Prediction of Air Mixing from High Sidewall Diffusers in Cooling Mode

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ABSTRACT

Computational Fluid Dynamics (CFD) modeling was used to evaluate the performance of high sidewall air supply in cooling mode. The research focused on the design, placement, and operation of air supply diffusers located high on a sidewall and return grilles located near the floor on the same sidewall. Parameters of the study are the supply velocity, supply temperature, diffuser dimensions and room dimensions. Thermal loads characteristic of high performance homes were applied at the walls and room temperature was controlled via a thermostat. The results are intended to provide information to guide the selection of high sidewall supply diffusers to provide proper room mixing for cooling of high performance homes.

INTRODUCTION

As Building America (BA) homes begin to reach the 40 to 50% performance levels, the downsizing of space conditioning systems will begin to reach the point where the air flow volumes required to meet the small remaining heating and cooling loads will not be large enough to maintain uniform mixing of room air. This project focused on resolving this technical barrier so that uniform temperatures can be maintained within future homes even though the cooling and heating capacities required to heat and cool the homes are significantly smaller than current homes. In addition, well insulated homes with advanced windows do not require perimeter heating, so centralized, compact duct systems in interior walls can be used. Compact systems are 20-25% less expensive than traditional designs, and offer higher system efficiency through reduced duct leakage and duct length, and by having the ducts in conditioned spaces (Griffiths and Zoeller, 2001). High sidewall diffusers are one of the simplest and most promising approaches of delivering conditioned air in very high performance homes since they are not blocked by furniture and, with proper design, can provide good mixing of the conditioned air in the space.

This report describes simulated performance of high sidewall diffusers for typical residential applications. The task addressed the evaluation of these systems for cases in which the room is recovering from a temperature setback state. This was considered to be a worst case operating condition for the high sidewall diffuser system, even though it may not represent the most common operating mode. However, if the system performs well in terms of providing good air mixing during transient setback recovery, it will likely perform well during steady-state, part load and design load operation.

The results from this project can be used to address comfort deficiencies that BA team members have found with high sidewall diffuser systems. The task produced guidelines for high sidewall diffuser systems that BA team members will be able to use in their designs for high performance houses.

A design guide to evaluate the performance of high sidewall air supply for residential applications was developed by Temple (2003). Following this guide, representative room and diffuser geometries, air supply flow rates, and temperatures were identified and modeled using Fluent CFD software. The model addressed thermal loads that are representative of high performance homes. Load density of 10 Btu/h.ft^2 (31.52 W/m^2) was applied at the ceiling and at the exterior wall opposite to the diffuser jet. A 4 in. (0.1 m) by 4 in. (0.1 m) thermostat with a dead-band between 70ºF (294.3 K) and 72ºF (295.4 K) was

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built into the model to control the room temperature. The thermostat was mounted on the wall parallel to the diffuser jet at 4 ft. (1.22 m) from the floor (see Figure 1).

NUMERICAL MODEL

The analysis considered two different rooms: Room A is 12 ft (3.66 m) long × 11 ft (3.35 m) wide × 9 ft (2.74 m) high, supplied by an 8 in. (0.2 m) × 4 in. (0.1 m) diffuser and Room B is 12 ft (3.66 m) long × 9 ft (2.74 m) wide × 9 ft (2.74 m) high, supplied by a 10 in. (0.25 m) × 6 in. (0.15 m) diffuser. The diffusers were based on available products and used the geometry and open area of the J & J 900V models. The simulations assumed straight vanes and completely open dampers with air being discharged in a uniform profile normal to the diffuser face. The diffusers have a 63% open area. In the simulations, the dimensions of the diffuser were reduced in both height and width, keeping the aspect ratio the same as the actual diffuser overall opening, in order to have the same average air speed from the simulated diffuser opening as from the actual diffuser. As other researchers have shown (Chen and Jiang, 1992), it was more important to keep jet speed the same between the actual and modeled diffuser openings than to keep the overall opening sizes the same. All diffusers were mounted with the upper edge of the opening 9 in. (0.23 m) from the ceiling.

