Work Paper SCE17RN003

**Revision 2**

**Southern California Edison**

**Insulation of Bare Refrigeration Suction Lines**

# At-a-Glance Summary

|  |  |
| --- | --- |
| **Measure Codes** | RF-82221, RF-46935 |
| **Measure Description** | Bare Suction Lines for Walk-in Coolers Insulation (RF-82221)  Bare Suction Lines for Walk-in Freezers Insulation (RF-46935) |
| **Base Case Description** | Refrigeration systems with un-insulated suction refrigeration pipes outside of the refrigerated space |
| **Units** | Per Linear Foot |
| **Energy Savings** | Refer to Excel Calculation, Attachment #1 |
| **Full Measure Cost ($/unit)** | Refer to Excel Calculation, Attachment #1 |
| **Incremental Measure Cost ($/unit)** | Refer to Excel Calculation, Attachment #1 |
| **Effective Useful Life** | 6.7 years, capped at the RUL of the host equipment (suction pipe). |
| **Measure Installation Type** | Retrofit Add-On (REA) |
| **Net-to-Gross Ratio** | 0.45 (DEER NTGR ID: NonRes-sAll-mPipeIns-deemed) |
| **Important Comments** | This work paper has a complementary Ex Ante Database data set that will be provided in a separate submission to the California Public Utilities Commission (CPUC). |

# Revision History

|  |  |  |  |
| --- | --- | --- | --- |
| **Rev** | **Date** | **Author** | **Summary of Changes** |
| 0 | 11/18/16 | Ramon Yll-Prous/TRC Energy Services | - This work paper is an update of SCE13RN003.1  - New calculation template for 2017 program year  - Baseline description added based on Title 24 (2016)  - Added all (16) California Climate zones to calculation template  - Updated costs based on RSMeans 2016  - Revised life per Retrofit Add on Guidance Document issued on April 20, 2015 |
| 1 | 10/16/17 | Yin Yin Wu/BASE Energy, Inc. | - DEER2017 values are checked. No change is made.  - Costs are updated to have relevant cost for each measure code, instead of the same costs for all measure codes.  - The work paper calculation template is updated using Version 6.7.4 |
| 2 | 10/19/18 | Jesse Manao (SCE) | - Updated NTG value per Resolution E-4952 issued on 10/12/18  - Workpaper calculation template is updated with Version 6.7.5 |

# Commission Staff and Cal TF Comments

|  |  |  |  |  |  |
| --- | --- | --- | --- | --- | --- |
| **Rev** | **Party** | **Submittal Date** | **Comment Date** | **Comments** | **WP Developer Response** |
|  |  |  |  |  |  |

Cal TF website: <http://www.caltf.org/>

# Section 1. General Measure & Baseline Data

## 1.1 Measure Description & Background

**Base, Standard, and Measure Cases**

|  |  |
| --- | --- |
| **Case** | **Description of Typical Scenario** |
| Measure | Refrigeration systems with insulated suction refrigeration pipes outside of the refrigerated space |
| Existing Condition | Refrigeration systems with un-insulated suction refrigeration pipes outside of the refrigerated space |
| Code/Standard | N/A |
| Industry Standard Practice | N/A |

Measures and Codes

|  |  |  |  |  |
| --- | --- | --- | --- | --- |
| **Measure Codes** | | | | **Measure Name** |
| SCG | SDG&E | SCE | PG&E |
|  |  | RF-82221 |  | Bare Suction Lines for Walk-in Coolers Insulation |
|  |  | RF-46935 |  | Bare Suction Lines for Walk-in Freezers Insulation |

This measure is applicable to bare suction lines insulated with closed-cell nitrite rubber or equivalent with at least 3/4 inch for medium-temperature and 1 inch for low-temperature systems. The suction line must be no more than 1 5/8 inch in diameter. Insulation R-values must be greater than equal to R-3.2 and R-4.3 for medium and low-temperature lines respectively. This measure also applies to the suction lines of medium-temperature (above 32oF) and low-temperature (below 32oF) refrigeration systems, and un-insulated suction lines in small refrigeration systems of restaurants (both fast food and sit-down), grocery stores, and food/convenience stores.

## 1.2 Technical Description

Insulation impedes heat transfer from the cold refrigeration lines, thereby reducing the undesirable system superheat. Excess superheat will increase the heat of compression and decrease the heat rejection ability of the condenser. Insulating suction lines will reduce the superheat and thereby, reduce compressor power and energy use.

Also, according to the California Commercial Saturation Survey [C], page 8-21:"The other measure reported here is Insulated Suction Lines. These lines contain refrigerant flowing into the compressor. The insulation on the line, from an efficiency perspective, ensures that no additional heat is being added into the system. Additional heat that may be added to the system may, not only cause the compressor to run poorly, but will also need to be ultimately removed by the system, increasing the energy usage. **Insulated suction lines are found on just under a quarter of all systems.**"

## 1.3 Installation Types and Delivery Mechanisms

The program type/install type for these measures is Retrofit Add-On (REA).

