**Workpaper PGE3PREF129-R1**

**FHPC on Singles**

**Revision 2**

**PECI**

**EnergySmart Grocer**

**Floating Head Pressure - Single Compressors**

**Measure Codes: R320, R321, R322, R323**

**PECI EnergySmart Grocer**

# At-a-Glance Summary

|  |  |  |
| --- | --- | --- |
| **Applicable Measure Codes:** | **R320, R321** | **R322, R323** |
| **Measure Description:** | **FHP on Single Compressor Systems-Air-cooled Condensing Unit** | **FHP on Single Compressor Systems- Air-cooled Remote Condenser** |
| **Energy Impact Common Units:** | Compressor motor HP | Compressor motor HP |
| **Base Case Description:** | Single compressor in an air-cooled condensing unit with a fixed head pressure control valve. | Multiple single compressors, each with a fixed head pressure control valve, cooled by a remote air-cooled condenser. |
| **Base Case Energy Consumption:** | Varies across climates and Low Temperature/Medium Temperature  Source: eQUEST-R calculations | Varies across climates and Low Temperature/Medium Temperature  Source: eQUEST-R calculations |
| **Measure Energy Consumption:** | Varies across climates and Low Temperature/Medium Temperature  Source: eQUEST-R calculations | Varies across climates and Low Temperature/Medium Temperature  Source: eQUEST-R calculations |
| **Energy Savings (Base Case – Measure)** | Varies across climates and Low Temperature/Medium Temperature, see At a Glance measure table  Source: eQUEST-R calculations | Varies across climates and Low Temperature/Medium Temperature, see At a Glance measure table  Source: eQUEST-R calculations |
| **Costs Common Units:** | Compressor motor HP | Compressor motor HP |
| **Base Case Equipment Cost ($/unit):** | $0  No code exists for this measure, and no equipment would be replaced | $0  No code exists for this measure, and no equipment would be replaced |
| **Measure Equipment Cost ($/unit):** | $347 per HP for low temperature and $418 per HP for medium temperature systems  Source: Distributors, Retailers, HVAC&R wholesaler | $106 per HP for low temperature and $146 per HP for medium temperature systems  Source: Distributors, Retailers, HVAC&R wholesaler |
| **Gross Measure Cost ($/unit):** | $347 per HP for low temperature  $418 per HP for medium temperature  Source: HVAC&R wholesaler, Distributor | $51 per HP for low temperature  $61 per HP for medium temperature  Source: HVAC&R wholesaler, Distributor |
| **Measure Incremental Cost ($/unit):** | $347 per HP for low temperature and $418 per HP for medium temperature systems  Source: Distributors, Retailers, HVAC&R wholesaler | $157 per HP for low temperature and $207 per HP for medium temperature systems  Source: Distributors, Retailers, HVAC&R wholesaler |
| **Effective Useful Life (years):** | 15 years  Source: DEER2008, D03-221 | 15 years  Source: DEER2008, D03-221 |
| **Measure Application Type:** | Retrofit Add-on (REA) | Retrofit Add-on (REA) |
| **Net-to-Gross Ratio:** | 0.60  DEER2011\_NTGR\_2012-05-16 | 0.60  DEER2011\_NTGR\_2012-05-16 |
| **Important Comments:** |  |  |

# Work Paper Approvals

The following Manager(s) approved this workpaper through the PG&E Electronic Data Routing System under Routing Requisition # \_\_\_\_\_\_\_\_\_\_\_\_\_\_\_

|  |
| --- |
|  |
| **Grant Brohard**  Manager, Technical Product Support |
| **Carolyn Weiner**  Principal, CES Products and Programs |

# Document Revision History

|  |  |  |  |
| --- | --- | --- | --- |
| **Revision #** | **Date** | **Section by Section Description of Revisions** | **Author (Company)** |
| **Revision 0** | 02/28/2011 | Original work paper. | Dustin Bailey, (PECI Engineering) |
| **Revision 1** | 05/24/2012 | Updated to PG&E 2013-2014 format. | Dustin Bailey,  (PECI Engineering) |
| **Revision 2** | 5/30/2014 | Update per weather file changes and PG&E format guidelines. | Ben Wright  (PECI Engineering) |

# Table of Contents

[At-a-Glance Summary ii](#_Toc389233289)

[Work Paper Approvals iii](#_Toc389233290)

[Document Revision History iv](#_Toc389233291)

[Table of Contents v](#_Toc389233292)

[List of Tables vi](#_Toc389233293)

[List of Figures vi](#_Toc389233294)

[Section 1. General Measure & Baseline Data 7](#_Toc389233295)

[1.1 Product Measure Description & Background 7](#_Toc389233296)

[1.2 Product Technical Description 7](#_Toc389233297)

[1.3 Measure Application Type 9](#_Toc389233298)

[1.4 Product Base Case and Measure Case Data 9](#_Toc389233299)

[1.4.1 DEER Base Case and Measure Case Information 9](#_Toc389233300)

[1.4.2 Codes & Standards Requirements Base Case and Measure Information 9](#_Toc389233301)

[1.4.3 EM&V, Market Potential, and Other Studies – Base Case and Measure Case Information 10](#_Toc389233302)

[1.4.4 Assumptions and Calculations from other sources—Base and Measure Cases 10](#_Toc389233303)

[1.4.5 Time-of-Use Adjustment Factor 10](#_Toc389233304)

[1.5 Summary of Inputs for Savings Calculations 10](#_Toc389233305)

[Section 2. Calculation Methods 12](#_Toc389233306)

[Measure Application Type 12](#_Toc389233307)

[Measure Life Basis 12](#_Toc389233308)

[First Baseline Period: Energy Savings Baseline 12](#_Toc389233309)

[Second Baseline Period: Energy Savings Baseline 12](#_Toc389233310)

[2.1 Electric Energy Savings Estimation Methodologies 12](#_Toc389233311)

[2.1.1 Overall Approach 13](#_Toc389233312)

[2.1.2 Variable Sensitivity Analysis 15](#_Toc389233313)

[2.1.3 eQUEST/DOE-2.2R Model Inputs 21](#_Toc389233314)

[2.2. Demand Reduction Estimation Methodologies 23](#_Toc389233315)

[2.3. Gas Energy Savings Estimation Methodologies 23](#_Toc389233316)

[Section 3. Load Shapes 23](#_Toc389233317)

[3.1 Base Case Load Shapes 23](#_Toc389233318)

[3.2 Measure Load Shapes 23](#_Toc389233319)

[Section 4. Base Case & Measure Costs 23](#_Toc389233320)

[4.1 Base Case(s) Costs 24](#_Toc389233321)

[4.2 Measure Case Costs 24](#_Toc389233322)

[4.3 Incremental & Full Measure Costs 26](#_Toc389233323)

[4.3.1 Full Measure Cost 26](#_Toc389233324)

[4.3.2 Incremental Measure Costs 26](#_Toc389233325)

[References: 28](#_Toc389233326)

# List of Tables

[Table 1 Measure Application Type 9](#_Toc389233273)

[Table 2 DEER Net-to-Gross Ratios 9](#_Toc389233274)

[Table 3- Summary of Inputs for Savings Calculations 10](#_Toc389233275)

