COMMERCIAL REFRIGERATION
BARE SUCTION PIPE INSULATION
SWCR010-01

CONTENTS

MEASURE NAME

Bare Suction Line Insulation

STATEWIDE MEASURE ID

SWCR010-01

TECHNOLOGY SUMMARY

This measure pertains to the addition of insulation on un-insulated refrigeration system suction pipes of medium-temperature coolers (case temperature at or above 32 $^{\circ}$F) and low-temperature freezers (case temperature below 32 $^{\circ}$F). Insulation impedes heat transfer from the cold refrigeration pipes and reduces the undesirable system superheat. Excess superheat will increase the heat of compression and decrease the heat rejection ability of the condenser. Insulating suction refrigeration pipes will reduce the superheat and therefore, reduce compressor power and energy use.

MEASURE CASE DESCRIPTION

The measure case is defined as a refrigeration system with insulated suction refrigeration pipes outside of the refrigerated space. The measure offerings are:

- Insulation of bare suction pipes for medium-temperature, walk-in coolers
- Insulation of bare suction pipes for low-temperature, walk-in freezers

BASE CASE DESCRIPTION

The base case is defined as a refrigeration system with un-insulated suction refrigeration pipes outside of the refrigerated space.

CODE REQUIREMENTS

Refrigeration system suction pipe insulation is not governed by federal or state. The California Building Energy Efficiency Regulations (Title 24) sets forth mandatory requirements for pipe insulation.[1] However, the code pertains to new systems only; the insulation of bare refrigeration suction pipes is add-on equipment to an existing system and therefore does not trigger Title 24 regulations.

The California Appliance Efficiency Standards (Title 20) regulate the construction and selected components used in refrigerated walk-ins, but refrigeration suction pipes are not currently part of the regulation.[2] The Energy Independence and Security Act of 2007 regulates the design of new walk-in

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[1] California Energy Commission. 2015. *2016 Building Energy Efficiency Standards for Residential and Nonresidential Buildings*. CEC-400-2015-037-CMF. Section 120.3

[2] California Energy Commission (CEC). 2014. *2014 Appliance Efficiency Regulations.* CEC-400-2014-009-CMF.

freezers and coolers,[3] but does not include specifications for suction pipes. Finally, test procedures for walk-in coolers are detailed in the Code of Federal Regulations, but suction pipes are not referenced.[4]

Applicable State and Federal Codes and Standards

Code	Applicable Code Reference	Effective Date
CA Appliance Efficiency Regulations – Title 20	None.	n/a
CA Building Energy Efficiency Standards – Title 24	None.	n/a
Federal Standards	None.	n/a

NORMALIZING UNIT

Linear feet (Len-ft.)

PROGRAM REQUIREMENTS

Measure Implementation Eligibility

All combinations of measure application type, delivery type, and sector that are established for this measure are specified below. Measure application type is a categorization based on the circumstances and timing of the measure installation; each measure application type is distinguished by its baseline determination, cost basis, eligibility, and documentation requirements. Delivery type is the broad categorization of the delivery channel through which the market intervention strategy (financial incentives or other services) is targeted. This table also designates the broad market sector(s) that are applicable for this measure.

Note that some of the implementation combinations below may not be allowed for some measure offerings by all program administrators.

Implementation Eligibility

Measure Application Type	Delivery Type	Sector
Add-on equipment	DnDeemed	Com
Add-on equipment	DnDeemed	Ag
Add-on equipment	DnDeemed	Ind
Add-on equipment	UpDeemed	Com
Add-on equipment	UpDeemed	Ag
Add-on equipment	UpDeemed	Ind
Add-on equipment	DnDeemDI	Com
Add-on equipment	DnDeemDI	Ag
Add-on equipment	DnDeemDI	Ind

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[3] H.R.6. – 110th Congress. The Energy Independence and Security Act of 2007. Pub.L. 110-140. 121 Stat. 1565–1568.

[4] Code of Federal Regulations at 10 CFR 431. Appendix B Subpart R.

Eligible Products

The eligible measure offerings for bare refrigeration pipe insulation are:

- Insulation of bare suction pipes for medium-temperature, walk-in coolers
- Insulation of bare suction pipes for low-temperature, walk-in freezers

The eligibility requirements are as follows:

- The refrigeration suction pipe on existing equipment must be no more than 1-5/8 inches in diameter.
- This measure is applicable to bare suction pipes insulated with closed-cell nitrite rubber or equivalent with at least ¾-inch for medium-temperature and 1-inch for low-temperature systems.
- Insulation R-values must be greater than or equal to R-3.2 for medium-temperature cooler pipes.
- Insulation R-values must be greater than or equal to R-4.3 low-temperature freezer pipes.

Eligible Building Types

This measure is applicable for any existing commercial building type of any vintage.

Eligible Climate Zones

This measure is applicable in all California climate zones.

PROGRAM EXCLUSIONS

None.

DATA COLLECTION REQUIREMENTS

None.

USE CATEGORY

ComRefrig

ELECTRIC SAVINGS (kWH)

The energy savings calculations for the bare refrigeration pipe insulation measure were based upon ASHRAE Fundamentals Handbooks, refrigeration documents, and thermal analysis software.

This section presents the assumptions specified for the refrigeration systems of walk-in coolers and freezers, the overall approach to calculate energy savings and demand impacts, and the methodology for

calculating the heat gain through bare and insulated suction pipes, and power usage for each scenario. The last section describes step-by-step demand and energy savings calculations.

Assumptions and Inputs

Assumptions and inputs for the energy savings calculations for refrigeration pipe insulation for low-temperature walk-in freezer and medium-temperature walk-in coolers are specified in the following tables. Note that the analysis focused on *exposed* suction pipes.