**Installation Type Descriptions**

|  |  |  |  |  |
| --- | --- | --- | --- | --- |
| **Installation Type** | **Savings** | | **Life** | |
| 1st Baseline (BL) | 2nd BL | 1st BL | 2nd BL |
| Retrofit Add-On (REA) | Above Customer Existing | N/A | EUL | N/A |

The delivery mechanisms are:

* Financial Support / Down-Stream Incentive – Deemed
* Financial Support / Direct Install
* Financial Support / On-bill Finance – loan
* Partnership / Down-Stream Incentive – Deemed
* Partnership / Direct Install
* Partnership / On-bill Finance - Loan

A delivery mechanism is a delivery method paired with an incentive method. Delivery mechanisms are used by programs to obtain program participation and energy savings.

**Delivery Method Descriptions**

|  |  |
| --- | --- |
| **Delivery Method** | **Description** |
| Financial Support | The program motivates customers, through financial incentives such as rebates or low interest loans, to implement energy efficient measures or projects. |
| Partnership | The program implements projects through a partnership between the utility and an institutional, government, or community-based organization. |

**Incentive Method Descriptions**

|  |  |
| --- | --- |
| **Incentive Method** | **Description** |
| Direct Install | The program implements energy efficiency measures for qualifying customers, at no cost to the customer. |
| Down-Stream Incentive-Deemed | The customer installs qualifying energy efficient equipment and submits an incentive application to the utility program. Upon application approval, the utility program pays an incentive to the customer. Such an incentive may be deemed or customized. |
| Down-Stream Incentive-Deemed-On-bill Finance – Loan (OBF) | The program offers financing for the cost of an efficient measure as part of the utility bill. This can be an add-on option to an existing program or can serve as an organizing principle for its own program. |

## 1.4 Measure Parameters

### 1.4.1 DEER Data

DEER Difference Summary

|  |  |
| --- | --- |
| **DEER Item** | **Used for Workpaper?** |
| Modified DEER methodology | No |
| Scaled DEER measure | No |
| DEER Base Case | No |
| DEER Measure Case | No |
| DEER Building Types | Yes |
| DEER Operating Hours | No |
| DEER eQUEST Prototypes | No |
| DEER Version | N/A |
| Reason for Deviation from DEER | DEER does not contain this type of measure. |
| DEER Measure IDs Used | N/A |

**Net-to-Gross Ratio**

The NTG values were obtained using the DEER READI tool. The relevant NTG values for the measures in this work paper are in the table below.

|  |  |  |  |  |  |
| --- | --- | --- | --- | --- | --- |
| **NTGR ID** | **Description** | **Sector** | **BldgType** | **Measure Delivery** | **NTGR** |
| NonRes-sAll-mPipeIns-deemed | All other EEMs with no evaluated NTGR; existing EEM in programs with same delivery mechanism for more than 2 years | NonRes | Any | Any, except upstream. | 0.45 |

**Spillage Rate**

Spillage rates are not tracked in work papers; they are tracked in an external document which will be supplied to the Commission Staff.

**Installation Rate**

The IR values were obtained using the DEER READI tool. The relevant IR values for the measures in this work paper are in the table below.

|  |  |  |  |  |  |
| --- | --- | --- | --- | --- | --- |
| **GSIA ID** | **Description** | **Sector** | **BldgType** | **ProgDelivID** | **GSIAValue** |
| Def-GSIA | Default GSIA values | Any | Any | Any | 1 |

**Effective and Remaining Useful Life**

The EUL and RUL values were obtained using the DEER READI tool. DEER defines the RUL as 1/3 of the EUL value. Additionally, the draft Retrofit Add on Guidance Document issued on April 20, 2015, defines the EUL for REA measures as: “the measure effective useful life (EUL) of the REA measure is the lesser of the remaining useful life (RUL) of the existing system to which the measure is added to or the EUL for the REA measure.” Also, according to Resolution E-4818 [A], the EUL of REA measures is capped at the RUL of the host equipment.

Refrigeration-grade copper tubing is the preferred material for refrigerant systems. According to an article [D] by the Copper Development Association Inc., the acceptable useful service life of the copper tubing was defined as having a remaining wall thickness of greater than 80% of the nominal at the end of 20 years in service. Therefore, the EUL of bare refrigerant pipe is considered to be 20 years for refrigerant pipe with a RUL of 6.7 years. The EUL from DEER2017 of the Refrigeration Insulation for Bare Suction Lines measure is 11 years. However, since the EUL of REA measures is capped at the RUL of the host equipment, the EUL of the measure is capped at 6.7 years.

