ADVANCED HVAC SYSTEMS FOR IMPROVING INDOOR ENVIRONMENTAL QUALITY AND ENERGY PERFORMANCE OF CALIFORNIA K-12 SCHOOLS

Draft / Final Research Report

CONSULTANT REPORT

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Introduction

The goal of the System Design Options task of the TDV project was to develop conceptual design solutions for displacement ventilation. The task started with a design charrette, which brought together leading architects and design engineers for K-12 schools in efforts to develop conceptual design solutions for displacement ventilation. Participants addressed issues such as air delivery options to displacement diffusers, location of HVAC equipment location and selection, and ductwork layout.

The results of the charrette were applied towards the development of conceptual design options for displacement ventilation, documented in the Outline Specification and Schematic Design Report (Deliverable 2.5b). CTG Energetics worked with Architectural Energy Corporation to define three HVAC system types for a typical classroom wing of eight classrooms. The designs included equipment specifications, schematic drawings, a comparison of advantages, disadvantages and limitations of each solution, and typical control sequences. The designs included packaged single zone DX units for each classroom, a packaged VAV multizone unit with VAV terminal units for classroom control, and a central plant solution.

This report covers additional HVAC design considerations for displacement ventilation systems. This report draws from completed research of the TDV project, the CFD analysis performed with Halton, and the results of the first demonstration classroom. The report addresses diffuser selection and layout, load calculations and system sizing and energy modeling options. The report also describes HVAC system requirements for displacement ventilation and control options.

For the design phase, this report covers design requirements for TDV, load calculation procedures, energy modeling and equipment selection. Architectural design requirements for TDV are based on prior research and the results of the CFD analysis performed under earlier tasks of this project. Recommendations for load calculations and system sizing are based on ASHRAE recommendations, CFD modeling results and monitored data. Energy simulation models must account for the thermal stratification in the space, to accurately predict the energy benefits of TDV.

For the construction phase, the report documents show typical diffuser locations and ductwork layout. The documents also provide control details and describe typical installation requirements. The intent of the documents is to focus on design details that are unique to displacement ventilation, particularly at the classroom level. The design procedure for the central chilled water plant, for instance, is essentially the same, whether overhead air distribution or displacement ventilation is used.

The design solutions and TDV design recommendations in this report form the basis for the subsequent design guidelines task in the TDV project (Task 2.9). A draft version of this report was developed in May 2005; the final report includes additional plan drawings, commissioning details, and revised energy modeling results.

Design Phase

Architectural Requirements

Unique architectural requirements for displacement ventilation should be specified early in the design phase. The CFD analysis and experimental mockup testing conducted for this project confirmed that displacement ventilation can provide good thermal comfort while handling design cooling loads up to 25 kBtu/h-ft^2. This means that displacement ventilation can maintain comfortable conditions for most California classroom load conditions without the use of supplemental cooling systems. However, there are a few architectural requirements for displacement ventilation, listed below.

- Ceiling Height – a minimum 9 ft high ceiling is adequate for displacement ventilation. Nine-foot ceiling heights are typical for new construction. From the CFD analysis, a 10 foot ceiling did not provide significant benefits, but a 12 foot ceiling did provide for benefits of stratification. Typical floor-to-floor heights for multi-story buildings will work for displacement ventilation. If the suspended ceiling is eliminated, there may be an opportunity to reduce floor-to-floor height. However, this will require strategies to minimize background noise from HVAC equipment.
- Building Envelope – no special modifications are required to the building envelope. TDV can effectively handle cooling loads for classrooms built to Title 24 standards. In inland and mountain climates, the use of double-paned, low-e windows will help reduce radiation from cold window surfaces, minimizing the potential for downdrafts that oppose displacement airflow.
- Equipment Location – the most common HVAC system type for California K-12 schools is packaged rooftop units for each classroom. There are no unique requirements for TDV in locating equipment. Background noise from mechanical equipment should be lower with TDV due to the lower air velocity.
- Air Distribution Methods – there are several options for air distribution for TDV. The most common option is to use two wall-mounted diffusers for each classroom. These can be mounted at the corners of an interior wall or at the quarter points along the wall. The diffusers can also be recessed into the wall. Another option is to integrate diffusers with the casework. A common pressurized supply plenum could deliver air to each classroom. This would require additional space in the building that is not common with modern school designs.
- Diffuser Location – the diffusers do take up a significant amount of classroom space and wall space. Two corner wall-mounted diffusers or semicircular diffusers mounted at the quarter-points along an interior wall will provide sufficient air distribution. A sheet metal cover is used to conceal flex ductwork that extends from the ceiling to the diffuser connection.

Load Calculations

An important first design step is determining the required supply air conditions for displacement ventilation.

Determining Required Airflow

TDV requires a different procedure for estimating the room loads in the occupied zone. Several procedures have been recommended:

1. ASHRAE design guidelines
2. REHVA design procedure
3. Energy Design Resources Simulation Guidebook
4. CFD analysis
5. UCSD displacement ventilation model (EnergyPlus)

The ASHRAE and REHVA design approaches use a procedure that is suitable for a manual calculation. The EDR design procedure can be used with a manual load calculation, or applied to a DOE-2 simulation. A common element among all approaches is a procedure to estimate the fraction of heat gains from lighting, building envelope, occupants and equipment that contributes to the load in the occupied zone. Once this is determined, the supply airflow and return air temperature can be easily determined from an energy balance.

CFD analysis is useful in predicting airflow patterns and detailed temperature profiles in the room. It is especially useful for specifying and locating diffusers to ensure thermal comfort. However, CFD is a time-intensive procedure that is not practical for load calculations.

EnergyPlus can also be used for determining the required system capacity. The user can choose between the UCSD displacement ventilation model and the Mundt nodal model. As part of this research project, the UCSD model has been evaluated for its use in modeling TDV for classrooms. The EnergyPlus models are described in more detail in the modeling section.

System Sizing for TDV

Once the supply airflow, supply air temperature and return air temperature are known, the sensible cooling load on the system (coil load) can be calculated in the same manner as with conventional systems. The equipment selected must be able to provide the sensible cooling capacity at design conditions.

A key difference with displacement ventilation is the higher supply air temperature. Conventional DX equipment cools the incoming air below its dewpoint temperature to provide dehumidification. For TDV, if a supply air temperature of 63-65°F is used off the coil, little or no dehumidification will occur. This has the effect of decreasing required cooling capacity significantly, at the expense of a higher indoor humidity. In inland valley and desert climates in California, outdoor humidity levels are below indoor design humidity levels throughout the year. Thus, outside ventilation air will also serve to offset latent loads from occupants. However, in southern California coastal climates, the outside air humidity is above indoor design levels at times. For these climates, the HVAC system used with displacement ventilation will need to cool a portion of the supply air to below its dewpoint temperature, to provide for dehumidification. Humidity control options are described later in this report.

For heating, the air distribution pattern is mixing ventilation, even if the low-velocity displacement diffusers are used. Thus, a conventional heating load calculation will be valid for TDV.

ASHRAE Design Guidelines: Sample Calculation

The ASHRAE TDV publication (Chen 2003) includes a procedure for calculating both the required supply air flow and supply air temperature for cooling. The procedure specifies a maximum allowable temperature difference between head and foot level, to provide proper thermal comfort.

An example of the ASHRAE load calculation for a representative classroom load conditions is shown in the table below. The example uses the design baseline of single-loaded classrooms for California Climate Zone 12, with a peak room sensible cooling load of 22,800 Btu/h. The example uses a space cooling setpoint of 74°F.