Assumptions for Energy Savings Calculations - Low-temperature Walk-in Freezer

Parameter	Value	Source
Design saturated evaporating temperature (SET)	-15 °F	eQuestRefrig 3.65-7175, Grocery prototype model, walk-in freezer. The "Sat Suction/Air TD is 4°F" (the design temperature differential between the air leaving the coil (supply air temperature), and the refrigerant suction temperature). The average Discharge air temperature is -11°F. Therefore, the Saturated suction temperature is (-11°F) – (4°F) = -15°F.
Discharge air temperature	-11 °F	eQuestRefrig 3.65-7175, Grocery prototype model, walk-in freezer. The space "Design Temperature is -5°F." The "Zone Entering Min Supply Temp is -13F" with "Cool Control Range of 4°F". Therefore, the average Discharge air temperature is -11°F.
Pipe nominal diameter	0.625 in	Professional judgement.
Actual pipe diameter	0.75 in	American Society for Testing and Materials (ASTM). 2014. *B88 Standard Specification For Seamless Copper Water Tube.* West Conshohocken (PA): ASTM International.
Insulation thickness	1.0 in	Professional judgement.
Closed-cell, flexible elastomeric thermal insulation with a thermal conductivity	0.233 Btu-in/hr-ft^2- °F	Professional judgement.
Total suction pipe length	40 ft	Professional judgement.
Refrigerant	R-22	Professional judgement.
Condenser temperature difference (TD)	15 °F	Per eQuestRefrig 3.65-7175, Grocery prototype model, walk-in freezer: "Rated T Differential of 15 °F" for LT_Condenser (the differential between saturated condensing temperature and outdoor drybulb temperature)
Change in compressor run time due to suction pipe insulation	None	
Average wind speed for exposed refrigeration pipes	7.5 mph	American Society of Heating, Refrigerating and Air Conditioning Engineers, Inc. (ASHRAE). 2005. *2005 ASHRAE Handbook Fundamentals.* Atlanta (GA): ASHRAE. Page 25.2, Table 1.
Wind speed for non-exposed refrigeration pipes	0.0 mph	Professional judgement.

Assumptions for Energy Savings Calculations - Medium-temperature Walk-in Cooler

Parameter	Value	Source
Design saturated evaporating temperature (SET)	23 °F	Per eQuestRefrig 3.65-7175, Grocery prototype model, walk-in cooler. The "Sat Suction/Air TD is 4°F" (the design temperature differential between the air leaving the coil (supply air temperature), and the refrigerant suction temperature). The average Discharge air temperature is 27°F. Therefore, the Saturated suction temperature is (27°F) – (4°F) = 23°F.
Discharge air temperature	27 °F	Per eQuestRefrig 3.65-7175, Grocery prototype model, walk-in cooler. The space "Design Temperature is 35°F." The "Zone Entering Min Supply Temp is 25F" with "Cool Control Range of 4°F". Therefore, the average Discharge air temperature is 27°F.
		Professional judgement.
Actual pipe diameter	0.75 in	American Society for Testing and Materials (ASTM). 2014. *B88 Standard Specification For Seamless Copper Water Tube.* West Conshohocken (PA): ASTM International.
Insulation thickness	0.75 in	Professional judgement.
Closed-cell, flexible elastomeric thermal insulation with a thermal conductivity	0.313 Btu-in/hr-ft^2- °F	Professional judgement.
Total suction pipe length	40 ft	Professional judgement.
Refrigerant	R-22	Professional judgement.
Condenser temperature difference (TD)	20 °F	Per eQuestRefrig 3.65-7175, Grocery prototype model, walk-in cooler. "Rated T Differential of 20 °F" for MT_Condenser (the differential between saturated condensing temperature and outdoor drybulb temperature)
Change in compressor run time due to suction pipe insulation	None	
Average wind speed for exposed refrigeration pipes	7.5 mph	American Society of Heating, Refrigerating and Air Conditioning Engineers, Inc. (ASHRAE). 2005. *2005 ASHRAE Handbook Fundamentals.* Atlanta (GA): ASHRAE. Page 25.2, Table 1.
Wind speed for non-exposed refrigeration pipes	0.0 mph	Professional judgement.

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 with the following calculation:

Calculation of Saturated Condensing Temperature

$$SCT_{cooler} = DB_{amb} + 20$$

$$SCT_{freezer} = DB_{amb} + 15$$

$$DB_{amb} \quad = \textit{design ambient dry-bulb temperature, by climate zone}[5]$$

Approach

The approach to estimate demand and energy savings for insulating bare suction pipes of walk-in coolers and freezers is comprised of five primary steps. The first two steps in the analysis were to *calculate compressor power usage for bare and insulated suction pipes*. This entailed a heat transfer analysis and a refrigeration cycle analysis:

Step 1: Conduct Heat Transfer Analysis. The objective of the heat transfer analysis for both bare (base case) and insulated (measure case) suction pipes of walk-ins was to determine 1) the overall heat transfer coefficient, and 2) the heat gain through bare and insulated pipes.

Step 2: Conduct Refrigeration Cycle Analysis. Conduct refrigeration cycle analysis for both bare (baseline) and insulated (post-retrofit) suction pipes to:

- Correlate refrigerant temperature and specific heat,
- Determine total system superheat by calculating refrigerant temperature at the suction port of the compressor for both bare and insulated suction pipes,
- Determine heat of compression for both bare and insulated suction pipe scenarios, and
- Determine the compressor power usage for both bare and insulated suction pipe scenarios.

The remaining three steps were necessary *to calculate the energy savings and demand reduction* due to insulating bare suction pipes. This was accomplished by calculating the compressor power savings per unit, calculating the equivalent full-load hours (EFLH) of operation, and then finally calculating the compressor energy savings per unit.

Step 3: Calculate Compressor Power Reduction per unit. Demand savings was derived from the bare and insulated suction pipe compressor power usage. This result was then converted to demand reduction per linear-ft. of exposed suction pipe for both walk-in coolers and freezers.

Step 4. Calculate the Equivalent Full-Load Hours (EFLH) of Operation. EFLH is determined by using annual available operation hours (8,760) and overall duty-cycle factor. Overall duty cycle factor is determined by considering compressor over-sizing factor, defrost periods and weather factor.

Step 5. Calculate Compressor Energy Savings. Energy savings are derived from the demand reduction and EFLH.

The energy savings and demand reduction calculations were repeated for all climate zones for both coolers and freezers.

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[5] Hollday, W.L. 1982. "Climatic Data for Region X."

