CODES AND STANDARDS ENHANCEMENT INITIATIVE (CASE)

2008 California Energy Commission Title 24 Building Energy Efficiency Standards
February, 2007

Final Report
Refrigerated Warehouses

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Overview

The standards have never addressed refrigerated warehouses or the processes around them; previous standards have focused on buildings that are heated and/or cooled for the purpose of human comfort. Refrigerated warehouses and the processes around them such as pre-coolers and food processing are extremely energy intensive and are fertile ground for additional energy savings and demand reductions.

HVAC systems for refrigerated warehouses are specialized equipment that is very different from equipment used to condition spaces intended for human occupancy. These differences will challenge the methods and procedures that we have used in the past to develop standards. Outside air ventilation is low or non-existent, refrigeration systems in large warehouses typically use ammonia rather than more conventional refrigerants, evaporators (essentially fan coils) are suspended or otherwise mounted in the cooler or freezer, and these are coupled to multiple compressors and condensers. Systems for large warehouses are typically custom designed, while small walk-in coolers may use packaged equipment.

Facilities can range from small walk-in coolers used in restaurants and grocery stores to very large food storage warehouses (250,000 ft^2 or more). Indoor design conditions can range from freezers to moderate temperature coolers. The actual freezer or cooler is the simpler and less energy intensive part of the operation. Pre-coolers are often a part of the operation and these are designed to rapidly cool the product before the product goes into the warehouse. Many refrigerated warehouses also are coupled with various types of food processing activities.

Refrigerated warehouses have long been the target of energy efficiency programs run by the IOUs. These programs have generally targeted shell and refrigeration equipment specifications. Shell requirements address wall and ceiling U-values, interior wall U-values, floor U-values for frozen food warehouses, and door U-values. Refrigeration systems requirements address condenser sizing, condenser fan and pump power, condenser fan controls, compressor motor efficiency, compressor capacity control, evaporator sizing, evaporator fan control, and evaporator fan motor efficiency. Refrigerant piping and storage vessels, when located outside, have maximum U-value requirements. Lighting generally defaults to Title 24 requirements for warehouse and/or C&I work area categories.

As part of this CASE Study, we carried out secondary research on refrigerated warehouse energy efficiency, conducted interviews with contractors and designers, and conducted detailed energy modeling and economic analysis on a series of potential measures that could be addressed within Title 24. Based on the results of these activities, we propose a set of changes to the Standards.

Description

The proposed changes to Title 24 affect the building shell insulation levels, evaporator fan controls, condenser fan power and control strategies, compressor plant controls and interior lighting levels for refrigerated warehouses. The equipment-related changes deal only with the storage part of the facility; standards for pre-coolers or other clearly process related equipment were not addressed.

Energy Benefits

The recommended energy conservation measures were tested against a common practice baseline established by Savings by Design. The energy benefits calculated in terms of kWh/ft^2-yr of refrigerated warehouse floor area are on the order of 0.3 kWh/SF for shell measures, 6.2 kWh/SF for evaporator fan controls, 1.4 kWh/SF for evaporative condensers and 4.1 kWh/SF for compressor controls, for a total of 12.0 kWh/SF.

Non-energy Benefits

Non-energy benefits associated with improved refrigerated warehouse energy efficiency include increased equipment reliability and stored product security. Strategies used to improve the efficiency of the refrigeration equipment reduce the operating pressures and temperatures, reducing stress on compressors, condensers and associated equipment. Improved U-value requirements for the insulated shell allow the warehouse to “coast” longer through power and equipment outages while keeping the stored product within an acceptable temperature range.

Research conducted in the Pacific Northwest for the Northwest Energy Efficiency Alliance indicated improved product quality and reduced mass loss in fruit stored in controlled atmosphere rooms with variable speed drive (VSD) controls on evaporator fans. VSDs applied to evaporator fans in freezers provided good temperature control while reducing wind-chill effects on warehouse employees.

Statewide Energy Impacts

A detailed analysis found that the first year’s implementation of the mandatory requirements for building shell, evaporator fan controls, evaporative condensers and compressor controls would reduce electricity energy consumption by 15.6 Gigawatt-hr per year, reduce electrical demand coincident with utility system peak by 1.8 Megawatts. There are no expected impacts on natural gas savings at the site. The discounted life cycle energy cost savings (3% discount rate, 15 year period) is $24.6 Million for one year’s new construction. After 10 years of this code measure the savings would be approximately tenfold or about $246 Million of present valued energy savings that accrue over the life of these buildings.

This estimate was based upon a unit energy savings estimate of 12 kWh/SF and expanded up to the population of one year’s new construction which is estimated to be 1.3 Million square feet per year for refrigerated warehouses. See the Results section of this report for a detailed description of how the statewide energy impacts were calculated.

Environmental Impact

Reductions in power plan emissions resulting from reductions in electricity consumption and demand is the principal environmental impact. There are no expected impacts on natural gas consumption, which is minimal at refrigerated warehouse sites. Leakage of ethylene glycol from underslab heating systems into groundwater is a potential environmental issue – however this proposal does not mandate the use of heat recovery for subslab heating nor the use ethylene glycol.

Type of Change

We are proposing a set of mandatory requirements for refrigerated warehouses. In order to implement a performance-based approach, the modeling software must be certified by the Energy Commission and must replicate energy performance as defined by the Title 24 reference method. The current Title 24 reference method (DOE-2.1E) is not suitable for refrigerated warehouse analysis. The program is limited to space temperatures greater than or equal to 0°F, limiting the ability of the program to evaluate the impacts of shell improvements in freezer facilities operated at temperatures below 0°F. The current reference method is also not capable of simulating industrial refrigeration systems used in refrigerated warehouses, due primarily to limitations in the supply air temperatures, which cannot be lower that 35°F. Given that DOE-2.1E will remain the reference software for 2008, it will not be possible to consider a performance-based approach for this round of the Standards. Thus, a set of mandatory requirements is proposed.

Technology Measures

Measures considered by this report included:

Insulation R-values:

- Freezer Ceiling, Exterior Wall, Floor
- Freezer to Cooler Wall
- Cooler Ceiling, Walls
- Dock Ceiling, Outdoor Wall, Floor

Refrigeration System Efficiency

- Minimum efficiency standards for compressor motors.
- Compressor minimum condensing temperature
- Limits on condenser fan and pump power
- Condenser sizing and approach

Refrigeration System Controls

- Floating head pressure
- Floating suction pressure
- Evaporator fan controls
- Condenser fan controls
- Compressor plant part-load controls

Measure Availability and Cost

The list of equipment manufacturers and engineering firms that design refrigerated warehouses in California is fairly small and well-known to the utilities, which have been active in this market for over 10 years. Engineering specifications from product literature were obtained and reviewed and interviews were conducted with engineering design firms and contractors to assess issues related to measure availability, costs, market capacity to supply equipment, product sources, and so on. A common-practice baseline established by the Savings by Design program for refrigerated warehouses and grocery store refrigeration systems was used as the baseline for this project.

Useful Life, Persistence and Maintenance

Envelope measures are expected to enjoy long life and savings persistence. Maintenance practices at large refrigerated warehouse facilities assessed during the interview process did not indicate any issues with measure life or maintenance. Given the size of these facilities and the risks to the stored product in the event of equipment failure, maintenance at these facilities is assumed to be fairly good. Contractors interviewed for the project cited potential issues with equipment resonant vibration when VSDs are installed on screw compressors, requiring testing during equipment startup and elimination of certain frequencies from the VSD operation. Leakage potential in glycol under-slab heating systems for freezer spaces was cited as a potential maintenance and environmental risk.

Performance Verification

Acceptance testing of refrigeration plant control systems and factory verification of evaporative condenser performance are performance verification options applicable to this effort. Development of detailed acceptance testing procedures is beyond the scope of the current work.

Cost Effectiveness

Virtually all measures evaluated by this project were shown to be cost effective. Shell measures were evaluated using on a life-cycle basis assuming a 30 year measure life, and mechanical measures were evaluated over a 15 year measure life. The Energy Commission time-dependent valuation (TDV) methodology[1] was used to estimate the value of the energy savings. (CEC 2005) The cost effectiveness of the variable speed evaporator fan measure and variable speed controls on ammonia screw compressors was extremely good, with benefit – cost ratios exceeding 10.

Analysis Tools

The energy savings were calculated using a DOE-2.2R building energy simulation program. DOE-2.2R is a variation on DOE-2.2 designed specifically for simulating refrigeration systems. DOE-2.2R can model spaces conditioned to low temperatures and provides the capability to simulate thermal distribution loops with fluids undergoing phase change, allowing for a detailed simulation of grocery store and refrigerated warehouse refrigeration systems. DOE2.2R is currently used to estimate savings for the refrigeration component of the statewide Savings by Design nonresidential new construction energy efficiency program operated by the California investor-owned utilities (IOUs).

Relationship to Other Measures

Issues relating to sizing and specific fan and pump power for cooling towers in commercial buildings are related to refrigerated warehouse condensers. Lighting issues in refrigerated warehouses are similar to those in non-refrigerated warehouses. Minimum efficiency requirements for motors in Title 24 will also apply to motors used in refrigerated warehouse equipment, such as compressors, condensers and evaporators. Title 20 addresses efficiency of walk-in coolers, which could potentially overlap with smaller refrigerated warehouse spaces. This initiative proposes language that clarifies the applicability of Title 20 standards for walk-in cooler and these proposed changes addressing refrigerated warehouses.

Methodology

To estimate the cost effectiveness of proposed changes addressing refrigerated warehouses, a series of DOE-2.2R prototype refrigerated warehouse models were developed for small and large warehouses. The small warehouse model was used to evaluate measures for reciprocating compressors and air-cooled condensers. The large refrigerated warehouse model was used to evaluate measures for screw compressors and evaporative condensers. Each model was run in climate zones 3 (representing a mild, coastal climate such as Salinas) and 13 (representing a hot, central valley climate such as Fresno). A description of the refrigerated warehouse prototypes used in this analysis is shown in Appendix A. Time dependent valuation multipliers were applied to the hourly outputs from the DOE-2.2R model to estimate the energy consumption and costs on a TDV basis.

Results

The measures evaluated in this report were generally cost effective on a TDV basis. Common practices for refrigerated warehouse design can be improved while remaining cost effective. However, given that refrigerated warehouses are not currently regulated, setting code minimum specifications that are more stringent then common practices may encounter resistance from the marketplace. Several of the contractors interviewed mentioned constructability or condensation control issues that may trump energy efficiency considerations. Several measures, such as interzone wall R-values and pipe and vessel R-values were removed from consideration based on these issues. A summary of the common practices as defined by the Savings by Design program, contractor interviews, and other sources are shown in Table 1 through Table 6.

[1] See Appendix D for a definition of time-dependent valuation

Table 1. Refrigerated Warehouse Shell Common Practices

Attribute	Savings by Design Baseline	Common practice from interviews	ASHRAE Recommendation
Freezer Ceiling	R-46	R-31 to R-50	R-45 to R-50
Freezer Exterior Wall	R-26	R-32 to R-56	R-35 to R-40
Freezer Floor R-value	R-30	R-18 to R-30	R-27 to R-32
Cooler Ceiling	R-23	R-24 to R-40	R-30 to R-35
Cooler Walls	R-20	R-23 to R-40	R-25
Dock Ceiling	R-23	Same as rest of facility	R-30 to R-35
Dock to outdoor wall	R-20	Same as rest of facility	R-25
Underfloor heating	No electric resistance	Heat recovery from condenser to underfloor glycol loop	None

Table 2. Evaporator Common Practices

Attribute	Savings by Design Baseline	Common practice from interviews
Evaporator fan speed control	Constant volume, constant operation	Constant volume, constant operation
Evaporator design approach temperature	10°F	Variable based on humidity requirements
Evaporator fan power (W/CFM)	Not addressed	No opinion

Table 3. Condenser Common Practices

Attribute	Savings by Design Baseline	Common practice from interviews
Condenser type	Not addressed	Evaporative condensers in ammonia facilities
Air-Cooled condenser fan speed control	Cycling one-speed fans	Cycling one-speed fans
Air-Cooled condenser design approach temperature	10°F to 15°F depending on suction temperature	10°F to 15°F depending on suction temperature
Air-Cooled condenser fan power	53 Btu/Watt-hr at 10°F approach temperature	No comment
Evaporative condenser fan speed control	Two speed fan	Two speed fan
Evaporative condenser design approach temperature	18°F to 25°F based on design wetbulb temperature	18°F to 20°F
Evaporative condenser fan and pump power	330 Btu/Watt-hr at 100°F saturated condensing temperature and 70°F wet bulb temperature	No comment

Table 4. Compressor Plant Common Practices

Attribute	Savings by Design Baseline	Common practice from interviews
Compressor capacity modulation	Not addressed	Slide valves on screw compressors; multiple compressor racks on reciprocating compressor plants
Compressor oil cooling	Not addressed	Not clear, new technology may be on the horizon

Table 5. Lighting Common Practice and Code Minimum Recommendations

Attribute	Savings by Design Baseline	Common practice from interviews
Lighting power density in warehouse spaces (W/SF)	0.6 W/SF	0.4 – 1.2 W/SF depending on application
Lighting controls	Not addressed	No control

Table 6. Refrigeration System Control Common Practice and Code Minimum Recommendations

Attribute	Savings by Design Baseline	Common practice from interviews
Suction pressure control	Not addressed	Fixed
Condensing temperature control	85°F minimum condensing temperature, fixed setpoint	Fixed
Defrost control	Not addressed	Time clock

Energy and Cost Savings

This section contains detailed energy and cost savings results that are summarized in the energy benefits section of the report. The results of the DOE-2.2R simulations of the prototypical building are presented in this section. Simulations were conducted in climate zone 3 (representing a mild coastal climate) and climate zone 13 (representing a warm, inland climate). Energy and cost savings are expressed per square foot of refrigerated warehouse floor space. TDV savings values were calculated by applying the 2005 hourly TDV multipliers by climate zone, and using updated 2008 net present value of energy costs per TDV unit. A life cycle of 30 years is used for shell measures, and a life cycle of 15 years is used for mechanical measures.[2]

Shell Measures

The shell measure analysis is based on results of simulations of the large warehouse prototype model, which is described in more detail in Appendix A. The energy and costs savings for shell insulation measures are shown in Table 7. Each value is expressed relative to the common practice baseline established by Savings by Design. The energy and cost savings are spread across the entire floorspace, thus savings from freezer, cooler and dock measures should be summed to obtain energy savings from shell measures at the whole facility level.

[2] Thirty year values for 2008 Standards development are $0.17592 per TDV kBtu; 15 year values are $0.09355 per TDV kBtu. See http://www.energy.ca.gov/title24/2008standards/documents/E3/TDVmethodology2008.doc for more information.

Table 7. Energy and Cost Savings for Shell Insulation Measures

Building Component	Insulation Level	Climate Zone 3		Climate Zone 13	
		Energy Savings kWh/SF	TDV Energy cost savings (PV $/SF)	Energy Savings kWh/SF	TDV Energy cost savings (PV $/SF)
Freezer Wall (R-26 base)	R-30	0.04	$0.13	0.07	$0.20
	R-35	0.08	$0.25	0.12	$0.36
	R-40	0.11	$0.33	0.12	$0.38
	R-45	0.13	$0.40	0.15	$0.46
	R-50	0.15	$0.46	0.19	$0.59
Cooler Wall (R-20 base)	R-25	0.01	$0.02	0.02	$0.05
	R-30	0.01	$0.04	0.03	$0.08
	R-35	0.02	$0.05	0.03	$0.11
	R-40	0.02	$0.06	0.02	$0.09
Dock Wall (R-20 base)	R-25	0.01	$0.04	0.01	$0.05
	R-30	0.02	$0.07	0.04	$0.13
	R-35	0.03	$0.09	0.07	$0.20
	R-40	0.03	$0.10	0.07	$0.21
Freezer Ceiling (R-46 base)	R-50	0.05	$0.15	0.06	$0.18
	R-55	0.10	$0.31	0.12	$0.38
	R-60	0.14	$0.45	0.17	$0.55
Cooler Ceiling (R-23 base)	R-25	0.01	$0.03	0.03	$0.09
	R-30	0.03	$0.09	0.06	$0.19
	R-35	0.04	$0.14	0.08	$0.27
	R-40	0.05	$0.17	0.10	$0.36
	R-45	0.05	$0.19	0.14	$0.47
Dock Ceiling (R-23 base)	R-25	0.00	$0.01	0.02	$0.06
	R-30	0.01	$0.03	0.03	$0.09
	R-35	0.02	$0.05	0.03	$0.10
	R-40	0.02	$0.06	0.04	$0.12
	R-45	0.02	$0.08	0.05	$0.16
Freezer Floor (R-30 base)	R-35	0.13	$0.38	0.13	$0.40
	R-40	0.22	$0.66	0.22	$0.67
	R-45	0.29	$0.89	0.32	$0.97
	R-50	0.35	$1.07	0.38	$1.16

A cool roof was applied to the base building to evaluate the effectiveness of this measure in what is already a highly insulated structure. The results of the analysis are shown in Table 8.

