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PNNL-23269

Enhancements to ASHRAE Standard 90.1 Prototype Building Models

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April 2014

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Richland, Washington 99352

Executive Summary

This report was completed by Pacific Northwest National Laboratory (PNNL) in support of the U.S. Department of Energy (DOE) Building Energy Codes Program. DOE supports the development and adoption of energy efficient and cost-effective residential and commercial building energy codes. These codes set the minimum requirements for energy efficient building design and construction and ensure energy savings on a national level. This report focuses on enhancements to prototype building models used to determine the energy impact of various versions of ANSI/ASHRAE/IES[1] Standard 90.1 (herein referred to as Standard 90.1).

Since the last publication of the prototype building models, PNNL has made numerous enhancements to the original prototype models compliant with the 2004, 2007, and 2010 editions of Standard 90.1. Those enhancements are described here and were made for several reasons: (1) to change or improve prototype design assumptions; (2) to improve simulation accuracy; (3) to improve simulation infrastructure; and (4) to add additional detail to the models needed to capture certain energy impacts from Standard 90.1 improvements. These enhancements impact simulated prototype energy use, and consequently impact the savings estimated from edition to edition of Standard 90.1. Table E1 shows the impact of all combined enhancements on the national weighted energy and energy cost savings between Standard 90.1-2004 and 90.1-2010, both with and without plug and process loads. Table E.2 and Table E.3 show the impact of all combined enhancements on energy use index (EUI) and energy savings for each prototype building both with and without plug and process loads.

Table E.1.1. National Average Savings Impact due to Model Enhancements

Standard 90.1-2010 compared to 90.1-2004 National-Weighted Energy Savings	With Plug and Process Loads		Without Plug and Process Loads	
	Pre-Enhancements	Post-Enhancements	Pre-Enhancements	Post-Enhancements
Site Energy	25.62%	23.43%	32.68%	30.39%
Energy Cost	23.16%	22.10%	29.47%	29.26%

[1] American National Standards Institute/American Society of Heating, Refrigerating, and Air-Conditioning Engineers/Illuminating Engineering Society of North America

Table E.0.2. EUI Impact for All Prototypes (with Plug and Process Loads) due to Model Enhancements

Prototype Name		Before Enhancements 2004 kBtu/ft^2	Before Enhancements 2010 kBtu/ft^2	After Enhancements 2004 kBtu/ft^2	After Enhancements 2010 kBtu/ft^2	Energy Savings (%) Before Enhance-ments	Energy Savings (%) After Enhance-ments
Office	Small Office	41.31	32.80	42.37	33.02	20.6%	22.1%
	Medium Office	51.62	37.34	49.49	36.79	27.7%	25.7%
	Large Office	45.99	33.35	84.54	71.88	27.5%	15.0%
Retail	Standalone Retail	75.98	49.53	79.52	53.35	34.8%	32.9%
	Strip Mall	80.40	56.90	83.66	60.40	29.2%	27.8%
Education	Primary School	73.41	50.22	80.08	60.10	31.6%	24.9%
	Secondary School	66.18	41.19	72.94	48.01	37.8%	34.2%
Healthcare	Outpatient Healthcare	163.29	123.61	157.43	120.23	24.3%	23.6%
	Hospital	157.44	118.43	170.45	131.26	24.8%	23.0%
Lodging	Small Hotel	78.52	66.62	73.34	63.62	15.2%	13.2%
	Large Hotel	163.90	125.93	123.47	96.85	23.2%	21.6%
Warehouse	Warehouse	26.28	18.99	25.54	18.23	27.7%	28.6%
Food Service	Fast Food Restaurant	570.07	519.91	653.62	604.35	8.8%	7.5%
	Sit-Down Restaurant	409.65	330.88	471.20	389.14	19.2%	17.4%
Apartment	Mid-Rise Apartment	46.99	41.19	52.12	46.34	12.3%	11.1%
	High-Rise Apartment	48.93	43.97	55.29	50.41	10.1%	8.8%
Totals		73.94	55.0	76.73	58.75		
National Weighted Average						25.6%	23.4%

Table E.0.3. EUI Impact for All Prototypes (without Plug and Process Loads) due to Model Enhancements

		Before Enhancements		After Enhancements		Energy Savings (%)	
Prototype Name		2004 kBtu/ft^2	2010 kBtu/ft^2	2004 kBtu/ft^2	2010 kBtu/ft^2	Before Enhance-ments	After Enhance-ments
Office	Small Office	32.21	24.36	33.27	24.54	24.4%	26.2%
	Medium Office	36.60	23.85	34.47	23.09	34.8%	33.0%
	Large Office	30.37	19.25	41.01	29.55	36.6%	28.0%
Retail	Standalone Retail	68.49	42.06	72.02	45.88	38.6%	36.3%
	Strip Mall	74.97	51.47	78.23	55.01	31.3%	29.7%
Education	Primary School	52.10	29.29	57.12	37.56	43.8%	34.2%
	Secondary School	51.75	27.14	57.69	33.00	47.6%	42.8%
Healthcare	Outpatient Healthcare	116.01	77.16	110.15	73.41	33.5%	33.4%
	Hospital	107.86	69.45	120.18	81.98	35.6%	31.8%
Lodging	Small Hotel	56.06	44.40	50.89	41.41	20.8%	18.6%
	Large Hotel	128.47	90.91	87.38	61.18	29.2%	30.0%
Warehouse	Warehouse	23.75	16.55	23.01	15.73	30.3%	31.6%
Food Service	Fast Food Restaurant	300.63	250.61	343.11	293.99	16.6%	14.3%
	Sit-Down Restaurant	256.29	178.07	299.87	218.01	30.5%	27.3%
Apartment	Mid-Rise Apartment	32.44	26.82	37.57	31.97	17.3%	14.9%
	High-Rise Apartment	35.73	31.02	42.10	37.46	13.2%	11.0%
Totals		56.81	38.24	58.09	40.44		
National Weighted Average						32.7%	30.4%

Acronyms and Abbreviations

AHRI	American Heating and Refrigeration Institute
AHU	air-handling unit
ANSI	American National Standards Institute
ASHRAE	American Society of Heating, Refrigerating, and Air-Conditioning Engineers
BECP	Building Energy Codes Program
Btu	British thermal unit
cfm	cubic feet per minute
DCV	demand controlled ventilation
DOAS	dedicated outdoor air system
DOE	U.S. Department of Energy
DX	direct expansion
EC	electronically commutated
EF	energy factor
EMS	energy management system
EUI	energy use index
gpm	gallons per minute
HVAC	heating, ventilation, and air conditioning
IES	Illuminating Engineering Society of North America
IT	information technology
kBtu	thousand British thermal units
LPD	lighting power density
MAU	make-up air unit
MBtu/h	million British thermal units per hour
MDP	minimum damper position
MSC	Mechanical Subcommittee
NC^3	National Commercial Construction Characteristics (database)
OA	outdoor air
PLR	part load ratio
PNNL	Pacific Northwest National Laboratory
PSC	permanent-split capacitor
PTAC	packaged terminal air conditioner
RCR	room cavity ratio
RE	recovery efficiency
SL	standby loss
SSPC	Standing Standard Project Committee
SWH	service water heating

TMY	typical meteorological year
UPS	uninterruptable power supply
VAV	variable air volume
VIFB	vertical integral face-and-bypass
w.g.	water gauge
WWR	window-to-wall ratio

Contents

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1.0 Introduction

The development of the prototype building models used by Pacific Northwest National Laboratory (PNNL) for analyzing the improvements to ANSI/ASHRAE/IES[1] Standard 90.1 (herein referred to as Standard 90.1) has been described in detail previously in *Achieving the 30% Goal: Energy and Cost Savings Analysis of ASHRAE Standard 90.1-2010* (Thornton et al. 2011), referred to here as *Analysis of 90.1-2010*. As noted in that report, PNNL developed a suite of 16 prototype buildings covering the majority of the commercial building stock and mid-rise to high-rise buildings. The prototypes used in the simulations are intended to represent a cross section of common commercial building types covering 80% of new commercial construction. The 16 prototype building models were reviewed extensively by building industry experts on ASHRAE 90.1 SSPC during development and assessment of multiple editions of ASHRAE Standard 90.1. These prototype models, their detailed characteristics and their development are described in detail on the Building Energy Codes Program (BECP) web site.[2] A detailed description of the prototypes may also be found in the completed savings analysis of ASHRAE Standard 90.1-2010: *Energy and Cost Savings Analysis of ASHRAE Standard 90.1-2010* that can be found on the Building Energy Codes Program (BECP) web site.[3]

Since the publication of that report, PNNL has made numerous enhancements to the original prototype models compliant with the 2004, 2007, and 2010 versions of Standard 90.1 (ASHRAE 2004, 2007, 2010). These enhancements were made for several reasons, including

- to change or improve prototype design assumptions, with input from the ASHRAE Standing Standard Project Committee (SSPC) 90.1;
- to improve simulation accuracy;
- to improve simulation infrastructure; and
- to add detail to the models to capture certain energy impacts from Standard 90.1 improvements.

These enhancements are described in the following sections. For those enhancements that have a substantial impact on prototype building energy use, that impact is shown after the enhancement description. Where the energy impact of an enhancement is shown, the energy use represents an intermediate stage of prototype development. The final energy consumption of the prototype buildings for all enhancements combined is shown in Section 0.

[1] American National Standards Institute/American Society of Heating, Refrigerating, and Air-Conditioning Engineers/Illuminating Engineering Society of North America.
[2] Prototype detail on BECP web site at www.energycodes.gov/development/commercial/90.1_models
[3] BECP web site at www.energycodes.gov/achieving-30-goal-energy-and-cost-savings-analysis-ashrae-ASHRAE Standard-901-2010.

2.0 Prototype Building Model Enhancements

2.1 Re-evaluation of Prototype Building Design

Several enhancements have been made to the prototype building models due to re-evaluation of the prototype design based on feedback from a Standard 90.1 subcommittee tasked with that role. These enhancements have been added to reflect developments in building design (change in window-to-wall ratio (WWR) for Multi-Family prototype building) or modifications in building functions (addition of data center to Large Office prototype building). These and other enhancements are detailed in the section below.

2.1.1 Large Office: Data Center

Based on literature review and a request from the SSPC 90.1 committee, a data center has been added to the Large Office prototype. The section below specifies the characteristics of the data center in terms of area, information technology (IT) equipment (computers, data switches, power supplies, monitors, uninterruptable power supplies, and associated equipment) load, and heating, ventilation, and air-conditioning (HVAC) system configuration.

2.1.1.1 Data Center Area

Literature review indicates that the percentage of data center area in a mixed-use building varies from less than 1% to about 10% of the total building floor area (LBNL 2001-2004; Richman et al. 2008; NREL 2009). For the Large Office prototype, 2.5% of the total building area has been designated as a data center. This includes 1.7% of building area used as a core data center (8476 ft^2) and the other 0.8% of building area used as IT closets (332 ft^2 each). The core data center is placed in the basement while the IT closets are evenly distributed on each of the 12 aboveground floors.

2.1.1.2 IT Equipment Loads

Equipment in data centers typically includes servers, storage devices, network equipment, and uninterruptable power supplies (UPS). Literature review indicates the IT load density for core data centers usually lies between 30 and 60 W/ft^2 for large office buildings (LBNL 2001-2004; Richman et al. 2008; NREL 2009). The IT closets are composed primarily of network equipment such as high power switches, routers, and UPS. In some cases, the IT closets can be small computer server rooms. Because there is no specific power density found in the literature for IT closets, we assume that the IT closets have power density at 20 W/ft^2, which is the minimum threshold value defined for computer rooms in Standard 90.1. Thus, the equipment load densities used are as follows:

- Core data center: 45 W/ft^2 of IT load
- IT closets: 20 W/ft^2

2.1.1.3 HVAC Systems

Typical data center HVAC systems were defined based on discussions with data center experts on the SSPC 90.1 Mechanical Subcommittee (MSC). For the Standard 90.1-2004 prototype, the core data center and IT closets are served by a water-cooled direct expansion (DX) system with a dry cooler and a constant speed fan. Standard 90.1-2004 does not require air-side economizers; hence, these haven't been modeled for the core data center. In the Standard 90.1-2010, requirements were added for economizers in data centers and variable flow fans for large (>110,000 Btu/h) DX systems. Therefore, for the Standard 90.1-2010 prototype, the core data center and IT closets are served by a water-cooled DX with dry-cooler, variable speed fan, and an air-side economizer.

This enhancement impacts the Large Office prototype energy use significantly. Table 2.1 highlights the impact of data center model enhancement on average national energy use for the Large Office prototype.

Table 2.1. Large Office Energy End Use Impact for Standard 90.1-2010 due to Data Center Addition

	Interior Lights (kBtu/ft^2)	Exterior Lights (kBtu/ ft^2)	Misc Loads (kBtu/ ft^2)	Fans (kBtu/ ft^2)	Pumps (kBtu/ ft^2)	Heat Recovery (kBtu/ ft^2)	Cooling (kBtu/ ft^2)	Heating (kBtu/ ft^2)	SWH (kBtu/ ft^2)	Site EUI (kBtu/ft^2)
Before Enhancement	7.30	1.04	14.05	1.58	0.82	0.02	3.69	3.77	0.65	32.9
After Enhancement	7.30	1.04	42.10	4.30	1.16	0.02	10.26	3.07	0.65	71.2

2.1.2 Mid-Rise and High-Rise Apartments: Window-to-Wall Ratio

2.1.2.1 High-Rise Apartments

Feedback from an ASHRAE 90.1 SSPC subcommittee[5] has indicated that a window-to-wall ratio (WWR) of 30% is a representative value for high-rise apartments. Based on this feedback, the High-Rise Apartment prototype WWR has been increased from 15% to 30%. Figure 2.1 shows the High-Rise Apartment prototype before and after this change.