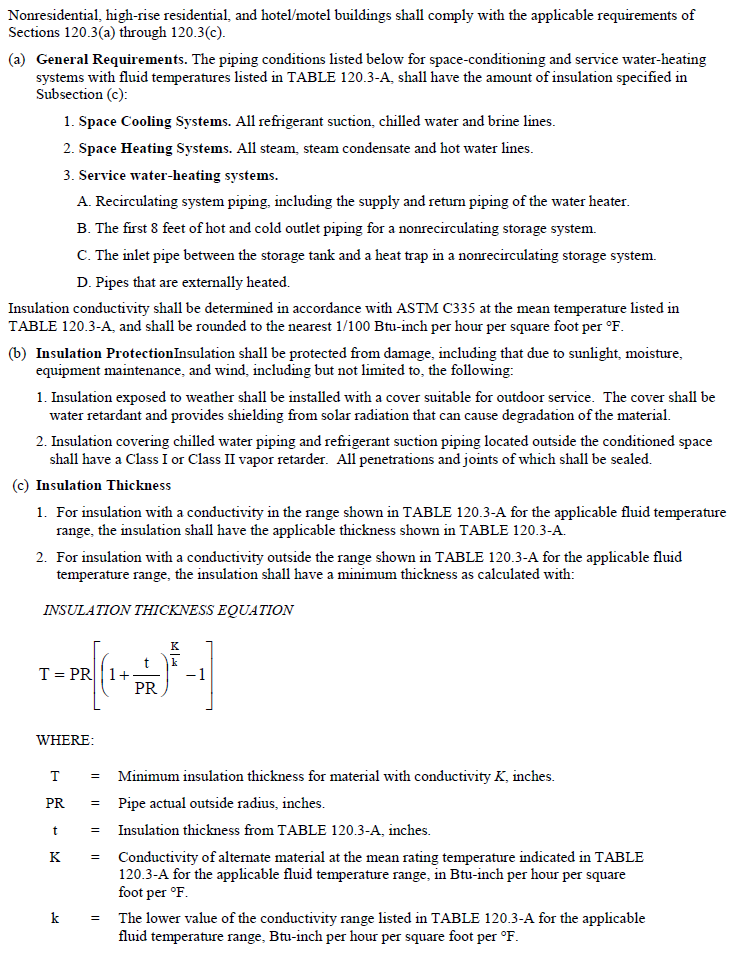
|  |  |  |  |  |  |
| --- | --- | --- | --- | --- | --- |
| **EUL ID** | **Description** | **Sector** | **UseCategory** | **EUL (Years)** | **RUL (Years)** |
| RefgWrhs-SLIns | Refrigeration Insulation for Bare Suction Lines | Com | ProcRefrig | 6.7\* | N/A |

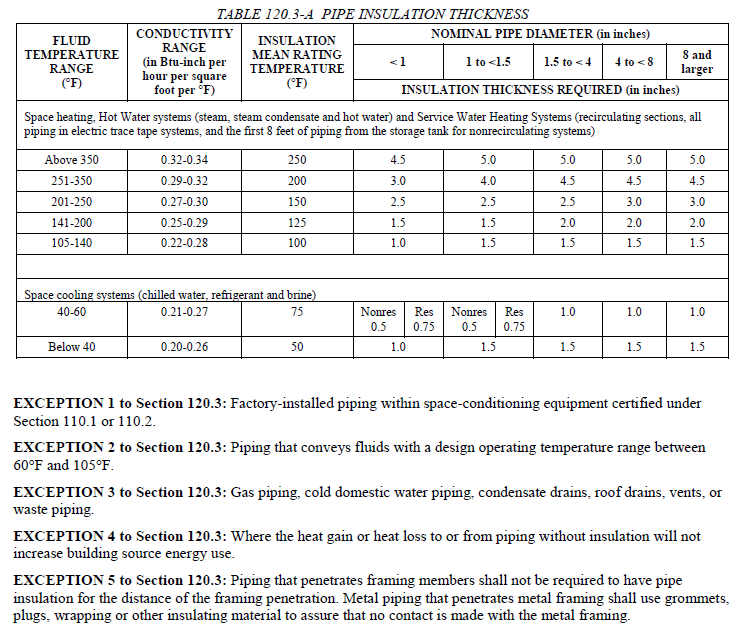
\* Capped at the RUL of the host equipment. See paragraph above for explanation.

### 1.4.2 Codes and Standards Analysis

Title 24 (2016), Section 120.3 [496] requires refrigeration suction line insulation for space cooling systems, which does not apply to commercial refrigeration system. Section 120.6 (b) is for commercial refrigeration, and it includes requirements for different components of the refrigeration systems, including: condensers serving refrigeration systems, compressor systems, refrigerated display cases and refrigeration heat recovery. However, Section 120.6 (b) does not have requirements for refrigeration pipe insulation. Therefore, the measure evaluated in this work paper is not impacted by code standards.

Section 120.3 presented below is for information purposes only.