The return grille was located directly below the supply diffuser, with the lower edge of the opening 4 in. from the floor. The grille area was much larger than any of the diffusers, in order to have a low air speed approaching the opening. All simulations shared uniform room temperature of 80ºF (299.8 K) as an initial condition. The thermostat deadband was set as 70ºF (294.3 K) and 72ºF (295.4 K). Air flow rates varied from 28 cfm (0.013 m^3/s) to 206 cfm (0.097 m^3/s), depending on diffuser open area and the desired supply air speed. Supply air speed, V_S, varied from 197 fpm (1 m/s) to 788 (4 m/s) fpm and supply air temperature, T_S, varied from 55ºF (285.9 K) to 65ºF (291.5 K).

Wall characteristics, gridding, and other solution parameters were chosen based on prior experience with models of this type. Room walls were considered massless and thermal loads were applied at the inner surfaces of the walls. Because the diffusers are placed in the middle of a wall, a plane of transverse symmetry exists. This allowed the simulations to model only one-half of the room, thus halving computational expense. Figure 1 shows the geometry outline for the room with thermal loads applied at the ceiling and at the right wall. The floor and the remaining walls of the room were defined to be adiabatic. The domain is shown with the plane of symmetry bisecting the room, through the diffuser and return grille.

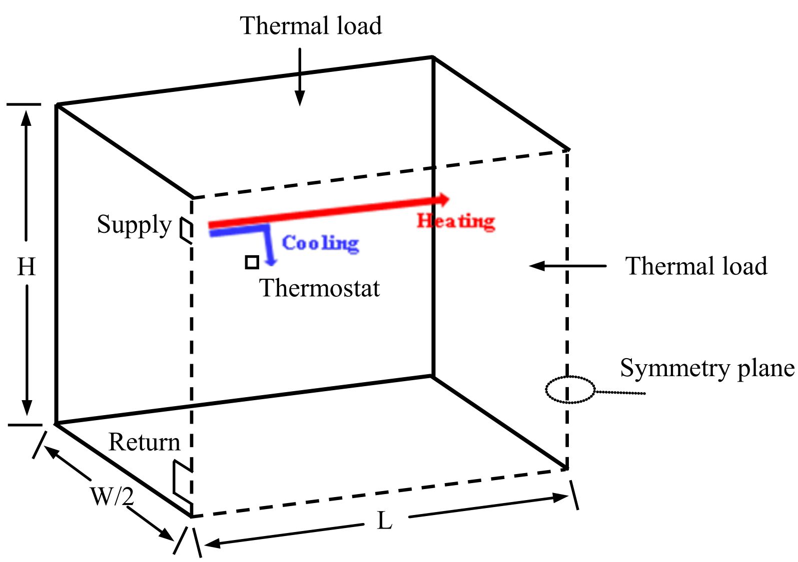


Figure 1 *Sketch of the computational domain (not to scale) showing the locations of supply, return, and thermostat in the room. Thermal loads are applied only to the ceiling and to the right wall to approximate load distribution in a perimeter zone.*

COMFORT CRITERIA

The air diffusion performance index (ADPI) was calculated at the end of the simulation when one air change was supplied to the room. ADPI is defined in a zone between 0.1 m and 1.7 m from the floor (ASHRAE 1992) and is a simple comfort criterion based on local temperature, average temperature in the whole room, and local air speed. ADPI uses effective draft temperature,

$$\theta = (T - T_{avg}) - 0.07(V_x - 30) \quad (1)$$

where T is the local air temperature [ºF], T_{avg} is the average temperature in the occupied zone [ºF], and V_x the local air speed [fpm]. A high percentage of occupants are comfortable in indoor environments when the effective draft temperature is between -3ºF (-1.5ºC) and 2ºF (1ºC) and local air speed below 70 fpm (0.35 m/s) for maximum comfort (ASHRAE 2009).

In the zone of interest, effective draft temperature and air speed were determined for each cell. The number of cells meeting these requirements was compared to the total number of cells to determine the ADPI for the simulation. Acceptable comfort level exists in the room when ADPI is in the range between 70 and 90%. Good comfort level is guaranteed when the ADPI is 90% or higher.