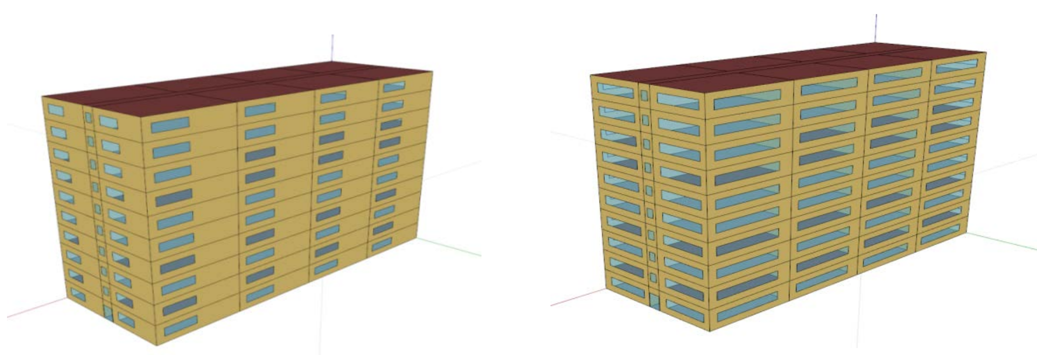

Figure 2.1. High-Rise Apartment Prototype Before (left) and After (right) WWR Enhancement

[5] ASHRAE 90.1 SSPC established the Simulation Working Group expressly to provide feedback on the modeling of improvements to Standard 90.1. The Simulation Working Group's function was superseded by the Advanced Energy Standards Working Group that reports directly to the ASHRAE 90.1 SSPC executive committee.

2.1.2.2 Mid-Rise Apartments

Based on similar feedback from the National Multi-family Housing Council, the WWR for the Mid-Rise Apartment prototype has been modified from 15% to 20%. Figure 2.2 shows the Mid-Rise Apartment prototype before and after this change.

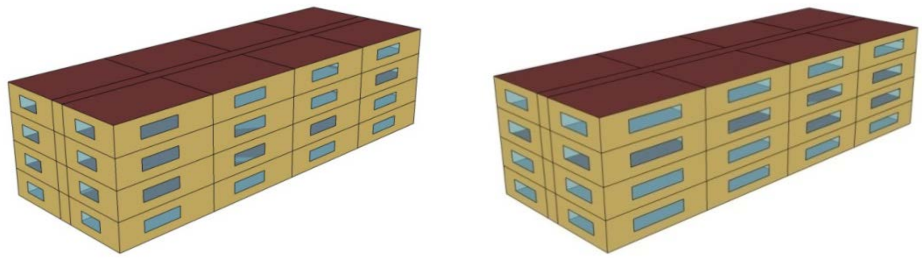

Figure 2.2. Mid-Rise Apartment Prototype Before (left) and After (right) the WWR Enhancement

2.1.3 Service Hot Water Enhancement

Service water heating (SWH) includes water heating uses such as restroom sinks in all prototypes as well as prototype-specific uses such as kitchens and laundry facilities. The main characteristics of the original SWH systems are described in Section 4.6 in *Analysis of 90.1-2010* (Thornton et al. 2011). Based on input from members of SSPC 90.1, it was decided to comprehensively review and update those assumptions. The 16 prototypes were analyzed to compare the SWH system loads and schedules to those in various ASHRAE publications as well as actual installations. The water heater inputs in each of the 16 prototypes were reviewed and analyzed to determine if each input matched the expected industry standards for the building types. The following characteristics of SWH systems were evaluated:

- SWH load
- SWH heater type
- SWH heater number
- SWH fuel type
- heater capacity
- peak efficiency
- part-load efficiency
- storage capacity
- recirculation pumps
- outlet water temperature
- thermal losses

The revised SWH equipment is summarized in Table 2.2 and additional details about the source of efficiency and analysis of the SWH energy use are provided in Appendix A. Since there is a large impact on baseline energy use resulting from this enhancement, Table 2.3 shows the energy impact of the SWH system modifications to all prototypes.

Table 2.2. Summary of Service Water Heating Equipment

Prototype Building	Type	Water Heater Energy Type	Storage Capacity (gal)	Heating Capacity (kBtu/h)	Total Peak Flow (gpm)
Small Office	Main	Electric	40	12 kW	0.06
Medium Office	Main	Gas	100	100	0.85
Large Office	Main	Gas	300	300	6.97
Standalone Retail	Main	Gas	40	40	0.30
Strip Mall	Main, each[a]	Electric	40	12 kW	0.03
Primary School	Main	Gas	200	200	1.67
	Dishwasher (DW) Booster	Electric	6	6 kW	1.00
Secondary School	Main	Gas	600	600	7.63
	DW Booster	Electric	6	19 kW	23.7
Outpatient Healthcare	Main	Gas	200	200	1.00
Hospital	Main	Gas	600	600	2.14
	DW Booster	Electric	6	3 kW	0.58
	Laundry	Gas	300	300	2.8
Small Hotel	Main	Gas	300	300	2.85
	Laundry	Gas	200	200	2.05
Large Hotel	Main	Gas	600	600	5.94
	DW Booster	Electric	6	8 kW	1.33
	Laundry	Gas	300	300	30.6
Warehouse	Main	Electric	20	6 kW	0.13
Quick-Service Restaurant	Main	Gas	100	100	1.52
Full-Service Restaurant	Main	Gas	200	200	2.22
	DW Booster	Electric	6	8 kW	1.33
Mid-Rise Apartment	Main, per apartment[b]	Electric	50	15 kW	0.06
High-Rise Apartment	Main	Gas	600	600	4.58

(a) There are seven water heaters, each serving one of the seven model zones.

(b) The Mid-Rise Apartment model includes 23 separate water heaters. Fifteen serve one apartment each on the ground and top floors. Eight serve two apartments each; these apartments are modeled as one apartment each with a multiplier of two, so water heater inputs must each account for two apartments.

Table 2.3. Baseline Impact through SHW Enhancements

	90.1-2004 Model			90.1-2010 Model		
Prototype Name	EUI ($kBtu/ft^2$) Before Enhancement	EUI ($kBtu/ft^2$) After Enhancement	% Change	EUI ($kBtu/ft^2$) Before Enhancement	EUI ($kBtu/ft^2$) After Enhancement	% Change
Small Office	39.29	39.75	1.17%	30.21	30.53	1.06%
Medium Office	51.64	49.42	-4.30%	38.22	35.99	-5.83%
Large Office	83.49	83.83	0.41%	71.11	71.47	0.50%
Standalone Retail	75.93	77.35	1.87%	50.85	52.20	2.64%
Strip Mall	79.80	80.00	0.25%	56.58	56.92	0.60%
Primary School	75.58	76.58	1.32%	52.96	53.77	1.53%
Secondary School	63.93	66.01	3.25%	44.01	46.38	5.36%
Outpatient HealthCare	163.38	165.15	1.08%	122.65	124.00	1.10%
Hospital	160.68	164.14	2.15%	123.80	127.33	2.85%
Small Hotel	72.02	73.72	2.36%	61.55	63.28	2.80%
Large Hotel	147.65	116.79	-20.90%	124.33	93.37	-24.90%
Warehouse	26.66	26.86	0.75%	19.28	19.47	0.96%
Fast Food Restaurant	585.43	628.22	7.31%	538.15	583.75	8.47%
Sit-Down Restaurant	419.62	455.42	8.53%	338.96	373.25	10.12%
Mid-Rise Apartment	48.65	51.70	6.27%	42.37	45.69	7.83%
High-Rise Apartment	53.19	57.95	8.95%	47.55	52.88	11.21%
National Weighted Average	74.83	75.09	0.35%	57.37	57.69	0.57%

2.1.4 Decentralization of Make-Up Air Unit for Small Hotel Prototype

In the original model for the Small Hotel prototype, a central make-up air unit (MAU) was used to supply outdoor air (OA) to the guestrooms and corridors, which resulted in an extremely large MAU serving the entire building. This MAU has been removed and the required OA is now assumed to enter each packaged terminal air conditioner (PTAC) directly, which is more typical of common practice for this building type, based on discussion with members of ASHRAE 90.1 SSPC. This change affects the energy use in the small hotel prototype in the following ways:

- OA is now being heated and cooled through the PTAC unit.
- Cooling energy use has been reduced due to slightly better cooling system efficiency for the PTAC unit compared to the MAU.
- Electric heating for PTACs (as compared to gas heating for the MAU) reduces site heating energy but increases source energy and raises energy costs.

2.2 Improvements to Simulation Accuracy

The prototype building models are continuously verified and the models are improved to enhance accuracy of the simulation. Improvements include modifications to the implementation of addenda; such as addendum cd to 90.1-2007—exterior lighting control; or improvements to previous modeling approaches, such as enhancements made due to improved methodologies for multi-zone system outdoor air calculations.

2.2.1 Small Office Prototype: Heat Pump Resistance Heat Lock Out

Based on a review of industry equipment literature and discussion with PNNL staff familiar with field characteristics of heat pump controls, values of parameters related to low-temperature operation have been modified for air source heat pumps used in the Small Office prototype. These include the minimum outdoor air temperature for compressor operation, crankcase heater capacity, maximum outdoor air temperature for crankcase heater operation and supplemental heater operation, and heat pump defrost strategy. Table 2.4 summarizes those enhancements.

Table 2.4. Small Office Enhancements to Air Source Heat Pump Parameters

Parameter	Minimum OA Temperature for Compressor Operation	Crankcase Heater Capacity	Maximum OA Temperature for Crankcase Heater Operation	Maximum OA Temperature for Supplemental Heater Operation	Heat Pump Defrost Strategy	Heat Pump Defrost Control
Before Changes	25°F	200 W	50°F	70°F	Resistive	On demand
After Changes	10°F	50 W	40°F	40°F	Reverse cycle	On demand

2.2.2 High-Rise Apartments: Vestibules

Previously, the vestibule requirements (i.e., addenda c to Standard 90.1-2004 and q to 90.1-2007) were not implemented in the High-Rise Apartment prototype because of misinterpretation of exceptions in the standard. This has been corrected. The Standard 90.1-2004 model has been modified to include air infiltration due to opening and closing of doors, and infiltration has been reduced for the Standard 90.1-2007 and 90.1-2010 models to account for vestibules.

Door opening frequency has been determined along with peak infiltration rates defined on a door area basis for the High-Rise Apartment prototype (Thornton et al. 2011). Similar calculations were made for the High-Rise Apartment prototype building. A peak infiltration rate of 3230 cfm for the ground floor corridor zone (that serves as a proxy for the building lobby) was added for Standard 90.1-2004 models, which then was reduced to a peak infiltration rate of 2125 cfm for climate zones 3 to 8 for Standard 90.1-2007 and 90.1-2010 models.

2.2.3 Lighting Power Density

Lighting power density (LPD) calculations were reviewed for consistency and updated for a few prototypes. The consistency improvements along with the enhancements are documented below.

2.2.3.1 Mid-Rise and High-Rise Apartments

The LPD for apartment prototypes was previously modeled to include only hardwired lighting, which accounts for 80% of the total lighting usage typical for single and multifamily housing (Hendron 2008). The other 20% of the lighting load, consisting of plug-in lighting, was previously unaccounted for. The apartment prototypes have been updated to include both hard-wired and plug-in lighting as shown in Table 2.5 for each apartment unit.

Table 2.5. LPD Calculations for Hardwired and Plug-in Lighting

Annual hard-wired lighting (pre and post enhancement)	972 kWh
Lighting use per day	2663.01 Wh
Peak lighting use	341.41 W
LPD input (hard-wired)	0.36 W/ft^2
Annual plug-in lighting (post enhancement only)	243 kWh
Lighting use per day	665.75 Wh
Peak lighting use	85.35 W
LPD input (plug-in)	0.09 W/ft^2

2.2.3.2 Large Hotel

The basement in the Large Hotel prototype used the LPD requirement determined through the building area method for "hotel" use type but the zone is classified[6] as an "office" space. The area of this space is quite large (21,300 ft^2) and it isn't classified as an enclosed office; hence, it has been modified to the LPD requirements determined through the building area method for "office" building area type. This LPD is lower than both the "hotel" building LPD and the "enclosed office" space LPD, so it more accurately reflects lighting levels in this area. Occupancy sensor savings have been applied to this space because of this change, using the space distribution in the Medium Office prototype. Those savings have been applied through a schedule multiplier, as explained in Section 2.2.4, Lighting Control. The resulting schedule reduction fractions for this enhancement are listed in Table 2.7.

2.2.3.3 Retail Standalone

This prototype used the LPD requirements specified through the building area method for "storage." This has been updated to use an area-weighted LPD determined using the space-by-space LPDs and the space distribution used previously for determining savings from occupancy sensors for addendum x to Standard 90.1-2007 (Thornton et al, 2011).

2.2.3.4 Retail Strip Mall

Standard 90.1-2004, Section 9.6.2, permits an additional lighting power allowance for lighting designed to highlight merchandise. Standard 90.1-2007 and 90.1-2010, in addition to the LPD allowance, include an additional 1000 W allowance unrelated to area. This 1000 W allowance was not previously included and has now been added for the Retail Strip Mall prototype for both the Standard 90.1-2007 and the 90.1-2010 versions of the model.