[Table 4- Baseline by Measure Application Type 12](#_Toc389233276)

[Table 5 - Typical Values for eQUEST Model Used for Analysis 12](#_Toc389233277)

[Table 6- Summary of Refrigeration Display Case Inputs 14](#_Toc389233278)

[Table 7 – Sensitivity Variables for the Remote Condenser 15](#_Toc389233279)

[Table 8 – Final Variables for Remote Condenser 19](#_Toc389233280)

[Table 9 - Sensitivity Variables for the Condensing Unit 19](#_Toc389233281)

[Table 10 – Final Variables for Condensing Unit 20](#_Toc389233282)

[Table 11 - Remote System Compressor/Condenser Design Conditions 21](#_Toc389233283)

[Table 12 - Condensing Unit System Compressor/Condenser Design Conditions 22](#_Toc389233284)

[Table 13 - Cost Information 23](#_Toc389233285)

[Table 14 - Condensing Unit System Final Cost 25](#_Toc389233286)

[Table 15– Remote Condenser Final Cost 25](#_Toc389233287)

[Table 16 - Summary of Final Costs 27](#_Toc389233288)

# List of Figures

[Figure 1 - Diagram of Condensing Temperature vs. Outside Air Temperature for FHP on Single 8](#_Toc389233331)

[Figure 2 - Multiple compressor lines being fed into a remote condenser 13](#_Toc389233332)

[Figure 3 - Multiple condensing units 13](#_Toc389233333)

[Figure 4 – Savings Sensitivity to Model Variables for the Remote Condenser 17](#_Toc389233334)

[Figure 5 - Savings Sensitivity to Model Variables for the Condensing Units, Medium Temp 20](#_Toc389233335)

# Section 1. General Measure & Baseline Data

## 1.1 Product Measure Description & Background

***Catalog Description***

Floating head pressure controls increase refrigeration system efficiency by running at a lower compressor head pressure during hours of lower outside air temperature. This measure addresses adding these controls to simple refrigeration systems including remote condensing systems and condensing units.

The savings are calculated for all 9 PG&E climate zones.

***Program Restrictions and Guidelines***

***Terms and Conditions***

The measure is applicable only to refrigeration systems having:

1. A single compressor of 1 HP motor or larger serving a suction group, either in a condensing unit or with a remote condenser,
2. Condenser intake air from outside ambient air, and
3. A fixed pressure head control valve.

Requirements:

1. Must replace fixed pressure head control valve with adjustable head pressure control valve (flood-back control valve) to lower minimum condensing head pressure.
2. Adjustable valve must be field adjusted to pressure equivalent of 70º F saturated temperature or lower and verified against a calibrated pressure gauge or transducer.
3. Must install either:
   1. A balanced-port valve or electronic expansion valve that is sized to meet the load requirement at 70º F condensing temperature.

Or

* 1. A device to supplement refrigerant feed to each evaporator attached to the condenser.

Exemption for 3): existing expansion valve is a balance port or electronic expansion valve.

Rebate is based on a “per compressor motor nameplate HP” basis. Pre-inspection is required.

***Market Applicability***

This measure is applicable to retail grocery stores of all sizes.

## 1.2 Product Technical Description

Typical refrigeration systems maintain high head pressure through a fixed head pressure control valve, which maintains a minimum head pressure equivalent to 90-95º F saturation temperature throughout the year. This temperature is chosen to create a pressure differential great enough to maintain refrigerant flow and to provide enough hot gas during hot gas defrosting. As ambient temperature drops the condenser will reach this minimum temperature, and the fixed head pressure control valve will modulate the flow of refrigerant to maintain operating pressure. This forces the compressors to operate at a high condensing temperature even during cooler outside air temperatures.

Compressor work can be reduced by installing a variable head pressure control valve and adjusting it to a lower pressure setting equivalent to 70º F saturation temperature. Because the difference between the evaporator (suction) pressure and the condensing discharge pressure directly affects compressor efficiency, lowering the condensing pressure saves energy by lowering the compression ratio. Savings from floating head pressure occur when the ambient outside air temperature falls below 80 to 85F, depending on refrigerant. The figure below demonstrates the advantage of a floating head pressure control over conventional fixed head pressure control.

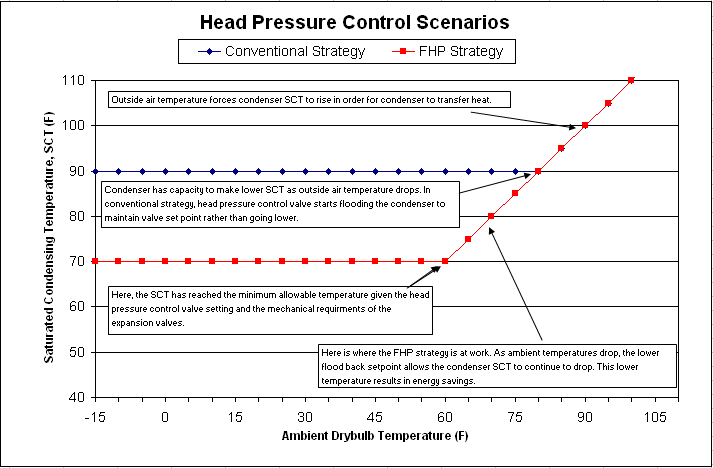


Figure 1 - Diagram of Condensing Temperature vs. Outside Air Temperature for FHP on Single

In order for a refrigeration system to operate at these lower head pressure conditions consideration must be taken to guarantee that loads are maintained properly. A variety of technologies can be used to maintain the refrigeration load at lower head pressures:

* Balanced port valves - these expansion valves have a mechanical bypass which allows valve operation at much lower head pressures than standard valves. These valves are rated to a minimum of 70º F head pressure, and are inexpensive and easy to install.
* Electronic expansion valves (EEV) - these expansion valves use advanced controls to maintain case temperature independent of head pressure. Although these valves allow for a minimum head pressure set point below 70º F, they are expensive and the additional savings of maintaining a temperature below 70º F is insignificant for the cost.
* Equipment specific technology – there are other technologies that feed refrigerant past the expansion valve directly to the evaporator. These systems maintain the proper pressure differential across the expansion valve under reduced head pressure conditions. The availability of this technology is limited but is allowed by the terms and conditions.

## 1.3 Measure Application Type

The program type for the measure is Retrofit Add-on (REA).

The DEER Measure Cost Data Users Guide, version 2.01, found on www.deeresources.com under DEER2011 Database Format hyperlink, DEER2011 for 13-14, spreadsheet SPTdata\_format-V0.97.xls, defines the term as follows:

**Table 1** **Measure Application Type[[1]](#endnote-2)**

*Identifies the measure application type in the Measure Implementation table in DEER2011.*

|  |  |  |
| --- | --- | --- |
| **Code** | **Description** | **Comment** |
| REA | Retrofit Add On | *Single baseline (above pre-existing), full measure costs required* |

## 1.4 Product Base Case and Measure Case Data

### 1.4.1 DEER Base Case and Measure Case Information

The DEER data does not contain the appropriate information for this measure. The Database for Energy Efficient Resources (DEER) does address *Floating Head Pressure, Fixed Set point*as measure D03D03-221[[2]](#endnote-3) which is for multiplex racks, but it does not address floating head controls for compressor systems that are not a part of multiplex rack. Therefore, a new analysis of savings was developed for floating head control installed on single compressor systems.

