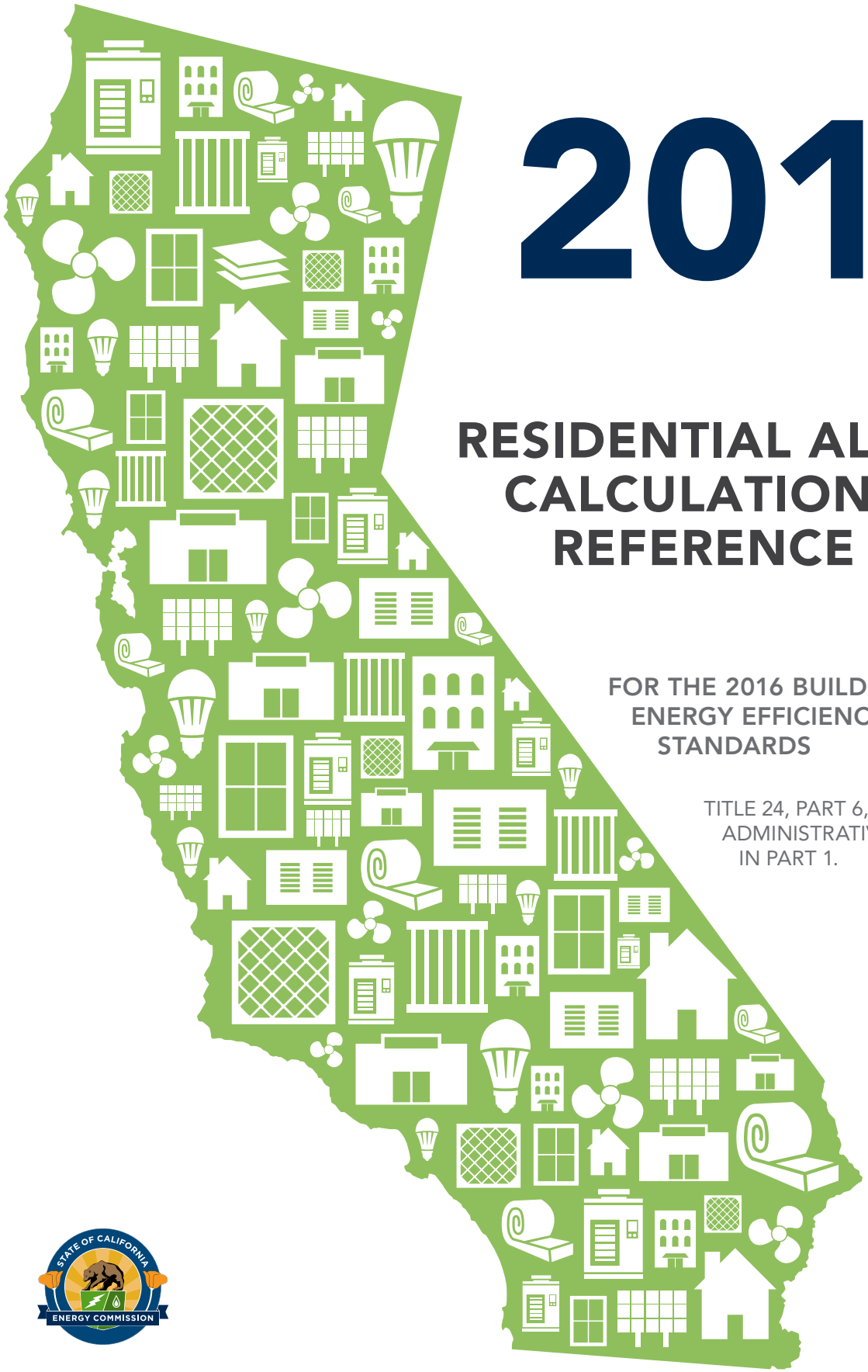

# 2016

# RESIDENTIAL ALTERNATIVE CALCULATION METHOD REFERENCE MANUAL

FOR THE 2016 BUILDING ENERGY EFFICIENCY STANDARDS

TITLE 24, PART 6, AND ASSOCIATED ADMINISTRATIVE REGULATIONS IN PART 1.

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**CALIFORNIA ENERGY COMMISSION**
Edmund G. Brown Jr., Governor

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# ACM Change Summary

| Page | Section/Topic | Correction |
|---|---|---|
| 5-6 | Solar PV credit | Update description to match current inputs. |
| 13 | Materials | Revise synthetic stucco names/properties (separating continuous insulation from stucco). |
| 39-40 | 2.4.6.11 Ducts | Change description of duct leakage. |
| 40 | 2.4.6.12 Low Leakage Air Handlers | Add a description for low leakage air handlers. |
| 44 | 2.4.9 IAQ Ventilation | Update standard design language for sensible heat recovery effectiveness. |
| 46 | Table 18 | Change whole house fan effectiveness. |
| 54-55 | 2.5.6.3 Exterior Walls | Include wording to show that garage and interior walls include cavity insulation only. |
| 69-70 | 2.9 Water heating efficiency | Add description of the new federal standard (effective June 12, 2017) rating water heaters using a uniform energy factor (UEF).<br>Update descriptions of water heating types to include the new federal categories of consumer, residential-duty commercial, and commercial. |
| 71 | 2.9 Standard distribution | Revise "standard" distribution to include pipe insulation (to match latest Plumbing Code). |
| 72 | 2.9.1 Dwelling unit standard design | Update wording of the standard design water heater to match the new ratings and water heater types. |
| 73 | 2.9.3 Solar water heating | Update the options for determining the solar fraction of the solar water heating system credit. |
| 78 | 2.10.3.4 Exterior shading | Revise exterior shading language for additions. |
| 82 | 2.10.3.11 Water Heating System | Update the standard design for water heating in additions and alterations. |
| 84 | Table 30 EDR | Correct outdated EDR adjustments for gas fuel type. |
| 86 | 3.4 Net Energy Metering (NEM) | New section language describing NEM limits on Solar PV credit. |
| 86 | 3.5 Battery Storage | New section describing how battery storage affects compliance credit and EDR scores. |

| | |
|---|---|
| Appendix A (Special Features) | Add battery storage and NEEA-rated water heater as a special features on the CF1R |
| Appendix B (Water Heating) | Include UEF to EF conversion<br>Include new definitions of water heater types<br>Update distribution multipliers to account for pipe insulation |

# ACKNOWLEDGMENTS

The *Building Energy Efficiency Standards* were adopted and put into effect in 1978 and have been updated periodically in the intervening years. The standards are a unique California asset, and have benefitted from the conscientious involvement and enduring commitment to the public good of many persons and organizations along the way. The *2016 Building Energy Efficiency Standards* (*2016 Standards*) development and adoption process continued the long-standing practice of maintaining the standards with technical rigor, challenging but achievable design and construction practices, public engagement, and full consideration of the views of stakeholders.

The revisions in the *2016 Standards* were conceptualized, evaluated, and justified through the excellent work of California Energy Commission staff and its consultants. This document was created by Energy Commission staff including Todd Ferris, Larry Froess, P.E., Jeff Miller, P.E., Dee Anne Ross, Peter Strait, and Danny Tam.

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# ABSTRACT


The *2016 Building Energy Efficiency Standards for Low-Rise Residential Buildings* allow compliance by either a prescriptive or performance method. Performance compliance uses computer modeling software to trade off efficiency measures. For example, to allow more windows, the designer will specify more efficient windows or, to allow more west-facing windows, install a more efficient cooling system. Computer performance compliance is typically the most popular compliance method because of the flexibility it provides in the building design.

Energy compliance software must be certified by the California Energy Commission. This document establishes the rules for creating a building model, describing how the proposed design (energy use) is defined, how the standard design (energy budget) is established, and ending with what is reported on the Certificate of Compliance (CF1R). This document **does not** specify the minimum capabilities of vendor-supplied software. The Energy Commission reserves the right to approve vendor software for limited implementations of what is documented in this manual.

This *Residential Alternative Calculation Method* (ACM) *Reference Manual* explains how the proposed and standard designs are determined.

The 2016 Compliance Manager is the simulation and compliance rule implementation software specified by the Energy Commission. The Compliance Manager, called California Building Energy Code Compliance (CBECC), models all features that affect the energy performance of the building. This document establishes the process of creating a building model. Each section describes how a given component, such as a wall, is modeled for the proposed design, standard design, and ends with what is reported on the Certificate of Compliance (CF1R) for verification by the building enforcement agency.

**Keywords:** ACM, Alternative Calculation Method, *Building Energy Efficiency Standards*, California Energy Commission, California Building Energy Code Compliance, CBECC, Certificate of Compliance, CF1R, compliance manager, compliance software, computer compliance, energy budget, energy standards, energy use, performance compliance, design, proposed design, standard design

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# 1 Introduction

## 1.1 Purpose

This manual documents the rules used for modeling residential buildings for performance compliance under California's *2016 Building Energy Efficiency Standards for Low-Rise Residential Buildings*, similar to the Alternative Calculation Methods (ACM) used in the past to document software rules. This document explains how the proposed design (energy use) and standard design (energy budget) are established for a given building.

The 2016 Compliance Manager is the simulation and compliance rule implementation software specified by the California Energy Commission. For example, attics, crawl spaces, basements, and attached unconditioned spaces (garages and storage) are defined in the building modeling software.

Documentation of detailed calculation algorithms is contained in the companion volume *Appendix E, 2016 Residential Alternative Calculation Method Algorithms*.

This document is designed to establish the process of creating a building model. Each section describes how the proposed design (energy use) is defined, how the standard design (energy budget) is established, and what is reported on the Certificate of Compliance (CF1R).

This *Reference Manual* documents the compliance analysis modeling rules for all aspects of the Energy Commission's ACM Reference Method. This document **does not** specify the minimum capabilities of vendor-supplied software. The Energy Commission reserves the right to approve vendor software for limited implementations of what is documented in this manual.

## 1.2 Other Documents

The basis of this document is the *2016 Building Energy Efficiency Standards*. Documents also relied upon include the *Reference Appendices for the 2016 Building Energy Efficiency Standards* and the *2016 Residential Compliance Manual (CEC-400-2015-032)*.

Detailed modeling information for the software user can be found in the *California Building Energy Code Compliance (CBECC) User Manual*.

# 2 The Proposed Design and Standard Design

## 2.1 Overview

This chapter describes how the proposed design is modeled and how the standard design is established.

### 2.1.1 Proposed Design

The building configuration is defined by the user through entries for floor areas, wall areas, roof and ceiling areas, fenestration (which includes skylight), and door areas. Each is entered along with performance characteristics such as U-factors, SHGC, thermal mass, and so forth. Information about the orientation and tilt is required for walls, fenestration, and other elements. The user entries for all these building elements are consistent with the actual building design and configuration. If the compliance software models the specific geometry of the building by using a coordinate system or graphic entry technique, the data generated are consistent with the actual building design and configuration.

### 2.1.2 Standard Design

For low-rise residential buildings, the standard design building, from which the energy budget is established, is in the same location and has the same floor area, volume, and configuration as the proposed design, except that wall and window areas are distributed equally among the four main compass points (north, east, south, and west). For additions and alterations, the standard design shall have the same wall and fenestration areas, and orientations as the proposed building. The details are described below.

The *energy budget* for the residential standard design is the energy that would be used by a building similar to the proposed design if the proposed building met the requirements of the prescriptive standards. The compliance software generates the standard design automatically, based on fixed and restricted inputs and assumptions. Custom budget generation shall not be accessible to program users for modification when the program is used for compliance or when compliance forms are generated by the program.

The basis of the standard design is prescriptive Package A (from Section 150.1(c) of the standards, Table 150.1-A). Package A requirements vary by climate zone. Reference Joint Appendix JA2, Table 2-1, contains the 16 California climate zones and representative cities. The climate zone can be found by city, county, and zip code in JA2.1.1.

The following sections present the details on how the proposed design and standard design are determined. For many modeling assumptions, the standard design is the same as the proposed design. When a building has special features, for which the Energy Commission has established

alternate modeling assumptions, the standard design features will differ from the proposed design so the building receives appropriate credit for its efficiency. Typically, these measures require verification. Alternate features, such as zonal control, are documented as *special features* on the Certificate of Compliance. Verified features are also documented on the CF1R.

## 2.2 The Building

#### PROPOSED DESIGN

The building is defined through entries for zones, surfaces, and equipment. Zone types include attic, conditioned space, crawl space, basements, and garages. The roof (such as asphalt shingles or tile) is defined as either part of the attic or as part of a cathedral ceiling (also called a *rafter roof*). Surfaces separating conditioned space from exterior or unconditioned spaces (such as garage or storage) are modeled as interior surfaces adjacent to the unconditioned zone. Exterior surfaces of an attached garage or storage space are modeled as part of the unconditioned zone.

The input file will include entries for floor areas, wall, door, roof and ceiling areas, and fenestration and skylight areas, as well as the water heating, space conditioning, ventilation, and distribution systems.

Each surface area is entered along with performance characteristics, including building materials, U-factor and SHGC. The orientation and tilt (see Figure 1) is required for envelope elements.

Building elements are to be consistent with the actual building design and configuration.

#### STANDARD DESIGN

To determine the standard design for low-rise buildings, a building with the same general characteristics (number of stories, attached garage, climate zone) and with wall and window areas distributed equally among the four main compass points is created by the software. Energy features are set to be equal to Section 150.1(c) and Table 150.1-A. For additions and alterations, the standard design for existing features in the existing building shall have the same wall and fenestration areas and orientations as the proposed building. The details are described below.

#### VERIFICATION AND REPORTING

All inputs that are used to establish compliance requirements are reported on the CF1R for verification.

### 2.2.1 Climate and Weather

#### PROPOSED DESIGN

The user specifies the climate zone based on the zip code of the proposed building. Compliance requirements, weather, design temperatures, and Time Dependent Valuation (TDV) of energy factors are a function of the climate zone. Compliance software assumes that the ground surrounding residential buildings has a reflectivity of 20 percent in both summer and winter.

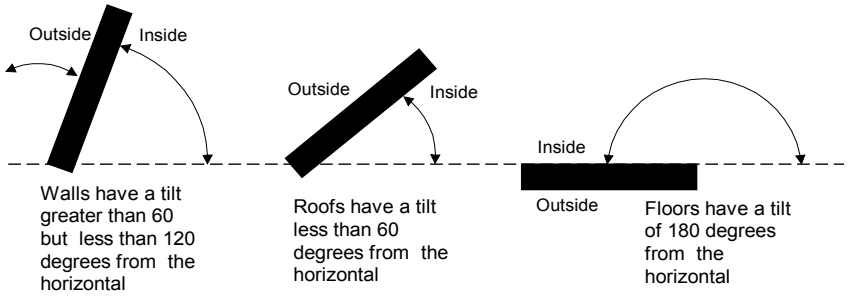
#### STANDARD DESIGN

The standard design climate zone is the same as the proposed design.

#### VERIFICATION AND REPORTING

The zip code and climate zone of the proposed design is reported on the CF1R for verification.

Figure 1: Surface Definitions




## 2.2.2 Standards Version

This input determines the appropriate federal appliance efficiency requirement for the standard design to compare with the proposed design.

#### PROPOSED DESIGN

The user inputs Compliance2017.

#### STANDARD DESIGN

The standard design cooling equipment efficiency is based on the federal requirements. A minimum SEER and EER (if applicable) that meet the current standard for the type of equipment is modeled.

#### VERIFICATION AND REPORTING

Compliance version is reported on the CF1R.

## 2.2.3 PV System Credit

The compliance credit available for photovoltaic (PV) systems is available for new construction only and is dependent on the climate zone and dwelling unit size. The credit may be used to tradeoff any efficiency measure, with limits as described below. The PV system must meet the eligibility and verification requirements of Residential Appendix RA4.6.1 and must meet the minimum system size described below.

The PV compliance credit for both single-family and multifamily buildings is calculated by the compliance software and is equal to:

$$PV_{credit} = TDV_{std} * PV_{maxpct} / 100.0 \qquad \textit{Equation 1}$$

Where:

$PV_{credit}$ = PV compliance credit (kTDV/ft²)
$TDV_{std}$ = Standard Design Compliance Total (kTDV/ft²)
$PV_{maxpct}$ = Maximum PV Credit Percentage from Table 1

The minimum PV system size for compliance credit is calculated by the compliance software and is equal to:

$$PV_{minsize} = ROUND((PV_{threshold} + PV_{addedsize}) * N_{dwellingunits}, 1) \qquad \textit{Equation 2}$$

For average dwelling units less than or equal to $CFA_{threshold}$:

$$PV_{addedsize} = 0 \qquad \textit{Equation 3}$$

For average dwelling units larger than $CFA_{threshold}$:

$$PV_{addedsize} = PV_{credit} * (CFA_{dwellingunit} - CFA_{threshold}) / PV_{generate} \qquad \textit{Equation 4}$$

Where:

$PV_{minsize}$ = Minimum PV System Size (kWdc) for compliance credit
$PV_{threshold}$ = Threshold PV System Size per dwelling unit (kWdc) from Table 2
$N_{dwellingunits}$ = Number of dwelling units
$PV_{addedsize}$ = Added PV System Size (kWdc) required
$CFA_{dwellingunit}$ = Average Conditioned floor area per dwelling unit (ft²)
$CFA_{threshold}$ = Average Threshold Conditioned floor per dwelling unit (ft²) from Table 2
$PV_{generate}$ = PV Generation Rate (kTDV/kWdc) from Table 1

The maximum PV credits in Table 1 are calculated by using a prototype analysis with the proposed features set equal to the 2016 prescriptive requirements except replacing the 2016 high performance attics (HPA) and high performance walls (HPW) with the 2013 prescriptive requirements. The percentages are calculated by dividing the compliance margin (kTDV/ft²) by the standard design compliance energy use (kTDV/ft²) and multiplying by -100. Climate zones 6 and 7 have no 2016 requirement for either HPA or HPW, so there is no PV credit in those climate zones.

#### *PROPOSED DESIGN*

The software allows the user to specify the use of PV compliance credit and establishes the minimum solar system in kilowatts direct current (DC). The software calculates the solar credit and subtracts it from the proposed design.

#### *STANDARD DESIGN*

The standard design has no PV system.

#### *VERIFICATION AND REPORTING*

A solar credit is reported as a special feature on the CF1R.

**Table 1: PV Credit Calculation Factors**

| Climate Zone | PV Generation Rate (kTDV/kWdc) | Maximum PV Credit for Single Family | Maximum PV Credit for Multi Family |
|---|---|---|---|
| 01 | 26762 | 8.0% | 4.4% |
| 02 | 30021 | 8.6% | 5.0% |
| 03 | 31137 | 6.9% | 3.1% |
| 04 | 30935 | 17.7% | 11.4% |
| 05 | 33490 | 7.6% | 2.3% |
| 06 | 30081 | 0.0% | 0.0% |
| 07 | 30701 | 0.0% | 0.0% |
| 08 | 29254 | 28.1% | 9.1% |
| 09 | 29889 | 25.9% | 11.0% |
| 10 | 30200 | 23.1% | 10.0% |
| 11 | 29693 | 17.7% | 8.7% |
| 12 | 29328 | 22.0% | 9.5% |
| 13 | 29553 | 19.8% | 9.2% |
| 14 | 31651 | 16.0% | 8.2% |
| 15 | 29177 | 16.3% | 7.3% |
| 16 | 30930 | 15.1% | 8.6% |

**Table 2: PV Threshold Factors**

| Dwelling Type | PV threshold (kWdc) | CFA threshold ($ft^2$) |
|---|---|---|
| Single Family | 2.0 | 2000 |
| Multi Family | 1.0 | 1000 |

## 2.2.4 Existing Condition Verified

These inputs are used for additions and alterations. The standard design assumption for existing conditions vary based on whether the existing conditions are verified by a home energy rating system (HERS) rater prior to construction. See Section 2.10.3 for more information.

#### *PROPOSED DESIGN*

The user inputs either yes or no. "Yes" indicates that the existing building conditions verified by a HERS rater. Default assumption is "no."

#### *STANDARD DESIGN*

The standard design assumption is based on Section 150.2(b), Table 150.2-C. If the user input is "no," the standard design for the existing component is based on the value in the second column. If the

proposed design response is "yes," the standard design value for the existing components is the value in the third column.

#### VERIFICATION AND REPORTING

Verification of existing conditions is a special feature and is reported in the HERS required verification listings on the CF1R.

### 2.2.5 Air Leakage and Infiltration

Air leakage is a building level characteristic. The compliance software distributes the leakage over the envelope surfaces in accordance with the building configuration and constructs a pressure flow network to simulate the air flows between the conditioned zones, unconditioned zones, and outside.

#### 2.2.5.1 Building Air Leakage and Infiltration (ACH50)

The air flow through a blower door at 50 Pascal (Pa) of pressure measured in cubic feet per minute is called cfm50. Cfm50 x 60 minutes divided by the volume of conditioned space is the air changes per hour at 50 Pa, called ACH50.

Specific data on ACH50 may be entered if the single-family building or townhouse will have verified building air leakage testing. In multifamily buildings, due to the lack of an applicable measurement standard, ACH50 is fixed at the above defaults.

#### PROPOSED DESIGN

ACH50 defaults to 5 for new construction in single-family buildings and townhomes, and 7 for all other buildings that have heating and/or cooling system ducts outside of conditioned space, and for buildings with no cooling system. In single-family buildings and townhomes with no heating and/or cooling system ducts in unconditioned space the default ACH50 is 4.4 and 6.2 for all others.

Specific data on ACH50 may be entered if the single-family building or townhouse will have verified building air leakage testing. User input of an ACH50 that is less than the default value becomes a special feature that requires HERS verification.

Due to the lack of an applicable measurement standard, ACH50 is fixed at the above defaults and is not a compliance variable in multi-family buildings.

#### STANDARD DESIGN

The standard design shall have 5 ACH50 for single-family buildings and 7 for other buildings (ducted space conditioning).

#### VERIFICATION AND REPORTING

When the user chooses verified building air leakage testing (any value less than the standard design), diagnostic testing for reduced infiltration, with the details and target values modeled in the proposed design, is reported in the HERS required verification listing on the CF1R.

#### 2.2.5.2 *Defining Air Net Leakage*

The compliance software creates an air leakage network for the proposed and standard design using the building description. Air leakage is distributed across the envelope surfaces according to the factors in Table 3. The air network is insensitive to wind direction. For buildings modeled with multiple conditioned zones, either a 20 square foot open door or 30 square foot open stairwell (in a multi-story building) is assumed between any two conditioned zones.

The only difference between the air network for the proposed and standard designs is the ACH50 if the user specifies a value lower than the default.

Multifamily buildings that have floors between dwelling units must define each floor as a separate zone or each dwelling unit as a separate zone.

**Table 3: Air Leakage Distribution**

| | % of Total Leakage by Surface | | | |
|---|---|---|---|---|
| **Configuration** | **Ceilings** | **Floors** | **Exterior Walls** | **House to Garage Surfaces** |
| Slab on Grade | 50 | 0 | | |
| Raised Floor | 40 | 10 | | |
| No Garage | | | 50 | 0 |
| Attached Garage | | | 40 | 10 |

### 2.2.6 *Insulation Construction Quality*

The compliance software user may specify either standard (unverified) or improved (verified high quality insulation installation, also called *quality insulation installation* or QII) for the proposed design. Buildings with standard insulation installation are modeled in the program with lower performing cavity insulation in framed walls, ceilings, and floors, and with added winter heat flow between the conditioned zone and attic to represent construction cavities open to the attic. (See Table 4.) Standard insulation does not affect the performance of continuous sheathing in any construction.

#### *PROPOSED DESIGN*

The compliance software user may specify improved quality insulation installation at the building level. The default is unverified/standard insulation installation. See Section 2.3.3 for information on modeling spray foam insulation.

#### *STANDARD DESIGN*

The standard design is modeled with standard insulation installation quality.

#### *VERIFICATION AND REPORTING*

The presence of improved/verified high quality insulation installation is reported in the HERS required verification listings on the CF1R. Improved quality insulation installation is certified by the installer and field verified to comply with RA3.5. Credit for verified quality insulation installation is applicable to ceilings/attics, knee walls, exterior walls and exterior floors.

**Table 4: Modeling Rules for Standard Insulation Installation Quality**

| Component | Modification |
|---|---|
| Walls, Floors, Attic Roofs, Cathedral Ceilings | Multiply the cavity insulation R-value/inch by 0.7 |
| Ceilings below attic | Multiply the blown and batt insulation R-value/inch by 0.96-0.00347*R |
| Ceilings below attic | Add a heat flow from the conditioned zone to the attic of 0.015 times the area of the ceiling below attic times (the conditioned zone temperature - attic temperature) whenever the attic is colder than the conditioned space |

For alterations to existing pre-1978 construction, if existing wall construction is assumed to have no insulation, no wall degradation is assumed for the existing wall.

## 2.2.7 Number of Bedrooms

#### PROPOSED DESIGN

The number of bedrooms in a building is used to establish the indoor air quality (IAQ) mechanical ventilation requirements and to determine if a building qualifies as a compact building for purposes of incentive programs.

#### STANDARD DESIGN

The standard design shall have the same number of bedrooms as the proposed design.

#### VERIFICATION AND REPORTING

The number of bedrooms is reported on the CF1R for use in field verification.

## 2.2.8 Dwelling Unit Types

Internal gains and indoor air quality (IAQ) ventilation calculations depend on the conditioned floor area and number of bedrooms. For multifamily buildings with individual IAQ ventilation systems, each combination of bedrooms and conditioned floor area has a different minimum ventilation cfm that must be verified. A *dwelling unit type* is one or more dwelling units in the building, each of which has the same floor area, number of bedrooms, and appliances.

#### PROPOSED DESIGN

For each dwelling unit type the user inputs the following information:

- Unit name
- Quantity of this unit type in building
- Conditioned floor area (CFA) in square feet per dwelling unit
- Number of bedrooms

#### STANDARD DESIGN

The standard design shall have the same number and type of dwelling units as the proposed design.

#### VERIFICATION AND REPORTING

The number of units of each type and minimum IAQ ventilation for each unit is reported on the CF1R for use in field verification.

### 2.2.9 Front Orientation

The input for the building front orientation is the actual azimuth of the front of the building. This will generally be the side of the building where the front door is located. The orientation of the other sides of a building viewed from the outside looking at the front door are called front, left, right, back, or a value relative to the front, and the compliance software calculates the actual azimuth from this input. Multiple orientation compliance can be selected for newly constructed buildings only.

#### PROPOSED DESIGN

The user specifies whether compliance is for multiple orientations or for a site-specific orientation. For site-specific orientation the user inputs the actual azimuth of the front in degrees from true north.

#### STANDARD DESIGN

The compliance software constructs a standard design building that has 25 percent of the proposed model wall and window areas facing each cardinal orientation regardless of the proposed model distribution of wall and window area.

#### VERIFICATION AND REPORTING

A typical reported value would be "290 degrees (west)". This would indicate that the front of the building faces north 70° west in surveyors terms. The closest orientation on 45° compass points should be reported in parenthesis (for example: north, northeast, east, southeast, south, southwest, west, or northwest). When compliance is shown for multiple orientations, "all orientations" or “cardinal” is reported as a special feature on the CF1R and the energy use results are reported for four orientations including north, east, south, and west.

### 2.2.10 Natural Gas Availability

The user specifies whether natural gas is available at the site. This is used to establish the TDV values from Reference Appendices JA3 used by the compliance software in determining standard and proposed design energy use.

#### PROPOSED DESIGN

The user specifies whether natural gas is available at the site. For newly constructed buildings, natural gas is available if a gas service line can be connected to the site without a gas main extension. For additions and alterations, natural gas is available if a gas service line is connected to the existing building.

#### STANDARD DESIGN

The standard design TDV values for space heating are as defined in Section 2.4.1 and for water heating are as defined in Section 2.9.

#### VERIFICATION AND REPORTING

Whether natural gas is or is not available is reported on the CF1R.

### 2.2.11 Attached Garage

The user specifies whether there is an attached garage. The garage zone is modeled as an unconditioned zone (see Section 2.8).

#### PROPOSED DESIGN

The user specifies whether there is an attached unconditioned garage.

#### STANDARD DESIGN

The standard design has the same attached garage assumption as the proposed design.

#### VERIFICATION AND REPORTING

Features of an attached garage are reported on the CF1R.

### 2.2.12 Lighting

Details of the calculation assumptions for lighting loads are included Appendix C and are based on the Codes and States Enhancement Initiative (CASE) report on plug loads and lighting (Rubin 2016, see Appendix D).

#### PROPOSED DESIGN

Fraction of portable lighting, power adjustment multiplier and the exterior lighting power adjustment multiplier (Watts/ft$^2$ – Watts per square foot) are fixed assumptions.

#### STANDARD DESIGN

The standard design lighting is set equal to the proposed design lighting.

#### VERIFICATION AND REPORTING

No lighting information is reported on the CF1R for compliance with Title 24, Part 6.

### 2.2.13 Appliances

Details of the calculation assumptions for appliances and plug loads are contained in Appendix C and based on the Codes and States Enhancement Initiative (CASE) report on plug loads and lighting (Rubin 2016, see Appendix D).

#### PROPOSED DESIGN

All buildings are assumed to have a refrigerator, dishwasher, and cooking appliance. Optionally, buildings can have a clothes washer and clothes dryer. The user can select fuel type as gas or electric for the clothes dryer and cooking appliance.

#### STANDARD DESIGN

The standard design appliances are set equal to the proposed appliances.

#### VERIFICATION AND REPORTING

No information for the appliance types listed above is reported on the CF1R for compliance with Title 24, Part 6.

## 2.3 Building Materials and Construction

### 2.3.1 Materials

Only materials approved by the Commission may be used in defining constructions. Additional materials may be added to the Compliance Manager.

Table 5 shows a partial list of the materials currently available for construction assemblies.

#### MATERIAL NAME

The material name is used to select the material for a construction.

#### THICKNESS

Some materials, such as three-coat stucco, are defined with a specific thickness (not editable by the compliance user). The thickness of other materials, such as softwood used for framing, is selected by the compliance user based on the construction of the building.

#### CONDUCTIVITY

The conductivity of the material is the steady state heat flow per square foot, per foot of thickness, or per degree Fahrenheit temperature difference. It is used in simulating the heat flow in the construction.

#### COEFFICIENT FOR TEMPERATURE ADJUSTMENT OF CONDUCTIVITY

The conductivity of insulation materials vary with their temperature according to the coefficient listed. Other materials have a coefficient of zero (0) and their conductivity does not vary with temperature.

**Table 5: Materials List**

| Material Name | Thickness (in.) | Conductivity (Btu/h-°F-ft) | Coefficient for Temperature Adjustment of Conductivity (°F(-1)) | Specific Heat (Btu/lb-°F) | Density (lb/ft$^3$) | R-Value per Inch (°F-ft$^2$-h/ Btu-in) |
|---|---|---|---|---|---|---|
| Gypsum Board | 0.5 | 0.09167 | 0.00122 | 0.27 | 40 | 0.9091 |
| Wood Layer | Varies | 0.06127 | 0.0012 | 0.45 | 41 | 1.36 |
| Synthetic Stucco | 0.375 | 0.2 | | 0.2 | 58 | 0.2 |
| 3 Coat Stucco | 0.875 | 0.4167 | | 0.2 | 116 | 0.2 |
| Carpet | 0.5 | 0.02 | | 0.34 | 12.3 | 4.1667 |
| Light Roof | 0.2 | 1 | | 0.2 | 120 | 0.0833 |
| 5 PSF Roof | 0.5 | 1 | | 0.2 | 120 | 0.0833 |
| 10 PSF Roof | 1 | 1 | | 0.2 | 120 | 0.0833 |
| 15 PSF Roof | 1.5 | 1 | | 0.2 | 120 | 0.0833 |
| 25 PSF Roof | 2.5 | 1 | | 0.2 | 120 | 0.0833 |
| TileGap | 0.75 | 0.07353 | | 0.24 | 0.075 | 1.1333 |
| SlabOnGrade | 3.5 | 1 | | 0.2 | 144 | 0.0833 |
| Earth | | 1 | | 0.2 | 115 | 0.0833 |
| SoftWood | | 0.08167 | 0.0012 | 0.39 | 35 | 1.0204 |
| Concrete | | 1 | | 0.2 | 144 | 0.0833 |
| Foam Sheathing | varies | varies | 0.00175 | 0.35 | 1.5 | varies |
| Ceiling Insulation | varies | varies | 0.00418 | 0.2 | 1.5 | varies |
| Cavity Insulation | varies | varies | 0.00325 | 0.2 | 1.5 | varies |
| Vertical Wall Cavity | 3.5 | 0.314 | 0.00397 | 0.24 | 0.075 | |
| GHR Tile | 1.21 | 0.026 | 0.00175 | 0.2 | 38 | |
| ENSOPRO | 0.66 | 0.03 | 0.00175 | 0.35 | 2 | |
| ENSOPRO Plus | 1.36 | 0.025 | 0.00175 | 0.35 | 2 | |
| Door | | | | | | |

### *Specific Heat*

The specific heat is the amount of heat in British thermal units (Btu) it takes to raise the temperature of one pound of the material one degree Fahrenheit.

### *Density*

The density of the material is its weight in pounds per cubic foot.

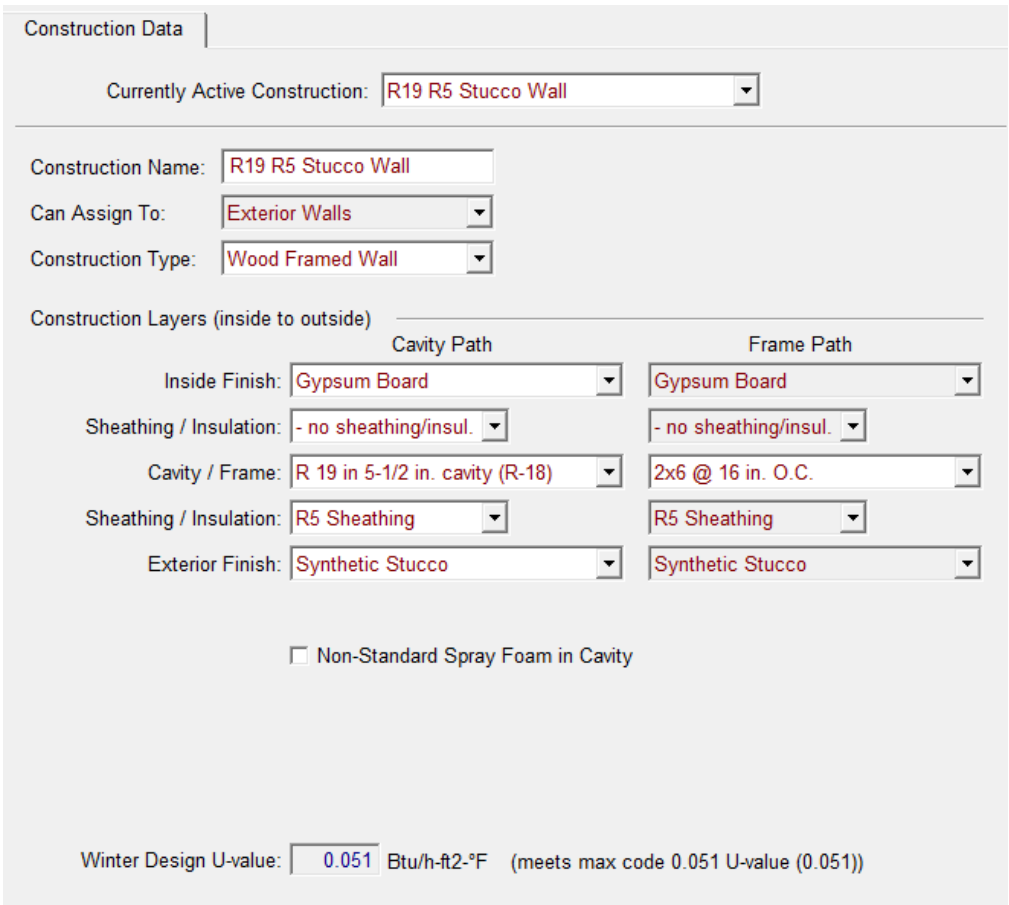
### *R-value per inch*

The R-value is the resistance to heat flow for a 1-inch thick layer.

## 2.3.2 Construction Assemblies

Constructions are defined by the compliance user for use in defining the building. The user assembles a construction from one or more layers of materials as shown in Figure 2. For framed constructions, there is a framing layer that has parallel paths for the framing and the cavity between the framing members. The layers that are allowed depend on the surface type. The compliance manager calculates a winter design U-factor that is compared to a construction that meets the prescriptive standard. The U-factor is displayed as an aid to the user. The calculations used in the energy simulation are based on each individual layer and framing rather than the U-factor.

Figure 2: Example Construction Data Screen




#### *Assembly Types*

The types of assemblies are:

Exterior wall
Interior wall
Underground walls
Attic roof
Cathedral roof
Ceiling below attic

Interior ceiling
Interior floor
Exterior floor (over unconditioned space or exterior)
Floor over crawl space

#### CONSTRUCTION TYPE:

1. Ceiling below attic (the roof structure is not defined here, but is part of the attic), wood framed. In a residence with a truss roof, the ceiling is where the insulation is located, while the structure above the ceiling is encompassed by the term attic or roof. The attic or roof consists of (moving from inside to outside) the radiant barrier, below deck insulation, framing, above deck insulation, and the roofing product, such as asphalt or tile roofing. See more in Section 2.4.5.

2. Cathedral ceiling (with the roof defined as part of the assembly), wood framed. Since there is no attic, the roof structure is connected to the insulated assembly at this point.

3. Roof, structurally insulated panels (SIP).

4. Walls (interior, exterior, underground), wood or metal framed, or structurally insulated panel (SIP).

5. Floors (over exterior, over crawl space, or interior).

6. Party surfaces separate conditioned space included in the analysis from conditioned space that may or may not be included in the analysis. Party surfaces for spaces that are modeled include surfaces between multifamily dwelling units. Party surfaces for spaces not included in the analysis include spaces joining an addition alone to the existing dwelling. Interior walls, ceilings, or floors can be party surfaces.

#### CONSTRUCTION LAYERS:

All assemblies have a cavity path and a frame path.

Spray foam insulation may use either default values with no special inspection requirements, or higher values when supported by an ESR number (see details Section 2.3.3 and RA3.5) verified by a HERS rater.

As assemblies are completed, the screen displays whether the construction meets the prescriptive requirement for that component.

#### PROPOSED DESIGN

The user defines a construction for each surface type included in the proposed design. Any variation in insulation R-value, framing size or spacing, interior or exterior sheathing, or interior or exterior finish requires the user to define a different construction. Insulation R-values are based on manufacturer-rated properties rounded to the nearest whole R-value. Layers such as sheetrock, wood sheathing, stucco, and carpet whose properties are not compliance variables are included as generic layers with standard thickness and properties.

Walls separating the house from an attached unconditioned attic or garage are modeled as interior walls with unconditioned space as the adjacent zone, which the compliance manager recognizes as a demising wall. Floors over a garage are modeled as floor over exterior. The exterior walls, floor, and ceiling/roof of the garage are modeled as part of the unconditioned garage zone.

#### STANDARD DESIGN

The compliance software assembles a construction that meets the prescriptive standards for each user-defined construction or assembly.

#### VERIFICATION AND REPORTING

All proposed constructions, including insulation, frame type, frame size, and exterior finish or exterior condition are listed on the CF1R. Non-standard framing (e.g., 24" on center wall framing, advanced wall framing) is reported as a special feature.

## 2.3.3 Spray Foam Insulation

The R-values for spray-applied polyurethane foam (SPF) insulation differ depending on whether the product is open cell or closed cell.

**Table 6: Required Thickness Spray Foam Insulation**

| Required R-values for SPF insulation | R-11 | R-13 | R-15 | R-19 | R-21 | R-22 | R-25 | R-30 | R-38 |
|---|---|---|---|---|---|---|---|---|---|
| Required thickness closed cell @ R5.8/inch | 2.00 inches | 2.25 inches | 2.75 inches | 3.50 inches | 3.75 inches | 4.00 inches | 4.50 inches | 5.25 inches | 6.75 inches |
| Required thickness open cell @ R3.6/inch | 3.0 inches | 3.5 inches | 4.2 inches | 5.3 inches | 5.8 inches | 6.1 inches | 6.9 inches | 8.3 inches | 10.6 inches |

Additional documentation and verification requirements for a value other than the default values shown in Table 6 is required (see RA3.5.6).

### 2.3.3.1 *Medium Density Closed-Cell SPF Insulation*

The default R-value for spray foam insulation with a closed cellular structure is R-5.8 per inch, based on the installed nominal thickness of insulation. Closed cell insulation has an installed nominal density of 1.5 to 2.5 pounds per cubic foot (pcf).

### 2.3.3.2 *Low Density Open-Cell SPF Insulation*

The default R-value for spray foam insulation with an open cellular structure is calculated as R-3.6 per inch, calculated based on the nominal required thickness of insulation. Open cell insulation has an installed nominal density of 0.4 to 1.5 pounds per cubic foot (pcf).

#### PROPOSED DESIGN

The user will select either typical values for open cell or closed cell spray foam insulation or higher than typical values, and enter the total R-value (rounded to the nearest whole value).

#### STANDARD DESIGN

The compliance software assembles a construction that meets the prescriptive standards for each assembly type (ceiling/roof, wall, and floor).

#### VERIFICATION AND REPORTING

When the user elects to use higher than typical R-values for open cell or closed cell spray foam insulation, a special features note is included on the CF1R requiring documentation requirements specified in RA4.1.7. Additionally, a HERS verification requirement for the installation of spray foam insulation using higher than default values is included on the CF1R.

## 2.4 Building Mechanical Systems

A space-conditioning system (also referred to as HVAC system) is made up of the heating subsystem (also referred to as heating unit or heating equipment or heating system), cooling subsystem (also referred to as cooling unit, cooling equipment, or cooling system) (if any), the distribution subsystem details and fan subsystem (if any). Ventilation cooling systems and indoor air quality ventilation systems are defined at the building level for single-family dwellings or as part of the dwelling unit information for multifamily buildings (see also Sections 2.4.9 and 2.4.10).

### *2.4.1 Heating Subsystems*

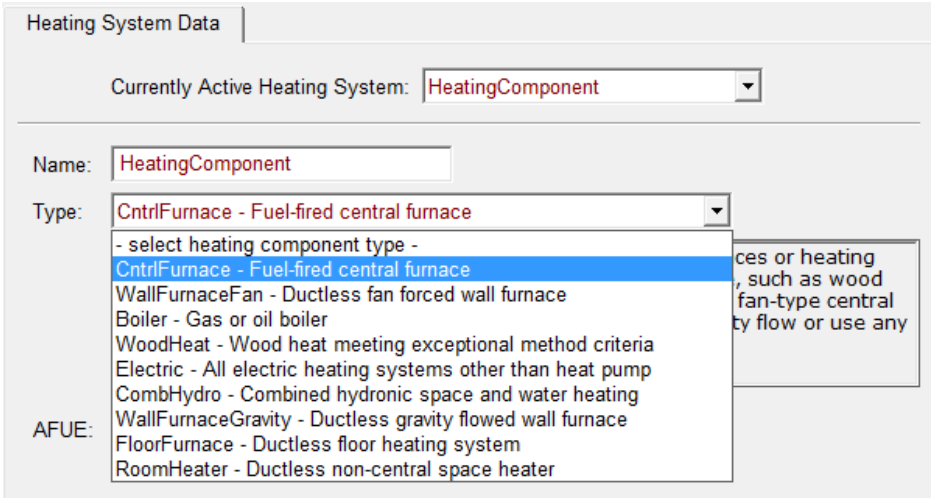
The heating subsystem describes the equipment that supplies heat to a space conditioning system. Heating subsystems are categorized according to the types shown in Table 7.

#### PROPOSED DESIGN

The user selects the type and supplies inputs for the heating subsystem in the heating system data screen shown in Figure 3. The user inputs the appropriate rated heating efficiency factor. Except for heat pumps, the rated heating capacity is not used as a compliance variable by the compliance software. For heat pumps the user inputs the rated heating capacity at 47°F and 17°F for the heat pump compressor to be installed and the software sizes the backup electric resistance heat for use in the simulation.

Until there is an approved compliance option for ductless heat pumps (ductless mini-split, multi-split, and VRF systems) these systems are simulated as a minimum efficiency split system equivalent to the standard design and with default duct conditions.

Figure 3: Heating System Data Screen




### *Standard Design*

When electricity is used for heating, the heating equipment for the standard design is an electric split system heat pump with default ducts in the attic and a heating seasonal performance factor (HSPF) meeting the *Appliance Efficiency Regulations* (CEC-400-2014-009) requirements for split systems with default ducts in the attic. The standard design heat pump compressor size is determined by the software as the larger of the compressor size calculated for air conditioning load, or the compressor with a 47°F rating that is 75 percent of the heating load (at the heating design temperature).

When electricity is not used for heating, the equipment used in the standard design building is a gas furnace (or propane if natural gas is not available) with default ducts in the attic and an annual fuel utilization efficiency (AFUE) meeting the *Appliance Efficiency Regulations* minimum efficiency for central systems. When a proposed design uses both electric and non-electric heat, the standard design is a gas furnace.

### *Verification and Reporting*

The proposed heating system type and rated efficiency are reported in the compliance documentation. For heat pumps, which are supplemented by electric resistance back-up heating, the rated heating capacity of each proposed heat pump is reported on the CF1R to verify that installed capacity is equal or larger than the capacities modeled at 47° and 17°.

**Table 7: HVAC Heating Equipment Types**

| Name | Heating Equipment Description |
|---|---|
| CntrlFurnace | Gas- or oil-fired central furnaces, propane furnaces or heating equipment considered equivalent to a gas-fired central furnace, such as wood stoves that qualify for the wood heat exceptional method. Gas fan-type central furnaces have a minimum AFUE=78%. Distribution can be gravity flow or use any of the ducted systems. (Efficiency metric: AFUE) |
| WallFurnaceGravity | Non-central gas- or oil-fired wall furnace, gravity flow. Equipment has varying efficiency requirements by capacity. Distribution is ductless. (Efficiency metric: AFUE) |
| WallFurnaceFan | Non-central gas- or oil-fired wall furnace, fan-forced. Equipment has varying efficiency requirements by capacity. Distribution is ductless. (Efficiency metric: AFUE) |
| FloorFurnace | Non-central gas- or oil-fired floor furnace. Equipment has varying efficiency requirements by capacity. Distribution is ductless. (Efficiency metric: AFUE) |
| GroundSourceHeatPump | A water-to-air heat pump or water-to-water heat pump using fluid flowing through underground piping as a heat source/heat sink, that provides heating, cooling, or heating and cooling conditioning with forced air or hydronic distribution. May also provide heat for domestic hot water. (Efficiency metric: COP) |
| RoomHeater | Non-central gas- or oil-fired room heaters. Non-central gas- or oil-fired wall furnace, gravity flow. Equipment has varying efficiency requirements by capacity. Distribution is ductless. (Efficiency metric: AFUE) |
| WoodHeat | Wood-fired stove. In areas with no natural gas available, a wood heating system with any back-up heating system, is allowed to be installed if exceptional method criteria described in the Residential Compliance Manual are met. (Efficiency Metric: N/A) |
| Boiler | Gas or oil boilers. Distribution systems can be Radiant, Baseboard, or any of the ducted systems. Boiler may be specified for dedicated hydronic systems. Systems in which the boiler provides space heating and fires an indirect gas water heater (IndGas) may be listed as Boiler/CombHydro Boiler and is listed under “Equipment Type” in the HVAC Systems listing. (Efficiency metric: AFUE) |
| SplitHeatPump | Heating side of a split heat pump heating system that has one or more outdoor units supply heat to each habitable space in the dwelling unit. Heat is at least partly distributed using one of the ducted systems. (Efficiency metric: HSPF) |
| SDHVSplitHeatPump | Heating side of a small Duct, High Velocity, Central split system that produces at least 1.2 inches of external static pressure when operated at the certified air volume rate of 220–350 CFM per rated ton of cooling and uses high velocity room outlets generally greater than 1,000 fpm that have less than 6.0 square inches of free area. (Efficiency Metric: HSPF) |
| DuctlessMiniSplitHeatPump: | Heating side of a heat pump system that has single outdoor section, and one or more ductless indoor sections. The indoor section(s) cycle on and off in unison in response to a single indoor thermostat. (Efficiency Metric: HSPF) |
| DuctlessMultiSplitHeatPump | Heating side of a heat pump system that has a single outdoor section and two or more ductless indoor sections. The indoor sections operate independently and can be used to condition multiple zones in response to multiple indoor thermostats. (Efficiency Metric: HSPF) |
| DuctlessVRFHeatPump | Heating side of a variable refrigerant flow (VRF) heat pump system that has one or more outdoor sections and two or more ductless indoor sections. The indoor sections operate independently and can be used to condition multiple zones in response to multiple indoor thermostats. (Efficiency Metric: HSPF) |
| RoomHeatPump | Same as DuctlessHeatPump except that heat is not supplied to each habitable space in the dwelling unit. (Efficiency metric: COP) |
| PkgHeatPump | Heating side of central packaged heat pump systems. Central packaged heat pumps are heat pumps in which the blower, coils, and compressor are contained in a single package, powered by single phase electric current, air cooled, and rated below 65,000 Btu/h. Distribution system is one of the ducted systems. (Efficiency metric: HSPF) |

| Name | Heating Equipment Description |
|---|---|
| LrgPkgHeatPump | Heating side of large packaged units rated at or above 65,000 Btu/h (heating mode). Distribution system is one of the ducted systems. These include water source and ground source heat pumps. (Efficiency metric: COP) |
| Electric | All electric heating systems other than space conditioning heat pumps. Included are electric resistance heaters, electric boilers and storage water heat pumps (air-water) (StoHP). Distribution system can be Radiant, Baseboard, or any of the ducted systems. |
| CombHydro | Water heating system can be storage gas (StoGas, LgStoGas), storage electric (StoElec) or heat pump water heaters (StoHP). Distribution systems can be Radiant, Baseboard, or any of the ducted systems and can be used with any of the terminal units (FanCoil, RadiantFlr, Baseboard, and FanConv). (Efficiency metric: AFUE) |
| AirToWaterHeatPump | An air-source heat pump that uses hydronic distribution to provide heating, cooling, or heating and cooling conditioning. May also provide heat for domestic hot water. (Efficiency metric: COP) |

## 2.4.2 Combined Hydronic Space/Water Heating

Combined hydronic space/water heating is a system whereby a water heater is used to provide both space heating and water heating. Dedicated hydronic space-heating systems are also a modeling capability. Space-heating terminals may include fan coils, baseboards, and radiant surfaces (floors, walls or ceilings).

For combined hydronic systems, the water-heating portion is modeled in the normal manner. For space heating, an effective AFUE is calculated for gas water heaters. For electric water heaters, an effective HSPF is calculated. The procedures for calculating the effective AFUE or HSPF are described below.

Combined hydronic space conditioning cannot be combined with zonal control credit.

#### *PROPOSED DESIGN*

When a fan coil is used to distribute heat, the fan energy and the heat contribution of the fan motor must be considered. The algorithms for fans used in combined hydronic systems are the same as those used for gas furnaces and are described in Chapter 3.

If a large fan coil is used and air-distribution ducts are located in the attic, crawlspace, or other unconditioned space, the efficiency of the air-distribution system must be determined using methods consistent with those described in Section 2.4.6. Duct efficiency is accounted for when the distribution type is ducted.

### 2.4.2.1 *Large or Small Storage Gas Water Heater*

When storage gas water heaters are used in combined hydronic applications, the effective AFUE is given by the following equation:

$$AFUE_{eff} = RE - \left[\frac{PL}{RI}\right] \qquad \textit{Equation 5}$$

Where:

$AFUE_{eff}$ = The effective AFUE of the gas water heater in satisfying the space heating load.

RE = The recovery efficiency (or thermal efficiency) of the gas storage water heater. A default value of 0.70 may be assumed if the recovery efficiency is unknown. This value is generally available from the Energy Commission appliance directory.

PL = Pipe losses (kBtu/h). This can be assumed to be zero when less than 10 feet of piping between the water heater storage tank and the fan coil or other heating elements are located in unconditioned space.

RI = The rated input of the gas water heater (kBtu/h) available from the Energy Commission appliance directory.

#### 2.4.2.2 *Instantaneous Gas Water Heater*

When instantaneous gas water heaters are used in combined hydronic applications, the effective AFUE is given by the following equation:

$$AFUE_{eff} = EF \qquad \textit{Equation 6}$$

Where:

$AFUE_{eff}$ = The effective AFUE of the gas water heater in satisfying the space heating load.

EF = The rated Energy Factor of the instantaneous gas water heater.

#### 2.4.2.3 *Storage Electric Water Heater*

The HSPF of storage water heaters used for space heating in a combined hydronic system is given by the following equations.

$$HSPF_{eff} = 3.413\left[1 - \frac{PL}{3.413kWi}\right] \qquad \textit{Equation 7}$$

Where:

$HSPF_{eff}$ = The effective HSPF of the electric water heater in satisfying the space heating load.

PL = Pipe losses (kBtu/h). This can be assumed to be zero when less than 10 feet of piping between the water heater storage tank and the fan coil or other heating elements are located in unconditioned space.

kWi = The kilowatts of input to the water heater available from the Energy Commission's appliance directory.

#### STANDARD DESIGN

When electricity is used for heating, the heating equipment for the standard design is an electric split system heat pump with an HSPF meeting the *Appliance Efficiency Regulations* requirements for split systems. The standard design heat pump compressor size is determined by the software based on the compressor size calculated for the air conditioning system.

When electricity is not used for heating, the equipment used in the standard design building is a gas furnace (or propane if natural gas is not available) with default ducts in the attic and an AFUE meeting the *Appliance Efficiency Regulations* minimum efficiency for central systems. When a proposed design uses both electric and non-electric heat, the standard design is a gas furnace.

### 2.4.3 Special Systems – Hydronic Distribution Systems and Terminals

[Not yet implemented]

#### PROPOSED DESIGN

This listing is completed for hydronic systems that have more than 10 feet of piping (plan view) located in unconditioned space. As many rows as necessary may be used to describe the piping system.

#### STANDARD DESIGN

The standard design is established for a hydronic system in the same way as for a central system, as described in Section 2.4.1.

#### VERIFICATION AND REPORTING

A hydronic or combined hydronic system is reported on the CF1R.

Other information reported includes:

- *Piping Run Length (ft).* The length (plan view) of distribution pipe located in unconditioned space, in feet, between the primary heating/cooling source and the point of distribution.
- *Nominal Pipe Size (in.).* The nominal (as opposed to true) pipe diameter in inches.
- *Insulation Thickness (in.).* The thickness of the insulation in inches. Enter "none" if the pipe is uninsulated.
- *Insulation R-value (hr-ft$^2$-°F/Btu).* The installed R-value of the pipe insulation. Minimum pipe insulation for hydronic systems is as specified in Section 150.0(j).

### 2.4.4 Ground-Source Heat Pump

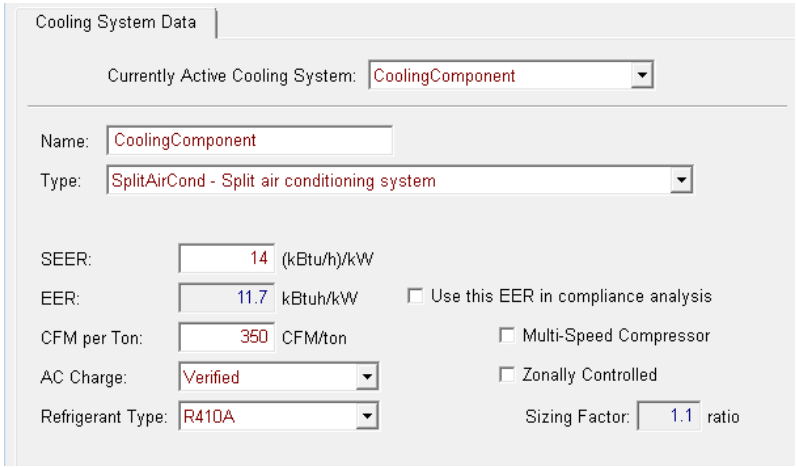
A ground-source heat pump system, which uses the earth as a source of energy for heating and as a heat sink for energy when cooling, is simulated as a minimum efficiency split system equivalent to

the standard design with default duct conditions in place of the proposed system. The mandatory efficiencies for ground-source heat pumps are a minimum coefficient of performance (COP) for heating and EER for cooling.

## 2.4.5 Cooling Subsystems

The cooling subsystem describes the equipment that supplies cooling to a space-conditioning system.

Figure 4: Cooling System Data




#### PROPOSED DESIGN

Cooling subsystems are categorized according to the types shown in Table 8. The user selects the type of cooling equipment and enters basic information to model the energy use of the equipment. Enter the cooling equipment type and additional information based on the equipment type and zoning, such as the SEER and EER. For some types of equipment, the user may also specify that the equipment has a multispeed compressor and if the system is zoned or not via checkboxes. For ducted cooling systems, the cooling air flow from the conditioned zone through the cooling coil is input as CFM per ton. The rated cooling capacity is not a compliance variable.

Until there is an approved compliance option for ductless heat pumps (ductless mini-split, multi-split, and variable refrigerant flow [VRF] systems), these systems are simulated as a minimum efficiency split system equivalent to the standard design with default duct conditions.

See sections below for the details of specific inputs.

#### STANDARD DESIGN

The cooling system for the standard design building is a non-zonal control split system air conditioner or heat pump meeting the minimum requirements of the *Appliance Efficiency Regulations*. The standard design system shall assume verified refrigerant charge in climate zones 2 and 8 through 15 for all ducted split systems, ducted package systems, mini-split, multi-split, and VRF systems. Mandatory fan efficacy is assumed in all climate zones.

**Table 8: HVAC Cooling Equipment Types**

| Name | Cooling Equipment Description |
|---|---|
| NoCooling | Entered when the proposed building is not cooled or when cooling is optional (to be installed at some future date). Both the standard design equivalent building and the proposed design use the same default system (refer to Section 2.4.8.3). (Efficiency metric: SEER) |
| SplitAirCond | Split air conditioning systems. Distribution system is one of the ducted systems. (Efficiency metric: SEER and EER) |
| PkgAirCond | Central packaged air conditioning systems less than 65,000 Btu/h cooling capacity. Distribution system is one of the ducted systems. (Efficiency metric: SEER and EER) |
| LrgPkgAirCond | Large packaged air conditioning systems rated at or above 65,000 Btu/h cooling capacity. Distribution system is one of the ducted systems. (Efficiency metric: EER) |
| SDHVSplitAirCond | Small duct, high velocity, split A/C system. (Efficiency Metric: SEER) |
| DuctlessSplitAirCond | One or more split air conditioning outdoor units that use refrigerant to transport cooling to at least one terminal in each habitable space in the dwelling unit. These include small ductless mini-split and multiple-split heat air conditioners and packaged terminal (commonly called "through-the-wall") units. Cooling is not distributed using ducts either inside or outside of the conditioned space. (Efficiency metric: EER) |
| RoomAirCond | Same as DuctlessSplitAirCond except that cooling is not supplied to each habitable space in the dwelling unit. (Efficiency metric: EER) |
| SplitHeatPump | Cooling side of split heat pump systems. Distribution system is one of the ducted systems. (Efficiency metric: SEER and EER<65,000 Btu/h EER>65,000 Btu/h) |
| PkgHeatPump | Cooling side of central single-packaged heat pump systems with a cooling capacity less than 65,000 Btu/h. Distribution system is one of the ducted systems. (Efficiency metric: SEER) |
| LrgPkgHeatPump | Cooling side of large packaged heat pump systems rated at or above 65,000 Btu/h cooling capacity. Distribution system is one of the ducted systems. (Efficiency metric: EER) |
| SDHVSplitHeatPump | Cooling side of a small Duct, High Velocity, Central split system that produces at least 1.2 inches of external static pressure when operated at the certified air volume rate of 220–350 CFM per rated ton of cooling and uses high velocity room outlets generally greater than 1,000 fpm that have less than 6.0 square inches of free area. (Efficiency Metric: SEER) |
| DuctlessMiniSplitAirCond | Ductless mini-split A/C system. (Efficiency Metric: SEER) |
| DuctlessMultiSplitAirCond | Ductless multi-split A/C system. (Efficiency Metric: SEER) |
| DuctlessVRFAirCond | Ductless variable refrigerant flow (VRF) A/C system. (Efficiency Metric: SEER) |
| DuctlessMiniSplitHeatPump: | Cooling side of a heat pump system that has single outdoor section and one or more ductless indoor sections. The indoor section(s) cycle on and off in unison in response to a single indoor thermostat. (Efficiency Metric: SEER) |
| DuctlessMultiSplitHeatPump | Cooling side of a heat pump system that has a single outdoor section and two or more ductless indoor sections. The indoor sections operate independently and can be used to condition multiple zones in response to multiple indoor thermostats. (Efficiency Metric: SEER) |
| DuctlessVRFHeatPump | Cooling side of a variable refrigerant flow (VRF) heat pump system that has one or more outdoor sections and two or more ductless indoor sections. The indoor sections operate independently and can be used to condition multiple zones in response to multiple indoor thermostats. (Efficiency Metric: SEER) |
| RoomHeatPump | Same as DuctlessHeatPump except that cooling is not supplied to each habitable space in the dwelling unit. (Efficiency metric: EER) |
| AirToWaterHeatPump | An air-source heat pump that uses hydronic distribution to provide heating, cooling, or heating and cooling conditioning. May also provide heat for domestic hot water. (Efficiency metric: EER) |

| Name | Cooling Equipment Description |
|---|---|
| GroundSourceHeatPump | A water-to-air heat pump or water-to-water heat pump using fluid flowing through underground piping as a heat source/heat sink, that provides heating, cooling, or heating and cooling conditioning with forced air or hydronic distribution. May also provide heat for domestic hot water. (Efficiency metric: EER) |
| EvapDirect | Direct evaporative cooling systems. Assume minimal efficiency air conditioner. The default distribution system location is DuctAttic; evaporative cooler duct insulation requirements are the same as those for air conditioner ducts. (Efficiency metric: SEER) |
| EvapIndirDirect | Indirect-direct evaporative cooling systems. Assume energy efficiency ratio of 13 EER. Requires air flow and media saturation effectiveness from the CEC directory. (Efficiency metric: EER) |
| EvapIndirect | Indirect cooling systems. The default distribution system location is DuctAttic; evaporative cooler duct insulation requirements are the same as those for air conditioner ducts. Assume energy efficiency ratio of 13 EER. Requires air flow and media saturation effectiveness from the CEC directory. (Efficiency metric: EER) |
| EvapCondenser | Evaporatively Cooled Condensers. A split mechanical system, with a water-cooled condenser coil. (Efficiency metric: EER) |

#### *VERIFICATION AND REPORTING*

Information shown on the CF1R includes cooling equipment type and cooling efficiency (SEER and/or EER). Measures requiring verification (see Table 9) are listed in the HERS verification section of the CF1R.

### 2.4.5.1 *Verified Refrigerant Charge or Fault Indicator Display*

Proper refrigerant charge is necessary for electrically driven compressor air-conditioning systems to operate at full capacity and efficiency. Software calculations set the compressor efficiency multiplier to 0.90 to account for the effect of improper refrigerant charge or 0.96 for proper charge.

#### *PROPOSED DESIGN*

The software allows the user to indicate if systems will have diagnostically tested refrigerant charge or a field-verified fault indicator display (FID). This applies only to ducted split systems and packaged air conditioners and heat pumps.

#### *STANDARD DESIGN*

The standard design building is modeled with either diagnostically tested refrigerant charge or a field-verified FID if the building is in climate zone 2 or 8 through 15, and refrigerant charge verification is required by Section 150.1(c) and Table 150.1-A for the proposed cooling system type.

#### *VERIFICATION AND REPORTING*

These features require field verification or diagnostic testing and are reported in the HERS required verification listings on the CF1R. Details on refrigerant charge measurement are discussed in *Reference Residential Appendix RA3.2*. Information on the requirements for FIDs is located in *Reference Joint Appendix JA6.1*.

**Table 9: Summary of Air Conditioning Measures Requiring Verification**

| Measure | Description | Procedures (Need Update) |
|---|---|---|
| Verified Refrigerant Charge | Air-cooled air conditioners and air-source heat pumps must be tested diagnostically to verify that the system has the correct refrigerant charge. The system must also meet the system airflow requirement. | RA3.2, RA1.2 |
| Verified Fault Indicator Display | A Fault Indicator Display can be installed as an alternative to refrigerant charge testing. | RA3.4.2 |
| Verified System Airflow | When compliance requires verified System Airflow greater than or equal to a specified criterion. | RA3.3 |
| Verified Air-handling Unit Fan Efficacy | To verify that Fan Efficacy (Watt/cfm) is less than or equal to a specified criterion. | RA3.3 |
| Verified EER | Credit for increased EER by installation of specific air conditioner or heat pump models. | RA3.4.3, RA3.4.4.1 |
| Verified SEER | Credit for increased SEER. | RA3.4.3, RA3.4.4.1 |
| Evaporatively Cooled Condensers | Must be combined with duct leakage testing, refrigerant charge, and verified EER. | RA3.1, RA3.1.4.3, RA3.2, RA1.2, RA3.4.3, RA3.4.4.1 |

### 2.4.5.2 *Verified System Airflow*

Adequate airflow from the conditioned space is required to allow ducted air-conditioning systems to operate at their full efficiency and capacity. Efficiency is achieved by the air distribution system design by improving the efficiency of motors or air distribution systems with less resistance to airflow. Software calculations account for the impact of airflow on sensible heat ratio and compressor efficiency.

A value less than 350 cfm/ton (minimum 150 cfm/ton) is a valid input only if zonally controlled equipment is selected and multispeed compressor is not selected.

The mandatory requirement in Section 150.0(m)13 is for an air-handling unit with a verified airflow rate greater than or equal to 350cfm/ton. Credit for a higher airflow rate requires diagnostic testing using procedures in *Reference Residential Appendix RA3.3*.

For Single Zone Systems:

- Installers may elect to use an alternative to HERS verification of 350 cfm/ton: HERS verification of a return duct design that conforms to the specification given in Table 150.0-B or C.
- The return duct design alternative is not an input to the compliance software but must be documented on the certificate of installation.

- If a value greater than 350 cfm/ton is modeled for compliance credit, the alternative return duct design method using Table 150.0-B or C is not allowed for use in demonstrating compliance.
- Multispeed or variable-speed compressor systems must verify airflow rate (cfm/ton) for system operation at the maximum compressor speed and the maximum air handler fan speed.

For Zonally Controlled Systems:

- The Table 150.0-B or C return duct design alternative is not allowed for zonally controlled systems.
- Multispeed, variable-speed and single-speed compressor systems must all verify airflow rate (cfm/ton) by operating the system at maximum compressor capacity and maximum system fan speed with all zones calling for conditioning.
- Single-speed compressor systems must also verify airflow rate (cfm/ton) in every zonal control mode.

#### *PROPOSED DESIGN*

The default cooling airflow is 150 cfm/ton for a system with zonally controlled selected and multi-speed compressor not selected (single-speed). The default cooling airflow is 350 cfm/ton for all other ducted cooling systems. Users may model a higher-than-default airflow and receive credit in the compliance calculation if greater verified system airflow is diagnostically tested using the procedures of *Reference Residential Appendix RA3.3*.

#### *STANDARD DESIGN*

The standard design shall assume a system that complies with mandatory (Section 150.0) and prescriptive (Section 150.1) requirements for the applicable climate zone.

#### *VERIFICATION AND REPORTING*

The airflow rate verification compliance target (cfm or cfm/ton) is reported in the HERS required verification listings of the CF1R. When there is no cooling system it is reported on the CF1R as a special feature.

### 2.4.5.3 *Verified Air-Handling Unit Fan Efficacy*

The mandatory requirement in Section 150.0(m)13 is for an air-handling unit fan efficacy less than or equal to 0.58 Watts/cfm as verified by a HERS rater. Users may model a lower fan efficacy (W/cfm) and receive credit in the compliance calculation if the proposed fan efficacy value is diagnostically tested using the procedures in *Reference Residential Appendix RA3.3*.

For Single Zone Systems:

- Installers may elect to use an alternative to HERS verification of 0.58 Watts/cfm: HERS verification of a return duct design that conforms to the specification given in Table 150.0-B or C.

- The return duct design alternative is not an input to the compliance software, but must be documented on the Certificate of Installation.
- If a value less than 0.58 Watts/cfm is modeled by the software user for compliance credit, the alternative return duct design method using Table 150.0-B or C is not allowed for use in demonstrating compliance.
- Multispeed or variable-speed compressor systems must verify fan efficacy (Watt/cfm) for system operation at the maximum compressor speed and the maximum air handler fan speed.

For Zonally Controlled Systems:

- The Table 150.0-B or C return duct design alternative is not allowed for zonally controlled systems
- Multispeed, variable-speed and single-speed compressor systems must all verify fan efficacy (Watt/cfm) by operating the system at maximum compressor capacity and maximum system fan speed with all zones calling for conditioning.
- Single-speed compressor systems must also verify fan efficacy in every zonal control mode.

#### PROPOSED DESIGN

The software shall allow the user to enter the fan efficacy. The default mandatory value is 0.58 W/cfm. Users may, however, specify a lower value and receive credit in the compliance calculation if verified and diagnostically tested using the procedures of *Reference Appendices, Residential Appendix RA3.3*.

If no cooling system is installed a default value of 0.58 W/cfm is assumed.

#### STANDARD DESIGN

The standard design shall assume a verified fan efficacy complying with the mandatory requirement for less than or equal to 0.58 Watts/cfm.

#### VERIFICATION AND REPORTING

For user inputs lower than the default mandatory 0.58 Watts/cfm, fan efficacy is reported in the HERS required verification listings of the CF1R.

For default mandatory 0.58 Watts/cfm, the choice of either fan efficacy or alternative return duct design according to Table 150.0-B or C is reported in the HERS required verification listings of the CF1R.

When there is no cooling system it is reported on the CF1R as a special feature.

### 2.4.5.4 *Verified Energy Efficiency Ratio (EER)*

#### *PROPOSED DESIGN*

Software shall allow the user the option to enter an EER rating for central cooling equipment. For equipment that is rated only with an EER (room air conditioners), the user will enter the EER. The *Appliance Efficiency Regulations* require a minimum SEER and EER for central cooling equipment. Only if a value higher than a default minimum EER is used is it reported as a HERS verified measure.

#### *STANDARD DESIGN*

The standard design for central air-conditioning equipment is 11.7 EER.

#### *VERIFICATION AND REPORTING*

EER verification is only required if higher than 11.7 EER is modeled. The EER rating is verified using rating data from AHRI Directory of Certified Product Performance at www.ahridirectory.org or another directory of certified product performance ratings approved by the Energy Commission for determining compliance. Verified EER is reported in the HERS required verification listings on the CF1R.

### 2.4.5.5 *Verified Seasonal Energy Efficiency Ratio (SEER)*

#### *PROPOSED DESIGN*

The software allows the user to specify the SEER value.

#### *STANDARD DESIGN*

The standard design is based on the default minimum efficiency SEER for the type of cooling equipment modeled in the proposed design, based on the applicable *Appliance Efficiency Regulations*. For central-cooling equipment, the minimum efficiency is 14 SEER and 11.7 EER.

#### *VERIFICATION AND REPORTING*

If a SEER higher than the default minimum efficiency is modeled in software, the SEER requires field verification. The higher than minimum SEER rating is verified using rating data from AHRI Directory of Certified Product Performance at www.ahridirectory.org or another directory of certified product performance ratings approved by the Commission for determining compliance. Verified SEER is reported in the HERS required verification listings on the CF1R.

### 2.4.5.6 *Verified Evaporatively-Cooled Condensers*

#### *PROPOSED DESIGN*

Software shall allow users to specify an evaporatively-cooled condensing unit. The installation must comply with the requirements of RA4.3.2 to ensure the predicted energy savings are achieved. This credit must be combined with verified refrigerant charge testing, EER, and duct leakage testing.

#### STANDARD DESIGN

The standard design is based on a split system air conditioner meeting the requirements of Section 150.1(c) and Table 150.1-A.

#### VERIFICATION AND REPORTING

An evaporatively-cooled condensing unit, verified EER, and duct leakage testing are reported in the HERS required verification listings on the CF1R.

### 2.4.5.7 *Evaporative Cooling*

Evaporative cooling technology is best suited for dry climates where direct and/or indirect cooling of the supply air stream can occur without compromising indoor comfort. Direct evaporative coolers are the most common system type currently available but provide less comfort and deliver more moisture to the indoor space. They are assumed to be equivalent to a minimum split-system air conditioner. The evaporative cooling modeling methodology addresses two performance issues. The first performance issue is the increase in indoor relative humidity levels during periods with extended cooler operation. Since modeling of indoor air moisture levels is beyond the capability of simulation models, a simplified algorithm is used to prohibit evaporative cooler operation during load hours when operation is expected to contribute to uncomfortable indoor conditions. The algorithm disallows cooler operation when outdoor wet bulb temperatures are 70°F, or above. The second performance issue relates to evaporative cooler capacity limitations. Since evaporative coolers are 100 percent outdoor air systems, their capacity is limited by the outdoor wet bulb temperature. Each hour with calculated cooling load, the algorithm will verify that the cooling capacity is greater than the calculated cooling load.

#### PROPOSED DESIGN

Software shall allow users to specify one of three types of evaporative cooling: (1) direct evaporative cooler is the most commonly available system type, (2) indirect, or (3) indirect-direct. Product specifications and other modeling details are found in the Energy Commission appliance directory for evaporative cooling. Direct system types are assigned an efficiency of 14 SEER (or minimum appliance efficiency standard for split system cooling). The default system type is evaporative direct. For indirect or indirect-direct, select the appropriate type, from the Energy Commission appliance directory and input a 13 EER as well as the air flow and media saturation effectiveness or cooling effectiveness from the Energy Commission appliance directory.

#### STANDARD DESIGN

The standard design is based on a split-system air conditioner meeting the requirements of Section 150.1(c) and Table 150.1-A.