RESULTS AND DISCUSSION

The results will be presented first for Room A and then for Room B by the temperature distribution and comfort level for different operating conditions. The time history of the thermostat temperature in Room A supplied by T_S of 55ºF (285.9 K) is presented in Figure 2 for two different supply velocities. The corresponding velocity variations at the diffuser are plotted in Figure 3. The time at which one air change was supplied to the room is marked on Figure 2. The temperature at the thermostat decreased quickly with the supply velocity of 591 fpm (3 m/s) and entered a quasi-steady state mode characterized by oscillation cycles around the thermostat deadband. A decrease in supply velocity to 394 fpm (2 m/s) delayed the room cooling and resulted in longer cooling/heating cycles.

We are interested in the flow patterns at the time when one air change was supplied to the room. This is presented in Figure 4 in terms of the distribution of temperature corresponding to V_S = 788 fpm (4 m/s) and T_S = 55 ºF (285.9 K). The contour plots are on the vertical mid-plane of symmetry at 25%, 50%, 75%, and 100% of the time for one air change. The isotherms showed a good mixing between the supply air and the room air. The jet was attached to the ceiling and mixed with the room air after hitting the opposite wall. Low supply velocities resulted in sinking jets, which may cause occupant discomfort.

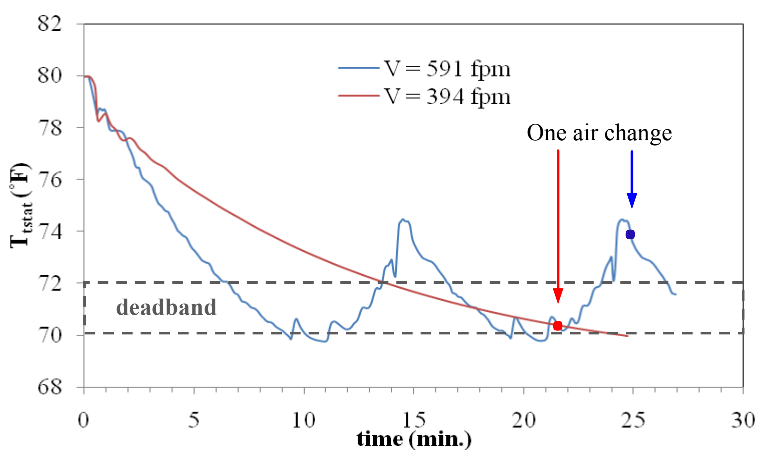


Figure 2 *Temporal evolution of air temperature near the thermostat in Room A when supplied by* T_S *= 55ºF (285.9 K).*

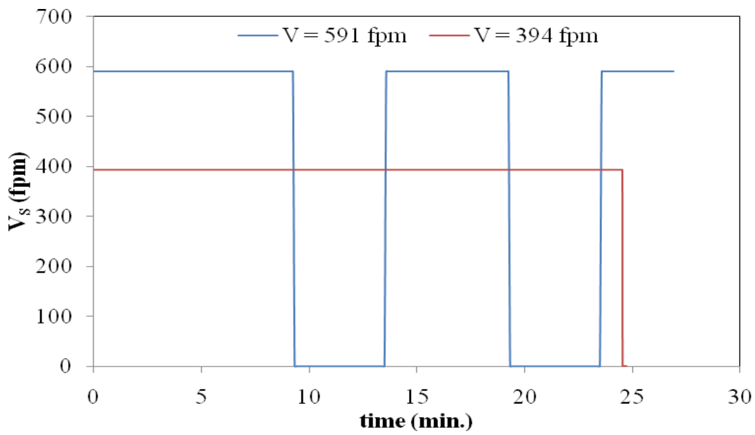


Figure 3 *Temporal evolution of the supply velocity at the register in Room A when supplied by T_S = 55ºF (285.9 K).*

