SOUTHERN CALIFORNIA EDISON
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CODES AND STANDARDS ENHANCEMENT INITIATIVE (CASE)

2008 CEC Title 20 Appliance Efficiency Standards Rulemaking

Preliminary CASE Report:

Analysis of Standards Options for Walk-in Refrigerated Storage

Prepared for:

Randall Higa & Ramin Faramarzi
SCE Codes and Standards Program

Prepared by:

Heschong Mahone Group

Date: January 31, 2008

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1. EXECUTIVE SUMMARY

Southern California Edison (SCE) Codes and Standards Enhancement (CASE) Initiative Project seeks to address energy efficiency opportunities through development of new and updated "Title 20 standards" California Energy Commission's (CEC) appliance efficiency regulations (Title 20, Cal. Code Regs,, §§ 1601 – 1608). This preliminary CASE report will document information and data helpful to the California Energy Commission (CEC) and other stakeholders in the development of these new and updated standards. This CASE report covers standards and options for walk-in refrigerated storage (walk-in coolers and walk-in freezers).

The 2006 Title 20 appliance regulations include a number of important efficiency measures for walk-in refrigerated storage. However, the SCE team has identified a number of additional cost-effective and feasible measures that should be applied to walk-in refrigerated storage; these measures are:

- Infiltration reduction with strip curtains or spring hinged doors
- High efficacy lighting or lighting controls (to match Federal standard)
- Freezer floor insulation of at least R-28 (to match Federal standard)
- Floating head pressure control
- Compressors capable of operating at 70ºF condensing temperatures
- Variable speed evaporator fans
- Temperature termination defrost controls
- Anti-sweat heater wattage limits and with humidity responsive controls

This is a preliminary CASE report on these very cost-effective measures. Most of these measures have simple paybacks that are close to one year and benefit/cost ratios well excess of the minimum B/C ratio of 1.0. This proposal will be refined after consultations with stakeholders and a draft CASE proposal will be submitted. We look forward to the input and assistance from these stakeholders that include manufacturers, end-users, researchers, utilities and CEC staff.

2.   BACKGROUND

In 2004, Pacific Gas and Electric (PG&E) had proposed appliance efficiency standards for walk-ins. These standards were adopted and incorporated into Title 20 Section 1605.3(a)4 "Energy Design Standards for Walk-In Refrigerators and Walk-In Freezers." This standard was primarily focused on: door closers, levels of insulation, motor efficiency and controlling operation or limiting capacity of anti-sweat heaters.

As a result of a negotiated proposal between ARI (Air-Conditioning and Refrigeration Institute), ACEEE (American Council for and Energy Efficient Economy) Federal walk-in efficiency standards for walk-ins were inserted into the Energy Independence and Security Act of 2007 (EISA 2007) (Pub. Law 110-140, H.R. 6). Additional code measures were also added at this time namely a minimum efficacy requirement for lighting and floor insulation required for freezers.

The California IOU's intervened and pointed out that there were additional cost-effective efficiency measures that were not included either in the Title 20 requirements or the proposed new Federal requirements. Given that both the NAECA (National Appliance Efficiency and Conservation Act) and the Energy Policy Act have provisions that pre-empt State appliance efficiency regulations and a future Title 20 requirement was likely; this proposal was seen as not a good deal for California appliance efficiency.

The negotiated settlement was that the California IOUs would drop their objections to a Federal walk-in efficiency regulation if California could have window of time where the State of California could develop and enforce an updated walk-in refrigerated storage appliance standard.   California would have to adopt the standard before January 2011. In 2012 the Department of Energy is required to complete the development of a performance based Federal walk-in regulation. When this Federal performance takes effect in 2015, it would pre-empt the California walk-in appliance standard.

Thus it is desirable to set a new California Title 20 walk-in appliance efficiency standard as soon as possible with all of the currently cost-effective measures included. This will provide a longer period during which the State can enjoy the benefits of high efficiency walk-ins and render it more likely that the features in a California walk-in appliance standard will migrate into the 2012 Federal walk-in appliance regulation.

3.   OVERVIEW

Description of Standards Proposal	See *Executive Summary.*		
California Stock and Sales		Coolers	Freezers
	Total number of Existing Walk-ins in CA	65,340	33,275
	Total Existing Sq Ft in CA	5,990,371	3,646,940
	Total new Sq Ft in CA	337,474	205,500
Energy Savings and Demand Reduction	See *Analysis and Results* section.		
Economic Analysis	See *Analysis and Results* section.		
Non-Energy Benefits	Besides the environmental benefits associated with reduced energy consumption, this proposal would also reduce the fluctuations in walk-in temperature resulting from defrost cycles that have not terminated once all the ice has melted from the evaporator coil.		
Environmental Impacts	Largest environmental impact results from reduced emissions due to energy savings. First year impacts would reduce Carbon Dioxide emissions by 44,000 tons/yr. See *Statewide Environmental Impact* section. Fluorescent lamps must be disposed of to keep mercury out of waste stream; this is true for this technology as a whole which supports much of California's lighting energy efficiency goals		
Acceptance Issues	Manufacturers, distributors and specifiers must be educated on the appliance regulations for walk-ins similar to other covered appliances. .		
AB 1109 (California Lighting Efficiency and Toxics Reduction Act)	The lighting efficiency measure in this proposal will help achieve the AB 1109 goals. However, the amount of lighting in walk-ins is fairly small.		

Federal Preemption or other Regulatory or Legislative Considerations	This standard will be eventually pre-empted in 2015. Special legislation allows this standard to be set and enforced by the State of California until 2015. This proposed standard may help set the baseline for the future Federal regulation.

4. METHODOLOGY

The analysis methods are described for each measure in the following subsections. These results draw upon refrigeration efficiency programs that SCE has delivered to its customers over time.

4.1 Cost of electricity

The economic value of electricity we are using for this analysis is based on the cost of energy embedded in the time dependent valuation files used in the Alternative Compliance Method (ACM) Manual. These costs are based on a forecast of energy costs over 15 years and discounted based upon a 3% real discount rate.

To maintain a conservative estimate of energy cost savings, our calculations are based upon the costs in the least expensive climate zone. For a 15 year period of analysis, a kWh of electricity costs $1.314 present valued dollars. This is equivalent to 11 cents per kWh and an annual basis. When the measure life is some other value than 15 years we multiply 11 cents per kWh by the uniform series present worth factor for that time series with a 3% real discount rate.

Table 1: Average Present Valued Costs of Electricity
(Nonresidential, 15 yr period of analysis, 3% real discount rate)

Climate Temperature Zone	Average PV $/kWh
CTZ1	$1.570
CTZ2	$1.568
CTZ3	$1.578
CTZ4	$1.574
CTZ5	$1.580
CTZ6	$1.789
CTZ7	$1.314
CTZ8	$1.786
CTZ9	$1.782
CTZ10	$1.778
CTZ11	$1.563
CTZ12	$1.565
CTZ13	$1.576
CTZ14	$1.778
CTZ15	$1.778
CTZ16	$1.564
min	**$1.314**
max	**$1.789**
average	**$1.634**

4.2 Measure Life

The Arthur D Little report on refrigeration efficiency opportunities provided estimates of design life ranging from 12 to 25 years for envelope components and from 8 to 12 years for refrigeration components. (ADL 1996) The prior Title 20 CASE report on Walk-in Coolers and Freezers assumed an average life of ten years for most refrigeration system components and 18 years for envelope components (PG&E 2004). This report will also use 18 years for envelope components and 10 years for controls A measure life of 4 years is used for strip doors in keeping with the earlier analysis conducted by Energy Power Solutions. (EPS 2007)

4.3 EPS walk-in simulation model

The analysis of floating head controls, variable speed evaporator fans and infiltration reduction (air curtains) was conducted by Energy and Power Solutions (EPS) to support SCE's intervention on the Federal walk-in regulation. To this end EPS designed a number of combined refrigerator and freezer walk-ins of different sizes to evaluate if there was a size breakpoint for the various measures we evaluated. This analysis was conducted using the refrigeration version of eQUEST (DOE-2.2R) the weather files for Baltimore Maryland. As will be shown the cost-effectiveness in this climate was overwhelming. If it is desired to conduct a similar analysis in the drier California climate, that can be performed.

