CODES AND STANDARDS ENHANCEMENT INITIATIVE (CASE)

# Refrigerated Warehouse

## *2013 California Building Energy Efficiency Standards*

California Utilities Statewide Codes and Standards Team October 31, 2011

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## Table of Contents

**CODES AND STANDARDS ENHANCEMENT INITIATIVE (CASE) ..... 1**
**1. Purpose ..... 6**
**2. Overview ..... 7**
**3. Methodology ..... 15**
3.1 Refrigerated Warehouse Prototype Definitions ..... 15
3.2 Simulation and Cost Effectiveness Methodology ..... 16
3.3 Stakeholder Meeting Process ..... 17
**4. Analysis and Results ..... 18**
4.1 Statewide Energy Savings ..... 18
4.2 Freezer Roof and Floor Insulation ..... 18
4.2.1 Roof Insulation Analysis Results by Climate Zone ..... 19
4.2.2 Floor Insulation Analysis Results by Climate Zone ..... 22
4.3 Evaporator Fan Control for Single Compressor Systems ..... 23
4.3.1 Evaporator Speed Control Analysis Results by Climate Zone ..... 24
4.4 Allow Air-Cooled Ammonia Condensers ..... 26
4.5 Condenser Specific Efficiency ..... 27
4.5.1 Incremental Analysis Results ..... 30
4.5.2 Condenser Specific Efficiency Analysis Results by Climate Zone ..... 31
4.6 Screw Compressor Part-Load Performance ..... 33
4.6.1 Compressor Variable Speed Control Analysis Results by Climate Zone ..... 35
4.6.2 Compressor Size Sensitivity Analysis ..... 36
4.7 Infiltration Barriers ..... 37
4.8 Acceptance Tests ..... 39
4.9 Code Language Changes Not Requiring Analysis ..... 40
4.10 Equipment Rating Accuracy, Standards and Certification ..... 42
**5. Recommended Code Language ..... 43**
5.1 Title 24 Draft Code Language ..... 43
5.2 Acceptance Test Language ..... 47
**6. Appendix A: Load Calculations and Equipment Selection ..... 54**
6.1 Load Calculations ..... 54
6.2 Equipment Selection ..... 62
**7. Appendix B: Base Case Prototype Descriptions ..... 64**
7.1 Base Case Facility Description ..... 64
**8. Appendix C: Measure Cost ..... 68**
8.1 Freezer Roof Measure Cost ..... 68
8.2 Evaporator Fan Control for Single Cycling-Compressor Systems ..... 73
8.3 Condenser Specific Efficiency ..... 76
8.4 Screw Compressor Part-Load Analysis ..... 79
8.5 Infiltration Barriers ..... 80
**9. Appendix D: Industry Interviews and Market Research ..... 83**
9.1 Insulation ..... 83
9.1.1 Rated R-Values ..... 83
9.1.2 Miscellaneous Insulation Comments from Contractors and Vendors ..... 84
9.2 Infiltration Barriers ..... 84

9.3 Condenser Specific Efficiency ..... 85
9.3.1 Evaporative Condenser Specific Efficiency ..... 85
9.3.2 Air-Cooled Condenser Specific Efficiency ..... 87
9.4 Screw Compressor Vi Research ..... 88
9.5 Acceptance Test Survey ..... 88
9.5.1 Implementation Time ..... 88
9.5.2 Required Equipment ..... 89
9.5.3 Control System Operator and Facility Owner Representative ..... 89
**10. Appendix E: Literature Review ..... 90**
10.1 Comparison of Title 24 to Title 20 ..... 90
10.2 Summary of Relevant Rating Standards ..... 91
10.2.1 AHRI Standard 460: Performance Rating of Remote Mechanical Draft Air-Cooled Refrigerant Condensers ..... 91
10.2.2 AHRI Standard 490: Remote Mechanical-Draft Evaporative-Cooled Refrigerant Condensers ..... 92
10.2.3 ARI Standard 420: Standard for Performance Rating of Forced-Circulation Free-Delivery Unit Coolers for Refrigeration ..... 92
10.2.4 ANSI/ASTM C177-76, ANSI/ASTM C236-66 and ANSI/ASTM C518-76 ..... 93
10.3 Compressor Selection Software ..... 93
10.4 Aircoil Literature Review ..... 96
**11. Appendix F: Savings By Design Databases ..... 97**
11.1 Condenser Specific Efficiency ..... 97
11.2 Insulation R-Values ..... 101
**12. Appendix G: Air-Cooled Ammonia Study ..... 104**
**13. Appendix H: Dropped Measures ..... 106**
13.1 Air Unit (Evaporator Coil and Fan) Specific Efficiency and Sizing Requirements ..... 106
13.1.1 Evaporator Specific Efficiency ..... 106
13.1.2 Evaporator Sizing and Test Standard ..... 106
13.2 Unitary Condenser Efficiency ..... 107
13.3 Compressor Staging ..... 108
**14. Appendix I: Full Condenser Specific Efficiency Analysis ..... 109**
14.1 Evaporatively-Cooled Ammonia Condensers ..... 109
14.2 Air-Cooled Halocarbon Condensers without EC Motors ..... 110
14.3 Air-Cooled Halocarbon Condensers with EC Motors ..... 111
14.4 Evaporatively-Cooled Halocarbon Condensers ..... 112
**15. Appendix J: Assumptions For Environmental Impact ..... 113**

## TABLE OF FIGURES

Figure 1: Prototype warehouse summary ..... 15
Figure 2: Summary of space utilization for each prototype warehouse ..... 15
Figure 3: Statewide energy and energy cost savings ..... 18
Figure 4: Insulation material assumptions ..... 19
Figure 5: Simulated freezer roof insulation thicknesses ..... 19
Figure 6: Freezer roof insulation analysis results ..... 21
Figure 7: Simulated freezer floor insulation thicknesses ..... 22
Figure 8: R-35 compared to R-30 freezer floor insulation analysis results ..... 22
Figure 9: R-40 compared to R-35 freezer floor insulation analysis results ..... 22
Figure 10: Base case assumptions for evaporator fan speed control measure ..... 23
Figure 11: Simulation summary for evaporator fan control measure ..... 24
Figure 12: Statewide savings results for evaporator fan control measure ..... 25
Figure 13: Typical screw compressor performance with ammonia and HFC refrigerant ..... 26
Figure 14: Description of prototype warehouses for condenser specific efficiency measure ..... 27
Figure 15: Graph of condenser capacity and power versus speed ..... 28
Figure 16: Condenser cost versus capacity at specific-efficiency rating conditions ..... 29
Figure 17: Example of incrementally increasing condenser size and resultant specific efficiency ..... 29
Figure 18: Example of building energy use and TDV energy cost versus specific efficiency ..... 30
Figure 19: Preliminary condenser specific efficiency results ..... 31
Figure 20: Analysis results by climate zone for condenser specific efficiency measure ..... 32
Figure 21: Part-load performance curves for slide valve and variable-speed control ..... 35
Figure 22: Savings analysis results for screw compressor variable speed measure ..... 36
Figure 23: Sensitivity analysis of screw compressor variable-speed measure ..... 37
Figure 24: Strip curtain savings and cost-effectiveness analysis results ..... 38
Figure 25: Acceptance test cost analysis results ..... 40
Figure 26: Description of three design climate zones ..... 54
Figure 27: Load calculations, 35°F cooler space (Prototype Warehouses #1 and 2) ..... 55
Figure 28: Load calculations, -10°F freezer space (Prototype Warehouses #1 and 2) ..... 56
Figure 29: Load calculations, 40°F dock space (Prototype Warehouse #1) ..... 57
Figure 30: Load calculations, 85°F dry storage space (Prototype Warehouse #2) ..... 58
Figure 31: Load calculations, 35°F cooler space (Prototype Warehouses #3 and 4) ..... 59
Figure 32: Load calculations, -10°F freezer space (Prototype Warehouses #1 and 2) ..... 60
Figure 33: Load calculations, 40°F dock space (Prototype Warehouse #3) ..... 61
Figure 34: Load calculations, 85°F dry storage space (Prototype Warehouse #4) ..... 62
Figure 35: Prototype Warehouse #1 and 2 compressor selection ..... 63
Figure 36: Prototype Warehouse #3 and 4 compressor selection ..... 63
Figure 37: Base case facility description ..... 67
Figure 38: Cost calculation worksheet for prefabricated urethane cam-lock panels ..... 69
Figure 39: Cost calculation worksheet for urethane and expanded polystyrene panels ..... 70
Figure 40: Cost calculation worksheet for polyisocyanurate overdeck insulation ..... 71
Figure 41: Example simultaneous analysis of cost regression and building energy use regression ..... 72
Figure 42: Cost regression analysis for expanded polystyrene floor insulation ..... 73
Figure 43: Measure cost calculator for fan speed control ..... 74
Figure 44: Maintenance cost calculator for fan speed control ..... 75

Figure 45: Measure cost calculator for fan staging control ..... 75
Figure 46: Maintenance cost calculator for fan staging control ..... 75
Figure 47: Cost versus capacity regression at specific efficiency rating conditions for axial-fan evaporative-cooled ammonia condensers ..... 76
Figure 48: Cost versus capacity regression at specific efficiency rating conditions for axial-fan air-cooled HFC condensers with standard motors ..... 77
Figure 49: Cost versus capacity regression at specific efficiency rating conditions for axial-fan air-cooled HFC condensers with BLDC motors ..... 77
Figure 50: Cost versus capacity regression at specific efficiency rating conditions for centrifugal-fan evaporative-cooled HFC condensers ..... 78
Figure 51: Additional materials and labor assumptions for variable-frequency drives versus soft-starts. ..... 79
Figure 52: Screw compressor part-load measure cost calculator for LT, MT, and booster suction groups ..... 79
Figure 53: Cost versus motor horsepower regressions for screw compressor speed control ..... 80
Figure 54: Cost assumptions for manual hard doors ..... 80
Figure 55: Cost assumptions for strip curtains ..... 81
Figure 56: Cost assumptions for standard- and high-speed automatic doors ..... 81
Figure 57: Cost assumptions for air curtains ..... 82
Figure 58: Survey of door opening speeds ..... 84
Figure 59: One manufacturer's infiltration barrier recommendations according to % door open time . 85
Figure 60: Minimum and maximum condenser catalog capacities for centrifugal-fan evaporative condensers and small axial-fan evaporative condensers ..... 86
Figure 61: Specific efficiency of centrifugal-fan and small axial-fan evaporative condensers at 100°F SCT, 70°F WBT ..... 87
Figure 62: Comparison of Title 20 and Title 24 ..... 91
Figure 63: Rating conditions for air-cooled condensers, as described by AHRI Standard 460 ..... 92
Figure 64: Rating conditions for evaporative-cooled condensers, as described in AHRI Standard 490 92
Figure 65: Rating conditions for air units (evaporator coils) described in ARI Standard 420 ..... 93
Figure 66: description of compressor manufacturer's software packages ..... 94
Figure 67: Low-temperature suction group pumping efficiency ..... 94
Figure 68: Medium-temperature suction group pumping efficiency ..... 95
Figure 69: Low-temperature booster suction group pumping efficiency ..... 95
Figure 70: Air-cooled axial-fan halocarbon condenser database ..... 98
Figure 71: Axial-fan evaporative-cooled ammonia condenser database ..... 99
Figure 72: Centrifugal fan evaporative-cooled halocarbon condenser database ..... 101
Figure 73: Insulation R-values from participants in the Savings By Design utility incentive program. ..... 103
Figure 74: Utility rate assumptions for air-cooled ammonia system evaluation ..... 104
Figure 75: Water assumptions for air-cooled ammonia system evaluation ..... 104
Figure 76: Energy and water savings for air-cooled compared to evaporative-cooled ammonia system on large warehouse ..... 105

# 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$^{o}$F freezers to moderate +50$^{o}$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:

1. freezer roof insulation
2. freezer floor insulation
3. evaporator fan control exception for single compressor systems
4. allow air-cooled ammonia condensers
5. screw compressor part-load performance

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

1. condenser specific efficiency
2. infiltration barriers
3. acceptance tests

The analysis included research of refrigerated warehouse energy efficiency, data collection from the Savings By Design 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:<br>▪ Freezer Roof Insulation – require a higher minimum R-value for freezer ceilings than 2008 requirements, which are below industry-standard practice and ASHRAE recommendations.<br>▪ 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.<br>▪ 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.<br>▪ 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.<br>▪ 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.<br>▪ Condenser Specific Efficiency – impose a maximum fan power per unit of capacity on refrigerant condensers utilized on refrigerated warehouse facilities.<br>▪ Infiltration Barriers – require devices at door openings of refrigerated spaces to minimize air infiltration, resulting in a reduction in refrigeration system load.<br>▪ Acceptance Tests – 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.<br><br>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 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 to roof and floor insulation material is expected to be negligible.

| | Mercury | Lead | Copper | Steel | Plastic | Others (insulation) |
|---|---|---|---|---|---|---|
| Pounds per Square Foot per Year | NC | NC | NC | NC | NC | NC |

**Material Increase (I), Decrease (D), or No Change (NC) Due to Insulation Change**

The removal of the evaporator fan speed control exemption for single, constant volume compressor systems and the requirement for VSD on single screw compressor systems will increase steel, copper, plastic and aluminum usage from addition of motor drives.

| | Mercury | Lead | Copper | Steel | Plastic | Others (Aluminum) |
|---|---|---|---|---|---|---|
| Pounds per Fan Motor per Year | NC | NC | I~ 0.1 | I~ 0.6 | I~ 0.1 | I~ 0.01 |
| Pounds per Compressor Motor per Year | NC | NC | I~ 5 | I~ 30.0 | I~ 5 | I~ 0.5 |

**Material Increase (I), Decrease (D), or No Change (NC)**
**Due to Evaporator Fan Speed Exemption Removal**

The change to allow the use of ammonia air-cooled condensers may result in a very small number of systems being designed with ammonia that otherwise would have required HFC refrigerants. However, because there has not been an air-cooled ammonia system designed in CA since the inception of RWH Savings by Design program, we estimate that this change in code language will net zero change.

| | Mercury | Lead | Copper | Steel | Plastic | Others |
|---|---|---|---|---|---|---|
| Pounds per MBTUH per Year | NC | NC | NC | NC | NC | NC |

**Material Increase (I), Decrease (D), or No Change (NC) Due to Ammonia Air-cooled Condensers**

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.

| | Mercury | Lead | Copper | Steel | Plastic | Others (Aluminum) | Others (Refrigerant) |
|---|---|---|---|---|---|---|---|
| Pounds per MBTUH per Year | NC | NC | I~0.07 | I~0.08 | NC | I~0.06 | D~0.05 |

**Material Increase (I), Decrease (D), or No Change (NC) Due to Condenser Specific Efficiency**

| | |
|---|---|
| g. Technology Measures | **Measure Availability:**<br>▪ 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.<br><br>▪ 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.<br><br>**Useful Life, Persistence, and Maintenance:**<br>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.<br><br>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 Compressor Systems | 15 | $30,016 | $1.15 | $11,251 | $0.43 | $304,716 | $12 | -263,449 | -10.33 |
| CTZ05 Santa Maria | Measure Life (Years) | Measure Cost ($) | 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 Compressor Systems | 15 | $30,016 | $1.15 | $11,251 | $0.43 | $304,303 | $12 | -263,037 | -10.69 |
| CTZ07 San Diego | Measure Life (Years) | Measure Cost ($) | 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 Compressor Systems | 15 | $30,016 | $1.15 | $11,251 | $0.43 | $314,723 | $12 | -273,456 | -10.69 |
| CTZ10 Riverside | Measure Life (Years) | Measure Cost ($) | 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 Compressor Systems | 15 | $30,016 | $1.15 | $11,251 | $0.43 | $310,973 | $12 | -269,706 | -10.58 |
| CTZ 12 Sacramento | Measure Life (Years) | Measure Cost ($) | 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 Compressor Systems | 15 | $30,016 | $1.15 | $11,251 | $0.43 | $306,675 | $12 | -265,408 | -10.41 |
| CTZ13 Fresno | Measure Life (Years) | Measure Cost ($) | 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 Compressor Systems | 15 | $30,016 | $1.15 | $11,251 | $0.43 | $308,621 | $12 | -267,354 | -10.49 |
| CTZ14 Palmdale | Measure Life (Years) | Measure Cost ($) | 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 Compressor Systems | 15 | $30,016 | $1.15 | $11,251 | $0.43 | $305,279 | $12 | -264,012 | -10.36 |

| | | | |
|---|---|---|---|
| | 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.<br><br>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.<br><br>The condenser specific efficiency analysis utilized the existing 2008 standard with floating head pressure and required condenser sizing as the reference baseline.<br><br>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.<br><br>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. The prototype warehouses used in this analysis were constructed to represent typical refrigerated warehouses conforming to 2008 Title 24 standards. System types, design loads, and operating schedules were assumed to represent industry-standard practice and typical operation for this building type, based on over 10 years of Savings By Design data. 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. 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**

A description of the refrigerated warehouse prototypes used in this analysis is shown in Appendix B. Not every prototype was used in the evaluation of each measure, either due to inappropriateness of size or system type for each particular measure.

