COMPUTER MODELING OF UNDERFLOOR AIR SUPPLY SYSTEM

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ABSTRACT
An underfloor air supply system is considered to provide better thermal distribution than a conventional overhead air supply system. A computer model of an environmental chamber under construction was developed to investigate the performance of these two systems. Computational fluid dynamics (CFD) was applied in this research to compare the temperature and air speed distribution patterns of both systems under different supply air grill locations. The CFD modeling has shown the underfloor air supply system to provide improved temperature distribution over the conventional overhead system. Under identical conditions, an underfloor air system can achieve lower air temperatures within the seated zone above the floor than possible with an overhead system leading to the possibility of energy savings. Initial results also indicated less air turbulence with an underfloor system.

INDEX TERMS
Underfloor air supply, Computational fluid dynamics, Temperature distribution, Thermal comfort

INTRODUCTION
Conventional HVAC systems for office buildings supply and return air from the ceiling. This typically results in inefficient cooling of a space. The mixing nature of overhead supply systems results in increased energy use due in part to lower air temperature and higher supply air volume needed to counteract the rising internal heat load. An underfloor air supply system can potentially resolve these issues. Using the natural forces of convection, an underfloor air supply system can achieve the same indoor temperature as that of overhead systems but with warmer air and lower volume. (Loudermilk 1999, Akimoto et al. 1995, Matsunawa et al. 1995) Underfloor air supply also eliminates the problem of short-circuiting fresh supply air by supplying air directly into the breathing zone.

This study examines the thermal and velocity distribution performances of an overhead supply system and variations of an underfloor air supply system to determine which is more effective in cooling an office environment. A computational fluid dynamics (CFD) model is utilized to visually and quantitatively investigate the temperature and velocity distributions of the different air supply methods with regard to supply grill layout and supply air volume.

METHODS
The simulated test room in this study is modeled after an environmental chamber currently under construction known as the Smart Building Module [SBM]. (See Figure 1.) The space is representative of a typical office environment consisting of 2 workstations with desktop computers, flat panel monitors, and 2 occupants. Including the light fixtures, the total internal load of the space is 1,040 W. For the ceiling supply and return system, there are 2 supply

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grills each measuring 0.3m x 0.3m and two return grills measuring 0.1m x 1.2m. For the underfloor supply system, each floor supply diffuser measures 0.3m x 0.3m and the ceiling return grills measure 0.1m x 1.2m. Temperature and velocity measurements were taken at the same three points in the center of the room for each simulation. (See Figure 2.)

Figure 1. Photo of Smart Building Module.

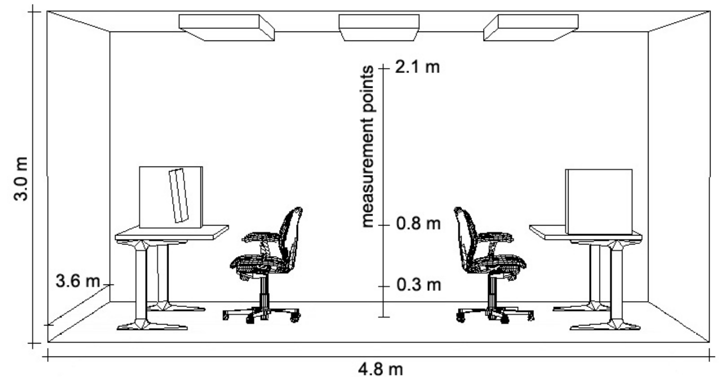
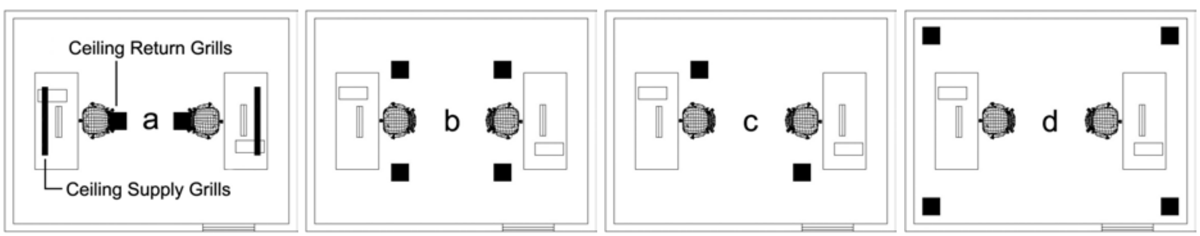