#### VERIFICATION AND REPORTING

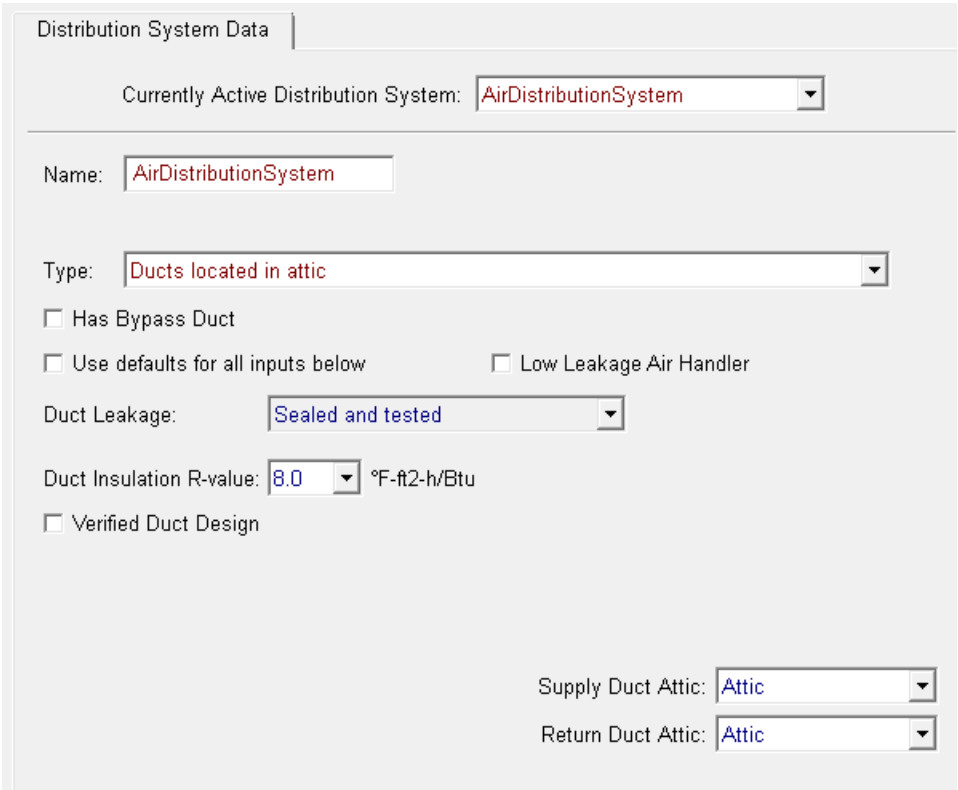
When a direct evaporative cooling system is modeled, the system type and minimum efficiency are shown in the appropriate section of the CF1R. When indirect or indirect-direct evaporative cooling is modeled, the EER verification is shown in the HERS verification section of the CF1R along with the system type, air-flow and system effectiveness.

## 2.4.6 Distribution Subsystems

If multiple HVAC distribution systems serve a building, each system and the conditioned space it serves may be modeled in detail separately or the systems may be aggregated together and modeled as one large system. If the systems are aggregated together they must be the same type and all meet the same minimum specifications.

For the purposes of duct efficiency calculations, the supply duct begins at the exit from the furnace or air handler cabinet.

Figure 5: Distribution System Data




### 2.4.6.1 Distribution Type

Fan-powered, ducted distribution systems that can be used with most heating or cooling systems. When ducted systems are used with furnaces, boilers, or combined hydronic/water heating systems, the electricity used by the fan is calculated. R-value and duct location are specified when a ducted system is specified.

#### PROPOSED DESIGN

The compliance software shall allow the user to select from the basic types of HVAC distribution systems and locations listed in Table 10. For ducted systems, the default location of the HVAC ducts and the air handler are in conditioned space for multifamily buildings and in the attic for all other buildings.

The software will allow users to select default assumptions or specify any of the verified or diagnostically tested HVAC distribution system conditions in the proposed design (see Figure 5 and Table 11), including duct leakage target, R-value, supply and return duct area, diameter and location.

**Table 10: HVAC Distribution Type and Location Descriptors**

| Name | HVAC Distribution Type and Location Description |
|---|---|
| Ducts located in attic (Ventilated and Unventilated) | Ducts located overhead in the attic space. |
| Ducts located in a crawl space | Ducts located under floor in the crawl space. |
| Ducts located in a garage | Ducts located in an unconditioned garage space. |
| Ducts located within the conditioned space (except < 12 linear ft) | Ducts located within the conditioned floor space except for less than 12 linear feet of duct, furnace cabinet, and plenums - typically an HVAC unit in the garage mounted on return box with all other ducts in conditioned space. |
| Ducts located entirely in conditioned space | HVAC unit or systems with all HVAC ducts (supply and return) located within the conditioned floor space. Location of ducts in conditioned space eliminates conduction losses, but does not change losses due to leakage. Leakage from either ducts that are not tested for leakage or from sealed ducts is modeled as leakage to outside the conditioned space. |
| Distribution system without ducts (none) | Air distribution systems without ducts such as ductless split system air conditioners and heat pumps, window air conditioners, through-the-wall heat pumps, wall furnaces, floor furnaces, radiant electric panels, combined hydronic heating equipment, electric baseboards, or hydronic baseboard finned-tube natural convection systems, etc. |
| Ducts located in outdoor locations | Ducts located in exposed locations outdoors. |
| Verified low leakage ducts located entirely in conditioned space | Duct systems for which air leakage to outside is equal to or less than 25 cfm when measured in accordance with Reference Residential Appendix RA3.1.4.3.8. |
| Ducts located in multiple places | Ducts with different supply and return duct locations. |

**Table 11: Summary of Verified Air-Distribution Systems**

| Measure | Description | Procedures |
|---|---|---|
| Verified Duct Sealing | Mandatory measures require that space-conditioning ducts be sealed. Field verification and diagnostic testing is required to verify that approved duct system materials are utilized and that duct leakage meets the specified criteria. | RA3.1.4.3 |
| Verified Duct Location, Reduced Surface Area and R-value | Compliance credit can be taken for improved supply duct location, reduced surface area, and R-value. Field verification is required to verify that the duct system was installed according to the duct design, including location, size and length of ducts, duct insulation R-value, and installation of buried ducts.[1] For buried duct measures Verified Insulation Construction Quality (QII) is required as well as duct sealing. | RA3.1.4.1, 3.1.4.1.1 |
| Low Leakage Ducts in Conditioned Space | When the standards specify use of the procedures in Section RA3.1.4.3.8 to determine if space conditioning system ducts are located entirely in directly conditioned space, the duct system location is verified by diagnostic testing. Compliance credit can be taken for verified duct systems with low air leakage to the outside when measured in accordance with Reference Appendices, Residential Appendix Section RA3.1.4.3.8. Field verification for ducts in conditioned space is required. Duct sealing is required. | RA3.1.4.3.8 |
| Low Leakage Air-Handling Units | Compliance credit can be taken for installation of a factory sealed air-handling unit tested by the manufacturer and certified to the Commission to have met the requirements for a Low Leakage Air-Handling Unit. Field verification of the air handler's model number is required. Duct sealing is required. | RA3.1.4.3.9 |
| Verified Return Duct Design | Verification to confirm that the return duct design conforms to the criteria given in Table 150.0-B or Table 150.0-C. as an alternative to meeting 0.58 W/cfm fan efficacy of Section 150.0(m)0. | RA3.1.4.4 |
| Verified Bypass Duct Condition | Verification to determine if system is zonally controlled and confirm that bypass ducts condition modeled matches installation. | RA3.1.4.6 |

1. Compliance credit for increased duct insulation R-value (not buried ducts) may be taken without field verification if the R-value is the same throughout the building, and for supply ducts located in crawl spaces and garages where all supply registers are either in the floor or within 2 feet of the floor. If these conditions are met, HERS rater verification is not required.

#### STANDARD DESIGN

The standard heating and cooling system for central systems is modeled with non-designed air distribution ducts located as described in Table 12, with duct leakage as specified in Table 15. The standard design duct insulation is determined by Package A (assuming attic option B) as R-6 in climate zones 3 and 5 through 7, and R-8 in climate zones 1, 2, 4, and 8 through16. The standard design building is assumed to have the same number of stories as the proposed design for determining the duct efficiency.

**Table 12: Summary of Standard Design Duct Location**

| Configuration of the Proposed Design | Standard Design | |
|---|---|---|
| | **Standard Design Duct Location** | **Detailed Specifications** |
| Attic over the dwelling unit | Ducts and air handler located in the attic | Ducts sealed (mandatory requirement)<br>No credit for verified R-value, location or duct design |
| No attic but crawl space or basement | Ducts and air handler located in the crawl space or basement | |
| Multi-family buildings and buildings with no attic, crawl space or basement | Ducts and air handler located indoors | |

*This table is applicable only when the standard design system has air-distribution ducts as determined in Table 10.*

#### VERIFICATION AND REPORTING

Distribution type, location, R-value, and whether tested and sealed will be shown on the CF1R. If there are no ducts, this is shown as a special feature on the CF1R. Any duct location other than attic (for example: crawl space) is shown as a special feature on the CF1R. Ducts in crawl space or basement shall include a special feature note if supply registers are located within 2 feet of the floor. Measures that require HERS verification will be shown in the HERS required verification section of the CF1R.

### 2.4.6.2 *Duct Location*

Duct location determines the external temperature for duct conduction losses, the temperature for return leaks, and the thermal regain of duct losses.

#### PROPOSED DESIGN

If any part of the supply or return duct system is located in an unconditioned attic, that entire duct system is modeled with an attic location. If no part of the supply or return duct system is located in the attic, but the duct system is not entirely in conditioned space, it is modeled in the unconditioned zone, which contains the largest fraction of its surface area. If the supply or return duct system is located entirely in conditioned space, the duct system is modeled in conditioned space.

For ducted HVAC systems with some or all ducts in unconditioned space, the user specifies the R-value and surface area of supply and return ducts, and the duct location.

Duct location and areas other than the defaults shown in Table 13 may be used following the verification procedures in *Reference Residential Appendix RA3.1.4.1*.

#### STANDARD DESIGN

The standard design duct location is determined from the building conditions (see Table 12).

#### VERIFICATION AND REPORTING

Duct location is reported on the CF1R. Ducts located entirely in conditioned space and verified low leakage ducts entirely in conditioned space are reported in the HERS required verification listing on the CF1R.

Default duct locations are as shown in Table 13. The duct surface area for crawl space and basement applies only to buildings or zones with all ducts installed in the crawl space or basement. If the duct is installed in locations other than crawl space or basement, the default duct location is "Other." For houses with two or more stories, 35 percent of the default duct area may be assumed to be in conditioned space as shown in Table 13.

The surface area of ducts located in conditioned space is ignored in calculating conduction losses.

**Table 13: Location of Default Duct Area**

| Supply Duct Location | Location of Default Duct Surface Area | |
|---|---|---|
| | One story | Two or more stories |
| All in crawl space | 100% crawl space | 65% crawl space, 35% conditioned space |
| All in basement | 100% basement | 65% basement, 35% conditioned space |
| Other | 100% attic | 65% attic, 35% conditioned space |

### 2.4.6.3 *Duct Surface Area*

The supply-side and return-side duct surface areas are treated separately in distribution efficiency calculations. The duct surface area is determined using the following methods.

### 2.4.6.4 *Default Return Duct Surface Area*

Default return duct surface area is calculated using:

$$A_{r,out} = K_r \times A_{floor} \qquad \textit{Equation 8}$$

Where $K_r$ (return duct surface area coefficient) is 0.05 for one-story buildings and 0.1 for two or more stories.

### 2.4.6.5 *Default Supply Duct Surface Area*

***STANDARD DESIGN***

The standard design and default proposed design supply duct surface area is calculated using Equation 9.

$$A_{s,\,out} = 0.27 \times A_{floor} \times K_S \qquad \textit{Equation 9}$$

Where $K_s$ (supply duct surface area coefficient) is 1 for one-story buildings and 0.65 for two or more stories.

### 2.4.6.6 *Supply Duct Surface Area for Less Than 12 feet of Duct In Unconditioned Space*

***PROPOSED DESIGN***

For proposed design HVAC systems with air handlers located outside the conditioned space but with less than 12 linear feet of duct located outside the conditioned space including air handler and plenum, the supply duct surface area outside the conditioned space is calculated using Equation 10. The return duct area remains the default for this case.

$$A_{s,out} = 0.027 \times A_{floor} \qquad \textit{Equation 10}$$

### 2.4.6.7 *Diagnostic Duct Surface Area*

Proposed designs may claim credit for reduced surface area using the procedures in *Reference Residential Appendix RA3.1.4.1*.

The surface area of each duct system segment shall be calculated based on its inside dimensions and length. The total supply surface area in each unconditioned location (attic, attic with radiant barrier, crawl space, basement, other) is the sum of the area of all duct segments in that location. The surface area of ducts completely inside conditioned space need not be input in the compliance software and is not included in the calculation of duct system efficiency. The area of ducts in floor cavities or vertical chases that are surrounded by conditioned space and separated from unconditioned space with draft stops are also not included.

### 2.4.6.8 *Bypass Duct*

Section 150.1(c)13 prohibits use of bypass ducts unless a bypass duct is otherwise specified on the certificate of compliance. A bypass duct may be needed for some single-speed outdoor condensing unit systems. The software allows users to specify a bypass duct for the system. Selection of a bypass duct does not trigger changes in the ACM modeling defaults, but verification by a HERS rater is required utilizing the procedure in *Reference Residential Appendix Section RA3.1.4.6*.

Note: specification of a single-speed condensing unit for the system will trigger a default airflow rate value of 150 cfm/ton for the calculations which reduces the compliance margin as compared to

systems that model 350 cfm/ton. Users may model airflow rates greater than 150 cfm/ton and receive credit in the calculations as described in Section 2.4.5.2.

#### PROPOSED DESIGN

Software shall allow users to specify whether a bypass duct is or is not used for a zonally controlled forced air system.

#### STANDARD DESIGN

The standard design is based on a split-system air conditioner meeting the requirements of Section 150.1(c) and Table 150.1-A. The system is not a zonally controlled system.

#### VERIFICATION AND REPORTING

An HVAC system with zonal control, and whether the system is assumed to have a bypass duct or have no bypass duct, is reported in the HERS required verification listings on the CF1R.

### 2.4.6.9 *Duct System Insulation*

For the purposes of conduction calculations in both the standard and proposed designs, 85 percent of the supply and return duct surface is assumed to be duct material at its specified R-value and 15 percent is assumed to be air handler, plenum, connectors, and other components at the mandatory minimum R-value.

The area weighted effective R-value is calculated by the compliance software using Equation 11 and including each segment of the duct system that has a different R-value.

$$R_{eff} = \frac{(A_1 + A_2 .... + A_N)}{\left[\frac{A_1}{R_1} + \frac{A_2}{R_2} .... + \frac{A_N}{R_N}\right]} \qquad \textit{Equation 11}$$

Where:

$R_{eff}$ = Area weighted effective R-value of duct system for use in calculating duct efficiency, (h-ft²-°F/Btu)

$A_N$ = Area of duct segment n, square feet

$R_n$ = R-value of duct segment n including film resistance (duct insulation rated R + 0.7) (h-ft²-°F/Btu)

#### PROPOSED DESIGN

The software user inputs the R-value of the proposed duct insulation and details. The default duct thermal resistance is based on Table 150.1-A, Attic Option B, which is R-6 in climate zones 3 and 5 through 7, R-8 in zones 1, 2, 4, and 8 through 16.

Duct location and duct R-value are reported on the CF1R. Credit for systems with mixed insulation levels, non-standard supply and return duct surface areas, or ducts buried in the attic require the compliance and diagnostic procedures in *Reference Residential Appendix RA3.1.4.1*.

If a verified duct design is selected, non-standard values for the supply-duct surface area and the return-duct surface area may be input by the user. A verified duct design must be verified by a HERS rater according to the procedures in *Reference Residential Appendix RA3.1.4.1.1*. Supply and return duct R-values, location, and areas are reported on the CF1R when non-standard values are specified.

#### STANDARD DESIGN

Package A required duct insulation R-values for the attic option B, for applicable climate zone are used in the standard design.

#### VERIFICATION AND REPORTING

Duct location, duct R-value, supply, and return duct areas are reported on the CF1R.

### 2.4.6.10 *Buried Attic Ducts*

Ducts partly or completely buried in blown attic insulation in dwelling units meeting the requirements for verified quality insulation installation may take credit for increased effective duct insulation.

The duct design shall identify the segments of the duct that meet the requirements for being buried, and these are input separately into the computer software. Ducts to be buried shall have a minimum of R-6.0 duct insulation prior to being buried. The user shall calculate the correct R-value based on the modeled attic insulation R-value, insulation type, and duct size for ducts installed on the ceiling, and whether the installation meets the requirements for deeply buried ducts for duct segments buried in lowered areas of ceiling.

The portion of duct runs directly on or within 3.5 inches of the ceiling gypsum board and surrounded with blown attic insulation of R-30 or greater may take credit for increased effective duct insulation as shown in Table 14. Credit is allowed for buried ducts on the ceiling only in areas where the ceiling is level and there is at least 6 inches of space between the outer jacket of the installed duct and the roof sheathing above.

Duct segments deeply buried in lowered areas of ceiling and covered by at least 3.5 inches of insulation above the top of the duct insulation jacket may claim effective insulation of R-25 for fiberglass insulation and R-31 for cellulose insulation.

#### PROPOSED DESIGN

The software shall allow the user to specify the effective R-value of buried ducts. This feature must be combined with duct sealing and verified quality insulation installation. The default is no buried ducts.

#### STANDARD DESIGN

The standard design has no buried ducts.

#### VERIFICATION AND REPORTING

Buried duct credit is reported in the HERS required verification listing on the CF1R.

**Table 14: Buried Duct Effective R-values**

| | Nominal Round Duct Diameter | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|
| Attic Insulation | 4'' | 5'' | 6'' | 7'' | 8'' | 10'' | 12'' | 14'' | 16'' |
| | Effective Duct Insulation R-value for Blown Fiberglass Insulation | | | | | | | | |
| R-30 | R-13 | R-13 | R-13 | R-9 | R-9 | R-4.2 | R-4.2 | R-4.2 | R-4.2 |
| R-38 | R-25 | R-25 | R-25 | R-13 | R-13 | R-9 | R-9 | R-4.2 | R-4.2 |
| R-40 | R-25 | R-25 | R-25 | R-25 | R-13 | R-13 | R-9 | R-9 | R-4.2 |
| R-43 | R-25 | R-25 | R-25 | R-25 | R-25 | R-13 | R-9 | R-9 | R-4.2 |
| R-49 | R-25 | R-25 | R-25 | R-25 | R-25 | R-25 | R-13 | R-13 | R-9 |
| R-60 | R-25 | R-25 | R-25 | R-25 | R-25 | R-25 | R-25 | R-25 | R-13 |
| | Effective Duct Insulation R-value for Blown Cellulose Insulation | | | | | | | | |
| R-30 | R-9 | R-4.2 | R-4.2 | R-4.2 | R-4.2 | R-4.2 | R-4.2 | R-4.2 | R-4.2 |
| R-38 | R-15 | R-15 | R-9 | R-9 | R-4.2 | R-4.2 | R-4.2 | R-4.2 | R-4.2 |
| R-40 | R-15 | R-15 | R-15 | R-9 | R-9 | R-4.2 | R-4.2 | R-4.2 | R-4.2 |
| R-43 | R-15 | R-15 | R-15 | R-15 | R-9 | R-4.2 | R-4.2 | R-4.2 | R-4.2 |
| R-49 | R-31 | R-31 | R-15 | R-15 | R-15 | R-9 | R-9 | R-4.2 | R-4.2 |
| R-60 | R-31 | R-31 | R-31 | R-31 | R-31 | R-15 | R-15 | R-9 | R-9 |

### 2.4.6.11 *Duct/Air Handler Leakage*

The total duct/air handler leakage shown in Table 15 is used in simulating the duct system. The supply duct leakage for each case is the table value times 0.585. The return leakage is the table value times 0.415.

#### PROPOSED DESIGN

For each ducted system, the software user specifies one of the duct/air handler leakage cases shown in Table 15.

#### STANDARD DESIGN

For ducted systems, the standard design is sealed and tested duct systems in existing dwelling units or new duct systems.

#### VERIFICATION AND REPORTING

Sealed and tested duct systems are listed in the HERS verification section of the CF1R. Duct leakage is measured in accordance with procedures and values specified in *Reference Appendices, Residential Appendix RA3.*

### 2.4.6.12 *Low Leakage Air Handlers*

A low leakage air handler may be specified as well as a lower duct leakage value (see Section 2.4.6.11). Installation requires installing one of the list of approved low leakage air handling units published by the Energy Commission. The manufacturer certifies that the appliance complies with the requirements of Reference Joint Appendices 9.2.1, 9.2.2, 9.2.3, and 9.2.4.

**Table 15: Duct/Air Handler Leakage**

| Case | Duct Leakage | Air Handler Leakage | Total Duct/Air Handler Leakage |
|---|---|---|---|
| Duct systems in existing single-family dwelling units | 15% | Included in duct leakage | 15% |
| Sealed and tested new or altered duct systems in unconditioned or conditioned space in a multi-family dwelling unit | 12% | Included in duct leakage | 12% |
| Sealed and tested new or altered duct systems in unconditioned or conditioned space in a townhome or single family dwelling unit | 5% | 2% | 7% |
| Verified low leakage ducts in conditioned space | 0% | 0% | 0% |
| Low leakage air handlers in combination with sealed and tested new duct systems | 5% or as measured | 0% | 5% or as measured |

#### *PROPOSED DESIGN*

Credit can be taken for installation of a factory sealed air-handling unit tested by the manufacturer and certified to the Energy Commission to meet the requirements for a low leakage air-handler. Field verification of the air handler model number is required.

#### *STANDARD DESIGN*

The standard design has a normal air handler.

#### *VERIFICATION AND REPORTING*

A low leakage air handler is reported on the compliance report and field verified in accordance with the procedures specified in *Reference Appendices, Residential Appendix RA3.1.4.3.9*.

#### 2.4.6.13 *Verified Low-Leakage Ducts in Conditioned Space*

***PROPOSED DESIGN***

For ducted systems the user may specify that all ducts are entirely in conditioned space and the software will model the duct system with no leakage and no conduction losses.

***STANDARD DESIGN***

The standard design has ducts in the default location.

***VERIFICATION AND REPORTING***

Systems that have all ducts entirely in conditioned space are reported on the compliance documents and this is verified by measurements showing duct leakage to outside conditions is equal to or less than 25 cfm when measured in accordance with *Reference Appendices, Residential Appendix RA3*.

## *2.4.7 Space Conditioning Fan Subsystems*

Fan systems move air for air conditioning, heating, and ventilation systems. The software allows the user to define fans to be used for space conditioning, indoor air quality, and ventilation cooling. Indoor air quality and ventilation cooling are discussed in Sections 2.4.9 and 2.4.10.

***PROPOSED DESIGN***

For the space conditioning fan system, the user selects the type of equipment and enters basic information to model the energy use of the equipment. For ducted central air conditioning and heating systems, the fan efficacy default is the mandatory minimum verified efficacy of 0.58 W/cfm (also assumed when there is no cooling system).

***STANDARD DESIGN***

The standard design fan shall meet the minimum Section 150.1(c) and Table 150.1-A requirements.

***VERIFICATION AND REPORTING***

Minimum verified fan efficacy is a mandatory requirement for all ducted cooling systems. Fan efficacy is reported in the HERS required verification listings on the CF1R.

## *2.4.8 Space Conditioning Systems*

This section describes the general procedures for heating and cooling systems in low-rise residential buildings. The system includes the cooling system, the heating system, distribution system, and mechanical fans.

If multiple systems serve a building, each system and the conditioned space it serves may be modeled in detail separately or the systems may be aggregated together and modeled as one large system. If the systems are aggregated together they must be the same type and all meet the same minimum specifications.

### 2.4.8.1 *Multiple System Types Within Dwelling*

#### *PROPOSED DESIGN*

For proposed designs using more than one heating system type, equipment type, or fuel type, and the types do not serve the same floor area, the user shall zone the building by system type.

#### *STANDARD DESIGN*

The standard design shall have the same zoning and heating system types as the proposed design.

#### *VERIFICATION AND REPORTING*

The heating system type of each zone is shown on the CF1R.

### 2.4.8.2 *Multiple Systems Serving Same Area*

If a space or a zone is served by more than one heating system, compliance is demonstrated with the most time-dependent valuation (TDV) energy-consuming system serving the space or the zone. For spaces or zones that are served by electric resistance heat in addition to other heating systems, the electric resistance heat is deemed to be the most TDV energy-consuming system unless the supplemental heating meets the Exception to Section 150.1(c)6. See eligibility criteria in *Residential Compliance Manual* Section 4.2.2 for conditions under which the supplemental heat may be ignored.

For floor areas served by more than one cooling system, equipment, or fuel type, the system, equipment, and fuel type that satisfies the cooling load is modeled.

### 2.4.8.3 *No Cooling*

#### *PROPOSED DESIGN*

When the proposed design has no cooling system, the proposed design is required to model the standard design cooling system defined in Section 150.1(c) and Table 150.1-A. Since the proposed design system is identical to the standard design system, there is no penalty or credit.

#### *STANDARD DESIGN*

The standard design system is the specified in Section 150.1(c) and Table 150.1-A for the applicable climate zone.

#### *VERIFICATION AND REPORTING*

No cooling is reported as a special feature on the CF1R.

### 2.4.8.4 *Zonally Controlled Forced-Air Cooling Systems*

Zonally controlled central forced-air cooling systems must be able to deliver, in every zonal control mode, an airflow to the dwelling of $\geq$ 350 CFM per ton of nominal cooling capacity, and operating at an air-handling unit fan efficacy of $\leq$ 0.58 W/CFM. This is a HERS verified measure, complying with *Residential Appendix RA3.3*.

An exception allows multispeed or variable-speed compressor systems, or single-speed compressor systems to meet the mandatory airflow (cfm/ton) and fan efficacy (Watt/cfm) requirements by operating the system at maximum compressor capacity and system fan speed with all zones calling for conditioning, rather than in every zonal control mode.

#### PROPOSED DESIGN

The user selects zonally controlled as a cooling system input.

#### STANDARD DESIGN

The standard design building does not have a zonally controlled cooling system.

#### VERIFICATION AND REPORTING

Zonally controlled forced air cooling systems are required to have the system bypass duct status verified by a HERS rater according to the procedures in *Reference Residential Appendix RA3.1.4.6*, and the fan efficacy and airflow rate are required to be verified according to the procedures in *RA3.3*.

### 2.4.9 Indoor Air Quality Ventilation

The standards require mechanical ventilation that complies with ASHRAE Standard 62.2 to provide acceptable indoor air quality for all newly constructed buildings and additions greater than 1,000 square feet. ASHRAE Standard 62.2 provides several ways to comply with the requirement for mechanical ventilation and these are described in the *Residential Compliance Manual*.

For the purposes of estimating the energy impact of this requirement in compliance software, the minimum ventilation rate is met either by a standalone indoor air quality (IAQ) fan system or a central air handler fan system that can introduce outdoor air. In many cases, this energy is substantially compliance neutral because the standard design is typically set equal to the proposed design.

The simplest IAQ fan system is an exhaust fan/bathroom fan that meets the criteria in ASHRAE Standard 62.2 for air delivery and low noise. More advanced IAQ fan systems that have a supply or both supply and exhaust fans are also possible. To calculate the energy use of standalone IAQ fan systems, the systems are assumed to be on continuously.

To calculate the energy use of central fan integrated ventilation, the systems are assumed to be on for at least 20 minutes each hour as described below. The fan flow rate and fan power ratio may be different than the values used when the system is on to provide for heating or cooling depending on the design or controls on the IAQ ventilation portion of the system.

The minimum ventilation rate for continuous ventilation of each single-family dwelling unit is given in Equation 12.

$$Q_{fan} = 0.01A_{floor} + 7.5(N_{br} + 1)$$ *Equation 12*

Where:

$Q_{fan}$ = fan flow rate in cubic feet per minute (cfm)

$A_{floor}$ = floor area in square feet ($ft^2$)

$N_{br}$ = number of bedrooms (not less than one)

The minimum ventilation rate for continuous ventilation of each multi-family dwelling unit is given in Equation 13.

$$Q_{fan} = 0.03A_{floor} + 7.5(N_{br} + 1)$$ *Equation 13*

Where:

$Q_{fan}$ = fan flow rate in cubic feet per minute (cfm)

$A_{floor}$ = floor area in square feet ($ft^2$)

$N_{br}$ = number of bedrooms (not less than one)

#### PROPOSED DESIGN

The proposed design shall incorporate a mechanical ventilation system fan. This requirement is a mandatory measure. The compliance software allows the user to specify the IAQ ventilation type (see Table 16) and the cfm of outdoor ventilation air which must be equal to or greater than what is required by ASHRAE Standard 62.2. The default is a standalone exhaust system meeting standard 62.2.

#### STANDARD DESIGN

The mechanical ventilation system in the standard design is the same as the proposed design. The air flow rate is equal to the proposed design. The sensible heat recovery effectiveness is 0. For standalone IAQ fan systems, the fan power ratio is equal to the proposed design value or 1.2 W/cfm, whichever is smaller. For central air handler fans, the fan power ratio is 0.58 W/cfm of central system airflow in ventilation mode.

#### VERIFICATION AND REPORTING

The required ventilation rate to comply with ASHRAE Standard 62.2 and the means to achieve compliance are indicated on the CF1R (see Table 17). The IAQ system characteristics are reported in the HERS required verification listing on the CF1R. The diagnostic testing procedures are in RA3.7.

**Table 16: IAQ Fans**

| Type | Description | Inputs |
|---|---|---|
| Standalone IAQ Fan<br>(exhaust, supply, or balanced) | Dedicated fan system that provides indoor air quality ventilation to meet or exceed the requirements of ASHRAE Standard 62.2. | cfm, Watts/cfm, recovery effectiveness for balanced only |
| Central Fan Integrated (CFI)<br>(variable or fixed speed) | Automatic operation of the normal furnace fan for IAQ ventilation purposes. Ventilation type uses a special damper to induce outdoor IAQ ventilation air and distribute it through the HVAC duct system. Mixing type distributes and mixes IAQ ventilation air supplied by a separate standalone IAQ fan system. | cfm, Watts/cfm |

**Table 17: CF1R Report – Indoor Air Quality**

| IAQ System Name | IAQ System Type | Whole Building IAQ Airflow Rate (cfm) | Standalone IAQ Fan Power Ratio (W/cfm) |
|---|---|---|---|
| SFam IAQVentRpt | Default | 28.5 | 0.25 |

## 2.4.10 Ventilation Cooling System

Ventilation cooling systems operate at the dwelling-unit level using fans to bring in outside air to cool the house when this can reduce cooling loads and save cooling energy. Ventilation cooling systems such as whole house fans involve window operation and attic venting. Central fan ventilation cooling systems use the HVAC duct system to distribute ventilation air. Ventilation cooling systems operate according to the schedule and setpoints shown in Table 19. Ventilation cooling systems that exhaust air through the attic require a minimum of the larger of 1 $ft^2$ of free attic ventilation area per 750 cfm of rated capacity for relief or the manufacturer specifications (see Section 150.1(c)12 of the standards).

#### *PROPOSED DESIGN*

Software allows the user to specify whether a ventilation cooling system (see Table 18) is included in conditioned and living zones. The user can specify the actual fan specifications or a default prescriptive whole house fan with a capacity of 1.5 CFM/$ft^2$ of conditioned floor area when there is a ventilated attic.

#### *STANDARD DESIGN*

The standard design building for a newly constructed building or for an addition greater than 1,000 square feet has a whole house fan in climate zones 8 through 14 and no ventilation cooling in other climate zones (see Section 150.1(c) and Table 150.1-A). The whole house fan has 1.5 CFM/$ft^2$ of conditioned floor area with 1 square foot of attic vent free area for each 750 CFM of rated whole house fan air-flow CFM.

#### VERIFICATION AND REPORTING

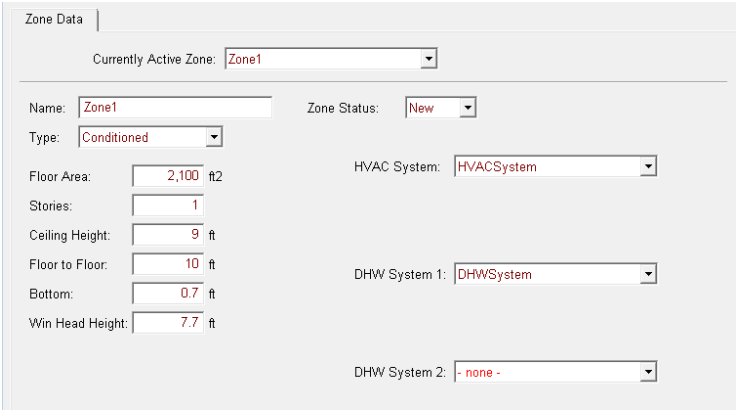
A ventilation cooling system is a special feature and the size and type is reported on the CF1R.

**Table 18: Ventilation Cooling Fans**

| Measure | Description |
|---|---|
| Whole House Fan | Traditional whole house fan mounted in the ceiling to exhaust air from the house to the attic, inducing outside air in through open windows. Whole house fans are assumed to operate between dawn and 11 p.m. only at 33 percent of rated cfm to reflect manual operation of fan and windows by occupant. Fans must be listed in the California Energy Commission's Whole House Fan directory. If multiple fans are used, enter the total cfm. |
| Central Fan Ventilation Cooling<br><br>Variable or fixed speed | Central fan ventilation cooling system. Ventilation type uses a special damper to induce outdoor air and distribute it through the HVAC duct system. |

## 2.5 Conditioned Zones

Figure 6: Zone Data




The software requires the user to enter the characteristics of one or more conditioned zones. Subdividing single-family dwelling units into conditioned zones for input convenience or increased accuracy is optional.

### 2.5.1 Zone Type

#### PROPOSED DESIGN

The zone is defined as conditioned, living, or sleeping. Other zone types include garage, attic, and crawl space.

#### STANDARD DESIGN

The standard design is conditioned.

#### VERIFICATION AND REPORTING

When the zone type is living or sleeping, this is reported as a special feature on the CF1R.

#### 2.5.1.1 *Heating Zonal Control Credit*

With the heating zonal control credit, the sleeping and living areas are modeled separately for heating, each with its own separate thermostat schedule and internal gain assumptions. Zonal control cannot be modeled with heat pump heating. The total non-closable opening area between zones cannot exceed 40 $ft^2$. Other eligibility criteria for this measure are presented in the *Residential Compliance Manual, Chapter 4*.

#### PROPOSED DESIGN

The user selects zonal control as a building level input with separate living and sleeping zones.

#### STANDARD DESIGN

The standard design building is not zoned for living and sleeping separately.

#### VERIFICATION AND REPORTING

Zonal control is reported as a special feature on the CF1R.

### *2.5.2 Conditioned Floor Area*

The total conditioned floor area (CFA) is the raised floor as well as the slab-on-grade floor area of the conditioned spaces measured from the exterior surface of exterior walls. Stairs are included in conditioned floor area as the area beneath the stairs and the tread of the stairs.

#### PROPOSED DESIGN

The compliance software requires the user to enter the total conditioned floor area of each conditioned zone.

#### STANDARD DESIGN

The standard design building has the same conditioned floor area and same conditioned zones as the proposed design.

#### VERIFICATION AND REPORTING

The conditioned floor area of each conditioned zone is reported on the CF1R.

## 2.5.3 Number of Stories

### 2.5.3.1 Number of Stories of the Zone

#### PROPOSED DESIGN

The number of stories of the zone.

#### STANDARD DESIGN

The standard design is the same as the proposed design.

### 2.5.3.2 Ceiling Height

#### PROPOSED DESIGN

The average ceiling height of the proposed design is the conditioned volume of the building envelope. The volume (in cubic feet) is determined from the total conditioned floor area and the average ceiling height.

#### STANDARD DESIGN

The volume of the standard design building is the same as the proposed design.

#### VERIFICATION AND REPORTING

The conditioned volume of each zone is reported on the CF1R.

### 2.5.3.3 Free Ventilation Area

Free ventilation area is the window area adjusted to account for bug screens, window framing and dividers, and other factors.

#### PROPOSED DESIGN

Free ventilation area for the proposed design is calculated as 5 percent of the fenestration area (rough opening), assuming all windows are operable.

#### STANDARD DESIGN

The standard design value for free ventilation area is the same as the proposed design.

#### VERIFICATION AND REPORTING

Free ventilation is not reported on the CF1R.

### 2.5.3.4 Ventilation Height Difference

Ventilation height difference is not a user input.

#### PROPOSED DESIGN

The default assumption for the proposed design is 2 feet for one-story buildings or one-story dwelling units, and 8 feet for two or more stories (as derived from number of stories and other zone details).

#### STANDARD DESIGN

The standard design modeling assumption for the elevation difference between the inlet and the outlet is two feet for one-story dwelling units and eight feet for two or more stories.

### 2.5.3.5 *Zone Elevations*

The elevation of the top and bottom of each zone is required to set up the air-flow network.

#### PROPOSED DESIGN

The user enters the height of the top surface the lowest floor of the zone relative to the ground outside as the "bottom" of the zone. The user also enters the ceiling height (the floor to floor height [ceiling height plus the thickness of the intermediate floor structure] is calculated by the software).

Underground zones are indicated with the number of feet below grade (for example, -8).

#### STANDARD DESIGN

The standard design has the same vertical zone dimensions as the proposed design.

### 2.5.3.6 *Mechanical Systems*

#### PROPOSED DESIGN

The software requires the user to specify a previously defined HVAC system to provide heating and cooling for the zone and an indoor air quality (IAQ) ventilation system. The user may also specify a ventilation cooling system that applies to this and other conditioned zones.

#### STANDARD DESIGN

The software assigns standard design HVAC, IAQ ventilation, and ventilation cooling systems based on Section 150.1(c) and Table 150.1-A for the applicable climate zone.

### 2.5.3.7 *Natural Ventilation*

Natural ventilation (from windows) is available during cooling mode when needed and available as shown in Table 19. The amount of natural ventilation used by computer software for natural cooling is the lesser of the maximum potential amount available and the amount needed to drive the interior zone temperature down to the natural cooling setpoint. When natural cooling is not needed or is unavailable no natural ventilation is used.

Computer software shall assume that natural cooling is needed when the building is in "cooling mode" and when the outside temperature is below the estimated zone temperature and the estimated zone temperature is above the natural cooling setpoint temperature. Only the amount of ventilation required to reduce the zone temperature down to the natural ventilation setpoint temperature is used and the natural ventilation setpoint temperature is constrained by the compliance software to be greater than the heating setpoint temperature.

**Table 19: Hourly Thermostat Setpoints**

| Hour | Cooling | Venting | Standard Heating | Zonal Control Heating | |
|---|---|---|---|---|---|
| | | | Single Zone | Living | Sleeping |
| 1 | 78 | Off | 65 | 65 | 65 |
| 2 | 78 | Off | 65 | 65 | 65 |
| 3 | 78 | Off | 65 | 65 | 65 |
| 4 | 78 | Off | 65 | 65 | 65 |
| 5 | 78 | Off | 65 | 65 | 65 |
| 6 | 78 | 68* | 65 | 65 | 65 |
| 7 | 78 | 68 | 65 | 65 | 65 |
| 8 | 83 | 68 | 68 | 68 | 68 |
| 9 | 83 | 68 | 68 | 68 | 68 |
| 10 | 83 | 68 | 68 | 68 | 65 |
| 11 | 83 | 68 | 68 | 68 | 65 |
| 12 | 83 | 68 | 68 | 68 | 65 |
| 13 | 83 | 68 | 68 | 68 | 65 |
| 14 | 82 | 68 | 68 | 68 | 65 |
| 15 | 81 | 68 | 68 | 68 | 65 |
| 16 | 80 | 68 | 68 | 68 | 65 |
| 17 | 79 | 68 | 68 | 68 | 68 |
| 18 | 78 | 68 | 68 | 68 | 68 |
| 19 | 78 | 68 | 68 | 68 | 68 |
| 20 | 78 | 68 | 68 | 68 | 68 |
| 21 | 78 | 68 | 68 | 68 | 68 |
| 22 | 78 | 68 | 68 | 68 | 68 |
| 23 | 78 | 68 | 68 | 68 | 68 |
| 24 | 78 | Off | 65 | 65 | 65 |

**Venting starts in the hour the sun comes up.*

## 2.5.4 Conditioned Zone Assumptions

### 2.5.4.1 Internal Thermal mass

Internal mass objects are completely inside a zone so that they do not participate directly in heat flows to other zones or outside. They are connected to the zone radiantly and convectively, and participate in the zone energy balance by passively storing and releasing heat as conditions change.

Table 20 shows the standard interior conditioned zone thermal mass objects and the calculation of the simulation inputs that represent them.

**Table 20: Conditioned Zone Thermal Mass Objects**

| Item | Description | Simulation Object |
|---|---|---|
| Interior walls | The area of one side of the walls completely inside the conditioned zone is calculated as the conditioned floor area of the zone minus ½ of the area of interior walls adjacent to other conditioned zones. The interior wall is modeled as a construction with 25 percent 2x4 wood framing and sheetrock on both sides. | Wall exposed to the zone on both sides |
| Interior floors | The area of floors completely inside the conditioned zone is calculated as the difference between the CFA of the zone and the sum of the areas of zone exterior floors and interior floors over other zones. Interior floors are modeled as a surface inside the zone with a construction of carpet, wood decking, 2x12 framing at 16 in. o.c. with miscellaneous bridging, electrical, and plumbing, and a sheetrock ceiling below. | Floor/ceiling surface exposed to the zone on both sides |
| Furniture and heavy contents | Contents of the conditioned zone with significant heat storage capacity and delayed thermal response, for example heavy furniture, bottled drinks and canned goods, contents of dressers and enclosed cabinets. These are represented by a 2 in. thick slab of wood twice as large as the conditioned floor area, exposed to the room on both sides. | Horizontal wood slab exposed to the zone on both sides |
| Light and thin contents | Contents of the conditioned zone that have a large surface area compared to their weight, for example, clothing on hangers, curtains, pots and pans. These are assumed to be 2 BTU per square foot of conditioned floor area. | Air heat capacity ($C_{air}$) = CFA * 2 |

#### *PROPOSED DESIGN*

The proposed design has standard conditioned zone thermal mass objects that are not user editable and are not a compliance variable. If the proposed design includes specific interior thermal mass elements that are significantly different from what is included in typical wood frame production housing, such as masonry partition walls, the user may include them. See also 2.5.6.4.

#### *STANDARD DESIGN*

The standard design has standard conditioned zone thermal mass objects.

### 2.5.4.2 *Thermostats and Schedules*

Thermostat settings are shown in Table 19. The values for cooling, venting, and standard heating apply to the standard design run and are the default for the proposed design run. See the explanation later in this section regarding the values for zonal control.

Systems with no setback required by Section 110.2(c) (gravity gas wall heaters, gravity floor heaters, gravity room heaters, non-central electric heaters, fireplaces or decorative gas appliances, wood stoves, room air conditioners, and room air-conditioner heat pumps) are assumed to have a constant heating set point of 68 degrees. The cooling set point from Table 19 is assumed in both the proposed design and standard design.

#### PROPOSED DESIGN

The proposed design assumes a mandatory setback thermostat meeting the requirements of Section 110.2(c). Systems exempt from the requirement for a setback thermostat are assumed to have no setback capabilities.

#### STANDARD DESIGN

The standard design has setback thermostat conditions based on the mandatory requirement for a setback thermostat. For equipment exempt from the setback thermostat requirement, the standard design has no setback thermostat capabilities.

#### VERIFICATION AND REPORTING

When the proposed equipment is exempt from setback thermostat requirements this is shown as a special feature on the CF1R.

### 2.5.4.3 *Determining Heating Mode vs. Cooling Mode*

When the building is in the heating mode, the heating setpoints for each hour are set to the "heating" values in Table 19, the cooling setpoint is a constant 78°F and the ventilation setpoint is set to a constant 77°F. When the building is in the cooling mode the heating setpoint is a constant 60°F, and the cooling and venting setpoints are set to the values in Table 19.

The mode is dependent upon the outdoor temperature averaged over hours 1 through 24 of eight days prior to the current day through two days prior to the current day (for example: if the current day is June 21, the mode is based on the average temperature for June 13 through 20). When this running average temperature is equal to or less than 60°F, the building is in a heating mode. When the running average is greater than 60°F, the building is in a cooling mode.

## *2.5.5 Internal Gains*

Internal gains assumptions are included in Appendix C and consistent with the CASE report on plug loads and lighting (Rubin 2016, see Appendix D).

## *2.5.6 Exterior Surfaces*

The user enters exterior surfaces to define the envelope of the proposed design. The areas, construction assemblies, orientations, and tilts modeled are consistent with the actual building design and shall equal the overall roof/ceiling area with conditioned space on the inside and unconditioned space on the other side.

### 2.5.6.1 *Ceilings Below Attics*

Ceilings below attics are horizontal surfaces between conditioned zones and attics. The area of the attic floor is determined by the total area of ceilings below attics defined in conditioned zones.

#### *PROPOSED DESIGN*

The software allows the user to define ceilings below attic, enter the area, and select a construction assembly for each.

#### *STANDARD DESIGN*

The standard design for new construction has the same ceiling below attic area as the proposed design. The standard design is a high performance attic with a ceiling constructed with 2x4 framed trusses, and insulated with the R-values specified in Section 150.1(c) and Table 150.1-A for the applicable climate zone assuming Option B. The roof surface is a 10 lb/ft$^2$ tile roof with an air space when the proposed roof is steep slope, or a lightweight roof when the proposed roof is low slope.

Climate zones 1 through 3 and 5 through 7 have R-0, and climate zones 4 and 8 through 16 have R-13 insulation between the roofing rafters in contact with the roof deck. Climate zones 1, 2, 4, and 8 through 16 have R-38 insulation on the ceiling. Climate zones 3 and 5 through 7 have R-30 insulation on the ceiling. Climate zones 2, 3, and 5 through 7 have a radiant barrier. Climate zones 1, 4, and 8 through 16 have no radiant barrier.

#### *VERIFICATION AND REPORTING*

Ceiling below attic area and constructions are reported on the CF1R. Metal frame or SIP assemblies are reported as a special feature on the CF1R.

### 2.5.6.2 *Non-Attic (Cathedral) Ceiling and Roof*

Non-attic ceilings, also known as cathedral ceilings, are surfaces with roofing on the outside and finished ceiling on the inside but without an attic space.

#### *PROPOSED DESIGN*

The software allows the user to define cathedral ceilings and enter the area, and select a construction assembly for each. The user also enters the roof characteristics of the surface.

#### *STANDARD DESIGN*

The standard design has the same area as the proposed design cathedral ceiling modeled as ceiling below attic with the features of Option B from Section 150.1(c) and Table 150.1-A for the applicable climate zone.

The standard design building has an area of ceiling below attic equal to the non-attic ceiling/roof areas of the proposed design. The standard design roof and ceiling surfaces are modeled with the same construction assembly and characteristics as Package A. The aged reflectance and emittance of the standard design are determined by Section 150.1(c) and Table 150.1-A for the applicable climate zone.

#### VERIFICATION AND REPORTING

Non-attic ceiling/roof area and constructions are reported on the CF1R. Metal frame or SIP assemblies are reported as a special feature on the CF1R.

### 2.5.6.3 *Exterior Walls*

#### PROPOSED DESIGN

The software allows the user to define walls, enter the gross area, and select a construction assembly for each. The user also enters the plan orientation (front, left, back, or right) or plan azimuth (value relative to the front, which is represented as zero degrees) and tilt of the wall.

The wall areas modeled are consistent with the actual building design, and the total wall area is equal to the gross wall area with conditioned space on the inside and unconditioned space or exterior conditions on the other side. Underground mass walls are defined with inside and outside insulation, and the number of feet below grade. Walls adjacent to unconditioned spaces with no solar gains (such as knee walls or garage walls) are entered as an interior wall with the zone on the other side specified as attic, garage, or another zone, and the compliance manager treats that wall as a demising wall. An attached unconditioned space is modeled as an unconditioned zone.

#### STANDARD DESIGN

The standard design building has high performance walls modeled with the same area of framed walls as is in the proposed design separating conditioned space and the exterior, with a U-factor equivalent to that as specified in Section 150.1(c)1.B. and Table 150.1-A for the applicable climate zone. Walls have 2″x6″ 16-in. on center wood framing with R-19 insulation between framing and R-5 continuous insulation in climate zones 1-5, 8-16 or 2″x4″ 16-in. on center wood framing with R-15 insulation between framing and R-4 continuous insulation in climate zones 6-7. Walls adjacent to unconditioned space, such as garage walls, are treated the same as exterior walls, except there is no continuous insulation.

The standard design building is modeled with the same area of above grade mass walls with interior and exterior insulation equivalent to the requirements in Section 150.1(c)1.B. and Table 150.1-A for the applicable climate zone.

The standard design building is modeled with the same area of below grade mass walls with interior insulation equivalent to the requirements in Section 150.1(c)1.B. and Table 150.1-A for the applicable climate zone.

The gross exterior wall area in the standard design is equal to the gross exterior wall area of the proposed design. If the proposed wall area is framed wall, the gross exterior wall area of framed walls in the standard design (excluding knee walls) contains wood framing and is equally divided between the four main compass points, north, east, south, and west.

Wall construction shall match wall construction and thermal characteristics of Section 150.1(c), Table 150.1-A. Window and door areas are subtracted from the gross wall area to determine the net wall area in each orientation.

#### VERIFICATION AND REPORTING

Exterior wall area and construction details are reported on the CF1R. Metal frame or SIP assemblies are reported as a special feature on the CF1R.

### 2.5.6.4 *Exterior Thermal Mass*

Constructions for standard exterior mass is supported, but not implemented beyond the assumptions for typical mass.

The performance approach assumes that both the proposed design and standard design building have a minimum mass as a function of the conditioned area of slab floor and non-slab floor (see Section 2.5.4.1).

Mass such as concrete slab floors, masonry walls, double gypsum board, and other special mass elements can be modeled. When the proposed design has more than the typical assumptions for mass in a building then each element of heavy mass is modeled in the proposed design, otherwise, the proposed design is modeled with the same thermal mass as the standard design.

#### PROPOSED DESIGN

The proposed design may be modeled with the default 20 percent exposed mass/80 percent covered mass, or with actual mass areas modeled as separate covered and exposed mass surfaces. Exposed mass surfaces covered with flooring materials that is in direct contact with the slab can be considered as exposed mass. Examples of such materials are tile, stone, vinyl, linoleum, and hard wood.

#### STANDARD DESIGN

The conditioned slab floor in the standard design is assumed to be 20 percent exposed slab and 80 percent slab covered by carpet or casework. Interior mass assumptions as described in Section 2.5.4.1 are also assumed. No other mass elements are modeled in the standard design. The standard design mass is modeled with the following characteristics:

- The conditioned slab floor area (slab area) shall have a thickness of 3.5 inches; a volumetric heat capacity of 28 Btu/ft$^3$-°F; a conductivity of 0.98 Btu-in/hr-ft$^2$-°F. The exposed portion shall have a surface conductance of 1.3 Btu/h-ft$^2$-°F (no thermal resistance on the surface) and the covered portion shall have a surface conductance of 0.50 Btu/h-ft$^2$-°F, typical of a carpet and pad.

- The "exposed" portion of the conditioned non-slab floor area shall have a thickness of 2.0 inches; a volumetric heat capacity of 28 Btu/ft$^3$-°F; a conductivity of 0.98 Btu-in/hr- ft$^2$-ºF; and a surface conductance of 1.3 Btu/h- ft$^2$-°F (no added thermal resistance on the surface). These thermal mass properties apply to the "exposed" portion of non-slab floors for both the

proposed design and standard design. The covered portion of non-slab floors is assumed to have no thermal mass.

#### VERIFICATION AND REPORTING

Exposed mass greater than 20 percent exposed slab on grade and any other mass modeled by the user shall be reported as a special feature on the CF1R.

### 2.5.6.5 *Doors*

#### PROPOSED DESIGN

The compliance software shall allow users to enter doors specifying the U-factor, area, and orientation. Doors to the exterior or to unconditioned zones are modeled as part of the conditioned zone. For doors with less than 50 percent glass area, the U-factor shall come from *JA4, Table 4.5.1* (default U-factor 0.50), or from NFRC certification data for the entire door. For unrated doors, the glass area of the door, calculated as the sum of all glass surfaces plus two inches on all sides of the glass (to account for a frame), is modeled under the rules for fenestrations; the opaque area of the door is considered the total door area minus this calculated glass area. Doors with 50 percent or more glass area are modeled under the rules for fenestrations using the total area of the door.

When modeling a garage zone, large garage doors (metal roll-up or wood) are modeled with a 1.0 U-factor.

#### STANDARD DESIGN

The standard design has the same door area for each dwelling unit as the proposed design. The standard design door area is distributed equally between the four main compass points—north, east, south and west. All doors are assumed to have a U-factor of 0.50. The net opaque wall area is reduced by the door area in the standard design.

#### VERIFICATION AND REPORTING

Door area and U-factor are reported on the CF1R.

### 2.5.6.6 *Fenestration*

Fenestration is modeled with a U-factor and solar heat gain coefficient (SHGC). Acceptable sources of these values are National Fenestration Rating Council (NFRC), default tables from Section 110.6 of the standards, and *Reference Appendix NA6*.

In limited cases for certain site-built fenestration that is field fabricated the performance factors (U-factor, SHGC) may come from *Nonresidential Reference Appendix NA6* as described in exception 4 to Section 150.1(c)3A.

There is no detailed model of chromogenic fenestration available at this time. As allowed by exception 3 to Section 150.1(c)3A, the lower rated labeled U-factor and SHGC may be used only when installed with automatic controls as noted in the exception. Chromogenic fenestration cannot be averaged with non-chromogenic fenestration.

#### PROPOSED DESIGN

The compliance software allows users to enter individual skylights and fenestration types, the U-factor, SHGC, area, orientation, and tilt.

Performance data (U-factors and SHGC) is from NFRC values or from the Energy Commission default tables from Section 110.6 of the standards. In spaces other than sunspaces, solar gains from windows or skylights use the California simulation engine (CSE) default solar gain targeting.

Skylights are a fenestration with a slope of 60 degrees or more. Skylights are modeled as part of a roof.

#### STANDARD DESIGN

If the proposed design fenestration area is less than 20 percent of the conditioned floor area, the standard design fenestration area is set equal to the proposed design fenestration area. Otherwise, the standard design fenestration area is set equal to 20 percent of the conditioned floor area. The standard design fenestration area is distributed equally between the four main compass points—north, east, south and west.

The standard design has no skylights.

The net wall area on each orientation is reduced by the fenestration area and door area on each facade. The U-factor and SHGC performance factors for the standard design are taken from Section 150.1(c) and Table 150.1-A (Package A). Where Package A has no requirement, the SHGC is set to 0.50.

#### VERIFICATION AND REPORTING

Fenestration area, U-factor, SHGC, orientation, and tilt are reported on the CF1R.

### 2.5.6.7 *Overhangs and Sidefins*

#### PROPOSED DESIGN

Software users enter a set of basic parameters for a description of an overhang and sidefin for each individual fenestration or window area entry. The basic parameters include fenestration height, overhang/sidefin length, and overhang/sidefin height. Compliance software user entries for overhangs may also include fenestration width, overhang left extension, and overhang right extension. Compliance software user entries for sidefins may also include fin left extension and fin right extension for both left and right fins. Walls at right angles to windows may be modeled as sidefins.

Figure 7: Overhang Dimensions

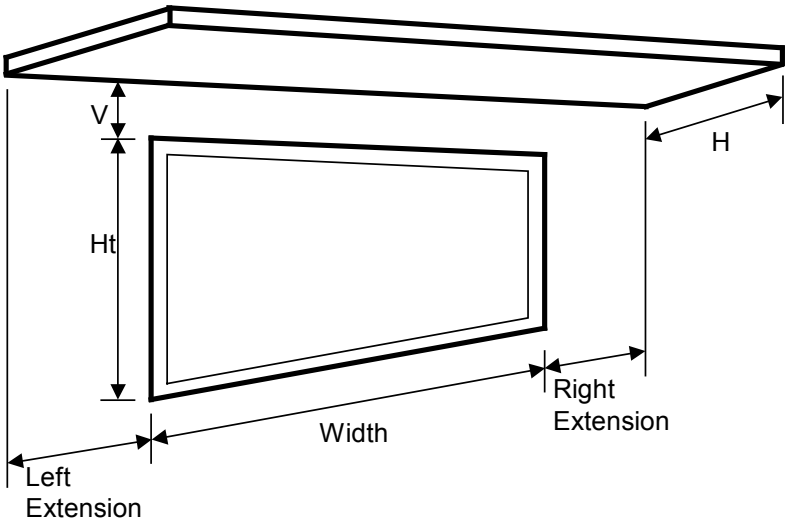
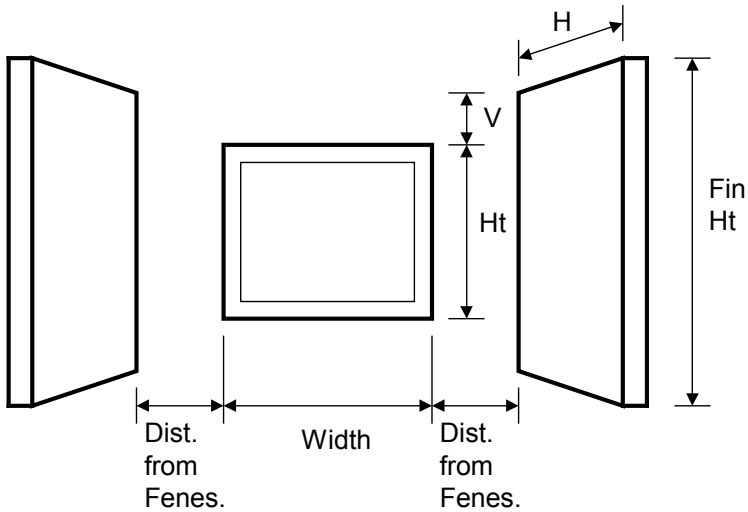



Figure 8: Sidefin Dimensions




#### *STANDARD DESIGN*

The standard design does not have overhangs or sidefins.

#### *VERIFICATION AND REPORTING*

Overhang and fin dimensions are reported on the CF1R.

### 2.5.6.8 *Interior Shading Devices*

For both the proposed and standard design, all windows are assumed to have draperies and skylights are assumed to have no interior shading. Window medium drapes are closed at night and half open in the daytime hours. Interior shading is not a compliance variable and is not user editable.

### 2.5.6.9 *Exterior Shading*

For both the proposed and standard design, all windows are assumed to have bug screens and skylights are assumed to have no exterior shading. Exterior shading is modeled as an additional glazing system layer using the ASHWAT calculation.

#### *PROPOSED DESIGN*

The compliance software shall require the user to accept the default exterior shading devices, which are bug screens for windows and none for skylights. Credit for shading devices that are allowable for prescriptive compliance are not allowable in performance compliance.

#### *STANDARD DESIGN*

The standard design shall assume bug screens. The standard design does not have skylights.

### 2.5.6.10 *Slab on Grade Floors*

#### *PROPOSED DESIGN*

The software allows users to enter areas and exterior perimeter of slabs that are heated or unheated, covered or exposed, and with or without slab-edge insulation. Perimeter is the length of wall between conditioned space and the exterior, but it does not include edges that cannot be insulated, such as between the house and the garage. The default condition for the proposed design is that 80 percent of each slab area is carpeted or covered by walls and cabinets, and 20 percent is exposed. Inputs other than the default condition require that carpet and exposed slab conditions are documented on the construction plans.

When the proposed heating distribution is radiant floor heating (heated slab), the software user will identify that the slab is heated and model the proposed slab edge insulation. The mandatory minimum requirement is R-5 insulation in climate zones 1 through 15 and R-10 in climate zone 16 (Section 110.8(g), Table 110.8-A).

#### *STANDARD DESIGN*

The standard design perimeter lengths and slab on grade areas are the same as the proposed design. Eighty percent of standard design slab area is carpeted and 20 percent is exposed. For the standard design, an unheated slab edge has no insulation with the exception of climate zone 16, which assumes R-7 to a depth of 16 inches. The standard design for a heated slab is a heated slab with the mandatory slab edge insulation of R-5 in climate zones 1 through 15 and R-10 in climate zone 16.

#### *VERIFICATION AND REPORTING*

Slab areas, perimeter lengths, and inputs of other than the default condition are reported on the CF1R.

### 2.5.6.11 *Raised Floors*

#### *PROPOSED DESIGN*

The software allows the user to input floor areas and constructions for raised floors over a crawl space, over exterior (garage or unconditioned), over a controlled ventilation crawl space, and concrete raised floors. The proposed floor area and constructions are consistent with the actual building design.

#### *STANDARD DESIGN*

The standard design has the same area and type of construction as the proposed design. The thermal characteristics meet Section 150.1(c) and Table 150.1-A. For floor areas that are framed construction, the standard design floor has R-19 in 2x6 wood framing, 16-in. on center. For floor areas that are concrete raised floor, the standard design floor is 6 inches of normal weight concrete with R-8 continuous insulation in climate zones 1, 2, 11, 13, 14, 16, R-4 in climate zoned 12 and 15, and R-0 in climate zones 3 through 10.

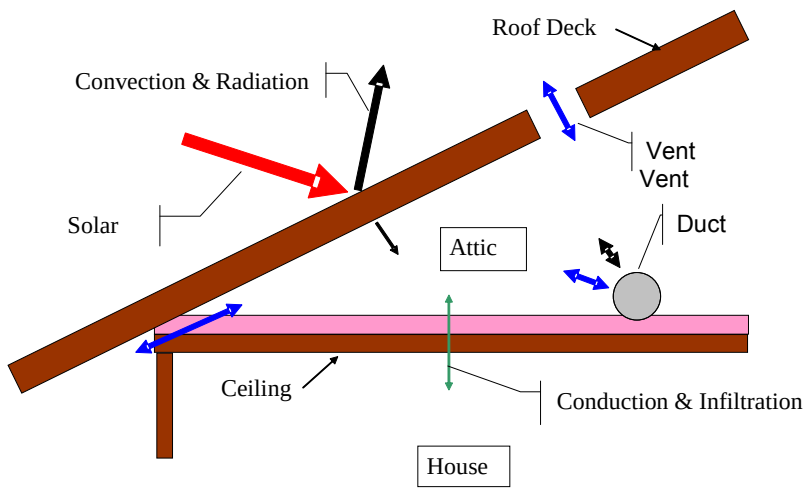
#### *VERIFICATION AND REPORTING*

Raised floor areas and constructions are reported on the CF1R.

## 2.6 Attics

The compliance software models attics as a separate thermal zone and includes the interaction with the air distribution ducts, infiltration exchange between the attic and the house, the solar gains on the roof deck, and other factors. These interactions are illustrated in Figure 9.

Figure 9: Attic Model Components



## 2.6.1 Attic Components

### 2.6.1.1 Roof Rise

This is the ratio of rise to run (or pitch), and refers to the number of feet the roof rises vertically for every 12 feet horizontally. For roofs with multiple pitches, the roof rise that makes up the largest roof area is used.

### 2.6.1.2 Vent Area

This value is the vent area as a fraction of attic floor area. This value is not a compliance variable and is assumed to be a value equal to attic floor area/300.

### 2.6.1.3 Fraction High

The fraction of the vent area that is high due to the presence of ridge, roof, or gable end mounted vents. Soffit vents are considered low ventilation. Default value is zero for attics with standard ventilation. Attics with radiant barriers are required to have a vent high fraction of at least 0.3.

### 2.6.1.4 Roof Deck/Surface Construction

Typical roof construction types are concrete or clay tile, metal tile, or wood shakes, or other high or low sloped roofing types.

### 2.6.1.5 Solar Reflectance

This input is a fraction that specifies the certified aged reflectance of the roofing material or 0.1 default value for uncertified materials. The installed value must be equal to or higher than the value specified here. Roof construction with a roof membrane mass of at least 25 lb/ft$^3$ or a roof area that has integrated solar collectors is assumed to meet the minimum solar reflectance.

### 2.6.1.6 Thermal Emittance

The certified aged thermal emittance (or emissivity) of the roofing material, or a default value. The installed value must be equal to or greater than the value modeled here. Default value is 0.85 if certified aged thermal emittance value is not available from the Cool Roof Rating Council (www.coolroofs.org). Roof construction with a roof membrane mass of at least 25 lb/ft$^2$ or roof area incorporated integrated solar collectors are assumed to meet the minimal thermal emittance.

#### PROPOSED DESIGN

The conditioning is either ventilated or unventilated. Each characteristic of the roof is modeled to reflect the proposed construction. Values for solar reflectance and thermal emittance shall be default or from the Cool Roof Rating Council.

Roofs with solar collectors or with thermal mass over the roof membrane with a weight of at least 25 lb/ft$^2$ may model the Package A values for solar reflectance and thermal emittance.

#### STANDARD DESIGN

The standard design depends on the variables of the climate zone and roof slope. Low-sloped roofs (with a roof rise of 2 feet in 12 or less) in climate zones 13 and 15 will have a standard design aged solar reflectance of 0.63 and a thermal emittance of 0.85.

Steep-sloped roofs in climate zones 10 through 15 will have a standard design roof with an aged solar reflectance of 0.20 and a minimum thermal emittance of 0.85.

Roofs with solar collectors or with thermal mass over the roof membrane with a weight of at least 25 lb/ft$^2$ are assumed to meet the standard design values for solar reflectance and thermal emittance.

#### VERIFICATION AND REPORTING

A reflectance of 0.20 or higher is reported as a cool roof, a value higher than the default but less than 0.20 is reported as a non-standard roof reflectance value.

### 2.6.2 Ceiling Below Attic

#### PROPOSED DESIGN

For each conditioned zone, the user enters the area and construction of each ceiling surface that is below an attic space. The compliance software shall allow a user to enter multiple ceiling constructions. Surfaces that tilt 60 degrees or more are treated as knee walls and are not included as ceilings. The sum of areas shall equal the overall ceiling area with conditioned space on the inside and unconditioned attic space on the other side.

The compliance software creates an attic zone with a floor area equal to the sum of the areas of all of the user input ceilings below an attic in the building. The user specifies the framing and spacing, the materials of the frame path, and the R-value of the insulation path for each ceiling construction.

The user inputs the proposed insulation R-value rounded to the nearest whole R-value. For simulation, all ceiling below attic insulation is assumed to have nominal properties of R-2.6 per inch, a density of 0.5 lb/ft$^3$ and a specific heat of 0.2 Btu/lb.

#### STANDARD DESIGN

The standard design shall have the same area of ceiling below attic as the proposed design. The ceiling/framing construction is based on the Package A prescriptive requirement and standard framing is assumed to be 2″x4″ wood trusses at 24 inches on center.

#### VERIFICATION AND REPORTING

The area, insulation R-value, and layer of each construction are reported on the CF1R.

## 2.6.3 Attic Roof Surface and Pitch

#### PROPOSED DESIGN

The roof pitch is the ratio of rise to run, (for example: 4:12 or 5:12). If the proposed design has more than one roof pitch, the pitch of the largest area is used.

The compliance software creates an attic zone roof. The roof area is calculated as the ceiling below attic area divided by the cosine of the roof slope where the roof slope is an angle in degrees from the horizontal. The roof area is then divided into four equal sections with each section sloping in one of the cardinal directions (north, east, south and west). Gable walls, dormers, or other exterior vertical surfaces that enclose the attic are ignored.

If the user specifies a roof with a pitch less than 2:12, the compliance software creates an attic with a flat roof that is 30 inches above the ceiling.

#### STANDARD DESIGN

The standard design shall have the same roof pitch, roof surface area, and orientations as the proposed design.

#### VERIFICATION AND REPORTING

The roof pitch is reported on the CF1R.

## 2.6.4 Attic Conditioning

Attics may be ventilated or unventilated. Insulation in a ventilated attic is usually at the ceiling level,but could also be located at the roof deck. Unventilated attics usually have insulation located at the roof deck and sometimes on the ceiling (Section 150.0(a)).

In an unventilated attic, the roof system becomes part of the insulated building enclosure. Local building jurisdictions may impose additional requirements.

#### PROPOSED DESIGN

A conventional attic is modeled as ventilated. When an attic will not be vented, attic conditioning is modeled as unventilated.

#### STANDARD DESIGN

Attic ventilation is set to ventilated for the standard design.

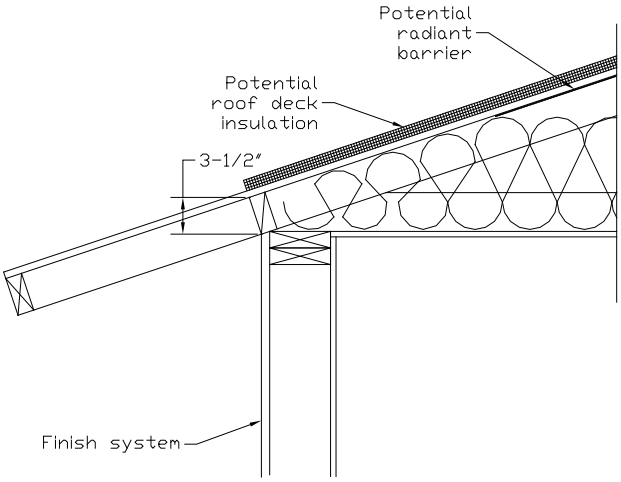
#### VERIFICATION AND REPORTING

The attic conditioning (ventilated or unventilated) is reported on the CF1R.

### 2.6.5 Attic Edge

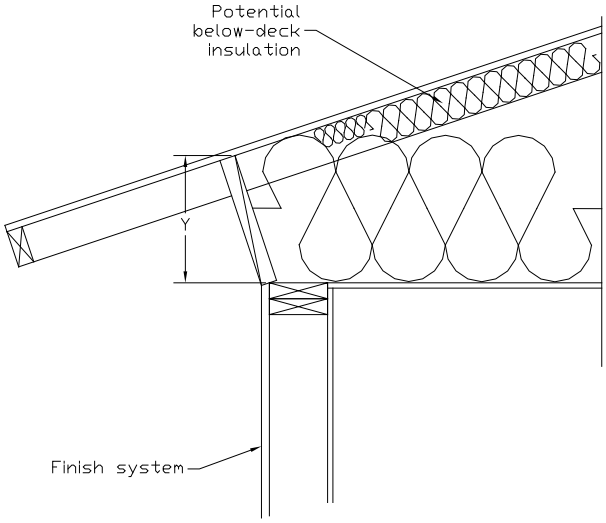
With a standard roof truss (Figure 10), the depth of the ceiling insulation is restricted to the space left between the roof deck and the wall top plate for the insulation path, and the space between the bottom and top chord of the truss in the framing path. If the modeled insulation completely fills this space, there is no attic air space at the edge of the roof. Heat flow through the ceiling in this attic edge area is directly to the outside both horizontally and vertically, instead of to the attic space. Measures that depend on an attic air space, such as radiant barriers or ventilation, do not affect the heat flows in the attic edge area.

Figure 10: Section at Attic Edge with Standard Truss




A raised heel truss (Figure 11) provides additional height at the attic edge that, depending on the height Y and the ceiling insulation R, can either reduce or eliminate the attic edge area and its thermal impact.

Figure 11: Section at Attic Edge with a Raised Heel Truss



For cases where the depth of insulation (including below deck insulation depth) is greater than the available height at the attic edge, the compliance software automatically creates cathedral ceiling surfaces to represent the attic edge area and adjusts the dimensions of the attic air space using the algorithms contained in the document *2016 Residential Alternative Calculation Method Algorithms*. If above deck insulation is modeled, it is included in the attic edge cathedral ceiling constructions, but radiant barriers below the roof deck are not.

#### PROPOSED DESIGN

The compliance software shall allow the user to specify that a raised heel truss will be used (as supported by construction drawings), with the default being a standard truss as shown in Figure 10. If the user selects a raised heel truss, the compliance software will require the user to specify the vertical distance between the wall top plate and the bottom of the roof deck (Y in Figure 11).

#### STANDARD DESIGN

The standard design shall have a standard truss with the default vertical distance of 3.5 inches between wall top plate and roof deck.

#### VERIFICATION AND REPORTING

A raised heel truss is a special feature and its vertical height above the top plate will be included on the CF1R.

### 2.6.6 The Roof Deck

The roof deck is the construction at the top of the attic and includes the solar optic properties of the exterior surface, the roofing type, the framing, insulation, air gaps, and other features. These are illustrated in Figure 12, which shows a detailed section through the roof deck.

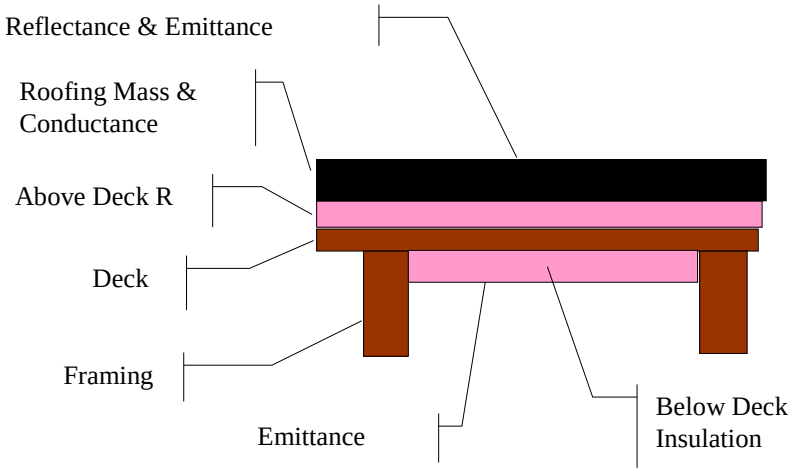



Figure 12: Components of the Attic Through Roof Deck

#### 2.6.6.1 *Radiant Barrier*

Radiant barriers are used to reduce heat flow at the bottom of the roof deck in the attic. A 0.05 emittance is modeled at the bottom surface of the roof deck if radiant barriers are used. If no radiant barrier is used, the value modeled is 0.9. If radiant barrier is installed over existing skip sheathing in a reroofing application, 0.5 is modeled.

##### *PROPOSED DESIGN*

The user shall specify whether or not the proposed design has a:

- Radiant Barrier
- No Radiant Barrier

##### *STANDARD DESIGN*

The standard design shall have a radiant barrier if required by the prescriptive standards (Section 150.1(c) and Table 150.1-A) for the applicable climate zone with Option B.

##### *VERIFICATION AND REPORTING*

Radiant barriers are reported as a special feature on the CF1R.

#### 2.6.6.2 *Below Deck Insulation*

Below deck insulation is insulation that will be installed below the roof deck between the roof trusses or rafters.

