**Work Paper PGE3PREF126**

**ECM for Walk-In Evaporator with Fan Controller**

**Revision # 2**

**Pacific Gas & Electric Company**

**Customer Energy Solutions**

**ECM for Walk-In Evaporator with Fan Controller**

**Measure Codes HA12, HA29, HA30, HA31, HA32, HA46**

***DNV GL BEST***

# At-a-Glance Summary

|  |  |
| --- | --- |
| **Applicable Measure Codes:** | **Use “At a Glance” list (next page);** |
| **Measure Description:** | Replacement of an continuously running standard efficiency shaded pole (SHP) or Permanent Split Capacitor (PSC) evaporator fan motor with an Electronically Commutated Motor (ECM) with fan cycling controls |
| **Energy Impact Common Units:** | Per motor / Each |
| **Base Case Description:** | Source: DNV GL  Shaded pole or permanent split capacitor motor of various sizes. |
| **Base Case Energy Consumption:** | Source: DNV GL  Dependent on motor size |
| **Measure Energy Consumption:** | Source: DNV GL  Dependent on motor size |
| **Energy Savings**  **(Base Case – Measure):** | Source: DNV GL  Dependent on motor size |
| **Costs Common Units:** | Per motor / Each |
| **Base Case Equipment Cost ($/unit):** | N/A |
| **Measure Equipment Cost ($/unit):** | Source: DEER 2011  $466.34 |
| **Gross Measure Cost ($/unit)** | Source: DEER 2011  $466.34 |
| **Measure Incremental Cost ($/unit):** | Source: DEER 2011  $466.34 |
| **Effective Useful Life (years):** | Source: DEER 2014  15 years |
| **Measure Application Type:** | ER |
| **Net-to-Gross Ratios:** | Source: DEER 2011  DI: 0.85  I: 0.60 |
| **Important Comments:** |  |

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| --- |
|  |
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# Document Revision History

|  |  |  |  |
| --- | --- | --- | --- |
| **Revision #** | **Date** | **Section-by-Section Description of Revisions** | **Author (Company)** |
| **Revision 0** | **6/7/2012** | **PGE3PREF126.0 – ECM for Walk-In Evaporator Fans with Fan Controller 3P** | **Joseph Flores (DNV KEMA)** |
| **Revision 0** | **8/29/12** | **Building Type and Building Vintage changed to “Any”. Unit Definition modified to “Each”. Added Incentive delivery option.** | **Arun Kumar Balaji (PG&E)** |
| **Revision 1** | **5/15/2014** | * **Section 2 - Revised calculation for EFLH and incorporated 2014 DEER weather data in calculation for Weather Factor.** * **Added category for 1, 1.5, and 2 HP motor sizes** * **Savings now dependent on case type (Medium Temp vs Low)** | **Joseph Flores, DNV GL** |
| **Revision 2** | **3/30/2016** | **Update to Ex Ante Format** | **Jim Wyatt (PG&E)** |

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# Section 1. General Measure & Baseline Data

## 1.1 Product Measure Description & Background

***Catalog Description –***

These measures are applicable to the replacement of a continuously running standard efficiency shaded pole (SHP) or permanent split capacitor (PSC) evaporator fan motor with an Electronically Commutated Motor (ECM) with fan cycling controls.

***Program Restrictions and Guidelines***

* This is an Early Retirement measure.
* Installation address must have a PG&E commercial electric account

***Terms and Conditions:***

* Controller must reduce the airflow of evaporator fans when the compressor cycles off and no refrigerant is flowing through the evaporator
* The retrofit must consist of the replacement of an existing standard efficiency shaded-pole or permanent split capacitor refrigerator evaporator fan motor in refrigerated display and walk-in cases
* This measure should not be installed in the following cases
  + If the compressor is running with a high-duty cycle
  + If the evaporator fan does not run at full-speed all the time
  + Evaporator does not use off-cycle or time-off to defrost

***Market Applicability:***

These measures are applicable to refrigerated display cases and walk-in coolers and freezers that are found in a variety of building types: schools, groceries, restaurants, lodging, hospitals, and others. However, these measures are predominantly implemented in grocery stores, small convenience markets, restaurants, and refrigerated warehouses.

## 1.2 Product Technical Description

An evaporator fan controller is defined as a device or system that reduces airflow across an evaporator in walk-in coolers and freezers when there is no refrigerant flow through the evaporator i.e., when the compressor is in an off-cycle; or when the controller receives a signal form the thermostat to stop the flow of refrigerant, i.e., turns the compressor off. The energy savings is typically accomplished by reducing the speed of the fan motors by at least 75% during the compressor off-cycle. The controller reduces air flow rather than turning fans off completely when the compressor is not operating because a minimum airflow may be required to provide defrosting and prevent the air in the cooler from stratifying into layers of higher and lower temperature.

A typical evaporator unit in a walk-in cooler contains one or more small fans with fractional horsepower motors that are operating continuously. A fan controller saves energy by reducing fan usage and by reducing the refrigeration load resulting from the fan's waste heat. When combined with a high efficiency EC Motor, potential energy savings can be very substantial.

## 1.3 Measure Application Type

The DEER Ex Ante Database Format defines the terms as follows:

Table 1 Measure Application Type[[1]](#endnote-1)

*Identifies the measure application type in the Measure Implementation table in DEER2014.*

|  |  |  |
| --- | --- | --- |
| **Code** | **Description** | **Comment** |
| ER | Early retirement | *Measure is more efficient than code/std; Dual baseline, full measure costs required* |
| ROB | Replace on Burnout | *Single baseline (above code), incremental or full costs* |
| NC | New Construction | *Single baseline (above code), incremental or full costs* |
| REA | Retrofit Add On | *Single baseline (above pre-existing), full measure costs required* |

EC Motor retrofits are considered an Early Retirement (ER) measure. Title 20, Title 24, and Federal codes are only applicable to newly constructed refrigerated cases, or in the case of Title 24, for the replacement of entire evaporator coil units. This measure applies specifically to the evaporator fan motor, which can be retrofitted without code triggering changes to the evaporator fan coil, refrigeration system, or the refrigerated case.