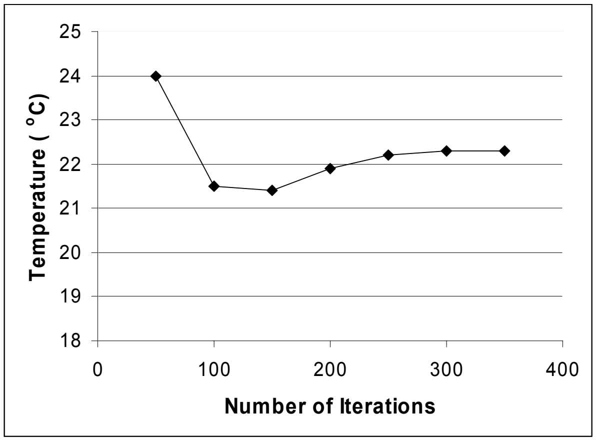
Figure 2. Schematic layout of test room.

Using the CFD model, the temperature and velocity distribution patterns of both overhead air supply and underfloor air supply systems were investigated. Only one layout was simulated for the overhead system and used as the base case, that of a typical office environment. For the underfloor system, three layouts were simulated involving different quantities and locations of the floor diffusers. (See Figures 3a-3d.)



Figures 3a-3d. Air supply and return grill layout of 4 test cases.
a. Ceiling supply and return (base case)
b. 4 center floor supply diffusers
c. 2 center floor supply diffusers
d. 4 perimeter floor supply diffusers
Note: Cases (b) through (d) use the same two linear return grills in case (a) as their returns.

Each simulation was run for 300 iterations using a 60 x 30 x 60 grid. Pre-testing simulations indicated 300 iterations to be sufficient for achieving convergence for this particular study. (See Figures 4 and 5.) The standard k-ε turbulence model was adopted for this study to predict turbulence behavior. For the first phase of this study, the supply air temperature was kept at 18°C (64.4°F) at a total volume of 18.88 l/sec (40 CFM). The second phase of the study involved increasing the total supply air volume in increments from 18.88 l/sec (40 CFM) to 151 l/sec (320 CFM) while maintaining the 18 °C supply air temperature.



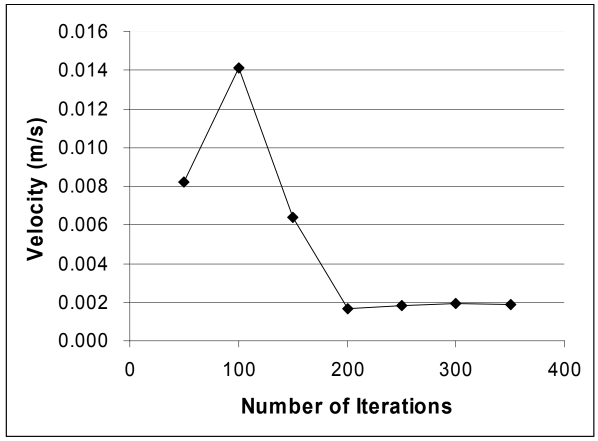
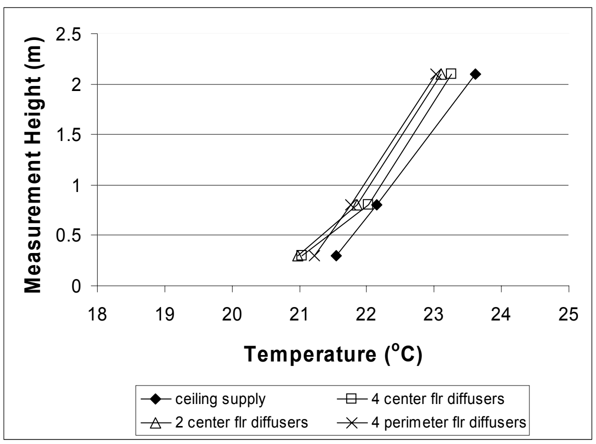