The ASHRAE design procedure uses the ASHRAE 55-2004 Standard limits for thermal stratification as a design criterion for displacement ventilation. The Standard defines a maximum allowable temperature difference between foot level (4”) and head level of seated occupants (40”) of 3.6°F. The procedure uses empirical load coefficients for estimating the fraction of heat gain from occupants, equipment and lighting that contribute to a load in the occupied zone.

Step 1. Calculate the load in the occupied zone.

$$Q_{occ} = a_{oe}\, Q_{oe} + a_l\, Q_l + a_{ex}\, Q_{ex}$$

Where Qocc, Qoe, Ql and Qenv are the loads to due occupants and equipment, loads due to lighting and envelope loads, respectively. Q_{occ} is the load in the occupied zone. The “delta T” is assumed to be 3.6°F from the ASHRAE 55-2004 Standard.

Step 2. An energy balance on the occupied zone is used to determine the required supply air flow rate:

$$Qocc = 1.1\ CFM\ (3.6°F)$$

Step 3. Once the supply airflow rate is determined, the supply air temperature is determined from correlations for convection near the floor surface.

Notice that for this procedure, the calculation of supply air flow rate did not require an assumption for supply air temperature. First, the air temperature near the floor is calculated by assuming the 3.6°F temperature difference:

$$T_{floor} = T_{sp} - 3.6°F$$

For example, for a space setpoint temperature of 74°F, the air temperature near the floor would be 71.4°F.

The supply air temperature is determined from a convection correlation defined by Mundt (Chen, p. 114). The convective and radiative heat transfer coefficients are assumed to be 1 Btu/h-ft^2 for this example.

Step 4. Determine return air temperature from an energy balance on the entire space.

Once the SAT and supply air flow rate are specificed, the return air temperature can be calculated from an energy balance on the entire space. With this step, a key assumption is made: the entire heat load is used in the space energy balance. In other words, under steady-state conditions, the heat extracted from the space equals the heat gain to the space.

Step 5. Specify outside air ventilation rate and design conditions to calculate system sensible loads.

The load on the HVAC system (coil load) is compromised of the space load and the load to cool outside ventilation air. In most cases, the HVAC system will use a mixture of outside air and recirculated, filtered air. Once supply air temperature, return air temperature, supply air flow rate and outside air ventilation rate are known, the system load calculation step is equivalent to the system load calculation for a mixing system.

The table below shows an example of the ASHRAE load calculation.

Table 1 – TDV Load Calculation for a typical Sacramento Classroom

	Overhead Mixing Air Distribution		Displacement Ventilation	
Step 1. Itemize loads—Estimate the load to the occupied zone for displacement ventilation.				
	Space Load		Occupied Zone Load	
Occupants and Equipment	8,300 Btu/h		x 0.295 = 2449 Btu/h	
Lighting	4,000 Btu/h		x 0.132 = 528 Btu/h	
Envelope	10,500 Btu/h		x 0.195 = 2048 Btu/h	
Space Load Subtotal (kBtu/h)	22,800 Btu/h		N/A	
Occupied Zone Load Subtotal	N/A		5,025 Btu/h	
Step 2. Airflow Calculation—Supply airflow is calculated from an energy balance on the occupied zone. The ΔT of 3.6 meets the ASHRAE Standard 55 comfort criterion. A space cooling setpoint of 74°F is assumed.				
ΔT	74 – 55 = 19F		3.6F (between 4” and 43” height)	
Supply Airflow	1090 cfm		1265 cfm	
Step 3. Determine SAT and RAT—DV Supply temperature is determined from ASHRAE design guidelines. Return air temperature is determined from an energy balance on the entire space.				
Room Setpoint	74 °F		74 °F	
SAT	55 °F		66.2 °F	
RAT	74 °F		82.7 °F	
Step 4. Calculate system load—Once the supply airflow, SAT and RAT are known, the required cooling capacity is determined in a similar manner to a mixing ventilation system.				
Outside air	600 cfm	100 °F DB	600 cfm	100 °F DB
Return air	490 cfm	74 °F DB	665 cfm	82.7 °F DB
Mixed air (entering coil condition)	1090 cfm	88.3°F DB	1265 cfm	90.9 °F DB
Supply air (leaving coil)	1090 cfm	55 °F	1265 cfm	66.2 °F
Sensible Cooling Capacity (Coil Load)	**39,940 kBtu/h**		**34,380 kBtu/h**	
Latent Occupant Load	4,600 Btu/h		4,600 Btu/h	
Outside Air Latent Load	-135 Btu/h		-4,600 Btu/h	
Latent Cooling Capacity	4,465 Btu/h		0 Btu/h	
Total Cooling Capacity	44,400 Btu/h		34,380 Btu/h	
Indoor Relative Humidity	50%		56.2%	

Table 1 shows a reduction in the required sensible cooling capacity for the cooling equipment. The system for TDV is assumed to provide no dehumidification, with the use of a higher SAT. The resulting indoor humidity will be higher in the displacement ventilation case.

Additional Design Considerations

The ASHRAE design procedure is applicable to a variety of spaces with varying load conditions and ceiling heights. However, the following observations have been made, based upon PIER research:

Ceiling Height. The PIER research indicated that ceiling heights of 12 feet or higher may increase thermal stratification affects. This implies that for spaces with high ceilings, a lower fraction of heat gains will become a cooling load in the occupied zone. The ASHRAE method uses a single set of load factors for all cases. A modified design procedure may be required for spaces with high ceilings.

Thermal Stratification Predictions. The ASHRAE guideline is a fairly "aggressive" model for predicting thermal stratification. For a space setpoint of 74°F, it predicts a return air temperature of nearly 83°F. The demonstration classrooms showed a smaller level of stratification (a return air temperature of 4-5°F warmer than the temperature at the thermostat).

The ASHRAE airflow calculation is based upon the assumption of a 3.6°F temperature difference between the air temperature near the floor and the air temperature at head level. In practice, the temperature stratification between 4" and 40" in the demonstration classrooms was 2.0°F to 2.5°F. A smaller temperature difference implies that a higher supply airflow is required.

Supply Air Temperature Definition. In practice, an HVAC design will specify the supply air temperature as a design condition. The ASHRAE model provides a method to calculate SAT. For classrooms, a minimum 62°F supply air temperature is recommended to avoid drafts and maintain comfort.

Energy Analysis and Modeling

Hourly building energy simulation tools (DOE-2, EnergyPlus, BLAST, i.e.) are useful in evaluating system design options. A program that is able to properly model unmixed flows that occur with displacement ventilation can predict energy use, and the effect of loads on airflow requirements. It can provide a comparison with conventional mixing systems to predict savings potential with TDV. Demonstration of energy savings will help designers make an appropriate decision during schematic design. Energy simulation tools are also useful in sizing cooling and heating capacity.

The objectives of this analysis are:

- To review commonly used methods for modeling TDV in energy simulation programs
- To recommend modeling procedures with DOE-2 and EnergyPlus
- To examine annual energy predictions of a typical displacement ventilation classroom

Displacement Ventilation Models

Although displacement ventilation is a simple concept, the physical processes that occur are complex and dynamic, and difficult to model. Several models have been developed for displacement ventilation. The first critical step is to determine the required airflow with TDV, at design cooling conditions. This is typically determined based on two design criteria: providing sufficient airflow for cooling the space, and providing sufficient airflow for ventilation (to maintain CO_2 levels below a recommended limit). For classrooms and other regularly occupied spaces, the airflow required for cooling will exceed the airflow required for ventilation. Current modeling methods are summarized in the table below.

Table 2 – Common Displacement Ventilation Models.