Code Summary

|  |  |  |
| --- | --- | --- |
| **Code** | **Reference** | **Effective Dates** |
| Title 24 (2016) | Section 120.3 – Requirements for Pipe Insulation | January 1, 2017 |
| Title 24 (2016) | Section 120.6 (b) – Mandatory Requiremtns for Commercial Refrigeration | January 1, 2017 |

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

This workpaper used ASHRAE Fundamentals Handbooks, Refrigeration documents, and Thermal Analysis Software to develop energy savings and demand reduction calculations. Refer to Section 2 for details.

## 1.6 Data Quality and Future Data Needs

N/A

# Section 2. Calculation Methodology

## 2.1 Energy Savings Estimation Methodologies

This section starts with assumptions used for the refrigeration systems of commercial walk-in coolers and freezers. The overall approach to calculate demand and energy savings are presented next. Following that, the methodology for calculating the heat gain through bare and insulated suction lines, and power usage for each scenario are outlined. The last section describes step-by-step demand and energy savings calculations.

### 

### Assumptions

The following are the key assumptions used in this analysis:

1. For low-temperature:
   * design saturated evaporating temperature (SET) of -10oF, per a typical delta T of 10 oF
   * design box or discharge air temperature of o0F, typical for freezer per U.S. Food & Drug Administration [E]
   * pipe nominal diameter of 5/8” [84] – actual pipe diameter of 3/4” [395]
   * insulation thickness of 1.25” [396]
   * closed-cell, flexible elastomeric thermal insulation with a thermal conductivity of 0.27 Btu-in/hr-ft2-oF [396]
   * 40 feet of total suction line [84]
   * Refrigerant: R-22
   * condenser temperature difference (TD) of 10oF
   * no change in compressor run time due to suction line insulation
2. For medium-temperature:

* design saturated evaporating temperature (SET) of +28oF, per a typical delta T of 10 oF
* design box or discharge air temperature of +38oF, typical for cooler per U.S. Food & Drug Administration [E]
* pipe nominal diameter of 5/8” [84] – actual pipe diameter of 3/4” [395]
* insulation thickness of 3/4” [396]
  + closed-cell, flexible elastomeric thermal insulation with a thermal conductivity of 0.27 Btu-in/hr-ft2-oF [396]
  + 40 feet of total suction line [84]
  + Refrigerant: R-22
  + condenser temperature difference (TD) of 15oF
  + no change in compressor run time due to suction line insulation

1. Average wind speed of 7.5 mph for exposed refrigeration lines [155, p. 25.2, Table 1]
2. Wind speed of 0 mph for non-exposed refrigeration lines

The table below summarizes design dry-bulb temperatures for all 16 CTZs. Also, the following table provides a description and lists the representative city in each climate zone.

Climate Zone Design Dry Bulb Temperature and Representative City



Source: ASHRAE 1982. Climatic Data for Region X. [397]

Based on the design dry-bulb temperatures and condenser TD, the saturated condensing temperature (SCT) for walk-in coolers and freezers for each climate zone was determined using:

 medium-temperature (coolers)

 low-temperature (freezers)

where

DBamb = design ambient dry-bulb temperature based on climate zone

### Overall Approach

The following outlines the approach used to estimate demand and energy savings for insulating bare suction lines of walk-in coolers and freezers applicable to restaurants and small grocery stores:

1. Heat transfer analysis:

Conduct a heat transfer analysis for both bare (baseline) and insulated (post-retrofit) suction lines of walk-ins to determine:

* 1. The overall heat transfer coefficient
  2. Heat gain through bare and insulated pipes

1. Refrigeration cycle analysis:

Conduct refrigeration cycle analysis for both bare (baseline) and insulated (post-retrofit) suction lines to:

* 1. Correlate refrigerant temperature and specific heat (Cp)
  2. Determine total system superheat by calculating refrigerant temperature at the suction port of the compressor for both bare and insulated suction lines
  3. Determine heat of compression for both bare and insulated suction line scenarios
  4. Determine the compressor power usage for both bare and insulated suction line scenarios

1. Compressor power savings per unit of measure (ΔkW/linear-ft):
   1. Determine demand savings by using bare and insulated suction line compressor power usage
   2. Determine demand savings per linear-ft of exposed suction lines for both walk-in coolers and freezers
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, defrost periods and weather factor
3. Compressor energy savings per unit of measure (ΔkWh/ft):
   1. Determine energy savings by using demand savings and EFLH
   2. Repeat energy savings calculation for all CTZs for both coolers and freezers and compile results

### 

### Heat Transfer Analysis

This section discusses necessary and crucial steps or methodologies in calculating the total heat gain through bare copper suction lines and insulated suction lines. These steps include:

1. Conducting heat transfer analysis, which includes
   * 1. Calculating heat transfer coefficients
     2. Calculating overall heat transfer coefficient
2. Calculating heat gain

#### Step 1: Heat transfer coefficients

Conducting a heat transfer analysis will identify key heat transfer coefficients. These heat transfer coefficients will ultimately determine the overall heat transfer coefficient, which then will be used to determine the heat gain for each scenario.

