Experimental and numerical study of air temperature distribution of an underfloor air distribution system

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ABSTRACT

A numerical model of an underfloor air distribution system was created to aid future research on the system's environmental performance. A validated numerical model can serve to minimize the need for costly experimental setups. This paper presents the results of an experimental and numerical study of the air temperature distribution of an underfloor air distribution system. A full scale mock up of an interior office space (26.75 m^2) was constructed for the experiment and computational fluid dynamics (CFD) was utilized for the numerical portion of the study. The numerical model was based on experimental data collected from four test cases involving supply air temperature of 18.3 ^oC and variations in supply air volume (56.6 L/s and 37.8 L/s) and diffuser throw angle (63^o and 90^o).

The findings of the study revealed a close match in air temperature measurements and vertical temperature distribution profiles between the experimental and numerical cases. The overall difference between measured and simulated values of temperature was ± 2.30 percent. That is a total difference of 4.60 percent which is much less than the 10 percent to 20 percent differences noted by Lemaire (1993), Heiselberg (1997), Roos (1999), and Bartak et al. (2001) in their reports on indoor airflow measurements and simulations. Overall, the simulated data tended to be 0.53 ^oC cooler than the measured data with an average difference of ± 0.95 ^oC. Considering the accuracy between measured and simulated results was 4.60 percent on average, less than the differences reported by other researchers by almost half or more, it is reasonable to conclude that the accuracy of this numerical model is sufficient enough to rely upon for future studies involving this underfloor air distribution system.

1. INTRODUCTION

Physical testing of environmental control systems can be time consuming and expensive. On the other hand, numerical studies of a system can save both time and money. However the results of the numerical models need to be measured against experimental data to determine the reliability of the numerical data.

Therefore, this research set out to conduct an experimental and numerical study of air temperature distribution of an underfloor air distribution (UFAD) system. By creating a validated numerical model of the UFAD system based on its physical performance data, further studies involving this UFAD system could be conducted in a more time efficient and cost effective manner. Such a model could also make it possible to study further scenarios not feasible with experimentation due to issues of cost, time, or other resource limitations.

2. EXPERIMENTAL STUDY

The experimental portion of this research utilized a functional full scale mock up of an interior office environment equipped with an UFAD system as shown in Figures 1 and 2. The office had an overall interior volume of

73.4 m^3 and was built inside an environmentally controlled space. The UFAD system was comprised of four square floor diffusers of 0.065 m^2 each and two conventional square 0.372 m^2 ceiling return grills. The test office also consisted of a total heat load of 960 W based on two occupants, two computer workstations, and lighting.

The test office was equipped with sensors that measured air temperature, air velocity, and humidity. Air temperature was measured in twenty six locations using a Solac V multi-channel thermocouple data acquisition system. Air velocity was measured in twelve locations using a Kanomax 1560 Multi-Channel Anemomaster configured with twelve omni-directional anemometers. Humidity was measured in two locations using two Thermo Recorder TR-72S loggers. The measurement locations for temperature are illustrated in Figures 1 and 2.

Air temperature distribution performance of the UFAD system was tested under four scenarios involving supply air temperature of 18.3 ^{o}C and two variations each in supply air volume (56.6 L/s and 37.8 L/s) and diffuser throw angle (63^{o} and 90^{o}) as shown in Figure 3. All scenarios were tested under a steady state condition. This physical test environment set the parameters and boundaries for the numerical model and the collected experimental data served to verify the accuracy of the numerical simulation results.

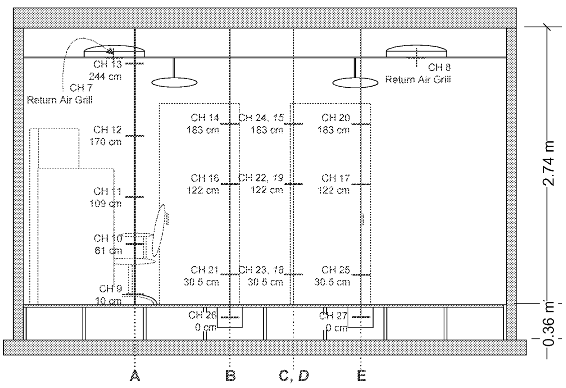


Figure 1: Section of test office. A-E indicate horizontal measurement locations.