Model	Basis	Implemented	Advantages	Disadvantages
UCSD	Salt-tank Physical Scale Model, CFD Analysis	EnergyPlus	Simple, Easy to use model Implemented in EnergyPlus, which includes thermal comfort predictions Employs a “flow decision model” at each time step, which determines whether conditions are valid for DV	Provides only a single factor to divide convective heat gains from lights, equipment and people Physical model does not account for radiative effects on displacement flow
Mundt	Experimental and theoretical studies	EnergyPlus	Implemented in EnergyPlus Based on a simple energy balance of a region near the floor	No guidance on how to define fraction of load that goes to the lower zone Assumes a linear temperature gradient with height
ASHRAE	CFD Analysis and Mockup Tests	ACCURACY (Chen) – uses Z transfer functions	Solid basis of experimental and simulation work Separately accounts for effects of occupant, envelope and lighting heat gains Incorporates ASHRAE 55 comfort requirements on temp stratification Predicts ventilation effectiveness	Load coefficients are average of large number of simulations – varying ceiling height does not affect calculation Limited load applicability range
REHVA		Manual load calculation	Simple Model suitable for manual calculation	Floor temperature via “50% rule” is a only a rough rule-of-thumb
Energy Design Resources	Load Factors	DOE-2 based programs (compatible with EnergyPro or eQuest)	Suitable for DOE-2; proper inclusion of loads in return air path	Must also include a procedure for determining when DV conditions are prevalent

There are several key modeling issues for displacement ventilation.

- Verify that the temperature difference between head and foot level meets the requirements of ASHRAE Standard 55-2004.
- Determine an effective cooling load for the occupied space. The different sources of heat gain, from occupants, building envelope, solar transmission and lighting have different effects on the load for the occupied space. The model should have the flexibility to treat these loads appropriately.
- Account for the effect of ceiling height. For a given load, a classroom with a 12 foot ceiling may require less supply airflow at design cooling conditions than a classroom with a 9 foot ceiling. The model should appropriately estimate the benefits of high ceilings.
- Account for radiative heat transfer from warm ceilings. The ceiling surface temperature will be significantly warmer than the room air near the occupants. As a result, the ceiling will radiate heat towards the floor to warm the floor. This will tend to reduce the thermal stratification in the space.

Modeling TDV with DOE-2

DOE-2 models each control zone as a fully mixed space. It cannot model the temperature stratification that occurs with displacement ventilation systems. As a result, the program cannot size the supply fan or cooling capacity properly, or accurately predict annual energy use. A separate algorithm is required to properly model displacement ventilation systems with DOE-2. One of the following approaches can be taken:

Provide an exceptional calculation method. Title 24 allows for an exceptional calculation method for special systems. A special calculation method could be developed that handles displacement ventilation. This could provide a method of compliance for displacement ventilation systems in the short-term. If the model proves effective, it could be incorporated into the DOE-2 program itself.

With DOE-2.2, designers have the capability to model user inputs (such as loads) as algebraic or logical functions. Addison and Nall (2001) describe a strategy for modeling underfloor air distribution or displacement ventilation with DOE-2.2. The occupied space could be defined as a conditioned zone of variable height, and the upper unoccupied zone modeled as a return air plenum. The loads would be defined first by determining a radiative / convective split by ASHRAE data. The distribution of convective heat gains to the occupied zone could incorporate ASHRAE's recent design guidelines (Chen, Glicksman 2003). Solar and other radiative heat gains would depend on window location and properties, interior surface properties and view factors. A very flexible approach could accommodate adjustments to the model from future research or field tests.

Modify DOE-2 to include a displacement ventilation model. EnergyPlus has implemented a nodal displacement ventilation model. This algorithm can be implemented by other simulation programs. The model divides the space into three "nodes" – a space near the floor, the occupied zone, and an upper unoccupied zone, above head level of the occupants. An energy balance on each node is used to determine the temperature of the node.

Certify other programs as an acceptable ACM. Since EnergyPlus can currently model displacement ventilation systems, this program could be certified as an alternate calculation method for Title 24 compliance. A concurrent project is underway to evaluate EnergyPlus as a code compliance tool.

Recommended DOE-2 Modeling Procedure

Architectural Energy Corporation looked at several displacement ventilation models for use with DOE-2. The model recommended for use in DOE-2 applies load factors to occupant, equipment and lighting loads to estimate the cooling load in the occupied zone. It also defines a return air plenum to provide a "return air path" for loads that do not reach the occupied zone (to properly account for the fact that the return air temperature will be warmer in the displacement ventilation classroom). The model was first introduced in an Energy Design Resources design brief (CTG 2004), and has since been suggested as a displacement ventilation and Underfloor air distribution model for the 2008 Title 24 Standards update. The model uses the following load factors:

Table 3 – Recommended Load Factors

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

The procedure is to define the upper, unconditioned portion of the space as a return air plenum. For instance, for lighting, half of the heat gain is assigned to the occupied space and the other half to the plenum. The total heat gain is unchanged. The effect is that the return air temperature is increased. This allows for a more accurate airflow sizing and load calculation.

The DOE-2 displacement ventilation model was tested under two different scenarios. The first scenario was to compare a constant air volume mixing classroom to a constant air volume DV classroom. The second test was to compare a VAV mixing to a VAV DV system. Each displacement ventilation model used the load factors for DV as described above. The tables below show the model assumptions and results.

Table 4 – DOE-2 Simulation Model Assumptions

Description	Overhead Supply	Displacement Ventilation
HVAC System		
DOE-2 System Type	PVVT (PSZ)	PVVT (PSZ)
Economizer	Yes	Yes
Equipment Capacity	Autosized	Autosized
Cooling Supply Air Temperature	55°F	62 to 65°F
Heating Supply Air Temperature	115°F	90°F
Cooling Efficiency	SEER 10	SEER 10
Fan CFM	Autosized	Autosized
Fan Control	VSD (FIXED)	VSD (FIXED)
Fan Efficiency	0.0004 bhp/cfm	0.0004 bhp/cfm
Heating Efficiency	Furnace 80%	Furnace 80%
OA CFM	15 CFM/Person	15 CFM/Person
Internal Loads		
Plenum Height	3 ft	6 ft
LPD = 1.1 W/sf	100% heat to space	DV: 50% to space; 50% to plenum
EPD = 0.75 W/sf	100% heat to space	DV: 50% to space; 50% to plenum
Number of occupants = 21	100% heat to space	DV: 67% to space; 33% to plenum
Envelope		
Minimally complies with Title 24 - 2005. East and west walls are adiabatic. Only south wall has windows		

The simulation results were verified by reviewing hourly reported temperatures of return air temperature, and comparing the model's prediction of thermal stratification with CFD simulation results and field data from the two demonstration classrooms. The model's reported return air temperatures (max. of 80°F for a space setpoint of 75°F) appear consistent with field data. Also, the airflow predictions appear consistent with simulation results, in estimating the required supply airflow to meet the design cooling loads. Use of the "autosizing" feature of DOE-2 with this model results in a supply airflow prediction that is approximately 15% higher than the mixing ventilation airflow prediction.

Table 5. Simulation Results for Packaged VAV System (Type PVVT)

Climate Zone	System	Supply CFM	Cooling kWh	Fan kWh	Total kWh	Heating Therm	Peak Cooling Load kBtu/h	Peak Heating Load kBtu/h
	Overhead Supply	935	495	120	615	110	20.7	18.1
3	Displacement Ventilation	1076	356	115	471	93	17.1	20.5
	Overhead Supply	971	1259	134	1393	67	27.8	14.2
8	Displacement Ventilation	1112	922	143	1065	65	24.2	16.3
	Overhead Supply	918	725	153	878	140	22.5	23.5
12	Displacement Ventilation	1060	543	155	698	114	20.2	26.2
	Overhead Supply	905	1002	181	1183	125	21.3	27.1
14	Displacement Ventilation	1033	843	163	1006	102	21.5	31.1
	Overhead Supply	891	251	148	399	275	20.8	27.2
16	Displacement Ventilation	1026	189	150	339	202	17.3	35.8

The results for the VAV system comparison show an overall electricity savings in all climates. As expected, the kWh savings is highest in the southern California coastal climates (CZ 8). This model predicted a slight savings in heating energy; however, a heating energy savings is not expected with DV systems.