### *3.2 Simulation and Cost Effectiveness Methodology*

The energy usage for each measure in each prototype warehouse was evaluated using DOE2 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, and uses 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 and is considered an acceptable measure to the standard. 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 benefit to cost ratio is presented for incremental analysis of measures. The benefit to cost ratio is calculated by dividing the TDV cost savings (benefit) by the measure incremental cost (cost). Any value over 1.0 is considered cost effective and an acceptable value for the measure.

[1] TDV methodology, Version 2

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.

Acceptance testing analysis assumed that the energy savings were captured in the 2008 CASE analysis for refrigerated warehouses. The 2008 CASE analysis measures were evaluated as if the equipment was working properly and did not include acceptance test cost. The measures addressed in the 2013 CASE acceptance test were evaluated for cost effectiveness by adding the Present Value (PV) of acceptance test costs to the incremental measure cost from 2008 CASE, then subtracting the 2008 TDV measure cost to get the LCC. This analysis ensured the cost effectiveness of the measures evaluated in 2008 that require an acceptance test once the costs of the acceptance tests were included. Assumptions for labor costs were gathered by survey and by protocol field tests. Survey results are presented in Appendix D.

This report also includes certain code changes recommendations which did not require energy 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 changes are described in Section 4.9 "Code Language Changes Not Requiring Analysis".

### *3.3 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 to evaluate the sensitivity of the cost-effectiveness results to roof and floor area (other prototypes were omitted because the results of such a study would be redundant—the freezer roof and floor area are the same between Prototypes #1 and 2, and between Prototypes #3, 4 and 5). 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, based on certified product information provided by insulation manufacturers.

| 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 75°F mean temperature | 1.80 lb/ft$^3$ | 0.29 Btu/(lb-°F) |

**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**

Energy-use analysis results for each climate zone are presented in Section 8.1 of Appendix C. The energy-use analysis results were used to establish a regression of simulated energy use versus insulation R-value.

### 4.2.1 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. The numbers in italics are those with B/C ratio less than 1.0, or not cost effective.

| Prototype Warehouse | | Annual Energy Savings | | TDV Cost Savings | | Incremental Cost ($) | Benefit/Cost Ratio |
|---|---|---|---|---|---|---|---|
| | | kWh | kWh/SF | $ | $/SF | | |
| **CTZ03 Oakland** | | | | | | | |
| #2 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 |
| #4 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** | | | | | | | |
| #2 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 |
| #4 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** | | | | | | | |
| #2 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 |
| #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** | | | | | | | |
| #2 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 |
| #4 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** | | | | | | | |
| #2 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 |
| #4 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** | | | | | | | |
| #2 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 |
| #4 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** | | | | | | | |
| #2 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 |

| | | | | | | | |
|---|---|---|---|---|---|---|---|
| #4 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 recommended R-value was the highest value found to be generally cost-effective, versus the 2008 standard, using LCC economic analysis methodology. The ASHRAE recommendation for -10°F to -20°F holding freezer roof insulation is R-45 to R-50.[2] The ASHRAE recommendation 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 mean temperatures (40$^o$F) which is higher than the rating basis for Title 24 standards (measured at 75$^o$F mean temperature) 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 cannot 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.

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

| 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**

### 4.2.2 Floor Insulation Analysis Results by Climate Zone

The freezer floor measure was simulated 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. Three climate zones were assumed to be sufficient to fully analyze this measure because the weather-dependent soil temperature was overridden, negating the utility of simulating a wide variety of climate zones. 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. Note that, in this case, the code intends to establish a minimum R-value while also giving flexibility to the design engineer to select the freezer floor insulation thickness that best optimizes the available heat from subcooling while minimizing the risk of freezing the soil below the floor. The optimum insulation value is likely to be greater than R-20, but less than R-35.

### *4.3 Evaporator Fan Control for Single 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." This measure evaluates the cost-effectiveness of replacing this exception with a requirement that evaporator fan speeds are reduced when the compressor is not pumping refrigerant. For this measure, a separate prototype (Prototype Warehouse #5) was developed based on the small Prototype Warehouse #3, which utilizes 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<br>Freezer System: 23°F<br>Dock System: 28°F | Cooler System: -22°F<br>Freezer System: 23°F<br>Dock System: 28°F | Cooler System: -22°F<br>Freezer System: 23°F<br>Dock System: 28°F |
| Condensing unit catalog capacity at design conditions | Cooler: 192.0 MBH<br>Freezer: 85.1 MBH<br>Dock: 209.9 MBH | Cooler: 160.7 MBH<br>Freezer: 68.6 MBH<br>Dock: 181.5 MBH | Cooler: 163.9 MBH<br>Freezer: 72.2 MBH<br>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<br>Freezer: 5<br>Dock: 2 | Cooler: 3<br>Freezer: 7<br>Dock: 3 | Cooler: 3<br>Freezer: 7<br>Dock: 3 |

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

Two control methods were considered based on discussion with evaporator coil manufacturers and other vendors: two-speed control, and variable speed control in a two-speed configuration. Both methods are implemented in different ways, depending on the size of the evaporator. For small, low-profile evaporators, the two-speed control method can be accomplished at almost no cost. These units are equipped with single-phase electronically-commutated (EC, also called brushless DC or BLDC) motors, which are inherently variable-speed capable. The motors accept an external speed signal (usually a 0-10 volt signal) provided by a speed controller. Although these evaporators are smaller than needed in most refrigerated warehouses, their capabilities may soon be available in larger

evaporators. For larger three-phase motors, external two-speed controllers are currently in the market, and variable-speed control methods can be used in a two-speed configuration. For the variable-speed option, two methods are possible: utilizing a variable speed drive, or control via EC motors which, as mentioned previously, may soon be available for these larger evaporators.

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 to speed reduction. As such, analysis of this alternative was performed.

Figure 11 summarizes the simulated runs for the evaluation of this measure.

| | |
|---|---|
| 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**

For the two-speed options (runs 1 through 6), this analysis assumed that control was accomplished with separate variable speed drives and output filters for each evaporator and a controller with associated control capability on each condensing unit. This was assumed to be the most expensive method of implementing this measure (note that the method of implementation only affects the real-

world measure cost and not the simulation methodology). For the fan-staging option (runs 7 through 9), the incremental cost was assumed to be equal to the cost of separate terminal blocks and extra relays at each evaporator.

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 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 13 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 13: 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.

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.

### *4.5 Condenser Specific Efficiency*

The cost-effectiveness of establishing a minimum condenser specific efficiency was analyzed. Condenser specific efficiency is the condenser Total Heat of Rejection (THR) capacity divided by the input electric power at 100 percent fan speed (including spray pump electric input power for evaporative condensers) at standard conditions. Figure 14 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 14: Description of prototype warehouses for condenser specific efficiency measure**

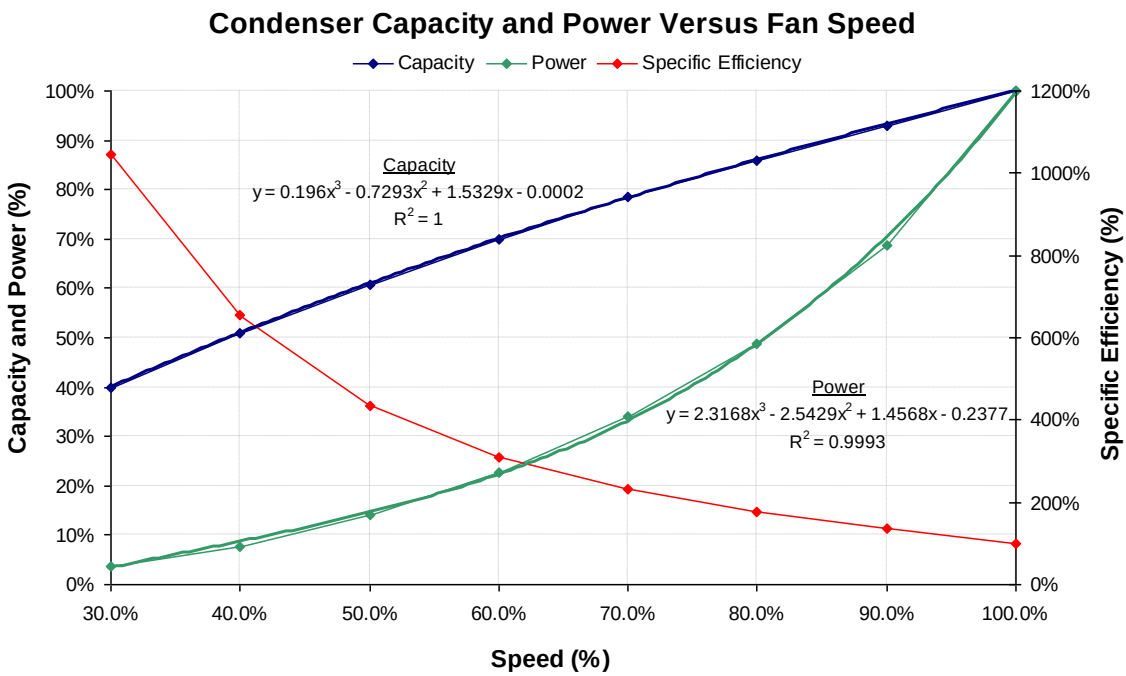
For this measure, the prototype warehouses were simulated with a 70°F minimum condensing temperature, an ambient-following control strategy and variable speed control of all condenser fans. DOE-2.2R simulation keywords explicitly applied the subject control strategy.

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.

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 15, showing normalized values for heat rejection capacity, fan power, and resultant specific efficiency as a function of fan speed, illustrates the sensitivity of specific efficiency to fan speed, with everything else held constant. Plots of capacity and power increase reference the left scale, while the plot of specific efficiency increase references the right scale.




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

**Figure 15: Graph of condenser capacity and power versus speed**

Figure 15 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 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 16 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. Figure 17 demonstrates this concept with a starting full-speed capacity of 8,537 MBH and a starting specific efficiency of 325 Btuh/Watt.

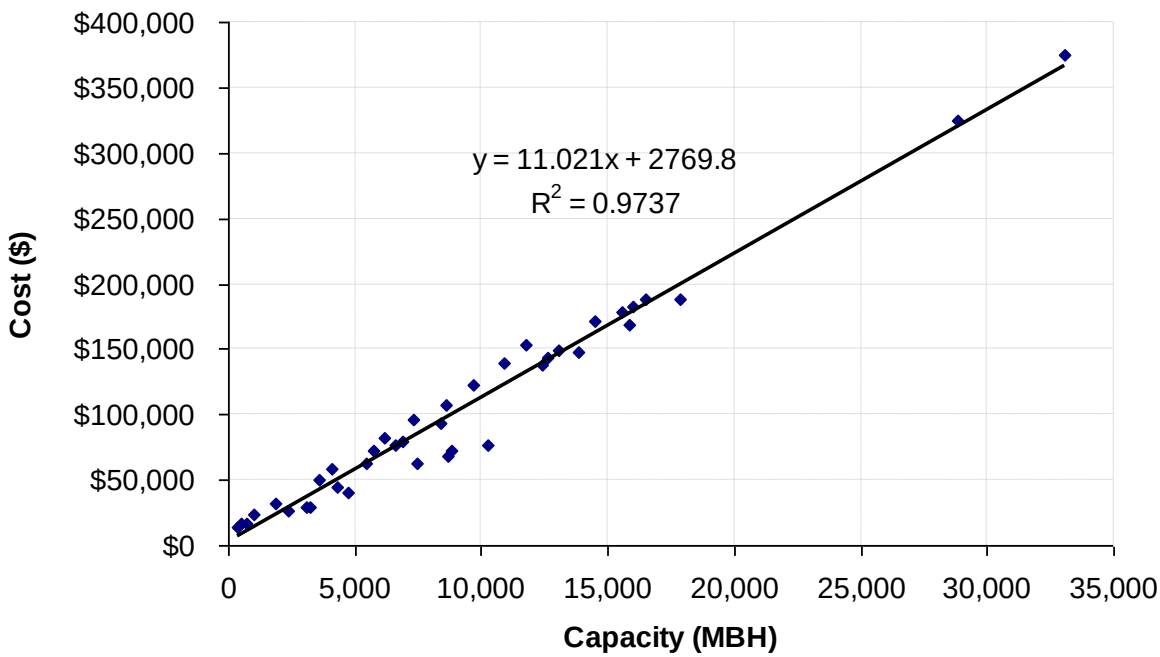



**Figure 16: Condenser cost versus capacity at specific-efficiency rating conditions**

| 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 17: 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 18 shows the simulation results for the large prototype warehouse with an evaporative-cooled condenser.

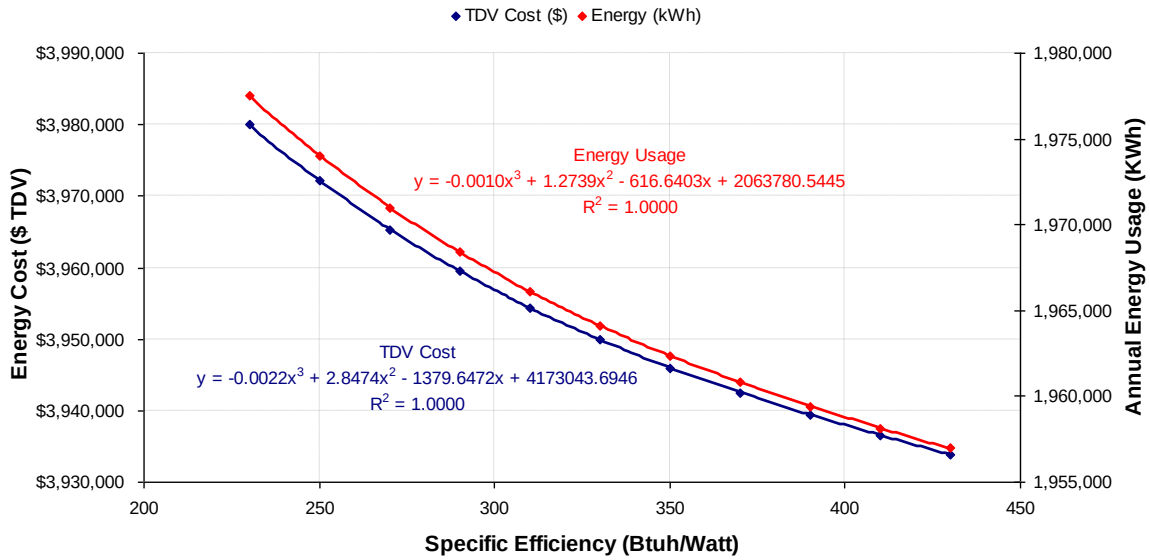



**Figure 18: Example of building energy use and TDV energy cost versus specific efficiency**

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).