Figure 4. Temperature convergence.

Figure 5. Velocity convergence.

RESULTS

In a cooling mode the overhead air supply system performs inefficiently, revealing a higher indoor air temperature relative to the three underfloor air supply system layouts given the same supply air temperature (18^{o}C) and total supply air volume (18.88 l/sec). (See Figure 8.) Also, there is greater fluctuation in air velocity for an overhead system than an underfloor air supply system. (See Figure 9.) For both temperature and velocity, there is not much variance in the values amongst the three underfloor system layouts.



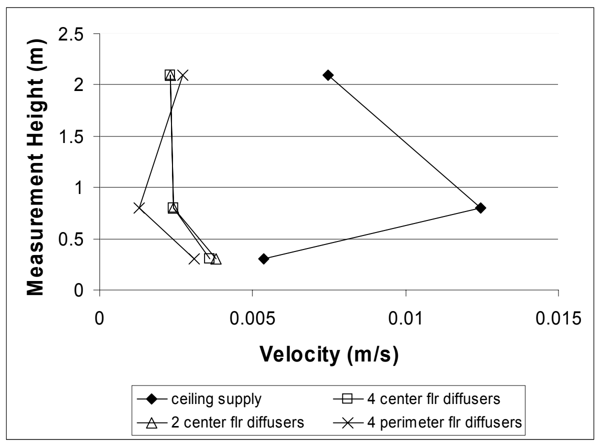


Figure 8. Supply grill location effect on temperature.

Figure 9. Supply grill location effect on velocity.

At a height of 0.8m above the floor, the simulation results indicate thermally comfortable conditions for all four cases. Given the assumptions in Table 1 below, the Predicted Mean Vote (PMV) values for all four cases present favorable thermal conditions without much variance from each other. (See Figure 10.)

Table 1. Environmental and personal parameter value assumptions for calculating PMV.

Environmental Parameters		*Personal Parameters*	
Ambient temperature (^oC)	Refer to Figure 8	Subject weight (kg)	70
Radiant temperature (^oC)	Same as ambient	Subject surface area (m^2)	1.8
Relative humidity (%)	50	Clothing insulation (clo)	0.6
Air velocity (m/s)	Refer to Figure 9	Metabolic rate (met)	1.2
		Exposure time (min)	60

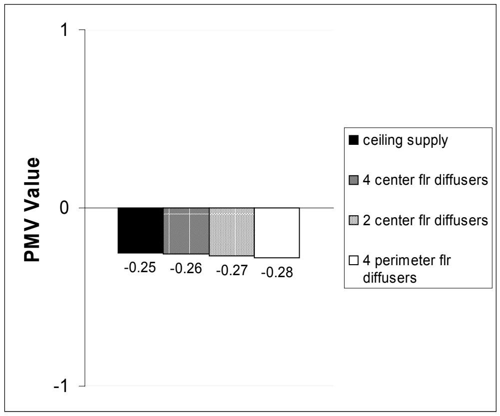


Figure 10. PMV values with 18^{o}C supply air temperature and total supply air volume of 18.88 l/sec.