The results for the constant air volume simulations (Table 3) tell a slightly different story. With these systems, fan energy use is comparable to or greater than cooling energy use in California climates. For this reason, the fan energy penalty associated with DV outweighs the cooling energy savings in many climates. In practice, the design supply airflow will be affected by cooling capacity and airflow capabilities of equipment (cfm/ton). Since the required cooling capacity is slightly lower with a DV system, it is reasonable to assume that the same supply airflow as a mixing system can meet the design load. The DOE-2 models predict supply airflow rates of 900-1000 cfm for mixing systems. In practice, systems are 3 tons to 5 tons in cooling capacity and have design airflow rates of 1,200 cfm to 2,000 cfm.

Table 6. Simulation Results for Packaged Constant Volume System (Type PSZ)

Climate Zone	System	Supply CFM	Cooling kWh	Fan kWh	Total kWh	Heating Therm	Peak Cooling Load kBtu/h	Peak Heating Load kBtu/h
	Overhead Supply	935	325	869	1194	72	20.3	15.7
3	Displacement Ventilation	1076	203	1000	1203	88	16.7	22.3
	Overhead Supply	971	1014	902	1916	51	28.6	12.2
8	Displacement Ventilation	1112	667	1033	1700	64	22.6	17.8
	Overhead Supply	918	592	853	1445	94	18.1	21.1
12	Displacement Ventilation	1060	527	985	1512	104	17.2	26.9
	Overhead Supply	905	860	840	1700	84	19.7	23.9
14	Displacement Ventilation	1033	793	960	1753	99	20.7	32.0
	Overhead Supply	891	214	828	1042	187	16.8	28.8
16	Displacement Ventilation	1026	184	953	1137	185	12.6	36.1

There are other design considerations when simulating a DV system with DOE-2:

1. **Load factors and heating.** The energy benefits of thermal stratification are only present in a DV system when the system is supplying cool or neutral air to the room. Therefore, the simulation should adjust the heat gain assignments to the plenum according to a schedule. When the space is in a heating mode, significant thermal stratification will not be present.
2. **Humidity control.** The simulation assumes a supply air temperature of 62°F to 65°F in cooling. These supply conditions will not allow for significant dehumidification. In climates and building conditions where dehumidification is necessary, energy benefits will be reduced. Some humidity control measures, such as return air bypass, are difficult to model in DOE-2.
3. **Ceiling height.** The recommended load factors are applicable to classrooms, administrative buildings, and other spaces with modest ceiling heights (9 feet to 13 feet). Spaces with very high ceilings may have a lower fraction of the heat gains transferred to the occupied zone, since there is more space for thermal stratification. Additional research is needed to determine appropriate load factors for spaces with high ceilings.

Modeling TDV with EnergyPlus

EnergyPlus has incorporated recent PIER research on displacement ventilation and Underfloor air distribution. The user has a choice between two nodal room air models:

- The Mundt model provides a means for estimating the air temperature near the floor. A constant temperature gradient between the ceiling and floor is assumed. Thus, after the air temperature near the floor is determined, a simple room energy balance is used to determine the return air temperature, and required supply airflow.
- The UCSD model is based on physical scale model testing and research on the fundamental driving forces of displacement ventilation. The room is divided into three nodes: a node near the floor, a node

representing the occupied zone, and a mixed unoccupied zone near the ceiling. The algorithm calculates surface convection coefficients and a single average temperature for each node.

The UCSD model is examined in this analysis. Required inputs are:

- Fraction of heat gains convected to the occupied zone (gp)
- Plumes per occupant – defaults to 1.
- Thermostat height – height of the thermostat. This can be set to 40" (1.0m) for a typical classroom.
- Comfort height – should be set to the same height as the thermostat. This is used in comfort calculations (see output).

For this model, a key user input is the fraction of convective heat gains to the occupied zone that remain in the occupied zone (gp). The heat gains considered are internal heat gains from occupants, equipment and task lighting. Overhead lighting and other heat sources are assumed to be in the unoccupied zone. This fraction is entered in the program as an hourly schedule. This input requires user judgment.

The advantage of this model is its simplicity, and its ability to predict the temperature stratification that occurs with displacement ventilation systems. In reality, the fraction of the heat gain that is carried to the upper unoccupied zone is not a constant, but rather depends upon the supply air temperature, air flowrate, and strength of the heat source, as well as ceiling height.

The following approaches can be used to determine gp:

1. Vary gp to correlate EnergyPlus airflow prediction with published results (Halton CFD or mockup tests, ASHRAE)
2. Determine a gp value that is consistent with other modeling approaches (ASHRAE)

A value of 0.2 for gp is consistent with the ASHRAE design guidelines (Chen & Glicksman 2003), which uses a fraction of 0.295 for the portion of heat gains from occupants and equipment that go to the occupied zone, 0.132 for the contribution of lighting heat gains, and 0.185 for the contribution from heat gains through the building envelope. ASHRAE's recommendations are based on a correlation against a large number of validated CFD cases, with varying loads and ceiling heights. Empirical coefficients represent the fraction of loads from occupants and equipment (a_{oe}), lighting (a_l), and building envelope (a_{ex}) that are convected to the occupied space. In the EnergyPlus implementation, lighting and building envelope loads are handled by determining an average convective coefficient and surface temperature for each surface in each zone. The effects of these loads on the cooling load of the occupied zone depend on the radiative heat exchange between surfaces.

The CFD analysis performed under Task 2.2 of this project shows that a room with a higher ceiling height requires less airflow to maintain the room to the cooling setpoint. Therefore, a lower fraction is justified for classrooms and spaces with high ceiling heights (12 ft or higher).

Model Outputs

The EnergyPlus displacement ventilation model is a big improvement over models that do not account for thermal stratification. It also provides additional outputs that are useful to the designer. The model outputs the maximum temperature gradient between head and foot level. This can be compared to the ASHRAE Standard 55 recommendation to ensure that the system maintain comfort. The model also can provide comfort assessments by estimating the Predicted Mean Vote (PMV).

Issues and Limitations

While the EnergyPlus displacement ventilation model provides a good model of the air distribution, there are limitations of EnergyPlus which limit its near-term application:

- Lack of easy to use interfaces ("front end") for building definition and output formatting
- HVAC capabilities: EnergyPlus cannot model static pressure reset, and cannot model dehumidification strategies, such as return air bypass, that are used with some systems

A concurrent research project is evaluating the use of EnergyPlus as an alternate calculation method (ACM) for compliance with the California Title 24 Energy Efficiency Standards. That project will provide much more details on the gaps between required ACM modeling features and EnergyPlus capabilities.

In the near term, a good use of EnergyPlus could be for use in incentive programs such as Savings By Design and with sustainable-based criteria such as CHPS and LEED.