Analysis data for evaporative-cooled ammonia condensers, evaporative-cooled halocarbon condensers, and air-cooled halocarbon condensers with both EC and non-EC motors, are presented in Appendix I.

### 4.5.1 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. Figure 19 summarizes the results from the preliminary analysis.

The base case specific efficiency for statewide savings analysis listed in Figure 19 was obtained from Savings By Design 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 Savings By Design 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 19: Preliminary condenser specific efficiency results**

### 4.5.2 Condenser Specific Efficiency Analysis Results by Climate Zone

Figure 20 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 | 3.2 |
| Indoor HFC evaporative-cooled | 356 | 0.01 | $765 | $0.03 | $279 | 2.7 |
| Outdoor HFC air-cooled | 1,556 | 0.06 | $6,461 | $0.25 | $2,531 | 2.5 |
| Outdoor HFC air-cooled BLDC motors | 1,556 | 0.06 | $6,461 | $0.25 | $4,939 | 1.3 |
| ***CTZ05 Santa Maria*** | | | | | | |
| Outdoor $NH_3$ evaporative-cooled | 7,909 | 0.09 | $17,523 | $0.19 | $5,812 | 3.0 |
| Indoor HFC evaporative-cooled | 350 | 0.01 | $748 | $0.03 | $279 | 2.7 |
| Outdoor HFC air-cooled | 1,443 | 0.06 | $4,396 | $0.17 | $2,531 | 1.7 |
| Outdoor HFC air-cooled BLDC motors | 1,443 | 0.06 | $4,396 | $0.17 | $4,939 | 0.9 |
| ***CTZ07 San Diego*** | | | | | | |
| Outdoor $NH_3$ evaporative-cooled | 8,548 | 0.09 | $18,556 | $0.20 | $5,812 | 3.2 |
| Indoor HFC evaporative-cooled | 387 | 0.02 | $810 | $0.03 | $279 | 2.9 |
| Outdoor HFC air-cooled | 1,779 | 0.07 | $5,669 | $0.22 | $2,531 | 2.2 |
| Outdoor HFC air-cooled BLDC motors | 1,779 | 0.07 | $5,669 | $0.22 | $4,939 | 1.1 |
| ***CTZ10 Riverside*** | | | | | | |
| Outdoor $NH_3$ evaporative-cooled | 8,850 | 0.10 | $20,612 | $0.224 | $6,178 | 3.3 |
| Indoor HFC evaporative-cooled | 378 | 0.02 | $801 | $0.031 | $281 | 2.8 |
| Outdoor HFC air-cooled | 5,594 | 0.22 | $23,967 | $0.922 | $4,110 | 5.8 |
| Outdoor HFC air-cooled BLDC motors | 5,594 | 0.22 | $23,967 | $0.922 | $6,472 | 3.7 |
| ***CTZ12 Sacramento*** | | | | | | |
| Outdoor $NH_3$ evaporative-cooled | 9,337 | 0.10 | $20,932 | $0.23 | $6,358 | 3.3 |
| Indoor HFC evaporative-cooled | 338 | 0.01 | $721 | $0.03 | $288 | 2.5 |
| Outdoor HFC air-cooled | 4,658 | 0.18 | $21,867 | $0.84 | $4,179 | 5.2 |
| Outdoor HFC air-cooled BLDC motors | 4,658 | 0.18 | $21,867 | $0.84 | $6,581 | 3.3 |

[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.

| | | | | | | |
|---|---|---|---|---|---|---|
| ***CTZ13 Fresno*** | | | | | | |
| Outdoor $NH_3$ evaporative-cooled | 9,612 | 0.10 | $21,822 | $0.24 | $6,358 | 3.4 |
| Indoor HFC evaporative-cooled | 344 | 0.01 | $730 | $0.03 | $288 | 2.5 |
| Outdoor HFC air-cooled | 7,680 | 0.30 | $28,096 | $1.08 | $4,179 | 6.7 |
| Outdoor HFC air-cooled BLDC motors | 7,680 | 0.30 | $28,096 | $1.08 | $6,581 | 4.3 |
| ***CTZ14 Palmdale*** | | | | | | |
| Outdoor $NH_3$ evaporative-cooled | 9,441 | 0.10 | $24,261 | $0.26 | $6,178 | 3.9 |
| Indoor HFC evaporative-cooled | 350 | 0.01 | $801 | $0.03 | $281 | 2.8 |
| Outdoor HFC air-cooled | 7,915 | 0.30 | $28,675 | $1.10 | $4,110 | 7.0 |
| Outdoor HFC air-cooled BLDC motors | 7,915 | 0.30 | $28,675 | $1.10 | $6,472 | 4.4 |

**Figure 20: Analysis results by climate zone for condenser specific efficiency measure**

Preliminary (incremental) analysis to identify the proposed specific efficiency levels was done in a cool, humid, coastal climate zone (CTZ05 Santa Maria), as well as hot, dry, inland climate zones (CTZ10 Riverside, and CTZ12 Sacramento). The result of the preliminary analysis was then applied in the remaining 4 climate zones to ensure cost-effectiveness in more areas.

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.

### *4.6 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.

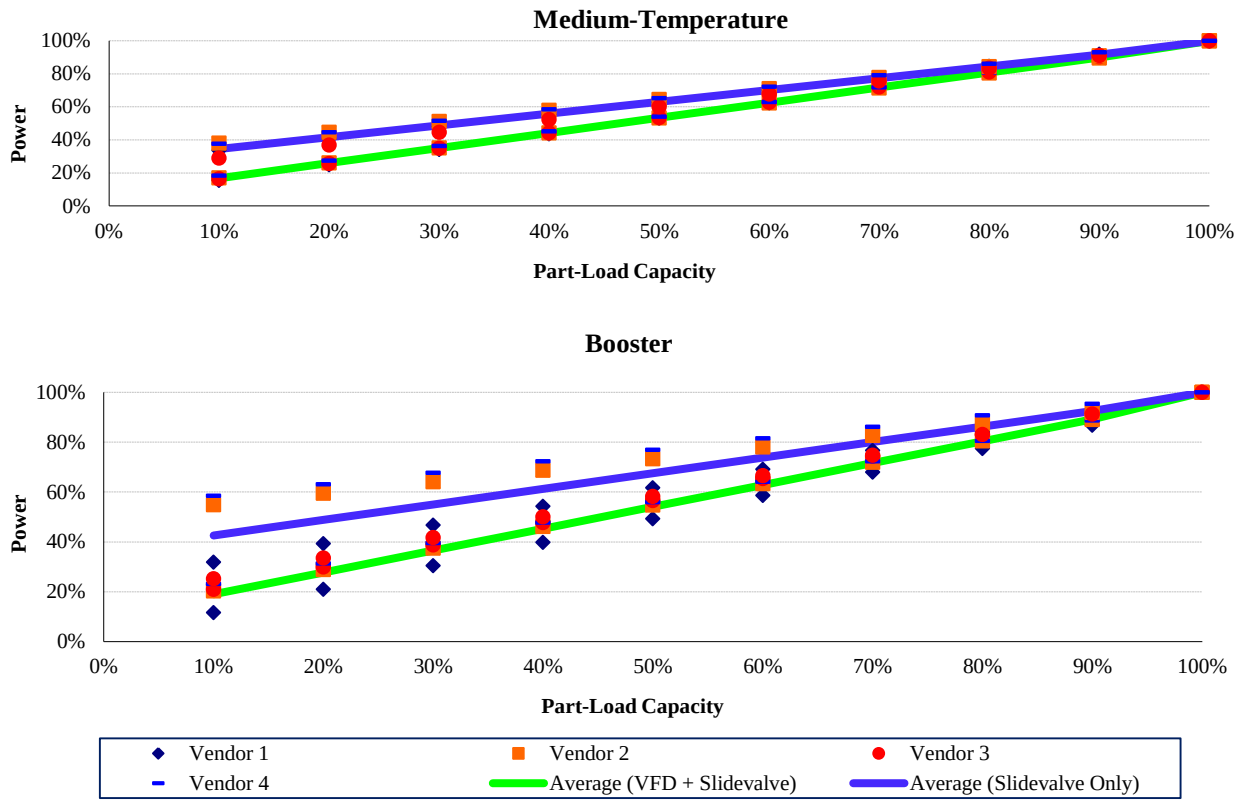
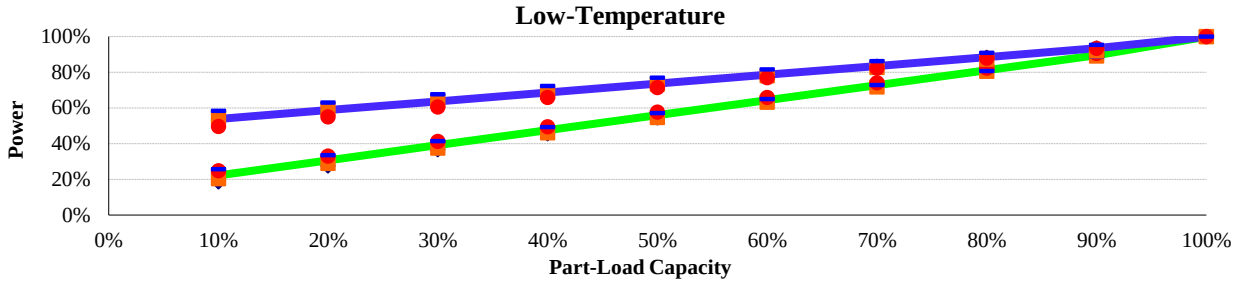



Figure 21 shows the simulated part-load performance curves for both the base case and the proposed case.



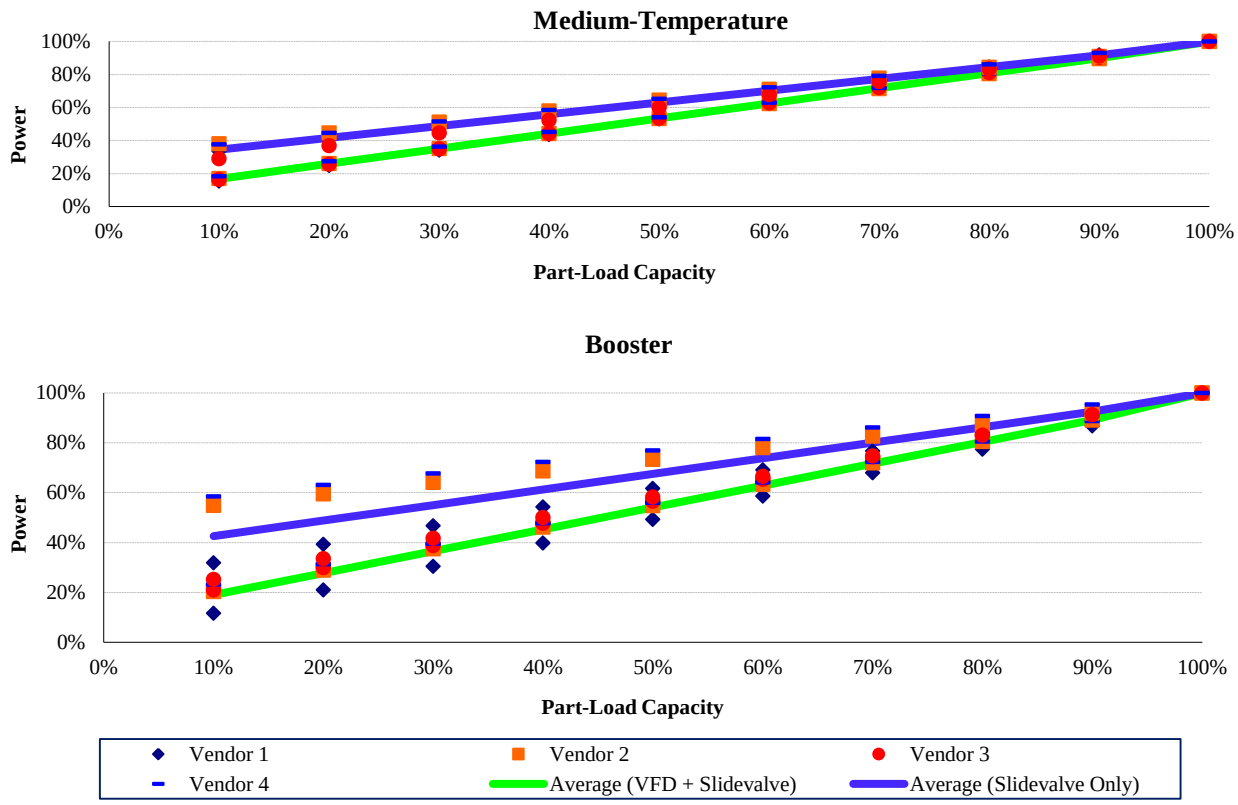



**Figure 21: Part-load performance curves for slide valve and variable-speed control**

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).

### 4.6.1 Compressor Variable Speed Control Analysis Results by Climate Zone

Figure 22 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 22: Savings analysis results for screw compressor variable speed measure**

Figure 22 shows that variable-speed compressor control is generally cost-effective across all applications and climate zones. However, variability in compressor sizing may affect conclusions. Therefore, a supplemental sensitivity analysis was performed; results are below.

### 4.6.2 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 23 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 23: 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.

## *4.7 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.

Both small and large warehouse prototypes were used to evaluate strip curtains 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). Strip curtains were also simulated between the 35°F cooler and the partially conditioned warehouse. For the base case, wide-open doors were

used. Strip curtains were assumed to be 85% effective[4] at arresting inter-zonal infiltration air versus wide-open doors.

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 24 shows the analysis results of simulating wide-open doors and incremental cost-effectiveness results for 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.

| | | 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 | 144,571 | 3,274 | $291,331 | $954 | 305 |
| | -10°F Freezer to Partially-Conditioned Warehouse | 461,458 | 10,745 | $956,262 | $954 | 1,002 |
| | -10°F Freezer to 40°F Refrigerated Dock | 127,529 | 3,336 | $296,902 | $954 | 311 |
| Large Warehouse | 35°F Cooler to Partially-Conditioned Warehouse | 218,929 | 5,179 | $460,941 | $1,908 | 242 |
| | -10°F Freezer to Partially-Conditioned Warehouse | 680,209 | 14,886 | $1,324,780 | $1,908 | 694 |
| | -10°F Freezer to 40°F Refrigerated Dock | 357,041 | 7,639 | $679,855 | $1,908 | 356 |

**Figure 24: Strip curtain savings and cost-effectiveness analysis results**

Figure 24 shows that strip curtains 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.