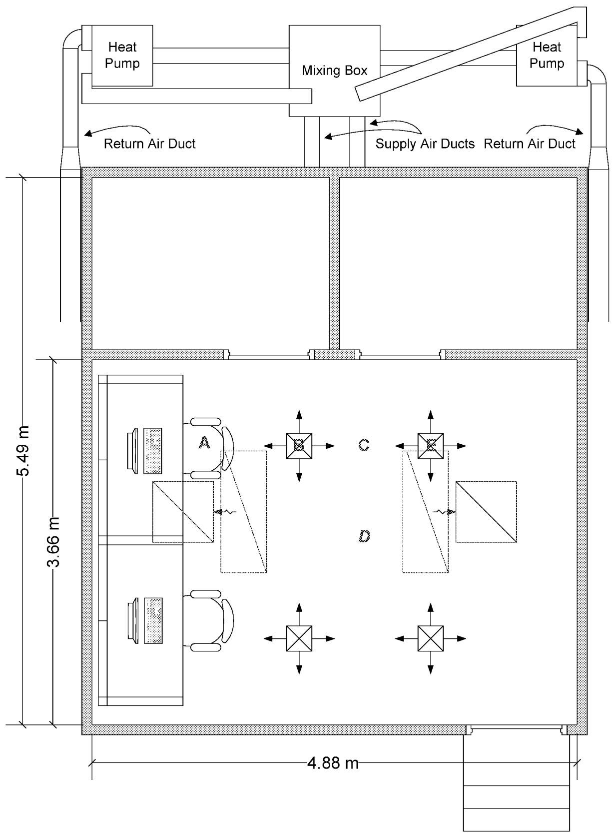


Figure 2: Floor plan of test office.

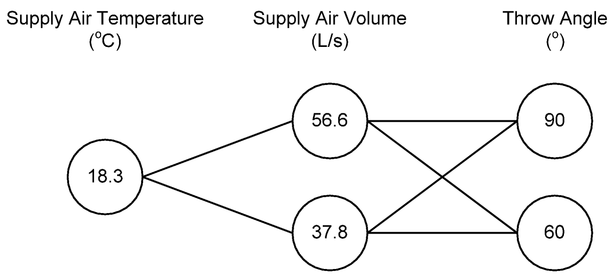


Figure 3: Variation of test variables.

3. NUMERICAL STUDY

The second research method, numerical simulation, involved the use of the computational fluid dynamics (CFD) software FLOVENT. The physical and performance data from the test office were used to develop the numerical version of the office.

3.1 Pre-testing of Numerical Model

Once the physical parameters of the test office were computer modeled, preliminary simulations were run for sensitivity analysis to determine the appropriate grid setting and number of iterations for the numerical model. Both the grid setting and number of iterations are important factors in obtaining accurate and timely simulation results. The preliminary simulations were based on the base case combination of 18.3 ^{o}C supply air temperature (SAT), 56.6 L/s supply air volume (SAV), and a throw angle (TA) of 90^{o}.

Because CFD calculates the forces of the environment through the sub-division of the three dimensional model into a set of non-overlapping, contiguous finite volumes referred to as nodes or cells over each of which calculations are performed that impact calculations of adjacent cells, selecting the proper grid or cell size is important to the accuracy of the simulation. In other words, the finer the grid or smaller the cell, the greater the number of cells that make up a model and the more detailed and accurate the measurements can be at a particular location.

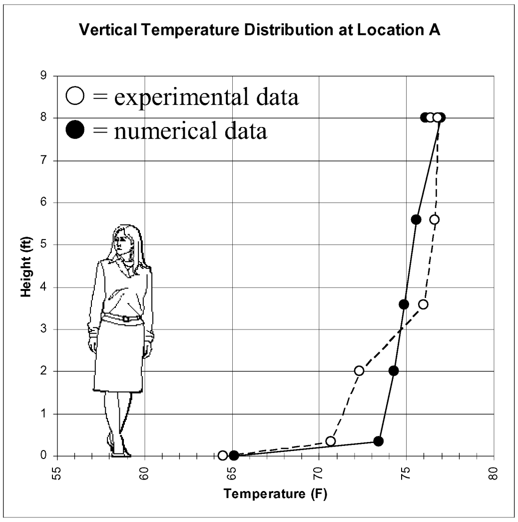
However, increasing the density of the grid results in a longer simulation run time due to the increase in the number of cells that require computation. Furthermore with increasing grid density, there is a point of diminishing return between simulation accuracy and grid size. To find this point of diminishing return that balances simulation accuracy with simulation run time, cell sizes ranging from a maximum of 30.48 cm to as fine as 2.54 cm were tested to determine an appropriate grid setting.