## 1.4 Product Base Case and Measure Case Data

## 1.4.1 DEER Base Case and Measure Case Information

The DEER 2011 data does not contain all of the appropriate information for these measures. Demand, Electric, or Gas energy savings are not included for these measures in DEER 2011. However, values for equipment costs, Net to Gross, and Equipment Useful Life are provided in the applicable DEER 2011 (or 2008) documentation. Information is displayed below where applicable.

**Base Case Costs and Measure Case Costs**

**Costs DEER Version and Impact Ids**

* The [Base Case / Measure Case / Incremental] Costs were downloaded from DEER directly. Pricing is a combination of “Evaporator Fan Controller for Walk-In Coolers” and “High Efficiency Evaporator Fan Motor, Electronically Commutated (SHP to ECM and match the intended measures for climate zones and building types and vintages.

|  |  |  |  |  |  |  |  |
| --- | --- | --- | --- | --- | --- | --- | --- |
|  |  |  | **Costs ($)** | | |  |  |
| **Building type** | **Bldg Vintage** | **Climate Zone** | **Base Case** | **Measure Case** | **IMC** | **DEER Version** | **Impact IDs** |
| **Any** | **Any** | **Any** | **N/A** | **466.34** | **466.34** | **2011** | **N/A** |

**Net-to-Gross Assumption:**

Table 2 below summarizes all applicable DEER based Net-to-Gross ratios for programs that may be used by this measure.

Table 2 DEER Net-to-Gross Ratios

|  |  |  |  |
| --- | --- | --- | --- |
|  |  | **DEER Spreadsheet** | |
| Program Approach | NTG | File name | Cell Number |
| Direct Install | 0.85 | DEER2011\_NTGR\_2012-05-16 | T59 |
| Incentive (Com-Default>2yrs) | 0.60 | DEER2011\_NTGR\_2012-05-16 | T50 |

**Effective Useful Life / Remaining Useful Life:**

**Effective Useful Life: DEER Version and Impact IDs**

* The Effective Useful Life estimates were downloaded directly from 2014 DEER, they match the intended measures for climate zones and building types and vintages. The lower EUL for High Efficiency Evaporator Fan Motors is used for this measure.

|  |  |  |  |  |  |  |
| --- | --- | --- | --- | --- | --- | --- |
| **Building type** | **Bldg Vintage** | **Climate Zone** | **EUL (yrs)** | **RUL (yrs)** | **DEER Version** | **Impact IDs** |
| **Any** | **Any** | **Any** | **15** | **5** | **2014** | **N/A** |

**In-service rate/first year installation rate**:

* The DEER data do not contain (not) the appropriate information for this (these) measure(s).

## 1.4.2 Codes & Standards Requirements Base Case and Measure Information

***Title 20:*** These measures fall under Title 20 of the California Energy Regulations. Starting January 1, 2009, walk-in coolers and freezers with evaporator fan motors of under one horsepower or less than 460 volts are required have ECM evaporator fan motors. Retrofit measures are applicable for units that pre-date this code change.

***Walk-ins:*** California’s Title 20 Appliance Efficiency Standards do regulate the efficiency of evaporator fans in newly constructed walk-in boxes, but not in motor-retrofit/replacement applications[[2]](#footnote-1). Debate on potential federal legislation for walk-in regulations is ongoing and the current proposed language specifically addresses ECM motors.

|  |  |  |  |  |
| --- | --- | --- | --- | --- |
| **Title 20 Std. Description** | **Base or Measure Case** | **Value** | **Units** | **Code Source or Reference** |
| ***Evaporator fan motor requirement for Walk-In Cooler/Freezers manufactured on or after January 1, 2009*** | *Base Case* | *Electronically Commutated Motor* | *motors* | Title 20 Appliance Efficiency Regulations, CEC-140 2013-002, Page 232, August 2013 |

***Title 24:*** California’s 2013 Title 24 Building Energy Efficiency Standards regulate evaporator motors for refrigerated warehouses that are greater or equal to 3,000 square feet. Newly installed evaporator units that utilize single phase fan motors, which are both less than 1 HP and less than 460 volts, must contain electronically commutated motors or motors with a minimum efficiency of 70 percent[[3]](#footnote-2).

New evaporator units served by either a suction group with multiple compressors, or by a single compressor with variable capacity capability, are required to be variable speed and controlled in response to space temperature or humidity. Exceptions to this requirement include the following:

* Addition, alteration, or replacement of less than all of the evaporators in an existing refrigerated space that does not have speed-controlled evaporators
* Coolers within refrigerated warehouses that maintain a Controlled Atmosphere for which a licensed engineer has certified that the stored product requires constant operation at 100% of design airflow
* Areas within refrigerated warehouses that are designed exclusively for quick chilling/freezing of products

New evaporator units served by a single compressor without variable capacity are required to utilize controls to reduce airflow by at least 40% for at least 75% of the time when the compressor is not running. Areas within refrigerated warehouses that are designed exclusively for quick chilling/freezing of products are exempt from this requirement.

Retrofit measures are applicable for units that do not trigger adherence to 2013 Title 24. This includes retrofits and repairs that do not increase the preexisting energy consumption of the repaired component, system, or equipment. Please refer to *2013 Building Energy Efficiency Standards* and the *2013 Nonresidential Compliance Manual[[4]](#footnote-3)* for more details.

|  |  |  |  |  |
| --- | --- | --- | --- | --- |
| **Title 24 Std. Description** | **Base or Measure Case** | **Value** | **Units** | **Code Source or Reference** |
| ***Evaporator fan motor requirement for Walk-In Cooler/Freezers installed in Refrigerated Warehouses equal or greater than 3,000 square feet*** | *Base Case* | *Electronically Commutated Motor* | *motors* | 2013 Building Energy Efficiency Standards, Title 24, Part 6, Section 120.6(a)3 |

***Federal Standards:*** EPACT 2005[[5]](#footnote-4) requires that walk-in coolers or freezers manufactured on or after January1, 2009 and have evaporator fan motors under 1 horsepower and less than 460 volts, us either 3-phase motors or electronically commutated motors.