The following equipment was selected to analyze the measures:

a. Case 1 (250 SF walk-in cooler and a 250 SF walk-in freezer):
   i. Cooler compressor(s): One (1) Carlyle 5F20 compressor
   ii. Cooler evaporator(s): One (1) Krack KR66A-310
   iii. Cooler air-cooled condenser(s): One (1) Bohn BRH023
   iv. Freezer compressor(s): One (1) Carlyle 5F60 compressor
   v. Freezer evaporator(s): Three (3) Krack KR66E-270
   vi. Freezer air-cooled condenser(s): One (1) Bohn BRH023

b. Case 2 (500 SF walk-in cooler and a 500 SF walk-in freezer):
   i. Cooler compressor(s): One (1) Carlyle 5F20 compressor
   ii. Cooler evaporator(s): Two (2) Krack KR66A-310
   iii. Cooler air-cooled condenser(s): One (1) Bohn BRH023
   iv. Freezer compressor(s): One (1) Carlyle 5H60 compressor
   v. Freezer evaporator(s): Six (6) Krack KR66E-270
   vi. Freezer air-cooled condenser(s): One (1) Bohn BRH049

c. Case 3 (1000 SF walk-in cooler and a 1000 SF walk-in freezer):
   i. Cooler compressor(s): One (1) Carlyle 5F40 compressor
   ii. Cooler evaporator(s): Four (4) Krack KR66A-310
   iii. Cooler air-cooled condenser(s): One (1) Bohn BRH031
   iv. Freezer compressor(s): One (1) Carlyle 5H120 compressor
   v. Freezer evaporator(s): Twelve (12) Krack KR66E-270
   vi. Freezer air-cooled condenser(s): One (1) Bohn BRH086

d. Case 4 (2500 SF walk-in cooler and a 2500 SF walk-in freezer):
   i. Cooler compressor(s): One (1) Carlyle 5H40 compressor
   ii. Cooler evaporator(s): Nine (9) Krack KR66A-310
   iii. Cooler air-cooled condenser(s): One (1) Bohn BRH065
   iv. Freezer compressor(s): Two (2) Carlyle 5H120 compressors
   v. Freezer evaporator(s): Twenty Eight (28) Krack KR66E-270

vi. Freezer air-cooled condenser(s): One (1) Bohn BRH202

The total refrigeration system capacity considered for each case is:

    a. Case 1:   2.6 Tons-R cooler; 6.8 Tons-R freezer.

    b. Case 2:   5.2 Tons-R cooler; 13.5 Tons-R freezer.

    c. Case 3:   10.3 Tons-R cooler; 27.0 Tons-R freezer.

    d. Case 4:   23.3 Tons-R cooler; 75.4 Tons-R freezer.

4.3.1 Assumptions and References

Some assumptions were made based on industry standards and other resources in order to simulate a working model. The following table shows a list of assumptions and their sources. For strip curtains, a 75% reduction in infiltration rates was assumed and for swing-type plastic hinged doors, a 95% reduction in the infiltration rates was assumed[1]. The base case compressors are not allowed to unload but allowed to cycle off and on when the suction temperature falls below a minimum cut-off point or goes above a maximum cut-in point respectively. Also, in the base case, the evaporator fans are assumed to operate at full speed throughout the year.

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[1] Based on information provided by Refrigeration and Thermal Test Center (RTTC) of SCE

Table 2 : Assumptions and References

Component	Case	Assumption		Source
Cooler temperature	All	35	ºF	
Freezer temperature	All	5	ºF	
Product load	All	70%	capacity	
Lighting load	All	1.0	W/SF	Heatcraft Refrigeration
Cooler infiltration rate	Case 1	1.00	Air changes per hour	Heatcraft Refrigeration
Cooler infiltration rate	Case 2	0.68	Air changes per hour	Heatcraft Refrigeration
Cooler infiltration rate	Case 3	0.46	Air changes per hour	Heatcraft Refrigeration
Cooler infiltration rate	Case 4	0.25	Air changes per hour	Heatcraft Refrigeration
Freezer infiltration rate	Case 1	0.78	Air changes per hour	Heatcraft Refrigeration
Freezer infiltration rate	Case 2	0.53	Air changes per hour	Heatcraft Refrigeration
Freezer infiltration rate	Case 3	0.36	Air changes per hour	Heatcraft Refrigeration
Freezer infiltration rate	Case 4	0.21	Air changes per hour	Heatcraft Refrigeration

4.4   Infiltration reduction

In walk-in freezers and coolers, product is loaded and unloaded by personnel at regular intervals. While doing so, for convenience of loading and unloading, the workers tend to leave the doors open. This causes infiltration of outside air (in the case of a grocery store, comfort conditioned air in the market space). This causes an increase in the infiltration load on the refrigeration system resulting in increased energy consumption. To reduce the effect of infiltration, strip curtains can be installed to minimize infiltration while making it convenient for the workers.

The calculation of savings from strip doors is based upon an assumed 75% reduction in infiltration loads.

4.5   High efficacy lighting

The requirements for high efficacy lighting and timer controls exceed the requirements of the Federal walk-in appliance regulation in the Energy Independence and Security Act of 2007 (EISA 2007) (Pub. Law 110-140, H.R. 6) because EISA requires only one measure of the other. Both replacing incandescent sources with fluorescent sources and installing occupancy controls are very cost effective.

4.6   Freezer floor insulation

The requirement for installing R-28 insulation on freezer floors matches the Federal walk-in appliance regulation in the Energy Independence and Security Act of 2007 (EISA 2007) (Pub. Law 110-140, H.R. 6).

Adding floor insulation decreases the overall UA (conductance area product) of the walk-in freezer envelope. For larger freezers there is the added benefit that this may prevent ground heaving associated with freezing the earth beneath the slab the freezer is sitting on.

Freezer floors are regularly insulated to save energy as well as to protect the integrity of the floor slab. Thus there is a well-developed market for providing insulated freezer floors.

As described earlier in the assumptions used for the computer simulations, we assume that freezers have a setpoint temperature of 5ºF. We also assume that the year round ground temperature is around 50ºF. We also assume that the floor slab is 6" poured concrete with a thermal resistance of 1.2 (hr·ft^2·ºF)/Btu[1]. The air film for this floor with flow of heat upward that a thermal resistance of 0.61 (hr·ft^2·ºF)/Btu[2]. Thus we are comparing the heat gains into the walk-in with two different floor U-factors:

Base case Freezer Floor U-factor = 1/ [1.2 + 0.61] = 0.55 Btu/ hr·ft^2·ºF

Proposed Freezer Floor U-factor = 1/ [1.2 + 28 + 0.61] = 0.034 Btu/ hr·ft^2·ºF

The annual electricity savings per square foot (kWh/yr·sf) resulting from insulating the freezer floor can be calculated as follows:

ES $\quad$ = (Ub – Up) x (Tg – Tf) x H x C / COP

Where,

U $\quad$ = $\quad$ Base Case Freezer Floor U-factor, 0.55 Btu/ hr·ft^2·ºF

U $\quad$ = $\quad$ Proposed Freezer Floor U-factor, 0.034 Btu/ hr·ft^2·ºF

H $\quad$ = $\quad$ hours per year, 8,760 h/yr

C $\quad$ = $\quad$ conversion coefficient, 0.000293 kW/Btu/h

COP $\quad$ = $\quad$ refrigerant plant coefficient of performance, no units

4.7 VSD Condenser fans & floating head pressure control

Floating the head pressure is a very common energy efficiency measure with very attractive economic benefits. Typically air-cooled condensing systems are set to condense at a fixed setpoint (usually set for the worst case scenario – the hottest day of the year). If a control system is installed to float this setpoint with the outside ambient dry-bulb temperature with an optimum approach, the compressor energy can be significantly reduced on cooler days and nights. This is because of the reduction of the lift (the difference in the pressures between which the compressor operates). A slight increase in the condenser fan energy is evident as the condenser fans work harder to achieve the lower setpoint on colder days and nights. But the increase in the fan energy

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[1] ASHRAE 2005. *Fundamentals Handbook*. Atlanta, Georgia, p. 25.8

[2] ASHRAE 2005. *Fundamentals Handbook*. Atlanta, Georgia, p. 25.2

is a small penalty compared to the decrease in the compressor energy. This measure has also been evaluated for all the scenarios.