Energy Analysis

The EnergyPlus displacement ventilation model developed by Linden (UCSD model) was applied to a typical classroom. A single classroom was modeled with displacement ventilation, and variable air volume control. The objectives of this modeling effort were to:

1) Predict TDV airflow and cooling requirement for a typical classroom
2) Estimate annual energy use and compare to a mixing system
3) Determine effect of user-defined parameters on energy calculations
4) Compare predictions of design airflow with other procedures (ASHRAE)

A single classroom was modeled in EnergyPlus. The classroom is situated in Sacramento, CA and has a building envelope meeting Title 24 standards. With an occupancy level of 30 students and the teacher, the design cooling load is 16.4 kBtu/h. The primary user input in the UCSD displacement ventilation model is $\mathbf{g_p}$, the fraction of convective heat gains that contribute to the cooling load in the occupied space. The user is allowed to enter a fraction as a schedule. This fraction was varied from 0.1 to 0.5, to test the sensitivity of the results to the user input. For energy use, a school calendar of August 15 to June 15 was assumed, and an HVAC operational schedule of 8:00 AM to 4:00 PM. The HVAC system used was a packaged variable air volume system with a fixed supply air temperature setpoint of 65°F in cooling. A minimum outside air ventilation of 15 cfm/person was used in the model.

Table 7 - EnergyPlus Simulation Results

gp	cfm	RAT	Cool kWh	Fan kWh	Total kWh
0.1	1010	79.8	602	591	1193
0.2	1030	79.5	616	608	1223
0.3	1050	79.2	628	625	1253
0.4	1070	79	641	643	1284
0.5	1091	78.7	655	661	1316
(Mixing Case)	670	75.2	1051	378	1429

The predicted airflow increases slightly as the fraction is increased. The sensitivity of the results to g_p is low: for an increase of 0.1 in the g_p fraction, there is only a 2% increase in required airflow. The cooling energy and fan energy also increase accordingly. The ASHRAE design guidelines (Chen 2003) specify fractions of the heat gains from occupants and equipment, lighting and envelope that contribute to the load in the occupied zone. ASHRAE developed empirical coefficients from a correlation to a large set of CFD simulation results. The coefficients are 0.295, 0.132 and 0.185 for occupants and equipment, lighting and envelope, respectively. Thus, entering a fraction of around 0.2 would be consistent with the ASHRAE guidelines.

Comparison with Mixing Case

Compared to the TDV system, the overhead mixing system will have a lower supply airflow and lower fan energy, if a VAV system is also employed. However, the mixing case has a higher cooling capacity requirement and a higher annual cooling energy. The net effect is a slightly higher electricity energy use for the mixing case. An EnergyPlus run was performed for the mixing case, also using a packaged VAV system. The supply air temperature used was 13°C (55.4°F) and the same minimum flow fraction, outside air ventilation, and room setpoints were used. For DV, the cooling energy savings more than offset increases in fan energy, so that the total annual electricity use of the compressor and fans is 7-15% lower than the mixing case. The analysis assumed a seasonal 10-month school year; a year-round operation would allow for more cooling energy savings with DV.

Applicable Codes and Standards

There are several ASHRAE Standards related to the design or evaluation of TDV systems. This section provides an overview of the design issues.

ASHRAE 55-2004 Thermal Environmental Conditions for Human Occupancy

This standard provides for comfort criteria for stratified space conditions that are common with TDV systems. Specifically, the following criteria are needed for thermal comfort:

- Temperature difference between head and foot level
- Surface temperature asymmetry
- Air velocity and draft temperature

The requirements are set for different comfort “classes”. This standard is recommended for design, but is not required for code compliance.

The temperature difference between head and foot level is a key design parameter for displacement ventilation systems. The thermal stratification in a space served by displacement ventilation is affected by the supply airflow. If the supply airflow is too low, a cooler SAT is needed to cool the space. The resulting temperature gradient would exceed the recommended maximum. If the supply airflow is too high, the thermal stratification in the space will be reduced. This will lower the return air temperature and reduce the energy benefits of TDV.

The Standard also provides a recommendation for the maximum allowable temperature difference between interior surfaces. A cold window surface during cold outdoor conditions could result in poor comfort, even if the space temperature is maintained at comfortable conditions. This is especially important when low-velocity displacement diffusers are used for heating. The use of high performance windows will reduce the heat loss rate and extend the range of outdoor conditions for which perimeter heating is not required.

The temperature differential between the ceiling surface and floor surface also must be considered. The warm ceiling in a room served by displacement ventilation could create a problem in some cases. A maximum temperature differential of 9°F between the warm ceiling surface and the room air is recommended to maintain comfort.

Draft, the unwanted local cooling provided by air movement, is a potential issue with displacement ventilation. The Standard provides an allowable mean air speed, as a function of local air temperature and turbulence intensity. Typically, a maximum air velocity of 40 fpm is recommended for TDV systems. Diffuser manufacturers publish data on the adjacent zone for a given supply temperature and airflow. The adjacent zone is the area near the diffuser where the local air velocity exceeds a specified limit (usually 40 fpm). Outside of this area, the air velocity is low and will normally not create drafts.

ASHRAE 62.1-2004 Ventilation for Acceptable Indoor Air Quality

The ASHRAE ventilation standard provides minimum outside air requirements for HVAC systems based on occupancy type. The ventilation requirement specifies a minimum cfm per person and minimum airflow per ft^2 of occupiable floor space. This corresponds to a combined outdoor rate of 15 cfm/person for ages 5-8 and 13 cfm/person for classrooms with ages 9 or older.

The Standard recognizes the improved ventilation effectiveness of TDV systems. It assumes an air distribution effectiveness (air change effectiveness) of 1.2 for TDV systems. Effectively, this means that 20% less outside air is required to meet the requirements of the Standard. Potentially, this could be used to justify a code change to lower outside air rates with displacement ventilation. However, another way to express it is that a mixing ventilation system would have to provide 20% more outside air (in cooling mode) to provide the same IAQ as a TDV system that meets the minimum requirements of the Standard.

ASHRAE 113 Method of Testing Room Air Diffusion

This Standard describes test methods for evaluating room air diffusion. Measurements of temperature and air velocity are taken at different locations. The variation in consecutive measurements over a three minute period is used to determine turbulence intensity. The Standard describes equipment accuracy requirements. Sensors used to measure air velocity should be able to read down to 20 fpm (0.1 m/s) with an accuracy of +/- 10 fpm or

5% of the reading. This is especially important when evaluating TDV systems, since air velocities in the space are very low.

The Appendix to the Standard describes a procedure that can be used to estimate the percentage of occupants that would be dissatisfied due to draft. This is also covered in ASHRAE Standard 55-2004.

ASHRAE 129 Standard Method of Measuring Air Change Effectiveness

ASHRAE Description: "This standard defines a method of measuring air-change effectiveness in mechanically ventilated buildings or spaces. The method involves an age-of-air approach to air-change effectiveness and employs tracer gas procedures to measure the age of air."

The mean age of air is also used with CFD simulation programs. From the Halton CFD analysis, a lower mean age of air is expected with TDV systems.

Title 24 Standards

The 2005 Title 24 Standards do not provide a distinct procedure for predicting the energy use of DV systems. The primary issues stem from the fact that with displacement ventilation, conventional procedures cannot be used to estimate supply airflow requirements for DV systems. This section summarizes the issues with modeling displacement ventilation systems.

Load Calculations

Section 144 contains prescriptive requirements for space conditioning systems. Load calculations are to be calculated by procedures outlined in the ASHRAE Fundamentals Volume. The load calculation procedure must provide a model of the thermal stratification to make an accurate estimate of the required supply airflow and cooling capacity. Assuming a return air temperature equal to the room setpoint for TDV systems would overestimate the fan size.

Zone Thermostat Controls

The standard (as well as ASHRAE 90.1) requires a deadband temperature range of 5°F where heating and cooling energy is shut off or reduced to a minimum. With TDV systems, care should be made to ensure that the supply temperature doesn't get too cold or warm. In some cases, a small amount of heating may be required if the mixed air temperature (return air with minimum outside air fraction) is cooler than 60-65°F.