##### *PROPOSED DESIGN*

The compliance software shall allow the user to specify the R-value of insulation that will be installed below the roof deck between the roof trusses or rafters. The default is no below deck roof insulation.

##### *STANDARD DESIGN*

The standard design has below deck insulation.

##### *VERIFICATION AND REPORTING*

The R-value of any below deck insulation is reported as a special feature on the CF1R.

#### 2.6.6.3 *Roof Deck and Framing*

The roof deck is the structural surface that supports the roofing. The compliance software assumes a standard wood deck and this is not a compliance variable. The size, spacing, and material of the roof deck framing are compliance variables.

##### *PROPOSED DESIGN*

The roof deck is wood siding/sheathing/decking. The compliance software shall default the roof deck framing to 2x4 trusses at 24 in. on center. The compliance software shall allow the user to specify alternative framing size, material, and framing spacing.

#### STANDARD DESIGN

The standard design is 2x4 trusses at 24 in. on center.

#### VERIFICATION AND REPORTING

Non-standard roof deck framing or spacing is reported as a special feature on the CF1R.

### 2.6.6.4 *Above Deck Insulation*

Above deck insulation represents the insulation value of the air gap in "concrete or clay tile" or "metal tile or wood shakes." The R-value of any user modeled insulation layers between the roof deck and the roofing is added to the air gap value.

#### PROPOSED DESIGN

This input defaults to R-0.85 for "concrete or clay tile" or for "metal tile or wood shakes" to represent the benefit of the air gap, but no additional insulation. The compliance software shall allow the user to specify the R-value of additional above deck insulation in any roof deck construction assembly.

#### STANDARD DESIGN

The standard design accounts for the air gap based on roofing type, but has no additional above deck insulation.

#### VERIFICATION AND REPORTING

Above deck insulation R-value is reported as a special feature on the CF1R.

### 2.6.6.5 *Roofing Type and Mass*

#### PROPOSED DESIGN

The choice of roofing type determines the air gap characteristics between the roofing material and the deck, and establishes whether other inputs are needed, as described below. The choices for roof type are shown below.

- **Concrete or clay tile.** These have significant thermal mass and an air gap between the deck and the tiles.
- **Metal tile or wood shakes.** These are lightweight, but have an air gap between the tiles or shakes and the deck. Note that tapered cedar shingles do not qualify and are treated as a conventional roof surface.
- **Other high slope roofing types.** This includes asphalt and composite shingles, and tapered cedar shingles. These products have no air gap between the shingles and the structural roof deck.
- **Low slope membranes.** These are basically flat roofs with a slope of 2:12 or less.
- **Above deck mass**. The above deck mass depends on the roofing type. The mass is 10 lb/ft² for concrete and clay tile and 5 lb/ft² for metal tile, wood shakes, or other high slope roofing types. For low slope roofs the additional thermal mass is assumed to be gravel or stone and the user chooses one of the following inputs that is less than or equal to the weight of the material being installed above the roof deck:

- No mass
- 5 lb/ft²
- 10 lb/ft²
- 15 lb/ft²
- 25 lb/ft²

#### STANDARD DESIGN

The roof slope shall match the proposed design. The roof type for a steep slope roof is 10 lb/ft² tile. The roof type for low-slope roof is lightweight roof.

#### VERIFICATION AND REPORTING

The roof type is reported on the CF1R.

### 2.6.6.6 *Solar Reflectance and Thermal Emittance*

#### PROPOSED DESIGN

The compliance software shall allow the user to default the solar reflectance and thermal emittance of the roofing. The solar reflectance default is 0.10 for all roof types. The thermal emittance default is 0.85.

The compliance software shall allow the user to input aged solar reflectance and thermal emittance of roofing material that are rated by the Cool Roof Rating Council. The installed value must be equal to or higher than the value specified here. Roof construction with a roof membrane mass of at least 25 lb/ft² or roof area incorporated integrated solar collectors are assumed to meet the minimal solar reflectance.

#### STANDARD DESIGN

The solar reflectance and thermal emittance of the standard design roofing are as specified in the prescriptive Standards.

#### VERIFICATION AND REPORTING

Thermal emittance and solar reflectance shall be reported on the CF1R. A reflectance of 0.20 or higher is reported as a cool roof, a value higher than the default but less than 0.20 is reported as a non-standard roof reflectance value.

## 2.7 Crawl Spaces

The crawl space type is either a (1) normal vented crawl space (has a conditioned space above with raised floor insulation), (2) insulated with reduced ventilation [as used in the Building Code], or (3) sealed and mechanically ventilated crawl space (also called a controlled ventilation crawl space or CVC).

#### PROPOSED DESIGN

The software user will model the crawl space as a separate unconditioned zone, selecting the appropriate crawl space type, with the perimeter of the crawlspace (in linear feet) and the height of the crawl space.

#### STANDARD DESIGN

The standard design has a typical vented crawlspace when a crawl space is shown. Otherwise the raised floor is assumed to be over exterior or unconditioned space.

#### VERIFICATION AND REPORTING

The crawl space zone type and characteristics shall be reported on the CF1R. A controlled ventilation crawl space shall be reported as a special feature on the CF1R.

## 2.8 Garage/Storage

An attached unconditioned space is modeled as a separate unconditioned zone. While the features of this space have no effect on compliance directly, it is modeled to accurately represent the building. The modeling of the garage will shade the walls adjacent to conditioned space and will also have a lower air temperature (than the outside) adjacent to those walls. The walls and door that separate the conditioned zone from the garage are modeled as part of the conditioned zone.

#### PROPOSED DESIGN

The software user will model the area and type for the floor, exterior walls (ignore windows), large metal roll-up or wood doors (assume a 1.0 U-factor), and roof/ceiling (typically an attic or the same as the conditioned zone).

#### STANDARD DESIGN

The standard design building has the same features as the proposed design.

#### VERIFICATION AND REPORTING

The presence of an attached garage or unconditioned space is reported as general information on the CF1R. The general characteristics of the unconditioned zone are reported on the CF1R.

## 2.9 Domestic Hot Water (DHW)

Water heating energy use is based on the number of dwelling units, number of bedrooms, fuel type, distribution system, water heater type, and conditioned floor area. Detailed calculation information is included in Appendix B.

Beginning June 12, 2017, Energy Factor (EF) ratings are replaced with the new federal standard for water heating energy efficiency, Uniform Energy Factor (UEF). The classification of small and

large water heater change to consumer and commercial heaters, respectively, and a new class of residential-duty commercial water heater.

For the remainder of the 2016 Standards cycle, any UEF input is converted to EF by the software using methods described in Appendix B.

#### *PROPOSED DESIGN*

The water heating system is defined by the heater type (gas, electric resistance, or heat pump), tank type, dwelling unit distribution type, central system distribution, efficiency (either EF, UEF or recovery efficiency with the standby loss), tank volume, exterior insulation R-value (only for indirect), rated input, and tank location (for electric resistance only).

Heat pump water heaters are defined by their energy factor, volume, and tank location or, for Northwest Energy Efficiency Alliance (NEEA) rated heat pumps, by selecting the specific heater brand, model, and tank location.

Water heater and tank types include:

- Consumer/small storage: ≤ 75,000 Btu/h gas/propane, ≤ 12 kW electric, or ≤ 24 amps heat pump. Rated with UEF or EF.
- Consumer/small instantaneous: ≤ 200,000 Btu/h gas or propane, or ≤ 12 kW electric. Instantaneous water heater is a water heater with an input rating of ≥ 4,000 Btu/h/gallon of stored water; rated with a UEF or EF.
- Residential-Duty Commercial storage: > 75,000 Btu/h, and ≤ 105,000 Btu/h gas/propane, ≤ 12 kW electric, ≤ 24 amps heat pump, and rated storage volume < 120 gallons; rated with a UEF.
- Residential-Duty Commercial (large) instantaneous: ≤ 200,000 Btu/h gas/propane, ≤ 58.6 kW electric, and rated storage volume ≤ 2 gallons; rated with a UEF.
- Commercial/large storage: > 75,000 Btu/h gas/propane, >105,000 Btu/h oil, or > 12 kW electric; rated with thermal efficiency and standby loss.
- Commercial/large instantaneous: >200,000 Btu/h gas/propane, > 12 kW electric. Instantaneous water heater is a water heater with an input rating of ≥ 4,000 Btu/h per gallon of stored water; rated with thermal efficiency.
- Mini-tank (only modeled in conjunction with an instantaneous gas water heater): a small electric storage buffering tank that may be installed downstream of an instantaneous gas water heater to mitigate delivered water temperatures (e.g., cold water sandwich effect). If the standby loss of this aftermarket tank is not listed in the Energy Commission appliance database, a standby loss of 35 W must be assumed.

- Indirect: a tank with no heating element or combustion device used in combination with a boiler or other device serving as the heating element.

- Boiler: a water boiler that supplies hot water, rated with thermal efficiency or AFUE.

Heater element type includes:

- Electric resistance
- Gas
- Heat pump

For water heating systems serving a single dwelling unit, a dwelling unit distribution type must be specified. Dwelling unit distribution system types for systems serving individual dwelling units include:

- Standard (all distribution pipes insulated)
- Point of use
- Central parallel piping
- Recirculation with non-demand control (continuous pumping)
- Recirculation with demand control push button
- Recirculation with demand control occupancy/motion sensor
- HERS required pipe insulation, all lines
- HERS required central parallel piping
- HERS required recirculation, demand control push button
- HERS required recirculation with demand control occupancy/motion sensor
- HERS required compact distribution system

When a multi-family building has central water heating, both a dwelling unit and a central system distribution type must be specified. Dwelling unit distribution types for this case include:

- Standard ( all distribution pipes insulated)
- HERS required pipe insulation, all lines

Multifamily central hot water heating central system distribution types include:

- No loops or recirculation system pump
- Recirculation with no control (continuous pumping)
- Recirculation demand control (standard design for new construction)
- Recirculation with temperature modulation control
- Recirculation with temperature modulation and monitoring

Some distribution systems have an option to increase the amount of credit received if the option for HERS verification is selected. See Appendix B for the amount of credit and *Reference Appendices, Residential Appendix Table RA2-1* for a summary of inspection requirements.

***STANDARD DESIGN***

## 2.9.1 Individual dwelling units

For systems serving individual dwelling units the standard design is a single gas or propane consumer instantaneous water heater for each dwelling unit. The standard design is natural gas except if the proposed water heater is propane; then the standard is modeled as propane. The consumer instantaneous water heater is modeled with an input of 200,000 Btu/h, a tank volume of zero gallons, high draw pattern, and a UEF meeting federal standards. The current federal standard for a high draw pattern instantaneous water heater is 0.81 UEF, or the equivalent of 0.82 energy factor for the standard system.

## 2.9.2 Multiple dwelling units

When the proposed design is a central water heating system, the standard design consists of the water heating devices, a recirculation system, and solar systems as follows:

**Water heating device.** The standard design consists of the same number of water heating devices as the proposed design using the efficiencies required in the Appliance Efficiency Standards. The standard design is natural gas except if the proposed water heater is propane, then the standard is modeled as propane. Each water heating device in the proposed system is examined separately. If the proposed water heating device is gas or propane, the standard design is set to the same type and characteristics as the proposed design. If the proposed water heating device is not natural gas or propane, then the standard design is converted to a gas or propane water heater of a similar type and characteristics as the proposed design. The appropriate efficiencies and standby losses for each standard water heating device are then assigned to match the minimum requirements of the definitions and Tables F-2 and F-3 of the 2015 Appliance Efficiency Standards.

**Recirculating system.** The standard design includes a recirculation system with controls that regulate pump operation based on measurement of hot water demand and hot water return temperature, and capable of turning off the system as described in Appendix B4 Hourly Recirculation Distribution Loss for Central Water Heating Systems. When a building has more than eight dwelling units, the standard design has one recirculation loop. When a building with eight or fewer dwelling units includes a recirculation system, the standard design has one recirculation loop.

**Solar.** The standard design has a solar water heating system meeting the installation criteria specified in *Reference Residential Appendix RA4* and with a minimum solar savings fraction of 0.20 in climate zones 1 through 9, or 0.35 in climate zones 10 through 16.

#### VERIFICATION AND REPORTING

All modeled features and the number of devices modeled for the water heating system are reported on the CF1R. Electric resistance and heat pump water heaters indicate the location of the water heater. NEEA-rated heat pumps are identified by the brand and model, which must be verified by the building inspector.

Where distribution systems specify HERS verification, those features are listed in the HERS required verification listings on the CF1R.

### 2.9.3 Solar Thermal Water Heating Credit

When a water heating system has a solar thermal system to provide part of the water heating, the Solar Fraction (SF) is determined using the Energy Commission Solar Water Heating Calculator, OG-100 calculation method, or the certified OG-300 rating. (Note: The OG-300 rating can only be used for system serving individual dwelling units and not central systems.) The calculation method requires that the user specify the climate zone and conditioned floor area, in addition to published data for the solar thermal water heating system.

## 2.10 Additions/Alterations

Addition and alteration compliance is based on standards Section 150.2. Alterations must model the entire dwelling unit. When there is no addition, Section 150.2(b)2 requires at least two components of the residence must be altered. Additions may be modeled as an entirely new building (whole building), addition alone, or as “existing+addition+alteration”.

### 2.10.1 Whole Building

The entire proposed building, including all additions and/or alterations, is modeled the same as a newly constructed building. The building complies if the proposed design uses equal or less energy than the standard design. This is a difficult standard to meet as the existing building usually does not meet current standards and must be upgraded.

### 2.10.2 Addition Alone Approach

The proposed addition alone is modeled the same as a newly constructed building except that the internal gains are prorated based on the size of the dwelling. None of the exceptions included for prescriptive additions, which are implemented in the existing plus addition compliance approach (see Section 2.10.3), are given to the addition alone approach (see Standards Section 150.2(a)2.B.) The addition complies if the proposed design uses equal or less space heating, space cooling, and water heating TDV energy than the standard design.

The addition alone approach shall not be used when alterations to the existing building are proposed. Modifications to any surfaces between the existing building and the addition are part of the addition and are not considered alterations.

#### PROPOSED DESIGN

The user shall indicate that an addition alone is being modeled and enter the conditioned floor area of the addition. Any surfaces that are between the existing building and the addition are either not modeled or are treated as adiabatic surfaces. All other features of the addition shall be modeled the same as a newly constructed building.

When an existing HVAC system is extended to serve the addition, the proposed design shall assume the same efficiency for the HVAC equipment as the standard design (see Sections 2.4.1 and 2.4.5).

When a dual-glazed greenhouse or garden window is installed in an addition or alteration, the proposed design U-factor can be assumed to be 0.32.

#### STANDARD DESIGN

The addition alone is modeled the same as a newly constructed building, with the following exceptions:

A. When roofing requirements are included in Table 150.1-A, they are included in the standard design if the added conditioned floor area is greater than 300 $ft^2$.

B. When ventilation cooling (whole-house fan) is required by Table 150.1-A, it is included in the standard design when the added conditioned floor area is greater than 1,000 $ft^2$. The capacity shall be based on 1.5 $cfm/ft^2$ of conditioned floor area for the entire dwelling unit conditioned floor area.

C. When compliance with indoor air quality requirements of Section 150.0(o) apply to an addition with greater than 1,000 $ft^2$ added, the conditioned floor area of the entire dwelling unit shall be used to determine the required ventilation airflow. For additions with 1,000 $ft^2$ or less of added conditioned floor area, no indoor air quality requirements shall apply.

### 2.10.3 Existing + Addition + Alteration Approach

Standards Sections 150.2(a)2 and (b)2 contain the provisions for additions and alterations to be modeled by including the existing building in the calculations. This is called the "Existing + Addition + Alteration" (or "E+A+A") performance approach.

#### PROPOSED DESIGN

The proposed design is modeled by identifying each energy feature as part of the existing building, the addition, or an alteration. The compliance software uses this information to create an E+A+A standard design using the rules in the standards that take into account whether altered components meet or exceed the threshold at which they receive a compliance credit and whether any measures are triggered by altering a given component.

For building surfaces and systems designated below, all compliance software must provide an input field with labels for the proposed design, which define how the standard design requirements are established based on the option selected by the software user:

- **Existing:** remains unchanged within the proposed design (both standard design and proposed design have the same features and characteristics).
- **Altered:** the surface or system is altered in the proposed design. No verification of existing conditions is assumed.
- **Verified Altered**: the surface or system is altered in the proposed design and the original condition is verified by a HERS rater (an optional selection).
- **New:** a new surface or system is added in the proposed design (may be in the existing building or the addition).

Features removed are not included in the proposed design.

The user chooses whether an altered feature includes "Third Party Verification" of an existing condition (see Section 150.2, Table 150.2-B):

**Altered** with no third party verification of existing conditions (the default selection). This compliance path does not require an on-site inspection of existing conditions prior to the start of construction. The attributes of the existing condition is undefined, with the standard design for altered components based on Section 150.2, Table 150.2-B and the climate zone. Energy compliance credit or penalty is a function of the difference between the value for that specific feature allowed in Table 150.2-B and the modeled/installed efficiency of the feature.

**Verified Altered** existing conditions. This compliance path requires that a HERS rater perform an on-site inspection of pre-alteration conditions prior to construction. If an altered component or system meets or exceeds the prescriptive alteration requirements, the compliance software uses the user-defined existing condition as the standard design value. Energy compliance credit is then based on the difference between the verified existing condition for that altered feature and the modeled/installed efficiency of the proposed design.

### 2.10.3.1 *Roof/Ceilings*

#### *STANDARD DESIGN*

The standard design roof/ceiling construction assembly is based on the proposed design assembly type as shown in Table 21. For additions less than or equal to 700 square feet, radiant barrier requirements follow Option C (Section 150.1(c)9B). The standard design for unaltered ceilings and roofs is the existing condition.

**Table 21: Addition Standard Design for Roofs/Ceilings**

| Proposed Design Roof/Ceiling Types | Standard Design Based on Proposed Roof/Ceiling Status | | | | |
|---|---|---|---|---|---|
| | Add ≤ 300 $ft^2$ | Add > 300 $ft^2$ and ≤ 700 ft | Addition > 700 $ft^2$ | Altered | Verified Altered |
| Roof Deck Insulation | NR | NR | CZ 4, 8-16 = R-13 below deck | NR | Existing |
| Ceilings Below Attic | R-22 / U-0.043 | R-22 / U-0.043 | CZ 1, 2, 4, 8-16 = R-38 ceiling<br>CZ 3, 5-7 = R-30 ceiling | R-19 / U-0.054 | Existing |
| Non-Attic (Cathedral) Ceilings and Roofs | R-22 / U-0.043 | R-22 / U-0.043 | Same as above | R-19 / U-0.054 | Existing |
| Radiant Barrier | CZ 2-15 REQ<br>CZ 1, 16 NR | CZ 2-15 REQ<br>CZ 1, 16 NR | CZ 2, 3, 5-7 REQ<br>CZ 1, 4, 8-16 NR | NR | Existing |
| Roofing Surface (Cool Roof) Steep Slope | NR | CZ 10-15 >0.20 Reflectance, >0.75 Emittance | CZ 10-15 >0.20 Reflectance, >0.75 Emittance | CZ 10-15 >0.20 Reflectance >0.75 Emittance | Existing |
| Roofing Surface (Cool Roof) Low Slope | NR | CZ 13, 15 > 0.63 Reflectance, >0.75 Emittance | CZ 13, 15 > 0.63 Reflectance, >0.75 Emittance | CZ 13, 15 > 0.63 Reflectance >0.75 Emittance | Existing |

### 2.10.3.2 *Exterior Walls*

#### *PROPOSED DESIGN*

Existing structures with R-11 insulation that are being converted to conditioned space using an E+A+A approach are allowed to show compliance using R-11 wall insulation, without having to upgrade to R-13 mandatory insulation requirements. The walls are modeled as an assembly with R-11 insulation.

#### *STANDARD DESIGN*

The areas, orientation and tilt of existing, new, and altered net exterior wall areas (with windows and doors subtracted) are the same in the existing and addition portions of standard design as the proposed design.

If the proposed wall area is framed, the gross exterior wall area (excluding knee walls) is equally divided between the four building orientations: front, left, back and right. The gross exterior wall area of any unframed walls is also equally divided between the four orientations in the standard design.

The standard design exterior wall construction assembly is based on the proposed design assembly type as shown in Table 22. Framed walls are modeled as 16-in. on center wood framing. The standard design for unaltered walls is the existing condition. The software does not implement the prescriptive provision that allows eliminating continuous insulation for walls being extended to an addition.

**Table 22: Addition Standard Design for Exterior Walls**

| Proposed Design Exterior Wall Assembly Type | Standard Design Values Based on Proposed Wall Status | | |
|---|---|---|---|
| | **Addition** | **Altered** | **Verified Altered** |
| **Framed Walls** | CZ 1-5, 8-16 = R19+R5 in 2x6 (U0.051)<br>CZ 6-7 = R15+R4 in 2x4 (U-0.065) | R-13 in 2x4<br>R-19 in 2x6 | Existing |
| **Framed Wall Adjacent to Unconditioned (Garage Wall)** | R-15 in 2x4<br>R-19 in 2x6 | R-13 in 2x4<br>R-19 in 2x6 | Existing |
| **Mass Interior Insulation** | CZ 1-15 = R-13<br>CZ 16 = R-17 | Mandatory requirements have no insulation for mass walls | Existing |
| **Mass Exterior Insulation** | CZ 1-15 = R-8<br>CZ 16 = R-13 | | Existing |
| **Below Grade Mass Interior Insulation** | CZ 1-15 = R-13<br>CZ 16 = R-15 | | Existing |

### 2.10.3.3 *Fenestration*

**Table 23: Addition Standard Design for Fenestration (in Walls and Roofs)**

| Proposed Design Fenestration Type | Standard Design Based on Proposed Fenestration Status | | | | |
|---|---|---|---|---|---|
| | **Add ≤ 400 ft$^2$** | **Add > 400 and ≤ 700 ft$^2$** | **Add > 700 ft$^2$** | **Altered** | **Verified Altered** |
| **Vertical Glazing: Area and Orientation** | 75 ft$^2$ or 30% | 120 ft$^2$ or 25% | 175 ft$^2$ or 20% | See full description below. | Existing |
| **West Facing Maximum Allowed** | CZ2, 4, 6 -16=60 ft$^2$ | CZ2, 4, 6 -16=60 ft$^2$ | CZ2, 4, 6 -16=70 ft$^2$ or 5% | NR | NR |
| **Vertical Glazing: U-Factor** | 0.32 | 0.32 | 0.32 | 0.40 | See below |
| **Vertical Glazing: SHGC** | CZ2, 4, 6 -16=0.25<br>CZ1,3 & 5=0.50 | CZ2, 4, 6 -16=0.25<br>CZ1,3 & 5=0.50 | CZ2, 4, 6 -16=0.25<br>CZ1,3 & 5=0.50 | CZ2, 4 & 6-16=0.35<br>CZ1,3 & 5=0.50 | Existing |
| **Skylight: Area and Orientation** | No skylight area in the standard design | No skylight area in the standard design | No skylight area in the standard design | NR | Existing |
| **Skylight: U-Factor** | 0.32 | 0.32 | 0.32 | 0.55 | Existing |
| **Skylight: SHGC** | CZ2, 4, 6 -16=0.25<br>CZ1,3 & 5=0.50 | CZ2, 4, 6 -16=0.30<br>CZ1,3 & 5=0.50 | CZ2, 4, 6 -16=0.30<br>CZ1,3 & 5=0.50 | CZ2, 4, 6 -16=0.30<br>CZ1,3 & 5=0.50 | Existing |

#### PROPOSED DESIGN

Fenestration areas are modeled in the addition as new. In the existing building they may be existing, altered, or new. Altered (replacement) fenestration is defined in Section 150.2(b)1.B as "*existing fenestration area in an existing wall or roof [which is] replaced with a new manufactured fenestration product...Up to the total fenestration area removed in the existing wall or roof....*" Added fenestration area in an existing wall or roof is modeled as new.

#### STANDARD DESIGN

Standard design fenestration U-factor and SHGC are based on the proposed design fenestration as shown in Table 23. Vertical glazing includes all fenestration in exterior walls such as windows, clerestories, and glazed doors. Skylights include all glazed openings in roofs and ceilings.

New fenestration in an alteration is modeled with the same U-factor and SHGC as required for an addition.

West-facing limitations are combined with maximum fenestration allowed and are not an additional allowance.

The standard design is set for fenestration areas and orientations as shown in Table 23:

- ***Proposed design ≤ allowed % total fenestration area:***

In the existing building, the standard design uses the same area and orientation of each existing or altered fenestration area (in its respective existing or altered wall or roof.)

In the addition, new fenestration is divided equally between the four project compass points similar to new gross wall areas in the addition described above.

- ***Proposed design > allowed % total fenestration area:***

The standard design first calculates the allowed total fenestration area as the total existing and altered fenestration area in existing or altered walls and roofs + % of the addition conditioned floor area.

### 2.10.3.4 *Overhangs, Sidefins and Other Exterior Shading*

#### STANDARD DESIGN

The standard design for a proposed building with overhangs, sidefins, and/or exterior shades is shown in Table 24. Exterior shading (currently limited to bug screens) is treated differently than fixed overhangs and sidefins as explained in Section 2.5.6.9.

**Table 24: Addition Standard Design for Overhangs, Sidefins and Other Exterior Shading**

| Proposed Design Shading Type | Standard Design Based on Proposed Shading Status | | |
|---|---|---|---|
| | **Addition** | **Altered** | **Verified Altered** |
| **Overhangs and Sidefins** | No overhangs or sidefins | Proposed altered condition | Same as altered |
| **Exterior Shading** | Standard (bug screens on fenestration, none on skylights) | Proposed altered condition | Existing exterior shading |
| **Window Film** | No window film | Proposed altered condition | Existing exterior shading |

### 2.10.3.5 *Window Film*

#### *PROPOSED DESIGN*

A window film must have at least a 10-year warranty and is treated as a window replacement. The values modeled are either the default values from Tables 110.6-A and 110.6-B or the NFRC Window Film Energy Performance Label.

### 2.10.3.6 *Floors*

#### *STANDARD DESIGN*

Table 150.2-C requires that the standard design is based on the mandatory requirements from Section 150.0(d). The standard design for floors is shown in Table 25.

**Table 25: Addition Standard Design for Raised Floor, Slab-on-Grade, and Raised Slab**

| Proposed Design Floor Type | Standard Design Based on Proposed Floor Status (Tag) | | |
|---|---|---|---|
| | **Addition** | **Altered (mandatory)** | **Verified Altered** |
| **Raised Floor Over Crawl Space or Over Exterior** | R-19 in 2x6 16” o.c. wood framing | R-19 in 2x6 16” o.c. wood framing | If proposed U $\leq$ 0.037, standard design = existing raised; if proposed U > 0.037, standard design = Altered |
| **Slab-on-Grade: Unheated** | CZ1-15: R-0<br>CZ16: R-7 16” vertical | Proposed design | Existing unheated slab-on-grade |
| **Slab-on-Grade: Heated** | CZ1-15: R-5 16” vertical<br>CZ 16: R-10 16” vertical | Proposed design | Existing heated slab-on-grade |
| **Raised Concrete Slab** | CZ1,2,11,13,14,16: R-8<br>CZ3-10: R-0<br>CZ12,15: R-4 | Proposed design | Existing raised concrete slab |

### 2.10.3.7 *Thermal Mass*

#### *STANDARD DESIGN*

The standard design for thermal mass in existing plus addition plus alteration calculations is the same as for all newly constructed buildings as explained in Section 2.5.4.1.

### 2.10.3.8 *Air Leakage and Infiltration*

#### *STANDARD DESIGN AIR LEAKAGE AND INFILTRATION*

The standard design for space-conditioning systems is shown in Table 26.

**Table 26: Addition Standard Design for Air Leakage and Infiltration**

| Proposed Air Leakage and Infiltration | Standard Design Air Leakage Based on Building Type | | |
|---|---|---|---|
| | **Addition** | **Altered** | **Verified Altered** |
| **Single Family Buildings** | 5 ACH50 | 5 ACH50 | Diagnostic testing of existing ACH50 value by HERS rater or 7.0 ACH50, whichever is less |
| **Multi-Family Buildings** | 7 ACH50 | 7 ACH50 | 7 ACH50 |

### 2.10.3.9 *Space Conditioning System*

#### *STANDARD DESIGN*

The standard design for space-conditioning systems is shown in Table 27.

When cooling ventilation is required by Sections 150.1 and 150.2, the capacity is 1.5 cfm/ft$^2$ of conditioned floor area for the entire dwelling unit.

When compliance with indoor air quality requirements of Section 150.0(o) apply to an addition with greater than 1,000 ft$^2$ added, the conditioned floor area of the entire dwelling unit is used to determine the required ventilation airflow. For additions with 1,000 ft$^2$ or less of added conditioned floor area, no indoor air quality requirements shall apply.

**Table 27: Addition Standard Design for Space Conditioning Systems**

| Proposed Design Space Conditioning System Type | Standard Design Based on Proposed Space Conditioning Status | | |
|---|---|---|---|
| | **Added** | **Altered** | **Verified Altered** |
| **Heating System:** | See Section 2.4 and 2015 Federal Appliance Stds based on fuel source and equipment type | Same as Addition | Existing heating fuel type and equipment type/efficiency |
| **Cooling System:** | See Section 2.4 and 2015 Federal Appliance Stds based on fuel source and equipment type | Same as Addition | Existing cooling equipment type/efficiency |
| **Refrigerant Charge** | Climate zones 2, 8-15:Yes<br>Climate zones 1, 3-7: No | Same as Addition | Existing |
| **Whole House Fan (WHF) applies only if addition > 1,000 ft2** | Climate zones 8-14; 1.5 cfm/ft$^2$ | | Existing condition. To count as Existing the WHF must be $\geq$ 1.5 cfm/ft$^2$ and be CEC-rated |
| **Indoor Air Quality applies only if addition > 1,000 ft2** | Meet mandatory ventilation for entire dwelling | N/A | N/A |

## 2.10.3.10 *Duct System*

***STANDARD DESIGN***

**Table 28: Addition Standard Design for Duct Systems**

| Proposed Design Duct System Type | Standard Design Based on Proposed Duct System Status | |
|---|---|---|
| | **Altered** | **Verified Altered** |
| **All** | CZ 1-10, 12-13: Duct insulation R-6 and duct sealing $\leq$ 15%<br>CZ 11, 14-16: Duct insulation R-8 and duct sealing $\leq$ 15% | Existing duct R-value and duct leakage of 15% |

Note 1: Refer to Section 150.2(b)1Diia for definition of an "Entirely New or Complete Replacement Duct System".

### 2.10.3.11 *Water Heating System*

***STANDARD DESIGN***

**Table 29: Addition Standard Design for Water Heater Systems**

| Proposed Design Water Heating System Type | Standard Design Based on Proposed Water Heating Status | | |
|---|---|---|---|
| | **Addition (adding water heater)** | **Altered** | **Verified Altered** |
| **Single Family** | Package A water heating system | Existing fuel type, proposed tank type, mandatory requirements (excluding any solar) | Existing water heater type(s), efficiency, distribution system. |
| **Multi-family: Individual Water Heater for Each Dwelling Unit** | Package A water heating system for each dwelling unit (see Section 2.9) | Existing fuel type, proposed tank type, mandatory requirements (excluding any solar) | Existing water heater type(s), efficiency, distribution system |
| **Multi-family: Central Water Heating System** | Central water heating system per Section 2.9 | Mandatory and Prescriptive requirements (excluding any solar) | Existing water heater type(s), efficiency, distribution system |

## 2.11 Documentation

The software shall be capable of displaying and printing an output of the energy use summary and a text file of the building features. These are the same features as shown on the CF1R when generated using the report manager.

See public domain software user guide or vendor software guide for detailed modeling guidelines

# 3 Energy Design Rating

The software can calculate an energy design rating (EDR) as required in the CALGreen energy provisions (Title 24, Part 11). The EDR implementation is limited to newly constructed single-family and multifamily dwellings.

The EDR is an alternate way to express the energy performance of a home using a scoring system where 100 represents the energy performance of a reference design building meeting the envelope requirements of the 2006 International Energy Conservation Code (IECC). The EDR is similar to the energy rating index in the 2015 IECC and the 2014 Residential Energy Services Network (RESNET) standard. Combining high levels of energy efficiency with generating renewable energy, a score of zero or less can be achieved.

Buildings complying with the current Building Energy Efficiency Standards are more efficient than the 2006 IECC, so most newly constructed buildings will have EDR scores below 100. Buildings with renewable generation like PV can achieve a negative score. If an EDR is calculated for an older inefficient home, the score would likely be well over 100.

When the user requests an EDR calculation, additional inputs are required specifying more details about PV systems, and an EDR screen is displayed at the end of a calculation and reported on the CF1R. EDR PV inputs and calculations are not used for compliance with the Title 24, Part 6.

## 3.1 Calculation Procedure

To calculate the EDR, the user enters the proposed home with additional inputs for PV as described in Section 3.3.

The software calculates the proposed design energy use and the standard design energy use in accordance with the normal compliance rules. A third annual simulation is performed to determine the reference design energy use as described in Section 3.2.

EDR calculations are based on the total energy use with units of kTDV/ft$^2$ with an adjustment to maintain similar ratings with different fuel types. The total energy use includes the energy use associated with the space heating, space cooling, and water heating used in compliance calculations plus the energy use for inside lighting, appliances, plug loads, and exterior lighting.

There are adjustments made to these EDR calculations to address the issue that the 2006 IECC includes efficiency specifications that result in significantly different levels of total energy use in the reference design for gas and electric equipment, and appliances. These adjustments are needed to ensure that homes using gas water heating equipment and gas appliances are not penalized in the EDR results, as compared to homes using electric water heating equipment and electric appliances. This adjustment provides EDRs for gas and electric homes that implement a fuel neutral approach in the reference design total energy use.

Four quantities are reported for the EDR ratings. The EDR of the standard design building is provided to illustrate how the 2016 standard design compares with the reference design. The EDR score of the proposed design is provided separately from the EDR value of installed PV so that the effects of efficiency and renewable energy can both be seen. The final EDR for the proposed building includes both the effects of efficiency and PV.

The EDR values are calculated for each end use and fuel type and then summed for the total EDR as follows:

$$EDR_{standard} = EU_{standard} / EU_{adjusted} \times 100 \qquad \textit{Equation 14}$$

$$EDR_{proposed\ without\ PV} = (EU_{proposed} - PV_{proposed}) / EU_{adjusted} \times 100 \qquad \textit{Equation 15}$$

$$EDR_{proposed\ PV} = PV_{proposed} / EU_{adjusted} \times 100 \qquad \textit{Equation 16}$$

$$EDR_{proposed\ final} = EU_{proposed} / EU_{adjusted} \times 100 \qquad \textit{Equation 17}$$

Where:

$EU_{standard}$ = Standard Design Energy Use (kTDV/ft²)

$EU_{reference}$ = Reference Design Energy Use (kTDV/ft²)

$EU_{proposed}$ = Proposed Design Energy Use (kTDV/ft²)

$PV_{proposed}$ = Proposed PV Energy Generation (kTDV/ft²)

$EU_{adjusted} = EU_{reference}$ / Adjustment (kTDV/ft²)

Adjustment = 1 for all end uses that are electric; or

Adjustment = Ratio from Table 30 for all end uses that are gas.

**Table 30: EDR Adjustments by End Use for Gas Fuel Type**

| Climate Zone | Single Family Space Heating | Single Family Water Heating | Single Family Appliances | Multi Family Space Heating | Multi Family Water Heating | Multi Family Appliances |
|---|---|---|---|---|---|---|
| 1 | 1.01 | 0.44 | 0.59 | 0.99 | 0.49 | 0.57 |
| 2 | 1.02 | 0.46 | 0.59 | 1.01 | 0.51 | 0.57 |
| 3 | 1.16 | 0.46 | 0.59 | 1.17 | 0.50 | 0.57 |
| 4 | 1.04 | 0.47 | 0.59 | 1.03 | 0.52 | 0.57 |
| 5 | 1.05 | 0.45 | 0.60 | 1.04 | 0.51 | 0.58 |
| 6 | 1.17 | 0.50 | 0.60 | 1.15 | 0.56 | 0.58 |
| 7 | 1.22 | 0.47 | 0.59 | 1.36 | 0.52 | 0.57 |
| 8 | 1.20 | 0.50 | 0.60 | 1.21 | 0.56 | 0.58 |
| 9 | 1.17 | 0.50 | 0.60 | 1.17 | 0.58 | 0.58 |
| 10 | 1.12 | 0.51 | 0.60 | 1.09 | 0.58 | 0.58 |
| 11 | 1.00 | 0.48 | 0.59 | 0.98 | 0.54 | 0.57 |
| 12 | 1.05 | 0.47 | 0.59 | 1.03 | 0.53 | 0.57 |
| 13 | 1.03 | 0.49 | 0.59 | 1.01 | 0.55 | 0.57 |
| 14 | 1.01 | 0.51 | 0.60 | 0.99 | 0.57 | 0.58 |
| 15 | 1.13 | 0.59 | 0.60 | 1.10 | 0.65 | 0.58 |
| 16 | 0.95 | 0.48 | 0.60 | 0.91 | 0.53 | 0.58 |

The ratios were calculated by using a prototype analysis once with gas fuel type for space heating, water heating, and appliances, and once with electric fuel types. The ratio is the end use energy for the EDR reference building run with gas (kTDV/ft²) divided by the run with electric (kTDV/ft²).

## 3.2 Reference Design

The reference design is calculated using the same inputs, assumptions and algorithms as the standard design except for the following requirements:

- Air handler power. The air handler power is 0.8 W/cfm.
- Air infiltration rate. The air infiltration rate is 7.2 ACH50.
- Cooling airflow. The air handler airflow is 300 CFM/ton.
- Duct R-value. The duct R-value is R-8.
- Duct leakage rate. The duct leakage rate is modeled as an HVAC distribution efficiency of 80 percent.
- Insulation Installation Quality (QII): QII is modeled as "improved".
- Wall construction. Climate zones 2-15 have 2x4 R-13 walls. Climate zones 1 and 16 have 2x6 R-19 walls.
- Roof/ceiling construction. Climate zones 2-15 have R-30 ceiling. Climate zones 1 and 16 have R-38 ceiling. No climate zones include radiant barriers or cool roofs.
- Floor construction. Climate zones 2-15 have 2x10 R-19 floors. Climate zones 1 and 16 have 2x10 R-30 floors.
- Slab edge insulation. Climate zones 1 and 16 include R-10 insulation 24 inches deep.
- Window U-factors. Climate zones 2-15 have 0.65 U-factor. Climate zones 1 and 16 have 0.35 U-factor.
- Window SHGC. All windows have 0.4 SHGC.
- Window area. When the window area is below 18 percent of the floor area, the reference design has the same area as the proposed design. Above 18 percent, the reference design has 18 percent.
- HVAC equipment efficiencies. HVAC equipment meets NAECA requirements in effect in 2006 such as 78 percent AFUE for gas central furnace, 13 SEER for central AC.
- Water heating efficiency. Water heating modeled as a 40 gallon storage water with a 0.594 EF if gas or a 0.9172 EF if electric.
- Appliance and plug load energy use and internal gains. Energy use and internal gains for appliance and miscellaneous plug loads are modeled as specified the ANSI/RESNET/ICC 301-2014 Standard.

## 3.3 Energy Design Rating PV System Credit

The PV credit reflected in the Energy Design Rating for both single-family and multifamily buildings uses calculations found in PVWatts technical documentation (see Appendix D). The PV system size and module type are required inputs. Users may select simplified or detailed inputs. With detailed inputs, the inverter efficiency must be included. The user can select either detailed installation information for the orientation and angle/tilt of the array or select California flexible installation.

Installation and verification must meet the requirements of *Residential Appendix RA4.6.1*.

## 3.4 Net Energy Metering

Net Energy Metering (NEM) sets rules for compensation of PV generated electricity. NEM sizing rules limit the PV size to a PV equal to the site annual kWh.

NEM compensation rules set the limits for compensating PV generation. Compensation varies by (1) behind the meter self-utilized kWh, (2) hourly exports, and (3) net annual surplus.

## 3.5 Battery Storage

The software provides an EDR credit for a battery storage system that is coupled with a PV array. If specified, the battery storage size must be 6 kWh or larger. For Part 6 compliance this credit has no impact on energy efficiency components. Including a battery storage system allows downsizing the PV system to reach a specific EDR target.

Two control options available are:

> Default Control. A simple control strategy that provides a modest EDR credit. With default control, the software assumes that the batteries are charged anytime PV generation (generation) is greater than the house load (load), and conversely, the batteries are discharged when load exceeds generation.
>
> Advanced/Utility or Aggregator-Controlled. A more sophisticated control strategy maximizes the TDV value of stored kWh. With advanced control, the software assumes that the batteries are charged when generation exceeds the load; batteries only discharge during the highest "anticipated" TDV intervals to maximize the value of the stored kWh.

For Part 6 compliance, the proposed design EDR credit is limited to a PV size equal to the site annual kWh. The effect of a larger PV array has a minimal impact on the EDR score. For above code projects, the software allows oversizing the proposed design PV system by a factor of 1.6 for full credit if a battery system of at least 6 kWh is coupled with the PV system.

Software includes an option to allow excess PV generation EDR credit for above code programs. This allows exceeding the 1.6 size limit. When selected, the software allows any size PV with full EDR credit.

# APPENDIX A – SPECIAL FEATURES

| *Measure* | *CF1R Documentation Requirement* |
|---|---|
| **GENERAL** | |
| Battery System Storage (kWh) | Special feature |
| Controlled-Ventilation Crawlspace (CVC) | Not yet implemented |
| Photovoltaic (PV) system credit (kWdc) | Special feature |
| Zonal heating controls | Special feature |
| **ENVELOPE** | |
| Above deck insulation | Special feature |
| Advanced wall framing | Special feature |
| Below deck insulation | Special feature |
| Building air leakage/reduced infiltration | HERS verification of reported ACH50 value |
| Cool roof | Special feature |
| Dynamic glazing | Not yet implemented |
| Exterior shading device | Not yet implemented |
| Exposed slab area greater than 20% | Special feature |
| High quality insulation installation (QII) | HERS verification |
| Metal-framed assembly | Special feature |
| Non-default spray foam insulation R-values | HERS verification |
| Overhangs and sidefins | Special feature |
| Raised heel truss (height above top plate) | Special feature |
| Spray foam insulation, closed cell >R-5.8/inch | HERS verification |
| Spray foam insulation, open cell >R-3.6/inch | HERS verification |
| Structurally insulated panel (SIP) assembly | Special feature |
| **MECHANICAL** | |
| Air handling unit fan efficacy | HERS verification |
| Airflow or system airflow (cfm) | HERS verification |

| | |
|---|---|
| Central fan ventilation cooling, fixed speed | Special feature |
| Central fan ventilation cooling, variable speed | Special feature |
| Energy efficiency ratio (EER) | HERS verification |
| Evaporatively-cooled condenser | HERS verification |
| Evaporative cooling, indirect, indirect/direct | Not yet implemented. |
| Fan efficacy | HERS verification |
| High seasonal energy efficiency ratio (SEER) | HERS verification |
| Indoor air quality ventilation | HERS verification |
| No cooling | Special feature |
| Refrigerant charge/fault indicator display (FID) | HERS verification |
| Ventilation cooling system, central fan | Special feature |
| Whole-building mechanical ventilation airflow | HERS verification |
| Whole house fan | Special feature |
| **DUCTS** | |
| Buried duct | HERS verification |
| Bypass duct conditions in a zonal system | HERS verification |
| Deeply buried duct | HERS verification |
| Duct leakage testing | HERS verification |
| Ducts in conditioned space | HERS verification |
| Ducts in crawl space | Special feature |
| High R-value ducts | Special feature |
| Low leakage air handler | HERS verification |
| Low leakage ducts in conditioned space | HERS verification |
| Non-standard duct leakage target | HERS verification |
| Non-standard duct location (any location other than attic) | Special feature |
| Return duct design | HERS verification |
| Supply duct location, surface area, R-value | HERS verification |

| | |
|---|---|
| **WATER HEATING** | |
| Compact distribution system (HERS verified) | HERS verification |
| Multifamily: Recirculating demand control | Special feature |
| Multifamily: No loops or recirc pump | Special feature |
| Multifamily: Recirculating with no control (continuous pumping) | Special feature |
| Multifamily: Recirculating with temperature modulation | Special feature |
| Multifamily: Recirculating with temperature modulation and monitoring | Special feature |
| Multifamily: Solar water heating credit | Special feature and additional documentation |
| Central parallel piping | Special feature |
| Central parallel piping (HERS verified) | HERS verification |
| Pipe insulation, all lines | Special feature |
| Pipe insulation, all lines (HERS verified) | HERS verification |
| Point of use | Special feature |
| Recirculation with demand control, occupancy/ motion sensor | Special feature |
| Recirculation with demand control, occupancy/ motion sensor (HERS verified) | HERS verification |
| Recirculation with demand control, push button | Special feature |
| Recirculation with demand control, push button (HERS verified) | HERS verification |
| Recirculation with non-demand control (continuous pumping) | Special feature |
| Solar water heating credit, single family building | Special feature and additional documentation |
| Northwest Energy Efficiency Alliance (NEEA) rated heat pump water heater; specific brand/model, or equivalent, must be installed | Special feature |
| **ADDITIONS/ALTERATIONS** | |
| Verified existing conditions | HERS verification |

# APPENDIX B – WATER HEATING CALCULATION METHOD

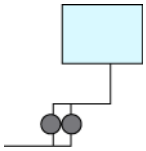
## B1. Purpose and Scope

This appendix documents the methods and assumptions used for calculating the hourly energy use for residential water heating systems for both the proposed design and the standard design. The hourly fuel and electricity energy use for water heating will be combined with hourly space heating and cooling energy use to come up with the hourly total fuel and electricity energy use to be factored by the hourly TDV energy multiplier. The calculation procedure applies to low-rise single family, low-rise multifamily, and high-rise residential.

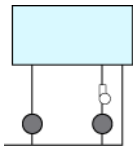
When buildings have multiple water heaters, the hourly total water heating energy use is the hourly water heating energy use summed over all water heating systems, all water heaters, and all dwelling units being modeled.

The following diagrams illustrate the domestic hot water (DHW) system types recognized by the compliance software.

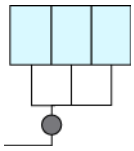
1 One distribution system with one or multiple water heaters serving a single dwelling unit. The system might include recirculation loops within the dwelling unit.



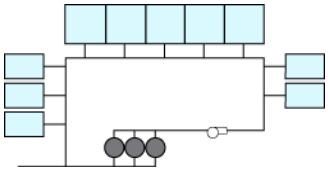
2 Two water heaters with independent distribution systems serving a single dwelling unit. One or more of the distribution systems may include a recirculation loop within the dwelling unit.



3 One distribution system without recirculation loop and with one or multiple water heaters serving multiple dwelling units.



4 One distribution system with one or multiple recirculation loops and with one or multiple water heaters serving multiple dwelling units.



## B2. Water Heating Systems

Water heating distribution systems may serve more than one dwelling unit and may have more than one water heating appliance. The energy used by a water heating system is calculated as the sum of the energy used by each individual water heater in the system. Energy used for the whole building is calculated as the sum of the energy used by each of the water heating systems. To delineate different water heating elements several indices are used.

i Used to describe an individual dwelling unit. For instance, $CFA_i$ would be the conditioned floor area of the $i^{th}$ dwelling unit. Nunit is the total number of dwelling units.

j Used to refer to the number of water heaters in a system. NWH is the total number of water heaters.

k Used to refer to a water heating system or distribution system. A building can have more than one system and each system can have more than one water heater.

l Used to refer to the $l^{th}$ unfired- or indirectly-fired storage tank in the $k^{th}$ system. $NL_k$ is the total number of unfired- or indirectly-fired storage tanks in the $k^{th}$ system. Temperature buffering tanks with electric heating (e.g., minitanks) shall not to be treated as unfired or indirectly-fired storage tanks.

| Symbol | Definition | Notes |
|---|---|---|
| CFA | Conditioned floor area, $ft^2$ | |
| NFloor | Number of floors in building | |
| Nunit | Number of units in building | |
| NK | Number of water heating systems | |
| $NWH_k$ | Number of water heaters in $k^{th}$ system | |
| $NLoop_k$ | Number of recirculation loops in kth system (multi-unit dwellings only) | |
| $CFA_i$ | Conditioned floor area of ith dwelling unit | |
| $CFAU_k$ | Average unit conditioned floor area served by kth system, $ft^2$ | |

## B3. Hot Water Consumption

The schedule of hot water use that drives energy calculations is derived from measured data as described in (Kruis, 2016). That analysis produced 365 day sets of fixture water draw events for dwelling units having a range of number of bedrooms. The draws are defined in the files DHWDUSF.TXT (for single-family) and DHWDUMF.TXT (for multifamily) that ship with CBECC-Res. Each draw is characterized by a start time, duration, flow rate, and end use. The flow rates given are the total flow at the point of use (fixture or appliance). This detailed representation allows

derivation of draw patterns at 1 minute intervals as is required for realistic simulation of heat pump water heaters.

The fixture flow events are converted to water heater (hot water) draws by 1) accounting for mixing at the point of use and 2) accounting for waste, distribution heat losses, and solar savings:

$$VS_k = VF_k \times f_{hot} \times f_{wh} \qquad \textit{Equation 1}$$

*Where*

$VS_k$ = Hot water draw at the k[th] water heating system's delivery point (gal)

$VF_k$ = Mixed water draw at an appliance or fixture (gal) served by the k[th] water heating system, as specified by input schedule

$f_{hot}$, $f_{wh}$ = End-use-specific factors from the following:

| End use | $f_{hot}$ | $f_{wh}$ |
|---|---|---|
| Shower | $\frac{105 - T_{inlet}}{T_s - T_{inlet}}$ | $WF_k \times \max\left(0, DLM_k - SSF_k\right)$ |
| Bath | | |
| Faucet | 0.50 | $1 - SSF_k$ |
| Clothes washer | 0.22 | |
| Dish washer | 1 | |

$T_s$ = Hot water supply temperature (ºF); ssumed to be 115 °F

$T_{inlet}$ = Cold water inlet temperature (ºF) as defined in Section B3.3

$WF_k$ = Hot water waste factor

- $WF_k$ = 0.9 for within-dwelling-unit pumped circulation systems (see Table B-1)
- $WF_k$ = 1.0 otherwise

$DLM_k$ = Distribution loss multiplier (unitless), see *Equation 5*

$SSF_k$ = Hourly solar savings fraction (unitless) for the k[th] water heating system, which is the fraction of the total water heating load that is provided by solar hot water heating. The annual average value for SSF is provided from the results generated by the Energy Commission approved calculations approaches for the OG-100 and OG-300 test procedure. A Commission-approved method shall be used to convert the annual average value for SSF to hourly $SSF_k$ values for use in compliance calculations.

The individual water heater draws are combined to derive the overall demand for hot water.

For each hour of the simulation, all water heater draws are allocated to 1 minute bins using each draw's starting time and duration. This yields a set of 60 $VS_{k,t}$ values for each hour that is used as input to the detailed heat pump water heating model (HPWH) as discussed in Section B7.7. For

hourly efficiency-based models used for some water heater types, the minute-by-minute values are summed to give an hourly hot water requirement:

$$GPH_k = \sum_{t=1}^{60} VS_{k,t} \qquad \textit{Equation 2}$$

In cases where multiple dwelling units are served by a common water heating system, the dwelling unit draws are summed.

In cases where there are multiple water heating systems within a dwelling unit, the draws are divided equally among the systems. For minute-by-minute draws (for HPWH models), this allocation is accomplished by assigning draws to systems in rotation within each end use. This ensures that some peak draw events get assigned to each system. Since heat pump water heater performance is non-linear with load (due to activation of resistance backup), allocation of entire events to systems is essential. Note that realistic assignment of draws to specific systems requires information about the plumbing layout of the residence and capturing that is deemed to impose an unacceptable user input burden.

## B4. Hourly Adjusted Recovery Load

The hourly-adjusted recovery load for the kth water heating system is calculated as:

$$HARL_k = HSEU_k + HRDL_k + \sum_{1}^{NL_k} HJL_l \qquad \textit{Equation 3}$$

*where*

$HSEU_k$ = Hourly standard end use at all use points (Btu), see *Equation 4*

$HRDL_k$ = Hourly recirculation distribution loss (Btu), see *Equation 11*; $HDRL_k$ is non-zero only for multi-family central water heating systems

$NL_k$ = Number of unfired or indirectly-fired storage tanks in the $k^{th}$ system

$HJL_l$ = Tank surface losses of the $l^{th}$ unfired tank of the $k^{th}$ system (Btu), see *Equation 40*

*Equation 4* calculates the hourly standard end use (HSEU). The heat content of the water delivered at the fixture is the draw volume in gallons (GPH) times the temperature rise ΔT (difference between the cold water inlet temperature and the hot water supply temperature) times the heat required to elevate a gallon of water 1 ºF (the 8.345 constant).

$$HSEU_k = 8.345 \times GPH_k \times \left(T_s - T_{inlet}\right) \qquad \textit{Equation 4}$$

*where*

$HSEU_k$ = Hourly standard end use (Btu)

$GPH_k$ = Hourly hot water consumption (gallons) from *Equation 2*

*Equation 5* calculates the distribution loss multiplier (DLM) which combines two terms: the standard distribution loss multiplier (SDLM), which depends on the floor area of the dwelling unit and the distribution system multiplier (DSM) listed in Table B-1.

$$DLM_k = 1 + (SDLM_k - 1) \times DSM_k \qquad \textit{Equation 5}$$

*where*

$DLM_k$ = Distribution loss multiplier (unitless)

$SDLM_k$ = Standard distribution loss multiplier (unitless), see *Equation 6*

$DSM_k$ = Distribution system multiplier (unitless), see Section 3.2

Several relationships depend on $CFA_k$, the floor area served (see below).

*Equation 6* calculates the standard distribution loss multiplier (SDLM) based on dwelling unit floor area. Note that in *Equation 6*, that floor area is capped at 2500 ft²--without that limit, *Equation 6* produces unrealistic $SDLM_k$ values for large floor areas.

$$\begin{aligned} SDLM_k &= 1.0032 \\ &+0.0001864 \times CFA_k \\ &-0.0000000215 \times CFA_k \times CFA_k \end{aligned}$$

*Equation 6*

*where*

$SDLM_k$= Standard distribution loss multiplier (unitless).

$CFAU_k$= Dwelling unit conditioned floor area (ft2) served by the kth system, calculated using methods specified in *Equation 7*.

Single dwelling unit,

$$CFAU_k = CFA / NK$$

For multiple dwelling units served by a central system:

$$CFAU_k = \frac{\sum_{\text{all units served by system k}} CFA_i}{Nunit_k} \qquad \textit{Equation 7}$$

*Method WH-CFAU*

Alternatively, if the system-to-unit relationships not known:

$$CFAU_k = \frac{\sum_{\text{all units served by any central system}} CFA_i}{\text{Number of units served by any central system}}$$

Note: "Method" designations are invariant tags that facilitate cross-references from comments in implementation code.

When a water heating system has more than one water heater, the total system load is assumed to be shared equally by each water heater, as shown in *Equation 8*.

$$HARL_j = \frac{HARL_k}{NWH_k} \qquad \textit{Equation 8}$$

*where*

$HARL_j$ = Hourly adjusted recovery load for the $j^{th}$ water heater of the $k^{th}$ system (Btu)

$HARL_k$ = Hourly adjusted total recovery load for the $k^{th}$ system (Btu)

$NWH_k$ = The number of water heaters in the $k^{th}$ system

## B4.1. Distribution Losses within the Dwelling Unit

The distribution system multiplier (DSM, unitless) is an adjustment for alternative water heating distribution systems within the dwelling unit. A value of 1.00 for "standard" distribution systems, defined as a non-recirculating system with the following mandatory requirements:

The full length of distribution piping is insulated in accordance with Section 150.0(j)2.

For all four system types, values for alternative distribution systems are given in Table B-1. Improved DSM values are available for cases where voluntary HERS inspections are completed, as per the eligibility criteria shown in Reference Residential Appendix RA4.4. Detailed descriptions of all of the distribution system measures are found in Residential Appendix RA 4.4.

***Table B-1. Distribution System Multipliers within a Dwelling Unit with One or More Water Heaters***

| Distribution System Types | Assigned Distribution System Multiplier | System Type 1 and 2 | System Type 3 and 4 |
|---|---|---|---|
| **No HERS Inspection Required** | | | |
| Trunk and Branch -Standard (STD) | 1.0 | Yes | Yes |
| | | | |
| Central Parallel Piping (PP) | 1.10 | Yes | |
| Point of Use (POU) | 0.30 | Yes | |
| Recirculation: Non-Demand Control Options (R-ND) | 9.80* | Yes | |
| Recirculation with Manual Demand Control (R-DRmc) | 1.75* | Yes | |
| Recirculation with Motion Sensor Demand Control (R-DRsc) | 2.60* | Yes | |
| **Optional Cases: HERS Inspection Required** | | | |
| Pipe Insulation (PIC-H) | 0.85 | Yes | Yes |
| Central Parallel Piping with 5' maximum length (PP-H) | 1.00 | Yes | |
| Compact Design (CHWDS-H) | 0.70 | Yes | |
| Recirculation with Manual Demand Control (R-DRmc-H) | 1.60* | Yes | |
| Recirculation with Motion Sensor Demand Control (RDRsc-H) | 2.40* | Yes | |
| | | | |
| *Recirculation DSMs reflect impact of reduced hot water consumption associated with recirculation systems. | | | |

## B4.2. Cold Water Inlet Temperature

The water heater inlet temperature is assumed to vary daily with the following relationship defined by the data included in the climate zone weather files. For each day of the year, $T_{inlet}$ will be calculated as follows:

$$T_{inlet} = T_{ground} \times 0.65 + T_{avg31} \times 0.35 \qquad \textit{Equation 9}$$

*where*

$T_{avg31}$ = Outdoor dry-bulb temperature averaged over all hours of the previous 31 days (note for January days, weather data from December will be used)

$T_{ground}$ = Ground temperature (°F ) for current day of year, calculated using *Equation 10*.

For each day (θ = 1 TO 365)

$$T_{ground}(\theta) = TyrAve - 0.5 \times (TyrMax - TyrMin) \times COS(2 \times \pi \times ((\theta - 1)/PB) - PO - PHI) \times GM \qquad \textit{Equation 10}$$

*where*

TyrAve = average annual temperature, °F

| | | |
|---|---|---|
| TyrMin | = | the lowest average monthly temperature, °F |
| TyrMax | = | the highest average monthly temperature, °F |
| PB | = | 365 |
| PO | = | 0.6 |
| DIF | = | 0.025 $ft^2$/hr |
| BETA | = | SQR(π/(DIF*PB*24))*10 |
| XB | = | EXP(-BETA) |
| CB | = | COS( BETA) |
| SB | = | SIN( BETA) |
| GM | = | SQR((XB*XB - 2.*XB*CB + 1)/(2.*BETA*BETA)) |
| PHI | = | ATN((1.-XB*(CB+SB)) / (1.-XB*(CB-SB))) |

## B5. Hourly Distribution Loss for Central Water Heating Systems

This section is applicable to the DHW system type 3 and 4, as defined in B1. The distribution losses accounted for in the distribution loss multiplier (DLM), *Equation 5*, reflect distribution heat loss within each individual dwelling unit. Additional distribution losses occur outside dwelling units and they include losses from recirculation loop pipes and branch piping feeding individual dwelling units. The hourly values of these losses, HRDL, shall be calculated according to *Equation 11*. Compliance software shall provide input for specifying recirculation system designs and controls according to the following algorithms.

$$HRDL_k = NLoop_k \times HRLL_k + HRBL_k \qquad \textit{Equation 11}$$

*where*

$HRDL_k$= Hourly central system distribution loss for $k^{th}$ system (Btu).

$HRLL_k$= Hourly recirculation loop pipe heat loss (Btu). This component is only applicable to system type 4, see *Equation 12*

$HRBL_k$= Hourly recirculation branch pipe heat loss (Btu), see *Equation 20*

$NLoop_k$= Number of recirculation loops in water heating system k; this component is only applicable to system type 4, see Section 4.3

A recirculation loop usually includes multiple pipe sections, not necessarily having the same diameter, that are exposed to different ambient conditions. The compliance software shall provide input entries for up to six pipe sections with three sections for supply piping and three sections for return piping for users to describe the configurations of the recirculation loop. For each of the six pipe sections, input entries shall include pipe diameter (inch), pipe length (ft), and ambient conditions. Ambient condition input shall include three options: outside air, underground,

conditioned or semi-conditioned air. Modeling rules for dealing with recirculation loop designs are provided in Section 4.3.

Outside air includes crawl spaces, unconditioned garages, unconditioned equipment rooms, as well as actual outside air. Solar radiation gains are not included in the calculation because the impact of radiation gains is relatively minimal compared to other effects. Additionally, the differences in solar gains for the various conditions (e.g., extra insulation vs. minimum insulation) are even less significant.

The ground condition includes any portion of the distribution piping that is underground, including that in or under a slab. Insulation in contact with the ground must meet all the requirements of Section 150.0(j), Part 6, of Title 24.

The losses to conditioned or semi-conditioned air include losses from any distribution system piping that is in an attic space, within walls (interior, exterior or between conditioned and unconditioned spaces), within chases on the interior of the building, or within horizontal spaces between or above conditioned spaces. It does not include the pipes within the residence. The distribution piping stops at the point where it first meets the boundaries of the dwelling unit.

## B5.1. Hourly Recirculation Loop Pipe Heat Loss Calculation

Hourly recirculation loop pipe heat loss ($HRLL_k$) is the hourly heat loss from all six pipe sections. There are two pipe heat loss modes, pipe heat loss with non-zero water flow (PLWF) and pipe heat loss without hot water flow (PLCD). The latter happens when the recirculation pump is turned off by a control system and there are no hot water draw flows, such as in recirculation return pipes.

Compliance software shall provide four options of recirculation system controls listed in Table B-2. A proposed design shall select a control type from one of the four options. The standard design shall use demand control.

***Table B-2. Recirculation Loop Supply Temperature and Pump Operation Schedule***

| Hour | No Control | | Demand Control | | Temperature Modulation | | Temperature Modulation with Continuous Monitoring | |
|---|---|---|---|---|---|---|---|---|
| | $T_{in,1}$ (°F) | $SCH_{k,m}$ | $T_{in,1}$ (°F) | $SCH_{k,m}$ | $T_{in,1}$ (°F) | $SCH_{k,m}$ | $T_{in,1}$ (°F) | $SCH_{k,m}$ |
| 1 | 130 | 1 | 130 | 0.2 | 120 | 1 | 115 | 1 |
| 2 | 130 | 1 | 130 | 0.2 | 120 | 1 | 115 | 1 |
| 3 | 130 | 1 | 130 | 0.2 | 120 | 1 | 115 | 1 |
| 4 | 130 | 1 | 130 | 0.2 | 120 | 1 | 115 | 1 |
| 5 | 130 | 1 | 130 | 0.2 | 120 | 1 | 115 | 1 |
| 6 | 130 | 1 | 130 | 0.2 | 125 | 1 | 120 | 1 |
| 7 | 130 | 1 | 130 | 0.2 | 130 | 1 | 125 | 1 |
| 8 | 130 | 1 | 130 | 0.2 | 130 | 1 | 125 | 1 |
| 9 | 130 | 1 | 130 | 0.2 | 130 | 1 | 125 | 1 |
| 10 | 130 | 1 | 130 | 0.2 | 130 | 1 | 125 | 1 |
| 11 | 130 | 1 | 130 | 0.2 | 130 | 1 | 125 | 1 |
| 12 | 130 | 1 | 130 | 0.2 | 130 | 1 | 125 | 1 |
| 13 | 130 | 1 | 130 | 0.2 | 130 | 1 | 125 | 1 |
| 14 | 130 | 1 | 130 | 0.2 | 130 | 1 | 125 | 1 |
| 15 | 130 | 1 | 130 | 0.2 | 130 | 1 | 125 | 1 |
| 16 | 130 | 1 | 130 | 0.2 | 130 | 1 | 125 | 1 |
| 17 | 130 | 1 | 130 | 0.2 | 130 | 1 | 125 | 1 |
| 18 | 130 | 1 | 130 | 0.2 | 130 | 1 | 125 | 1 |
| 19 | 130 | 1 | 130 | 0.2 | 130 | 1 | 125 | 1 |
| 20 | 130 | 1 | 130 | 0.2 | 130 | 1 | 125 | 1 |
| 21 | 130 | 1 | 130 | 0.2 | 130 | 1 | 125 | 1 |
| 22 | 130 | 1 | 130 | 0.2 | 130 | 1 | 125 | 1 |
| 23 | 130 | 1 | 130 | 0.2 | 130 | 1 | 125 | 1 |
| 24 | 130 | 1 | 130 | 0.2 | 125 | 1 | 120 | 1 |

Pipe heat loss modes are determined by recirculation control schedules and hot water draw schedules. For each pipe section, hourly pipe heat loss is the sum of heat loss from the two heat loss modes.

Hourly heat loss for the whole recirculation loop ($HRLL_k$) is the heat loss from all six pipe sections, according to the following equation:

$$HRLL_k = \sum_n \left[ PLWF_n + PLCD_n \right] \qquad \textit{Equation 12}$$

*where*

$PLWF_n$= Hourly pipe heat loss with non-zero water flow (Btu/hr), see *Equation 13*

$PLCD_n$= Hourly pipe heat loss without water flow (Btu/hr), see *Equation 18*

n= Recirculation pipe section index, 1through 6

$$PLWF_n = Flow_n \times \left(1 - f_{noflow,n}\right) \times \rho \times C_p \times \left(T_{n,in} - T_{n,out}\right)$$ *Equation 13*

*where*

$Flow_n$ = $Flow_{recirc} + Flow_{n,draw}$ (gph), assuming

$Flow_{n,draw}$ = Average hourly hot water draw flow (gph); for supply sections, n=1, 2, or 3, $Flow_{n,draw} = GPH_k/NLoop_k$; for return pipes, n=4, 5, and 6, $Flow_{n,draw} = 0$

$Flow_{recirc}$ = Hourly recirculation flow (gph) is assumed to be 360 gallons based on the assumption that the recirculation flow rate is 6 gpm

$f_{noflow,n}$ = Fraction of the hour for pipe section n to have zero water flow, see *Equation 14*

$\rho$ = Density of water, 8.345 (lb/gal)

$C_p$ = Specific heat of water, 1 (Btu/lb-°F)

$T_{n,in}$ = Input temperature of section n (°F); for the first section (n=1), $T_{1,in}$ shall be determined based on Table B-2. The control schedule of the proposed design shall be based on user input. The standard design is demand control. For other sections, input temperature is the same as the output temperature the proceeding pipe section, $T_{n,in} = T_{n-1,out}$

$T_{n,out}$ = Output temperature of section n (°F), see *Equation 15*

$$f_{noflow,n} = \left(1 - SCH_{k,m}\right) \times NoDraw_n$$ *Equation 14*

*where*

$NoDraw_n$ = Fraction of the hour that is assumed to have no hot water draw flow for pipe section n; $NoDraw_1 = 0.2$, $NoDraw_2 = 0.4$, $NoDraw_3 = 0.6$, $NoDraw_4 = NoDraw_5 = NoDraw_6 = 1$

$SCH_{k,m}$ = Recirculation pump operation schedule, representing the fraction of the hour that the recirculation pump is turned off, see Table B-2. $SCH_{k,m}$ for the proposed design shall be based on proposed recirculation system controls. Recirculation system control for the standard design is demand control.

$$T_{out,n} = T_{amb,n} + \left(T_{in,n} - T_{amb,n}\right) \times e^{-\frac{UA_n}{\rho C_p Flow_n}}$$ *Equation 15*

*where*

$T_{Amb,n}$ = Ambient temperature of section n (°F), which can be outside air, underground, conditioned, or semi-conditioned air. Outside air temperatures shall be the dry-bulb temperature from the weather file. Underground temperatures shall be obtained from *Equation 10*. Hourly conditioned air temperatures shall be the same as conditioned

space temperature. For the proposed design, $T_{amb,n}$ options shall be based on user input. The standard design assumes all pipes are in conditioned air.

$UA_n$ = Heat loss rate of section n (Btu/hr-°F), see *Equation 16*

$$UA_n = Len_n \times min\left(U_{bare,n}, f_{UA} \times U_{insul,n}\right) \qquad \textit{Equation 16}$$

*where*

$Len_n$ = Section n pipe length (ft); for the proposed design, use user input; for the standard design, see *Equation 27*

$U_{bare,n}$, $U_{insul,n}$ = Loss rates for bare (uninsulated) and insulated pipe (Btu/hr-ft-°F), evaluated using *Equation 17* with section-specific values, as follows:

- $Dia_n$ = Section n pipe nominal diameter (inch); for the proposed design, use user input; for the standard design, see *Equation 28*
- $Thick_n$ = Pipe insulation minimum thickness (inch) as defined in the Title 24 Section 120.3, TABLE 120.3-A for service hot water system
- $Cond_n$ = Insulation conductivity shall be assumed = 0.26 (Btu inch/h·sf·F)
- $h_n$ = Section n combined convective/radiant surface coefficient (Btu/hr-ft2-F) assumed = 1.5

$f_{UA}$ = Correction factor to reflect imperfect insulation, insulation material degradation over time, and additional heat transfer through connected branch pipes that is not reflected in branch loss calculation. It is assumed to be 2.0.

*Equation 17* defines general relationships used to calculate heat loss rates for both loop and branches using appropriate parameters.

$$Dia_o = Dia + 0.125$$

$$U_{bare} = h \times \pi \times Dia_o / 12$$

$$Dia_x = Dia_o + 2 \times Thick \qquad \textit{Equation 17}$$

$$U_{insul} = \frac{\pi}{\dfrac{ln\left(Dia_x / Dia_o\right)}{2 \times Cond/12} + \dfrac{12}{h \times Dia_x}}$$

*where*

Dia = Pipe nominal size (in)

$Dia_o$ = Pipe outside diameter (in)

$Dia_x$ = Pipe + insulation outside diameter (in)

Thick = Pipe insulation thickness (in)

Cond = Insulation conductivity (Btu in/hr-ft$^2$-$^o$F)

h = Combined convective/radiant surface coefficient (Btu/hr-ft$^2$-$^o$F)

Pipe heat loss without water flow shall be calculated according to the following equations:

$$PLCD_n = Vol_n \times \rho \times C_p \times \left(T_{n,start} - T_{n,end}\right) \qquad \textit{Equation 18}$$

*where*

$Vol_n$ = Volume of section n (gal) is calculated as $7.48 \times \pi \times \left(Dia_o/24\right)^2 \times Len_n$ where 7.48 is the volumetric unit conversion factor from cubic feet to gallons. Note that the volume of the pipe wall is included to approximate the heat capacity of the pipe material.

$T_{n,start}$ = Average pipe temperature ($^o$F) of pipe section n at the beginning of the hour. It is the average of $T_{n,in}$ and $T_{n,out}$ calculated according to *Equation 15* and associated procedures.

$T_{n,end}$ = Average pipe temperature ($^o$F) of pipe section n at the end of pipe cool down, see *Equation 19*

$$T_{n,end} = T_{amb,n} + \left(T_{n,start} - T_{amb,n}\right) \times e^{-\frac{UA_n \times f_{noflow,n}}{Vol_n \times \rho \times C_p}} \qquad \textit{Equation 19}$$

*Equation 19* calculates average pipe temperature after cooling down, so the pipe heat loss calculated by *Equation 18* is for pipe with zero flow for fraction $f_{noflow,n}$ of an hour. Recirculation pumps are usually turned off for less than an hour and there could be hot water draw flows in the pipe. As a result, recirculation pipes usually cool down for less than an hour. The factor $f_{noflow,\,n}$ calculated according *Equation 14* is used to reflect this effect in *Equation 19*.

## B5.2. Hourly Recirculation Branch Pipe Heat Loss Calculation

The proposed design and standard design shall use the same branch pipe heat loss assumptions. Branch pipe heat loss is made up of two components. First, pipe heat losses occur when hot water is in use (HBUL). Second, there could be losses associated with hot water waste (HBWL) when hot water was used to displace cold water in branch pipes and hot water is left in pipe to cool down after hot water draws and must be dumped down the drain.

The Total Hourly Branch Losses ($HRBL_k$) shall include both components and be calculated as:

$$HRBL_k = Nbranch_k \times \left(HBUL + HBWL\right) \qquad \textit{Equation 20}$$

*where*

HBUL = Hourly pipe loss for one branch when water is in use (Btu/hr), see *Equation 21*

HBWL = Hourly pipe loss for one branch due to hot water waste (Btu/hr), see *Equation 24*

$Nbranch_k$ = Number of branches in water heating system k, see *Equation 29*

The hourly branch pipe loss while water is flowing is calculated in the same way as recirculation pipe heat loss with non-zero water flow (PLWF) using the following equations:

$$HBUL = \left(\frac{GPH_k}{NBranch_k}\right) \times \rho \times C_p \times \left(T_{b,in} - T_{b,out}\right) \qquad \textit{Equation 21}$$

*where*

$T_{b,in}$ = Average branch input temperature (°F). It is assumed to be equal to the output temperature of the first recirculation loop section, $T_{1,out}$

$T_{b,out}$ = Average branch output temperature (°F), see *Equation 22*

$$T_{b,out} = T_{amb,b} + \left(T_{b,in} - T_{amb,b}\right) \times e^{-\frac{UA_b}{\rho \times C_p \times Flow_b}} \qquad \textit{Equation 22}$$

*where*

$T_{amb,b}$ = Branch pipe ambient temperature (°F) Branch pipes are assumed to be located in the conditioned or semi-conditioned air.

$UA_b$ = Branch pipe heat loss rate (Btu/hr-°F), see Equation 23

$Flow_b$ = Branch hot water flow rate during use (gal/hr). It is assumed to be 2 gpm or 120 gal/hr.