4.8   Variable speed evaporator fan control

Variable speed control of evaporator fans allows the fans to unload based on the room temperature. When loads are low in the walk-in cooler or freezer, air flow can be minimized thereby reducing the fan speed. Since power is a cubic function of the speed, a small reduction in flow and speed results in an increased energy savings. In addition, the motor load (placed in the air flow which contributes heat) also reduces thereby reducing the refrigeration energy consumption.

The walk-in cooler and freezer models assume that the base case is an evaporator fan that is operating continuously. Our proposed case assumes that fan speed modulates with load.

4.9   Temperature termination defrost controls

According to the USDOE, temperature termination defrost controls are already widely used on commercial refrigeration equipment and save energy. "Defrost cycle control can reduce energy consumption by reducing the frequency and duration of defrost periods. The majority of equipment currently manufactured already uses partial defrost cycle control in the form of cycle termination control."[1]

4.10   Anti-sweat heater humidity responsive controls

ASHs are electric resistance heaters installed in the frames and doors of refrigerated cases. Typical case construction requires three sets of heaters for glass doors (Figure 1):
- Case mullion heaters – located inside the case frame to keep the doors from freezing shut (green in Figure 1).
- Door frame heaters – located in the door frame to keep the doors from freezing shut and provide some heat to the glass (red in Figure 1).
- Glass heaters – located on the glass itself to raise its surface temperature and prevent condensation (blue in Figure 1).

Case manufacturers specify that only the glass heaters may be cycled with an ASH controller.

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[1]   Federal Register: July 26, 2007 (Volume 72, Number 143)] Proposed Rules Page 41161-41210
http://www.epa.gov/EPA-IMPACT/2007/July/Day-26/i3640.htm

Figure 1: Typical ASH locations

The temperature at which condensation will occur (also known as dew point (DP)) depends on the amount of moisture in the air of the surrounding environment. More humid air will condense at a higher temperature than drier air. Condensation takes place when moist air contacts a cold surface with a temperature below the DP temperature of air. Absent ASH, formation of condensation on glass doors and door frames of refrigerated display cases is a likely occurrence in supermarkets.

In standard installations, the ASHs operate at full power 100% of the time. ASH controllers monitor the DP temperature of air in the sales area and adjust the duty cycle of the heaters accordingly. For example, when the air is dry and its dew point is low, the ASHs operate at a low duty cycle and surface is allowed to get cold since condensation will not form. On the other hand, when the air is humid and dew point is high, the ASHs operate at 100% duty cycle to keep the surface warm and above the dew point temperature. Between these extremes, the duty cycle is adjusted according to the measured DP.

Some of the heat generated by ASHs ends up as a load on the refrigeration system. Therefore, any reduction in ASH power not only will reduce the ASH electric demand, it will also result in secondary benefits on the refrigeration side. As a result, compressor run time and energy consumption are reduced.

This measure applies to ASHs on glass doors of both low temperature ("freezer" – below 32°F) and medium temperature ("cooler" – above 32°F) display cases and walk-ins. The E3 calculator lists the savings and cost figures on a linear-foot of doors basis. Calculations of savings and cost on a linear-foot of doors basis for both coolers and freezers were carried out for 8 climate zones in SCE's service territory (CTZs 6, 8, 9, 10, 13, 14, 15, and 16) for grocery store applications. The detailed calculation methodology is given in Appendix 1: Anti-sweat Energy Savings Estimation Methodology.

5. ANALYSIS AND RESULTS

5.1 Infiltration reduction

As described earlier the space infiltration was assumed to be decreased by 75% by the presence of strip doors. The measure life is assumed to be 4 years which results in a present worth factor of 3.72 or a present valued electricity cost of PV$0.41/kWh. Given a cost of $126 for a strip door, the energy savings pay for the cost of the door in under a year for all size walk-ins.

Table 3 : Strip Door Cost-Effectiveness Calculations

Sizes (sf)	Savings kWh/yr	Savings %	Savings $/yr	Savings PV$	Incremental Cost ($)	Benefit / Cost Ratio
250	3,457	1.5%	$380	$1,414	$126	11.2
500	2,319	0.8%	$255	$948	$126	7.5
1,000	4,143	0.7%	$456	$1,694	$126	13.4
2,500	5,987	0.4%	$659	$2,448	$126	19.4

5.2 High efficacy lighting and controls

From an earlier report that evaluated lighting power in walk-ins, the lighting power density of walk-ins illuminated with incandescent lighting is approximately 1 W/sf (ADL 1996). If the same total lumens are delivered by a fluorescent source the efficacy is at least 3 times greater and thus the lighting power density for the same light levels is approximately 1/3 that with incandescent lighting.

Note that the incremental cost for the larger space is the same as the smaller space. The smaller space would likely have a hardwired compact fluorescent luminaire while the larger space would likely have installed a linear fluorescent luminaire.

Table 4 : Calculated savings due to high efficacy lighting

	Freezer	Cooler
Typical sq ft	80	240
Manual controls: Operating hours 75% on	6,570	6,570
Incandescent LPD (W/sf)	1.0	1.0
Total Watts	80	240
Incandescent energy consumption (kWh/yr)	526	1,577
High efficacy LPD (W/sf)	0.33	0.33
Total Watts	26	79
High efficacy energy consumption (kWh/yr)	173	520
High efficacy energy savings (kWh/yr)	352	1,056
High efficacy energy cost savings ($/yr)	$38.74	$116.21
High efficacy discounted PV energy savings over 10 years	$330.43	$991.30
Incremental cost of high efficacy fixture	$100	$100
Benefit Cost ratio	3.3	9.9

The following calculations are based on an assumption that walk-ins with manual lighting controls are lighted approximately 75% of the time and that the installation of a time switch or an occupancy sensor will reduce the fraction of the time the lights are on to 25% of the time. If this is indeed the case, the controls are cost effective.

	Freezer	Cooler
Typical sq ft	80	240
High efficacy LPD (W/sf)	0.33	0.33
Total Watts	26	79
Manual controls: Operating hours 75% on	6,570	6,570
Manual controls: High efficacy energy consumption (kWh/yr)	173	520
Timer or Occ. sensor controls: Operating hours 25% on	2,190	2,190
Timer or Occ. sensor controls: High efficacy energy consumption (kWh/yr)	58	173
Energy savings (kWh/yr)	116	347
Energy cost savings ($/yr)	$12.72	$38.16
Discounted PV energy savings over 10 years	$108.50	$325.50
Cost of controls	$50	$200
Benefit Cost ratio	2.2	1.6

5.3   Freezer floor insulation

As described in the methodology section, the annual electricity savings per square foot (kWh/yr·sf) resulting from insulating the freezer floor can be calculated as follows:

ES   = (Ub – Up) x (Tg – Tf) x H x C / COP

Where,

U   =   Base Case Freezer Floor U-factor, 0.55 Btu/ hr·ft^2·ºF

U   =   Proposed Freezer Floor U-factor, 0.034 Btu/ hr·ft^2·ºF

H   =   hours per year, 8,760 h/yr

C   =   conversion coefficient, 0.000293 kW/Btu/h

COP   =   refrigerant plant coefficient of performance, no units

For this calculation we are assuming that the refrigeration plant has a COP of 3. The total plant COP may be lower, in this case the energy savings from insulating the freezer floor will be higher.

