Assume fan motors are 460 V 3 phase
Assume 1 microcontroller per condensing unit, 3 condensing units for cooler space
Assume 3 condensing units for cooler space, 7 for freezer space, 3 for dock space
Assume 1 VFD per evaporator, at $600/VFD.
Assume 1 sine filter per VFD, $300/sine filter
Assume 6 evaporators in cooler space, 7 in freezer, 6 in dock

Cooler:	Microcontroller cost:	$ 140.00
	Deduct thermostat and time clock	$ (77.00)
	Number of Required Microcontrollers	3
	Total Microcontroller cost	$ 189.00
	VFD cost	$ 3,600.00
	Sine Filter	$ 1,800.00
	Additional technical costs	$ 500.00
	Total:	$ 6,089.00
	Manufacturer mark-up:	50%
	Contractor mark-up and tax:	30%
	Cost to owner:	**$ 11,873.55**
Freezer:	Microcontroller cost:	$ 140.00
	Deduct thermostat and time clock	$ (77.00)
	Number of Required Microcontrollers	7
	Total Microcontroller cost	$ 441.00
	VFD cost	$ 4,200.00
	Sine Filter	$ 2,100.00
	Additional technical costs	$ 500.00
	Total:	$ 7,241.00
	Manufacturer mark-up:	50%
	Contractor mark-up and tax:	30%
	Cost to owner:	**$ 14,119.95**
Dock:	Microcontroller cost:	$ 140.00
	Deduct thermostat and time clock	$ (77.00)
	Number of Required Microcontrollers	3
	Total Microcontroller cost	$ 189.00
	VFD cost	$ 3,600.00
		$ 1,800.00
	Additional technical costs	$ 500.00
	Total:	$ 6,089.00
	Manufacturer mark-up:	50%
	Contractor mark-up and tax:	30%
	Cost to owner:	**$ 11,873.55**
	Total First Cost to Owner:	**$ 37,867.05**

Figure 46: Measure cost calculator for fan speed control

Maintenance Costs for VFD Drive Option

Yearly maintenance (not including equipment replacement)
Estimate 1/2 hour/condensing unit per year

Hours per year:	6.50	
Labor cost:	$60	/hour
Total Labor Cost:	$390	/year

Equipment replacement cost: assume 1 VFD/year

Cost per VFD:	$600	
Time to Replace:	4	hours/VFD
Labor cost:	$60	/hour
Total Replacement Cost:	$840	/year

Total Maintenance Cost:	$1,230
Discount Rate:	3%
15-year present value of maintenance costs:	**$15,124**

Figure 47: Maintenance cost calculator for fan speed control

Fan Cycling Option

Separate Terminal Blocks and Extra Relays:	$300	per evaporator
Installation Time (hours):	1	
Labor Cost:	$60	
Taxes and Permits:	10%	
Contractor Mark-up:	20%	
Total Cost per evaporator	$456	
Number of Evaporators	19	
Total First Cost to Owner:	**$8,664**	

Figure 48: Measure cost calculator for fan staging control

Maintenance Costs for Fan Cycling Option

Yearly maintenance (not including equipment replacement)
Estimate 1/2 hour/condensing unit per year

Hours per year:	6.50	
Labor cost:	$60	/hour
Total Labor Cost:	$390	/year
Discount Rate:	3%	
15-year present value of maintenance costs:	**$4,795**	

Figure 49: Maintenance cost calculator for fan staging control

8.3 Condenser Specific Efficiency

The methodology for calculating costs for the condenser specific efficiency measure is detailed in Section 4.3. Figure 50-Figure 53 illustrate the assumed condenser cost versus capacity at specific efficiency rating conditions for each condenser type analyzed. Condenser costs for air-cooled halocarbon condensers and evaporative-cooled centrifugal-fan halocarbon condensers were based on catalog costs multiplied by typical contractor multipliers ranging from 0.22 to 0.30, depending on the equipment manufacturer. Contractor multipliers were obtained through contractor and vendor interviews, and represent a typical multiplier value for a national contractor in good standing with the equipment manufacturer. Axial-fan evaporative-cooled ammonia condenser costs were obtained directly from equipment manufacturers, and were assumed to already have the contractor multiplier factored into the cost. Costs for all units assumed a 15 percent contractor mark-up, an 8 percent sales tax and a 5 percent delivery cost. Figure 50-Figure 53 show the cost-regressions for the condenser types included in this analysis.

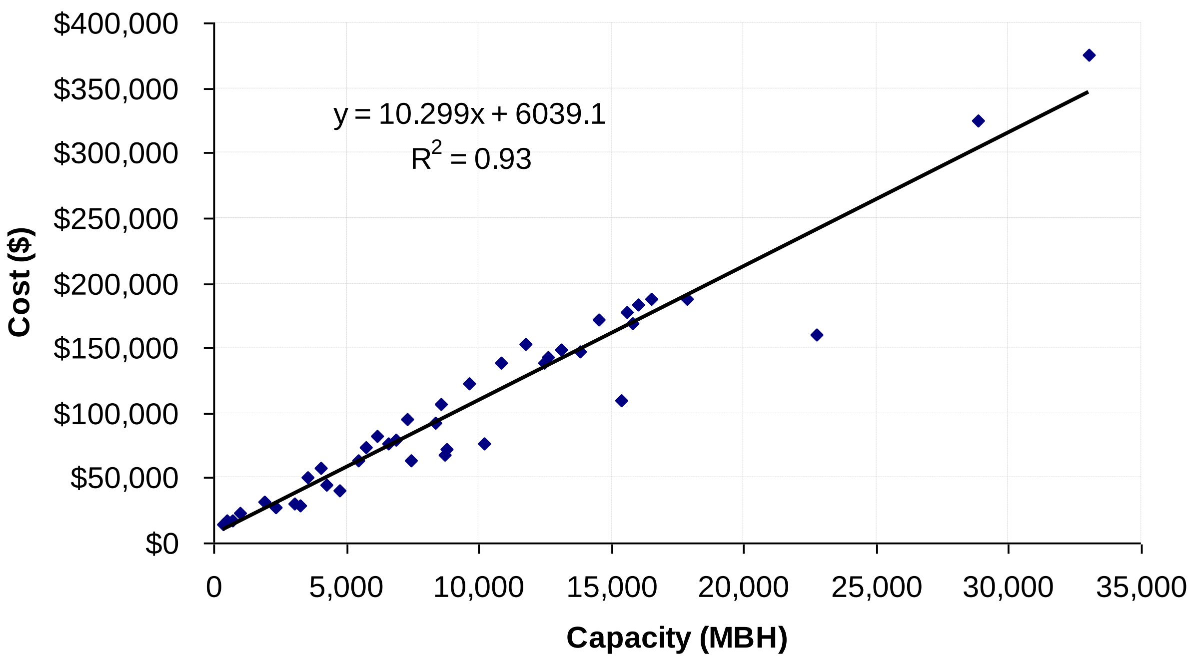


Figure 50: Cost versus capacity regression at specific efficiency rating conditions for axial-fan evaporative-cooled ammonia condensers

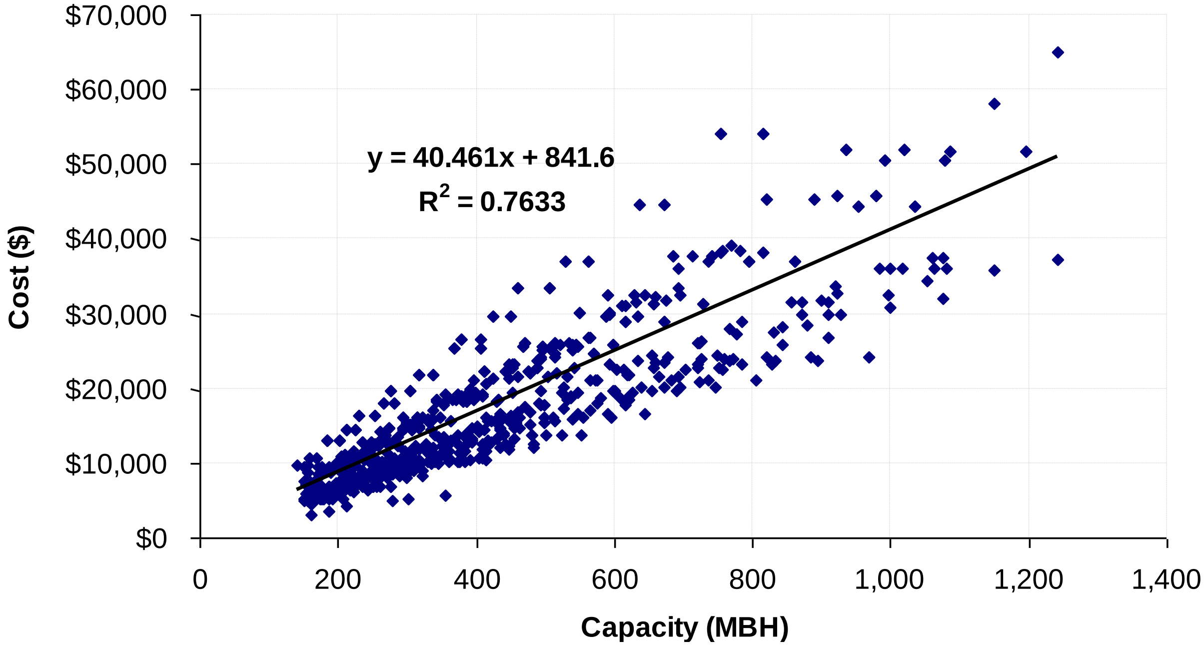


Figure 51: Cost versus capacity regression at specific efficiency rating conditions for axial-fan air-cooled HFC condensers with standard motors

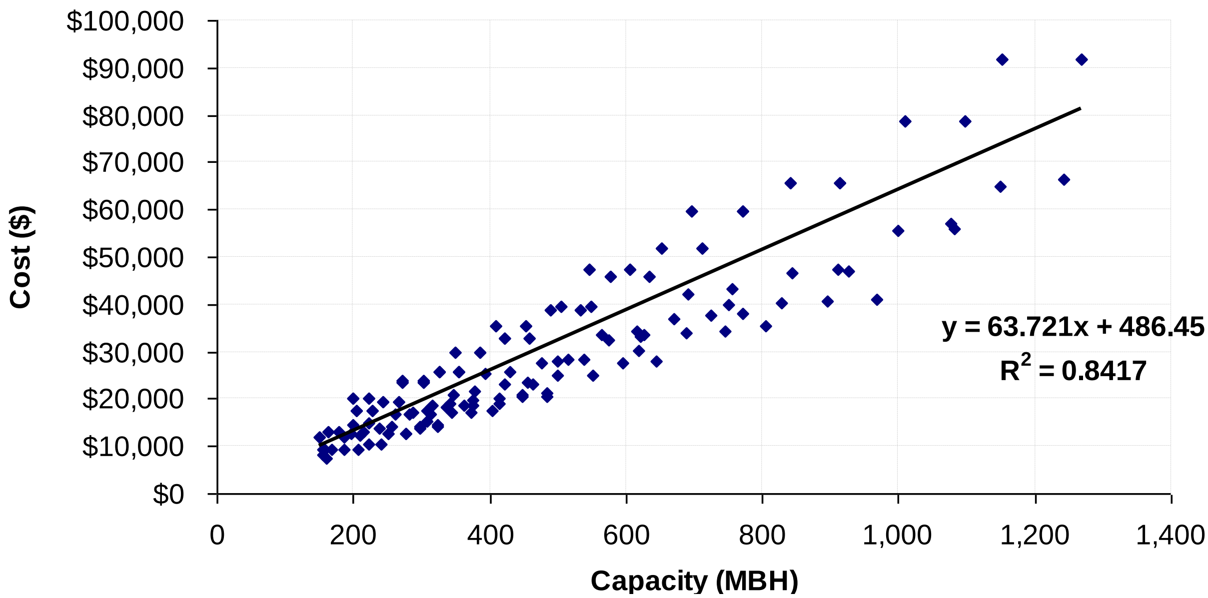


Figure 52: Cost versus capacity regression at specific efficiency rating conditions for axial-fan air-cooled HFC condensers with BLDC motors