Even with incremental increases in total supply air volume, the ceiling supply system continues to remain at an unfavorably higher air temperature measured at 0.8m above the floor than the underfloor system layouts. (See Figure 11.) One exception is the two center floor diffuser layout at 151 l/sec. This case results in a slightly higher temperature than the overhead system. The more dramatic changes resulting from increased supply air volume is apparent with velocity. (See Figure 12.) It was also revealed that the overhead supply system to be more susceptible to increases in supply air volume in regards to velocity distribution.

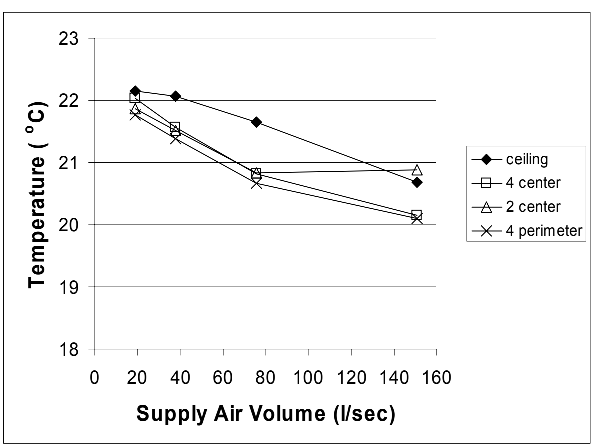


Figure 11. Temperature variation with supply air volume.

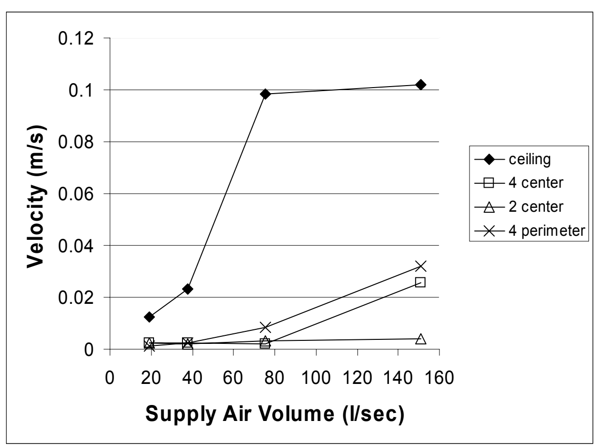


Figure 12. Air velocity variation with supply air volume.

Even with the differences in temperature and velocity for the four cases, simulation results indicate thermally neutral conditions at 0.8m above the floor with 75.51 l/sec of total supply air volume. Given the same assumptions as in Table 1 above (but substituting the values of ambient temperature and air velocity with those in Figures 11 and 12), the PMV values for all four cases again present favorable thermal conditions without much variance from each other. (See Figure 13.)

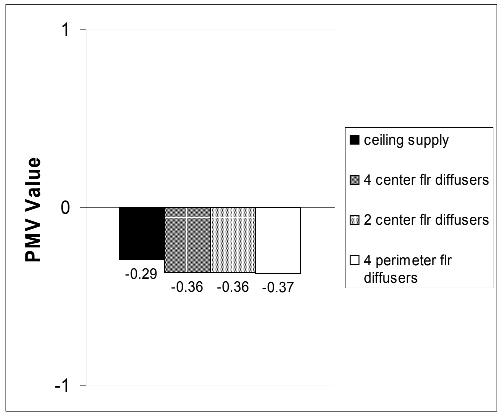


Figure 13. PMV values with 18^{o}C supply air temperature and total supply air volume of 75.51 l/sec (6.5 air changes per hour).

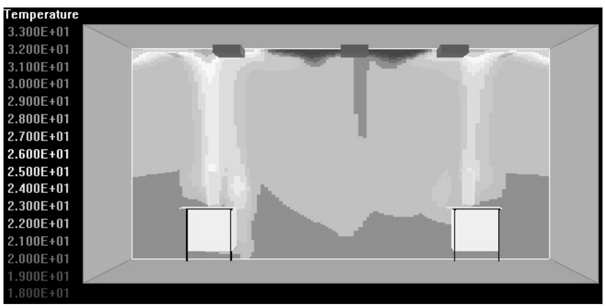


Figure 14. Temperature distribution for ceiling supply.