[6] In development of the prototypes, the assumed characteristics of each zone are fully documented. See Footnote 2.

The Retail Strip Mall prototype previously used an area-weighted LPD based on the building area method specified LPDs for “Retail” and “Office” use types. Five percent of the area of the building was assumed to be “other spaces” with the LPD being reduced proportionally. This methodology has been updated to use space-by-space LPD requirements in the area-weighting calculation. The 5% “other spaces” area has been re-distributed amongst the other areas proportionally for ensuring consistency. The additional lighting allowance, which is only available if the space-by-space approach is used, is now applied correctly.

2.2.3.5 Warehouse

The Warehouse prototype used the LPD from the building area method for the office use-type. However, the LPD value for Standard 90.1-2010 prototype was previously specified at 1.0 W/ft^2. This has been corrected to be 0.9 W/ft^2. The occupancy sensor savings calculations from addendum x to Standard 90.1-2007 assume that the office portion of the Warehouse prototype is a combination of open and enclosed offices. Since the area of the office zone is quite large (2550 ft^2), it is also intended to include areas with lower LPDs such as corridors and restrooms. Thus, the occupancy sensor savings calculations have been revised to use the area distribution for small office from the National Commercial Construction Characteristics (NC^3) database (Richman et al. 2008). The occupancy sensor savings calculation methodology is explained in Section 2.2.4 and the resulting schedule reduction fractions for this enhancement are listed in Table 2.7. .

2.2.3.6 Hospital

The Hospital prototype uses the enclosed office space LPD for the “basement” zone. However, the basement zone is too large (40,235 ft^2) to include just enclosed office spaces; hence, it has been updated to use the office LPD requirements from the building area method.[7] Occupancy sensor savings have been applied to this space using the space distribution in the Medium Office prototype. The occupancy sensor savings calculation methodology is explained in Section 2.2.4 and the resulting schedule reduction fractions for this enhancement are listed in Table 2.7.

2.2.3.7 Corridors in all prototypes

Corridors in all prototypes were assigned an average room cavity ratio (RCR) allowance based on NC^3 data in the *Analysis of 90.1-2010* (Thornton et al. 2011). A re-evaluation of the corridors as modeled in the Primary School prototypes indicated that all corridors are less than 8 ft wide and thus the additional allowance isn’t applicable. Hence, the additional allowance has been rolled back for corridors wider than 8 ft. The previous analysis used the space areas from the NC^3 database for RCR calculations. This approach is now modified to use the actual space areas and dimensions in the prototype.

Table 2.6 summarizes the LPD values for the prototypes affected by the enhancements. Table 2.7 summarizes the occupancy sensor multipliers for the affected prototypes.

[7] See similar discussion for hotel prototype in Section 2.2.3.2 and footnote 6.

Table 2.6. Prototypes Affected by LPD Enhancement

		90.1-2004 W/ft^2		90.1-2007 W/ft^2		90.1-2010 W/ft^2	
Prototype	Zone	Before Enhance-ment	After Enhance-ment	Before Enhance-ment	After Enhance-ment	Before Enhance-ment	After Enhance-ment
Large Hotel	Basement	1.0	1.0	1.0	1.0	1.0	0.9
	Corridors	0.5	0.5	0.5	0.5	0.67	0.66
Standalone Retail	Back Space	0.8	0.839	0.8	0.839	0.63	0.703
Strip Mall Retail	Type 1	1.3	1.475	1.3	1.475	1.1875	1.425
	Type 2	1.3	1.475	1.3	1.475	1.1875	1.425
	Type 3	1.3	1.475	1.3	1.475	1.1875	1.425
Warehouse	Office	1.0	1.0	1.0	1.0	1.0	0.9
Hospital	Office	1.11	1.0	1.11	1.0	1.11	0.9
	Corridors	1.0	1.0	1.0	1.0	0.9	0.89
Apartment High-Rise and Mid-Rise	Corridors	0.5	0.5	0.5	0.5	0.67	0.792
Primary and Secondary School	Corridors	0.5	0.5	0.5	0.5	0.67	0.66
Small Hotel	Corridors	0.5	0.5	0.5	0.5	0.67	0.792
	Exercise Room	0.9	0.9	0.9	0.9	0.76	0.864
Outpatient Healthcare	Anesthesia	1.5	1.5	1.5	1.5	1.71	1.66
	Clean Room	1.5	1.5	1.5	1.5	1.71	1.66
	Corridors	1.5	1.5	1.5	1.5	0.67	0.792
	Examination	1.5	1.5	1.5	1.5	1.71	1.66
	Soiled	1.5	1.5	1.5	1.5	1.71	1.66
	Utility Room	1.5	1.5	1.5	1.5	1.71	1.66

Table 2.7. Occupancy Sensor Savings Multipliers

Prototype	Zone	Occupancy Sensor Multiplier
Hospital	Basement	0.938
Large Hotel	Basement	0.938
Warehouse	Office	0.855

2.2.4 Lighting Control

Automatic lighting shutoff controls were previously modeled as a reduction in LPD as a surrogate for reduced hours of operation. This strategy was used for occupancy sensor requirements in lecture halls, training rooms, supply and storage rooms (up to 1000 ft^2), office spaces (up to 250 ft^2), restrooms, dressing rooms, locker rooms, and fitting rooms (addendum x to Standard 90.1-2007), interior stairways (addendum cf to Standard 90.1-2007), and bathroom lighting in hotels and motels (addendum aw to Standard 90.1-2007).

To provide more accuracy and a truer diversity of loads, this approach has been revised by reducing the schedule fractions for the zones of the affected space types instead of the LPD. This enhancement changed the LPD of affected space types back to the requirements of Standard 90.1-2010. Then space-specific lighting schedules were created for the spaces affected by the requirement and the schedule fractions were modified to reflect the impact of lighting controls. Table 2.8 shows the percentage of time various space types are assumed to be unoccupied during otherwise occupied hours and thus available for savings from occupancy sensors (Thornton et al. 2011). The schedule reduction fractions are calculated using values from Table 2.8 as follows:

$$Schedule\ Reduction\ Fraction\ = Space\ Type\ Fraction \times Occupancy\ Sensor\ Reduction$$

Table 2.8. Occupancy Sensor Control Lighting Reduction by Space Type

Space types	Occupancy Sensor Reduction Estimate
Pre-K to 12 Classrooms	32%
Storage and Supply (50-1,000 ft^2)	48%
Offices (private up to 250 ft^2)	22%
Restrooms	34%
Dressing/Fitting Rooms	10%

The example below for the Large Hotel prototype demonstrates the calculation of reduction fractions for occupancy sensors in the storage room, which are then applied to the lighting schedules based on the occupancy of the space. Table 2.9 shows the breakdown of storage space types in the Large Hotel prototype.

Table 2.9. Large Hotel Percentage of Storage Area by Space Type

Space Type	Space Type Fraction
Active storage $\geq$ 50 ft^2 and $\leq$ 1000 ft^2	86%
Active storage < 50 ft^2 and > 1000 ft^2	14%

The reduction estimate is then applied to the space area fraction of the impacted zone area. In the case of the Large Hotel, Addendum x savings are applicable only to storage rooms 50 to 1000 ft^2in area, which forms 86% of the total storage space in the large hotel prototype. Hence, a saving of 86% × 48% = 41% has been applied to the lighting schedule of all storage spaces.

Addendum cf to Standard 90.1-2007 requires stairwell lighting to be controlled automatically so that lighting power is reduced to 50% within 30 minutes of all occupants leaving the zone. The occupancy percentage of stairwells is assumed to be 10%, based on the supporting information in the foreword to the public review of this addendum. The stairwell is therefore unoccupied 90% of the time. The control is calculated as a 50% reduction in lighting when unoccupied as required by the addendum. This addendum was also previously implemented by reducing the LPD of stairwells in the prototype models. For the modified approach, the schedule reduction fraction is calculated based on the stairwell area within the building estimated through the NC[3] database (Richman et al. 2008). Estimated savings are calculated using the equation below and applied to the lighting schedule applicable to stairwells.

$$Schedule\ Reduction\ Fraction = \frac{Stairwell\ Area\ Fraction\ \times\ Unoccupied\ Fraction\ (90\%) \times\ Savings\ from\ Controls\ Requirement\ (50\%)}{Actual\ Space\ Area\ Fraction}$$

Addendum aw to Standard 90.1-2007 requires bathrooms within hotel and motel guestrooms to have a separate control device capable of turning off the bathroom lighting, except night lighting not exceeding 5 W, within 60 minutes of an occupant leaving the space. This was also previously modeled as an LPD reduction for guestrooms in hotel and motels and has been modified to calculate a schedule reduction fraction based on the fraction of lighting comprised by bathroom luminaires of the total guestroom lighting. The schedule reduction fraction is calculated using the following equation and the fraction of bathroom lighting and occupancy sensor savings as described in the *Analysis of 90.1-2010* (Thornton et al. 2011:

$$Schedule\ Reduction\ Fraction = \ Fraction\ of\ Bathroom\ Lighting \times Occupancy\ Sensor\ Savings$$

Addendum aa to Standard 90.1-2007 included requirements for occupancy sensors to be manual on instead of automatic on. Savings was assumed to occur in perimeter offices when daylight was available (Thornton et al. 2011). Previously savings was taken in the three office prototypes as a reduction in LPD. This has been modified to calculate a schedule reduction based on the assumed savings from manual-on occupancy sensors and the applicable area fraction using the following equation:

$$Schedule\ Reduction\ Fraction = Percentage\ of\ Enclosed\ Daylit\ Office\ Area \times Maual\ On\ Control\ Savings$$

The calculations and schedule reduction fractions for the three office prototypes are summarized in Table 2.10.

Table 2.10. Manual-On Occupancy Sensor Schedule Reductions for Office Buildings

Prototype Building	Total Building Area (ft^2)	Enclosed Daylit Office Area (ft^2)	Enclosed Daylit Office Area Fraction	Manual-On Savings Percentage	Occupied Hours Schedule Reduction Fraction
Small Office	5,500	1,595	0.29	0.10	0.029
Medium Office	53,600	10,023	0.19	0.09	0.017
Large Office	498,600	69,494	0.14	0.09	0.013

As described in Section 2.2.3.2, the basement for Large Hotel prototype is classified as office space and this enhancement applies occupancy sensor savings using the space type distribution for the Medium Office prototype. The occupied hours reduction fraction is applied as a schedule multiplier to the lighting schedules in the basement zone. Table 2.11 shows the savings calculations for the Large Hotel Basement due to occupancy sensors.

Table 2.11. Occupancy Sensor Savings for the Basement in Large Hotel Prototype

Savings Space (from Add. X methodology)	Occupancy Sensor Savings (from Add. X methodology)	Zone Area (ft^2)	Space Area Fraction	Occupied hours Savings Fraction
Active storage ≥ 50 ft^2 and ≤ 1000 ft^2	0.48	21300	0.019	0.009
Office - enclosed	0.22	21300	0.187	0.041
Restrooms	0.34	21300	0.036	0.012
Occupied Hours Reduction Fraction				0.938

To provide a detailed example of the LPD and lighting controls enhancements discussed above, Table 2.12 summarizes all the revised LPDs and the lighting schedule adjustment fractions for the Large Hotel prototype. Figure 2.3 illustrates the changes to the lighting schedules in the Large Hotel. Note that the schedules before-enhancement for the storerooms and corridors are almost identical.

Table 2.12. Schedule Multipliers for Large Hotel Prototype to Account for Occupancy Sensors

	LPD (W/ft^2)			Schedule Reduction Fractions					
Zone Space Type	90.1-2004	90.1-2007	90.1-20010	Occupancy Sensor (Addendum x)	Interior Stairways (Addendum cf)	Guestroom bathroom lighting (Addendum AW)	Automatic Shutoff (Addendum AA)	Total Occupied Hours Reduction Fraction	Occupied Hours Schedule Multiplier
Basement	1.0	1.0	1.0	0.9375	0.0000	0.0000	0.0000	0.9375	0.0625
Retail	1.5	1.5	1.4	0.0000	0.0000	0.0000	0.0000	0.0000	1.0000
Mechanic	1.5	1.5	0.95	0.0000	0.0000	0.0000	0.0000	0.0000	1.0000
Storage	0.8	0.8	0.63	0.4145	0.0000	0.0000	0.0000	0.4145	0.5855
Laundry	0.6	0.6	0.6	0.0000	0.0000	0.0000	0.0000	0.0000	1.0000
Dining	1.3	1.3	0.89	0.0000	0.0000	0.0000	0.0000	0.0000	1.0000
Lobby	1.1	1.1	1.06	0.0000	0.0000	0.0000	0.0000	0.0000	1.0000
Guestroom	1.1	1.1	1.11	0.0000	0.0000	0.0310	0.0000	0.0310	0.9690
Corridor_flr3	0.5	0.5	0.67	0.0000	0.1657	0.0000	0.0000	0.1657	0.8343
Corridor_flr6	0.5	0.5	0.67	0.0000	0.1657	0.0000	0.0000	0.1657	0.8343
Kitchen	1.2	1.2	0.99	0.0000	0.0000	0.0000	0.0000	0.0000	1.0000

