

PART 1: Underfloor Air Distribution and Thermal Displacement Ventilation

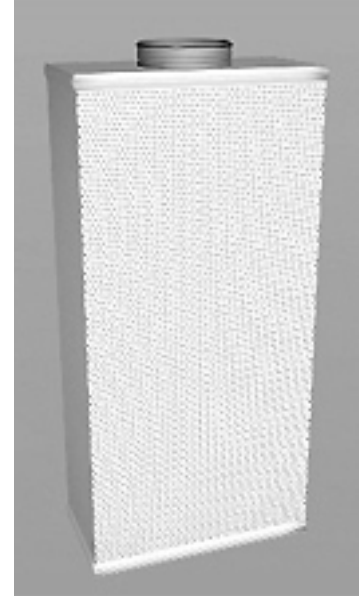
Underfloor air distribution (UFAD) and thermal displacement ventilation (TDV) have become increasingly common in commercial new construction because they are energy-efficient, enhance indoor air quality, and increase flexibility for space reconfiguration. However, conflicting opinions exist concerning the benefits of UFAD and TDV. This often leads to inappropriate analysis and unrealistic customer expectations. There are many different notions regarding the energy efficiency of UFAD and TDV systems, with some people claiming that these systems save little or no energy, while others suggest that they can cut HVAC energy usage by fifty percent or more. To help the energy modeler evaluate the energy benefits of UFAD and TDV, this simulation guidebook identifies the key characteristics that distinguish UFAD and TDV systems from traditional overhead systems and presents a logical, engineering-based method for analyzing UFAD and TDV with DOE-2-based simulation programs.

What are Underfloor Air Distribution and Thermal Displacement Ventilation?

This simulation guidebook is concerned with methods for analyzing air distribution systems that deliver cooling and heating air at floor level instead of from the ceiling. An example of such a system is underfloor air distribution (UFAD), where conditioned air is delivered at a moderate velocity (650 to 800 feet-per-minute) via a 10" to 16" plenum space underneath an access floor system (Figure 1a). Another example is a Thermal Displacement Ventilation (TDV) system that delivers supply air horizontally at low velocity (50 to 100 feet-per-minute) from wall-mounted diffusers without using an underfloor plenum (Figure 1b).

FIGURE 1:
(A) A TYPICAL UNDERFLOOR AIR DISTRIBUTION SYSTEM CONSISTS OF A RAISED ACCESS FLOOR, A 10" TO 16" UNDERFLOOR PLENUM, AND AIR DELIVERY DIFFUSERS. (B) A THERMAL DISPLACEMENT VENTILATION SYSTEM DELIVERS LOW VELOCITY AIR AT FLOOR LEVEL. AN ACCESS FLOOR IS NOT USUALLY EMPLOYED FOR SUCH A SYSTEM.

Diffuser for Underfloor Air Distribution System¹ (1a-left) versus Diffuser for Thermal Displacement System² (1b-right)



While there are differences in the performance characteristics between UFAD and TDV systems, the modeling methodology described in this simulation guidebook applies to both. Energy modelers must exercise judgment to adjust the methods to suit either system. Guidelines are provided throughout this guidebook that can be applied to account for differences between the two systems.

What differentiates UFAD and TDV systems from overhead distribution systems?

Traditional space conditioning systems supply heated or cooled air from diffusers mounted in a suspended ceiling grid. The design assumption made is that supply air completely mixes with the air in the room, and as a result, all of the air within the conditioned space reaches a homogeneous temperature (Figure 2).

Designers go to great effort to select diffusers that promote this mixing effect so that cold air does not “dump” onto the occupants below. In an overhead mixing system, cold supply air mixes with hot air that accumulates near the ceiling as a result of heat generated

¹ Source: Tate Access Floors

² Source: Halton Group

by people, lights, and equipment. While an overhead mixing system concept can provide good occupant comfort, it wastes energy by providing comfortable conditions from the floor all the way to the ceiling. It would be more efficient to limit the distribution of heated or cooled air only to the lower volume (for example, up to seven feet above the floor) of the room where the occupants are located.

Overhead Air Delivery Provides Homogeneous Temperature Distribution³

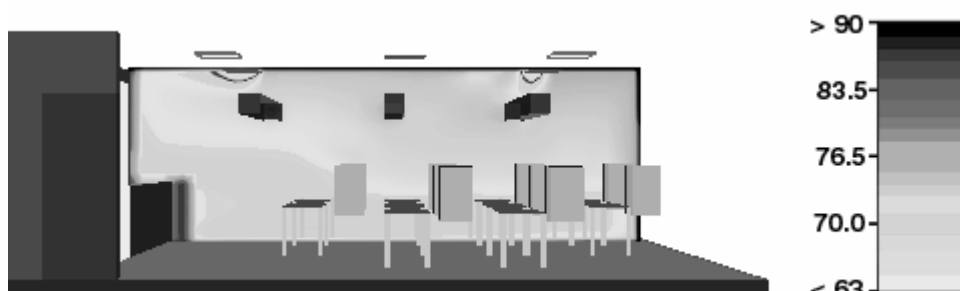


FIGURE 2:
TRADITIONAL OVERHEAD AIR DISTRIBUTION SYSTEMS ARE DESIGNED TO COMPLETELY MIX SUPPLY AIR WITH ROOM AIR. THE GOAL IS TO PROVIDE A UNIFORM TEMPERATURE DISTRIBUTION FROM FLOOR TO CEILING.