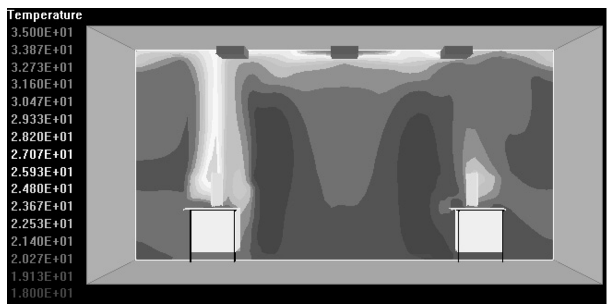


Figure 15. Temperature distribution for 4 center floor diffusers.

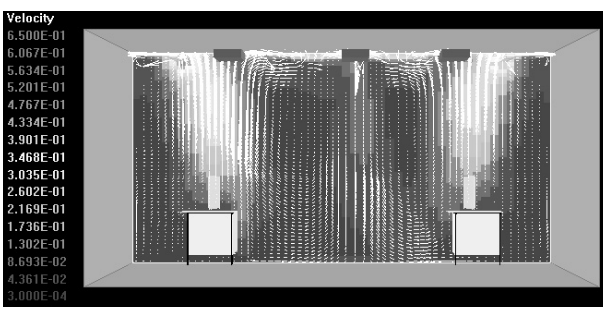


Figure 16. Velocity distribution for ceiling supply.

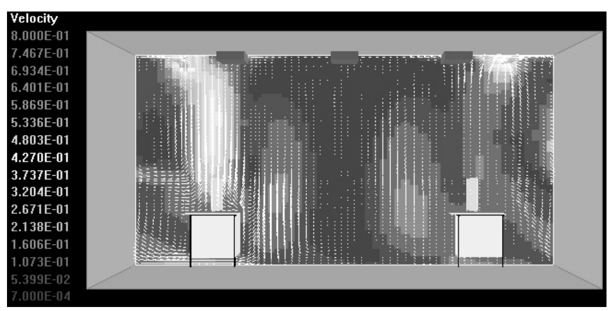


Figure 17. Velocity distribution for 4 center floor diffusers.

At a total supply air volume of 75.51 l/sec delivered at 18^{o}C, the CFD simulation graphic outputs above illustrate the temperature and velocity distribution patterns between the ceiling supply and the four center floor diffuser layouts. A comparison between Figures 14 and 15 show the underfloor system to be more effective at maintaining a cooler temperature within the occupied zone of the space. A comparison between Figures 16 and 17 indicate there is potentially more turbulent air movement with the overhead system than underfloor system.

This is evident in the velocity vector swirl patterns of Figure 16. The air movement in Figure 17 is predominantly in a straight upward motion indicating less turbulence.

DISCUSSIONS AND CONCLUSIONS

Given the close PMV values for all the cases under the various conditions, thermal conditions are comfortable for all four cases. The underfloor system maintains overall lower temperatures in the occupied zone of the space compared to the overhead system. Furthermore, the three underfloor diffuser configurations tested did not result in major differences in temperature and velocity values in the center of the room. This suggests it is possible to reduce the number of floor diffusers while increasing supply air volume without causing much negative effect on thermal comfort conditions.

From this study, it can be concluded that the underfloor system can potentially provide similar thermal conditions as an overhead system but with greater energy savings. Because temperatures in the occupied zone of the space were lower with an underfloor system, such an air supply system can provide conditioned air at a higher temperature than possible with an overhead system. The higher supply air temperature results in energy savings over the conventional system.

The realizable energy savings possible with the higher supply air temperature of an underfloor air supply system presents a potentially effective means of addressing the need for reduced energy consumption in commercial buildings today and in the future.

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