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. Infiltration barriers are standard design practice for nearly all

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

refrigerated warehouse new construction projects, and this requirement is just establishing a baseline (a baseline can be established with no economic basis if the majority of new construction includes it).

Since stakeholders generally agree that a strip curtain is already the minimum infiltration barrier and because cost analysis shows that it has the least capital cost, this measure intends to establish strip curtains as the code-minimum infiltration barrier, 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.

### *4.8 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.

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 eight 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 $4,700.

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

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

The cost for evaporator acceptance testing is approximately $5,200 in small warehouses and approximately $7,000 in large warehouses.

Results of the cost analysis of acceptance tests are shown in Figure 25. Results are the average of all climate zones and of small and large warehouses.

| Acceptance Test Cost Results | TDV Cost Savings ($/ft$^2$) | Measure Cost ($/ft$^2$) | Benefit/ Cost | LCC ($/ft$^2$) |
|---|---|---|---|---|
| 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 25: Acceptance test cost analysis results**

### *4.9 Code Language Changes Not Requiring Analysis*

A number of changes to the code language were made that were 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.

### *4.10 Equipment Rating Accuracy, Standards and Certification*

Currently, the equipment performance requirements in this standard are defined without the benefit or use of rating standards (e.g. AHRI, ASHRAE, CTI) or certification of manufacturers' data. Whereas today most air conditioning equipment is rated to common standards and independently certified, this is generally not the case for refrigeration condensers and evaporators and some compressors. The data provided in manufacturers' catalogs was relied upon in the development of this CASE report, and will be the basis upon which owners and engineers will design systems and determine compliance with code requirements. As a result, the minimum performance requirements derived herein are necessarily somewhat conservative, as well as reflecting rather general assumptions regarding the facts and characteristics of how actual equipment operates vs. catalog values.

Requiring equipment ratings to be published in accordance with common standards and requiring certification of these ratings was considered and discussed. Existing test and rating standards that might be used were considered, along with the real-world application considerations that would affect standards and application of equipment. The conclusion was that considerable work was required to develop appropriate standards and that equipment data would likely change substantially (to a greater extent for smaller equipment). These facts, along with the costs of labs, testing and product line changes would impose a large cost on industry. Considering these facts, it was determined that the proposed code requirements could be reasonably undertaken with the existing "state of the art" concerning performance data, as long as the level of stringency was carefully moderated.

Throughout the development of this CASE report, stakeholders often noted support for prospective test standards and certification for the subject equipment. Equipment designed and rated to common standards would be beneficial to manufacturers and end-users of the equipment by creating a "level playing field", allowing better system design and ultimately leading to greater system efficiency and greater trust in performance values, ultimately reduced first costs through system right sizing. Future code minimum performance requirements and cost-effectiveness could certainly be more refined and exacting. Perhaps most importantly, standard ratings would be a first step towards performance definitions sufficient for a Performance Compliance Option, which industry stakeholders have strongly noted should be available for refrigerated warehouses.

Continued work is recommended to support development of the relevant equipment standards and methods to allow consideration in the next code cycle.

# 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 condenser Total Heat of Rejection (THR) capacity divided by the input electric power at 100 percent fan speed (including spray pump electric input power for evaporative condensers) at standard conditions.

**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.

TABLE 126-B FAN-POWERED CONDENSERS – MINIMUM EFFICIENCY REQUIREMENTS

| **Condenser Type** | **Refrigerant Type** | **Minimum Efficiency [a]** | **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 | |
| [a] See section 101 for definition of condenser specific efficiency | | | |

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 into the control system for the functional test steps. Allow 5 minutes for system to normalize.

Step 2: Using the control system, lower test temperature or humidity 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 was 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.[5] 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 26 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<br>CTZ05 – Santa Maria<br>CTZ07 – San Diego (Lindbergh) |
| 2 | Medium-Temperature, Central Valley | Sacramento | 104°F/74°F | CTZ12 – Sacramento Executive Airport<br>CTZ13 – Fresno |
| 3 | Hot Temperature, Inland Empire | Riverside | 106°F/75°F | CTZ10 – Riverside<br>CTZ14 – Palmdale |

**Figure 26: Description of three design climate zones**

Figure 27 through Figure 34 represent example load calculation worksheets used to size refrigeration equipment in the prototype warehouses.

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

VaCom Technologies — DATE: 12/31/2009

Prototype Warehouse 1 and 2

BOX: **Cooler**

TEMP (° F): 35

LENGTH (ft.): 200 — AREA (S.F.): 40,000

WIDTH (ft.): 200 — VOLUME (ft$^3$): 1,200,000

HEIGTH (ft.): 30

INSULATION TYPE: Polyisocyanurate

| | THICKNESS (in.) | "R/INCH" | "R" | "U" |
|---|---|---|---|---|
| CEILING: | 5.283 | 5.3 | 28 | 0.0357 |
| FLOOR: | 8.000 | 0.66 | 5.28 | 0.1894 |
| 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 | 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 = | 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 |
| LOAD WITH SAFETY FACTOR (Btuh): | 1,170,935 | 115.0% | 97.6 | 410 |
| COIL DESIGN LOAD (Btuh): | 1,277,384 | 125.5% | 106.4 | 376 |

SAFETY FACTOR: 1.15

COIL OP. HOURS: 22

**Figure 27: Load calculations, 35°F cooler space (Prototype Warehouses #1 and 2)**

VaCom Technologies — DATE: 12/31/2009

Prototype Warehouse 1 and 2

BOX: **Freezer**

TEMP (° F): -10

LENGTH (ft.): 200 — AREA (S.F.): 40,000

WIDTH (ft.): 200 — VOLUME (ft³): 1,200,000

HEIGTH (ft.): 30

INSULATION TYPE: Polyisocyanurate

| | THICKNESS (in.) | "R"/INCH" | "R" | "U" |
|---|---|---|---|---|
| CEILING: | 6.792 | 5.3 | 36 | 0.0278 |
| FLOOR: | 6.792 | 5.3 | 36 | 0.0278 |
| 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) | ip. 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% |

SAFETY FACTOR: 1.15

COIL OP. HOURS: 22

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

**Figure 28: Load calculations, -10°F freezer space (Prototype Warehouses #1 and 2)**

VaCom Technologies — DATE: 12/31/2009

Prototype Warehouse 1

BOX: **Dock**
TEMP (°F): 40
LENGTH (ft.): 400
WIDTH (ft.): 30
HEIGTH (ft.): 30

AREA (S.F.): 12,000
VOLUME (ft³): 360,000

INSULATION TYPE: Polyisocyanurate

| | THICKNESS (in.) | "R/INCH" | "R" | "U" |
|---|---|---|---|---|
| CEILING: | 5.283 | 5.3 | 28 | 0.0357 |
| FLOOR: | 8.000 | 0.66 | 5.28 | 0.1894 |
| 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) | op. 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 |
| LOAD WITH SAFETY FACTOR: | 1,015,525 | 115.0% | 84.6 | 142 |
| COIL DESIGN LOAD: | 1,107,846 | 125.5% | 92.3 | 130 |

SAFETY FACTOR: 1.15
COIL OP. HOURS: 22

**Figure 29: Load calculations, 40°F dock space (Prototype Warehouse #1)**

VaCom Technologies — DATE: 12/31/2009

Prototype Warehouse 2

BOX: **Dry Warehouse**
TEMP (° F): 85
LENGTH (ft.): 400
WIDTH (ft.): 50
HEIGTH (ft.): 30

AREA (S.F.): 20,000
VOLUME (ft$^3$): 600,000

INSULATION TYPE: Polyisocyanurate

| | THICKNESS (in.) | "R/INCH" | "R" | "U" |
|---|---|---|---|---|
| CEILING: | 3.585 | 5.3 | 19 | 0.0526 |
| FLOOR: | 8.000 | 0.66 | 5.28 | 0.1894 |
| 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 |
| LOAD WITH SAFETY FACTOR: | 642,994 | 115.0% | 53.6 | 373 |
| COIL DESIGN LOAD: | 642,994 | 115.0% | 53.6 | 373 |

SAFETY FACTOR: 1.15
COIL OP. HOURS: 24

**Figure 30: Load calculations, 85°F dry storage space (Prototype Warehouse #2)**

VaCom Technologies — DATE: 12/31/2009

Prototype Warehouse 3 and 4

BOX: **Cooler**

TEMP (°F): 35

LENGTH (ft.): 100 — AREA (S.F.): 10,000

WIDTH (ft.): 100 — VOLUME (ft³): 300,000

HEIGTH (ft.): 30

INSULATION TYPE: Polyisocyanurate

| | THICKNESS (in.) | "R/INCH" | "R" | "U" |
|---|---|---|---|---|
| CEILING: | 5.283 | 5.3 | 28 | 0.0357 |
| FLOOR: | 8.000 | 0.66 | 5.28 | 0.1894 |
| 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 |
| LOAD WITH SAFETY FACTOR (Btuh): | 318,804 | 115.0% | 26.6 | 376 |
| COIL DESIGN LOAD (Btuh): | 347,787 | 125.5% | 29.0 | 345 |

SAFETY FACTOR: 1.15

COIL OP. HOURS: 22

**Figure 31: Load calculations, 35°F cooler space (Prototype Warehouses #3 and 4)**

VaCom Technologies — DATE: 12/31/2009

Prototype Warehouse 3 and 4

BOX: **Freezer**

TEMP (° F): -10

LENGTH (ft.): 100 — AREA (S.F.): 10,000

WIDTH (ft.): 100 — VOLUME (ft³): 300,000

HEIGTH (ft.): 30

INSULATION TYPE: Polyisocyanurate

| | THICKNESS (in.) | "R"/INCH" | "R" | "U" |
|---|---|---|---|---|
| CEILING: | 6.792 | 5.3 | 36 | 0.0278 |
| FLOOR: | 6.792 | 5.3 | 36 | 0.0278 |
| 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) | ip. 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 |
| LOAD WITH SAFETY FACTOR: | 404,354 | 115.0% | 33.7 | 297 |
| COIL DESIGN LOAD: | 441,113 | 125.5% | 36.8 | 272 |

SAFETY FACTOR: 1.15

COIL OP. HOURS: 22

**Figure 32: Load calculations, -10°F freezer space (Prototype Warehouses #1 and 2)**

VaCom Technologies — DATE: 12/31/2009

Prototype Warehouse 3

BOX: **Dock**
TEMP (°F): 40
LENGTH (ft.): 200
WIDTH (ft.): 30
HEIGTH (ft.): 30

AREA (S.F.): 6,000
VOLUME (ft³): 180,000

INSULATION TYPE: Polyisocyanurate

| | THICKNESS (in.) | "R/INCH" | "R" | "U" |
|---|---|---|---|---|
| CEILING: | 5.283 | 5.3 | 28 | 0.0357 |
| FLOOR: | 8.000 | 0.66 | 5.28 | 0.1894 |
| 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) | ip. 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 |
| LOAD WITH SAFETY FACTOR: | 370,033 | 115.0% | 30.8 | 195 |
| COIL DESIGN LOAD: | 403,672 | 125.5% | 33.6 | 178 |

SAFETY FACTOR: 1.15
COIL OP. HOURS: 22

**Figure 33: Load calculations, 40°F dock space (Prototype Warehouse #3)**

VaCom Technologies — DATE: 12/31/2009

Prototype Warehouse 4

BOX: **Dry Warehouse**
TEMP (° F): 85
LENGTH (ft.): 200 — AREA (S.F.): 10,000
WIDTH (ft.): 50 — VOLUME (ft$^3$): 300,000
HEIGTH (ft.): 30

INSULATION TYPE: Polyisocyanurate

| | THICKNESS (in.) | "R/INCH" | "R" | "U" |
|---|---|---|---|---|
| CEILING: | 3.585 | 5.3 | 19 | 0.0526 |
| FLOOR: | 8.000 | 0.66 | 5.28 | 0.1894 |
| 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 |
| LOAD WITH SAFETY FACTOR: | 493,091 | 115.0% | 41.1 | 243 |
| COIL DESIGN LOAD: | 493,091 | 115.0% | 41.1 | 243 |

SAFETY FACTOR: 1.15
COIL OP. HOURS: 24

**Figure 34: 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 35, while Prototype Warehouses #3 and 4 are summarized in Figure 36.

| | **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 35: 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 36: 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 37 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<br>Cooler: 35°F<br>Dock: 40°F | Freezer: -10°F<br>Cooler: 35°F<br>Dry Storage Cooling: 85°F<br>Dry Storage Heating: 70°F | Freezer: -10°F<br>Cooler: 35°F<br>Dock: 40°F | Freezer: -10°F<br>Cooler: 35°F<br>Dry Storage Cooling: 85°F<br>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)<br>Cooler/Dock: 3.29” (R-28) | Freezer: 4.23” (R-36)<br>Cooler: 3.29” (R-28)<br>Partially-Conditioned Warehouse: 6.13” (R-19) | Freezer: 4.23” (R-36)<br>Cooler/Dock: 3.29” (R-28) | Freezer: 4.23” (R-36)<br>Cooler: 3.29” (R-28)<br>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)<br>Dry storage area: 4.19” (R-13) | Cooler/Dock: 3.29” (R-28) | Cooler: 3.29” (R-28)<br>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.<br>(2) 10’ x 10’ doors between freezer and dock.<br>Doors are automatic bi-parting break-away warehouse doors | (2) 10’ x 10’ doors between cooler and dock.<br>(2) 10’ x 10’ doors between freezer and dock.<br>Doors are automatic bi-parting break-away warehouse doors | (1) 10’ x 10’ doors between cooler and dock.<br>(1) 10’ x 10’ doors between freezer and dock.<br>Doors are automatic bi-parting break-away warehouse doors | (1) 10’ x 10’ doors between cooler and dock.<br>(1) 10’ x 10’ doors between freezer and dock.<br>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)<br>Cooler: 10°F (45°F to 35°F) | | | |
| Average Product Specific Heat | Freezer: 0.50 Btu/lb-°F<br>Cooler: 0.65 Btu/lb-°F | | | |
| Product Throughput | Freezer: 400,000 lb/day<br>Cooler: 400,000 lb/day | | Freezer: 100,000 lb/day<br>Cooler: 100,000 lb/day | |
| Product Pull down Load | Freezer: 41.7 MBH<br>Cooler: 108.3 MBH | | Freezer: 10.4 MBH<br>Cooler: 27.1 MBH | |
| Respiring Product Load | Freezer: none<br>Cooler: 171.9 MBH (750 tons of respiring product @ 5,500 Btuh/ton respiration rate) | | Freezer: none<br>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.<br>Assumed 20 MBH/forklift, 10 MBH/pallet lift | | 10 forklifts plus 3 pallet lifts distributed evenly throughout facility.<br>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<br>LT Suction Group: -23°F SST<br>MT Suction Group: 22°F SST | | Fixed SST set point with 1°F throttling range<br>LT System: -23°F SST<br>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<br>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<br>Cooler: -20°F<br>Dock: 30°F<br>10°F design TD in all spaces | Freezer: 25°F<br>Cooler: -20°F<br>10°F design TD in all spaces | Freezer: 25°F<br>Cooler: -20°F<br>Dock: 30°F<br>10°F design TD in all spaces | Freezer: 25°F<br>Cooler: -20°F<br>10°F design TD in all spaces |
| Air Unit Specific Efficiency | 34 Btuh/Watt at 10°F TD | | | |
| Defrost Method | Freezer: Hot Gas<br>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 37: 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 38, Figure 39, and Figure 40 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**