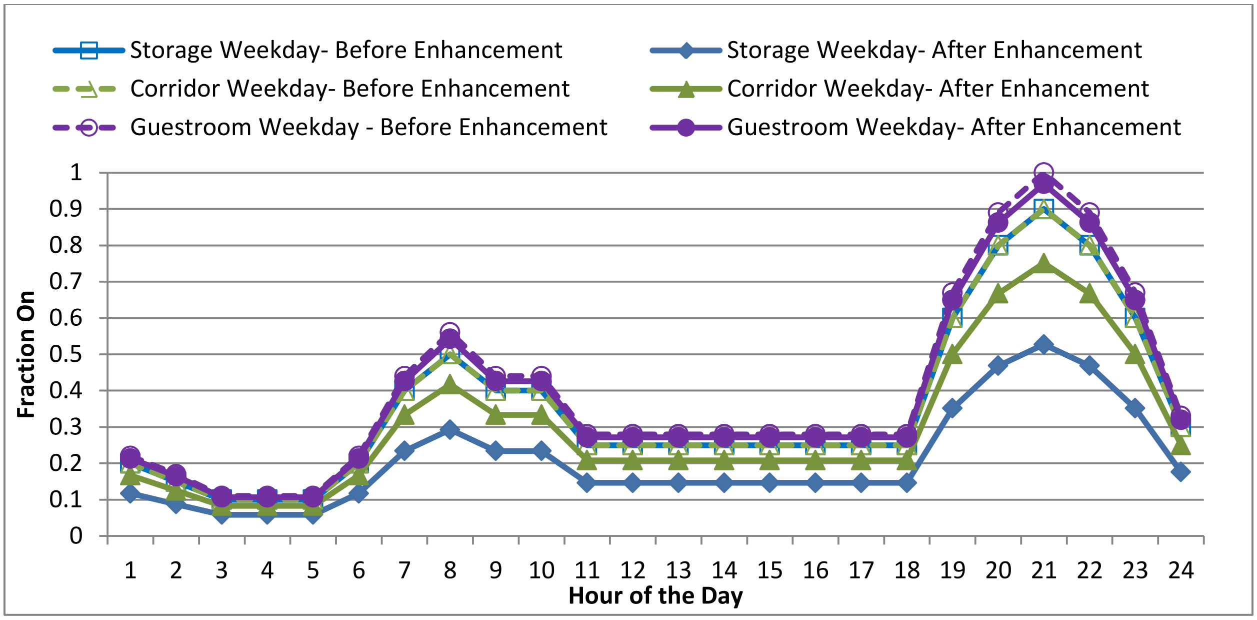

Figure 2.3. Schedule Modification to Reflect Lighting Controls in Large Hotel Prototype

2.2.5 Exterior Lighting Control

Standard 90.1-2010, Section 9.4.1.7c, requires exterior lighting (except building façade and landscape lighting) to be controlled by a device that automatically reduces the connected lighting power by at least 30% for either night time hours (defined to be from midnight or within 1 hour of the end of business operations, whichever is later, until 6 am or business opening, whichever is earlier) or when no activity has been detected for a maximum period of 15 minutes. This requirement had been previously implemented as a 70% reduction of total exterior lighting and has now been corrected to reflect a 30% reduction during the night time hours.

2.2.6 Multi-Zone System Ventilation Calculation Enhancements

Section 5.2.2.21 in *Analysis of 90.1-2010* (Thornton et al. 2011) describes how the prototypes with multiple-zone variable air volume (VAV) systems were modeled for outdoor air ventilation optimization control in the Standard 90.1-2010 models. That section also describes how multiple-zone system outdoor air intake rate (V_{ot}) and minimum damper positions (MDPs) were calculated for the previously published Standard 90.1 models (see subsection "Calculating Standard 62.1 Multiple-zone System Outdoor Air Flow" under Section 5.2.2.21 in *Analysis of 90.1-2010*). The current enhancement modifies the calculation process for V_{ot} and MDP, and the outdoor air ventilation optimization control modeling method remains the same. The enhancement changes the applicability of ventilation optimization control shown in Table 5.27 in *Analysis of 90.1-2010* (Thornton et al. 2011). This section only discusses calculation methods for V_{ot} and MDP. The methods consist of three key steps: (1) calculate zone ventilation efficiency; (2) calculate system ventilation efficiency; and (3) adjust MDP for a target system ventilation efficiency. Steps 1 and 2 remain the same as before. The original method in Step 3 resulted in some unrealistic MDP and V_{ot} modeling inputs, and manual adjustments were conducted for some prototypes based on trial and error methods (Thornton et al. 2011). The process cannot be automated and the manual adjustments may not produce repeatable results. Therefore, the key of the enhancement is to revise Step 3. The whole process is described below and Step 3 before and after the enhancement is explained.

Step 1. Calculation for zone ventilation efficiency

$$V_{oz,i} = V_{bz,i}/E_{z,i}$$

$$V_{dz,i} = V_{pz,i} \cdot MDP_i$$

$$Z_{d,i} = V_{oz,i}/V_{dz,i}$$

$$E_{vz,i} = 1 + X_s - Z_{d,i}$$

Where

$V_{oz,i}$ is zone outdoor airflow rate for zone i.

$V_{bz,i}$ is breathing zone outdoor airflow rate for zone i. It is calculated based on zone ventilation requirements in ASHRAE Standard 62.1.

$E_{z,i}$ is zone air distribution effectiveness for zone i. It is assumed to be 1 for all zones.

$V_{dz,i}$ is minimum expected zone discharge airflow rate for zone i.

$V_{pz,i}$ is primary airflow rate for design purposes. It is calculated during the design day simulation and the larger of calculated zone heating and cooling flow rates is used.

MDP_i is minimum damper position for zone i. This is taken from prescriptive requirements in Standard 90.1.

$Z_{d,i}$ is zone discharge outdoor air fraction. It is calculated based on the minimum expected zone discharge airflow based on the minimum damper position.

$E_{vz,i}$ is zone ventilation efficiency for zone i. The efficiency with which the system distributes air from the outdoor air intake to the breathing zone in any particular ventilation zone.

X_s is average outdoor air fraction: at the primary air handler, the fraction of outdoor air intake flow in the system primary airflow. See below for its calculation method.

Step 2. Calculation for system ventilation efficiency

$$V_{o,u} = \sum V_{oz,i}$$

$$X_s = V_{o,u}/V_{p,s}$$

$$E_v = \min(E_{vz,i})$$

Where

$V_{o,u}$ is uncorrected outdoor air intake flowrate.

$V_{p,s}$ is system primary airflow rate. It is calculated during the design day simulation.

E_v is system ventilation efficiency, the efficiency with which the system distributes air from the outdoor air intake to the breathing zone in the ventilation-critical zone, which requires the largest fraction of outdoor air in the primary air stream.

At this point, a total system outdoor air intake flow rate V_{ot} can be calculated using the formula below and the minimum damper position, MDP_i, is the prescriptive requirement in the standards. These two parameters are needed as EnergyPlus inputs.

$$V_{ot} = V_{o,u}/E_v$$

Where

V_{ot} is outdoor air intake flow rate.

Step 3 (before enhancement). Adjustments for target minimum system ventilation efficiency

All versions of Standard 90.1 allow an exception for the prescriptive MDP requirement. For example, in Standard 90.1-2004, Exception 5 to Section 6.5.2.1 indicates that one can use higher MDPs if the designer can demonstrate that overall system annual energy usage is reduced by offsetting reheat/recooling energy losses through reduction in outdoor air intake in accordance with the multiple space requirements defined in ASHRAE Standard 62.1. The overall annual energy savings are realized by increasing the system ventilation efficiency E_v and reducing system outdoor air intake, potentially resulting in a net reduction in energy needed to temper outdoor air. In the calculation method above, it is assumed that the system ventilation efficiency equals the minimum zone ventilation efficiency among all the zones, i.e., $E_v = \min(E_{vz,i})$. Therefore, to increase system ventilation efficiency, the MDP for critical zones, which reduces the system ventilation effectiveness, needs to be increased. For this purpose, for all versions of Standard 90.1, PNNL developed a calculation method to adjust the critical zone MDPs, and the method was implemented in the previously published models. Before the current enhancement, the target minimum system ventilation efficiency values for different prototypes were not the same. After the enhancement, the target was set to 0.6 for all applicable prototypes. Following is the previously developed adjustment method using the target of 0.6 as an example.

$$\mathrm{MDP}_{a,i} = \begin{cases} V_{oz,i}/V_{pz} \cdot (1 + X_s - 0.6), & \text{if } E_{vz,i} < 0.6 \\ \mathrm{MDP}_i, & \text{if } E_{vz,i} \geq 0.6 \end{cases}$$

$$V_{dz,a,i} = V_{pz,i} \cdot \mathrm{MDP}_{a,i}$$

$$Z_{d,a,i} = V_{oz,i}/V_{dz,a,i}$$

$$E_{vz,a,i} = 1 + X_s - Z_{d,a,i}$$

Where

Subscript "a" stands for "adjusted"

System efficiency and outdoor air intake calculation after zone ventilation efficiency adjustment is as follows:

$$E_{v,a} = \min(E_{vz,a,i})$$

$$V_{ot,a} = V_{o,u}/E_{v,a}$$

Step 3 (after enhancement). Adjustments for target minimum system ventilation efficiency

The calculation procedure above is the general method used for developing the previous Standard 90.1 models. Some special cases were documented at the end of Section 5.2.2.21 in *Analysis of 90.1-2010* (Thornton et al. 2011). The special cases are that some zone ventilation efficiencies were too small or negative and/or the adjusted minimum damper position $\mathrm{MDP}_{a,i}$ was larger than 1. *Analysis of 90.1-2010* (Thornton et al. 2011) used manual trial and error methods to address the special cases in different ways and suggested that the method be enhanced to provide further refinement. During the model enhancement, those special cases were reviewed and it was found that since the primary airflow rate for design purposes $V_{pz,i}$(calculated based on sizing simulation) is so small that it is close to or smaller than the required zone outdoor flowrate, $V_{oz,i}$. To address this issue, the method to adjust minimum damper position $\mathrm{MDP}_{a,i}$ was changed as follows.

When, $E_{vz,i} < 0.6$, meaning $Z_{d,i} > 1 + X_s - 0.6$

$$Z_{d,a,i} = 1 + X_s - 0.6$$

$$V_{dz,a,i} = V_{oz}/Z_{d,a,i}$$

$$\mathrm{MDP}_{a,i} = \min(V_{dz,a,i}/V_{pz,i}, 1)$$

At the second-to-last equation above, the adjusted minimum zone discharge airflow rate ($V_{dz,a,i}$) was found to be larger than the design primary airflow rate ($V_{pz,i}$) in a small number of zones in the prototype models, i.e., $V_{dz,a,i}/V_{pz} > 1$, which means the design primary airflow may not be sufficient to bring in enough ventilation air to the zones. Theoretically, in such cases $V_{pz,i}$ should be increased, but that would require changing the sizes of VAV boxes, supply fans, and heating and cooling equipment after they are calculated through the EnergyPlus sizing simulations. Developing an automatic process to change $V_{pz,i}$ and its related modeling inputs is very challenging because it may need to override EnergyPlus' sizing results, on which the implementation of many Standard 90.1 equipment efficiency requirements is dependent. We decided not to change $V_{pz,i}$, but to leave the dampers fully open in the critical zones at all times for more ventilation air. Therefore, the $MDP_{a,i}$ is taken as the smaller of $V_{dz,a,i}/V_{pz}$ and 1.

The calculation for system ventilation efficiency and system outdoor air intake flow remains the same as before and it is repeated here

$$E_{vz,a,i} = 1 + X_s - Z_{d,a,i}$$

$$E_{v,a} = \min(E_{vz,a,i})$$

$$V_{ot,a} = V_{o,u}/E_{v,a}$$

The enhancements affected all prototypes with multiple-zone VAV systems, i.e., Medium Office, Large Office, Primary School, Secondary School, Outpatient Healthcare, and Hospital.

2.2.7 Kitchen Exhaust Fan Modeling Strategy

Six prototype buildings (Large Hotel, Quick Service Restaurant, Hospital, Full-Service Restaurant, Primary School, and Secondary School) have kitchen zones and transfer air from the adjacent zones to replace kitchen exhaust airflow. The models use the ZoneMixing object in EnergyPlus, which only affects the energy balance of the kitchen zones and does not affect the energy or airflow balance in the "source" zones or the airflow balance of the kitchen zones. However, to capture both the air and the energy balance during the process, the models were developed with "dummy" kitchen exhaust fans.

Before the model enhancements, kitchen exhaust fan objects couldn't be modeled using the actual flow rate inputs because it would cause the air-handling unit (AHU) serving the kitchen zone to bring in excessive outdoor airflow to balance the exhaust flow rate regardless of the existence of transfer air. Thus, previously the energy usage of the actual kitchen exhaust fans was modeled as a plug load and the Fan:ZoneExhaust objects had zero pressure rise and 100% total fan efficiency. Though this approach effectively accounted for the fan electric energy consumption, it caused issues in reporting of regulated and unregulated energy end uses. EnergyPlus V8.0 provided a new input parameter, "Balanced Exhaust Fraction Schedule Name," under the FanZoneExhaust object, which considers a specified fraction of the exhaust air flow to be balanced by transfer air. In addition to this new capability, implementation of addendum aj to Standard 90.1-2010 (motor efficiency for motors between 1/12 and 1 hp) required a modification to the exhaust fan modeling approach to effectively capture the impact of the addendum. Therefore, as an improvement to the exhaust fan modeling strategy for kitchens, the actual exhaust fans are modeled with an assumed static pressure rise of 0.5 in. w.g. The fans are assumed to be small centrifugal fans or similar fans with 0.55% fan mechanical efficiency. The "Balanced Exhaust Fraction Schedule Name" inputs are added. The modeled motor efficiency is documented in Section 2.4.2 of this document.