The branch pipe heat loss rate is

$$UA_b = Len_b \times U_{insul,b} \qquad \textit{Equation 23}$$

*where*

$Len_b$ = Branch pipe length (ft), see *Equation 31*

$U_{insul,b}$ = Loss rate for insulated pipe (Btu/hr-ft-°F), evaluated using *Equation 17* with branch-specific values, as follows:

- $Dia_b$ = Branch pipe diameter (inch), see *Equation 30*
- $Thick_b$ = Branch pipe insulation minimum thickness (inch) as defined in the Title 24 Section 120.3, TABLE 120.3-A for service hot water system.
- $Cond_b$ = Branch insulation conductivity, assumed = 0.26 Btu in/hr-ft²-°F

$h_b$ = Branch combined convective/radiant surface coefficient (Btu/hr-ft$^2$-$^o$F) assumed = 1.5

The hourly pipe loss for one branch due to hot water waste is calculated as follows:

$$HBWL = N_{waste} \times SCH_{waste,m} \times f_{vol} \times 7.48 \times \pi \times \left(\frac{Dia_b + 0.125}{24}\right)^2 \times Len_b \times \rho \times C_p \times (T_{b,in} - T_{inlet})$$

*Equation 24*

*where*

$N_{waste}$ = Number of times in a day for which water is dumped before use. This depends on the number of dwelling units served by a branch. Statistically, the number of times of hot water waste is wasted is inversely proportional to the number of units a branch serves, see *Equation 25.*

$SCH_{waste,m}$ = Hourly schedule of water waste, see Table B-3

$f_{vol}$ = The volume of hot water waste is more than just the volume of branch pipes, due to branch pipe heating, imperfect mixing, and user behaviors. This multiplier is applied to include these effects and is assumed to be 1.4.

$T_{in,b}$ = Average branch input temperature ($^o$F) is assumed to equal the output temperature of the first recirculation loop section, $T_{OUT,1}$

$T_{inlet}$ = The cold water inlet temperature ($^o$F) according to Section 3.3 Cold Water Inlet Temperature

$$N_{waste} = 19.84 \times e^{-0.544 \times Nunit_b}$$

*Equation 25*
*Method WH-BRWF*

*where*

$Nunit_b$= Number of dwelling units served by the branch, calculated using *Equation 26* (note that $Nunit_b$ is not necessarily integral).

$$Nunit_b = \frac{Nfloor}{2}$$

*Equation 26*
*Method WH-BRNU*

***Table B-3. Branch Water Waste Schedule***

| Hour | $SCH_{waste,m}$ |
|---|---|
| 1 | 0.01 |
| 2 | 0.02 |
| 3 | 0.05 |
| 4 | 0.22 |
| 5 | 0.25 |
| 6 | 0.22 |
| 7 | 0.06 |
| 8 | 0.01 |
| 9 | 0.01 |
| 10 | 0.01 |
| 11 | 0.01 |
| 12 | 0.01 |
| 13 | 0.01 |
| 14 | 0.01 |
| 15 | 0.01 |
| 16 | 0.01 |
| 17 | 0.01 |
| 18 | 0.01 |
| 19 | 0.01 |
| 20 | 0.01 |
| 21 | 0.01 |
| 22 | 0.01 |
| 23 | 0.01 |
| 24 | 0.01 |

## B5.3. Recirculation System Plumbing Designs

A recirculation system can have one or multiple recirculation loops. Each recirculation loop consists of many pipe sections, which are connected in sequence to form a loop. Each pipe section could have different pipe diameter, length, and location. The compliance software shall use six pipe sections, with three supply pipe sections and three return pipe sections, to represent a recirculation loop. When multiple recirculation loops exist, all recirculation loops are assumed to be identical. The compliance software shall provide default and standard recirculation system designs based on building geometry according to the procedures described in the following sections. The default design reflects typical recirculation loop design practices. The standards design is based on one or two loops and is used to set recirculation loop heat loss budget.

The first step of establishing recirculation system designs is to determine the number of recirculation loops, $Nloop_k$, in water heating system k. The standard design has one recirculation loop, $Nloop_k$ =1, when Nunit <= 8, or two recirculation loops, $Nloop_k$ =2 for buildings with Nunit > 8. The proposed design is allowed to specify more than one loop only if the design is verified by a HERS rater. Otherwise, the proposed design can only be specified to have one recirculation loop.

The standard and default recirculation loop designs are based on characteristics of the proposed building. There could be many possibilities of building shapes and dwelling unit configurations, which would determine recirculation loop pipe routings. Without requiring users to provide detailed

dwelling unit configuration information, the compliance software shall assume the proposed buildings to have same dwelling units on each floor and each floor to have a corridor with dwelling units on both sides. Recirculation loops start from the mechanical room (located on the top floor), go vertically down to the middle floor, loop horizontally in the corridor ceiling to reach the dwelling units on both ends of the building, then go vertically up back to the mechanical room. At each dwelling unit on the middle floor, vertical branch pipes, connected to the recirculation loop supply pipe, are used to provide hot water connection to dwelling units on other floors above and below.

Both the standard and default recirculation loop designs are assumed to have equal length of supply sections and return sections. The first section is from the mechanical room to the middle floor. The second section serves first half branches connected to the loop and the third section serves the rest of the branches. The first and second sections have the same pipe diameter. Pipe size for the third section is reduced since less dwelling units are served. Return sections match with the corresponding supply pipes in pipe length and location. All return sections have the same diameter. For both the standard and default designs, mechanical room is optimally located so that only vertical piping is needed between the mechanical room and the recirculation pipes located on the middle floor. Pipe sizes are determined based on the number of dwelling units served by the loop, following the 2009 Uniform Plumbing Code (UPC) pipe sizing guidelines. The detailed recirculation loop configurations are calculated as follows:

Pipe Length in the mechanical room (ft): $L_{mech}=8$

Height of each floor (ft): $H_{floor}$=user input floor-to-floor height (ft)

Length of each dwelling unit (ft): $L_{unit} = \sqrt{CFAU_k}$ (see *Equation 7*)

Length of recirculation pipe sections (ft):

$$Len_1 = Len_6 = L_{mech} + H_{floor} \times \frac{Nfloor}{2}$$

$$Len_2 = Len_3 = Len_4 = Len_5 = L_{unit} \times \frac{Nunit_k}{4 \times Nloop_k \times Nfloor}$$

*Equation 27*
*Method WH-LOOPLEN*

Pipe diameters for recirculation loop supply sections depend on the number of dwelling units being served and return section diameters depend only on building type, as follows:

$Dia_1$, $Dia_2$, and $Dia_3$: derived from Table B-4 based on $Nunit_1$, $Nunit_2$, and $Nunit_3$

$Dia_4 = Dia_5 = Dia_6 = 0.75$ in for low-rise multi-family building and hotel/motel less than four stories

$Dia_4 = Dia_5 = Dia_6 = 1.0$ in for high-rise multi-family and hotel/motel more than three stories

*Equation 28*
*Method WH-LOOPSZ*

*where*

$Nunit_1$ = Number of dwelling unit served by the loop section 1 = $\frac{Nunit_k}{Nloop_k}$

$Nunit_2 = Nunit_1$

$Nunit_3 = \frac{Nunit_1}{2}$

Note that Nunit values are not necessarily integers.

Branch pipe parameters include number of branches, branch length, and branch diameter. The number of branches in water heating system k is calculated as (note: not necessarily an integer):

$$Nbranch_k = \frac{Nunit_k}{Nunit_b}$$

*Equation 29*
*Method WH-BRN*

The branch pipe diameter shall be determined as follows:

$Dia_b$: derived from Table B-4 based on $Nunit_b$

*Equation 30*
*Method WH-BRSZ*

The branch length includes the vertical rise based on the number of floors in the building plus four feet of pipe to connect the branch to the recirculation loop.

$$Len_b = 4 + H_{floor} \times Nfloor / 2$$

*Equation 31*
*Method WH-BRLEN*

Proposed designs shall use the same branch configurations as those in the standard design. Therefore, compliance software does not need to collect branch design information.

***Table B-4. Pipe Size Schedule***

| Number of dwelling units served $NUnit_n$ or $NUnit_b$ | Loop pipe nominal size $Dia_n$ in | Branch pipe nominal size $Dia_b$ in |
|---|---|---|
| < 2 | 1.5 | 1 |
| 2 ≤ N < 8 | 1.5 | |
| 8 ≤ N < 21 | 2 | |
| 21 ≤ N < 42 | 2.5 | |
| 42 ≤ N < 68 | 3 | |
| 68 ≤ N < 101 | 3.5 | |
| 101 ≤ N < 145 | 4 | |
| 145 ≤ N < 198 | 5 | |
| N >= 198 | 6 | |

## B6. High Rise Residential Buildings, Hotels and Motels

Simulations for high-rise residential buildings, hotels, and motels shall follow all the rules for central or individual water heating with the following exceptions:

- For central systems which do not use recirculation but use electric trace heaters the program shall assume equivalency between the recirculation system and the electric trace heaters.
- For individual water heater systems which use electric trace heating instead of gas, the program shall assume equivalency.

## B7. Energy Use of Individual Water Heaters

Once the hourly adjusted recovery load is determined for each water heater, the energy use for each water heater is calculated as described below and summed.

### B7.1. Small/Consumer Gas or Oil Storage Water Heaters

The hourly energy use of storage gas or oil storage water heaters is given by the following equation.

$$WHEU_j = \frac{HARL_j \times HPAF_j}{LDEF_j} \qquad \textit{Equation 32}$$

*where*

$WHEU_j$ = Hourly energy use of the water heater (Btu for fuel or kWh for electric); *Equation 33* provides a value in units of Btu. For electric water heaters, the calculation result needs to be converted to the unit of kWh by dividing 3413 Btu/kWh.

$HARL_j$ = Hourly adjusted recovery load (Btu)

$HPAF_j$ = 1 for all non-heat pump water heaters

$LDEF_j$ = The annual load dependent energy factor (LDEF) is given by *Equation 33*. This equation adjusts the nominal EF rating for storage water heaters for different load conditions.

$$LDEF_j = \min\left[LDEF_{\max}, \max\left(LDEF_{\min}, \ln\left(\frac{AAHARL_j \times 24}{1000}\right)\left(a \times EF_j + b\right) + \left(c \times EF_j + d\right)\right)\right] \qquad \textit{Equation 34}$$

*where*

a,b,c,d = Coefficients from the table below based on the water heater type

**Table B-5. LDEF Coefficients**

| Coefficient | Storage Gas |
|---|---|
| a | -0.098311 |
| b | 0.240182 |
| c | 1.356491 |
| d | -0.872446 |
| $LDEF_{min}$ | .1 |
| $LDEF_{max}$ | .90 |

$AAHARL_j$ = Annual average hourly adjusted load (Btu) = $\frac{1}{8760}\sum_{1}^{8760} HARL_j$ ; calculation of $AAHARL_j$ requires a preliminary annual simulation that sums $HARL_j$ values for each hour.

$EF_j$ = Energy factor of the water heater (unitless). This is based on the DOE test procedure. EF for storage gas water heaters with volume less than 20 gallons must be assumed to be 0.58 unless the manufacturer has voluntarily reported an actual EF to the California Energy Commission.

## B7.2. Small/Consumer Gas or Oil Instantaneous Water Heater

The hourly energy use for instantaneous gas or oil water heaters is given by *Equation* 34, where the nominal rating is multiplied by 0.92 to reflect the impacts of heat exchanger cycling under real world load patterns.

$$WHEU_j = \frac{HARL_j}{EF_j \times 0.92} \qquad \textit{Equation 35}$$

*where*

$WHEU_j$ = Hourly fuel energy use of the water heater (Btu)

$HARL_j$ = Hourly adjusted recovery load

$EF_j$ = Energy factor from the DOE test procedure (unitless) taken from manufacturers' literature or from the CEC Appliance Database

0.92 = Efficiency adjustment factor

## B7.3. Small/Consumer Electric Instantaneous Water Heater

The hourly energy use for instantaneous electric water heaters is given by the following equation.

$$WHEU_{j,elec} = \frac{HARL_j}{EF_j \cdot 0.92 \cdot 3413} \qquad \textit{Equation 36}$$

*where*

$WHEU_{j,elec}$ = Hourly electric energy use of the water heater (kWh)

$HARL_j$ = Hourly adjusted recovery load (Btu)

$EF_j$ = Energy factor from DOE test procedure (unitless)

0.92 = Adjustment factor to adjust for overall performance

3413 = Unit conversion factor (Btu/kWh)

## B7.4. Mini-Tank Electric Water Heater

Mini-tank electric heaters are occasionally used with gas tankless water heaters to mitigate hot water delivery problems related to temperature fluctuations that may occur between draws. If mini-tank electric heaters are installed, the installed units must be listed in the CEC Appliance Database and their reported standby loss (in Watts) will be modeled to occur each hour of the year. (If the unit is not listed in the CEC Appliance Database, a standby power consumption of 35 W should be assumed.)

$$WHEU_{j,elec} = MTSBL_j / 1000 \qquad \textit{Equation 37}$$

*where*

$WHEU_{j,elec}$ = Hourly standby electrical energy use of mini-tank electric water heaters (kWh)

$MTSBL_j$ = Mini-tank standby power (W) for tank j (if not listed in CEC Appliance directory, assume 35 W)

## B7.5. Large/Commercial Gas or Oil Storage Water Heater

Energy use for large storage gas is determined by the following equations. Note: large storage gas water heaters are defined as any gas storage water heater with a minimum input rate of 75,000 Btu/h.

$$WHEU_j = \frac{HARL_j}{EFF_j} + SBL_j \qquad \textit{Equation 38}$$

*where*

$WHEU_j$ = Hourly fuel energy use of the water heater (Btu)

$HARL_j$ = Hourly adjusted recovery load (Btu)

$SBL_j$ = Total Standby Loss (Btu/hr). Obtain from CEC Appliance Database or from AHRI certification database. This value includes tank losses and pilot energy. If standby rating is not available from either of the two databases, it shall be calculated as per Table F-2 of the 2015 Appliance Efficiency Regulations, as follows:

$SBL = Q/800 + 110\ (V)^{1/2}$, where Q is the input rating in Btu/hour, and V is the tank volume in gallons.

$EFF_j$ = Efficiency (fraction, not %). Obtained from CEC Appliance Database or from manufacturer's literature. These products may be rated as a recovery efficiency, thermal efficiency or AFUE.

## B7.6. Large/Commercial Instantaneous, Indirect Gas, and Hot Water Supply Boilers

Energy use for these types of water heaters is given as follows:

$$WHEU_j = \frac{HARL_j}{EFF_j \times 0.92} + PILOT_j \qquad \textit{Equation 39}$$

*where*

$WHEU_j$ = Hourly fuel energy use of the water heater (Btu), adjusted for tank insulation.

$HARL_j$ = Hourly adjusted recovery load. For independent hot water storage tank(s) substitute $HARL_j$ from Section B3.

$EFF_j$ = Efficiency (fraction, not %) to be taken from CEC Appliance Database or from manufacturers literature. These products may be rated as a recovery efficiency, thermal efficiency or AFUE.

$PILOT_j$ = Pilot light energy (Btu/h) for large instantaneous. For large instantaneous water heaters, and hot water supply boilers with efficiency less than 89 percent assume the default is 750 Btu/hr if no information is provided in manufacturer's literature or CEC Appliance Database.

0.92 = Adjustment factor used when system is not supplying a storage system.

## B7.7. Small/Consumer Storage Electric or Heat Pump Water Heaters

Energy use for small electric water heaters is calculated as described in the HPWHsim Project Report (Ecotope, 2015 and 2016). The HPWH model uses a detailed, physically based, multi-node model that operates on a one-minute time step implemented using a suitable loop at the time-step level within CSE. Tank heat losses and heat pump source temperatures are linked to the CSE zone heat balance as appropriate. Thus, for example, the modeled air temperature of a garage containing a heat pump water heater will reflect the heat extracted.

HPWH can model three classes of equipment:

- Specific air-source heat pump water heaters identified by manufacturer and model. These units have been tested by Ecotope and measured parameters are built into the HPWH code.
- Generic air-source heat pump water heaters, characterized by EF and tank volume. This approach provides compliance flexibility. The performance characteristics of the generic model are tuned to use somewhat more energy than any specific unit across a realistic range of EF values.

- Electric resistance water heaters, characterized by EF, tank volume, and resistance element power.

Several issues arise from integration of a detailed, short time step model into an hourly framework. HPWH is driven by water draw quantities, not energy requirements. Thus to approximate central system distribution and unfired tank losses, fictitious draws are added to the scheduled water uses, as follows:

$$V_{j,t} = \frac{VS_{k,t} + \dfrac{HRDL_k + \sum_{1}^{NL_k} HJL_l}{60 \times 8.345 \times \left(t_s - t_{inlet}\right)}}{NWH_k} \qquad \textit{Equation 40}$$

*where*

$V_{j,t}$ = Hot water draw (gal) on j[th] water heater for minute t

Another issue is that the HPWH hot water output temperature varies based on factors like control hysteresis and tank mixing. For compliance applications, it is required that all system alternatives deliver the same energy. To address this, the HPWH tank setup point is modeled at 125 °F and delivered water is tempered to $t_s$. If the HPWH output temperature is above $t_s$,it is assumed that inlet water is mixed with it (thus reducing $V_{i,t}$). If the output temperature is below $t_s$, sufficient electrical resistance heating is supplied to bring the temperature up to $t_s$ (preventing under capacity from being exploited as a compliance advantage).

## B7.8. Jacket Loss

The hourly jacket loss for the l[th] unfired tank or indirectly fired storage tank in the k[th] system is calculated as:

$$HJL_l = \frac{TSA_l \times \Delta TS}{RTI_l + REI_l} + FTL_l \qquad \textit{Equation 41}$$

*where*

$HJL_l$ = The tank surface losses of the l[th] unfired tank of the k[th] system

$TSA_l$ = Tank surface area (ft$^2$), see *Equation 41*

$\Delta TS$ = Temperature difference between ambient surrounding tank and hot water supply temperature (ºF). Hot water supply temperature shall be 124ºF. For tanks located inside conditioned space use 75ºF for the ambient temperature. For tanks located in outside conditions use hourly dry bulb temperature ambient.

$FTL_l$ = Fitting losses; a constant 61.4 Btu/h

$REI_l$ = R-value of exterior insulating wrap; no less than R-12 is required

$RTI_l$ = R-value of insulation internal to water heater; assume 0 without documentation

Tank surface area (TSA) is used to calculate the hourly jacket loss (HJL) for unfired or indirectly fired tanks. TSA is given in the following equation as a function of the tank volume.

$$TSA_l = \left(1.254 \times VOL_l^{0.33} + .531\right)^2 \qquad \textit{Equation 42}$$

*where*

$VOL_l$ = Tank capacity (gal)

## B7.9. Electricity Use for Circulation Pumping

For single-family recirculation systems, hourly pumping energy is fixed as shown in Table B-6.

***Table B-6. Single Family Recirculation Energy Use (kWh) by Hour of Day***

| Hour | Non-Demand Controlled Recirculation | Demand-controlled Recirculation |
|---|---|---|
| 1 | 0.040 | 0.0010 |
| 2 | 0.040 | 0.0005 |
| 3 | 0.040 | 0.0006 |
| 4 | 0.040 | 0.0006 |
| 5 | 0.040 | 0.0012 |
| 6 | 0.040 | 0.0024 |
| 7 | 0.040 | 0.0045 |
| 8 | 0.040 | 0.0057 |
| 9 | 0.040 | 0.0054 |
| 10 | 0.040 | 0.0045 |
| 11 | 0.040 | 0.0037 |
| 12 | 0.040 | 0.0028 |
| 13 | 0.040 | 0.0025 |
| 14 | 0.040 | 0.0023 |
| 15 | 0.040 | 0.0021 |
| 16 | 0.040 | 0.0019 |
| 17 | 0.040 | 0.0028 |
| 18 | 0.040 | 0.0032 |
| 19 | 0.040 | 0.0033 |
| 20 | 0.040 | 0.0031 |
| 21 | 0.040 | 0.0027 |
| 22 | 0.040 | 0.0025 |
| 23 | 0.040 | 0.0023 |
| 24 | 0.040 | 0.0015 |
| Annual Total | 350 | 23 |

Multifamily recirculation systems typically have larger pump sizes, and therefore electrical energy use is calculated based on the installed pump size. The hourly recirculation pump electricity use (HEUP) is calculated by the hourly pumping schedule and the power of the pump motor as in the following equation.

$$HEUP_k = \frac{0.746 \times PUMP_k \times SCH_{k,m,}}{\eta_k}$$ *Equation 43*

*where*

$HEUP_k$ = Hourly electricity use for the circulation pump (kWh)

$PUMP_k$ = Pump brake horsepower (bhp)

$\eta_k$ = Pump motor efficiency

$SCH_{k,m}$ = Operating schedule of the circulation pump, see Table 0-2. The operating schedule for the proposed design shall be based on user input control method. The standard design operation schedule is demand control.

## B7.10. Conversion between Energy Factor and Uniform Energy Factor

For consumer water heaters and residential-duty commercial water heaters, Energy Factor (EF) can be converted into Uniform Energy Factor (UEF) and vice versa. The mathematical conversion factors translate values determined under the energy factor, thermal efficiency, and standby loss test procedures for consumer water heaters and certain commercial waters heaters to those determined under the new UEF test procedure. The mathematical conversion factors are detailed in the DOE final rule under docket EERE-2015-BT-TP-0007.

For the remainder of the *2016 Building Energy Efficiency Standards* cycle, UEF input is converted to EF by CBECC using these methods.

# APPENDIX C – PLUG LOADS AND LIGHTING MODELING

## 1.1 Appliances, Miscellaneous Energy Use and Internal Gains

Full details of the assumptions for lighting and appliance loads are found in the Codes and Standards Enhancement Initiative (CASE) Plug Loads and Lighting Modeling (Rubin 2016, see Appendix D).

### 1.1.1 Background

Rulesets for all plug loads (including appliances and miscellaneous electric loads (MELs)) and lighting loads were updated in 2016. The CASE report describes the methodology, data sources, and assumptions used to develop the rulesets. The updated methodology replaces the rulesets from the 2013 *Residential Alternative Calculation Method (ACM) Reference Manual* (ACM Reference Manual), which in turn referenced the 2008 California Home Energy Rating System (HERS) Technical Manual.

The rulesets were modified to reflect efficiency levels assuming 2017 federal code baseline or 2017 projected market average performance, depending on whether or not a product is regulated by federal energy efficiency standards. Miscellaneous loads were disaggregated so that the three largest loads in this group—televisions, set-top boxes, and computers and monitors—are modeled individually. The remaining miscellaneous loads are modeled in aggregate. Garage lighting is also disaggregated from interior lighting. Assumptions about how energy use scales with building size were updated for all plug load and lighting end uses.

Updated load profiles were proposed for the majority of the modeled plug load and lighting end uses. The proposed updates include revisions to both the hourly schedules and seasonal multipliers. The updated load profiles are based on the water heating models described in section 2.9 of the ACM Reference Manual for the applicable end uses and otherwise on recent submetering studies.

### 1.1.2 Approach

Rulesets for all modeled end uses reflect the estimated energy consumption of those devices in new homes built during the 2016 Title 24 Code Cycle. The plug load rulesets estimate annual energy consumption (AEC) as a function of number of bedrooms (BRperUnit) and the lighting rulesets estimate AEC as a function of conditioned floor area (CFAperUnit). The relationship between AEC and BRperUnit for dishwashers, clothes washers, and clothes dryers was based on the usage assumptions in the water heating model. The relationship between all other plug load AEC and BRperUnit was generally derived from the 2009 Residential Appliance Saturation Survey (RASS), through a statistical and engineering analysis that applied modern efficiency assumptions to estimate what the AEC of plug loads within homes included in the 2009 RASS would be if they were built during the 2016 Title 24 code cycle. The relationship between lighting AEC and CFAperUnit was derived using a similar analysis completed on the RASS data but using data from the 2012 California Lighting and Appliance Saturation Survey.

With additional user inputs, the default AEC equations for primary refrigerators, clothes washers, and clothes dryers can be modified to reflect the efficiency of the devices that are actually installed in the building. That is, the modeled energy use can be adjusted downward if more efficient devices are installed (the software tool can also adjust energy use upward if devices are less efficient).

Updated load profiles are derived from the following data sources:

- **Dishwashers, clothes washers, and clothes dryers:** updated to be consistent with the usage patterns assumed by water heating models described in section 2.9 of the ACM Reference Manual.
- **Ovens, cooktops, and televisions:** based on data from the Phased Deep Retrofit (PDR) study conducted by the Florida Solar Energy Center (FSEC), which submetered 60 Florida homes in 2012.
- **Set-top boxes, computers, and monitors:** based on the Northwest Energy Efficiency Alliance (NEEA) Residential Building Stock Assessment (RBSA), released in 2014. This study monitored 100 homes in the Pacific Northwest over the course of one year, submetering major end uses at 15 minute intervals.
- **Exterior lighting:** the proposed hourly schedule for exterior lighting is derived from the NEEA RBSA light logging data; the proposed exterior lighting seasonal multipliers are no longer constant, but instead equivalent to the interior and garage seasonal multipliers.

Load profiles for interior lighting, garage lighting, and residual MELs were not updated in 2016. The current hourly schedules for interior lighting are based on the 1999 Heschong Mahone Group (HMG) study "Lighting Efficiency Technology Report: California Baseline." The current hourly schedule for residual MELs is derived from the 2008 Building America House Simulation Protocol, which in turn relied on data from a 1989 Pacific Northwest submetering study conducted by the End-Use Load and Consumer Assessment Program (ELCAP).

Refrigerators and freezers use PDR data to adjust estimated energy use on an hourly basis depending on the modeled indoor temperature (using the Title 24 compliance software) in the space where the refrigerator is installed.

### 1.1.3 Problems

The plug load and lighting rulesets have some limitations. The rulesets generally do not account for differences in energy use patterns between single-family and multi-family housing. For example, they do not account for the energy use of laundry equipment in multi-family residences that is installed in common areas—only laundry equipment in the dwelling units.

The plug load and lighting rulesets were developed to apply to new homes built during the 2016 Title 24 Code Cycle, and thus should not be used for estimating energy use for existing homes.

### 1.1.4 Inputs

#### 1.1.4.1 *AEC Inputs and Algorithms*

Table 1 summarizes the user inputs that determine the plug load and lighting annual energy consumption (AEC) estimates. The variable 'BRperUnit' refers to the number of bedrooms in a single-family home or the number of bedrooms in each dwelling unit of a multi-family building. Similarly, 'CFAperUnit' refers to the conditioned floor area per dwelling unit. AEC equations are to be applied to each dwelling unit within a multi-family building, not the building as a whole. Users also specify the zone where certain major appliances are located; however, this affects the modeled internal gains from equipment and lighting, not their estimated energy use of the plug load or lighting load and is therefore not included in the table below. The Optional inputs are not implemented in CBECC-Res 2016.2, but may be allowed in future releases.

**Table 1: User Inputs Affecting Estimated Plug Load and Lighting Energy Use**

| End Use | User Inputs that Determine Estimated Energy Use | Notes |
|---|---|---|
| Primary Refrigerator/ Freezer | - BRperUnit<br>- Optional: rated annual kWh usage from the Energy Guide label of the installed device | - Default kWh can be overridden with the rated annual kWh usage input on the Energy Guide label; however, there is a maximum allowable kWh credit dependent on BRperUnit.<br>- Energy use adjusted on an hourly basis depending on the indoor temperature in the kitchen simulated in the software. |
| Non-Primary Refrigerators and Separate Freezers | - BRperUnit<br>- Single-family or multi-family housing | - Assumed to be installed in the garage in new, single-family homes.<br>- Assumed to be absent in multi-family dwelling units. |
| Dishwasher | - BRperUnit<br>- Presence of device<br>- Single-family or multi-family | - Ruleset estimates machine energy use only.<br>- Energy use is only included if user indicates the device will be present.<br>- Assumed different usage patterns in single family and multi-family when developing algorithms. |
| Clothes Washer | - BRperUnit<br>- Presence of device<br>- Single-family or multi-family<br>- Optional: whether installed device will comply with the 2015 federal efficiency standards (credit for installing new or nearly-new device) | - Ruleset estimates machine energy use only.<br>- Energy use is only included if user indicates the device will be present.<br>- Assumed different usage patterns in single family and multi-family when developing algorithms.<br>- Default energy use can be reduced if the user specifies the device will meets the 2015 federal standard, which can be determined by looking up the model on the California Appliance Efficiency Database. |

| End Use | User Inputs that Determine Estimated Energy Use | Notes |
|---|---|---|
| Clothes Dryer | - BRperUnit<br>- Presence of device<br>- Fuel type (natural gas, propane, or electric)<br>- Single-family or multi-family<br>- Optional: percent remaining moisture content (RMC) of the clothes washer | - Energy use is only included if user indicates the device will be present.<br>- User can select fuel type. If user indicates natural gas is available at the site (see Section 2.2.10 of RACM), then the default fuel type is natural gas. If user indicates that natural gas is not available at the site then the default fuel type is electric. User cannot select natural gas as the fuel type if natural gas is not available at the site.<br>- Default energy use can be reduced if the user specifies that the installed clothes washer has a rated RMC of less than 50 percent. |
| Oven | - BRperUnit<br>- Presence of device<br>- Fuel type (natural gas, propane, or electric) | - Energy use is only included if user indicates the device will be present.<br>- User can select fuel type, but default assumption is natural gas if user indicates that natural gas is available on-site and electric if user indicates natural gas is not available on-site |
| Cooktop | | |
| Televisions<br>Set-Top Boxes<br>Computers and Monitors<br>Residual MELs | - BRperUnit | |
| Interior Lighting<br>Exterior Lighting | - CFAperUnit | |
| Garage Lighting | - CFAperUnit<br>- Presence of garage | - Energy use is only included if user indicates there is a garage present.<br>- Garage lighting is assigned to multi-family buildings if there is at least once garage present.<br>- Carport lighting is covered under the exterior lighting ruleset. |

Table 2 summarizes the proposed AEC algorithms for plug load and lighting. These linear equations take the following general form where the homes size metric is BRperUnit for plug loads and CFAperUnit for lighting:

$$y = mx + b$$

Where: y = Estimated AEC measured in kWh/yr or therms/yr
m = how AEC changes with home size
x = home size as measured in BRperUnit for plug loads or CFAperUnit for lighting
b = minimum energy use (energy use at y-intercept)

BR-based equations are capped at 7 bedrooms, meaning that units with eight or more bedrooms have the same estimated AEC as a 7-bedroom unit. CFA-based equations are capped at 4,150 square feet. For those end uses that list 'presence of device' as a user input in Table 2, the AEC equation is only applied if the device is present. Similarly, for the AEC equations for end uses that can be gas or electric are only applied according to the user-specified fuel type. Gas algorithms apply to devices that use natural gas or propane.

**Table 2: Algorithms for Plug Load and Lighting Annual Energy Use**

| End Use | Standard Design Fuel Type | kWh or therms | Intercept | Slope | Per-Unit BR or CFA |
|---|---|---|---|---|---|
| Primary Refrigerator/Freezer | Electricity | kWh | 454 | 37.0 | BR |
| Non-Primary Refrigerators and Separate Freezers (Single-Family only) | Electricity | kWh | 0 | 71.0 | BR |
| Oven | Electricity | kWh | 138 | 16 | BR |
| Oven | Gas | therms | 6.0 | 0.95 | BR |
| Oven | Gas | kWh | 41 | 4.79 | BR |
| Cooktop | Electricity | kWh | 84 | 5.68 | BR |
| Cooktop | Gas | therms | 5.0 | 0.30 | BR |
| Cooktop | Gas | kWh | 0 | 0 | BR |
| Televisions | Electricity | kWh | 265 | 31.8 | BR |
| Set-Top Boxes | Electricity | kWh | 76 | 59.4 | BR |
| Computers and Monitors | Electricity | kWh | 79 | 55.4 | BR |
| Residual MELs | Electricity | kWh | 672 | 235 | BR |
| Interior Lighting | Electricity | kWh | 100 | 0.1775 | CFA |
| Exterior Lighting | Electricity | kWh | 8.0 | 0.0532 | CFA |
| Garage Lighting | Electricity | kWh | 20 | 0.0063 | CFA |

Table 3 and Table 4 summarize the AEC algorithms for dishwashers, clothes washers and clothes dryers. These rulesets only include machine energy use from dishwashers and clothes washers. Energy use for water heating is accounted for in the water heating model.

**Table 3: Single-Family Residence Algorithms for Dishwasher, Clothes Washer, and Clothes Dryer Annual Energy Use**

| BRperUnit | Dishwashers (kWh/yr) | Clothes Washers (kWh/yr) | Electric Clothes Dryers (kWh/yr) | Natural Gas Clothes Dryers | |
|---|---|---|---|---|---|
| | | | | Natural Gas Use (therms/yr) | Electricity Use (kWh/yr) |
| 0 | 83 | 84 | 634 | 22 | 32 |
| 1 | 83 | 84 | 634 | 22 | 32 |
| 2 | 91 | 85 | 636 | 22 | 32 |
| 3 | 100 | 99 | 748 | 26 | 37 |
| 4 | 99 | 101 | 758 | 27 | 38 |
| 5+ | 119 | 227 | 877 | 31 | 44 |

**Table 4: Multi-Family Dwelling Unit Algorithms for Dishwasher, Clothes Washer, and Clothes Dryer Annual Energy Use**

| BRperUnit | Dishwashers (kWh/yr) | Clothes Washers (kWh/yr) | Electric Clothes Dryer (kWh/yr) | Gas Clothes Dryers | |
|---|---|---|---|---|---|
| | | | | Natural Gas Usage (therms/yr) | Electricity Usage (kWh/yr) |
| 0 | 56 | 66 | 496 | 17 | 25 |
| 1 | 68 | 70 | 527 | 19 | 26 |
| 2 | 96 | 99 | 745 | 26 | 37 |
| 3 | 94 | 97 | 733 | 26 | 37 |
| 4 | 121 | 118 | 885 | 31 | 44 |
| 5+ | 114 | 107 | 805 | 28 | 40 |

#### 1.1.4.2 *AEC Algorithms for High-Efficiency Appliances*

As indicated in Table 5, if allowed in the software, users could override the default AEC rulesets for the primary refrigerator, clothes washer and clothes dryer if the software user has additional information about the device that will be installed.

For the primary refrigerator, the default AEC ruleset could be replaced with the rated AEC listed on the refrigerator's Energy Guide label. If using this option, the user will input AEC measured in kWh per year, and that value will replace the AEC value for the primary refrigerator calculated using the equation below. The default AEC of the primary refrigerator cannot be adjusted below a certain value, which is dependent on BRperUnit as described in the following equation:

$$MinPrimaryRefrigAEC\frac{kWh}{yr} = \left(8.4\frac{kWh}{BRperUnityr} \times BRperUnit\right) + 291\frac{kWh}{yr}$$

Users could reduce the estimated primary refrigerator AEC to this value, but no lower.

**Table 5: Minimum primary refrigerator AEC that builders may claim by BRperUnit**

| BRperUnit | Default Primary Refrigerator AEC (kWh/yr) | Minimum Allowable Primary Refrigerator AEC (kWh/yr) |
|---|---|---|
| 0 | 470 | 291 |
| 1 | 496 | 299 |
| 2 | 523 | 308 |
| 3 | 550 | 316 |
| 4 | 577 | 325 |
| 5 | 603 | 333 |
| 6 | 630 | 341 |
| 7+ | 657 | 350 |

For clothes washers, if allowed in the software, the user could specify that the installed clothes washer meets the 2015 federal standards (as documented on the CEC Appliance Efficiency Database). This effectively provides credit if the clothes washer is new or nearly new. Table 6 presents the AEC values used if the washer is compliant with the 2015 federal standards.

**Table 6: Minimum allowable high-efficiency AEC for clothes washers**

| BRperUnit | Single Family | | Multi-Family | |
|---|---|---|---|---|
| | Default AEC (kWh/yr) | High-Efficiency Clothes Washer AEC[1] (kWh/yr) | Default AEC (kWh/yr) | High-Efficiency Clothes Washer AEC[1] (kWh/yr) |
| 0 | 84 | 68 | 66 | 53 |
| 1 | 84 | 68 | 70 | 57 |
| 2 | 85 | 68 | 99 | 80 |
| 3 | 100 | 80 | 98 | 79 |
| 4 | 101 | 81 | 118 | 95 |
| 5+ | 117 | 94 | 107 | 86 |

[1]Applicable to clothes washers that meet the 2015 federal efficiency standards

For clothes dryers, if allowed in the software, the user could specify the percent remaining moisture content (RMC) of the installed clothes washer (as documented on the CEC Appliance Efficiency Database) to override the default clothes dryer AEC ruleset. The RMC-adjusted clothes dryer AEC should be calculated using the equations provided below. For natural gas dryers the RMC-adjusted AEC modifies natural gas use but does not impact electricity use.

### Electric Dryer: RMC-adjusted AEC (kWh/yr)

$$RMCadjusted\ AEC\ \frac{kWh}{yr} = 12.67\ \frac{kWh}{yr} + \left[\left(3.80\frac{kWh}{cycle}\left(RMC_{User,Input}\right) + 0.25\ \frac{kWh}{cycle}\right) \times \frac{cycles}{yr}\right]$$

### Gas Dryer: RMC-adjusted AEC (therms/yr)

$$RMCadjusted\ AEC\ \frac{therms}{yr} = \left[0.136\frac{therms}{cycle}\left(RMC_{User,Input}\right) + 0.00853\ \frac{therms}{cycle}\right] \times \frac{cycles}{yr}$$

**Table 7: Annual clothes dryer cycles estimated based on BrperUnit**

| BRperUnit | Dishwasher Cycles Per Year | |
|---|---|---|
| | Single-Family | Multi-Family |
| 0 | 290 | 227 |
| 1 | 290 | 241 |
| 2 | 291 | 341 |
| 3 | 342 | 335 |
| 4 | 346 | 405 |
| 5+ | 401 | 368 |

#### 1.1.4.3 *Load Profiles*

Dishwashers and clothes washers loads are specified in the water heating load profiles. Clothes dryers have the same usage assumptions as clothes washers, but shifted one hour later.

The estimated energy use for refrigerators is adjusted for each hour of the year depending on the simulated indoor temperature in the thermal zone where the refrigerator or freezer is installed (user input).Multi-family housing is assumed to have no energy use for non-primary refrigerators or separate freezers.

The following tables summarize the hourly load profiles and seasonal multipliers for the remaining plug load and lighting end uses.

**Table 8: Hourly Multiplier – Weekdays**

| Hour | Oven and Cooktop | Televisions | Set-Top Boxes | Computers and Monitors | Residual MELs | Interior and Garage Lighting | Exterior Lighting |
|---|---|---|---|---|---|---|---|
| 1 | 0.005 | 0.035 | 0.040 | 0.036 | 0.037 | 0.023 | 0.046 |
| 2 | 0.004 | 0.026 | 0.040 | 0.033 | 0.035 | 0.019 | 0.046 |
| 3 | 0.004 | 0.023 | 0.040 | 0.032 | 0.034 | 0.015 | 0.046 |
| 4 | 0.004 | 0.022 | 0.040 | 0.032 | 0.034 | 0.017 | 0.046 |
| 5 | 0.004 | 0.021 | 0.040 | 0.031 | 0.032 | 0.021 | 0.046 |
| 6 | 0.014 | 0.021 | 0.040 | 0.032 | 0.036 | 0.031 | 0.037 |
| 7 | 0.019 | 0.025 | 0.040 | 0.034 | 0.042 | 0.042 | 0.035 |
| 8 | 0.025 | 0.032 | 0.041 | 0.036 | 0.044 | 0.041 | 0.034 |
| 9 | 0.026 | 0.038 | 0.040 | 0.039 | 0.037 | 0.034 | 0.033 |
| 10 | 0.022 | 0.040 | 0.040 | 0.043 | 0.032 | 0.029 | 0.028 |
| 11 | 0.021 | 0.038 | 0.040 | 0.045 | 0.033 | 0.027 | 0.022 |
| 12 | 0.029 | 0.038 | 0.040 | 0.045 | 0.033 | 0.025 | 0.015 |
| 13 | 0.035 | 0.041 | 0.040 | 0.046 | 0.032 | 0.021 | 0.012 |
| 14 | 0.032 | 0.042 | 0.040 | 0.046 | 0.033 | 0.021 | 0.011 |
| 15 | 0.034 | 0.042 | 0.041 | 0.046 | 0.035 | 0.021 | 0.011 |
| 16 | 0.052 | 0.041 | 0.041 | 0.047 | 0.037 | 0.026 | 0.012 |
| 17 | 0.115 | 0.044 | 0.042 | 0.048 | 0.044 | 0.031 | 0.019 |
| 18 | 0.193 | 0.049 | 0.043 | 0.049 | 0.053 | 0.044 | 0.037 |
| 19 | 0.180 | 0.056 | 0.044 | 0.049 | 0.058 | 0.084 | 0.049 |
| 20 | 0.098 | 0.064 | 0.045 | 0.049 | 0.060 | 0.117 | 0.065 |
| 21 | 0.042 | 0.070 | 0.046 | 0.049 | 0.062 | 0.113 | 0.091 |
| 22 | 0.020 | 0.074 | 0.047 | 0.048 | 0.060 | 0.096 | 0.105 |
| 23 | 0.012 | 0.067 | 0.045 | 0.044 | 0.052 | 0.063 | 0.091 |
| 24 | 0.010 | 0.051 | 0.045 | 0.041 | 0.045 | 0.039 | 0.063 |

**Table 8: Hourly Multiplier – Weekends**

| Hour | Oven and Cooktop | Televisions | Set-Top Boxes | Computers and Monitors | Residual MELs | Interior and Garage Lighting | Exterior Lighting |
|---|---|---|---|---|---|---|---|
| 1 | 0.005 | .035 | 0.041 | 0.036 | 0.037 | 0.023 | 0.046 |
| 2 | 0.004 | 0.027 | 0.041 | 0.034 | 0.035 | 0.019 | 0.046 |
| 3 | 0.003 | 0.022 | 0.040 | 0.033 | 0.034 | 0.015 | 0.045 |
| 4 | 0.003 | 0.021 | 0.041 | 0.033 | 0.034 | 0.017 | 0.045 |
| 5 | 0.003 | 0.020 | 0.040 | 0.032 | 0.032 | 0.021 | 0.046 |
| 6 | 0.005 | 0.020 | 0.040 | 0.033 | 0.036 | 0.031 | 0.045 |
| 7 | 0.010 | 0.022 | 0.040 | 0.033 | 0.042 | 0.042 | 0.044 |
| 8 | 0.027 | 0.029 | 0.040 | 0.035 | 0.044 | 0.041 | 0.041 |
| 9 | 0.048 | 0.037 | 0.041 | 0.038 | 0.037 | 0.034 | 0.036 |
| 10 | 0.048 | 0.043 | 0.042 | 0.042 | 0.032 | 0.029 | 0.030 |
| 11 | 0.046 | 0.042 | 0.042 | 0.044 | 0.033 | 0.027 | 0.024 |
| 12 | 0.055 | 0.039 | 0.041 | 0.045 | 0.033 | 0.025 | 0.016 |
| 13 | 0.063 | 0.040 | 0.041 | 0.046 | 0.032 | 0.021 | 0.012 |
| 14 | 0.059 | 0.042 | 0.041 | 0.047 | 0.033 | 0.021 | 0.011 |
| 15 | 0.062 | 0.045 | 0.041 | 0.047 | 0.035 | 0.021 | 0.011 |
| 16 | 0.068 | 0.048 | 0.042 | 0.048 | 0.037 | 0.026 | 0.012 |
| 17 | 0.091 | 0.051 | 0.042 | 0.049 | 0.044 | 0.031 | 0.019 |
| 18 | 0.139 | 0.052 | 0.043 | 0.049 | 0.053 | 0.044 | 0.038 |
| 19 | 0.129 | 0.056 | 0.044 | 0.048 | 0.058 | 0.084 | 0.048 |
| 20 | 0.072 | 0.061 | 0.044 | 0.048 | 0.060 | 0.117 | 0.060 |
| 21 | 0.032 | 0.065 | 0.045 | 0.048 | 0.062 | 0.113 | 0.083 |
| 22 | 0.014 | 0.069 | 0.045 | 0.047 | 0.060 | 0.096 | 0.098 |
| 23 | 0.009 | 0.064 | 0.044 | 0.044 | 0.052 | 0.063 | 0.085 |
| 24 | 0.005 | 0.050 | 0.039 | 0.041 | 0.045 | 0.039 | 0.059 |

**Table 9: Seasonal Multipliers**

| Month | Oven and Cooktop | Televisions | Set-Top Boxes | Computers and Monitors | Residual MELs and Lighting |
|---|---|---|---|---|---|
| Jan | 1.094 | 1.032 | 1.02 | 0.98 | 1.19 |
| Feb | 1.065 | 0.991 | 0.84 | 0.87 | 1.11 |
| Mar | 1.074 | 0.986 | 0.92 | 0.89 | 1.02 |
| Apr | 0.889 | 0.990 | 0.98 | 1.11 | 0.93 |
| May | 0.891 | 0.971 | 0.91 | 1.14 | 0.84 |
| Jun | 0.935 | 0.971 | 0.94 | 0.99 | 0.80 |
| Jul | 0.993 | 1.002 | 1.05 | 1.05 | 0.82 |
| Aug | 0.920 | 1.013 | 1.06 | 1.01 | 0.88 |
| Sep | 0.923 | 1.008 | 1.06 | 0.96 | 0.98 |
| Oct | 0.920 | 1.008 | 1.14 | 0.97 | 1.07 |
| Nov | 1.128 | 1.020 | 1.03 | 0.99 | 1.16 |
| Dec | 1.168 | 1.008 | 1.05 | 1.04 | 1.20 |

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# 2016 Residential Alternative Calculation Method

## Algorithms

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# 1 Introduction

# 2 California Simulation Engine (CSE)

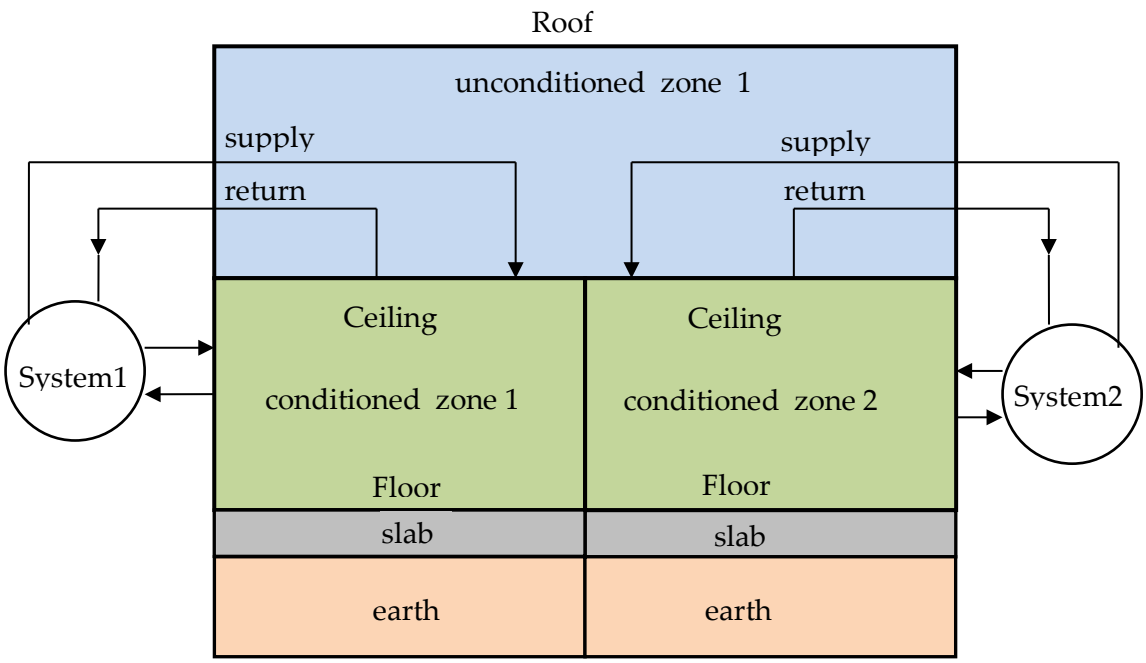
## 2.1 Overview

The building modeled can have multiple conditioned and unconditioned zones. Each conditioned zone has an air handler associated with it, and each air handler can have supply and/or return ducts in an unconditioned zone (nominally the attic), and in the conditioned zone itself. Air handlers can operate independently in either a heating, cooling, or off mode. See Figure 1.

Every time step (nominally two minutes), the zone model updates the heat transfers to and from the zones and the zone mass temperatures. Each zone's conditions are updated in succession and independently, based on the conditions in the adjacent zones in the last time step.

The conditioned zone thermostat algorithms determine whether an air handler should be in a heating or cooling mode, or floating, and if heating or cooling, the magnitude of the load that must be met by the air handler to keep the conditioned zone at its current setpoint. If the setpoints cannot be satisfied, the conditioned zone floats with heating, cooling, or ventilation, at full capacity. In the off mode case the zones are modeled during the time step without duct or air handler effects.

**Figure 1: Schematic of Zones and Air Handler Systems**




Although shown partly outside of the envelope, all ducts are assumed to be in either the conditioned or unconditioned zones only.

The duct system model determines duct losses, their effect on the conditions of the unconditioned and conditioned zones, and their effect on the heating or cooling delivery of the air handler system.

The duct system model allows unequal return and supply duct areas, with optional insulation thicknesses. The ducts can have unequal supply and return leakages, and the influence of unbalanced duct leakage on the unconditioned and conditioned zones infiltration and ventilation is taken into account. Every time step it updates the air handler and duct system heat transfers, and HVAC energy inputs, outputs, and efficiency.

For each window, the ASHWAT window algorithm calculates the window instantaneous shortwave, longwave, and convective heat transfers to the zones.

The AIRNET infiltration and ventilation algorithm calculates the instantaneous air flow throughout the building based on the air temperatures in the zones, and on the outside wind and air temperature. AIRNET also handles fan induced flows.

In the update processes, a zones mass-node temperatures are updated using a forward-difference (Euler) finite difference solution, whereby the temperatures are updated using the

driving conditions from the last time step. For accuracy, this forward-difference approach necessitates a small time-step.

The small time-step facilitates the *no-iterations* approach we have used to model many of the interactions between the zones, and allows the zones to be updated independently.

For example, when the zone energy balance is performed for the conditioned zone, if ventilation is called for, the ventilation capacity, which depends on the zone temperatures (as well as maximum possible ventilation openings and fan flows), is determined from the instantaneous balance done by AIRNET. To avoid iteration, the ventilation flows, and the accompanying heat transfers are based on the most recently available zone temperatures.

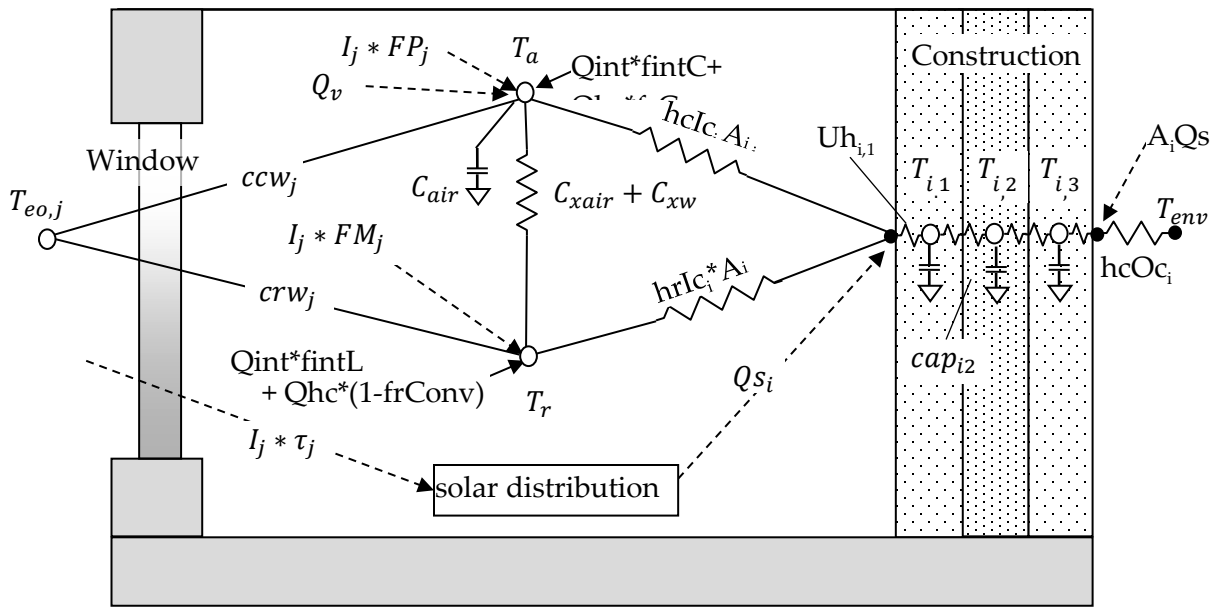
To avoid iteration, a similar use of the last time-step data is necessary is dealing with inter-zone wall heat transfer. For example, heat transfer through the ceiling depends on the conditions in both zones, but these conditions are not known simultaneously. Thus, ceiling masses are treated as belonging to the attic zone, and updated at the same time as other attic masses, partly based on the heat transfer from the conditioned zone to the ceiling from the last time step. In turn, when the conditioned zone is updated it determines the ceiling heat transfer based on the ceiling temperature determined two-minutes ago when the attic balance was done.

Similarly, when the conditioned zone energy balance is performed, if for example heating is called for, then the output capacity of the heating system needs to be known, which requires knowing the duct system efficiency. But the efficiency is only known after the air handler simulation is run. To avoid iteration between the conditioned zone and attic zones, the most recent duct efficiency is used to determine the capacity in the conditioned zones thermostat calculations. When the attic simulation is next performed, if the conditioned zone was last running at capacity, and if the efficiency now calculated turns out to be higher than was assumed by the thermostat calculations, then the load will have exceeded the limiting capacity by a small amount depending on the assumed vs. actual efficiency. In cases like this, to avoid iteration, the limiting capacity is allowed to exceed the actual limit by a small amount, so that the correct energy demand is determined for the conditioned zone load allowed.

### 2.1.1 Schematic of Zone Thermal Network

Figure 2 shows a schematic of the zone model network. It models a single zone whose envelope consists of any number of walls, ceilings, floors, slabs, and windows, and can be adjacent to other conditioned or unconditioned zones. The envelope constructions can be made of any number of layers of different materials of arbitrary thermal conductivity and heat capacity. Each layer is modeled with one or more "T" networks in series. Each T has the layer heat capacitance, $cap_{ij}$, centered between by two thermal conductances, where the first subscript corresponds to the wall construction number and the second to the layer number. Framed constructions are treated as two separate surface areas, the surface area of the part between framing, and the surface area of the part containing the framing itself; the heat flow is assumed to follow independent and parallel paths through these two surfaces.

**Figure 2: Schematic of Simulation Network**




The room air, represented by the mass node $Ta$, is assumed to be well-mixed and have heat capacitance $C_{air}$ (Btu/F). The air is shown in Figure 2 to interact with all of the building interior construction surfaces via convection coefficients $hcIc_i$ for surface $i$. The overall conductance through the window between $T_a$ and an effective outdoor temperature $T_{eo}$ is $ccw_j$ for window surface $j$. The conductances $ccw_j$ and the corresponding radiant value $crw_j$ are outputs of the ASHWAT windows algorithm applied to window $j$ each time step.

A mean radiant temperature node, $T_r$, acts as a clearinghouse for radiant exchange between surfaces. With conductances similar to those of the air node: $hrIc_i$ and $crw_j$.

Depending on the size of the zone and the humidity of the air, the air is assumed to absorb a fraction of the long-wave radiation and is represented by the conductance $C_{Xair}$.

The internal gains, $Qint$, can be specified in the input as partly convective (fraction $fintC$), partly long wave ($fintLW$), and partly shortwave (fintSW). The heating or cooling heat transfers are shown as $Qhc$ (+ for heating, - for cooling). If $Qhc$ is heating, a fraction ($frConv$) can be convective with the rest long-wave. The convective parts of $Qint$ and $Qhc$ are shown as added to the air node. The long wave fraction of Qint and Qhc are shown added to the $T_r$ node.

Additional outputs of the ASHWAT algorithm are $FP_j$ the fraction of insolation $I_j$ incident on window $j$ that ultimately arrives at the air node via convection, and $FM_j$, the fraction that arrives at the radiant node as long-wave radiation.

The term $Qs_i$ is the the total solar radiation absorbed by each construction surface i, as determined by the solar distribution algorithm. The short wave part of the internal gains, Qint $*$

fintSW, is distributed diffusely, with the same diffuse targeting as the diffusely distributed solar gains.

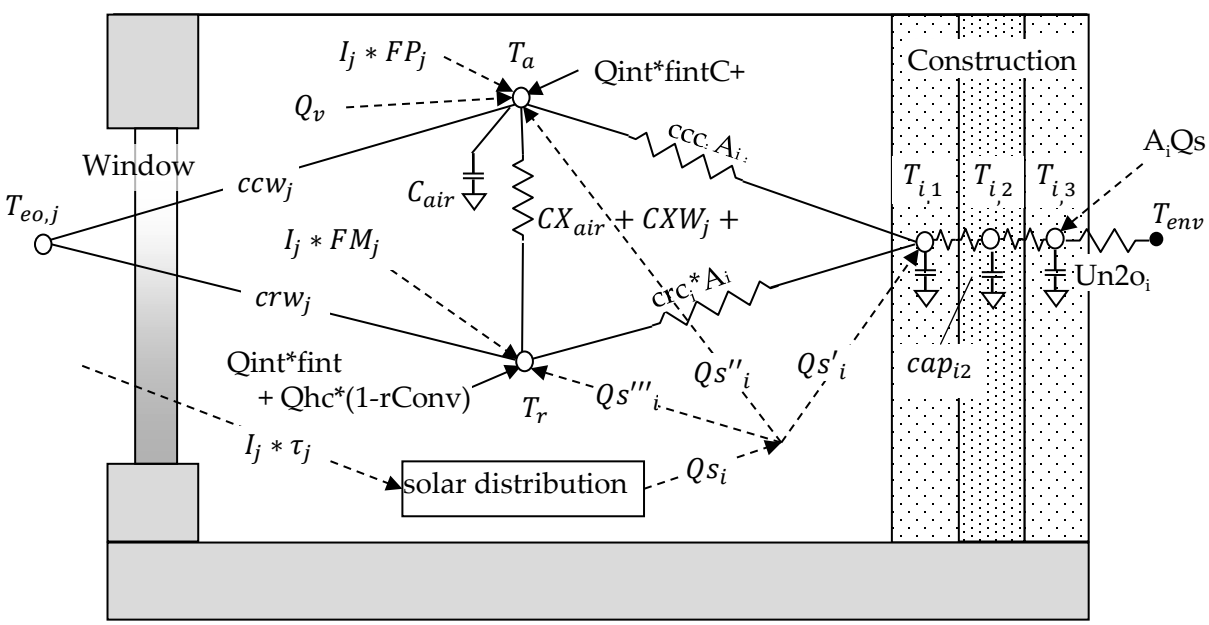
Solar gains absorbed at the outside surface of constructions are represented by $Qs_o$ in Figure 2.

The slab is connected to the Ta and Tr in a similar fashion as the wall surfaces, although the slab/earth layering procedure is different than for walls.

## 2.1.2 Schematic of Reduced Thermal Network

Before a zone energy balance is formulated it is convenient to dissolve all the massless nodes from the network of Figure 2 (represented by the black dots), except for the mean radiant temperature node $Tr$. Figure 3 shows the resulting reduced network. A massless node is eliminated by first removing the short-wave gains from the node by using the current splitting principle (based on superposition), to put their equivalent gains directly onto adjacent mass nodes and other nodes that have fixed temperatures during a time step. Then the massless node can be dissolved by using Y-Δ transformations of the circuit.

**Figure 3: Network after Dissolving Massless Nodes**




For example, to eliminate the massless surface node of layered mass in Figure 2, the gain $Qs_i$ absorbed by the surface node is split into three parts: $Qs_i'$ to the $T_{i,1}$ node, $Qs_i''$ to the Ta node and $Qs_i'''$ to the Tr node. For example, by current splitting,

$$Qs_i' = Qs_i \frac{Uh_{i,1}}{hcIc_i + hrIc_i + Uh_{i,1}}$$

*Equation 1*

A Y-Δ transformation of the remaining Y circuit gives the $ccc_i$ and $crc_i$ conductances, as well as an additional cross conductance $CXC_i$ that is added to CXair. For example,

$$ccc_i = \frac{hcIc_i * Uh_{i,1}}{hcIc_i + hrIc_i + Uh_{i,1}}$$

*Equation 2*

### 2.1.3 Zone Balance Calculation Sequence

The temperatures in the zone are determined using a thermal balance method. The following procedure is followed each time step.

At the start of the simulation, say time t, assume all temps $Ta(t), Tr(t), T_{i,1}(t), T_{i,2}(t)$, etc. are known along with all the solar gains, internal gains, etc.

(1) First, the layered mass temperatures are updated using the explicit Euler routine (see Section 2.2), giving $T_{i,1}(t+dt), T_{i,2}(t+dt)$, etc. The Euler method determines each of these mass temperatures assuming that all the boundary conditions (temperatures and heat sources) that cause the change in the mass temperatures, are conditions at time t. Thus the mass node temperatures can be in any order, independently of each other.

(2) Next, a steady-state instantaneous energy balance at the Ta and Tr nodes is made at time t+ dt. This balance involves the mass temperatures determined for time t + dt in Step-1, as well other heating or cooling sources at time t+ dt. The balance in this step involves querying the HVAC control algorithm which allows heating, cooling and ventilation (forced or natural) in response to scheduled setpoints. The idealized control system is assumed to keep the zone at exactly the scheduled setpoint unless Ta is in the deadband or if the HVAC capacity is exceeded, whereupon the system runs at maximum capacity, and Ta floats above or below the relevant setpoint. While the heating, cooling and forced ventilation system capacities are scheduled inputs, the natural ventilation capacity is dependent on the current zone and environment conditions.

Thus, the energy balance at the Ta and Tr nodes returns either the heating, the cooling or the ventilation required to meet the setpoint, or else returns the floating Ta that results at the capacity limits or when Ta is in the deadband.

At this stage the conditions have been predicted for the end of the time step, and steps 1 and 2 and repeated. The various boundary conditions and temperature or air flow sensitive coefficients can be recalculated as necessary each time step at the beginning of step (1), giving complete flexibility to handle temperature sensitive heat transfer and control changes at a time step level.

Note that step (2) treats the energy balance on Ta as a steady state balance, despite the fact that air mass makes it a transient problem. However, as shown in Section 2.3.1, if the air mass temperature is updated using an implicit-difference method, the effect of the air mass can be

duplicated by employing a resistance, $\Delta t/Cair$, between the Ta node and a fictitious node set at the beginning of the time-step air temperature TaL = Ta(t), and shown as such in Figure 3.

The overall CSE Calculation Sequence is summarized below:

*Hour*

Determine and distribute internal gains.

*Sub-hour*

Determine solar gain on surfaces.

Determine surface heat transfer coefficients.

Update mass layer temperatures.

Find AirNet mass flows for non-venting situation (building leakage + last step HVAC air flows).

Find floating air temp in all zones / determine if vent possibly useful for any zone.

If vent useful

- find AirNet mass flows for full venting
- find largest vent fraction that does not sub-cool any zone; this fraction is then used for all zones.
- if largest vent fraction > 0, update all floating zone temperatures assuming that vent fraction

Determine HVAC requirements for all zones by comparing floating temp to setpoints (if any)

- System heating / cooling mode is determined by need of 1st zone that requires conditioning
- For each zone, system indicates state (t and w) of air that could be delivered at register (includes duct loss effects). Zone then requests air flow rate required to hold set-point temperature

Determine HVAC air flow to zones (may be less than requested); determine zone final zone air temperatures.

Determine system run fraction and thus fuel requirements.

Determine zone humidity ratio for each zone.

Calculate comfort metrics for each zone.

## 2.2 Updating Layered Mass Temperatures

The heat transfer through the layered constructions is assumed to be one dimensional. The heat conduction equation ( $\frac{\partial^2 T}{\partial x^2} = \frac{1}{\alpha}\frac{\partial T}{\partial t}$ ) is solved by using finite differences (Δt and Δx) to approximate the differential increments in time and distance; $\alpha$ is the thermal diffusivity. The smaller the finite increments, the more accurate the solution. The homogeneous layers are divided into lumps Δx thick, and the lumps are represented by the two-conductance/one-

capacitance "T" circuits shown for each layer in Figure 2. Frequently the actual layer thicknesses as sufficiently thin that Δx can be taken as the layer thickness. However, at times the actual layer of homogeneous material must be divided into smaller thicknesses. See Section 2.4– Discretization Errors for the criterion used to determine Δx and Δt.

The temperatures of the mass nodes are updated every time step using the Euler explicit numerical integration method (see Press et al), whereby the change in temperature of the mass during the time step is based only on the boundary conditions at the beginning of the time step. The boundary conditions are the temperatures of the surrounding nodes and other heat flow sources.

To update $T_{i,1}$ in Figure 3, for example, if the rate of heat transfer into $T_{i,1}$ is equated to its rate of change in internal energy, resulting in the differential equation for mass temperature $T_{i,1}$:

$$\frac{dT_{i,1}}{dt} = \frac{Tss_{i,1} - T_{i,1}}{\tau}$$

*Equation 3*

where $T_{i,1}$ is the surface layer mass temperature, and $Tss_{i,1}$ is the temperature $T_{i,1}$ would have if steady state were reached:

$$Tss_{i,1} = \frac{ccc_{i,1}Ta + crc_{i,1}Tr + Ubn_{i,1}T_{i,2} + Qs_i'}{ccc_{i,1} + crc_{i,1} + Ubn_{i,1}}$$

*Equation 4*

$Qs_i'$ is given by Equation 1, $ccc_{i,1}$ by Equation 2, $T_{i,2}$ is the temperature of mass node 2, and $\tau$ is the time constant of mass node 1 given by:

$$\tau = \frac{cap_{i,1}}{ccc_{i,1} + crc_{i,1} + Ubn_{i,1}}$$

*Equation 5*

The heat capacity of layer-1 is $cap_{i,1}$ (Btu/ft²-F). $Ubn_{i,1}$ is the conductance between nodes 1 and 2, given by:

$$Ubn_{i,1} = \frac{1}{\frac{1}{Uh_{i,1}} + \frac{1}{Uh_{i,2}}}$$

*Equation 6*

To integrate of Equation 3 over a time step, the Euler procedure assumes that the right hand side of the equation remains constant over the time step at its value at the beginning of the time step. In this case the mass temperature at the end of the time step becomes:

$$T_{i,1}(t + \Delta t) = T_{i,1}(t)\left(1 - \frac{\Delta t}{\tau}\right) + Tss\left(\frac{\Delta t}{\tau}\right)$$

*Equation 7*

If the capacitance of any layer is zero (a convecting air layer for example) its updated temperature is set equal to $Tss$. That is, the temperature at the central node is determined by a steady state energy balance.

All of the mass nodes are updated in an analogous fashion each time step. The order in which the masses are updated is irrelevant because they are updated based only on the values of variables at the beginning of the time step, not on the values that may have been updated since.

## 2.3 Zone Energy Balance

### 2.3.1 Implicit Update of Air Temperature

Similar to the energy balance on the construction mass nodes, an energy balance on the air node gives the differential equation:

$$\frac{dT_a}{dt} + \frac{T_a}{\tau_a} = \frac{Tss}{\tau_a}$$

*Equation 8*

where $Tss$, the asymptotic steady state temperature of $Ta$, includes all the sources connected to $Ta$. For simplicity, if the zone only contained the one construction ($i$ = 1) and one window ($j$=1), like in Figure 3, then from a steady state energy balance Tss is given by:

$$T_{ss} = [T_{out}(U_{inf} + U_v) + ccw_1 Awin_1 T_{out} + ccc_1 Acon_1 T_{1,1} + Acon_1 Qs_1'' + Awin_1 Qsw_1'' \\ + Qint * fintC + Qhc * frConv + CX * Tr]/Usum$$

*Equation 9*

where

$$CX = CXair + CXW_1 + CXC_1$$

*Equation 10*

$$Usum = U_{inf} + U_v + ccw_1 Awin_1 + ccc_1 Acon_1 + CX$$

*Equation 11*

and the air time constant is:

$$\tau_a = \frac{Cair}{Usum}$$

*Equation 12*

Equation 8 is solved using an full implicit (or backward time) difference, similar to the Euler explicit method except here the right hand side of the equation remains constant over the time step at its value at the end of the time step, not its value at the beginning as in the Euler method. Thus, Equation 8 then becomes:

$$T_a(t+\Delta t) = \frac{\frac{T_a(t)\tau_a}{\Delta T} + Tss(t+\Delta t)}{\frac{\tau_a}{\Delta t} + 1}$$

*Equation 13*

Where the times $t$ and $t+\Delta t$ in parenthesis indicate the terms are evaluated at the beginning and end of the time step, respectively. Substituting Equation 12 for $\tau_a$, Equation 13 can be put in the convenient form:

$$T_a(t+\Delta t) = \frac{\frac{T_a(t)Cair}{\Delta T} + Usum * Tss(t+\Delta t)}{\frac{Cair}{\Delta t} + Usum}$$

*Equation 14*

As this equation shows, with the implicit difference the effect of the air mass can be thought of as a resistance, $\Delta t/Cair$, between the $Ta$ node and a fictitious node set at the air temperature at the value it was at the beginning of the time step, $TaL = Ta(t)$. This alternative is known as an 'associated discrete circuit'. Leaving out the explicit time references, Equation 14 can be written:

$$Ta = \frac{\frac{TaL * Cair}{\Delta T} + Usum * Tss}{\frac{Cair}{\Delta t} + Usum}$$

*Equation 15*

where $Ta$ and $Tss$ are evaluated at the end of the time step, and $TaL$ stands for $T_a(t)$ at the beginning of the time step. Note that Equation 15 still contains the variable $Tr$ (hidden in $Tss$) which is unknown. $Tr$ can be eliminated by making an energy balance on the $Tr$ node and substituting the expression for $Tr$ into Equation 15. This is done for the complete set of equations that follow.

## 2.3.2 Zone Balance Equations

The complete set of zone energy balance equations for multiple windows and constructions are given below. Terms containing $Qv$ and $Qhc$ are kept separate so that the resulting equations can be solved for $Qv$ or $Qhc$ when $Ta$ is fixed at a setpoint.

### 2.3.2.1 Air Node Balance

The energy balance equation on the Ta node, comparable to Equation 15 above is:

$$Ta = \frac{Qv + Qhc \cdot frConv + Nair + CX \cdot Tr}{Dair + CX}$$

*Equation 16*

The Equation 16 form, using $Q_v$, is used when heat is transferred to a conditioned zone with ventilation or infiltration air. When heat is transferred to an unconditioned zone due to ventilation or infiltration, $Q_v$ is replaced by the essentially equivalent form given by Equation 17, wherein $Q_v$ is replaced by $Q_v = mC_p\Delta T$ such that $mdot * c_p * T$ is added to the numerator and $mdot * c_p$ is added to the denominator. This was implemented to eliminate oscillations in Ta.

$$Ta = \frac{ST + Qhc \cdot frConv + Nair + CX \cdot Tr}{SB + Dair + CX}$$

*Equation 17*

where,

$$ST = \sum mdot * c_p * T,$$

*Equation 18*

where T is the temperature of the air in the zone supplying the infiltration or ventilation air.

$$SB = \sum mdot * c_p$$

*Equation 19*

$$CX = CXair + \sum^{con} Acon_i \cdot cxc_i + \sum^{win} Awin_i \cdot cxw_i$$

*Equation 20*

with the sum's for all constructions and all windows respectively.

$$Nair = TaL\left(\frac{Cair}{dt}\right) + Qint \cdot fintC + \sum^{con} Acon_i \cdot \left(ccc_i \cdot T_{i1} + \frac{Q_{si} \cdot hcIc_i}{hcIc_i + hrIc_i + Uh_{i1}}\right) + \sum^{win} \left[Awin_i\left(ccw_i \cdot Teo_i + I_j \cdot FP_j\right)\right]$$

*Equation 21*

$$Dair = \frac{Cair}{dt} + \sum^{con} Acon_i \cdot ccc_i + \sum^{win} Awin_i \cdot ccw_i$$

*Equation 22*

Qv is the heat transfer to the air node due to infiltration and forced or natural ventilation.

#### 2.3.2.2 Radiant Node Balance

An energy balance on the $Tr$ node gives Equation 23.

$$Tr = \frac{Qhc \cdot (1 - frConv) + N_{rad} + CX \cdot Ta}{Drad + CX}$$

*Equation 23*

where,

$$Nrad = Qint \cdot fintLW + \sum^{con} Acon_i \left[ crc_i \cdot T_{i1} + Qsi_i \cdot \frac{hrIc_i}{hcIc_i + hrIc_i + Uh_{i1}} \right]$$

$$+ \sum^{win} Awin_i \left[ crw_i \cdot Teo_i + I_j \cdot FM_j \right]$$

*Equation 24*

$$Drad = \sum^{con} Acon_i \cdot crc_i + \sum^{win} Awin_i \cdot crw_i$$

*Equation 25*

### 2.3.2.3 Simultaneous Solution of Ta and Tr Equations

Equation 16 and Equation 23 can be solved simultaneously to eliminate $Tr$ and give $Ta$ explicitly:

$$Ta = \frac{(Qv + Qhc \cdot frConv + Nair)(Drad + CX) + CX(Nrad + Qhc(1 - frConv)}{(Dair + Drad)CX + Dair \cdot Drad}$$

*Equation 26*

Similar to Equation 16 and Equation 17), the alternate form of Equation 26 is given by Equation 27.

$$Ta = \frac{(ST + Qhc \cdot frConv + Nair)(Drad + CX) + CX(Nrad + Qhc(1 - frConv)}{(SB + Dair + Drad)CX + (SB + Dair)\mathrm{D}rad}$$

*Equation 27*

Substituting $Ta$ from Equation 26 into Equation 23 gives $Tr$.

### 2.3.2.4 Qhc and Qv Equations

When *Ta* is at either the heating or cooling setpoints, Equation 26 is solved to determine the required *Qhc*. In this case *Qv* is set *to QvInf.*

$$Qhc = \frac{Ta(Dair * Drad + CX(Dair + Drad)) - (Nair + Qv)(Drad + CX) - Nrad * CX)}{frConv * Drad + CX}$$

*Equation 28*

Similarly, when Ta is at the ventilation setpoint, Equation 26 can be solved for Qv to give:

$$Qv = \frac{(Dair * Drad + CX(Dair + Drad))Ta - CX(Nrad + Qhc(1 - frConv))}{Drad + CX} - (Qhc * frConv + Nair)$$

*Equation 29*

With $Qhc$ = 0 this becomes:

$$\text{Qv} = \frac{(Dair * Drad + CX(Dair + Drad))Ta - Nair(Drad + CX) - CX * Nrad}{Drad + CX}$$

*Equation 30*

The zone balance is essentially an instantaneous balance, so all the temp inputs are simultaneous values from the end of the time step (with the exception of $TaL$; see Section 2.3.1). Although the balance is with contemporary temperatures, many of the heat flows in $Nair$ etc., are based on last time step conditions.