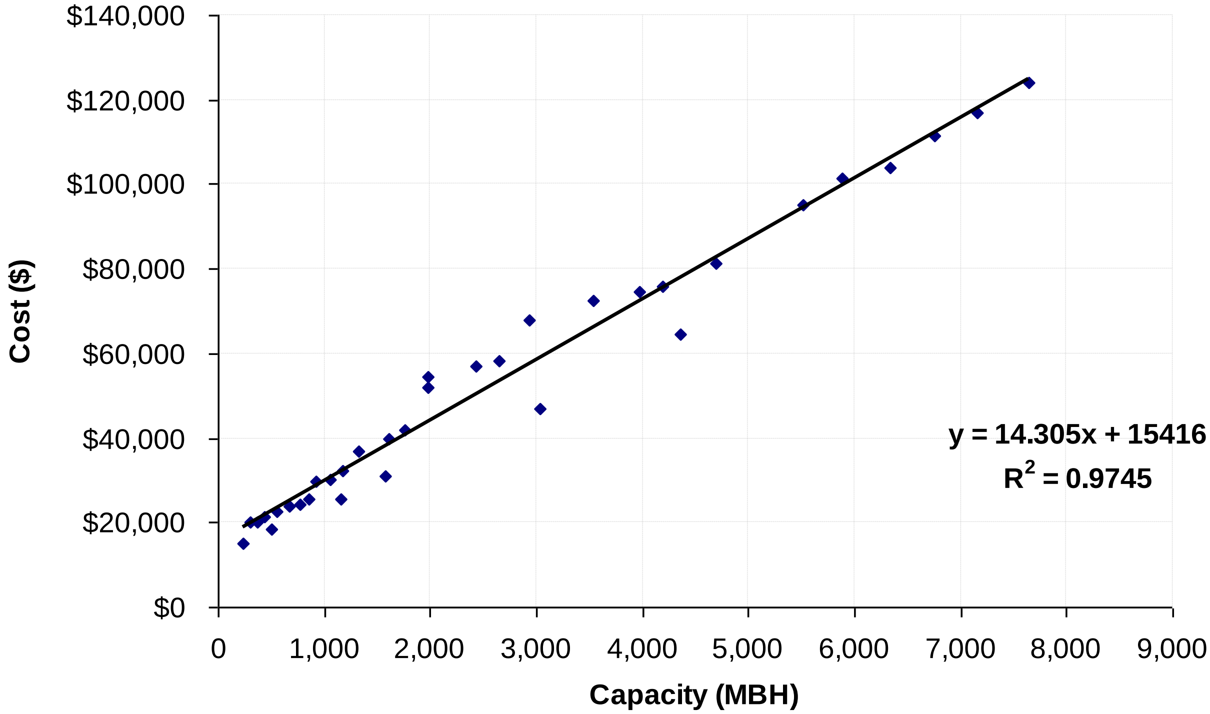


Figure 53: Cost versus capacity regression at specific efficiency rating conditions for centrifugal-fan evaporative-cooled HFC condensers

8.4 Screw Compressor Part-Load Analysis

The screw compressor part-load analysis assumed the cost of an appropriately-sized VFD, rather than the cost of a soft-starter in the base case. Costs were obtained from the four largest screw compressor manufacturers in the state. Both the VFD and soft-start costs were assumed to be costs to the end-user, and thus included typical contractor multipliers. An 8 percent sales tax was assumed. Figure 54 to Figure 56 describe the assumptions for the screw compressor part-load analysis measure.

Materials	Cost (est)	Contractor Margin	Total
PLC control, interface wiring and additional panel materials:	$2,500	0.35	$3,846
Additional electrical wiring materials:	$800	0.35	$1,231

Additional Labor and Subcontracts (vs. Soft-Start)	Labor Rate	Hours	Total
Electrical installation labor:	$32	65	$2,080
Programming, start-up and fine-tuning labor:	$60	85	$5,100

Electrical installation labor Includes mounting additional panel, conduit deltas and tie-ins
Programming, start-up and fine-tuning labor includes dditional logic for VFD and bypass operation

Figure 54: Additional materials and labor assumptions for variable-frequency drives versus soft-starts.

	Nom. HP	Applied Power (HP)	Capacity (TR)	Power/Capacity (HP/TR)	VFD Cost*	Soft-Start Cost*	Total VFD Cost	Total Soft-Start Cost	Difference
LT System									
Vendor 1	350	315.0	130.4	2.42	$33,533	$2,357	$48,473	$2,546	$45,927
Vendor 2	350	338.7	140.9	2.40	$30,635	$4,950	$45,343	$5,346	$39,997
Vendor 3	350	323.5	138.8	2.33	$32,272	$7,655	$47,111	$8,267	$38,843
Vendor 4	350	339.0	132.3	2.56	$30,588	$5,352	$45,292	$5,780	$39,512
						AVERAGE:	$46,554	$5,485	$41,070
MT System									
Vendor 1	250	242.7	220.4	1.10	$28,619	$2,314	$43,165	$2,499	$40,666
Vendor 2	250	240.7	219.1	1.10	$23,545	$4,280	$37,686	$4,622	$33,063
Vendor 3	250	232.9	215.9	1.08	$24,918	$6,354	$39,168	$6,862	$32,306
Vendor 4	250	229.4	219.4	1.05	$23,087	$4,629	$37,191	$4,999	$32,192
						AVERAGE:	$39,303	$4,746	$34,557
Booster System									
Vendor 1	125	105.6	122.0	0.87	$20,449	$2,428	$34,342	$2,622	$31,720
Vendor 2	125	110.8	134.1	0.83	$16,595	$3,315	$30,180	$3,580	$26,599
Vendor 3	150	125.9	141.0	0.89	$20,361	$6,080	$34,247	$6,566	$27,680
Vendor 4	125	109.8	128.2	0.86	$15,731	$3,588	$29,246	$3,875	$25,371
						AVERAGE:	$32,004	$4,161	$27,843

* NOTE: VFD and Soft-Start costs already include contractor multipliers

Figure 55: Screw compressor part-load measure cost calculator for LT, MT, and booster suction groups

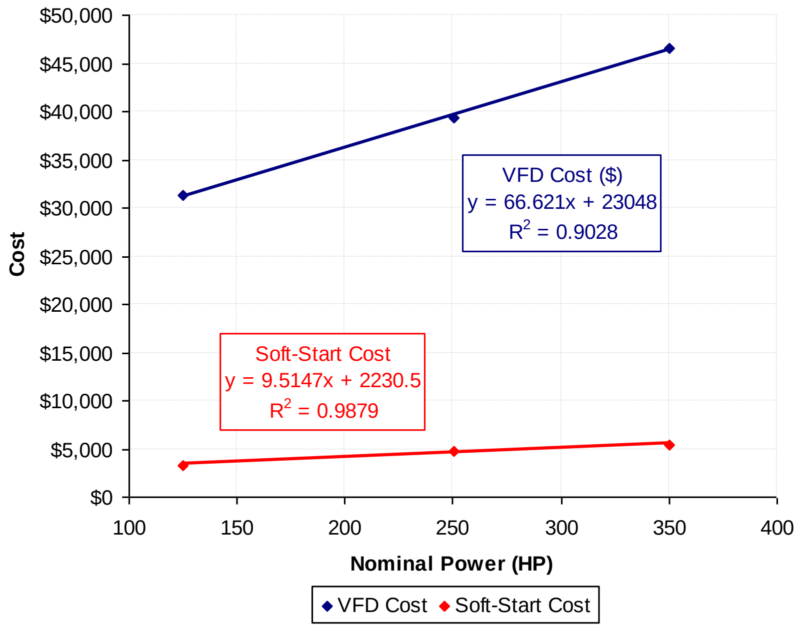


Figure 56: Cost versus motor horsepower regressions for screw compressor speed control

8.5 Infiltration Barriers

Several different infiltration barrier types were evaluated for this analysis, including manual hard doors, strip curtains, low- and high-speed vertical and bi-parting doors, and air curtains. Costs were obtained from one or more vendors for each barrier type. Figure 57 - Figure 60 are cost calculators made to quantify costs for each infiltration barrier type.

Manual Door

Sales Tax:	8%
Shipping:	25%

	Hours/Door	Labor Rate	Total
Installation:	1	$35	$35

Cost per Door	Tax	Shipping	Installation	Total Cost
$2,000	$160	$500	$35	$2,695

	Measure Cost
Small Warehouse:	$2,695.00
Large Warehouse:	$5,390.00

Figure 57: Cost assumptions for manual hard doors

Strip Curtains

Equipment Cost:	$6.68	per S.F. based on DEER analysis work
Installation Cost:	$2.86	per S.F. based on DEER analysis work

	Small Warehouse	Large Warehouse
Total Doors per Building:	1	2
Door Area (SF/door):	100	100
Total Area per Building:	100	200
Equipment Cost:	$668.00	$1,336.00
Installation Cost:	$286.00	$572.00

	Measure Cost
Small Warehouse:	$954.00
Large Warehouse:	$1,908.00

Figure 58: Cost assumptions for strip curtains

Standard-Speed Automatic Door

Sales Tax: 8%

Manufacturer	Model	Type	Opening Speed (in/sec)	Cost	Tax	Shipping	Installation	Total Cost
A	1	Bi-Parting	60	$13,995	$1,120	$950	$1,980	$18,045
A	2	Rollup	50	$11,595	$928	Included	Included	$12,523
							AVERAGE:	$15,284

Installation and shipping estimated by vendors

	Measure Cost
Small Warehouse:	$15,283.60
Large Warehouse:	$30,567.20

High-Speed Automatic Door

Sales Tax: 8%

Make	Model	Type	Opening Speed (in/sec)	Cost	Tax	Shipping	Installation	Total Cost
B	1	Rollup	96	$13,439	$1,075	$1,200	$1,750	$17,464
B	2	Rollup	96	$12,258	$981	$1,200	$1,750	$16,189
A	3	Rollup	100	$13,990	$1,119	$850	$2,075	$18,034
							AVERAGE:	$17,229

Installation and shipping estimated by vendors

	Measure Cost
Small Warehouse:	$17,228.99
Large Warehouse:	$34,457.97

Figure 59: Cost assumptions for standard- and high-speed automatic doors

Air Curtains

Sales Tax:	8%
Shipping:	25%

	Hours/Door	Labor Rate	Total
Installation:	1	$35	$35

Material Cost per Door	Tax	Shipping	Installation	Total Cost
$2,000	$160	$500	$35	$2,695

	Measure Cost
Small Warehouse:	$2,695.00
Large Warehouse:	$5,390.00

Figure 60: Cost assumptions for air curtains

9. Appendix D: Industry Interviews and Market Research

The following information summarizes the industry interviews and market research conducted as part of this study.

9.1 Insulation

Refrigerated warehouses in California are typically steel-framed buildings or concrete construction (either concrete block or concrete tilt-up construction). For concrete refrigerated warehouses, the insulation panels may be on the exterior of the support structure (structurally interior), or on the inside (structurally exterior). Local building codes may dictate the type of refrigerated warehouse construction allowed in some cities or counties. For example, Los Angeles County requires concrete tilt-up or block construction with a box-within-a-box insulation configuration, whereas steel-framed buildings are allowed in the nearby inland empire.