Thus, the annual energy savings from insulating the freezer floor with R-28 insulation for each square foot of floor area is:

ES   = (0.55 – 0.034) x (50 – 5) x 8,760 x 0.000293/ 3 = 19.9 kWh/sf

At a conservatively low electrical cost of $0.11/kWh, the annual energy cost savings from insulating the floor is approximately $2.18/sf·yr. With a discounted present value cost of electricity being PV $ 1.51/kWh over an 18 year period of analysis, the present valued cost savings resulting from insulating the freezer floor is $30/sf. From discussion with walk-in manufacturers, adding the insulated floor to a walk-in freezer is approximately $1,000 for a 100 sf walk-in or approximately $10/sf. Thus the benefit to cost ratio is approximately 3 to 1.

5.4   Floating head pressure control

As shown in Table 5, the cost of floating head pressure controls is relatively constant regardless of systems size, yet the savings grow with the increasing loads associated with larger walk-in coolers and freezers. However, even for systems that have a 250 sf cooler and a 250 sf freezer, installing the controls is still very cost-effective. The period of analysis used for this calculation was 10 years and thus the present value of the cost of electricity over this time was PV$0.94/kWh.

Table 5: Head Pressure Controls Cost-Effectiveness Calculations

Sizes (sf)	Savings kWh/yr	Savings %	Savings $/yr	Savings PV$	Incremental Cost ($)	Benefit / Cost Ratio
250	31,506	13.7%	$3,466	$29,563	$5,340	5.5
500	23,459	7.8%	$2,580	$22,012	$5,340	4.1
1,000	48,866	7.7%	$5,375	$45,852	$5,340	8.6
2,500	142,027	10.2%	$15,623	$133,267	$5,340	25.0

5.5 Evaporator fan continuous speed control

The analysis summary in Table 6 compares fixed speed control of evaporator fans to fans that reduce their speed in response to space or product temperature. With fans smaller than 1 hp and less than 460 V the fans will have electrically commutated motors (brushless DC motors) and are very easy to control their speed. Larger motors will make use of adjustable speed drive controls. Simple paybacks for the controls are less than a year. With a 10 year period of analysis, fan speed control often has benefit cost ratios greater than 10.

Table 6: Evaporator Fan Speed Control Cost-Effectiveness Calculations

Sizes (sf)	Savings kWh/yr	Savings %	Savings $/yr	Savings PV$	Incremental Cost ($)	Benefit / Cost Ratio
250	18,994	8.3%	$2,089	$17,822	$1,700	10.5
500	27,104	9.0%	$2,981	$25,432	$3,400	7.5
1,000	59,218	9.3%	$6,514	$55,566	$5,100	10.9
2,500	140,040	10.1%	$15,404	$131,403	$8,500	15.5

Variable speed evaporator fan controls are also required in refrigerated warehouses by Section 126 of the Title 24 building efficiency standards. With this requirement, the manufacturers of refrigeration evaporators will be receiving a consistent message that their products regardless of application are required to have speed controls.

5.6 Temperature termination defrost controls

Defrost termination energy savings range from 2% to 5% of refrigeration system load (Kelsey 2004, Kosar & Dumitrescu 2005). Costs vary depending on control method. With a 2% savings, on average the control will save approximately 8 kWh/sf-yr. For the smallest system we considered: a 250 sf walk-in cooler and 250 sf walk-in freezer, the energy savings are approximately 4,000 kWh/yr. At $0.11/kWh, the annual savings are approximately $440/yr. The discounted present valued savings over 10 years for this control is approximately PV$3,760.

Typical paybacks are between 2.5 and 6 years (Kelsey 2004). With an a period of analysis of 10 years, having a simple payback less than 8.5 years results in Benefit/Cost ratio greater than 1.0.

This measure provides non-energy benefits as well; temperature fluctuations in the walk-in are reduced, which can improve product quality.

5.7   Anti-sweat heater humidity responsive controls

5.7.1 EM&V, Market Potential, and Other Studies

Installation of ASH controllers on medium and low temperature display cases was included in the 2004-5 EnergySmart® Grocer Program. Contractors who participated in the program estimated that approximately 41% of stores they service already have some type of ASH control in operation. Thus there are substantial levels of market penetration for this control. Increasing the penetration of this type of control would not cause undue burden on refrigeration equipment industry.

5.7.2 Measure Effective Useful Lives

ASH controller measures currently use a EUL of 12 years (CALMAC 2000).

5.7.3 Measure Costs

Wisconsin Focus on Energy lists new ASH controllers installed cost at $85 per door. (WFE 2004) Doors are typically 2.5 feet wide, giving a cost of approximately $34 per linear foot.

5.7.4 Energy savings and benefit cost estimates

The measured average connected load for ASH was approximately 0.37 amps per foot of case, and heaters were on 8% of the time with the controllers in place. This was in very close agreement with the deemed reduction value of 343 kWh/ft per year used by the program.

The report concludes that 90% direct ASH energy savings should be expected in most applications but does not discuss secondary effects on the refrigeration system. (PWP 2006) This program reported an annual net impact of 1,332,514 kWh/yr for installation of 4,046 ln-ft (329 kWh/ln-ft) of ASH on low temp doors and 7,576,500 kWh/yr for installation of 23,009 ln-ft (329 kWh/ln-ft) of ASH controls on medium temp doors.

Though the expected life of the control is 12 years, this report will conservatively estimate the life cycle costs based on a 10 year period of analysis. The present worth factor over 10 years at a 3% real discount rate is 8.5. Thus the present valued cost of annual electricity savings over 10 years is $0.94 per annual kWh/yr saved. Thus the anti-sweat heater controls which save 329 kWh/yr·lf save $36/yr·lf or a discounted present value of $309/lf over the 10 year period of analysis.

The historical average cost of retrofitting the ASH controls has been approximately $34/lf. Approximately half of this cost is labor. Thus even retrofitted the controls pay for themselves in about a year and have a benefit/cost ratio of 9.1 to 1.

6.   STATEWIDE SAVINGS ESTIMATES

Most of the calculations of savings can be readily normalized to a savings per square foot of floor area of the walk-ins. The statewide areas used in these calculations are taken from Table 7. These values were derived from the Arthur D Little Report and the PG&E Walk-in CASE report.

Table 7: California Statewide Quantities of Walk-in Coolers and Freezers

	Coolers	Freezers	Coolers + Freezers
Total Number of Existing Walk-ins in CA	65,340	33,275	98,615
Total Existing Sq Ft in CA	5,990,371	3,646,940	9,637,311
Total new Sq Ft in CA	337,474	205,500	542,974

The anti-sweat controls savings calculations were based upon on linear feet of door area. To convert this figure to a savings per sf of floor area, an assumption was made that there is 20 sf of floor area for each linear foot of door area. In addition the floor area to which anti-sweat heaters apply was cut in half as not all walk-ins have anti-sweat heaters.

The figures here are the technical savings potential from this CASE proposal. However, since these measures are very cost-effective, a good fraction of the walk-ins have incorporated some of these measures. Nonetheless, the first year after the effective date of this standard, tens of Gigawatt hours will be saved with increasingly more energy saved each subsequent year.