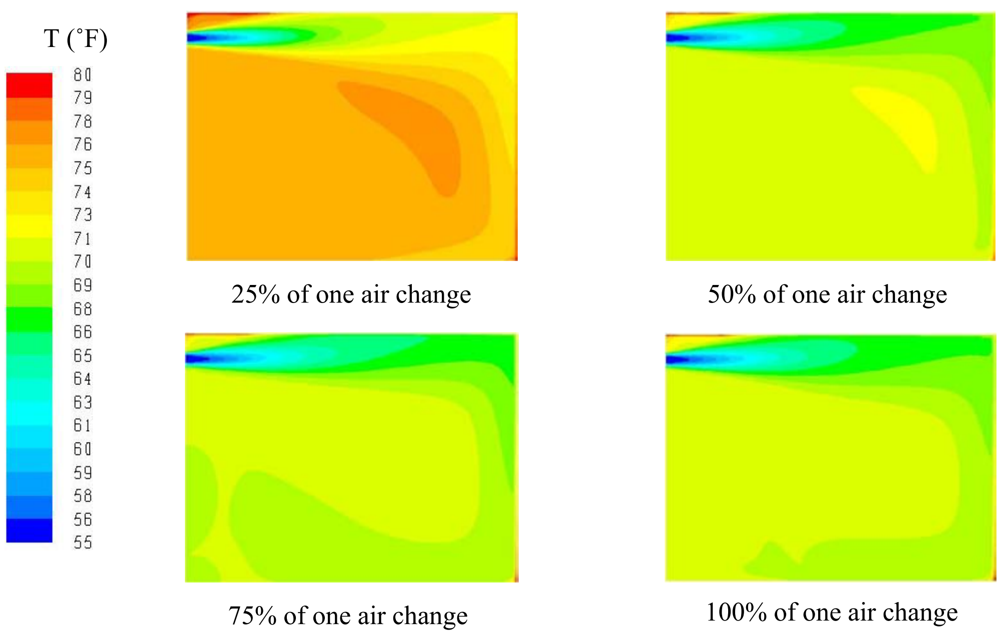
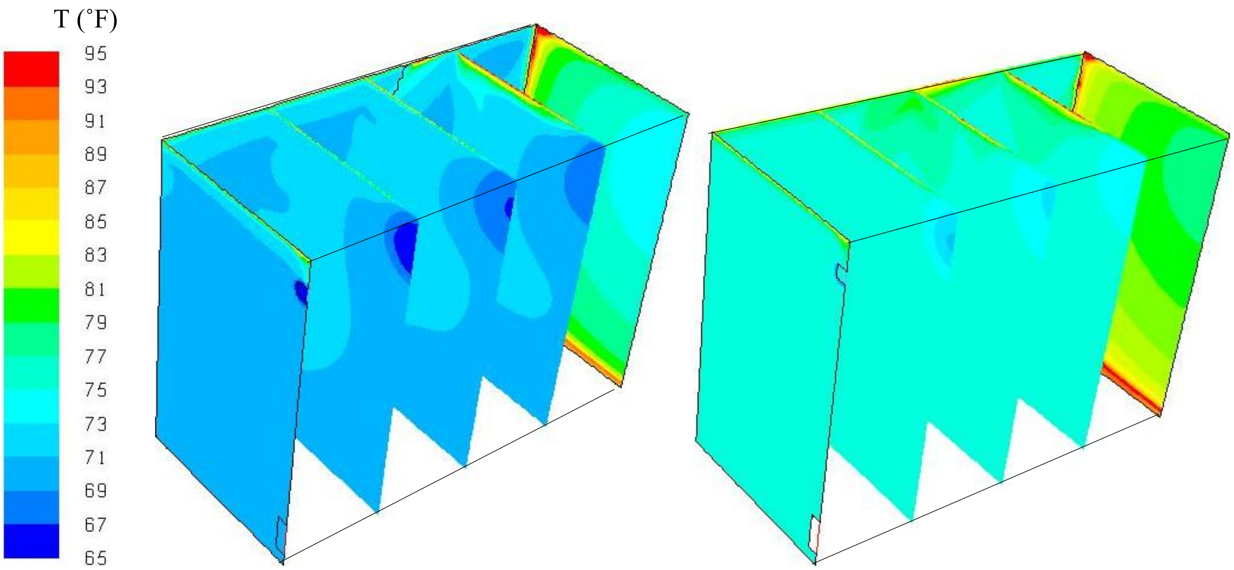