|  |  |  |  |  |
| --- | --- | --- | --- | --- |
| **Federal Std. Description** | **Base or Measure Case** | **Value** | **Units** | **Code Source or Reference** |
| ***Evaporator fan motor requirement for Walk-In Cooler/Freezers manufactured on or after January 1, 2009*** | *Base Case* | *Electronically Commutated Motor* | *motors* | EPACT 2005, Subpart R, Section 431.306. <http://www.ecfr.gov/cgi-bin/text-idx?SID=8f2b0bd78ff466ba84889adff20227db&node=10:3.0.1.4.19&rgn=div5#10:3.0.1.4.19.18> |

## 1.4.3 EM&V, Market Potential, and Other Studies – Base Case and Measure Case Information

***1.4.3.1 Study #1– Food Service Technology Center – GE ECM Evaporator Fan Motor Energy Monitoring******[[6]](#endnote-2)***

A study was conducted Food Service Technology Center (FSTC) in San Ramon, California which quantified the energy savings when ECM fans are used in walk-in evaporators. Researchers at the FSTC installed and monitored two GE ECM motors in a walk-in freezer to document the energy savings potential.

The FSTC’s laboratory walk-in freezer evaporator unit originally came equipped with two 1/15 HP shaded pole motors. Pre-retrofit motor power was measured for one day, along with rotational speed and airflow. Then the ECM units were installed with the original fan blades. Fan speed and airflow were measured post retrofit to ensure that they were the same as the basecase. The combined energy consumption of both ECM fans was then monitored for two weeks.

The initial measurement of the two standard shaded pole motors indicated 135.5 watts each. Measurement of the replacement ECMs showed an average input power of 44 watts each. This represented a savings of 91.5 watts each (0.0915 kW), with annual energy savings of approximately 802 kWh. There is also the added benefit of reduced internal heat load within the refrigerated space. With this reduced internal heat load taken into account, the FSTC study arrived at a combined daily kWh energy savings of 3.294 kWh per fan. Assuming continuous operation, the annual kWh savings is 1202 kWh per fan.

**Energy Savings Assumption (ΔW, ΔTherms):**

* Electric savings were taken from FSTC Study, however they differ from the measures by;
  + - FSTC study was only for 1/15 HP motors
    - Savings as a result of reduced internal heat load were not quantified in the FSTC studied. Combined calculations assumed a system COP of 2.

|  |  |  |  |  |  |
| --- | --- | --- | --- | --- | --- |
| **Building type** | **Bldg Vintage** | **Climate Zone** | **Electric Savings Watts** | **Study units** | **Specific study reference** |
| **ANY** | **ANY** | **ANY** | **91.5** | **Per fan** |  |

## 1.4.4 Assumptions and Calculations from other sources—Base and Measure Cases

**Energy Savings Assumption (ΔW):**

* Wattage measurements for both baseline and retrofit motors are provided by Energy Control Equipment, which specializes in the retrofit of refrigeration evaporator shaded pole and permanent split capacitor motors. This data was collected for the purpose of estimating savings for ECM and Evaporator Fan Controller installations. These wattages were used in the Energy Savings Calculations described in Section 2.1.

Table 3 Baseline and EC Motor Wattages

|  |  |  |  |  |
| --- | --- | --- | --- | --- |
| **Motor Model** | **Motor Specification** | **SHP Motor Wattage** | **EC Motor Wattage, Hi Speed** | **EC Motor Wattage, Low Speed** |
| 2SB13 | 16 Watt / 120V | 91.85 | 48 | 3 |
| MA59 | 1/15 or 1/20 Hp | 186.35 | 72 | 5 |
| MA14215 | 1/5 Hp / 240V | 361.5 | 210 | 14 |
| MA14213 | 1/3 Hp / 240V | 563.35 | 349 | 24 |
| MA14212 | 1/2 Hp / 240V | 799.00 | 524 | 37 |
| MA14234 | 3/4 Hp / 240V | 1087.1 | 788 | 53 |
| MA14210 | 1 Hp / 240V | 1393.0 | 843 | 91 |
| N/A | 1.5 Hp / 240V | 1985 | 1205 | 139 |
| N/A | 2 Hp / 240V | 2415 | 1551 | 172 |

***1.5 Summary of Inputs for Savings Calculations***

The following table provides references to sections that document the inputs for calculation:

|  |  |  |  |  |  |
| --- | --- | --- | --- | --- | --- |
| **Input Variable** | **Variations** | **Base Case 1 Average Value** | **Base Case 2 Average Value** | **Measure Case Average Value** | **Reference Section** |
| **Electric Savings** |  |  |  | *Varies* | *1.4.4* |
| **Gas Savings** | N/A |  |  |  |  |
| **Hours of operation** | N/A |  |  |  |  |
| **Full Cost** | ER | $466.34 |  |  | 1.4.1 |
| **Incremental Cost** | ER | $466.34 |  |  | 1.4.1 |
| **EUL /RUL** | ER | 15 / 5 |  |  | 1.4.1 |
| **NTG** | One | 0.85 |  |  | 1.4.1 |
| **ISR** | Applies -- Yes / No |  |  |  |  |
| **TOU Factor** | *A/C projects only* | *N/A* |  |  | *N/A* |

# Section 2. Calculation Methods

Table 4 Baseline by Measure Application Type

|  |  |  |  |
| --- | --- | --- | --- |
| ****Measure Application Type**** | ****Measure Life Basis**** | ****First Baseline Period: Energy Savings Baseline**** | ****Second Baseline Period: Energy Savings Baseline**** |
| ***ER* (early retirement)** | **EUL** | Customer Average Baseline | Code Baseline |
| ***ROB* (replace-on-burnout)** | **EUL** | Code Baseline | N/A |
| ***NC* (new construction)** | **RUL/EUL-RUL** | Code Baseline | N/A |

Notes:

* For ROB measures, First Baseline is the baseline for the full EUL. There is no second baseline.
* For ER measures, First Baseline Period is the period for the RUL(remaining useful life),defined by the CPUC as RUL=1/3 EUL. Second baseline period for ER is Code baseline for the period EUL-RUL.

## 2.1 Electric Energy Savings Estimation Methodologies

* This measure is an Early Retirement measure. The following analysis is applicable to both the EUL and RUL Baseline Periods. Since EC Motors for Walk-In Evaporator fans are required under Federal and State codes only for new evaporator coils or refrigerated cases, or cases undergoing major renovations, many installations will continue in their existing condition for their entire rated EUL or longer.

