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Title
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Permalink
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Publication Date
2013-08-01

CBE EnergyPlus Modeling Methods for UFAD Systems

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Revised 8-2013

STRATIFICATION MODELING: GAMMA-PHI FORMULATIONS

This report documents the UFAD module specifications for EnergyPlus v3.1 and higher. We have conducted extensive validation of the interior zone model for Price swirl and York MIT1 variable area (VA) diffusers. These correlations should not be considered to be applicable to other diffuser types. (See below for user procedures to support other diffuser brands and types.)

INTERIOR ZONE

We have shown [Bauman et. al. 2006; Webster et. al. 2008] that diffusion plots for interior zones show a good correlation between the parameters PHIoz-avg and Gamma as described in the following:

$$PHI = \frac{T_{oz-avg} - T_{SAT}}{T_R - T_{SAT}} \qquad \text{Equation 1}$$

Where,

T_R return air temperature [°F]

T_{oz-avg} occupied zone average temperature [°F], calculated as average temperature between heights of 4 inches and 67 inches in room

T_{SAT} diffuser supply air temperature [°F].

PHI represents the stratification in the room; lower values mean more stratification. [1] Bauman, et al. shows the definition of Gamma for interior zones derived from theory for point source thermal plumes.

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[1] The temperature difference between the upper and lower layers in the room is only an approximation to the actual stratification that we would expect. Stratification normally occurs primarily in the lower zone and is characterized by the difference between the temperature near the floor and one at head height. This defines the occupied zone (OZ) which is the region between 0.1m (4 inches) and 1.7 m (67 inches) from the floor. For transition heights fixed at 1.7 m (67 inches) we can obtain a rough estimate of the "real" stratification by assuming a linear profile where the average is the lower layer temperature; in this case the stratification will be two times the upper to lower temperature difference.

PHI is equal to the term Kc used in the Energyplus documentation. It is the ratio of the heat gain in the lower layer to that of the entire room.

$$\Gamma = \frac{\left(Q \cdot \cos \varphi \right)^{\frac{3}{2}}}{m \cdot \left(\frac{n}{m} \cdot A_{eff} \right)^{\frac{5}{4}} \cdot \left(0.0281 \cdot W \right)^{\frac{1}{2}}}$$
Equation 2

Q	total room airflow [m³/s]
$\cos(\varphi)$	discharge angle from vertical [°]
n	number of diffusers [1]
m	number of plumes (i.e., workstations) [1]
A_{eff}	effective diffuser area [m²]
W	room convective heat extraction (i.e., cooling load) [kW]
Γ	Gamma, a dimensionless number representing the ratio of momentum to buoyancy forces

Interior correlations

Figure 3 shows regressions from full scale laboratory testing of typical office workstation configurations for a wide range of operating conditions.
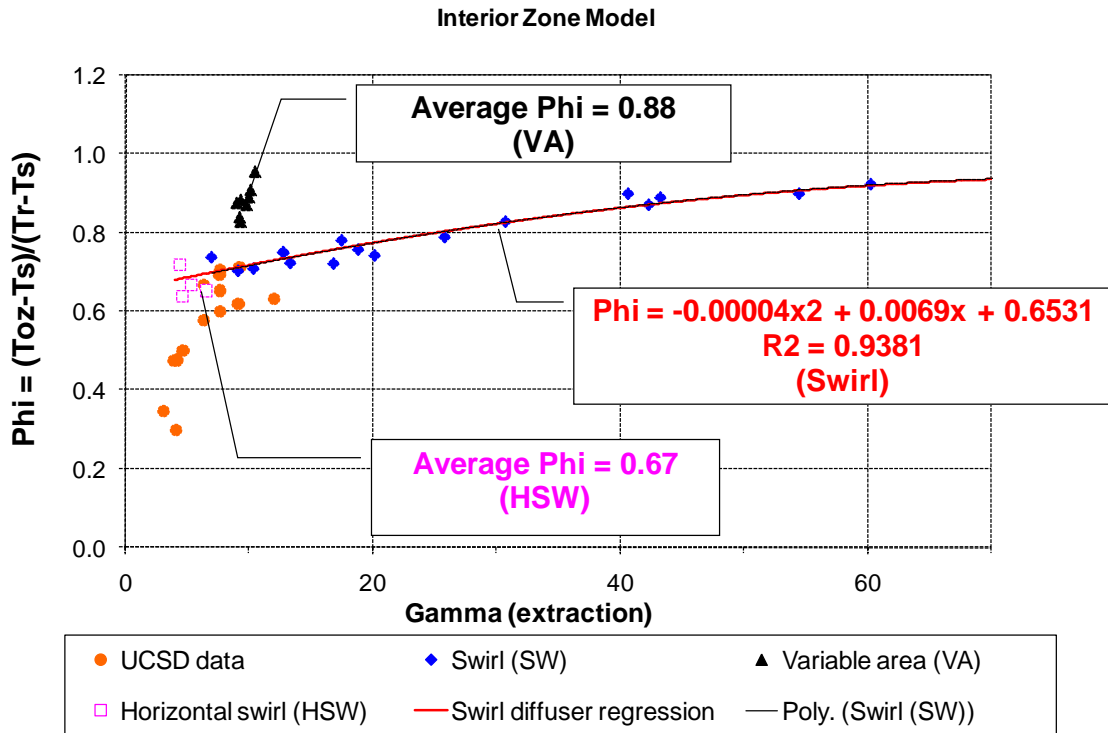

Figure 3. Interior zone correlations

PERIMETER ZONE

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For perimeter zones, Gamma is developed based on theory of line plumes generated by heat gain from exterior windows and walls. In this case a linearized formulation for Gamma is used:

$$\Gamma_L = \frac{(Q \cdot \cos \varphi)}{(n \cdot A_{eff}) \cdot (B_L)^{\frac{1}{3}}}$$

Equation 3

Where,

Q	total perimeter zone airflow [m³/s]
(φ)	discharge angle from vertical [°]
n	number of diffusers [1]
A_{eff}	effective diffuser area of single diffuser [m²]
B_L	buoyancy per unit length of zone [m³/s³]

Converting buoyancy to heat gain and following the strategy we used for interior zones of using the room extraction rate for the buoyancy term we have:

$$\Gamma_L = \frac{(Q \cdot \cos \varphi)}{(n \cdot A_{eff}) \cdot (0.0281 \cdot W_L)^{\frac{1}{3}}}$$

Equation 4

Where,

W_L zone extraction rate per unit length of zone [kW/m]

Perimeter correlations

Figure 4 shows (revised based on testing conducted in 2010) correlations for two tested diffuser types: York/JCI MIT2), and linear bar grilles (Titus and Walnut 6" wide linear). Note that based on previous testing EnergyPlus applies a consistent reduction of 0.2 Phi for all diffusers when blinds are down. Linear bar grille tests were conducted without interior zone diffusers, i.e., diffusers only in perimeter zone that included internal workstation loads as well as solar gains derived from a solar simulator consisting of an array of HID lamps. For these reasons, only the variable area and linear bar grilles are valid types to use in perimeter zones at the present time. **Note that the older correlations for York lab testing are no longer recommend to be used, see discussion under diffusers below.**
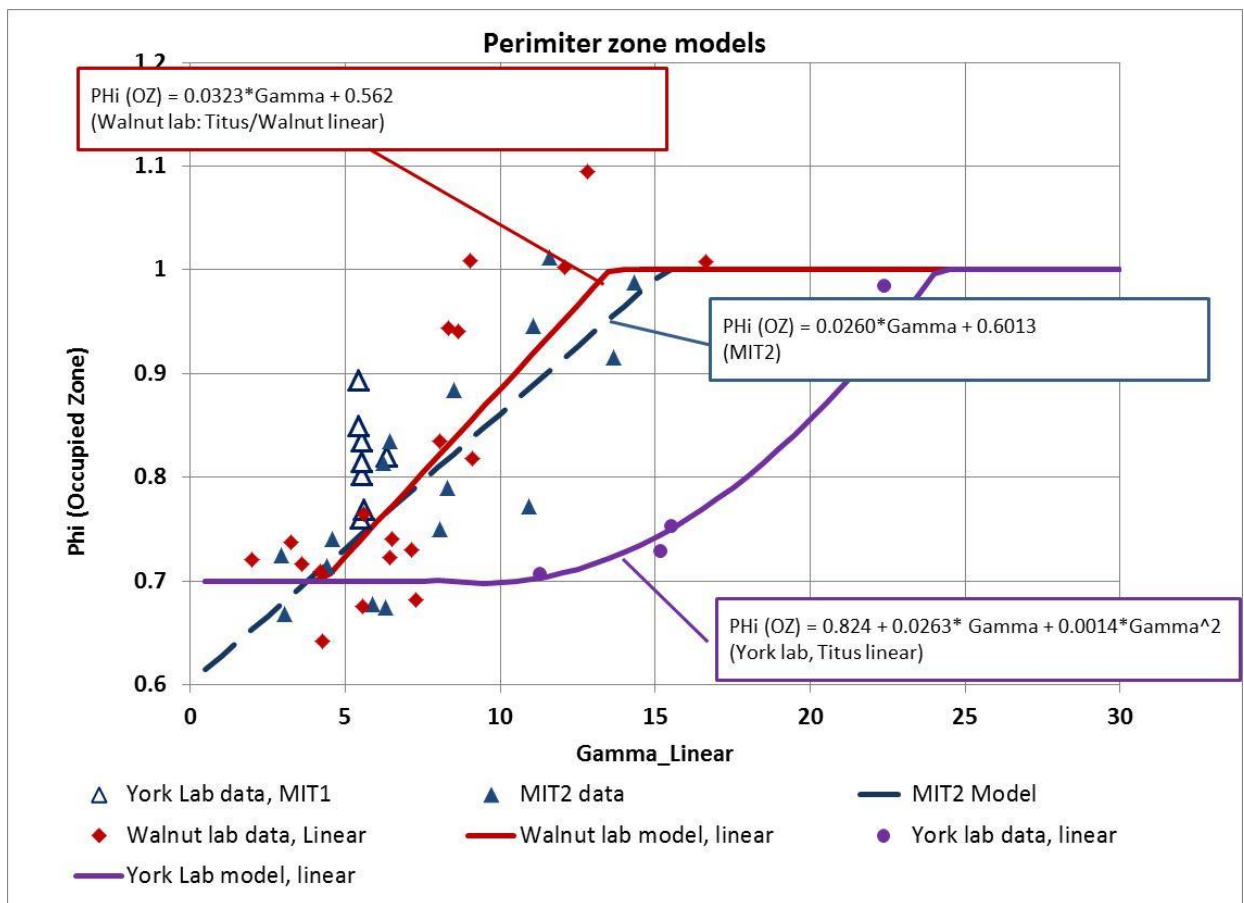
*Figure 4. Perimeter zone stratification correlations (**NOTE: DO NOT USE York lab results**)*

ENERGYPLUS UFAD INPUT REQUIREMENTS

Complete descriptions of required inputs and their defaults can be found in the EnergyPlus Input/Output Reference document. Most defaults are based on tested diffusers discussed above. The following provides more detail on selected input parameters to the help the user better understand EnergyPlus methods and the connection to plume theory.

Default values for all the required inputs are embedded in EnergyPlus and are activated when the *Auto-calculate* command is invoked. This is intended to provide reasonable default values where information is left out or lacking. It is not the same as "*Autosize;*" there are no sizing calculations done for these inputs (other than those performed in the CBE EnergyPlus/UFAD User Interface; see below).