Table 8. Energy and Cost Savings for Cool Roofs

Building Component	Reflectance	Climate Zone 3		Climate Zone 13	
		Energy Savings kWh/SF	TDV Energy cost savings (PV $/SF)	Energy Savings kWh/SF	TDV Energy cost savings (PV $/SF)
Cool Roof (base reflectance of 0.20)	0.70	0.16	$0.30	0.30	$0.54

Evaporator Measures

Limits on specific fan power for evaporator fan coil units were investigated. A compilation of evaporator air flow rate and motor horsepower data from manufacturers' catalogs was used to characterize the current market conditions. The data compilation represents units manufactured by Frick division of York Refrigeration and the Lennox/Heatcraft companies (Bohn, Chandler Refrigeration, and Larkin). Data were compiled for a total of 3760 individual products serving the refrigerated warehouse market, and include evaporator fan coil units and remote air handlers. The distribution of motor size for all units compiled is shown in Figure 1.

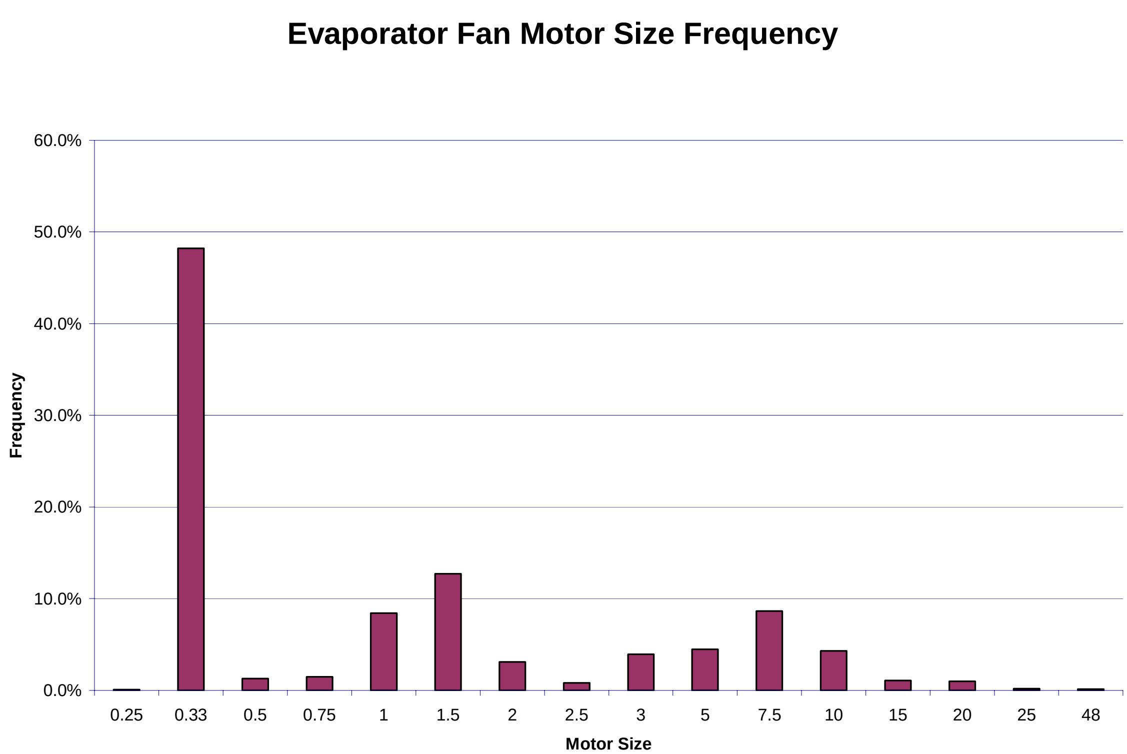

Figure 1. Evaporator Fan Motor Size Frequency

Note that 1/3 hp is the most common motor size, with 1 and 1.5 hp motors also used. Larger motors were used primarily in centrifugal fan evaporators, which are used primarily in ducted, remote evaporator systems. The distribution of the specific fan power for all units compiled is shown in Figure 2.

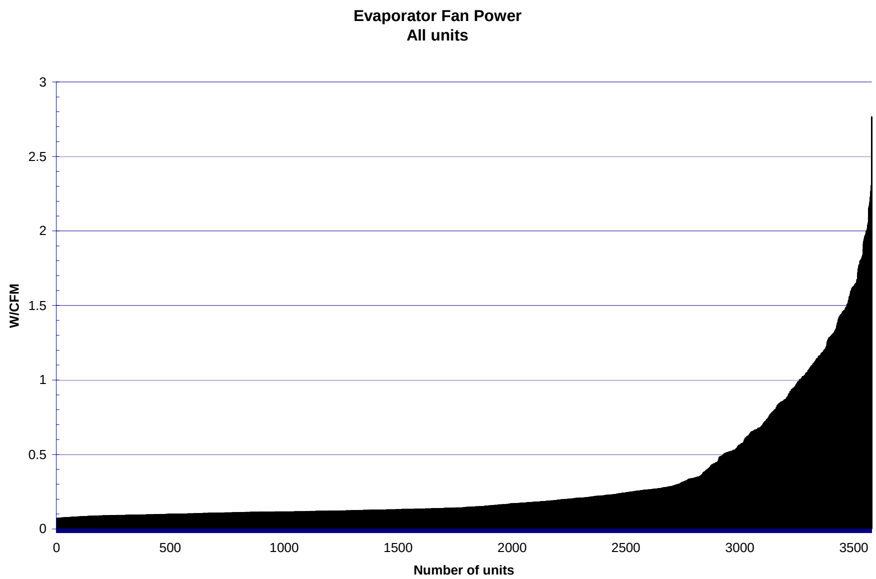

Figure 2 Evaporator Specific Fan Power

Note, the median value for specific fan power is 0.14 W/CFM. Many units exceed the limit of 0.8 W/CFM established in the Standards for constant volume air handlers for comfort air conditioning.. When the data are filtered to eliminate remote, centrifugal fan air handlers, showing only in-space fan coil units with axial fans, the distribution of specific fan power is shown in Figure 3.

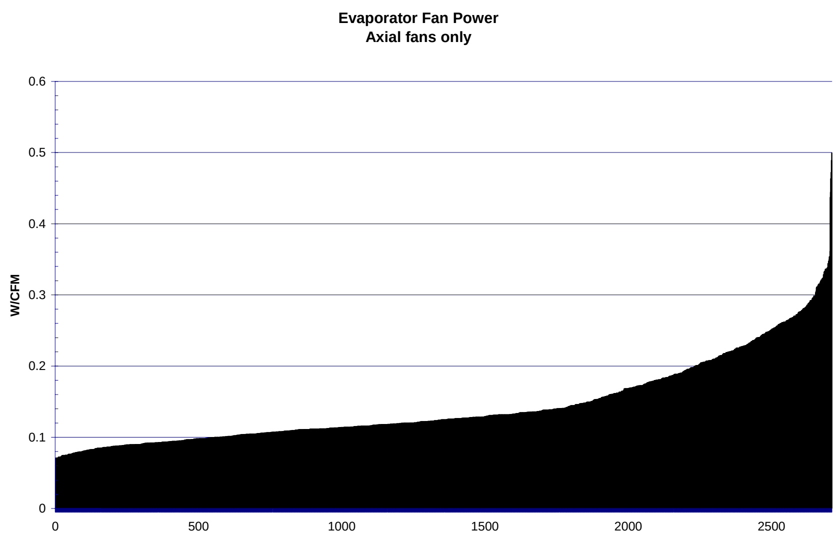

Figure 3 Specific Fan Power for Axial Fan Evaporators Only

The median value is 0.125 W/CFM. Note that the majority of the units consume 0.3 W/CFM or less, thus remote air handlers driven by centrifugal fans are the cause of the majority of the very high fan power units. The relationship between specific fan power and evaporator coil face velocity is shown in Figure 4.

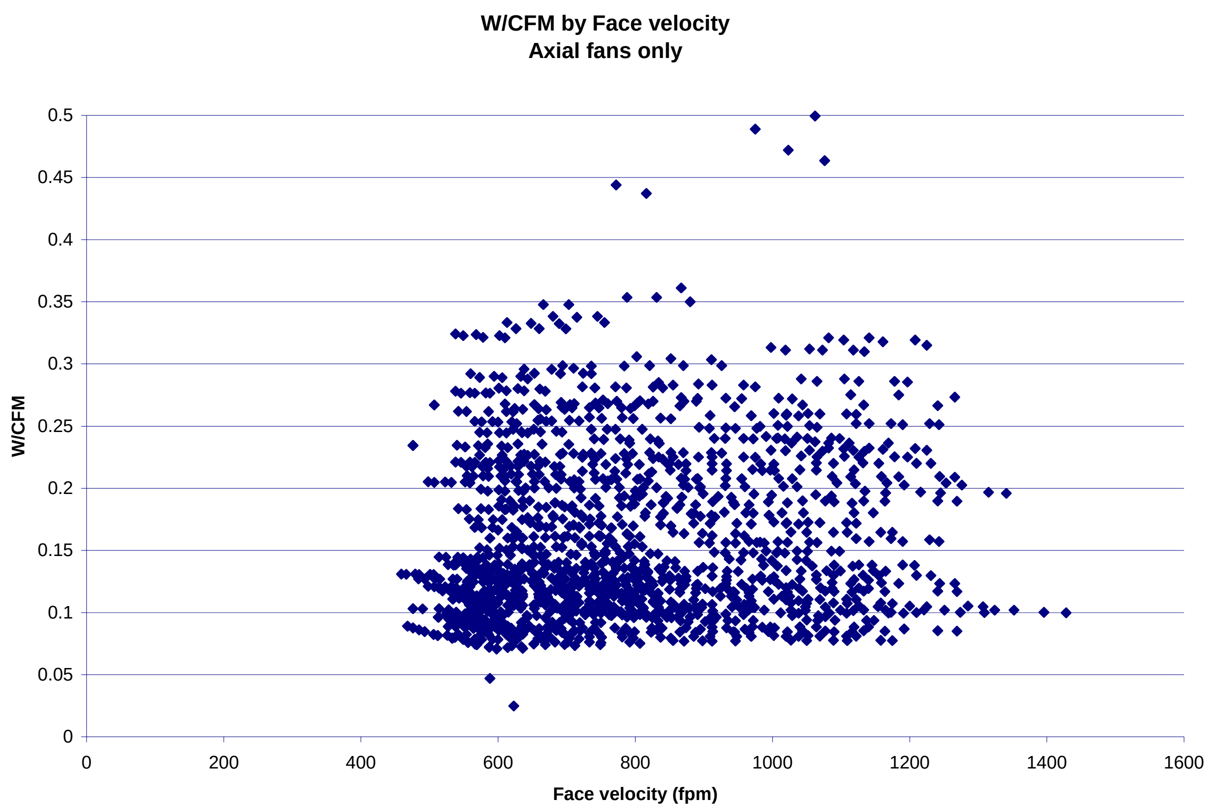

Figure 4 Evaporator Specific Fan Power as a Function of Face Velocity

These data show very little relationship between fan power and face velocity, indicating that the face velocity, which is related to the fan coil unit throw is not limited by the specific fan power of the unit. Similar analysis revealed no relationship between specific fan power and the number of rows per coil or coil fin spacing..

The Northwest Energy Efficiency Alliance (NEEA) has conducted a study on the application of evaporator fan variable speed drive for refrigerated warehouses. The project investigated the benefits of variable speed drives on evaporator fans in controlled-atmosphere fruit storage warehouses and in long-term storage warehouses. For controlled atmosphere warehouses, the study examined energy savings and the impacts of variable speed evaporator fan operation on stored product quality. The study involved detailed instrumentation of four refrigerated warehouse facilities in the Northwest. Activities included monitoring of evaporator fan electric power and room conditions affecting product quality such as temperature, CO_2 level and relative humidity in rooms with and without variable speed fan controls. Fruit quality was also assessed. The study results showed evaporator fan energy savings ranging from 24% to 78% with no impact on room temperature or CO_2 concentration distribution. Fruit quality improved in rooms with variable speed evaporator fans.[3] For storage warehouses, the study showed significant energy savings with no loss in stored product quality. Worker comfort was also increased due to reduced evaporator fan noise and wind chill.

[3] Morton, Robert D. and Mike McDevitt (2002), "Evaporator Fan VFD Effects on Energy and Fruit Quality," Cascade Energy Engineering, Walla Walla, WA. www.**cascadeenergy**.com

The energy and costs savings resulting from variable speed evaporator motors are shown in Table 9. The savings were evaluated relative to the baseline model specifications shown in Appendix A, which were based on the common practice baseline established by Savings by Design. The large warehouse prototype model was used in the analysis (see Appendix A for the model description). The impact of variable speed fans on evaporator motors was examined over a range of evaporator sizing conditions, from the minimum capacity needed to meet the load in the prototype model (sizing factor = 1.0) to twice the this capacity (sizing factor = 2.0) to meet unexpected internal or pull-down loads. As installed capacity increases, the energy and cost savings increase dramatically.

Table 9. Energy and Cost Savings for Variable Speed Evaporator Fan Measure

Evaporator Sizing	Climate Zone 3		Climate Zone 13	
	Energy Savings kWh/SF	TDV Energy cost savings (PV $/SF)	Energy Savings kWh/SF	TDV Energy cost savings (PV $/SF)
SIZING FACTOR = 1.0	2.98	$4.58	3.67	$5.69
SIZING FACTOR = 1.2	3.77	$5.91	4.48	$7.07
SIZING FACTOR = 1.4	4.34	$6.86	5.26	$8.30
SIZING FACTOR = 1.6	4.93	$7.79	6.06	$9.61
SIZING FACTOR = 1.8	5.53	$8.76	6.80	$10.77
SIZING FACTOR = 2.0	6.11	$9.68	7.60	$12.01

Evaporative Condenser Measures

The evaporative condenser analysis is based on results of simulations of the large refrigerated warehouse prototype model, with a multiplex ammonia screw compressor lineup for each suction group. The analysis considered the following measures:

- Floating head pressure with fixed setpoint. The condensing temperature is allowed to float down to a minimum of 70°F. The condensing temperature setpoint is fixed at 70°F for all conditions.

- Floating head pressure with variable setpoint. The condensing temperature setpoint is allowed to float down to a minimum of 70°F. The setpoint is calculated as a function of outdoor wetbulb temperature. The simulation assumed that the setpoint is equal to the ambient wetbulb temperature plus 9°F.

- Variable speed condenser fans. Once the condensing temperature setpoint is reached, the condenser fan speed is continuously varied to maintain the setpoint.

- Efficient, oversized condensers. The impact of efficient, oversized condensers was examined by reducing the design condensing temperature over a range from 4°F to 12°F, while limiting the specific fan and pump power to 400 Btu/watt-hr.

The savings were evaluated relative to the baseline model specifications shown in Appendix A, which were based on the common practice baseline established by Savings by Design. This base case is: an ammonia screw compressor with a 85°F condensing temperature, served by a condenser with a 23°F wet-bulb approach, condenser pump and fan power of 330 Btu/W-hr, and two speed fan controlled to the 85°F condensing temperature,. The energy and costs savings are shown in Table 10.