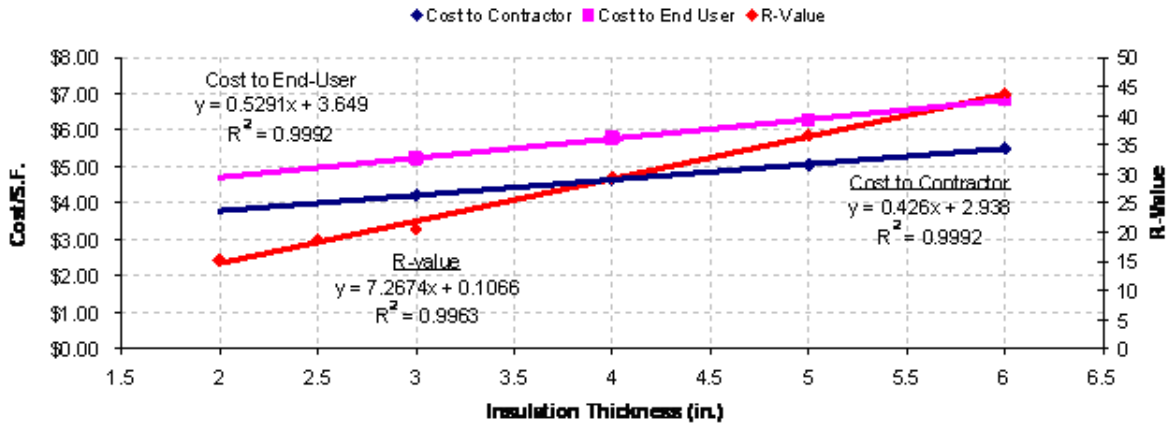
| | | |
|---|---|---|
| Reseller Mark-Up: | 20% | (est) |
| Contractor Mark-up: | 20% | (est) |

| 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 38: Cost calculation worksheet for prefabricated urethane cam-lock panels**

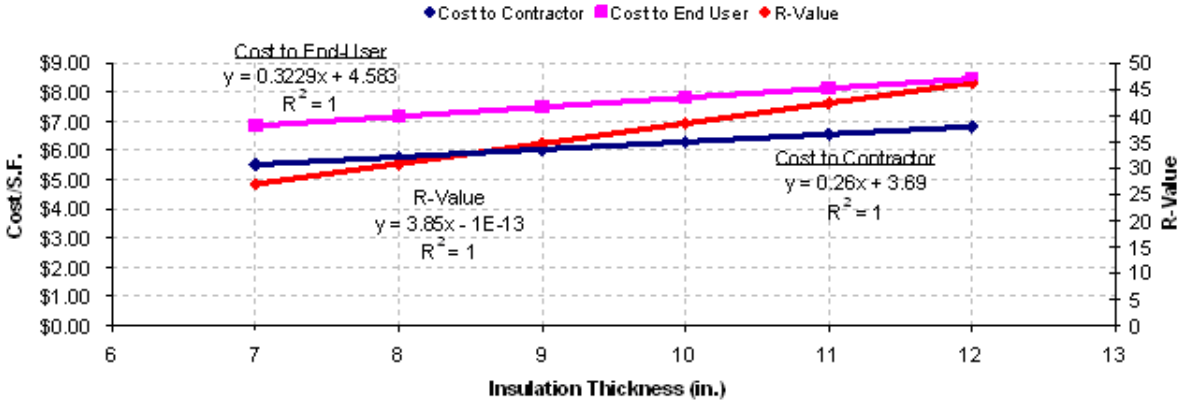
**Polyurethane**

| 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 |



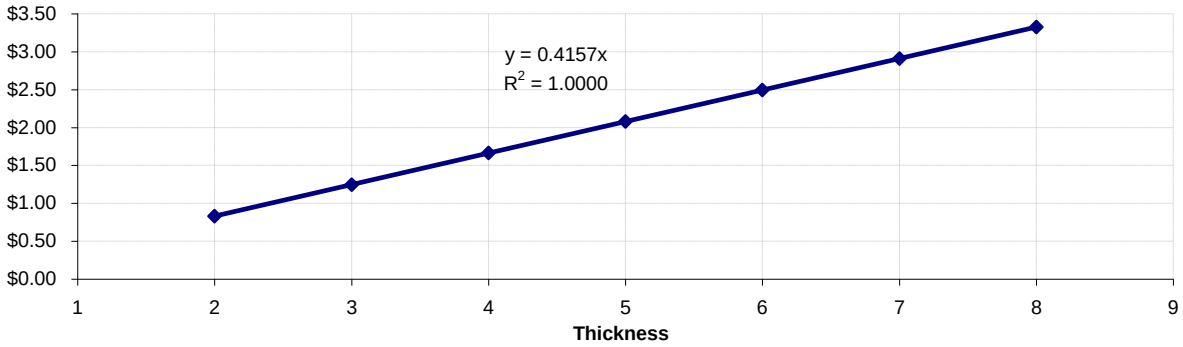

**Notes:**

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 39: 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 |



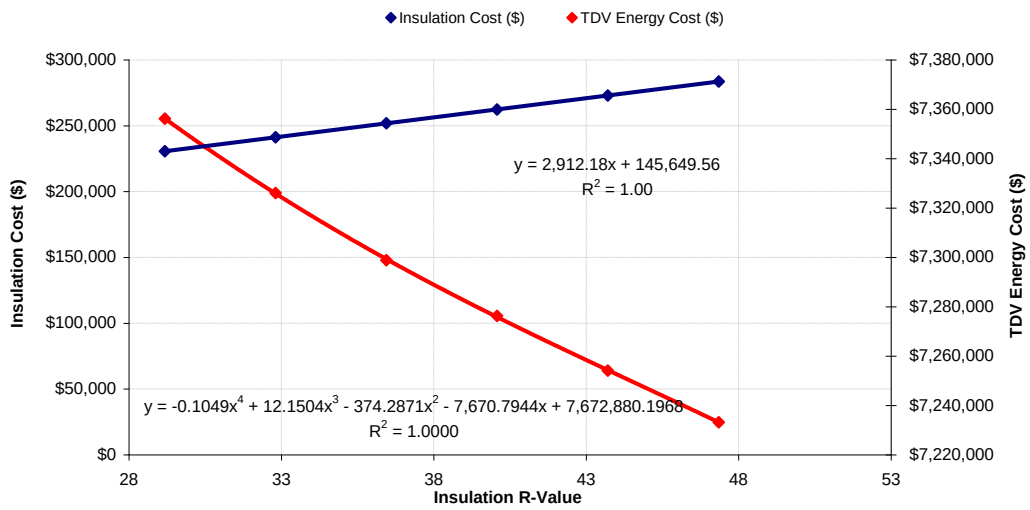

**Notes**

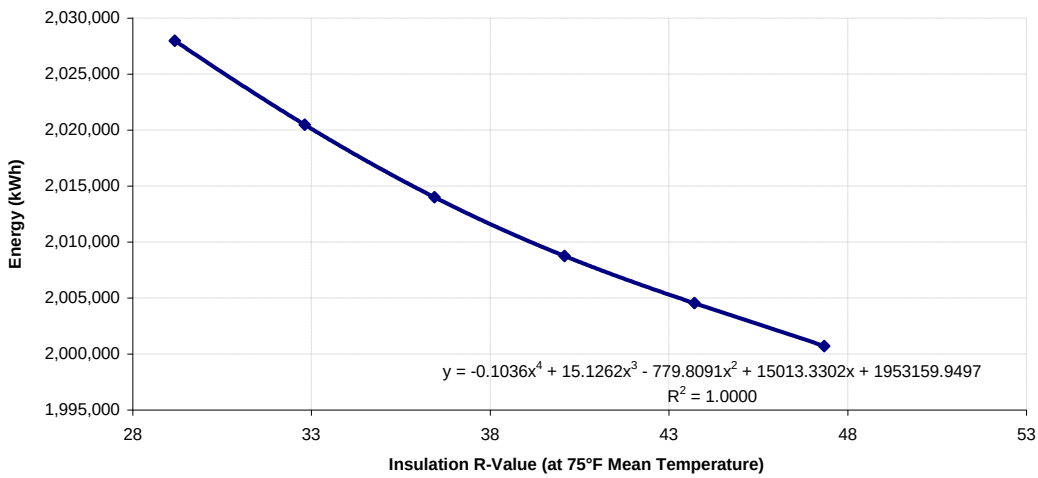
R/inch is about 6.25 based on a test method called LTTR (long-term thermal resistance), in which they take a wafer of iso board and prematurely age it, then test the resistance. Cost per SF for a 1" increment is about $0.30-$0.32 from the manufacturer to the first-time buyer (usually a distributor) for 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 40: Cost calculation worksheet for polyisocyanurate overdeck insulation**

Figure 41 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 41: 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 42 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 |

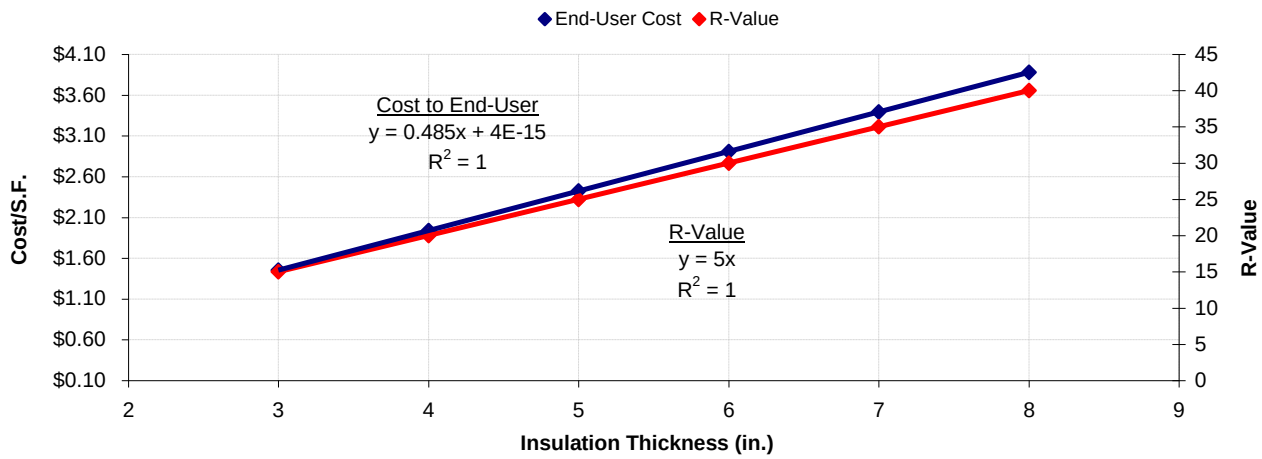



**Figure 42: Cost regression analysis for expanded polystyrene floor insulation**

Extruded polystyrene R-value is R-5.0/inch at 75$^{o}$F mean temperature and R-5.4/inch at 40$^{o}$F per American Society for Testing and Materials (ASTM) C518. Prices were estimated by a refrigeration subcontractor and include installation labor but not vapor barriers or adhesives. The 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: $0.44/ft$^2$
40 psi: $0.53/ft$^2$
60 psi: $0.64/ft$^2$

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 43 through Figure 46 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 43: 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 44: 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 45: 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 46: 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 47 through Figure 50 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 47-Figure 50 show the cost-regressions for the condenser types included in this analysis.

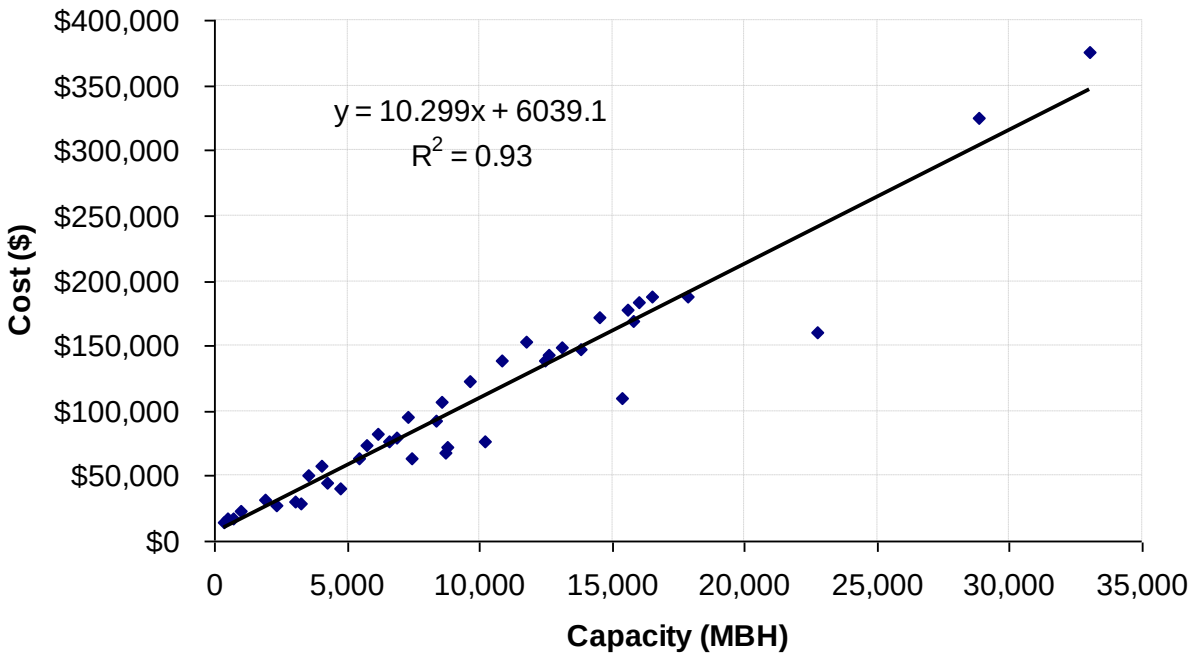



**Figure 47: Cost versus capacity regression at specific efficiency rating conditions for axial-fan evaporative-cooled ammonia condensers**

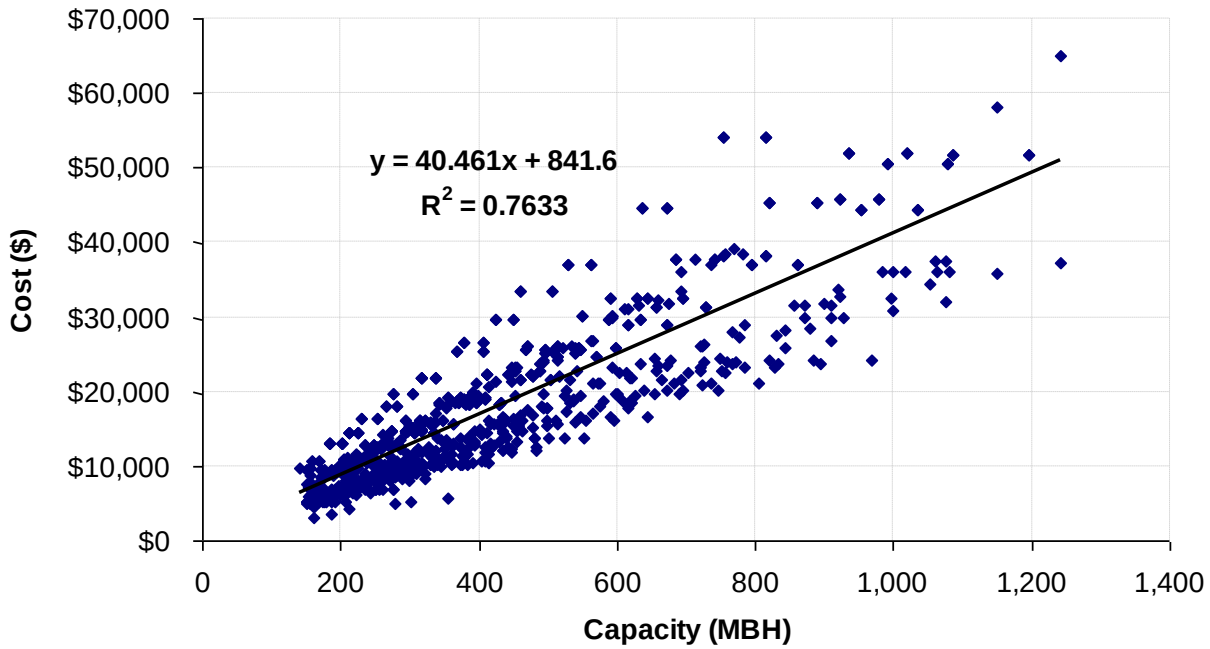



**Figure 48: Cost versus capacity regression at specific efficiency rating conditions for axial-fan air-cooled HFC condensers with standard motors**