According to two contractors, prefabricated polyurethane foam-in-place panels are used in nearly all projects featuring insulation exterior to the building structure. For interior insulation, approximately 75 percent of new construction projects currently utilize polyurethane foam-in-place panels. Expanded polystyrene boards are typically the second most prevalent insulation type.

9.1.1 Rated R-Values

Insulation is typically rated according to either a 40°F or 75°F mean temperature difference. All insulation materials have a conductivity curve; insulation will exhibit higher real thermal resistance at lower mean temperature differences. R-values at 75°F mean temperature are the basis for Title 24 compliance, as described in Part 6 of California Title 24.

There is some disagreement in the industry regarding rated R-values for urethane, polyisocyanurate, and expanded polystyrene. All insulation vendors and contractors interviewed stated that prefabricated polyurethane foam-in-place panels currently have the majority of the refrigerated warehouse insulation market share. Contractors stated that polyurethane is typically selected by clients due to the fact that published R-values for polyurethane are disproportionately higher than the cost on a per-inch basis than polyisocyanurate or expanded polystyrene. The rated R-values for polyurethane panels typically represent new-condition material, whereas polyisocyanurate manufacturers publish aged R-values. One polyurethane panel vendor stated that polyurethane panel performance degrades much slower than polyisocyanurate due to the manufacturing process used in the foam-in-place polyurethane panel industry. The degradation of insulation effectiveness is caused by gas escaping the insulating material as it ages; since polyurethane is 'foamed in place' between metal skins, the gas is effectively trapped, preventing degradation. Therefore it is acceptable that the polyurethane manufacturers publish new-condition R-values. Polyisocyanurate board manufacturers, however, contest this assumption. One polyisocyanurate manufacturer stated that the assumption that the metal skins trap escaping gas is true only if the bond between the insulation and the metal skin is perfect, which is generally not true. Small voids created from imperfect bonding between the foam-in-place polyurethane and the metal skin allows gas build-up and degradation of the panel's insulating properties. Furthermore, the polyurethane insulation is exposed on the edges of a foam-in-place panel, so trapping all gas is impossible.

9.1.2 Miscellaneous Insulation Comments from Contractors and Vendors

- All contractors interviewed for this analysis indicated that the industry-standard practice for selecting interior, inter-zonal insulation between adjacent refrigerated spaces is to use insulation with the same R-value as the exterior walls of the colder space. Contractors stated that refrigerated spaces should be insulated assuming that adjacent refrigerated spaces will be converted to unconditioned areas. Contractors noted that coolers are often converted to dry storage or conditioned work areas, which leads to problems with under-insulation and condensation on adjacent freezer walls if less resistive insulation was used.
- It is not possibly to comply with the 2008 Title 24 freezer floor insulation requirement (R-36) using non-custom insulation available on the market. Extruded polystyrene insulation, the material of choice for freezer floor insulation due to its high compressibility strength, is typically available in 2” and 3” thick boards. Since polystyrene floor insulation boards can be stacked, a combination of 2” and 3” boards can be used to make any combined thickness in 1” increments. The rated R-value of extruded polystyrene is R-5/inch. In order to abide by 2008 Title 24 requirements without using custom insulation panels, floors need to be insulated to R-40.
- Insulation cost is an economy of scale. Savings can be up to 30 percent on large buildings, especially if many contractors are bidding on a project.
- Roof insulation R-values are sometimes a consequence of structural concerns rather than thermal insulation concerns. Thicker insulation panels can span longer distances with less structural support than thinner panels. If the cost of a thicker panel is cheaper than the cost of a thinner panel plus the cost of structural supports, the thicker panel will be selected for the project.

9.2 Infiltration Barriers

Figure 61 outlines the opening speed of refrigerated warehouse doors offered by the four manufacturers surveyed as part of this analysis.

Manufacturer	Door Type	Opening Speed	Maximum Dimensions
A	Bi-parting (sliding doors)	84"/ second combined	10' W x 16' H
A	Bi-parting (folding doors)	84"/ second combined	35' W x 25' H (Standard Model) 12' W x 16' H (Freezer Model)
A	Rollup	50"/ second	30' W x 24' H
A	Rollup	50"/ second max (variable)	16' W x 15' H
A	Rollup	100"/ second max (variable)	12' W x 16' H
A	Rollup	101"/ second average	12' W x 16' H
B	Rollup	100"/ second (variable)	
B	Bi-parting (side-rolling doors)	120"/sec combined	
B	Bi-parting (sliding doors)	60"/sec combined	
B	Rollup	48"/ second	14' W x 14' H
C	Rollup	96"/ second	39' W x 18' H
C	Rollup	96"/ second	15' W x 15' H
D	Bi-Parting	96”/second combined	
D	Bi-Parting	80”/second combined	

Figure 61: Survey of door opening speeds

For rollup doors, two of the four manufacturers surveyed offered a “standard speed” door, with published door opening speeds of 48-50 inches per second. Published door opening speed of 96-101 inches per second were available in the “high speed” models. For bi-parting doors, one manufacturer offered a standard-speed door with a published combined opening speed of 60 inches per second, and a 120 inch per second high-speed option. The remaining manufacturers offered bi-parting opening speeds of 80, 84, and 96 inches per second. For all manufacturers, the typical door closing speed was restricted to 48-50 inches per second due safety concerns. For low-temperature applications, three of the four manufacturers interviewed recommended heated blower elements to reduce condensate on the warm side of the door, and to prevent ice buildup on the door mechanisms which might prevent the door from operating properly. The blowers circulate warm air over the warmer side of the door. The fourth manufacturer instead recommended a door with a higher insulation R-value, which eliminates the need for warm air blowers, infra-red heaters, or electric resistance mechanism heaters which consume energy as well as increase the load in the refrigerated space. This manufacturer stated that, in general, heating elements are added to doors that were designed for general (unrefrigerated) industrial work. Instead of a heating element, the manufacturer offered a door insulation R-value of R-32 for freezer applications, where the TD across the door was greater than 60°F. An R-10 door is recommended for other applications.

Figure 62 outlines one manufacturer’s guidelines for selecting the appropriate infiltration barrier based on the percent of time the door is open:

Door Open Time (%)	Recommended Door Type
<1-4	Hard door (non-hittable), no automatic opening mechanism
5-15	Hittable door with automatic opening mechanism
11-20	Hybrid system: automatic bi-parting or single sliding door with horizontal air curtain
>21	Double horizontal air curtain with manual door closed only during non-business hours (when air curtains are off), or other wide-open passageway solution.

Figure 62: One manufacturer’s infiltration barrier recommendations according to % door open time

9.3 Condenser Specific Efficiency

9.3.1 Evaporative Condenser Specific Efficiency

The proposed code language will mandate a minimum specific efficiency for evaporative condensers based on the installed location of the condenser (160 Btuh/Watt for indoor condensers, 350 Btuh/Watt for outdoor condensers). Indoor condensers are embodied by centrifugal-fan condensers while outdoor condensers are embodied by axial condensers. There are no centrifugal-fan condensers on the market that are capable of meeting the outdoor condenser efficiency requirement. This is a problem, as end users might require a centrifugal-fan condenser due to noise restrictions, needing multiple circuits, static pressure concerns, or needing a smaller condenser.

Industry Interviews

When questioned about the problem stated above, condenser manufacturers offered the following information:

I. *Noise Concerns* – One manufacturer commented that low-sound axial fans are available, but not in the low-capacity sizes that are comparable to centrifugal units. Axial fan units can also be sold with sound baffles, but these add cost and mitigate performance.

II. Multi-Circuiting – Three major evaporative condenser manufacturers both commented that axial-fan units of any size can be made with multiple circuits, within reason. There would be limits to the number of circuits that can physically fit into smaller packages.

III. Static Pressure –Two major manufacturers stated that they do not have a product that can be used in an outdoor application where static pressure is a concern that can meet the 350 Btuh/Watt requirements.

IV. Size Restrictions – Three major manufacturers recently introduced new induced-draft axial-fan product lines small enough to overlap with centrifugal fan product lines. Figure 63 shows the size limitations of the mentioned product lines compared to the size limits of centrifugal-fan condensers available on the market from the major centrifugal-fan condenser manufacturers.

	Minimum Size (MBH)	Maximum Size (MBH)
Centrifugal Condensers	232.7	7,644.0
Axial-Fan Condensers	639.5	59,073.4

Figure 63: Minimum and maximum condenser catalog capacities for centrifugal-fan evaporative condensers and small axial-fan evaporative condensers

As Figure 63 shows, there is a gap in availability between 233 MBH and 640 MBH where only centrifugal-fan condensers are available. Up to 7,644 MBH, both centrifugal and axial-fan condensers are available, and only axial-fan condensers are available above 7,644 MBH.

Figure 64 maps specific efficiency for the subject models in the 640 to 7,644 MBH range, where both axial-fan and centrifugal-fan condensers are available on the market.

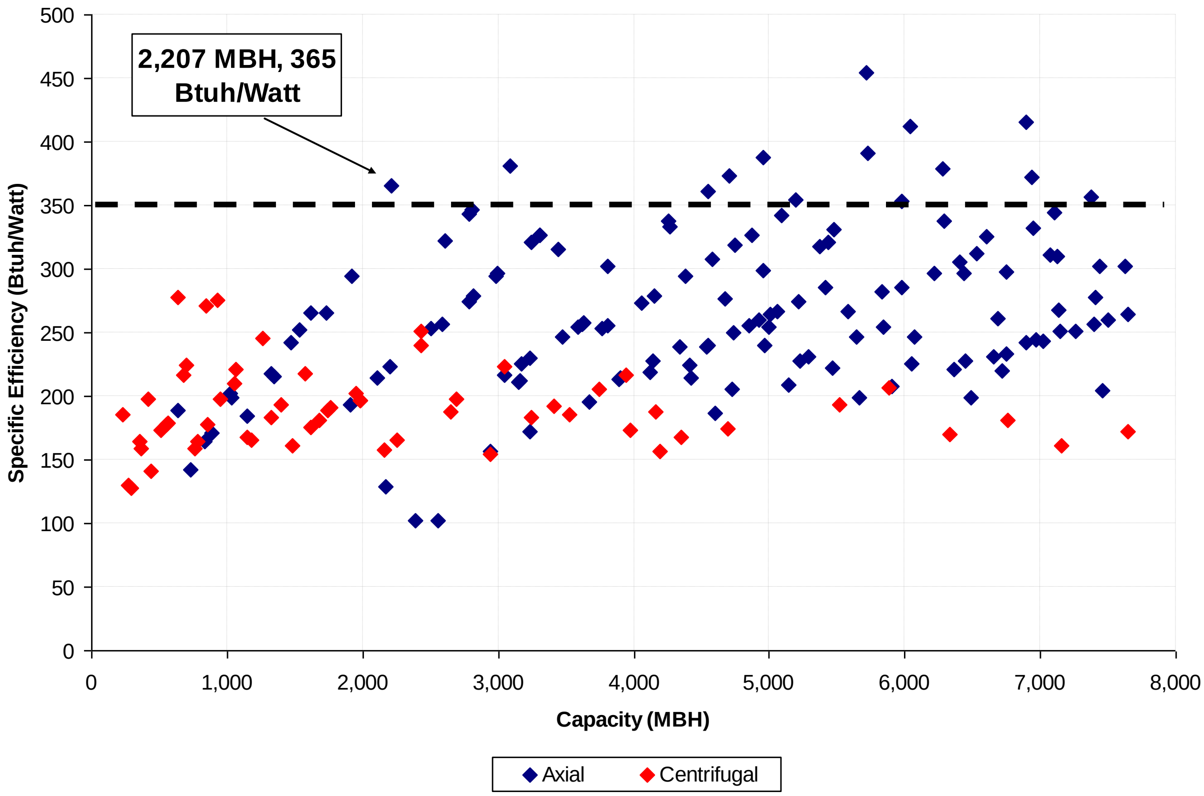


Figure 64: Specific efficiency of centrifugal-fan and small axial-fan evaporative condensers at 100°F SCT, 70°F WBT

Figure 64 shows that the centrifugal fan condenser population is more heavily-weighted at smaller sizes, where axial-fan evaporative condensers are more heavily-weighted at larger sizes. Furthermore, axial fan condensers are not available in the 233-640 MBH size range, and axial-fan condensers that exceed 350 Btuh/Watt specific efficiency are not available below 2,200 MBH (and are only sparingly available from 2,200-8,000 MBH).