Table 2 DEER Net-to-Gross Ratios

|  |  |  |  |
| --- | --- | --- | --- |
|  |  | **DEER Spreadsheet** | |
| Program Approach | NTG | File name | Cell Number |
| EnergySmart Grocer | 0.60 | DEER2011\_NTGR\_2012-05-16 | T56 |

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

***Title 20:*** These measures do not fall under Title 20 of the California Energy Regulations. According to Section 1601 of Title 20, only new appliances sold in California are regulated by Title 20[[3]](#endnote-4). Additionally, there are no floating-head pressure controls in Title 20. Because these measures are retrofit additions, they are not subject to new appliance requirements. Condensing temperature control strategies fall under Title 24 building efficiency guidelines.

***Title 24:*** These measures do not fall under Title 24 of the California Energy Regulations. Although condensing temperature control strategies are covered by Title 24, per Section 100.0(a)2 a building permit would invoke any Title 24 code requirements. The implementation of this measure does not require a building permit to be filed or renewed and therefore no code implications are considered[[4]](#endnote-5).

***Federal Standards:*** These measures do not fall under Federal DOE or EPA Energy Regulations. Condensing temperature control strategies fall under Title 24 building efficiency guidelines.

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

This is a relatively new measure and has had few EM&V studies that evaluate its performance.

***1.4.3.1 Summit Blue BPA Energy Smart Grocery Evaluation[[5]](#endnote-6)***

Summit Blue evaluated a variety of measures as a part of their evaluation of the EnergySmart Program in the Pacific Northwest (PNW). Based on their evaluation, the floating head pressure control on multiplex systems energy savings was found to be 937 kWh/yr-HP. Summit Blue did verification across nine sites in the BPA territory. Although these results are for a multiplex system, they can be used as a reasonable comparison to the final results for single compressor systems.

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

**Effective Useful Life**

For DEER measure D03D03-221, *Floating Head Pressure, Fixed Set point*, DEER 2008 reports an effective useful life (EUL) of 15 years[[6]](#endnote-7). Since equipment in this measure is comparable to equipment used for FHP in multiplex compressor systems, the same EUL is used.

### 1.4.5 Time-of-Use Adjustment Factor

As directed by the CPUC in decision 06-06-063 dated June 29, 2006, time-of-use (TOU) adjustment factors are to be applied for residential A/C and commercial A/C (packaged and split-system direct-expansion cooling) measures only. This measure is assigned a DEER08 load shape, i.e. the load shape starts with “DEER:” so the TOU assigned to this measure is zero.

## 1.5 Summary of Inputs for Savings Calculations

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

Table 3- Summary of Inputs for Savings Calculations

| **Input Variable** | **Variations** | **Base Case 1 Average Value** | **Base Case 2 Average Value** | **Measure Case Average Value** | **Reference Section** |
| --- | --- | --- | --- | --- | --- |
| **Electric Savings- Remote Condenser** | CZ | 491 kWh | 491 kWh | 491 kWh |  |
| **Electric Savings- Condensing Unit** | CZ | 637 kWh | 637 kWh | 637 kWh |  |
| **Gas Savings** | N/A | N/A | N/A | N/A |  |
| **Hours of operation** | N/A | N/A | N/A | N/A |  |
| **Full Cost- Remote Condenser** | REA | $182 | $0 | $182 | *Section 4.3* |
| **Full Cost- Condensing Unit** | REA | $383 | $0 | $383 | *Section 4.3* |
| **Incremental Cost- Remote Condenser** | REA | $182 | $0 | $182 | *Section 4.2* |
| **Incremental Cost- Condensing Unit** | REA | $383 | $0 | $383 | *Section 4.2* |
| **EUL /RUL** | REA | *5* | *10* | 15 | *Section 1.4.4* |
| **NTG** | One | 0.7 | *0.7* | 0.7 | *Section 1.4.1* |
| **ISR** | Applies -- Yes | 1 | 1 | 1 |  |
| **TOU Factor** | *A/C projects only* | N/A | N/A | N/A | *Section 1.4.5* |

# 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** |
| ***REA (retrofit add on)*** | **EUL** | Code Baseline | N/A |

An energy savings forecast from the implementation of the Floating Head Pressure (FHP) on Single Compressor Systems- Air-Cooled Condenser measure was generated from the results of an eQUEST simulation model and engineering calculations. As shown in Table 2, a customer average baseline is used for both the first and second baseline periods.

## 2.1 Electric Energy Savings Estimation Methodologies

Data was collected from multiple sources including but not limited to: manufacturer data, PECI field staff, refrigeration experts, and the DEER 2005 Grocery model. These data were used to establish a range of various conditions found in the field.

This measure modeled a change in the condenser head pressure and the savings that occurred in the refrigeration system components. Most non-refrigeration system components in the model had negligible effect on the savings of this measure. Table 3 describes typical values for non-refrigeration system components in the eQUEST models.

Table 5 - Typical Values for eQUEST Model Used for Analysis

|  |  |
| --- | --- |
| **Building Characteristic** | **Typical Input Value** |
| Envelope (Walls, Windows, Roof) | DEER 2005 grocery envelope values |
| Lighting Power Density | 2.8 Watts/SF |
| Miscellaneous Equipment | 1.1 Watts/SF |
| Infiltration | 0.07 CFM/SF |
| HVAC System Type | Package Variable Temperature Constant Volume Supply Fan |
| Design Supply Air | 3,604 CFM |
| OSA Air | 30% of supply air |
| Refrigeration System | Described in detail below |
| Fan Operating Hours | 16 Hours/day, 365 days per year |
| Sales Area | 2,650 SF |
| Cooling Capacity | Sized to climate zone |
| Heating Capacity (All Zones) | Sized to climate zone |
| Fan System Efficiency | 0.000652 kW/cfm |
| Fan speed | Constant 100% |
| Occupancy schedule | DEER 2005 grocery schedule and 125 ft2 per person |

None of these values had a significant effect on the measure savings. The refrigeration variables had more significant impact on the measure savings and are discussed in the Variable Sensitivity Analysis section.

Two main models were created to represent this measure. The remote condenser system has multiple compressors attached to one central remote condenser (see Figure 2 - Multiple compressor lines being fed into a remote condenser). This system uses outside air temperature to control the condenser fans, typically staging the fans in ten degree increments; i.e. one fan on all the time, a second fan on when ambient temperature rises to 60 degrees, a third fan on when ambient temperature rises to 70 degrees, etc. until all fans are on.



Figure 2 - Multiple compressor lines being fed into a remote condenser

The second system modeled is the condensing unit system which has one compressor attached to one condenser (see Figure 3 - Multiple condensing units). For this system, the condenser fans cycle on and off with the compressor.