Table 8: Statewide Savings Estimate First Year and After Full Replacement

Measure	kWh/sf	PV$/sf	Statewide 1st Year GWh/yr	Statewide 1st Year PV$ Million	Statewide Full Replacement GWh/yr	Statewide Full Replacement PV$ Million
Infiltration	6.9	$2.8	3.8	$1.5	66.6	$27.3
High eff lighting	5.9	$5.5	3.2	$3.0	56.4	$52.8
Freezer floor insulation	19.9	$2.2	4.1	$0.4	72.6	$8.0
Floating head pressure	63.0	$59.1	34.2	$32.1	607.3	$569.8
Variable speed evap fan	38.0	$35.6	20.6	$19.4	366.1	$343.5
Defrost controls	8.0	$7.5	4.3	$4.1	77.1	$72.5
Anti-sweat controls	16.5	$15.5	4.5	$4.2	79.3	$74.4
Totals	**158.1**	**$128.2**	**74.7**	**$64.7**	**1,325.3**	**$1,148.2**

7. STATEWIDE ENVIRONMENTAL IMPACT

The emissions impacts of this g measure are calculated by multiplying the change in statewide electricity and natural gas consumption by the respective emissions factor values generated by the California Energy Commission for evaluating the environmental impacts of the 2005 standards as shown in Table 9 below.[1]

Table 9: Emissions Factors used to calculate the air emissions reductions resulting from end-use reductions in electricity and natural gas consumption

Emissions factors	NOx	CO	CO2	PM10
Electricity, Western States (lbs/MWh)	0.383	0.23	1200	0.06

Given the electricity savings of 75 GWh/yr the first year after adoption and the electricity savings of 1,325GWh/yr after all walk-ins have been upgraded to the this proposed standard, the air emissions savings are given in below.

Table 10: Statewide Emission Reductions

		Tons of Emission Reductions			
	GWh/yr	NOx	CO	CO2	PM10
Statewide 1st Year	75	14	9	44,802	2
Statewide Full Replacement	1,325	254	152	795,191	40

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[1] Table 1, Appendix B page 2, Initial Study/Proposed Negative Declaration for the 2005 Building Energy Efficiency Standards for Residential and Nonresidential Buildings September 2003 P400-03-018 http://www.energy.ca.gov/reports/2003-09-12_400-03-018.PDF Values provided by the CEC System Assessment and Facilities Siting Division.

8. RECOMMENDATIONS

We recommend that the California Energy Commission move as soon as possible to adopt new Title 20 appliance efficiency standards for walk-in refrigerated storage. As described in the Analysis and Results section of this CASE report there are a number of efficiency measures that are extremely cost-effective and can save a substantial fraction of the energy consumption of refrigerated storage. The following changes to the Title 20 standards are proposed:

- Change the designation of "walk-in refrigerators" to "walk-in coolers" – this reflects common usage and is compatible with the language in Energy Independence and Security Act of 2007 (EISA 2007) (Pub. Law 110-140, H.R. 6). Using the same designation as in EISA 2007 will help assure that manufacturers are aware that California has additional requirements for walk-ins beyond that of the Federal regulations.

- Add a definition of floating head controls

- Add requirement for infiltration reduction with strip curtains or spring hinged doors

- Add a requirement for high efficacy lighting <u>and</u> lighting controls. The Federal standard required either high efficacy or lighting controls. This change would require both.

- Match the new Federal standard requirement of freezer floor insulation of at least R-28

- Add requirement for floating head pressure control down to a condensing temperature of at least 70ºF.

- Require compressors capable of operating at 70ºF condensing temperatures

- Require evaporator fans speed controls that respond to space conditions.

- Require that the coil defrosting cycle is terminated upon reaching a coil or coil leaving air temperature

- Modify the anti-sweat hater requirements so that anti-sweat heaters have both wattage limits <u>and</u> humidity responsive controls

8.1 Proposed standards language

The existing language from the 2007 Title 20 is shown as black unformatted text. Proposed changes to the standards language will use <u>underlining to indicate new language</u> and ~~strikethroughs to show deleted language~~.   The existing language is excerpted from: the California Energy Commission, *2007 Appliance Efficiency Regulations*, CEC publication number CEC-400-2007-016-REV1 Revised December 29, 2007

and downloaded from http://www.energy.ca.gov/2007publications/CEC-400-2007-016/CEC-400-2007-016-REV1.PDF

Section 1601. Scope.

This Article applies to the following types of new appliances, if they are sold or offered for sale in California, except those sold wholesale in California for final retail sale outside the state and those designed and sold exclusively for use in recreational vehicles, or other mobile equipment. Each provision applies only to units manufactured on or after the effective date of the provision.

Note: For the applicability of these regulations to appliances installed in new building construction, see Sections 110 and 111 of Part 6 of Title 24 of the California Code of Regulations.

(a) Refrigerators, refrigerator-freezers, and freezers that can be operated by alternating current electricity, including but not limited to refrigerated bottled or canned beverage vending machines, automatic commercial ice-makers, refrigerators with or without doors, freezers with or without doors, walk-in ~~refrigerators~~ coolers, walk-in freezers, and water dispensers, but excluding the following types:

(1) consumer products with total refrigerated volume exceeding 39 ft^3;

(2) commercial refrigerators, commercial refrigerator-freezers, and commercial freezers with total refrigerated volume exceeding 85 ft^3; except that walk-in ~~refrigerators~~ coolers and walk-in freezers are not excluded.

Section 1602. Definitions.
(a) General.
In this Article the following definitions apply. If a term is not defined here, the applicable definition in NAECA, EPAct, or the test methods listed in Section 1604 shall apply where it is reasonable to do so. …

"Envelope" of a ~~refrigerator~~ cooler or walk-in freezer means the walls and ceiling of the walk-in refrigerator or walk-in freezer but not the doors or floors.

"Floating head pressure" is a condenser fan control system which allows the compressor discharge pressure to float lower than design condensing pressure during periods of low ambient air temperature.

"Remote," in reference to any refrigerator, freezer, refrigerator-freezer, reach-in cabinet, pass through cabinet, roll-in or roll-through cabinet, walk-in ~~refrigerator~~ cooler, or walk-in freezer means an appliance that:
(1) receives refrigerant fluid from a condensing unit located externally to its cabinet assembly; and
(2) is capable of being purchased and installed with different types of compressor or condenser, so that its efficiency depends on the type of compressor or condenser applied by the purchaser, installer, or user.

"Walk-in freezer" means a space refrigerated to temperatures below 32° F that can be walked into.

"Walk-in ~~refrigerator~~ cooler" means a space refrigerated to temperatures at or above 32° F that can be walked into.

Section 1604. Test Methods for Specific Appliances.
(a) **Refrigerators, Refrigerator-Freezers, and Freezers.**
…
(5) There is no test method for walk-in refrigerators or walk-in freezers.

Section 1605.3. State Standards for Non-Federally-Regulated Appliances.
(a) **Coolers, Refrigerators, Refrigerator-Freezers, and Freezers.**
…
(4) **Energy Design Standards for Walk-In ~~Refrigerators~~ Coolers and Walk-In Freezers.**
(i) **All Walk-in ~~Refrigerators~~ Coolers and Walk-in Freezers**. Walk-in ~~refrigerators~~ coolers and walk-in freezers, with an enclosed floor area of 3,000 sf or less, manufactured on or after the effective dates shown in Table A-6, with the applicable ~~motor types~~ criteria shown in Table A-6, shall be manufactured with the required components shown in Table A-6.