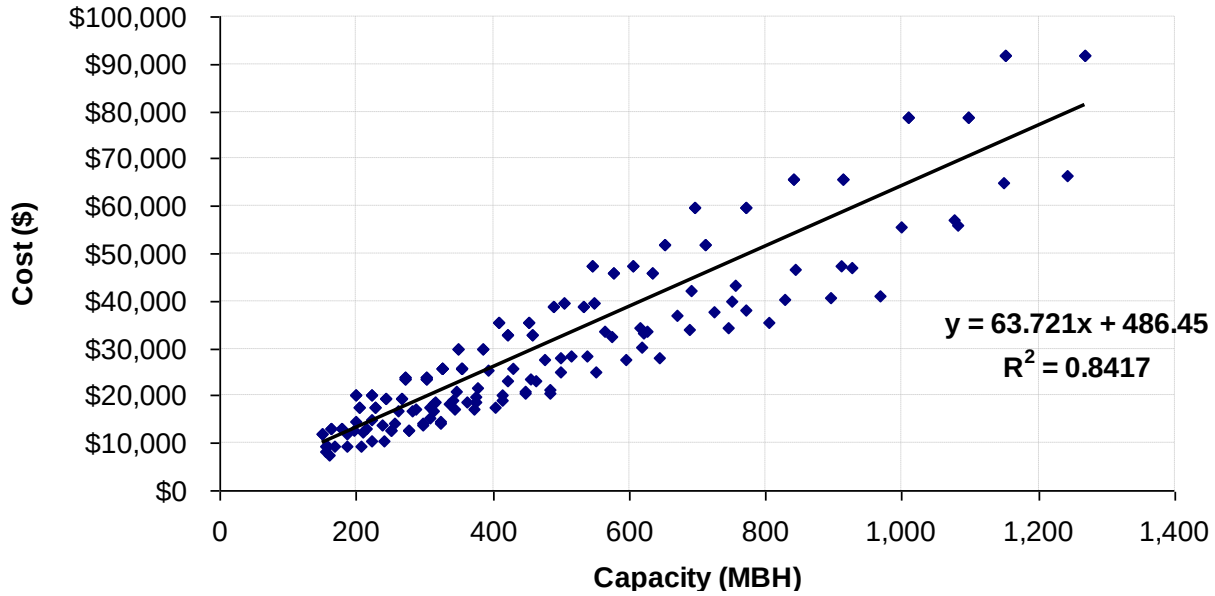



**Figure 49: Cost versus capacity regression at specific efficiency rating conditions for axial-fan air-cooled HFC condensers with BLDC motors**

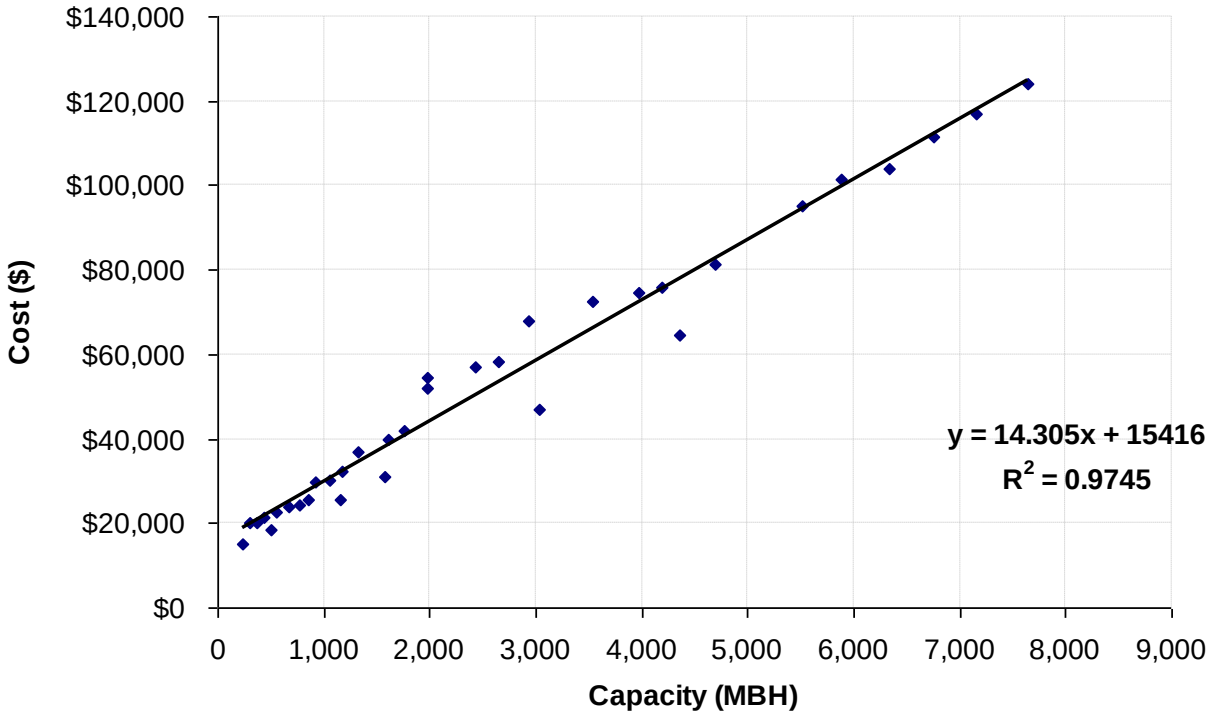



**Figure 50: Cost versus capacity regression at specific efficiency rating conditions for centrifugal-fan evaporative-cooled HFC condensers**

### *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 51 to Figure 53 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 51: 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 52: Screw compressor part-load measure cost calculator for LT, MT, and booster suction groups**

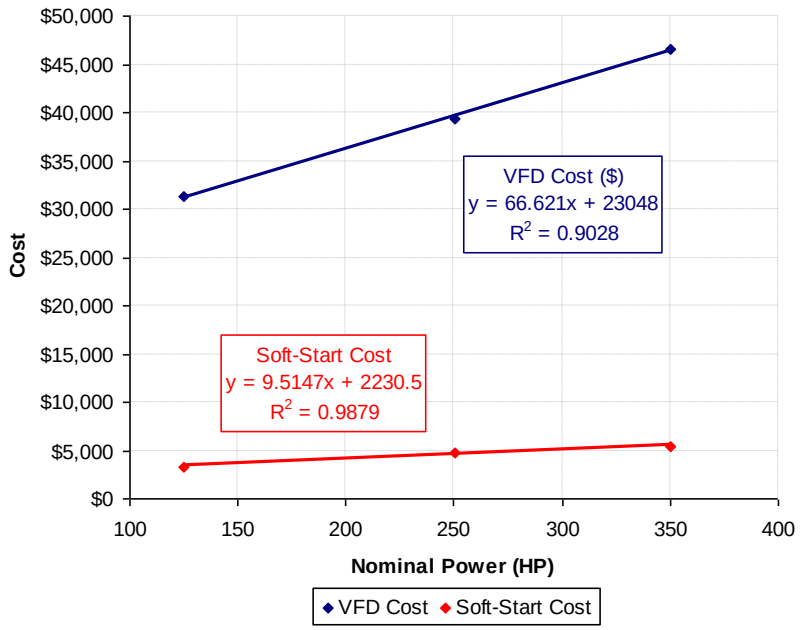



**Figure 53: Cost versus motor horsepower regressions for screw compressor speed control**

### *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 54 - Figure 57 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 54: 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 55: 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 56: 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 57: 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 58 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)<br>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 58: 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 59 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 59: 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 60 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 60: Minimum and maximum condenser catalog capacities for centrifugal-fan evaporative condensers and small axial-fan evaporative condensers**

As Figure 60 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 61 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.

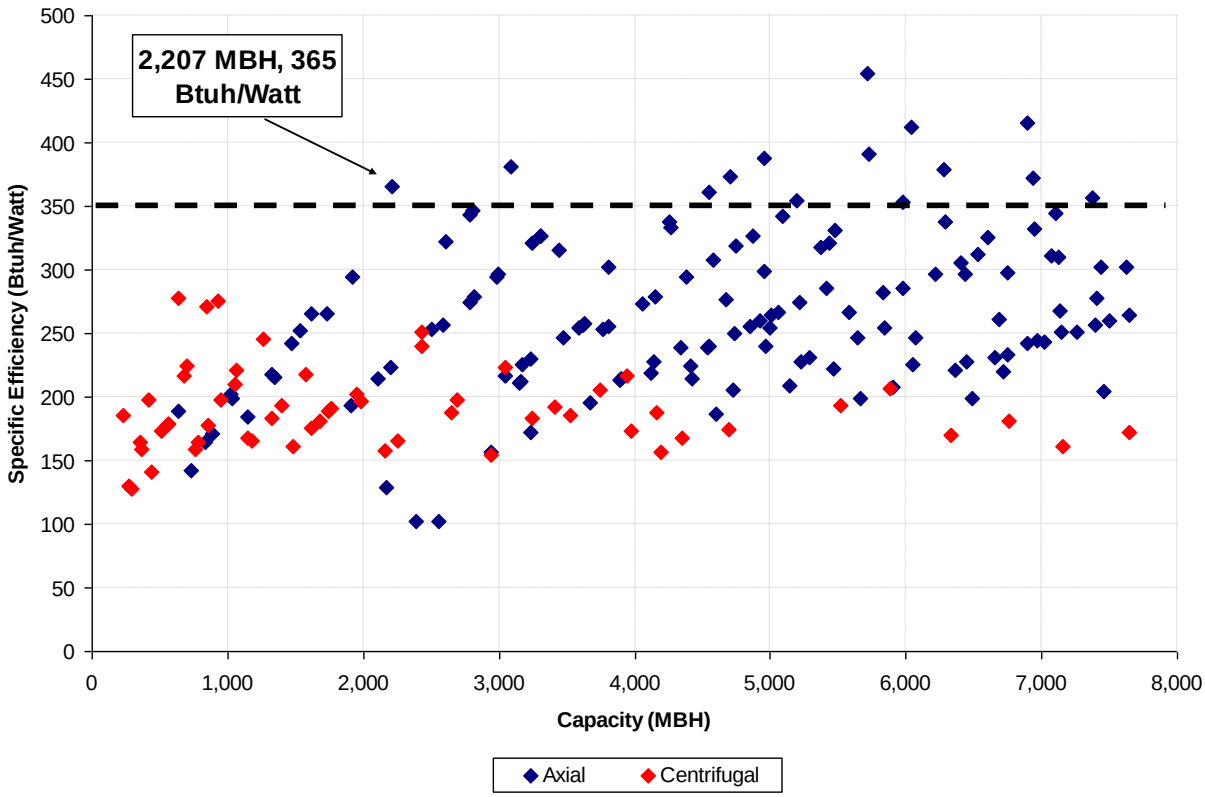



**Figure 61: Specific efficiency of centrifugal-fan and small axial-fan evaporative condensers at 100°F SCT, 70°F WBT**

Figure 61 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 - $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 steps) 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 - 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 ensure 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).<br>Almost no existing examples of variable speed evaporator fan control on single systems in the field.<br>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.<br>Recommendations given on system integration (i.e., suction control), but not mandatory. Same issue with walk-ins.<br>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.<br>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.<br>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.<br>T20 doesn't address evaporative condensers, although they are used in California supermarkets. | No requirements in 2008 Title 24 standards.<br>Proposed 2011 updates:<br>Same issues as Title 20 regarding lack of test standards.<br>Halocarbon air cooled condensers for refrigerated warehouses are the same condensers as used for supermarkets included in Title 20.<br>Ammonia air-cooled condensers will be investigated and will include a specific efficiency requirement.<br>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.<br>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:<br>Evaporative condenser design TDs are |

| | | |
|---|---|---|
| | | specified for various design wetbulb temperatures.<br>Air cooled remote condenser TDs: 10°F for freezer systems, 15°F for cooler systems<br>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.<br>2011:<br>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:<br>• Continuous fan operation when compressor on with holdback low limit (smaller units).<br>• Or variable speed solely.<br>• Variable speed with holdback at minimum speed.<br>• 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.<br>2011 standards to require acceptance testing |

**Figure 62: 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), American National Standards Institute (ANSI) and 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 60, 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** 63**: 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 64. 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 64: 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 64, 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, minimum data requirements for published ratings, marking and nameplate data, and conformance conditions

for forced-circulation free-delivery unit coolers (evaporator coils) for refrigeration applications. 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 2.0°F. Test requirements for ARI Standard 420 are provided in Figure 65.

| 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 65: 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 66 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 66: description of compressor manufacturer's software packages**

Figure 67 through Figure 69 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.