Thermal Displacement Ventilation Encourages Stratification⁴



FIGURE 3:
BY DELIVERING COOL AIR AT FLOOR LEVEL AND DRAWING WARMER AIR FROM THE CEILING, TDV ENCOURAGES THERMAL STRATIFICATION.

Barriers to Modeling UFAD and TDV in DOE-2

Uniform temperatures are assumed throughout the entire conditioned space

Most simulation programs based on DOE-2.1e or DOE-2.2 (such as eQUEST and EnergyPro) determine space cooling loads as a summation of all heat losses and heat gains within a space, without regard to how the loads are influenced by airflow patterns and the buoyancy of warm air. Stated another way, most simulation programs are not aware that hot air rises, and therefore assume a uniform temperature throughout the conditioned volume.

³ Source: CTG Energetics, Inc.

⁴ Source: CTG Energetics, Inc.

For example, consider the following internal load calculation:

People (sensible + latent)	
30 occupants x 500 Btu/hr-person	15,000 Btu/hr
Lights	
900 SF x 1.5 W/SF x 3.413 Btu/hr-W	4,608 Btu/hr
Equipment	
900 SF x 1.0 W/SF x 3.413 Btu/hr-W	3,072 Btu/hr
<hr/>	
TOTAL	22,680 Btu/hr

Most simulation programs calculate the required cooling capacity to meet these internal loads as the sum of the loads. The fact that hot air rises (producing warmer temperatures near the ceiling and cooler temperatures near the floor) is not accounted for. For overhead mixing-type air distribution this approach is satisfactory because overhead diffusers are selected and placed to promote mixing of supply air and room air. The flow of supply air from the ceiling pushes the hot air near the ceiling down to the level of occupants.

Calculation of cooling loads isn't available

When conditioned air is delivered at floor level at low velocity, it does not significantly mix with the hot ceiling air. Accordingly, floor-supplied air is not as disruptive to thermal stratification as overhead delivery (Figure 3). This is advantageous because the hot ceiling air can be drawn directly into the return air system and exhausted from the space instead of neutralized by mixing with cold air. The effect of thermal stratification is that cooling loads are reduced relative to those of overhead delivery systems, but most simulation programs do not reflect this change. Referring to the previous load calculation example, a UFAD system would reduce the cooling load resulting from internal heat gains by nearly 50 percent. More complex and time-consuming analysis methods, such as computational fluid dynamics (CFD), must be employed if one wishes to calculate cooling loads that account for thermal stratification. In most cases it is not practical to perform CFD analysis, and such analysis cannot be submitted to show Title 24 compliance.

THE EFFECT OF THERMAL STRATIFICATION IS THAT COOLING LOADS ARE REDUCED RELATIVE TO THOSE OF OVERHEAD DELIVERY SYSTEMS, BUT MOST SIMULATION PROGRAMS DO NOT REFLECT THIS CHANGE.

Thermal Displacement Ventilation and Underfloor Air Distribution Modeling

The strategy for modeling TDV and UFAD systems is to move a portion of the heat gain from people, lights and equipment from the conditioned space to an unconditioned plenum space. While project-specific information about how much of each internal load should be apportioned to the plenum is highly desirable, such data is infrequently available. Table 1 provides reasonable estimates for both UFAD and TDV systems. Figure 4 and Figure 5

show how lighting and receptacle loads would be redistributed in a typical underfloor air system. A similar approach would be employed for occupant heat gain.

Internal Load Distribution Values for Typical Underfloor Air and Thermal Displacement Ventilation Encourages Systems⁵

Load Component	Percent Load to Space		Percent Load to Plenum	
	Underfloor Air Distribution	Thermal Displacement Ventilation	Underfloor Air Distribution	Thermal Displacement Ventilation
People	75%	67%	25%	33%
Lights	67%	50%	33%	50%
Equipment	67%	50%	33%	50%

Source: CTG Energetics, Inc.