DIFFUSER OBJECT INPUT FIELDS

Detailed input fields for diffuser characteristics are describe here and in the EnergyPlus Input/Output Reference. These inputs are used to calculate Gamma which is in turn is used to calculate Phi from the regression equations. As stated above, Phi is then used to apportion the zone convective gains between the upper and lower layers of the EnergyPlus/UFAD two layer room model. This allows a dynamic hour by hour simulation of stratification.

The primary inputs for diffusers are:

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- Effective area (Ae) – the effective discharge area for one diffuser; determined from manufacturers data or by testing (Methods for determining effective area are discussed below)

- Number of diffusers (n) – Total number of diffusers per zone; user assigned (i.e., derived from peak flow rates and diffuser design airflow or other criteria) or defaulted to design number of occupants or based on assumption of a nominal 9.3 m^2 (100 ft$^{2)}$ per occupant.

- Diffuser discharge angle (φ) – Angle from vertical of air discharge from the diffuser. Does not include angles used to induce swirl.

- Diffuser type – Type of diffuser in a zone; only one type is allowed per zone. The options are Swirl, Horizontal Swirl, Variable area, and Linear bar grille any of which can be specified in the Diffuser Type object. If "Custom" is entered for diffuse type, the default types and their corresponding parameters are ignored and the detailed user entered parameters (including regression coefficients) are used instead. (See Other Diffusers below.)

All of these parameters have defaults that are used when a specific diffuser type is selected; the defaults are listed in Table 1.

Alternative diffusers

Other diffusers can be accommodated in EnergyPlus UFAD modeling. All that is required is empirically derived Gamma vs. Phi correlations for the particular diffuser of interest, as well as the other diffuser characteristics, effective area and discharge angle. The regression coefficients can be input into EnergyPlus in either power or polynomial equation formats.

THERMAL PLUMES

These parameters are only relevant to interior zones with internal loads that generate thermal plumes. Although perimeter zones have internal plume generating loads also, we assume that all loads (internal and solar) behave like an equivalent wall plume which is represented by the Gamma linear vs. Phi formulation. Likewise, for interior zones we do not distinguish between individual load components (people, computers, etc.) rather we represent these as a single (coalesced) plume; i.e., one plume per workstation. In both of these constructs, the room extraction rate is assumed to be the total convective plume power generated by m number of (interior) zones; one wall plume is assumed for perimeter zones) plumes.

The primary inputs for interior zone plumes are:

- Power-per-plume (PpP) – This is the power (kW) that represents the full load power of an individual plume; i.e., equivalent full load plume power. Users can input a value or use the EnergyPlus default via the auto-calculate command. Defaults are derived from the total design convective heat gains for people and equipment divided by the design number of occupants; i.e., assumes one occupant per workstation or load center. If occupants are not included, one plume per 9.3 m^2 (100 ft^2) is assumed.

Once calculated, power-per-plume is used internally to calculate the number of plumes (m). During simulation m is determined by dividing the total room convective heat gains (~extraction rate) by power-per-plume; m then is the number of equivalent full load plumes for a given hour.

For perimeter zones, plumes are modeled as one continuous wall plume whose strength is proportional to the room cooling load.

TRANSITION HEIGHT

The transition height represents the stratification level, the division between the upper and lower layer. According to plume theory this height varies depending on the balance between buoyancy (via thermal plumes) and momentum (via diffuser airflow) forces. Transition height increases with higher airflow from the diffuser for a given load. However, it also depends on the number and characteristics of the diffusers and is a function of Gamma as is the case for Phi. Gamma is used to calculate the transition height, h. This can result in a complex dynamic relationship between the control point and the occupied zone average temperature.

The transition height can be fixed by user input or auto-calculated from its Gamma correlation. This correlation has not been validated since the height is nearly impossible to measure in real systems. This results from the fact that in most real situations there is not a clear or single valued stratification height; it is most likely a band due to unequal plume strengths and load component vertical locations.

Simulations have been conducted to determine the sensitivity of using fixed Phi and transition height parameters. The transition has been found to have little or no impact in interior zones, and only a minor impact in perimeter zones. These results indicate that using a fixed height as a simplification (e.g., fixed at 67 inches to allow the occupied zone temperature to be the controlled temperature) results in very little error.

DIFFUSER CHARACTERISTICS

The critical parameters required for properly simulating diffusers are effective area, discharge angle, and a corresponding design airflow as noted above. The following describes the basis for the default models installed in EnergyPlus.

SWIRL

The effective area of swirl diffusers is based on the Price model RFTD with an effective diffuser area of 0.0075 m² (0.081 ft^2) and a discharge angle of $28°$. These values were derived from bench testing and correspond closely to results from catalogue data. It is also noted that this area was identical for other swirl diffuser brands (i.e., Trox, Nailor, Titus). However, the Gamma-Phi correlations only represent Price diffusers; others need to be tested to provide a corresponding regression. These diffusers are most commonly used in interior zones, but rarely in perimeter zones.

HORIZONTAL SWIRL

The effective area for horizontal discharge diffusers (Price ARFHD) is 0.0060 m² (0.0648 ft^2). These diffusers have horizontal discharge therefore the discharge angle from vertical is much greater than for standard swirl diffusers. We used $73°$ based on choosing a reasonable comparison to the salt-tank data from UCSD. Only four tests using HSW diffusers were conducted. These are only used in interior zones.

VARIABLE VOLUME

These diffusers (York MIT generation 2 (MIT2) diffusers) operate with constant underfloor plenum pressure (i.e. 0.05 iwc during laboratory tests) and constant air velocities leaving the diffusers. Consequently, in order to vary the amount of airflow entering the room, the variable volume function is derived from pulse-width modulation of the diffuser inlet damper.

By assuming a constant supply air velocity of 2 m/s (400 fpm), the effective area is calculated using Equations 5 and 6, below. This assumed velocity was verified by staff at York/JCI. For fully open diffusers airflow measurements reveal a flow of about 255 m³/h (150 cfm). If the area of a fully open

diffuser is 0.035 m² (0.38 ft²) the outlet velocity at the grille is about 2 m/s (400 fpm). The estimated discharge angle for MIT diffusers is 45°.

Although the grille area does not change (as it did in previous versions of these products) pulse-width modulation creates a variable volume function, therefore the area and discharge velocity is fixed for the open condition. We therefore consider these "constant Phi" diffusers. These diffusers are commonly used in both interior and perimeter zones.

LINEAR BAR GRILLES

These diffusers are standard products used and are commonly employed in UFAD systems for heating and cooling. The effective area (for Titus diffusers) was derived from bench testing and corroborated by catalogue data. They come in various lengths and widths and usually have a lip on the bars that directs the flow away from the window at about a 15° angle. Sometimes they are equipped with vanes that spread the flow along the window. When this is the case $\cos(\varphi)$ is a compound angle.

Recent testing (2010 at Walnut labs) demonstrated that there is some difference in performance due to the form factor of these diffusers. The original testing was done with small diffusers (Titus CT-481, 3 in x 18 in) with inlet vanes. The results from Walnut lab testing were for a more common size, nominally Titus or Walnut, 6 in wide by 6 ft long without inlet vanes. Other brands with a different form factor (e.g., Price LFG series).have not been tested and we would expect the performance to be somewhat different. To support simulations we have assumed nominal configuration data shown in Table 2 below. For LIN2 this represents a nominal 6 in (0.154 m) wide by 3 ft (1.21m) long diffuser that was extrapolated from the 6 ft long tested diffusers. These represent the default configurations used in EnergyPlus for the respective object label. (See the Appendix A for details of how these values are used for sizing purposes.) The Titus and Walnut data for diffuser LIN2 was combined to create the correlation equations. Note that the data for LIN2 were collected with the diffuser wide open (full flow mode) but due to the pulse width modulated control sequence used, in operation these diffusers operate at a fixed phi (and gamma) of about 0.85.

LIN1 diffusers were tested completely different and no longer need to be used. These were tested along with swirl diffusers that supply the internal loads for the zone in a large 676 sf chamber. (Walnut testing had only linear bar grilles, and a 256 sf chamber).[2] The curves shown in Figure 4 and Table 2 of LIN1 diffusers is inaccurate since the full room flow was assumed to flow through the nominal size of the 3" wide bar grilles (which also have diverting vanes installed). This is incorrect for two reasons; (1) the airflow at the linear diffusers were controlled by a flapper so the affective area (Ae) needs to be corrected for this fact, and (2) the Phi – Gamma relationship is complicated by the fact of using two different types of diffusers. We have attempted several ways to calculate gamma considering these factors, but have yet to determine a scientifically valid way to represent this data. **Therefore, we recommend that the LIN1 option not be used for simulations. This should have little practical effect on results since linear bar grilles produce little stratification in simulation and in practice based on our experience; either model replicates this effect.**

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[2] In other words, York data assumes swirl diffusers in the perimeter zone, not an uncommon approach for real systems, but Walnut data assumes only linear bar grilles in the perimeter zone.

Although other form factors and various lengths could be derived from the basic unit form factor data (i.e., a wider unit would have a larger form factor, so the equivalent length would be shorter) this option requires that the CUSTOM specification be used with user supplied object fields.

For reference purposes, Figures 5a through 5c show catalog performance information for the Titus CT-481 diffusers, and Price swirls. Table 2 below shows simulation specifications used for the default diffuser types.

York/JCI diffusers

Although the phi-gamma curves for the Walnut and Titus diffusers are virtually the same from the testing, there is a fundamental difference in how they are applied. The Walnut diffusers are supplied by York/JCI packaged in a unit that uses their pulse width modulated (PWM) air valves, just as is done with the MIT diffuser units. The effect of this is the same; these diffusers all operate a fixed discharge velocity (equal to that of the design airflow) when the air valve damper is open. This has the effect of making these diffusers "constant phi" diffusers. We make the assumption that these diffusers are selected and controlled to deliver the design airflow when open. This is contrary to how these would work when connected to a fan coil unit (as Titus and other linear bar grilles are normally configured) where the airflow is varied for a fixed area.

Design volumes

Design volumes listed in Table 2 have been scaled to nominal values for simulation purposes and based on our analysis and testing and assumptions about what we consider appropriate default sizes for simulation purposes; they may not conform precisely to manufacturers catalogue data. We have standardized on design pressures as follows: a plenum pressure of 0.050 iwc for swirl and MIT diffusers, and a drop across the diffuser plate of about 0.019 iwc; i.e., pressure just below the diffuser plate, for linear bar grilles. Unfortunately, this pressure was not accurately measured during testing to verify this nominal value that was estimated from Titus diffuser grille data (see Figure 5a below). Walnut specifies plenum pressures to be controlled to 0.05 iwc so we conclude that there is a significant pressure drop through the damper box at full flow to yield these low pressures at the diffuser plate. In real systems it represents the pressure at the inside of the diffuser as supplied by a fan coil unit for Titus diffusers and fan coil unit for the Walnut based systems.