DNV GL has developed a refrigeration savings calculator in SCCR – ECM -xxxx.xlsx[[7]](#footnote-5). The spreadsheet uses a cooling load calculation to calculate the refrigeration load of a typical refrigerated case, walk-in cooler or freezer found in convenience stores, grocery stores, or restaurants. The calculator is designed to be most applicable to convenience stores. It is not applicable for stand-alone display cases without a walk-in main door. Savings calculated in the spreadsheet are attributed to decreased cooling load and compressor usage.

Cooling load calculations are based on ASHRAE methodology[[8]](#footnote-6) for typical refrigeration loads. Details of the analysis are provided in a separate attachment and spreadsheets. The total cooling load of a refrigerated space requires the calculation of the following:

1. Transmission or conduction load
2. Anti-sweat heater (ASH) load
3. Internal load (load due to evaporator fan motors, lighting, and people)
4. Product load (product shelving and product pull-down load)
5. Infiltration load

Additional assumptions must be made regarding the air properties of the refrigerated and adjacent spaces, number of doors, door type, and door size. Current values are based on DNV GL field observations in California, SCE Workpaper assumptions[[9]](#footnote-7), and ADM evaluation results of gasket and strip curtain installations[[10]](#footnote-8). All assumptions and their source are documented in the spreadsheet.

Savings estimates for different measures can be calculated by adjusting these parameters and comparing the pre-retrofit and post-retrofit annual energy consumptions. The spreadsheet calculator contains the details. The calculator is set up for cooler walk-in, freezer walk-in, cooler reach-in, and freezer reach-in. The difference between the two in this document is that the reach-in is a walk-in with glass doors. Stand-alone refrigerated cases are not applicable to the calculator. The analysis adjustments per measure are discussed below.

**Calculator Shortcomings**

The calculator methodology is based on assumptions that require further research to validate. However based on the available information, they are satisfactory for calculating deemed savings. DNV GL believes this approach uses the most up to date data available and building up from the basics. Much of the calculator is based on the methodology and assumptions found in the SCE refrigeration workpapers[[11]](#footnote-9). The SCE methodology assumes that the system is comprised of a single reciprocating compressor and an air cooled condenser. Refrigeration system configurations vary widely depending on capacity and use. For example, many systems found at large commercial grocery stores are comprised of multiplex systems with water cooled condensers. There was also disagreement over the calculation methodology regarding the defrost heater internal load, which resulted in this internal load being omitted from the spreadsheet calculations[[12]](#footnote-10).

In addition, the methodology for determining the EER for both medium and low temperature applications uses SCE’s internal review of reciprocating compressor manufacturer performance curves to calculate EER. Their data and analysis is not available for review. Questions have arose about whether this data is applicable to different areas of the country, since these performance curves are dependent on saturated condensing temperature, cooling load, and the cooling capacity of the compressor. Further research is recommended to account for different types and how they would affect overall system efficiency and energy usage. Weather normalization analysis can be improved by using TMY3 8760 hourly weather data. However, only a simplified normalization is currently used.

Savings are quantified for the motor sizes outlined in Table 2 for all PG&E Climate Zones, utilizing weather data from 2014 DEER. Backup data for PGE Climate Zone and Weather Factors are provided in the backup documentation***[[13]](#endnote-3)***.

The following lists all the key assumptions:

1. Building Construction Data –assumptions in the following table (based on SCE workpapers):

Table 5 Refrigerator Construction Assumptions by Unit Type

|  |  |  |  |  |  |  |
| --- | --- | --- | --- | --- | --- | --- |
| **Parameter** | **Notation** | **Walk-in Cooler** | **Walk-in Freezer** | **Cooler with glass doors** | **Freezer with glass doors** | **Source** |
| Temp of surroundings | DBadj, °F | 70 | 70 | 70 | 70 | ADM |
| RH of surroundings | RHadj, % | 37.5 | 37.5 | 37.5 | 37.5 | ADM |
| Box temp | DBrefrig, °F | 40 | 0 | 40 | 0 | ADM |
| Box RH | RHrefrig, % | 78.3 | 63.5 | 78.3 | 63.5 | ADM |
| Walk-in Length | L, ft | 10 | 10 | 30 | 15 | Assumption |
| Walk-in Width | W, ft | 10 | 10 | 15 | 10 | Assumption |
| Walk-in Height | H, ft | 8 | 8 | 10 | 10 | Assumption |
| Panel thickness | D, in | 2.5 | 2.5 | 2.5 | 2.5 | SCE |
| Panel thermal conductivity | kpanel, Btu-in / hr-ft2-°F | 0.16 | 0.16 | 0.16 | 0.16 | SCE |
| Main door quantity |  | 1 | 1 | 1 | 1 | SCE |
| Main door Height | Hdoor, ft | 7 | 7 | 7 | 7 | SCE |
| Main door Width | Wdoor, ft | 3 | 3 | 3 | 3 | SCE |
| Glass door quantity |  | 0 | 0 | 8 | 4 | Assumption |
| Glass door Height | Hglass, ft | -- | -- | 4.25’ |  | SCE |
| Glass door Width | Wglass, ft | -- | -- | 2.5’ |  | SCE |
| Glass thickness | dglass, in | -- | -- | 1 | 1 | SCE |
| Glass thermal conductivity | kglass, Btu-in / hr-ft2-°F | -- | -- | 3.26 | 3.26 | SCE |
| Number of Evap Fan Motors |  | 2 | 2 | 6 | 6 | Assumption |