TABLE 1

Room Cross Section with Loads in Space⁶

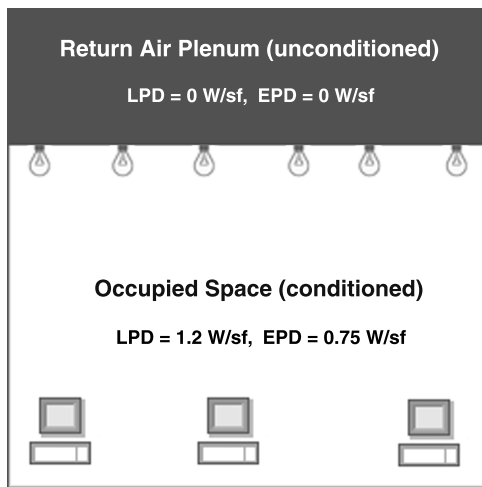


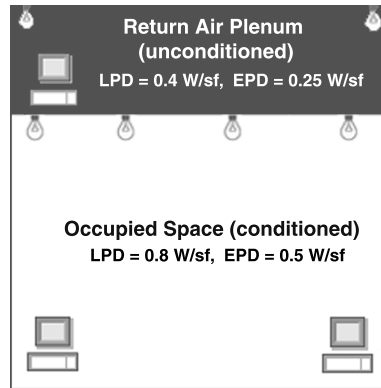
FIGURE 4:
CROSS-SECTION VIEW OF TYPICAL ROOM SHOWING CONDITIONED AND UNCONDITIONED SPACES, WITH TYPICAL LIGHTING POWER DENSITY (LPD) AND EQUIPMENT POWER DENSITY (EPD). THE TOTAL INTERNAL HEAT GAIN FROM THESE SOURCES IS 1.95 W/FT².

⁵ Source: CTG Energetics, Inc.
⁶ Source: CTG Energetics, Inc.

Room Cross Section with Loads in Plenum⁷

FIGURE 5:

TO SIMULATE THE REDUCED HEAT GAIN FROM LIGHTING AND RECEPTACLE LOADS, MUCH OF THE LOAD IS REASSIGNED TO THE UNCONDITIONED RETURN AIR PLENUM. THE EFFECT IS THAT COOLING LOADS IN THE CONDITIONED SPACE ARE REDUCED WHILE THE RETURN AIR TEMPERATURE IS INCREASED. THE TOTAL GAIN FROM LIGHTS AND RECEPTACLES IS UNCHANGED; HOWEVER, IT HAS BEEN APPORTIONED DIFFERENTLY BETWEEN CONDITIONED AND UNCONDITIONED SPACE.



Modeling Issues

There are a number of issues that must be considered when modeling thermal displacement or underfloor air distribution systems. These issues include:

- System Selection
- Supply Air Temperature
- Dehumidification
- Air Volume
- Static Pressure
- Economizer Controls
- Building Skin Loads
- Perimeter Systems

⁷

Source: CTG Energetics, Inc.

System Selection

The energy modeler must take care to choose the appropriate HVAC system type from those available within the simulation program. The exact system choice should reflect whether the system operates in constant or variable-volume fashion, the source of heating and cooling, and the way that outside air ventilation is managed. The following are examples of potential system selections:

Example #1: Access Floor System with Manually Adjustable Diffusers. This sort of system, which may include a large number of round, manually adjustable “swirl” diffusers (approximately one diffuser per 75 to 100 ft² of conditioned area), has become increasingly common in office buildings. Because occupants have some control over the airflow in their workspace - but cannot completely shut off the air supply - the system operates essentially as a variable-air-volume system with a high minimum airflow rate. Such systems are most commonly employed as part of a chilled water cooling system. In the DOE-2 simulation environment, such a system could be modeled using system type VAVS (variable-air-volume, with chilled water cooling). The minimum airflow rate (MIN-CFM-RATIO) would be set high to reflect the diversity of loads in the conditioned space and also the limited turndown offered to occupants. It is common for turndown ratios to be 70 to 80 percent of full flow, though modeling assumptions should be verified with the HVAC engineer. In many cases, a 100% outside-air-economizer cycle will be employed and the specific program inputs should reflect the fact that the cooling requirements can be met using warmer air than with overhead systems (i.e. 64°F to 67°F).

Example #2: Access Floor System with Thermostatically Controlled VAV Zone Terminals. This system is similar in some respects to Example #1, but the large number of “swirl” diffusers is replaced with a reduced number of thermostatically-controlled VAV terminals located in the underfloor plenum. Zoning for such systems is often comparable to overhead systems in terms of the average area per zone. Such systems usually offer higher turndown than the manual “swirl” diffusers and are automatically controlled based on space temperature. As a result, the minimum airflow ratio (MIN-CFM-RATIO) will frequently be lower with this air distribution strategy. Reviewing the zone schedule prepared by the mechanical engineer should provide information about the minimum airflow for each VAV terminal. Without such data, it is reasonable to assume a minimum airflow ratio of 50 percent until more detailed information is available.

Example #3: Thermal Displacement System with Constant Volume Delivery. This system design is most frequently used in classrooms or other assembly areas. A common configuration for thermal displacement systems consists of four-pipe fan coil units for each zone (or classroom), with a central air-handling unit distributing outside air to each unit. The fan coil units deliver constant volume supply air horizontally at low velocities from wall-mounted diffusers. The system can be modeled in DOE-2 using system type FPFC (four-pipe fan coil). The energy model system inputs should reflect the supply air temperature and volume design conditions associated with TDV, and fan energy inputs for each fan coil should account for the contributions from the central outside air supply unit.