Table 1. Default characteristics of tested diffusers

	EPlus label	Test lab	Make and model	Nominal Size	
				in, ft	m
Interior	Swirl	York/JCI	Price RFTD	8 in dia	0.2
	Horiz Swirl	York/JCI	Price ARFHD	8 in dia.	0.2
	MIT	York/JCI	York/JCI, MIT1	11.4 in x 11.4 in	0.29 x 0.29
	MIT2	Walnut	York/JCI, MIT2	11.4 in x 11.4 in	0.29 x 0.29
Perimeter	LIN1	York/JCI	Titus CT-481	3" x 1.5 ft	0.077 x 0.46
	MIT1	York/JCI	York/JCI, MIT1	11.4 in x 11.4 in	0.29 x 0.29
	LIN2	Walnut	Titus CT-481 & Walnut CLEMIT-3-27-06-015-L6F	6" x 6 ft	0.154 x 1.83
	MIT2	Walnut	York/JCI, MIT2	11.4 in x 11.4 in	0.29 x 0.29

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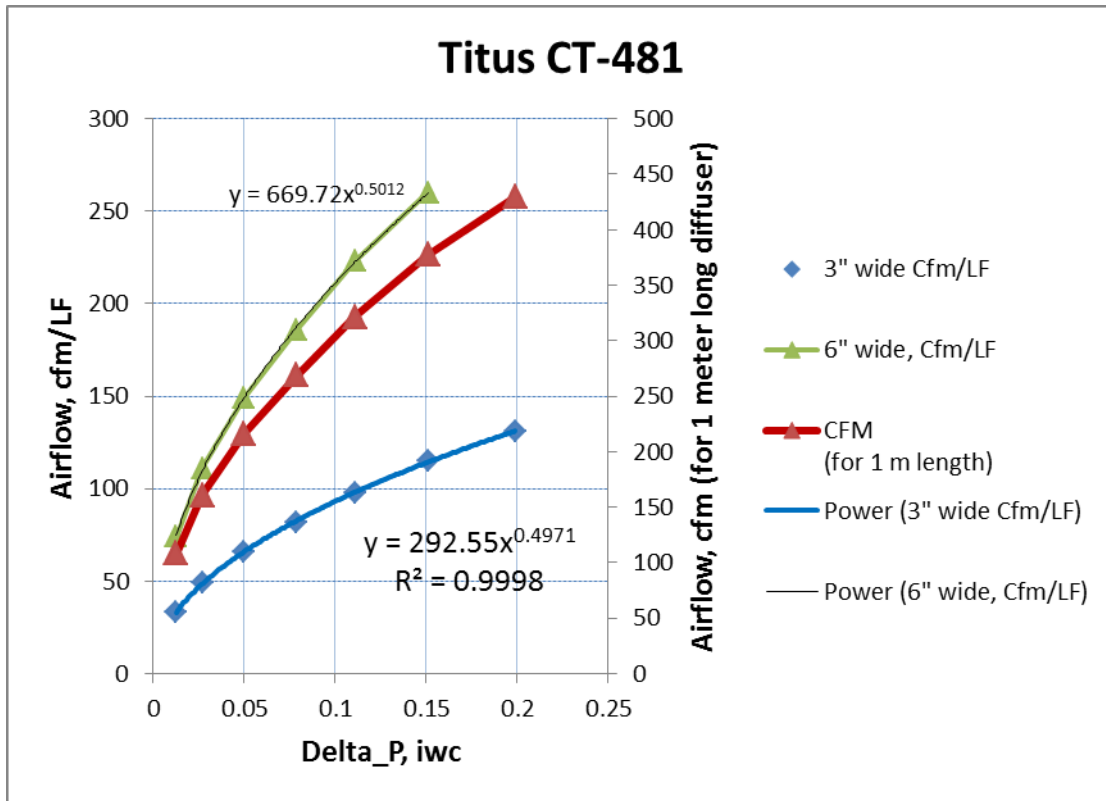| Area, Square Feet*	Nominal Duct Width, Inches	Total Pressure		.0125	.027	.050	.079	.111	.151	.199	.252	.310
.082	3"	CFM per Foot		33	49	66	82	98	115	131	148	164
		NC		–	12	20	26	31	36	40	43	46
		Throw, Feet	Sill or Floor	2 - 2 - 2	6 - 6 - 6	9 - 9 - 9	12 - 12 - 13	15 - 15 - 16	17 - 18 - 19	20 - 21 - 22	21 - 22 - 23	23 - 23 - 24
			Side Wall	4 - 6 - 9	7 - 10 - 13	9 - 13 - 17	12 - 16 - 21	15 - 20 - 25	17 - 22 - 28	20 - 26 - 32	21 - 28 - 35	23 - 31 - 39
.186	6"	CFM per Foot		74	111	149	186	223	260	298	335	372
		NC		–	14	22	28	34	39	42	46	49
		Throw, Feet	Sill or Floor	4 - 4 - 4	10 - 10 - 10	14 - 14 - 14	18 - 18 - 18	22 - 22 - 22	25 - 25 - 25	28 - 28 - 28	30 - 30 - 30	32 - 32 - 32
			Side Wall	8 - 10 - 13	11 - 14 - 18	15 - 19 - 23	19 - 23 - 27	23 - 27 - 31	25 - 30 - 35	28 - 33 - 39	31 - 37 - 43	34 - 40 - 46

Figure 5a. Titus Linear bar grille performance data

Figure 5b. Titus Linear bar grille performance

DETERMINING DIFFUSER EFFECTIVE AREA

In this section we describe procedures for determining the effective area for diffusers. Given adequate data, this can be accomplished with manufacturer's catalog data.
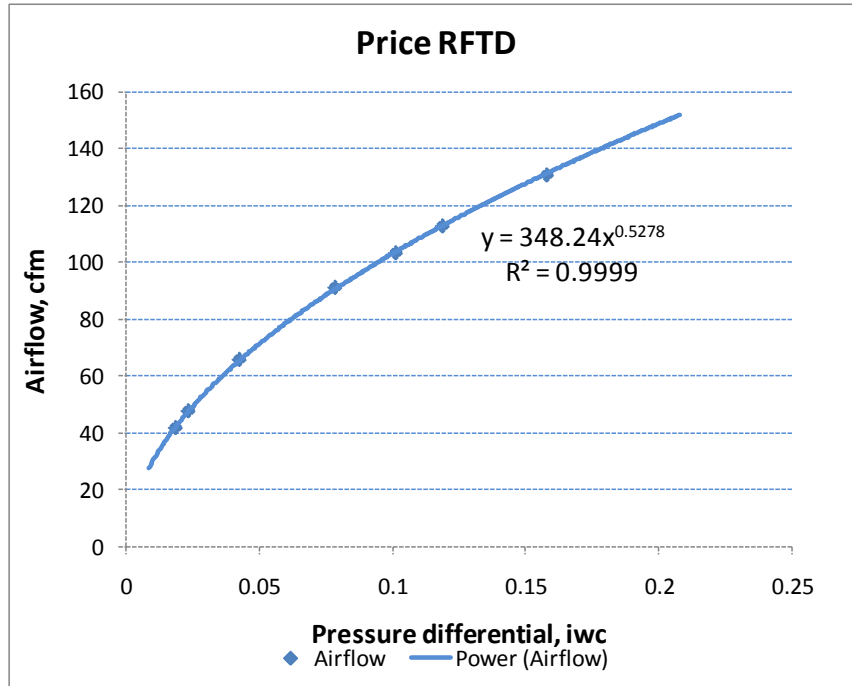
Figure5c. Price swirl diffuser airflow performance

Using results from testing or data from catalogs for a given diffuser the effective area can be determined as follows using the example regression equation from Figure 5c:

To convert results from the linear regression to square root form; select typical operating pressure (e.g., 0.05 iwc) and compute as follows.

$$A_{eff} = \frac{K}{4005}$$
Equation 5

Where,

$$K = \frac{348.2 * 0.05^{.0.50}}{0.05^{0..5278}}$$
Equation 6

DEFAULT DIFFUSER INPUT SPECIFICATIONS SUMMARY

Table 2 summarizes default specifications and correlation equations for all diffusers tested. The design volumes were derived from the test diffuser specifications shown in Table 1.

Table 2. Default specifications for EnergyPlus/UFAD

Zone Type	Diffuser Type (Eplus lable)	Default total effective area per diffuser (unit area)		Discharge Angle from vertical	PHI – Gamma correlations (X = Gamma, X_L = Gamma linear)	Phi limits		Nominal Design volume[3] (flow/Length)	
		ft^2	m^2			min	max	cfm	m^3/s
Interior	Swirl[4]	0.081	0.0075	28°	Phi = -0.00004x^2 + 0.0069x + 0.6531	NA	NA	72	0.034
	Horiz Swirl[5]	.0644	0.0060	73°	Phi = 0.67 (constant for all Gamma)	NA	NA	55	0.026
	MIT1	0.38	0.035	45°	Phi = 0.88 (constant for all Gamma)	NA	NA	150	0.071
	MIT2	0.38	0.035	45°	Phi = 0.025x$_L$ + 0.6013	NA	1.0	150	0.071
Perimeter	LIN1[6]	0.126 (0.084/Lft)	0.0115 (0.025/Lm)	15°	Phi = 0.0014x^2 - 0.0263x + 0.8214	0.7	1.0	47 (33.4/Lft)	0.022
	LIN2[7]	0.52 (0.18/Lft)	0.048 (0.055/Lm)	15°	Phi = 0.0323x$_L$ + 0.56	0.7	1.0	225 (75/Lft)	0.106
	MIT2	0.38	0.035	45°	Phi = 0.025x$_L$ + 0.6013[8]	NA	NA	150	0.071

EnergyPlus phi-gamma coefficients format: Phi = A*Gamma**B + C + D*Gamma + E*Gamma**2
(A and B coefficients allow for more complex curve fits other than 2nd order polynomial.)

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[3] Flow at 0.05 iwc plenum pressure for swirl and MIT and estimated 0.0127 iwc for linear bar grilles (behind the grille). Design volumes are used in the CBE UFAD interface to facilitate calculation of the number of diffusers.

[4] Airflow based on CBE measurements at .05 iwc

[5] Airflow based on Price catalogue data at 0.05 iwc

[6] Based on 1.5 LF diffuser

[7] Assumes Titus or Walnut 6" wide based on 3ft nominal length diffuser

[8] Although this table shows a gamma-phi curve for MIT2 diffusers, in practice this is used only for design sizing; the associated phi values remains fixed at the design value for subsequent daily and annual runs. We assume MIT2 represents all York/JCI products that use the modulating damper. (See Appendix A)

SUPPLY PLENUM MODELING

Figure 7-1 presents a schematic diagram of how the simplified plenum model can be configured within EnergyPlus. Conditioned air from the air handler enters the underfloor plenum (Plenum 1) at the desired flow rate and plenum inlet temperature (T_{in1}). Similar to other conditioned zones, EnergyPlus performs an energy balance on the plenum, producing a single well-mixed temperature ($T_{plenum1}$). To calculate the energy balance, surface convection coefficients developed by CBE are specified for the slab (h_{s1}) and raised floor (h_{f1}). The well-mixed plenum temperature ($T_{plenum1}$) serves as the average diffuser discharge air temperature (T_{out1}) entering the conditioned space (Zone 1). Note that more than one thermal zone can be served by a single underfloor plenum, and in fact, unless the user has more detailed information about the expected temperature distribution in the underfloor plenum across the entire floor plate, this is the preferred plenum configuration (one single underfloor plenum) for each floor of the building. However, if desired, one or more additional plenum zones may be added in series to the first plenum zone. In this case, as shown in Figure 7-1, T_{out1} is equal to the plenum inlet temperature (T_{in2}) for the second plenum zone (Plenum 2), i.e., series plenums. This permits the possibility of simulating an interior plenum zone with a cooler supply air temperature and a perimeter plenum zone with a warmer supply air temperature due to thermal decay in the plenum. Note that in this case, the airflow entering the second plenum will be reduced by the volume of air delivered into Zone 1. Due to the complexity of most underfloor plenum airflow and temperature distributions only the series plenum configuration is assumed as the default for current versions of EnergyPlus. Only knowledgeable users should attempt to configure the model for the configurations listed below. Airflow paths are dictated in the AirLoop objects in EnergyPlus. This includes zones, terminal equipment and splitters and mixers. Currently (2011) only splitters are allowed for supply plenums and mixers for return plenums.
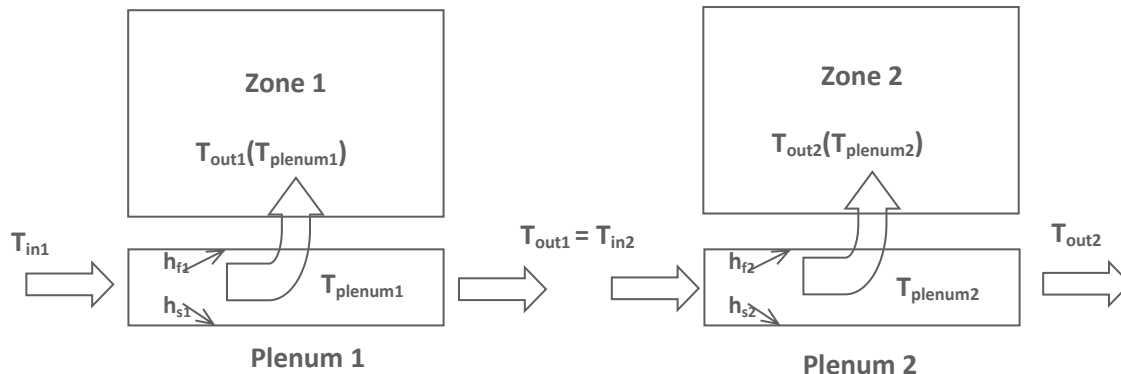