Figure 3 - Multiple condensing units

### Overall Approach

Once the model conditions were chosen for the baseline, the measure was modeled by changing the head pressure control valve set point from 94 ºF for the baseline model to 70 ºF for the retrofit model. The 94 ºF was chosen to represent R-22 at 180 psig, the factory setting for fixed head pressure valves for R-22. The 70 ºF represents the new variable head pressure valve setting required in the Terms and Conditions (T&Cs) of the measure. This setting does not simulate the condenser running at this temperature all the time but instead simulates the condenser temperature floating down during hours of lower ambient temperatures. Hours of lower ambient temperature will therefore result in lower condenser refrigeration temperature and pressure, lower pressure at the compressor and lower energy of the refrigeration system.

Three major design components were considered when designing the model: compressor system design, refrigerated case design and condenser system design.

**Compressor System Design**

The size of the compressor for the remote condenser system was assumed to be 10 HP. This was based on a refrigeration schedule review of a variety of remote compressors (n=26) which showed that the coefficient of performance (COP) of the compressors did not change significantly over 7.5 HP, and common remote compressor sizes were typically 10 to 20 HP. Since the rebate is per HP, the size of the compressor in the model is not important as long as the COP is consistent.

Condensing units sizes typically varied from 1 to 10 HP. COP varies across this size range so motor weighting was done to calculate the final savings based on audited motor sizes (n=897).

Compressors were modeled based on data from Copeland product specification sheets.[[7]](#endnote-8) The compressor capacity curves presented in the detailed manufacturer specification sheets were used for compressor capacity. In order to properly model the power usage of the compressors, the COP was calculated for each compressor at its specific design condition. Using the calculated COP, the default eQUEST COP power curve was used to model the power of the compressors.

Equation 1- Compressor COP



Where,

CompCOP= Compressor Coefficient of Performance (unit-less)

Comp­cap= Compressor capacity from manufacturer’s specification sheets at design conditions (BTU/hr)

Comppower­= Compressor input power from manufacturer’s specification sheets in units of BTU/hr (BTU/hr=3.412 W)

**Case Design**

The case length was calculated to determine a refrigeration load that fit the compressor capacity with sizing. Refrigeration load was calculated from the compressor capacity and the compressor sizing factor that was found from analysis of refrigeration schedules of single compressor systems (n=21). The sizing factor represents a percentage that the compressor is sized relative to the refrigeration load.

Equation 2- Calculate Refrigeration Load from Compressor Sizing



Where,

Rload= Refrigeration Load (BTU/hr)

CSF= Compressor sizing factor (%)

Two different cases were chosen from the eQUEST library to represent the refrigerated cases in this model. The model inputs are shown in Table 4. The refrigeration load is equivalent to coil capacity shown below.

Table 6- Summary of Refrigeration Display Case Inputs

|  |  |  |  |  |  |  |
| --- | --- | --- | --- | --- | --- | --- |
|  | **Model #** | **TYPE** | **Conduction** | **Infiltration Load** | **Rated Supply T** | **Coil Capacity\*** |
|  |  |  | **BTU/hr-ft** | **BTU/hr-ft** | **Deg F** | **BTU/hr-ft** |
| LT Reach in case | RCA | Multideck Doors | 110 | 26 | -12 | 616 |
| MT Open Multideck | M4ALG1 | Multideck | 100 | 899 | 25 | 1400 |

\* Coil capacity per door = 1540 BTU/hr, assume 2.5 ft door

To calculate the case length for each model (LT and MT), the refrigeration load was divided by coil capacity per length:

Equation 3- Case Length



Where,

Caselength = Case length (ft)

CoilCap= Coil capacity based on case type as shown in Table 4 (BTU/hr-ft)

**Condenser System Design**

Condenser capacity (size) was calculated two different ways for the two different types of condenser systems and was based on the available information collected from single compressor system refrigeration schedules. The remote condenser capacity was calculated usingsizing factors calculated from the average of remote condenser system refrigeration schedules. The condensing unit capacity was calculated by summing the evaporator (refrigeration) load and the compressor design power.

Equation 4 - Condenser Capacity –Remote Condenser



Equation 5 - Condenser Capacity -Condensing unit



Where,

CondCapR= Condenser capacity for the remote system (BTU/hr)

CondSF= Condenser sizing factor (%)

CondcapCU= Condenser capacity for the condensing unit system (BTU/hr)

All condensers were designed with 1 HP condenser fans. Typically condenser fan size is chosen with concern for first-cost (for example, ½ HP fans cost more because more fans are needed) and noise restrictions (1.5 HP fans make more noise).

### Variable Sensitivity Analysis

Sensitivity analysis was conducted using a tornado diagram method. The tornado diagram was constructed by first listing variables that affect the energy savings and finding the “typical,” “maximum” and “minimum” values for each. The variables were chosen by their estimated impact on the measure energy savings. The typical, maximum and minimum values were chosen by surveying PECI field staff, refrigeration design engineers and refrigeration technicians. The parametric analysis was conducted using eQUEST/DOE-2.2R, changing one variable at a time, holding all other variables constant. The outputs shown in the tornado diagram are the savings values obtained from subtracting the DOE-2.2R refrigeration system energy use in the measure case minus the DOE-2.2R refrigeration system energy use in the base case. The tornado diagram shows variables by the order of magnitude of effect on the energy savings so that the output looks like a tornado with the typical values as the “eye.”

Oregon, Idaho and Washington climate zones were used in the sensitivity analysis to look at climate zone effect, but the final measure results were run in each CA climate zone.

**2.1.2.1 Sensitivity Analysis-Remote Condenser System**

The “typical” values used in this analysis represent what is typically found in existing single compressor refrigeration systems with remote compressors, according to the survey of experts. The results of testing the effects of these variables were used to guide further investigation of key variables. The list of variables and their respective typical, maximum and minimum values can be found in Table 5. Each variable was independently changed from its typical value in the DOE2.2R model to assess sensitivity.

Table 7 – Sensitivity Variables for the Remote Condenser

|  |  |  |  |  |
| --- | --- | --- | --- | --- |
| **Base Case Variables** | | | | |
| **Variable** | **Units** | **Min** | **Typical** | **Max** |
| Compressor COP | unit less | N/A | 2.7 | 3.1 |
| Compressor Sizing Factor | % | 110% | 120% | 130% |
| Condenser Sizing Factor | % | 130% | 170% | 210% |
| Number of Condenser Fans | Number | 4 | 5 | 6 |
| Condenser Set Point | Degree F | 85 | 90 | 95 |
| Suction Temperature | Degree F | -30 | 15 | 40 |
| **Measure Case Variables** | | | | |
| **Variable** | **Units** | **Min** | **Typical** | **Max** |
| EE Condenser Minimum Set Point | Degree F | 60 | 70 | N/A |

The output from the sensitivity analysis is shown in Figure 4. The sensitivity analysis was used to provide guidance into what values need to be further investigated based on the magnitude of significance on the energy savings. Below is the explanation of what each of the variables represent, how the measure analysis was modified and how final model assumption values were chosen for each variable:

* Condenser Set Point – this represents the base case set point of the head pressure control valve on the condenser which also represents the minimum condensing set point. As seen in Figure 4 this value had the greatest effect on the energy savings of the model. For this reason more research was done to find an accurate representation of this value.