*Bare Suction Line*:

The heat transfer components for bare suction pipes include conduction through the thickness of pipe, and convection and radiation through the surface of the pipe. The conduction through the thickness of pipe is a function of the pipe’s inner radius (ri), the pipe’s outer radius (ro) and the thermal conductivity of the pipe (kpipe). The convection and radiation, however, act in parallel at the surface of the pipe. Accordingly, the thermal circuit can be constructed as follows (Figure 1):



Figure 1 Network Representation of Radiative, Convective and Conductive Exchange between Ambient and Vapor Refrigerant for Bare Suction Pipes

However, for simplicity purposes, *the pipe’s wall thermal resistance will be neglected*. It is assumed that the surface of the pipe has the same temperature as the vapor refrigerant. As shown in Figure 2, the only components that will contribute to the heat gain of vapor refrigerant are convection and radiation. Therefore, the total thermal resistance between a bare suction pipe and ambient air include effective resistance associated with convection and radiation.

Figure 2 Schematic and Network Representation of Radiative and Convective Exchange between Ambient and Vapor Refrigeration for Bare Suction Pipes

The convective and radiant heat transfer coefficients for bare pipes can be calculated using [398, p22.17]:



where,

hconvection = convective heat transfer coefficient, Btu/hr-ft2-oF

C = constant depending on shape and heat flow condition

(C = 1.016 for horizontal pipes)

d = pipe diameter, inches

tavg = average temperature of air film, oF

Δt = surface to air temperature difference, oF

V = wind speed, mph

To estimate the average air film temperature, the ambient air temperature and the pipe’s surface temperatures were used. Since the assumption is that the pipe’s surface temperature is the same as the vapor refrigerant temperature, the pipe’s surface temperature equated to the refrigerant vapor temperature. That is:



where,

ta = air temperature, oF

ts = surface temperature, oF

The following illustrates radiation heat transfer coefficient.



where,

hradiation = radiant heat transfer coefficient, Btu/hr-ft2-oF

Ɛ = surface emittance, 0.44 for dull bare copper pipe [398, p22.18, Table 12]

The convective and radiant heat transfer coefficients are used to determine the total thermal resistance (RTotal). The total thermal resistance for heat flow through resistances in parallel can be obtained using: [155, p3.18]

 🡺 

where,

RTotal = total thermal resistance, oF-ft2-hr/Btu

Once the total thermal resistance is determined, the overall heat transfer coefficient, U, can be obtained by following relationship:

 🡺 

where,

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

*Insulated Suction Lines*:

The heat transfer components for insulated suction pipes include conduction through the thickness of both the pipe and of the insulations, and convection and radiation through the pipe’s insulation surface. The following thermal circuit illustrates each component.



Figure 3 Network Representation of Radiative, Convective and Conductive Exchange between Ambient and Vapor Refrigerant for Insulated Suction Pipes

However, for simplicity purposes the following assumptions are made:

1. The pipe’s wall thermal resistance is negligible
2. The insulation exterior surface temperature is equal to ambient air temperature   
   (Tinsulation = Tambient), therefore:
   1. Convection through the pipe’s insulation surface is omitted
   2. Radiation through the pipe’s insulation surface is omitted

Thus, the only component that will contribute to the heat gain of vapor refrigerant is the thermal resistance of insulation or conduction through the insulated material.



Figure 4 Schematic and Network Representation of Convective Exchange between Ambient and Vapor Refrigerant through Insulation

The thermal resistance (Rcond) associated with using insulation is a function of the inner radius of insulation (in this case pipe radius), outer radius of insulation, and thermal conductivity of the insulation. The following illustrates the relationship: [398, p20.9]



where,

Rcond-insulation = conduction through insulation, oF-ft2-hr/Btu

rsurface = outer radius of insulation, inches

rpipe = inner radius of insulation or pipe radius, inches

kinsulation = thermal conductivity of insulation, Btu-in/hr-ft2-oF

It is assumed that the insulation material is flexible, closed-cell elastomeric with thermal conductivity (k value) of 0.27 Btu-in/hr-ft2-oF [396]. Also, note that rsurface can be determined by adding radius of the pipe and thickness of the insulation. In other words:

rsurface = rpipe + (insulation thickness)

Thus, the overall heat transfer coefficient, U, can be obtained as follows:

 OR 

where,

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

#### Step 2: Heat Gain

The heat gain is a function of overall heat transfer coefficient, surface area and temperature difference. The following illustrates this relationship: [155, p3.18]



where,

Q = heat gain, Btu/hr

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

A = surface area, ft2

ta = ambient temperature, oF

ts = vapor refrigerant temperature, oF

The surface area for cylindrical shapes (i.e., pipes) can be obtained by using the following relationship:

For bare suction lines: A = 2πrpipel

For insulated suction lines: A = 2πrsurfacel

where,

rpipe = pipe radius, ft

rsurface = outer radius of insulation, ft

l = pipe length, ft

### Refrigeration Cycle Analysis

This section discusses analytical methodologies for correlating heat gain through suction pipes to increase in overall system superheat. The increase in overall system superheat is then correlated to compressor power increase. These analytical methodologies are supported by experimental data. The following steps illustrate the methodologies for correlating heat gain and superheat, as well as superheat and compressor power.