THE ENERGY MODELER
MUST TAKE CARE TO
CHOOSE THE APPROPRI-
ATE HVAC SYSTEM TYPE
FROM THOSE AVAILABLE
WITHIN THE SIMULATION
PROGRAM.

Supply Air Temperature

Since TDV and UFAD generally introduce conditioned air in close proximity to the building occupants, the air is delivered at a temperature only slightly (5°F to 10°F) below space temperature set points. This corresponds to a 64°F to 67°F supply air temperature set point (MIN-SUPPLY-T or COOL-SET-T) as opposed to a 55°F set point for traditional mixing systems.

Dehumidification

Due to the elevated supply air temperature associated with TDV and UFAD, the mechanical designer must give close attention to humidity control for these systems. With the exception of cool, dry climates, cooling coils provide inadequate removal of latent load when cooling to only 64°F or 67°F. Consequently, most TDV and UFAD designs need to implement supplemental humidity control features to avoid the decreased comfort and indoor air quality associated with high space humidity conditions. In a common TDV or UFAD humidity control scheme, the chilled water coil cools a mixture of outside air and return air down to 55°F, and this conditioned air is then mixed with the remainder of the return air to increase the temperature back up to the supply air temperature set point. Although the limitations of DOE-2 prevent the accurate modeling of this humidity control scheme, energy modelers should keep in mind that this form of humidity control will achieve less energy savings than projected by a DOE-2 model with a high supply-air temperature set point.

WITH THE EXCEPTION OF
COOL, DRY CLIMATES,
COOLING COILS PROVIDE
INADEQUATE REMOVAL OF
LATENT LOAD WHEN COOL-
ING TO ONLY 64°F OR
67°F.

Air Volume

Design supply airflow calculations for the space must account for both the elevated supply air temperature and the redistribution of a portion of the occupant, plug and lighting loads from the space to the return air. Ignoring the high supply-air temperature for TDV and UFAD systems will result in an underestimation of supply air volume, and neglecting to redistribute a portion of the space loads to the plenum will result in supply air-flow rates that are up to two times greater than the amount required to condition the space. Typically, supply air flows for a true TDV system exceed those of a corresponding overhead mixing system by only five to twenty percent.⁸ Supply air flow rates for UFAD systems range from twenty-five percent less to fifteen percent more than traditional overhead systems.⁹

Static Pressure

In most UFAD systems, the underfloor plenum serves as the primary source of air distribution. Consequently, UFAD systems generally use far less ductwork than corresponding overhead systems, resulting in reduced static pressure at the supply fans when compared against standard overhead systems. However, due to the wide variance in UFAD

⁸ “Underfloor Air Distribution and Access Floors.” Energy Design Resources Design Brief.

⁹ Webster, Tom, Bauman, Fred, and Reese, Jim. “Underfloor Air Distribution: Thermal Stratification.” ASHRAE Journal. May 2002. Vol. 44, No. 5, Pg. 34.

design, energy analysts should confirm estimated values for static pressure with the mechanical designer prior to modeling savings associated with reduced fan static pressure. The fan energy savings linked to lower fan static pressures will generally not be reflected in Title-24, since the standard case changes with the proposed case for inputs related to fan power.

Economizer Controls

TDV and UFAD systems can often take advantage of increased hours of economizer operation due to the higher temperature of air delivered to the space. In most California climate zones, raising the supply air temperature from 55°F to 65°F can extend economizer operation by 2,000-2,500 hours per year.¹⁰ However, the humidity control requirements in many of these climates will limit the hours of additional economizer operation, resulting in reduced free cooling benefits. In climate zones that require additional dehumidification, the economizer operation must be integrated with the humidity control to maintain proper humidity conditions. This requires differential enthalpy-based economizer operation to ensure that the humidity of the outside air remains lower than that of the return air. In DOE-2, differential enthalpy control is modeled using the ENTHALPY keyword for OA-CONTROL at the system level.

IN MOST CALIFORNIA CLIMATE ZONES, RAISING THE SUPPLY AIR TEMPERATURE FROM 55°F TO 65°F CAN EXTEND ECONOMIZER OPERATION BY 2,000-2,500 HOURS PER YEAR.

Building Skin Loads

If return grilles are located directly above the windows in perimeter spaces served by UFAD or TDV systems, a significant portion of the convective cooling load associated with the building skin can be funneled directly into the return air plenum.¹¹ A precise energy model for UFAD and TDV systems can account for the energy savings associated with this phenomenon by reapportioning some of the glazing and exterior walls in the occupied space to the adjacent plenum. However, this methodology may result in the loss of legitimate automated daylighting control savings in DOE-2-based programs. Furthermore, this modeling approach has not yet been approved for demonstrating UFAD system savings in 2005 Title-24.