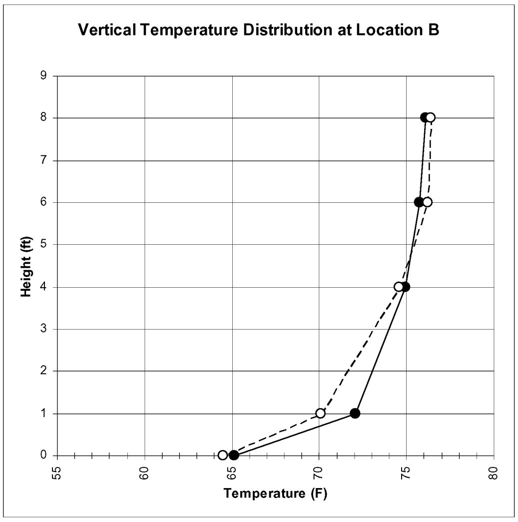
Because the details of the grid and iteration testing are beyond the scope of this paper, only the main findings of the pre-testing of the numerical model are presented. Although the tests showed there were no large temperature differences between the 7.62 cm, 5.08 cm, and 2.54 cm grid settings, the 5.08 cm setting was selected for three reasons: economy of time compared to the 2.54 cm grid (13 times faster), higher resolution output than 7.62 cm grid, and lower standard of deviation compared to the 7.62 cm grid. The 5.09 and 2.54 cm grids had a standard deviation of 0.19 whereas it was 0.93 when compared to the 7.62 cm grid.

All the CFD simulations were run using the k-ε turbulence model with stratification on. The solution was set for flow and heat transfer and run under steady state.

4. RESULTS AND DISCUSSION

Verification of the numerical models was based on a comparison of the experimental and numerical data. Data from the twenty six air temperature measurement points in the experiments were compared to the same locations in the numerical models. Charts 1 and 2 present the vertical temperature distribution comparison between the experimental and numerical data at locations A through E for two of the four test cases.