Step 1: Conduct Heat Transfer Analysis

This section discusses necessary and crucial steps or methodologies to calculate the total heat gain through bare copper suction pipes and insulated suction pipes. These steps include: 1) heat transfer analysis, which includes calculating heat transfer coefficients and calculating overall heat transfer coefficient; and 2) heat gain calculation.

Step 1a: Overall Heat Transfer Coefficient

The objective of the heat transfer analysis was to identify key heat transfer coefficients. These heat transfer coefficients were used to determine the overall heat transfer coefficient, which was then used to determine the heat gain for each scenario.

Bare Suction Pipe. 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 inner radius (r_i), the pipe outer radius (r_o) and the thermal conductivity of the pipe (k_{pipe}). The convection and radiation, however, act in parallel at the surface of the pipe. Accordingly, the thermal circuit can be constructed as follows:

Network Representation of Radiative, Convective and Conductive Exchange between Ambient and Vapor Refrigerant for Bare Suction Pipes
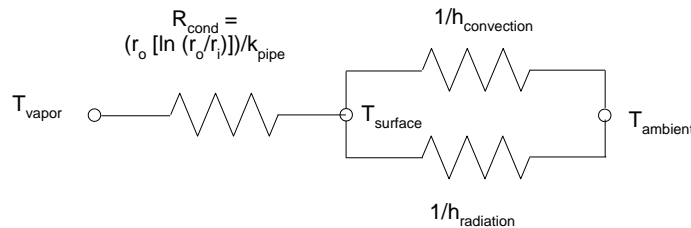
However, for simplicity purposes, *the pipe wall thermal resistance will be neglected*. In other words, the surface of the pipe is assumed to have the same temperature as the vapor refrigerant. As in the figure below, the only components that will contribute to the heat gain of vapor refrigerant are convection and radiation. Therefore, the total thermal resistance between a bare suction pipe and ambient air include effective resistance associated with convection and radiation.

Schematic and Network Representation of Radiative and Convective Exchange between Ambient and Vapor Refrigeration for Bare Suction Pipes
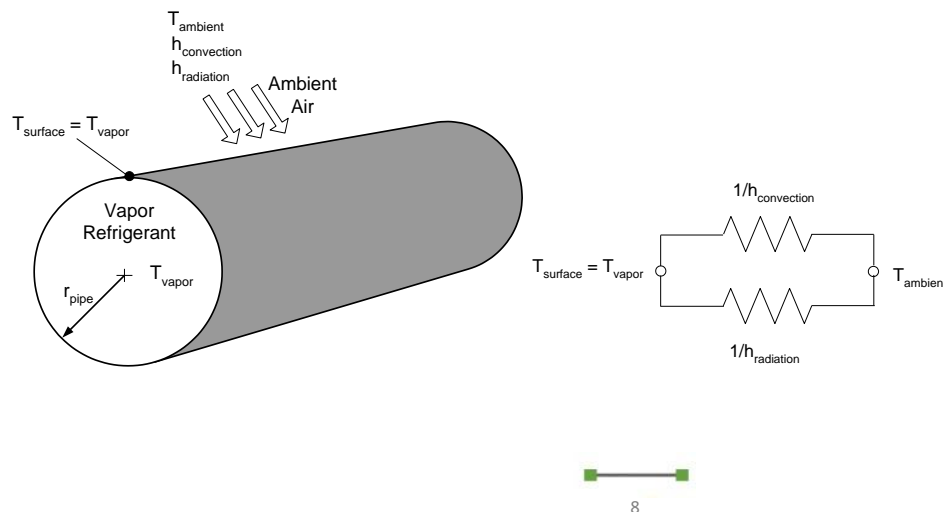
The convective and radiant heat transfer coefficients for bare pipes were calculated as follows: [6]

Convective and Radiant Heat Transfer Coefficients

$$h_{covection} = C * \left(\frac{1}{d}\right)^{0.2} * \left(\frac{1}{t_{avg}}\right)^{0.181} * \Delta t^{0.266} * \sqrt{1 + 2.77 * V}$$

where:

$h_{convection}$	= *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*
t_{avg}	= *average temperature of air film, ᵒF*
Δt	= *surface to air temperature difference, ᵒF*
V	= *wind speed, mph*

The ambient air temperature and the pipe surface temperatures were used to estimate the average air film temperature, as shown by the calculation below. Since the assumption is that the pipe surface temperature is the same as the vapor refrigerant temperature, the pipe surface temperature equated to the refrigerant vapor temperature.

Average Air Film Temperature

$$t_{avg} = \frac{t_a + t_s}{2}$$

t_a	= *air temperature, ᵒF*
t_s	= *pipe surface temperature, ᵒF*

The following equation illustrates radiation heat transfer coefficient.

Radiant Heat Transfer Coefficient

$$h_{radiation} = \frac{\varepsilon * 0.1713 * 10^{-8} * \left[(t_a + 459.6)^4 - (t_s + 459.6)^4\right]}{(t_a - t_s)}$$

$h_{radiation}$	= *radiant heat transfer coefficient, Btu/hr-ft²-ᵒF*
ε	= *surface emittance, 0.44 for dull bare copper pipe* [7]

[6] American Society of Heating, Refrigerating and Air Conditioning Engineers (ASHRAE). 1993. *1993 ASHRAE Handbook Fundamentals.* Atlanta (GA): ASHRAE. Page 22.17.

[7] American Society of Heating, Refrigerating and Air Conditioning Engineers (ASHRAE). 1993. *1993 ASHRAE Handbook Fundamentals.* Atlanta (GA): ASHRAE. Page 22.18, Table 12.

The convective and radiant heat transfer coefficients are used to determine the total thermal resistance. The total thermal resistance for heat flow through resistances in parallel can be obtained using the following calculation.[8]

Total Thermal Resistence

$$\sum R_{Total} = \left[\frac{1}{\left(\frac{1}{h_{convection}}\right)} + \frac{1}{\left(\frac{1}{h_{radiation}}\right)} \right]^{-1} \rightarrow \sum R_{Total} = \left(h_{convection} + h_{radiation} \right)^{-1}$$

where:

R_{Total} = *Total thermal resistance, $^oF\text{-}ft^2\text{-}hr/Btu$*

Once the total thermal resistance is determined, the overall heat transfer coefficient can be calculated as the following.