Perimeter Systems

Perimeter system approaches vary widely for both UFAD and TDV systems. In some cases, perimeter underfloor air plenums for UFAD systems are separated from interior underfloor air plenums with dividers; in another approach, underfloor ductwork provides perimeter spaces with a separate source of supply air, and sometimes perimeter spaces are entirely served by overhead systems. Baseboard heating can also be provided as the primary heating source for perimeter zones served by TDV or UFAD systems. The Underfloor Air Distribution Design Guide (ASHRAE, 2003) provides a good overview of

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“The Case for TDV in California Schools.” California Energy Commission’s Public Interest Energy Research (PIER) Program. http://www.archenergy.com/ieq-k12/thermal_displacement/thermal_displace_background.htm

¹¹

Bauman, Fred S. and Daly, Alan. Underfloor Air Distribution (UFAD) Design Guide. Atlanta: ASHRAE, 2003.

the range of perimeter system designs commonly applied in conjunction with underfloor air distribution. Energy analysts should use their judgment to select the type of space heating and zone terminal units in DOE-2 that most closely represent the perimeter system design for their project.

Modeling Methodology

In native DOE-2 (the BDL input file),¹² energy modelers can apply the modeling strategies described above for TDV and UFAD. However, strict alternative calculation methods (ACMs) published by the CEC for the 2001 Title-24 standards have prevented the simple application of TDV and UFAD modeling strategies to the Title-24 compliance modules of EnergyPro and eQUEST. The ACMs defined for 2005 Title-24 standards do permit software developers to offer an optional system type for Underfloor Air Distribution (UFAD), which will make accommodations for the user to assign a percentage of the occupant, lighting and plug loads to the return air plenum.⁶ The following step-by-step TDV modeling methodologies for DOE-2, EnergyPro and eQUEST assume that a UFAD system has not yet been implemented.

Modeling TDV and UFAD in DOE-2.1e

In native DOE-2.1e, energy analysts can define TDV or UFAD systems using the following process:

Step 1. Define at least one (but no more than three)⁷ return air plenum(s) for each system. In cases where the building design defines a return air plenum, model the plenum as drawn in the plans. If no return air plenum is defined in the plans, add a plenum with a height of three feet, and an area equal to the building area served by the system.

Step 2. For each space served by a TDV or UFAD system, multiply the number of occupants, the lighting power density, and the equipment power density by the Percent Load to Space factor for each load defined in Table 1. If occupant density is defined using the AREA/PERSON keyword, divide the AREA/PERSON by the Percent Load to Space factor for people. Input the revised occupant, lighting, and equipment data in each space.

Step 3. Input the lighting, equipment and occupant schedules defined for the occupied spaces into the return air plenum(s).

¹² This simulation guidebook refers to “Native DOE-2” as the BDL (or plain text) DOE-2 input file. This is contrasted against a graphical user interface program such as EnergyPro, VisualDOE, or eQUEST.

¹³ “NonResidential ACM Manual.” October 2003 Draft Language, Commission Proposed Standards. California Energy Commission.

¹⁴ DOE-2.1e does not allow more than three plenums per zone.

COMMON SIMULATION
SOFTWARE

THIS GUIDEBOOK USES
RESEARCH GENERATED
FROM THE FOLLOWING
ENERGY SIMULATION
SOFTWARE PACKAGES:
ENERGYPRO V. 3.142,
EQUEST V. 3.44 WITH
DOE2.2 RELEASE 42K6,
AND
DOE-2.1E RELEASE 134.

KEEP IN MIND THAT THIS
SOFTWARE IS CONSTANTLY
UPDATED. REVIEW THE
DOCUMENTATION OF
LATER RELEASES FOR ANY
CHANGES TO SOFTWARE
INPUTS OR KEYWORDS
THAT MIGHT IMPACT THE
MODELING METHODOLO-
GY DISCUSSED IN THIS
SIMULATION GUIDEBOOK.

Step 4. Calculate the AREA/PERSON, lighting power densities, and equipment power densities using equations 1, 2, & 3 shown in Figure 6.

Step 5. Insert the values calculated in step four above for plenum area per person, equipment power density, and lighting power density into the plenum using DOE-2 keywords AREA/PERSON, EQUIPMENT-W/SQFT, and LIGHTING-W/SQFT respectively.

Step 6. Select the DOE-2 system type in accordance with the system selection instructions above. Set the return air path (RETURN-AIR-PATH) to plenum (PLENUM-ZONES).

Step 7. For variable volume systems, set the MIN-CFM-RATIO for each zone in accordance with the system selection instructions above; if the supply fan design includes a variable speed drive, be sure to set the FAN-CONTROL equal to FAN-EIR-FPLR, and the FAN-EIR-FPLR equal to ANY-FAN-W/VSD.

Step 8. Set the keyword for cooling supply air temperature (MIN-SUPPLY-T or COOL-SET-T) as defined in the design (generally 64-67 °F).

Step 9. Model the system economizer controls as designed. In most California climate zones, a differential enthalpy economizer should be used (OA-CONTROL = ENTHALPY).