According to manufacturers, the fixed valve is adjusted at the factory and is not field adjustable. The valve has a +/- 5 psi calibration accuracy which corresponds to about 2 °F for R-404A, R-22 and R-507. The assumption was made that the same number of valves are calibrated high as are calibrated low. The valve recommended for R-22 is set at 180 psig by one manufacturer and 185 psig[[8]](#endnote-9) by another. To be conservative, 180 psig was chosen as the final model value. The model used a fixed head pressure set point of 94ºF for R-22, the saturated temperature at 180 psig.

Because this was the most significant sensitivity, analysis was conducted for other refrigerants that would have different pressure temperature relationships. In particular, R-404A was carefully investigated. Manufacturers use different pressure set points for each refrigerant that result in similar temperature baseline conditions. The impact of using different refrigerants was negligible to the final energy savings of the model.

* Suction Temperature – this represents the evaporator temperature of the load attached to the system. This variable had the second greatest effect on the final savings, so two separate measures were created: one measure uses -20ºF to represent low temperature systems and the other uses 15ºF to represent medium temperature systems. The values of -20ºF and +15ºF were chosen from an analysis of PECI ‘S GrocerSmartTM audit data[[9]](#footnote-2) of stores both in CA and in the PNW (n=16,576).

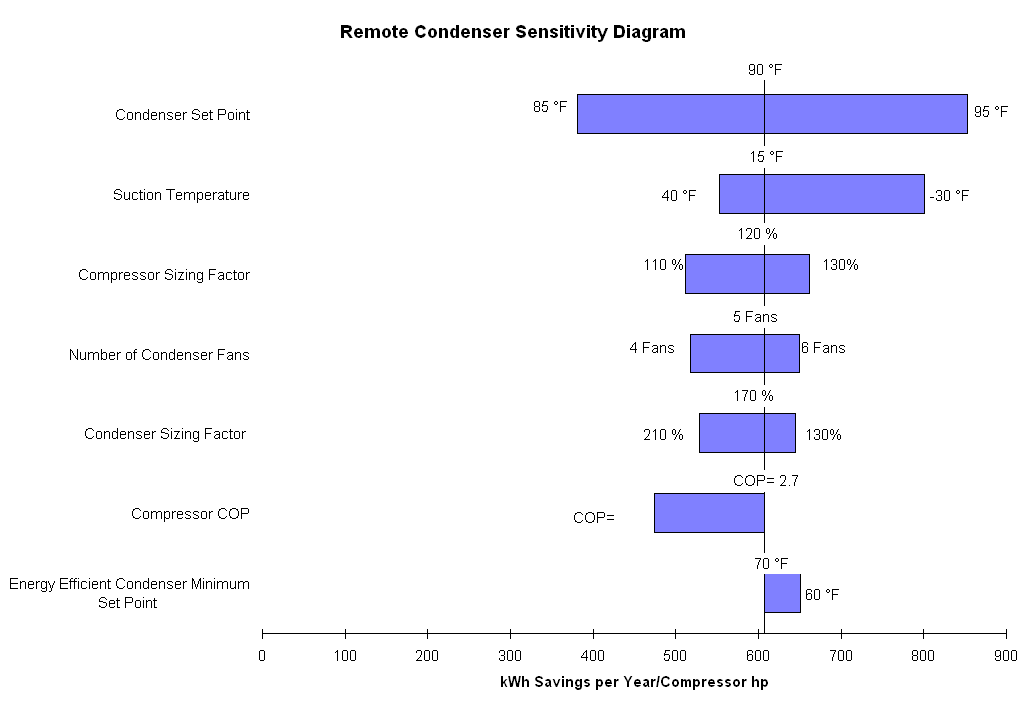


Figure 4 – Savings Sensitivity to Model Variables for the Remote Condenser

* Compressor Sizing Factor – this represents the ratio of the compressor capacity and the refrigeration load that is attached to the compressor. This variable also had a significant effect on the final savings of this measure and was further investigated.

Refrigeration schedules that included single compressor systems were obtained. Analysis of these refrigeration schedules showed a median value of 117% sizing factor for low temperature systems and 113% sizing factor for medium temperature systems (n=26).

* Number of Condenser Fans – the number of fans had an impact on the savings due to the simplistic fan controls found on these systems. The fans operate based on outside air temperature and depending on the number of fans the baseline energy changes in the model.

The final number of fans was chosen based on fan sizes and condenser capacity. Typically condenser fan size is chosen with concern for first-cost (for example, ½ HP fans cost more because more fans are needed) and noise restrictions (1.5 HP fans make more noise). For this reason the final model used the appropriate number of 1 HP fans to match manufacturer design and condenser capacity.

* Condenser Sizing Factor – this represents the ratio of the condenser capacity and the refrigeration load attached the condenser. Refrigeration schedules that included remote condensers systems were obtained. Analysis of refrigeration schedules of single compressors with remote condenser (n=6, small sample size) showed a median value of 185% sizing factor of condenser for low temperature systems and 150% sizing factor of condenser for medium temperature systems. The difference between the minimum and maximum values resulted in less than 10% energy savings difference. Since there was a small sample of condensers available and typically the condenser is sized by location, the values obtained from the refrigeration schedule review were used for this measure.
* Compressor COP – this was used to represent the effects of compressor motor size on the savings. There is a strong correlation between motor size and efficiency at small motor sizes, but above 7.5 HP the COP doesn’t change significantly. Remote compressor systems typically use compressors larger than 7.5 HP, so average COP was calculated from collected refrigeration schedules (n=26).
* Energy Efficient Condenser Minimum Set Point – the use of a more complex (and more expensive) expansion valve allows for lower minimum head pressure. The sensitivity analysis showed that a lower condenser set point has small incremental energy savings for the large incremental cost of the more advanced expansion valve. For this reason it was decided to require only balance port valves that allow for a 70 ºF minimum condenser set point.

Table 8 – Final Variables for Remote Condenser

|  |  |  |  |
| --- | --- | --- | --- |
| **Final Model Variables** | | | |
| **Variable** | **Units** | **MT** | **LT** |
| Compressor COP | 10 HP | 2.49 | 1.45 |
| Compressor Sizing Factor | % | 113% | 117% |
| Condenser Sizing Factor Factor | % | 150% | 185% |
| Number of Condenser Fans | Number | 4 | 4 |
| Condenser Head Pressure Set Point | Degree F | 94 | 94 |
| Suction Temperature | Degree F | 15 | -20 |
| **Measure Case Variables** | | | |
| **Variable** | **Units** | **MT** | **LT** |
| Condenser Minimum Set Point | Degree F | 70 | 70 |

For more information on the other model variables, see the detailed model calculations below.

**2.1.2.2 Sensitivity Analysis-Condensing Unit System**

A sensitivity analysis similar to that for the remote condenser system was conducted for the condensing unit system. Again, the “typical” values used in this analysis represent what experts felt were typically found in existing single compressor refrigeration systems with remote compressors. The results of testing the effects of these variables were used to guide further investigation of key variables. Each variable was independently changed from its typical value in the DOE2.2R model to assess sensitivity to it.