Figure 7-1. Schematic diagram of EnergyPlus plenum model structure for series plenum

Underfloor supply plenums can be installed and zoned in a variety of configurations. Based on current research and practice, it is believed that the choice of plenum configuration can have important impacts on the thermal performance (distribution of heat gains) of the supply plenum. Plenum configurations now available in the CBE development version of EnergyPlus v6.0 are described and illustrated briefly below. The descriptions are somewhat idealized compared to actual practice, but are intended to capture the expected performance characteristics of different approaches to plenum design and operation.

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Figure 7-2 shows plan and section schematic views of a portion of the (default) series plenum configuration available in the public release version of EnergyPlus. This configuration represents an approach that has been used in many UFAD installations in which an open plenum with minimal ductwork serves both an interior and perimeter zone of the building. Supply air from the air handler is delivered from the service core into the interior part of the plenum. Swirl diffusers are used in the interior and linear bar grilles are used at the perimeter. The approximate temperature distribution in the plenum and room (for demonstration purposes) can be read using the color temperature scale. In this case, the series plenum refers to the concept that cool air entering into the plenum (60°F [15.6°C]) will first gain heat as it flows through the interior portion of the plenum (reaching 64°F [17.8°C]), before entering the perimeter portion of the plenum, where it gains additional heat (68°F [20°C]) before entering the room. As will be discussed further, this approach is expected to produce the highest perimeter diffuser discharge temperatures, and should be avoided if possible.

Figure 7-3 shows a reverse series plenum configuration. This is not yet available in EnergyPlus, but should be available later in 2013. This represents an alternative plenum design that uses ductwork (or other means) to deliver the cool supply air from the air handler directly into the perimeter portion of the plenum. Supply air temperature rise proceeds in the reverse direction with the incoming (60°F [15.6°C]) air gaining some heat from the perimeter zone producing (64°F [17.8°C]) supply air at the perimeter bar grilles, before flowing back to the interior where additional supply air temperature rise produces the highest supply temperature. This configuration provides cooler supply air at the perimeter to handle peak cooling loads and warmer temperatures in the interior for comfort purposes.

Figure 7-4 shows the "common" plenum configuration. This refers to a situation where air is delivered into the plenum from a variety of locations (using ductwork or air highways, as shown) and with a variety of inlet velocities. The net result is that while there will be variations in individual diffuser discharge temperatures across the floorplate, the average supply temperature entering the interior zone will be very similar to that entering the perimeter zone. In this case, the term "common" plenum means that the plenum delivers the same average temperature everywhere. This characteristic behavior has been observed in completed UFAD projects.

Figure 7-5 shows a parallel, or zoned, plenum configuration. This configuration requires the installation of a plenum divider separating (in this case) the perimeter and interior portions of the plenum. This is an approach that may be used when the designer wants to ensure improved control of a special zone with unique load requirements. It does result in a more complicated plenum configuration with barriers restricting some of the flexibility. Supply air from the air handler must be delivered directly to each isolated zone.

One other plenum configuration, ducted perimeter, is not illustrated, but represents a combination of the parallel and single plenum schemes. In this case, AHU supply air is ducted directly to the perimeter zone diffusers via an assumed VAV box. The interior portion of the plenum is served in the same manner as parallel plenum configuration.

All of the configurations described above, except reverse series, can be simulated with appropriate changes to the air path objects and careful construction of the zone to plenum boundary surfaces objects. These are currently (2012) implemented in the CBE prototype building .imf files that can be used as an example for other building models.
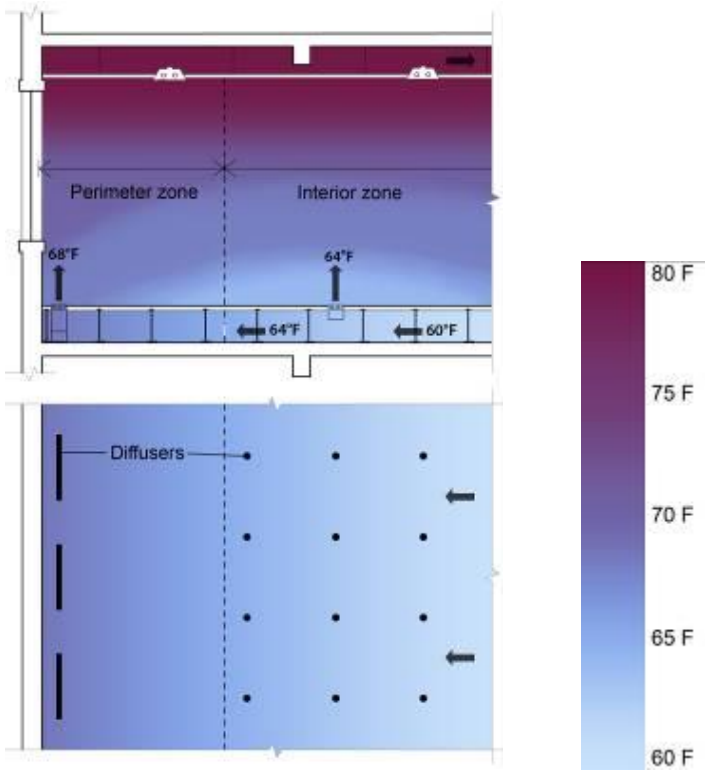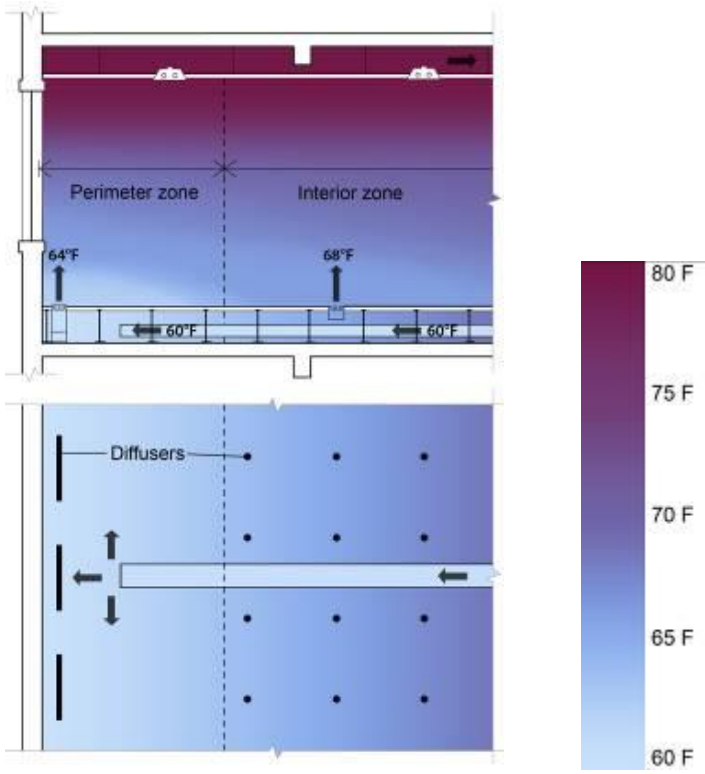
Figure 7-2. Series plenum configuration

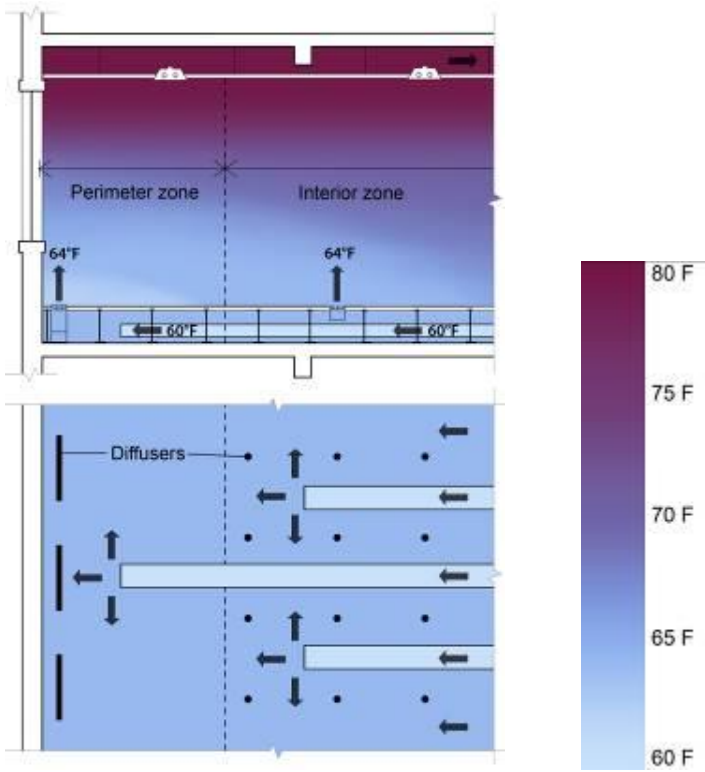*Figure 7-3. Reverse series plenum configuration*

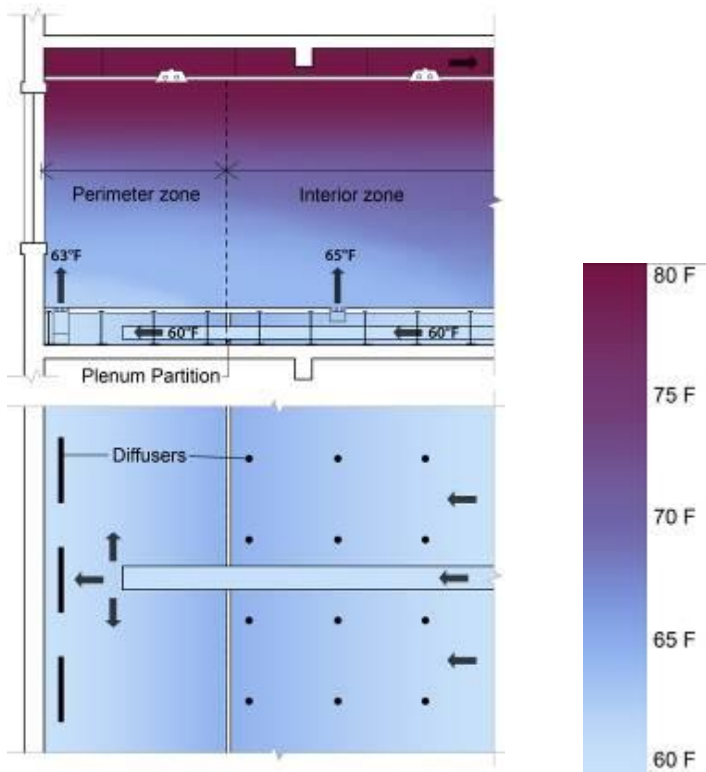*Figure 7-4. Common plenum configuration*

Figure 7-5 Parallel (zoned) plenum configuration

CBE ENERGYPLUS UFAD HVAC MODELING GUIDE

In this section we outline methods used to model various system and plant components typically used for UFAD systems. Each run is assumed to be sized anew for central system and plant components, either using the EnergyPlus auto-sizing capability or some form of manual sizing. This approach assumes that systems are redesigned[9] for the new design conditions implicit in the parametric factors under study. Note also that a number of the features and design work-arounds discussed below are only implemented in CBE's development version of EnergyPlus.