1. Compressor, condenser, and evaporator are the same for pre and post-retrofit (evaporator fan changes only for ECM measure).
2. Compressor (SCE workpaper):
   * Single reciprocating compressor
   * 15% compressor over-sizing factor
   * For weather factor analysis, assume sealed compressor
3. Condenser (SCE workpaper/engineering assumption):
   * Air cooled condenser
   * Condenser temperature difference (TD) of 20oF for medium-temperature applications (coolers)
   * Condenser TD of 15oF for low-temperature applications (freezers)
4. Evaporator (SCE workpaper):
   * Evaporator TD of 10oF for both temperature groups (coolers and freezers)
   * 135.5 W motor[[14]](#footnote-11) at 70% efficiency
5. Anti-sweat heater load (SCE workpaper):
   * 35% of heat dissipated due to connected electrical load contributes to the refrigeration load
   * For main door of walk-in freezers, connected electrical load is 0.30 kW per door
   * No ASH for main door of walk-in coolers
   * For reach-in glass doors of walk-in coolers, connected electrical load is 0.045 kW per door
6. Internal loads (SCE workpaper):
   * Evaporator fan motors (see above)
   * Lighting intensity of 0.75 watts per square-foot of walk-ins
   * People, one person inside the walk-ins
   * Electric defrost (only for walk-in freezers) is currently not considered in the anlaysis
7. Product load (SCE workpaper):
   * 40% space utilization factor
   * Product density (ρ) of 2 lb/ft3 for coolers and 1 lb/ft3 for freezers
   * Product specific heat (Cp) of 0.45 Btu/lb-°F
   * Post-defrost product temperature rise (ΔTpost-defrost) of 3°F
   * Temperature difference of 5oF between initial and final product temperature when shelving (ΔTshelving)
   * Time required to lower the product temp. after shelving, 60 min
   * Defrost frequency, 4 defrosts per day
8. Infiltration load (ADM):
   * Proportion of doorway open time (Dt),time door simply stands open (minutes/day): 72 (cooler), 50 (freezer), 5 (glass door)
   * Doorway flow factor (Df): 0.552 (cooler), 0.643 (freezer)
   * Effectiveness against infiltration (E): 0.40
   * All main doors and glass doors are assumed to be tightly sealed
   * No air leakage through the walk-in panels

**Calculation Analysis**

The following are the guidelines used to calculate the potential savings of refrigeration measures.

1. Cooling load analysis:

The cooling load should change for existing and retrofit systems. The following lists the necessary variables for calculating cooling load (the sum of the individual loads) of walk-ins:

* 1. Air properties of refrigerated and adjacent spaces
  2. Transmission or conduction load
  3. Anti-sweat heater (ASH) load
  4. Internal load – includes evaporator fans, lighting, and people (electric defrost could be included in this category)
  5. Product load
  6. Infiltration load

1. EER and demand of the compressor:
   1. Calculate using saturated condensing temperature (SCT), cooling load and compressor capacity for both pre and post retrofit.
   2. Calculate power (kW) requirements by using EER and cooling load.
2. Equivalent-full-load-hours (EFLH) of operation:
   1. EFLH is determined by using annual available operation hours (8,760) and overall duty cycle factor.
   2. Overall duty cycle factor is determined by taking into account compressor over-sizing factor (or part-load ratio), defrost periods, and weather.
3. Compressor energy usage:
   1. Determine energy consumption by using power requirements and EFLH.

**Cooling Load Calculations**

This section discusses the steps in calculating the total cooling load of the refrigerated spaces. These steps are repeated for each measure’s baseline and retrofit assumptions.

1. Air properties of refrigerated and adjacent spaces
2. Transmission or conduction load
3. Anti-sweat heater (ASH) load
4. Internal load (load due to evaporator fan motors, lighting, people, and defrost)
5. Product load (product shelving and product pull-down load)
6. Infiltration load
7. Total cooling load

**Step 1: Air properties of refrigerated and adjacent spaces**

Calculate air density, enthalpy and absolute humidity ratio based on assumed dry-bulb and relative humidity for the surroundings and inside the walk-in. These are done by using Excel plug-in for psychometric chart (by HillsBrother).

**Step 2: Transmission or conduction load**

Calculate transmission or conduction load through the walls of the walk-in and glass doors.

1. Calculate overall heat transfer coefficient (U) for conduction only:

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where

∑RTotal = Total thermal resistance, Btu/hr-ft2-°F

Rcond = thermal resistance for conduction, Btu/hr-ft2-°F

k = thermal conductivity, Btu-in/hr-ft2-°F

d = thickness, in

1. Calculate total surface area (Atotal)[[15]](#footnote-12) with no glass doors:



Calculate total surface area (Atotal) with glass doors:





where

n = number of glass doors

1. Calculate temperature differential (ΔT):



where

DBrefrig = dry-bulb inside the refrigerated zone (box temperature), oF

DBadj = dry-bulb of adjacent space where the walk-in reside, oF

1. Calculate the conduction or transmission load (Qcond):23

With no glass doors:



With glass doors:







where

U = overall heat transfer coefficient, Btu/hr-ft2-oF

A = total surface area of insulated panels or doors, ft2

ΔT = temperature differential between refrigerated and adjacent zones, oF

**Step 3: Anti-sweat heater (ASH) load**

Main doors and glass doors are typically equipped with anti-sweat heaters (ASH). The heat dissipated due to the connected electrical load of ASH is also considered in the load.

Calculate ASH load for the main or glass doors (QASH)





Where

n = number of glass doors

kW = ASH power for door, kW

KASH = 35%, heat dissipated to refrigeration system due to electrical load

Conv. Factor: 3,413 Btu/hr = 1 kW



**Step 4: Internal Load**

The internal load in refrigerated spaces is assumed to be evaporator fan motors, lighting, people and defrost.

1. Load due to evaporator fan motors for walk-ins (Qevap-fans):



where

n = number of fans

hp = motor horsepower, hp

η = motor efficiency, %

Conv. Factor: 0.7457 kW = 1 hp

1. Load due to lighting for walk-ins (Qlights):



where

ρlight = light intensity, watts/ft2

L = length of the walk-ins, ft

W = width of the walk-ins, ft

Conv. Factor: 3,413 Btu/hr = 1 kW

For the retrofit case, loads are multiplied by the compressor % runtime (describe below) to account for their low-speed operation.

1. Load due to people inside the walk-in (Qpeople-total):

Load due to people is dependent on room dry-bulb temperature and comprised of two components, latent and sensible:

Qpeople-total = Qpeople-sensible + Qpeople-latent



inside the refrigerated zone (box temperature), oF

Qpeople-sensible = 61% Qpeople-total[[16]](#footnote-13)

**Step 5: Product Load**

This section describes the calculation of the load due to product shelving and post-defrost pull down load.