2.2.8 Warehouse Roof Solar Reflectance and Emittance

The solar reflectance and emissivity of the Warehouse prototype roof was modified to be consistent with other prototypes not required to have a high solar reflectance and emittance (cool roof). The changes are shown in Table 2.13.

Table 2.13. Warehouse Roof Solar Reflectance and Emittance (all Standards)

Roof Surface Property	Before Enhancement	After Enhancement
Solar Reflectance	0.3	0.23
Thermal Emittance	0.7	0.9

2.2.9 Demand Controlled Ventilation Enhancement

The structure and functionality of the mechanical ventilation controller object (Controller:MechanicalVentilation) was improved in EnergyPlus version 8.0. In the previous versions of EnergyPlus, it was not possible to separate the control of demand controlled ventilation (DCV) and dynamic ventilation reset in multi-zone systems. With the new features in EnergyPlus version 8.0, it is now possible to enable or disable DCV regardless of whether dynamic reset of ventilation is required. This change affects only the multi-zone VAV system in the Large Hotel prototype.

DCV was not working correctly in the high occupancy spaces in the Secondary School prototype. When DCV is required, the minimum OA flow rate needed to be set to zero in the Controller:OutdoorAir object. This rule was applied to systems serving the Gymnasium, Auxiliary Gymnasium, and Auditorium spaces in the Secondary School prototype.

2.2.10 Large Hotel: Dedicated Outdoor Air System

This enhancement improves the modeling strategy for the dedicated outdoor air system (DOAS) used in the Large Hotel prototype. Since this particular enhancement creates a large change in baseline energy use, the energy impacts for this enhancement are shown in Table 2.14. The following modifications were made as a part of this enhancement:

- The Large Hotel prototype was previously modeled with two dedicated outdoor air systems, one serving floor 6 of the model and one serving floor 3. The prototype has been modified to combine the two separate systems. This has minimal energy use impacts.
- The second part of this enhancement corrects the zone and system outdoor air rates used. This improvement results in over 2% whole building EUI increase in the Standard 90.1-2004 models and a 0.5% whole building EUI increase in the Standard 90.1-2010 model as shown in Table 2.14.

Table 2.14. Weighted Site EUI Impact for Large Hotel Prototype DOAS and Ventilation Enhancements

	90.1-2004 EUI kBtu/ft^2	90.1-2010 EUI kBtu/ft^2
Before Enhancement	120.78	96.13
After Enhancement	123.35	96.42
Percentage Change	2.13%	0.30%

2.3 Simulation Infrastructure Updates

EnergyPlus has been under continuous development by the U.S. Department of Energy (DOE) since 1996 (DOE 2013), with new versions released periodically. During development of Standard 90.1-2013 (ASHRAE 2013), the prototype models were updated from EnergyPlus version 6.0 to version 8.0 and the weather files were updated from typical meteorological year (TMY) 2 to TMY3. These enhancements are discussed in the sections below.

2.3.1 Use of TMY3 Weather Files

Location-specific TMY weather files are used in the simulation to represent average weather for each representative city. TMY2 data representing a 30-year average from 1961 through 1990 was replaced with TMY3 data representing a 30-year average from 1975 through 2005. Weather data for each climate zone was used from the same location as previously. Since this enhancement created a large change in energy use, Table 2.15 shows the impact of the switch to TMY3 on the energy use of the prototype building models.

Table 2.15. National Average EUI Impact for Upgrade from TMY2 to TMY3 Weather Files

	90.1-2004 kBtu/ft^2			90.1-2010 kBtu/ft^2		
Prototype Name	TMY2	TMY3	% Change	TMY2	TMY3	% Change
Small Office	42.38	42.37	-0.02%	33.01	32.99	-0.06%
Medium Office	49.90	49.49	-0.82%	36.88	36.59	-0.79%
Large Office	85.16	84.53	-0.74%	71.96	71.63	-0.46%
Standalone Retail	80.42	79.44	-1.22%	53.55	53.23	-0.60%
Strip Mall	81.86	81.16	-0.86%	57.84	57.28	-0.97%
Primary School	81.66	81.28	-0.47%	57.42	57.14	-0.49%
Secondary School	72.79	72.53	-0.36%	48.80	48.97	0.35%
Outpatient Healthcare	157.42	157.13	-0.18%	119.45	119.24	-0.18%
Hospital	171.52	170.67	-0.50%	133.05	132.22	-0.62%
Small Hotel	73.68	73.34	-0.46%	63.21	63.15	-0.09%
Large Hotel	121.64	120.78	-0.71%	96.77	96.15	-0.64%

Table 2.15 (continued)

	90.1-2004 kBtu/ft^2			90.1-2010 kBtu/ft^2		
Prototype Name	TMY2	TMY3	% Change	TMY2	TMY3	% Change
Warehouse	25.82	25.55	-1.05%	18.47	18.24	-1.25%
Fast Food Restaurant	662.37	657.87	-0.68%	609.84	605.78	-0.67%
Sit-Down Restaurant	478.06	475.34	-0.57%	395.74	392.51	-0.82%
Mid-Rise Apartment	51.78	51.51	-0.52%	45.73	45.46	-0.59%
High-Rise Apartment	55.27	55.20	-0.13%	50.12	50.05	-0.14%
National Weighted Average	76.94	76.45	-0.64%	58.65	58.37	-0.48%

2.3.2 EnergyPlus V8.0 Update

Modifications to versions of the EnergyPlus simulation engine include changes to syntax used to describe modeling parameters, addition of components and controls strategies, new output capabilities, and updates to the algorithms and calculations used to generate simulation results (DOE 2013). These updates have a modest impact on the energy use results as shown in Table 2.16.

Table 2.16. National Average EUI Impact for EnergyPlus Version Upgrade from V6.0 to V8.0

	90.1-2004 kBtu/ft^2			90.1-2010 kBtu/ft^2		
Prototype Name	V6	V8	% Change	V6	V8	% Change
Small Office	42.41	42.37	-0.09%	33.04	32.99	-0.15%
Medium Office	49.98	49.49	-0.98%	36.86	36.59	-0.73%
Large Office	86.52	84.53	-2.30%	73.61	71.63	-2.69%
Standalone Retail	80.49	79.44	-1.30%	55.06	53.23	-3.32%
Strip Mall	81.96	81.16	-0.98%	57.90	57.28	-1.07%
Primary School	79.59	78.73	-1.08%	56.65	55.73	-1.62%
Secondary School	72.86	70.50	-3.24%	49.67	47.30	-4.77%
Outpatient Healthcare	158.39	157.13	-0.80%	120.00	119.24	-0.63%
Hospital	175.59	170.67	-2.80%	137.00	132.22	-3.49%
Small Hotel	73.41	73.34	-0.10%	62.95	63.15	0.32%
Large Hotel	120.92	120.78	-0.12%	96.15	96.15	0.00%
Warehouse	25.93	25.55	-1.47%	18.62	18.24	-2.04%
Fast Food Restaurant	654.41	657.87	0.53%	600.18	605.78	0.93%
Sit-Down Restaurant	478.72	475.34	-0.71%	394.39	392.51	-0.48%
Mid-Rise Apartment	51.76	51.51	-0.48%	45.73	45.46	-0.59%
High-Rise Apartment	58.31	55.20	-5.33%	53.16	50.05	-5.85%
National Weighted Average	77.30	76.12	-1.53%	59.35	58.12	-2.07%

2.4 Enhancements to Provide More Detail

During the process of evaluating new requirements in Standard 90.1-2013, it sometimes became apparent that the prototype models did not contain sufficient detail to capture some of the nuances of the addenda. Adding details to the baseline models (Standard 90.1-2004, 2007, and 2010 compliant models) allowed for the capture of the resultant savings from changes to Standard 90.1-2013. Those enhancements are described below.

2.4.1 Enhancements to Steam Humidification System and Preheat Coil Controls

Evaluation of addendum as to Standard 90.1-2010 identified two specific clauses that required changes to the baseline models.

- Section 6.5.2.4, as revised for Standard 90.1-2013, requires humidification system dispersion tube hot surfaces in the air stream of ducts or AHUs to be insulated with a product with an insulating value of at least R-0.5. Prior to this enhancement, heat gain to the airstream from these hot surfaces was not accounted for in the simulation.
- Section 6.5.2.5, new in Standard 90.1-2013, requires preheat coils to have controls that prevent their heat output whenever mechanical cooling, including economizer operation, is on.

To capture savings from both of these requirements, the baseline models had to be altered.

According to Wasner and Lundgreen (2007), the heat gain from a typical steam dispersion assembly results in a temperature rise of the airstream of 2.58°F if the steam dispersion tubes are not insulated. To simulate the impact of heat gain from the steam dispersion tubes, an electric heating coil has been added to the affected AHUs, which is simulated to be on during humidifier operation. For the baseline model (Standard 90.1-2010) the electric coil causes a supply air temperature rise of 2.58°F. This impacts the Large Hospital, Outpatient Healthcare, and Large Office baseline models.

Section 6.5.2.5 in Standard 90.1-2013 intends to avoid/reduce the uncontrolled heat transfer from a preheat coil to the bypass air when AHUs are in the cooling mode, including economizing mode. This happens with systems using a vertical integral face-and-bypass (VIFB) type steam preheating coil as typically found in healthcare facilities in cold climates. With these systems, the steam coil valve is fully open below a fixed outdoor air temperature, and instead of modulating the steam flow when preheat is not needed, the airstream goes through the bypass. Based on field measurements conducted by a member of the SSPC 90.1 MSC, it is assumed that due to air traveling across the face of the coil, there is 18% of peak coil design heat transfer to the airstream when no preheat is needed. Peak and minimum capacity for the preheat coils are shown in Table 2.17.

Table 2.17. Minimum Capacity for Preheat Coil Operation

Unit Type	Coil Type	Peak Capacity (Btu/h/cooling cfm)	Minimum Capacity (Btu/h/cooling cfm)
Operating Room AHUs	VIFB Steam	46.1	8.5
Other Medical Areas AHU	VIFB Steam	16.2	3.0
Non-medical AHU	Steam Distributing	9.8	0.0

To capture the impact of this uncontrolled heat transfer from the preheat coil to bypass air, a hot water coil has been added to each AHU serving medical areas in baseline models. For baseline healthcare prototype models (Outpatient Healthcare and Hospital), in climate zones 6 and above, the added hot water coils are simulated to be on at the minimum capacity in Table 2.17 when the outdoor air temperature is below 50°F, irrespective of the AHU operating mode.

2.4.2 Fan Motor Efficiency

Addendum aj to Standard 90.1-2010 requires motors from 1/12 hp to under 1 hp to be electronically commutated (EC) motors or have a minimum efficiency of 70%. The intent is to replace standard permanent-split capacitor (PSC) motors with more-efficient EC motors. Intended applications include toilet and elevator exhaust fans, series fan-powered VAV boxes, and fan-coil units. Exemptions to this requirement include motors in an airstream where only heating is provided, motors in packaged equipment, and capacitor-start capacitor-run, capacitor-start induction-run, and polyphase motors.

To capture savings from this addendum, changes to the baseline assumptions were required. Baseline motors were assumed to be PSC motors. Sources give a range with peak efficiencies as high as 65%, but this is very sensitive to the design load, and operating off the design load gives efficiencies in the range of 12% to 45% (Taylor Engineering 2011). Research presented to the California Energy Commission, considering EC motors for California Title 24, used 29% efficiency for PSC motors (Taylor Engineering 2011). The ENERGY STAR program uses a criterion for small exhaust fans of a minimum of 2.8 cfm/W tested at 0.25 in w.g., and requires a 60% efficient fan for rated airflow under 90 cfm, and 70% efficient fan for rated airflow from 90-500 cfm.[8] This implies a motor efficiency as low as 12%. A motor efficiency of 29% was used as an intermediate value between highest potential efficiency and lowest efficiency. The minimum required for EC motors as per the addendum is 70%. This is close to the average typical EC motor efficiency, and therefore this was the value used for the analysis.

The Primary School, Secondary School, Hospital, High-Rise Apartment, Mid-Rise Apartment, Small Hotel, Large Office, and Medium Office prototype buildings were affected. The fan motors impacted by the addendum are those associated with fan coil units, general exhaust fans, kitchen exhaust fans, and elevator exhaust fans. Table 2.18 shows the new values for fan efficiency and fan power before and after the enhancements.

Table 2.18. Enhancements to Fractional hp Fan Motor Efficiency

	Pre- Enhancement		Post-Enhancement	
Fan System Type	W/cfm	Fan motor eff.	W/cfm	Fan motor eff.
Fan-Coil Units	0.3	80%	0.8	29%
Exhaust Fans	0.06-0.37	11%-67%	0.15	29%
Kitchen Exhaust Fans	0.18	33%	0.37	29%
Elevator Fans	0.33	-	0.33	29%

2.4.3 Cooling Capacity and Economizer Control

Addendum aq to Standard 90.1-2010 introduced several new requirements related to DX cooling capacity control, air economizer integration, and fan air flow control:

[8] http://www.energystar.gov/ia/partners/prod_development/revisions/downloads/vent_fans/Vent_Fans_Draft_V3.0.pdf.

- For DX units ≥65,000 Btu/h (effective 1/1/2016), that control cooling capacity based on space temperature (usually serving a single zone), a minimum of 2 stages of mechanical cooling capacity is required.
- For DX units that control cooling capacity based on space temperature (usually serving a single zone), a minimum of 2 stages of fan control shall be used. Low or minimum speed shall not exceed 66% of full speed.
- DX cooling capacity control shall be interlocked with air economizer controls such that 100% outdoor air can be supplied when mechanical cooling is on and outdoor airflow is only reduced when the discharge air temperature is below 45°F.