Table A-6:
Energy Design Standards for Walk-In ~~Refrigerators~~ Coolers and Walk-In Freezers

~~Motor Type~~ Criteria	Effective Date	Required Components
All	January 1, 2006	Automatic door closers that firmly close all reach-in doors
All	January 1, 2006	Automatic door closers on all doors no wider than four foot or higher than seven foot, that firmly close walk-in doors that have been closed to within one inch of full closure
All	January 1, 2009	All entry doors shall have spring loaded doors, strip curtains, or other equivalent method of reducing infiltration when the primary doors are open as determined by the Executive Director
~~All~~ Coolers	January 1, 2006	Envelope insulation > R-28 for ~~Refrigerators~~ Coolers
~~All~~ Freezers	January 1, 2006	Envelope insulation > R-36 for Freezers
Freezers	January 1, 2009	Floor insulation > R-28 for freezers
All	January 1, 2009	All interior lights use light sources with an efficacy of at least 40 lumen/Watt including ballasts losses (if any) and light is controlled by a 15 minute timer switch or an occupancy sensor.
All	January 1, 2011	Refrigerant systems serving walk-in coolers shall be equipped with floating head pressure controls. The minimum condensing temperature setpoint shall be less than or equal to 70°F.
All	January 1, 2011	Compressors shall be designed to operate at a minimum condensing temperature of 70°F or less
Condenser Fan Motors < 1 HP	January 1, 2006	(i) Electronically commutated motors, (ii) permanent split capacitor-type motors, (iii) polyphase motors > ½ HP, or (iv) motors of equivalent efficiency as determined by the Executive Director
~~Single-phase Evaporator Fan Motors < 1 HP and < 460 volts~~	~~January 1, 2006~~	~~(i) Electronically commutated motors or (ii) permanent split capacitor-type motors~~
All	January 1, 2011	All evaporator fans shall be continuously variable speed and the speed shall be controlled in response to space conditions
Single-phase Evaporator Fan Motors < 1 HP and < 460 volts	January 1, 2008	Electronically commutated motors

~~Motor Type~~ Criteria	Effective Date	Required Components
All	January 1, 2011	Ice defrost cycles shall terminate based on temperature of evaporator coil or temperature of air leaving evaporator.

(ii) **Walk-in Refrigerators and Walk-in Freezers with Transparent Reach-in Doors.**

In addition to the requirements in (i), walk-in refrigerators and walk-in freezers with transparent reach-in doors that are manufactured on or after January 1, 2006 shall meet the following requirements:

(I) transparent reach-in doors shall be of triple-pane glass with either heat-reflective treated glass or gas fill;

(II) if the appliance has an anti-sweat heater ~~without anti-sweat heat controls~~, then: the appliance shall have a total door rail, glass, and frame heater power draw of no more than 40 watts (freezers) or 17 watts (refrigerators) per foot of door frame width; and

(III) if the appliance has an anti-sweat heater ~~with anti-sweat heat controls, and the total door rail, glass, and frame heater power draw is more than 40 watts (freezers) or 17 watts (refrigerators) per foot of door frame width, then:~~ the anti-sweat heat controls shall reduce the energy use of the anti-sweat heater in an amount corresponding to the relative humidity in the air outside the door or to the condensation on the inner glass pane.

9.   BIBLIOGRAPHY AND OTHER RESEARCH

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10. APPENDIX 1: ANTI-SWEAT ENERGY SAVINGS ESTIMATION METHODOLOGY

The ASH controller senses DP temperature in the store and modulates power supplied to the heaters accordingly. DP inside a building is primarily dependent on the moisture content of outdoor ambient air and secondarily on moisture generated by internal factors such as occupants. In a previous research project (SCE 2006), indoor and outdoor temperature and humidity were recorded at a supermarket in 5 minute intervals for nearly 4 months. Figure 2 shows the relationship observed between the outdoor and indoor dew point temperatures.
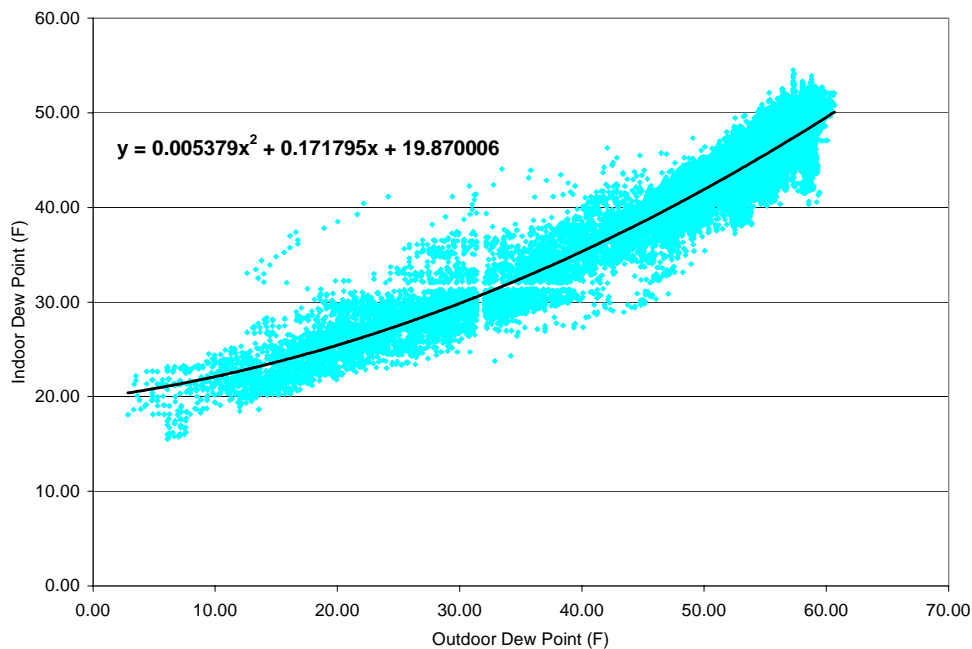

The chart shows Indoor Dew Point (F) on the y-axis (0.00 to 60.00) versus Outdoor Dew Point (F) on the x-axis (0.00 to 70.00) with the fitted equation:

$$y = 0.005379x^2 + 0.171795x + 19.870006$$

Figure 2: Indoor and Outdoor Dew Point at a Supermarket in Fullerton, CA. (Oct. 2005 – Jan. 2006, 5-minute data)

Because the outdoor dew point varies between climate zones, weather data from each climate zone must be analyzed to obtain a dew point profile. Using the correlation from Figure 2, a profile of the indoor dew point can be estimated, which allows modeling of ASH operation for an entire year for each climate zone.

Weather data for each of the 8 SCE CTZs was obtained from archived SCE weather station data for the cities listed in Table 11. These files contain 15-minute reads of

temperature and relative humidity for 2006. Analysis was conducted to calculate ASH power as a function of DP for each 15-minute reading. Additionally, secondary effects on compressor power and energy were calculated for each 15-minute interval.

Table 11: SCE Climate Zone and Corresponding Weather Stations Used in Analysis

Climate Zone	SCE Weather Station
6	El Segundo
8	Santa Ana
9	Rosemead
10	Rialto
13	Tulare
14	Ridgecrest
15	Cathedral City
16	Mammoth Lakes

The following calculations were carried out for each 15-minute reading:

1. Convert DB temperatures from °F to Rankine (R):

   R = °F + 459.67

2. For DB < 32 °F, using following equation to calculate water vapor saturation pressure (P_{ws} in psia):[1]

$$P_{ws} = \exp\left[\left(\frac{C_1}{R}\right) + C_2 + (C_3 * R) + (C_4 * R^2) + (C_5 * R^3) + (C_6 * R^4) + (C_7 * \ln(R))\right]$$

   where

   R = dry-bulb temperature in Rankine

   $C_1 = -1.0214165 \, E + 04$

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[1] ASHRAE 2005. Fundamentals Handbook. Atlanta, Georgia, p. 6.2