Table 10. Energy and Cost Savings for Evaporative Condenser Sizing and Control Strategies

Measure	Climate Zone 3		Climate Zone 13	
	Energy Savings (kWh / SF)	TDV Energy Cost Savings (PV $ / SF)	Energy Savings (kWh / SF)	TDV Energy Cost Savings (PV $ / SF)
Floating head pressure, fixed at 70°F	0.96	$1.45	1.61	$2.39
Floating head pressure, min of 70°F, 9°F wetbulb offset	0.97	$1.46	1.77	$2.71
VSD condenser fans	0.13	$0.20	0.12	$0.18
Oversized, efficient condenser (4°F reduction in wetbulb approach, 400 Btu/watt)	0.05	$0.08	0.07	$0.10
Oversized, efficient condenser (6°F reduction in wetbulb approach, 400 Btu/watt)	0.08	$0.12	0.11	$0.14
Oversized, efficient condenser (8°F reduction in wetbulb approach, 400 Btu/watt)	0.10	$0.14	0.13	$0.17
Oversized, efficient condenser (10°F reduction in wetbulb approach, 400 Btu/watt)	0.12	$0.18	0.17	$0.22
Oversized, efficient condenser (12°F reduction in wetbulb approach, 400 Btu/watt)	0.15	$0.22	0.20	$0.27

Selecting a compressor that safely operates down to a 70°F condensing temperature and resetting the condenser setpoint yields the greatest savings. Wetbulb offset controls have a small modest incremental savings in CTZ 3 but large savings in CTZ13, due to the increased number of hours in CTZ 13 with higher wetbulb temperatures. VSD condenser fans save a moderate amount of energy – the two speed fans in the base case save a substantial amount of energy relative to single speed fans. As the wetbulb approach is decreased, savings increase.

Air-Cooled Condenser Measures

The air cooled condenser analysis is based on results of simulations of the small refrigerated warehouse prototype model, with a multiplex halocarbon reciprocating compressor lineup for each suction group. The analysis considered the following measures:

- Floating head pressure with fixed setpoint. The condensing temperature is allowed to float down to a minimum of 70°F. The condensing temperature setpoint is fixed at 70°F for all conditions.

- Floating head pressure with variable setpoint. The condensing temperature setpoint is allowed to float down to a minimum of 70°F. The setpoint is calculated as a function of outdoor drybulb temperature. The simulation assumed that the setpoint is equal to the ambient drybulb temperature plus 5°F.

- Variable speed condenser fans. Once the condensing temperature setpoint is reached, the condenser fan speed is varied to maintain the setpoint.

- Efficient (reduced fan power) condenser. The impact of efficient condensers was examined by limiting the specific fan power over a range from 60 - 120 Btu/watt-hr, relative to the standard practice specific fan power of 53 Btu/watt-hr

A database of condenser fan energy normalized to condenser heat rejection capacity was developed from manufacturers' catalog data. These data are expressed as specific fan power in Btu/hr-watt at a 10°F temperature difference between saturated condensing temperature and outdoor drybulb temperature. Individual fan motor size generally varies from 0.5 hp to 1.5 hp, depending on fan diameter and RPM. The distribution of condenser specific fan power is shown in Figure 5.

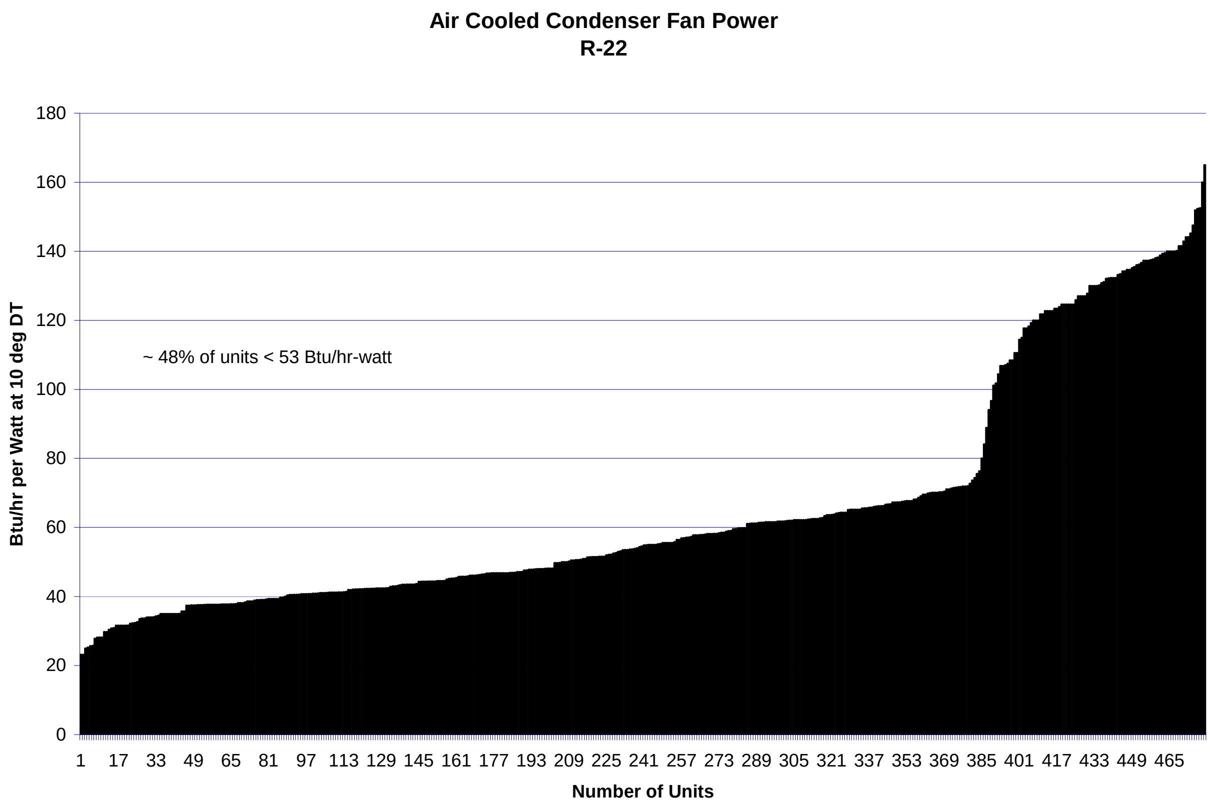

Figure 5 Air Cooled Condenser Specific Fan Power Distribution

Note, about half of the units on the market are less efficient than the common practice baseline (53 Btu/W-hr) used in the Savings by Design program. The specific fan power as a function of unit size is shown in Figure 6. Condensers serving larger loads tend to require more fan power per Btu/h of heat rejection. Thus the availability of equipment meeting the high efficiency specifications in Savings by Design (84 Btu/hr-watt) is limited in larger condenser sizes.

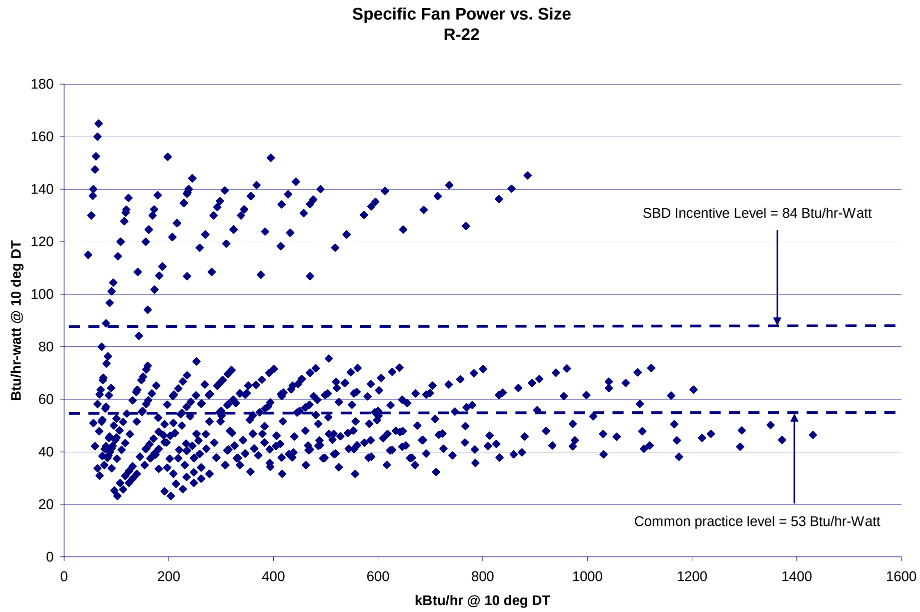

Figure 6 Air Cooled Condenser Specific Fan Power as a Function of Condenser Size

Savings associated with air cooled condenser fan power and control strategies were evaluated relative to the baseline model specifications shown in Appendix A, which were based on the common practice baseline established by Savings by Design. This baseline is a HFC refrigerant compressor with a condensing temperature of 85°F, 10°F drybulb approach for freezers and 15°F approach for coolers, and a condenser fan power of 53 Btu/W-hr. The energy and costs savings are shown in Table 11

Table 11. Energy and Cost Savings for Air-Cooled Condenser Fan Power and Control Strategies

Measure	Climate Zone 3		Climate Zone 13	
	Energy Savings (kWh / SF)	TDV Energy Cost Savings (PV $ / SF)	Energy Savings (kWh / SF)	TDV Energy Cost Savings (PV $ / SF)
Float head pressure with fixed setpoint	1.93	$2.87	< 0	< 0
Float head pressure with drybulb following	2.03	$3.04	2.06	$3.63
VSD condenser fans	1.48	$2.41	1.50	$2.32
Efficient condenser 60 Btu/hr-watt	0.15	$0.27	0.44	$0.87
Efficient condenser 80 Btu/hr-watt	0.43	$0.77	1.27	$2.52
Efficient condenser 100 Btu/hr-watt	0.59	$1.08	1.77	$3.51
Efficient condenser 120 Btu/hr-watt	0.71	$1.28	2.10	$4.18

Similar the results of the evaporative condensers the measure with the largest savings is to select a compressor that can easily handle a 70°F condensing temperature and reset the condenser fans for a 70°F condensing temperature setpoint. The drybulb following control optimizes the trade-offs between compressor power at lower condensing temperatures and the condenser fan energy needed to achieve a given condensing temperature – savings are substantially higher in CTZ 13 than 3.

Screw Compressor Capacity Control Measures

The energy and costs savings of a set of ammonia screw compressor capacity control strategies are shown in Table 12. The savings were evaluated relative to the large refrigerated warehouse baseline model specifications shown in Appendix A. The baseline model assumes a three compressor parallel-unequal compressor line for each suction group. The first run shows the energy and cost savings from applying a VSD to the smaller of the three compressors in each suction group. An additional run was done using a three compressor parallel-equal compressor line, and applying a VSD to one of the three compressors in each suction group. The energy savings resulting from applying a VSD to a parallel-equal system was much greater, due to the greater capacity of the VSD-controlled compressor and the poorer part-load performance of a parallel equal compressor line.

Table 12. Energy and Cost Savings from Screw Compressor Capacity Control Strategies

	Climate zone 3		Climate zone 13	
	Energy Savings (kWh / SF)	TDV Energy Cost Savings (PV $ / SF)	Energy Savings (kWh / SF)	TDV Energy Cost Savings (PV $ / SF)
VSD trim compressor (parallel unequal baseline)	0.02	$0.05	0.17	$0.25
VSD trim compressor (parallel equal baseline)	2.71	$4.16	4.10	$6.29

Cost-effectiveness

The cost effectiveness of the proposed measures are calculated from the estimated incremental cost associated with the measure installation and the net present value of the TDV energy savings calculated from the DOE-2.2R simulation model. The large refrigerated warehouse model with an evaporatively-cooled ammonia screw compressor refrigeration plant was used for the shell, evaporator, evaporative condenser, and screw compressor control measures. The small refrigerated warehouse model with an air cooled, halocarbon, parallel multiplex reciprocating compressor refrigeration plant was used to evaluate the air cooled condenser measures. The net present value of the TDV savings was evaluated assuming a 30 year measure life for shell insulation measures, and a 15 year measure life for cool roofs and mechanical system measures. Incremental maintenance costs are assumed to be zero based on interviews with contractors. The TDV savings are normalized to the same units as the measure cost data, to allow for a direct comparison of the savings versus the measure cost.

Shell Measures

The energy cost savings and total incremental measure costs for improved shell insulation is shown in Table 14. The analysis assumes that additional rigid insulation (at R-5 per inch) was applied to meet the specified insulation level. Incremental insulation material costs were estimated based on the 2005 R.S. Means “CostWorks” construction cost estimating CD. The fixed costs for installation and operations and maintenance are assumed to be independent of R-value. The total incremental costs assumed in the analysis are shown in Table 13.

Table 13. Insulation R-value and Cost Assumptions

Insulation System	R-Value per inch	Total incremental cost ($/SF-in)
Extruded Polystyrene (floor)	5.0	$0.32
Polyisocyanurate (roof)	7.1	$0.25
Polyurethane (wall)	5.0	$0.63

The TDV savings and incremental costs, expressed in terms of square foot of insulation applied relative to the baseline assumptions are shown, along with the benefit cost ratio (BCR). Measures with a BCR > 1 are deemed cost effective on a life cycle basis. In order to not constrain the analysis to R-values above the common practice baseline, the analysis was expanded to consider values below the common practice baseline. Note, the insulation parametrics were run using the large refrigerated warehouse model, assuming the refrigeration plant recommendations are also implemented. The cost effectiveness of the insulation requirements is very sensitive to the refrigeration plant efficiency, thus higher benefit cost ratios would result under a standard practice ammonia refrigeration plant design or in a facility with an air-cooled halocarbon refrigeration plant.

Table 14. Shell Insulation Measure Cost Effectiveness

Building Component	Insulation Level	Climate zone 3			Climate zone 13		
		TDV Savings/SF_{wall}	Incr Cost/ SF_{wall}	BCR	TDV Savings/ SFwall	Incr Cost/ SFwall	BCR
Freezer Wall (R-20 base)	R-25	$1.91	$0.63	3.0	$2.12	$0.63	3.4
	R-30	$3.19	$1.26	2.5	$4.08	$1.26	3.2
	R-35	**$4.10**	**$1.89**	**2.2**	**$5.30**	**$1.89**	**2.8**
	R-40	$4.76	$2.52	1.9	$5.50	$2.52	2.2
	R-45	$5.29	$3.15	1.7	$6.08	$3.15	1.9
	R-50	$5.72	$3.78	1.5	$7.13	$3.78	1.9
Cooler Wall (R-10 base)	R-15	$0.61	$0.63	1.0	$0.81	$0.63	1.3
	R-20	$0.91	$1.26	0.7	$1.66	$1.26	1.3
	R-25	$1.10	$1.89	0.6	$2.03	$1.89	1.1
	R-30	**$1.22**	**$2.52**	**0.5**	**$2.29**	**$2.52**	**0.9**
	R-35	$1.31	$3.15	0.4	$2.49	$3.15	0.8
	R-40	$1.37	$3.78	0.4	$2.36	$3.78	0.6
Dock Wall (R-10 base)	R-15	$0.80	$0.63	1.3	$1.34	$0.63	2.1
	R-20	$1.27	$1.26	1.0	$1.93	$1.26	1.5
	R-25	$1.53	$1.89	0.8	$2.23	$1.89	1.2
	R-30	**$1.72**	**$2.52**	**0.7**	**$2.80**	**$2.52**	**1.1**
	R-35	$1.86	$3.15	0.6	$3.27	$3.15	1.0
	R-40	$1.95	$3.78	0.5	$3.34	$3.78	0.9

Building Component	Insulation Level	Climate zone 3			Climate zone 13		
		TDV Savings/SF_{wall}	Incr Cost/ SF_{wall}	BCR	TDV Savings/ SFwall	Incr Cost/ SFwall	BCR
Freezer Ceiling (R-30 base)	**R-35**	**$0.93**	**$0.18**	**5.2**	**$1.21**	**$0.18**	**6.7**
	R-40	$1.65	$0.36	4.6	$1.87	$0.36	5.2
	R-45	$2.21	$0.54	4.1	$2.59	$0.54	4.8
	R-50	$2.66	$0.72	3.7	$3.11	$0.72	4.3
	R-55	$3.03	$0.90	3.4	$3.59	$0.90	4.0
	R-60	$3.34	$1.08	3.1	$3.97	$1.08	3.7
Cooler Ceiling (R-15 base)	R-20	$0.32	$0.18	1.8	$0.63	$0.18	3.5
	R-25	$0.53	$0.36	1.5	$1.02	$0.36	2.8
	R-30	**$0.67**	**$0.54**	**1.2**	**$1.26**	**$0.54**	**2.3**
	R-35	$0.77	$0.72	1.1	$1.44	$0.72	2.0
	R-40	$0.84	$0.90	0.9	$1.64	$0.90	1.8
	R-45	$0.91	$1.08	0.8	$1.90	$1.08	1.8
Dock Ceiling (R-15 base)	R-20	$0.47	$0.18	2.6	$0.26	$0.18	1.4
	R-25	$0.73	$0.36	2.0	$1.02	$0.36	2.8
	R-30	**$0.91**	**$0.54**	**1.7**	**$1.26**	**$0.54**	**2.3**
	R-35	$1.04	$0.72	1.4	$1.31	$0.72	1.8
	R-40	$1.14	$0.90	1.3	$1.47	$0.90	1.6
	R-45	$1.22	$1.08	1.1	$1.79	$1.08	1.7
Freezer Floor (R-20 base)	R-25	$1.83	$0.32	5.7	$1.91	$0.32	6.0
	R-30	$3.06	$0.64	4.8	$3.19	$0.64	5.0
	R-35	**$3.92**	**$0.96**	**4.1**	**$4.10**	**$0.96**	**4.3**
	R-40	$4.58	$1.28	3.6	$4.73	$1.28	3.7
	R-45	$5.10	$1.60	3.2	$5.42	$1.60	3.4

Cool roof cost effectiveness was evaluated based on the 15 year present value of the TDV savings and the incremental cost of cool roofs in low slope applications as reported in the 2005 Title 24 proceeding (PG&E, 2002). Incremental costs for cool roofs over conventional roofing products used in the 2005 Title 24 proceeding varied from no cost to $0.20 per square foot. Using $0.20 per square foot, the cost effectiveness for cool roofs is shown in Table 15.