Existing features within EnergyPlus were not sufficient to capture the impact of these requirements, specifically, the impact of staged DX units on fan speed control and on economizer effectiveness. In addition to the requirements of addendum aq, the existing economizer simulation in EnergyPlus was thought to be inadequate and required a new method of calculation. The main issue was that economizers were more optimistically simulated than they operate in actual practice, so the energy use was already lower than the new requirements would create.

Economizer Simulation Improvements

EnergyPlus overstates the reduction in DX cooling from economizers because the system simulation in EnergyPlus models a partial capacity of the coil during the time step, after accounting for full economizer benefit. This is how an economizer works with a hydronic coil operated by a mixed air control, as the hydronic cooling coil can match the remaining cooling load needed. While the simulation applies the full economizer benefit at all times, most DX roof-top unit economizers operate in many conditions with the economizer airflow reduced to avoid icing on the coil or discomfort. A single-stage DX cooling coil has a fairly steady temperature difference of about 20°F, so when the outside air temperature is lower than 65°F the economizer must reduce the outside air to avoid discharge temperatures below 45°F that can result in either discomfort or freezing coils or both. Because the simulation assumes the cooling capacity is adjusted to maintain the required supply air temperature rather than model the on and off operation of the DX coil with a varying discharge air temperature, the economizer savings is overstated. This actual operation results in less outside air being provided by the economizer than is simulated. Further, the default maximum outside air fraction for economizers is generally 100% outside air, while field measurements show that 70% is more typical (Davis et al. 2002).

To improve the economizer simulation, the following changes were made to prototype models:

1. The maximum OA fraction that the economizer is allowed to supply was reduced from 1.0 to 0.7. This is based on field measurements and it incorporates leakage from the return air dampers, allowing room for improvement in the future if leakage requirements are strengthened. The *M*aximum Fraction of Outdoor Air Schedule Name field in the Controller:OutdoorAir object was used to turn down the maximum OA fraction through a schedule. The same schedule is changed by an EnergyPlus energy management system (EMS) program to simulate economizer operation more accurately, as described below.
2. To improve the economizer simulation, a separate calculation was completed at a range of conditions and then a regression was developed to determine the economizer effectiveness as a function of cooling load and outside conditions. The fraction of time for which the system operates in full economizer, partial economizer, and full cooling mode was determined during the simulation run to

determine an effective average economizer fraction for a given time step. This fraction can be thought of as the economizer effectiveness for the time step, given the outdoor air conditions, return air temperature, the cooling load, and the available cooling capacity. The economizer effectiveness is adjusted by changing the maximum outside air schedule that controls the amount of outside air available at a time step. This simulates the limit the controller puts on the economizer to avoid low temperature discharge air and results in the correct amount of outside air being introduced during the time step, reducing mechanical cooling equal to the actual two mode operation.

The improved economizer simulation is controlled through the EMS within EnergyPlus. The EMS allows control of certain internal calculations within EnergyPlus and also allows the user to program new features that can be modeled in real time during the simulation. The improved simulation reduced economizer integration in the baseline and allowed the capture of savings from better economizer integration when staged cooling is required as per addendum aq to Standard 90.1-2010.

The prototypes and the systems where these changes were applied are given in Table 2.19. Only single-zone DX systems in Standard 90.1-2010 were affected by these changes.

Table 2.19. HVAC Systems Affected by Economizer Simulation Improvements

Building Prototype	HVAC Systems Affected
Standalone Retail	All systems
Strip Mall	All systems
Primary School	PSZ-AC_2:5, PSZ-AC_1:6, PSZ-AC_2:7
Secondary School	PSZ-AC_1:5, PSZ-AC_2:6, PSZ-AC_3:7, PSZ-AC_4:8, PSZ-AC_5:9
Quick-Service Restaurant	PSZ-AC_1:1, PSZ-AC_2:2
Full-Service Restaurant	PSZ-AC_1:1, PSZ-AC_2:2
Small Hotel	SAC_FRONTOFFICE, SAC_FRONTLOUNGE, SAC_MEETINGROOM, SAC_EXC_EMPLGE_RESTRM
Warehouse	PSZ-OFFICE, PSZ-FINE Storage

Fan Airflow Control Improvements

Standard 90.1-2010 has fan airflow control requirements for single zone systems. These requirements were modeled using a VAV fan (Thornton et al. 2011). However, this approach overstated savings from fan speed control for a multi-speed fan (Hart et al. 2013). Addendum aq added staging requirements in addition to fan airflow control requirements. An EMS program was written to model the new requirements as well as to improve the previous modeling of single zone fan airflow controls. In each time step, the compressor speed ratio is used to determine the percentage of time when the compressor runs at its rated speed. The DX coil runtime fraction is then used to determine the percentage of time in ventilation mode and economizing mode. The logic works as follows:

1. If the compressor speed ratio is greater than 0, the percentage of first-stage DX cooling is equal to 1 minus compressor speed ratio.
2. If the compressor speed ratio is equal to 0, the DX cooling coil runtime fraction is the percentage of time for first-stage DX cooling.
3. Next, if the DX coil runtime fraction is greater than 0, the percentage of time for ventilation mode is equal to 1 minus DX coil runtime fraction.

4. If the DX coil runtime fraction is equal to 0, the unit is in either ventilation mode or economizing mode for the whole time step.
5. To differentiate between ventilation and economizing mode, the current outdoor airflow is compared to the minimum. If it is higher than minimum, the unit is in the economizing mode; otherwise, it is in the ventilation mode.

Table 2.20 shows the HVAC systems affected by the change of approach to modeling the fan airflow control requirements in Standard 90.1-2010.

Table 2.20. HVAC Systems Affected by Fan Airflow Control Improvements

Building Prototype	HVAC Systems Affected
Standalone Retail	PSZ-AC:1, PSZ-AC:2
Strip Mall	PSZ-AC_1:1
Primary School	PSZ-AC_2:5, PSZ-AC_1:6
Secondary School	PSZ-AC_1:5, PSZ-AC_2:6, PSZ-AC_4:8
Full-Service Restaurant	PSZ-AC_1:1

The economizer improvements result in less outdoor air being supplied for cooling, resulting in higher energy consumption. The fan control changes result in a slightly higher fan energy use, as the time averaged fan power is greater than the flow averaged fan power that is the EnergyPlus default calculation.

2.4.4 Boiler Capacity Control

Addendum am to Standard 90.1-2010 requires minimum boiler turndown ratios as specified in Table 2.21. However, the baseline boilers did not model performance changes at part load conditions. To provide more accurate simulation and to capture savings from addendum am, the baseline models for prototypes with hot water boilers were modified to include a boiler part load efficiency curve.

The part load efficiency curve is determined by the minimum turndown ratio. The American Heating and Refrigeration Institute (AHRI) database for certified boilers[9] indicates 55% of boilers with capacity range greater than 1000 MBtu/h but less than 5000 MBtu/h have single stage boiler control and 80% of the certified products between 5000 MBtu/h and 10,000 MBtu/h, have single stage capacity control. The AHRI database doesn't have any certified boilers with capacity range larger than 10,000 MBtu/h. Therefore, the baseline models were assumed to have single stage capacity control.

[9] http://www.ahridirectory.org/ahridirectory/pages/cblr/defaultSearch.aspx

Table 2.21. Addendum am Boiler Minimum Turndown Ratio

Boiler System Design Input (Btu/h)	Minimum Turndown Ratio
≥ 1,000 MBTU/h and less than or equal to 5,000MBtu/h	3 to 1
> 5,000 MBtu/h and less than or equal to 10,000 MBtu/h	4 to 1
> 10,000 MBtu/h	5 to 1

Based on research by Bertagnolio and Andre (2010), single stage boilers are modeled with the following part load curve:

$$SingelStageCurve = 0.907 + 0.320 * PLR - 0.420 * PLR^2 + 0.193 * PLR^3$$

Where

PLR: Part load ratio

This enhancement affects all prototypes with boilers, including the Large Office, Hospital, Large Hotel, Outpatient Healthcare, Secondary School, and High-Rise Apartment prototypes. It results in a slight increase in the baseline heating energy use for all prototypes except Outpatient Healthcare. The reduction of energy use in the Outpatient Healthcare prototype is because a boiler curve from the EnergyPlus dataset was used prior to this enhancement and that curve represents a lower part load performance than the proposed curve. For a uniform implementation of this enhancement, the previous curve has been modified with the one mentioned above and results in lower EUI for the Standard 90.1-2004, 2007, and 2010 cases.

2.4.5 Setback and Optimum Start Controls

Optimum start controls in Standard 90.1 are defined as controls that are designed to automatically adjust the start time of an HVAC system each day with the intention of bringing the space to desired occupied temperature levels immediately before scheduled occupancy. The setback and optimal start control requirements in Standard 90.1 are the same in the 2004, 2007, and 2010 versions. Due to the absence of relative performance differences to be evaluated, the previously published prototype models did not address the optimal start requirement. Addendum cb to Standard 90.1-2010 added a few new requirements and clarifications, which triggered modifications of the modeling strategies to provide more detail. The following modifications are applied to all thermal zones that are not expected to be constantly occupied.

Cooling setback was disabled in all climate zones except for 1B, 2B, and 3B as allowed by Standard 90.1-2004, 2007, and 2010. Heating setback for non-constantly-occupied zones remained the same as before.

An EnergyPlus EMS program is used to detect the AHU supply fan size. When the fan size is large than 10,000 cfm, an optimum start control is implemented; otherwise, the non-optimum start control is used as allowed by Standard 90.1-2004, 2007, and 2010. The implementation has been carried out through thermostat schedule modifications to differentiate between optimum and non-optimum start controls.

- For non-optimum start control, the thermostat setpoint is set at the occupied setpoint for 2 full hours before the occupancy period starts.
- For optimum start control, the thermostat temperature setpoint ramps up (for heating) or down (for cooling) through two intermediate setpoints during 2 hours before the occupancy period starts. Each increment between the two adjacent setpoints is about one-third of the difference between the setback and occupied setpoints.

2.4.6 Water to Air Heat Pump Efficiency

Addendum h to Standard 90.1-2010 improves the minimum energy efficiency standards for water-to-air heat pumps. This required an examination of the way water-to-air heat pump efficiency was simulated. The High-Rise Apartment building includes water-loop heat pumps and is the only one of the 16 prototype buildings that is impacted.

In the Standard 90.1-2004, 2007, and 2010 models, the water-loop heat pump efficiency at peak capacity was input in the High-Rise Apartment building model. This efficiency, together with part load performance, was developed from ClimateMaster product catalog data (ClimateMaster 2009). The part load performance data was used to generate curve coefficients for input in the EnergyPlus models. The method used was to select the highest cooling or heating capacity and corresponding efficiency in the catalog data as the normalized values for coefficient generation. The heat pump working conditions at the highest capacities are very different from the standard-rated conditions, which are used to establish the code required efficiency values. Thus, the performance curve coefficients needed to be updated based on the standard conditions. Table 2.22 and Table 2.23 list the performance curve coefficients for cooling and heating before and after changing the normalized conditions.

Table 2.22. Performance Curve Coefficients for Water-to-Air Heat Pump (Cooling)

	Before Addendum h to 90.1-2010			After Addendum h to 90.1-2010		
	TotalCoolCap	SensCoolCap	CoolPower	TotalCoolCap	SensCoolCap	CoolPower
Coefficient 1	-3.431103402	4.969610285	-14.49180587	-4.302669873	6.001944481	-5.677759764
Coefficient 2	5.729871883	18.73769703	1.12046501	7.185369905	22.63006772	0.438988157
Coefficient 3	-1.785831488	-22.18715403	14.91937455	-2.239467145	-26.7960784	5.845277342
Coefficient 4	0.111637779	-1.427262757	0.361431607	0.139995928	-1.7237472	0.141605667
Coefficient 5	0.081864912	0.406253917	-0.43065797	0.10266018	0.490644802	-0.168727936
Coefficient 6		0.057390285			0.069311935	

Table 2.23. Performance Curve Coefficients for Water-to-Air Heat Pump (Heating)

	Before Addendum h to 90.1-2010		After Addendum h to 90.1-2010	
	TotalHeatCap	HeatPower	TotalHeatCap	HeatPower
Coefficient 1	0.204873159	-3.57324736	0.237847463	-3.791755292
Coefficient 2	-2.89266411	3.192752155	-3.358237961	3.387992395
Coefficient 3	3.278698626	1.415690223	3.806404674	1.502261208
Coefficient 4	0.154356726	-0.167415851	0.179200417	-0.177653511
Coefficient 5	0.110777566	-0.097139669	0.128607198	-0.103079864

2.5 Impact of Enhancements on Energy Use

Table 2.24 and Table 2.25 show the combined impact of all enhancements on EUI for each of the Standard 90.1-2004 and 2010 prototype buildings with and without plug and process loads. The most significant difference occurs in the Large Office prototype, which has a significant reduction in energy savings due to a doubling in energy use from the addition of the data center. Table 2.26 shows the combined impact of all enhancements on the national weighted energy and energy cost savings between Standard 90.1- 2004 and 90.1-2010, both with and without plug and process loads.