$C_2 = -4.8932428 \text{ E} + 00$

$C_3 = -5.3765794 \text{ E} - 03$

$C_4 = 1.9202377 \text{ E} - 07$

$C_5 = 3.5575832 \text{ E} - 10$

$C_6 = -9.0344688 \text{ E} - 14$

$C_7 = 4.1635019 \text{ E} + 00$

If $DB > 32^oF$, use the following equation to calculate water vapor saturation pressure (P_{ws} in psia):

$$P_{ws} = \exp\left[\left(\frac{C_8}{R}\right) + C_9 + (C_{10} * R) + (C_{11} * R^2) + (C_{12} * R^3) + (C_{13} * \ln(R))\right]$$

where

R = dry-bulb temperature in Rankine

$C_8 = -1.0440397 \text{ E} + 04$

$C_9 = -1.1294650 \text{ E} + 01$

$C_{10} = -2.7022355 \text{ E} - 02$

$C_{11} = 1.2890360 \text{ E} - 05$

$C_{12} = 2.4780681 \text{ E} - 09$

$C_{13} = 6.5459673$

3. Partial pressure of water vapor (p_w in psia) is the product of relative humidity and water vapor saturation pressure:[1]

$$p_w = RH \times p_{ws}$$

where

RH = relative humidity, %

---

[1] ASHRAE 2005. *Fundamentals Handbook*. Atlanta, Georgia, p. 6.12

4. To find outdoor dew point ($t_{d\text{-}out}$), between dew points of 32 and 200°F use:[1]

$$t_{d\text{-}out} = C_{14} + C_{15}\alpha + C_{16}\alpha^2 + C_{17}\alpha^3 + C_{18}(p_w)^{0.1984}$$

below 32°F use:

$$t_{d\text{-}out} = 90.12 + 26.412\alpha + 0.8927\alpha^2$$

where

$\alpha = \ln p_w$

C14 = 100.45

C15 = 33.193

C16 = 2.319

C17 = 0.17074

C18 = 1.2063

5. Indoor dew point ($t_{d\text{-}in}$) is related to outdoor dew point according to the equation in Figure 2.

$$t_{d\text{-}in} = 0.005379 \times t_{d\text{-}out}^2 + 0.171795 \times t_{d\text{-}out} + 19.870006$$

Direct ASH Power

6. Typical indoor humidity-based ASH controllers are programmed with two setpoints, All ON and All OFF. All ON refers to the high end of the humidity scale where heaters must operate all the time (100% duty cycle). The All OFF setpoint refers to the low end of the humidity scale where heaters are not needed (0% duty cycle). These setpoints can usually be changed based on the requirements of a particular store location. However, it is assumed that common practice is to set All OFF at 42.89°F DP (35%RH) and All ON at 52.87°F DP (50%RH) for a typical 72°F supermarket. Between these values, the ASH duty cycle changes proportionally:

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[1] ASHRAE 2005. *Fundamentals Handbook.* Atlanta, Georgia,, p. 6.13

$$ASH\,ON\% = \frac{DP_{meas} - All\,OFF\,Set\,Point}{All\,ON\,Set\,Point - All\,OFF\,Set\,Point}$$

where

DP_{meas} = measured DP temperature inside the sales area

Note: The controller only changes run time via the number of ASH pulses per unit of time, while instantaneous electric demand does not change.

7. In order to obtain the annual operating profile of ASH, calculations 1 through 6 were carried out for each 15-minute reading using each SCE weather station data for 2006. This process was repeated for all 8 SCE climate zones. Figure 3 shows the July ASH operation profile for CTZ 8. Table 12 gives the resulting duty cycle (% of hours ASH are on) for each of the eight SCE CTZs.
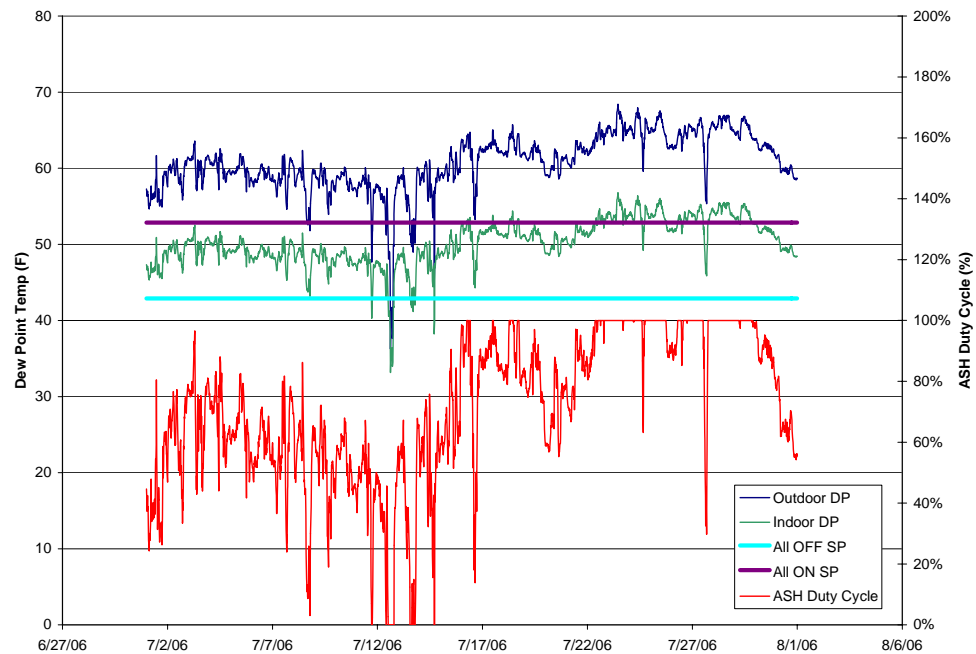

Figure 3: Outdoor DP, Indoor DP and ASH Operating Profile for CTZ 8 (based on July 2006 data)

Table 12: Annual ASH Run Time and Duty Cycle for 8 SCE CTZs

Climate Zone	SCE Weather Station	ASH Annual Run Time (hrs)	ASHDuty Cycle
6	El Segundo	2790	31.85%
8	Santa Ana	1455	16.61%
9	Rosemead	1840	21.00%
10	Rialto	1025	11.71%
13	Tulare	1139	13.01%
14	Ridgecrest	43	0.49%
15	Cathedral City	966	11.03%
16	Mammoth Lakes	1	0.01%

8. Because the controller only changes the run-time of the heaters, instantaneous ASH power (kW_{ASH}) as a resistive load remains constant at:

$$kW_{ASH} = (0.37 \text{ A/ft})(115 \text{ V}) = 0.04255 \text{ kW/ft}$$

9. ASH energy consumption for each 15-minute period is a product of power and run-time.

$$kWh_{ASH-15} = kW_{ASH} \times \text{ASH ON\%} \times \frac{15 \text{ min}}{60 \text{ min}}$$

10. Total annual ASH energy consumption is the sum of all 15-minute consumption values.

$$kWh_{ASH} = \sum kWh_{ASH-15}$$

Interactive Effects

11. Based on the results from prior SCE research, it is assumed that 35% of sensible heat generated by the ASH ends up as a cooling load (Q_{ASH}) inside the case. (SCE 1999) The cooling load contribution from ASH can be given by:

$$Q_{ASH} = 0.35 \times kW_{ASH} \times \frac{3413\,\text{Btu/hr}}{1\,\text{kW}} \times ASH\,ON\%$$

Once the ASH cooling load was determined, the following steps were followed to determine the compressor power requirements and energy savings. Because ASHs are present on many different types of equipment with various refrigeration system configurations, compressor analysis is limited to the cooling load imposed by the ASH, not the total cooling load of a particular display case or walk-in. The compressor power requirements are based on calculated cooling load and energy-efficiency ratios (EER) obtained from manufacturers' data.

12. Determine the saturated condensing temperature (SCT)

For medium temperature (MT): $SCT = DB_{adj} + 15$

For low temperature (LT): $SCT = DB_{adj} + 10$

where

DB_{adj} = dry-bulb temperature (°F) of ambient or adjacent space where the compressor/condensing units reside. Defaults are based on climate zone design values in Table 13.