#### Step 1: Average refrigerant mass flow rates

From the compressor manufacturer’s catalog, two sets of 0.5, 0.75 and 1.0 horsepower (hp) reciprocating compressors were selected to represent both low- and medium-temperature compressors [399]. The following, lists the selected compressor models for both medium- and low-temperature applications using refrigerant R-22:

Medium-temperature compressor models: KANB-0050, KAMB-0075, KAJB-0100

Low-temperature compressor models: HAG2-0050, KAN2-0075, KAR2-0100

Accordingly, the arithmetic average mass flow rates at various SET and SCT were calculated for the selected low- and medium-temperature compressors. The table shown below shows the average mass flow rates of refrigerant for medium-temperature compressors. The table shown shows the average mass flow rates of refrigerant for low-temperature compressors. These mass flow rates (corresponding to appropriate SET and SCT) were later used to calculate the increase in superheat as a function of heat gain for each climate zone for both walk-in coolers (medium-temperature) and walk-in freezers (low-temperature).

**Average Mass Flow Rate of R-22 for Selected Medium-Temperature Compressors**



**Average Mass Flow Rate of R-22 for Selected Low-Temperature Compressors**



#### Step 2: Correlation between specific heat of refrigerant (Cp) and temperature

Since the specific heat of refrigerant (Cp) is a function of refrigerant temperatures, refrigerant Cp was correlated to refrigerant temperatures. The data used to find the correlation between Cp and the refrigerant temperature were based on thermo physical properties of R-22 [155, p20.5]. Figure 5 and its accompanying equation illustrate this correlation.



**Figure 5 Correlation between Refrigerant Specific Heat and Temperature for R-22**

#### Step 3: Temperature at compressor inlet

Once the methodology for heat gain and correlation between Cp and refrigerant vapor temperatures were determined, the temperature at the inlet of the compressor was determined for both bare and insulated suction lines. The entire suction line length was segmented and for each segment heat gain, and corresponding temperature and Cp were calculated. The following equation illustrates the relationship between the heat gain (), average refrigerant mass flow rate () from step 1, specific heat (Cp), and temperature increase (ΔT) due to heat transfer between the ambient air and the suction line [400, p18].



where,

 = heat gain through the suction pipe (Btu/hr)

 = specific heat of vapor refrigerant R22 (Btu/lb-ºF)

 = change in temperature of the refrigerant (oF)

 = mass flow rate of the refrigerant (lb/hr)

The necessary steps to estimate the vapor refrigerant temperature at the inlet of the suction port of the compressor are outlined below:

1. Determine the initial vapor refrigerant temperature for the first segment of the suction pipe assuming a design superheat of 7oF and using SET of refrigerant. This first segment of suction pipe represents the suction pipe run right after the evaporator coil. That is:

Tinitial = SHdesign + SET

where,

Tinitial  = initial temperature of refrigerant at the beginning of segment (oF)

SHdesign = design evaporator superheat (oF)

SET = saturated evaporating temperature (oF)

1. Determine the Cp for the first segment of suction pipe using the correlation (from step 2) between Cp and initial vapor temperature obtained from step i.
2. Determine the heat gain for the first segment of suction pipe using initial vapor temperature (step i) and climate zone information using methodologies discussed in “Heat Transfer Analysis” section.
3. Determine the final vapor refrigerant temperature for the first segment of the suction pipe using initial vapor refrigerant temperature (step i), refrigerant Cp (step ii), heat gain (step iii), and refrigerant mass flow rate (step 2). The following equation shows the relationship between these parameters.



where,

Tfinal = final temperature of refrigerant at the end of segment (oF)

 = heat gain through a segment of the suction pipe (Btu/hr)

 = mass flow rate of the refrigerant (lb/hr)

 = specific heat of refrigerant R-22 at a segment of suction pipe (Btu/lb-ºF)

Tinitial  = initial temperature of refrigerant at the beginning of segment (oF)

**Note:** The final vapor refrigerant temperature for the first segment is essentially equal to the initial vapor refrigerant temperature for the second segment of the suction pipe.

1. Repeat steps ii, iii and iv, discussed above for subsequent segments. These steps are repeated until the suction pipe’s last segment where the vapor refrigerant enters the suction port of the compressor or the vapor refrigerant temperature equals ambient temperature.

#### Step 4: Enthalpy and entropy of refrigerant at inlet and outlet of the compressor

Using the SCT and SET of the system, the corresponding discharge and suction pressures were found using refrigerant property software [401].