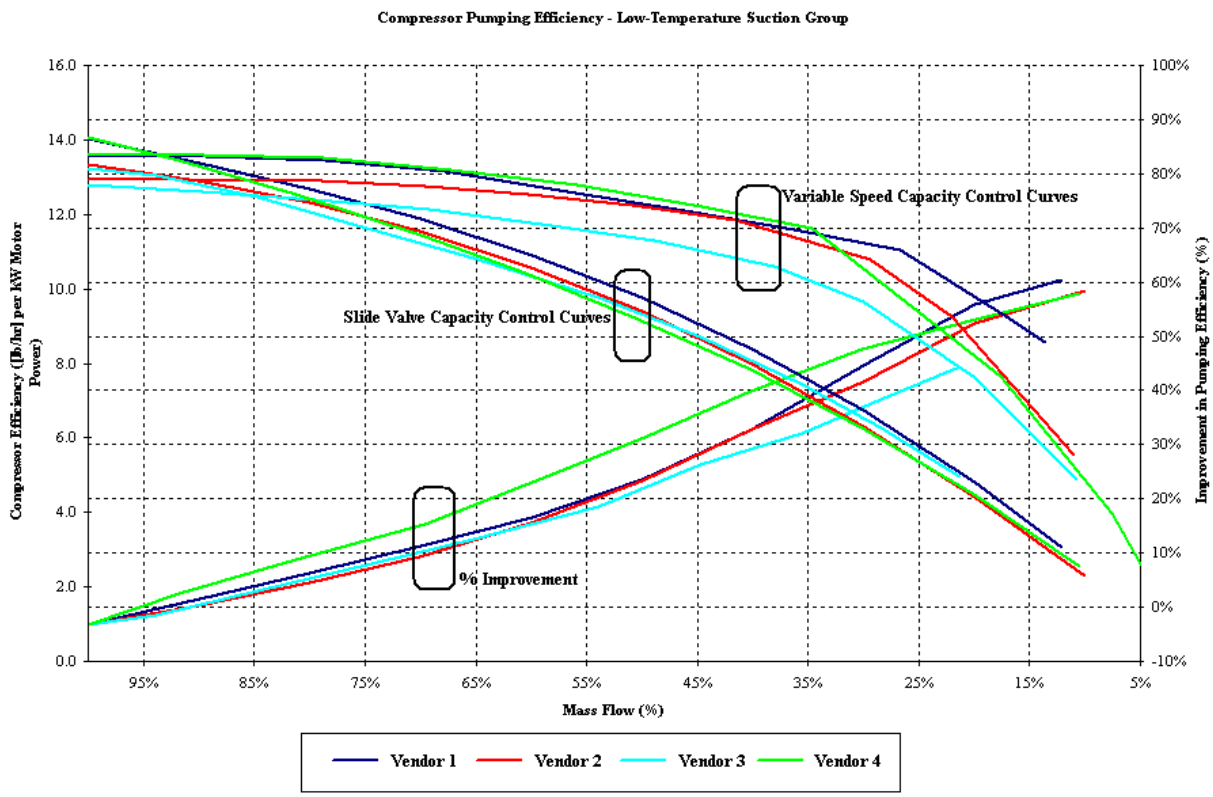



**Figure 67: Low-temperature suction group pumping efficiency**

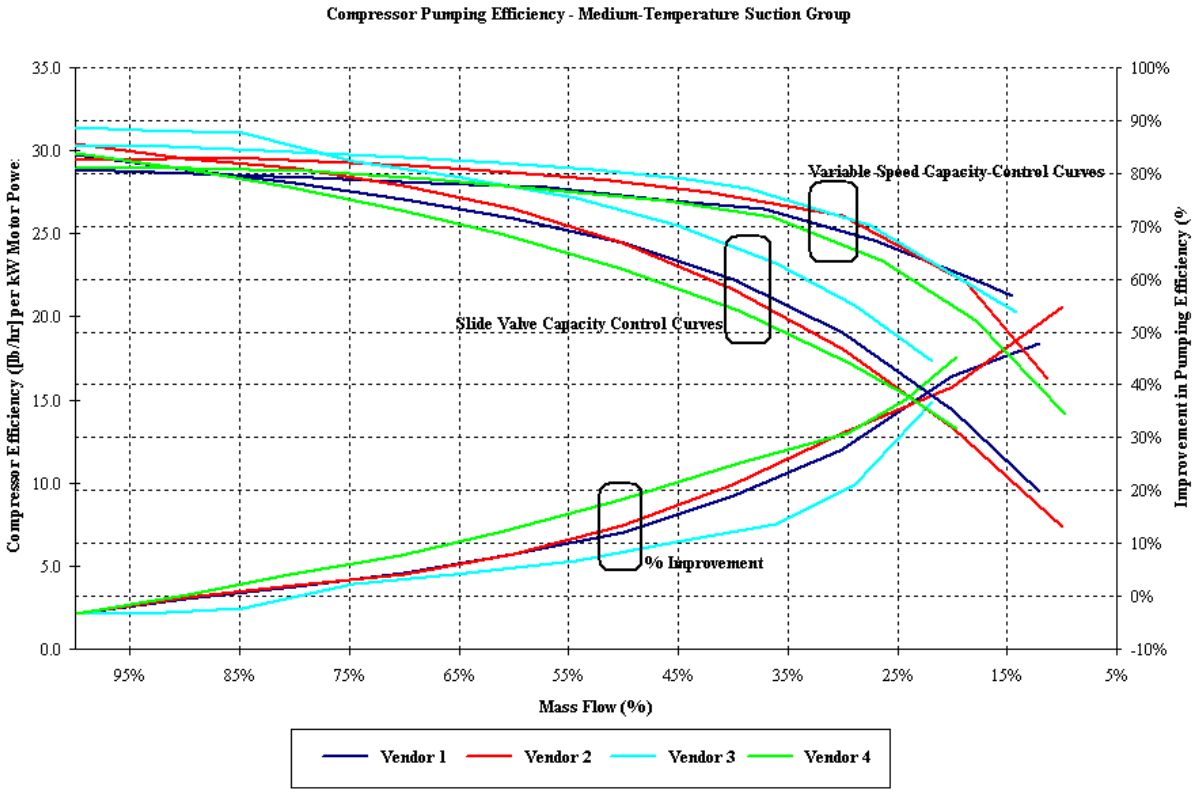



**Figure 68: Medium-temperature suction group pumping efficiency**

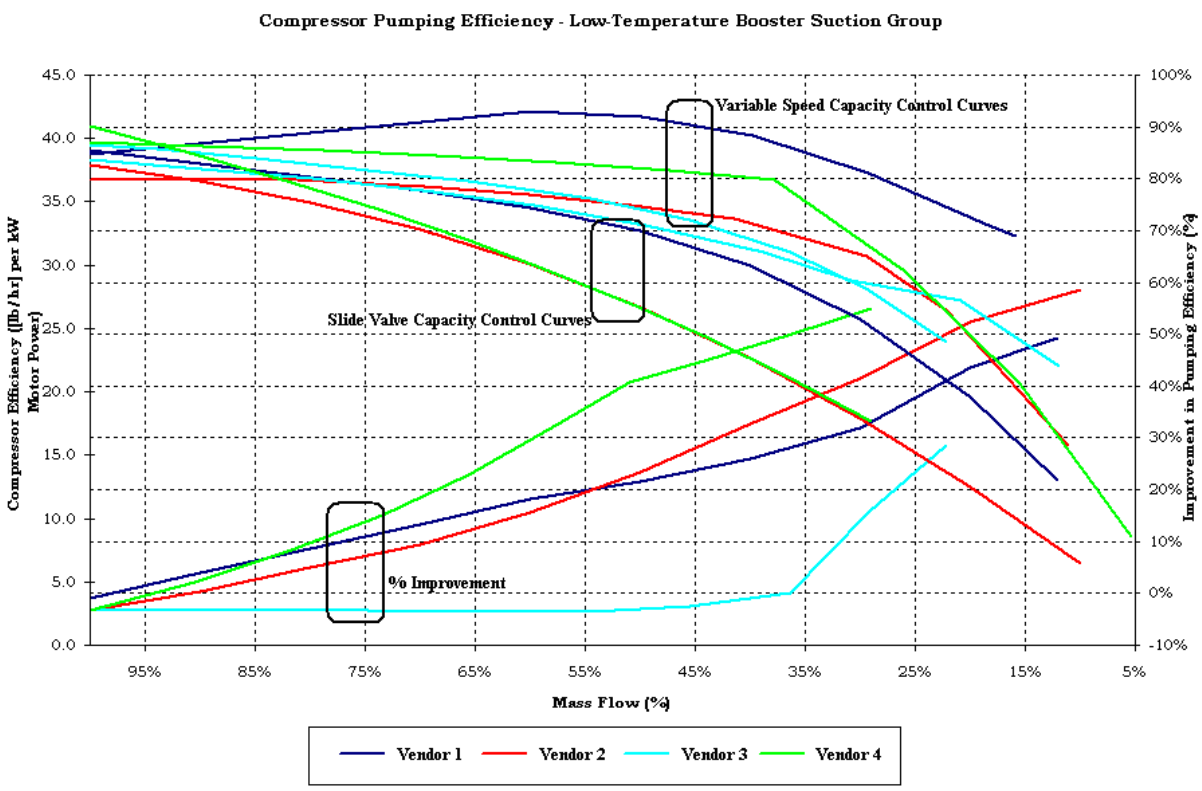



**Figure 69: Low-temperature booster suction group pumping efficiency**

### *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 70 through Figure 72 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 70: 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 71: 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 72: 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.

| Savings By Design 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 | Freeze r | -20 | R-38 | R-38 | R-32 |
| 2008 | SCE | Commerce | Freeze r | -50 | R-75 | R-75 | R-36 |

**Figure 73: Insulation R-values from participants in the Savings By Design 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 74 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 74: Utility rate assumptions for air-cooled ammonia system evaluation**

The simulated utility rate was based on PG&E E-20 rates. Figure 75 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 75: 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 76: 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:

- 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).
- 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.
- 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,
- conducting a literature review of multiple compressor controls for all types of compressors and refrigerants that may be used in refrigerated warehouses,
- developing an acceptance test for all types of compressors and refrigerants that may be used in a refrigerated warehouse,
- researching control system and instrumentation requirements and cost, and
- calculating 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.

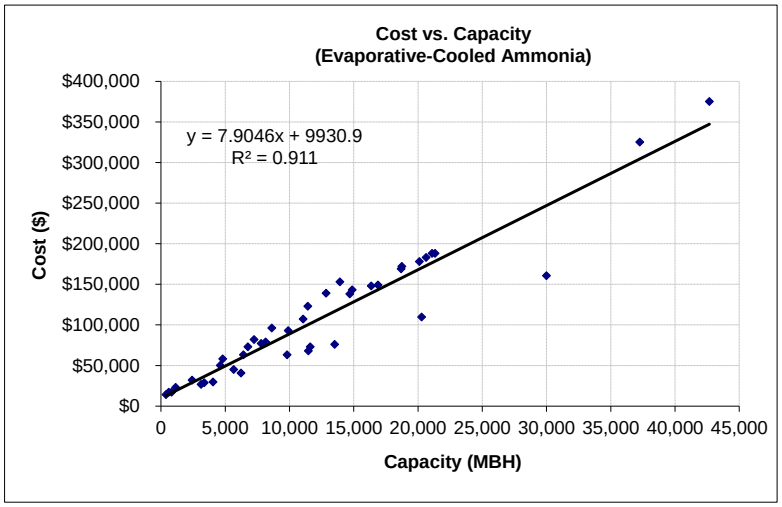
This measure was dropped from consideration. Technical feasibility is the main issue. Industrial refrigeration compressors are not tested and rated by a common standard and there is no certification of compressor ratings. Given the difficulty of measuring vapor flow, there is little research information 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.

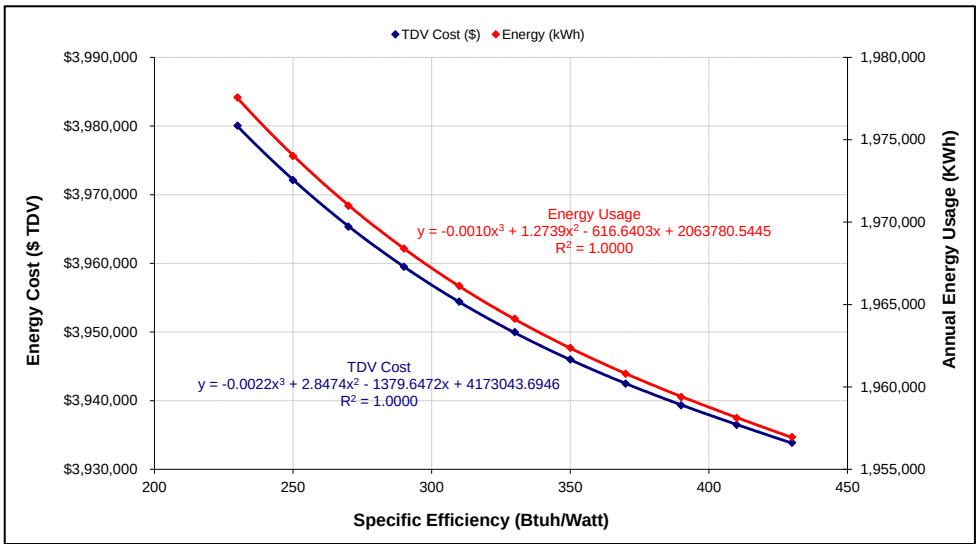
# 14. Appendix I: Full Condenser Specific Efficiency Analysis

This appendix includes figures for cost versus capacity, specific efficiency analysis, and energy analysis results versus specific efficiency for evaporatively-cooled condensers (ammonia and halocarbon) and air-cooled halocarbon condensers (with and without EC motors)

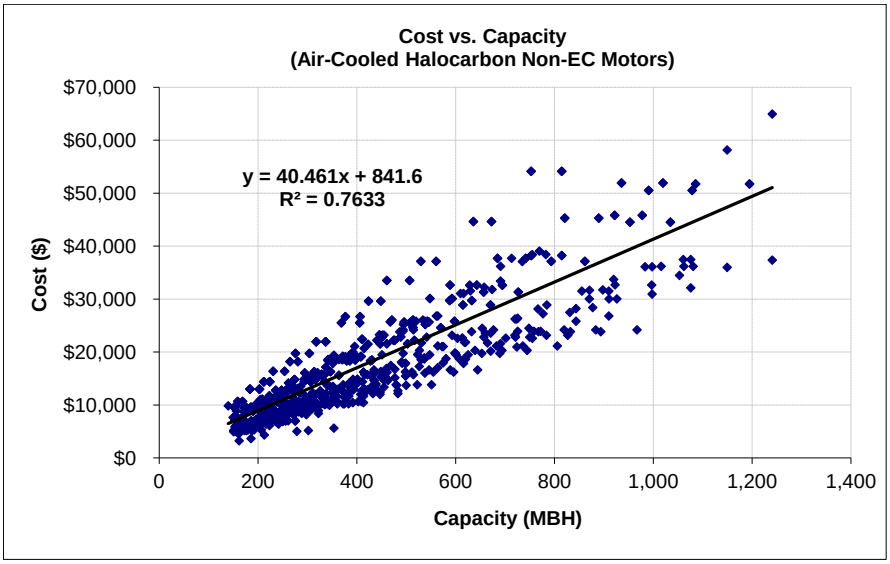
## *14.1 Evaporatively-Cooled Ammonia Condensers*



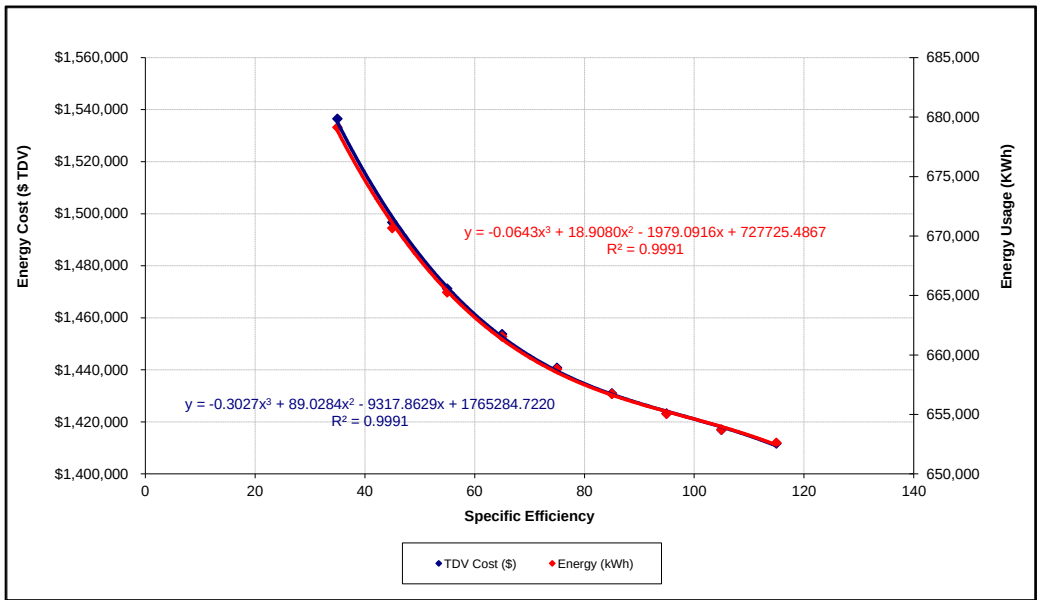

| % Increase in Condenser Size | Capacity of Larger Condenser at 100% Speed | Power of Larger Condenser at 100% Speed | Required % Full-Speed Capacity of Larger Condenser | Maximum Speed of New Condenser to Match Required Capacity* | New Power at Reduced Speed (kW) | New Specific Efficiency (Btuh/Watt) |
|---|---|---|---|---|---|---|
| 1% | 8,622 | 26.53 | 99.0% | 98.6% | 25.1303 | 339.7 |
| 2% | 8,708 | 26.79 | 98.0% | 97.1% | 24.1588 | 353.4 |
| 3% | 8,793 | 27.06 | 97.1% | 95.7% | 23.2416 | 367.3 |
| 4% | 8,878 | 27.32 | 96.2% | 94.4% | 22.3754 | 381.5 |
| 5% | 8,964 | 27.58 | 95.2% | 93.0% | 21.5572 | 396.0 |