### 2.3.3 Thermostat Logic

At the end of each time step the program finds the floating temperature of the zone without HVAC ($Qhc = 0$) and with venting Qv = QvInf. This floating temperature found from Equation 26 is defined as TS1. Next, the venting capacity is determined (see Section 2.9.3.10, Heat Flow), and Equation 26 is solved for Ta at the full venting capacity. This Ta is defined as TS2.

TS1 will satisfy one of the four cases:

- TS1>TC
- TC > TS1 > TD
- TD > TS1 > TH
- TH > TS1

Similarly, TS2 will satisfy one of the four cases:

- TS2 > TC,
- TC > TS2 > TD
- TD > TS2 > TH
- TH > TS2

where TC, TD, and TH are the scheduled cooling, ventilation, and heating setpoints, with TC > TD > TH.

Based on the cases that TS1 and TS2 satisfy, nested logic statements determine the appropriate value of heating, cooling, venting, or floating .

For example, if TS1 and TS2 are both > TC, then $Q_v$ is set *QvInf* and Ta is set to TC, and Equation 28 is solved for the required cooling, Qhc. If Qhc is smaller than the cooling capacity at this time step then Qhc is taken as the current cooling rate and the zone balance is finished and the routine is exited. If Qhc is larger than the cooling capacity then Qhc is set to the cooling capacity, and Equation 26 is solved for Ta, floating above TC due to the limited cooling

capacity. If Ta < TS2 then Ta and Qhc are correct and the zone balance routine is exited. If this Ta > TS2 then Ta is set equal to TS2, Qhc is set to zero, and Equation 29 is solved for the ventilation rate $Qv$, and the Zone Balance routine is complete.

Similar logic applies to all other logically possible combinations of the TS1 and TS2 cases above.

### 2.3.4 Limiting Capacities

The limiting capacity of the heating and cooling system is determined each time step by multiplying the scheduled nominal air handler input energy capacity by the duct system efficiency. To avoid iteration between the conditioned zone and unconditioned zone simulations, the duct system efficiency is taken from the last time-step's unconditioned zone simulation, or unity if the system mode (heating, cooling, venting, or floating) has changed.

## 2.4 Discretization Errors

The temperatures predicted by Equation 7, which updates the layered mass temperatures, is subject to errors due to the finite lump size chosen to represent real wall homogeneous layers. It is also subject to errors due to the finite time step $\Delta t$. Similarly Equation 14 for updating the air mass temperature is subject to error due to the finite time step chosen.

Discretization errors can be made negligible by reducing the layer thicknesses and time step to very small values. However for practical run time minimization purposes it is useful to have large $\Delta t$ and $\Delta x$ layers, insofar as accuracy allows. The range of choices of $\Delta t$ and $\Delta x$ is narrowed if accurate results are only required for a limited range of frequencies of the driving boundary conditions. Only extremely thin lumped layers have the correct frequency response at high frequencies. To model environmental influences, 3 cycles/day (8-hr period sinusoid) is likely the highest frequency necessary to consider when determining the frequency response of buildings (Goldstein, Anderson and Subbarao). Higher frequencies may be desirable for accurately modeling things like control step changes. During the program development, accuracy was measured by analyzing the frequency response at 3 cycles/day.

The exact frequency response of a layered wall can be obtained using the matrix method (Section 3.7 of Carslaw & Jaeger) which gives the inside driving point admittance (from the inside air node), the outside driving point admittance, and the transfer admittance, for any frequency. The magnitude of the inside driving point admittance is the principle parameter used to assess algorithm accuracy.

At the frequency chosen, 3 cycles/day say, the exact driving point admittance of the real wall (with homogeneous layers) can be obtained from the matrix method. Similarly the exact driving point admittance of the lumped wall which the user has chosen to represent the real wall, can also be determined by the matrix method. Comparing these two results shows the accuracy of the lumping assumptions, independent of time step considerations.

The time discretization error associated with Equation 7 at the frequency chosen can be assessed by comparing the driving point admittance predicted by the CSE code, when the air node is driven with a sinusoidal temperature at the chosen frequency, to the theoretical admittance of the lumped wall. Note that this procedure measures the global discretization error, larger potentially than the per time-step error.

Using this procedure for typical lightweight residential construction, we have confirmed that the errors in the temperature predictions made by the CSE finite difference algorithms indeed tend toward zero as $\Delta t$ and $\Delta x$ are reduced toward zero.

### 2.4.1 Layer Thickness of a Homogeneous Material

The lumped layer thickness, $\Delta x$, should be is chosen thin enough that the single temperature of the lumped layer is a good measure of the average temperature over a width $\Delta x$ of the sinusoidal temperature distribution in the material. That is, the temperature of the sinusoidal wave should not vary much over the layer width. This criterion is similar to that used by Chirlian (1973) to determine the appropriate lump sizes in electrical circuits.

The wave length of the temperature distribution in a particular material is given by

$$\lambda = 2\pi d_p$$

*Equation 31*

where $d_P$ , the penetration depth, an intrinsic characteristic of the material, is given by

$$d_p = \sqrt{2\alpha/\omega}$$

*Equation 32*

where the angular frequency $\omega = 2\pi/period$ , $\alpha$ is the thermal diffusivity of the layer material, and $\omega$ is the highest angular frequency of the environmental boundary conditions for which good frequency response is desired. As a general guideline it is suggested that the lumped layer thicknesses, $\Delta x$, be chosen to be thinner than the penetration depth for the layer. That is, select

$$\Delta x \lesssim d_p$$

*Equation 33*

Substituting Equation 32 into Equation 33 shows that the rule of Equation 33 limits the lump size $\Delta x$ to about 16% of the wavelength:

$$\Delta x \lesssim dp = \lambda/2\pi = 0.16\lambda$$

*Equation 34*

The Equation 33 rule is more important for the modeling of layers on the inner side of the wall, where the layers are subjected to the higher frequency harmonics of inside driving conditions.

Deeper into the wall the high frequency harmonics begin to be damped (by about a factor of $e^{-\frac{\Delta x}{d_p}}$ ), so accurate modeling is of less significance.

### 2.4.2 Choosing the Time Step

The time step used in the code is input by the user. For high accuracy Equation 7 and Equation 14 should be applied using a time step that is a small fraction of the smallest time constant of any layer.

$$\Delta t \ll \tau$$

*Equation 35*

Thin layers of a material have a smaller time constant τ than thick layers. The time constant of a layer scales as $\sim \beta^2$, where $\beta$ is the a layers dimensionless thickness defined as $\beta = \Delta x/d_p$. Thus, if a layers dimensionless thickness is reduced by a factor of two, the time constant is reduced by a factor of four. Therefore the time to run an annual simulation can increase rapidly for small $\beta$'s. Small tau layers have cv increased such that tau = dt.

Note that the Euler mass layer update algorithm of Equation 7becomes unstable when $\Delta t > \tau$ . The predicted temperatures will oscillate with increasing amplitude each time step. The code outputs warnings whenever a mass node update is performed for which $\Delta t > \tau$.

Like the explicit Euler method, the implicit differencing used at the air node is most accurate for small time steps relative to the air's time constant (Equation 12). The implicit difference method is never unstable, and time steps larger than the air time constant give useful, if somewhat inaccurate predictions. The air balance could have been solved using an Euler difference, but since the air time constant is likely the smallest in the zone, it would dictate smaller time steps than is afforded using the implicit method

## 2.5 Surface Heat Transfer Coefficients

The radiation coefficients for surfaces inside the conditioned zone are given in Section 2.6.1 where the long-wave radiant network model is discussed.

### 2.5.1 Local Wind Velocity Terrain and Height Correction

The wind velocity as a function of height at the house site is obtained from the meteorological station wind measurement by making adjustments for terrain and height differences between the meteorological station and the house site.

#### 2.5.1.1 Sherman-Grimsrud method

This method uses Equation 36 which determines the wind velocity $V(z)$, in ft/sec, at any height $z$ (ft) based on the wind velocity, $V_{met}$ in ft/sec, measured at a location with a Class II terrain (see Table 1) and at a height of 10-meters (32.8 ft):

$$V(z) = SC * V_{met} * \alpha * \left(\frac{z}{32.8}\right)^{\gamma}$$

*Equation 36*

where,

$\alpha$ and $\gamma$ are obtained from Table 1 for the terrain class at the building location.

$SC$ = shielding coefficient from Table 2 for the building location.

$V(z)$ = wind velocity at height z at the building location (ft/sec).

$V_{met}$ = wind velocity (ft/sec) measured at 10-meters height in a Class II location.

The terrain factor of Table 1 is a general factor describing the influence of the surroundings on a scale on the order of several miles. The shielding factor of Table 2 is a local factor describing the influence of the surroundings on a scale of a few hundred yards.

**Table 1: Parameters for Standard Terrain Classifications**

| Class | γ | α | Description |
|---|---|---|---|
| I | 0.10 | 1.30 | ocean or other body of water with at least 5km of unrestricted expanse |
| II | 0.15 | 1.00 | flat terrain with some isolated obstacles(buildings or trees well separated) |
| III | 0.20 | 0.85 | rural areas with low buildings, trees, etc. |
| IV | 0.25 | 0.67 | urban, industrial, or forest areas |
| V | 0.35 | 0.47 | center of large city |

**Table 2: Local Shielding Parameters**

| Class | C' | SC | Description |
|---|---|---|---|
| I | 0.324 | 1.00 | no obstructions or local shielding |
| II | 0.285 | 0.880 | light local shielding with few obstructions |
| III | 0.240 | 0.741 | moderate local shielding, some obstructions within two house heights |
| IV | 0.185 | 0.571 | heavy shielding, obstructions around most of the perimeter |
| V | 0.102 | 0.315 | very heavy shielding, large obstructions surrounding the perimeter within two house heights. |

#### 2.5.1.2 Implementation

If it is assumed that the default value of the terrain classification at the building location is Class IV terrain of Table 1, and the default local shielding coefficient is SC= 0.571 of Class IV of Table 2, then the wind velocity at the building site at height ***z*** is given by:

$$V(z) = SC * V_{met} * \alpha * \left(\frac{ze}{32.8}\right)^{\gamma} = 0.571 * V_{met} * 0.67 * \left(\frac{z}{32.8}\right)^{0.25}$$

or,

$$V(z) = 0.16 * z^{0.25} * V_{met}$$

For example, for 1, 2, and 3 story buildings, of 9.8 ft (3-m), 19.7 ft (6-m), and 29.5 ft (9-m), respectively, then the local eave height wind velocities are:

$V(9.8) = 0.16 * 9.8^{0.25} * V_{met}$ = 0.28 $V_{met}$ for a 1-story building.

V(19.7) = 0.34 $V_{met}$ for a 2-story building.

V(29.5) = 0.38 $V_{met}$ for a 3-story building.

(References: Sherman & Grimsrud (1980), Deru & Burns (2003), Burch & Casey (2009), European Convention for Constructional Steelwork (1978).)

## 2.5.2 Convection Coefficient for the Inside and Outside Surfaces of the Zones

The schematic buildings in Figure 4 and Figure 5 show all of the possible interior heat transfer situations for which the convection heat transfer coefficients are determined. The figures symbolically show the nature of the heat transfer boundary layer, and the heat flow direction. The symbols used are explained at the end of this document. Similar schematics have not been done for the outside surfaces.

The equations are developed that give the heat transfer coefficient for each of the Figure 4 and Figure 5 situations, and for the building outside surfaces. The heat transfer coefficients depend on the surface tilt angle θ ($0 \leq \theta \leq 90$), the surface and air temperatures, and on whether the heat flow of the surface has an upward or downward facing component.

The results, which apply to both the UZ and CZ zones, can be summarized as follows:

### 2.5.2.1 Inside surfaces

For floors, and either vertical walls, or walls pulled-in-at-the-bottom:

If Tair > Tsurf use Equation 53. (heat flow down)

If Tair < Tsurf use Equation 52. (heat flow up)

For ceilings (horiz or tilted), and walls pulled-in-at-the-top:

If Tair > Tsurf use Equation 52. (heat flow up)

If Tair < Tsurf use Equation 53. (heat flow down)

### 2.5.2.2 Outside surfaces

For all vertical walls, and walls with moderate tilts use Equation 54.

For horizontal or tilted roof, use Equation 57.

**Figure 4: Heat Flow Down Situations**

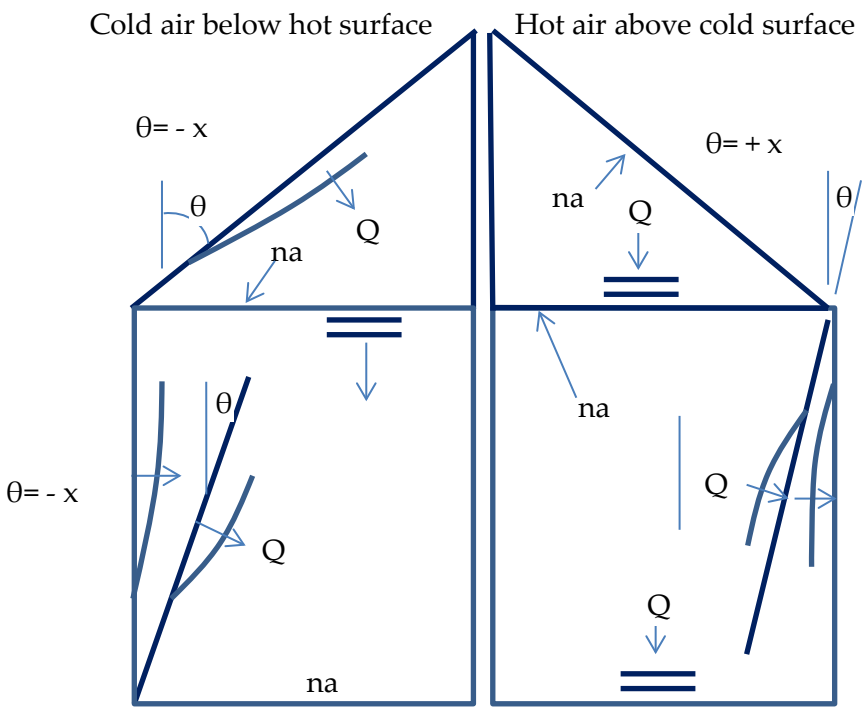
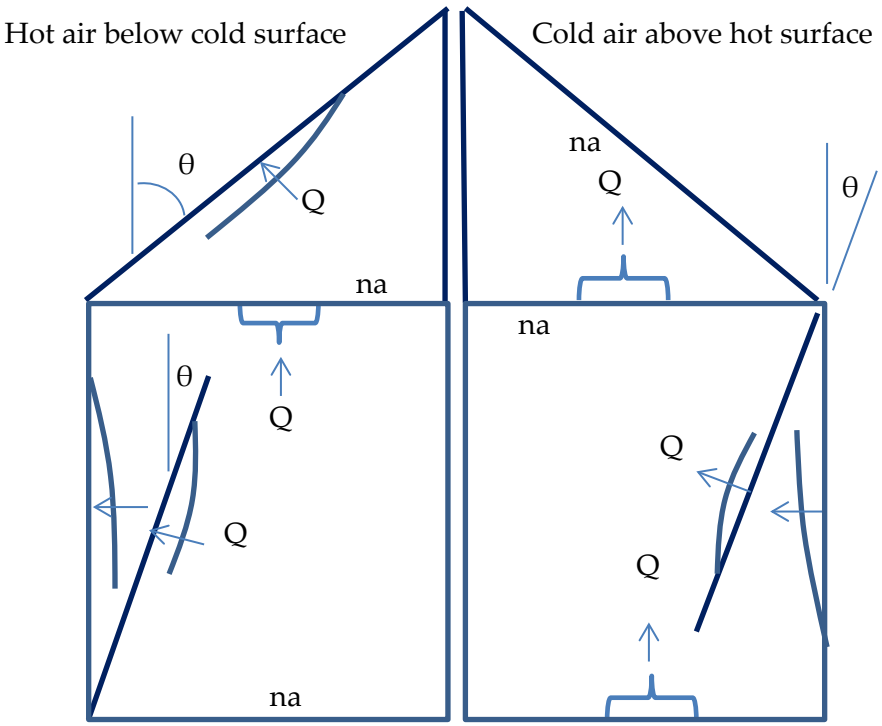
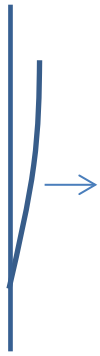



**Figure 5: Heat Flow Up Situations**



**Explanation of Symbols**



symbolizes the direction in which the boundary layer is thickening, and the direction of positive heat flow.

symbolizes the air is stratified next to the surface and the direction the heat flow would be if there is any--the Figs. 1 eqns give zero heat flow in this case.

na

symbolizes the equations are n/a (not applicable) to the adjacent surface, because the surface doesn't have an an upward or downward facing component, as the case requires .

symbolizes plumes of air buoyantly leaving the adjacent surface, and the direction of heat flow, due to Rayleigh-Benard instability.

### 2.5.2.3 Natural convection equations

Equation 37, from Churchill and Chu (see Eq. 4.86, Mills (1992)), is used to determine the natural convection coefficients for tilted surfaces. The choice of this equation is partly informed by the work of Wallenten (2001), which compares the Churchill and Chu equation with other correlations and experimental data.

Equation 37 is for turbulent convection ($10^9 < \mathrm{Ra} < 10^{12}$), expected to be the dominant case in room heat transfer.

Equation 37 applies to either side of a tilted surface for angles ($0 \leq \theta \leq 88^o$) if the heat flow has a downward component, or the heat flow is horizontal.

Equation 37 also applies to either side of a tilted surface for angles $\theta < 60^o$ if the heat flow has an upward component, or the heat flow is horizontal.

$$Nu = 0.68 + 0.67(Ra \cdot \psi)^{0.25}(1 + 1.6 * 10^{-8} Ra \cdot \psi)^{\frac{1}{12}}$$

*Equation 37*

where,

$Ra$ = the Rayleigh number.

$Nu$ = the Nusselt number.

$Pr$ = the Prandtl number.

$$\psi = \left[1 + \left(\frac{0.492}{Pr}\right)^{\frac{9}{16}}\right]^{-\frac{16}{9}}$$

Using $\psi = 0.349$ for $Pr = 0.72$, Equation 37 reduces to:

$$Nu = 0.68 + 0.515Ra^{0.25}(1 + 5.58 * 10^{-9}Ra)^{1/12}$$

*Equation 38*

For high Ra [$Ra \approx> 10^9$] , neglecting the additive terms "1" and "0.68" in Equation 37 gives:

$$Nu = 0.1057Ra^{1/3}$$

*Equation 39*

By the definition of the Nusselt number, the natural convection heat transfer coef, $h_n$ is:

$$h_n \equiv Nu * \frac{k(air)}{L_{char}}$$

*At 70F,* $Ra = 1.66x10^6L^3|\Delta T|cos(\theta)$, *and* $k = 0.0148$, *Equation 39 reduces to:*

$$h_n = 0.185\left(|\Delta T|cos(\theta)\right)^{1/3}$$

*Equation 40*

Note that $h_n$ is independent of characteristic length $L_{char}$.

***Downward heat flow***

According to Mill's(1992) Equation 37 doesn't apply to downward heat flow for $\theta > 88^o$. At 70 F, for a 20 ft characteristic length, the 0.68 term predicted by Equation 38 for $\theta = 90^o$ corresponds to $h_n = \frac{0.68k}{L} = 0.0005$; essentially zero. Although the downward heat flow is ideally stably stratified (three cases shown in Figure 4**Error! Reference source not found.**), most measurements and modeling practice indicate h may be larger than zero. We use the equation of Clear et al. for the minimum, for heat flow down:

$$Nu = 0.27Ra^{0.25}$$

*Equation 41*

Clear's equation reduces to:

$$h_n = 0.27(0.0148)\left(\frac{4}{L}\right)\left(1.642E6\Delta T\left(\frac{L}{4}\right)^3\right)^{0.25}$$

or

$$h_n = 0.202\left|\frac{\Delta \mathrm{T}}{\mathrm{L_{char}}}\right|^{\frac{1}{4}}$$

*Equation 42*

where, $L_{char}$ is the wall characteristic length; see Equation 51 definitions.

Adding this h to Equation 40 gives:

$$h_{down} = MAX\left[0.185\left(|\Delta T|cos(\theta)\right)^{\frac{1}{3}}, 0.202\left|\frac{\Delta \mathrm{T}}{\mathrm{L_{char}}}\right|^{\frac{1}{4}}\right] \qquad 0 \le \theta \le 90$$

*Equation 43*

The following simplification is made, where the exponent of the second term is changed to 1/3, so that $|\Delta T|^{1/3}$ can be factored out:

$$h_{down} = |\Delta T|^{1/3} MAX\left[0.185\left(cos(\theta)\right)^{\frac{1}{3}}, 0.202\mathrm{L}_{\mathrm{char}}^{-\frac{1}{3}}\right] \qquad 0 \le \theta \le 90$$

*Equation 44*

Changing the exponent means Equation 44 gives same answer as Equation 43 only when $\frac{\Delta T}{L_{char}} = 1$. But Equation 44 would have acceptable error for other $\frac{\Delta T}{L_{char}}$ ratios, and gives more or less the right dependence on $\Delta T$. If in addition, one assumes a typical $L_{char} = 15$, say, then the minimum term becomes: $0.202\mathrm{L}_{\mathrm{char}}^{-\frac{1}{3}} = 0.08$, giving the final reasonable form:

$$h_{down} = |\Delta T|^{1/3} MAX\left[0.185\left(cos(\theta)\right)^{\frac{1}{3}}, 0.08\right] \qquad 0 \le \theta \le 90$$

*Equation 45*

***Upward heat flow for $\theta \le 60^o$***

For the inside & outside of walls where the heat flow has an upward (or horizontal heat flow at the limit $\theta = 0^o$), and the outside of roofs, Equation 40 applies:

$$h_n = 0.185\left(|\Delta T|cos(\theta)\right)^{1/3}$$

***Upward heat flow for $\theta > 60^o$***

To handle cases of upward heat flow for $\theta > 60^o$, $h_{up}$ is found by interpolating between Equation 40, evaluated at $\theta = 60^o$, and Equation 47 at $90^o$. Equation 46, for heat transfer from a

horizontal surface ($\theta = 90$), is from Clear et al. (Eq. 11a). It is close to the much used McAdams equation suggested by both the Mills(1992) and Incropera-Dewitt textbooks.

$$Nu = 0.15Ra^{1/3}$$

*Equation 46*

At 70-F, Equation 46 reduces to

$$h_n = 0.26(\Delta T)^{1/3}$$

*Equation 47*

Interpolating, for upward heat flow cases with $\theta \geq 60^o$:

$$h_{up} = 0.185\left(\Delta T cos(60)\right)^{1/3} + \frac{\left(0.26(\Delta T)^{1/3} - 0.185\left(\Delta T cos(60)\right)^{1/3}\right)(\theta - 60)}{30}$$

which reduces to:

$$h_{up} = (0.00377 * \theta - 0.079)|\Delta T|^{\frac{1}{3}} \qquad \text{for } 60^o \leq \theta \leq 90$$

*Equation 48*

where $\theta$ is in degrees.

### 2.5.2.4 Inside forced convection equation

Measured forced convection heat transfer coefficients are frequently correlated using an equation of the form

$$h_{ach} = h_{forcedIN} = C_{ach} * ACH^{0.8}$$

*Equation 49*

The RBH model (Barnaby et al. (2004) suggests using $h_f = 0.88$ Btu/hr-ft$^2$F at ACH = 8. This gives $C_{ach} = 0.167$. Walton (1983) assumes h = 1.08 when the "air handler system is moving air through the zone." If this was at 8 ach, then this implies $C_{ach} = 0.205$.

### 2.5.2.5 Outside forced convection equation for all walls and all roofs

From Clear et al. (2001, Eq. (11a)),

$$Nu = W_f R_f 0.037 Re^{0.8} Pr^{1/3}$$

*Equation 50*

Clear et al. used the Reynolds number based on a free-stream wind velocity 26.2 ft (8 m) above the ground.

At 70F, Equation 50 reduces to:

$$h_V = k * \frac{Nu}{L} = 0.527 W_f R_f \frac{V^{0.8}}{L^{0.2}}$$

*Equation 51*

where for walls,

$L =$ wall $L_{char} = 4\frac{Wall\ Area}{Wall\ Perimeter}$ $\left(= 4\frac{Height*Width}{2(Height+width)} \approx height\ of\ square\ wall = Z_{eave}\right)$

$V =$ wind velocity at eave height at building location, in ft/sec, $= 0.16 * Z_{eave}^{0.25} * V_{met}$ from Section 2.5.1.2.

$V_{met} =$ freestream wind velocity, in ft/sec, 10 m (32.8 ft) above the ground at the meteorological station site.

$R_f =$ Table 3 value.

$W_f = 0.63$

The wind direction multiplier, $W_f$, is defined as the average $h$ of all of the vertical walls, divided by the $h$ of the windward wall, with this ratio averaged over all wind directions. We estimated $W_f$ using the CFD and wind tunnel data of Blocken et al. (2009) for a cubical house. Blocken's Table 6 gives a windward surface convection coefficient of $h_c \approx 4.7V^{0.84}$ (SI units), averaged over wind direction. Blocken's Figure 9 gives $\overline{h_c} \approx 7.5$ averaged over all vertical surfaces, for wind speed $V_{met} = 3$-m/s. Thus, we estimate $W_f = \frac{7.5}{4.7V^{0.84}} = 0.63$.

and for roofs,

$L =$ roof $L_{char} = 4\frac{Roof\ Plan\ Area}{Roof\ Perimeter}$ $\left(= 4\frac{Length*Width}{2(Length+width)} \approx \sqrt{Roof\ Area}\ for\ square\ roof \approx Z_{eave}\right)$

$V =$ wind velocity 9.8 ft (3 m) above the eave height at building location, in ft/sec.

$= 0.16 * (Z_{eave} + 9.8)^{0.25} * V_{met}$

$W_f = 1$

$R_f =$ Table 3 value.

Walton (1983) assumed that the ASHRAE roughness factors of Table 3 apply to the convection coefficient correlations. The Clear et al. (2001) experiments tend to confirm the validity of these factors. Blocken et al. (2009) says, "The building facade has been assumed to be perfectly smooth. Earlier experimental studies have shown the importance of small-scale surface roughness on convective heat transfer. For example, Rowley et al. found that the forced convection coefficient for stucco was almost twice that for glass. Other studies showed the important influence of larger-scale surface roughness, such as the presence of mullions in glazed areas or architectural details on the facade, on the convection coefficient."

**Table 3: Surface Roughness Parameter $R_f$ (Walton 1981)**

| Roughness Index | Rf | Example |
|---|---|---|
| 1 (very rough) | 2.1 | Stucco |
| 2 (rough) | 1.67 | Brick |
| 3 (medium rough) | 1.52 | Concrete |
| 4 (Medium smooth) | 1.13 | Clear pine |
| 5 (Smooth) | 1.11 | Smooth plaster |
| 6 (Very Smooth) | 1 | Glass |

### 2.5.2.6 Inside combined natural and forced convection

The combined convection coefficient is assumed to be the direct sum of the natural and forced convection coefficients:

For upward and horizontal heat flow:

$$h_{combined} = h_{up} + h_{ach}$$

*Equation 52*

where,

$h_{up} =$ Equation 40 or Equation 48 depending on whether $\theta$ is < or > 60°.

$h_{ach} =$ Equation 49

For downward heat flow:

$$h_{combined} = h_{down} + h_{ach}$$

*Equation 53*

where,

$h_{down} =$ Equation 45

$h_{ach} =$ Equation 49

### 2.5.2.7 Outside combined natural and forced convection

The conclusion of Clear et al. (2001) is that the combined convection coefficient is best correlated by assuming it to be the sum of the natural and forced coefficients. For roofs, Clear et al. (2001) assumes that the natural and forced convection are additive, but that natural convection is suppressed by the factor $\eta$ given by Equation 56 when forced convection is large ($\eta \rightarrow 0$ as the

Reynolds number becomes large). We also assume this attenuation of the natural convection applies to the outside of the walls.

For all vertical walls, and walls with moderate tilts:

$$h_{combined} = \eta h_n + h_V$$

*Equation 54*

where,

$h_n$ = Equation 40

$h_V$ = Equation 51

$\eta = \frac{1}{1+\frac{1}{\ln\left(1+\frac{0.06L|\Delta T|}{V^2}\right)}}$(to avoid divide by zero, if V= 0, could set to V = 0.001)

where $L$ & $V$ are the same as used in Equation 51 for walls.

For roofs, Clear et al. (2001) assumes that the natural and forced convection are additive, but that natural convection is suppressed by the factor $\eta$ when forced convection is large ($\eta \rightarrow 0$ as the Reynolds number becomes large). Clear gives $\eta$ as:

$$\eta = \frac{1}{1+\frac{1}{\ln\left(1+\frac{Gr_L}{Re_L^2}\right)}}$$

*Equation 55*

At 70F, with $Gr = 2.28x10^6 L^3|\Delta T|$, $\mathrm{Re}^2 = (6140\mathrm{VL})^2$, and $\mathrm{L} = \mathrm{L_{char}}$ for surface , Equation 55 reduces to:

$$\eta = \frac{1}{1+\frac{1}{\ln\left(1+\frac{0.06L|\Delta T|}{V^2}\right)}}$$

*Equation 56*

For roofs:

$$h_{combined} = \eta h_n + h_V$$

*Equation 57*

where,

$h_n =$ Equation 45 for downward heat flow.

$h_n =$ Equation 47 for upward heat flow.

$h_V =$ Equation 51 for upward or downward heat flow.

η is from Equation 56

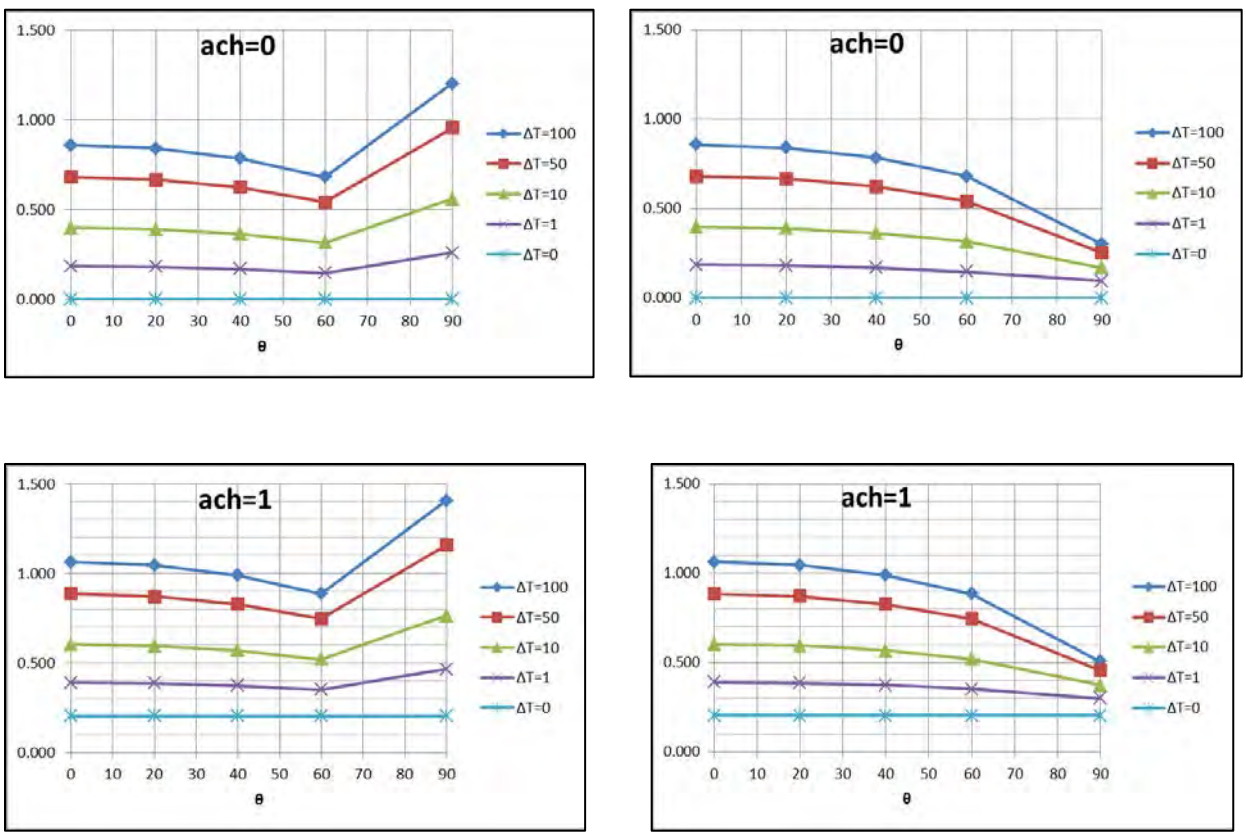
$L$ & $V$ are the same as used in Equation 51 for roofs.

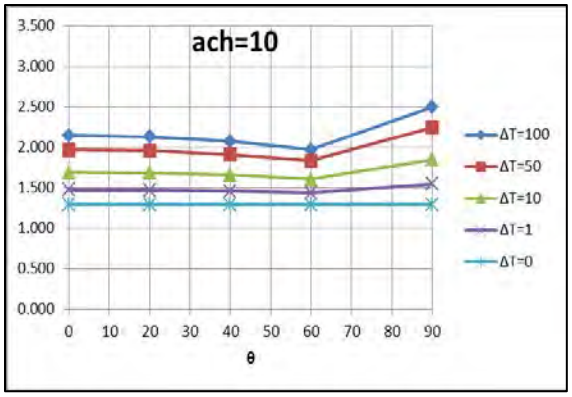
### 2.5.2.8 Plots of equations

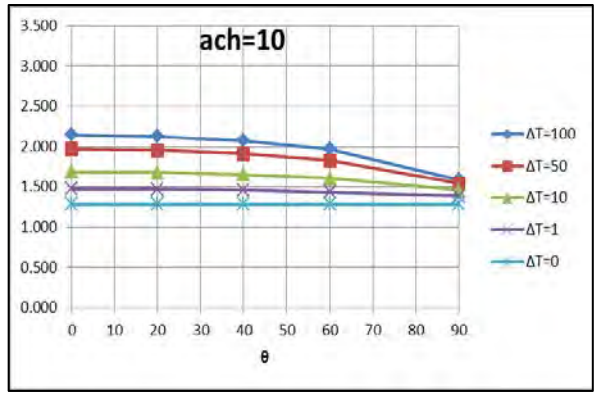
In Figure 6, the left hand column of plots are of Equation 53, for the downward heat flow cases shown in Figure 4. The right hand side plots are of Equation 52, for upward heat flow cases of Figure 5. All of the plots assume $T_{film} = 70F$, and $L_{char} = Z_{eave} = 20$ ft.

**Figure 6: Plots of Equations for Downward and Upward Heat Flow**

*DOWNWARD HEAT FLOW (Equation 53):* *UPWARD HEAT FLOW (Equation 52):*



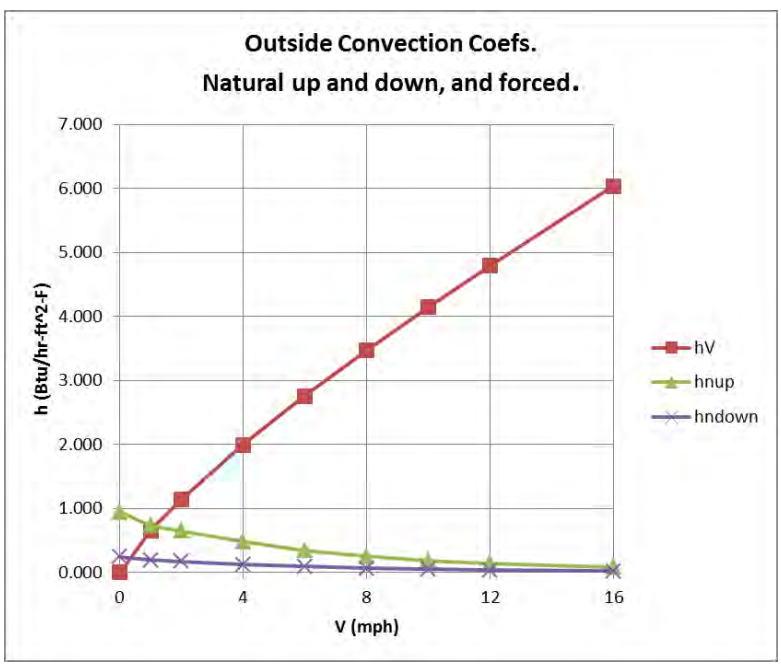



<u>OUTSIDE surface convection coefficient Plot for a horizontal roof:</u>

For $\Delta T = 50\ F$, $L_{char} = 20$ ft, $R_f = 1.67$:

**Figure 7: Outside Convection Coefficients, Natural Up and Down, and Forced**




## 2.5.3 Outside Radiation Coefficients

### 2.5.3.1 Wall surfaces

The net long wave radiation heat exchange between the outside surface and the environment is dependent on surface temperature, the spatial relationship between the surface and the surroundings, and the properties of the surface. The relevant material properties of the surface, emissivity $\varepsilon$ and absorptivity $\alpha$, are complex functions of temperature, angle, and wavelength. However, it is generally assumed in building energy calculations that the surface emits or reflects diffusely and is gray and opaque ($\alpha = \varepsilon, \tau = 0, \rho = 1 - \varepsilon$).

The net radiant heat loss from a unit area of the outside of a construction surface to the outside environment is given by:

$$q_{rad} = \varepsilon\epsilon_g\sigma F_{gnd}(T_s^4 - T_g^4) + \varepsilon\sigma F_{sky}\beta\left(T_s^4 - T_{sky}^4\right) + \varepsilon\sigma F_{sky}(1-\beta)(T_s^4 - T_a^4)$$

*Equation 58*

where,

$\varepsilon$ = surface emissivity.

$\epsilon_g$ = ground emissivity is assumed to be 1.

$\sigma$ = Stephan-Boltzmann constant.

$T_s$ = outside surface temperature.

$T_a$ = outside dry bulb temperature.

$T_g$ = ground surface temperature.

$T_{sky}$ = effective temperature of sky.

$F_{gnd}$ = view factor from surface to ground.

$F_{sky}$ = view factor from surface to sky.

$\beta = \cos\left(\frac{\phi}{2}\right)$

The sky irradiance is taken as a $\beta$ weighted average of that from $T_{sky}$ and that from $T_a$.

### 2.5.3.2 Fsky, Fgnd, and β

Howell (1982, #C-8, p.94), gives the fraction of the radiation leaving the window surface and reaching the sky by:

$$F_{sky} = \frac{1 + \cos\phi}{2}$$

The fraction leaving the window incident on the ground is:

$$F_{gnd} = \frac{1 - \cos\phi}{2}$$

where, $\varphi$ = surface tilt angle, the angle between ground upward normal and window outward normal ($0^o$ corresponds to a horizontal skylight, $90^o$ to a vertical surface).

The parameter β accounts for the sky temperature's approach to the air temp near the horizon. β is the fraction of the sky effectively at Tsky; (1-β) is the fraction of the sky effectively at Ta. β is used by Walton (1983), and Energy Plus (2009), but appears to have little theoretical or experimental basis.

Walton (1983) give β as:

$$\beta = \cos\left(\frac{\phi}{2}\right)$$

Since $\cos\left(\frac{\varphi}{2}\right) = \sqrt{\frac{1+\cos\varphi}{2}}$, it is noted that $F_{sky} = \beta^2$, and $F_{sky}\beta = \beta^3$.

#### 2.5.3.3 Net radiant heat loss from a unit area

Equation 58 can be written as

$$q_{rad} = h_{rg}(T_s - T_g) + h_{rsky}(T_s - T_{sky}) + h_{rair}(T_s - T_a)$$

*Equation 59*

where,

$$h_{rg} = \varepsilon\sigma F_{gnd}(T_s^2 + T_g^2)(T_s + T_g)$$

$$h_{rsky} = \epsilon\sigma\, F_{sky}\beta(T_s^2 + T_{sky}^2)(T_s + T_{sky})$$

$$h_{rair} = \varepsilon\sigma\, F_{sky}(1-\beta)(T_s^2 + T_a^2)(T_s + T_a).$$

$T_g$ is assumed to be equal to $Ta$, so Equation 59 becomes

$$q_{rad} = h_{rsky}(T_s - T_{sky}) + h_{ra}(T_s - T_a)$$

*Equation 60*

where,

$$h_{rsky} = \epsilon\sigma\, F_{sky}\beta\,(T_s^2 + T_{sky}^2)(T_s + T_{sky})$$

*Equation 61*

$$h_{rair} = \varepsilon\sigma\,(F_{sky}(1-\beta) + F_{gnd})(T_s^2 + T_a^2)(T_s + T_a)$$

*Equation 62*

For a vertical surface, $F_{sky}\beta = 0.354$ , and $F_{sky}(1-\beta) + F_{gnd} = 0.646$, so

$$h_{rsky} = 0.354\epsilon_s\sigma\,(T_s^2 + T_{sky}^2)(T_s + T_{sky}) \approx 4(0.354)\varepsilon_s\sigma\bar{T}^3$$

$$h_{rair} = (0.146 + 0.5)\varepsilon_s\sigma(T_s^2 + T_a^2)(T_s + T_a) \approx 4(0.646)\varepsilon_s\sigma\bar{T}^3$$

#### 2.5.3.4 Total effective conductance and outside effective temperature, Tenv, for walls

Adding the exterior convection coefficient, $hco$, of Equation 40 to Equation 60 gives the total net heat transfer from the outside surface :

$$q_{rad+conv} = h_{rsky}(T_s - T_{sky}) + (h_{rair} + h_{co})(T_s - T_a)$$

*Equation 63*

This can be written as,

$$q_{rad+conv} = h_o(T_s - T_{env})$$

*Equation 64*

where $h_o$ is the effective exterior conductance to the conductance weighted average temperature, $T_{env}$.

$$h_o = h_{rsky} + h_{rair} + h_{co}$$

*Equation 65*

$$T_{env} = \frac{h_{rsky}T_{sky} + (h_{rair} + h_{co})T_a}{h_{rsky} + h_{rair} + h_{co}}$$

*Equation 66*

#### 2.5.3.5 Outside window surfaces

The ASHWAT window algorithm of Section 2.7 utilizes the irradiation intercepted by the window. From Equation 58 this can be deduced to be:

$$G = F_{gnd}\sigma T_{gnd}^4 + F_{sky}\beta\sigma T_{sky}^4 + F_{sky}(1-\beta)\sigma T_{air}^4$$

*Equation 67*

### 2.5.4 Sky Temperature

It is possible to approximate the long wave radiation emission from the sky as a fraction of blackbody radiation corresponding to the temperature of the air near the ground. The sky emittance $\varepsilon_{sky}$ is defined such that the sky irradiation on a horizontal surface is $\sigma\varepsilon_{sky}T_a^4$.

The effective temperature of the sky is obtained by equating the blackbody emissive power of the sky at $T_{sky}$, to the sky irradiation:

$$\sigma\, T_{sky}^4 = \sigma\varepsilon_{sky}T_a^4$$

or,

$$T_{sky} = \varepsilon_{sky}^{0.25}T_a,$$

*Equation 68*

where $T_{sky}$ and $T_a$ are in degrees Rankine.

The value of $\varepsilon_{sky}$ depends on the dewpoint temperature, cloud cover, and cloud height data. Martin and Berdahl (1984) give the $\varepsilon_{sky}$ for clear skies as $\varepsilon_o$:

$$\varepsilon_o = 0.711 + 0.56\frac{T_{dew}}{100} + 0.73\left(\frac{T_{dew}}{100}\right)^2 + 0.013\cos\left(\pi\frac{hr}{12}\right) + 0.00012\,(P_{atm} - 1000)$$

*Equation 69*

where,

$T_{dew}$ = the dewpoint temperature in Celsius.

$hr$ = hour of day (1 to 24).

$P_{atm}$ = atmospheric pressure in millibars.

#### 2.5.4.1 Palmiter version of Martin-Berdahl model

The clear sky emissivity is corrected to account for cloud cover by the following algorithm, developed by Larry Palmiter (with Berdahl's imprimatur), that represents the Martin and Berdahl model when weather tape values of cloud ceiling height, and total and opaque cloud fractions are available.

$$\epsilon_{sky} = \varepsilon_0 + (1 - \varepsilon_0)(n_{op}\varepsilon_{op}\Gamma_{op} + n_{th}\varepsilon_{th}\Gamma_{th})$$

*Equation 70*

where,

$n_{op}$ = the opaque cloud fraction

$n_{th}$ = the thin cloud fraction: $n_{th} = n - n_{op}$

n = the total sky cover fraction

$\varepsilon_{op}$ = the opaque cloud emittance is assumed to be 1.

$\varepsilon_{th}$ = the thin cloud emittance; assumed to be 0.4.

The cloud factor $\Gamma$ is used to adjust the emissivity when the sky is cloudy due to the increasing cloud base temperature for decreasing cloud altitudes. The cloud base temperature is not available on the weather tapes, so assuming a standard lapse rate of 5.6°C/km, $\Gamma$ is correlated with the more commonly measured cloud ceiling height, h (in meters), giving by the general expression:

$$\Gamma = e^{-\frac{h}{8200}}$$

For thin clouds, $\Gamma_{th}$ is determined using an assumed cloud height of 8000-m, so,

$$\Gamma_{th} = e^{-\frac{8000}{8200}} = 0.377$$

*Equation 71*

For opaque clouds,

$$\Gamma_{op} = e^{-\frac{h}{8200}}$$

*Equation 72*

If ceiling height data is missing (coded 99999 on TMY2), the Palmiter model assumes that the opaque cloud base is at $h = 2000\ m$. If ceiling height is unlimited (coded as 77777) or cirroform (coded 88888), it is assumed that the opaque cloud base is at $h = 8000\ m$.

Using the assumed cloud cover and emissivity factors, Equation 70 becomes:

$$\epsilon_{sky} = \epsilon_0 + (1 - \epsilon_0)\left[n_o\Gamma_{op} + (n - n_o) * 0.4 * 0.377\right]$$

or,

$$\epsilon_{sky} = \epsilon_0 + (1 - \epsilon_0)\left[n_o e^{-\left(\frac{h}{8200}\right)} + 0.151(n - n_o)\right]$$

*Equation 73*

### 2.5.4.2 When opaque cloud cover data, $n_o$, is missing

In this case it is assumed that the cloud cover is opaque, $n_o = n$, when the ceiling height is less than 8000-m, and half opaque, $n_o = \frac{n}{2}$, when the ceiling height is equal or greater than 8000. That is,

for $h < 8000\ m$ (from Equation 73 with $n_o = n$):

$$\epsilon_{sky} = \epsilon_0 + (1 - \epsilon_0)ne^{-\left(\frac{h}{8200}\right)}$$

*Equation 74*

For $h \geq 8000\ m$ (from Equation 73 with $n_{op} = n_{th} = \frac{n}{2}$ ):

$$\epsilon_{sky} = \epsilon_0 + (1 - \epsilon_0)n\left[\frac{1}{2}e^{-\left(\frac{h}{8200}\right)} + 0.0754\right]$$

*Equation 75*

### 2.5.4.3 When both opaque cloud cover and ceiling height data is missing

When only total sky cover is available using an h of 2000-m reduces Equation 74 to:

$$\epsilon_{sky} = \epsilon_0 + 0.784(1 - \epsilon_0)n$$

*Equation 76*

# 2.6 Distribution of SW and LW Radiation inside the Zone

## 2.6.1 Long Wave Radiation Distribution

### 2.6.1.1 Carroll model

The radiant model used in CSE is based on the "MRT Network Method" developed by Joe Carroll (see Carroll 1980 & 1981, and Carroll & Clinton 1980 & 1982). It was chosen because it doesn't require standard engineering view factors to be calculated, and yet gives a relatively accurate radiant heat distribution for typical building enclosures (see Carroll 1981).

It is an approximate model that simplifies the "exact" network (see Appendix C) by using a mean radiant temperature node, Tr, that act as a clearinghouse for the radiation heat exchange between surfaces, much as does the single air temperature node for the simple convective heat transfer models. For n surfaces this reduces the number of circuit elements from (n-1)! in the exact case, to n with the Carroll model.

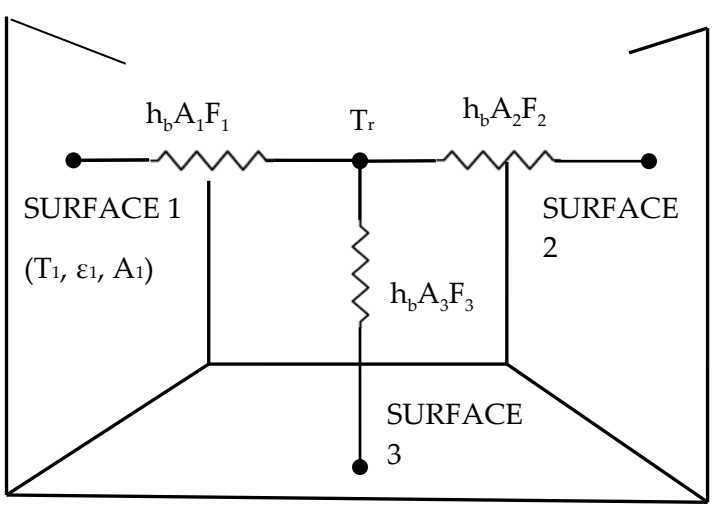
For black surfaces the radiant network is shown in Figure 8. For n surfaces, $T_r$ floats at the conductance, $A_iF_i$, weighted average surface temperature:

$$T_r = \frac{\sum_1^n A_i F_i T_i}{\sum_1^n A_i F_i}$$

*Equation 77*

The actual areas, $A_i$ , need not be equal, nor limited to three.

**Figure 8: Carroll Network for Black Surfaces**



The factor $F_i$, in the radiant conductance between the $T_i$ surface node and the $T_r$ node is Carroll's "MRT view factor", that corrects for the self-weighting (see Appendix D) of $T_i$ in the temperature $T_r$. The $F_i$ factors are obtained from the set of n nonlinear equations for n surfaces:

$$F_i = \frac{1}{1 - \frac{A_i F_i}{\sum_1^n A_j F_j}}$$

*Equation 78*

Given the surface areas, these equations are easily solved at the beginning of the simulation by successive substitution, starting with all $F_i = 1$. This converges for realistic enclosures, but won't necessarily converge for enclosures having only two or three surfaces, particularly if there are large area disparities.

> $F_i$ is always larger than 1 because it's role is to raise the conductance between $T_r$ and $T_i$ to compensate for the potential difference $|T_r - T_i|$ being smaller than it would be had $T_i$ not been part of the conductance weighted average $T_r$. The $F_i$ values can be seen to be close to 1, since

Equation 78 is roughly approximated by $F_i \approx 1 + (A_i / A_{all\ surfaces})$.

The net radiant heat transfer [Btu/hr] from surface $i$ is:

$$q_i = h_b A_i F_i (T_i - T_r)$$

*Equation 79*

Using a Y-Δ transformation, the Figure 8 circuit can put in the form of the exact solution network of Figure C-1 in Appendix C, showing the implicit view factors $F_{ij}$ to be:

$$F_{ij} = \frac{F_i A_j F_j}{\sum_{k=1}^n A_k F_k}$$

*Equation 80*

Thus the implicit view factors are independent of the relative spacial disposition of the surfaces, and almost directly proportional to the surface area $Aj$ of the viewed by surface $i$. Also, without special adjustments (see Carroll (1980a)), all surfaces see each other, so coplanar surfaces (a window and the wall it is in) radiate to each other.

> Equation 79 is exact (i.e., gives same answers as the Appendix C model) for cubical rooms; for which

Equation 78 gives $F_i = 1.20$. Substituting this into Equation 80 gives the implied $F_{ij} = 0.2$. This is the correct $F_{ij}$ for cubes using view-factor equations Howell(1982). It is likely accurate for all of the regular polyhedra.

***Grey surfaces***

Carroll's model handles gray surfaces, with emissivities $\varepsilon_i$, by adding the Oppenheim surface conductance $\frac{A_i\varepsilon_i}{1-\varepsilon_i}$ in series with the conductances $h_b A_i F_i$ . As shown in Figure 9, the conductance between $T_i$ and $T_r$ becomes $h_b A_i F_i'$, where the $F_i'$ terms are:

$$F_i' = \frac{1}{\frac{1}{F_i} + \frac{1-\varepsilon_i}{\varepsilon_i}}$$

*Equation 81*

The net radiant heat transfer [Btu/hr] from surface $i$ is given by:

$$q_i = h_b A_i F_i' (T_i - T_r)$$

*Equation 82*

where for grey surfaces $T_r$ is the "$h_b A_i F_i'$" weighted average surface temperature given by:

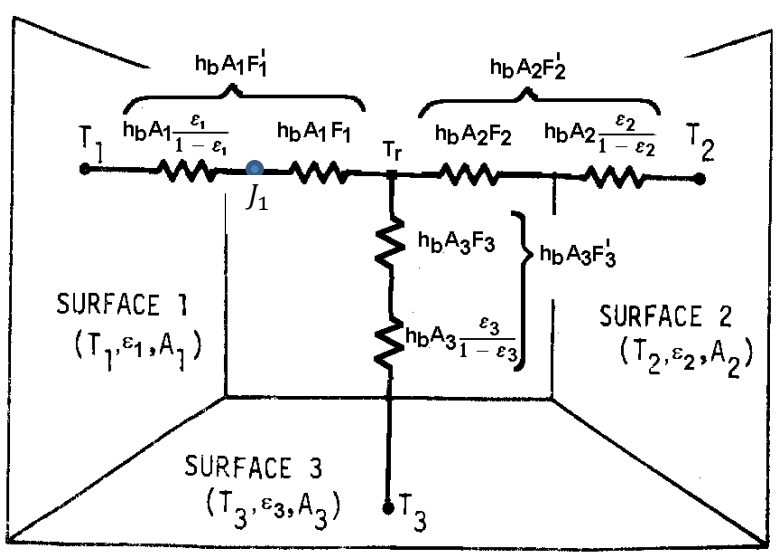
$$T_r = \frac{\sum_1^n A_i F_i' T_i}{\sum_1^n A_i F_i'}$$

*Equation 83*

Similar to Equation 77 for a black enclosure, Equation 83 shows that $T_r$ for grey surfaces is the conductance, $A_i F_i'$, weighted average surface temperatures.

The role of $F_i$ hasn't changed, but since the conductance $A_i F_i$ is now connected to the radiosity node rather than the surface node, $E_r (= \sigma T_r^4)$ can be thought of as the $A_i F_i$-weighted average radiosity of the surfaces, rather than the $A_i F_i$-weighted average emissive power of the surfaces as in the black enclosure case.

**Figure 9: Carroll Radiant Network for Grey Surfaces**



This completes the description of the basic Carroll model. The principle inputs are the interior surface areas in the zone, the emissivities of these surfaces, and the typical volume to surface area ratio of the zone (see Section 2.6.1.3). All of the interior surfaces, including ducts, windows, and interior walls, are assumed to exchange heat between each other as diffusely radiating gray body surfaces.

Longwave radiant internal gains can be added, in Btu/hr, to the radiant node Tr. This distributes the gains in proportion to the conductance $A_i F_i'$.

***Conversion to delta***

Using a Y-Δ transformation, the radiant network of Figure 9 can be converted to the Appendix C, Figure C-3 circuit form, eliciting the $F_{ij}'$ interchange factors implicit in Carroll's algorithm. Similar in form to Equation 80,

$$A_i F_{ij}' = \frac{A_i F_i' A_j F_j'}{\sum_{k=1}^{n} A_k F_k'}$$

*Equation 84*

Using these $A_i F_{ij}'$ values, $q_{ij}$ can be obtained from

$$q_{ij} = h_b A_i F_{ij}' (T_i - T_j)$$

*Equation 85*

The total net heat transfer from surface *i* (i.e., the radiosity minus the irradiation for the un-linearized circuit) is given by summing Equation 85 for all the surfaces seen by surface i:

$$q_i = \sum_{j=1}^{n} h_b A_i F_{ij}' (T_i - T_j)$$

*Equation 86*

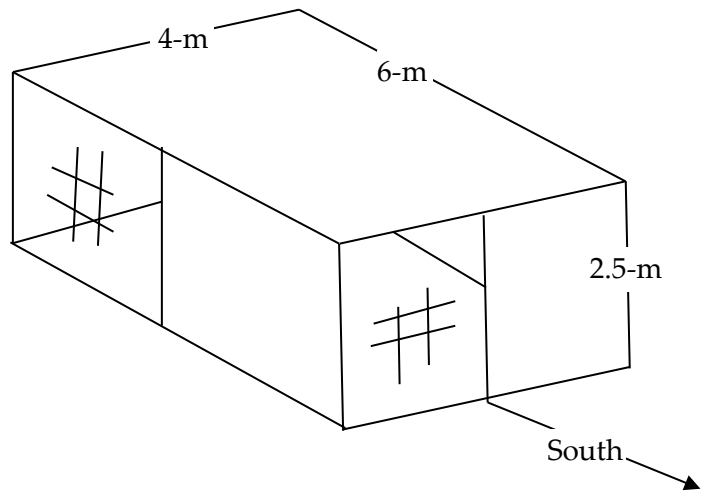
which will agree with the result of Equation 82.

### 2.6.1.2 Accuracy of Carroll model

The Carroll model of Figure 9 is exact for cubical enclosures with arbitrary surface emissivities. It is surprisingly accurate for a wide variety of shapes, such as hip roof attics and geodesic domes.

Carroll (1981) compared his model, and other simplified models, with the exact solution for the enclosure shown in Figure 10. Half the south wall and half of the west wall are glass with ε = 0.84, and the rest of the interior surfaces have ε = 0.9.

**Figure 10: Test Room of Walton (1980)**




Comparisons were made primarily regarding three types of errors:

***Heat balance errors***

The first law requires that the sum of the net radiation emitted by each of the surfaces, plus any internal gain source of long-wave radiation, must equal zero. That is, $q_{int} + \sum_{i=1}^{n} q_i = 0$.

Due to their fixed conductance circuits, both the Carroll method and the Walton(1983) method are inherently free of heat balance errors. Carroll found BLAST and NBSLD algorithms to have rms heat balance errors of 9.8% (12%) and 1.7%(3.4%) for the Figure 10 enclosure.

***Individual surface net heat transfer errors***

For a given enclosure, these are errors in an individual surfaces net heat flow, $q_i$, compared to the exact method. For Carroll's method, this finds the error in $q_i$ determined from the $A_i F'_{ij}$ values ofEquation 84, compared to the $q_i$ values found using the exact $A_i F'_{ij}$ values (obtainable from Figure C-3 of Appendix C).

Carroll found the % rms error in the $q_i$ values for a given enclosure in two different ways.

The first method, Equation 87, gives the rms error of $q_i$ for each surface divided by the rms of the $n$ net heat transfers from each surface:

$$Err = \left( \frac{\frac{1}{n}\sum_{i=1}^{n} \Delta q_i^2}{\frac{1}{n}\sum_{i=1}^{n} \bar{q}_i^2} \right)^{\frac{1}{2}} * 100$$

*Equation 87*

where

$\Delta q_i = q_i - \bar{q}_i$, is the error in $q_i$.

$q_i = \sum_{j=1}^{n} h_b A_i F'_{ij}(T_i - T_j)$ , using $F'_{ij}$ values from Carroll's model, Equation 87.

$\bar{q}_i = \sum_{j=1}^{n} h_b A_i F'_{ij}(T_i - T_j)$ , using the exact $F'_{ij}$ values of Figure C-3 of Appendix C

$T_i - T_j = 1^0 F$ assumed in all cases.

$n =$ the number of surfaces

The second method, Equation 88, gives the rms of the percentage error in $q_i$ of each surfaces. This method increases the weight of smaller surfaces such as windows.

$$Err = \frac{1}{n}\sum_{i=1}^{n}\left(\frac{\Delta q_i}{q_i}\right)^2 * 100$$

*Equation 88*

***Results***

For the enclosure of Figure 10, Carroll found his method gives *Err* = 0.11% for the first method and 0.19% for the second method.

These results are shown in Table 4, along with the results for other shape enclosures, and the errors determined by Carroll using the radiant interchange algorithms of Walton (1980) and NBSLD and BLAST simplified models.

**Table 4: $Err$ = % rms Error in $q_i$ from Equation 87 and Equation 88 in Parenthesis**

| | Figure 10 room<br>2.5:4:6<br>ε = 0.9 (.84 wdws) | Corridor<br>10:1:1<br>ε = 0.9 | Warehouse<br>10:10:1<br>ε = 0.9 |
|---|---|---|---|
| Carroll | 0.11 (0.19) | 0.06(0.05) | 0.07 (0.04) |
| Walton(1980) | 1.9 (1.30) | 0.6 (0.6) | 4.4 (3.0) |
| NBSLD, BLAST | 3.2 (2.1) | 3.2 (2.6) | 7.5 (4.4) |

***Errors in an individual surface's distribution of heat transfer to other surfaces***

These are errors in $q_{ij}$, the heat exchanged between surfaces i and j (both directly and by reflections from other surfaces), relative to the exact total net heat transfer from surface *i* given by Figure C-3 of Appendix C.

Carroll gives two percentage error results.

By the first method, for each surface *i*, the rms of the error, $\Delta q_{ij}$, in heat exchange to each of the *n-1* other *j* surfaces is obtained. Then the rms of these *n* rms error values is obtained, giving a

representative distribution error for the enclosure. Dividing this by the rms value of the exact net surface heat transfers, $q_i$, of all the surfaces gives the final distribution error in percent:

$$Err = \frac{\sqrt{\sum_{i=1}^{n}\left(\frac{\sum_{j=1}^{n}\Delta q_{ij}^2}{n(n-1)}\right)}}{\sqrt{\sum_{i=1}^{n}\left(\frac{\bar{q}_i^2}{n}\right)}} * 100$$

*Equation 89*

where

$\Delta q_{ij} = q_{ij} - \bar{q}_{ij}$,

$q_{ij} = h_b A_i F'_{ij}(T_i - T_j)$ with $F'_{ij}$ values from Equation 84.

$\bar{q}_{ij} = h_b A_i F'_{ij}(T_i - T_j)$ with the exact $F'_{ij}$ values of Figure C-3 of Appendix C.

By the second method, for each surface $i$, the rms of the percentage error in heat exchange $q_{ij}$, relative to the exact net heat transfer from that surface, $\bar{q}_i$, is obtained.

$$Err = 100 * \sqrt{\sum_{i=1}^{n}\left(\frac{\sum_{j=1}^{n}\left(\frac{\Delta q_{ij}}{\bar{q}_i}\right)^2}{n(n-1)}\right)}$$

*Equation 90*

***Distribution error results***

For the Figure 10 room, Carroll's model gives errors of 2.1% and 3.9% for methods 1 and 2 respectively. Walton's model has corresponding errors of 2.4% and 3.7%. Equation 91 was used for the results in parenthesis.

**Table 5: % rms Error in $q_i$ from Equation 90**

| | Figure 10 room<br>2.5:4:6<br>ε = 0.9 (.84 wdws) | Corridor<br>1:10:1<br>ε = 0.9 | Warehouse<br>1:10:10<br>ε = 0.9 |
|---|---|---|---|
| Carroll | 2.1 (3.9) | 3.3 (2.8) | 0.6 (1.9) |
| Walton(1980 ) | 2.4 (3.7) | 3.3 (2.8) | 2.8 (3.0) |
| BLAST | 2.8 (4.4) | 3.4 (4.4) | 3.4 (15) |
| NBSLD | 1.7 (3.5) | 1 (0.83) | 3.3(1.9) |

(Equation 91 was used for the results in parenthesis.)

Carroll's model is seen to give very respectable results, despite giving no special treatment to coplanar surfaces.

### 2.6.1.3 Air absorption

The Carroll model also accounts for the absorption of long-wave radiation in the air, so that the air and mrt nodes are thermally coupled to each other as well as to the interior surfaces. Carroll (1980a) gives an air emissivity by the following dimensional empirical equation that is based on Hottel data from McAdams(1954):

$$\varepsilon_a = 0.08\varepsilon_s \ln\left(1 + \left(\frac{4V}{\varepsilon_s A} R P_{atm}\right) e^{(TaF-32)/30.6}\right)$$

*Equation 91*

The logarithm is natural, and,

$\varepsilon_s$ = the area-weighted average long-wave emissivity for room surfaces, excluding air.

V/A = the room volume to surface area ratio, in meters.

R = the relative humidity in the zone. (0 ≤ R ≤ 1).

Patm = atmospheric pressure in atmospheres.

Ta = zone air temperature, in °F.

Following a heuristic argument Carroll assigns an effective area $A_a$ to the air that is the product of $\varepsilon_a$ and the sum of all of the zone surface areas, as if the absorbing part of the air were consolidated into a surface of area $A_a$.

$$A_a = \varepsilon_a \sum A_i$$

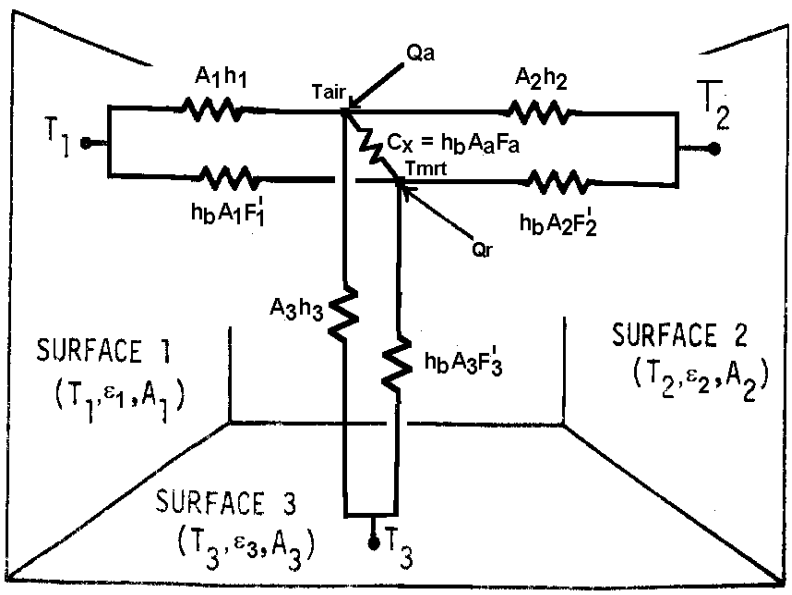
*Equation 92*

Using this area, the value of $F_a$ for this 'surface' can be calculated along with the other $F_i$ by Equation 80. The value of the conductance between the air and radiant nodes in Figure 11 is given by:

$$C_x = h_b A_a F_a$$

*Equation 93*

**Figure 11: Like Figure 9 but with Convective Network Added**




***Facets***

Suppose one of the interior surfaces of total area $A_i$ is composed of $N_i$ identical flat sub-surfaces, each at the same temperature, and similar views to each other, like the facets of a geodesic dome. The $F_i$ values would be the same if each facet is treated as a separate surface. To avoid redundant solutions to Equation 80, it is easy to show that $A_i$ can be treated as one surface in Equation 6-4 if $N_i$ is introduced into Equation 80 as follows:

$$F_i = \frac{1}{1 - \frac{A_i \cdot F_i / N_i}{\Sigma(A_i \cdot F_i)}}$$

*Equation 94*

The facet feature is utilized in the simulation to represent attic truss surfaces.

Short Wave Radiation Distribution

This routine was used in the development code for this program. It is not currently implemented in CSE, being replaced by a simplified but similar routine.

The short wave radiation (solar insolation from hourly input) transmitted by each window can, at the users discretion, be all distributed diffusely inside the zone, or some of the insolation from each window can be specifically targeted to be incident on any number of surfaces, with the remaining untargeted radiation, if any, from that window, distributed diffusely. The insolation incident on any surface can be absorbed, reflected, and/or transmitted, depending on the surface properties inputted for that surface. The radiation that is reflected from the surfaces is distributed diffusely, to be reflected and absorbed by other surfaces ad infinitum.

Since some of the inside surfaces will be the inside surface of exterior windows, then some of the solar radiation admitted to the building will be either lost out the windows or absorbed or reflected by the widows.

#### 2.6.1.4 Radiation removed at each surface of a zone by a single source of targeted insolation

Assume a spherical zone with total insolation S(Btu/hr) admitted into the zone through one window. Assume that the portion $a_i Q_i$ (ft²*Btu/(hr-ft²)) of $S$(Btu/hr) is targeted to surface i with area $a_i$ such that,

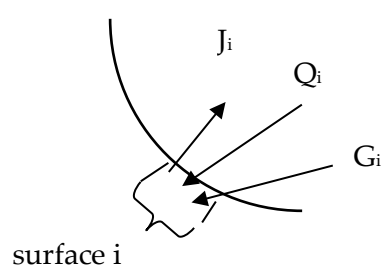
$$\sum_i a_i Q_i = S$$

*Equation 95*

where the sum is over all surfaces *i*. The total spherical area is $a_S = \sum_i a_i$. Also incident on surface i will be the irradiation $G_i$ (Btu/hr-ft²) from other surfaces that have reflected a portion of the radiation they have received. We distinguish between the $Q_i$ incident on the surface directly from the window, and the irradiation $G_i$ which is composed of radiation reflected to i from all the surfaces, and that reflected by windows. All radiation (including Incident beam) is assumed to be reflected diffusely.

Each surface i will also reflect short-wave radiation, with a radiosity $J_i$ [Btu/hr-sf].

**Figure 12: Radiation Terminology**



The derivation below determines the equations to obtain $J_i$ and $G_i$ for known $Q_i$ values, for all the surfaces of the sphere, i = 1 to n.

**First a relationship between Gi and Ji is developed:**

Since $G_i$ is composed only of reflected radiation,

$$a_i G_i = \sum_k J_k a_k F_{ik}$$

*Equation 96*

where the sum is over all surfaces n of the sphere of area $a_s$.

Using the view-factor reciprocity principle,

$$a_k F_{ki} = a_i F_{ik}$$

$G_i$ becomes

$$G_i = \sum_k J_k F_{ik}$$

For spherical geometry, the view factor is $F_{ik} = a_k / a_S$, where $a_S = \sum a_k$, so $G_i$ can be written

$$G_i = \frac{1}{a_S} \sum_k J_k a_k$$

*Equation 97*

The right hand side is the area-weighted average radiosity, showing that $G_i$ is independent of i,

$$G_i = \bar{J}$$

*Equation 98*

**Next a separate relationship between $J_i$ and $G_i$ is obtained, $G_i$ eliminated and $J_i$ solved for explicitly:**

The radiosity of surface *i* is composed of the reflected part of both the irradiation and the targeted solar

$$\mathrm{J_i} = \rho_\mathrm{i}(\mathrm{G_i} + \mathrm{Q_i})$$

*Equation 99*

Substituting Equation 97 for $G_i$ gives

$$\frac{J_i}{\rho_i} = \frac{1}{a_S} \sum_k J_k a_k + Q_i$$

*Equation 100*

Since by Equation 98 $G_i$ is independent of *i*, then Equation 99 shows that the radiosity of any surface *i* is related to the radiosity of any surface k by the relationship

$$\frac{J_i}{\rho_i} - Q_i = \frac{J_k}{\rho_k} - Q_k$$

Substituting this into Equation 100 gives

$$\frac{J_i}{\rho_i} = \frac{1}{a_S}\sum_k \left( a_k \rho_k \left( \frac{J_i}{\rho_i} + Q_k - Q_i \right) \right) + Q_i$$

This can be solved explicitly for $J_i$:

$$J_i = \frac{\frac{1}{a_S}\rho_i}{1-\bar{\rho}} \left( \sum_k a_k \rho_k Q_k \right) + Q_i \rho_i$$

*Equation 101*

From Equation 101, the area weighted average *J* is

$$\bar{J} = \frac{\frac{1}{a_S}\bar{\rho}}{1-\bar{\rho}} \left( \sum_k a_k \rho_k Q_k \right) + \frac{1}{a_S}\sum_i a_i Q_i \rho_i$$

*Equation 102*

where $\bar{\rho}$ is the area weighed average reflectivity.

**Now that $J_i$ and $G_i$ are known an energy balance will give the net heat transfer:**

The net energy rate (Btu/hr) absorbed and/or transmitted by surface i, is:

$$Qnet_i = (G_i + Q_i - J_i)a_i = (\bar{J} - J_i + Q_i)a_i$$

*Equation 103*

Substituting Equation 101 and Equation 102 into this gives

$$Qnet_i = \frac{a_i}{a_S}\left(\frac{1-\rho_i}{1-\bar{\rho}}\right)\sum_k a_k \rho_k Q_k + a_i Q_i (1-\rho_i)$$

*Equation 104*

The first term in Equation 104 is from the absorption and/or transmission of radiation that reached and is absorbed by surface i after having been reflected, ad infinitum, by the interior surfaces. The second term is from the absorption of the "initially" incident insolation $Q_i$ on surface *i*.