From the sensitivity analysis of the remote condenser system, the measure was divided into two measures, one for low temperature systems and one for medium temperature systems. Also, the system components were better defined as they came in as in packaged units. For this reason the sizing factor for the compressor and condenser fan motor sizing was fixed based manufacturer specifications. This reduced the number of variables requiring investigation in the sensitivity analysis. Table 7 shows the list of variables and the typical, maximum and minimum values.

Table 9 - Sensitivity Variables for the Condensing Unit

|  |  |  |  |  |
| --- | --- | --- | --- | --- |
| **Base Case Variables** | | | | |
| **Variable** | **Units** | **Min** | **Max** | **Typical** |
| Compressor COP | unit less | 0.9 | 2.9 | 2.7 |
| Compressor Sizing Factor | % | 110% | 130% | 120% |
| Condenser Minimum Set Point (SCT) | Degree F | 92.5 | 96.1 | 94.3 |

The output from the energy savings sensitivity analysis is shown in Figure 5. The sensitivity analysis was used to provide guidance into what values needed further investigation.

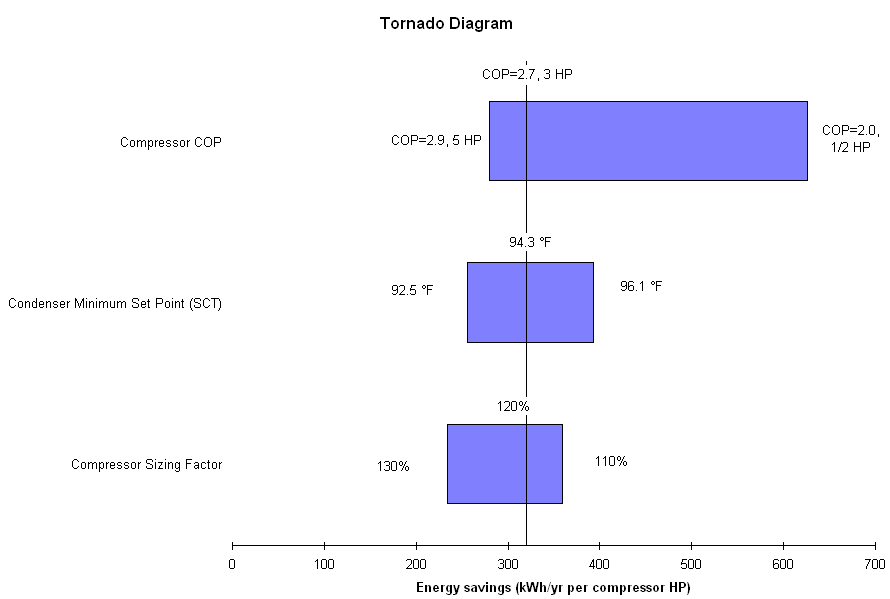


Figure 5 - Savings Sensitivity to Model Variables for the Condensing Units, Medium Temp

Below is the explanation of what each of the variables represents, how the measure analysis was modified and how typical assumption values were verified:

* Compressor Efficiency - this was used to represent the effects of different size compressor motors on the savings. There is a strong correlation between motor size and motor efficiency for sizes less than 7.5 HP, and condensing units are commonly 10 HP or less.

Since this parameter had such a large effect, as shown in Figure 5, motor weighting was used. In order to get a weighted COP, models were created for motor sizes from 1 to 10 HP. The results were weighted by percent of each motor size for single compressor systems as found in the GrocerSmartTM audit data (n=897).

* Condenser Set Point - this represents the set point of the head pressure control valve on the condenser (see section *2.1.2.1 Remote Condenser Systems*). The set point is the same for remote condensers and condensing units
* Compressor Sizing Factor - this represents the ratio of compressor capacity to the amount of attached case load. As this variable had a large effect on the final savings, refrigeration schedules of condensing unit systems were collected (n=21). Analysis of refrigeration schedules showed a median value of 121% sizing factor for low temperature and medium temperature systems.

Table 10 – Final Variables for Condensing Unit

|  |  |  |  |
| --- | --- | --- | --- |
| **Final Model Variables** | | | |
| **Variable** | **Units** | **MT** | **LT** |
| Compressor Efficiency | Mbtu/h-compressor | weighted | weighted |
| Compressor Sizing Factor | % | 121% | 121% |
| Condenser set point | Degree F | 94 | 94 |
| Suction Temperature | Degree F | 15 | -20 |

For details of the other model variables see the detailed model calculations in section 2.1.

### eQUEST/DOE-2.2R Model Inputs

Applying the equations in section 2.1.1 and the final values for the variables found from the sensitivity analysis in section 2.1.2 resulted in the key model inputs described below. These variables are part of the refrigeration design of the model only. Other factors such as HVAC and lighting are described in Table 3 and had negligible impact on the final energy savings results.

**2.1.3.1 Remote Condenser System**

Using the equations discussed in section 2.1.1, a single compressor with remote condenser system was created using four identical, 10 HP motor, single compressors attached to one central condenser for low temperature cases and four identical, 10 HP, single compressors attached to one condenser for medium temperature cases. Four compressors were chosen based on refrigeration system reviews and discussions with refrigeration technicians and PECI field staff. The number of compressors and the compressor motor size should not have any large effect on the energy savings as the savings is being deemed per motor HP. The medium temperature and low temperature systems were modeled separately in order to simplify the results from DOE2.2R model. Typically medium temperature and low temperature compressors would be on separate remote condensers, and if they were on the same condenser because of the simple outside air fan controls on remote condensers, the medium temperature and low temperature systems would have little interactivity. The design conditions were determined using the sensitivity results as described in section 2.1.2.1 and the condenser design temperature difference (TD) was chosen to represent typical design TD. The results of applying the equations in section 2.1.1 and other important model inputs are shown below:

Table 11 - Remote System Compressor/Condenser Design Conditions

|  |  |  |  |  |  |  |  |
| --- | --- | --- | --- | --- | --- | --- | --- |
| **Low Temp** |  | Design conditions  -25F SST and 105F SCT | | |  |  |  |
| Comp motor | Compressor COP | Compressor Capacity | Condenser Capacity | Refrigeration Load | Case length | Cond design TD | Compressor Model for mass curve |
| (HP) |  | (BTU/hr) | (BTU/hr) | (BTU/hr) | (ft) | (deg F) |  |
| 10 | 1.45 | 38,487 | 60,855 | 32,895 | 53 | 10 | 3DS3F46KE-TFC |
|  |  |  |  |  |  |  |  |
| **Medium Temp** |  | Design conditions 20F SST and 105F SCT | | |  |  |  |
| Comp motor | Compressor COP | Compressor Capacity | Condenser Capacity | Refrigeration Load | Case length | Cond design TD | Compressor for mass curve |
| (HP) |  | (BTU/hr) | (BTU/hr) | (BTU/hr) | (ft) | (deg F) |  |
| 10 | 2.49 | 83,784 | 111,712 | 74,474 | 53 | 15 | 9RC1-1015-THC-100 |

**2.1.3.2 Condensing Unit System**

Using the equations discussed above, the condensing unit systems were modeled for compressor nominal motor HP from 1 to 10 HP. Unlike the remote condenser runs, separate models had to be run for a variety of motor sizes. As discussed in section 2.1.2.2 this was done to account for changes in COP for the range of smaller motor sizes. The motor weighting process is described in section 2.1.3.3. The results of applying the equations in section 2.1.1 and other important model inputs are shown in Table 10 below