Table 13: Climate Zone Design Dry Bulb Temperature and Representative City

CTZ	Description	Representative City	Summer Design Dry Bulb (ASHRAE Climatic Region X Data, 0.5% Column) [°F]
1	North Coast	Arcata	69
2	North Coast	Santa Rosa	96
3	North Coast	Oakland	89
4	North Coast	Sunnyvale	88
5	North Coast	Santa Maria	83
6	South Coast	Los Angeles	84
7	South Coast	San Diego	83
8	South Coast	El Toro	89
9	South Inland	Pasadena	94
10	South Inland	Riverside	100
11	Central Valley	Red Bluff	104
12	Central Valley	Sacramento	100
13	Central Valley	Fresno	101
14	Desert	China Lake	108
15	Desert	El Centro	111
16	Mountain	Mount Shasta	89

Source: ASHRAE 1982. Climatic Data for Region X.

13. Determine the EER for both MT and LT applications
    a. Compressor performance curves were obtained from a review of manufacturer data for reciprocating compressors as a function of SCT, cooling load, and cooling capacity of compressor.
    b. Part-load ratio (PLR):

    It is the ratio of total cooling load (from Cooling Load Calculation Section) to compressor capacity. It indicates the percentage of compressor capacity needed to remove the total cooling load. It is calculated by following equation:

    $$PLR = \frac{Q_{cooling}}{Q_{capacity}}$$

    where,

    PLR $\qquad$ = Part Load Ratio

    $Q_{coolinge}$ $\quad$ = cooling load

    $Q_{capacity}$ $\qquad$ = total compressor capacity

NOTE: Compressor capacity is determined by multiplying baseline cooling load by compressor over-sizing factor of 15%:

$$Q_{capacity} = Q_{cooling} * 1.15$$

Therefore, $PLR = 1/1.15 = 0.87$

Because the ASH cooling load is very small in relation to the cooling load of a typical display case, to simplify the analysis it is assumed that PLR remains constant for the post-retrofit condition.

c. For medium temperature compressors, the following equation is used to determine the EER_{MT} (Btu/hr/watts)

$$EER_{MT} = a + (b * SCT) + (c * PLR) + (d * SCT^2) + (e * PLR^2) + (f * SCT * PLR) + (g * SCT^3) + (h * PLR^3) + (i * SCT * PLR^2) + (j * SCT^2 * PLR)$$

where,

a	$= 3.75346018700468$
b	$= -0.049642253137389$
c	$= 29.4589834935596$
d	$= 0.000342066982768282$
e	$= -11.7705583766926$
f	$= -0.212941092717051$
g	$= -1.46606221890819E\text{-}06$
h	$= 6.80170133906075$
i	$= -0.020187240339536$
j	$= 0.000657941213335828$
PLR	$= PLR_{baseline}$ or $PLR_{post\text{-}retrofit}$

d. For low temperature compressors, the following equation is used to determine the EER_{LT} (Btu/hr/watts)

$$EER_{LT} = a + (b * SCT) + (c * PLR) + (d * SCT^2) + (e * PLR^2) + (f * SCT * PLR) + (g * SCT^3) + (h * PLR^3) + (i * SCT * PLR^2) + (j * SCT^2 * PLR)$$

where,

a	= 9.86650982829017
b	= -0.230356886617629
c	= 22.905553824974
d	= 0.00218892905109218
e	= -2.48866737934442
f	= -0.248051519588758
g	= -7.57495453950879E-06
h	= 2.03606248623924
i	= -0.0214774331896676
j	= 0.000938305518020252
PLR	= $PLR_{baseline}$ or $PLR_{post-retrofit}$

14. Power used by the ASH and compressor (kW), and power savings (ΔkW)
    a. Power used by the compressor to remove heat imposed by the ASH is determined based on calculated cooling load and EER, as outlined below.

$$kW_{comp\text{-}15} = \frac{Q_{ASH}}{(EER * 1000)}$$

where

$kW_{comp\text{-}15}$ = compressor kW as a function of baseline load for 15-minute interval

Q_{ASH} = ASH cooling load

EER = EER for LT and/or MT (Btu/hr/watts)

    b. Once the compressor power requirements for baseline and post-retrofit scenarios are determined, power savings (ΔkW_{comp}) can be calculated based on the average compressor power.

$$\Delta kW_{comp} = kW_{comp\text{-}baseline\text{-}average} - kW_{comp\text{-}post\text{-}retrofit\text{-}average}$$

c. Total power savings would include both direct ASH and compressor savings. Because the controller does not alter the instantaneous demand of the ASH, no direct peak demand savings are claimed.

$$\Delta kW_{Total} = \Delta kW_{comp} + \Delta kW_{ASH} \Rightarrow \Delta kW_{comp}$$

where

$$\Delta kW_{ASH} = 0$$

15. Energy usage (kWh) and savings (ΔkWh):
   a. Compressor energy consumption for each 15-minute period of weather data is calculated based on compressor power and the length of each analysis time period (15 minutes).

$$kWh_{comp\text{-}15} = kW_{comp-15} \times \frac{15\,min}{60\,min}$$

where

$kWh_{comp\text{-}15}$ = compressor kWh for 15-minute interval

   b. Annual energy used by the compressor is the sum of all calculated 15-minute compressor energy values.

$$kWh_{comp} = \sum kWh_{comp-15}$$

where

kWh_{comp} = annual compressor energy

   c. Once the annual compressor energy for baseline and post-retrofit scenarios are determined, annual compressor energy savings (ΔkWh_{comp}) can be calculated.

$$\Delta kWh_{comp} = kWh_{comp\text{-}baseline} - kWh_{comp\text{-}post\text{-}retrofit}$$

d. Total energy savings (kWh$_{Total}$) combines compressor and ASH savings

$$\Delta kWh_{Total} = \Delta kWh_{comp} + (kWh_{ASH\text{-}baseline} - kWh_{ASH\text{-}post\text{-}retrofit})$$

e. The resulting demand (ΔkW_{Total}) and energy (ΔkWh_{Total}) savings are expressed per unit of measure, because the baseline ASH power was per linear foot of display case.

Table 14 summarizes the demand and energy savings and ASH duty cycle of MT and LT applications for all 8 SCE climate zones. Demand savings values are based solely on compressor savings while energy savings values include compressor and direct ASH savings. Compressor power savings are lower for milder climate zones, partly due to increased compressor efficiency at lower ambient temperatures.

Table 14: Demand and Energy Savings for Grocery Application for All 8 SCE Climate Zones

Climate Zone	Representative City and Temperature (Summer Design Dry-Bulb, ASHRAE Climatic Region X Data, 0.5% Column)	Measure Unit	MT ASH Controller		LT ASH Controller	
			Demand Savings (ΔkW/unit)	Energy Savings (ΔkWh/unit)	Demand Savings (ΔkW/unit)	Energy Savings (ΔkWh/unit)
6	Los Angeles - 84°F	/Ln.ft.	0.004486	293	0.005731	304
8	El Toro - 89°F	/Ln.ft.	0.005909	363	0.007603	377
9	Pasadena - 94°F	/Ln.ft.	0.006041	347	0.007808	363
10	Riverside - 100°F	/Ln.ft.	0.007425	394	0.009619	413
13	Fresno - 101°F	/Ln.ft.	0.007436	389	0.009634	409
14	China Lake - 108°F	/Ln.ft.	0.009569	455	0.012384	479
15	El Centro - 111°F	/Ln.ft.	0.009020	411	0.011661	434
16	Mount Shasta - 89°F	/Ln.ft.	0.007085	435	0.009116	453

The "Refrigeration" load shape should be used for this measure because ASHs operate all the time, like a refrigeration system. The compressor operates continuously to maintain the proper temperature and the ASH operates at 100% duty cycle, so energy use occurs throughout the day and night.

The "Refrigeration" load shape should also be used for this measure because for the majority of the time, the ASH controller will reduce the power demands consistently. Therefore, the difference load shape should also be "Refrigeration".