If none of the insolation is specifically targeted, and instead S is assumed to be distributed isotropically then $Q_i$ is the same for each surface:

$$Q_i = \frac{S}{a_S}$$

*Equation 105*

Substituting this into Equation 104 gives $Qnet_i$ for isotropically distributed insolation:

$$Qnet_i = \frac{a_i}{a_S}\left(\frac{1-\rho_i}{1-\bar{\rho}}\right)S$$

*Equation 106*

### 2.6.1.5 Radiation removed at each surface of a zone by multiple window sources of targeted insolation

The targeting can be different for each window. Adding an additional subscript "j" to Equation 104 allows it to represent the energy removal for each surface separately for each window j. That is, Equation 104 becomes Equation 107, the rate of energy removal at each surface due to insolation $S_j$, that is distributed according to the assigned targeted values $Q_{ji}$.

$$Qnet_{ji} = \frac{a_i}{a_S}\left(\frac{1-\rho_i}{1-\bar{\rho}}\right)\sum_k a_k\rho_k Q_{jk} + a_i Q_{ji}(1-\rho_i)$$

*Equation 107*

The targeting fractions $H_{jk}$, to be user input, are defined as the fraction of insolation from window $j$ that is incident on surface k:

$$H_{jk} = \frac{a_k Q_{jk}}{S_j}$$

*Equation 108*

With this definition, Equation 107 can be written as

$$Qnet_{ji} = a_i S_j(1-\rho_i)\left(\frac{H_{ji}}{a_i} + \frac{1}{a_S(1-\bar{\rho})}\sum_k \rho_k H_{jk}\right)$$

*Equation 109*

The effective absorptivity of the targeted surfaces is defined as

$$\alpha effT_{ji} = Qnet_{ji}/S_j$$

*Equation 110*

Replacing the spherical surfaces $a_i$ in Equation 109 by $a_i = A_i F_i$, and substituting Equation 109 into Equation 110 gives the targeted gain equation used in the CZM code:

$$\alpha effT_{ji} = A_i F_i(1-\rho_i)\left(\frac{H_{ji}}{A_i F_i} + \frac{1}{(1-\bar{\rho})\sum_i A_i F_i}\sum_k \rho_k H_{jk}\right)$$

*Equation 111*

If $\sum_k H_{jk} < 1$ then it is assumed that the remaining insolation $S_j(1 - \sum_k H_{jk})$is distributed isotropically. From Equation 105 it is

$$isotropic\ Qnet_{ji} = \frac{a_i}{a_S}\left(\frac{1-\rho_i}{1-\bar{\rho}}\right)S_j(1-\sum_k H_{jk})$$

*Equation 112*

The definition of the effective absorptivity for isotropic insolation is:

$$\alpha eff I_{ji} = Qnet_{ji}/S_j$$

*Equation 113*

Changing Equation 112 to utilize zone areas, $a_i = A_i F_i$, and substituting Equation 112 into Equation 113 gives the amount of the diffuse part of the insolation from each window j that is absorbed in each surface i. This is used in the CZM code.

$$\alpha eff I_{ji} = \frac{A_i F_i}{\sum_k A_k F_k}\left(\frac{1-\rho_i}{1-\bar{\rho}}\right)(1-\sum_k H_{jk})$$

*Equation 114*

Note that no distinction has been made between surfaces that are opaque like walls, and partially transparent window surfaces. They are treated equally. The difference is that the energy removed by an opaque wall is absorbed into the wall, whereas that removed by the window surfaces is partly transmitted back out the window, and partly absorbed at the window inside surface. The CZM development code lets the user specify a fraction of the radiation that is absorbed in the room-side surface of the window, which slightly heats the window and thus the zone.

Adding Equation 111 and Equation 114 gives the total effective absorptivity of surface i from the insolation admitted through window j:

$$\alpha eff_{ji} = A_i F_i (1-\rho_i)\left(\frac{1}{(1-\bar{\rho})\sum_k A_k F_k}\left[1-\sum_k (1-\rho_k)H_{jk}\right] + \frac{H_{ji}}{A_i F_i}\right)$$

*Equation 115*

The net radiation absorbed in surface i from window j is thus

$$Qnet_{ji} = A_i F_i (1-\rho_i) S_j\left(\frac{1}{(1-\bar{\rho})\sum_k A_k F_k}\left[1-\sum_k (1-\rho_k)H_{jk}\right] + \frac{H_{ji}}{A_i F_i}\right)$$

*Equation 116*

Summing this over all windows gives the total SW radiation absorbed and/or transmitted by surface i as:

$$Qnet_i = A_i F_i (1-\rho_i) \sum_j \left[ S_j \left( \frac{1}{(1-\bar{\rho}) \sum_k A_k F_k} \left[ 1 - \sum_k (1-\rho_k) H_{jk} \right] + \frac{H_{ji}}{A_i F_i} \right) \right]$$

*Equation 117*

## 2.7 Window Model

The ASHWAT algorithm is used to model complex windows with diatherminous layers and curtains, etc. (Wright and Kotey 2006, Wright, J.L. 2008). Given the environmental conditions on each side of the window, ASHWAT determines the long wave, short wave and convection heat transfers to the conditioned space.

For the following input and output discussion, ASHWAT is treated as a black box.

### 2.7.1 Inputs

Each time step, for each window, ASHWAT is given the environmental inputs:

$I$ = insolation incident on window system.

$I_{refl}$ =insolation reflected diffusely from the other room surfaces.

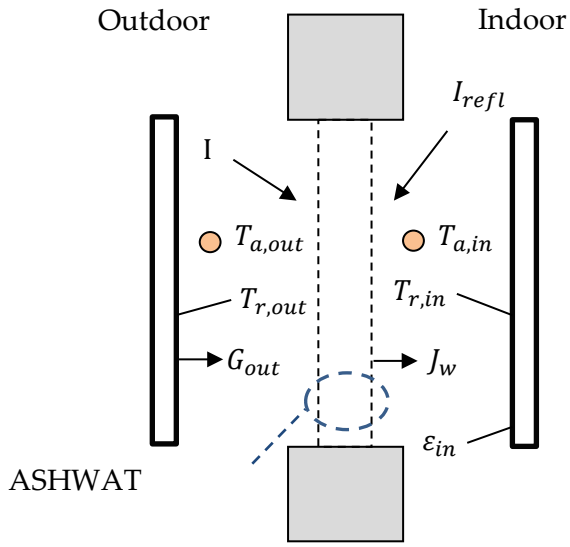
$T_{a,out}$ = outside dry bulb air temperature.

$T_{a,in}$ = inside dry bulb air temperature.

$T_{r,in}$ = the temperature of the indoor plate.

$T_{r,out}$ = the temperature of the outdoor plate.

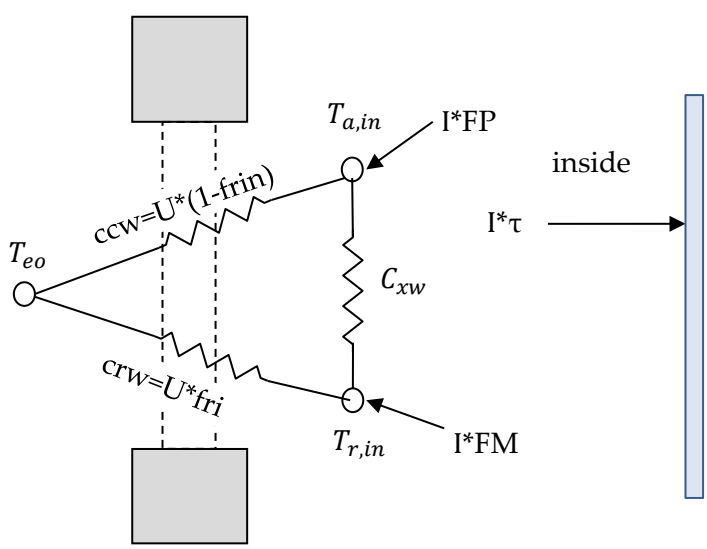
**Figure 13: ASHWAT Inputs and Nomenclature**




## 2.7.2 Outputs

ASHWAT's output gives heat transfer rates and circuit elements of Figure 14. The circuit of Figure 14 is part of the conditioned zone radiant network of Figure 2 and Figure 3 (with some different nomenclature).

**Figure 14: Window System Representation in CSE**




In Figure 14,

$FP$ = fraction of the heat from Insolation absorbed in the various window layers that ends up being transferred to the inside radiant node.

$FM$ = fraction of the heat from Insolation absorbed in the various window layers that ends up being convected to the inside air node.

$frin$ = fraction of total non-solar heat transfer that goes to the inside radiant node; dimensionless.

frout = fraction of non-solar heat transfer to the outside that goes to the outside radiant node.

$U$ = conductance between the inside and outside effective temperatures $T_{ei}$ and $T_{eo}$; Btu/(hr-sf-F), where $T_{ei} = T_{a,in}(1 - frin) + T_{r,in}frin$.

$T_{eo} = T_{a,out} * (1 - frout) + T_{r,out} * frout$ = the effective outdoor temperature; F.

$C_{xw}$ = the cross coupling term; Btu/(hr-sf-F).

$\tau$ = the short wave transmissivity of the window system.

Note that the solar heat gain coefficient is: $SHGC = \tau + FP + FM$.

Net energy into zone via window, per unit COG area = + $I\ (\tau + FP + FM)$ - $I_{refl}$ + $U(T_{eo} - T_{ei})$

## 2.7.3 Matching ASHWAT to CSE Radiant Network

### 2.7.3.1 Outside boundary conditions

ASHWAT models the irradiation on the outside of the window system as if it were emitted by a black plate parallel to the window at temperature $T_{r,out}$, as shown in Figure 13. The irradiation on the window system from the outside plate is thus $G_{out} = \sigma T_{r,out}^4$, so

$$T_{r,out} = \left(\frac{G_{out}}{\sigma}\right)^{0.25} = \left[F_{sky}\beta\, T_{sky}^4 + \left[F_{gnd} + F_{sky}(1-\beta)\right]T_{air}^4\right]^{0.25}$$

*Equation 118*

where $G_{out}$ has been replaced by Equation 67.

### 2.7.3.2 Inside boundary conditions

From Figure 13, the equivalent network between the radiosity of the window system, $J_w$, and the inside plate is shown in Figure 15. The circuit parameters are in the conductance form. The "1" is the view factor between the plate and the window.

**Figure 15: Equivalent Network between the Radiosity of the Window System, $J_w$, and the Inside Plate**

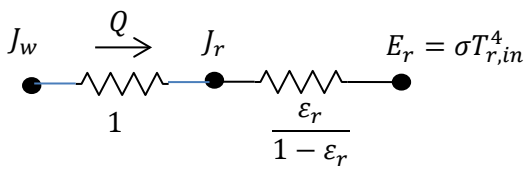
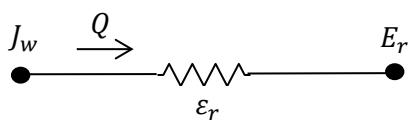



Figure 15 reduces to:

**Figure 16: Reduced Figure 15**



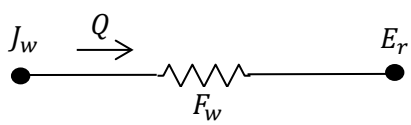

Thus the heat transfer rate per unit area, with $Q$ positive from window to room, is:

$$Q = \varepsilon_r (J_w - E_r)$$

*Equation 119*

From Figure 9 the network between the radiosity of a surface and the mean radiant temperature node is shown in Figure 17. This corresponds to Figure 16 for the ASHWAT algorithm:

**Figure 17: Network between the Radiosity of a Surface and the Mean Radiant Temperature Node**




with the corresponding heat transfer rate:

$$Q = F_w (J_w - E_r)$$

*Equation 120*

Comparing Equation 119 and Equation 120 shows that to obtain the heat flow consistent with the Carroll network ASHWAT must model the window by setting inside plate's emissivity to the value of $F_w$.

$$\varepsilon_r = F_w$$

*Equation 121*

$F_w$ is the Carroll MRT view factor defined in Section 2.6.1. $F_w$ is slightly larger than 1, and serves to increase the heat transfer between $J_w$ and $E_r$ to compensate for the fact that $|J_w - E_r|$ is smaller than it would if $T_{r,in}$ had not included the window temperature in its average. This MRT view factor effect cannot be simulated by a parallel plate model without the trick of artificially raising the emissivity of the inside plate to the value $F_w$.

# 2.8 Slab Model

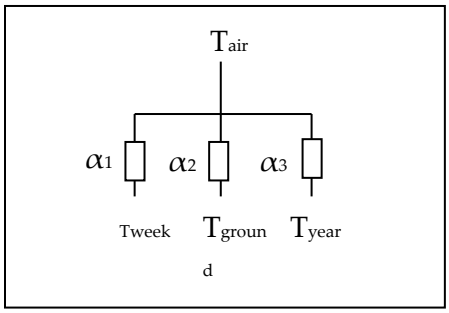
## 2.8.1 Bajanac Simplified Model

The CZM slab model is partly based on the Simplified Method for the calculation of heat flow through foundations, presented by Bazjanac et al. (2000). They divide a slab into two regions.

### 2.8.1.1 Perimeter region

The perimeter area of the slab is defined as a 2 ft wide strip along external walls. Through this perimeter path, the interior air is assumed to be coupled via conductances $\alpha_1$, $\alpha_2$, and $\alpha_3$ to three environmental temperatures: $T_{week}$, , $T_{ground}$, *and* $T_{year}$:

**Figure 18: Perimeter Coupling**




Thus the instantaneous heat flow from the room Temp node to perimeter slab, in Btu/hr-sf-F, is given by:

$$Qperim = \left[\alpha_1(T_{air} - T_{week}) + \alpha_2\left(T_{air} - T_{ground}\right) + \alpha_3\left(T_{air} - T_{year}\right)\right]$$

*Equation 122*

where,

$T_{air}$ = the current interior-space effective temperature (involving both Ta and Tr).

$T_{week}$ = the average outside air temperature of the preceding two-weeks.

$T_{ground}$ = the current average temperature of the earth from the surface to a 10 ft depth.

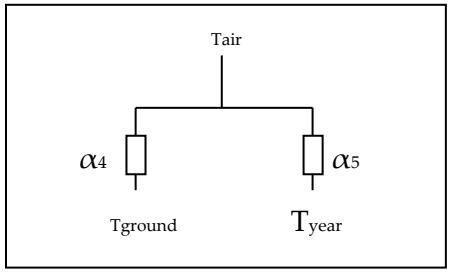
$T_{year}$ = the average yearly dry bulb temperature.

$\alpha's$ = conductances from Table 3 of Bazjanac et al; Btu/sf-hr-F.

### 2.8.1.2 Core region

The core region couples $T$ to $T_{ground}$ and $T_{year}$, via conductances $\alpha_4$ and $\alpha_5$.

**Figure 19: Core Coupling**




$$Qcore = [\alpha_4(T_{air} - T_{ground}) + \alpha_5(T_{air} - T_{year})]$$

*Equation 123*

Bazjanac et al. determined the conductances $\alpha_1$, through $\alpha_5$ by multi-linear regression analysis of the numerical results from a two-dimensional finite-difference slab-earth model. The conductances were determined for 52 slab foundation conditions and given in their Table 3.

### 2.8.1.3 Properties

The Bozjanac Table 3 conductances were obtained assuming the following properties:

1. Properties of earth:
   - conductivity = 1 Btuh/ft-F. (The k chosen was justified by assuming that lawns and other vegetation around California houses was watered during the dry season).
   - density =115 lbm/ft$^3$
   - specific heat = 0.2 Btu/lbm-F
   - thermal diffusivity = 0.0435 ft$^2$/hr.
2. Slab: "heavy construction grade concrete"
   - thickness = 4-inches
   - conductivity = 0.8
   - density = 144
   - specific heat = 0.139
3. Rrug = 2.08 hr-ft$^2$-F/Btu (ASHRAE 2005HF, p.25.5 'carpet fibrous pad').
4. Rfilm = 0.77 Btu/hr-ft$^2$F, the inside surface-to-room-temperature combined convective and radiative conductance.

### 2.8.1.4 Ground temperature

The above model uses the ground temperature determined by Kusuda and Achenbach (1965). Using the classical semi-infinite medium conduction equations for periodic surface temperature variation (Carslaw and Jaeger), they found the average ground temperature from the surface to a depth of 10 ft to be given by:

$$\mathrm{T_{ground}} = \mathrm{T_{yrAve}} - \mathrm{GM}\left(\frac{\mathrm{TyrMax-TyrMin}}{2}\right) cos\left(\left(\frac{2\pi}{8760}\right)\theta - PO - \phi\right)$$

*Equation 124*

where,

$T_{yrAve}$ = average outdoor temperature over year; F.

$T_{yrMax}$ = highest average monthly outdoor temperature for the year; F.

$T_{yrMin}$ = lowest average monthly outdoor temperature for the year; F.

$GM = \sqrt{\frac{e^{-2\beta} - 2e^{-\beta}\cos\beta + 1}{2\beta^2}}$ = dimensionless amplitude for integrated depth average.

$\beta = L\sqrt{\frac{\pi}{D*PY}}$ = dimensionless depth.

L = 10 ft, the depth over which average is taken.

D = thermal diffusivity of soil, $ft^2$/hr.

PY = 8760 hr = period of 1 year.

$\theta = 24\left(\frac{365M}{12} - 15\right) \approx$ elapsed time from Jan-1 to middle of month M; hours.

M = month, 1→12.

$\phi = atan\left(\frac{1-e^{-\beta}(cos\beta+sin\beta)}{1-e^{-\beta}(cos\beta-sin\beta)}\right)$ = phase angle for depth averaged $T_{ground}$; radians.

PO = 0.6 *radians* = phase lag of ground surface temperature (assumed equal to air temperature) relative to January 1. From measured data, see Fig. 7 in Kusuda and Achenbach.

## 2.8.2 Addition of a Layered Slab and Earth

The Bazjanac model assumes a constant indoor temperature, so cannot be applied directly to a whole building thermal-balance simulation model that allow changing indoor temperatures. To apply this model to CZM, with changing indoor temperatures, requires incorporating the dynamic effects of the slab and earth due to changing inside conditions.

This is done by putting a one-dimensional layered construction, representing the slab and some amount of earth mass, into the steady-state Bazjanac model circuit--replacing part of its resistance by a thermal impedance (which is equal to the resistance for steady state conditions). In this way the correct internal temperature swing dynamics can be approximated.

First, the circuit of Figure 18 is alternately expressed as shown in Figure 20(a), with Equation 122 taking the form:

$$Q = A * U_g(T_{air} - T_{geff})$$

*Equation 125*

where $T_{geff}$ is the $\alpha$-weighted average ground temperature:

$$T_{geff} = \frac{\alpha_1 T_{week} + \alpha_2 T_{ground} + \alpha_3 T_{yrAve}}{\alpha_1 + \alpha_2 + \alpha_3}$$

*Equation 126*

and

$$R_g = \frac{1}{\alpha_1 + \alpha_2 + \alpha_3}$$

*Equation 127*

Similarly for the core region,

$$T_{geff} = \frac{\alpha_4 T_{ground} + \alpha_5 T_{yrAve}}{\alpha_4 + \alpha_4}$$

*Equation 128*

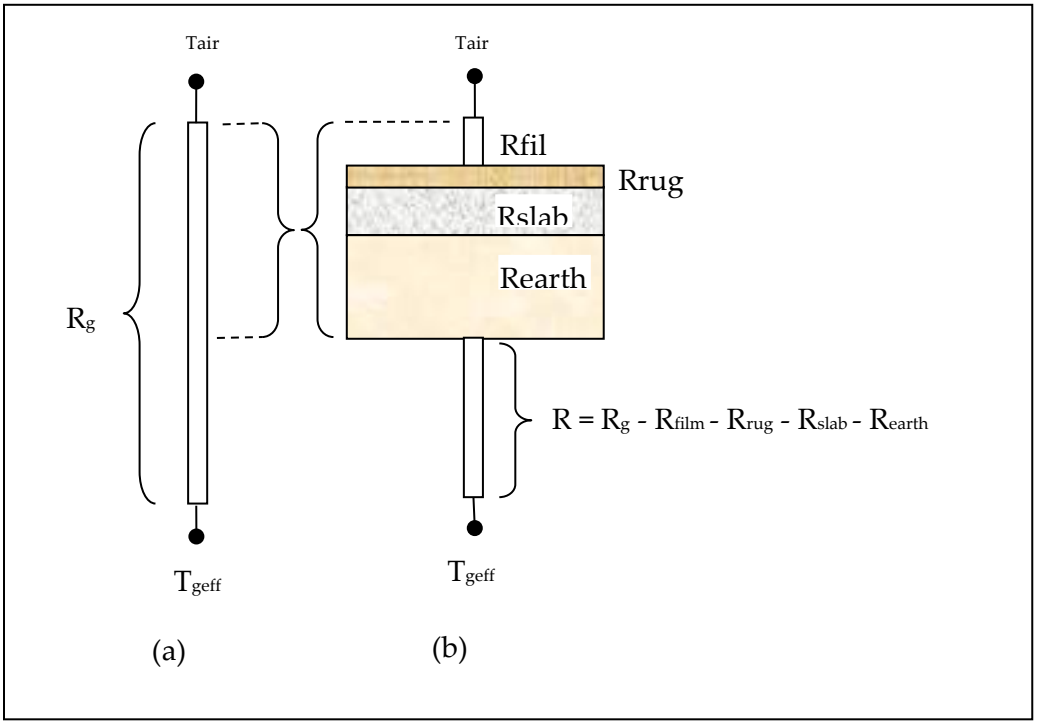
$$R_g = \frac{1}{\alpha_4 + \alpha_5}$$

*Equation 129*

Now a one-dimensional layered construction is added into the circuit as shown in Figure 20(b), consisting of a surface film layer, a carpet (if any), the concrete slab, and earth layer. The bottom of the earth layer is then connected to $T_{geff}$ through the what's left of $R_g$.

A one-dimensional representation of the mass is appropriate for the core region. It is a bit of a stretch for the perimeter slab modeling, because the real perimeter heat flow is decidedly 2-dimensional, with the heat flow vectors evermore diverging along the path of heat flow.

**Figure 20: Addition of Film, Rug, Slab, and Earth**




The earth thicknesses required to adequately model the dynamic interaction between the room driving forces (sun and temperature) and the slab/earth model was determined by considering the frequency response of the slab earth model of Figure 20(b). In the frequency domain, the periodic heat flow from the Tair node is given by Equation 130.

$$\tilde{Q}_{air} = \tilde{T}_{air}X - \tilde{T}_{geff}Y \ .$$

*Equation 130*

where,

X = the driving point admittance at the air (or combined air/radiant effective temp) node, in the units of Btu/hr-sf-F. It is the contribution to $\tilde{Q}_{air}$ per degree amplitude of $\tilde{T}_{air}$. X and Y are complex numbers determined from the layer properties (conductivity, heat capacity, density) of the circuit layers in Figure 20(b). See Carslaw and Jaeger; Subbarao and Anderson.

Y = the transfer admittance at the air node. It is the contribution to $\tilde{Q}_{air}$ per degreee amplitude of $\tilde{T}_{geff}$. [The same value of transfer admittance applies to the $\tilde{T}_{geff}$ node, even if the circuit is not symmetrical, being the contribution to the $\tilde{T}_{geff}$ node per degree amplitude of $\tilde{T}_{air}$]

$\tilde{Q}_{air}$ = the amplitude (Btu/hr-ft²-F) and phase of the heat transfer rate leaving $\tilde{T}_{air}$, and is composed of the contribution from all of the frequencies that may be extant in the driving temperatures $\tilde{T}_{air}$ and $\tilde{T}_{geff}$.

Note that the layers shown, when modeled as a mass construction, may need to be subdivided into thinner layers, particularly the earth, in order to satisfy the discretization procedure discussed in Section 2.4; but this subdivision is irrelevant to the slab model discussion in this section.

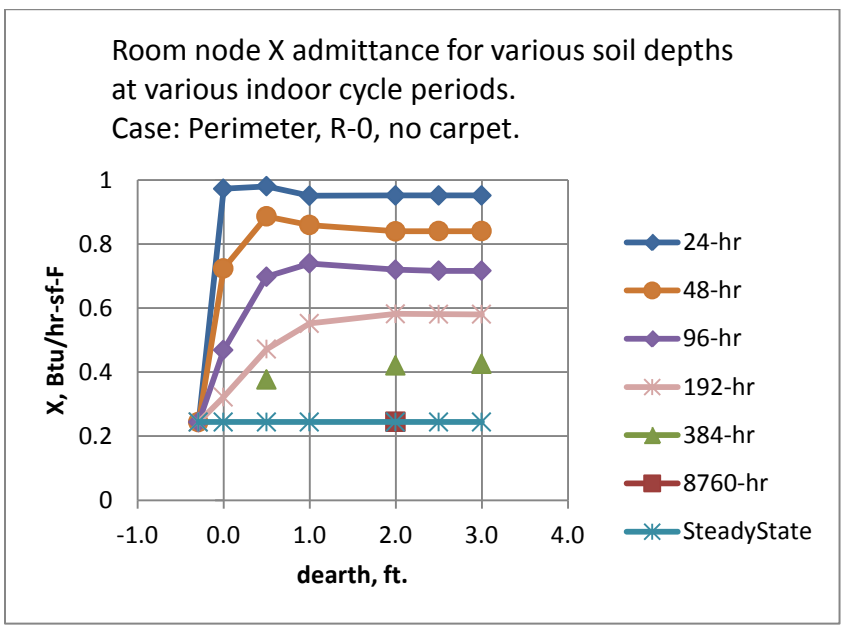
The maximum possible thickness of the earth layer is limited by the need for R to be positive. The limiting maximum possible thickness value, dmax, occurs in the perimeter case, when the foundation is uninsulated (i.e., the foundation insulation value R-0 in Bazjanac et al), and the slab is uncarpeted. In this case, dmax = 2.9 ft. The corresponding numbers for an uncarpeted core slab case is 11.8 ft

A depth of 2 ft is implemented in the code, for both the perimeter and core slab earth layers.

The 2 ft value was chosen primarily because, for the frequencies of concern, the magnitude of the X admittance from the Tair node was almost independent of earth layer depth for earth layer depths greater than 2 ft. See Figure 21. The phase shift is similarly essentially independent of depth after 2 ft. This was also the case for the core region.

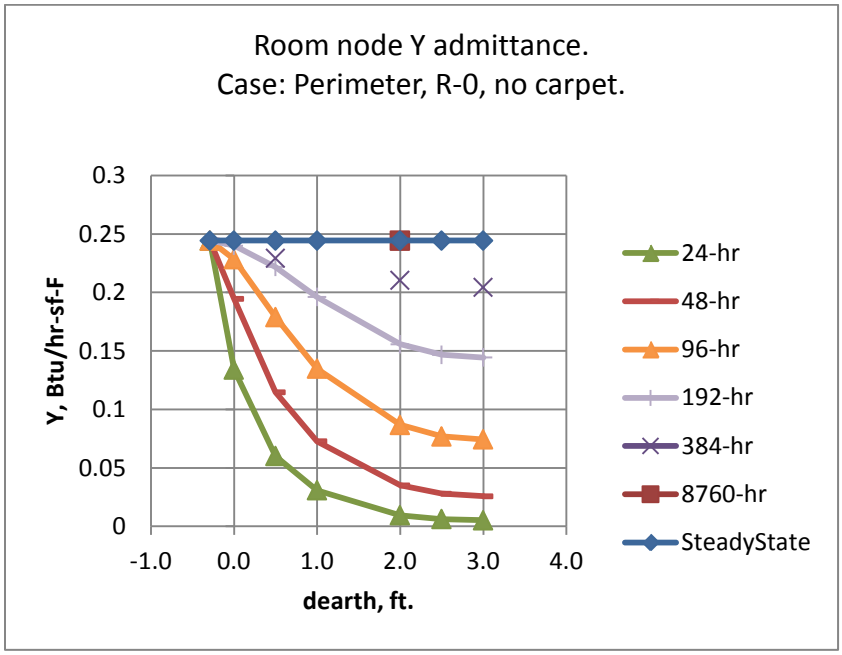
This is the case for all indoor driving frequencies periods of up to at least 384-hr = 16-days. Thus 2 ft of earth is able to portray the dynamics resulting from a cycle of 8-cloudy days followed by 8 sunny days.

**Figure 21: Room Node X Admittance**




The transfer admittance Y shown in Figure 22 also contributes to $\tilde{Q}_{air}$ according to the frequencies extant in the driving temperature $\tilde{T}_{geff}$.

**Figure 22: Room Node Y Admittance**




As seen in Equation 128, for the slab core region, $\tilde{T}_{geff}$ has the same frequency content as $T_{ground}$ and $T_{year}$. $T_{year}$ is a constant, i.e., zero frequency, steady-state.

As seen in Equation 124, $T_{ground}$ contains only the annual 8760-hr period. Figure 22 shows that the 8760-hr waves are transmitted unaffected by the mass layer. That is, $Y$ becomes essentially equal to the steady state transfer admittance which is the U factor of the assembly, the reciprocal of the $R_g$ value. Thus, for the core region, the magnitude of the slab loss rates produced by Equation 125 are preserved and unaffected by the added earth layers.

However, although the mass layers don't affect the magnitude of the Bazjanac model slab losses, they do introduce a time lag that is in addition to that already implicit in the $\tilde{T}_{geff}$ values. For a 2 ft earth layer the lag is ~40-hours. A 22-day lag is already included by $\phi$ of Equation 124. To eliminate double-counting, the 40-hrs could be subtracted from phi, but this has not been done since 40-hr is inconsequential compared to 22 days.

For the perimeter region, $\tilde{T}_{geff}$ has the the additional frequency content of the $T_{week}$, the two-week running average outdoor temperature. $T_{week}$ is dominated by the annual period, but has small amplitude 6-month period component, and a bit of signal at higher frequencies. Like the annual cycle, the 6-month period component is transmitted through the layered construction without damping, but again with a small but inconsequential phase lag.

Thus it was concluded that 2 ft of earth thicknesses below a 4-inch concrete slab adequately models changes in room side conditions, and at the same time adequately preserves the same average "deep earth" slab losses and phase lags of the Bazjanac model.

The validity of the response of the core slab construction is expected to be better than for the perimeter slab construction since the perimeter layers added do not properly account for the perimeter two-dimensional effects.

#### 2.8.2.1 Warm-up time

The longest pre-run warm-up time is expected to be for a carpeted core slab with the 2 ft earth layer. Using the classical unsteady heat flow charts for convectively heated or cooled slabs (Mills) , the time to warm the slab construction 90% (of its final energy change) was found to be about 20-days. Most of the heat-up heat transfer is via the low resistance rug and air film, with less through the higher ground resistance (R in Figure 20(b)), so the 20- day estimate is fairly valid for the complete range of foundation insulation options given in Bazjanac's Table 3.

#### 2.8.2.2 Input properties

Strictly speaking, the same properties assumed in the Bazjanac model in Section 2.8.1.3 should also be used in describing the rug, the concrete slab, and the earth in the layered constructions inputs.

This is particularly true for the carpet, if a carpet is specified, because the regression coefficients (the conductances a1, a2...) obtained for the carpeted slabs were sensitive to the Rrug value used. While inputting a different value than Rrug = 2.08 may give the desired carpeted room admittance response, the heat conducted from the deep ground will still give the heat flow based on Rrug = 2.08.

Small differences between the inputted and above properties is less important for the other layers, and is violated in the code with regard to Rfilm; its value is calculated each time-step and is used instead of 0.77, even though 0.77 is still the value subtracted from Rg in the code. This allows the correct modeling of the admittance of the slab floor, at the expense of a slight error in the overall resistance of the slab earth circuit.

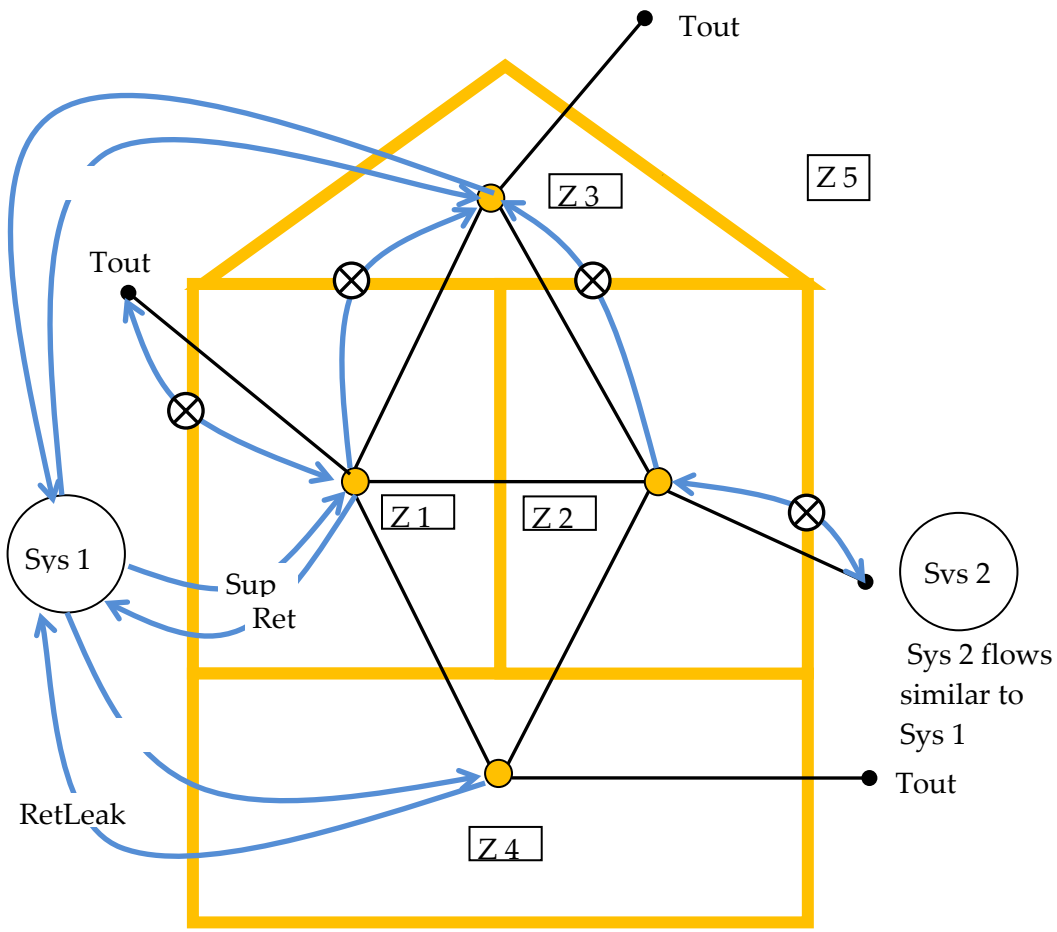
# 2.9 Ventilation and Infiltration Air Network

## 2.9.1 Overview

This section describes the flow network algorithm used to model infiltration and ventilation air flows between conditioned zones, unconditioned zones, and the outdoors based on pressure and density differences and leakage areas between the zones.

Figure 23 shows the flow network interconnecting two conditioned zones (Z1 and Z2), the unconditioned attic and crawl space zones (Z3 and Z4), and the outside zone (Z5).

**Figure 23. Schematic of Flow Network**




The black lines represent one or more pressure difference driven and/or buoyancy driven flows between zones.

The blue lines in Figure 23 represent scheduled fan flows not directly dependent on zone-to-zone pressure differences. These include individual house fans (circled x's) and fan driven duct system supply, return, and leakages flows. The fans will affect the zone pressures, but the

pressures won't affect the fan flow. The duct flows, determined by the load and air handler capacity, are assumed to not constitute leakage paths when the air handler is not operating.

Small leakage or ventilation openings will be modeled as orifices using a power law equations of Section 2.9.3 with an exponent of 0.5. Infiltration leaks are modeled with the power law equation exponent of 0.65.

Large vertical holes or infiltration surfaces, large enough that the vertical pressure difference distribution allows two-way flow, are modeled as two vertically separated small holes using the Wolozyn method (see Section 2.9.5– Large Vertical Openings).

The following kind of elements are modeled using the power law equations:

- Wall infiltration for vertical envelope walls, vertical interzone walls, and roof decks
- Ceiling, floor, and wall base infiltration
- Interzone doors, door undercuts, jump ducts, relief vents
- Openable window flow
- Attic soffit vents, gable vents, roof deck vents, ridge vents
- Crawl space vents
- Trickle Vents
- Fire Place leakage.
- Infiltration to garage.

Additional equations are used to model large horizontal openings, like stairwells; Section 2.9.4. This type of opening would typically be between zones Z1 and Z2 in Figure 23 when the zones are stacked vertically. The algorithm used is based on that implemented in Energy Plus (2009). In addition to using the power law equation above, the algorithm calculates buoyancy induced flows that can occur when the density of the air above the opening is larger than the density of the air below the opening, causing Rayleigh-Taylor instability.

To determine the flow rates at each time step, the flow through each flow element in the building is determined for an assumed set of zone reference pressures. If the flow into each zones does not match the flow out of the zone, the pressures are adjusted by the Newton-Raphson iterative method until the flows balance in all the zones within specified tolerances.

#### 2.9.1.1 Wind direction independent air-network solution

For energy standards application, the air network for the four zone building model is designed to give results that are wind direction independent. In computing ventilation or infiltration air flows from holes in vertical walls exposed to outdoors, the program automatically calculates the sum of the flows through 4 holes each ¼ the area, one with each cardinal compass orientation, or an offset thereof. Thus, there will be wind induced flows through the envelope leakages that approximate the average flow expected over long periods, and they will be independent of wind direction. This approach is applied to all zone ventilation or infiltration flow elements connected to the outdoor conditions.

## 2.9.2 Vertical Pressure Distribution

The pressure at a given elevation in a zone, including outdoors, is a combination of stack and wind effects added to the zones reference pressure. The difference in pressure in the zones on each side of a leakage element connecting the zones determines the flow rate through the element.

The pressure on the zone *i* side of a flow element is given by:

$$p_i = Pz_i - \rho_i g z_i$$

*Equation 131*

$z_i$ is the height of the element above some datum $z = 0$. The datum is arbitrary but is nominally taken as ground level. $Pz_i$ is zone $i's$ reference pressure. This is the pressure zone *i* would have at elevation $z = 0$, regardless of whether the zone actually extends to this level. For the interior zones the $Pz_i$ reference pressures are the unknowns that are solved for using the Newton-Raphson method. This method determines what values of zone pressures simultaneously result in a balanced flow in each zone. The value of $Pz_i$ for the outdoor side of a flow element is given by (*i* is 5 if there are 4 conditioned and unconditioned zones):

$$Pz_i = patm + CP * Pu$$

*Equation 132*

The weather tape atmospheric pressure, patm, is assumed to exist at the elevation z = 0 far from the building. Patm is taken as zero so that the unknown zones pressures will be found relative to the weather tape atmospheric pressure. Of course the actual weather tape atmospheric pressure is used in determining inside and outside zone air densities.

The wind velocity pressures, Pu, is:

$$Pu = \rho_{out}(S * U)^2/2$$

*Equation 133*

where,

*U* is the wind velocity at eave height.

S is the shelter coefficient equal to SC of Table 2: Local Shielding Parameters.

$\rho_{out}$ is the outside air density.

*CP* is the orientation sensitive pressure coefficient.

### 2.9.2.1 Pressure coefficients used

The wall pressure coefficients in Table 6 are those used by Walker et al. (2005). They are for the four vertical walls of an isolated rectangular house, with the wind perpendicular to the long wall (short wall = ½ long wall). As discussed in regard to hip roofs below, only data for the

normal wind direction is used. These coefficients are used for all ventilation and infiltration holes in the walls. Soffit vents also use these values since they are assumed to have the same pressure coefficient as the walls under them. This assumption is roughly corroborated by the data of Sharples (1997).

**Table 6: Pressure Coefficients for Wind Normal to One Wall**

| Wind Pressure Coefficient, *CP* | | |
|---|---|---|
| Upwind wall | Side walls | Downwind Wall |
| +0.6 | -0.65 | -0.3 |

Table 7 gives hip roof's pressure coefficients for a range of roof angles. These are used to determine the outside pressure on ridge vents and roof deck vents.

There is little data available for hip roof surface pressure coefficients, or for ridge pressure coefficients (needed to model ridge vents) for any roof type. The data in Table 7 is a simplified synthesis of the data given by Xu (1998) and Holmes (1993, 2003, etc.), informed by a review of ASHRAE, EU AIVC, and other data sets and research papers.

Xu used a wind tunnel to measure pressure coefficients for a hip roofed building which was otherwise identical to the gable roof building wind tunnel data obtained by Holmes. The building had an aspect ratio of 2:1, with $0^o$ wind direction normal to the long side (and normal to the gable ridge and hip roof top ridge). The building eave height was 0.4 the length of the short side. The building had a relatively large eave overhang of about 35% of the eave height. Xu and Holmes presented data for this building for roof pitch angles of 15, 20, and $30^o$ . Other Holmes data, for both larger and smaller roof angles was used to estimate the pressure coefficients beyond the 15 to 30 degree range. Neither Xu nor Holmes presented average surface pressures, so the average surface data and average ridge pressures given in the table are based on estimates from their surface pressure contour data.

The table is for wind normal to the long side of the building. Similar tables were obtained from Xu's data for the 45 and 90 degree wind angles. Table 7 would ideally be wind direction independent, implying some kind of average pressure coefficient; for example, for each surface take the pressure coefficient that is the average for the 0, 45, and 90 degree angles. However, infiltration flows depend on pressure differences, and the average of the pressure differences is not necessarily indicative of the difference of the average pressures. The soffit vents flows, driven by the pressure difference between the adjacent wall and the various roof vents complicate any averaging schemes.

Comparison of the pressure coefficients for the three wind directions, while showing plausible differences, arguably does not show a discernable pattern that would obviate just using the normal wind direction data. Given the variety of roofs and building shapes that will be represented by these coefficients, the variety of vent locations and areas, and the deficiencies of the data, using a consistent set of data for only one wind direction is deemed appropriate.

**Table 7: Hip Roof Wind Pressure Coefficients**

| Roof Pitch $\psi$ | Hip Roof Wind Pressure Coefficient, *CP* | | | |
|---|---|---|---|---|
| | Upwind roof | Side hip roofs | Downwind Roof | Ridge |
| $\psi < 10^o$ | -0.8 | -0.5 | -0.3 | -0.5 |
| $10 \leq \psi < 15$ | -0.5 | -0.5 | -0.5 | -0.8 |
| $15 \leq \psi < 25$ | -0.3 | -0.5 | -0.5 | -0.5 |
| $25 \leq \psi < 35$ | +0.1(pos) | -0.5 | -0.5 | -0.3 |
| $35 \leq \psi < 50$ | +0.3 (pos) | -0.5 | -0.5 | -0.2 |

#### 2.9.2.2 Density

Zone $i$'s air density $\rho_i$ is assumed to be only a function of zone temperature $T_i$. That is, assuming the air is an ideal gas, at standard atmospheric conditions, the pressure change required to change the density by the same amount as a change in temperature of 1°F is $\frac{\partial \rho}{\partial T}$ / $\frac{\partial \rho}{\partial p} = -\rho R_{air}$ , which is approximately - 200 Pascals/F. Since zone pressure changes are much smaller than 200 Pa, they are in the range of producing the same effect as only a fraction of a degree F change in zone temperature; thus the density is assumed to always be based on patm. (This has been changed in code so that $\rho_i$ depends on both $T_i$ and $Pz_i$).

Using the ideal gas approximation, with absolute temperature units,

$$\rho_i = \frac{P_{atm}}{R_{air} T_i}$$

*Equation 134*

The pressure difference across the flow element is given by

$$\Delta p_{ij} = p_i - p_j = Pz_i - Pz_j - gz_i(\rho_i - \rho_j)$$

*Equation 135*

### *2.9.3 Power Law Flow Equation*

#### 2.9.3.1 Orifice flow power law

For an orifice, with fixed density of air along the flow path (from inlet to vena contracta), Bernoulli's equation gives:

$$m = C_D A \sqrt{2\rho_{in} g_c} (\Delta p)^{\frac{1}{2}}$$

*Equation 136*

where

$C_D$ is the dimensionless orifice contraction coefficient.

$C_D = \frac{\pi}{\pi+2}$ = Kirchoff's irrotational flow value for a sharp edge orifice.

$C_D =$ 0.6 default for CSE windows

$C_D =$ 1 for rounded inlet orifice as used in ELA definition, and consistent with no vena contracta due to rounded inlet.

$A$ = Orifice throat area, $ft^2$.

$\rho_{in}$ = density of air entering the orifice; $lb_m/ft^3$.

$g_c = 32.2 \frac{lb_m ft}{lb_f\, sec^2}$

### 2.9.3.2 Infiltration flow power law

The following is based on Sherman (1998). English units are used herein. Measured blower door infiltration data is expressed empirically as a power law:

$$Q = \kappa \Delta P^n$$

*Equation 137*

or

$$m = \rho_{in} \kappa \Delta P^n$$

*Equation 138*

where

$Q$ = volume flow in $ft^3$/sec.

$m$ = mass flow in $lb_m/sec$.

$\rho_{in}$ = entering air density, $lb_m/ft^3$

$\Delta P$ = pressure difference in $lb_f/ft^2 = psf$ .

$n$= measured exponent, assumed to be n = 0.65 if measured value is unavailable.

$\kappa$ = measured proportionality constant.

Equation 137 and Equation 138 are dimensional equations. Thus $\kappa$ is not a dimensionless number but implicitly has the dimensions $ft^{3+2n}/(sec{*}lb_f^n)$. See Section 2.9.3.8–Converting Units of κ.

Sherman defines equivalent leakage area, ELA, as the area of a rounded-entrance orifice that gives the same flow as the infiltration of Equation 137 when the pressure difference $\Delta P$ is equal

to the reference pressure $P_r$ = 0.08354 psf (= 4 Pa) By Equation 136, a rounded-entrance nozzle with throat area $ELA$ and $\Delta P = P_r$ has a flow rate:

$$m = ELA\sqrt{2\rho_{in}g_c}\,(P_r)^{\frac{1}{2}}$$

*Equation 139*

Equation 137 and Equation 139 with $\Delta P = P_r$ gives the ELA as:

$$ELA = \kappa P_r^{\,n-\frac{1}{2}}\sqrt{\frac{\rho_{in}}{2g_c}}$$

*Equation 140*

Solving Equation 140 for κ, gives

$$\kappa = ELA\sqrt{\frac{2g_c}{\rho_{in}}}\,P_r^{\frac{1}{2}-n}$$

*Equation 141*

Substituting Equation 140 into Equation 137 gives the general equation, equivalent to Equation 137, that is the infiltration flow at any pressure difference $\Delta P$:

$$m = ELA\sqrt{2\rho_{in}g_c}\,P_r^{\frac{1}{2}-n}\,\Delta P^n$$

*Equation 142*

(Note that substituting Equation 140 into Equation 142 recovers the empirical Equation 137).

### 2.9.3.3 General power law flow equation

CSE uses Equation 136 to model flow through elements such as windows, doors, and vents. Equation 142 is used for infiltration flows for elements with a defined ELA.

Both equations are special cases of the generalized flow power law Equation 143. For flow from zone $i$ to zone $j$.

$$m_{i,j} = SP * A_e\sqrt{2\rho_{in}\,g_c}\,|\Delta p_{i,j}|^{n_g}$$

*Equation 143*

SP is the sign of the pressure difference $\Delta p_{i,j} = p_i - p_j$, utilized to determine the sign of the flow, defined as + from $i$ to $j$. The exponent is $n_g$, "g" for generalized.

Equation 143 reduces to the orifice Equation 136 if:

- $A_e = A * C_D$ with $C_D = 0.6$.
- $n_g = \frac{1}{2}$

Equation 143 reduces to the infiltration Equation 142 if:

- $A_e = \left(P_r^{\frac{1}{2}-n}\right) ELA$ , where n here is the measured exponent.
- $n_g = n$
- $P_r = 0.08354 \frac{lb_f}{ft^2}$

[Although $C_D$ is dimensionless in Equation 136, the generalization to Equation 143 requires $C_D$ to implicitly have the units of$(lb_m)^{\frac{1}{2}-n_g}(ft)^{2n_g-1}$].

### 2.9.3.4 Dealing with unbounded derivative at $\Delta P = 0$

The partial derivative of the mass flow of Equation 143 with respect to the pressure in zone $i$ is given by:

$$\frac{\partial m_{i,j}}{\partial p_i} = A_e n_g \sqrt{2\rho_{in}} |\Delta p_{i,j}|^{n_g - 1}$$

*Equation 144*

Since $n_g < 1$ this derivative →∞ as $\Delta P$→ zero, potentially causing problems with the Newton-Raphson convergence. In order to make the derivative finite for small pressure drops, whenever the pressure difference is below a fixed value, $\Delta P_L$, the power law is extended to the origin by a linear power law ($n_g = 1$),

$$m_{i,j} = SP * A_{elinear} \sqrt{2\rho_{in}\, g_c}\, |\Delta p_{i,j}|^1$$

*Equation 145*

as shown in Figure 24.

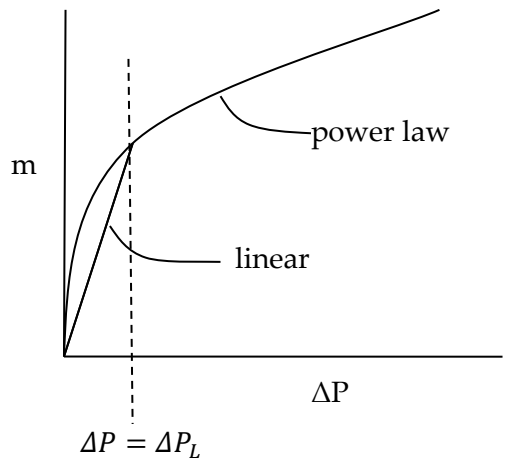
So that the flow rates match when $\Delta P = \Delta P_L$, $A_{elinear}$ is determined by equating Equation 145 to Equation 143 with $\Delta p = \Delta P_L$, giving:

$$A_{elinear} = A_e \Delta P_L{}^{ng-1}$$

*Equation 146*

Note that the derivative of $m$ will be discontinuous when $\Delta p = \Delta P_L$, which conceivably could also cause Newton-Raphson problems, but during extensive code testing, none have occurred.

**Figure 24: Mass Flow m versus Pressure Drop $\Delta P$**




### 2.9.3.5 Summary of inputs to the generalized flow equation

The generalized flow equation, Equation 143,

$$m_{i,j} = SP * A_e\sqrt{2\rho_{in}\, g_c}\,|\Delta p_{i,j}|^{ng}$$

is used with the following parameter values, depending on element type and pressure drop $\Delta P$.

### 2.9.3.6 For windows, doors, and vents

If $\Delta P > \Delta P_L$:

- $A_e = A * C_D$
- $A$ = area of flow element; $ft^2$.
- $C_D = 0.6$.
- $n_g = \frac{1}{2}$

If $\Delta P < \Delta P_L$:

- $Ae = C_D\, A * \Delta P_L{}^{0.5-1} = C_D\, A/\sqrt{\Delta P_L}$
- $A$ = area of flow element; $ft^2$.
- $C_D = 0.6$
- $n_g = 1$
- $\Delta P_L$ = determined by computational experiment.

### 2.9.3.7 For infiltration leakage elements

If $\Delta P > \Delta P_L$:

- $A_e = \left(P_r^{\frac{1}{2}-n}\right) ELA$ , $n$ here is the measured exponent, or 0.65 if not known. (Note that if $n = 0.65$, $A_e = 1.45 * ELA$, used in CEC ACM manual).
- $n_g = n$
- $P_r = 0.08354 \frac{lb_f}{ft^2}$
- $ELA$ is determined from either:
  - the measured parameters $\kappa$ and $n$ using Equation 140. See Section 2.9.3.8– Converting units of κ.
  - code regulations, in which case $n = 0.65$ is assumed.

If $\Delta P < \Delta P_L$:

- $A_e = \Delta P_L{}^{n-1} \left(P_r^{\frac{1}{2}-n}\right) ELA$ , where n here is the measured value, or 0.65.
- $n_g = 1$
- $P_r = 0.08354 \frac{lb_f}{ft^2}$

### 2.9.3.8 Converting units of κ

The $\kappa$ in Equation 137 is not dimensionless, so κ changes value depending on the units of $Q$ and $\Delta P$ in Equation 137. The analysis herein (Section 2.9) assumes $Q$ in $ft^3/sec$ and $\Delta P$ in $lb_f/ft^2$.

However, conventionally κ is obtained from measured data with $Q$ in $\left(\frac{ft^3}{\text{min}}\right)$ and $\Delta P$ in $Pascals$. With these units Equation 137 takes the form:

$$Q[cfm] = \kappa'\left(\Delta P(Pa)\right)^n$$

*Equation 147*

Using dimensional analysis the value of κ to be used in Equation 137 with Q in $(\text{ft}^3/\text{sec})$ and ΔP in $\left(\frac{\text{lb}_\text{f}}{\text{ft}^2}\right)$ is:

$$\kappa = \left(\frac{47.88^n}{60}\right)\kappa'$$

*Equation 148*

where the numbers are from the conversion factors $47.88 \frac{\text{Pa}}{\text{lb}_\text{f}/\text{ft}^3}$ and 60 sec/min. $\kappa = 0.206\kappa'$ for $n = 0.65$.

$\kappa'$ = the measured value from data with $Q$ in $\left(\frac{ft^3}{\text{min}}\right)$ and $\Delta P$ in $Pascals$.

### 2.9.3.9 ACM Manual relationship between CFM50 and ELA

Using Equation 142 (in volume flow form) the infiltration volume flow, CFS50, with 50 Pa pressurization is:

$$\text{CFS50} = \text{ELA}\sqrt{\frac{2g_c}{\rho}} P_r^{\frac{1}{2}-n} \Delta P^n = ELA\sqrt{\frac{64.4}{0.075}}\, 0.08354^{-0.15}\, 1.04428^{0.65} = 43.738 * ELA$$

*Equation 149*

where

$\Delta P$ = 50 Pa = 1.04428 psf

CFS50 = flow in units of $ft^3/sec$.

For cfm units, and ELA in square inches,

CFM50 = (60/144)CFS50 = 18.224*ELA

*Equation 150*

or alternately,

ELA [in$^2$] = 0.055*CFM50

*Equation 151*

This is the equation used to get ELA from blower door data at 50 Pa pressure difference.

#### 2.9.3.10 Heat Flow

When the flow $m_{i,j}$ is positive, the heat delivered to zone j by this flow is given by

$$Q_j = m_{i,j} C_p (T_i - T_j),$$

*Equation 152*

while the heat delivered to zone *i* by the flow $m_{i,j}$ is zero:

$$\mathrm{Q_i} = 0$$

*Equation 153*

When the flow $m_{i,j}$ is negative, the heat delivered to zone j is zero,

$$Q_j = 0$$

*Equation 154*

while the heat delivered to zone *i* by the flow $m_{i,j}$ is:

$$Q_i = m_{i,j} C_p (T_i - T_j)$$

*Equation 155*

### *2.9.4 Large Horizontal Openings*

An additional set of equations is needed to model large horizontal openings such as stairwells. The algorithm used is similar to that implemented in Energy Plus, which is based on that given by Cooper (1989). In addition to pressure driven flow using the power law equations of Section

2.9.3.3 this algorithm involves buoyancy induced flows that can occur when the density of the air above the opening is larger than the density of the air below the opening, causing Rayleigh-Taylor instability.

For a given rectangular opening this algorithm can produce three separate flows components between the zones:

a) a forced orifice flow in the direction dictated by the zone to zone pressure difference, Δp. This flow is independent of the following instability induced flows.
b) a buoyancy flow downward when the air density in the upper zone is greater than that is the lower zone, i.e., $T_{upper\text{-}zone} < T_{lower\text{-}zone}$. This flow is maximum when Δp is zero, and linearly decreases with increasing Δp until the buoyancy flow is zero, which occurs when the pressure difference is large enough that the forced flow "overpowers" the instability flow. The latter occurs if Δp is greater than the "flooding" pressure ΔpF.
c) an upward buoyancy flow equal to the downward buoyancy flow.

These three flows are modeled by two flow-elements. The first element handles the forced flow (a) and in addition whichever of the buoyancy flow component, (a) or (c), that is in the same direction as the forced flow. The second element handles the alternate buoyancy flow component.

### 2.9.4.1 Pressure driven flow

The pressure forced flow is modeled as orifice flow using Equation 143, except the area $A$ in the Section 2.9.3.5 is replaced by:

$$Aeff = L1 * L2 * sin(StairAngle) * (1 + cos(StairAngle))$$

*Equation 156*

*L1* and *L2* are the dimensions of the horizontal rectangular hole. To include the effect of stairs, a $StairAngle$ can be set, where $StairAngle = 90\ deg$ corresponds to vertical stairs. The angle can be set to 90 degrees to exclude the effect of the stairs.

Equation 144 is used for the partial derivative of the flow, with the area $A_e$ using $Aeff$ in place of $A$.

### 2.9.4.2 Buoyancy flow

When the zone on top has a higher density than the zone on the bottom, the maximum possible buoyancy flow, mbm, occurs when the pressure difference across the hole is zero:

$$mbm = 0.055\sqrt{g\bar{\rho}|\Delta\rho|Dhyd^5}$$

*Equation 157*

The 0.055 factor is dimensionless ; g = 32.2 ft/s^2.

The hydraulic diameter of the hole is defined as:

$$Dhyd = 2 * Aeff/(L1 + L2)$$

*Equation 158*

When the zone on top has a higher density than the zone on the bottom, and the pressure difference is lower than the flooding pressure, then the buoyancy flow is given by:

$$mb \ = \ mbm * \left(1 - \frac{|\Delta p|}{\Delta pF}\right)$$

*Equation 159*

The flooding pressure difference $\Delta pF$ is defined as:

$$\Delta pF = C_s^2 g|\Delta\rho|Dhyd^5/(2\mathrm{A}_{\mathrm{eff}}^2)$$

*Equation 160*

The shape factor $C_s$ is

$$C_s = 0.942\left(minimum\left(\frac{L1}{L2}, \frac{L2}{L1}\right)\right)$$

*Equation 161*

If the top zone density is lower than the bottom zones, or if $|\Delta p| > \Delta pF$ then the buoyancy flow mb is zero.

The partial derivatives of the buoyancy flows with respect to adjacent zone pressures are all zero since the buoyancy flows are equal and opposite. That is, although the buoyancy flow magnitudes are sensitive to zone pressures, they have no influence on the zone mass balance.

Although the buoyancy flows don't directly influence zone pressures, they do affect the heat transfer rates.

When buoyancy flows exists, the heat transfer due to the buoyancy flow to the upper zone, *i* say, is

$$\mathrm{Q_i} = \ \mathrm{mb} * \mathrm{C_p}(\mathrm{T_j} - \mathrm{T_i})$$

*Equation 162*

and to the lower zone is

$$\mathrm{Q_j} = \ \mathrm{mb} * \mathrm{C_p}(\mathrm{T_i} - \mathrm{T_j})$$

*Equation 163*

## 2.9.5 Large Vertical Openings

The flow through large vertical rectangular openings are handled using the method suggested by Woloszyn (1999).

Woloszyn uses a simplified version of the common integrate-over-pressure-distribution scheme as used by Walker for example. Rectangular holes are divided in two, with the flow through the top half driven by a constant Δp equal to the pressure difference ¾ the way up the opening (the midpoint of the top half of the opening area). Similarly, the flow through the bottom half uses the Δp at ¼ the way up the hole, and assumes it is constant over the bottom half. Although approximate compared to the integration methods, it is expected to be able to reasonably accurately, if not precisely, portray one and two way flows through such elements. This procedure has the virtue of eliminating the calculation of the neutral level, thereby greatly reducing the number of code logic branches and equations. It also eliminates a divide by zero problem when $\Delta\rho \rightarrow 0$ in the exact integration methods.

Besides being used for large vertical holes, like open windows and doorways, the method is also used for distributed infiltration. That is, a rectangular wall with an effective leakage area ELA is represented by two holes, each of area ELA/2, located at the ¼ and ¾ heights. These holes are then modeled using Equation 138, Equation 141, Equation 144, and Equation 145.

### 2.9.5.1 Triangular surfaces

The method is generalized further to treat the tilted triangular surfaces assumed for hip roofs. In this case the lower Woloszyn hole, of area ELA/2, is placed at the height that is above ¼ of the area of the triangle. This can be shown to be a height of:

$$H_{lower\ hole} = Z_{soffit} + \left(1 - \frac{\sqrt{3}}{2}\right)(Z_{ridge} - Z_{soffit})$$

*Equation 164*

Similarly, the top hole is placed at the height above ¾ of the area of the triangle:

$$H_{upper\ hole} = Z_{soffit} + (1/2)(Z_{ridge} - Z_{soffit})$$

*Equation 165*

## 2.9.6 Newton-Raphson Solution

Assume there are a total of *nuc* conditioned and unconditioned zones with unknown pressures. The outside conditions, of known pressure, are assigned a zone number *nout* = *nuc*+*1*.

The mass flow rate from zone *i* to zone *j* (including *j*=*nout*) is designated as $m_{i,j}$, and can be positive (flow out of zone i) or negative (flow into zone i).

$$m_{i,j} = \sum_{k=1}^{K_{i,j}} m_{i,j,k}$$

*Equation 166*

where $m_{i,j,k}$ is the flow rate through the $k$'th element of the $K_{i,j}$ elements in surface $i,j$. By symmetry,

$$m_{i,j,k} = -\, m_{j,i,k}$$

*Equation 167*

and

$$\mathrm{m_{i,j}} \quad = -\, \mathrm{m_{j,i}}$$

*Equation 168*

From Equation 167, $\mathrm{m_{i,j,}}_k$ values are functions of the zone pressure difference $(P_i - P_j)$.

$$\mathrm{m_{i,j,k}} = +\, \mathrm{A_e}\sqrt{2\rho_{\mathrm{in}}}\left(\mathrm{P_i} - \mathrm{P_j}\right)^{\mathrm{n}} \qquad \text{for positive } \Delta\mathrm{P}$$

$$\mathrm{m_{i,j,k}} = -\mathrm{A_e}\sqrt{2\rho_{\mathrm{in}}}\left(\mathrm{P_j} - \mathrm{P_i}\right)^{\mathrm{n}} \quad \text{for negative } \Delta\mathrm{P}$$

*Equation 169*

This shows that in general,

$$\frac{\partial m_{i,j,k}}{\partial P_i} = -\frac{\partial m_{i,j,k}}{\partial P_j}$$

*Equation 170*

The net flow leaving zone *i* (*i=1 to nuc)* is the defined as the residual $r_i$:

$$r_i = \sum_{j=1, j\neq i}^{nout} m_{i,j} = \sum_{j=1, j\neq i}^{j=nout} \sum_{k=1}^{K_{i,j}} m_{i,j,k}$$

*Equation 171*

The $j \neq i$ criterion on the sums eliminates summing $m_{i,i}$ terms which are zero by definition. The zone pressures $\mathrm{P_i}$ are to be determined such that the residuals $\mathrm{r_i}$ all become zero.

Equation 169 and Equation 171 constitute a set of n = nuc nonlinear equations with n = nuc unknown pressures. To linearize the equations, a Taylor's series is used to determine the residual $r_i'$ at the pressure $P_j'$ near the guessed value of pressures $P_j$, where the residual is $r_i$. Keeping only first order terms:

$$r_i' = r_i + \sum_{j=1}^{nuc} \frac{\partial r_i}{\partial P_j}\left(P_j{}' - P_j\right)$$

*Equation 172*

In matrix form this is written:

$$\boldsymbol{r}' = \boldsymbol{r} + \boldsymbol{J}(\boldsymbol{P}' - \boldsymbol{P})$$

*Equation 173*

where $\boldsymbol{r}'$ is the vector with elements $r_i'$, and $\boldsymbol{r}$ is the vector with elements $r_i$.

$\boldsymbol{J}$ is the *nuc*-by-*nuc* Jocobian matrix with elements:

$$J_{i,l} = \frac{\partial r_i}{\partial P_l} = \sum_{j=1}^{nout} \frac{\partial m_{i,j}}{\partial P_l} = \sum_{j=1, j\neq i}^{j=nout} \sum_{k=1}^{k=K_{i,j}} \frac{\partial m_{i,j,k}}{\partial P_l}$$

*Equation 174*

where $i = 1\ to\ nuc, and\ l =\ 1\ to\ nuc$.

Setting $r_i' = 0$ and solving for $P_j{}'$, Equation 173 becomes

$$\boldsymbol{P}' = \boldsymbol{P} - \boldsymbol{J}^{-1}\boldsymbol{r}$$

*Equation 175*

$$\boldsymbol{P}' = \boldsymbol{P} - \boldsymbol{C}$$

*Equation 176*

where $\boldsymbol{C}$ is the correction vector:

$$\boldsymbol{C} = -\boldsymbol{J}^{-1}\boldsymbol{r}$$

*Equation 177*

$$P_i{}' = P_i - C_i$$

*Equation 178*

Equation 178 gives the pressures $P_i{}'$ that are predicted to make $r_i'$ zero.

### 2.9.6.1 Convergence

Convergence is attained when the residuals $r_i$ are sufficiently small. As employed by Energy Plus and Clarke, both absolute and relative magnitude tests are made.

Convergence is assumed when the absolute magnitude of the residual in each zone *i* is less than a predetermined limit *ResMax*:

$$|res_i| < ResMax$$

*Equation 179*

OR, the magnitude of the residual divided by the sum of the magnitudes of the flow through each element connected to zone *i*, is less than a predetermined limit $ResErr$:

$$\frac{|res_i|}{resmag_i} < ResErr$$

*Equation 180*

where

$$resmag_i = \sum_{j=1}^{nout} |m_{i,j}|$$

*Equation 181*

(code uses: sum of magnitude of flows to & from zone iz, resmag(iz) += ABS(mdot(iz,jz,ke)).

### 2.9.6.2 Relaxation

Equation 178 is more generally written as

$$P_i' = P_i - relax * C_i$$

*Equation 182*

where $relax$ is the relaxation coefficient, a factor less than one that reduces the correction applied to $P_i$. Relaxation factors on the order of 0.75 have been shown to reduce the number of iterations in cases normally having slowly decreasing and oscillating corrections. But a fixed value of 0.75 can slow what were formerly rapidly converging cases. The following approach is used to reduce the relaxation factor only when necessary.

Following Clarke, when the corrections $C_i$ from one iteration to the next changes sign, and the latest $C_i$ has a magnitude over half as big as the former $C_i$, then it is assumed that the convergence is probably slow and oscillating. This symptom is typically consistent over a few iterations, and if this were precisely the case, the correction history would follow a geometric progression with a negative common ratio $\frac{C_i}{C_i^{last}}$. Thus by extrapolation a better estimate of correct solution will be obtained if the relaxation factor is taken as the sum of the infinite termed geometric progression:

$$relax = \frac{1}{1 - \frac{C_i}{C_i^{last}}}$$

*Equation 183*

Thus, whenever, during an iteration for zone $i$,

$$\frac{C_i}{C_i^{last}} < -0.5$$

*Equation 184*

then Equation 178 is replaced by

$$P_i{}' = P_i - \frac{1}{1 - \frac{C_i}{C_i^{last}}} * C_i$$

*Equation 185*

Insofar as the extrapolation is warranted, this should give a better prediction of the pressure $P_i{}'$ than would using *relax* = 1 for this iteration. For the iteration following that using Equation 185, *relax* = 1 is reverted to (i.e., Equation 178) so that only unrelaxed correction values are used to evaluate $\frac{C_i}{C_i^{last}}$. The following iteration, if any, is then again tested by Equation 184. The first iteration is always done with *relax = 0.75* since at this point there is no value available for $C_i^{last}$.

It would be reasonable to add a max $Ci$ limit; i.e., max pressure change allowed, a la Clarke, but code testing has not shown the need.

### 2.9.6.3 Off diagonal terms

Consider Equation 174 for off-diagonal terms. Since $i \neq l$, zone $i's$ flow $m_{i,j}$ varies with $P_l$ only if $j = l$. Thus setting $j = l$, and $i \neq l$, Equation 174 reduces to:

$$J_{i,l,i\neq l} = \sum_{k=1}^{K_{i,l}} \frac{\partial m_{i,l,k}}{\partial P_l}$$

*Equation 186*

where $i = 1\ to\ nuc, and\ l =\ 1\ to\ nuc$.

Equation 186, along with Equation 170, shows that all off diagonal terms have a negative magnitude. Since $m_{i,l,k} = -\ m_{l,i,k}$,

$$J_{i,l,i\neq l} = \sum_{k=1}^{K_{i,l}} \frac{\partial m_{i,l,k}}{\partial P_l} = -\sum_{k=1}^{K_{i,l}} \frac{\partial m_{l,i,k}}{\partial P_l}$$

*Equation 187*

Using Equation 170, Equation 187 becomes:

$$J_{i,l,i\neq l} = \sum_{k=1}^{K_{i,l}} \frac{\partial m_{l,i,k}}{\partial P_i} = J_{l,i,i\neq l}$$

*Equation 188*

Thus the Jacobian matrix is symmetric:

$$J_{i,l} = J_{l,i}$$

*Equation 189*

Thus only the upper (or lower) diagonal terms need be determined, with the other half determined by transposition. The off diagonal terms only involve partials of flows between zones with unknown pressures.

### 2.9.6.4 Diagonal terms

For $i = j$ Equation 174 gives:

$$J_{i,i} = \frac{\partial r_i}{\partial P_i} = \sum_{j=1}^{nout} \frac{\partial m_{i,j}}{\partial P_i} = \sum_{j=1}^{j=nout} \sum_{k=1}^{k=K_{i,j}} \frac{\partial m_{i,j,k}}{\partial P_i}$$

*Equation 190*

where $i = 1\ to\ nuc$.

Equation 190 terms can be regrouped to show a simpler numerical way to determine $J_{i,i}$, by using the off diagonal terms already calculated:

$$J_{i,i} = \frac{\partial m_{i,nout}}{\partial P_i} - \sum_{k=1,k\neq i}^{k=nuc} J_{k,i}$$

*Equation 191*

This shows that the diagonal elements use the derivatives of mass flows to the outdoors minus the off diagonal terms in the same column of the Jacobian.

Equation 191 shows that matrix will be singular if $\frac{\partial m_{i,nout}}{\partial P_i} = 0$, so that at least one connection to outdoors is necessary.

# 2.10 Duct System Model

## 2.10.1 Description of Model

The duct model builds on the procedure given by Palmiter (see Francisco and Palmiter, 2003), that uses a steady state heat exchanger effectiveness approach to get analytical expressions for instantaneous duct loss and system efficiencies. The duct model, developed for this program by Palmiter, makes use of many of the same fundamental steady state equations and approach, but given the considerable complexity of the multiple duct systems, does not do a simultaneous solution of all the equations which a generalized Francisco and Palmiter scheme may imply. Instead the approach takes advantage of the small time steps used in the code, and in effect decouples the systems from each other and the zone by basing all losses and other heat transfers occurring during the time step on the driving conditions of Tair and Tmrt known at the beginning of the time step, similar to how heat transfers are determined during mass temperature updates .

Other assumptions made in the duct program: mass and thermal siphon effects in the duct system are ignored.

The duct system performance is analyzed at every time step. The duct air temperatures are calculated assuming they are operating at steady state, in equilibrium with the thermal conditions at the beginning of the time-step in the attic. Heat capacity effects of the ducts are ignored.

During each time step, the following steps are taken to find the duct system operating conditions such as the air temperatures in each duct, the losses, the heating or cooling delivered, etc.

Initially, for each time step, the duct systems performance is determined when operating at full capacity, independent of the load. The procedure starts at the return registers in each conditioned zone, where the duct air temperatures are the current timesteps conditioned zone air temperatures. The conditioned zone air entering the return register heats or cools, or both, as it traversed through each component of the duct system: the return duct, the return plenum, the heating/cooling device, and the supply ducts. That is, the duct air temperature rises or drops immediately downstream of the return register (where returns leaks are assigned to occur) due to mixing of leakage air at the air temperature in the unconditioned zone in which the return duct is located with the return air from the conditioned zone. It may also increase or decrease in temperature in the return plenum as it mixed with the air from the return duct in another unconditioned zone. After being heated or cooled by the air handler at its applicable heating/cooling capacity, it is then additionally heated or cooled by supply duct conductive gains/losses to the interior of the unconditioned zone.

Summing all the gains and losses in temperature of the duct air as it travels through the system gives the supply temperature for the supply duct, allowing the heat delivered at full capacity, $Qdel$, to be determined.

If the above useful heat delivered at full capacity is more than required by the load, then the equipment capacity is reduced to meet the load by assuming the system is only running the fraction $Qload/\ Qdel$ of the time step. The needed capacity, $Qneed$, is this fraction of the nominal capacity. The duct losses for the time step are also reduced by this fraction.

The above calculations are done each time step and the average Qneed summarized in the hourly output.

The above steps are presented in detail in the following sections, in the same sequence as described above.

## 2.10.2 Duct System Inputs

### 2.10.2.1 Subscripts

In most cases in this section, the subscripted variables stand for arrays.

The subscript u stands for the unconditioned zone in which the duct is located.

The subscript c stands for conditioned zone number and its associated air handler system.

The subscript m stands for the mode of air handler operation: 0 off, 1 heating, 2 cooling.

### 2.10.2.2 Annual run inputs

The following data is input to model the duct/air handler system(s):

***Duct inside areas***

$Asd_{c,u}$ = supply duct inside area for air handler c in unconditioned zone u.

$Ard_{c,u}$ = return duct inside area for airhandler c in unconditioned zone u.

***Duct insulation rated R values***

$Rsd_{c,u}$ = supply duct rated R for air handler c in unconditioned zone u; hr-ft$^2$-F/Btu.

$Rrd_{c,u}$ = return duct rated R for air handler c in unconditioned zone u; hr-ft$^2$-F/Btu.

***Inside duct area and inside area based resistance, and the outside duct area and outside area based resistance when there is a single duct segment in the return and supply branches***

Consider one duct of constant inside diameter, $d_i$, and length $L$. The duct is insulated with insulation having a thermal conductivity k, and rated R value, $R_{rate}$. All R values herein are in the units of (hr-ft2-F/Btu).