Table 12 - Condensing Unit System Compressor/Condenser Design Conditions

|  |  |  |  |  |  |  |  |
| --- | --- | --- | --- | --- | --- | --- | --- |
| **Low Temp** |  | Design conditions -25F SST and 105F SCT | | |  |  |  |
| Comp motor | Compressor COP | Compressor Capacity | Condenser Capacity | Refrigeration Load | Case length | Cond design TD | Compressor Model for mass curve |
| (HP) |  | (BTU/hr) | (BTU/hr) | (BTU/hr) | (ft) | (deg F) |  |
| 10 | 1.48 | 46,370 | 77,610 | 38,197 | 62 | 11.7 | 3DS3F46KE-TFC-100 |
| 7.5 | 1.48 | 37,500 | 62,919 | 30,890 | 50 | 16.1 | 3DB3F33KE-TFC-100 |
| 5 | 1.51 | 26,410 | 43,953 | 21,755 | 35 | 16.7 | ZF18K4E-TF5-251 |
| 3 | 1.28 | 12,150 | 21,665 | 10,008 | 16 | 15.9 | LAHB-0311-CAB-100 |
| 2 | 1.30 | 7,650 | 13,547 | 6,302 | 10 | 14.8 | EADA-0200-TAC-100 |
| 1 | 1.27 | 4,165 | 7,447 | 3,431 | 6 | 18.7 | KAJB-0100-CAV-100 |
|  |  |  |  |  |  |  |  |
| **Medium Temp** |  | Design conditions 20F SST and 105F SCT | | |  |  |  |
| Comp motor | Compressor COP | Compressor Capacity | Condenser Capacity | Refrigeration Load | Case length | Cond design TD | Compressor Model for mass curve |
| (HP) |  | (BTU/hr) | (BTU/hr) | (BTU/hr) | (ft) | (deg F) |  |
| 10 | 3.1 | 84,100 | 111,318 | 69,276 | 49 | 21 | 9RC1-1015-THC-100 |
| 7.5 | 2.7 | 56,200 | 76,769 | 46,294 | 33 | 22.5 | MRH4-0760-TFC-100 |
| 5 | 2.6 | 37,600 | 52,070 | 30,972 | 22 | 22 | CRN5-0500-TF5-270 |
| 3 | 2.6 | 23,355 | 32,322 | 19,238 | 14 | 22.5 | CRJ3-0300-PFV-271 |
| 2 | 2.5 | 16,000 | 22,299 | 13,180 | 9 | 20 | CRD1-0200-PFV-272 |
| 1 | 2.4 | 7,440 | 10,601 | 6,129 | 4 | 27.3 | KAR2-0100-CAV-100 |

**2.1.3.3 Condensing Unit- Motor Weighting**

In order to reduce the number of measures to a reasonable number, a single motor size was chosen to represent the savings for condensing unit measure for each climate zone. The results of the trials were weighted according to motor size using motor size data from GrocerSmartTM audit data for stores in CA with single compressor systems (n=897). The weightings are shown in Table 12 in section 4.2.

The results of the motor weighting were compared to the individual motor trials, and the motor size that most closely matched the weighted savings was used as the “representative” baseline for all climate zones. The low temperature model used the 10HP model and the medium temperature system used the 3 HP model.

## 2.2. Demand Reduction Estimation Methodologies

There is no anticipated demand reduction associated with this measure. Since this measure does not provide savings during the peak hours of the year when there are high ambient temperatures, there are no demand reductions for these measures.

## 2.3. Gas Energy Savings Estimation Methodologies

There are no gas energy savings associated with this measure.

# *Section 3. Load Shapes*

## 3.1 Base Case Load Shapes

The base case load shape, characterized by the PG&E E3 Calculator, is “Commercial Refrigeration.”

## 3.2 Measure Load Shapes

There are no measure case load shapes applicable to this measure. The base case shapes are to be used in the cost avoidance calculation.

# 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)** |
| ***REA (retrofit add on)*** | EUL | Calculated as Full Gross Measure Cost | N/A |

The measure costs are based on the following inputs:

Table 13 - Cost Information

|  |  |  |  |  |  |  |
| --- | --- | --- | --- | --- | --- | --- |
| Equipment | Equipment Cost | Installation Labor Cost | Unit | Total | Notes | Cost Source |
| Adjustable Head Pressure Control Valve | $166.40 | $140 | Per Compressor | $307 | 1 control valve per compressor | Sporlan |
| Bypass valve | $79.70 | $140 | Per Compressor | $220 | 1 bypass valve per compressor | Sporlan |
| Filter drier | $41.75 | $18 | Per Compressor | $59 | 1 filter drier per compressor | Sporlan, HVAC/R Wholesaler |
| Balance Port Valve | $129.64 | $35 | Per Evaporator | $165 | One evaporator and one TXV per 4 ft of MT case and per 4 door of LT case | Sporlan, Grainger |

An assumption was made that the average reach-in case has one evaporator and one expansion valve for every four doors (low temperature system) and that the average open multi-deck case has one evaporator and one valve every four feet (medium temperature systems). These are reasonable and conservative assumptions. It was also assumed for the analysis that each compressor is independently plumbed to the condenser so that there is one head pressure control valve per compressor. This is a conservative assumption as some condensers may have a common header going into the condensers and therefore be less costly since only one adjustable head pressure control valve, filter and bypass valve per condenser would be required.

## 4.1 Base Case(s) Costs

The following Measure Application Types is are appropriate to these measures. The Base Case Costs are:

|  |  |  |  |  |  |  |
| --- | --- | --- | --- | --- | --- | --- |
| ***Measure Code*** | **Measure Application Type** | **Baseline** | **Equipment Cost** | **Labor / Installation Cost** | **Maintenance / Other Cost** | **Total Base Case Cost** |
| R320 | REA | Existing | $203.00 | $144.00 | $0 | $347.00 |
| R321 | REA | Existing | $247.00 | $171.00 | $0 | $418.00 |
| R322 | REA | Existing | $106.00 | $51.00 | $0 | $157.00 |
| R323 | REA | Existing | $146.00 | $61.00 | $0 | $207.00 |

*All costs are noted as $ per measure unit*

## 4.2 Measure Case Costs

The following Measure Application Type is appropriate to these measures. The Measure Case Costs are:

|  |  |  |  |  |  |  |
| --- | --- | --- | --- | --- | --- | --- |
| ***Measure Code*** | **Measure Application Type** | **Baseline** | **Equipment Cost** | **Labor / Installation Cost** | **Maintenance / Other Cost** | **Total Measure Case Cost** |
| R320 | REA | Existing | $203.00 | $144.00 | $0 | $347.00 |
| R321 | REA | Existing | $247.00 | $171.00 | $0 | $418.00 |
| R322 | REA | Existing | $106.00 | $51.00 | $0 | $157.00 |
| R323 | REA | Existing | $146.00 | $61.00 | $0 | $207.00 |