Table 2.24. EUI Impact for all Prototypes (with Plug and Process Loads) due to Model Enhancements

		Before Enhancements		After Enhancements		Energy Savings (%)	
Prototype Name		2004 kBtu/ft^2	2010 kBtu/ft^2	2004 kBtu/ft^2	2010 kBtu/ft^2	Before Enhance-ments	After Enhance-ments
Office	Small Office	41.31	32.80	42.37	33.02	20.6%	22.1%
	Medium Office	51.62	37.34	49.49	36.79	27.7%	25.7%
	Large Office	45.99	33.35	84.54	71.88	27.5%	15.0%
Retail	Standalone Retail	75.98	49.53	79.52	53.35	34.8%	32.9%
	Strip Mall	80.40	56.90	83.66	60.40	29.2%	27.8%
Education	Primary School	73.41	50.22	80.08	60.10	31.6%	24.9%
	Secondary School	66.18	41.19	72.94	48.01	37.8%	34.2%
Healthcare	Outpatient Healthcare	163.29	123.61	157.43	120.23	24.3%	23.6%
	Hospital	157.44	118.43	170.45	131.26	24.8%	23.0%
Lodging	Small Hotel	78.52	66.62	73.34	63.62	15.2%	13.2%
	Large Hotel	163.90	125.93	123.47	96.85	23.2%	21.6%
Warehouse	Warehouse	26.28	18.99	25.54	18.23	27.7%	28.6%
Food Service	Fast Food Restaurant	570.07	519.91	653.62	604.35	8.8%	7.5%
	Sit-Down Restaurant	409.65	330.88	471.20	389.14	19.2%	17.4%
Apartment	Mid-Rise Apartment	46.99	41.19	52.12	46.34	12.3%	11.1%
	High-Rise Apartment	48.93	43.97	55.29	50.41	10.1%	8.8%
Totals		73.94	55.0	76.73	58.75		
National Weighted Average						25.6%	23.4%

Table 2.25. EUI Impact for all Prototypes (without Plug and Process Loads) due to Model Enhancements

Prototype Name		Before Enhancements 2004 kBtu/ft²	Before Enhancements 2010 kBtu/ft²	After Enhancements 2004 kBtu/ft²	After Enhancements 2010 kBtu/ft²	Energy Savings (%) Before Enhance-ments	Energy Savings (%) After Enhance-ments
Office	Small Office	32.21	24.36	33.27	24.54	24.4%	26.2%
	Medium Office	36.60	23.85	34.47	23.09	34.8%	33.0%
	Large Office	30.37	19.25	41.01	29.55	36.6%	28.0%
Retail	Standalone Retail	68.49	42.06	72.02	45.88	38.6%	36.3%
	Strip Mall	74.97	51.47	78.23	55.01	31.3%	29.7%
Education	Primary School	52.10	29.29	57.12	37.56	43.8%	34.2%
	Secondary School	51.75	27.14	57.69	33.00	47.6%	42.8%
Healthcare	Outpatient Healthcare	116.01	77.16	110.15	73.41	33.5%	33.4%
	Hospital	107.86	69.45	120.18	81.98	35.6%	31.8%
Lodging	Small Hotel	56.06	44.40	50.89	41.41	20.8%	18.6%
	Large Hotel	128.47	90.91	87.38	61.18	29.2%	30.0%
Warehouse	Warehouse	23.75	16.55	23.01	15.73	30.3%	31.6%
Food Service	Fast Food Restaurant	300.63	250.61	343.11	293.99	16.6%	14.3%
	Sit-Down Restaurant	256.29	178.07	299.87	218.01	30.5%	27.3%
Apartment	Mid-Rise Apartment	32.44	26.82	37.57	31.97	17.3%	14.9%
	High-Rise Apartment	35.73	31.02	42.10	37.46	13.2%	11.0%
Totals		56.81	38.24	58.09	40.44		
National Weighted Average						32.7%	30.4%

Table 2.26. National Average Savings Impact due to Model Enhancements

Standard 90.1-2010 compared to 90.1-2004 National-weighted Energy Savings	With Plug and Process Loads		Without Plug and Process Loads	
	Pre-Enhancements	Post-Enhancements	Pre-Enhancements	Post-Enhancements
Site Energy	25.62%	23.43%	32.68%	30.39%
Energy Cost	23.16%	22.10%	29.47%	29.26%

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Appendix A

Service Water Heating

Appendix A

Service Water Heating

Service water heating (SWH) includes water heating uses such as restroom sinks in all prototypes as well as prototype-specific uses such as kitchens and laundry facilities. Details of the SWH in the prototype models were previously described in *Analysis of 90.1-2010* (Thornton et al. 2011). Pacific Northwest National Laboratory (PNNL) reviewed all SWH modeling inputs in the prototype building models and made enhancements as necessary. This review included working with an engineering consulting firm, which examined the SWH assumptions in reference to typical design practice, the *2011 ASHRAE Handbook: HVAC Applications* (ASHRAE 2011a, Chapter 50, Service Water Heating) and specific existing building projects. PNNL considered these and other sources to develop a revised set of SWH model inputs.

This review resulted in numerous changes and additions to the SWH model inputs. The description of these is organized into four sections:

- main SWH equipment
- dedicated water heating equipment, kitchen and laundry
- pumping
- pipe losses

A.1 Main Service Water Heating Equipment

The prototypes typically include one or more water heaters providing general SWH. To model water heaters in EnergyPlus the following input parameters are required:

- water heater type – natural gas or electric
- water heater efficiency
- peak usage (defined at water heater level, or in most cases at each zone)
- storage volume
- maximum heating capacity
- hot water usage schedule
- make-up water inlet temperature
- hot water supply temperature
- location of water heater
- storage tank losses
- pilot light loss

A.1.1 Water Heater Type

All modeled water heaters are electric or natural gas storage type water heaters. Potential exists to change to on-demand or other types in the future with possible code changes. The Small Office and Strip Mall water heaters were changed from natural gas to electricity during the model enhancement. Electric water heaters are substantially cheaper to purchase and install and are often used in low-use SWH applications. The High-Rise Apartment was changed from individual electric water heaters in each apartment to a central natural gas water heating system reflecting a common configuration in large multi-family buildings. The Mid-Rise Apartment water heaters remain individual electric water heaters in each apartment.

A.1.2 Water Heater Efficiency

The performance requirements in ANSI/ASHRAE/IES Standard 90.1 are specified in energy factor (EF), thermal efficiency (E_t), and standby loss (SL), which vary with the type, capacity, and storage volume of the water heater. In EnergyPlus, water heater efficiency (E_h_t) input is described as the thermal conversion efficiency from fuel energy to heat energy for the heater element or burner. EnergyPlus does not allow a separate input for a performance curve for the water heater so this efficiency value input reflects the average efficiency. Another related EnergyPlus input is the skin loss coefficient "UA" (W/k) (the UA inputs for on-cycle and off-cycle are assumed the same for each water heater). The calculation methods for E_h_t and UA for different heater types are described below. The calculation method was not changed except to accommodate changes in the size of the storage tanks, and in a few cases to incorporate the assumption that part of the storage was in external tanks only. The calculation of inputs required for by EnergyPlus for different types and sizes of storage water heaters were developed based on the test procedure for each class of equipment (ASHRAE 2010) and are as follows:

For electrical storage water heater with capacity equal or smaller than 12 kW:

$$E\text{-}h_t = 1$$

$$UA = \frac{41094 \times \left(\frac{1}{EF} - 1\right)}{24 \times 67.5}$$

For electrical storage water heater with capacity larger than 12 kW:

$$E\text{-}h_t = 1$$

$$UA = \frac{SL \times 1}{70}$$

For gas-fired storage water heater with capacity equal or smaller than 75,000 Btu/h:

$$E\_h_t = 0.82$$

UA is calculated by solving the following two equations together:

$$UA = \frac{\left(\frac{1}{EF} - \frac{1}{RE}\right)}{67.5 \cdot \left(\frac{24}{41094} - \frac{1}{RE \cdot P\_on}\right)}$$

$$0.82 = \frac{UA \cdot 67.5 + P\_on \cdot RE}{P\_on}$$

For gas-fired storage water heater with capacity larger than 75,000 Btu/h:

$$E\_h_t = \frac{UA \times 70 + P\_on \times E_t}{P\_on}$$

$$UA = \frac{SL \times E_t}{70}$$

Where,

EF is energy factor

E_t is thermal efficiency, defined in American National Standards Institute (ANSI) Z21.10.3 as the ratio of the heat content of water leaving the tank at a constant rate to the sum of the higher heating value of rate of fuel input to the tank plus direct electrical consumption of the water heater (ANSI 2011). For the thermal efficiency metric, the water temperature leaving the tank is 70°F higher than the temperature entering the tank. The thermal efficiency metric thus includes the effect of tank shell losses.

RE is recovery efficiency as measured in the DOE residential water heater test procedure[1] (DOE 2010), and it is calculated by summing the heat content of hot water removed during the first hour (first draw cycle) with the change in the heat content of stored water before and after the first draw cycle and dividing the sum by the total energy used during this first draw including any auxiliary energy. The resulting efficiency metric is similar to E_t, but in the residential test procedure, the average delivery temperature is allowed to change during the draw. In both E_t and RE, the efficiency metric accounts for any shell losses from a storage tank during the draw.

SL is the standby loss as defined in ANSI Z21.10.3 (ANSI 2011). It is the ratio of the average energy input to the tank on a per hour basis (measured during a 48 hour+ standby period, with no water draws, minus any change in stored hot water energy) to the heat content of the stored water in the tank. Thus, it includes the impact of burner efficiency and is not just a measure of shell conductive heat transfer. Standards and codes specify the SL limit for some water heater types. For some water heater types, rated water heater tank volume (V) is required to calculate SL. When the total storage volume is equal to or smaller than 100 gallons, this volume is used as V. When the total storage volume is larger than 100 gallons, this volume is split to a 100-gallon water heater tank for the SL limit calculation and a storage tank. Standards and codes do not specify SL for storage tanks. Assumptions for SL of the storage tank were made based on samples of commercial products and the SL used does not change for different standards or codes.

[1] Test procedures specified in 10 Code of Federal Regulations (CFR) 430 Appendix E to Subpart B.

When the total storage volume is larger than 100 gallon, the total SL is the sum of SL from the water heater tank and the storage tank.

P_on is the nameplate input capacity (assumed to be 75,000 Btu/h for gas-fired storage water heaters)

A.1.3 Peak Usage

Hot water usage is determined by peak usage multiplied by the hourly schedule for each hour over the course of the year. Peak usage values were reviewed by considering the resulting total gallons per day during weekdays. Changes were made where usage was considerably different from the usage for actual buildings (that had been monitored by the consultant) similar to the prototypes or when compared to the average daily usage in the *2011 ASHRAE Handbook: HVAC Applications* (ASHRAE 2011a) referred to here as *HVAC Applications*. For most water heaters, usage is entered for each zone under the EnergyPlus water usage equipment object and the flow rate is for the hot water, not the mixed water at the tap. In some cases, a single value is entered with the water heater object.

A.1.4 Total Storage Volume and Water Heater Capacity

In most cases, the tank storage volume and water heater capacity were changed according to recommendations from the engineering consultant. The consultant provided values that are consistent with available water heating equipment that has the capacity to meet the peak load and usage demands. The consultant reviewed data from actual buildings similar to the prototypes, and the *HVAC Applications* (ASHRAE 2011a) to determine usage and peak load. Storage volume was generally related to the capacity in a ratio of 1 gallon of storage to 1 kBtu/h of capacity, a common ratio in available water heaters. PNNL also compared the recommended peak capacity and storage values to those in the *HVAC Applications*, Chapter 50, Figures 16 to 23, which graphically represent a range of values relating recovery capacity and storage capacity. At lower storage capacity, higher recovery capacity is needed to meet the peak load; no storage capacity means a water heater is an instantaneous water heater with recovery capacity equal to the highest peak load.

A.1.5 Hot Water Usage Schedule

Modeled energy usage for service hot water is primarily driven by assigning a peak usage multiplied by an hourly operating schedule as described above. The SWH schedules were reviewed and compared to actual building monitored usage (where this information was available) and relative to daily usage estimated from other sources such as *HVAC Applications*. The consultant identified that the schedules used were reasonable profiles of typical usage. PNNL also reviewed the modeled schedules against default modeling schedules published in the ASHRAE Standard 90.1 User’s Manual (ASHRAE 2011b). In many cases, for example the Medium Office, retail, and school prototype buildings, the schedules were modified to fit the building operating hours where necessary. During the review, schedules were changed to zero values in unoccupied building hours (if not already at zero) because off-hour losses are covered by “dump” losses described below.

A.1.6 Make-up Water Inlet Temperature

The make-up water inlet temperature comes from the Site:WaterMainsTemperature input and was unchanged during the review. The water temperature is calculated using the correlation option from EnergyPlus, which calculates the water main temperature from the annual average outdoor air temperature and the maximum difference in monthly average outdoor air temperature.

A.1.7 Hot Water Supply Temperature

The water heater model in EnergyPlus requires an input for tank setpoint temperature and changes were made to some prototype buildings so that all main water heaters use a setpoint of 140°F to avoid conditions that may increase the chance of Legionnaire's disease (ASHRAE 2000). It is acknowledged that some setpoint temperatures in practice may be kept lower than that.

When specifying water temperature at the tap, the hot water temperature (140°F) is used instead of mixed water temperature. This is consistent with the water flow rate input, which represent the hot water load and not mixed water use.

A.1.8 Location of Water Heater

The review identified that the water heaters did not have defined locations in the prototype buildings. As a result, storage tank losses were not affecting space loads. Except the High-rise Apartment, water heaters in all prototypes were assigned to an appropriate zone during the enhancement. The High-rise Apartment prototype was modified to include a central water heater instead of individual apartment water heaters, and there is no appropriate zone to include the central water heater in. Presumably, this equipment would be in a mechanical penthouse or basement—such a zone could be added to the model in the future if desired, although there would be no impact on building energy use, as a mechanical room is typically unconditioned.