Figure 4 *Temperature distribution at the symmetry plane of Room A when V_S = 788 fpm (4 m/s) and T_S = 55˚F (285.9 K).*

A different view of the flow structure is illustrated in Figure 5 at selected cross sections along the room. This was at an intermediate supply velocity V_S = 591 fpm (3 m/s). The left image was for T_S = 55ºF (285.9 K) and the right image was for T_S = 65ºF (291.5 K), both at one air change. The temperature fields showed well mixed air within the room with some hot regions near the ceiling and at the wall opposite to the diffuser throw. These hot spots, which were a result of thermal loads applied at these locations, were more intense at higher supply temperature of 65ºF (291.5 K).

Figure 6 shows the plots of acceptable draft temperature, between -3˚ and 2˚F, on the top row and the plots of draft temperature at full scale on the bottom row. Three supply velocities of 394 fpm (2 m/s), 591 fpm (3 m/s), and 788 fpm (4 m/s) were tested and the ADPI was evaluated at one air change. The red line limits the occupied zone. Regions of acceptable draft temperature are larger at low velocity and decrease as the velocity increases. As a result, the velocities of 394 fpm (2 m/s) and 591 fpm (3 m/s) provided good occupant comfort with ADPIs of 97% and 92%, respectively.



(a): T_S = 55˚F (285.9 K)

(b): T_S = 65˚F (291.5 K)

Figure 5 *Temperature distribution in Room A at one air change when supplied by V_S = 591 fpm (3 m/s): (a) T_S = 55˚F (285.9 K) and (b) T_S = 65˚F (291.5 K).*