Table 15. Cool Roof Cost Effectiveness

Building Component	Reflect	Climate zone 3			Climate zone 13		
		TDV Savings/ SF_{roof}	Incr Cost/ SF_{roof}	BCR	TDV Savings/ SF_{roof}	Incr Cost/ SF_{roof}	BCR
Cool Roof (base reflectance of 0.20)	0.70	$0.30	$0.20	1.5	$0.54	$0.20	2.7

Since the 2005 Title proceedings, several stakeholders have requested that this analysis be revisited with an incremental cost of $0.50/sf. With this conservatively high estimate of cost, cool roofs are cost-effective for all nonresidential buildings except those in climate zones 1, 3, 5 and 16. This analysis finds the same result for refrigerated warehouses, cool roofs are cost effective in CTZ 13 but not in CTZ 3.

Evaporator Measures

The energy cost savings and incremental measure costs for VSDs applied to evaporator fan motors are shown in Table 16. Costs for VSDs applied to evaporator fan motors were obtained from the Evaporator Fan VFD Market Transformation Initiative conducted by the Northwest Energy Efficiency Alliance.[4]

Table 16. Evaporator Fan VSD Cost Effectiveness Relative to Continuous Fan Operation

Evaporator sizing	Climate Zone 3			Climate Zone 13		
	TDV Savings/ hp	Incr Cost / hp	BCR	TDV Savings/ hp	Incr Cost / hp	BCR
SIZING FACTOR = 1.0	$10,775	$577	18.7	$10,972	$577	19.0
SIZING FACTOR = 1.2	$11,598	$577	20.1	$11,360	$577	19.7
SIZING FACTOR = 1.4	$11,526	$577	20.0	$11,436	$577	19.8
SIZING FACTOR = 1.6	$11,454	$577	19.9	$11,586	$577	20.1
SIZING FACTOR = 1.8	$11,447	$577	19.8	$11,540	$577	20.0
SIZING FACTOR = 2.0	$11,392	$577	19.7	$11,585	$577	20.1

The data in Table 16 are based on the baseline assumption that evaporator fans are operated continuously and do not cycle with the refrigeration load. A separate series of runs was done to evaluated the savings of VSD controlled evaporator fans relative to the case where the evaporator fans cycle on and off in response to the space refrigeration load. The results of these simulations are shown in Table 17.

Table 17. Savings from VSD Controlled Evaporator Fans Relative to Cycling Fan Baseline

Evaporator sizing	Energy Savings kWh/SF	TDV Savings/ hp	Incr Cost / hp	BCR
SIZING FACTOR = 1.0	3.5	$10,483	$577	18.2
SIZING FACTOR = 1.2	4	$10,283	$577	17.8
SIZING FACTOR = 1.4	4.3	$9,505	$577	16.5
SIZING FACTOR = 1.6	4.4	$8,745	$577	15.2
SIZING FACTOR = 1.8	4.4	$7,770	$577	13.5
SIZING FACTOR = 2.0	4.2	$6,801	$577	11.8

The runs shown in Table 17 were done in Climate zone 13 only, but similar results are expected in other climates. The energy savings are not a dramatic as the case with continuously operating fans, but the energy savings are still substantial with a benefit cost ratio exceeding 10 in all cases studied.

4 Evaporator Fan VFD Market Transformation Initiative Market Progress Evaluation Report No. 3. Prepared for the Northwest Energy Efficiency Alliances by Pacific Energy Associates and MetaResearch Group.

Condenser Measures

The energy cost savings and incremental measure costs for floating head pressure controls VSD condenser fan controls and close approach, efficient evaporative condensers are shown in Table 18. Costs for these measures were obtained from the 2005 DEER Measure Cost Study.[5] The DEER study evaluated close approach condensers at a 5°F reduction in the condensing approach temperature, resulting in a reduction in the approach temperature in Climate Zone 13 from 23°F to 18°F under design conditions. The cost from the DEER study ($88.26 per ton) was scaled up and down based on the range of approach temperature reductions modeled. A measure combination including floating head pressure with fixed setpoint and VSD fans was also analyzed.

Table 18. Evaporative Condenser Sizing and Control Cost Effectiveness

Measure	Climate Zone 3			Climate Zone 13		
	TDV Savings/ ton	Incr Cost/ ton	BCR	TDV Savings/ ton	Incr Cost/ ton	BCR
Floating head pressure, fixed at 70°F	$561	$20	27.88	$752	$20	37.34
Floating head pressure, min of 70°F, 9°F wetbulb offset	$566	$27	20.80	$852	$27	31.34
VSD condenser fans	$76	$100	0.76	$58	$100	0.58
Oversized, efficient condenser (4°F reduction in wetbulb approach, 400 Btu/watt)	$30	$67	0.45	$32	$67	0.48
Oversized, efficient condenser (6°F reduction in wetbulb approach, 400 Btu/watt)	$46	$112	0.41	$44	$112	0.39
Oversized, efficient condenser (8°F reduction in wetbulb approach, 400 Btu/watt)	$55	$168	0.33	$54	$168	0.32
Oversized, efficient condenser (10°F reduction in wetbulb approach, 400 Btu/watt)	$68	$240	0.28	$68	$240	0.28
Oversized, efficient condenser (12°F reduction in wetbulb approach, 400 Btu/watt)	$86	$335	0.26	$85	$335	0.25
Floating head pressure, fixed at 70°F, VSD condenser fans	$613	$119	5.14	$799	$119	6.70

As described earlier, dropping the condensing from 86°F to 70°F, results in the greatest savings and is extremely cost-effective, with a benefit cost ratio greater than 20 to 1. In climate zone 3, the addition of wetbulb following controls was not cost-effective (incremental savings of $5/ton, incremental cost of $7/ton) as compared to a base case with a 70°F condensing temperature and fixed temperature control for cycling the two speed fans. However, the wetbulb following control was very cost-effective in CTZ 13 (incremental savings of $100/ton, incremental cost of $7/ton) because there are more hours with higher wetbulb temperatures where it is not cost-effective to run the fans at full flow to try and achieve a 70°F condensing temperature. Variable speed controls are cost-effective only if combined with floating head pressure controls. Continuously variable speed controls (VSD) have a relatively low savings by themselves as they are compared with a base case of fans that are already variable speed (two speed

[5] 2005 DEER Measure Cost Study, conducted by Summit Blue Consulting.

fans), however, savings are maximized through a combination of floating head pressure and continuously variable speed fan controls.

The cost effectiveness of improved condenser efficiency on air-cooled condensers was investigated through a series of simulations. Air cooled condenser list prices were obtained from the manufacturers, and are shown along with the specific fan power data in Figure 7.

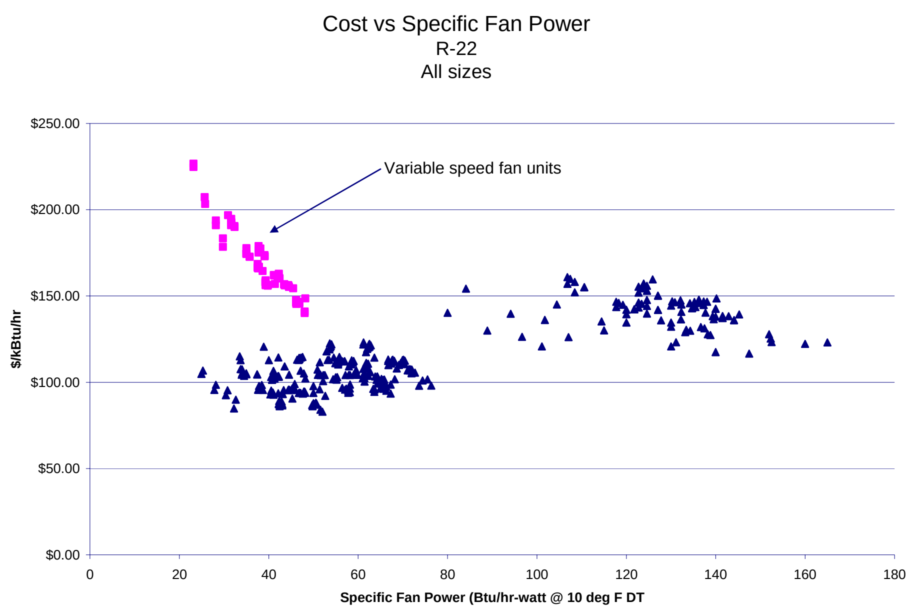

Figure 7 Air Cooled Condenser Costs as a function of Specific Fan Power

Note, the first cost of condenser increases slightly as the condenser efficiency improves, with the exception of a series of variable speed condensers from one manufacturer. For the cost effectiveness analysis, we eliminated the variable speed units from the database and filtered the units based on size to get a relationship between specific fan power and first cost in the size range appropriate for the small refrigerated warehouse prototype model. This relationship is shown in Figure 8.

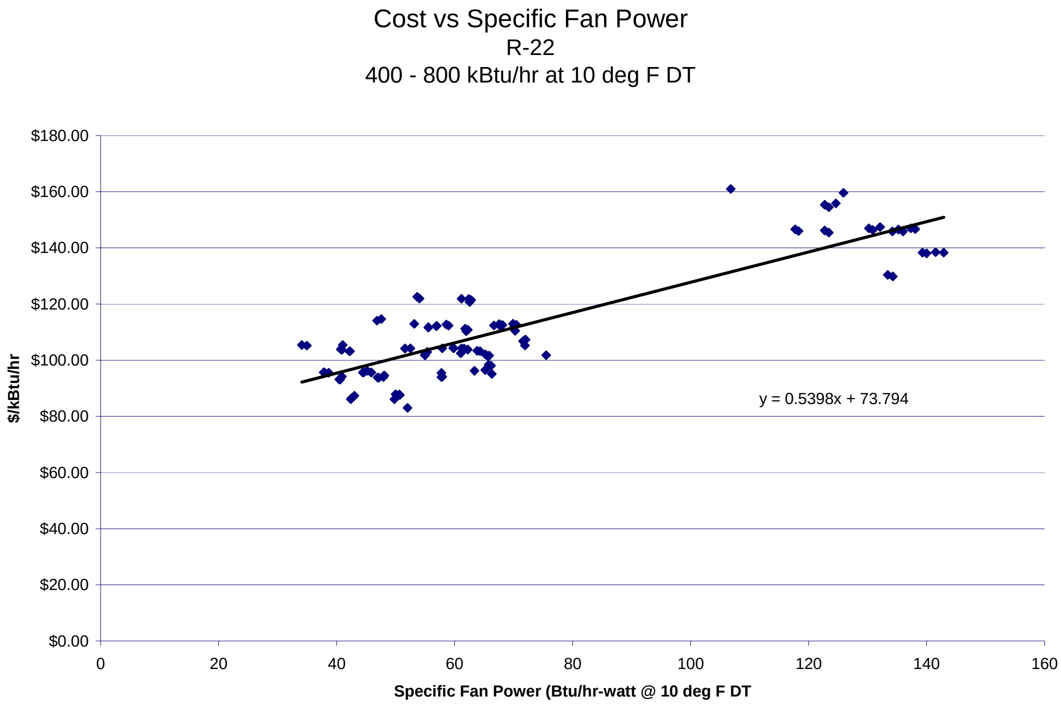

Figure 8 Air Cooled Condenser Cost Correlation

The energy cost savings and incremental measure costs for efficient air cooled condensers, floating head pressure controls and VSD condenser fan controls are shown in Table 19. The small warehouse prototype model with a parallel-equal reciprocating compressor line for each suction group was used to develop the savings estimates. A full description of the prototype simulation model used to develop the energy savings estimates is shown in Appendix A. Costs for these measures were obtained from the 2005 DEER Measure Cost Study[6] and the cost/performance relationship shown in Figure 8, normalized per compressor ton.

[6] 2005 DEER Measure Cost Study, conducted by Summit Blue Consulting.

Table 19. Air-Cooled Condenser Efficiency and Control Cost Effectiveness

Measure	Climate Zone 3			Climate Zone 13		
	TDV Savings/ ton	Incr Cost/ ton	BCR	TDV Savings/ ton	Incr Cost/ ton	BCR
Floating head pressure, fixed at 70°F	$1,578	$22	70.70	< 0	$20	< 0
Floating head pressure, min of 70°F, 5°F drybulb offset	$1,675	$37	45.09	$1,567	$27	42.17
VSD condenser fans	$1,325	$290	4.57	$1,003	$100	3.46
Efficient condenser with specific fan power = 60 Btu/hr-watt	$147	$45	3.24	$377	$67	8.32
Efficient condenser with specific fan power = 80 Btu/hr-watt	$425	$175	2.43	$1,090	$112	6.23
Efficient condenser with specific fan power = 100 Btu/hr-watt	$592	$304	1.94	$1,518	$168	4.99
Efficient condenser with specific fan power = 120 Btu/hr-watt	$703	$434	1.62	$1,804	$240	4.16
Floating head pressure, min of 70°F, 5°F drybulb offset, VSD condenser fans	$3,165	$328	9.66	$3,470	$335	10.59

Selecting compressors that can withstand condensing temperatures below 70°F and resetting air-cooled condenser fan controls to a 70°F condensing temperature setpoint is extremely cost-effective in climate zone 3. Floating head pressure controls combined with drybulb offset are cost-effective in all climate zones. Variable speed drives are cost-effective with or without floating head pressure controls, since the fan energy per unit of heat rejection is higher for air cooled condensers and the base case is a single speed fan.

Compressor Measures

The energy cost savings and incremental measure costs for compressor control measures are shown in Table 20. Incremental costs for variable speed compressors and floating suction pressure controls were obtained from the 2005 DEER Measure Cost Study.

Table 20. Compressor Control Cost Effectiveness

	Climate zone 3			Climate zone 13		
	TDV Savings/ ton	Incr Cost/ ton	BCR	TDV Savings/ ton	Incr Cost/ ton	BCR
VSD trim compressor (parallel unequal baseline)	$74	$171	0.43	286	$171	1.7
VSD trim compressor (parallel equal baseline)	$3,020	$171	17.6	$3,661	$171	21.4

With parallel equal compressor lineups, variable speed compressor controls are cost effective in climate zones 3 and 13. When applied to a parallel unequal compressor lineup, variable speed compressor controls are cost effective only in climate zone 13.

Statewide Energy Savings

A series of runs was done to look at the relative contribution of each measure to the overall energy savings. The runs started with applying the most effective measures first and moving to the least effective. The runs start with VSD evaporator fans, then add VSD trim compressor control, then add condenser measures (float head pressure to 70°F, VSD condenser fan control, 20°F approach temperature limit) and finally add the shell measures. The impact of these measure groups on the overall energy consumption of the large warehouse prototype building is shown in Figure 9.