A.1.9 Storage Tank Losses

Storage tank loss inputs are modeled to match the Standard 90.1 allowed losses. The calculation method is the same as described earlier in the calculation of water heater efficiency (Section A.1).

A.1.10 Pilot Light Loss

Many of the water heaters included a loss input to account for a continuously operating pilot light. Pilot light losses were removed because new commercial water heaters typically use an electronic ignition and do not have pilot lights.

A.2 Dedicated Water Heating Equipment for Kitchen and Laundry

For the kitchens, booster heaters were added to account for the need for 180°F water for dishwasher sterilization, instead of just the 140°F provided by the already defined main water heaters. Chemicals are sometimes used instead of a booster heater, but it is assumed that hot water is used, except for the Quick Service Restaurant, which is assumed to use chemicals. Thus, the Quick-service Restaurant prototype does not have a booster heater. Booster heaters were defined for the Primary School, Secondary School, Hospital, Large Hotel, and Full Service Restaurant prototypes. These are defined as electric on-demand water heaters meeting Standard 90.1 efficiency and heat loss requirements for electric water heaters. The calculation method for efficiency and tank heat loss is the same as for the main water heaters. Dishwashing is estimated to use 60% of the kitchen hot water based on an example in *HVAC Applications* (ASHRAE 2011a, pg. 50.21). The booster heater capacity is defined based on 60% of the peak flow requirements for the kitchen load with a 40°F temperature difference and the same operating schedule as the main water heater.

Laundry service is included in the Hospital, Small Hotel, and Large Hotel prototype buildings. Commonly, dedicated water heaters are used for this type of load, which also requires 180°F water. These loads were separated out or newly defined for these prototypes with a separate water heater from the main water heater. These are gas storage water heaters with Standard 90.1 efficiency. The calculation method for efficiency and tank heat loss is the same as for the main water heaters. Laundry loads were defined from a laundry sizing calculation method from a laundry equipment manufacturer (CLEC 2012) based on the number of beds for the hospital and rooms in the hotels.

Table A.1 summarizes the main water heaters and the dishwasher booster and laundry water heaters.

Table A.1. Water Heating Equipment

Prototype Building	Type	Water Heater Energy Type	Storage Capacity (gal)	Heating Capacity, (kBtu/h)	Total Peak Flow (gpm)
Small Office	Main	Electric[c]	40	12	0.06
Medium Office	Main	Gas	100	100	0.85
Large Office	Main	Gas	300	300	6.97
Standalone Retail	Main	Gas	40	40	0.30
Strip Mall	Main, each[a]	Electric[c]	40	12	0.03
Primary School	Main	Gas	200	200	1.67
	Dishwasher (DW) Booster	Electric[c]	6	6	1.00
Secondary School	Main	Gas	600	600	7.63
	DW Booster	Electric[c]	6	19	23.7
Outpatient Healthcare	Main	Gas	200	200	1.00
Hospital	Main	Gas	600	600	2.14
	DW Booster	Electric[c]	6	3	0.58
	Laundry	Gas	300	300	2.8
Small Hotel	Main	Gas	300	300	2.85
	Laundry	Gas	200	200	2.05
Large Hotel	Main	Gas	600	600	5.94
	DW Booster	Electric[c]	6	8	1.33
	Laundry	Gas	300	300	30.6
Warehouse	Main	Electric[c]	20	6	0.13
Quick-service Restaurant	Main	Gas	100	100	1.52
Full-service Restaurant	Main	Gas	200	200	2.22
	DW Booster	Electric[c]	6	8	1.33
Mid-Rise Apartment	Main, per apartment[b]	Electric[c]	50	15	0.06
High-Rise Apartment	Main	Gas	600	600	4.58

(a) There are 7 water heaters, each serving one of the seven model zones.
(b) The Mid-Rise Apartment includes 23 separate water heaters. 15 serve one apartment each on the ground and top floors. Eight serve two apartments each; these apartments are modeled as one apartment each with a multiplier of two, but don't have a zone water use input, so water heater inputs must each account for 2 apartments.
(c) Electric water heater capacity is expressed in kilowatts.

A.3 Pumping

Many SWH systems include pumps and provide continuous circulation of water. This allows the system to quickly deliver hot water at the tap. The energy use of these pumps is accounted for in the simulations. The review included consideration of whether or not each prototype should have (or continue to have) a circulating system based on the size of the system and typical practice, and for those with pumps, a common method of calculating the pump power.

The Small Office and Quick Service Restaurant prototypes were changed to non-circulating systems. Standalone Retail, Strip Mall, Warehouse and Mid-Rise Apartment prototype buildings all remain non-circulating systems. The High-Rise Apartment, which is changed to a central system as described above, is changed to a circulating system with a pump. Other prototypes remain circulating systems.

The pump power is modeled in EnergyPlus with three inputs: pump head, motor efficiency, and circulation flow. The circulation flow is the design flow in EnergyPlus. However, circulation flow in real design is typically much less than the design flow needed to meet the peak demand for hot water at the tap or appliance. Circulation flow is instead determined to allow reasonably hot water to be delivered at the tap despite the temperature loss that occurs in the piping. The pump head is modeled to allow the pump power to reflect a calculated recirculation flow when the model uses the design flow. The following items are considered in developing the pumping inputs and adjusted values:

- length of pipe
- diameter of pipe
- circulation flow rate
- pipe friction and calculated pump head
- motor efficiency
- calculated pump power
- adjusted pump head
- pump control

A.3.1 Pipe Length

Pipe length is used to estimate the pump head and pipe losses. The pipe length is estimated using the formula below (Sezgen and Koomey 1995). The formula is related to the perimeter of each floor and 10 feet of length of riser per floor of the building other than the top floor. This length is assumed to provide an estimate of the effective pipe length including valves and fittings for purposes of applying pipe friction loss factors. A minimum effective length of ten feet is used to account for the minimum required pipe and fittings even if the system is near the water heater.

$$\textit{Pipe length} = 2\cdot[(\textit{floor area}/\textit{number of floors})^{1/2} + 10\cdot(\textit{number of floors}-1)]$$

A.3.2 Pipe Diameter

For simplicity, all pipe is assumed to be 0.75-inch copper tubing. The need for larger diameter pipe for larger flows is neglected, as the pumping head will likely be comparable. Design and layout of piping networks was beyond the scope and resources of the project.

A.3.3 Circulation Flow Rate

The circulation flow rate is calculated based on pipe thermal losses. The pipe thermal loss methodology is described in Section A.4. The flow is sized to ensure that the temperature drop from the tank to the tap does not exceed 20°F. The calculation results in a recirculation pump flow capacity equal to the pipe heat loss divided by the change in the heat content of the water in the pipe system with a 20°F temperature change. This calculation is from *HVAC Applications*, pg. 50.7, equation 9 (ASHRAE 2011a). A minimum flow of 1 gpm is used if the calculated value is less than 1 gpm. Losses were calculated for 1/2 inch and 1 inch thick insulation depending on the Standard 90.1 version applicable for each simulation. Resulting values with the different thicknesses of insulation were not substantially different and the pump energy is quite small; therefore, an average of the circulating flow for both cases was used, and the same pump circulation flow was used for all simulations with a given prototype to simplify the analysis.

A.3.4 Pipe Friction and Calculated Pump Head

Pipe head loss factor was estimated from pipe friction tables in the *ASHRAE Handbook – Fundamentals* (ASHRAE 2009, pg. 22.7, Fig. 5) based on the selected flow rate. The pump head is calculated as the head loss factor multiplied by the pipe length divided by 100 feet.

A.3.5 Motor Efficiency

All of the pumps identified are fractional horsepower. These are assumed to use permanent-split capacitor motors with 30% efficiency consistent with similar motors described in Section 2.4.2.

A.3.6 Pump Power

EnergyPlus does not have a direct input for pump power, instead it is calculated in the simulation based on the flow, head, pump efficiency, and motor efficiency. However, as mentioned previously, recirculation flow is not the same as design demand flow and cannot be entered separately without changing the hot water usage. In order to allow the simulated pump power to be consistent with the circulation flow instead of the design flow, the pump head input is adjusted as described below. The first step is to calculate pump power manually using the previously calculated circulation flow.

calculated break horsepower =

circulation flow (gpm) · pump head (ft) / [3,960 · pump impeller efficiency (0.78)]

The pump impeller efficiency is fixed internally by EnergyPlus at 78%.

A.3.7 Adjusted Pump Head

Since the flow used in EnergyPlus has to be the design flow to allow the correct hot water usage to be calculated, the pump head entered into EnergyPlus is adjusted from the manually calculated pipe head loss. The adjustment is made in proportion to the ratio of the circulation flow and the design flow.

adjusted head (ft) =

calculated head (ft) · circulation flow (gpm) / design flow (gpm)

A.3.8 Pump Control

The pump speed control input in seven of the prototypes was previously set to variable speed. This is changed to constant speed in all cases.

Table A.2 shows the pipe length, circulation flow rate, calculated pump head and calculated pump power for the prototype buildings.

Table A.2. Circulation Pump Parameters for Prototype Buildings

Prototype	Pipe Length (ft)	Flow Rate (gpm)	Pump Head (ft)	Pump Power (bhp)
Medium Office	308	1.00	10.0	0.0032
Large Office	632	2.07	10.0	0.0062
Primary School	544	1.00	10.0	0.0032
Secondary School	670	1.00	11.4	0.0037
Outpatient Healthcare	274	1.11	10.0	0.0033
Hospital	501	2.20	10.0	0.0066
Small Hotel	268	1.17	10.0	0.0035
Large Hotel	393	1.72	10.0	0.0051
Full-service Restaurant	148	1.00	10.0	0.0032
High-Rise Apartment	401	1.77	10.0	0.0053

A.4 Pipe Thermal Losses

Total pipe thermal losses include pipe losses when the building and the SWH are in use, and “dump” losses, which are once a day losses due to the heating up of water that has cooled down in the pipes during unoccupied hours when there is no usage and circulation pumps are turned off.

The analysis included calculated pipe losses for heating one complete volume of the water in the pipes per day, and the heat loss from the pipes through the pipe walls and any insulation during the occupied hours. Dump losses were totaled and computed as heat loss per hour. Pipe losses are also calculated as an hourly loss. These are added and modeled as occurring every hour during occupied hours using the on cycle parasitic loss input in the water heater object in EnergyPlus.

A.4.1 Dump Losses

Dump losses are the heating energy required for the volume of water in the pipes assumed to be dumped to the drain when the taps are first opened during occupied hours. Daily dump loss per foot of pipe per hour is calculated the same for all prototypes and climate zones as a simplification using an average make-up water temperature of 61°F. This is the volume of water in one foot of ¾ inch pipe multiplied by the heat capacity of water and a temperature difference of 79°F (140°F to 61°F) divided by 24 hours, resulting in a heat loss of 0.689 Btu/ft of pipe per hour.

A.4.2 Pipe Thermal Losses

Pipe thermal losses through the walls of the pipe and any insulation while the system is in use are calculated using heat loss values from *HVAC Applications* (ASHRAE 2011a, pg. 50.6, Table 1). Values are available per foot of 3/4-inch copper pipe with the insulation thickness and flow condition depending on whether the system is circulating or not circulating. Values for no insulation, 1/2-inch and 3/4-inch insulation are provided and values for 1 inch were extrapolated. The values are provided in the units Btu/h·ft·°F. To calculate the heat loss per foot of pipe, this was multiplied by a 70°F delta-T corresponding to 140°F supply water temperature and an ambient air temperature of 70°F. Table A.3

shows the pipe losses for different insulation thicknesses. All versions of Standard 90.1 require 0, 1/2, or 1 inch of insulation for this diameter pipe and water heating temperatures.

Table A.3. Pipe Losses by Insulation Thickness and Type of Flow

Insulation Thickness (in.)	Non-circulating (Btu/hr·ft)	Circulating (Btu/hr·ft)
0.0	28.07	30.80
0.5	12.25	17.50
1.0	11.27	16.10

The values for non-circulating systems were estimated with the heat loss values with flowing water 25% of the time, and stationary water 75% of the time. Circulating values assume flowing water all of the time. Non-circulating values assume that the temperature of the water in the pipes remains high as intermittent usage continues to refill the pipes so that the heat transfer remains similar to that at the 140° F supply temperature.

For non-circulating systems, Standard 90.1 requires insulation only for the first 8 feet of pipe near the exit of the water heater. The total heat loss input for the non-circulating systems include insulation loss factors for the first 8 feet and no insulation for the remaining pipe. Table A.4 shows the total pipe heat loss by prototype.

Table A.4. Total Pipe Thermal Loss For Prototype Buildings

Prototype	Non-Circulating[a] (Btu/h)	Circulating 1.0 in. Insulation (Btu/h)	Circulating 0.5 in. Insulation (Btu/h)
Small Office	572		
Medium Office		1,383	1,277
Large Office		3,001	2,771
Standalone Retail	1,862		
Strip Mall	174		
Primary School		843	784
Secondary School		943	879
Outpatient Healthcare		1,613	1,488
Hospital		3,193	2,938
Small Hotel		1,704	1,568
Large Hotel		2,501	2,301
Warehouse	483		
Quick-service Restaurant	722		
Full-service Restaurant		776	716
Mid-Rise Apartment	90		
High-Rise Apartment		2,562	2,357

(a) With 1/2 inch insulation for the first 8 feet of pipe. Losses are slightly less with 1.0 inch of insulation for first 8 feet of pipe.

A.5 References

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