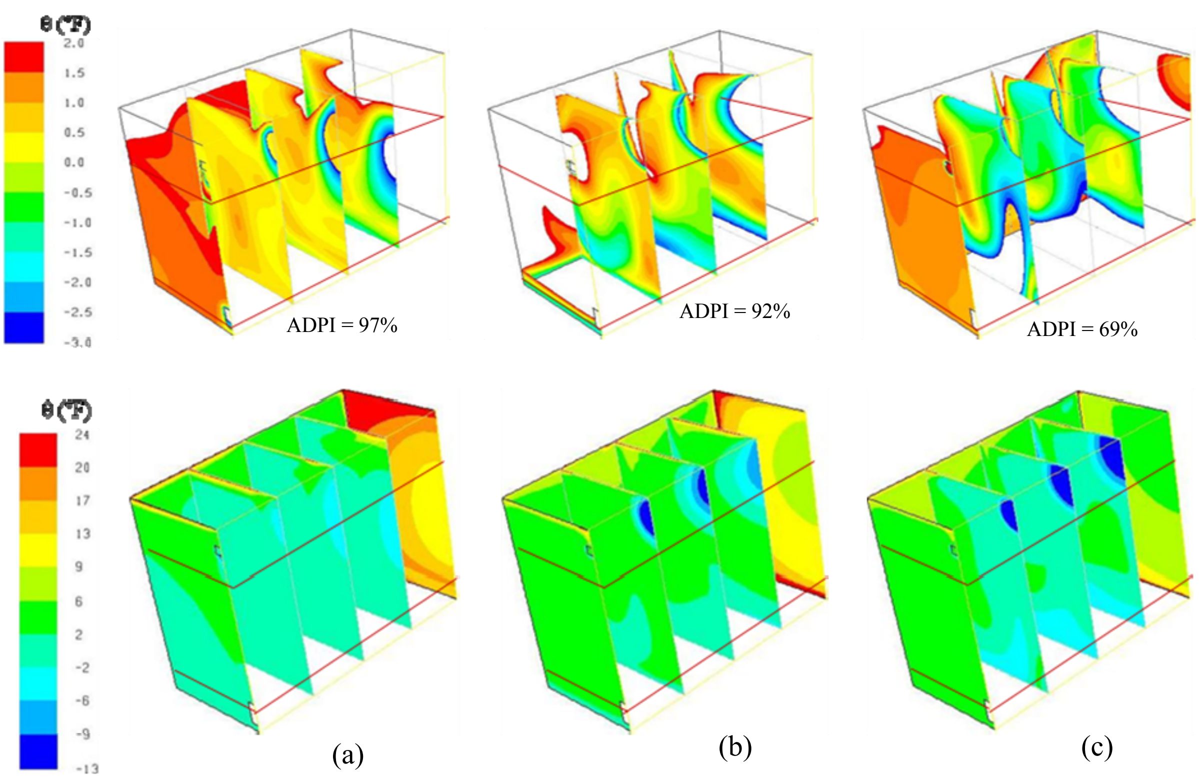


Figure 6 *Distribution of draft temperature in Room A when T_S = 55˚F (285.9 K): (a) V_S = 394 fpm (2 m/s), (b) V_S = 591 fpm (3 m/s), and (c) V_S = 788 fpm (4 m/s). Top row shows acceptable draft temperature, between -3 and 2˚F, and the bottom row shows the draft temperature at full scale.*

The simulation results for Room B are discussed in the remainder of this section. This room is relatively smaller than Room A and is supplied by a larger diffuser. At a fixed supply velocity, one air change will be reached in Room B much faster than in Room A.

Illustrated in Figure 7 is the variation with time of the thermostat temperature of Room B. Two supply air temperatures of 55˚F (285.9 K) and 65˚F (291.5 K) sharing a supply velocity of 788 fpm (4 m/s) were considered. Both curves started at a high value of about 80˚F (299.8 K) and decreased monotonically with time to reach a minimum of about 70˚F (294.3 K) and then entered an oscillation mode as a result of thermostat control. The large temperature values at the start of the simulation were attributed to the initial condition of uniform room temperature of 80˚F (299.8 K). With a supply of 55˚F (285.9 K), the room cooled quickly and stayed within the comfort zone set by the thermostat. However, with 65˚F (291.5 K) air provided at the diffuser, the cooling took longer and for short periods of time, the room temperature went above the comfort zone.

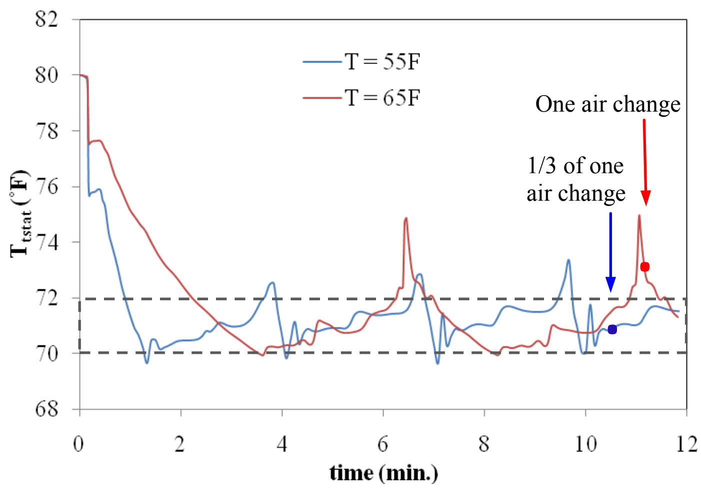


Figure 7 *Temporal evolution of air temperature near the thermostat in Room B when supplied by V_S = 788 fpm (4 m/s).*

The flow visualization in Room B is presented in Figure 8 by the distributions of temperature and velocity at the symmetry plane. Three velocities of 329 fpm (1.67 m/s), 558 fpm (2.83 m/s), and 788 fpm (4 m/s) supplying 55˚F (285.9 K) air were simulated. At low velocity of 329 fpm (1.67 m/s), the air jet lost momentum towards the middle of the room and sank into the occupied zone, which may result in occupants' discomfort. A different scenario was observed at 558 fpm (2.83 m/s) and 788 fpm (4 m/s), where the jet had enough momentum to cross the room and circulate along the wall. Better mixing occurred at the higher velocity of 788 fpm (4 m/s) as depicted by the velocity vectors in Figure 8(d).