1. The product load (Qproduct-load) is calculated using the following equation:



where

WFprod = weight flow of product, lb/hr (see discussion below)

Cp = product specific heat, Btu/lb-°F

ΔT = temperature rise due to product shelving or defrost periods, °F

* 1. Determine the total product volume (Vprod)



where

Vprod = product volume, ft3

kSU = space utilization factor, %

* 1. Determine the total product mass (mprod)



where

mprod = mass of product, lb

ρprod = product density, lb/ft3

* 1. Determine the total product weight flow (WFprod) assuming all product in the walk-in is replaced each day:



WFprod = weight flow of product, lb/hr

However, considering the assumptions made, the above equation can be simplified to determine the product shelving load and product pull-down load:

1. Determine product shelving load (Qshelving-load):



where

Cp = product specific heat, Btu/lb-°F

ΔTshelving = Temperature difference between initial and final product temperature when shelving, °F

1. Determine product pull-down load (Qpull-down-load):



where

t = time to lower product temp (during post-defrost), hr

ΔTpost-defrost = post-defrost product temperature rise, °F

n = number of defrost periods

**Step 6: Infiltration Load**

The infiltration load or the heat gain is a product of the following parameters.

1. Refrigeration load for fully established flow (q)
2. Portion of time doorway is open (Dt)
3. Doorway flow factor (Df)
4. Effectiveness of doorway protective device (E)
5. Density factor (Fm)
6. Portion of time doorway is closed (Dc)
7. Calculate the decimal portion of time doorway is open (Dt):24



1. Calculate the density factor (Fm):



where

ρr = density of air inside the refrigerated spaces (from step 1), lb/ft3

ρi = density of adjacent space air that infiltrates into the refrigerated spaces (from step 1), lb/ft3

1. Calculate refrigeration load for fully established flow (q):



where

A = doorway area, ft2

hi = enthalpy of adjacent space air that infiltrates into the refrigerated spaces (from step 1), Btu/lb

hr = enthalpy of air inside the refrigerated spaces (from step 1), Btu/lb

ρi = density of adjacent space air that infiltrates into the refrigerated spaces (from step 1), lb/ft3

ρr = density of air inside the refrigerated spaces (from step 1), lb/ft3

H = doorway height, ft

Fm = density factor

1. Calculate the door open infiltration load (Qopen):



where

q = refrigeration load, Btu/hr

Dt = doorway open-time factor

Df = doorway flow factor

E = effectiveness of doorway protective device

n = number of doorways (number of doors)

For the purpose of this analysis and to estimate the effectiveness of strip curtains for the baseline scenario, it was assumed that 50% of walk-ins have no strip curtains (zero effectiveness) and 50% of walk-ins have strip curtains where one of the strips is missing with 2 cm gap at the bottom (80% effectiveness). Accordingly, the average effectiveness of strip curtains for the baseline scenario is 40% or 0.40.

Infiltration through gaskets of closed main and glass doors (assuming no infiltration or air leakage through the walk-in panels):

1. Calculate the decimal portion of time doorway is closed (Dc):



1. Calculate the infiltration rate of the box ( (ft3/h))[[17]](#footnote-14):



where

Δp = pressure differential between inside and outside of walk-in, mmWC

K = conversion factor, 35.315 ft3/m3

**Note**: The analysis focuses on infiltration through the door gaskets when the doors are closed.

1. Calculate refrigeration load for infiltration through gaskets (Qclosed):



where

hi = enthalpy of adjacent space air that infiltrates into the refrigerated spaces (from step 1), Btu/lb

hr = enthalpy of air inside the refrigerated spaces (from step 1), Btu/lb

ρi = density of adjacent space air that infiltrates into the refrigerated spaces (from step 1), lb/ft3

1. Calculate the total infiltration load (Qinf):



**Step 7: Total Cooling Load**

*Calculate the total cooling load of refrigerated spaces (Qtotal-cooling) by summing the calculated loads from the steps stated above.*



**Compressor EER and Demand**

To calculate compressor parameters:

1. Determine the saturated condensing temperature (SCT)
2. Determine the energy-efficiency ratio (EER) of the compressor based on SCT, cooling load, and cooling capacity of compressor
3. Determine the power requirements (kW) of the compressor based on calculated EER and cooling load
4. Determine the equivalent-full-load-hours (EFLH) of operation
5. Determine the annual energy usage (kWh) by using compressor power (kW) and EFLH

**Step 1: Determine the saturated condensing temperature (SCT)**

For medium temperature (MT): 

For low temperature (LT): 

where

DBadj = dry-bulb temperature (oF) of ambient or adjacent space where the compressor/condensing units reside. Defaults are based on climate zone design values representative PG&E climate zone cities

**Step 2: Determine the EER for both MT and LT**

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. To calculate the EER the following is needed:

Part-load ratio (PLR):

It is the ratio of total cooling load to compressor capacity. It indicates the percentage of compressor capacity needed to remove the total cooling load. It is calculated by following equation:

PLR = 

**Note:** Compressor capacity is determined by multiplying baseline cooling load by compressor capacity factor of 15%. Compressor capacity remains unchanged even when cooling capacity decreases with a measure.

Qcapacity = Qcooling-baseline \* 1.15

For calculating compressor efficiency, EER (Btu/hr/watts),use the following:

EER = a + (b \* SCT) + (c \* PLR) + (d \* SCT2) + (e \* PLR2) + (f \* SCT \* PLR) + (g \* SCT3) + (h \* PLR3) + (i \* SCT \* PLR2) + (j \* SCT2 \* PLR)

Where the coefficients are summarized in the table below.

Table 6 Coefficients for Compressor Efficiency Calculation

|  |  |  |
| --- | --- | --- |
| **Coefficient** | **Medium Temp** | **Low Temp** |
| a | 3.753460187 | 9.866509828 |
| b | -0.049642253 | -0.230356887 |
| c | 29.45898349 | 22.90555382 |
| d | 0.000342067 | 0.002188929 |
| e | -11.77055838 | -2.488667379 |
| f | -0.212941093 | -0.24805152 |
| g | -1.46606E-06 | -7.57495E-06 |
| h | 6.801701339 | 2.036062486 |
| i | -0.02018724 | -0.021477433 |
| j | 0.000657941 | 0.000938306 |

**Step 3: Power used by the compressor (kW)**

Power used by the compressor is determined based on calculated cooling load and EER, as outlined below.