Using the suction pressure and the temperature of the refrigerant at the inlet of the compressor (from step 2), the enthalpy at the inlet of the compressor was found via refrigerant property software for R-22 [401]. Similarly, the entropy at the inlet of the compressor was found using the suction pressure and the refrigerant temperature at the compressor inlet using R-22 refrigerant property software [401].

Assuming that the compressor performance is constant entropy, the entropy at the compressor outlet can be assumed to be the same as the compressor inlet. Using this entropy and the discharge pressure of the system, the corresponding enthalpy at the compressor outlet was found using R-22 refrigerant property software [401].

#### Step 5: Heat of compression

Once the enthalpies at the inlet and outlet of the compressor were determined (step 3), the heat of compression or work of compression was calculated using the following equation [188, p34.12].



where,

 = heat of compression of the compressor (Btu/lb)

 = refrigerant enthalpy at the outlet of the compressor (Btu/lb)

 = refrigerant enthalpy at the inlet of the compressor (Btu/lb)

#### Step 6: Compressor power

Once the heat of compression (step 4) and mass flow rate of refrigerant (step 1) were determined, the compressor power usage for bare and insulated suction lines was obtained. Inefficiencies of the compressor were also accounted for in the energy balance equation. The overall efficiency of the compressor includes isentropic, motor and mechanical efficiencies. For the purpose of this analysis an overall compressor efficiency of 0.5355 was used [402]. The following illustrates the equation for calculating compressor power usage as a function of refrigerant mass flow rate, heat of compression and efficiency of the compressor [403, p223].



where,

kWcomp = compressor power usage (kW)

 = mass flow rate of the refrigerant (lb/hr)

 = heat of compression of the compressor (Btu/lb)

ηoverall = overall efficiency of the compressor

k = conversation factor (3,413 Btu/hr/kW)

**Note:** Steps 3 through 6 were repeated for both bare and insulated suction line scenarios for walk-in coolers and freezers.

### Methodology for Calculating Demand and Energy Savings

This section discusses the necessary steps or methodologies for calculating the demand and energy savings due to insulating bare suction lines. The previous section detailed methodologies for calculating compressor power usage for bare and insulated suction lines.

#### Step 1: Compressor power savings (ΔkW)

Once the compressor power usage was determined for bare and insulated suction lines for walk-in coolers and freezers, the compressor power savings were calculated using the following equation:

ΔkWcomp = kWcomp-bare – kWcomp-insulated

where,

ΔkWcomp = demand savings due to insulating bare suction lines (kW)

kWcomp-bare = compressor power usage for bare suction line scenario (kW)

kWcomp-insulated = compressor power usage for insulated suction line scenario (kW)

Once the total power savings were determined, the obtained value was divided by total linear-feet of suction line. In this case, a length of 40 linear-feet of was used.

#### Step 2: Equivalent-full-load hours (EFLH) of operation

The equivalent-full-load hours (EFLH) was determined by multiplying annual available operation hours (8,760) by the overall duty cycle factor. Duty cycle is a function of the 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 part-load ratio (PLR) and is determined by subtracting PLR from 1. Note that PLR is the ratio of total cooling load to compressor capacity. It is a common practice for refrigeration systems to be designed with a PLR of 87%. The defrost factor depends on the number and duration of defrost. The weather factor, however, is a function of CTZ. Using DOE-2 simulation results for a typical supermarket, the weather factors were determined for each CTZ (see Table 6) [404, pp20-21, Table 2].

Duty cycle = Capacity factor x Defrost factor x Weather factor

where,

Capacity factor = function of PLR, (1 - PLR) or (1 – 0.87)

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

Defrost factor = for coolers, 10.0% (2.4 hrs/ 24 hrs), (1 – 0.1)

Weather factor = function of CTZ (see discussion below)

To estimate the weather factor for each CTZ, the annual energy usage of a refrigeration system for a typical supermarket in each CTZ was used. This was based on a DOE-2 computer simulation [404, pp20-21, Table 2]. Using CTZ 15 as a benchmark with an 85% weather factor, the weather factors for the other 15 CTZs were estimated. The following equation shows this methodology. The table below illustrates the annual energy usage of refrigeration for each CTZ and corresponding weather factors.

WFCTZ =  WFCTZ-15

where,

WFCTZ = weather factor for each CTZ

Annual kWh CTZ = annual energy usage of refrigeration system for each CTZ

Annual kWh CTZ15 = annual energy usage of refrigeration system for CTZ 15

WFCTZ-15 = weather factor for CTZ 15, 85%

**Weather Factor According to Climate Zone**



Accordingly, duty cycles and EFLH were determined as follows:

Duty cycleLT = (1 – PLR) x DFLT x WFCTZ = (1 – 0.87) x (1 – 0.05) x WFCTZ

Duty cycleMT = (1 – PLR) x DFMT x WFCTZ = (1 – 0.87) x (1 – 0.1) x WFCTZ

EFLHLT = 8,760 x Duty cycleLT hrs/yr

EFLHMT = 8,760 x Duty cycleMT hrs/yr

where,

Duty CycleLT = duty cycle for freezers (low-temperature systems)