CENTRAL AIR HANDLER MODELING

For these studies the fan design volume (and fan size) will be different for each run. For these we assume that the new operating points all use the same AHU Fan design static pressure (FSP). This implies that the fans are assumed to be resized for each new design point such that the internal losses (filters, coils, etc.) are the same, AND the fan is sized to maintain the same (near peak) design efficiency. Therefore, each UFAD design point is input with the same peak efficiency and UFAD design static pressure, but a new airflow.

Due to problems with auto-sizing the airflow, we use a CBE sizing strategy where the design day output is used for the maximum airflow which is then increased by a global sizing factor of 1.2.

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[9] As opposed to maintaining a given size while changing the parameters as would be done if differences in operation were being investigated

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Fan design efficiencies

EnergyPlus AHU models require design airflow, static pressure, fan efficiency, and motor efficiency as inputs. For the supply and return fans we assume a variable frequency drive with a high efficiency motor and a belt drive to the fan. Typical values are shown in Table 2.

Table 2. Central air handler efficiencies

	Individual	Combined
Fan static efficiency	75%	75%
Motor efficiency	90%	84%
Belt drive efficiency	95%	
VFD efficiency (100%)	98%	
Total Efficiency	63%	63%

Variable frequency drive (VFD) efficiencies are a function of operating frequency [Wray, Personal communication] (i.e., speed) which are in the range of 60-70% at 30% speed and 98% at 100% speed. We have included only the design point VFD efficiency; a new algorithm will be required to simulate the VFD part load performance accurately. The resulting motor plus overall drive efficiencies are incorporated in the motor efficiency as shown in the Combined column in Table 2.

Fan design static pressure

Static pressure breakdown

To guide the user, Table 3 shows a typical component breakdown of static pressure losses for air system components and how UFAD and overhead systems compare.

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Table 3. Fan static pressure guide

Pressure loss item	OH		UFAD		comments
	Office supply, iwc	Office return/ relief, iwc	Office supply, iwc	Office return/ relief, iwc	
AHU (clean filters, coils, louvers, etc.)	1.1	0.3	1.1	0.3	Include system effect
Dampers	0.1	0.1	0.1	0.1	Economizer dampers
Dirty filter allowance	0.75		0.75		
Ductwork, shafts	0.08	0.03	0.08	0.03	
Ductwork, branches	1.25	0.02	0.75	0.02	Return side = return plenum UFAD: varies depending on plenum ductwork configuration
VAV boxes	0.5		0		
VAV box reheat coil	0.2		0		
Diffusers and low pressure ductwork	0.3	0.1	0.05	0.1	RA grille
Safety factor					
Total	**4.28**	**0.55**	**2.83**	**0.55**	
Fan design static	4.53	0.55	3.08	0.55	Assumes return relief fan. Supply fan static = combined supply and return losses
Fan design eff	0.75	0.54	0.75	0.54	
Motor eff	0.9	0.9	0.9	0.9	
Drive/VFD eff	0.98	0.9	0.98	0.9	
Min fan static (shutoff)	1.5 -0.5	0	0.5	0	Assume 0.5 = static pressure reset

Impact of UFAD plenum configurations

As shown by Table 3 the primary differences between typical OH and UFAD systems is due to branch and low pressure ductwork; Table 3 shows UFAD to be 60% of OH for this component. However, the value for UFAD depends on the configuration of the plenum air distribution system. It can vary widely between extremes of no ductwork to fully ducted (equivalent to an OH system); "air highway" systems are somewhere in between. The user should adjust the values on Table 3 to reflect the design being considered. Examples are shown in Table 3a below.

Table 3a. Design static pressures for various plenum configurations

System/UFAD Plenum configuration	AHU Design FSP
Overhead system	4.3 iwc (1075 Pa)
Common open plenum	2.8 iwc (700 Pa)
Series open plenum	2.8 iwc (700 Pa)
Ducted perimeter	4.3 iwc (1075 Pa)

Air handler supply temperature

Two methods can be used to specify air handler supply temperatures (SAT): (1) fixed, where the cooling coil discharge is maintained at a constant temperature, or (2) reset, where the SAT is reset based on outside air temperature or by load demand. Both of these strategies are available in EnergyPlus, but the load based reset has not been shown to be reliable in our experience. Reset is required by ASHRAE standard 90.1-2010.

Part load performance

During simulation, the power at each time step is determined by multiplying the design power by a part load factor from the part load curves shown in Figure 8. Shutoff pressures[10] characterize different control methods and system designs, as indicted in Figure 8. For overhead systems using constant static pressure control strategies, shutoff pressures vary between 0.250 kPa (1.0 iwc) for well-designed systems and 0.375 kPa (1.5 iwc) for typical practice design. [Daly 2009] For UFAD systems, shutoff pressure is assumed to be 0.125 kPa (0.5 iwc) for partly ducted distribution systems, our baseline assumption.
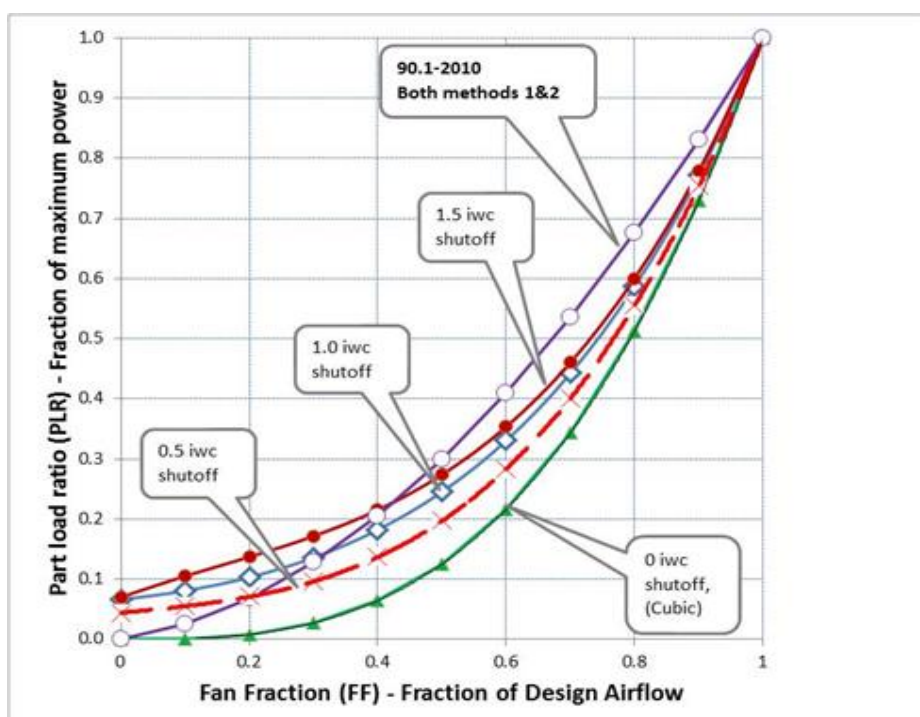

Figure 8. Fan part load curves for central airfoil fans with difference shutoff assumptions and for variable speed VAV terminal units.

Table 4 summarizes the recommended equations used for various fan types.

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[10] Shutoff pressure represents the design static pressure setpoint for a VAV system without static pressure reset. When static pressure reset is simulated, the shutoff pressure is assumed to be near zero (e.g., 0.5 iwc).

Table 4: Part load curves

	PLR curves	Remarks
Supply fan, typical overhead system, without static reset	$y = 0.2939x^4 + 0.2227x^3 + 0.3063x^2 + 0.1131x + 0.0662$	Assume airfoil fan with 1.0 min static pressure
Relief fan, and VSFCUs	$y = 1x^3 + 2E\text{-}13x^2 - 6E\text{-}14x - 4E\text{-}15$	Assume propeller relief fan follows cubic curve
UFAD supply with static pressure reset, Return fans	$y = 0.0636x^4 + 0.7993x^3 - 0.0102x^2 + 0.1037x + 0.0441$	Assume airfoil fan with 0.5 min static pressure

ZONE HVAC MODELING

UFAD system models

Although other system types are possible to model with EnergyPlus depending on the skill and knowledge of the user, this guide focuses on the two system types discussed below that we have modeled successfully.

- **System 1, engineered:** UFAD system configured with swirl diffusers in the interior controlled by varying the supply plenum pressure and a perimeter system with a variable speed fan powered terminal unit's (VSFCU) that supplies linear bar grille diffusers.
- **System 2, packaged:** This system is exemplified by the packaged system sold by York/JCI. In this system the plenum pressure is held constant and both interior and perimeter VAV floor diffuser boxes modulate airflow for cooling. For heating, a constant volume fan coil unit provides heat through perimeter diffusers that are switched between heating and cooling as required.

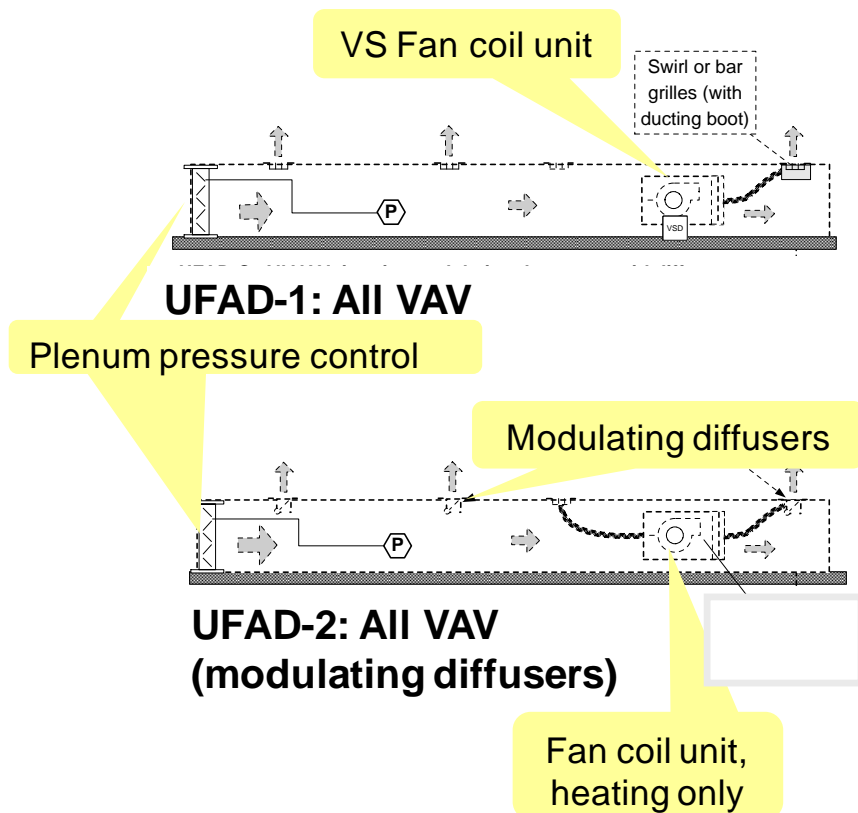Figure 9 shows illustrations of these two system types.