9.3.2 Air-Cooled Condenser Specific Efficiency

Major air-cooled condenser manufacturers were contacted to discuss air cooled condenser specific efficiency, the impact on EC fan motor design, and potential for speed control. Interview data concluded that it is feasible to speed-limit EC motors in the field or as a factory feature to "set" the specific efficiency. One manufacturer also provided test information on capacity and power at reduced fan speed.

Manufacturers were questioned about the specific efficiency of condensers with EC motors falling below the efficiency of units with standard motors, when EC units were marketed as an energy-efficient alternative. Manufacturers responded that the technology offers flexibility in setting the efficiency (described above), and also commented that combining EC technology with a micro-channel heat exchange surface is attractive from an efficiency standpoint. One manufacturer responded that they will be increasing production of these units in the near future. Preliminary performance data was obtained for an upcoming product line with the subject technology, and high-end specific efficiency for these units was comparable to other units on the market.

9.4 Screw Compressor Vi Research

In order to determine the reasonableness of requiring new screw compressors to automatically adjust Vi (i.e., no user interface required to adjust Vi in response to operating conditions), the four primary compressor manufacturers were surveyed about the typical size open drive screw compressors at RWH application conditions to determine:

1. Whether automatically-variable Vi is standard?
2. If not, is it available and what is the option cost?
3. Or, what is their offer, in terms of Vi adjustments?

One manufacturer responded that their screw compressors have automatically-variable Vi as standard, included in the standard cost. Larger sizes all have continuously variable Vi, while smaller models have three-step Vi, but it is still fully automatic.

The second manufacturer responded that automatically-variable Vi (continuously variable) is an option on all compressors, and is approximately $2,000 to $5,000 depending on model.

The third manufacturer responded that automatically-variable Vi (continuously variable) is standard on all new compressors.

The final manufacturer responded that automatically-variable Vi (three step) is an available option on the current series of compressors, and is approximately $1,000. The manufacturer will soon be releasing their newest product line, which will have auto-Vi as standard.

9.5 Acceptance Test Survey

Phone and email interview were conducted with six commercial/industrial refrigerated warehouse builders, designers and contractors between June 14 and July 8, 2010. Interview questions were directed at understanding the applicability of an on-site functional test of the refrigerated warehouse mechanical systems as an additional tool to assure compliance with the 2011 T24 Code for refrigerated warehouses in California. Contractors were asked to review the First Draft Test Protocol developed by PECI, then surveyed on the ability of the test to correctly assess equipment functionality and operation. Contractors were also asked questions about their costs to conduct the test, including the cost of equipment needed to perform the test and the amount of time spent traveling to the site, conducting the test and filling out forms.

Most contractors stated the First Draft Test Protocol would be straightforward enough to implement with some minor changes and revisions to the protocol's language. In general, there was consensus among those interviewed about the duration of the test, the equipment required and the involvement of other parties while conducting the test.

9.5.1 Implementation Time

Contractors were asked to give an estimate of how long the test would take to implement in one of their facilities. Each contractor mentioned that the duration of the test could vary greatly depending on the size of the facility, number of systems, and the time of year the test was implemented. However, when asked to estimate the test duration at a typical newly constructed site, all respondents stated the test would take eight hours. This would include time to set up the equipment, run through the test, and restore equipment to its normal operation. Also included in that estimate was the time required to fill out the certificate of acceptance forms. In addition to the eight hours required to implement the test, contractors were also asked to estimate travel time to and from the site. For this most contractors mentioned requiring between 4-8 hours to and from the test. Thus, the average estimate for total labor hours to conduct the acceptance test is between 16- 24 hours.

9.5.2 Required Equipment

Required equipment includes calibrated RH meters, sling psychrometers, calibrated thermometers, calibrated pressure transducers, amp or power meters, and other NIST traceable calibrated measurement devices. Most contractors surveyed already owned the equipment necessary to perform the test. The additional cost to purchase extra equipment would be $0.

Additional stakeholders were contacted during the field demonstration of the acceptance test. These contractors did not have NIST traceable calibrated equipment. Additional research was conducted on the cost of calibrating instruments at labs with NIST traceable standards.

The protocol requires on-site calibration of control point sensors. Most respondents said that calibration could be an issue if the calibration criteria far exceeded the criteria used by the industry for those sensors. Each respondent gave adequate input into the standard sensor accuracy for each control point measurement, and the criteria were set to a level acceptable by many of the stakeholders.

9.5.3 Control System Operator and Facility Owner Representative

When asked how much involvement would be necessary from the facility owner while performing the test, roughly half of the respondents stated that the owner or an owner's representative should at least be on-site and available during the test. For the most part, contractors stated this was necessary because the test could disrupt other activity on site. The other half of the respondents stated they did not require someone representing the facility to be on site because the installing contractor would be fully capable of operating the equipment. However, if the installing contractor is not the entity tasked with performing the acceptance test, someone representing the installing contractor would be required on-site in order to run the equipment through the functions required by the test. An additional eight hours of labor time is required for control system operator, if necessary (approximately half of the tests).

10. Appendix E: Literature Review

10.1 Comparison of Title 24 to Title 20

The California Appliance Efficiency Regulations (Title 20) applies to refrigerated spaces less than 3,000ft^2. Title 20 is an appliance efficiency standard that defines the minimum performance requirements for walk-in coolers and freezers. Due to the nature of the standard, there are several topics and issues that are applicable to both the Title 20 walk-in standard as well as the Title 24 refrigerated warehouse standard. The following table highlights several of the similarities and differences in the California Title 20 walk-in standard and the Title 24 refrigerated warehouse efficiency standard:

	Title 20 Walk-In Standard	2008 Title 24 RWH Standard
Evaporator Control	Variable speed fan control evaluated and proposed, with adjustable two-speed fan speed control for on-off single systems, and fully variable speed as the primary temperature control method for systems with compressor capacity regulation (unloading, or compressor staging for larger multi-compressor unitary systems and parallel systems). Almost no existing examples of variable speed evaporator fan control on single systems in the field. Issues with system integration; suction set point, suction regulator set point, or liquid solenoid control has to come after variable speed evaporator fan control.	2008 requirements: mandatory variable speed on all evaporators, with exception for unitary condensing units with no staging. Recommendations given on system integration (i.e., suction control), but not mandatory. Same issue with walk-ins. Proposed 2011 updates: Acceptance tests have to deal with the integration topic—A small change in suction setting could result in all fans running 100% speed.
Remote Condenser Specific Efficiency	The main application and segment is remote air cooled condensers on supermarkets. A few remote condensers are used on single systems for walk-ins. There is a class of products for food service and restaurants, mostly, that include multi-circuit air-cooled condensers in multi-compressor packages. All customized for specific applications. Analysis of supermarket condenser cost effectiveness concluded that lower TD with lower power *and* variable speed is not cost effective. Variable speed is most cost effective, followed by a balance of specific efficiency and lower TD. However, when evaluating 8°F TD vs. 10°F rated TD and both are extrapolated from a published 30°F test point (which is not confirmed or certified), the optimization is suspect. This highlights the need for certification of equipment performance in order to properly mandate an efficiency standard. T20 doesn't address evaporative condensers, although they are used in California supermarkets.	No requirements in 2008 Title 24 standard. Proposed 2011 updates: Same issues as Title 20 regarding lack of test standards. Halocarbon air cooled condensers for refrigerated warehouses are the same condensers as used for supermarkets included in Title 20. Ammonia air-cooled condensers will be investigated and will include a specific efficiency requirement. Adding mandatory variable speed and all fans running in unison (in 2008 standard) rather than on/off fan cycling changed the marginal economics of low power condensers. Industry discussion on evaporative condenser ratings indicates a significant past effort which resulted in a stalemate. CTI may be a better source for standard on larger evaporative condensers for ammonia (large players in evaporative condenser market are the same as those providing large cooling towers and fluid coolers.)
Remote Condenser Sizing	No requirements for sizing on remote condensers.	2008 requirements: Evaporative condenser design TDs are

		specified for various design wetbulb temperatures. Air cooled remote condenser TDs: 10°F for freezer systems, 15°F for cooler systems The same air-cooled TD requirements apply to the largest unitary condensing units, in order to close a loophole of what could be called unitary, just because it is packaged with a compressor and control system. Problem with this is that TD was not sufficiently defined for a catalog system applicable over a wide range of conditions. 2011: Ammonia air cooled condensers to be added, which will include sizing requirements.
Air Cooled Condensing Unit (Unitary Condensing Unit) EER	Started out in T20 as only requirements for the condenser within air cooled condensers. Evolved to looking at the EER of the condensing unit.	To be evaluated for 2011 revisions as a reach code.
Condenser Control	Proposed floating head pressure (FHP), with varying requirements per system size and type: • Continuous fan operation when compressor on with holdback low limit (smaller units). • Or variable speed solely. • Variable speed with holdback at minimum speed. • Fixed set point control.	2008 standards mandated that all fans be controlled with variable speed in unison. Ambient following (DBT or WBT) with floating head pressure to 70°F SCT or lower. 2011 standards to require acceptance testing

Figure 65: Comparison of Title 20 and Title 24

10.2 Summary of Relevant Rating Standards

Below are summaries of equipment rating standards for air-cooled and evaporative condensers, air units, and insulation, which are published by the Air-conditioning Heating and Refrigeration Institute (AHRI), and ANSI/ASTM.

10.2.1 AHRI Standard 460: Performance Rating of Remote Mechanical Draft Air-Cooled Refrigerant Condensers

AHRI Standard 460 applies to remote, mechanical draft air-cooled condensers; the standard excludes evaporative-cooled condensers and air-cooled condensers included in packaged unitary equipment. The standard intends to establish testing requirements, rating requirements, minimum data requirements for published ratings, marking and nameplate data, and conformance conditions for air-cooled condensers.

Testing requirements for AHRI Standard 460 are established by ANSI/ASHRAE Standard 20. Rating requirements are given in

Figure 66 below. The rating conditions apply to all refrigerants.

Parameter	Rating Condition for All Refrigerants
Barometric Pressure	29.92 In. Hg
Entering Air Dry-Bulb Temperature	95°F
Saturated Condensing Temperature	125°F (30°F TD)
Refrigerant Temperature Entering Condenser	190°F (65°F Superheat)
External Static Pressure	0 in. H_20

Figure 66: Rating conditions for air-cooled condensers, as described by AHRI Standard 460

All claims to ratings within the scope of Standard 460 are required to publish the THR capacity ratings for air-cooled condensers as well as the TD at which the THR capacity applies. The standard allows the THR capacity to be published for any TD, and the capacity can be scaled linearly from the 30°F rated TD to any TD. The standard also requires the publication of the fan motor input watts at the rated conditions.

AHRI Standard 460 is not utilized by any of the condensers in the general market.

10.2.2 AHRI Standard 490: Remote Mechanical-Draft Evaporative-Cooled Refrigerant Condensers

AHRI Standard 490 applies to remote, mechanical draft evaporative-cooled condensers; the standard excludes air-cooled condensers and evaporative-cooled condensers included in packaged unitary equipment. Additionally, the standard is limited to ammonia (R-717) and chlorodifluoromethane (R-22) refrigerant. The standard intends to establish testing requirements, rating requirements, minimum data requirements for published ratings, marking and nameplate data, and conformance conditions for air-cooled condensers.