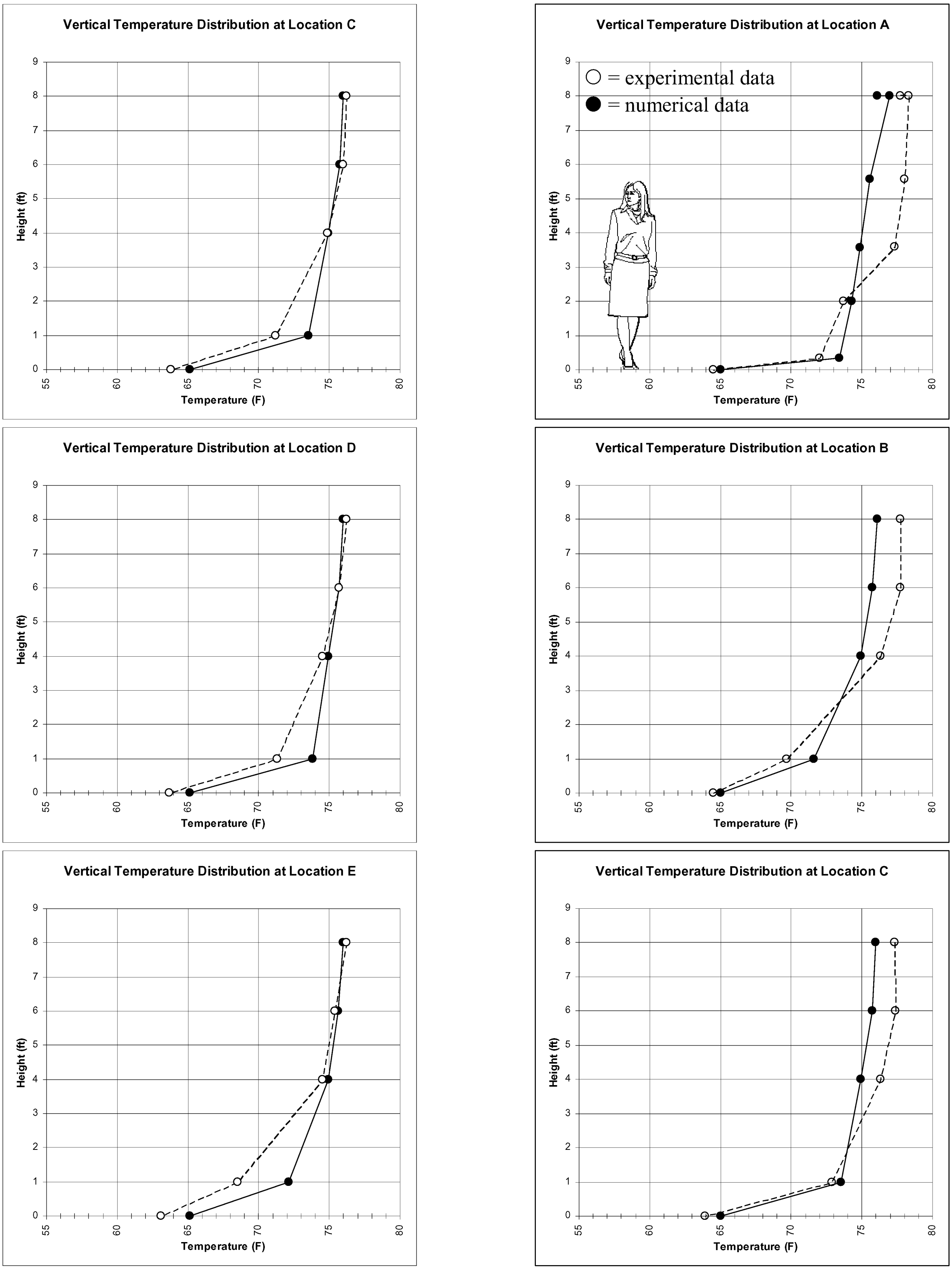


Chart 1: Experimental and numerical vertical temperature comparison for test case 18.3°C SAT_37.8 L/s SAV_63° TA.

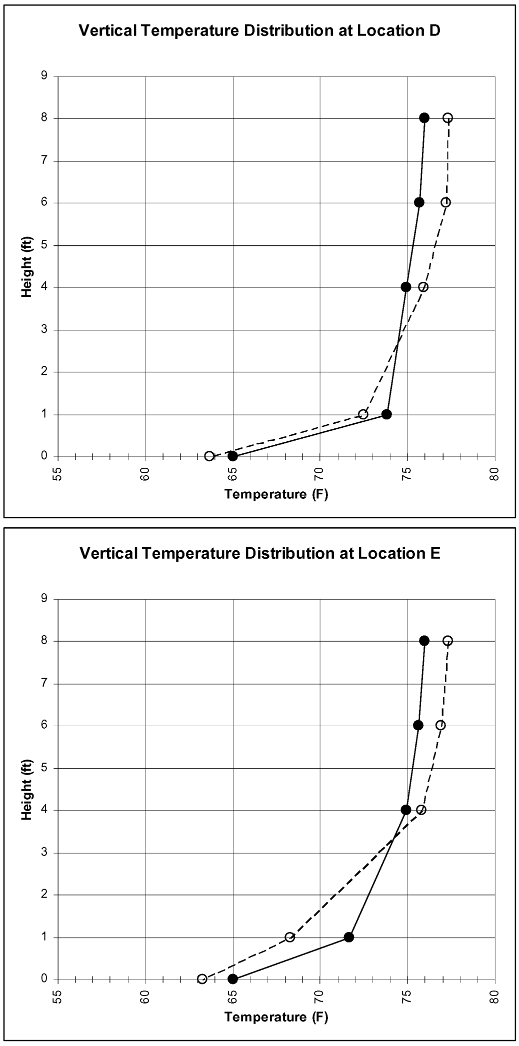


Chart 2: Experimental and numerical vertical temperature comparison for test case 18.3°C SAT_37.8 L/s SAV_90° TA.