Figure 9: UFAD System configurations modeled by EnergyPlus

Terminal units modeling

Fan powered terminal unit efficiencies

Small fan terminal units typically used in VAV and UFAD systems use small direct drive fans, usually with forward curve blades. Direct drive makes it difficult to measure actual fan shaft power thus data sheets typically do not show power performance along with their flow vs. static pressure performance. Also, static pressure performance is usually stated in terms of external static, not total fan static pressure across the fan itself. For these reason, fan efficiency cannot be determined from catalog data. For this project, aid of manufactures' was enlisted to provide data from which the overall unit efficiency (fan plus drive) could be calculated. Data was obtained for throttling performance at constant speed for various ATU models.

From the data provided, the largest units were selected with an assumed design operating point of 0.10 kPa (0.4 iwc) (includes hot water coil and discharge ducting) at maximum speed. Based on interviews of design professionals, this strategy is a reasonable representation of how these units are selected in actual practice; where possible, several large units selected to operate in parallel at their maximum speed to meet the zone design airflow requirements. Design efficiency for this operating point was found to be ~15 percent. The user can modify this parameter based on more recent data. We assume heating fan coils used for System 2 use constant volume fans with a similar overall efficiency.

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Fan powered terminal unit part load performance

Since variable speed terminal units are not controlled to static pressure set points, they may can be assumed to throttled along a cubic curve to zero pressure and zero flow. For simulation purposes this is characterized by a cubic curve as shown in Figure 8. This performance was verified by actual measurements provided by an ATU vendor.

Zone control models

Interior zones

Since for both types of system, temperature is controlled by varying the volume, these zones are modeled with a non-reheat VAV box. The control model is shown in Figure 10.
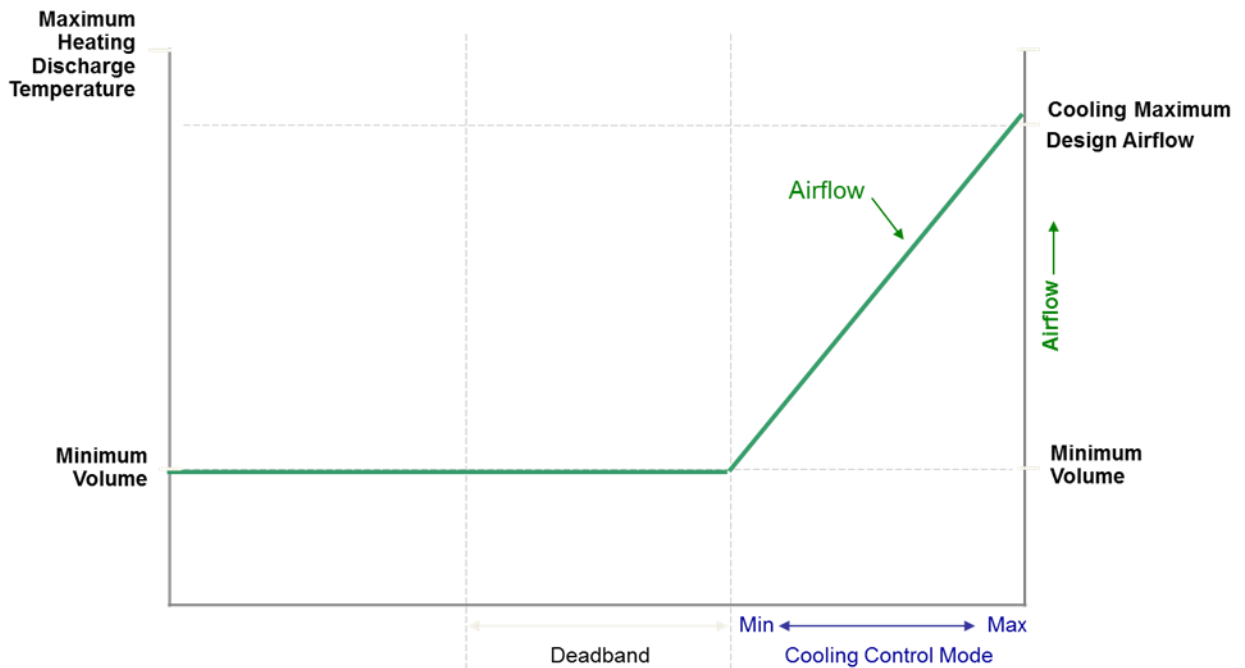
Figure 10. EnergyPlus/UFAD interior zone VAV box control model

Perimeter zones

System 1

The EnergyPLus control model for VSFCU's is shown in Figure 11. This control scheme is a "dual max" (i.e., a maximum for each cooling and heating) system with "dual minimums." Dual minimums reflect how these units are typically operated with the unit turned off in the deadband. The lowest minimum reflects leakage flows. The other minimum reflects the minimum speed airflow of ECM powered terminal units when heating starts, typically ~12%. The heating coil controls the output unit the heating maximums discharge temperature is achieved and then the motor speed is increased up to the maximum heating volume. The dual minimum strategy discussed here is only included in CBE's development version.
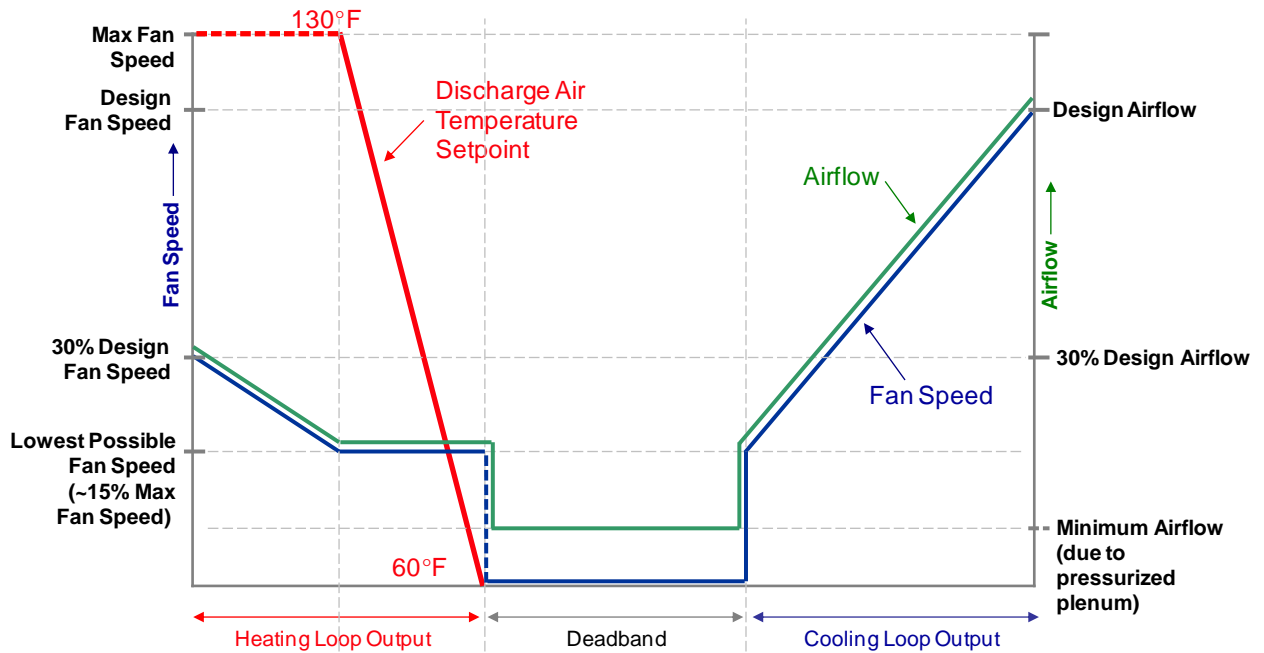
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Figure 11. EnergyPlus variable speed terminal unit for UFAD with dual minimums

System 2

Although this system type looks similar to a parallel fan powered VAV box, it is not. Fan powered boxes do not allow for zone air (more specifically, lower layer temperature air) to pass through the box during heating. Therefore, modeling this system requires separating the heating and cooling functions but with limitations imposed by EnergyPLus modeling; two terminal units can be used as long as one has its airflow path entirely within the zone. Figure 12 illustrated the EnergyPlus possibilities.
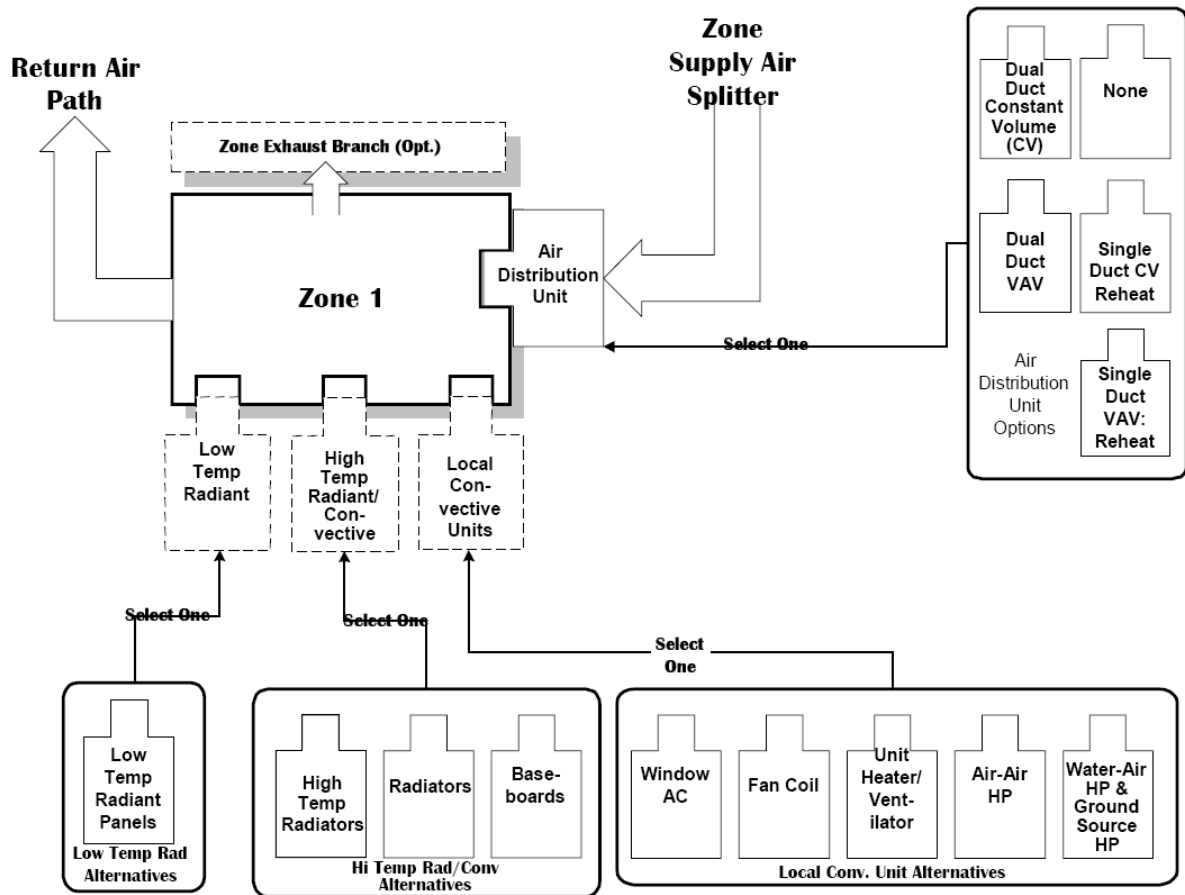
Figure 12. EnergyPlus terminal unit options