Layed flat, the thickness the insulation layer is:

$$t\ =\ Rrate * k$$

*Equation 192*

If the insulation is wrapped at this thickness around a duct of diameter $d_i$, the outside diameter, $d_o$, of the insulation will be:

$$d_o = d_i + 2 * Rrate * k$$

so,

$$\frac{d_o}{d_i} = 1 + \frac{2kR_{rate}}{d_i}$$

*Equation 193*

Conduction heat transfer texts gives the overall conductance $C$ of length $L$ of an annular insulation layer as:

$$C = \frac{2\pi kL}{ln\left(\frac{d_o}{d_i}\right)}$$

*Equation 194*

Dividing this by inside area, $A_i = \pi d_i L$, gives the conductance per unit inside area:

$$C_i = \frac{2k}{d_i ln\left(\frac{d_o}{d_i}\right)}$$

The duct resistance value per unit inside area is the reciprocal,

$$R_i = \frac{d_i ln\left(\frac{d_o}{d_i}\right)}{2k}$$

*Equation 195*

This can be written in terms of areas, and length $L$, as:

$$R_i = \frac{A_i ln\left(\frac{A_o}{A_i}\right)}{2\pi kL}$$

*Equation 196*

The duct resistance value based on outside area can be determined from $R_i and\ A_i$ as:

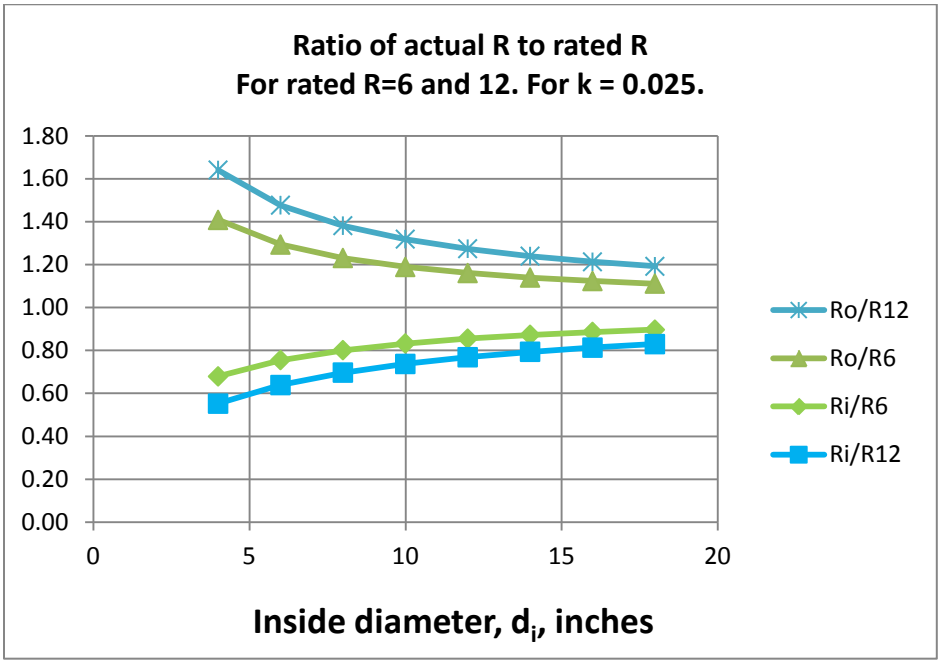
$$R_o = \frac{d_o}{d_i} R_i$$

*Equation 197*

$$A_o = \frac{d_o}{d_i} A_i$$

The $R$ values of Equation 195 and Equation 197, divided by $R_{rated}$, are plotted in Figure 25 as a function of the inside diameter of the duct branch.

**Figure 25: Ratio of Actual R to Rated R**




***Duct system composed of multiple segments in the supply and return branches***

Suppose the supply ducts from an air handler system are branched, with each branch having different sizes, lengths, rated insulation Rrate, and conductivity k values, and all the branches are in one unconditioned zone. These could be combined into one equivalent duct as follows.

The duct branches, j=1→ n, are combined, each of inside areas $A_i(j)$, outside areas $A_o(j)$, conductivities $k(j)$, and inside area based resistances $R_i(j)$. Using the method of Palmiter and Kruse (2003), the overall conductance of the branched duct system, based on inside area, is the sum of the conductances of each branch:

$$'UA' = \sum_{j=1\to n} \left(\frac{A_i(j)}{R_i(j)}\right)$$

*Equation 198*

where $R_i$ for each branch segment $j$ is given by Equation 196 as

$$R_i(j) = \frac{A_i(j) ln\left(\frac{A_o(j)}{A_i(j)}\right)}{2\pi k(j)L(j)}$$

*Equation 199*

With $A_i(j) = \pi d_i(j)L(j)$, and using Equation 193, this can be written as

$$R_i(j) = \frac{d_i(j) ln\left(1 + \frac{2k(j)R_{rate}(j)}{d_i(j)}\right)}{2k(j)}$$

*Equation 200*

The total branch inside area is:

$$A_i = \sum_{j=1\to n} A_i(j)$$

*Equation 201*

The effective overall resistance of the branched duct, based on inside area $A_i$, is thus:

$$R_i = \frac{A_i}{'UA'} = \frac{A_i}{\sum_{j=1\to n}\left(\frac{A_i(j)}{R_i(j)}\right)} = \frac{A_i}{\sum_{j=1\to n}\left[\frac{A_i(j)}{\frac{d_i(j) ln\left(1 + \frac{2k(j)R_{rate}(j)}{d_i(j)}\right)}{2k(j)}}\right]}$$

*Equation 202*

The values of the terms needed for each branch segment, shown on the right hand side of Equation 202 are not available since the former ACM manual only requires that the following R is known:

$$\text{"R"} = \frac{A_i}{\sum_{j=1\to n}\frac{A_i(j)}{R_{rate}(j)}}$$

*Equation 203*

Equation 202 and Equation 203 would be equivalent if Equation 203 had the term $R_i(j)$ in place of $R_{rate}(j)$. As it is, Equation 203 gives the area weighted average $R_{rate}$, not $R_i(j)$.

The total outside area is:

$$A_o = \sum_{j=1\to n} A_o(j)$$

*Equation 204*

Based on outside area, the effective duct system resistance would be:

$$R_o = \frac{A_o}{A_i} R_i$$

*Equation 205*

#### 2.10.2.3 Emissivities

$epss_{c,u}$ = supply duct emissivity for air handler c in unconditioned zone u.

$epsr_{c,u}$ = return duct emissivity for air handler c in unconditioned zone u.

#### 2.10.2.4 Duct leakage

$Ls_{c,u}$ = the fraction of the flow through the system c air handler fan that is leaked from the supply duct in unconditioned zone u. The leak is assigned to occur near the supply register so that the leakage air is at the supply register temperature.

$Lr_{c,u}$ = the fraction of the flow through the system c air handler fan that is leaked into the return duct in unconditioned zone u. The leak is assigned to occur at the return register. The air leaking into the duct is at the unconditioned zone temperature.

#### 2.10.2.5 System flow

$Flow_{m,c}$ = the flow rate in cfm (at standard conditions) through the air handler for the cooling and heating modes, for of each system.

#### 2.10.2.6 Flow distribution

How much of the air handler flow of system c goes through each of its return and supply ducts is given by the per run input flow fractions:

$Fmr_{c,u}$ = fraction of flow of system c in the return duct located in unconditioned zone u.

$Fms_{c,u}$ = fraction of flow of system c in the supply duct located in unconditioned zone u.

$Fmrc_c$ = fraction of flow of system c in the return duct located in conditioned zone c.

$Fmsc_c$ = fraction of flow of system c in the supply duct located in conditioned zone c.

For a given system c, the sum of the return duct fractions must add to one: $Fmr_{c,1}$ + $Fmr_{c,2}$ + $Fmrc_c$ = 1. Similarly for the supply duct fractions.

### *2.10.3 Return Duct Air Temperatures*

Following the procedure indicated in Section 2.10.1, the return duct air temperatures are determined first. Utilizing the heat exchanger effectiveness approach (see Mills (1992), and Appendix A), the temperature of the system c return duct air entering the return plenum from a return duct located in unconditioned zone number u is given by:

$$Tout_{c,u} = Er_{m,c,u} Teqr_{c,u} + (1 - Er_{m,c,u}) \cdot Tmix_{c,u}$$

where $Er_{m,c,u}$ is the effectiveness of the return duct of system c in unconditioned zone u when operating in mode m:

$$Er_{m.c.u} = 1 - e^{\frac{-Urtot_{c,u}}{Mcpr_{m,c,u}}}$$

where $Urtot_{c,u}$ is the total conductance between the return duct air and the equivalent surroundings temperature $Teqr_{c,u}$:

$$Teqr_{c,u} = (Frda_{c,u} \cdot Tair_u + Frdr_{c,u} \cdot Tmrt_u),$$

$Frda_{c,u}$ is the fraction of return duct (dissolved surface node) conductance that goes to the $Tair_u$ node.

$$Frda_{c,u} = \frac{Urc_{c,u}}{(Urc_{c,u} + Urr_{c,u}}$$

$Frdr_{c,u}$ is the fraction of the conductance from the c,u return duct air that goes to the $Tmrt_u$ radiant node.

$$Frdr_{c,u} = \frac{Urr_{c,u}}{Urc_{c,u} + Urr_{c,u}}$$

The U terms are the conductances from the duct air to the mrt and air nodes, determined as described in Appendix A. These conductance values, and the similar supply duct values of Section 2.10.6 are used in the energy balance of the unconditioned zone(s) containing ducts.

$Urr_{cu}$ = conductance from return duct air to Tmrt.

$Urc_{c,u}$ = conductance from return duct air to Tair.

$$Urtot_{c,u} = Urc_{c,u} + Urr_{c,u}$$

The term $Mcpr_{m,c,u}$ is the flow conductance (see below) for the return duct flow:

$$Mcpr_{m,c,u} = Mcp_{m,c} Fmr_{c,u}$$

The total system flow, $Mcp_{m,c}$ is in the "flow conductance" form with the units Btu/hr-F:

$$Mcp_{m,c} = Flow_{m,c} \cdot c_p$$

where $c_p$ is the volumetric heat capacity, which is taken as 1.08 Btu/(hr-F-cfm) for dry air at the ASHRAE standard conditions of density = 0.075 $lb_m/ft^3$ and $c_p$ = 0.24 Btu/$lb_m$-F.

The term $Tmix_{c,u}$ is the mixed air just downstream of the return duct leakage, given by:

$$Tmix_{c,u} = Lr_{c,u} Tair_u + (1 - Lr_{c,u}) Temp_c$$

where $Temp_c$ is the temperature of conditioned zone c's air, assumed to be well-mixed.

### 2.10.4 Return Plenum Temperature and Return Duct Conductive Heat Losses

The heat loss rate from the return duct via convection and radiation, needed in the unconditioned zone energy balance, is:

$$qlr_{c,u} = Mcpr_{m,c,u} \cdot (Tmix_{c,u} - Tout_{c,u})$$

The final return plenum temperature of system c is found by summing the contributions to its plenum temperature from the return ducts in each unconditioned zone and the return ducts located in the conditioned zone. That is,

$$Trp_c = Fmrc_c \cdot Temp_c + \sum_{all\ u} Fmr_{c,u} \cdot Tout_{c,u}$$

### 2.10.5 Temperature Rise through Air Handler Heating or Cooling Equipment

If the mode is heating or cooling, the temperature rise through the air handler heating or cooling equipment of system c at sensible capacity $Cap_c$ is given by:

$$dte_c = \frac{Cap_c}{Mcp_{m,c}}$$

*Equation 206*

The program considers no heat losses or gains from the air handler components other than from the ducts.

### 2.10.6 Supply Plenum and Supply Register Temperatures

The supply plenum temperature is given by:

$$Tsp_c = Trp_c + dte_c$$

*Equation 207*

The supply register temperature for the supply duct of system c in unconditioned space u is:

$$Tsr_{c,U} = Teqs_{c,u} + (1 - Es_{m,c,u}) \cdot (Tsp_c - Teqs_{c,u})$$

*Equation 208*

where $Es_{m,c,u}$ is the effectiveness of the supply duct of system c in unconditioned zone u when operating in mode m:

$$Es_{m,c,u} = 1 - e^{\frac{-Ustot_{c,u}}{Mcps_{m,c,u}}}$$

Substituting the $Tsp_c$ equation above into Equation 208and rearranging gives:

$$Tsr_{c,u} = (1 - Es_{m,c,u})dte_c + Tsrhx_{m,c,u}$$

*Equation 209*

where

$$Tsrhx_{m,c,u} = \left(1 - Es_{m,c,u}\right)Trp_c + Es_{m,c,u}Teqs_{c,u}$$

$Tsrhx$ is the temperature that would be delivered to the supply register with the current mode's flow rate but with zero capacity such that $dte_c = 0$. The duct system is then acting as a heat exchanger (thus the 'hx') between the connected conditioned and unconditioned zones.

The term $s_{c,u}$ , similar to $Teqr_{c,u}$ of Section 2.10.3, is an equivalent environmental temperature defined by

$$Teqs_{c,u} = (Fsda_{c,u} \cdot Tair_u + Fsdr_{c,u} \cdot Tmrt_u),$$

where

$$Fsda_{c,u} = \frac{Usc_{c,u}}{Usc_{c,u} + Usr_{c,u}}$$

$$Fsdr_{c,u} = \frac{Usr_{c,u}}{Usc_{c,u} + Usr_{c,u}}$$

$Usr_{c,u}$= conductance from supply duct air to Tmrt.

$Usc_{c,u}$ = conductance from supply duct air to Tair.

$$Ustot_{c,u} = Usc_{c,u} + Usr_{c,u}$$

The supply duct flow rate is:

$$Mcps_{m,c,u} = Mcp_{m,c} \cdot Fms_{c,u}$$

## 2.10.7 Heating/Cooling Delivered and Supply Duct Conductive Heat Loss

Given $Tsr_{c,u}$, from above, the heat delivered to the conditioned zones by way of the supply ducts located in one or more of the unconditioned zones is given by summing the sensible heat delivered via each unconditioned zones:

$$\text{Q delivered from ducts} = \sum_u Mcpsr_{m,c,u} \cdot (Tsr_{c,u} - Temp_c)$$

*Equation 210*

where $Mcpsr_{m,c,u}$, the flow out the supply register after the supply leakage is removed, is given by:

$$Mcpsr_{m,c,u} = (1 - Ls_{c,u}) \cdot Mcps_{m,c,u}$$

The heat delivered to the conditioned zones by way of ducts in the conditioned zone, which are assumed to have no losses or unbalanced leakage, is given by:

$$\text{Q delivered directly to conditioned zone} = Fmsc_c \cdot Cap_c$$

*Equation 211*

Adding the Q's of Equation 210 and Equation 211 gives the net heating (+), or cooling (-), delivered by the system c as:

$$Qdel_c = Fmsc_c \cdot Cap_c + \sum_{over\ u}[Mcpsr_{m,c,u} \cdot (Tsr_{c,u} - Temp_c)]$$

Substituting the expression for $Tsr_{c,u}$ from Equation 209 into this, $Qdel_c$ can be put in the form:

$$Qdel_c = Qdel1_c + Qdel2_c,$$

where $Qdel1_c$ is the part of Qdel that is independent of air handler capacity. That is, it is the Q delivered if dte is zero, and is the heat exchanged between the unconditioned and conditioned zones via the duct system:

$$Qdel1_c = \sum_{all\ u} [Mcpsr_{m,c,u}(Tsrhx_{m,c,u} - Temp_c)].$$

$Qdel2_c$ is the part of $Qdel$ that is linearly dependent on the air handler capacity:

$$Qdel2_c = Fmsc_c \cdot Cap_c + \sum_{over\ u} [Mcpsr_{m,c,u} \cdot (1 - Es_{m,c,u}) \cdot dte_c]$$

The rate of supply duct conduction losses this time step is given by:

$$qls_{c,u} = Mcps_{m,c,u} \cdot (Tsp_c - Tsr_{c,u})$$

### 2.10.8 Duct System Performance when the Load is Less than the Heat Delivered at Full Capacity

If $Qld_c$ is smaller than the capacity $Qdel_c$, then the system runs only part of the time step. In this case the run time fraction is:

$$Frun_c = \frac{Qld_c}{Qdel_c}$$

The capacity required to meet the load is $Qneed_c$:

$$Qneed_c = Frun_c \cdot Cap_c$$

The duct conductive and leakage losses are also reduced by the same $Frun_c$ fraction.

### 2.10.9 Duct System Performance when the Load is Greater than the Heat Delivered at Full Capacity

In principle this won't occur because the conditioned zone load is limited to the system capacity when it is calculated by the conditioned zone thermostat logic However, the capacity thus calculated is based on the duct efficiency [defined as η=Qload/Qneed] determined for the unconditioned zone during the last time-step, and as a result the load might exceed the capacity determined by the duct model efficiency this time-step.

That is, when the conditioned zone energy balance is performed, and for example heating is called for, then the output capacity of the heating system needs to be known, which requires

knowing the duct system efficiency. But the duct efficiency is only known after the attic simulation is run.

To avoid iteration between the conditioned zone and attic zone modules, the most recent duct efficiency is used to determine the capacity in the conditioned zones thermostat calculations. When the attic simulation is next performed, if the conditioned zone was last running at capacity, and if the efficiency now calculated turns out to be higher than was assumed by the thermostat calculations, then the load will have exceeded the limiting capacity by a small amount depending on the assumed vs. actual efficiency. In cases like this, to avoid iteration, the limiting capacity is allowed to exceed the actual limit by a small amount, so that the correct air handler input energy demand is determined for the conditioned zone load allowed.

In this case, the system is set to run for the full sub-hour time step and the air handler meets the load by increasing its capacity with the following procedure. This procedure, a carryover from the 2008 Residential Building Standards ACM procedures, wherein no capacity limits were imposed on the air handler systems, is as follows.

From the $Qdel1$ and $Qdel2$ equations it can be seen that the capacity needed in this case is:

$$Qneed_c = \frac{Qld_c - Qdel1_c}{Qdel2_c} Cap_c$$

Thus, the temperature rise through the air handler needs to be:

$$dte_c = \frac{Qneed_c}{Mcp_{m,c}}$$

The supply plenum temperature becomes:

$$Tsp_c = Trp_c + dte_c$$

The supply register temperatures is determined reusing Equation 208:

$$Tsr_{c,u} = Teqs_u + (1 - Es_{m,c,u}) \cdot (Tsp_c - Teqs_{c,u})$$

The supply duct losses now become:

$$qls_{c,u} = Mcps_{m,c,u} \cdot (Tsp_c - Tsr_{c,u})$$

The $Qneed_c$'s from each of the time steps during the hour are summed over the hour and reported in the output as $Qneed_c$. The supply and return duct conduction loss terms $qls_{c,u}$ and $qlr_{c,u}$ are used in the energy balance of the unconditioned zone each time step.

## 2.11 Variable Insulation Conductivity

The following correlation is used. It is based on the correlation used in EnergyGauge USA (Parker, et al, 1999) which is based on Wilkes (1981) data:

$k = (kn) \cdot (1 + 0.00418(T_{insul} - 70))$ ; temperatures in °F.

where,

$k$ = insulation conductivity (Btu/hr-ft-R) at the average insulation temperature, $T_{insul}$(F).

$kn$ = nominal insulation conductivity (Btu/hr-ft-R) for insulation at 70 F.

## 2.12 Ceiling Bypass Model

A simple model was implemented to simulate ceiling bypass heat transfer, the heat that is transported from the conditioned zone to the attic via miscellaneous inter-wall cavities in the conditioned zone that may be partially open to the attic, as for example around a fireplace unit. Natural convection in the cavity when the conditioned zone is hotter than the attic is assumed to be the main mechanism for the bypass heat transfer. The conductance, when the conditioned zone air temperature $Tair_c > Tair_u$, the attic air temperature:

$$qbp = U(Tair_c - Tair_u)$$

where, the conductance follows a simple power law dependence on the temperature difference:

$$U = U_{bp}(Temp_1 - Tair_u)^{nbp}$$

$Ubp$ is a coefficient depending on the cavity geometry. Although an exponent of $nbp$ on the order of 1/4 can be assumed for laminar convection, there is no current empirical basis for determining the exponent. If the ACM rule of $U = 0.02Aceil$ were implemented, then $nbp$ would be chosen as zero.

## 2.13 Zone Humidity Balance

### 2.13.1 Zone Humidity Balance

Given a zone with various flows, $m_j$, with humidities $w_j$, entering the zone, and with a scheduled source of water vapor, $m_{sched}$, a water mass balance on the zone gives:

$$\frac{dMw}{dt} = \sum_j m_j(w_j - w) + m_{sched}$$

*Equation 212*

which can be written as:

$$M\frac{dw}{dt} = \sum_j m_j(w_j - w) + m_{sched} - w\frac{dM}{dt}$$

*Equation 213*

where,

$M$ = mass of dry air in the zone; lbm of dry air.

$\frac{dw}{dt}$ = the rate of change of humidity ratio in zone.

$m_j$ = air flow rate from source $j$ into zone; lbm-dry-air/unit-time. Source $j$ can be outdoors, a supply register, adjacent zone, etc.

$w_j$ = humidity ratio of air coming from source $j$; lbm $H_2O$/lbm dry air.

$w$ = humidity ratio of air in zone; lbm $H_2O$/lbm dry air.

$m_{sched}$ = scheduled rate of moisture addition to zone; lbm $H_2O$/unit time.

Using the air perfect gas equation the last term in Equation 213 can be written

$$w\frac{dM}{dt} = -w\frac{M}{T}\frac{dT}{dt}$$

so that Equation 213 becomes

$$M\frac{dw}{dt} = \sum_j m_j(w_j - w) + m_{sched} + w\frac{M}{T}\frac{dT}{dt}$$

*Equation 214*

where $T$ is the air temperature in absolute degrees.

This equation is solved using a forward difference rather than a backward or central difference since a forward difference uncouples the moisture balance equations of each of the zones. Integrating from time $t$ to time $t + \delta t$, where $\delta t$ is the time step, using a forward difference, gives:

$$w(t+\delta t) = \left(m_{sched}(t) + \sum_j m_j(t)w_j(t)\right)\frac{\delta t}{M(t)} + w(t)\left(1 - \frac{\delta t}{M(t)}\sum_j m_j(t) - \frac{T(t+\delta t) - T(t)}{T(t)[\deg R]}\right)$$

*Equation 215*

Notice that all of the values on the right hand side of Equation 215 are determined at $t$ (the beginning of the integration period) except for the $T(t + \delta t)$ term which represents the zone air temperature at the end of the integration period. $T(t + \delta t)$ is known from the zone energy

sensible energy balance at time $t$ (see Section 2.3). The term $\frac{T(t+\delta t)-T(t)}{T(t)[\deg R]}$ is assumed to be negligible and not included in the CSE code.

### 2.13.2 Stability of Solution

The time series solution of Equation 215 will become unstable unless the second term is positive. That is, stability requires

$$\left(\frac{\delta t}{M(t)}\sum_j m_j(t) + \frac{T(t+\delta t)-T(t)}{T(t)[\deg R]}\right) < 1$$

*Equation 216*

Solving for $\delta t$, stability requires

$$\delta t < \frac{M(t)}{\sum_j m_j(t)}\left(1 - \frac{T(t+\delta t)-T(t)}{T(t)[\deg R]}\right)$$

*Equation 217*

Since the zone air changes per unit time is $AC = \frac{\sum_j m_j(t)}{M(t)}$ then the stability requirement can be written in terms of air changes as:

$$AC < \frac{1}{\delta t}\left(1 - \frac{T(t+\delta t)-T(t)}{T(t)[\deg R]}\right)$$

*Equation 218*

If the solution is unstable at the given $\delta t$, the zone air mass M(t) can be temporarily boosted up such that:

$$M(t) > \frac{\delta t \sum_j m_j(t)}{\left(1 - \frac{T(t+\delta t)-T(t)}{T(t)[\deg R]}\right)}$$

This will lead to a higher latent capacity for the zone air, introducing some error in the zone humidity prediction. This will also lead to a zone latent heat imbalance unless this artificial increase in zone air is accounted for.

### 2.13.3 Hygric Inertia of Zone

The absorption/desorption of moisture in the zone is accounted for using the hygric inertial model of Vereecken et al. whereby a multiplier X is added to the M(t) term of Equation 10 and Equation 11. An appropriate value of X can be measured for the complete zone and all of its furnishings by using the protocol given by [Vereecken E, Roels S, Janssen H, 2011. In situ

determination of the moisture buffer potential of room enclosures, Journal of Building Physics, 34(3): 223-246.]

## 2.14 Zone Comfort Algorithm

CSE includes an implementation of the ISO 7730 comfort model. The model is documented in ASHRAE Standard 55-2010 (ASHRAE 2010) among other places. The model calculates Predicted Mean Vote (PMV) and Predicted Percent Dissatisfied (PPD) for each zone at each time step. These statistics are averaged over days, months, and the full year.

The inputs to the ISO 7730 model are:

- Air dry-bulb temperature
- Air humidity ratio
- Mean radiant temperature
- Air velocity
- Occupant metabolic rate
- Occupant clothing level

Zone conditions calculated by CSE are used for the first three of these inputs. The remaining inputs are set by user input. They can be varied during the simulation using the CSE expression capability.

## 2.15 HVAC Equipment Models

Air conditioning systems shall be sized, installed, tested and modeled according to the provisions of this section.

### 2.15.1 Compression Air-Conditioner Model

The Compliance Software calculates the hourly cooling electricity consumption in kWh using Equation 219. In this equation, the energy for the air handler fan and the electric compressor or parasitic power for the outdoor unit of a gas absorption air conditioner are combined. The Compliance Software calculates the hourly cooling gas consumption in therms using Equation 219.

$$AC_{kWh} = \frac{Fan_{Wh} + Comp_{Wh}}{1{,}000}$$

*Equation 219*

Where:

$AC_{kWh}$ = Air conditioner kWh of electricity consumption for a particular hour of the simulation. This value is calculated for each hour, combined with the TDV multipliers, and summed for the year.

$Fan_{Wh}$ = Indoor fan electrical energy for a particular hour of the simulation, Wh.

$Comp_{Wh}$ = Electrical energy for all components except the indoor fan for a particular hour of the simulation, Wh. This value includes consumption for the compressor plus outdoor condenser fan and is calculated using Equation 221.

CSE calculates the energy for electrically driven cooling using the algorithms described in this section.

**Primary model parameters**. The following values characterize the AC unit and are constant for a given unit:

Cap95 = AHRI rated total cooling capacity at 95 °F, Btuh

$CFM_{per\ ton}$ = Air flow rate per ton of cooling capacity, cfm/ton.

$$= \frac{\text{Operating air flow rate, cfm}}{\text{Cap95 / 12000}}$$

EFan = Fan operating electrical power, W/cfm. Default = 0.365.

SEER = AHRI rated Seasonal Energy Efficiency Ratio, Btuh/W. EER shall be used in lieu of the SEER for equipment not required to be tested for a SEER rating.

EER = AHRI rated energy efficiency ratio at 95 °F, Btuh/W. If EER is not available, it is derived from SEER as follows:

| If SEER | >=16 | >=13 & <16 | <13 |
|---|---|---|---|
| EER | 13 | 11.3 + 0.57 x (SEER-13) | 10+0.84×(SEER– 11.5) |

$F_{chg}$ = Refrigerant charge factor, default = 0.9. For systems with a verified charge indicator light (Reference Residential Appendix RA3.4) or verified refrigerant charge (Reference Residential Appendix RA3), the factor shall be 0.96.

$F_{size}$ = Compressor sizing factor, default = 0.95. For systems sized according to the Maximum Cooling Capacity for compliance software Credit (see Section <TODO>), the factor shall be 1.0.

**Derived model parameters**. The following values are used in the formulas below and depend only on model parameters.

Tons = Nominal cooling capacity defined as Cap95 / 12000

$QFan_{rat}$ = Assumed fan heat included at AHRI test conditions, Btuh

| Fan motor type | $QFan_{rat}$ |
|---|---|
| PSC | 500 x Cap95 / 12000 |
| BPM | 283 x Cap95 / 12000 |

$QFan_{op}$ = Fan heat assumed during operation (i.e., during simulation), Btuh

$$QFan_{op} = \frac{CFM_{per\ ton} \times Cap95 \times EFan \times 3.413}{12000}$$

*Equation 220*

**Model inputs**. The following values vary at each time step in the simulation and are used in the formulas below to determine AC unit performance under for that time step.

$DB_t$ = Dry bulb temperature of air at the condensing unit, °F (typically outdoor air temperature).

$WB_{ec}$ = Coil entering air wet bulb temperature, °F (return air temperature adjusted for blow-through fan heat if any)

$DB_{ec}$ = Coil entering air dry bulb temperature, °F (return air temperature adjusted for blow through fan heat if any)

Qneed = Cooling system sensible cooling output, Btuh. Qneed is calculated across the unit and thus includes both the building load and distribution losses.

Compressor energy for a particular time step of the simulation shall be calculated using Equation 221.

$$Comp_{wh} = \frac{QFan_{op} + Qneed}{CE_t}$$

*Equation 221*

Where:

$Fan_{wh}$ = Fan power for this time step, Wh.

$CE_t$ = Sensible energy efficiency at current conditions, Btuh/W. This is calculated using Equation 222 below.

$$CE_t = EER_t \times SHR$$

*Equation 222*

Where:

$EER_t$ = Energy efficiency ratio at current conditions, Btuh/W. $EER_t$ is calculated using Equation 226 below.

SHR = Sensible Heat Ratio (sensible capacity / total capacity), derived as follows:

$$
\begin{aligned}
SHR = \text{minimum}(1, \quad & A_{SHR} \times DB_{ec} + \\
& B_{SHR} \times WB_{ec} + \\
& C_{SHR} \times DB_t + \\
& D_{SHR} \times CFM_{per\ ton} + \\
& E_{SHR} \times DB_{ec} \times DB_t + \\
& F_{SHR} \times DB_{ec} \times CFM_{per\ ton} + \\
& G_{SHR} \times WB_{ec} \times DB_t + \\
& H_{SHR} \times WB_{ec} \times CFM_{per\ ton} + \\
& I_{SHR} \times DB_t \times CFM_{per\ ton} + \\
& J_{SHR} \times WB_{ec}^2 + \\
& K_{SHR} / CFM_{per\ ton} + \\
& L_{SHR})
\end{aligned}
$$

SHR coefficients:

| | |
|---|---|
| $A_{SHR}$ | 0.0242020 |
| $B_{SHR}$ | -0.0592153 |
| $C_{SHR}$ | 0.0012651 |
| $D_{SHR}$ | 0.0016375 |
| $E_{SHR}$ | 0 |
| $F_{SHR}$ | 0 |
| $G_{SHR}$ | 0 |
| $H_{SHR}$ | -0.0000165 |
| $I_{SHR}$ | 0 |
| $J_{SHR}$ | 0.0002021 |
| $K_{SHR}$ | 0 |
| $L_{SHR}$ | 1.5085285 |

$CAP_{nf}$ = Total cooling capacity across coil (that is, without fan heat) at current conditions, Btuh

$$CAP_{nf} = \left(Cap95 + QFan_{rat}\right) \times F_{chg} \times F_{size} \times F_{cond_cap}$$

*Equation 223*

$F_{cond_cap}$ = $A_{CAP} \times DB_{ec}$ +
$B_{CAP} \times WB_{ec}$ +
$C_{CAP} \times DB_{t}$ +
$D_{CAP} \times CFM_{per\ ton}$ +
$E_{CAP} \times DB_{ec} \times DB_{t}$ +
$F_{CAP} \times DB_{ec} \times CFM_{per\ ton}$ +
$G_{CAP} \times WB_{ec} \times DB_{t}$ +
$H_{CAP} \times WB_{ec} \times CFM_{per\ ton}$ +
$I_{CAP} \times DB_{t} \times CFM_{per\ ton}$ +
$J_{CAP} \times WB_{ec}^{2}$ +
$K_{CAP} / CFM_{per\ ton}$ +
$L_{CAP}$

Coefficients as follows:

| SHR Condition | SHR < 1 | SHR = 1 |
|---|---|---|
| $A_{CAP}$ | 0 | 0.009483100 |
| $B_{CAP}$ | 0.009645900 | 0 |
| $C_{CAP}$ | 0.002536900 | -0.000600600 |
| $D_{CAP}$ | 0.000171500 | -0.000148900 |
| $E_{CAP}$ | 0 | -0.000032600 |
| $F_{CAP}$ | 0 | 0.000011900 |
| $G_{CAP}$ | -0.000095900 | 0 |
| $H_{CAP}$ | 0.000008180 | 0 |
| $I_{CAP}$ | -0.000007550 | -0.000005050 |
| $J_{CAP}$ | 0.000105700 | 0 |
| $K_{CAP}$ | -53.542300000 | -52.561740000 |
| $L_{CAP}$ | 0.381567150 | 0.430751600 |

$CAP_{sen}$ = Sensible capacity including fan heat, Btuh

$$CAP_{sen} = CAP_{nf} \times SHR - QFan_{op}$$

*Equation 224*

$CAP_{lat}$ = Latent capacity, Btuh

$$CAP_{lat} = CAP_{nf} - CAP_{sen}$$

*Equation 225*

Note: The air leaving the AC unit is limited to 95% relative humidity. If that limit is invoked, $CAP_{lat}$ is reduced and $CAP_{sen}$ is increase.

$EER_t$ is calculated as follows:

When

| | |
|---|---|
| $DB_t \leq 82$ ºF | $EER_t = SEER_{nf}$ |
| $82 < DB_t < 95$ °F | $EER_t = SEER_{nf} + ((DB_t - 82)*(EER_{nf} - SEER_{nf}) / 13)$ |
| $DB_t \geq 95$ °F | $EER_t = EER_{nf}$ |

*Equation 226*

Where:

$SEER_{nf}$ = Seasonal energy efficiency ratio at current conditions without distribution fan consumption ("nf" = no fans). This is calculated using Equation 227.

$EER_{nf}$ = Energy efficiency ratio at current conditions without distribution fan consumption ("nf" = no fans) and adjusted for refrigerant charge and airflow. This is calculated using Equation 228.

$$SEER_{nf} = \frac{F_{chg} \times F_{size} \times F_{cond_SEER} \times (1.09 \times Cap95 + QFan_{rat})}{1.09 \times Cap95/SEER - QFan_{rat}/3.413}$$

*Equation 227*

$$\begin{aligned} F_{cond_SEER} = F_{cond_cap} / \quad & (A_{SEER} \times DB_{ec} + \\ & B_{SEER} \times WB_{ec} + \\ & C_{SEER} \times DB_t + \\ & D_{SEER} \times CFM_{per\ ton} + \\ & E_{SEER} \times DB_{ec} \times DB_t + \\ & F_{SEER} \times DB_{ec} \times CFM_{per\ ton} + \\ & G_{SEER} \times WB_{ec} \times DB_t + \\ & H_{SEER} \times WB_{ec} \times CFM_{per\ ton} + \end{aligned}$$

$I_{SEER} \times DB_t \times CFM_{per\ ton} +$

$J_{SEER} \times WB_{ec}^2 +$

$K_{SEER} / CFM_{per\ ton} +$

$L_{SEER})$

Coefficients as follows:

| SHR Condition | SHR < 1 | SHR = 1 |
|---|---|---|
| $A_{SEER}$ | 0 | 0.0046103 |
| $B_{SEER}$ | -0.0202256 | 0 |
| $C_{SEER}$ | 0.0236703 | 0.0125598 |
| $D_{SEER}$ | -0.0006638 | -0.000512 |
| $E_{SEER}$ | 0 | -0.0000357 |
| $F_{SEER}$ | 0 | 0.0000105 |
| $G_{SEER}$ | -0.0001841 | 0 |
| $H_{SEER}$ | 0.0000214 | 0 |
| $I_{SEER}$ | -0.00000812 | 0 |
| $J_{SEER}$ | 0.0002971 | 0 |
| $K_{SEER}$ | -27.95672 | 0 |
| $L_{SEER}$ | 0.209951063 | -0.316172311 |

$$EER_{nf} = \frac{Cap_{nf}}{F_{cond_EER} \times (Cap95/EER - QFan_{rat}/3.413)}$$

*Equation 228*

Where:

$F_{cond_EER} = (A_{EER} \times DB_{ec} +$

$B_{EER} \times WB_{ec} +$

$C_{EER} \times DB_t +$

$D_{EER} \times CFM_{per\ ton} +$

$E_{EER} \times DB_{ec} \times DB_t +$

$F_{EER} \times DB_{ec} \times CFM_{per\ ton} +$

$G_{EER} \times WB_{ec} \times DB_t +$

$H_{EER} \times WB_{ec} \times CFM_{per\ ton} +$

$I_{EER} \times DB_t \times CFM_{per\ ton} +$

$J_{EER} \times WB_{ec}^2 +$

$K_{EER} / CFM_{per\ ton} +$

$L_{EER})$

Coefficients as follows:

| SHR Condition | SHR < 1 | SHR = 1 |
|---|---|---|
| $A_{EER}$ | 0 | 0.004610300 |
| $B_{EER}$ | -0.020225600 | 0 |
| $C_{EER}$ | 0.023670300 | 0.012559800 |
| $D_{EER}$ | -0.000663800 | -0.000512000 |
| $E_{EER}$ | 0 | -0.000035700 |
| $F_{EER}$ | 0 | 0.000010500 |
| $G_{EER}$ | -0.000184100 | 0 |
| $H_{EER}$ | 0.000021400 | 0 |
| $I_{EER}$ | -0.000008120 | 0 |
| $J_{EER}$ | 0.000297100 | 0 |
| $K_{EER}$ | -27.956720000 | 0 |
| $L_{EER}$ | 0.015003100 | -0.475306500 |

## 2.15.2 Air-Source Heat Pump Model (Heating mode)

The air source heat pump model is based on methods presented in AHRI Standard 210/240-2008.

**Primary model parameters**. The following values characterize the ASHP and are constant for a given unit:

Cap47 = Rated heating capacity at outdoor dry-bulb temperature = 47 °F

COP47 = Coefficient of performance at outdoor dry bulb = 47 °F (if available, see below)

| | | |
|---|---|---|
| Cap35 | = | Heating capacity under frosting conditions at outdoor dry-bulb temperature = 35 °F (if available, see below) |
| COP35 | = | Coefficient of performance at outdoor dry bulb = 35 °F (if available, see below) |
| Cap17 | = | Rated heating capacity at outdoor dry-bulb temperature = 17 °F |
| COP17 | = | Coefficient of performance at outdoor dry bulb = 17 °F (if available, see below) |
| HSPF | = | Rated Heating Seasonal Performance Factor, Btuh/Wh |
| COPbu | = | COP of backup heating, default = 1 (electric resistance heat) |
| Capbu | = | Available backup heating capacity, Btuh |

**Derived model parameters.**

| | | |
|---|---|---|
| Inp47 | = | Electrical input power at 47 °F = Cap47 / COP47, Btuh (not W) |
| Inp17 | = | Electrical input power at 17 °F = Cap17 / COP17, Btuh (not W) |

**Estimation of unavailable model parameters.**

$$\text{COP47} = 0.3038073 \times \text{HSPF} - 1.984475 \times \frac{\text{Cap17}}{\text{Cap47}} + 2.360116$$

$$\text{COP17} = 0.2359355 \times \text{HSPF} + 1.205568 \times \frac{\text{Cap17}}{\text{Cap47}} - 0.1660746$$

$$Cap35 = 0.9 \times [Cap17 + 0.6 \times (Cap47 - Cap17)]$$

$$Inp35 = 0.985 \times [Inp17 + 0.6 \times (Inp47 - Inp17)]$$

$$COP35 = \frac{Cap35}{Inp35}$$

**Simulation**

Full-load capacity and input power of the ASHP is determined each time step as a function of outdoor dry-bulb temperature T, as follows --

If (17 °F < T < 45 °F)

$$Cap(T) = Cap17 + \frac{(Cap35 - Cap17) \times (T - 17)}{35 - 17}$$

$$Inp(T) = Inp17 + \frac{(Inp35 - Inp17) \times (T - 17)}{35 - 17}$$

Else

$$Cap(T) = Cap17 + \frac{(Cap47 - Cap17) \times (T - 17)}{47 - 17}$$

$$Inp(T) = Inp17 + \frac{(Inp47 - Inp17) \times (T - 17)}{47 - 17}$$

**Resistance heat.**

Load in excess of Cap(T) is met with backup heating at COPbu.

### 2.15.3 Equipment Sizing

CSE determines the capacity of HVAC equipment via an auto-sizing capability. Autosizing is conducted prior to the main annual simulation. It is done by using the hourly simulator for a set of design days and increasing capacity as needed to maintain thermostat set points. Each design day is repeated several times until the required capacity converges. The set of design days includes one cold day with no solar gain and several hot days at with clear-sky solar at different times of the year. This ensures that maximums of both heating and cooling loads will be found. Equipment characteristics other than capacity are specified on a per-unit basis (e.g. "cfm per ton"), so a full description of the system can be derived from the primary capacity.

The sizing procedure uses the equipment models in an inverse mode. For example, the sensible cooling load for a given set up conditions is back-converted to the required rated total capacity (Cap95) by using inverted forms of the model equations. The general simulation calculation sequence is used, but the logic of the HVAC models is altered during the autosizing phase.

Note that for air-source heat pumps, only the backup heating capacity is autosized. In addition, modeled duct sizes are not sized and must be specified.

The equipment sizes calculated by CSE are used for compliance analysis only and are not substitutes for load calculations used for selecting equipment or meeting other code requirements.

# 3 Compliance Manager

## 3.1 Overview

## 3.2 One-dimensional Roof Edge Heat Transfer Model

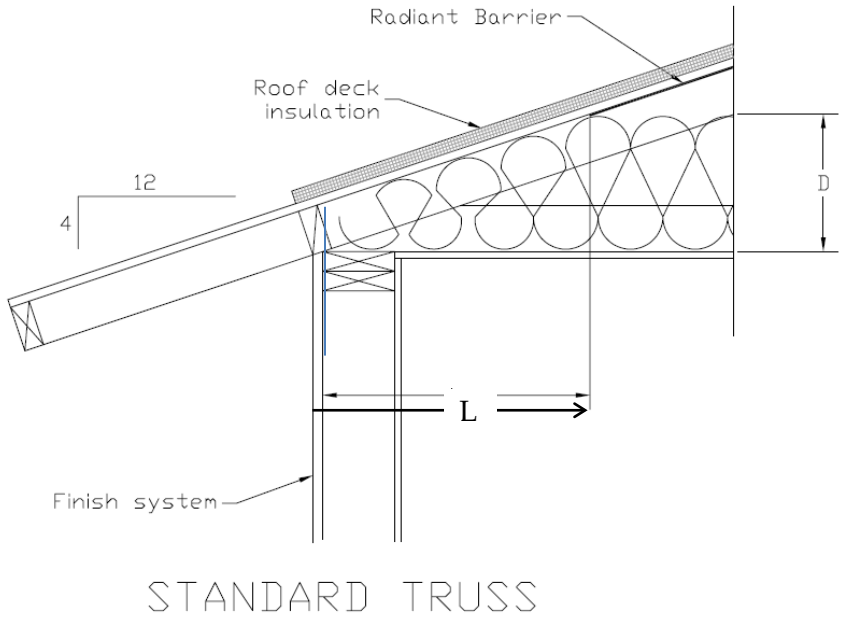
### *3.2.1 Construction Practice*

This document describes the one-dimensional model used to represent the heat flow between the conditioned zone and the outdoors through the portion of the ceiling insulation, along the outside edge of the attic, through which the heat flows to the outdoors without passing through the attic air. This portion will be modeled in CSE as cathedral ceilings, and is referred herein as the roof edge. The rest of the heat flow path through the ceiling insulation will be modeled as part of the attic zone, and is not discussed here.

Two types of roof construction are considered, standard-heel and raised-heel trusses, shown in Figure 26 and Figure 27, with the geometries assumed to be representative of current practice. The roof trusses are assumed to be framed with 2x4's. Although the figures are for a roof with a 4-in-12 pitch, the 1-D model will handle any standard pitch. The distance between the wall plate and roof deck (shown, for example, as 12 inches in Figure 27) is also not restricted to the distances implied by Figure 26 and Figure 27.

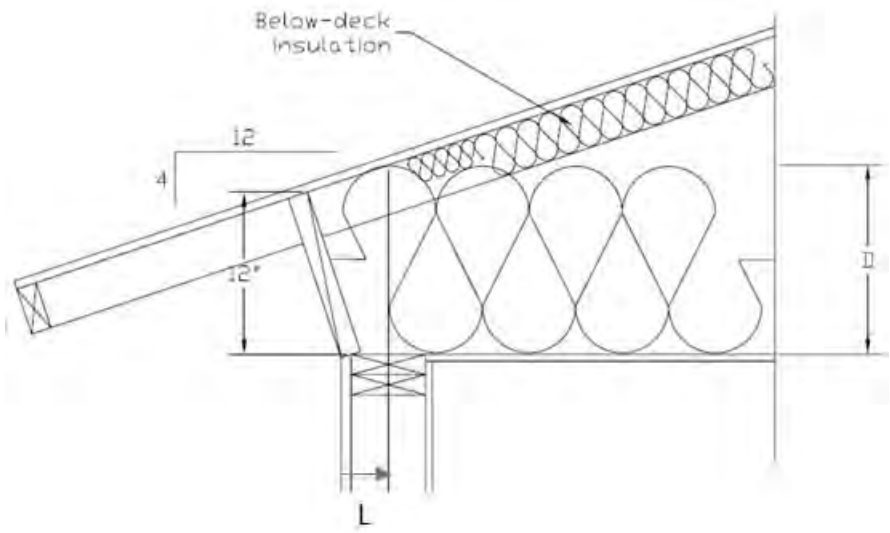
The 1-D model is developed in order to simplify the heat transfer calculation for roof edges, while preserving the steady state and transient characteristics (layer mass) of the typical roof constructions addressed. The 1-D model produces the dimensions of the construction layers needed to represent the roof edges.

**Figure 26: Standard-Heel Geometry**




L and D determine where the top of the ceiling insulation meets the roof deck plane.

**Figure 27: Raised-Heel Geometry**




L and D determine where the top of the ceiling insulation meets the roof deck plane.

## 3.2.2 One-Dimensional Model

Using the parallel path method, the heat transfer is determined separately for the insulation and framing paths of the constructions.

First consider modeling the standard-heel truss of Figure 26.

### 3.2.2.1 Standard heel insulation path

For the path through the insulation, Figure 26 is approximated as the simpler 2-D configuration of Figure 28 and Figure 30, with the left vertical edge assumed to be adiabatic and of height $Y$. The right vertical edge is also assumed to be adiabatic. To partly compensate for not allowing heat flow out the left side tilted edge board in Figure 26, the ceiling is assumed to extend to the outer edge of the vertical wall.

The width, $W_A$ of roof edge path A, is taken as the width of the vertical path of solid wood in the framing section view of Figure 30.

**Figure 28: Standard-Heel Simplified Geometry for Insulation Path**

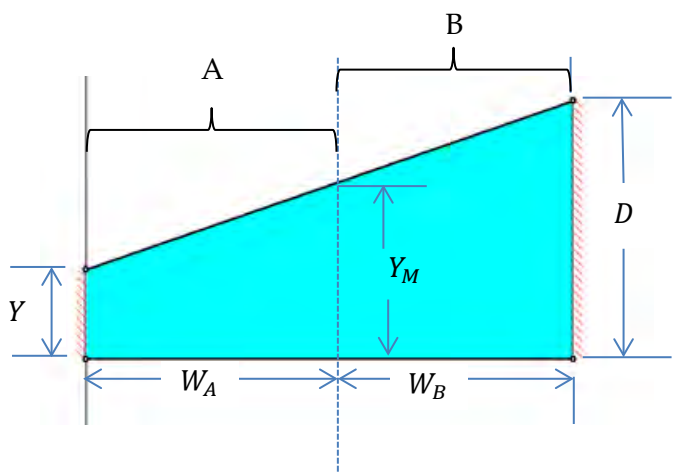
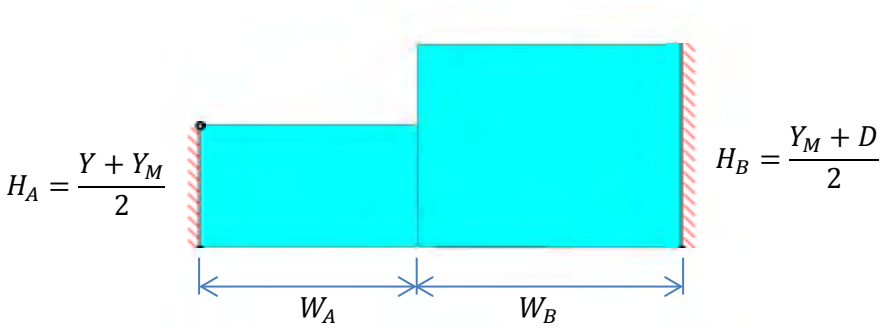



Figure 28 is then reduced to the 1-D form shown in Figure 29, where the layer thicknesses are taken as the average height of the layer in Figure 28.

**Figure 29: Standard-Heel 1-D Geometry for Insulation Path**




The left hand portion represents the insulation path of the 1-D model of roof edge A. The right side represents the insulation path of the 1-D model of roof edge B.

The 1-D model just considers the ceiling insulation and framing. When implemented as part of a cathedral ceiling in CSE, a sheetrock layer would be added to the bottom of Figure 29 paths. Layers added to the tops of the layers in Figure 29 would be decking, asphalt shingles, and tile, for example.

### 3.2.2.2 Standard heel framing path

Similar to the insulation path, the framing heat transfer path starts with Figure 30, which is reduced to Figure 31. The widths of A and B are the same as for the insulation path figures. $H_{B1}$, and $H_{B3}$ are the vertical thickness of the 2x4's and $H_{B2}$ is the average thickness of insulation.

**Figure 30: Standard-Heel Simplified Geometry for Framing Path**

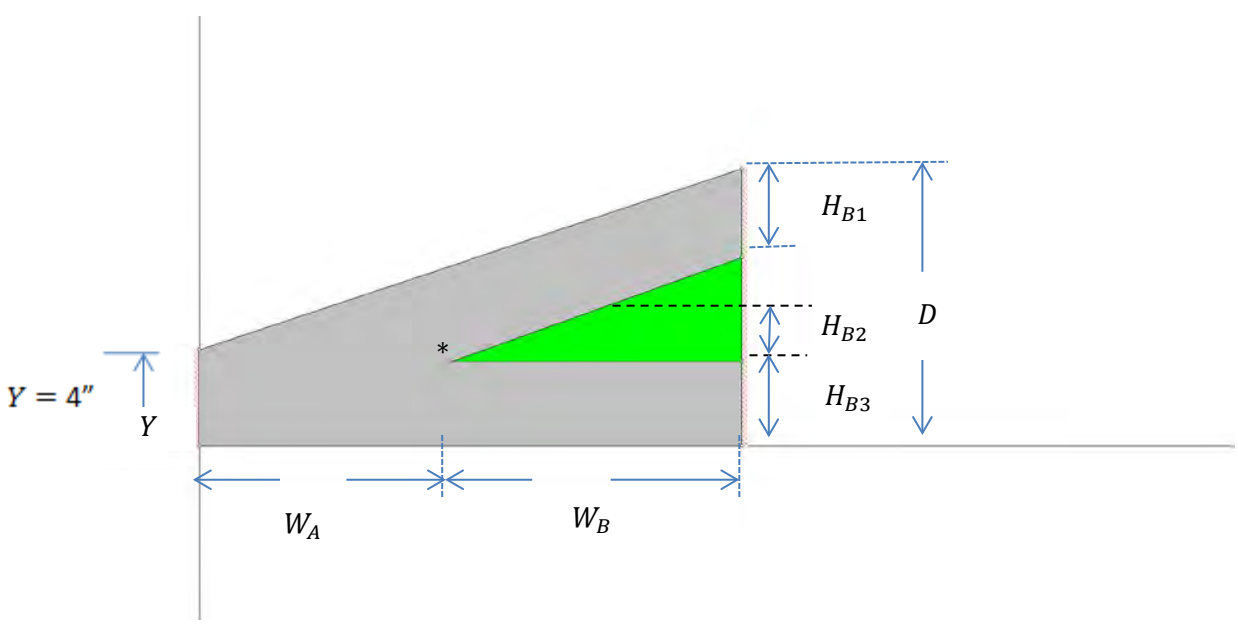
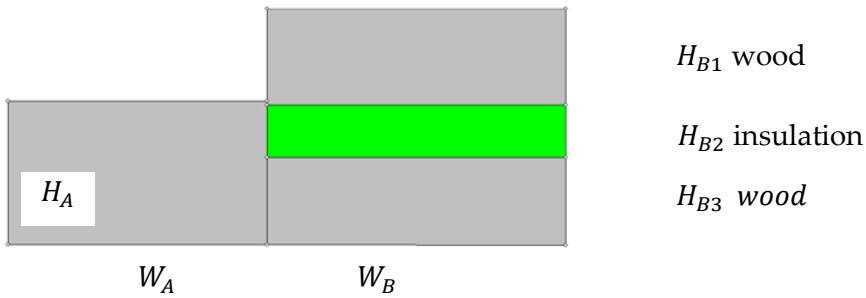


**Figure 31: Standard-Heel 1-D Geometry for Framing Path**



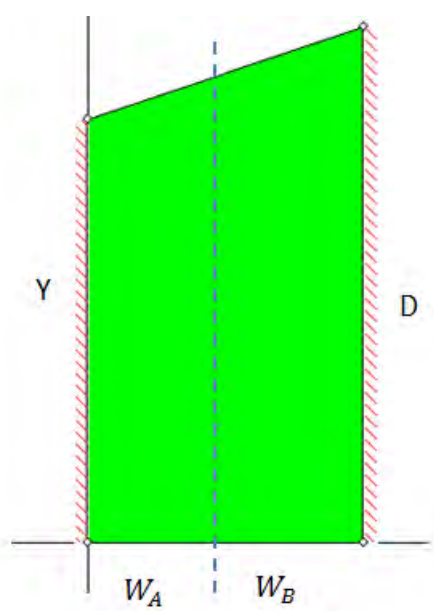

### 3.2.2.3 Raised heel

The 1-D model for the raised-heel case of Figure 27 is different than for the standard-heel case of Figure 26. The geometry is illustrated in Figure 32 and Figure 33 for a ceiling insulation of R38.

As the edge height, $Y$, in Figure 30 is increased, the deck and ceiling 2x4's separate vertically near the roof edge, and a vertical 2x4 is assumed to fill the gap. That is, as $Y$ is increased, the standard truss geometry of Figure 28 and Figure 30 morphs into the raised truss geometry of Figure 32 and Figure 33. The 1-d roof edge algorithm below, gives the layering outputs for both the standard-heel truss and the raised-heel truss and everything in between.

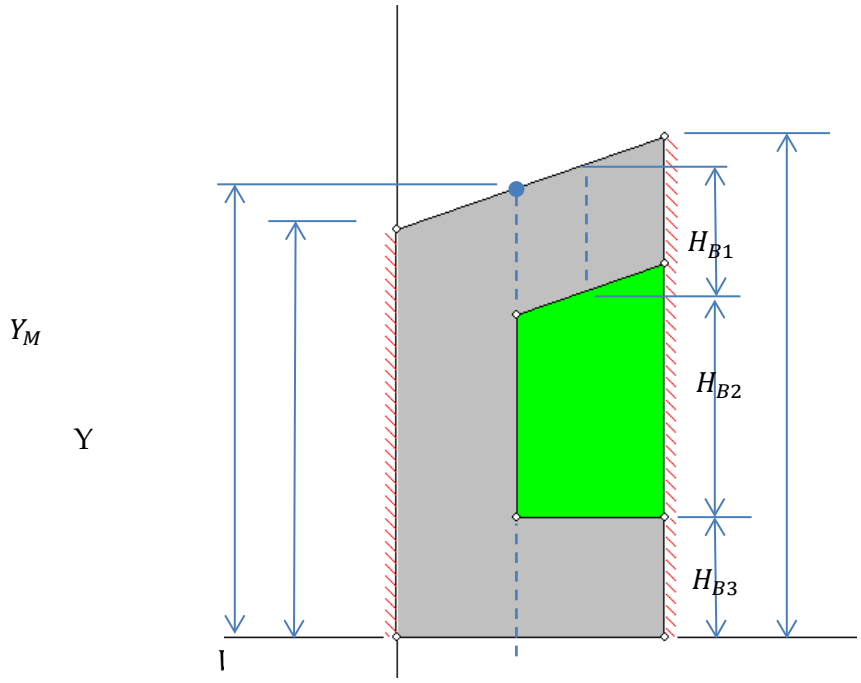
### 3.2.2.4 Raised heel insulation path

**Figure 32: Raised-Heel 1-D Insulation Path Geometry**



### 3.2.2.5 Raised heel frame path

**Figure 33: Raised-Heel 1-D Framing Path Geometry**




$H_{B1}$, and $H_{B3}$ are the vertical thickness of the 2x4's and $H_{B2}$ is the average thickness of insulation.

### 3.2.2.6 Roof edge algorithm

The following algorithm, written in succinct pseudo-code form, determines the layer widths and thicknesses for roof edge paths A and B. [This algorithm is implemented in RoodEdgeAlgorithm-1.xlsx].

All dimensions are assumed to be feet.

**Input:**

$Y$ = edge height, ft.

$w$ = framing width, ft. [nominally (3.5/12)-ft]

$Rtot$ = total R value of ceiling insulation, hr-ft²-F/Btu.

$R_{perin}$ = R value of the ceiling insulation for one inch thickness, hr-ft²-F/Btu.

$P$ = pitch = rise/12

**Calculation of $W_A, W_B, H_A, H_B, H_{B1}$, $H_{B2}$, and $H_{B3}$:**

$D = \frac{Rtot}{12*R_{perin}}$ insulation depth, ft

$Y_I = w\left(1 + \sqrt{1 + P^2}\right)$ vertical thickness at position $x = X_I$.

$X_I = \frac{Y_I - Y}{P}$ horiz distance from left edge to intersection of deck and ceiling 2x4's (see * in Figure 30). In Figure 33, the corresponding point would be outside the roof section, and near the left edge of the page, and as opposed to the * point, this $X_I$ will is a negative number since it is to left of origin at outside surface of top wall plate.

IF $X_I \geq w$:

$Y_M = Y_I$ $Y_M$ is the height of roof section at the vertical line between A and B.

ELSE $X_I < w$:

$Y_M = (w - X_I)P + Y_I$

END IF

IF $D \geq Y_M$:

$$W_B = \frac{D - Y_M}{P}$$

$$H_A = \frac{Y + Y_M}{2}$$

$$H_B = \frac{Y_M + D}{2}$$

$$H_{B1} = w\sqrt{1 + P^2}$$

$$H_{B2} = H_B - Y_I$$

$$H_{B3} = w$$

IF $X_I \geq w$:

$W_A = X_I$

ELSE $X_I < w$:

$W_A = w$

END IF

ELSE IF $D < Y_M$ AND $D > Y$:

$$W_A = \frac{D - Y}{P}$$

$$H_A = \frac{D + Y}{2}$$

$W_B = H_B = H_{B1} = H_{B2} = H_{B3} = 0$

ELSE (D ≤ Y)

$W_A = H_A = 0$

$W_B = H_B = H_{B1} = H_{B2} = H_{B3} = 0$

END IF

PRINT OUTPUT: $W_A, W_B, H_A, H_B, H_{B1}, H_{B2}, H_{B3}$

## 3.2.3 Roof Edge Model Validation

The roof edge heat transfer is basically a 3-D problem. The 1-D model makes a number of simplifications. For example, the parallel insulation and framing path assumption ignores lateral heat transfer between the insulation and framing path, and leads to an underestimation of the overall heat transfer. The assumption of an adiabatic right hand border, where in reality the heat flow lies are not quite vertical, also underestimates the heat transfer through the cathedral ceilings with an accompanying overestimation of the remaining heat transfer through the attic portion of the ceiling insulation. The assumption of the layer thicknesses taken as the average layer thickness ignores 2-D effects. The complicated 2-D heat transfer at the junction of the vertical wall and roof is simplified by assuming the left border is adiabatic, and the ceiling continuation to the outside of the wall. Corner effects for the roof edge, where vertical walls meet at right angles, results in a 3-D heat flow situation that can only be estimated.

Because of these complexities, it is difficult to assess the accuracy of the 1-D model.

However, in order to obtain some perspective on the accuracy of the 1-D model, the heat transfer was calculated for two cases, of different insulation depths, using both the 1-D roof edge algorithm, and a 2-D (using FEHT finite-element program) solution with the roof edge 2-D geometry of Figure 26. The 2-D model still requires many of the assumptions made in the 1-D model, including the parallel path assumption.

### 3.2.3.1 1-D model, Rtot = 30, Y = 4-inches

Using the 1-D Roof Edge Algorithm, the heat transfer rates through roof edges A and B was calculated for the following inputs.

**Input to Roof Edge Algorithm**

$R_{tot} = 30$ hr-ft²-F/Btu.
$P = 4/12$
$Y = 0.3333$ ft

$w = 0.2917$ ft

$R_{perin} = 2.6$ hr-ft²-F/Btu.

**Output of Roof Edge Algorithm**

$W_A$ =0.797 ft

$W_B$ = 1.087 ft

$H_A$ =0.466 ft

$H_B$ =0.780 ft

$H_{B1}$ = 0.292 ft

$H_{B2}$ = 0.182 ft

$H_{B3}$ = 0.307 ft

**Insulation Path Results**

The insulation conductivity is $k = \frac{1}{12R_{perin}} = 0.03205$ Btu/hr-ft-F.

The thermal resistance of A and B are:

$$R_A = \frac{H_A}{k_{insul}} = 14.546 \text{ hr-ft}^2\text{-F/Btu}$$

$$R_B = \frac{H_B}{k_{insul}} = 24.346 \text{ hr-ft}^2\text{-F/Btu}$$

If no additional layers are added (sheetrock, etc.), and the top and bottom surface temperature difference is 100 F, the heat transfer rate in this case, per foot of roof perpendicular to the section, becomes:

$$Q_A = \frac{W_A * \Delta T}{R_A} = \frac{0.797*100}{14.546} = 5.48 \text{ Btu/hr-ft}$$

$$Q_B = \frac{W_B * \Delta T}{R_B} = \frac{1.087*100}{24.346} = 4.465 \text{ Btu/hr-ft}$$

The total is:

$$Q_{insulpath} = Q_A + Q_B = 9.944 \text{ Btu/hr-ft}$$

**Framing Path Results**

Assume wood framing conductivity k = 0.084 hr-ft-F/Btu.

The thermal resistance of A, per foot of roof edge perpendicular to the section:

$$R_A = \frac{H_A}{k_{wood}} = 5.55 \text{ hr-ft}^2\text{-F/Btu}$$

The thermal resistance of path B; sum of layer resistances:

$$R_B = \frac{H_{B1}}{k_{wood}} + \frac{H_{B2}}{k_{insul}} + \frac{H_{B3}}{k_{wood}} = 3.66+5.654+3.473=12.786 \text{ hr-ft}^2\text{-F/Btu}$$

The sum of the heat transfers in this case is (from CathedralWorksheet.xlsx).

$$Q_A = \frac{W_A * \Delta T}{R_A} = \frac{0.797*100}{5.550} = 14.36$$

$$Q_B = \frac{W_B * \Delta T}{R_B} = \frac{1.087*100}{12.786} = 8.50$$

$$Q_{framingpath} = Q_A + Q_B = 22.86 \text{ Btu/hr-ft}$$

### 3.2.3.2 1-D model, Rtot = 60, Y = 4-inches

**Input to Roof Edge Algorithm**

Rtot = 60; other inputs the same as in Rtot = 30 case above.

**Output of Roof Edge Algorithm**

$W_A = 0.797$ ft

$W_B = 3.972$ ft

$H_A = 0.466$ ft

$H_B = 1.261$ ft

$H_{B1} = 0.292$ ft

$H_{B2} = 0.662$ ft

$H_{B3} = 0.307$ ft

**Insulation Path Results**

Similar to the Rtot = 30 case, the thermal resistance of A and B are:

$$R_A = \frac{H_A}{k_{insul}} = 14.546 \text{ hr-ft}^2\text{-F/Btu}$$

$$R_B = \frac{H_B}{k_{insul}} = 39.346 \text{ hr-ft}^2\text{-F/Btu}$$

The heat transfer rates are:

$$Q_A = \frac{W_A * \Delta T}{R_A} = \frac{0.797*100}{14.546} = 5.48$$

$$Q_B = \frac{W_B * \Delta T}{R_B} = \frac{3.972*100}{39.346} = 10.095$$

$$Q_{framingpath} = Q_A + Q_B = 15.57 \text{ Btu/hr-ft}$$

**Framing path results**

The thermal resistance of A, per foot of roof edge perpendicular to the section:

$$R_A = \frac{H_A}{k_{wood}} = 5.55 \text{ hr-ft}^2\text{-F/Btu}$$

The thermal resistance of path B is the sum of layer resistances. k = 0.084 hr-ft-F/Btu is assumed.

$$R_B = \frac{H_{B1}}{k_{wood}} + \frac{H_{B2}}{k_{insul}} + \frac{H_{B3}}{k_{wood}} = \text{hr-ft}^2\text{-F/Btu}$$

$$= 3.472 + 20.66 + 3.66 = 27.79$$

The sum of the heat transfers in this case is:

$$Q_A = \frac{W_A * \Delta T}{R_A} = \frac{0.797*100}{5.55} = 14.36$$

$$Q_B = \frac{W_B * \Delta T}{R_B} = \frac{3.972*100}{27.79} = 14.29$$

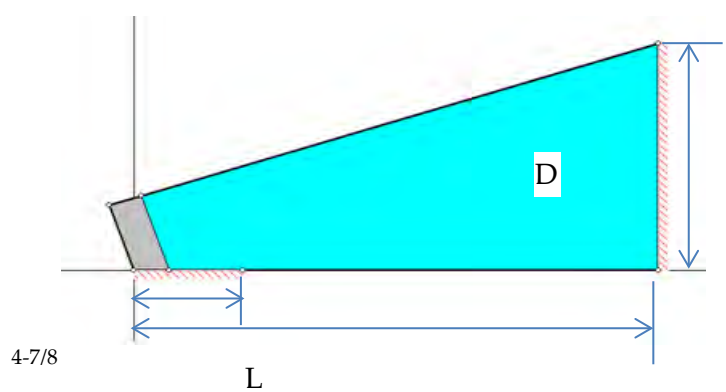
$$Q_{framingpath} = Q_A + Q_B = 28.65 \text{ Btu/hr-ft}$$

### 3.2.3.3 2-D Model, Rtot = 30, Y = 4-inches

***Insulation Path***

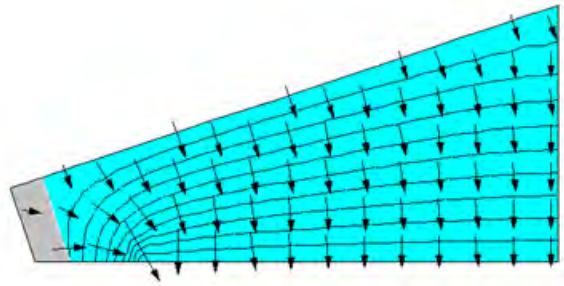
The Figure 26 case is modeled with the simplified geometry of Figure 34, shown for a ceiling insulation of R30. The top of the plate capping the vertical wall is assumed to be adiabatic. The tilted block assumed to be wood, exposed to ambient conditions on its outside sides. The outside of the wood and insulation assumed to be at a uniform 100F. The ceiling side of the insulation is set to at 0 F. The same material properties were used as in the 1-D model.

**Figure 34: Standard Truss, Insulation Path, 2-Dimensional Heat Transfer Model Geometry**




The resulting isotherms and heat transfer vectors are shown in Figure 35.

**Figure 35: Standard-Heel, Insulation Path, 2-Dimensional Heat Transfer Isotherms and Heat Transfer Vectors**



The overall heat transfer, per foot of perimeter, for this case was determined (RUN std30.FET) to be:

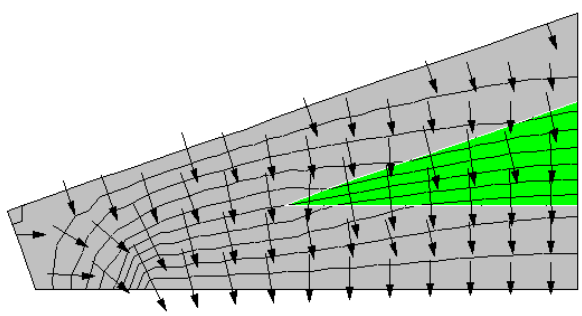
$$Q = 9.67 \text{ Btu/hr-ft}$$

*Equation 229*

### *Framing Path*

The frame path was modeled similarly, with the Figure 36 graphic results.

**Figure 36. Standard-Heel, Frame Path, 2-Dimensional Heat Transfer Isotherms and Heat Transfer Vectors**



The overall heat transfer, per foot of perimeter, for this case was determined (RUN std30F.FET) to be:

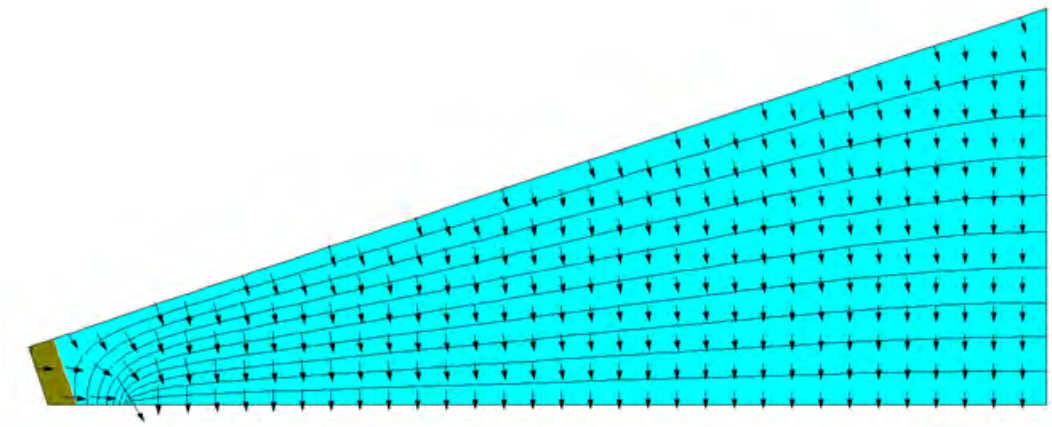
$$Q = 21.94 \text{ Btu/hr-ft}$$

*Equation 230*

### 3.2.3.4 2-D Model, Rtot = 60, Y = 4-inches

Similar to the R30 case above, Figure 37 and Figure 38 show the insulation and framing path 2-D results.

**Figure 37: 2-D Results for Insulation Path of R-60 Standard-Heel**



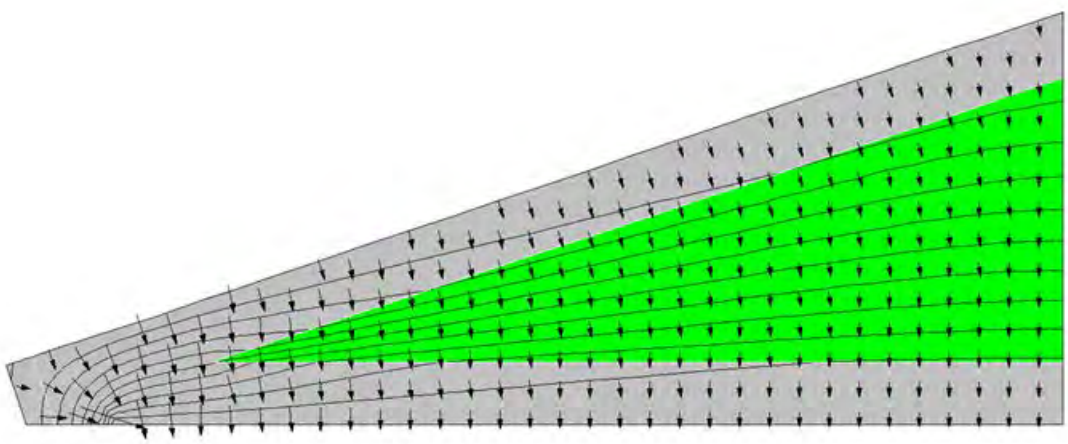
.

The overall heat transfer, per foot of perimeter, for this case was determined (RUN: std60.fet) to be:

$$Q = 16.342 \text{ Btu/hr-ft}$$

*Equation 231*

**Figure 38: 2-D Results for Framing Path of R-60 Standard-Heel**



The overall heat transfer, per foot of perimeter, for this case was determined (RUN: std60F.FET) to be:

$$Q = 31.543 \text{ Btu/hr-ft}$$

*Equation 232*

#### 3.2.3.5 Comparison of 1-D and 2-D results

The above 2-D results are compared with the simplified 1-D model in Table 8.

**Table 8: Comparison of 1-D and 2-D Results**

| | **R30 ceiling insulation** | | **R60 ceiling insulation** | |
|---|---|---|---|---|
| | insulation path | framing path | insulation path | framing path |
| $Q_{1D}$ | 9.94 | 22.86 | 15.57 | 28.65 |
| $Q_{2D}$ | 9.67 | 21.94 | 16.34 | 31.543 |
| $Q_{1D}$ is | $3\% > Q_{2d}$ | $4\% > Q_{2d}$ | $5\% < Q_{2d}$ | $9\% < Q_{2d}$ |

Because the 2-D model used is itself of limited accuracy considering the numerous approximations made, the above results are not considered to be definitive. However they do indicate that a number of the assumptions made in the 1-D model are reasonably accurate. While this comparison is limited to steady state heat transfer performance, mass effects are expected to have comparable accuracy.

## 3.3 How to Build an Airnet

### *3.3.1 Background*

### *3.3.2 Approach*

IZXFER is the building block input for an Airnet. There will be many IZXFERs in an input file, each representing a single air transfer object (leak, vent, window, fan, duct leak). Each IZXFER needs a unique name if detailed reports on its activity are needed. IZXFER is a command at the same level as HOLIDAY.

MATERIAL, CONSTRUCTION, METER, ZONE and REPORT which means that it can be located anywhere except inside another object (like a zone).

The main objects in the Airnet are:

- Infiltration
- Window ventilation
- IAQ ventilation
- Mechanical cooling ventilation

- Duct leakage

Units are $ft^2$.

CBECC inputs to add for Airnet:

| | |
|---|---|
| WinHHTop | Head height of the highest windows in the zone. Used to get the vertical location of the window ventilation holes. In the development program this was done on a building wide basis:<br><br>#define WnHeadHeight 7.67 // Average head height above the floor of operable windows |
| Ventilation Height Difference | This rule needs to be changed to refer to Zone instead of Building: "The default assumption for the proposed design is 2 feet for one story buildings and 8 feet for two or more stories. Greater height differences may be used with special ventilation features such as high, operable clerestory windows. In this case, the height difference entered by the user is the height between the average center height of the lower operable windows and the average center height of the upper operable windows. Such features shall be fully documented on the building plans and noted in the Special Features Inspection Checklist of the CF-1R." (2008 RACM pp 3-9) |
| Floor Height | The height of each floor over outdoors, crawl or garage is needed to set the Z dimension of the hole in the floor. |
| Soffit height | The height of the attic floor. Probably can be determined by the height of the ceiling below attic. Trouble for Split level? |
| Interzone Door | May need input for whether an interzone door exists between each 2 conditioned zones. Assuming it for now. |
| ReturnRegister | The conditioned zone(s) where the return/exhaust register is located. Make this an input on the HVAC System Data screen |

#### 3.3.2.1 Problems

- The window scheme doesn't work for 3 story zones!!!
- The Econ and NightBreeze cooling ventilation systems are multizone and use ducts. I suggest we set them up as part of the duct system.