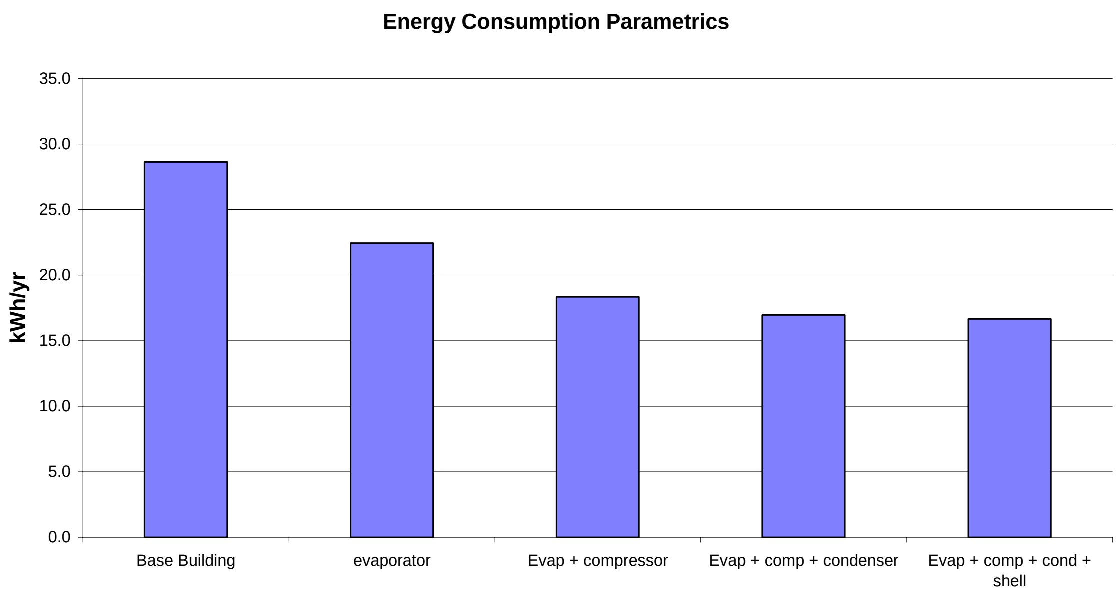

Figure 9. Energy Consumption Parametric Runs

Based in the order of implementation described above, the relative contribution of each measure group to the total energy savings is shown in Figure 10.

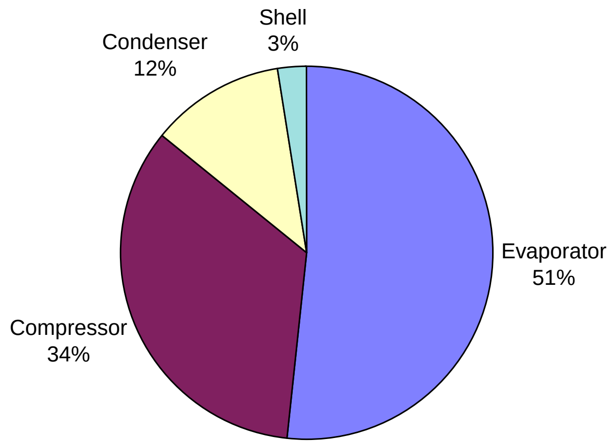

Figure 10. Relative Contribution of Each Measure to Overall Savings

The energy savings potential for the recommended measures total approximately 12 kWh/SF and 1.4 W/SF. Expanding this to the statewide estimate of refrigerated warehouse new construction estimate of 1.3 million square feet results in an overall statewide energy savings of 15.6 GWh and 1.8 MW per year.

Recommendations

Based on the interviews and analysis presented above, a set of recommendations for changes to Title 24 have been developed. The recommendations consider a range of factors, including the magnitude of the energy savings, measure cost-effectiveness, and practical issues associated with applicability of the measure over a wide range of refrigerated warehouse designs, availability of performance data or industry standards on rating and reporting equipment performance, and measure enforceability.

Applicability

The new requirements would apply to cold storage or frozen storage areas.

Exemptions:

Areas within refrigerated warehouses that are designed solely for the purpose of quick chilling or freezing of products. The scope is for cold or frozen storage as opposed to process cooling.

Walk-in coolers or freezers with a floor area less than 3,000 square feet which meets Title 20 requirements for walk-in cooler or freezers.

The purpose of these exceptions to make clear that all cold storage areas are either regulated by Title 20 for small walk-ins and all of the larger spaces are regulated by these new Title 24 requirements.

Minimum R-values for freezers and coolers

Title 20 has established minimum R-values for walls and ceilings of walk-in coolers and freezers. Manufactured interlocking panels used to construct walk-in coolers and freezers can also be used to construct refrigerated warehouses, so consistency between Title 20 and Title 24 is desired. The minimum R-values specified in Title 20 are well below the economic optimum R-values indicated in this study in most applications; however the energy savings from improved wall and ceiling insulation are small relative to the other measures investigated, so we recommend adopting the Title 20 minimum specifications to reduce confusion in the marketplace. We recommend that Title 24 addresses minimum floor insulation R-values for freezer spaces. The recommended values are as follows:

- R-36 Wall
- R-36 Ceiling
- R-36 Floor

Minimum R-values for coolers

- R-28 Wall
- R-28 Ceiling

The R-value of foam insulation products varies as a function of temperature. In order to reduce ambiguity on the reported R-values, the R-values should be evaluated at the mean temperature of the insulation, where the mean temperature is defined as the mean of the average annual outdoor temperature and the storage temperature. Cool roofs were found to be cost effective, and are recommended on roof surfaces common with refrigerated storage.

Limit on electric resistance underfloor heating

According to interviews with refrigerated warehouse designers, electric underslab heating is not very common in large warehouses, but may be used in small frozen storage spaces. To accommodate this application, we recommend that electric underslab heating be allowed if it is thermostatically controlled and is controlled off during summer on-peak periods

Evaporators

Limits on evaporator fan power are not recommended at this time, even though they are shown to be cost effective. The lack of manufacturer data and industry standards on how to measure and report evaporator fan motor power makes regulation difficult. Title 20 standards for walk-in coolers and freezers, while not limiting specific fan power, require EC (electronically commutated) motors on evaporator fan motors less than 1 hp. These motors are more efficient that standard motors. Since many of the products used in walk-in coolers can also be used in refrigerated warehouses, consistency between the standards is important. The majority of the savings are from the variable speed fans, which are easily implemented when EC motors are used.

We are also recommending that small evaporator and condensers fans less than 1 hp be electronically commutated. As shown in the Title 20 Refrigerated Walk-in CASE report (PG&E 2004), the benefit/cost ratio of EC motors in evaporators is greater than 2. Having this common requirement in both Title 24 and Title 20 provides a consistent signal to manufacturers is easier to comply with and to enforce and yields significant savings.

Variable speed control on evaporator fan motors was shown to be the most effective measure evaluated in this study, in terms of energy savings and cost effectiveness. Therefore, we recommend a mandatory requirement for variable speed controls on evaporator fan motors. This requirement is compatible with the Title 20 requirement for EC motors on walk-in evaporators. The measure is also shown to be cost effective using variable frequency drives to control motor speed in evaporators with fan motors greater than 1 hp. Thus for all motor sizes, the motor speed should be controlled based on space temperature, with a provision for a minimum speed setting that can be defined by the operators of the refrigerated warehouse. In discussions with stakeholders, it was identified that small stand alone split refrigeration circuits served by a single compressor having limited unloading capacity would be adversely affected by variable speed evaporator fans. For this reason, we recommend exempting these systems from the variable speed evaporator fan requirement, but would require EC motors if less than 1 hp.

Condensers

We recommend a mandatory requirement for evaporative condensers on ammonia based refrigeration systems. According to interviews with refrigerated warehouse designers, the application of evaporative condensers is standard practice in the industry, so this requirement should not pose any hardship on the industry. We do not recommend establishing minimum requirements for specific fan power for air-cooled and evaporative condensers at this time, due to the lack of standards and reliable industry data to calculate the specific fan power. The results of this study indicate that high-efficiency air cooled condensers are cost effective, but the lack of products on the market that meet the high efficiency specifications are a barrier at this time. However, the Title 20 Refrigerated Walk-in CASE report (PG&E 2004), reported a benefit/cost ratio for EC motors in condenser fan applications greater than 3. The current Title 20 requirements specify either permanent split capacitor (PSC) or EC motors on condenser fans. Having this common requirement in both Title 24 and Title 20 provides a consistent signal to manufacturers, is easier to comply with and to enforce, and yields significant savings.

Oversized condensers with specific fan power limits were not shown to be cost effective, and are not recommended. Variable speed fans when combined with floating head pressure controls were shown to be cost effective, and are recommended. These measures work well in combination, providing better condensing temperature control stability while maximizing condenser savings. The condenser recommendations are summarized as follows:

- Require evaporative condensers on all ammonia systems
- Limits on evaporative condenser wetbulb approach temperature of 20°F or less
- Limits on air-cooled condenser drybulb approach temperature of 15°F or less in coolers and 10°F or less in freezers
- Require floating head pressure control to a minimum of 70°F on all condensers, with drybulb offset controls on air-cooled condensers.
- Require variable speed control on condenser fan motors
- Require permanent split capacitor (PSC) or electronically commutated (EC) motors on single phase condenser fan motors less than 1 hp.

Compressors

Variable speed controls on screw compressors as a method to control compressor output was shown to be cost effective relative to conventional slide valve capacity controls in parallel-equal compressor lineups. We recommend a requirement for variable speed controls on at least one compressor per suction group on refrigeration plants with screw compressors, or a combination of slide valve controls and parallel-unequal compressor sizing strategies that can attain an equivalent part-load performance to a compressor line with one VSD compressor. Since floating head pressure was found to be cost effective, it is important that all compressor systems are supplied with the capability to operate at reduced condensing temperatures. Thus, we recommend a requirement that all compressors and accessories supplied by manufacturers are capable of operating at a minimum condensing temperature of 70°F.

Lighting

We recommend that the current requirements for non-refrigerated warehouses be applied to refrigerated warehouses as follows:

- Maximum lighting power density of 0.6 W/SF (this requirement is already in Section 146).
- Require bi-level lighting controls in storage spaces

Defrost

We are not proposing a requirement to eliminate electric resistance defrost at this time. According to designer interviews, hot gas defrost is common practice in large refrigerated warehouse spaces, but is not practical for split systems with a single, dedicated condensing unit for each evaporator.

Proposed Standards Language

The Standards currently do not address refrigerated warehouses, so it is recommended that the mandatory measures be assigned to an unused section of the Standards. The refrigerated warehouse provisions would be included in a completely new Section 120 at the end of Subchapter 2: ALL OCCUPANCIES—MANDATORY REQUIREMENTS FOR THE MANUFACTURE, CONSTRUCTION AND INSTALLATION OF SYSTEMS, EQUIPMENT AND BUILDING COMPONENTS. Current sections 120 through 125 would have to be renumbered to accommodate the new section 120. Placing the requirements in Section 126 would be a problem as this falls under subchapter 3 which contains mandatory requirements for space-conditioning and service water-heating systems only and this proposal includes insulation requirements. The following language is recommended:

SECTION 100 – SCOPE

TABLE 100-A APPLICATION OF STANDARDS

Occupancies	Application	Mandatory	Prescriptive	Performance	Additions/Alterations
General Provisions		100, 101, 102, 110, 111			
Nonresidential, High-Rise Residential, And Hotels/Motels	General	140	142	141	149
	Envelope (conditioned)	116, 117, 118	143		
	Envelope (unconditioned, process spaces)		143 (c)		
	HVAC (conditioned)	112, 115, ~~120-125~~ 121-126	144		
	Water Heating (conditioned)	113, ~~123~~ 124	145		
	Indoor Lighting (conditioned, process spaces)	119, 130, 131	143 (c), 146		
	Indoor Lighting (unconditioned)	119, 130, 131	143 (c), 146	N.A.	
	Outdoor Lighting	119, 130, 132	147		
Refrigerated Warehouse	Envelope and HVAC	120			
Signs	Indoor and Outdoor	130, 132	148		
Low-Rise Residential	General	150	151 (a, f)	151 (a-e)	152
	Envelope (conditioned)	116, 117, 118, 150 (a-g, l)			
	HVAC (conditioned)	112, 115, 150 (h, i, m)			
	Water heating (conditioned)	113, 150 (j)			
	Indoor Lighting (conditioned and parking garages)	119(d), 150 (k)			
	Outdoor Lighting	119(d), 150 (k)			

SECTION 101 – DEFINITIONS AND RULES OF CONSTRUCTION

Cold storage is an area where space temperatures are maintained between 32°F and 55°F.

Frozen storage is an area where space temperatures are maintained below 32°F.

PROCESS is an activity or treatment that is not related to the space conditioning, lighting, service water heating, or ventilating of a building as it relates to human occupancy, cold storage or frozen storage.

PROCESS SPACE is a space that is thermostatically controlled to maintain a process environment temperature less than 55º F or to maintain a process environment temperature greater than 90º F for the whole space that the system serves, or that is a space with a space-conditioning system designed and controlled to be incapable of operating at temperatures above 55º F or incapable of operating at temperatures below 90º F at design conditions.

PROCESS LOAD is a load resulting from a process.

Refrigerated warehouse is a building constructed for storage of products, where mechanical refrigeration is used to maintain the space temperature at 55°F or less.

SECTION 110 – SYSTEMS AND EQUIPMENT—GENERAL

Sections 111 through ~~119~~ 120 establish requirements for the manufacture, construction, and installation of certain systems, equipment and building components that are installed in buildings regulated by Title 24, Part 6. Systems, equipment and building components listed below may be installed only if:

(a) The manufacturer has certified that the system, equipment or building component complies with the applicable manufacture provisions of Sections 111 through ~~119~~ 120; and

(b) The system, equipment or building component complies with the applicable installation provisions of Sections 111 through ~~119~~ 120.

No system, equipment or building component covered by the provisions of Sections 111 through ~~119~~ 120 that is not certified or that fails to comply with the applicable installation requirements may be installed in a building regulated by Title 24, Part 6.

The systems, equipment and building components covered are:

Appliances regulated by the Appliance Efficiency Regulations (Section 111).

Other space-conditioning equipment (Section 112).

Other service water-heating systems and equipment (Section 113).

Pool and spa heating systems and equipment (Section 114).

Gas appliances (Section 115).

Doors, windows, and fenestration products (Section 116).

Joints and other openings (Section 117).

Insulation and Cool Roofs (Section 118).

Lighting control devices (Section 119).

Refrigerated warehouses (Section 120).

All of the language is new for a new Section 120 that is at the end of SUBCHAPTER 2 ALL OCCUPANCIES—MANDATORY REQUIREMENTS FOR THE MANUFACTURE, CONSTRUCTION AND INSTALLATION OF SYSTEMS, EQUIPMENT AND BUILDING COMPONENTS. Sections 120 through 125 in the 2005 standards are renumbered to account for this new section. Language though new is not underlined for clarity sake.

SECTION 120 – MANDATORY REQUIREMENTS FOR REFRIGERATED WAREHOUSES

Refrigerated warehouses containing either cold storage or frozen storage areas shall meet the requirements of this section.

(a) Insulation Requirements. Exterior surfaces of refrigerated warehouses shall be insulated at least to the R-values in Table 120-A.

Table 120-A

Space	Surface	Minimum R-value (°F·hr·sf/Btu)
Frozen Storage	Roof	R-36
	Wall	R-36
	Floor	R-36
Cold Storage	Roof	R-28
	Wall	R-28

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

EXCEPTION to Section 120 (b): Underslab heating systems controlled such that the electric resistance heat is thermostatically controlled and disabled during the summer on-peak period, as 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 shall be electronically commutated motors
2. Evaporator fans shall be variable speed and speed shall be controlled in response to space conditions.

EXCEPTION to Section 120 (c)2:. Areas within facility served by one or more split refrigeration systems with a single condensing unit serving a single evaporator.

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

1. Condensers for systems utilizing ammonia shall be evaporatively cooled
2. Condensing temperatures for evaporative condensers under design conditions shall be less than or equal to the design wetbulb temperature plus 20°F.
3. 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 and shall be less than or equal to the design drybulb temperature plus 15°F for systems serving cold storage.
4. Condenser fans for evaporative condensers shall be continuously variable speed.

5. Condenser fans for air-cooled condensers shall be continuously variable speed and controlled in response to ambient drybulb temperature or refrigeration system load.

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

(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 screw compressors shall be controllable in response to the refrigeration load on at least one compressor per suction group, or the input power to all compressors on a suction group shall be controlled to be less than or equal to 40% of full load input power when operated at 50% of full refrigeration capacity.