Reheating with Zone Controls

The Title 24 Standard does not allow reheating, unless the room is served by a variable air volume system. This precludes the use of reheat with small constant air-volume packaged units to raise the supply air temperature to 65°F.

Outdoor Ventilation Loads

Outdoor air requirements are defined in Section 121 of the 2005 Title 24 Standard. The code-required minimum for classrooms is the greater of 0.15 cfm/ft^2 of floor space, or 15 cfm/occupant. A possible change is to include a minimum that varies with the ventilation effectiveness.

ASHRAE 90.1-2004

The ASHRAE 90.1 is the national energy efficiency standard for buildings. Like Title 24, the Standard defines mandatory, prescriptive and performance-based requirements for demonstrating compliance.

The fundamental issue is that programs such as DOE-2 that are used for compliance don't model air distribution systems that provide unmixed airspaces. This will affect system sizing and hourly cooling load calculations. Specific code-related sections to address:

1. Supply airflow requirements: section 11.3.2 (g) defines a supply air-room air temperature difference of 20°F for airflow calculations. For DV, the typical temperature difference is 15°F (+/- 2-3°F). This is needed to accurately estimate supply airflow requirements.

2. Equipment sizing (section 11.3.2 (g)): due to stratification and the use of warmer supply air temperatures, the required sensible cooling capacities are typically ~10% lower with DV systems. (Moreover, if the space has dehumidification requirements that are low enough to be met by outside ventilation air, the required latent system capacity is zero.)
3. Fan system efficiency (section 11.3.2 (h)): a DV system with a central air handler would likely have higher fan efficiency due to the reduced pressure drop (lower BHP/cfm). I don't have data to quantify this, however.
4. Outside air (section 11.3.2 (d)): ASHRAE 90.1 states that the budget and proposed designs use the same outside air ventilation rates. One could argue that the minimum be reduced with DV systems, to be consistent with ASHRAE 62.1-2004, which states a ventilation efficiency of 1.2 for DV systems. (However, the minimum OA in heating should not be reduced, since ventilation effectiveness benefits are only apparent in cooling/ventilation.)

Several informative sections in Appendix G would also need updating:

G.3.1.2.2 – Equipment Capacities

G.3.1.2.2.1 – Sizing Runs

G.3.1.2.8 – Supply air to room air temperature difference for design airflow calcs

G.3.1.3.1.2 – SAT reset (systems 5-8)

G.3.1.3.8 Chilled water setpoint temperature

G.3.1.3.9 Chilled water reset

There are also modeling-related issues that are not purely a code issue but rather an inability to model certain system types or configurations.

Equipment Specification

HVAC Selection

HVAC equipment used for DV should be able to deliver the 65°F supply air temperature and provide for a tight control of supply temperature (within 3-4°F) under varying load conditions. The best cooling source for a displacement ventilation system is a chilled water coil. The control valve in a hydronic system allows supply of constant 63ºF to 65ºF air. A typical direct expansion (DX) system is designed to provide colder 50ºF to 55ºF air while the compressor is running and cycles on and off to meet space loads. This lower temperature and larger temperature fluctuations would create a comfort problem in displacement ventilation when the supply air comes in contact with occupants. However, larger DX systems with several compressors and temperature-reset capabilities can be used as an alternative to a chilled water system. For example, a packaged rooftop variable air volume (VAV) system serving several classrooms should be able to provide the necessary supply air temperature control.

The Outline Specification and Schematic Design Report developed for this project describes general criteria used to compare HVAC system types. Systems were compared on the basis of room temperature control (comfort), energy efficiency, first cost and ease of maintenance. This report describes displacement ventilation requirements for each equipment type.

Packaged DX Equipment Requirements

The required SAT and level of SAT control is not possible with currently available packaged DX equipment. A general recommendation is to control the supply air temperature to within 3°F of the SAT setpoint. This would require multiple cooling stages and additional controls and sensors for control of the supply air temperature.

An air-side economizer can use either a differential dry-bulb or differential enthalpy controller. The economizer should have a low-ambient lockout, to prevent the discharge of cold supply air when the outside air temperature drops below a specified limit (i.e., 60°F).

A morning warm-up sequence is an important control strategy for displacement ventilation systems. DV systems are not as effective in heating. If warm air is supplied near the floor of the space at a low velocity, some of the supply air may rise towards the ceiling exhaust before it can effectively heat the space. A morning warm-up strategy heats the room to the occupied setpoint temperature by using recirculated air. An hour prior to occupancy, outside air is introduced to the space to meet code ventilation requirements. Once the classroom is occupied, heat gains from lighting, equipment and occupants normally outweigh envelope losses to the outdoors. Thus, a morning warm-up will minimize heating requirements during occupied hours.

Humidity Control

For overhead air distribution systems, indoor space humidity is controlled indirectly by cooling the supply air to near 55°F. The dehumidified supply air offsets latent heat generation from occupants. Displacement ventilation typically requires a supply air of 65°F or higher for school applications. With the higher supply air temperature, less moisture is removed from the outside air. For most California climates, this does not pose a problem. The outdoor humidity ratio, the pound of moisture per pound of dry air, is lower than indoor design conditions for almost all hours of the year. For an indoor space condition of 75°F DB, 60% RH, indoor humidity can be maintained with the use of 65°F air.

In warm, coastal California climates (Climate Zones 6, 7 and 8), the humidity of the outdoor air is above design conditions for a significant portion of the year. Although cooling the supply air to 65°F may provide for some dehumidification of the outside air, the resulting supply air will not be dry enough to remove latent heat gains. As a result, indoor humidity will exceed 60%. In warm valley climates (Climate Zones 9 and 10), there is only a slight requirement for dehumidification of outside air. For spaces with high latent loads, humidity control options can be considered. For cool, temperate climates (Climate Zone 3) and desert climates (Climate Zones 12-15), indoor humidity is normally not an issue, even with the use of warmer supply air.

There are several means to control humidity:

Return air bypass. *When the outside air requires dehumidification, a bypass damper can direct up to 100% of the return air to bypass the cooling coil. Since less air passes over the cooling coil, this lowers the air temperature leaving the coil. The dehumidified air off the coil (near 55°F) is mixed with the bypassed warmer return air to achieve the 65°F supply air temperature. Dampers control the fraction of return air that bypasses the coil. As the humidity in the space decreases, the amount of return air directed to the cooling coil can increase, to increase the leaving air temperature and capture energy savings.*

Mixed air bypass. *Face-and-bypass dampers allow the entering air (a mix of outside air and return air) to bypass the cooling coil. While this is a low-cost option, it does not dehumidify as effectively during humid outdoor conditions, since some of the outside air also bypasses the coil.*

Condenser heat recovery. *A "run-around" coil can capture heat rejected from the condenser and warm the supply air downstream of the cooling coil.*

Heat recovery or total energy recovery. *Outside air is preconditioned by the exhaust air stream, removing both heat and moisture with total energy recovery. With this option, 100% outside air can be used.*

For coastal climates, the specification of a differential enthalpy economizer will help to maintain space humidity levels during mild, humid outdoor conditions. Maintaining enthalpy-based sensors can be a challenge, however. Humidity control can be a challenge, since in California, high humidity levels often occur at part-load conditions, when the air temperature is 75°F to 85°F outside. At part-load conditions, as the required air volume decreases, the fraction of outside air increases. The system must be able to provide sufficient dehumidification without overcooling the space.