Equations Used in Modeling UFAD and TVD

$$\text{AREA/PERSON}_{Pln} = \frac{\left[\text{AREA}_{Pln} \times \sum_{spaces} \frac{\text{AREA/PERSON}_{space}}{\text{AREA}_{space}} \right]}{\left[\% \text{ Load to Plenum} \right]_{People}} \tag{1}$$

$$\text{E - W}_{Pln} = \frac{\left[\sum_{spaces} \text{E - W}_{space} \times \text{AREA}_{space} \right] \times \left[\% \text{ Load to Plenum} \right]_{Equip}}{\text{AREA}_{Pln}} \tag{2}$$

$$\text{LIGHTING - W/SQFT}_{Pln} = \frac{\left[\sum_{spaces} \text{LIGHTING - W/SQFT}_{space} \times \text{AREA}_{space} \right] \times \left[\% \text{ Load to Plenum} \right]_{Lights}}{\text{AREA}_{Pln}} \tag{3}$$

FIGURE 6:
THESE EQUATIONS CALCULATE AREA PER PERSON, LIGHTING POWER DENSITY, AND EQUIPMENT POWER DENSITY FOR UNDERFLOOR AIR DISTRIBUTION SYSTEM ENERGY MODELING.

- Pln = Return air plenum serving the selected spaces
- AREA/PERSON = Area per person
- Equip = Equipment
- E-W = Equipment watts per square foot
- Lighting-W/SQFT = Lighting watts per square foot
- % Load to Plenum is the percentage defined in Table 1.

DOE-2.1e Sample Text for an Access Floor System with Manually Adjustable Diffusers

FIGURE 7:
 THE SYSTEM THAT CORRESPONDS TO THIS SAMPLE TEXT SERVES THREE SPACES AND IS MODELED WITH A RETURN AIR PLENUM TO SIMULATE THE IMPACTS OF UNDERFLOOR AIR DISTRIBUTION. ONE THIRD OF THE LIGHTING AND EQUIPMENT LOADS AND ONE QUARTER OF THE OCCUPANT LOAD IS REDISTRIBUTED TO THE PLENUM. THE SYSTEM OPERATES WITH A VARIABLE SPEED FAN AND A DIFFERENTIAL ENTHALPY ECONOMIZER AND SUPPLIES AIR AT AN ELEVATED SUPPLY AIR TEMPERATURE. THE MINIMUM AIR FLOW TO THE SPACE IS ASSUMED TO BE 75%.

\$ SAMPLE SPACE-DEFINITION - OFFICE WITH UNDERFLOOR AIR CONDITIONING:

\$ E. Office \$

ZONE-1 = SPACE

ZONE-TYPE = CONDITIONED

PEOPLE-SCHEDULE = SCHED-26

LIGHTING-SCHEDULE = SCHED-25

EQUIP-SCHEDULE = SCHED-24

INF-SCHEDULE = SCHED-23

AREA/PERSON = 133 \$ AREA/PERSON = 100/75% = 133

\$ where 75% = Percent to space factor

PEOPLE-HG-SENS = 250

PEOPLE-HG-LAT = 200

LIGHTING-W/SQFT = 0.871 \$ LIGHTING-W/SQFT = 1.3 * 67% = 0.871

\$ where 66% = Percent to space factor

EQUIPMENT-W/SQFT = 1.0 \$ EQUIPMENT-W/SQFT = 1.5 * 67% = 1.0

\$ where 66% = Percent to space factor

EQUIP-SENSIBLE = 1.0

EQUIP-LATENT = 0.0

INF-METHOD = AIR-CHANGE

AIR-CHANGES/HR = 0.2027

AREA = 1710

VOLUME = 17100

..

\$ SAMPLE ZONE-DEFINITION - OFFICE WITH UNDERFLOOR AIR CONDITIONING, UFAD DIFFUSERS WITH VSD ON FANS

ZONE-1 = ZONE

ZONE-TYPE = CONDITIONED

DESIGN-HEAT-T = 70.0

DESIGN-COOL-T = 74.0

THROTTLING-RANGE = 4.0

HEAT-TEMP-SCH = SCHED-20

COOL-TEMP-SCH = SCHED-19

OA-CFM-PER = 15

SIZING-OPTION = ADJUST-LOADS

INDUCED-AIR-ZONE = ZONE-1

TERMINAL-TYPE = SVAV

MIN-CFM-RATIO = 0.75 \$ MINIMUM CFM SET TO 75%

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$ SAMPLE PLENUM SPACE DEFINITION - OFFICE WITH UNDERFLOOR AIR
CONDITIONING
(SERVES THREE SPACES LIKE THE E. OFFICE SPACE ABOVE)
$ Plenum Zone - Space $
PlnZone = SPACE
  ZONE-TYPE = PLENUM
  PEOPLE-SCHEDULE = SCHED-26      $Same schedule defined for space occ
  LIGHTING-SCHEDULE= SCHED-25     $Same schedule defined for space ltg
  EQUIP-SCHEDULE  = SCHED-24      $Same schedule defined for space
                                  equip
  AREA/PERSON    = 400            $ AREA/PERSON = 100/25% = 400
                                  $ where 25% = Percent to plenum factor