**Overall Heat Transfer Coefficient**

$$U = \frac{1}{\sum R_{Total}} \rightarrow U = \frac{1}{\left(h_{convection} + h_{radiation} \right)^{-1}} = \left(h_{convection} + h_{radiation} \right)$$

U = *overall heat transfer coefficient, $Btu/hr/ft^{2}\text{-}^oF$*

Insulated Suction Pipes: 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 insulation surface. The thermal circuit shown below illustrates each component.

Network Representation of Radiative, Convective and Conductive Exchange between Ambient and Vapor Refrigerant for Insulated Suction Pipes
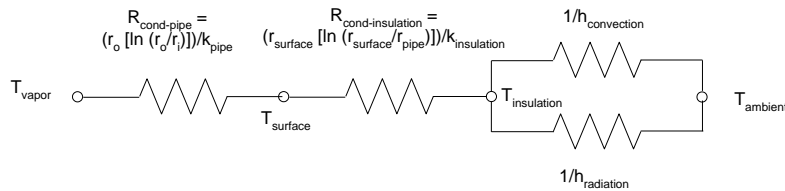
For simplicity purposes the following assumptions are made:

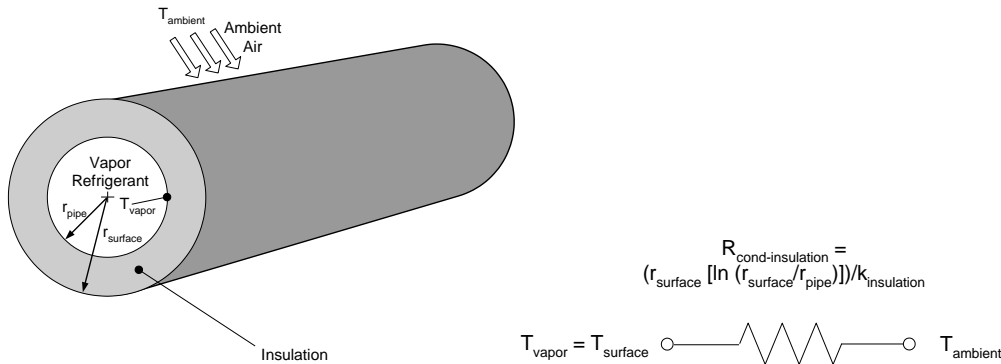1. The pipe wall thermal resistance is negligible.
2. The insulation exterior surface temperature is equal to ambient air temperature ($T_{insulation}$ = $T_{ambient}$), therefore:
   - Convection through the pipe insulation surface is omitted, and

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[8] American Society of Heating, Refrigerating and Air Conditioning Engineers, Inc. (ASHRAE). 2005. 2005 ASHRAE Handbook Fundamentals. Atlanta (GA): ASHRAE. Page 3.18.

– Radiation through the pipe 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.

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

$$R_{cond\text{-}insulation} = (r_{surface} [ln (r_{surface}/r_{pipe})])/k_{insulation}$$

$$T_{vapor} = T_{surface}$$ ———/\/\/\——— $$T_{ambient}$$

The thermal resistance associated with using insulation is a function of the inner radius of insulation (i.e., the pipe radius), the outer radius of insulation, and the thermal conductivity of the insulation.[9]

Thermal Resistance

$$R_{cond\text{-}insulation} = \frac{\left[12\left(\ln\left(\frac{r_{surface}}{r_{pipe}}\right)\right)\right]}{2\pi k_{insulation}}$$

where:

$R_{cond\text{-}insulation}$	= *conduction through insulation, $^oF\text{-}ft^2\text{-}hr/Btu$*
$r_{surface}$	= *outer radius of insulation, inches*
r_{pipe}	= *inner radius of insulation or pipe radius, inches*
$k_{insulation}$	= *thermal conductivity of insulation, $Btu\text{-}in/hr\text{-}ft^2\text{-}{}^oF$*

This analysis assumed that the insulation material is flexible, closed-cell elastomeric with thermal conductivity (k value) of 0.233 Btu-in/hr-ft^2-°F for low-temperature line and 0.313 Btu-in/hr-ft^2-°F for medium-temperature line.[10] Note that r$_{surface}$ can be determined by adding radius of the pipe and thickness of the insulation.

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[9] American Society of Heating, Refrigerating and Air Conditioning Engineers (ASHRAE). 2009. 2009 ASHRAE Handbook Fundamentals. Atlanta (GA): ASHRAE. Page 23.4.

[10] Southern California Edison (SCE) 2018. *Solutions Directory 2018.* 23rd Edition. 3rd Quarter 2018.

Outer Radius of Insulation

$r_{surface} = r_{pipe} + (insulation\ thickness)$

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

Overall Heat Transfer Coefficient

$$U = \frac{1}{\sum R_{Total}} \quad or \quad U = \frac{1}{R_{cond-insulation}}$$

where:

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

Step 1b: Calculate Heat Gain

The heat gain is a function of overall heat transfer coefficient, surface area and temperature difference.[11]

Heat Gain

$$Q = U * A * (t_a - t_s)$$

Q = *heat gain, Btu/hr*
U = *overall heat transfer coefficient, Btu/hr/ft²-ºF*
A = *surface area, ft²*
t_a = *ambient temperature, ºF*
t_s = *vapor refrigerant temperature, ºF*

The surface area for cylindrical shapes (i.e., pipes) can be derived as the following:

Surface Area

$A = 2\pi r_{pipe} l$ *For bare suction pipes*

$A = 2\pi r_{surface} l$ *For insulated suction pipes*

r_{pipe} = *pipe radius, ft.*
$r_{surface}$ = *outer radius of insulation, ft.*
l = *pipe length, ft.*

For low-temperature line insulation, "Insulation R-values must be R-4.3 or greater."

[11] American Society of Heating, Refrigerating and Air Conditioning Engineers, Inc. (ASHRAE). 2005. 2005 ASHRAE Handbook Fundamentals. Atlanta (GA): ASHRAE. Page 3.18.