**Equivalent-full-load-hours (EFLH) of Operation**

For compressors of walk-in cooler and freezer, EFLH was determined by multiplying annual available operation hours of 8,760 by the overall duty cycle factors. Duty cycle is a function of the compressor capacity, defrost and weather factors.

The following equation shows the relationship between the capacity, defrost and weather factors. The capacity factor is a function of both compressor capacity and cooling load. In other words, the capacity factor is a function of PLR. The defrost factor depends on the number and duration of defrost and is determined by simply dividing defrost duration (in hours) by 24-hours. The weather factor, however, is a function of weather. Using 2014 DEER weather data values for dry bulb temperature, SCT, and design dry bulb temperature

Duty cycle = Capacity factor x Defrost factor x Weather factor

where,

Capacity factor = function of PLR, (PLRbaseline) or (PLRpost-retrofit)

Defrost factor = 5.0% (1.2 hrs / 24 hrs), for freezers, (1 – 0.05)

Defrost factor = 10.0% (2.4 hrs /24 hrs), for coolers (1 – 0.10)

Weather factor = function of weather (described below)

To estimate the weather factor for each weather zone, use the following steps.

1. Calculated the net refrigeration effect (assume a sealed compressor). Calculation is based on ASHRAE HVAC Systems and Equipment Handbook, CH 35 Table 1.

For medium temperature (coolers),



For low temperature (freezers),



1. Calculate the net compressor load

1 – Net refrigeration effect

1. Sum the net compressor load across the 213 DEER hourly weather data.
2. Calculate the net compressor load for the design temperature.
3. The ratio of the sum of the compressor load divided by the net compressor load for design temperature is the full load hours.
4. Normalize the full load hours to 8760 to calculate the weather factor.

Accordingly,

EFLH = 8,760 x Duty cycle (hrs/yr)

**Compressor Annual Energy Usage**

Annual energy used by the compressor is determined based on calculated compressor power and EFLH.

kWhcomp = kWcomp x EFLH

**Savings due to cycling of evaporator fans**

Annual savings attributed to evaporator fan cycling assume a baseline fan operation of 8760 hours, utilizing a Shaded Pole or Permanent Split Capacitor motor. Post retrofit usage assumes the installation of an Electronically Commutated motor and fan cycling controls. These controls reduce the operating speed of the evaporator fans when the compressor is cycled off. These savings are dependent on the Compressor Duty Cycle.

The estimated Compressor Duty Cycle during the winter is assumed to be 40% and for summer 50%. 2014 DEER weather data is utilized to determine the number of annual hours for each operating mode. It is assumed that hourly readings below 32 degrees F° utilize a winter duty cycle. Hourly bins above 32 F° utilize a summer duty cycle. Total evaporator fan cycling savings are calculated using the following formula:

Final savings are a sum of compressor savings and savings attributed to evaporator fan cycling. A weighted average is then taken across case types to arrive at an average savings value for each PG&E climate zone.

## 2.2. Demand Reduction Estimation Methodologies

* This measure is an Early Retirement measure. The following analysis is only applicable to the First Baseline Period. Since EC Motors for Walk-In Evaporator fans are required under Federal and State codes for new evaporator coils or refrigerated cases, no savings will be claimed under the Second Baseline Period.

Demand savings are derived from annual kWh savings by dividing by 8760.

## 2.3. Gas Energy Savings Estimation Methodologies

* There are no gas energy savings associated with this measure.

# *Section 3. Load Shapes*

Load Shapes are an important part of the life-cycle cost analysis of any energy efficiency program portfolio. The net benefits associated with a measure are based on the amount of energy saved and the avoided cost per unit of energy saved. For electricity, the avoided cost varies hourly over an entire year. Thus, the net benefits calculation for a measure requires both the total annual energy savings (kWh) of the measure and the distribution of that savings over the year. The distribution of savings over the year is represented by the measure’s load shape. The measure’s load shape indicates what fraction of annual energy savings occurs in each time period of the year. An hourly load shape indicates what fraction of annual savings occurs for each hour of the year. A Time-of-Use (TOU) load shape indicates what fraction occurs within five or six broad time-of-use periods, typically defined by a specific utility rate tariff. Formally, a load shape is a set of fractions summing to unity, one fraction for each hour or for each TOU period. Multiplying the measure load shape with the hourly avoided cost stream determines the average avoided cost per kWh for use in the life cycle cost analysis that determines a measure’s Total Resource Cost (TRC) benefit.

## 3.1 Base Case Load Shapes

The base case load shape is expected to follow a typical non-residential refrigeration end use load shape

## 3.2 Measure Load Shapes

The measure load shape for this measure is determined based on the applicable *non-residential* market sector and the *refrigeration* end-use. This load shape is different from the base case due to the savings impact of the measures and is shown by the load shapes listed below.

# Section 4. Base Case & Measure Costs

|  |  |  |  |
| --- | --- | --- | --- |
| **Measure Application Type** | **Measure Life Basis** | **First Baseline Period Full Measure Cost (RUL)** | **Second Baseline Period Full Measure Cost (EUL – RUL)** |
| ***NC (new construction)*** | EUL | Calculated as Incremental Measure Cost | N/A |
| ***ROB(replace on burnout)*** | EUL | Calculated as Incremental Measure Cost | N/A |
| ***ER (early retirement)*** | RUL/  EUL-RUL | Calculated as Full Gross Measure Cost | Calculated as Negative Full Gross Base Case Cost |

## 4.1 Base Case(s) Costs

Base costs are not given in DEER 2011 Cost Data.