  PEOPLE-HG-SENS = 250
  PEOPLE-HG-LAT  = 200
  0.429LIGHTING-W/SQFT = 0.429    $ LIGHTING-W/SQFT = 1.3 * 33% =
                                  $ where 33% = Percent to space factor

  EQUIPMENT-W/SQFT = 0.5          $ EQUIPMENT-
                                  $ where 33% = Percent to space factor

  EQUIP-SENSIBLE = 1.0
  EQUIP-LATENT   = 0.0
  AREA           = 5130
  VOLUME         = 15390
..

$SAMPLE PLENUM ZONE DEFINITION - OFFICE WITH UNDERFLOOR AIR CONDITIONING
(SERVES THREE SPACES LIKE THE E. OFFICE SPACE ABOVE)
PlnZone = Zone
  ZONE-TYPE = PLENUM
..

$ SAMPLE SYSTEM DEFINITION - OFFICE WITH UNDERFLOOR AIR CONDITIONING
SYSTEM-1 = SYSTEM
  SYSTEM-TYPE      = VAVS
  ZONE-NAMES = (
    ZONE-1,
    ZONE-2,
    ZONE-3,
  )
  PLENUM-NAMES = (
PLNZONE,)
  RETURN-AIR-PATH = PLENUM-ZONES
  FAN-SCHEDULE    = SCHED-18
  OA-CONTROL      = ENTHALPY      $Differential enthalpy economizer

```

ECONO-LOCKOUT	=	NO	
MAX-OA-FRACTION	=	1.0	
FAN-CONTROL	=	FAN-EIR-FPLR	
FAN-EIR-FPLR	=	ANY-FAN-W/VSD	\$Variable Speed Fan Controls
MIN-FAN-RATIO	=	0.30	
SUPPLY-CFM	=	5000	
SUPPLY-KW	=	0.001	
SUPPLY-DELTA-T	=	2.79	
DUCT-AIR-LOSS	=	0.00	
COOL-CONTROL	=	RESET	
COOL-RESET-SCH	=	COLD-DECK-RESET	
COOLING-CAPACITY	=	153000	
COOL-SH-CAP	=	122400	
MIN-SUPPLY-T	=	65	\$ ELEVATED SUPPLY AIR TEMPERATURE
..			

Modeling TDV and UFAD in eQUEST

If eQUEST users are not using the Title-24 compliance module, they can define TDV and UFAD systems using the following process:

Step 1. If the building design includes a return air plenum, model the plenum in the Building Footprint screen of the eQUEST wizard by selecting floor-to-floor height and floor-to-ceiling height as shown in the plans. If no return air plenum is defined in the plans, a plenum must be defined in the eQUEST detailed edit interface. The plenum should have a height of three feet, and an area equal to the building area served by the system.

Step 2. From the **Occupied Loads by Activity Area** screen of the eQUEST wizard, multiply the installed lighting power density and the equipment power density for each occupancy type by the **Percent Load to Space** factor for each load defined in Table 1. Input the revised lighting and equipment data in each space.

Step 3. From the **HVAC System Definitions** screen, select the system type as outlined in the system selection guidelines above.

Step 4. From the **HVAC Zones: Temperatures and Air Flows** screen, set the supply air temperature as defined in the plans. For VAV systems, define the VAV minimum flow for both core and perimeter spaces, as described in the system selection guidelines.

Step 5. Switch to **Detailed Edit Mode** by selecting **File / Mode / Detailed Edit Mode**.

Step 6. From the Internal Loads module, select the Spreadsheet tab, and then select Occupancy. Divide the AREA/PERSON for each space by the Percent Load to Space factor for people defined in Table 1. Input the revised data in each space. In the return air plenum space, select the occupancy schedule to be the same as the occupied spaces.

Use equation 1 to calculate the AREA/PERSON for the plenum.

Step 7. Again from the **Spreadsheet** tab, select **Lighting**. In the return air plenum, select the same lighting schedule as is defined for the occupied spaces. Use equation 3 to calculate the lighting power density for the plenum. Repeat this process for **Equipment**, using equation 2 to calculate the plenum equipment power density.

Sample eQuest Input Screen

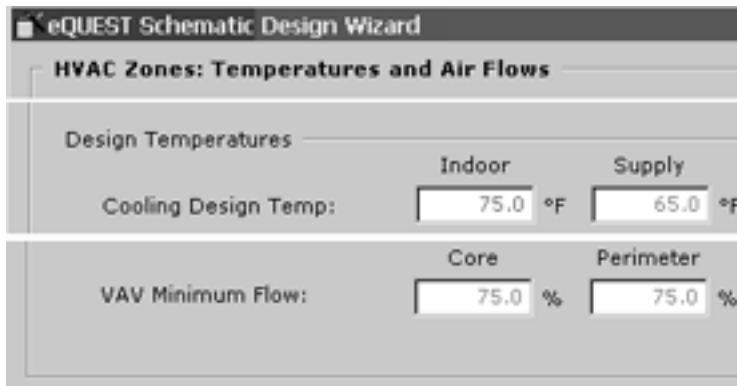


FIGURE 8:
EQUEST ZONE TEMPERATURE AND AIR FLOW SELECTIONS FOR AN ACCESS FLOOR SYSTEMS WITH MANUALLY ADJUSTABLE DIFFUSERS.

Defining TDV or UFAD for Title-24 Comparisons

When using the Title-24 compliance module, eQUEST users must define TDV or UFAD inputs using a slightly more complex process:

Step 1. Complete steps 1, 3, and 4 described above for the eQUEST non-compliance TDV and UFAD modeling process.

Step 2. Run the Title-24 simulation using the **Perform Compliance Analysis** option.

Step 3. Load the DOE-2 input file for the Title-24 proposed case (titled [FileName]- T24 Proposed Building.inp) into eQUEST by selecting **File / Open / Files of Type / DOE-2.2 BDL Input Files**, and then selecting the appropriate file.