Diffuser Selection Criteria

Diffuser Location Options

Common locations for the displacement diffusers in classrooms are at the corners of an interior wall or at the "quarter-points" of an interior wall. There may also be an opportunity to integrate diffusers with casework.

Performance Criteria

Displacement diffusers must supply low-velocity air in cooling, to minimize entrainment of room air that will cause mixing conditions. Typical diffusion performance indices such as ADPI and throw do not apply to displacement diffusers. The primary diffuser criteria are:

1. Adjacent zone – this is defined as the region near the diffuser where the air velocity is above a specified threshold, typically 40 fpm. In this area the potential for draft exists when the local air temperature is low. This region depends upon the airflow and the temperature difference between the supply air and room air. Typically the minimum supply air temperature (at the diffuser) for classrooms is 65°F. Lower supply temperatures should not be used unless a careful analysis has verified thermal comfort.
2. Noise Criteria – increasing the outlet area of the diffuser will lower the noise levels. Manufacturers publish NC ratings for diffusers. Individual diffusers should be selected for an NC rating of 25-30 at design airflow. Using two diffusers with an NC rating of 25 or lower will help to meet background noise level requirements of the voluntary 2002 ANSI S12.60 Standard, 35 dBA for core-learning spaces.
3. The size and shape of the diffuser may be limited by architectural constraints.
4. A diffuser with a variable aperture or means of increasing the discharge velocity in heating is desired, to promote mixing in the space.
5. CFD analysis may be useful for selecting and locating diffusers for large spaces. For classrooms, two wall-mounted diffusers installed in the corners of the room or quarter-points of the interior wall will provide good comfort for most applications.

Perimeter Heating

For most California climates, supplemental perimeter heating is not required. The low-velocity "displacement" diffusers can be used for morning warm-up heating. They can also be used to supply warm air during occupied hours. The supply air flow should be designed to provide a higher air velocity in heating. For the typical diffuser, this requires the air flow to be set to the maximum (1000-1200 cfm per classroom). The supply air temperature should be kept to 80-85°F in heating because of the proximity of the diffusers to the students.

For climates that have a winter design dry-bulb of 10°F or less, supplemental perimeter heating is required, to offset radiant effects from cold window surfaces. Typically only the mountain climates of California would experience this condition. For other climates, the use of high performance windows and building envelope is important in moderating interior surface temperatures. This will eliminate the possibility of a cold downdraft from windows in winter that would oppose the displacement airflow.

Control Options

With DV, the space temperature can be controlled by varying the supply air temperature, the supply air volume, or both. Variable air volume (VAV) systems will allow for fan energy savings at part-load conditions. Control strategies that allow for variation of both the supply air volume and supply air temperature have the greatest potential for energy savings. A 65°F SAT is often only needed at design cooling conditions. SAT reset will maximize the potential for free cooling.

Because of the temperature stratification, the thermostat location is an important design consideration. It should be located at a height approximately equal to head level of seated occupants (40 in.). The thermostat should be located outside of the adjacent zone of the diffuser, to avoid cool drafts at foot level. A location at least 6 ft from the nearest diffuser will work well for space temperature control.

Economizer Operation: With the higher SAT, DV extends the period of free cooling. However, it does require a low ambient temperature lockout on the economizer, which closes the outside air damper to the minimum position when the outside air falls below a certain temperature (i.e., 60°F). When the outside air temperature is below this point, a mix of outside air and return air must be used, to avoid discharging cold air into the space. A differential dry-bulb lockout with return air temperature will provide for efficient operation in dry, inland climates. A differential-enthalpy economizer is a good option in more humid, coastal climates. However, maintenance with enthalpy sensors can be a challenge.

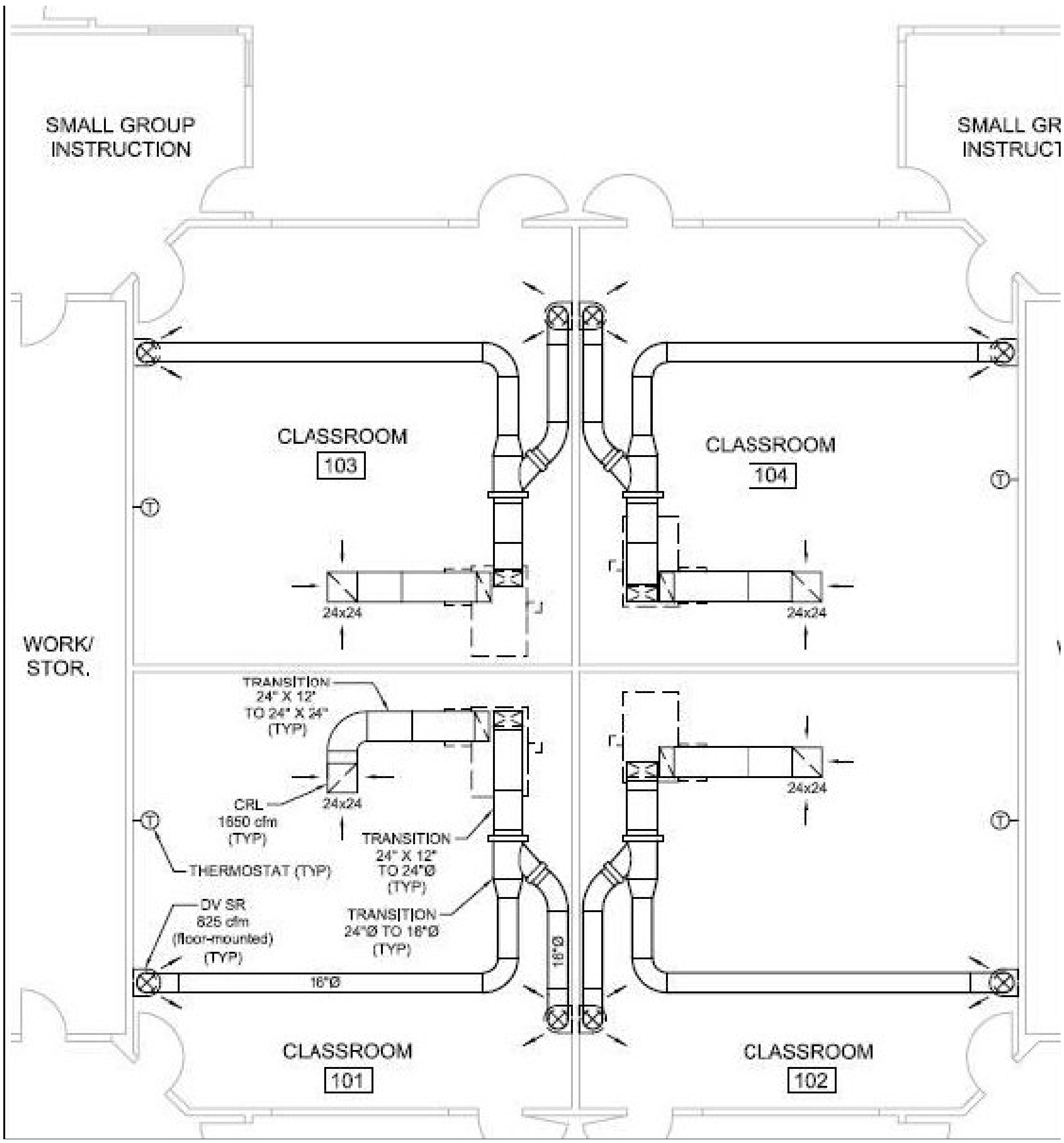
In some mild coastal climates, the outside air temperature will rarely exceed the air temperature exhausted from the space of a DV classroom (typically, 78°F to 82°F). It may be possible to use 100% outside air in these climates, with little energy penalty. However, 100% outside air cannot be used during the winter months when the outside air temperature frequently falls below 60°F.