Testing requirements for AHRI Standard 460 are established by ANSI/ASHRAE Standard 64. Rating requirements are given in
Figure **67**. The rating conditions apply to all refrigerants:

Parameter	Rating Condition	
	R-22	R-717
Barometric Pressure	29.92 in. Hg	29.92 in. Hg
Entering Air Wet-Bulb Temperature	78.0°F	78.0°F
Saturated Condensing Temperature	105°F (27°F TD)	96.3°F (18.3°F TD)
Refrigerant Temperature Entering Condenser	140°F (35°F Superheat)	140°F (35°F Superheat)
External Static Pressure	0 in. H_20	0 in. H_20

Figure 67: Rating conditions for evaporative-cooled condensers, as described in AHRI Standard 490

AHRI Standard 490 does not require the publication of the fan or spray pump input power, but shaft (output) power for both devices is required. For the THR capacity and fan shaft power at conditions other than the rating conditions described in Figure 67, AHRI Standard 490 allows the use of the manufacturer's published THR and fan power correction factors, which are not based on any rating standard.

AHRI Standard 490 is not utilized by any of the condenser manufacturers in the general market.

10.2.3 ARI Standard 420: Standard for Performance Rating of Forced-Circulation Free-Delivery Unit Coolers for Refrigeration

ARI Standard 420 establishes definitions, test requirements, rating requirements, requirements for published ratings, marking and nameplate data, and conformance for forced-circulation free-delivery unit coolers (evaporator coils) for refrigeration The standard does not apply to air-conditioning (comfort cooling) equipment, equipment installed in ductwork, or unit coolers using zeotropic refrigerants with glides greater than Test requirements for ARI Standard 420 are provided in

Figure 68.

Condition Number	Coil Condition	Entering Air				Refrigerant Saturation Temperature	Temperature Difference
		Dry-Bulb Temperature	Wet-Bulb Temperature	Relative Humidity	Dew-Point Temperature		
1	Wet	50	46.1	75%	-	35°F	15°F
2	Dry	50	-	<45	<30	35°F	15°F
3	Dry	35	-	<50	<20	25°F	10°F
4	Dry	10.0	-	<46	<-5.0	0.00°F	10°F
5	Dry	-10.0	-	<43	<-25	-20.0°F	10°F

Figure 68: Rating conditions for air units (evaporator coils) described in ARI Standard 420

AHRI Standard 490 does not refer to ASHRAE 25 test methods. Rather, Standard 490 outlines a test method within the standard itself. AHRI Standard 490 includes five rating conditions at four temperatures. The standard mandates the publication of input electric power for single-phase electric service, but mandates fan motor shaft (output) power for three-phase electric service.

AHRI Standard 490 is not referenced by any of the evaporator manufacturers in the general market.

10.2.4 ANSI/ASTM C177-76, ANSI/ASTM C236-66 and ANSI/ASTM C518-76

Part 12 of Title 24 mandates that insulation shall be tested according to the procedures described in ANSI/ASTM C177-76: “Standard Test Method for Steady-State Thermal Transmission Properties by Means of the Guarded Hot Plate”, ANSI/ASTM C518-76: “Standard Test Method for Steady-State Thermal Transmission Properties by Means of the Heat Flow Meter”, or ANSI/ASTM C236-66: “ASTM C236-89(1993)e1 Standard Test Method for Steady-State Thermal Performance of Building Assemblies by Means of a Guarded Hot Box.” Test conditions are specified in Title 24 Part 12 Sections 12-13-1553 which state, “All thermal performance tests shall be conducted on materials which have been conditioned at 73.4°F +/- 3.6°F and a relative humidity of 50% +/- 5 percent for 24 hours immediately preceding the tests. The average testing temperature shall be 75°F +/-2°F with at least a 40°F temperature difference.”

10.3 Compressor Selection Software

For screw compressors, product selection programs from four manufacturers were utilized in this analysis. Figure 69 describes the features available in each of the vendor’s software packages.

	Vendor A	Vendor B	Vendor C	Vendor D
Inputs				
User-defined oil cooling method	X	X	X	X
User-defined suction superheat/liquid subcooling	X	X	X	X
User-defined suction/discharge pressure drop	X	X	X	X
Supports multiple refrigerants	X	X	X	X
User-defined volume ratio (Vi)				X
User can vary part-load capacity by speed or slide-valve	X	X	X	X
Outputs				
Capacity (in TR or Btuh)	X	X	X	X
Capacity (in mass flow rate)			X	
Absorbed power	X	X	X	X
Nameplate motor power	X			
Oil flowrate/temperature	X	X		
Compressor minimum speed	X	X	X	X
Slide valve indicator position		X		
Package price	X			
Motor price	X			
Manufacturer cites rating standard				

Figure 69: description of compressor manufacturer's software packages

Figure 70-Figure 72 show the improvement in pumping efficiency for the compressors by using speed-reduction capacity control compared to slide valve only. The top curves (the left axis) show the mass flow pumping efficiency of using VFD/slide valve combination, as well slide valve only. The VFD curves include an assumed 2 percent fixed drive loss and an assumed 2 percent variable drive loss. The bottom curves (the right axis) show the percent improvement in pumping efficiency with VFDs, compared to the respective base case.