Step 4. From the **Internal Loads** module, select the **Spreadsheet** tab, and then select **Occupancy**. Divide the AREA/PERSON for each space by the **Percent Load to Space** factor for people defined in **Table 1**. Input the revised data in each space. In the return air plenum space, select the occupancy schedule to be the same as the occupied spaces. Use equation 1 to calculate the AREA/PERSON for the plenum.

Step 5. Again from the **Spreadsheet** tab, select **Lighting**. Multiply the installed lighting power density for each space by the **Percent Load to Space** factor for defined in **Table 1**. Input the revised lighting power density in each space. In the return air plenum, select the same lighting schedule as is defined for the occupied spaces. Use equation 3 to calculate

the lighting power density for the plenum.

Step 6. Repeat the process above for **Equipment** using the equipment power densities, equipment schedules, and calculating the plenum equipment power density with equation 2.

Step 7. Rerun the energy model to obtain a revised Title-24 proposed case. The Title-24 standard case should remain the same. As an error-checking routine, the energy modeler should confirm that the lighting and equipment energy usage for the original Title-24 proposed case is equal to those shown in the revised Title-24 proposed case.

Modeling TDV and UFAD in EnergyPro

An energy modeler can simulate TDV or UFAD with EnergyPro by:

- Inputting system inputs directly in EnergyPro;
- Using the EnergyPro Win/DOE module to generate a DOE-2 input file, and
- Revising the lighting, equipment, and occupant inputs directly in DOE-2. This step is explained below.

Revising the Lighting, Equipment, and Occupant Inputs Directly in DOE-2

Step 1. Define at least one but no more than three return air plenums for each system. In cases where the building design defines a return air plenum, model the plenum as drawn in the plans. If no return air plenum is defined in the plans, add a plenum with a height of three feet, and an area equal to the building area served by the system.

Step 2. Select the DOE-2 system type as outlined in the system selection instructions above. Set the cooling supply air temperature in the cooling tab for each system. Model the economizer type as designed (often differential enthalpy for TDV systems). For each system, confirm that any variable speed fans are appropriately defined under the **Fans** tab.

Step 3. For variable volume systems, select a zonal system from the mechanical tab for each zone, with minimum air flow set in accordance with the system selection guidelines.

Step 4. From the File menu, select **Calc Manager / Options / Win/DOE**. Confirm that Delete DOE files after run is unchecked.

Step 5. To generate the DOE-2 files, select **Calc Manager / Calculate**.

Step 6. From your EnergyPro Win/DOE directory, open the Title-24 proposed input file titled [FileName]-Proposed.doe (where filename is the name you entered for the project in EnergyPro).

Step 7. Complete steps 2-5 for Modeling TDV in DOE-2 as outlined above.

Step 8. Create a text file in your Win/DOE directory using the following syntax: doe21e

“[FileName]-Proposed.doe” [EnergyProWeatherPath]\[WeatherFile]. [FileName] represents the name of your project, [EnergyProWeatherPath] represents the path to the EnergyPro weather directory, and [WeatherFile] represents the name of the weather file used for your project. For example, the text file for a project titled Office and located in Sacramento, CA (climate zone 12) would contain the text doe21e “office-Proposed.doe” C:\EP3\Weather\CZ12RV2.WY2, assuming that the EnergyPro directory was located in C:\EP3.

Step 9. Change the extension of the text file to .bat to create a batch file that can run your project in MS DOS. (To run the simulation, navigate to the .bat file using either Windows Explorer or My Computer, and double-click on the .bat file)

Step 10. As an error-checking routine, the energy modeler should confirm that the lighting and equipment energy usage for the original Title-24 proposed case are equal to those shown in the revised Title-24 proposed case.