Step 2: Conduct Refrigeration Cycle Analysis

This section discusses analytical methods 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 2a: Average refrigerant mass flow rates

Two sets of 0.5-hp, 0.75-hp, and 1.0-hp reciprocating compressors were selected from the compressor manufacturer catalog to represent both low- and medium-temperature compressors.[12] The selected compressor models for both medium- and low-temperature applications using refrigerant R-22 are:

Medium-temperature compressor models: KANB-0050, KAMB-0075, KAJB-0100
Low-temperature compressor models: HAG2-0050, KAN2-0075, KAR2-0100

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

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

Refrigerant (R-22) Mass Flow Rates (lbs/hr)										
MT		Evaporating Temperature								
	°F	0	5	10	15	20	25	30	35	45	55
Condensing Temperature	70	56.67	64.33	73.67	83.67	94.67	107.00	120.67	135.67	169.00	208.33
	80	53.67	61.67	70.33	80.00	91.00	103.00	116.00	130.67	163.33	201.00
	90	51.00	58.67	67.00	76.67	87.33	99.33	111.67	125.67	157.67	195.00
	100	48.33	56.00	64.00	73.33	84.00	95.33	107.67	121.33	152.33	188.67
	110	46.00	53.00	61.00	70.33	80.33	91.33	103.33	116.67	147.00	182.00
	120	43.00	50.00	58.00	67.00	77.00	87.67	99.33	112.00	141.33	175.67
	130	40.00	47.00	54.67	63.00	73.00	83.33	94.67	107.33	136.00	169.00
	140	36.33	43.33	51.00	59.00	68.33	78.33	89.67	102.00	129.33	162.00

[12] Emerson. (n.d.). *Copeland Condensing Units. Selection guide for commercial refrigeration.*

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

		Refrigerant (R-22) Mass Flow Rates (lbs/hr)								
LT		Evaporating Temperature								
	°F	-40	-35	-30	-25	-20	-15	-10	-5	0
Condensing Temperature	70	25.7	30.3	35.7	41.7	48.3	56.0	64.7	74.3	85.0
	80	23.0	28.0	33.3	39.0	46.0	53.7	62.0	71.7	82.0
	90	21.3	25.7	31.0	36.7	43.7	51.0	59.3	68.7	78.7
	100	18.7	23.7	28.7	35.0	41.3	48.7	57.0	65.7	75.7
	105	17.7	22.3	28.0	34.0	40.3	47.7	55.3	64.7	74.0
	110	17.0	21.3	27.0	32.7	38.7	46.0	54.3	63.0	72.7
	120	14.7	19.3	24.3	30.0	36.3	43.3	51.0	59.7	69.0
	130	11.7	16.7	21.7	27.3	33.3	40.0	47.7	56.0	65.0

Step 2b: Correlation between refrigerant specific heat and temperature

Since the specific heat of refrigerant is a function of refrigerant temperatures, the specific heat of refrigerant was correlated to refrigerant temperatures. The data used to find the correlation between the refrigerant specific heat and temperature were based on thermo-physical properties of R-22.[13] The figure below illustrates this correlation.

Correlation Between Refrigerant Specific Heat and Temperature for R-22

$$y = 3E\text{-}14x^6 - 3E\text{-}12x^5 - 4E\text{-}10x^4 + 5E\text{-}08x^3 + 3E\text{-}06x^2 + 0.0003x + 0.1594$$
$$R^2 = 0.999$$
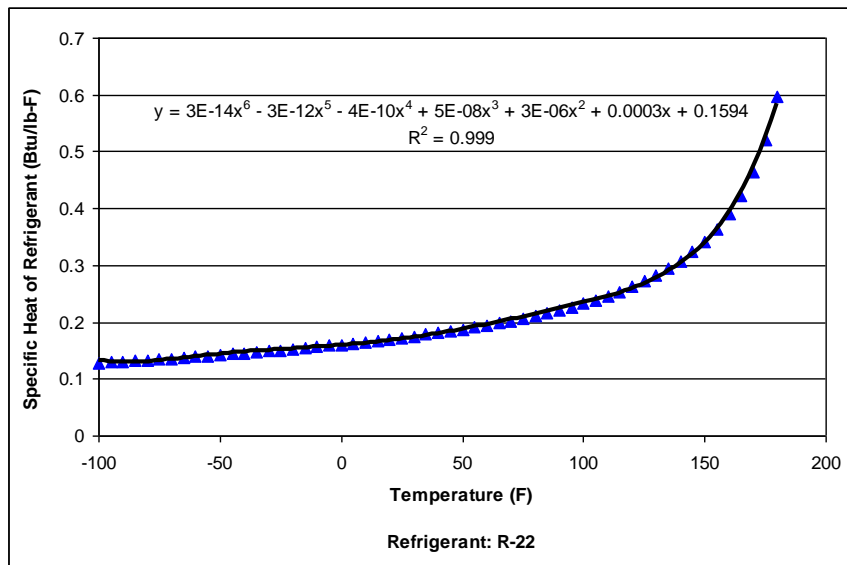
Step 2c: Temperature at compressor inlet

Once the methodology for heat gain and correlation between heat of refrigerant and refrigerant vapor temperatures were determined, the temperature at the inlet of the compressor was determined for both

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[13] American Society of Heating, Refrigerating and Air Conditioning Engineers, Inc. (ASHRAE). 2005. *2005 ASHRAE Handbook Fundamentals.* Atlanta (GA): ASHRAE. Page 20.5.

bare and insulated suction pipes. The entire suction pipe length was segmented and for each segment heat gain, and corresponding refrigerant specific heat and temperature. The equation below illustrates the relationship between the heat gain, average refrigerant mass flow rate, specific heat, and temperature increase due to heat transfer between the ambient air and the suction pipe.[14]

$$Q = mc_p\Delta T$$

Q = heat gain through the suction pipe (Btu/hr)
m = mass flow rate of the refrigerant (lb/hr)
c_p = specific heat of vapor refrigerant R22 (Btu/lb-ºF)
ΔT = change in temperature of the refrigerant (ºF)

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 7 ºF and using SET of refrigerant. This first segment of suction pipe represents the suction pipe run right after the evaporator coil.