*All costs are noted as $ per measure unit*

The cost of this measure per motor HP depends on the number of balance port valves on the system and the size of the compressor. Table 10 - Condensing Unit System Compressor/Condenser Design Conditions shows the total length of display case for low temperature and medium temperature by HP. The assumption for the number of valves per foot of display case and detailed cost information is shown in Table 11. Equations 6 and 7 show the calculation of number of expansion valves for each motor size and cost per motor HP for each motor size as presented in Tables 12 and 13:

Equation 6- Calculation of # expansion valves



ValveMT#= number of expansion valve for the MT cases (valve)

Caselength= Case length (ft) from section 2.1.3

LengthExp= Case length per expansion valve ( 4 ft/valve)



ValveLT#= number of expansion valve for the LT cases (valve)

CaseDoors= Number of case door (doors)

DoorExp= Number of case doors per expansion valve ( 4 doors/valve)

Equation 7- Calculation of Measure Cost per HP



CostHP = The total cost per HP ($/HP)

Costcomp= Cost per compressor ($)

Costvalve= Cost per valve ($/valve)

Valve#= Number of expansion valves (valve)

CompHP= Compressor hp (HP)

The costs for Condensing Units are weighted and shown in Table 12.

Table 14 - Condensing Unit System Final Cost

|  |  |  |  |  |  |  |
| --- | --- | --- | --- | --- | --- | --- |
| **Cost for LT by motor size assuming avg # exp valves / motor** | | | | | | |
| HP | # of exp valves | Equip Cost per compressor | Labor Cost per compressor | Equip Cost per motor nom. HP | Labor Cost per motor nom. HP | Weighting |
| 10 | 6 | $1,066 | $508 | $106 | $51 | 6% |
| 7.5 | 5 | $937 | $473 | $125 | $63 | 13% |
| 5 | 4 | $807 | $438 | $162 | $88 | 18% |
| 3 | 1.5 | $483 | $350 | $161 | $117 | 24% |
| 2 | 1 | $418 | $333 | $209 | $166 | 24% |
| 1 | 1 | $418 | $333 | $418 | $333 | 15% |
|  | **LT Condensing Unit FHP Measure Cost / HP** | | | |  | **$347** |
|  | | | | | | |
| **Cost for MT by motor size assuming avg # exp valves / motor** | | | | | | |
| HP | # of exp valves | Equip Cost per compressor | Labor Cost per compressor | Equip Cost per motor nom. HP | Labor Cost per motor nom. HP | Weighting |
| 10 | 9 | $1,455 | $613 | $146 | $61 | 3% |
| 7.5 | 6 | $1,066 | $508 | $142 | $68 | 9% |
| 5 | 4.5 | $871 | $455 | $174 | $91 | 13% |
| 3 | 2.5 | $612 | $385 | $204 | $128 | 22% |
| 2 | 1.5 | $483 | $350 | $241 | $175 | 34% |
| 1 | 1 | $418 | $333 | $418 | $333 | 19% |
|  | **MT Condensing Unit FHP Measure Cost / HP** | | | |  | **$418** |

Although the remote condenser motor sizes can be larger than 10 HP, the costs calculated are conservative for all sizes above 10 HP. Costs are shown in Table 13 represent the remote condenser unit and is assumed to be a 10 HP compressor.

Table 15– Remote Condenser Final Cost

|  |  |  |  |  |  |  |
| --- | --- | --- | --- | --- | --- | --- |
| **LT Remote Condenser FHP Measure Cost** | | | | | | |
| HP | # of exp valves | Equip Cost per compressor | Labor Cost per compressor | Equip Cost per motor nom. HP | Labor Cost per motor nom. HP | Total Cost per motor nom. HP |
| 10 | 6 | $1,066 | $508 | $106 | $51 | $157 |
|  | | | | | | |
| **MT Remote Condenser FHP Measure Cost** | | | | | | |
| HP | # of exp valves | Equip Cost per compressor | Labor Cost per compressor | Equip Cost per motor nom. HP | Labor Cost per motor nom. HP | Total Cost per motor nom. HP |
| 10 | 9 | $1,455 | $613 | $146 | $61 | $207 |

## 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** |
| REA | 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 REA with a single baseline, so the Full Measure Cost (FMC) is represented by the equation below:

FMC = Measure Equipment Cost + Measure Labor Cost

# *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 REA**.** There is no base case to which to compare the measure, so the Incremental Measure Cost (IMC) is represented by the equation below:

IMC = Measure Equipment Cost + Measure Labor Cost

*IMC = $ per (unit)+ $ per (unit) = $ per (unit)*

**Summary Table for Section 4**

|  |  |  |  |  |  |
| --- | --- | --- | --- | --- | --- |
| **Measure ID** | **Measure Application Types** | **Base Case Total Cost** | **Measure Case Total Cost** | **Full Measure Case Cost** | **Incremental Measure Cost** |
| R320 | REA | $0.00 | $347.00 | $347.00 | $347.00 |
| R321 | REA | $0.00 | $418.00 | $418.00 | $418.00 |
| R322 | REA | $0.00 | $157.00 | $157.00 | $157.00 |
| R323 | REA | $0.00 | $207 | $207 | $207 |

The incremental and full measure costs are the same for retrofit measures. The incremental costs are the difference between the measure cost and the base cost as shown in Table 14:

Table 16 - Summary of Final Costs

|  |  |
| --- | --- |
| **MEASURE RUN NAME** | **INCREMENTAL AND FULL MEASURE COSTS** |
| **($ / HP)** |
| FHP Single, LT Condensing Unit | $347.00 |
| FHP Single, LT Condensing Unit | $418.00 |
| FHP Single, LT Remote  Condenser | $157.00 |
| FHP Single, MT Remote  Condenser | $207.00 |

# References:

1. [↑](#endnote-ref-2)
2. 2004-2005. Database for Energy Efficiency Resource (DEER). 2005. Itron, Inc,. p.7-87. [↑](#endnote-ref-3)
3. California Code of Regulations, Title 20: Division 2, Chapter 4, Article 4, Sections 1601 – 1608: Appliance Efficiency Regulations, p. 1. [↑](#endnote-ref-4)
4. California Code of Regulations, Title 24: Part 6, Section 100(a)2. Edition: May 2012, 15 Day, p. 32. [↑](#endnote-ref-5)
5. Sep 28 2009, BPA EnergySmart Grocer Program Process and Impact Evaluations, Summit Blue Consulting, p.67-68 [↑](#endnote-ref-6)
6. 2008 DEER [EUL/RUL Values and Summary Documentation](http://www.deeresources.com/deer0911planning/downloads/EUL_Summary_10-1-08.xls) (Updated 10 October 2008). [↑](#endnote-ref-7)
7. http://www.emersonclimate.com/en-US/resources/online\_product\_information/Pages/online\_product\_information.aspx [↑](#endnote-ref-8)
8. Emerson Climate Technologies bulletin HP/HPC Headmaster Head Pressure Controls, p1. [↑](#endnote-ref-9)
9. The GrocerSmartTM auditing and energy simulation software tool supports PECI’s rate payer funded energy efficiency

   programs by enabling audits of the major electromechanical and mechanical systems of commercial retail refrigeration equipped

   facilities. [↑](#footnote-ref-2)