Comfort issues are discussed here by measuring the ADPI in the occupied zone of Room B. The outcome is shown in Figure 9 for the supply temperature of 65˚F (291.5 K) and velocities of 329 fpm (1.67 m/s), 558 fpm (2.83 m/s), and 788 fpm (4 m/s). The plots show cells in the occupied zone with acceptable draft temperature and air speed. The occupied zone, which was defined above, is limited by the red line on each plot. Similar to the results obtained for Room A, ADPI was higher at low velocities and decreased as the velocity increased. Room B presented lower comfort levels at high velocities when compared to Room A. Despite the good mixing obtained at high velocity, ADPI decreased due to the condition on air speed to be below 70 fpm.

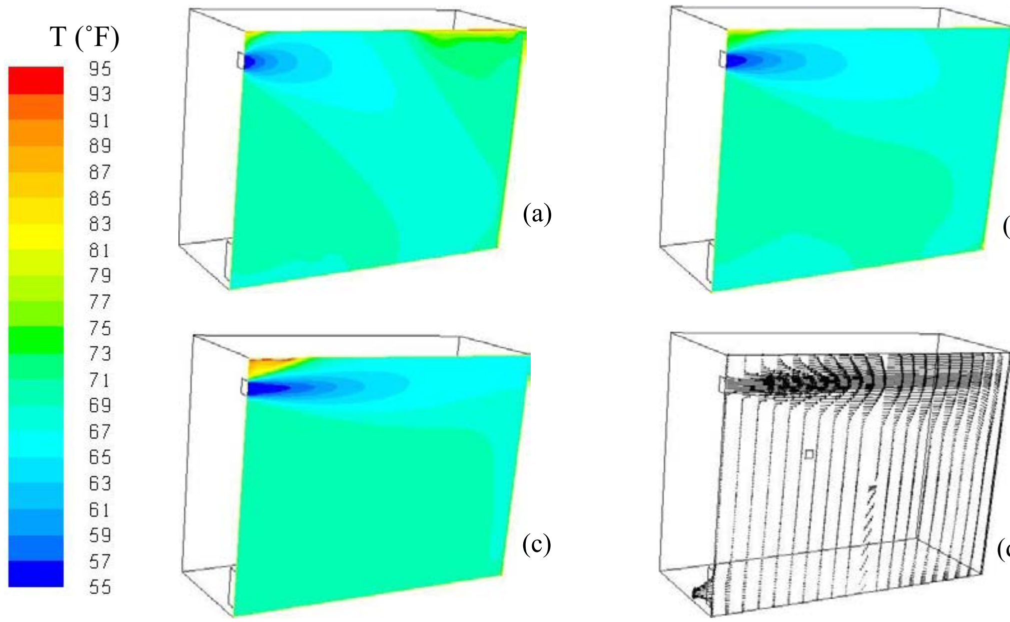


Figure 8 *Temperature distribution (a-c) and velocity vectors (d) at the symmetry plane of Room B: (a) V_S = 329 fpm (1.67 m/s), (b) V_S = 558 fpm (2.83 m/s), and (c-d) V_S = 788 fpm (4 m/s). The supply temperature is 55F (285.9 K).*