In this case we combine a single duct VAV box without reheat with a constant volume unit heater/ventilator local convection unit. The control diagram is shown in Figure 13.
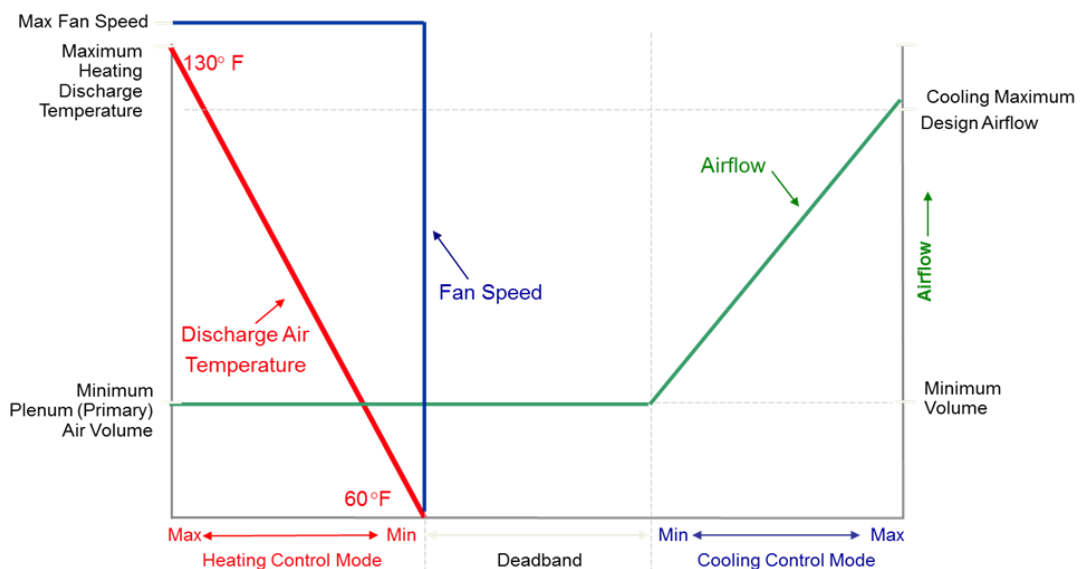

Figure 13. EnergyPlus System 2 control model

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PLANT EQUIPMENT

Although ASHRAE 90.1 Appendix G specifies DX packaged rooftop units to be used for medium sizing buildings, for consistency and ease of debugging etc. we use a chilled water system regardless of the size of building we are simulating.

Chiller assumptions

Chillers: As specified in ASHARE 90.1, Appendix G, two screw chillers (using Chiller:EIR object) sized at 50% each (based on autosizing) are served by a single variable speed primary pump. Chiller staging is controlled using an idealized strategy where the first chiller is staged up to its optimum operating point; the second chiller loads to its optimum point, then both chillers increase capacity together to full load. Leaving chilled water setpoint is reset based on outside air temperature using the schedule shown in Figure 14.
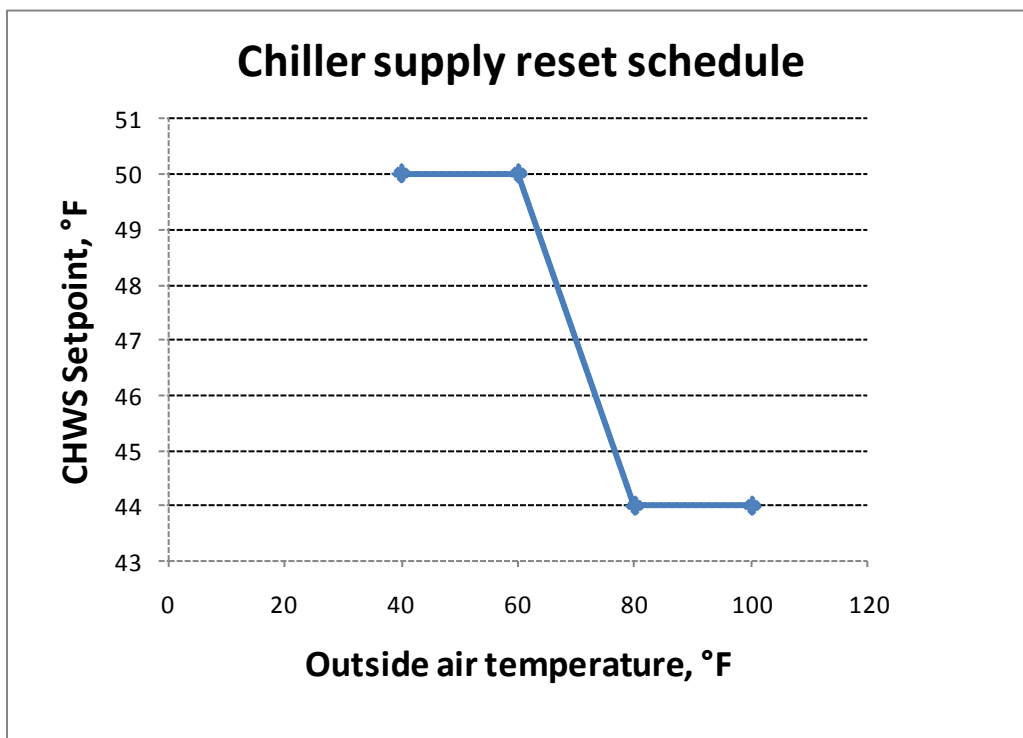

Figure 14. Example chilled water supply reset schedule

Chilled water pump (and boiler) variable speed pumps are modeled with parameters shown in Table 5 and part load curve shown in Figure 15.

Table 5. Chilled water pump performance

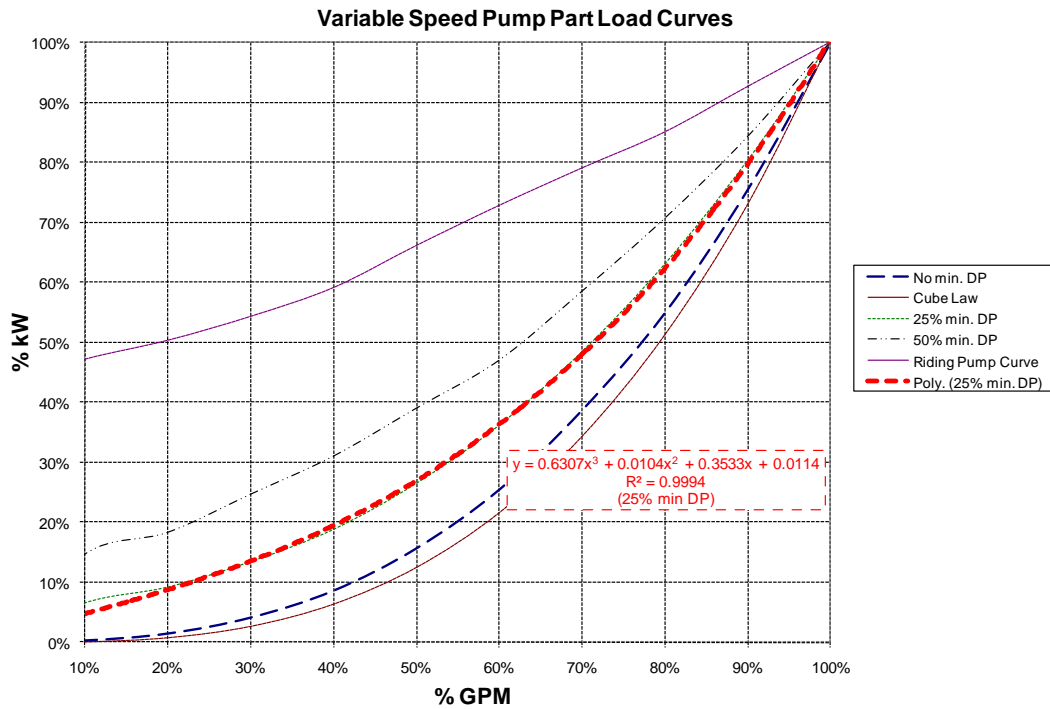| Chilled water pump eff	0.75
CHW pump design head	80 ft
Chilled water pump min DP	33%
Motor eff	0.90
Drive/VFD eff	0.98
VS pump PLR	FFP = 0.6307*PLF^3 + 0.0104*PLR^2 + 0.3533*PLR + 0.0114

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Figure 15. Chilled water and hot water variable speed pump performance curve

Boiler assumptions

Two non-condensing forced draft boilers are assumed using a 50/50% staging strategy. Boiler part load performance is modeled with DOE-2 forced draft curves shown in Figure 16. EnergyPlus uses a normalized part load curve (indicated in red) in its input structure. Hot water supply is reset according to the schedule shown in Figure 17. However, EnergyPlus has limited curve fit correlations none of which represent the type of curve shown. Therefore, we have adopted a two part linear curves shown to more accurately represent the low end performance.