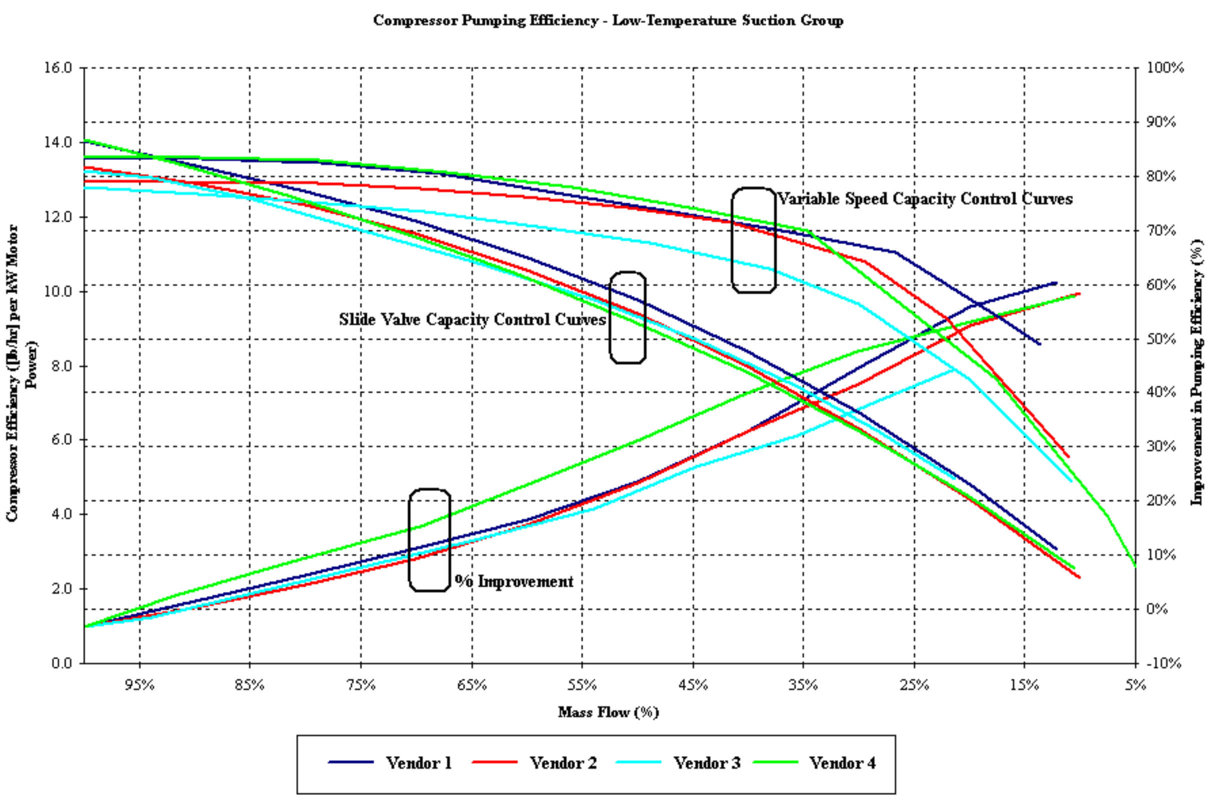


Figure 70: Low-temperature suction group pumping efficiency

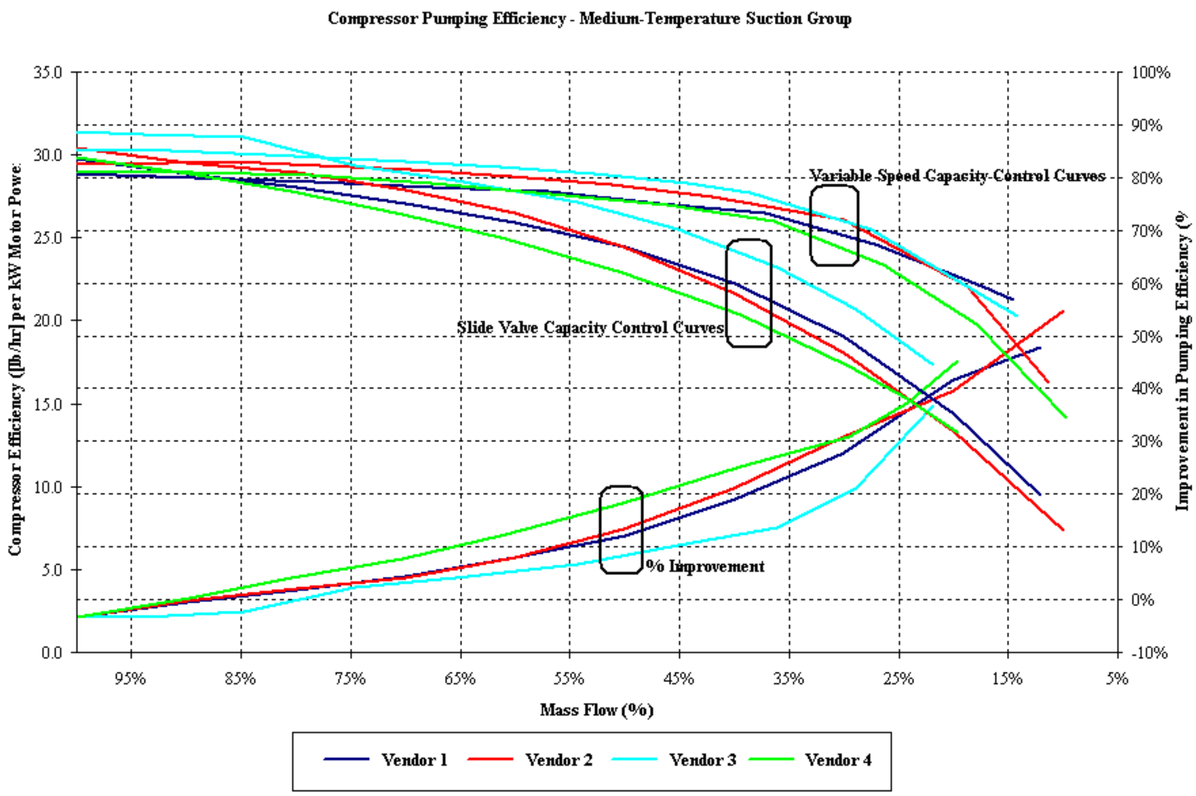


Figure 71: Medium-temperature suction group pumping efficiency

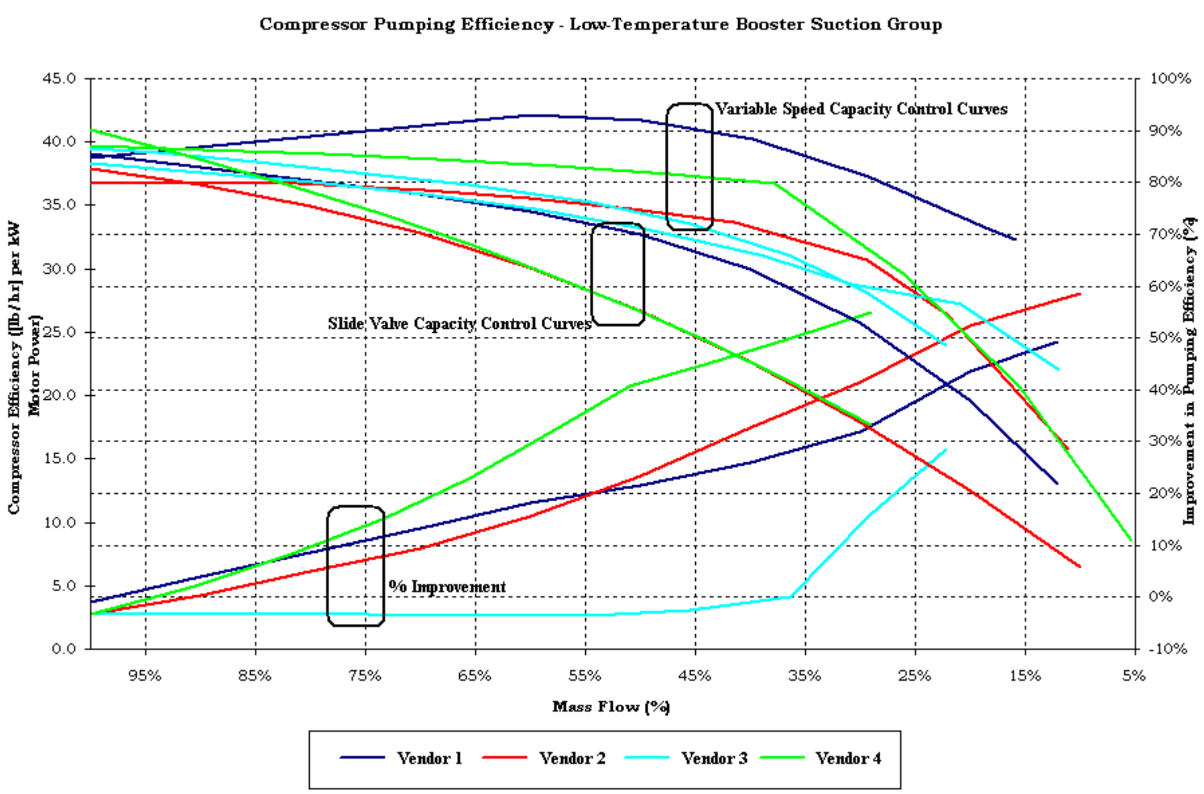


Figure 72: Low-temperature booster suction group pumping efficiency

10.4 Aircoil Literature Review

One manufacturer publishes regular (0” static pressure), long throw/45° penthouse (1/4” static pressure), and 90° penthouse (1/2” static pressure) data. For ¼” static pressure, the catalog recommends an increase in fan nameplate HP of ½ (example: regular motors are 1 HP, 45° penthouse motors are 1 hp), and ½” static pressure motors are an additional ½ HP. Another manufacturer recommends the same motor for all applications.

11. Appendix F: Savings By Design Databases

11.1 Condenser Specific Efficiency

Figure 73-Figure 75 show a database of condenser specific efficiencies utilized to calculate base case specific efficiency for the condenser efficiency measure. The condenser efficiencies come from projects that participated in the Savings By Design new construction incentive program. Both warehouses and supermarkets are included in the database; there is some equipment overlap between supermarkets and small refrigerated warehouses, and a concurrent Title 24 CASE study is striving to mandate condenser efficiencies. Both the supermarket and refrigerated warehouse efficiency mandates utilize the database depicted here.

Year	Utility	Project Type	Location	Configuration	Conditioned Area (SF)	Specific Efficiency (Btuh/Watt)
2008	PG&E	Grocery	Orcutt	Air-Cooled		150
2008	PG&E	Grocery	Lompoc	Air-Cooled		150
2008	SCE	Grocery	Oxnard	Air-Cooled		150
2008	PG&E	Grocery	Santa Cruz	Air-Cooled		139
2007	SCE	Grocery	Marina Del Rey	Air-Cooled		139
2007	PG&E	Grocery	Novato	Air-Cooled		139
2007	PG&E	Grocery	Milpitas	Air-Cooled		134
2007	PG&E	Grocery	Novato	Air-Cooled		134
2007	SCE	Grocery	Marina Del Rey	Air-Cooled		130
2007	SCE	Grocery	La Verne	Air-Cooled		130
2007	PG&E	Grocery	San Jose	Air-Cooled		82
2007	PG&E	Grocery	Redwood City	Air-Cooled		82
2008	PG&E	Grocery	Santa Cruz	Air-Cooled		78
2008	PG&E	Grocery	Santa Cruz	Air-Cooled		78
2007	PG&E	Grocery	San Jose	Air-Cooled		77
2007	PG&E	Grocery	Redwood City	Air-Cooled		77
2008	PG&E	Grocery	Novato	Air-Cooled		77
2007	PG&E	Grocery	Antioche	Air-Cooled		77
2010	SDG&E	Warehouse	San Diego	Air-Cooled	13,000	76
2007	SCE	Grocery	Irvine	Air-Cooled		75
2008	SCE	Grocery	Lakewood	Air-Cooled		74
2008	SCE	Grocery	Hawthorne	Air-Cooled		74
2008	PG&E	Grocery	Pittsburg	Air-Cooled		74
2008	PG&E	Grocery	Pittsburg	Air-Cooled		74
2008	PG&E	Grocery	Pittsburg	Air-Cooled		74
2008	PG&E	Grocery	Pittsburg	Air-Cooled		74
2008	SCE	Grocery	Apple Valley	Air-Cooled		74
2008	SCE	Grocery	Apple Valley	Air-Cooled		74
2008	SCE	Grocery	Apple Valley	Air-Cooled		74
2008	SCE	Grocery	Apple Valley	Air-Cooled		74
2008	SCE	Grocery	Apple Valley	Air-Cooled		74
2008	PG&E	Grocery	Pittsburg	Air-Cooled		74
2007	SCE	Grocery	Irvine	Air-Cooled		71
2008	SCE	Grocery	Seal Beach	Air-Cooled		71
2008	SCE	Grocery	Tustin	Air-Cooled		71

2008	PG&E	Grocery	Santa Cruz	Air-Cooled		71
2007	SCE	Grocery	Claremont	Air-Cooled		62
2008	PG&E	Grocery	Santa Cruz	Air-Cooled		62
2008	PG&E	Grocery	Santa Cruz	Air-Cooled		62
2007	SCE	Grocery	Torrance	Air-Cooled		61
2008	PG&E	Grocery	Santa Cruz	Air-Cooled		60
2007	SCE	Grocery	Marina Del Rey	Air-Cooled		60
2007	SCE	Grocery	Marina Del Rey	Air-Cooled		60
2007	SCE	Grocery	La Verne	Air-Cooled		60
2007	PG&E	Grocery	Novato	Air-Cooled		60
2007	SCE	Grocery	La Verne	Air-Cooled		60
2008	PG&E	Grocery	Santa Cruz	Air-Cooled		57
2008	PG&E	Grocery	Santa Cruz	Air-Cooled		57
2007	SCE	Grocery	Norwalk	Air-Cooled		55
2008	PG&E	Grocery	Santa Cruz	Air-Cooled		54
2008	PG&E	Grocery	Santa Cruz	Air-Cooled		54
2007	SCE	Grocery	Norwalk	Air-Cooled		51
2010	SCE	Warehouse	Buena Park	Air-Cooled	30,000	49.6
2007	SCE	Grocery	Claremont	Air-Cooled		48
2008	SCE	Grocery	Long Beach	Air-Cooled		48
2008	PG&E	Grocery	Santa Cruz	Air-Cooled		48
2007	SCE	Grocery	Malibu	Air-Cooled		46
2008	SCE	Grocery	Rancho Temecula	Air-Cooled		46
2010	SCE	Warehouse	Buena Park	Air-Cooled	30,000	41.3
2007	SCE	Warehouse	Santa Barbara	Air-Cooled	25,200	41.1
2007	SCE	Grocery	Torrance	Air-Cooled		40
2007	SCE	Grocery	Malibu	Air-Cooled		40
					Base Case	70
					Avg. Below Base Case	53

Figure 73: Air-cooled axial-fan halocarbon condenser database

Year	Utility	Project Type	Location	Configuration	Square Ft.	Specific Efficiency
2006	PG&E	Warehouse	Tracy	Axial-Fan Evaporative	76,900	553
2009	PG&E	Warehouse	Santa Maria	Axial-Fan Evaporative	87,300	518
2008	PG&E	Warehouse	Gilroy	Axial-Fan Evaporative	21,000	474
2008	PG&E	Warehouse	Fresno	Axial-Fan Evaporative	47,000	454
2006	SDG&E	Warehouse	San Diego	Axial-Fan Evaporative	131,200	425
2008	PG&E	Warehouse	Wasco	Axial-Fan Evaporative	31,500	411
2008	PG&E	Warehouse	Soledad	Axial-Fan Evaporative	60,400	398
2010	SCE	Warehouse	Pomona	Axial-Fan Evaporative	17,150	374
2008	SCE	Warehouse	Oxnard	Axial-Fan Evaporative	50,300	369
2009	SCE	Warehouse	City of Industry	Axial-Fan Evaporative	120,000	360
2010	SCE	Warehouse	Hanford	Axial-Fan Evaporative	11,000	355
2007	PG&E	Warehouse	Guadalupe	Axial-Fan Evaporative	135,100	348
2007	PG&E	Warehouse	Guadalupe	Axial-Fan Evaporative	135,100	348
2007	PG&E	Warehouse	Kingsburg	Axial-Fan Evaporative	31,400	348
2010	PG&E	Warehouse	West Sacramento	Axial-Fan Evaporative	80,788	341

2008	PG&E	Warehouse	Stockton	Axial-Fan Evaporative	215,000	332
2006	PG&E	Warehouse	Santa Maria	Axial-Fan Evaporative	55,000	322
2006	PG&E	Warehouse	Coalinga	Axial-Fan Evaporative	27,700	318
2009	SCE	Warehouse	Carson	Axial-Fan Evaporative	246,470	316
2009	SCE	Warehouse	Carson	Axial-Fan Evaporative	246,470	316
2006	SCE	Warehouse	Rialto	Axial-Fan Evaporative	47,000	310
2008	SCE	Warehouse	Oxnard	Axial-Fan Evaporative	67,800	302
2007	PG&E	Warehouse	Arvin	Axial-Fan Evaporative	10,000	300
2008	SCE	Warehouse	Delano	Axial-Fan Evaporative	184,000	292
2006	SCE	Warehouse	City of Industry	Axial-Fan Evaporative	122,200	283
2007	PG&E	Warehouse	Wasco	Axial-Fan Evaporative	26,761	279
2010	SCE	Warehouse	Riverside	Axial-Fan Evaporative	139,100	272
2010	SCE	Warehouse	Riverside	Axial-Fan Evaporative	139,100	272
2010	SCE	Warehouse	Riverside	Axial-Fan Evaporative	139,100	272
2009	PG&E	Warehouse	Santa Maria	Axial-Fan Evaporative	87,300	265
2008	SCE	Warehouse	Buena Park	Axial-Fan Evaporative	70,000	263
2010	PG&E	Warehouse	Chico	Axial-Fan Evaporative	21,672	252
2006	PG&E	Warehouse	Union City	Axial-Fan Evaporative	18,000	252
2006	PG&E	Warehouse	Fresno	Axial-Fan Evaporative	180,600	246
2009	PG&E	Warehouse	Fresno	Axial-Fan Evaporative	139,000	246
2009	SCE	Warehouse	Fontana	Axial-Fan Evaporative	317,000	241
2007	SCE	Warehouse	Chino	Axial-Fan Evaporative	56,700	241
2007	PG&E	Warehouse	Kerman	Axial-Fan Evaporative	35,000	213
2010	PG&E	Warehouse	Santa Maria	Axial-Fan Evaporative	68,000	206
2007	SCE	Warehouse	Visalia	Axial-Fan Evaporative	38,000	205
2010	SCE	Warehouse	Delano	Axial-Fan Evaporative	18,980	182
2007	PG&E	Warehouse	Bakersfield	Axial-Fan Evaporative	36,000	150
2008	SCE	Warehouse	Commerce	Axial-Fan Evaporative	76,000	118
2006	PG&E	Warehouse	West Sacramento	Axial-Fan Evaporative	45,000	102
					Base Case	350
					Avg. Below Base Case	265