For each case the temperature difference was calculated for each of the twenty six corresponding measurement points in the experimental and numerical data sets. The average values for each case is shown in Table 1. Standard deviation values were also calculated between the experimental and numerical data points for each case. The percentage difference between the experimental and numerical temperature data for each case was calculated by calculating the percentage difference of standard deviation from the mean measured temperature.

Table 1: Summary of experimental and numerical temperature data of the four test cases.

Test Cases (SAT_SAV_TA)	Avg. Temp. Cooler or Warmer: Numerical vs. Experimental Data (°C)	Avg. Temp. Difference: Numerical vs. Experimental Data (°C)	Standard Deviation Between Numerical and Experimental Data	% Temp. Difference of Numerical from Experimental Data
18.3°C_37.8 L/s_63°	0.71 warmer	± 1.02	0.96	± 1.41
18.3°C_37.8 L/s_90°	0.36 cooler	± 1.45	0.62	± 1.97
18.3°C_56.6 L/s_63°	2.00 cooler	± 2.01	1.29	± 2.77
18.3°C_56.6 L/s_90°	2.17 cooler	± 2.23	1.13	± 3.06
Overall Values				
	0.95 cooler	± 1.70	1.94	± 2.30

As noted in Table 1 above, the overall difference between experimental and numerical values of temperature was ± 2.30 percent. That is a total difference of 4.60 percent which is less than the 10 percent to 20 percent differences noted by Lemaire (1993), Heiselberg (1997), Roos (1999), and Bartak et al. (2001) in their reports on indoor airflow measurements and simulations. It is also less than the differences of up to 25 percent reported by Davidson and Olsson (1987). The increased accuracy between measured and simulated data of this study compared to earlier studies is probably best attributed to advances in CFD technology over the past twenty plus years, particularly in the areas of grid generation and turbulence modeling. It should also be noted that while this and the prior studies performed by others all investigated the measured and simulated data of indoor airflow, this study looked at an underfloor air distribution system while the others were based on overhead air distribution systems.

Overall, the numerical data tended to be 0.53°C cooler than the experimental data with an average difference of ± 0.95°C. However, when the 37.76 L/s and 56.63 L/s combinations were analyzed separately, the 37.76 L/s combinations provided a smaller percentage difference than the 56.63 L/s combinations, ± 1.69 percent versus ± 2.68 percent with a standard deviation of 0.79 versus 1.21 respectively.

Considering the accuracy between experimental and numerical results was 4.60 percent on average, or 3.38 percent and 5.36 percent if calculated separately by L/s, less than the differences reported by other researchers by almost half or more, it is reasonable to conclude that the accuracy of the computer model is sufficient enough to rely upon for future studies.

There were also several limitations related to this research. The clearest limitation was the fact that the experimental data was collected in a very controlled setting and both the experiment and simulations were conducted under steady state conditions. For the results to be more representative of the real world environment, future studies need to be conducted under transient state conditions.

Furthermore, the physical test space had a lower than desirable floor to ceiling height of 2.4 m instead of 2.7 m or higher which is more representative of office environments. The height of the test space was a physical limitation of the space in which it was constructed. Because of the impact vertical distance has on thermal stratification, it would be prudent to investigate the significance of floor to ceiling height on temperature distribution of the UFAD system.

Another future direction of the research would be a study of the significance of cooling load on the temperature distribution performance of the UFAD system. Because UFAD works with heat using thermal stratification and buoyancy to supply and distribute conditioned air into the occupied zone the influence of internal heat sources on air distribution is another important issue to address.

5. CONCLUSIONS

With an overall difference between experimental and numerical data points of 4.60 percent, less than half the difference reported in other studies, the numerical model proved to be a fair representation of the experimental set up. There however still remains much room for improvement, in particular scenarios dealing with transient state and environmental conditions beyond the boundaries of this study. So while further studies in areas such as those mentioned in the previous section are needed to create a more accurate CFD model of the UFAD system, this study has shown that the numerical model used in this study is reliable enough for future studies on the system. Once validated with experimental data, numerical models have the ability to serve as more time and cost efficient alternatives to experimentation within the limits of the validated model.

ACKNOWLEDGEMENTS

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