Refrigerant Temperature at Beginning of Segment

$$T_{initial} = SH_{design} + SET$$

$T_{initial}$ = initial temperature of refrigerant at the beginning of segment (ºF)
SH_{design} = design evaporator superheat (ºF)
SET = saturated evaporating temperature (ºF)

2. Determine the refrigerant specific heat for the first segment of suction pipe using the correlation between refrigerant specific heat and initial vapor temperature.
3. Determine the heat gain for the first segment of suction pipe using initial vapor temperature and climate zone information using methodologies discussed in the "Heat Transfer Analysis" section.
4. Determine the final vapor refrigerant temperature for the first segment of the suction pipe using initial vapor refrigerant temperature, refrigerant specific heat, heat gain, and refrigerant mass flow rate. The following equation shows the relationship between these parameters.

Refrigerant Temperature at End of Segment

$$T_{final} = \frac{Q}{mc_p} + T_{initial}$$

T_{final} = final temperature of refrigerant at the end of segment (ºF)
Q = heat gain through a segment of the suction pipe (Btu/hr)
m = mass flow rate of the refrigerant (lb/hr)
c_p = specific heat of refrigerant R-22 at a segment of suction pipe (Btu/lb-ºF)
$T_{initial}$ = initial temperature of refrigerant at the beginning of segment (ºF)

[14] Kreider, J., P. Curtiss, and A. Rabl. 2002. *Heating and Cooling of Buildings, Design for Efficiency*. 2nd ed. New York (NY): McGraw-Hill. Page 18.

Note that 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.

5. Repeat steps discussed above for subsequent suction pipe segments. These steps are repeated until the suction pipe last segment where the vapor refrigerant enters the suction port of the compressor or the vapor refrigerant temperature equals ambient temperature.

Step 2d: 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 derived using refrigerant property software.[15]

Using the suction pressure and the temperature of the refrigerant at the inlet of the compressor, the enthalpy at the inlet of the compressor was derived via refrigerant property software for R-22.[16] Similarly, the entropy at the inlet of the compressor was derived using the suction pressure and the refrigerant temperature at the compressor inlet using R-22 refrigerant property software.[17]

If 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 derived using R-22 refrigerant property software. [18]

Step 2e: Heat of compression

Once the enthalpies at the inlet and outlet of the compressor were determined, the heat of compression was calculated with the following equation.[19]

Heat of Compression

$$\Delta h_{hc} = h_{outlet} - h_{inlet}$$

Δh_{hc} = *heat of compression of the compressor (Btu/lb)*

h_{outlet} = *refrigerant enthalpy at the inlet of the compressor (Btu/lb)*

[15] Optimized Thermal Systems, Inc. (formerly Thermal Analysis Partners, LLC). (n.d.) Xprops® Thermophysical Property Software, version 1.3.

[16] Optimized Thermal Systems, Inc. (formerly Thermal Analysis Partners, LLC). (n.d.) Xprops® Thermophysical Property Software, version 1.3.

[17] Optimized Thermal Systems, Inc. (formerly Thermal Analysis Partners, LLC). (n.d.) Xprops® Thermophysical Property Software, version 1.3.

[18] Optimized Thermal Systems, Inc. (formerly Thermal Analysis Partners, LLC). (n.d.) Xprops® Thermophysical Property Software, version 1.3.

[19] American Society of Heating, Refrigerating and Air Conditioning Engineers, Inc. (ASHRAE). 2004. 2004 ASHRAE Handbook HVAC Systems and Equipment. Atlanta (GA): ASHRAE. Page 34-12.

h_{inlet} = refrigerant enthalpy at the outlet of the compressor (Btu/lb)

Step 2f: Compressor power usage

Once the heat of compression and mass flow rate of refrigerant were determined, the compressor power usage for bare and insulated suction pipes 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. An overall compressor efficiency of 0.5355 was used for the purpose of this analysis.[20] The calculation of compressor power usage as a function of refrigerant mass flow rate, heat of compression, and efficiency of the compressor is shown below.[21]

Compressor Power Usage

$$kW_{comp} = \frac{m\Delta h_{hc}}{n_{overall}\, k}$$

kW_{comp} = compressor power usage (kW)

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

Δh_{hc} = heat of compression of the compressor (Btu/lb)

$\eta_{overall}$ = overall efficiency of the compressor

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

Step 3: Calculate Compressor Power Reduction

Once the compressor power usage was determined for bare (base case) and insulated (measure case) suction pipes for walk-in coolers and freezers, the compressor power reduction was calculated using the equation below.

$$\Delta kW_{comp} = kW_{comp\text{-}bare} - kW_{comp\text{-}insulated}$$

ΔkW_{comp} = demand savings due to insulating bare suction pipes (kW)

$kW_{comp\text{-}bare}$ = compressor power usage for bare suction pipe scenario (kW)

$kW_{comp\text{-}insulated}$ = compressor power usage for insulated suction pipe scenario (kW)

The calculated total compressor power reduction was then divided by total linear feet of suction pipe to derive power reduction per linear foot.

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[20] Southern California Edison (SCE), Refrigeration and Thermal Test Center. 2007. *Evaluation of Heat Gain in Suction Line on Compressor Power.*

[21] Dossat, R.J. 1997. *Principles of Refrigeration.* 4th ed. Upper Saddle River (NJ): Prentice-Hall. Page 223.

Step 4: Calculate 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 equation below 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 1.0. 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 climate zone. Using DOE-2 simulation results for a typical supermarket, the weather factors were determined for each climate zone.[22]

Duty cycle = Capacity factor x Defrost factor x Weather factor

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 climate zone (see discussion below)*

The annual energy usage of a refrigeration system for a typical supermarket in each climate zone was used to estimate the weather factor for each climate zone. This was based on a DOE-2 computer simulation.[23] Using climate zone 15 as a benchmark with an 85% weather factor, the weather factors for the other 15 climate zone were estimated. The calculation below represents this methodology. The following table illustrates the annual energy usage of refrigeration for each climate zone and corresponding weather factors.

$$\text{WF}_{CTZ} = \left[\frac{\text{Annual kWh CTZ}}{\text{Annual kWh CTZ}_{15}} \right] \times \text{WF}_{CTZ\text{-}15}$$

WF_{CTZ} = weather factor for each climate zone
Annual kWh CTZ = annual energy usage of refrigeration system for each climate zone
Annual kWh CTZ_{15} = annual energy usage of refrigeration system for climate zone 15
$WF_{CTZ\text{-}15}$ = weather factor for climate zone 15, 85%

Weather Factors by Climate Zone

Climate Zone	Refrigeration Annual Energy Usage (kWh/yr) [a]	Weather Factor (%)
1	473,398	74.3%
2	476,862	74.8%

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[22] The original reference for this information could not be located: *2001 Express Efficiency, New Refrigeration Measure Computer Simulation Report*. Page 20-21, Table 2.