EXCEPTION 1 to Section 120: Areas within refrigerated warehouses that are designed solely for the purpose of quick chilling or freezing of products.

EXCEPTION 2 to Section 120: Walk-in refrigerators and walk-in coolers with a floor area less than 3,000 sf that meet all of the requirements of California Appliance Efficiency Regulations[7]

Alternate Calculation Manual

The proposed changes are assigned as mandatory requirement, so most of the ACM will not be affected. Language relating to the scope of Title 24 will need to be revised to include refrigerated warehouses.

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[7] California Code of Regulations, Title 20, Sections 1601 through 1608 dated December 2006

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Appendix A. Prototype Refrigerated Warehouse DOE-2.2 Model Descriptions

The characteristics of the large and small refrigerated warehouse prototype models are shown below:

Table 21. Prototypical Large Refrigerated Warehouse Model Description

Model Parameter	**Value**
Shape	Rectangular (400 ft by 230 ft)
Floor area	Freezer: 40,000 SF Cooler: 40,000 SF Shipping Dock: 12,000 SF Total: 92,000 SF
Number of floors	1
Floor to ceiling height	30 ft
Exterior wall construction	Insulated metal panel
Ext wall R-Value	Cooler and loading dock – R-20 Freezer – R-26
Infiltration rate	Cooler and Freezer: 0.1 ACH Loading Dock: 0.3 ACH
Roof construction	Insulated low mass roof
Roof R-values	Cooler and loading dock – R-23 Freezer – R-46
Roof absorptivity	0.80
Lighting power density	0.6 W/SF
Equipment power density	0.7 W/SF (covers fork lifts and miscellaneous plug loads and equipment)
Operating schedule	24 / 7
No. People	184 max
Evaporator type	Constant volume, continuous fan operation
Evaporator Size (climate zone 13)	Cooler: 102 ton (392 SF/ton) Freezer: 136 ton (295 SF/ton) Dock: 55 ton (218 SF/ton)
Evaporator CFM (climate zone 13)	Cooler: 172,000 cfm (4.3 cfm/SF) Freezer: 131,400 cfm (4.79 cfm/SF) Dock: 55,300 cfm (7.9 cfm/SF)
Compressor type	Ammonia screw compressor with slide valve capacity control (Frick RWF –100 typical)
Compressor configuration	Parallel equal, 3 compressors per suction group, size ratio 0.5, 0.5, 0.5
Suction groups	Low temperature (freezer): -20°F High temperature (cooler and dock): 30°F
Room temperature	Cooler: 40°F Freezer: -10°F Dock: 40°F

Model Parameter	Value
Evaporator fan power	0.15 W/CFM (0.32 hp per ton)
Condenser type	Evaporative condenser
Minimum condensing temperature	85
Condenser fan and pump power	330 Btu/watt
Condenser design approach temperature	23°F (CZ 13, design wetbulb = 73°F) 25°F (CZ 3, design wetbulb = 64°F)

Table 22. Prototypical Small Refrigerated Warehouse Model Description

Model Parameter	Value
Shape	Rectangular (200 ft by 130 ft)
Floor area	Freezer: 10,000 SF Cooler: 10,000 SF Shipping Dock: 6,000 SF Total: 26,000 SF
Number of floors	1
Floor to ceiling height	30 ft
Exterior wall construction	Insulated metal panel
Ext wall R-Value	Cooler and loading dock – R-20 Freezer – R-26
Infiltration rate	Cooler and Freezer: 0.1 ACH Loading Dock: 0.3 ACH
Roof construction	Insulated low mass roof
Roof R-values	Cooler and loading dock – R-23 Freezer – R-46
Roof absorptivity	0.80
Lighting power density	0.6 W/SF
Equipment power density	0.7 W/SF (covers fork lifts and miscellaneous plug loads and equipment)
Operating schedule	24 / 7
No. People	52 max
Evaporator type	Constant volume, continuous fan operation
Evaporator Size (climate zone 13)	Cooler: 18 ton (550 SF/ton) Freezer: 26 ton (380 SF/ton) Dock: 17 ton (360 SF/ton)
Evaporator CFM (climate zone 13)	Cooler: 30,000 cfm (3 cfm/SF) Freezer: 35,000 cfm (3.5 cfm/SF) Dock: 28,000 cfm (4.7 cfm/SF)
Compressor type	HFC reciprocating compressor with unloader
Compressor configuration	Parallel equal, 2 compressors per suction group, size ratio 0.6, 0.6

Model Parameter	Value
Suction groups	Low temperature (freezer): -20°F High temperature (cooler and dock): 30°F
Room temperature	Cooler: 40°F Freezer: -10°F Dock: 40°F
Evaporator fan power	0.15 W/CFM (0.32 hp per ton)
Condenser type	Air-cooled condenser
Minimum condensing temperature	85
Condenser fan power	53 Btu/hr-watt
Condenser design approach temperature	15°F for cooler and dock, 10°F for freezer

An eQUEST representation of the building is shown in Figure 11:

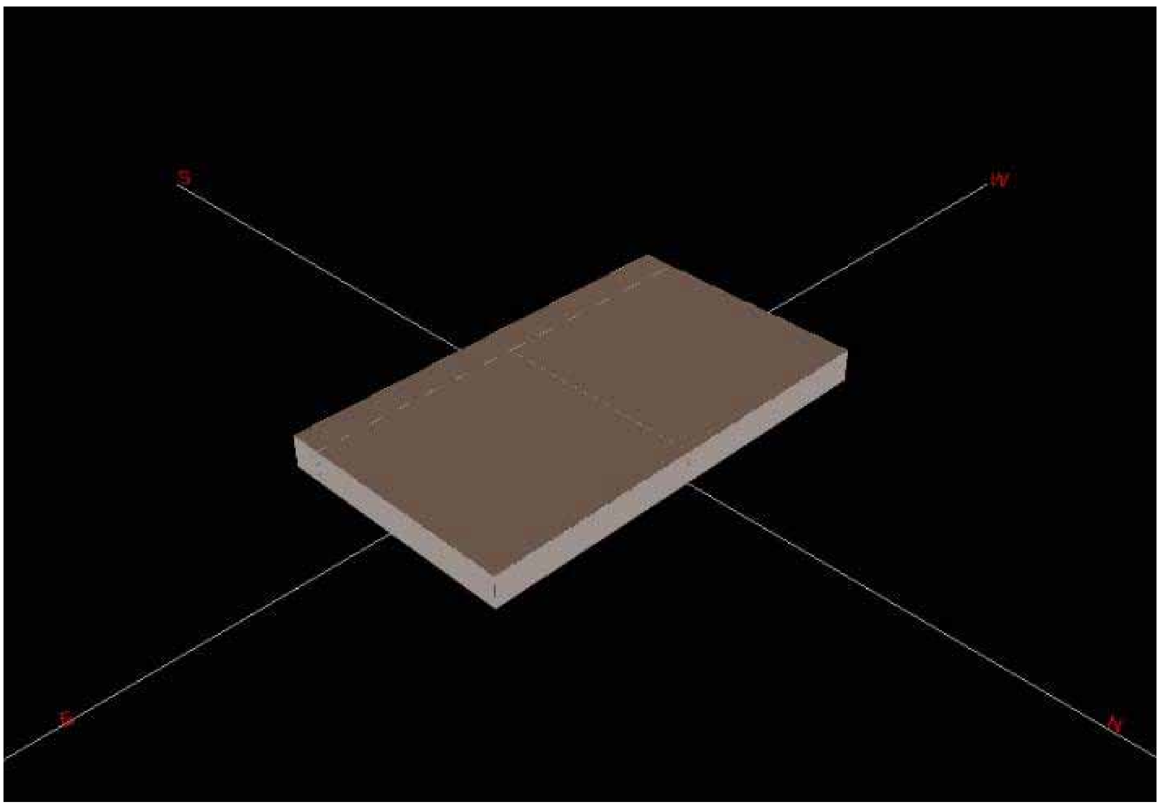

Figure 11. eQUEST representation of prototypical building model

Appendix B. Summary of Contractor Interviews

Building Shell

Two of the five people interviewed did not care to offer any answers regarding the building shell and typical construction practices. No contractor offered any information on the cost of the building shell. Most stated the recent volatile oil market has caused the price of insulation to be unpredictable. It was generally perceived that the only concern with regards to availability and market capacity was price.

There did not appear to be any variability in construction or insulation performance due to climate regions.

Freezers/Coolers Ceilings, Walls and Floors

Freezer ceiling construction for larger facilities would most likely place the panels on the outside over a metal “B” deck with EPDM roofing membrane over the insulation that would also act as the vapor barrier. Layers of 5” Isocyanurate achieving R values from R-31 to R-50 is typical. This same construction is used for both ice-cream and holding freezers. A code minimum of R-50 may receive some resistance.

For smaller facilities, a 6” expanded urethane metal clad sandwich panel is typical for the ceiling.

Typical freezer wall construction consists of the same 5” or 6” expanded urethane metal clad panels with R values that range from R-32 to R-56. The metal acts as the vapor barrier. In addition to the thermal performance characteristics, the thickness of the wall also becomes a function of the wall height. A code minimum of R-26 may be too low. A code minimum of R-32 appears reasonable and may achievable.

Due to the constructability of the facility, it is common for the cooler walls and ceilings to be the same thickness as the freezer walls and ceilings for facilities that house both coolers and freezers. It is also common for loading docks adjacent to coolers to share the same wall and ceiling thickness.

The wall separating a cooler from a freezer may be built to the same thickness due to the height of the structure. The height would dictate the wall thickness. This may also be true for the wall separating the cooler and freezer from the loading dock area. These separation walls, however, may not have the same thermal characteristics.

The wall separating the cooler from the freezer and the freezer from the loading dock typically has 5” to 6” of urethane for insulation and has a metal frame structure. A code minimum may not be necessary for these types of walls due to the fact that the end user is more concerned with condensation forming on the warm side of the wall and the thermal characteristics of the wall would have to be high enough to prevent condensation from occurring. These walls are typically built to a higher R value than the suggested code minimum of R-26.

Freezer floors are typically insulated from R-18 to R-30, depending on the soil and ground characteristics. They are typically constructed with glycol tubes set in a mud slab with 4” of rigid styrene over the mud slab and 6” of reinforced concrete poured over that. The glycol tubes may also be extended two feet out under the dock. One contractor mentioned a shift away from glycol piping to electric resistance heat due to liability issues from leaking glycol pipes. The size of the facility may also influence the economic viability of installing an under-slab glycol heating system and may favor electrical resistance heating on smaller floor prints. The suggested code minimum of R-30 appears reasonable and may be achievable. The thickness of the insulating panels may be limited by the structural characteristics required and achieving R-30 appears to not pose any structural problems.

In facilities that house only cooler storage, it is typical for the ceilings to be constructed of wood frame plywood with 4” of blown-on urethane insulation on the underside with R values from R-24 to R-40. The walls could be either 4” to 5” expanded urethane metal clad panels or sandwiched concrete panels with R values from R-23 to R-40. The code minimum of R-20 appears to be too low. A code minimum of R-25 may be reasonable and achievable.

Several contractors mentioned the extensive use of pre-stressed concrete beams with poured concrete slab on the roofs and concrete tilt-up panels utilized for wall construction. This building assembly allows for stable temperature and humidity levels. The end user of these types of buildings typically store fresh fruit commodities such as grapes.

Cooler floors are typically un-insulated concrete slab on grade.

Loading dock construction is highly dependant on how the rest of the facility is constructed and will typically be built to the same characteristics as the adjacent space.

Doors

Contractors typically do not get involved with door design and the frequency of use appears seasonal. A larger issue may be the type of seals use and/or the amount of infiltration that is allowed.

Cool Roof

Most contractors interviewed did not have much experience with roofing design and wondered how much a highly reflective surface would help a building that already ahs a large amount of insulation. One contractor that has performed experiments with cool roof designs has witnessed a drop in roof surface temperatures from ~140°F to ambient on days with a 100°F dry bulb temperature.

Low-E Paint

Most contractors liked to see the insides of the facilities painted mainly for the reflective quality and the possible reduction in required electric lighting. Some questioned whether the paint could be used on facilities regulated by the USDA and one contractor did not see how the paint could have any beneficial effect.

Underfloor Heating

The issue of leaking glycol piping under the slab should be investigated further to understand the extend of this concern. The use of air to heat under the slab appears to be under utilized. The installation of under-slab glycol heating appears to be the industry norm.

Refrigeration Systems

A typical response to questions of cost was that the cost are project specific and not evaluated on a cost per ton basis. Also that no two system are similar enough to produce reliable numbers.

Questions regarding fan and/or pump power (W/cfm and/or BTU/watt) did not receive much input. This strengthens the concern brought to light by one interviewee that 90% of cold storage facilities do not have load calculations done, that equipment is selected on a square foot per ton basis or based on the last facility done.

Evaporators

Evaporator Fan Speed Control

Common design practice is single speed and perhaps fan cycling. VFDs are becoming more common but do require a certain level of control to utilize. Several contractors did not find it reasonable to require VFDs for fan speed control but only offered for comments situations that would not require any evaporator fan modulation, in other words, they viewed it unreasonable because there may be situations or products that require constant air movement. Requiring the installation of VFDs for fan speed control could only work for the storage of products that do not require constant air movement. The availability and market capacity both appear high to medium high.

Evaporator Design Temperature Difference

The common design practice for evaporator design temperature difference is 10°F to 12°F for cold storage warehouses. This delta T could be shaved down to 7°F to 8°F for a higher performance design. However, several contractors were concerned about limiting the temperature difference for situations or products that require a lower humidity. This is also a concern for evaporators serving loading docks that may be designed with a 12°F to 15°F temperature difference. This appears to be a very product dependant or application specific design parameter. For fresh fruits that require a higher level of humidity to be maintained an even lower delta T would be used. The availability and market capacity appear high for evaporators designed for lower temperature differences and the only concern is designing for commodities requiring a lower level of humidity.

Evaporator Fan Power

Very little feedback was received regarding evaporator fan power. It is common for the designer not to be concerned with evaporator fan power.   The fan is selected to deliver a certain amount of air to a certain distance though a specific thickness of coil. For this reason, it is unclear what the market availability or the market capacity is for such a code minimum. For this reason, it also may be unreasonable to require a minimum watt/cfm for evaporator fan power.

Condensers

Condenser Type

The most commonly used condenser type for larger refrigerated storage facilities using ammonia refrigerant is evaporative condensing. High performance design considerations would include pre-cooling the water used (which is rarely done) or increasing the surface area. On smaller systems that utilize a halocarbon refrigerant, the evaporative condenser is considered for high performance designs only.

The limitations to utilizing an evaporative condenser on ammonia systems would be the water source and/or quality of available water. Limitations for utilizing an evaporative condenser on halocarbon refrigerant systems would also include first cost.

The availability of evaporative condensers and market capacity appears high and the only objections to requiring evaporative condensing are the cost impact to smaller systems and water availability.

Air Cooled Condenser Fan Speed Control

The common design practice for air cooled fan speed control is to cycle single speed fans. The high performance consideration is to control fan speed with a VFD. It is generally believed that because air cooled condensers are typically used on smaller systems and utilize several smaller fans which offers a wide range of modulation, the use of VFDs is often economically unattractive. However, the only comment regarding cost of adding a VFD to an air cooled condenser was an increase of 5% with no suggested increase in maintenance costs.

The code minimum specification of requiring a VFD to control fan speed on air cooled condensers appears to not be reasonable with tree responses of No, one Yes and one No comment (due to lack of experience with air cooled condensers). All contractors interviewed had more experience with evaporative condensers than with air cooled.

Air Cooled Condenser Design Approach Temperature

The responses for air cooled condenser design approach temperature varied greatly from 10°F to 30°F. When asked if having a code maximum of 10°F would be reasonable, the responses were equally varied from, it wouldn’t work, the condenser would be excessively large, to, remote condensers are always 0°F for low temp. and 15°F for medium temp.

The exceptions to the 10°F code minimum were only that the capitol cost would be too much. The market availability responses were two Highs, one Low and one No comment. The market capacity responses were one High, one Medium, one Low and one No comment. The only response to first cost was a suggested increase of 10% with no added maintenance costs.

All contractors interviewed were more familiar with evaporative condenser design and selection than with air cooled condensers. If air cooled condenser manufacture’s offer condensers sized for a 10°F design approach temperature, it appears reasonable for said code minimum to be achievable.