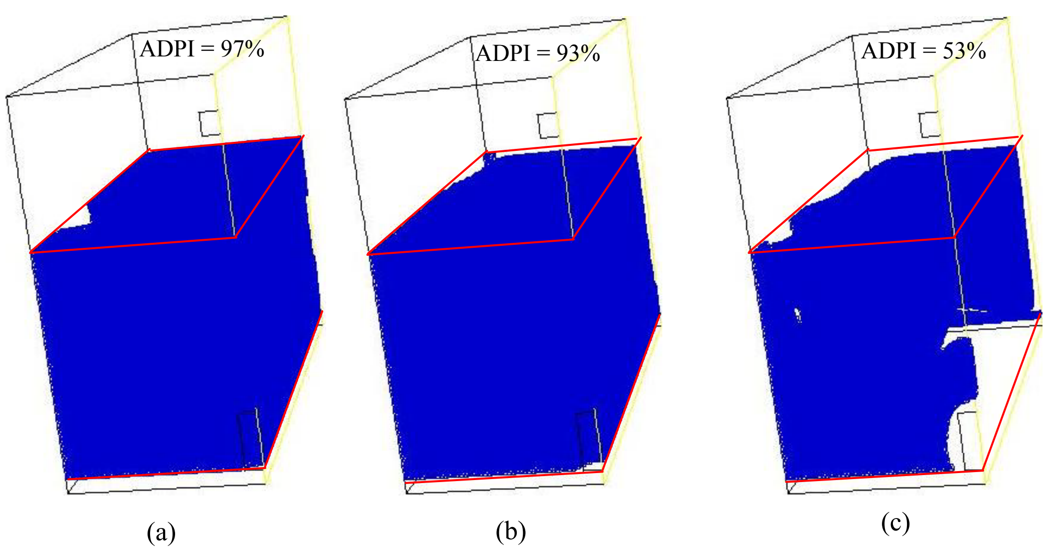


Figure 9 *Cells with acceptable draft temperature and air speed, in the occupied zone of Room B, when T_S = 65˚F (291.5 K): (a) V_S = 329 fpm (1.67 m/s), (b) V_S = 558 fpm (2.83 m/s), and (c) V_S = 788 fpm (4 m/s).*

CONCLUSIONS

The application of high sidewall supply diffusers in residential buildings was simulated in cooling mode. Two diffuser dimensions, based on available products, were considered and evaluated for a wide range of operating conditions. Thermal loads characteristic of high performance homes were applied at the walls of the room and the airflow was controlled by a thermostat. The three-dimensional numerical model was solved using the finite volume method under the platform of FLUENT software. The simulations ended when one air change is supplied to the room.

The effect of thermostatic control and thermal loads on the behavior of these systems is summarized in Figure 10 by the variations of the ADPI as a function of air flow conditions. The comfort levels measured by the ADPI decreased with the supply velocity for all the cases considered. The supply air temperature of 65ºF (291.5 K) provided higher ADPIs, in Room A, compared to 55ºF (285.9 K). Also, under similar operating conditions, Room A with a smaller diffuser (smaller flow rate) showed better performance. The supply air with higher density tends to sink and easily mixes with the room air. At high velocities, however, the ADPI decreased due to the draft.

In summary, high sidewall supply diffusers were proven to achieve good mixing in the room and provide acceptable levels of comfort. It is recommended to use air supply velocity in the range between 200 fpm (1.02 m/s) and 600 fpm (3.05 m/s). In this range, both supply temperatures of 55ºF (285.9 K) and 65ºF (291.5 K) provided comparable comfort levels. These recommendations are valid for both 8 in. × 4 in. and 10 in. × 6 in. diffusers.

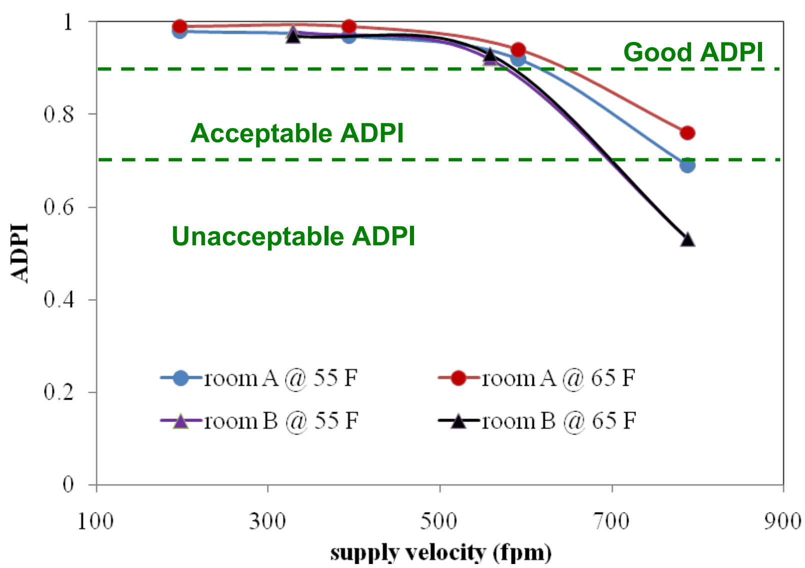


Figure 10 *Variations of the Air Diffusion Performance Index (ADPI) as a function of supply velocity.*

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