[23] The original reference for this information could not be located: *2001 Express Efficiency, New Refrigeration Measure Computer Simulation Report*. Page 20-21, Table 2.

Climate Zone	Refrigeration Annual Energy Usage (kWh/yr) [a]	Weather Factor (%)
3	484,793	76.0%
4	496,426	77.9%
5	486,175	76.3%
6	502,982	78.9%
7	508,252	79.7%
8	506,024	79.4%
9	500,936	78.6%
10	495,565	77.7%
11	477,810	75.0%
12	485,318	76.1%
13	496,488	77.9%
14	491,056	77.0%
15 [b]	541,873	85.0%
16	452,877	71.0%

a. Source: DOE-2 simulations for a typical supermarket.
b. Climate zone 15 was used as the reference climate zone.

Accordingly, duty cycles and EFLH for low- and medium-temperature walk-ins were calculated with the equation below.

Duty Cycle and Effective Full-Load Hours Calculations

$Duty\ cycle_{LT}$ $= (1 – PLR)\ x\ DF_{LT}\ x\ WF_{CTZ} = (1 – 0.87)\ x\ (1 – 0.05)\ x\ WF_{CTZ}$

$Duty\ cycle_{MT}$ $= (1 – PLR)\ x\ DF_{MT}\ x\ WF_{CTZ} = (1 – 0.87)\ x\ (1 – 0.1)\ x\ WF_{CTZ}$

$EFLH_{LT} = 8,760\ x\ Duty\ cycle_{LT}$

$EFLH_{MT} = 8,760\ x\ Duty\ cycle_{MT}$

$Duty\ Cycle_{LT}$ $= duty\ cycle\ for\ freezers\ (low\text{-}temperature\ systems)$
$Duty\ Cycle_{MT}$ $= duty\ cycle\ for\ coolers\ (medium\text{-}temperature\ systems)$
PLR $= part\text{-}load\ ratio$
DF_{LT} $= defrost\ factor\ for\ freezers$
DF_{MT} $= defrost\ factor\ for\ coolers$
WF_{CTZ} $= weather\ factor\ for\ each\ climate\ zone$
$EFLH_{LT}$ $= annual\ operation\ hours\ for\ freezers$
$EFLH_{MT}$ $= annual\ operation\ hours\ for\ coolers$

Step 5. Calculate Compressor Energy Savings per Unit

The annual energy savings for both coolers and freezers was calculated as the product of the compressor power reduction and the EFLH, as shown in the following equations.

Annual Compressor Energy Savings for Cooler

$\Delta kWh_{comp\text{-}MT} = \Delta kW_{comp\text{-}MT}\ x\ EFLH_{MT}$

$\Delta kWh_{comp\text{-}MT}$ $= annual\ compressor\ energy\ savings\ for\ coolers$
$\Delta kW_{comp\text{-}MT}$ $= compressor\ demand\ savings\ for\ coolers$

$EFLH_{MT}$ = annual operation hours for coolers

Annual Compressor Energy Savings for Freezer

$$\Delta kWh_{comp\text{-}LT} = \Delta kW_{comp\text{-}LT} \times EFLH_{LT}$$

$\Delta kWh_{comp\text{-}LT}$ = annual compressor energy savings for freezers
$\Delta kW_{comp\text{-}LT}$ = compressor demand savings for freezers
$EFLH_{LT}$ = annual operation hours for freezers

PEAK ELECTRIC DEMAND REDUCTION (KW)

Peak demand reduction was derived with the methodology presented in the Electric Savings section.

GAS SAVINGS (THERMS)

Gas energy savings was derived with the methodology presented in the Electric Savings section.

LIFE CYCLE

Effective useful life (EUL) is an estimate of the median number of years that a measure installed through a program is still in place and operable. Remaining useful life (RUL) is an estimate of the median number of years that a technology or piece of equipment replaced or altered by an energy efficiency program would have remained in service and operational had the program intervention not caused the replacement or alteration. The RUL is only applicable to the first baseline period for a retrofit measure with an applicable code baseline.

The methodology to calculate the RUL conforms with Version 5 of the Energy Efficiency Policy Manual, which recommends "one-third of the effective useful life in DEER as the remaining useful life until further study results are available to establish more accurate values."[24] This approach provides a reasonable RUL estimate without the requiring any a priori knowledge about the age of the equipment being replaced.[25] Further, as per Resolution E-4807, the California Public Utilities Commission (CPUC) revised add-on equipment measures so that the EUL of the measure is equal to the lower of the RUL of the modified system or equipment or the EUL of the add-on component." [26]

The EUL and RUL established for bare suction pipe insulation for walk-in coolers and freezers are specified below.

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[24] California Public Utilities Commission (CPUC), Energy Division. 2013. *Energy Efficiency Policy Manual Version 5*. Page 32.

[25] KEMA, Inc. 2008. "Summary of EUL-RUL Analysis for the April 2008 Update to DEER." Memorandum submitted to Itron, Inc.

[26] California Public Utilities Commission (CPUC). 2016. Resolution E-4807. December 16. Page 13.