We assume 180°F maximum boiler leaving water temperature reset to 150°F with outside air over the range of 20°F to 50°F, respectively.
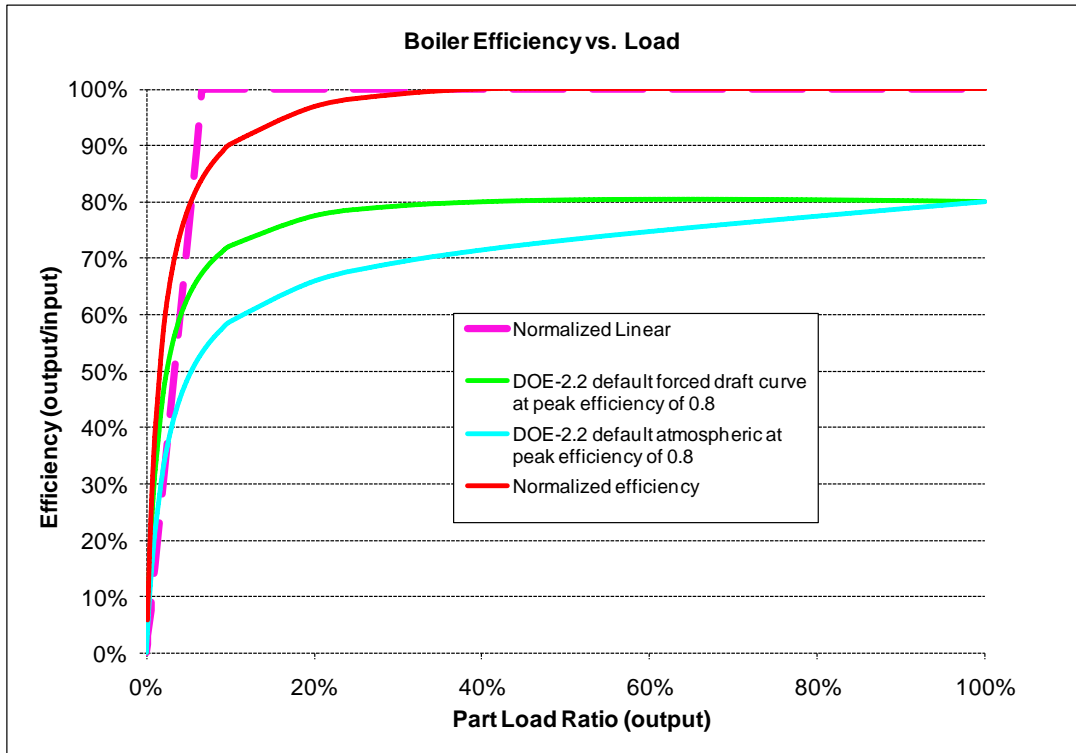
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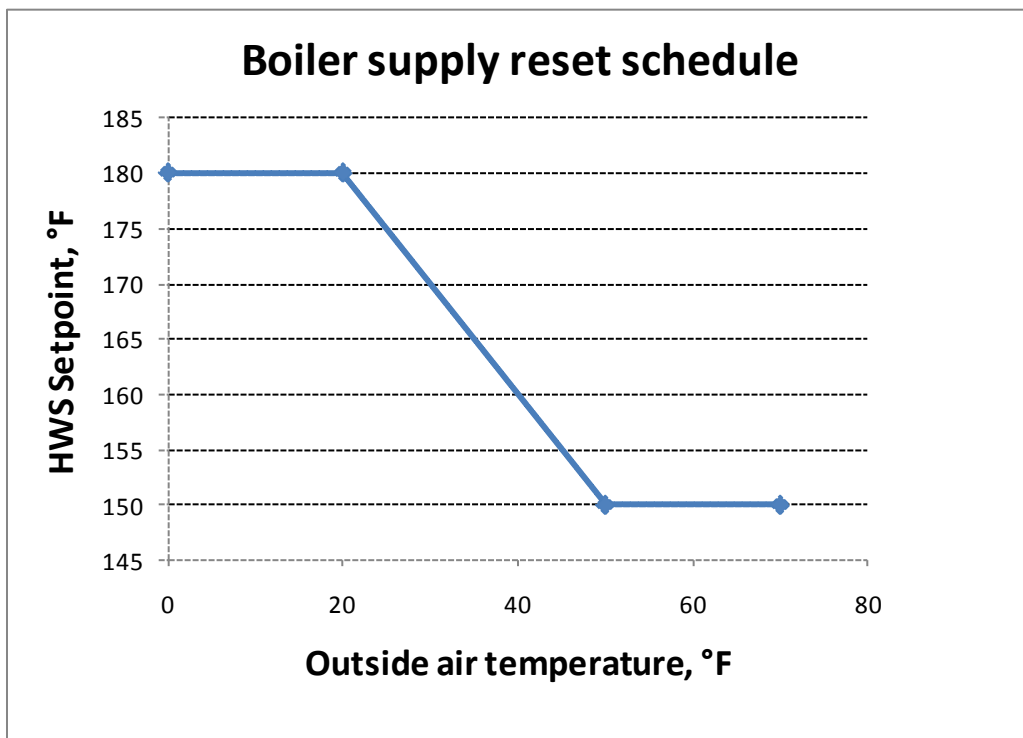*Figure 16. Forced draft boiler curve*

Figure 17. Boiler reset schedule

SIZING STRATEGIES

As mentioned previously in this report, CBE research indicates that EnergyPlus auto-sizing routines result in properly sized AHU fans for OH systems, but usually cause UFAD fans to be oversized. Proper sizing and specification of part load performance for air handlers is critical to accurate results. Erroneous results were encountered when sizing was not correctly specified or EnergyPlus autosizing routines were used for UFAD systems. One lesson learned is that the results from autosizing should always be checked before final simulations are conducted.

CHILLER SIZING

EnergyPlus chiller auto-sizing is not adequate for UFAD systems, therefore the following work-around was implemented to overcome these deficiencies.. A cooling design day simulation is run where the peak plant loop cooling demand is selected and a sizing factor of 0.6 is applied to each chiller to size the chillers for the annual run. Two chillers having sizing factor of 0.6 results in 20% over-sizing overall. Then the annual simulation is performed based on the chiller size determined from this work around.

BOILER SIZING

Since problems similar to chiller sizing were encountered with boiler sizing, an alternative to the EnergyPlus autosizing, is implemented with CBE sizing strategies. A heating design day simulation is run where the peak plant loop heating demand is selected and a sizing factor of 0.6 is applied to each boiler to size the boilers for the annual run. Two boilers having sizing factor of 0.6 results in 20% oversizing overall. Then the annual simulation is performed based on the boiler size determined from this work around.

TERMINAL UNIT SIZING

OH systems

Sizing of VAV boxes is not a straightforward process and requires special care since it is closely related to the sizing of other system components such as chillers, coils, boilers, pumps, cooling towers and fans and has an impact on reheat energy. The user must explicitly specify the VAV box cooling design supply air temperature (SAT) for each VAV box (EnergyPlus object name: Zone Sizing). Using this SAT input, EnergyPlus sizes each VAV box based on the peak cooling load determined from a design day calculation. The higher the design SATs, the larger the VAV box size. For OH systems, we set the VAV box design SAT 0.5°C (0.9°F) higher than the air handling unit (AHU) SAT to provide a small safety factor to meet the high cooling loads during the annual simulation. The same design SATs are used for OH regardless of the different zones since the actual discharge air temperature from the diffusers are assumed to be equal to the AHU SAT without duct heat gains.[11] As of V6.0 release, EnergyPlus now allows minimum flowrates for VAV boxes to be set at a fixed fraction (of design cooling airflow) or a fixed amount (typically equal to the minimum ventilation requirement). Also, implemented is a "dual max" strategy where both cooling and heating maximums can be specified. These minimum settings are determined by a

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[11] This represents a serious deficiency when comparing UFAD and OH, since it is known that OH systems also have duct heat gain; however, EnergyPlus has yet to implement a method for simulating this.

variety of factors but are primarily based on standards T-24 and ASHRAE 90.1, Appendix G. (See Appendix B.)

UFAD systems

On the other hand for UFAD systems, different design SATs are used for each zone of the UFAD model to account for the impact of thermal decay in the supply plenum. These procedures are covered in more detail in Appendix A.

Determining the number of diffusers is another important element of UFAD sizing and it directly affects   UFAD performance because it impacts stratification. Since determining number of diffusers has not been addressed in EnergyPlus autosizing procedures, we have implemented the work-around described in Appendix A.

Appendix A: Terminal Unit Sizing Procedures

OBJECTIVES

This sizing procedure aims to accurately estimate thermal decay with a preliminary design day run. This estimate serves to supply the zone design supply air temperature, which is used for sizing the maximum airflow rate of the ATUs. Thermal decay can be affected by many things, such as floor mass, window configuration, climate, and slab insulation, for example. Accurate sizing of the ATUs is important because it affects stratification heavily, as well as ATU minimum volumes which affect airflow and reheat energy.

BACKGROUND

ORIGINAL METHOD FOR ATU SIZING

Thermal Decay was previously accounted for in Air Terminal Unit (ATU) sizing with a coarse correction to the zone design supply air temperature. This input is found in the Sizing:Zone object and is parameterized in the interface as Zone_Cool_SAT_Perim[] and Zone_Cool_SAT_Core[] (at the time of the writing of this note, there is an obsolete variable called Zone_Cool_SAT_Serv[], intended to be the design SAT for the service core zones). For perimeter zones, 4°C was added to the design SAT, and in the core, 1.25 was added. The final design calculations proceed from here, so only one design simulation is performed.

This includes using higher design SATs for perimeter zones than for interior zones. Currently we add 1.3°C (2.3°F) to the AHU SAT setpoint for interior UFAD zones and 3.9°C (7.0°F) for perimeter zones. (This strategy was used circa 6/2011and is being revised to more accurately determine box size, especially for perimeter zones.)

APPROACH

DESIGN DAY FOR THERMAL DECAY

The new approach using (Excel macros) has been implemented as follows: A design day run is performed in order to estimate the thermal decay, and appropriately set the design SATs for the ATUs. In this initial run, the design airflow is oversized greatly so all peak loads will be covered; we assume 6 °C of thermal decay in the perimeter, and 2 °.C in the core zones. After the simulation is completed, the thermal decay for each zone can be accurately determined from the design day run. It is defined as the difference between EnergyPlus output "Zone/Sys Air Temperature" and the AHU SAT at the peak airflow condition. This is calculated separately for the middle floor core, and then averaged over the middle floor perimeter zones, for the zone design SAT for the core and perimeter zones respectively.

Design days for each exposure

This sizing strategy can be improved for both the peak airflow calculation and the thermal decay calculation. In order to better estimate the annual peak airflow and thermal decay for each exposure, a special design condition for each exposure can be added to the design day simulation. For example, the south zone peaks during the winter, due to a low sun angle.

ATU and number of diffusers sizing

The following procedure is used to size the ATUs for UFAD systems.

Select default diffuser type – Certain diffuser types for which we have laboratory test data for phi-gamma are included as defaults in EnergyPlus UFAD code. Table 2 summarizes the default types and their characteristics. Other types and/or modification of the defaults can be specified via the Custom command; users should contact the CBE team before attempting this, however. Note that the only factor that is not specified in code is the diffuser design airflow which is applied via the VBA macro procedures in the Excel use interface. [And in python code in the JEPlus parametric generator]

1) Specify the design Phi and "fake" thermal decay (oversizing ATU SAT) – This parameter provides a means for accurately sizing both the ATU and the number of diffusers

2) Run initial "oversizing" design day with autosizing on – Run all of the design days (see above) with the oversizing ATU SAT and design phi. This yields an accurate ATU inlet temperature for all zones since the airflow will be modulated to something less than the large box size.

3) Run second final sizing run with autosizing off? - From previous run, enter the ATU SATs and rerun design day with design phi. This provides an accurate box size.

4) Number of diffusers - Once the peak airflow (ie. Box size) is determined, the number of diffusers for each zone is found by dividing the design airflow by the diffuser nominal design airflow (See Table 2).

   a) York/JCI systems - The design phi is updated to correspond to the phi determined from the gamma-phi equation at the gamma for the peak design condition. As noted in Table 2, the MIT2 gamma-phi correlation is assumed represent all diffusers for York/JCI products. This phi is then fixed for all runs; i.e., the gamma-phi calculation is overridden.

   The design number of diffusers can also be entered by the user to override the design calculation above (e.g., to study impact of increasing number of diffusers). When this is done for the Walnut/JCI products it assumes that the plenum pressure has been lowered to provide a lower design volume; the constant phi is then entered corresponding to the new design gamma based on the user entered number of diffusers.

   b) For other diffusers, the phi is calculated hourly from the gamma-phi equation after the design number of diffusers is entered.

   c) For linear bar grilles the total length of n diffusers should not exceed the length of the zone; there is no check for this.

5) Conduct annual or other runs – The interface (and python code) allows for making "operating runs" or "design runs". In the case of operating runs, the sizing procedure is run once and then all subsequent runs are made with the same sizing results.

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Webster, T., F. Bauman, F. Buhl, A. Daly. 2008. "Modeling Of Underfloor Air Distribution (UFAD) Systems," *Proceedings, SimBuild 2008*. Berkeley CA. July, 8 pp.