Duty CycleMT = duty cycle for coolers (medium-temperature systems)

PLR = part-load ratio

DFLT = defrost factor for freezers

DFMT = defrost factor for coolers

WFCTZ = weather factor for each CTZ

EFLHLT = annual operation hours for freezers

EFLHMT = annual operation hours for coolers

#### Step 3: The annual energy savings (ΔkWh)

Once the compressor power savings and EFLH were determined, the annual energy savings for both coolers and freezers were calculated using the following relationship:

ΔkWhcomp-MT = ΔkWcomp-MT x EFLHMT

where

ΔkWhcomp-MT = annual compressor energy savings for coolers

ΔkWcomp-MT = compressor demand savings for coolers

EFLHMT = annual operation hours for coolers

ΔkWhcomp-LT = ΔkWcomp-LT x EFLHLT

where

ΔkWhcomp-LT = annual compressor energy savings for freezers

ΔkWcomp-LT = compressor demand savings for freezers

EFLHLT = annual operation hours for freezers

The full calculations can be found in the attached “Refrigeration Calculations.xlsx” [2], and are summarized in the calculation spreadsheet [1].

# Section 3. Load Shapes

The ideal load shape for net benefits estimates would represent the difference between the base case and measure case. The closest load shapes that are applicable to the measures in this work paper are listed in the table below.

Building Types and Load Shapes

|  |  |  |
| --- | --- | --- |
| **Building Type** | **Load Shape** | **E3 Alternate Building Type** |
| Grocery | Refrigeration | Grocery\_Store |
| Lodging - Hotel | Refrigeration | Hotel\_Motel |
| Restaurant - Fast-Food | Refrigeration | Fast\_Food\_Restaurant |
| Restaurant - Sit-Down | Refrigeration | Sit\_Down\_Restaurant |

# Section 4. Costs

## 4.1 Base Case Cost

For REA measures, the base case cost is assumed to be zero because these are discretionary modifications to the customers’ existing equipment. Their alternative is to make no changes to their existing system.

The base case cost is $0/linear-foot.

## 4.2 Measure Case Cost

In the case of REA, the customer is making a conscious decision to add-on to existing, working equipment. Since this is a discretionary choice by the consumer, the cost invoked is the full cost of the equipment and installation of the energy efficient equipment.

Cost estimates for this measure were obtained from the 2016 edition of RS Means Mechanical Cost Data [501]. Insulation thickness of ¾” is considered for walk-in coolers, and 1.25” is considerd for walk-in freezers. Refer to Attachment [3], “SCE17RN003.0 – Costs.xlsx”, for more information. These costs are adjusted for unit length and for each climate zone in the calculation spreadsheet [1].

For direct install programs, SCE utilizes one or more contractors as part of the program. The actual cost can vary by contractor, the date in which the work occurred, and by the volume of business. Contractor costs are confidential information and are based upon contractually agreed upon pricing as established in their purchase order with SCE; therefore, the SCE program tracking system is the only source for this data.

|  |  |  |  |  |
| --- | --- | --- | --- | --- |
| **Measure** | **Unit** | **Equipment Cost** | **Labor Cost** | **Total** |
| Bare Suction Lines for Walk-in Coolers Insulation | linear-ft of insulation | $2.69 | $3.76 | $6.45 |
| Bare Suction Lines for Walk-in Freezers Insulation | linear-ft of insulation | $3.50 | $3.85 | $7.35 |

## 4.3 Full and Incremental Measure Cost

**Full and Incremental Measure Cost Equations**

|  |  |  |  |
| --- | --- | --- | --- |
| **Installation Type** | **Incremental Measure Cost** | **Full Measure Cost** | |
| **1st Baseline** | **2nd Baseline** |
| REA | MEC + MLC | MEC + MLC | N/A |

MEC = Measure Equipment Cost; MLC = Measure Labor Cost

BEC = Base Case Equipment Cost; BLC = Base Case Labor Cost

**Full and Incremental Costs**

|  |  |  |  |  |
| --- | --- | --- | --- | --- |
| **Measure** | **Installation Type** | **Incremental Measure Cost** | **Full Measure Cost** | |
| **1st Baseline** | **2nd Baseline** |
| RF-82221 | REA | $7.38 | $7.38 | N/A |
| RF-46935 | REA | $7.38 | $7.38 | N/A |

# Attachments

* + 1. SCE17RN003.2 A1 - Calculation Template\_FINAL
    2. SCE17RN003.2 A2 - Savings Calcs
    3. SCE17RN003.2 A3 - Cost Calcs

# References



[84]

[155]

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[E] Are You Storing Food Safely? U.S. Food & Drug Administration. April 9, 2014. Retrived from https://www.fda.gov/ForConsumers/ConsumerUpdates/ucm093704.htm