Effective Useful Life and Remaining Useful Life

Parameter	Insulation of Bare Suction Pipes for Walk-in Coolers	Insulation of Bare Suction Pipes for Walk-in Freezers	Source
EUL (years) – host equipment, refrigeration compressor	15.00	15.00	California Public Utilities Commission (CPUC). 2008. "EUL_Summary_10-1-08.xlsx." California Public Utilities Commission (CPUC), Energy Division. 2003. *Energy Efficiency Policy Manual v 2.0*. Page 16.
EUL (yrs) – refrigeration insulation for bare suction lines	11.00	11.00	California Public Utilities Commission (CPUC), Energy Division. 2014. "DEER2014-EUL-table-update_2014-02-05.xlsx."
RUL (yrs) – host equipment, refrigeration compressor	5.00	5.00	RUL = 1/3 EUL of host equipment

BASE CASE MATERIAL COST ($/UNIT)

Insofar as the installation of refrigeration pipe insulation is add-on equipment, the base case assumes that the existing suction refrigeration pipes do not have insulation. Therefore, the base case material cost for *all delivery types* is $0.

MEASURE CASE MATERIAL COST ($/UNIT)

For add-on equipment, 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 is the full cost of the equipment and installation of the measure.

The material costs for *all delivery types* were derived from cost estimates of insulation, pipe covering, rubber tubing flexible closed cell foam, 1" wall, from 3/4" to 1-5/8" iron pipe size obtained from the 2016 online edition of RSMeans Mechanical Cost Data.[27] The final measure case material costs were calculated as the average of RSMeans cost data and includes sales tax.

BASE CASE LABOR COST ($/UNIT)

Insofar as the installation of refrigeration pipe insulation is add-on equipment, the base case assumes that the existing suction refrigeration pipes do not have insulation. Therefore, the base case labor cost for *all delivery types* is $0.

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[27] Southern California Edison (SCE). 2016. "SCE17RN003.0 A3 - Insulation of Bare Refrigeration Suction Costs Calcs 2016.xlsm."

MEASURE CASE LABOR COST ($/UNIT)

The measure case labor costs for *all delivery types* were derived from cost estimates to install insulation, pipe covering, rubber tubing flexible closed cell foam, 1" wall, from 3/4" to 1-5/8" iron pipe size obtained from the 2016 online edition of RSMeans Mechanical Cost Data.[28] The final labor costs were calculated as the average of RSMeans cost data.

NET-TO-GROSS (NTG)

The net-to-gross (NTG) ratio represents the portion of gross impacts that are determined to be directly attributed to a specific program intervention. For *downstream delivery types*, the NTG adopted for this measure was specified for the Database for Energy Efficient Resources (DEER) 2019 update and is associated with non-HVAC pipe insulation or domestic hot water applications. The original source to support this value is unknown. For *all other (upstream) delivery types*, the adopted NTG values are based upon the average of all NTG ratios for all evaluated 2006 – 2008 commercial, industrial, and agriculture programs, as documented in the 2011 DEER Update Study conducted by Itron, Inc. These sector average NTGs ("default NTGs") are applicable to all energy efficiency measures that have been offered through commercial, industrial, and agriculture sector programs for more than two years and for which impact evaluation results are not available.

Net-to-Gross Ratios

Parameter	Insulation of Bare Suction Pipes for Walk-in Coolers	Insulation of Bare Suction Pipes for Walk-in Freezers	Source
Downstream Delivery Types			
NTG - downstream	0.45	0.45	California Public Utilities Commission (CPUC), Energy Division. 2018. "*SupportTable-NTG2020-rev18Sep2018.xlsx.*"
Upstream Delivery Types			
NTG – Commercial	0.60	0.60	Itron, Inc. 2011. *DEER Database 2011 Update Documentation.* Prepared for the California Public Utilities Commission. Page 15-4 Table 15-3.
NTG – Industrial	0.60	0.60	
NTG - Agriculture	0.60	0.60	

GROSS SAVINGS INSTALLATION ADJUSTMENT (GSIA)

The gross savings installation adjustment (GSIA) rate represents the ratio of the number of verified installations of the measure to the number of claimed installations reported by the utility. This factor varies by end use, sector, technology, application, and delivery method. This GSIA rate is the current "default" rate specified for measures for which an alternative GSIA has not been estimated and approved.

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[28] Southern California Edison (SCE). 2016. "SCE17RN003.0 A3 - Insulation of Bare Refrigeration Suction Costs Calcs 2016.xlsm."

Gross Savings Installation Adjustment Rate

Parameter	Insulation of Bare Suction Pipes for Walk-in Coolers	Insulation of Bare Suction Pipes for Walk-in Freezers	Source
GSIA	1.0	1.0	California Public Utilities Commission (CPUC), Energy Division. 2013. *Energy Efficiency Policy Manual Version 5*. Page 31.

NON-ENERGY IMPACTS

Non-energy impacts for this measure have not been quantified.

DEER DIFFERENCES ANALYSIS

This section provides a summary of DEER-based inputs and methods, and the rationale for inputs and methods that are not DEER-based.

Table 1. DEER Difference Summary

DEER Item	Comment / Used for Workpaper
Modified DEER methodology	No
Scaled DEER measure	No
DEER Base Case	No
DEER Measure Case	No
DEER Building Types	No
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
NTG	Source: DEER2011 and DEER2020 update. The value of 0.45 is associated with NTG ID: *NonRes-sAll-mPipeIns-deemed.* The value of 0.60 is associated with NTG ID *Com-Default>2yrs, Ag-Default>2yrs, Ind-Default>2yrs*
GSIA	Source: DEER READI tool version 2.5.1. The value of 1.0 is associated with GSIA ID: *Def-GSIA.*
EUL/RUL	Source: DEER2011 and DEER2020 Update. The value of 15 years is associated with EUL ID: *RefgWrhs-Comp*

REVISION HISTORY

Measure Characterization Revision History

Revision Number	Date of Revision Completion	Primary Author, Title, Organization	Revision Summary and Rationale for Revision Effective Date and Approved By
01	9/30/2017	Jennifer Holmes, Cal TF Staff	Draft of consolidated text for this statewide measure is based upon: SCE17RN003, revision 1 (December 4, 2017) CE17RN003, revision 0 (November 18, 2016) Consensus reached among Cal TF members.
	11/29/2018	BASE Energy	Updated some values and references under the *Assumptions and Inputs* and *Approach* sections, and updated savings calculations accordingly.
	12/26/2018	Jennifer Holmes, Cal TF Staff	Final edits for submittal of version 01.