### 3.3.3 Inputs

| | |
|---|---|
| ACH50 | 7.6 (Air Changes per Hour at 50 Pascals pressure difference that leak through the envelope of the conditioned zones) |
| Avent | 1/300 (ratio of "free area of attic vents to AceilGross) |
| Fraction High | 0.3 (fraction of the attic vent area located in the upper part of the attic, check precise definition) |

1. *Infiltration Setup.* Infiltration is uncontrolled air leakage through the cracks and intentional vents in the building. The first step is to determine the total size of the openings and then distribute them over the conditioned zones in proportion to surface areas.

   It is modeled in a single conditioned zones with 8 holes (IZXFERs) to represent the leakage in vertical walls and 1 hole each in the floor and ceiling.

   Calculate:

   a. For conditioned zones the total Effective Leakage Area ELAtot = CFA*ACH50/(2*10000) (CFA is conditioned floor area)
   b. Determine Envelope Areas
      1. ExCeiltotSF =sum (AceilGross + area of exterior ceilings) (exterior ceilings are surfaces in conditioned zones of type ceiling whose outside condition is Ambient, Ignore Knee walls for infiltration (walls between the conditioned zone and the attic)
      2. ExWalltotSF =sum (Gross Area of Exterior Walls) (walls in conditioned zones whose outside condition is Ambient)
      3. ExFloortotSF =sum (Gross Area of Exterior floors) (floors in conditioned zones whose outside condition is Ambient, Crawl or GROUND)
      4. ExFloorSlabSF =sum (Gross Area of Exterior slab on grade floors) (slab on grade floors in conditioned zones)
      5. SlabRatio =ExFloorSlabSF/ExFloortotSF
      6. GaragetotSF =sum (Gross Area of Surfaces to Garage) (walls and floors in conditioned zones whose outside condition is Garage)
   c. Determine leakage distribution:
      1. ELAceilsf = ELAtot* (.4+.1*SlabRatio)/( AceilGross + area of exterior ceilings)
      2. ELAraisedFloorsf = ELAtot* (.2* 1-SlabRatio)/( ExFloortotSF-ExFloorSlabSF)

   #If there is a garage zone

      3. ELAGaragesf = ELAtot* 0.1/GaragetotSF

4. ELAwallsf = ELAtot* (.3 + .1*SlabRatio)/ExWalltotSF

#Else

3. ELAGaragesf = 0

4. ELAwallsf = ELAtot* (.4 + .1*SlabRatio)/ExWalltotSF

#endif

2. *Cooling Ventilation Setup:* Four types: Windows only (all types have windows for some part of the year), Whole house fan, Smart Vent, NightBreeze

Set up seasonal window control

#if Smart Vent or NightBreeze //Windows are on in Winter, but off in summer when mechanical ventilation is on

#redefine Windowmode select( @weather.taDbAvg07 >60., 0.00001,default 1.)

#define VentDiffMult select( @top.tDbOSh < (@znRes[Single].prior.S.tAir-VentDiff), 1,default 0.000001) //Vent off if Tin-Vendiff > Tout

#Else //everything but Econ and NightBreeeze Windows are on year round

#reDefine Windowmode 1. //Always available

#Define VentDiff 0 // Differential. No differential for windows or WWF

// multiplier for window and whole house fan vent availability, .00001 is proxy for Off Revised to start at dawn end at 11 PM.

#redefine Win_hr select( $hour < 24, select($radDiff <1., select($hour>12,1.0, default .00001 ), default 1.0 ), default .00001 )

3. *Airnet for Each Conditioned Zone:*
   a. Calculate

ELA_Aceil(zone) = ELAceilsf * AceilGross(zone)

ELA_Xceil(zone) = ELAceilsf * (AEdge(zone) + area of exterior ceilings(zone))
//AEdge is determined in the Ceiling Surface setup BAW 120517

ELAXwall(zone) = ELAwallsf * Gross Area of Exterior Walls(zone)

ELAGwall(zone) = ELAGaragesf * Gross Area of walls and floors next to the Garage(zone)

ELAfloor(zone) = ELAraisedFloorsf * AreaExtfloor(zone) (gross area of floors whose outside condition is Ambient, Crawl)

ZoneBotZ = Bottom(zone) - height of the lowest floor in the zone

ZoneTopZ = ZoneBotZ + FloortoFloor(zone)*NumofStories(zone)

ZoneHeightZ = ZoneTopZ - ZoneBotZ

WinHHTop = ZoneBotZ + FloortoFloor(zone)*(NumofStories(zone)-1) + Window head height

b. Exteror wall of conditioned zones infiltration objects Calculate height of bottom and top holes.

// All infiltration leaks in walls are assumed to be spread uniformly over the exposed wall surfaces areas. There are no LEAKS associated with windows, doors etc.

//8 Wall Holes in each zone to Outdoors 1 upwind, 2 side walls, 1 downwind. Sidewalls are identical so combine them into 1 hole with 2*area

// Low is at 1/4 of wall height, high is at 3/4 of wall height

// izCpr (default = 0) = Wind Coef Upwind wall +0.6Side walls -0.65 Downwind Wall -0.3

ELAXwall(zone) = ELAwallsf * Gross Area of Exterior Walls(zone)

WH = ELAXwall(zone)*1.45/8 //Wall Hole size. Conversion from ELA to airnet infiltration opening is 1.45*

Write Airnet Objects to CSE Input WlLU stands for Wall Low Upwind etc.

IZXFER (ZoneName)WlLU izNVTYPE = AirNetExt izZN1 = (ZoneName) izALo = WH izHD = ZoneBotZ + (0.25 * ZoneHeightZ) izNVEff = 1 izExp=0.65 izCpr=0.6

IZXFER (ZoneName)WlLS izNVTYPE = AirNetExt izZN1 = (ZoneName) izALo = 2*WH izHD = ZoneBotZ + (0.25 * ZoneHeightZ) izNVEff = 1 izExp=0.65 izCpr=-.65

IZXFER (ZoneName)WlLD izNVTYPE = AirNetExt izZN1 = (ZoneName) izALo = WH izHD = ZoneBotZ + (0.25 * ZoneHeightZ) izNVEff = 1 izExp=0.65 izCpr=-0.3

IZXFER (ZoneName)WlHU izNVTYPE = AirNetExt izZN1 = (ZoneName) izALo = WH izHD = ZoneBotZ + (0.75 * ZoneHeightZ) izNVEff = 1 izExp=0.65 izCpr=0.6

IZXFER (ZoneName)WlHS izNVTYPE = AirNetExt izZN1 = (ZoneName) izALo = 2*WH izHD = ZoneBotZ + (0.75 * ZoneHeightZ)izNVEff = 1 izExp=0.65 izCpr=-.65

IZXFER (ZoneName)WlHD izNVTYPE = AirNetExt izZN1 = (ZoneName) izALo = WH izHD = ZoneBotZ + (0.75 * ZoneHeightZ) izNVEff = 1 izExp=0.65 izCpr=-0.3

c. Windows

// Operable window openings for ventilation. Assumes effect of screens is included in open area Revised 120409 BAW

// IZXFER izALo and izAHi are the min and max vent areas. Both are hourly.

//8 Window Holes in zone Single to Outdoors Assumes no orientation so 1/4 each orientation, 1/8 low and 1/8 high. Sidewalls are identical so combine them into 1 hole with 2*area

// high is at 1/2 default Hdiff below Window WinHHTop, Low is at WinHdiff below.high

//Note that this scheme doesn't work for 3 story zones!!!

Inputs

| | |
|---|---|
| WnVentArea | // ft2, Nonzero - operable window open area. Default is 10 percent of the window area. Assume a single window is 4 feet high with openings centered at -1 and -3' from the top |
| WnVentHDiff 2.0 | // Window vent height difference between center of high opening and low opening |
| WinHHTop | // Head height of highest windows in the zone |

Calculate

WnHole = 0.5*(WnVentArea/8.)*Win_hr*Windowmode // 1/8th in each hole, ft2. 1/2 of nominal area to account for screens etc. Hourly and seasonal availability

Write Airnet Objects to CSE Input WnLU stands for Window Low Upwind etc

IZXFER (ZoneName)WnLU izNVTYPE = AirNetExt izZN1 = (ZoneName) izALo=.00001 izAHi = WnHole izHD = WinHHTop - (3+ WnVentHDiff) izNVEff =.5 izCpr=0.6

IZXFER (ZoneName)WnLS izNVTYPE = AirNetExt izZN1 = (ZoneName) izALo=.00001 izAHi = 2*WnHole izHD = WinHHTop - (3+WnVentHDiff) izNVEff =.5 izCpr=-.65

IZXFER (ZoneName)WnLD izNVTYPE = AirNetExt izZN1 = (ZoneName) izALo=.00001 izAHi = WnHole izHD = WinHHTop - (3+WnVentHDiff) izNVEff =.5 izCpr=-0.3

IZXFER (ZoneName)WnHU izNVTYPE = AirNetExt izZN1 = (ZoneName) izALo=.00001 izAHi = WnHole izHD = (WinHHTop-1) izNVEff =.5 izCpr=0.6

IZXFER (ZoneName)WnHS izNVTYPE = AirNetExt izZN1 = (ZoneName) izALo=.00001 izAHi = 2*WnHole izHD = (WinHHTop-1) izNVEff =.5 izCpr=-.65

IZXFER (ZoneName)WnHD izNVTYPE = AirNetExt izZN1 = (ZoneName) izALo=.00001 izAHi = WnHole izHD = (WinHHTop-1) izNVEff =.5 izCpr=-0.3

d. Ceiling

Calculate

If ceiling below attic

ELA_Aceil(zone) = ELAceilsf * AceilGross(zone) //knee walls to attic not included in determining the conditioned to attic leakage distribution of

If Whole House fan, Ceiling leak through WWF when off

If Improved WHFela = .05 // Average of Motor Damper Models

else WHFela = .11 // Average of Gravity Damper Models

If ceiling to outside

ELA_Xceil(zone) = ELAceilsf * AEdge(zone) + area of exterior ceilings(zone)

CeilHole = (ELA_Aceil(zone) + WHFela )*1.45 //Ceil Hole size. Conversion from ELA to airnet infiltration opening is 1.45*

CathCeilHole = ELA_Xceil(zone) * 1.45 //Cathedral Ceil Hole size. Conversion from ELA to airnet infiltration opening is 1.45*

Write Airnet Objects to CSE Input

IZXFER (ZoneName)xAttic izNVTYPE = AirNetIZ izZN1=(ZoneName) izALo=CeilHole izHD = ZoneTopZ izNVEff=1. izExp=0.65 izZN2 = Attic

IZXFER (ZoneName)CC izNVTYPE = AirNetExt izZN1=(ZoneName) izALo=CathCeilHole izHD = ZoneTopZ izNVEff=1. izExp=0.65

e. Floor over outside

For each floor over outside calculate:

ELAfloor(Name) = ELAraisedFloorsf * AreaExtfloor * 1.45 (floors whose outside condition is Ambiant)

Write Airnet Object to CSE Input

IZXFER (Name) izNVTYPE = AirNetExt izZN1 = (ZoneName) izALo = ELAfloor(Name) izHD = Floor Height izNVEff = 1 izExp=0.65 izCpr=0. // located at the Extfloor elevation, no wind effect

f. Floor over Crawl

For each floor over outside calculate:

ELAfloor(Name) = ELAraisedFloorsf * AreaCrawlfloor * 1.45 (floors whose outside condition is Crawl)

Write Airnet Object to CSE Input

IZXFER (Name)xCrawl izNVTYPE = AirNetIZ izZN1=(ZoneName) izALo=ELAfloor(Name) izHD = Floor Height izNVEff=1. izExp=0.65 izZN2 = Crawl

g. Floor over Garage

For each floor over garage calculate:

ELAfloor(Name) = ELAGaragesf * AreaGarfloor * 1.45 (floors whose outside condition is Garage)

Write Airnet Object to CSE Input

IZXFER (Name)xGarage izNVTYPE = AirNetIZ izZN1=(ZoneName) izALo=ELAfloor(Name) izHD = Floor Height izNVEff=1. izExp=0.65 izZN2 = Garage

h. Garage wall

Calculate:

GWH = ELAGaragesf * Gross Area of walls next to the Garage(zone)/2 // size of the 2 holes (high and low) betwen zone and garage

GWalltopZ = Min(ZoneTopZ(zone),(ZoneTopZ(Garage)) //The top of the shared wall

GWallBotZ = Max(ZoneBotZ(zone),(ZoneBotZ(Garage)) //The bottom of the shared wall

GwallH = GWalltopZ - GWallBotZ // Height of shared wall

GWHhZ = GwallBotZ + .75 GwallH // Height of top hole

GWHlZ = GwallBotZ + .25 GwallH // Height of bottom hole

Write Airnet Objects to CSE Input

IZXFER (ZoneName)xGarageH izNVTYPE = AirNetIZ izZN1=(ZoneName) izALo=GWH izHD = GWHhZ izNVEff=1. izExp=0.65 izZN2 = Garage

IZXFER (ZoneName)xGarageL izNVTYPE = AirNetIZ izZN1=(ZoneName) izALo=GWH izHD = GWHlZ izNVEff=1. izExp=0.65 izZN2 = Garage

4. *Airnet for Each Unconditioned Zone:*

a. Attic

If Ventilated attic Calculate (4 soffit vents at attic floor elevation plus sloped deck vents at 2/3 of Attic high if frac high > 0

// Pitch types for roof wind pressure coeffs: 0 deg, <10deg, <15 deg,<25, <35 ,all the rest. Flat same as low slope.

#define PitchType select( Pitch <= 0, 1,Pitch <= 0.18, 1,Pitch <= 0.27, 2,Pitch <= 0.47, 3,Pitch <= 0.7, 4,default 5)

AventTot = AceilGross * AVent

SoffitVent 0.5*0.25 *(1.-FracHigh)*Max(AventTot, AtticRelief) //Attic relief is minimum vent needed to vent mechanical cooling air dumped to attic

Deckvent 0.5*0.25*FracHigh*Max(AventTot, AtticRelief)

If sealed attic [to be developed]

Write Airnet Objects to CSE Input

IZXFER AtticSU izNVTYPE = AirNetExt izZN1 = Attic izALo = SoffitVent izHD = SoffitHeight izNVEff = .6 izExp=0.65 izCpr=0.6

IZXFER AtticSS izNVTYPE = AirNetExt izZN1 = Attic izALo = 2*SoffitVent izHD = SoffitHeight izNVEff = .6 izExp=0.65 izCpr=-.65

IZXFER AtticSD izNVTYPE = AirNetExt izZN1 = Attic izALo = SoffitVent izHD = SoffitHeight izNVEff = .6 izExp=0.65 izCpr=-0.3

IZXFER AtticDU izNVTYPE = AirNetExt izZN1 = Attic izALo = DeckVent izHD = 0.67 * AtticHeight + SoffitHeight izNVEff=.6 izExp=0.5 izCpr=testx*choose(Pitchtype,-.5,-.8,-.5,-.3,.1,.3)

IZXFER AtticDS izNVTYPE = AirNetExt izZN1 = Attic izALo = 2*DeckVent izHD = 0.67 * AtticHeight + SoffitHeight izNVEff=.6 izExp=0.5 izCpr=testx*choose(Pitchtype,-.5,-.5,-.5,-.5,-.5,-.5)

IZXFER AtticDD izNVTYPE = AirNetExt izZN1 = Attic izALo = DeckVent izHD = 0.67 * AtticHeight + SoffitHeight izNVEff=.6 izExp=0.5 izCpr=testx*choose(Pitchtype,-.5,-.3,-.5,-.5,-.5,-.5)

b. Garage – Assume California garage with a water heater and combustion air vents so it is pretty leaky Guess 1 ft2 of free area. Ignore other infiltration

Calculate

Gvent = 1/4

Write Airnet Objects to CSE Input

IZXFER GarageU izNVTYPE = AirNetExt izZN1 = Garage izALo = Gvent izHD = GarageBotZ +1 izNVEff = .6 izExp=0.65 izCpr=0.6

IZXFER GarageS izNVTYPE = AirNetExt izZN1 = Garage izALo = 2*Gvent izHD = GarageBotZ +1 izNVEff = .6 izExp=0.65 izCpr=-.65

IZXFER GarageD izNVTYPE = AirNetExt izZN1 = Garage izALo = Gvent izHD = GarageBotZ +1 izNVEff = .6 izExp=0.65 izCpr=-0.3

c. Vented crawl space [To Be Developed]
d. Sealed crawl space [To Be Developed]
e. Basement [To Be Developed]

5. *Interzone Holes* – Assume an open door or stair between any twp conditioned zones with common surfaces, except between units in multi-family

If 2 or more conditioned zones

Error if not at least one common surface for every conditioned zone (a surface in zone A whose outside condition is another conditoned zone)

Door calculation for each pair of zones with a common wall surface (zoneA<>zoneB, zoneB<>zoneC, zoneA<>zoneC, etc)

DoortopZ = Min(ZoneTopZ(zone A),(ZoneTopZ(zone B)) //The top of the shared wall

DoorBotZ = Max(ZoneBotZ(zone A),(ZoneBotZ(zone B)) //The bottom of the shared wall

DoorH = DoortopZ - DoorBotZ // Height of shared opening

DH = 20/ // Area of half of assumed door

DHhZ = GwallBotZ + .75 GwallH // Height of top hole

DHHlZ = GwallBotZ + .25 GwallH // Height of bottom hole

For each zone pair write Airnet Objects to CSE Input

IZXFER (ZoneNameA)DHx(ZoneNameB)DH izNVTYPE = AirNetIZ izZN1=(ZoneNameA) izALo=DH izHD = DHhZ izNVEff=1. izExp=0.5 izZN2 = (ZoneNameB)

IZXFER (ZoneNameA)DLx(ZoneNameB)DL izNVTYPE = AirNetIZ izZN1=(ZoneNameA) izALo=DH izHD = DHlZ izNVEff=1. izExp=0.5 izZN2 = (ZoneNameB)

Stair calculation for each pair of zones with only a floor/ceiling surface (zoneA<>zoneB, zoneB<>zoneC, zoneA<>zoneC, etc)

StairZ = Max(ZoneBotZ(zone A),(ZoneBotZ(zone B)) //The height of the stair hole is at the upper floor

For each zone pair write Airnet Objects to CSE Input Note that izZN1 MUST be the lower of the 2 zones or the model doesn't work

IZXFER (ZoneNameA)Sx(ZoneNameB)S izNVType = AIRNETHORIZ izZN1=(ZoneName of lowerzone) izZN2 = (ZoneName of upper zone) izL1=3 izL1=10 izHD =StairZ

6. *IAQ ventilation*

Inputs for each zone

IAQVentCFM // CFM of IAQ vent

IAQfanWperCFM // W/CFM of IAQ vent

Type IAQExhaust // "IAQExhaust", "IAQSupply", "IAQBalanced" "NoIAQVent"

IAQVentHtRcv 0.0 // Heat recovery efficiency of Balanced type, frac

Write Airnet Objects to CSE Input

If Exhaust

IZXFER (Zone)IAQfan izNVTYPE = AirNetExtFan izZN1 = (Zone) izVFmin=-IAQVentCFM izVFmax=-IAQVentCFM izFanVfDs=IAQVentCFM izFanElecPwr=IAQfanWperCFM izFanMtr=IAQventMtr

If IAQSupply

IZXFER (Zone)IAQfan izNVTYPE = AirNetExtFan izZN1 = (Zone) izVFmin=IAQVentCFM izVFmax=IAQVentCFM izFanVfDs=IAQVentCFM izFanElecPwr=IAQfanWperCFM izFanMtr=IAQventMtr

If IAQBalanced // Needs heat recovery

IZXFER (Zone)IAQfanS izNVTYPE = AirNetExtFan izZN1 = (Zone) izVFmin=IAQVentCFM izVFmax=IAQVentCFM izFanVfDs=IAQVentCFM izFanElecPwr=IAQfanWperCFM izFanMtr=IAQventMtr

IZXFER (Zone)IAQfanE izNVTYPE = AirNetExtFan izZN1 = (Zone) izVFmin=-IAQVentCFM izVFmax=-IAQVentCFM izFanVfDs=IAQVentCFM izFanElecPwr=IAQfanWperCFM izFanMtr=IAQventMtr

*7. Mechanical Cooling Ventilation* // The following does not work for multi-zone systems with Econ, NightBreeze. Revise along with ducts model

For each Cooling Ventilation System

Inputs

CoolVentType //type of MECHANICAL cooling ventilation, Choice of WHF, Econ, NightBreeze

CoolVentCFM //Rated air flow of the mechanical cooling system

CoolVent W/CFM //

ReturnRegister // If WHF the conditoned zone where it is located

Calculate

Relief = CoolVentCFM/375 // The minimum size of the attic vents required to let the WHF flow out of the attic

If WHF // Whole House Fan

Calculate

Relief = CoolVentCFM/375 // The minimum size of the attic vent required for this fan to let the WHF flow out of the attic

Write Airnet Objects to CSE Input

IZXFER (Zone)WHF izNVTYPE=AirNetIZFan izZN1=(Zone) izVFmin=0. izVFMax=-CoolVentCFM*Win_hr izFanVfDs=CoolVentCFM izZn2=Attic izFanElecPwr=CoolVentWperCFM izFanMtr=CoolVentMtr

If Econ // Economizer ventilation option on the Central Forced Air System such as Smartvent

Calculate

Relief = CoolVentCFM/375 // The minimum size of the attic vent required for this fan to let the WHF flow out of the attic

Write Airnet Objects to CSE Input

ZXFER Econ# izNVTYPE=AirNetExtFan izZN1=(zone) izVFmin=0. izVFMax=CoolVentCFM*Coolmode*VentDiffMult izFanVfDs=CoolVentCFM izFanElecPwr=CoolVentWperCFM izFanMtr=CoolVentMtr //!! 110413

IZXFER Relief# izNVTYPE=AirNetIZFlow izZN1=ReturnRegister izZn2=Attic izVFmin=0 izVFmax=-CoolVentCFM*Coolmode*.9*VentDiffMult

If NightBreeze //Model for NightBreeze variable flow night ventilation system !!needs lower limit @ CFA<1000/unit and multiple systems @ CFA> 3333 ft2

Calculate

Relief = CoolVentCFM/375 // The minimum size of the attic vent required for this fan to let the WHF flow out of the attic

Write Airnet Objects to CSE Input

IZXFER NightBreeze izNVTYPE=AirNetExtFan izZN1=(zone) izVFmin=0. izFanMtr=CoolVentMtr

izFanVfDs=CoolVentCFM * CFA //CoolVentCFM = CFM/CFA for NightBreeze. Default is 0.6

izFanElecPwr = (616.47-0.6159*CFA + .000246 *CFA*CFA)/(CoolVentCFM * CFA) //W/CFM DEG 9/29/2010 Equation 1

izVFMax=CoolVentCFM*Coolmode*VentDiffMult*CFA / max((17.91554 - 3.67538 * logE(@weather.taDbPvPk)),1) //DEG 9/29/2010 Equation 3 110411

izFanCurvePy = 0, -0.026937155, 0.187108922, 0.839620406, 0 //Fit to DEG flow^2.85

IZXFER NBRelief izNVTYPE=AirNetIZFlow izZN1=ReturnRegister izZn2=Attic izVFmin=0 izVFmax=-CoolVentCFM * CFA * Coolmode*.9 *VentDiffMult

Next Zone

Calculate

AtticRelief = Sum(CoolVentCFM)/375 // The sum of all zonal cool vent CFM determines the minimum size of the attic vents required to let the vent air out of the attic

//Used in Attic Zone AirNet above

//Min Attic Vent area for relief Tamarac http://www.tamtech.com/userfiles/Fan%20size%20and%20venting%20requirements(3).pdf

7. *Duct system leaks and pressurization.* These are generated by automatically by CSE based on the duct system inputs.

## 3.4 How to Create CSE Conditioned Zone Internal Mass Inputs

### 3.4.1 Background

### 3.4.2 Approach

Internal mass objects are completely inside a zone so that they do not participate directly in heat flows to other zones or outside. They are conected to the zone radiantly and convectively and participate in the zone energy balance by passively storing and releasing heat as conditions change. For now only in Conditioned Zones.

The main internal mass objects in the are:

- Interior walls
- Interior floors
- Furniture
- Cair
- Specific masses (for addition later)

CBECC inputs to add:

- Specific masses (for addition later)

### 3.4.3 Inputs

Floor Area of zone

For each Conditioned Zone

1. Interior Floor Setup. Input for inside the conditioned zone interior floors as mass elements.

   Calculate:

   a. Xflr = sum of the area of floors to ground, crawl space, exterior or other zones
   b. IntFlr = Floor Area-Xflr
2. Interior Wall Setup. Input for inside the conditioned zone interior walls as mass elements.

   Calculate:

   a. IZwall = sum of the area of interior walls to other conditioned zones
   b. Intwall(zone) floor area - .5 * IZwall
3. Write objects to the CSE input

Light stuff

1. znCAir=floor area * 2
2. Interior wall if Intwall(zone) > 0

   SURFACE IntWallC(zone) sfType=Wall sfArea=0.75*Intwall(zone) sfCon=IntwallCav; sfAZM=0 sfExCnd=ADJZN sfAdjZn=(zone)

   SURFACE IntWallF(zone) sfType=Wall sfArea=0.25*Intwall(zone) sfCon=IntwallFrm; sfAZM=0 sfExCnd=ADJZN sfAdjZn=(zone)

3. Furniture

   SURFACE Furniture(zone) sfType=wall sfArea= Floor Area * 2.; sfCon=FurnCon; sfAZM=0 sfExCnd=ADJZN sfAdjZn=Zone

4. Interior Floor if IntFlr(zone) >0

   // floor construction for interior mass. Assumes 2x10 @ 16" OC. Both floor and ceiling are in the conditioned zone

   SURFACE IntFlrFrm sfType=Floor sfCon=IntFFrm2x10 sfArea=0.1 * RaisedFlr; sfExCnd=ADJZN sfAdjZn=(Zone)

   SURFACE IntFlrCav sfType=Floor sfCon=IntFCav2x10 sfArea=0.9 * RaisedFlr; sfExCnd=ADJZN sfAdjZn=(Zone)

4. Constructions

CONSTRUCTION FurnCon // 2.5" wood Revised Layers

Layer lrMat="SoftWood" lrThk=2.5/12

CONSTRUCTION IntwallCav // 2x4 Revised Layers

Layer lrMat="Gypsum Board"

Layer lrMat="Gypsum Board"

CONSTRUCTION IntwallFrm // 2x4 Revised Layers

Layer lrMat="Gypsum Board"

Layer lrMat="SoftWood" lrThk=3.5/12.

Layer lrMat="Gypsum Board"

CONSTRUCTION IntFFrm2x10 // 9.25" (2x10)

Layer lrMat="Carpet"

Layer lrMat="Wood layer"

Layer lrMat="SoftWood" lrThk=9.25/12.

Layer lrMat="Gypsum Board"

CONSTRUCTION IntFCav2x10 // 9.25" (2x10)

Layer lrMat="Carpet"

Layer lrMat="Wood layer"

Layer lrMat="Carpet" // Air space with 1 psf of stuff (cross bracing wiring, plumbing etc) approximated as 1" of carpet

Layer lrMat="Carpet" // Air space with 1 psf of stuff (cross bracing wiring, plumbing etc) approximated as 1" of carpet

Layer lrMat="Gypsum Board"

# 3.5 Appliances, Miscellaneous Energy Use and Internal Gains

## 3.5.1 Background

This model is derived from the 2008 HTM (California Energy Commission, HERS Technical Manual, California Energy Commission, High Performance Buildings and Standards Development Office. CEC-400-2008-012). This is a major change from the 2008 RACM in that internal gains are built up from models for refrigerator, people, equipment and lights instead of the simple constant plus fixed BTU/ft$^2$ used there. The HTM derived model has been used in the 2013 Development Software throughout the 2013 revision process.

This model has another significant change beyond the HTM model with the addition of latent gains required as input for the new CSE air conditioning model. There was no information on latent gains in either the 2008 RACM or the HTM. The latent model here was created by applying the best available information on the latent fraction of internal gains to the HTM gains model.

## 3.5.2 Approach

The approach here is to calculate the Appliances and Miscellaneous Energy Use (AMEU) for the home and use that as the basis for the internal gains. This will facilitate future expansion of the procedure to calculate a HERS Rating.

### 3.5.2.1 Problems

The procedure here (also used in the 2013 development program) does not work correctly for multifamily buildings unless all of the units are the same (CFA and number of bedrooms). I don't believe this problem was considered in developing the HTM. I believe that the only exactly correct solution involves simulating each unit as a separate zone with a different internal gain. For now we will ignore this problem and assume that average values are OK.

The HTM equations do not work if there is a gas range and electric oven.

The allocation of internal gain to zones is not specified in either the RACM or the HTM. A proposed approach is presented here.

## 3.5.3 Inputs

| | |
|---|---|
| Units | Number of dwelling units in the building. |
| BRperUnit | Bedrooms/DwellingUnits rounded to an integer |
| CFA | Conditioned Floor Area in the building |
| CFAperUnit | CFA/DwellingUnits |

New CBECC input at the building level: an Appliances Input Screen (for a single conditioned zone, most of these default, we are assuming that MF buildings will be done as one zone):

| | |
|---|---|
| Refrigerator/Freezer | Efficiency (Choice of Default = 669 kWh/year, no other choices at this time), Location (Choice of zones if multiple conditioned zones). // HTM assumes all Dwelling units have refrigerators. Different for additons and alterations when we get to them. |
| Dishwasher | Efficiency (Choice of Default, no other choices at this time), Location (Choice of zones if multiple conditioned zones). // HTM assumes all Dwelling units have refrigerators. Different for additons and alterations when we get to them. |
| Clothes Dryer | Location (Choice of zones if multiple zones, No Dryer space or hookup provided) Dryer power (Choice Electric, Gas or other) //Assuming gas for now |
| Clothes Washer | Location (Choice of zones if multiple zones), No Washer space or hookup provided) |
| Range/Oven | Location (Choice of zones if multiple conditioned zones, No Range/Oven space and hookup provided) Range/Oven power (Choice Electric, Gas or other) Assumes gas for now. |

Assumes CSE Meters are set up elsewhere:

Mtr_Elec

Mtr_NatGas

Mtr_Othewr //PropaNE

Write Constants to the CSE input:

#redefine Intgain_mo choose1($month, 1.19,1.11,1.02,0.93,0.84,0.8,0.82,0.88,0.98,1.07,1.16,1.21) //The monthly internal gain multiplier (same as 2008 RACM).

#redefine Lights_hr hourval(0.023,0.019,0.015,0.017,0.021,0.031,0.042,0.041,0.034,0.029,0.027,0.025,\

0.021,0.021,0.021,0.026,0.031,0.044,0.084,0.118,0.113,0.096,0.063,0.038) // Changed 0.117 to 0.118 to add to 1

#redefine OutdoorLights_hr hourval(0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0.25,0.25,0.25,0.25) //

#redefine People_hr
hourval(0.035,0.035,0.035,0.035,0.035,0.059,0.082,0.055,0.027,0.014,0.014,0.014, \

0.014,0.014,0.019,0.027,0.041,0.055,0.068,0.082,0.082,0.070,0.053,0.035)

#redefine Equipment_hr
hourval(0.037,0.035,0.034,0.034,0.032,0.036,0.042,0.044,0.037,0.032,0.033,0.033, \

0.032,0.033,0.035,0.037,0.044,0.053,0.058,0.060,0.062,0.060,0.052,0.045)

1. Setup the gains that are distributed across the zones per CFA of the zone and write to CSE input: Calculations are generally more complicated in future for HERS
   a. //Lights Returns Btu/day-CFA - based on ElectricityInteriorLights = (214+ 0.601×CFA)×(FractPortable + (1-FractPortable)×PAMInterior ) //HTM Eqn 11, p. 30

      #define FractPortable .22 //fixed for now, variable later for HERS

      #define Paminterior 0.625 //fixed for now, variable later for HERS

      #Redefine LightsGainperCFA (((214. + 0.601 * CFAperUnit) * (FractPortable + (1-FractPortable) * Paminterior ) * 3413. / 365) * DwellingUnits /CFA)

   b. People Returns BTU/day-CFA - 100% is internal gain 57.3% sensible, 42.7% latent Based on HTM and BA existing bldgs Sensible 220, Latent 164 BTU

      #redefine PeopleperUnit (1.75 + 0.4 * BRperUnit)

      #Redefine PeopleGainperCFA ((3900/0.573) * PeopleperUnit * DwellingUnits / CFA)

   c. Misc Electricity Returns BTU/day-CFA - 100% is internal gain

      #Redefine MiscGainperCFA ((723. + (0.706 * CFAperUnit))* DwellingUnits * 3413. / 365.)/CFA

2. Setup the gains that are point sources located in a particular zone and write to CSE input. Calculations are generally more complicated in future for HERS
   a. Refrigerator. In the HTM all Standard Design refrigerators use the same amount of electricity (669 kWh/year) regardless of the size of dwelling unit or number of bedrooms. The proposed use is based on the energy label of the actual refrigerator installed or if that is not available the default. For existing home HERS calculations the default is (775 kWh/year). Refrigerators run at a constant power 24 hours per day, regardless of the interior air temperature or number of times the door is opened.

      Returns BTU/day - 100% is internal gain. Installed refrigerator rating is input for proposed in HERS later

      #Redefine RefrigeratorGain (DwellingUnits * 669. * (3413. / 365.))

b. Dishwasher. 0 based choose returns BTU/day // uses Table based in INTEGER bedrooms per dwelling.

#Redefine DishwasherGain (choose (BRperUnit,90,90,126,126,126,145,145,174,174,174,default 203) * DwellingUnits * 3413. / 365.)

c. Stove and Oven – Assumes both are gas with electonic igniter Returns BTU/day - Full Energy Use, 90% is internal Gain

define CookGain (((31. + (.008 * CFAperUnit))* 0.43* 0.9)* DwellingUnits * 100000. / 365.) //Added the 0.43 for the electronic ignition 12/4 BAW

d. Clothes Washer - // Returns BTU/day

#Redefine WasherGain ((-64 + 0.108 * CFAperUnit) * DwellingUnits * 3413. / 365.)

e. Clothes Dryer - Assumes gas with electonic igniter Returns BTU/day - Full energy Use, 30% is internal gain

define DryerGAin (13. + (.01 * CFAperUnit))* DwellingUnits * 100000. / 365. //Added the 0.43 for the electronic ignition //120831

f. Exterior Lights Returns Btu/day - based on HTM Eqn 14

#define PamExterior 0.49 //fixed for now, variable later for HERS

#Redefine ExtLightGain (-81+ 0.152 × CFA)×PAMExterior * 3413. / 365)

3. For each conditioned zone: //Write GAIN objects inside each conditioned zone

GAIN Lights(zone) gnPower= LightsGainperCFA*CFA(Zone)*Lights_hr*Intgain_mo gnFrRad=0.4 gnEndUse=Lit gnMeter= Mtr_Elec

GAIN People(zone) gnPower= PeopleGainperCFA*CFA(Zone)*People_hr*Intgain_mo gnFrRad=0.3 gnFrLat=0.427 // Free Energy so not metered

GAIN Misc(zone) gnPower= MiscGainperCFA*CFA(Zone)*Equipment_hr*Intgain_mo gnFrRad=0.3 gnFrLat=0.03 gnEndUse=Rcp gnMeter= Mtr_Elec

Write any of the following if the source is located in this zone:

GAIN Refrigerator gnPower= RefrigeratorGain/24 gnFrRad=0 gnEndUse=Refr gnMeter= Mtr_Elec // No *Intgain_mo, change fro 2013 DevProg

GAIN Dishwasher gnPower= DishwasherGain*Equipment_hr*Intgain_mo gnFrRad=0 gnFrLat=0.25 gnEndUse=Dish gnMeter= Mtr_Elec //

GAIN Cooking gnPower= CookGain*Equipment_hr*Intgain_mo gnFrRad=0 gnFrLat=0.67 gnEndUse=Cook gnMeter= Mtr_NatGas gnFrZn=.9 //

GAIN Washer gnPower= WasherGain*Equipment_hr*Intgain_mo gnFrRad=0 gnEndUse=Wash gnMeter= Mtr_Elec //

GAIN Dryer gnPower= DryerGAin*Equipment_hr*Intgain_mo gnFrRad=0 gnFrLat=0.5 gnEndUse=Dry gnMeter= Mtr_NatGas gnFrZn=.3 //

Write the following to the 1st zone only (one gain per building):

GAIN ExtLights gnPower= ExtLightGain*OutdoorLights_hr gnFrZn=.0 gnEndUse=Ext gnMeter= Mtr_Elec // outside lights, no internal gain

4. For each unconditioned zone write the following if the source is located in this zone: //Garage or Basement Maybe 2nd refrigerator in garage later?

GAIN Washer gnPower= WasherGain*Equipment_hr*Intgain_mo gnFrRad=0 gnEndUse=Wash gnMeter= Mtr_Elec //

GAIN Dryer gnPower= DryerGAin*Equipment_hr*Intgain_mo gnFrRad=0 gnFrLat=0.5 gnEndUse=Dry gnMeter= Mtr_NatGas gnFrZn=.3 //

## 3.6 Seasonal Algorithm

These are constant control rules. You could substitute values for defined terms in some cases like Winter_Vent Winter_Cool Summer_heat and Sumr_Vent_Temp

//Thermostats and associated controls

//Heat Mode

#redefine SZ_Heat_hr
hourval(65,65,65,65,65,65,65,68,68,68,68,68,68,68,68,68,68,68,68,68,68,68,68,65)

#redefine Liv_Heat_hr
hourval(65,65,65,65,65,65,65,68,68,68,68,68,68,68,68,68,68,68,68,68,68,68,68,65)

#redefine Slp_Heat_hr
hourval(65,65,65,65,65,65,65,68,65,65,65,65,65,65,65,65,65,65,65,65,68,68,65)

#redefine Winter_Vent 77

#redefine Winter_Cool 78

//Cool Mode

#redefine SZ_Cool_hr
hourval(78,78,78,78,78,78,78,83,83,83,83,83,83,82,81,80,79,78,78,78,78,78,78,78)

#redefine Liv_Cool_hr
hourval(83,83,83,83,83,83,83,83,83,83,83,83,83,82,81,80,79,78,78,78,78,78,78,83)

```
#redefine Slp_Cool_hr
hourval(78,78,78,78,78,78,78,83,83,83,83,83,83,83,83,83,83,83,83,83,78,78,78)

#redefine Summer_Heat 60

#redefine Sumr_Vent_Temp 68      //


//  Summer Winter mode switch based on 7 day average temp.  Winter<=60>Summer

#redefine Coolmode select( @weather.taDbAvg07 >60., 1,default 0)

#redefine HeatSet select( @weather.taDbAvg07 >60., Summer_Heat, default SZ_Heat_hr )

#redefine CoolSet select( @weather.taDbAvg07 >60., SZ_Cool_hr, default Winter_Cool )

#redefine Tdesired select( @weather.taDbAvg07 >60., Sumr_Vent_Temp, default Winter_Vent )

// Window interior shade closure

#define SCnight 0.8    //  when the sun is down.  80%

#define SCday 0.5      //  when the sun is up  50%

#define SCcool 0.5     //  when cooling was on previous hour.  50%?
```

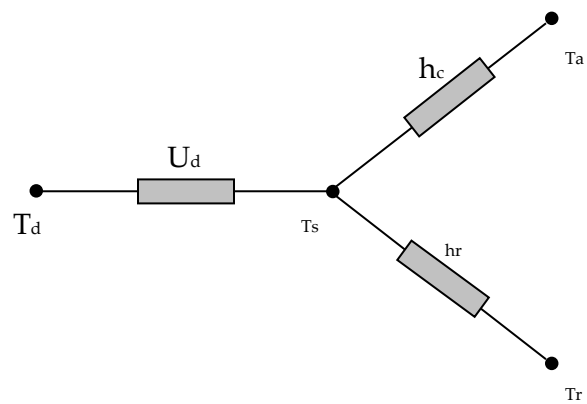
# APPENDICES

## Appendix A. Derivation of Duct Loss Equations Using Heat Exchanger Effectiveness and Y-Delta Transformations

This derivation is for one zone only, and the nomenclature is specific to this appendix alone.

Heat transfer through the duct walls can be illustrated in the electrical analogy in Figure A-1. The first node on the left represents the temperature of the air in the duct ($T_d$) and is connected to the temperate on the surface of the duct ($T_s$) by the conductance through the duct wall ($U_d$). The convective heat transfer coefficient ($h_c$) connects the surface temperature to the duct zone air temperature ($T_a$). The radiation heat transfer coefficient ($h_r$) connects the surface temperature to the duct zone radiant temperature ($T_r$).

**Figure A-1: Electrical Analogy of Heat Transfer through a Duct Wall**




The temperatures of the duct zone are assumed to be constant; the duct surface temperature is not. The duct surface temperature can be removed from the analysis by using a Y-Δ transform. Figure A-2 shows the result of this transformation with direct connections between the duct air temperature, the duct zone radiant and air temperatures through combined coefficients defined in Equation A- 1.

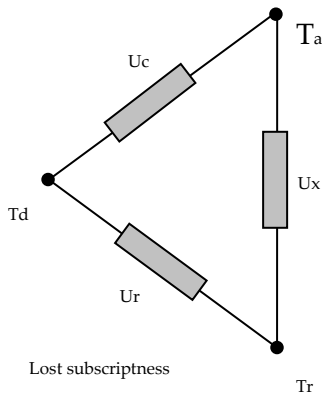
$$U_r = \frac{U_d h_r}{D} \qquad U_c = \frac{U_d h_c}{D} \qquad U_x = \frac{h_c h_r}{D}$$

*Equation A- 1*

where

$$D = U_d + h_c + h_r$$

**Figure A-2: Heat Transfer through a Duct Wall with Surface Temperature Removed**




Using an energy balance, the rate of change of heat flow along the length ( $x$ ) of duct must equal the heat flow through the duct wall, or

$$-mc_p \frac{dT_d(x)}{dx} = U_c P(T_d(x) - T_a) + U_r P(T_d(x) - T_r)$$

*Equation A- 2*

where

$mc_p$ = capacitance flow rate of the air in the duct

$T_d$ = temperature of air in the duct

$U_c$ = equivalent heat transfer coefficient (see Equation A- 1)

$P$ = perimeter of duct

$T_a$ = temperature of air in duct zone

$U_r$ = equivalent heat transfer coefficient (see Equation A- 1)

$T_r$ = radiant temperature in duct zone

Regrouping by temperature terms

$$mc_p \frac{dT_d(x)}{dx} = -(U_c P + U_r P)T_d(x) + U_c P T_a + U_r P T_r$$

*Equation A- 3*

and dividing through by the quantity $(U_c P + U_r P)$ gives

$$\frac{mc_p}{(U_c P + U_r P)} \frac{dT_d(x)}{dx} = -T_d(x) + T_{amb}$$

*Equation A- 4*

where

$$T_{amb} = \frac{U_c P}{(U_c P + U_r P)} T_a + \frac{U_r P}{(U_c P + U_r P)} T_r$$

*Equation A- 5*

Let $y(x)$ be

$$y(x) = T_{amb} - T_d(x)$$

*Equation A- 6*

The derivative of which is

$$dy = -dT_d$$

*Equation A- 7*

Substituting Equation A- 6 and Equation A- 7 into Equation A- 4 gives

$$-\frac{mc_p}{(U_c + U_r)P} \frac{dy}{dx} = y(x)$$

*Equation A- 8*

Rearranging

$$\frac{1}{y(x)} dy = -\frac{(U_c + U_r)P}{mc_p} dx$$

*Equation A- 9*

and integrating from entrance ( $x = 0$ ) to exit ( $x = L$ )

$$\int_0^L \frac{1}{y(x)} dy = \int_0^L -\frac{(U_c + U_r)P}{mc_p} dx$$

*Equation A- 10*

Gives

$$\ln y(L) - \ln y(0) = -\frac{(U_c + U_r)PL}{mc_p}$$

*Equation A- 11*

Recalling the definition in Equation A- 6 and replacing the product of the perimeter and length with the surface area ( $A$ ) of the duct, and a bit of manipulation yields the following relationships

$$\frac{y(L)}{y(0)} = \frac{T_d(L) - T_{amb}}{T_d(0) - T_{amb}} = \exp\left(-\frac{(U_c + U_r)A}{mc_p}\right)$$

*Equation A- 12*

*Equation A- 13*

Let

$$\beta = \exp\left(-\frac{(U_c + U_r)A}{mc_p}\right)$$

*Equation A- 14*

Then

$$\frac{T_d(L) - T_{amb}}{T_d(0) - T_{amb}} = \beta$$

*Equation A- 15*

Solving for the exit temperature gives

$$T_d(L) = \beta(T_d(0) - T_{amb}) + T_{amb}$$

*Equation A- 16*

The temperature change in length L of duct is

$$T_d(0) - T_d(L) = -\beta(T_d(0) - T_{amb}) - T_{amb} + T_d(0)$$

*Equation A- 17*

This can be rewritten as

$$T_d(0) - T_d(L) = (1 - \beta)(T_d(0) - T_{amb})$$

*Equation A- 18*

Let $\varepsilon$ be the sensible heat exchanger effectiveness

$$\varepsilon = (1 - \beta)$$

*Equation A- 19*

Then the conduction loss from the duct to the duct zone can then be written as

$$Q_{loss} = mc_p(T_d(0) - T_d(L)) = \varepsilon mc_p(T_d(0) - T_{amb})$$

*Equation A- 20*

# Appendix B. Screen Pressure Drop

*The following algorithms are not currently implemented in the code, but are here for future code use, and in the interim are useful to manually determine the effects of screens on window ventilation flow pressure drop.*

The references cited are a few of the papers reviewed to ascertain state of the art regarding screen pressure drop. In one of the more recent papers, Bailey et al. (2003) give the pressure drop through a screen as:

$$\Delta p = K\frac{\rho w^2}{2g_c} = K\frac{m^2}{2g_c\rho A^2}$$

*Equation B- 1*

where,

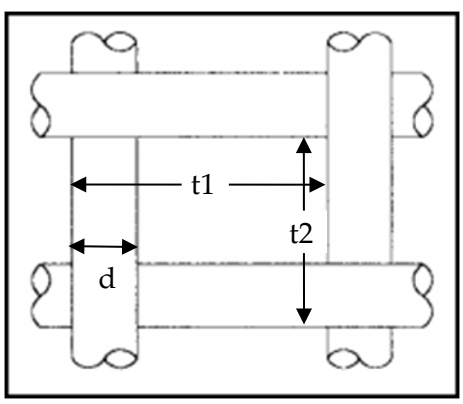
$$K = \left(\frac{1}{\beta^2} - 1\right)\left[\frac{18}{Re} + \frac{0.75}{log(Re + 1.25)} + 0.055\, log(Re)\right]$$

*Equation B- 2*

and,

β = screen porosity = open area/total area perpendicular to flow direction.

$$\beta = \left(1 - \frac{d}{t1}\right)\left(1 - \frac{d}{t2}\right)$$




$Re = \frac{wd}{\nu} = \frac{md}{\rho A \nu}$ = Renolds number.

$w$ = face velocity = $\frac{m}{\rho A}$, ft/sec.

$m$ = mass flow rate, $lb_m$/sec.

$d$ = wire diameter, ft.

$\nu$ = viscosity ≈ 1.25E-4 + 5.54E-07T(degF); $ft^2$/sec. = 1/6100 ft2/sec at 70-F.

$\rho$ = air density, $lb_m/ft^3$.

$g_c$ = 32.2 $lb_m ft/lb_f\text{-}sec^2$.

The first term, intended for portraying Re < 1 pressure drops, is the dominate term. The third term, intended for Re > 200, is relatively negligible, and the second term is a bridge between the first and third terms.

The Reynolds number for the screen flow $Re = \frac{wd}{\nu}$ is roughly 5 times the face velocity in ft/sec. For a velocity of 1 ft/sec, Re ~ 5. For the expected range of wind speeds of concern for ventilation (see note #1), and with the motive of making the partial derivatives simple (see below), Equation B- 2 is approximated as

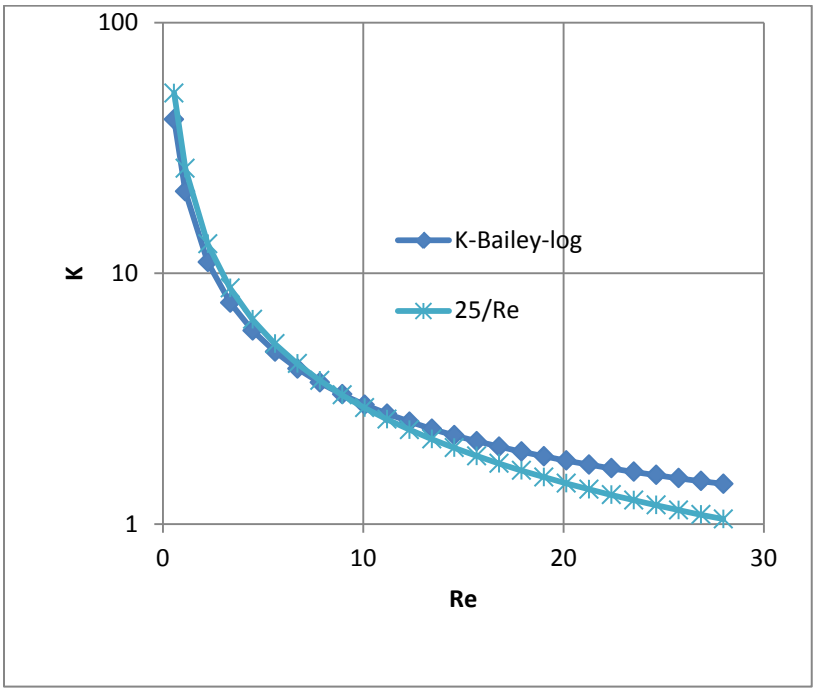
$$K = \left(\frac{1}{\beta^2} - 1\right)\frac{25}{Re}$$

*Equation B- 3*

[note #1: California CZ12 ave yearly Vmet ≈ 11 ft/sec. Correcting for height and shielding gives Vlocal ≈ 11*0.5*0.32 ≈ 1.8 ft/sec. For max flow case of windows on windward and leeward walls, with typical wall wind pressure coefficients, w ≈ 0.5*Vlocal. Thus, the maximum window velocity expected for the annual average wind velocity of 11 ft/sec is w ≈ 0.5*1.8 = 0.9 ft/sec. In a building with windows in multiple directions, the average *w* is expected to be much lower, perhaps 0.5 ft/sec. *Stack Effect:* Using old ASHRAE equation, $w(fpm) = 9\sqrt{H[ft](\Delta T[F])} = 9 * \frac{\sqrt{10*10}}{60} = 1.5\ ft/sec$. Together, the wind and stack may be on the order of 1-ft/sec].

The constant 25 in this approximate formula was determined by forcing Equation B- 2 and Equation B- 3 to match when the window air velocity is at the characteristic value $w_c = 21\psi\nu C_d^2$, defined below. As discussed there, at this velocity the pressure drop through the screen is equal to that through the window orifice.

**Figure B-1: Screen Pressure Drop**




Equation B- 3 can be written as

$$K = \frac{25\psi\nu}{w}, \text{or alternately} \frac{25\psi\rho A\nu}{m}$$

*Equation B- 4*

where, as a result of the approximation Equation B- 3, the screen inputs can be combined into one characteristic screen parameter $\psi$ (of dimension ft$^{-1}$):

$$\psi = \frac{1}{d}\left(\frac{1}{\beta^2} - 1\right)$$

*Equation B- 5*

$\psi$ encapsulates all that needs to be known about the screen for pressure drop purposes. This is only true when $K$ varies in the form assumed byEquation B- 3.

The flow rate through a screenless window is modeled by Airnet as a sharp edged orifice of opening area A.

$$m = C_d A(2\rho g_c \Delta p)^{\frac{1}{2}}$$

*Equation B- 6*

[Idelchik says this is valid for Re > 10^4].

At the equal pressure point the window orifice Reynolds number is $\mathrm{Re_{wdw}} = (\mathrm{D_h}/\mathrm{d})25\psi \mathrm{d}\mathrm{C_d^2}$.

$w_c$ =1.89 ft/sec for std 14x18x0.011 screen &Cd=0.6; 14&18 are wires/inch.

$Re_c$ = 10.6 for std 14x18x0.011 screen &Cd=0.6

$Re_{wdw}$= (~1.5*12/0.011)*(10.6)=1,7345 >$10^4$.

Thus Re is not > $10^4$ when w < ~1 ft/sec. But this is when the pressure drop starts to be dominated by the screen, so the orifice drop accuracy is not so important.

Solving for $\Delta p$,

$$\Delta p = \frac{1}{C_d^2}\frac{\rho w^2}{2g_c} = \frac{1}{C_d^2}\frac{m^2}{2g_c\rho A^2}$$

*Equation B- 7*

Adding Equation B- 1 and Equation B- 7 gives the total pressure drop for a window and screen in series:

$$\Delta p = \left(K + \frac{1}{C_d^2}\right)\frac{m^2}{2g_c\rho A^2}$$

*Equation B- 8*

Solving for mass flow rate,

$$m = AC_d\left(\frac{1}{1 + C_d^2 K}\right)^{\frac{1}{2}}(2\rho g_c \Delta p)^{\frac{1}{2}}$$

*Equation B- 9*

or,

$$m = AC_d S(2\rho g_c \Delta p)^{\frac{1}{2}}$$

*Equation B- 10*

where S is the ratio of the flow with a screen to the flow rate without a screen, as a function of velocity $w$, viscosity, and screen and window orifice parameters.

$$S = \left(\frac{1}{1 + C_d^2 K}\right)^{\frac{1}{2}} = \left(\frac{1}{1 + C_d^2\frac{25\psi\nu}{w}}\right)^{\frac{1}{2}}$$

*Equation B- 11*

Equating Equation B- 1 and Equation B- 7 shows that the velocity when the screen and window pressure drops are equal is given by:

$$w_c = 25\psi\nu C_d^2$$

*Equation B- 12*

The corresponding Reynolds number is $Re_c = 25\psi d C_d^2$.

Substituting Equation B- 12 into Equation B- 11shows that for this condition,

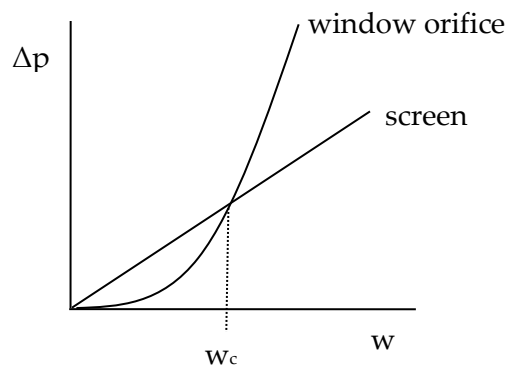
$$S_c = \frac{1}{\sqrt{2}} = 0.707$$

*Equation B- 13*

Equation B- 12 and Equation B- 13 show that $w_c$, which, besides viscosity, only depends on the screen constant $\psi$ and window-orifice coefficient $C_d$, can be considered a "characteristic" velocity, the velocity at which the flow is reduced by (1 - 0.707) ~ 29.3% by the addition of a screen to the window.

Figure B-2 shows that the typical Δp vs. flow curves for a screen and an orifice separately, not in series. The curves cross at velocity $w_c$. To the left of $w_c$, the laminar flow pressure drop dominates the window orifice pressure drop; to the right the orifice pressure drop progressively dominates. Screens give a greater flow reduction at low wind speeds than at high wind speeds.

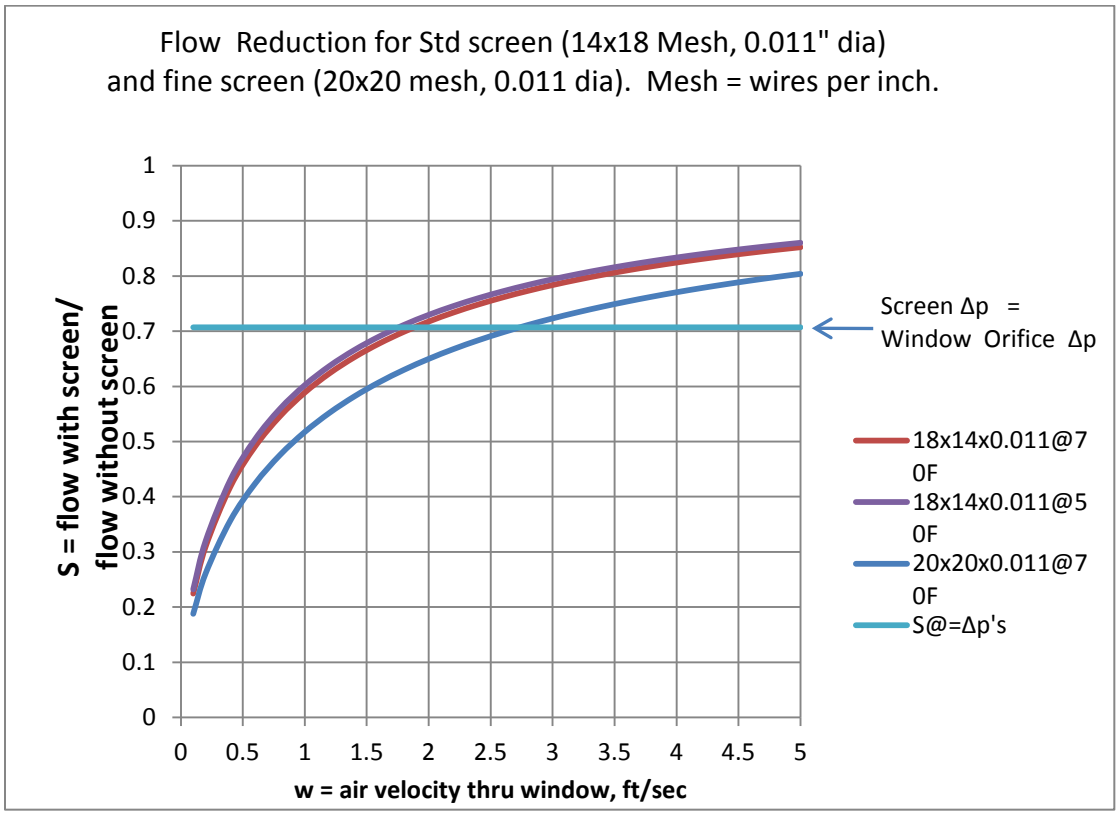
**Figure B-2: Pressure vs. Flow Characteristics**




In Figure B-2, to the left of $w_c$ the laminar-flow screen pressure drop is higher, and to the right the orifice pressure drop dominates.

Figure B-3 shows S as a function of air velocity *w* for two common screen sizes. For the Standard screen, $w_c$ is 1.9-ft/sec. [At w = ~ 1-ft/sec taken as typical according to note #1, S = 0.6, corresponding to a 40% reduction in flow].

**Figure B-3: Standard Screen Flow Reduction**




**Partial Derivatives for use in Airnet:**

From Equation B- 4 and Equation B- 8,

$$\Delta p = \left(\frac{25\psi\rho A v}{m} + \frac{1}{C_d^2}\right)\frac{m^2}{2g_c\rho A^2}$$

*Equation B- 14*

This can be written in the quadratic form for m:

$$m^2 + bm - a\Delta p = 0$$

*Equation B- 15*

Where,

$$a = 2g_c\rho C_d^2 A^2$$

$$b = 25\psi\rho v C_d^2 A$$

The single real root of the quadratic Equation B- 15 gives the mass flow rate through a screen in series with a window-orifice as a function of screen and window properties and overall pressure drop:

$$m = \frac{b}{2}\left\{\left(1 + \frac{4a\Delta p}{b^2}\right)^{\frac{1}{2}} - 1\right\}$$

*Equation B- 16*

If $\Delta p$ is taken as $\Delta p = P_1 - P_2$, then the partial derivative of m with respect to $P_1$ is

$$\frac{\partial m}{\partial P_1} = \frac{a}{b\sqrt{1 + \frac{4a\Delta p}{b^2}}}$$

*Equation B- 17*

$$\frac{\partial m}{\partial P_2} = -\frac{\partial m}{\partial P_1}$$

*Equation B- 18*

These derivatives are needed in the Newton-Raphson procedure. The derivatives Equation B-17 and Equation B- 18 do not become unbounded when $\Delta p = 0$, as does the orifice Equation B-6, so that no special treatment is needed near $\Delta p = 0$.

But the derivatives do become a little peculiar near zero $\Delta p$ as shown in Figure B-4 and Figure B-5. The value of $a/b \approx 25$, and $\frac{4a}{b^2} \approx 88$ for thse plots for the standard screen of Figure B-3. It is possible this could cause problems in the N-R method, but testing AirNet with this type of element is the easiest way to find out.

**Figure B-4: For Small Δp**

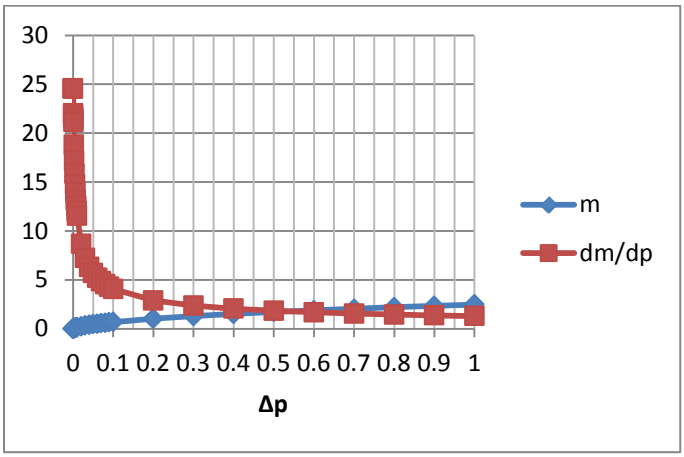
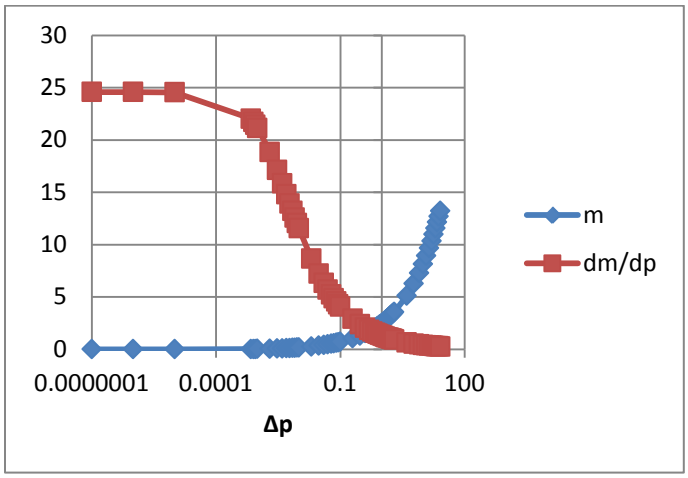



**Figure B-5: For Large Δp**



# Appendix C. Exact Longwave Radiation Model

Figure C-1 shows the standard Heat Transfer Engineering method of determining the long wavelength radiation exchange between black-body surfaces at uniform temperatures (Oppenheim(1956), Mills(1992)). The areas need not be equal, or symmetrically disposed, but are drawn that way for simplicity. The surfaces are assumed to be isothermal, and each surfaces temperature node is connected to all other surface temperature nodes via conductances $h_b A_i F_{ij}$ .

The following methodology is referred to as the "exact" solution in the discussions of Section 2.6. However, it is recognized that it still is an idealization. For instance, surfaces are generally not isothermal. Although the heat transfer $q_{ij}$ [Btu/hr], of Equation C- 2 is accurate if surfaces $i$ and $j$ are isothermal, the *local* surface heat transfer $q'_{ij}$ [Btu/hr-ft²] on the surfaces is nonuniform because the local view factors are different than the integrated value $F_{ij}$. For example, if the two surfaces are connected along a common edge, then near the edge $q'_{ij}$ will be higher than the average $q_{ij}/A_i$, which will tend to change the temperatures of each wall near the edge faster than away from the edge. For the same reason, the radiation intensities are also non-uniform over a surface, which affects the accuracy of the treatment of the emissivity effects by the Oppenheim surface conductance term, which assumes uniform irradiation.

From the Stefan-Boltzmann equation, the net heat transfer rate between surfaces $i$ and $j$ is:

$$q_{ij} = h_b A_i F_{ij}(T_i - T_j)$$

*Equation C- 1*

where,

$h_b = 4\sigma\bar{T}^3 =$ black body radiation coefficient; Btu/(hr-ft²-F).

$\sigma = 0.1714x10^{-8}$ Btu/hr-ft²-R⁴, the Stefan-Boltzmann constant.

$\bar{T}^3 \approx \frac{T_i+T_j}{2}$; degrees R.

The $F_{ij}$ term is the standard view factor, equal to the fraction of radiation leaving surface $i$ that is intercepted by surface $j$. $F_{ij}$ depends on the size, shape, separation, and orientation of the surfaces, and at worst requires a double integration. Reciprocity requires that $A_i F_{ij} = A_j F_{ji}$.

Equation (C-1) is in the linearized form of the Stefan-Boltzmann equation, where for small temperature differences, $\left(T_i^4 - T_j^4\right)$ is approximated by $4\bar{T}^3(T_i - T_j)$.

**Figure C-1: View-Factor Method's Radiant Network for Black-Body Surfaces**

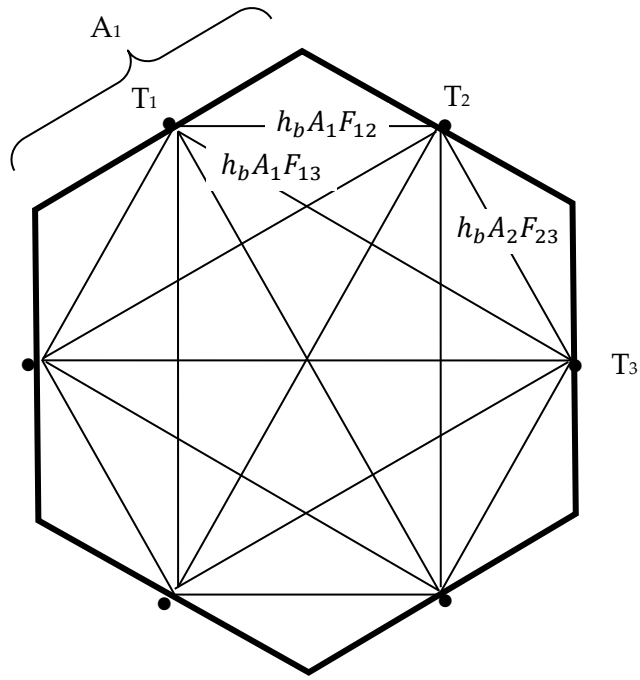
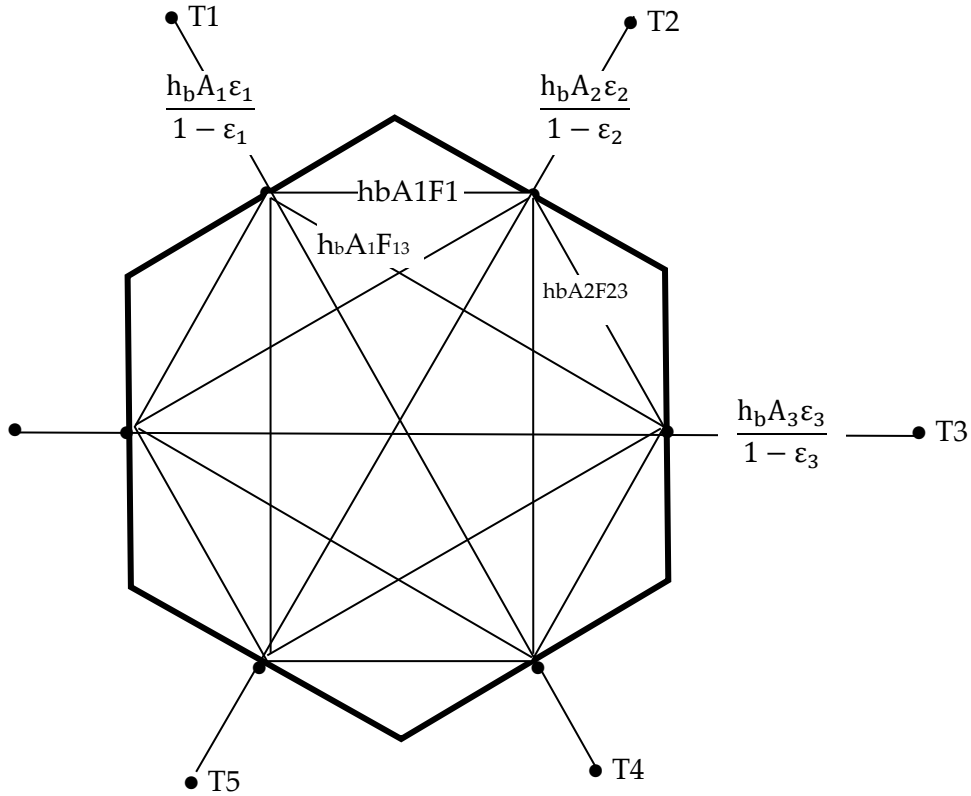


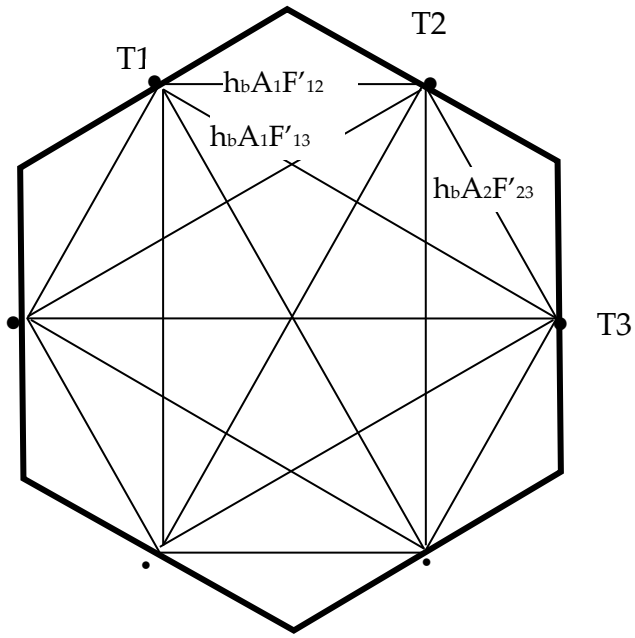

Figure C-2 shows the Figure C-1 black surface case extended to handle diffuse gray surfaces (ε = $\alpha$ = constant over temperature range of interest) with emissivities $\varepsilon_i$, by adding the Oppenheim radiant surface conductances $\frac{A_i \varepsilon_i}{1-\varepsilon_i}$ between the surface temperature nodes and the black body network. (Figure C-2 also represents the un-linearized Stefan-Boltzmann circuit if the surface temperatures are replaced by the emissive power of the surfaces. In this case the surface radiosities are the potentials at the floating nodes.)

**Figure C-2: View-Factor Method's Network for Grey Surfaces**




By dissolving the radiosity nodes using Y-delta transformations, Figure C-2 converts into Figure C-3 showing the same circuit form as the black surface circuit of Figure C-1. The transformation provides the conductances $A_i F'_{ij}$ implicit in the conductances of Figure C-2.

**Figure C-3: View-Factor Method's Network for Grey Surfaces Reduced to Star Network**




$F'_{ij}$ are the 'radiant interchange factors'. As with the black surfaces view factors, reciprocity holds: $A_i F'_{ij} = A_j F'_{ji}$. The net heat transfer between surface *i* and *j* (both directly and via reflections from other surfaces) is given by:

$$q_{ij} = h_b A_i F'_{ij}(T_i - T_j)$$

*Equation C- 2*

The total net heat transfer from surface *i* (i.e., the radiosity minus the irradiation for the un-linearized circuit) is given by summing Equation C- 2 for all the surfaces seen by surface *i, j≠i*:

$$q_i = \sum_{j=1}^{n} h_b A_i F'_{ij}(T_i - T_j)$$

*Equation C- 3*

The above methodology is referred to as the "exact" solution in the discussion of Section 2.6.1. However, as discussed by Carroll, it is recognized that it is still an idealization. For instance, surfaces are generally not isothermal. Although the heat transfer, $q_{ij}$ [Btu/hr], of Equation C- 2 is accurate if surfaces $i$ and $j$ are isothermal, the *local* surface heat transfer $q'_{ij}$ $q'$[Btu/hr-ft$^2$] on the surfaces is nonuniform because the local view factors are different than the integrated value $F_{ij}$. For example, if the two surfaces are connected along a common edge, then near the edge $q'_{ij} q'$ will be higher than the average $q_{ij}/A_i$, which will tend to change the temperatures of each

wall near the edge faster than away from the edge. For the same reason, the radiation intensities are also non-uniform over a surface, which affects the accuracy of the treatment of the emissivity effects by the Oppenheim surface conductance term, which assumes uniform irradiation.

# Appendix D. Determining the Form of the Self-weighting Term $F_i$

Consider a flat black surface of area $A_1$and temperature $T_1$viewing the rest of the room of area $A_s$ and surface temperature $T_s$, with the view factor $F_{1s} = 1$. By Equation C- 1 of Appendix C, the net q from surface $A_1$ is given by:

$$q_1 = h_b A_1 F_{1s}(T_1 - T_s) = h_b A_1 (T_1 - T_s)$$

*Equation D- 1*

With Carroll's model applied to this geometry,

$$q_1 = h_b A_1 F_1 (T_1 - T_r)$$

*Equation D- 2*

where

$$T_r = \frac{A_1 F_1 T_1 + A_s F_s T_s}{A_1 F_1 + A_s F_s}$$

*Equation D- 3*

Equating Equation D- 1 and Equation D- 2 and solving for $F_1$ gives:

$$F_1 = \frac{1}{1 - \frac{A_1 F_1}{A_1 F_1 + A_s F_s}}$$

*Equation D- 4*

The net heat transfer rate from surface 1 is $q_1 = h_b A_1 F_1 (T_1 - T_r)$, with similar expressions for $F_s$ and $q_s$. This is then generalized to the form of Equation 78 of Section 2.6.1.

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