Air Cooled Condenser Fan Power

No responses were received to offer any insight into common design practices with regards to BTU per watt of condenser fan power. Fan power appears to not be part of the design selection process for air cooled condensers. Contractors interviewed had no idea what would be a reasonable number for BTU per watt or whether it would have any cost impact.

The performance of air cooled condensers would need to be analyzed to determined the most common BTU per watt and if any room for improvement exists. This would need to include an analysis of manufacture's performance data.

Evaporative Condenser Fan Speed Control

Both fan cycling and VFD fan speed control appear to be equally common. Utilization of VFDs to control fan speed on evaporative condensers has increased in recent years as the price of VFDs has come down. High performance designs would typically utilize VFD control. The availability of VFD fan speed control appears very high and the market capacity also appears high to medium. Only one contractor offered comments regarding price with a suggested 10% increase with minimal maintenance cost.

The only concerns stated for requiring a VFD to control fan speed is that it may not be necessary. However, VFDs are becoming more common due to price decreases and a growing number of contractors and facility operators that understand the control logic for optimization of head pressure control. The minimum proposed code specification of utilizing VFDs to control fan speed on evaporative condensers appears reasonable and achievable.

Evaporative Condenser Design Approach Temperature

The common design practice for evaporative condenser approach temperature ranges from 18°F to 20°F. High performance design practices range from 9°F to 15°F. The suggested code maximum design approach temperature of 20°F appears too high.

The availability and market capacity for providing an evaporative condenser with an approach temperature of 20°F appears very high. This codes minimum specification should be reviewed for opportunities to tighten this approach temperature down. A reasonable approach temperature appears to be in the range of 9°F to 15°F.

Evaporative Condenser Fan and Pump Power

Only two contractors offered comments regarding evaporative condenser fan and pump power. This code minimum specification appears similar to the fan power minimums for air cooled condensers in that it is not considered when selecting an evaporative condenser. The one response suggested that 200 BTUh/watt would be more realistic.

Some comments regarding the selection design data suggest that using 100°F saturated condensing temperature with 70°F entering wet bulb temperature may not be common. A more common selection design criteria appears to be 98°F saturated condensing temperature with a 78°F entering wet bulb temperature and 85/76 for high performance design.

The performance of evaporative condensers would need to be analyzed to determined the most common BTU per watt and if any room for improvement exists. This would need to include an analysis of manufacture's performance data.

Compressors

Compressor Capacity Modulation

The most common design practice for compressor capacity modulation is the use of slide valves on screw compressors. The high performance design practice would utilize a VFD to control the screw compressor's speed. When asked if requiring the utilization of a VFD for compressor capacity modulation was reasonable for a code minimum specification, four of five contractors responded with No. But when asked to provide situations were an exception should be made, the responses were more mixed and those that suggested there should be exceptions could not offer specific situations other than designing a plant with more compressors to allow for more stages of modulation.

The availability of VFDs appears high and the market capacity appears medium to high. Only one contractor offered a comment on price suggesting an increase of 30% with minimal maintenance costs. It is becoming more common for compressor manufacture's to offer VFD control as a package. A code minimum specification of requiring VFD control for compressor capacity modulation may be reasonable and achievable for the trim compressor of a multiple compressor plant or for a single compressor system that will see variations in load. For systems that may require a constant load or a minimum load that could be matched to the full load of the smallest compressor, requiring a VFD for compressor modulation may encounter some resistance.

Volume Ratio

Fixed and variable volume ratios are common and appear to be available from most compressor manufactures. Variable volume ratios would be considered a higher performance design practice.

Compressor Plant Capacity Modulation

Compressor plant capacity modulation is typically a function of facility requirements. Most common design practices appear to utilize multiple compressor of either equal or unequal sizes with multiple unequal size compressors being more common. High performance design practices would utilize VFD control on the trim compressor.

Compressor Oil Cooling

Thermosyphon oil cooling is the common design practice with liquid injection as a less preferred practice. Thermosyphon oil cooling is also considered the higher performance practice. Limitations to thermosyphon oil cooling are the physical locations of the thermosyphon vessel with respect to the compressor plant. These limitations are mostly negated for new construction.

When asked if a code minimum specification of requiring thermosyphon oil cooling would be reasonable, most contractors answered No. There appears to be a growing movement from compressor manufactures toward more efficient ways to perform liquid injection oil cooling. Liquid injection may also be easier and offer a greater sense of assuredness for oil cooling.

The availability of thermosyphon oil cooling appears high and the market capacity appears medium. A code minimum specification of requiring thermosyphon oil cooling may not be necessary since most contractors already offer it at minimum (5%) additional cost and there appears to be a potential for compressor manufacture's to provide a more efficient application of liquid injection.

Compressor Plant Stages

The number of compressor plant stages appears to be specific to the facility requirements. For facilities with only coolers, single stage is common. For facilities with both coolers and freezers, two stages without an economizer and with an inter-cooler or one stage with economizer are both common. The two stage system is considered the higher performance system.

Number of Suction Groups

The number of suction groups is also very dependant on the facility requirements. If more than one evaporator suction temperature is required and the difference between the two temperatures is large enough, it appears common to design for more than one suction group.

Use of Economizers on Single Stage Systems

The general consensus for use of economizers on single stage systems is that it would be more beneficial to add another smaller compressor or design a two stage system or even add another suction group.

Use of Intercooling on Multistage systems
Both flash and shell and coil appear equally common. The use of flash intercooling is considered the higher performance design practice. Intercooling appears to be commonly utilized and the use of flash intercooling appears to be more popular.

Lighting

Two contractors interviewed did not care to comment on lighting. No contractor offered any comments on lighting costs.

Lighting power Density
Common design practices range from 0.4 to 1.2 W/SF with 0.6 W/SF. appearing to be more acceptable. High performance design considerations include the use of T-5 bulbs and the use of occupancy sensors.

The availability and market capacity appears medium to high. A code maximum specification of 0.6 W/SF appears reasonable and achievable. The only concerns offered were regarding process facilities and the safety requirements for workers.

Issues Affecting Lighting and Power Density
Ceiling height, rack height and placement and labor policies appear to be the main issues affecting lighting and power density. There is also concern on how the new high performance fixtures will perform under the extreme conditions of refrigerated storage.

Lighting Controls
Contractors admitted to not being aware of the options available but have noticed an increase in bi-level lighting controls and even occupancy sensors. It is perceived as available and reasonable to require bi-level lighting control for a code minimum specification. The concerns offered include the special requirements in process applications and labor practices.

Controls

Computerized Control Systems
It appears to be common, especially on larger systems, for a facility wide controls system to be installed. High performance design practices would include more sensors and control points. This does allow for control code minimum specifications to be easily implemented.

Suction Pressure Control
Common design practice is to utilized a fixed suction pressure control and the high performance design would utilize a floating suction pressure control. The only limitations to floating suction pressure control are specific cases that may require a constant suction pressure and the addition of a control system. As stated above, a capable control system appears to be common on larger systems.

The availability and market capacity of adding floating suction pressure control appears to be high. The only cases offered as exceptions were process systems that may require lower humidity and therefore, a constant suction pressure. Only one contractor offered a comment on price with a suggested increase of 3% and minimal maintenance cost.

Requiring floating suction pressure control as a code minimum specification for refrigerated warehouses appears reasonable and achievable.

Condensing Temperature Control

The common design practice for condensing temperature control is fixed and the high performance design would utilize floating control from the wet bulb offset. The only limitations to utilizing floating head pressure control would be specific situations requiring a minimum head pressure for defrost requirements and/or liquid injection.

Utilizing floating head pressure control appears to be more acceptable than floating suction pressure control. Wet bulb offset may not be the only algorithm for controlling head pressure and there is some concern with applying a limitation to how the floating head pressure could be controlled.

The availability and market capacity appear to be high for implementing floating head pressure control. Only one contractor offered a comment on price with a suggested increase of 2-3% and minimal maintenance cost.

Requiring floating head pressure control as a code minimum specification for refrigerated warehouses appears reasonable and achievable but may not need to be limited to wet bulb offset control.

Defrost Type and Control

The common design practices for defrost type on larger ammonia systems is air for storage above 40°F and water for storage below 40°F and hot gas defrost. Hot gas defrost is considered the higher performance design practice.

The common design practice for defrost control is to utilize a time clock for scheduling defrost times and terminating the defrost based on time or temperature. High performance defrost controls utilize an accumulative run time or on-demand defrost.

Every contractor agreed that the use of electrical resistance heat for defrost on larger system should not be allowed and that requiring a code minimum specification of on-demand controls would appear reasonable and achievable. It appears that electrical resistance defrost is only used on smaller systems.

Demand Response

All contractors interviewed are familiar with demand response practices at some facilities. The product types that might be candidates for demand response strategies would include frozen packaged products, frozen juices and frozen products that do not require a minimum temperature.

These products can typically tolerate a 5°F drift in temperature and depending on the amount of product stored, and the traffic in the storage facility, this 5°F drift may take from 4 to 6 hours.

Other factors that will affect the length of time for this 5°F drift to occur include lighting, fan requirements and insulation performance.

Products that would not tolerate this strategy include frozen vegetables and frozen meats. Fluctuations in temperature will cause moisture migration in these products.

It appears common for 10-15% spare capacity to be designed into the refrigeration systems.

Other Opportunities

Most contractors agree that refrigerated warehouse could be designed and operated more efficiently. Areas to consider include VFD control of fans and compressors, improved door design and performance, lighting improvements and lighting controls, better control packages and building simulation to be performed on all large warehouses in order to optimize building envelope and refrigeration system options.

Appendix C. Contractor Interview Survey Instrument

Refrigerated Warehouse Case Initiative
Interview Guide for Refrigeration Industry Interviews
Version 5
Architectural Energy Corporation

General Information

Contact Name	
Title	
Company	
City and State	
Phone	

Area of expertise: ❐ Large warehouses (typically >50kSF with central refrigeration plant) OR ❐ Small warehouses (typically <50kSF with packaged DX refrigeration) (if both, complete a questionnaire for both types)

Questions:

1. Building Shell

1.1 Typical construction practices.

In your experience, what are typical industry design practices for (list attribute)

Attribute	Wall Construction Type (concrete tilt up, steel frame, Prefabricated metal insulated panel,, other	Construction Details (wall/ roof thickness, support spacing and thickness, insulation location and application)	Insulation R-value or Insulation Type and Thickness	Variability by Climate region? If yes, explain
Freezer Ceiling (Ice Cream)				
Freezer Ceiling (Holding Freezer)				
Freezer Exterior Wall				
Freezer to Cooler Wall				
Freezer to Dock Wall				
Freezer Floor R-value				
Cooler Ceiling				
Cooler Walls				
Dock Ceiling				
Dock to outdoor wall				
Dock Floor				

Other Design Details

Attribute	Typical practices and materials
Under floor heating	
Roofing materials	
Interior doors (freezer to dock)	Low freq of use (< cpd): Med freq of use (100-400 cpd) Hi freq of use (> 400 cpd)
Interior doors (cooler to dock)	Low freq of use (< cpd): Med freq of use (100-400 cpd) Hi freq of use (> 400 cpd)
Exterior doors	

1.2 Measure list and specifications

We are considering including (list attribute) in the Title 24 Standards. The minimum proposed specifications for the code are (list code minimum). Does this specification seem reasonable to you? (Yes or no, list reasons)

Attribute	Code minimum specification	Reasonable (Y/N)	ASHRAE Recommendations	Comments
Freezer Ceiling (Ice Cream)	R = 50		R-50 to R-60	
Freezer Ceiling (Holding Freezer)	R = 46		R-45 to R-50	
Freezer Exterior Wall)	R = 26		R-35 to R-40	
Freezer to Cooler Wall	R = 26			
Freezer to Dock Wall	R = 26			
Freezer Floor R-value	R = 30		R-27 to R-32	
Cooler Ceiling	R = 23		R-30 to R-35	
Cooler Walls	R = 20		R-25	
Dock Ceiling	R = 23			
Dock to outdoor wall	R = 20			
Dock Doors	R = 15			
Cool Roof	Solar reflectance > 0.70, thermal emittance > 0.75		Same as commercial	
Underfloor heating	No electric resistance			

1.3 Exceptions

Can you think of any situations where an exception should be made? (Yes or no)

What are the circumstances for an exception, and what should an alternative minimum specification be under the circumstance? (list circumstances and alternative specification)

Attribute	Code minimum specification	Exceptions (Y/N)	Circumstances and alternative specification
Freezer Ceiling (Ice Cream)	R = 50		
Freezer Ceiling (Holding Freezer)	R = 46		
Freezer Exterior Wall	R = 26		
Freezer to Cooler Wall	R = 26		
Freezer to Dock Wall	R = 26		
Freezer Floor R-value	R = 30		

Attribute	Code minimum specification	Exceptions (Y/N)	Circumstances and alternative specification
Cooler Ceiling	R = 23		
Cooler Walls	R = 20		
Dock Ceiling	R = 23		
Dock to outdoor wall	R = 20		
Dock Doors	R = 15		
Cool Roof	Solar reflectance > 0.70, thermal emittance > 0.75		

1.4 Measure availability and market capacity

How would you rate the availability of the equipment and/or materials? (high – available through my normal sources; medium – available through sources that I don't normally use; low – not available anywhere to my knowledge). If Title 24 were to include (list code minimum) for (list attribute), what, in your opinion, is the capacity of the market to supply the materials and equipment necessary to meet the demand? (high- can meet demand with little to no supply disruption; medium – some early disruption in supply anticipated, low –persistent lack of equipment for the foreseeable future).

Attribute	Code minimum specification	Availability (High, Medium, Low)	Market Capacity (High, Medium, Low)
Freezer Ceiling (Ice Cream)	R = 50		
Freezer Ceiling (Holding Freezer)	R = 46		
Freezer Exterior Wall)	R = 26		
Freezer to Cooler Wall	R = 26		
Freezer to Dock Wall	R = 26		
Freezer Floor R-value	R = 30		
Cooler Ceiling	R = 23		
Cooler Walls	R = 20		
Dock Ceiling	R = 23		
Dock to outdoor wall	R = 20		
Dock Doors	R = 15		
Cool Roof	Solar reflectance > 0.70, thermal emittance > 0.75		

1.5 Costs

What are the equipment and/or material first costs (absolute costs) for (list attribute)? How much of a difference is this relative to standard practices (list standard practice and % difference). Are there any incremental maintenance costs associated with (list attribute, annual maintenance costs and units (SF, ton, etc.).

Attribute	Code minimum specification	First Cost ($/SF)	% diff from common practice	Incremental Maintenance Cost ($/SF)
Freezer Ceiling (Ice Cream)	R = 50			
Freezer Ceiling (Holding Freezer)	R = 46			
Freezer Exterior Wall)	R = 26			
Freezer to Cooler Wall	R = 26			
Freezer to Dock Wall	R = 26			
Freezer Floor R-value	R = 30			
Cooler Ceiling	R = 23			

Attribute	Code minimum specification	First Cost ($/SF)	% diff from common practice	Incremental Maintenance Cost ($/SF)
Cooler Walls	R = 20			
Dock Ceiling	R = 23			
Dock to outdoor wall	R = 20			
Dock Doors	R = 15			
Cool Roof	Solar reflectance > 0.70, thermal emittance > 0.75			

Refrigeration System

2. Evaporators

2.1 Typical design practices.

In your experience, what are typical industry design practices for (list attribute)? and what changes to common design specification if any are commonly applied for refrigerated warehouses where the client is especially motivated to minimize energy consumption? What are the limitations to applying these design practices to all refrigerated warehouses?

Attribute	Common design practice	High performance design practice	High performance practice limitations or problems
Evaporator fan speed control (single speed, two speed, variable speed, other (list))			
Evaporator design temperature difference (Room temperature – refrigerant evaporation temperature)			
Evaporator fan power (Watts of fan power per cfm of air flow rate)			
Other evaporator design options			

2.2 Measure list and specifications

We are considering including (list attribute) in the Title 24 Standards. The minimum proposed specifications for the code are (list code minimum). Does this specification seem reasonable to you? (Yes or no, list reasons)

Attribute	Code minimum specification	Reasonable (Y/N)	Comments
Evaporator fan speed control	VSD		
Evaporator design approach temperature	10°F		
Evaporator fan power (W/CFM)	0.30		

2.3 Exceptions

Can you think of any situations where an exception should be made? (Yes or no)

What are the circumstances for an exception, and what should an alternative minimum specification be under the circumstance? (list exception and alternative specification)

Attribute	Code minimum specification	Exceptions (Y/N)	Circumstances and alternative specification
Evaporator fan speed control	VSD		
Evaporator design approach temperature	10°F		
Evaporator fan power (W/CFM)	0.30		

2.4 Measure availability and market capacity

How would you rate the availability of the equipment and/or materials? (high – available through my normal sources; medium – available through sources that I don't normally use; low – not available anywhere to my knowledge). If Title 24 were to include (list code minimum) for (list attribute), what, in your opinion, is the capacity of the market to supply the materials and equipment necessary to meet the demand? (high- can meet demand with little to no supply disruption; medium – some early disruption in supply anticipated, low –persistent lack of equipment for the foreseeable future).