Figure 74: Axial-fan evaporative-cooled ammonia condenser database

Year	Utility	Project Type	Location	Configuration	Square Ft.	Specific Efficiency
2007	SCE	Grocery	South El Monte	Centrifugal-Fan Evap		278
2008	SCE	Grocery	Buena Park	Centrifugal-Fan Evap		261
2008	SCE	Grocery	Pomona	Centrifugal-Fan Evap		240
2007	PG&E	Warehouse	Petaluma	Centrifugal-Fan Evap	18,720	234
2007	SCE	Warehouse	Ontario	Centrifugal-Fan Evap	39,000	226
2007	PG&E	Grocery	Paso Robles	Centrifugal-Fan Evap		214
2008	SCE	Grocery	Chino	Centrifugal-Fan Evap		193
2010	PG&E	Warehouse	Gonzales	Centrifugal-Fan Evap	21,000	192
2010	PG&E	Warehouse	Gonzales	Centrifugal-Fan Evap	21,000	192
2008	SCE	Grocery	Corona	Centrifugal-Fan Evap		191
2008	SCE	Grocery	Moreno Valley Frederick	Centrifugal-Fan Evap		191
2008	SCE	Grocery	Moreno Valley Heacock	Centrifugal-Fan Evap		191
2008	SCE	Grocery	Palm Springs	Centrifugal-Fan Evap		191

2008	SCE	Grocery	Pedley	Centrifugal-Fan Evap		191
2008	PG&E	Grocery	Bakersfield-Brimhall	Centrifugal-Fan Evap		191
2008	PG&E	Grocery	Bakersfield-Hageman	Centrifugal-Fan Evap		191
2008	PG&E	Grocery	Bakersfield-Olive	Centrifugal-Fan Evap		191
2008	PG&E	Grocery	Bakersfield-Planz	Centrifugal-Fan Evap		191
2008	PG&E	Grocery	Bakersfield-Stine	Centrifugal-Fan Evap		191
2008	PG&E	Grocery	Bakersfield-Stockdale	Centrifugal-Fan Evap		191
2008	PG&E	Grocery	Fresno-Tulare	Centrifugal-Fan Evap		191
2008	PG&E	Grocery	Lemoore	Centrifugal-Fan Evap		191
2008	PG&E	Grocery	Wasco	Centrifugal-Fan Evap		191
2008	SCE	Grocery	Alhambra	Centrifugal-Fan Evap		191
2008	SCE	Grocery	Baldwin Park	Centrifugal-Fan Evap		191
2008	SCE	Grocery	Loma Linda	Centrifugal-Fan Evap		191
2008	SCE	Grocery	Ontario-Euclid	Centrifugal-Fan Evap		191
2007	SCE	Grocery	Upland	Centrifugal-Fan Evap		191
2008	SCE	Grocery	Temecula	Centrifugal-Fan Evap		191
2008	SCE	Grocery	West Covina	Centrifugal-Fan Evap		191
2008	SCE	Grocery	Chino Hills	Centrifugal-Fan Evap		191
2008	SCE	Grocery	Covina	Centrifugal-Fan Evap		191
2008	SCE	Grocery	Fontana	Centrifugal-Fan Evap		191
2008	SCE	Grocery	Fountain Valley Harbor	Centrifugal-Fan Evap		191
2008	PG&E	Grocery	Fresno-1st St	Centrifugal-Fan Evap		191
2008	PG&E	Grocery	Fresno-Cedar	Centrifugal-Fan Evap		191
2008	SCE	Grocery	Compton	Centrifugal-Fan Evap		191
2008	SCE	Grocery	Delano	Centrifugal-Fan Evap		191
2008	SCE	Grocery	Fountain Valley 1082	Centrifugal-Fan Evap		191
2008	SCE	Grocery	Glendora	Centrifugal-Fan Evap		191
2008	SCE	Grocery	Hesperia	Centrifugal-Fan Evap		191
2008	SCE	Grocery	Long Beach	Centrifugal-Fan Evap		191
2008	SCE	Grocery	Moreno Valley Perris	Centrifugal-Fan Evap		191
2008	SCE	Grocery	Newbury Park	Centrifugal-Fan Evap		191
2008	SCE	Grocery	Norwalk	Centrifugal-Fan Evap		191
2008	SCE	Grocery	Oak Park	Centrifugal-Fan Evap		191
2008	SCE	Grocery	Palmdale	Centrifugal-Fan Evap		191
2008	SCE	Grocery	Paramount	Centrifugal-Fan Evap		191
2008	SCE	Grocery	Pico Rivera	Centrifugal-Fan Evap		191
2008	SCE	Grocery	Rialto	Centrifugal-Fan Evap		191
2008	SCE	Grocery	San Jacinto	Centrifugal-Fan Evap		191
2008	SCE	Grocery	Simi Valley	Centrifugal-Fan Evap		191
2008	SCE	Grocery	Upland	Centrifugal-Fan Evap		191
2008	SCE	Grocery	Yucaipa	Centrifugal-Fan Evap		191
2007	SCE	Grocery	Arcadia	Centrifugal-Fan Evap		191
2007	SCE	Grocery	Buena Park	Centrifugal-Fan Evap		191
2007	SCE	Grocery	Eagle Rock	Centrifugal-Fan Evap		191
2007	SCE	Grocery	Hemet	Centrifugal-Fan Evap		191
2007	SCE	Grocery	Huntington Beach	Centrifugal-Fan Evap		191
2007	SCE	Grocery	La Mirada	Centrifugal-Fan Evap		191
2007	SCE	Grocery	Laguna Hills	Centrifugal-Fan Evap		191
2007	SCE	Grocery	West Covina	Centrifugal-Fan Evap		191
2007	SCE	Grocery	Moreno Valley	Centrifugal-Fan Evap		189
2008	SCE	Grocery	Victorville	Centrifugal-Fan Evap		188

2007	SCE	Grocery	Visalia	Centrifugal-Fan Evap		188
2007	SCE	Grocery	Irvine	Centrifugal-Fan Evap		187
2007	SCE	Grocery	Victorville	Centrifugal-Fan Evap		186
2007	SCE	Grocery	Moreno Valley	Centrifugal-Fan Evap		186
2007	SCE	Grocery	Lake Forest	Centrifugal-Fan Evap		186
2008	SCE	Grocery	Anaheim Hills	Centrifugal-Fan Evap		175
2008	SCE	Grocery	Lakewood	Centrifugal-Fan Evap		175
2008	SCE	Grocery	City of Industry	Centrifugal-Fan Evap		175
2008	SCE	Grocery	La Habra	Centrifugal-Fan Evap		175
2008	SCE	Grocery	Moorpark	Centrifugal-Fan Evap		175
2008	SCE	Grocery	Moreno Valley Alessandro	Centrifugal-Fan Evap		175
2007	PG&E	Warehouse	Chico	Centrifugal-Fan Evap	9,100	175
2008	PG&E	Grocery	Manteca	Centrifugal-Fan Evap		173
2007	PG&E	Grocery	Woodland	Centrifugal-Fan Evap		173
2008	PG&E	Grocery	Madera	Centrifugal-Fan Evap		173
2008	SCE	Grocery	Duarte	Centrifugal-Fan Evap		172
2008	SCE	Grocery	Manhattan Beach	Centrifugal-Fan Evap		172
2008	SCE	Grocery	Palm Desert	Centrifugal-Fan Evap		172
2007	PG&E	Grocery	Martell	Centrifugal-Fan Evap		170
2007	PG&E	Grocery	Fresno	Centrifugal-Fan Evap		168
2007	PG&E	Grocery	San Francisco	Centrifugal-Fan Evap		168
2007	SCE	Grocery	Oxnard	Centrifugal-Fan Evap		155
2008	SCE	Grocery	Victorville	Centrifugal-Fan Evap		155
					Base Case	160
					Avg. Below Base Case	155

Figure 75: Centrifugal fan evaporative-cooled halocarbon condenser database

11.2 Insulation R-Values

Shown below for reference is a database of insulation R-values from refrigerated warehouses that participated in the Savings By Design new construction incentive program.

SBD Year	Utility	Location	Space	Space Temperature (°F)	Proposed Wall Insulation	Proposed Roof Insulation	Proposed Floor Insulation
2010	SCE	Pomona	Cooler	35/40	R-34	R-34	unknown
2008	PG&E	Stockton	Cooler	34/60	R-25	R-25	unknown
2008	PG&E	Gilroy	Cooler	33/45	R-32	R-33	unknown
2008	PG&E	American Canyon	Storage	63	R-10.8	R-30.6	un-insulated
2008	PG&E	American Canyon	Storage	63	R-10.8	R-30.6	un-insulated
2009	PG&E	American Canyon	Cooler	55	R-19	R-19	unknown
2011	PG&E	Tracy	Cooler	45	R-33	R-41	un-insulated
2008	SCE	Oxnard	Cooler	41	R-18	R-18	unknown
2010	SCE	Fontana	Cooler	40	R-33	R-33	unknown
2008	SCE	Commerce	Dock	38	R-38	R-38	R-27
2008	SDG&E	San Diego	Cooler	38	R-22	R-36	unknown
2010	PG&E	Salinas	Cooler	36	R-33	R-33	un-insulated
2010	SCE	Ontario	Dock	36	R-25	R-37	un-insulated
2008	PG&E	Union City	Cooler	36	R-28	R-28	unknown

2008	SCE	City of Industry	Cooler	36	R-32.6	R-23.9	un-insulated
2010	PG&E	West Sacramento	Docks	35	R-28.6	R-35	un-insulated
2010	SDG&E	San Diego	Cooler	35	R-45	R-45	un-insulated
2009	SCE	Carson	Cooler	35	R-26	R-36	unknown
2009	SCE	Fontana	Cooler	35	R-43	R-40	un-insulated
2008	SCE	Commerce	Cooler	35	R-38	R-38	R-32
2010	PG&E	Gonzales	Cooler	34	R-32	R-33	unknown
2009	SCE	Compton	Cooler	34	R-28.6	R-37	un-insulated
2010	PG&E	Santa Maria	Cooler	33	R-33	R-41	un-insulated
2010	PG&E	Santa Maria	Dock	33	R-33	R-41	un-insulated
2009	PG&E	Santa Maria	Cooler	33	R-32 ext/ R-24 int	R-32	unknown
2009	PG&E	Lemoore	Cooler	30	R-49 ext/R-33 int	R-33	un-insulated
2012	PG&E	Tracy	Cooler	28	R-33	R-41	un-insulated
2008	PG&E	Union City	Freezer	-9	R-35	R-35	unknown
2013	PG&E	Tracy	Freezer	-10	R-41	R-41	R-30
2010	PG&E	West Sacramento	Freezer	-10	R-35.7	R-50	R-30
2010	SCE	Ontario	Freezer	-10	R-37.6	R-53	R-30
2010	SDG&E	San Diego	Freezer	-10	R-45	R-45	R-30
2009	PG&E	Lemoore	Freezer	-10	R-49	R-49	unknown
2009	SCE	Buena Park	Freezer	-10	R-50	R-60	R-43
2008	PG&E	Stockton	Freezer	-10	R-33	R-33	R-33
2008	SDG&E	San Diego	Freezer	-10	R-30	R-36	unknown
2010	SCE	Pomona	Freezer	-15	R-50	R-50	R-33
2008	SCE	Commerce	Freezer	-20	R-38	R-38	R-32
2008	SCE	Commerce	Freezer	-50	R-75	R-75	R-36

Figure 76: Insulation R-values from participants in the SBD utility incentive program.