## 4.2 Measure Case Costs

The following Measure Application Type is appropriate to these measures. These costs are sourced from DEER 2011 Cost data. The Measure Case Costs are:

|  |  |  |  |  |  |  |
| --- | --- | --- | --- | --- | --- | --- |
| ***Measure Code*** | **Measure Application Type** | **Baseline** | **Equipment Cost** | **Labor / Installation Cost** | **Maintenance / Other Cost** | **Total Measure Case Cost** |
|  | ER | Existing | $300.63 | $165.71 | $0 | $466.34 |

*All costs are noted as $ per measure unit*

## 4.3 Incremental & Full Measure Costs

|  |  |  |  |
| --- | --- | --- | --- |
| **Measure Application Type** | **Full Measure Cost**  **(RUL Period/First Baseline)** | **Full Measure Cost**  **(EUL-RUL Period/ Second Baseline)** | **Incremental Measure Cost** |
| ER | Measure Equipment Cost  +Measure Labor Cost | (-1)x(Base Equipment Cost  + Base Labor Cost) | Measure Equipment Cost  – Base Case Equipment Cost |
| ROB | Measure Equipment Cost  – Base Case Equipment Cost | N/A | Measure Equipment Cost  – Base Case Equipment Cost |
| NC | Measure Equipment Cost  – Base Case Equipment Cost | N/A | Measure Equipment Cost  – Base Case Equipment Cost |

# *4.3.1 Full Measure Cost*

Full Measure Cost is the cost to install an energy efficient measure per the CPUC calculators. This definition implies a different meaning depending on the Measure Application type.

This measure application type is **ER** for the First baseline period only (RUL), so the Full Measure Cost (FMC) is represented by the equation below:

FMC = Measure Equipment Cost + Measure Labor Cost

FMC = *[$69.69 per unit (controller equipment) + $230.94 per unit (EC Motor equipment)] + [$92.06 per unit (controller labor)+ $73.65 per unit (EC Motor labor)] = $ 466.34 per motor controlled*

There are no 2011 DEER base case costs attributed to these measures. Therefore there are no FMC costs associated with the second baseline period.

\*Note: Various complicated price fluctuations are not addressed in these equations, such as future costs due to inflation in labor, future costs due to deflation in material cost, and other variables that cannot be accurately described at this time.

# *4.3.2 Incremental Measure Costs*

Incremental Measure Cost is the premium cost to install an energy efficient measure over a standard efficiency measure or code baseline measure. While IMC has a straightforward definition depending on the Measure Application type, the equation does vary.

This Measure Application Types is **ER.** There are no 2011 DEER base case costs attributed to these measures, so the Incremental Measure Cost (IMC) is represented by the equation below:

IMC = Measure Equipment Cost + Measure Labor Cost

IMC = *[$69.69 per unit (controller equipment) + $230.94 per unit (EC Motor equipment)] + [$92.06 per unit (controller labor)+ $73.65 per unit (EC Motor labor)] = $ 466.34 per motor controlled*

**Summary Table for Section 4**

|  |  |  |  |  |  |
| --- | --- | --- | --- | --- | --- |
| **Measure ID** | **Measure Application Types** | **Base Case Total Cost** | **Measure Case Total Cost[[18]](#endnote-4)** | **Full Measure Case Cost** | **Incremental Measure Cost** |
|  | ER | **N/A** | **466.34** | **466.34** | **466.34** |

# Calculations



# References

1. The DEER Measure Cost Data Users Guide found on [www.deeresources.com](http://www.deeresources.com) under *DEER2011 Database Format* hyperlink, DEER2011 for 13-14, spreadsheet *SPTdata\_format-V0.97.xls.* [↑](#endnote-ref-1)
2. Title 20 Appliance Efficiency Regulations, CEC-140 2013-002, Page 232, August 2013 [↑](#footnote-ref-1)
3. 2013 Building Energy Efficiency Standards, Title 24, Part 6, Section 120.6(a)3 [↑](#footnote-ref-2)
4. 2013 Nonresidential Compliance Manual, Title 24, Part 6, CEC-400-2013-002-CMF, June 2013 [↑](#footnote-ref-3)
5. EPACT 2005, Subpart R, Section 431.306. <http://www.ecfr.gov/cgi-bin/text-idx?SID=8f2b0bd78ff466ba84889adff20227db&node=10:3.0.1.4.19&rgn=div5#10:3.0.1.4.19.18> [↑](#footnote-ref-4)
6. GE ECM Evaporator Fan Motor Energy Monitoring, FSCT Report #5011.05.13 (Revised). Food Service Technology Center, July 2006. Fisher Nickel [↑](#endnote-ref-2)
7. Calculation sheets will be provided Supplemental Calculation Documentation. [↑](#footnote-ref-5)
8. ASHRAE 2002. Refrigeration Handbook. Atlanta, Georgia. pp. 12.1 [↑](#footnote-ref-6)
9. Southern California Edison Company. WPSCNRRN002.1 – Infiltration Barriers – Strip Curtains, October 2007. [↑](#footnote-ref-7)
10. “Commercial Facilities Contract Group Direct Impact Evaluation Draft Final Report: HIM Appendices”. ADM Associates, Inc., prepared for the California Public Utilities Commission, December 8, 2009. [↑](#footnote-ref-8)
11. Southern California Edison Company. WPSCNRRN002.1 – Infiltration Barriers – Strip Curtains, October 2007. [↑](#footnote-ref-9)
12. Defrost heater load should be considered in future iterations. [↑](#footnote-ref-10)
13. Weather Factor Calculator - 2014 DEER - PGE CZ 5.2014.xlsx. Submitted as Supplemental Calculation Documentation

     [↑](#endnote-ref-3)
14. Motor size is from FSTC report. [↑](#footnote-ref-11)
15. Assume floor is adiabatic. [↑](#footnote-ref-12)
16. According to ASHRAE, when doing “very light work” at room temperature of 75oF, 61% of people load is sensible and 39% latent. These ratios, however, are a function of room temperature. As the room temperature decreases, the contribution of sensible load is increased. Due to lack of such information about the relationship between these variables, a 61% sensible load ratio is used in the current analysis, although higher ratios are expected at lower room temperatures. [↑](#footnote-ref-13)
17. This value is a point of controversy. The ADM report discusses measuring the infiltration rate, but does not quantify it in the study as a function of temperature, gasket condition, or other factor. Since the load calculated for infiltration using the approach discussed is small, the analysis at this time is based on the approach documented in the SCE workpaper. [↑](#footnote-ref-14)
18. SCE, Measure Cost Revision 5 revised for PG&E by S.L. Blanc 2012

     [↑](#endnote-ref-4)