Attribute	Code minimum specification	Availability (High, Medium, Low)	Market Capacity (High, Medium, Low)
Evaporator fan speed control	VSD		
Evaporator design approach temperature	10°F		
Evaporator fan power (W/CFM)	0.30		

2.5 Costs

What are the equipment and/or material first costs (absolute costs) for (list attribute)? How much of a difference is this relative to standard practices (list standard practice and % difference). Are there any incremental maintenance costs associated with (list attribute, annual maintenance costs and units (SF, ton, etc.).

Attribute	Code minimum specification	First Cost ($/TR)	% diff from common practice	Incremental Maintenance Cost ($/TR)
Evaporator fan speed control	VSD			
Evaporator design approach temperature	10°F			
Evaporator fan power (W/CFM)	0.30			

3. Condensers

3.1 Typical design practices.

In your experience, what are typical industry design practices for (list attribute)

Attribute	Common design practice	High performance design practice	High performance practice limitations or problems
Condenser type (air cooled or evaporative by system size and climate region)			
Air cooled condenser fan speed control (1 speed, two speed, variable speed, other (list))			
Air cooled condenser design approach temperature (saturated condensing temperature – ambient drybulb temperature; list evaporator temp)			
Air cooled condenser fan and pump power (Btu per watt @ 10°F delta T)			
Evaporative condenser fan speed control (1 speed, two speed, variable speed, other (list))			
Evaporative condenser design approach temperature (saturated condensing temperature – ambient wetbulb temperature at design conditions; list design WBT)			
Evaporative condenser fan and pump power (Btu per watt @ 100°F SCT, 70°F EWB)			

3.2 Measure list and specifications

We are considering including (list attribute) in the Title 24 Standards. The minimum proposed specifications for the code are (list code minimum). Does this specification seem reasonable to you? (Yes or no, list reasons)

Attribute	Code minimum specification	Reasonable (Y/N)	Comments
Condenser type	Evaporative condenser required in inland applications above xx TR		
Air cooled condenser fan speed control	Variable Speed		
Air cooled condenser design approach temperature	10°F		
Air cooled condenser fan and pump power	53 BTU/Watt at 10°F TD and sea level)		

Attribute	Code minimum specification	Reasonable (Y/N)	Comments
Evaporative condenser fan speed control	Variable Speed		
Evaporative condenser design approach temperature	20°F		
Evaporative condenser fan and pump power	330 BTU/Watt at 100°F SCT and 70°F WBT)		

3.3 Exceptions

Can you think of any situations where an exception should be made? (Yes or no)

What are the circumstances for an exception, and what should an alternative minimum specification be under the circumstance? (list exception and alternative specification)

Attribute	Code minimum specification	Exceptions (Y/N)	Circumstances and alternative specification
Condenser type	Evaporative condenser required in inland applications above xx TR		
Air cooled condenser fan speed control	Variable Speed		
Air cooled condenser design approach temperature	10°F		
Air cooled condenser fan and pump power	53 BTU/Watt at 10°F TD and sea level)		
Evaporative condenser fan speed control	Variable Speed		
Evaporative condenser design approach temperature	20°F		
Evaporative condenser fan and pump power	330 BTU/Watt at 100°F SCT and 70°F WBT)		

3.4 Measure availability and market capacity

How would you rate the availability of the equipment and/or materials? (high – available through my normal sources; medium – available through sources that I don't normally use; low – not available anywhere to my knowledge). If Title 24 were to include (list code minimum) for (list attribute), what, in your opinion, is the capacity of the market to supply the materials and equipment necessary to meet the demand? (high- can meet demand with little to no supply disruption; medium – some early disruption in supply anticipated, low –persistent lack of equipment for the foreseeable future).

Attribute	Code minimum specification	Availability (High, Medium, Low)	Market Capacity (High, Medium, Low)
Condenser type	Evaporative condenser required in inland applications above xx TR		
Air cooled condenser fan speed control	Variable Speed		
Air cooled condenser design approach temperature	10°F		
Air cooled condenser fan and pump power	53 BTU/Watt at 10°F TD and sea level)		
Evaporative condenser fan speed control	Variable Speed		
Evaporative condenser design approach temperature	20°F		
Evaporative condenser fan and pump power	330 BTU/Watt at 100°F SCT and 70°F WBT)		

3.5 Costs

What are the equipment and/or material first costs (absolute costs) for (list attribute)? How much of a difference is this relative to standard practices (list standard practice and % difference). Are there any incremental maintenance costs associated with (list attribute, annual maintenance costs and units (SF, ton, etc.).

Attribute	Code minimum specification	First Cost ($/ton of refrigeration capacity)	% diff from common practice	Incremental Maintenance Cost ($/ton of refrigeration capacity)
Condenser type	Evaporative condenser required in inland applications above xx TR			
Air cooled condenser fan speed control	Variable Speed			
Air cooled condenser design approach temperature	10°F			
Air cooled condenser fan and pump power	53 BTU/Watt at 10°F TD and sea level)			
Evaporative condenser fan speed control	Variable Speed			
Evaporative condenser design approach temperature	20°F			
Evaporative condenser fan and pump power	330 BTU/Watt at 100°F SCT and 70°F WBT)			

4. Compressors

4.1 Typical design practices.

In your experience, what are typical industry design practices for (list attribute)

Attribute	Common design practice	High performance design practice	High performance practice limitations or problems
Compressor capacity modulation (slide valve, VSD, other (list))			
Volume ratio (fixed, variable, other (list))			
Compressor plant capacity modulation (multiple, equal sized compressors, multiple unequal sized compressors, VSD trim compressor, slide valve trim compressor, other (list))			
Compressor oil cooling (refrigerant injection, water cooling, thermosyphon, other (list))			
Compressor plant stages (one, two, other (list))			
Number of suction groups			
Use of economizer on single stage systems			
Use of intercooling on multistage systems Intercooler type (flash, shell and coil, other (list))			

4.2 Measure list and specifications

We are considering including (list attribute) in the Title 24 Standards. The minimum proposed specifications for the code are (list code minimum). Does this specification seem reasonable to you? (Yes or no, list reasons)

Attribute	Code minimum specification	Reasonable (Y/N)	Comments
Compressor capacity modulation	VSD on at least one compressor per suction group above xx TR, or limit over-sizing and document constant load application		
Compressor oil cooling	Thermosyphon		

4.3 Exceptions

Can you think of any situations where an exception should be made? (Yes or no)

What are the circumstances for an exception, and what should an alternative minimum specification be under the circumstance? (list exception and alternative specification)

Attribute	Code minimum specification	Exceptions (Y/N)	Circumstances and alternative specification
Compressor capacity modulation	VSD on at least one compressor per suction group above xx TR, or limit over-sizing and document constant load application		
Compressor oil cooling	Thermosyphon		

4.4 Measure availability and market capacity

How would you rate the availability of the equipment and/or materials? (high – available through my normal sources; medium – available through sources that I don’t normally use; low – not available anywhere to my knowledge). If Title 24 were to include (list code minimum) for (list attribute), what, in your opinion, is the capacity of the market to supply the materials and equipment necessary to meet the demand? (high- can meet demand with little to no supply disruption; medium – some early disruption in supply anticipated, low –persistent lack of equipment for the foreseeable future).

Attribute	Code minimum specification	Availability (High, Medium, Low)	Market Capacity (High, Medium, Low)
Compressor capacity modulation	VSD on at least one compressor per suction group above xx TR, or limit over-sizing and document constant load application		
Compressor oil cooling	Thermosyphon		

4.5 Costs

What are the equipment and/or material first costs (absolute costs) for (list attribute)? How much of a difference is this relative to standard practices (list standard practice and % difference). Are there any incremental maintenance costs associated with (list attribute, annual maintenance costs and units (SF, ton, etc.).

Attribute	Code minimum specification	First Cost ($/TR)	% diff from common practice	Incremental Maintenance Cost ($/TR)
Compressor capacity modulation	VSD on at least one compressor per suction group above xx TR, or limit over-sizing and document constant load application			
Compressor oil cooling	Thermosyphon			

5. Lighting

5.1 Typical design practices.

In your experience, what are typical industry design practices for (list attribute)

Attribute	Common design practice	High performance design practice	High performance practice limitations or problems
Lighting power density in warehouse spaces (W/SF)			
Issues affecting lighting power density (lamp type, storage temperature, ceiling height, rack height, aisle width, other (list))			
Lighting controls (bi-level lighting, occupancy sensors, time clocks, other (list))			

5.2 Measure list and specifications

We are considering including (list attribute) in the Title 24 Standards. The minimum proposed specifications for the code are (list code minimum). Does this specification seem reasonable to you? (Yes or no, list reasons)

Attribute	Code minimum specification	Reasonable (Y/N)	Comments
Lighting power density in warehouse spaces (W/SF)	0.6 W/SF		
Lighting controls	Bi-level lighting		

5.3 Exceptions

Can you think of any situations where an exception should be made? (Yes or no)

What are the circumstances for an exception, and what should an alternative minimum specification be under the circumstance? (list exception and alternative specification)

Attribute	Code minimum specification	Exceptions (Y/N)	Circumstances and alternative specification
Lighting power density in warehouse spaces	0.6 W/SF		
Lighting controls	Bi-level lighting		

5.4 Measure availability and market capacity

How would you rate the availability of the equipment and/or materials? (high – available through my normal sources; medium – available through sources that I don’t normally use; low – not available anywhere to my knowledge). If Title 24 were to include (list code minimum) for (list attribute), what, in your opinion, is the capacity of the market to supply the materials and equipment necessary to meet the demand? (high- can meet demand with little to no supply disruption; medium – some early disruption in supply anticipated, low –persistent lack of equipment for the foreseeable future).

Attribute	Code minimum specification	Availability (High, Medium, Low)	Market Capacity (High, Medium, Low)
Lighting power density in warehouse spaces	0.6 W/SF		
Lighting controls	Bi-level lighting		

5.5 Costs

What are the equipment and/or material first costs (absolute costs) for (list attribute)? How much of a difference is this relative to standard practices (list standard practice and % difference). Are there any incremental maintenance costs associated with (list attribute, annual maintenance costs and units (SF, ton, etc.).

Attribute	Code minimum specification	First Cost ($/SF)	% diff from common practice	Incremental Maintenance Cost ($/TR)
Lighting power density in warehouse spaces	0.6 W/SF			
Lighting controls	Bi-level lighting			

6. Controls

6.1 Typical design practices.

In your experience, what are typical industry design practices for (list attribute)

Attribute	Common design practice	High performance design practice	High performance practice limitations or problems
Computerized control systems (facility wide, component only (list), none)			
Suction pressure control (fixed, floating, other (list))			
Condensing temperature control (fixed pressure control,- control point wetbulb offset – control point other (list))			
Defrost type (air, electric resistance, hot gas, hot water, other (list))			
Defrost controls (time clock, on-demand, other (list))			

6.2 Measure list and specifications

We are considering including (list attribute) in the Title 24 Standards. The minimum proposed specifications for the code are (list code minimum). Does this specification seem reasonable to you? (Yes or no, list reasons)

Attribute	Code minimum specification	Reasonable (Y/N)	Comments
Suction pressure control	Floating		
Condensing temperature control	Wetbulb offset		
Defrost control	On-demand controls required for electric resistance defrost		

6.3 Exceptions

Can you think of any situations where an exception should be made? (Yes or no)

What are the circumstances for an exception, and what should an alternative minimum specification be under the circumstance? (list exception and alternative specification)

Attribute	Code minimum specification	Exceptions (Y/N)	Circumstances and alternative specification
Suction pressure control	Floating		
Condensing temperature control	Wetbulb offset		
Defrost control	On-demand controls required for electric resistance defrost		

6.4 Measure availability and market capacity

How would you rate the availability of the equipment and/or materials? (high – available through my normal sources; medium – available through sources that I don't normally use; low – not available anywhere to my knowledge). If Title 24 were to include (list code minimum) for (list attribute), what, in your opinion, is the capacity of the market to supply the materials and equipment necessary to meet the demand? (high- can meet demand with little to no supply disruption; medium – some early disruption in supply anticipated, low –persistent lack of equipment for the foreseeable future).

Attribute	Code minimum specification	Availability (High, Medium, Low)	Market Capacity (High, Medium, Low)
Suction pressure control	Floating		
Condensing temperature control	Wetbulb offset		
Defrost control	On-demand controls required for electric resistance defrost		

6.5 Costs

What are the equipment and/or material first costs (absolute costs) for (list attribute)? How much of a difference is this relative to standard practices (list standard practice and % difference). Are there any incremental maintenance costs associated with (list attribute, annual maintenance costs and units (SF, ton, etc.).

Attribute	Code minimum specification	First Cost ($/TR)	% diff from common practice	Incremental Maintenance Cost ($/TR)
Suction pressure control	Floating			
Condensing temperature control	Wetbulb offset			
Defrost control	On-demand controls required for electric resistance defrost			

7. *Demand response*

Demand response, using the thermal mass of the building and stored product in frozen food warehouses is an idea that Pacific Gas and Electric is interested in exploring. The overall concept is to reduce the product temperature during off-peak periods and turn the refrigeration plant off during on-peak periods, allowing the product to float up to the nominal storage temperature. This strategy could be followed on a regular basis, or only during an anticipated peak demand emergency. I would like to ask a few questions regarding this topic to assess the overall feasibility of the approach:

7.1 Are you aware of any facilities where demand response using stored product thermal mass is practiced?

7.2 What product types might be candidates for this strategy?

7.3 What level of temperature variation can be tolerated by the products listed above?

7.4 What duration (in minutes or hours per day) of refrigeration plant interruption can be tolerated?

7.5 What factors influence the maximum acceptable load interruption duration (temperature excursion, RH control, other (list))?

7.6 How much spare capacity (refrigeration plant tons / peak refrigeration load tons) is typically available in frozen food warehouses (express as a percentage):

7.7 What factors or conditions might make this approach infeasible?

8. Other measures

What other measures or energy savings strategies do you think should be considered for inclusion into Title 24?

Appendix D. Time Dependent Valuation

The Title 24 building standards are based upon the cost-effectiveness of efficiency measures that can be incorporated into new buildings in California. The standards promote measures that have a greater value of energy savings than their cost. The Title 24 standards are flexible enough to allow building designers to make trade-offs between energy saving measures using computer analysis methods that evaluate the relative energy performance. For example, the energy losses from having more windows in a building design can be offset by better insulation or a higher efficiency air conditioner. Historically, within the Title 24 methodology, the value of energy efficiency measure savings had been calculated on the basis of a “flat” source energy cost, which does not vary by season, or by day-of-the-week, or by time-of-day.

Beginning with the 2005 standards update, time-dependent valuation (TDV) has been used in the cost-effectiveness calculation for Title 24. This allows the Title 24 efficiency standards to provide more realistic signals to building designers, encouraging them to design buildings that perform better during periods of high energy cost. The concept behind TDV is that energy efficiency measure savings should be valued differently at different times to better reflect the actual costs to users, to the utility system, and to society. For example, the savings of an energy measure that is very efficient during hot summer weekday afternoons would be valued more highly than a measure that achieves efficiency during the off-peak. This kind of savings valuation reflects the realities of the energy market, where high system demand on summer afternoons drives electricity prices much higher than during, say, night time hours in mild weather.

TDV is based on a series of 8760 values of energy cost, one for each hour of the typical CEC weather year. TDV values are available for each of the sixteen climate zones, for residential and for nonresidential buildings, and for electricity, natural gas and propane. Hourly energy savings estimates for a typical year are developed for a given building using a CEC-approved computer simulation tool, and those savings are then multiplied by each hour’s TDV value. The sum of these values is the annual savings.