12. Appendix G: Air-Cooled Ammonia Study

Air-cooled ammonia condensers are prohibited on refrigerated warehouses by 2008 standards, though there was no cost analysis to justify this requirement. This analysis evaluates air-cooled ammonia systems to see if they are cost-effective in certain climate zones, which would justify revising the current standard. Prototype Warehouse #1 was used to evaluate this measure. The operating costs of both air-cooled condensers and comparably sized evaporative-cooled condensers were evaluated in all climate zones. The analysis includes water procurement and treatment costs as well as utility costs. For both the air-cooled and evaporative-cooled condenser evaluations, the prototype warehouse was simulated using a 70°F minimum saturated condensing temperature with an ambient-following control strategy and variable speed control of all evaporator fans. DOE-2.2R simulation keywords exactly replicate the proposed control strategy.

To accurately capture utility costs for both measures, the utility rate schedule described in Figure 77 was simulated.

Period	Applicable dates	Applicable time	Energy cost	Demand cost
Winter off-peak	January 1 – April 30 and November 1 – December 31	12 AM to 9 AM and 10 PM to 12 AM on weekdays, and all-day on weekends/holidays	$0.08067/kWh	$0.00/kW
Winter part-peak		9 AM to 10 PM on weekdays	$0.09113/kWh	$1.12/kW
Summer off-peak	May 1 through October 31	12 AM to 9 AM and 10 PM to 12 AM on weekdays, and all-day on weekends/holidays	$0.08339/kWh	$0.00/kW
Summer part-peak		9 AM to 12 PM and 6 PM to 10 PM on weekdays	$0.10168/kWh	$2.81/kW
Summer peak		12 PM to 6 PM on weekdays	$0.14606/kWh	$12.67/kW
			Non-coincident demand cost:	$8.56/kW

Figure 77: Utility rate assumptions for air-cooled ammonia system evaluation

The simulated utility rate was based on PG&E E-20 rates. Figure 78 describes the assumptions made to calculate water consumption and cost.

Evaporation rate	Bleed rate	Drift rate	Procurement cost	Treatment cost
1,843.7 to 1,965.2 gallons x 1,000, depending on climate zone	1,316.9 to 1,403.7 gallons x 1,000, depending on climate zone	676.4 gallons x 1,000	$0.0084/gallon	$600/month

Figure 78: Water assumptions for air-cooled ammonia system evaluation

Evaporation rates were calculated based on the total heat rejected from the condenser, based on simulation results. Condenser water drift rates were estimated as 0.18 percent of the condenser recirculation rate, which was assumed to be 715 GPM. Bleed rate was calculated based on 2.4 cycles of concentration, which was calculated based on dissolved mineral content in typical municipal water, and tolerable dissolved mineral concentrations in sump water from condenser manufacturer data. Water procurement costs are based on water and wastewater costs for commercial and industrial customers for the top 50 cities in California. Treatment costs are based on surveys with building operators. Incremental costs are not included so there is no LCC analysis.

	Energy Savings		Water Savings			Energy Cost Savings		Total Cost Savings	
	kWh	kWh/SF	Gallons	$	$/SF	$	$/SF	$	$/SF
CTZ03 Oakland	8,917	0.097	3,883,850	$39,669	$0.43	($12,028)	($0.13)	$27,641	$0.30
CTZ05 Santa Maria	2,327	0.025	3,837,061	$39,278	$0.43	($10,571)	($0.12)	$28,707	$0.31
CTZ07 San Diego	7,814	0.085	3,986,436	$40,527	$0.44	($5,196)	($0.06)	$35,331	$0.38
CTZ10 Riverside	-108,317	-1.177	4,024,818	$40,847	$0.44	($38,079)	($0.41)	$2,768	$0.03
CTZ12 Sacramento	-163,592	-1.778	3,966,384	$40,359	$0.44	($36,313)	($0.40)	$4,046	$0.04
CTZ13 Fresno	-232,248	-2.524	4,016,431	$40,777	$0.44	($46,311)	($0.50)	($5,534)	($0.06)
CTZ14 Palmdale	-190,413	-2.07	4,045,402	$41,020	$0.45	($46,197)	($0.50)	($5,177)	($0.06)

Figure 79: Energy and water savings for air-cooled compared to evaporative-cooled ammonia system on large warehouse

13. Appendix H: Dropped Measures

This appendix summarizes the measures that were considered for inclusion in the 2013 standards, but were later dropped from consideration after initial research. These include:

- Evaporator Coil Specific Efficiency
- Evaporator Coil Sizing
- Unitary Condensing Unit Efficiency
- Compressor Staging

13.1 Air Unit (Evaporator Coil and Fan) Specific Efficiency and Sizing Requirements

The feasibility of mandating an air unit specific efficiency standard, an air unit sizing standard, and test requirement were evaluated for inclusion in the 2013 Title 24 refrigerated warehouse standards.

13.1.1 Evaporator Specific Efficiency

Evaporator coil specific efficiency (Btu/hr/Watt at a standard condition) was considered for inclusion in the Title 24 standard. Research was to be conducted for as many as five or more families of evaporator coils, including consideration of coil sizes, refrigerant feed type (direct expansion or flooded/recirculated), considerations for long-throw and penthouse (ducted) configurations, freezer and cooler coils, fans required for air mixing (throw length), with potential to research other variants. Existing work has already been completed for smaller evaporators as part of the 2008 Title 20 appliance efficiency standards, where an initial study of a large portion of the available evaporator coils showed a very wide range in evaporator fan power per unit of capacity (specific efficiency).

Initial research into the feasibility of this measure revealed several challenges:

- Evaporator coils are not rated to any performance standard. Capacity is not published per AHRI standards. Power is often not published at all, and when available is almost always the nominal motor power, not the applied power. Furthermore, for smaller units, the nominal motor power is typically regarded as a generalization, with actual shaft power often differing from nominal power by as much as 100 percent. Until evaporators are rated and published according to a standard, the actual performance will remain largely unknown, and it is very likely that evaporators will increase in size if they are rated, tested and certified to a standard.
- Setting requirements to ratings at AHRI conditions (and certified ratings) would very likely cause extensive changes to evaporator coil ratings since the catalog values now are "commercialized" by most accounts, at least on smaller models.

While mandating an efficiency requirement to prohibit the least-performing models would yield significant savings, it is recommended that this measure be deferred until certification and testing is widely implemented for this equipment.

13.1.2 Evaporator Sizing and Test Standard

Standards for evaporator size (i.e., a limit on the difference between saturated evaporating temperature and design space temperature) was considered to determine if there was a way to establish efficient design temperature differential (TD) and still be mindful of widely varying application conditions, operating requirements, defrosting and liquid feed constraints. Research was to include:

- determining whether an ASHRAE standard could be used for rating and/or testing the evaporators,
- determining whether non-evaporative cooling heat exchangers should be included (e.g., glycol secondary fluid that stays single-phase),
- acceptance testing of this measure for all types of evaporators that are found to be economically and technically feasible, and
- calculations of annual energy savings and cost-effectiveness for new construction of small and large refrigerated warehouses with both ammonia and HFC central systems.

This measure was dropped from consideration in the 2013 standards. Technical feasibility was the main issue. Standards for evaporator size (TD) are a distant opportunity for several reasons:

1. It was discovered that in many cases the TD is currently driven by humidity or defrost loading requirements, which may already drive evaporators to be as large as is economically justified (assuming they are controlled to full utilization which has essentially been accomplished in the 2008 Standard).
2. The evaporator size is a function of the design cooling load and many engineers use rule-of-thumb load calculations or use simple calculations with sum of non-coincident peaks, then assume a run time that is less than 24 hours, leading to significant oversizing of cooling equipment. This oversizing may mask a significant amount of shortfall in actual system performance (especially direct-expansion HFC systems). Until more accurate load calculations and engineering methods are employed, evaporator sizing standards may be somewhat meaningless, and easily counterproductive in many instances.
3. Fin spacing is a consideration; denser fin configurations increase capacity but also increase defrost frequency and pose cleaning issues. Currently there are inadequate engineering methods and product offerings to allow airflow and cooling requirements to be optimized separately.

13.2 Unitary Condenser Efficiency

Information was gathered condensing unit designs, with a focus on the condenser sizing and performance. Key facts include:

- Condensing unit capacities do not refer to AHRI test standards.
- Ratings for a given unit condensing cover a very wide application range (with a doubling of capacity and heat through condenser between extremes), and manufacturers use different “cut-off” temperatures for their product lines as well as some having intermediate ranges.
- Most published ratings are calculated based on compressor capacity, not testing, which results in an overstatement of as much as 30 percent due to the legacy compressor rating points. The AHRI 1250P test conditions are the first instance to address realistic return gas temperatures for refrigeration systems, which underlies this broad and systematic error.

13.3 Compressor Staging

Staging of refrigeration compressors was to be researched to determine if there is a way to reasonably set a minimum standard. Certain systems operate in on/off mode, which may be reasonably efficient and effective, given the large thermal capacitance in refrigerated warehouses. Larger industrial compressors have continuously variable capacity. This measure was to include:

- determining if there is an ASHRAE or AHRI standard for multiple compressor control that can be referenced,
- literature review of multiple compressor control for all types of compressors and refrigerants that may be used in refrigerated warehouses,
- development of an acceptance test for all types of compressors and refrigerants that may be used in a refrigerated warehouse,
- research of control system and instrumentation requirements and cost, and
- calculation of annual energy savings and cost-effectiveness for new construction of small and large refrigerated warehouses with both ammonia and HFC central systems with multiple compressors.

This measure was dropped from consideration. Technical feasibility is the main issue. Industrial refrigeration compressors are not tested and rated by a common standard and there is no certification of compressor ratings. Given the difficulty of measuring vapor flow, there is little research information on installed systems to compare with manufacturer ratings. Also, the application conditions and compressors are unique to each warehouse, making acceptance testing definition and testing of part-load prohibitively expensive. Cooling loads are very slow moving in refrigerated warehouses, and there is a high degree of “capacitance” in the system. For these system times, many steps of capacity are sometimes not necessary for maximum efficiency (and uneven compressor sizes are also not feasible in some warehouse refrigeration systems), making a general compressor staging measure difficult. Finally, there are considerations for the individual compressor pumping efficiency, however the system generally includes several compressors in parallel. Thus the overall system efficiency is also at issue, inclusive of control strategies and dynamic response in conjunction with the system regulator valves. The premise of feedback control, to maintain an instant response to system pressure vs. set-point, may be less efficient than a load-based control that operates to deliver required ton-hours in a 24-hour period, with compressors operated fully loaded for variable time increments.

There is no viable path towards a standard that could be applied generally across the many system situations. Therefore, this measure was dropped from consideration.