Construction and Commissioning Phase

This section summarizes construction and commissioning details that are unique to displacement ventilation. Also refer to the project Commissioning reports for additional details.

Ductwork Layout

A schematic design for a typical classroom wing was developed from a copy of a school plan. This schematic design assumes rooftop packaged units serving each classroom. In the most common installation, the units will be located directly above classrooms. Supply ductwork is routed down to a supply plenum, with branch takeoffs towards the two sidewall diffusers. A single return grille is used for each classroom.



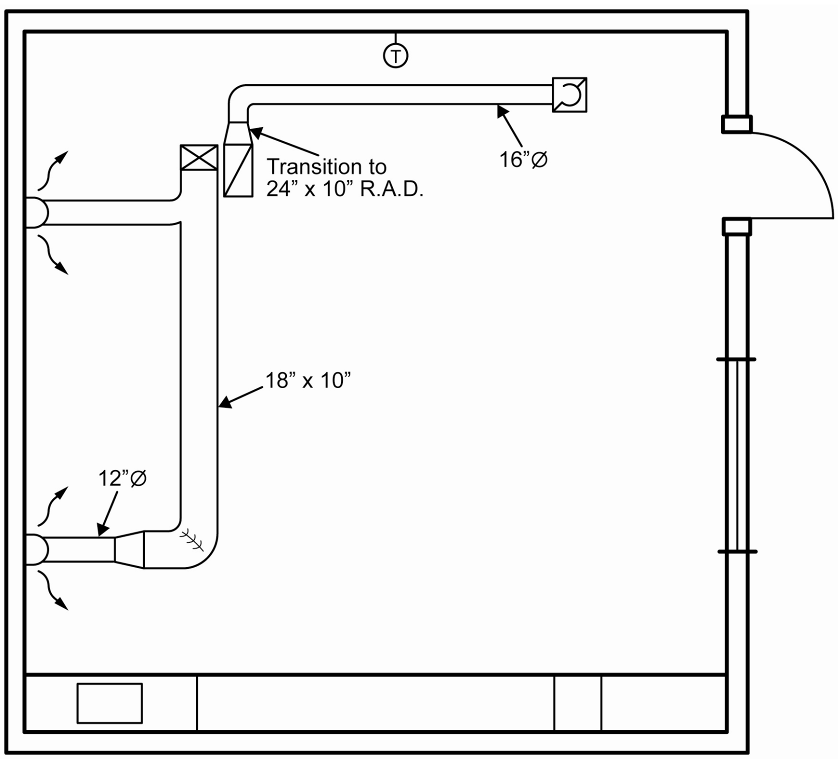
DV Classroom Building Plan (Partial). (not to scale)

Ductwork should be sized to limit velocities to meet acoustics requirements, and be sized to meet system static pressure requirements. Sizing and layout of ductwork follows SMACNA guidelines, as with overhead mixing distribution systems.

Diffuser Layout

The typical layout for a classroom uses two sidewall diffusers per classroom. The supply diffusers can be located on an interior wall of the classroom. The diffusers can be located in the corners or at quarter-points along the interior wall. Two diffusers provide a good room air distribution for rooms with depths of 30-35 feet or less. For larger spaces (such as a library or lecture hall), additional diffusers may be required.

The return air grille can be located at any point at the ceiling (it is not critical that it be located on the opposite side of the room as the displacement diffusers).



DV Classroom Ductwork Details. (not to scale)

A duct drops to each duct connection at the top of the diffuser (typically a 10” round or 12” round connection). The final duct drop can be circular duct or flex duct. The vertical duct run above the diffuser should be at least three times the duct diameter, for proper sound attenuation near the diffuser.

Diffuser installation is quite simple. However, the job usually includes the attachment of a duct cover to conceal ductwork below the ceiling. The job will also require two ceiling tiles to be cut out to provide room for the duct drops.

Thermostat Location

The thermostat should be located far enough away from the displacement diffusers to avoid any “local” air effects of the adjacent zone near the diffuser. It should be located at least six feet away from the nearest supply diffuser. The height is also an important design consideration, since the temperature of the space will vary from floor to ceiling. For best control, it should be located at a height approximately equal to the head level of the seated student (i.e., 40”). If it is located much higher (52”-66”), the thermostat setting should be adjusted upwards by 1°F to 2°F, since the air will be slightly warmer at this height.

Commissioning Details

Initial Setup and Checkout – Sample Checklist

The following list shows checks for air handling equipment and hydronic lines for central cooling equipment. This list does not take the place of checks recommended by the manufacturer or installing contractor.

Check	**Y/N**	**Initials**
Casing condition good: no dents, leaks, door gaskets tight		
General condition appears good		
Pipe connections are complete according to the specification		
Piping is properly supported		
Refrigerant and water connection points are clearly marked or readily identified		
Condensate drain in place and properly tapped		
Correct refrigerant charge		
Compressors and piping were leak tested		
Disconnects in place and clearly labeled		
All electric connections tight		
All dampers stroke fully and easily		
All dampers close tightly		
Alignment of motor driven components correct		
Supply fan belt: tension and condition ok		
Supply fan acceptable noise and vibration		
Exhaust fan belt condition ok		
Filters clean and tight fitting		
Verify that ductwork is insulated and verify that any flex duct is not compressed or excessively bent so as to obstruct flow		
(System-specific checks)		
Return air temperature and SAT sensors properly secured		
Return water temperature sensor properly installed		
3-way control valve properly installed		

Refrigerant charge-robber is installed properly		
Hydronic components securely mounted to skid		
TAB: verify air handler supply air flow rate		
TAB: verify chilled water flow rate		
TAB: verify even air flow to Halton diffusers		
Other:		
Calibrate or verify calibration of sensors and thermostat		

Functional Testing

In addition to regular start-up and checkout procedures, the following checks should be performed with a displacement ventilation system during the commissioning process:

1. Measurement of supply air temperature. Verify that the SAT provided during cooling is maintained at or within 2-3°F of the supply air temperature setpoint. Report supply air temperatures if they drop below 60-61°F and remain low for a period of more than two minutes.
2. Airflow Measurements: estimates of supply airflow cannot be made with a traditional flow hood (balometer). The diffuser opening and airflow velocity does not permit accurate measurements. Approximate measurements of airflow in the occupied zone can be made with an omni-directional hot-wire anemometer. Air velocity should be maintained to 40 fpm or less in the occupied zone. Instead, airflow rate for packaged DX rooftop units can be measured directly at the air handler, with a device such as the Energy Conservatory Trueflow meter.
3. Verify compressor lockout: verify that the compressor does not come on for cooling when the outside air temperature is cooler than the supply air temperature setpoint (i.e., 65°F).
4. Verify economizer low-limit operation: verify that during cool outdoor conditions, the outside air dampers modulate closed whenever the outside air temperature drops below the SAT setpoint.
5. For VAV systems, verify that outside air damper minimum position adjusts to provide the minimum outside airflow required to meet Title 24 ventilation requirements (i.e, 15 cfm/person).
6. For central plant systems, verify that the chilled water temperature is being maintained to the chilled water setpoint (for DV systems, a chilled water temperature of 52°F to 54°F is typical).
7. If specified, verify morning warm-up sequence: when the space temperature is below the heating setpoint prior to occupancy, the heat should turn on and outside air dampers closed to heat the space. Verify that outside air dampers open 1 hour prior to actual occupancy to meet code requirements.
8. Verify supply air temperature in heating mode meets specifications (typically, a supply air temperature of 80-90°F is recommended when heating with displacement ventilation).

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