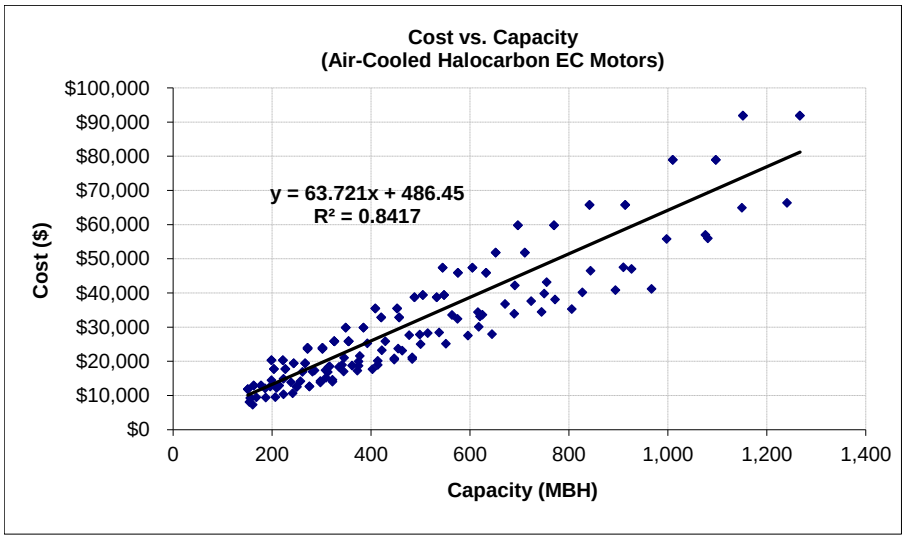
## 14.2 Air-Cooled Halocarbon Condensers without EC Motors



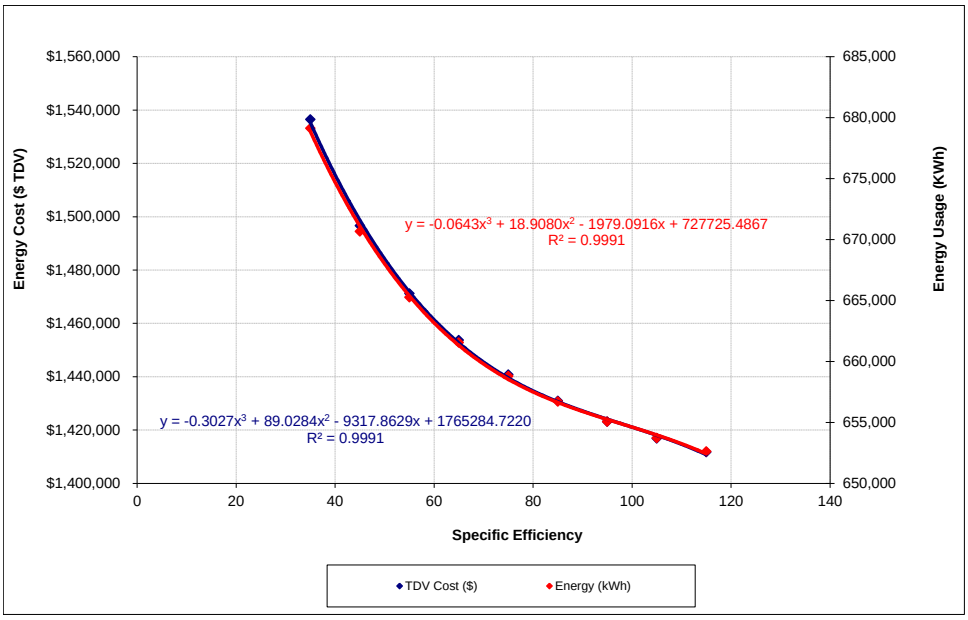

| % Increase in Condenser Size | Capacity of Larger Condenser at 100% Speed | Power of Larger Condenser at 100% Speed | Required % Full-Speed Capacity of Larger Condenser | Maximum Speed of New Condenser to Match Required Capacity* | New Power at Reduced Speed (kW) | New Specific Efficiency (Btuh/Watt) |
|---|---|---|---|---|---|---|
| 1% | 2,016 | 36.65 | 99.0% | 98.6% | 34.7195 | 57.5 |
| 2% | 2,036 | 37.02 | 98.0% | 97.1% | 33.3773 | 59.8 |
| 3% | 2,056 | 37.38 | 97.1% | 95.7% | 32.1101 | 62.2 |
| 4% | 2,076 | 37.74 | 96.2% | 94.4% | 30.9134 | 64.6 |
| 5% | 2,096 | 38.11 | 95.2% | 93.0% | 29.7830 | 67.0 |



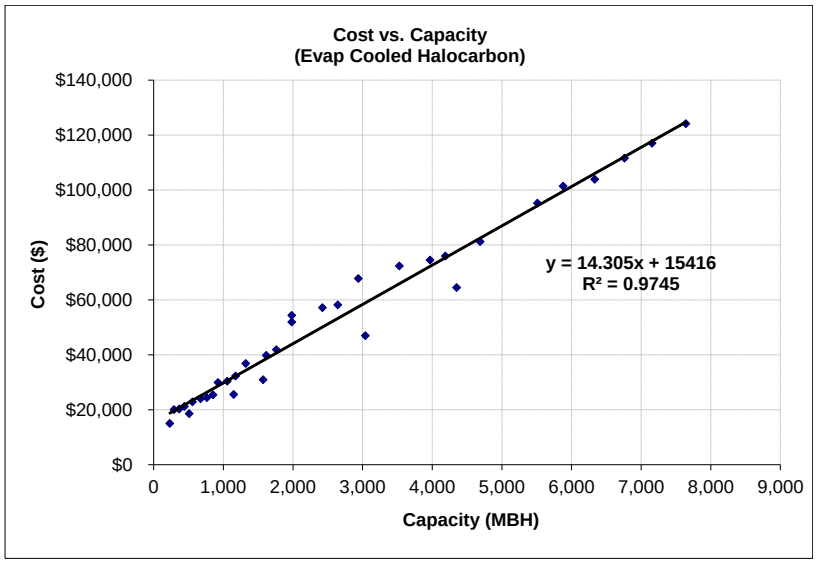
## 14.3 Air-Cooled Halocarbon Condensers with EC Motors



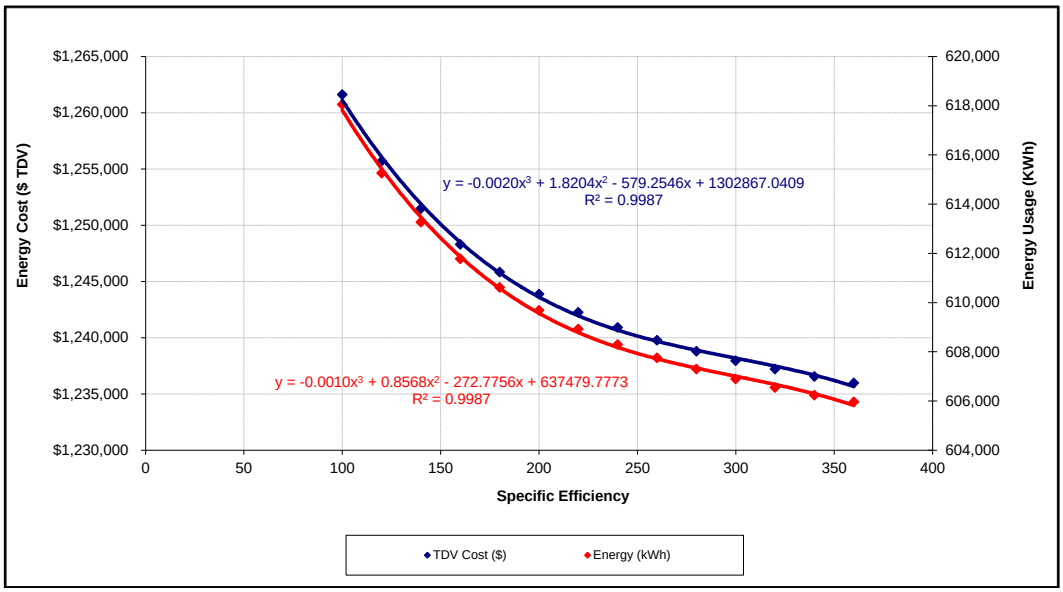

| % Increase in Condenser Size | Capacity of Larger Condenser at 100% Speed | Power of Larger Condenser at 100% Speed | Required % Full-Speed Capacity of Larger Condenser | Maximum Speed of New Condenser to Match Required Capacity* | New Power at Reduced Speed (kW) | New Specific Efficiency (Btuh/Watt) |
|---|---|---|---|---|---|---|
| 1% | 2,016 | 36.65 | 99.0% | 98.6% | 34.7195 | 57.5 |
| 2% | 2,036 | 37.02 | 98.0% | 97.1% | 33.3773 | 59.8 |
| 3% | 2,056 | 37.38 | 97.1% | 95.7% | 32.1101 | 62.2 |
| 4% | 2,076 | 37.74 | 96.2% | 94.4% | 30.9134 | 64.6 |
| 5% | 2,096 | 38.11 | 95.2% | 93.0% | 29.7830 | 67.0 |



## 14.4 Evaporatively-Cooled Halocarbon Condensers




| % Increase in Condenser Size | Capacity of Larger Condenser at 100% Speed | Power of Larger Condenser at 100% Speed | Required % Full-Speed Capacity of Larger Condenser | Maximum Speed of New Condenser to Match Required Capacity* | New Power at Reduced Speed (kW) | New Specific Efficiency (Btuh/Watt) |
|---|---|---|---|---|---|---|
| 1% | 2,943 | 21.02 | 99.0% | 98.6% | 19.9130 | 146.3 |
| 2% | 2,972 | 21.23 | 98.0% | 97.1% | 19.1432 | 152.2 |
| 3% | 3,001 | 21.44 | 97.1% | 95.7% | 18.4164 | 158.2 |
| 4% | 3,031 | 21.65 | 96.2% | 94.4% | 17.7301 | 164.4 |
| 5% | 3,060 | 21.86 | 95.2% | 93.0% | 17.0817 | 170.6 |



# 15. Appendix J: Assumptions For Environmental Impact

This appendix includes assumptions used to report material impact presented in section 2f.

**Insulation:**

Floor insulation will decrease 1" and roof insulation will increase 1" from 2008 to 2013 proposed code. Insulation densities are similar so projected change in material is 0.

**Evaporator fan speed control for single compressor systems:**

The deletion of this exemption applies to a very small number of evaporator fan motors. Assume VFD addition to 1 to 7.5 HP motor. VFD catalogs provide a wide range of sizes and weights so a medium weight of 4 lbs is selected for analysis. Assume shell is plastic, ½ volume is steel, ¼ volume is copper and ¼ volume is aluminum. Assume VFD life is 5 years.

**Air-cooled ammonia condensers:**

Given historical trends, it is unlikely that designers will choose an air-cooled ammonia system. However, if building is in areas of water restrictions, designer should be able to use ammonia instead of HFC for refrigerant. Because of lack of historical evidence that there will be any air-cooled ammonia systems, the material impact is projected to be 0.

**Condenser specific efficiency:**

Assumed the following base case and proposed case condenser models with properties from manufacturers' catalogs and 15 year life for condensers.

Air-cooled condensers

"Proposed" condensers

- Bohn BNX-D06-A039: spec. eff. = 64.8 Btuh/Watt, 51.55 lb/MBH, 2.8 lbR404a/MBH
- Bohn BNX-S03-A020: spec. eff. = 64.8 Btuh/Watt, 48.19 lb/MBH, 2.8 lbR404a/MBH

"Base Case" condensers

- Bohn BNL-D04-A032: spec. eff. = 52.5 Btuh/Watt, 45.4 lb/MBH, 3.1 lbR404a/MBH
- Bohn BNL-D08-A065: spec. eff. = 52.5 Btuh/Watt, 44.7 lb/MBH, 5.3 lbR404a/MBH
- Bohn BNL-S02-A016: spec. eff. = 52.5 Btuh/Watt, 43.1 lb/MBH, 3.1 lbR404a/MBH
- Bohn BNL-S04-A032: spec. eff. = 52.5 Btuh/Watt, 41.3 lb/MBH, 5.2 lbR404a/MBH

Evaporative-cooled condensers

"Proposed" condensers

- Evapco ATC-135E-1g: spec. eff. = 351 Btuh/Watt, 4.03 lb/MBH, 0.065 lbR717/MBH
- Evapco ATC-598E-1g: spec. eff. = 350 Btuh/Watt, 3.34 lb/MBH, 0.070 lbR717/MBH

"Base Case" condensers

- Evapco ATC-1861E-1g: spec. eff. = 325 Btuh/Watt, 3.38 lb/MBH, 0.072 lbR717/MBH
- Evapco ATC-442E-1g: spec. eff. = 325 Btuh/Watt, 3.10 lb/MBH, 0.062 lbR717/MBH

Change in refrigerant weight is estimated at 1.36 lbR404a/MBH decrease for air-cooled condenser and no change for evaporative-cooled condenser. Change in condenser weight is an increase of 6 lb/MBTUH for air-cooled condensers. Change in condenser weight is an increase of 0.44 lb/MBTUH for evaporative-cooled condensers. Assume 50 percent of installed condensers are air-cooled and 50 percent are evaporative-cooled. Assume all weight increase in evaporative-cooled condensers is from steel. For air-cooled condensers, assume 1/3 of weight

increase is steel in frame and motors, 1/3 is increase in aluminum in fins and 1/3 is copper in tubes and motors.

**Speed Control on Single Screw Compressors:**
Assume VFD addition to 125-500 HP motor. VFD catalogs provide a wide range of sizes and weights so a medium weight of 200 lbs is selected for analysis. Assume shell is plastic, ½ volume is steel, ¼ volume is copper and ¼ volume is aluminum. Assume VFD life is 5 years. Since this measure only applies to single screw compressors, it is not applicable to all systems.

**Infiltration Barriers:**
The infiltration barrier measure is included as a base case measure to define standard practice. Because it is standard practice, there is no change in material usage.