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Thermal Comfort Evaluation of a Heat Pump System using Induced-air Supply Unit

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ABSTRACT

Traditional heat pump systems supply conditioned air to space at certain temperatures such as, in summer, about 16°C. When the supply-air temperature drops too low, most occupants tend to feel uncomfortable. On the other hand, a certain amount of dehumidification has to be carried out and sometimes, the velocity of supply-air has to be high which in turn creates a draught. This paper introduces a new air supply method to reduce fan power consumption as well as to improve thermal comfort of occupants. Using this method, colder fresh air at 13°C is generated by air handling units and pumped to the ceiling supplies. As supply air temperature is lower, less air flow rate is required to meet the latent load. The reduced air flow rate decreases the fan power consumption. The supply air units mounted on the ceiling are specially designed so that space air flows upwards to inside due to negative pressure created by the supply unit and mixed with the 13°C cold air. The mixed air consists of roughly 60% return air and 40% fresh air at 13°C. Due to the mixing process, the supply air temperature is around 19°C and supply air velocity is around 0.7 m/s. Both the supply temperature and velocity are more thermally-favored compared to those from traditional units. A commercially-available computational fluid dynamics (CFD) tool was used to evaluate the operative temperature as well as air velocity distribution inside an office-setting equipped with induced-air supply unit for both heating and cooling cases. Since the air velocity may change due to various latent loads, Proper Orthogonal Decomposition (POD) technique was applied to study the thermal comfort conditions when air velocity changes between 0.1 m/s to 1 m/s. The POD method uses a sophisticated interpolation method based on a set of CFD snapshots which are generated once beforehand. The method can reduce the computation time from hours for CFD to seconds when predicting temperature and velocity fields at different supply air velocities. The model outputs are compared with field measured data for model validation. The simulation results show that the induced-air supply unit creates a good thermal comfort conditions for the occupants in terms of predicted mean vote (PMV). The average PMV of a 45-person office shows a value of - 0.2 in the summer and - 0.6 in winter.

1. INTRODUCTION

For a heat pump system, the determination of supply air temperature should consider both sensible load and the latent load from space. The refrigerant temperature in the evaporator has to be low enough so that sufficient moisture can be removed. This requires the supply air temperature to be typically lower than the dew point temperature of room air. On the other hand, as the supply air temperature decreases, certain reheat process becomes necessary for the purpose of thermal comfort. The reheat process, if achieved by using an electric heater, will increase the compressor power consumption. An alternative option to obtain the same dehumidification capacity is to increase supply air flow rate. Though this option does not lead to a larger compressor power input, it leads to an increase in fan power consumption.

In the paper, a novel heat pump system using induced-air supply units is introduced. The schematic of the induced-air unit is shown in Figure 1. The fresh air from an air handling unit is sent to the unit. The fresh air has a temperature of 13°C which is lower than that of traditional unit, but because of careful design of heat exchangers (not discussed here for brevity), the evaporating temperature is maintained at 10°C. For the same amount of latent capacity, the 13°C fresh air requires less air flow rate and therefore consumes less fan power. As the fresh air is received from the top, the unit is capable of introduce room air through the bottom inlets located at the two sides by induction. Although the fresh air flow is low, the total mixed supply air flow rate is actually higher than that of conventional units. Since the induction does not require any power input, the larger air flow rate does not lead to the increase in fan power consumption. The amount of induced air is proportional to that of fresh air supplied and is typically made up of 60% room air induced with 40% fresh air supply. The total supply air is sent to the space through the middle part at the bottom of the unit. The surface area of the supply unit is relatively larger than that of conventional outlets (for a rated 300 m^3/h supply air flow rate, the surface area of that unit is 0.3 m^2) and the supply air velocity is between 0.2 m/s to 0.9 m/s. The relatively lower air velocity brings the benefit of minimal possibility of draught and hence better thermal comfort. Moreover, due to the mixing with room air, the supply air temperature is around 19°C. Such a temperature does not require any forms of reheat, and consequently provide occupants with comfortable supply air at no extra fan power penalty.

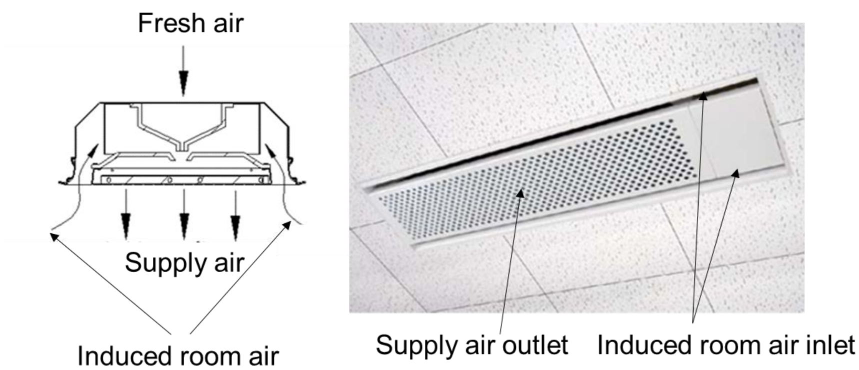

Figure 1: Schematic of induced-air supply unit (left) and the picture of unit after installation (right)

It should be noted that the induced-air supply unit can be used in both heating and cooling cases, and is capable of providing better thermal comfort and lower power consumption compared to traditional systems. However, in this paper, the focus is on the thermal comfort analysis of induced-air supply unit under cooling and heating operation.

2. INDOOR AIR TEMPERATURE CALCULATION

This section describes the methodology used for predicting the indoor air temperature and velocity. A simulation tool based on POD method is developed and used to analyze the thermal comfort created by the induced-air unit. Traditionally, air temperature inside the room has to be done using CFD tools, which are also computationally very expensive. The POD method can assist in significantly reducing the computational effort for interpolation cases.

2.1 CFD modeling and simulation

A commercially-available CFD (ANSYS, 2006) package was chosen for the modeling. Several room models with multiple induced-air units are modeled using 3D double precision option. The size of rooms varies depending on the number of supply units. For one- and two-supply cases, the room dimension is 3 m long, 1.5 m wide and 2.7 m high. For three and four supplies cases, the room is modeled as 4.5 m long, 3 m wide and 2.7 m high. The supplies are uniformly distributed at the top of the room. Each supply is around 1.1 m in length and 0.6m in width. The mesh size of smaller room is 500,000 and 1,260,000 for larger room. Figure 2 shown the computational domain for a room with two outlets. The supply unit is modeled as a face with two return air inlets at two sides and supply air outlet in the middle just as shown in Figure 1. The Bousinessq assumption is applied to enable the simulation of natural convection of indoor air. According to the characteristic length and temperature difference of the model, the Rayleigh number exceeds 10^9, and therefore k-ω SST model is enabled as the turbulent viscous model. The conditioned space also receives solar radiation through frontal wall. The solar heat flux is assumed to 800 W/m^2. Figures 3 to 6 show sample results of indoor air temperature profiles and velocity profiles of one supply case at the middle plane of the room in both heating and cooling conditions. The air temperature profile shows a clear stratification due to air density difference, however the temperature difference from floor to a point 1.5 meter high is within 2K. The velocity profile shows a maximum velocity of 0.9 m/s at the supply duct outlet. The bulk air region has a negligible air velocity showing minimal signs of draught. It also can be found that the return air flows back to the supply unit to represent the induction effect.

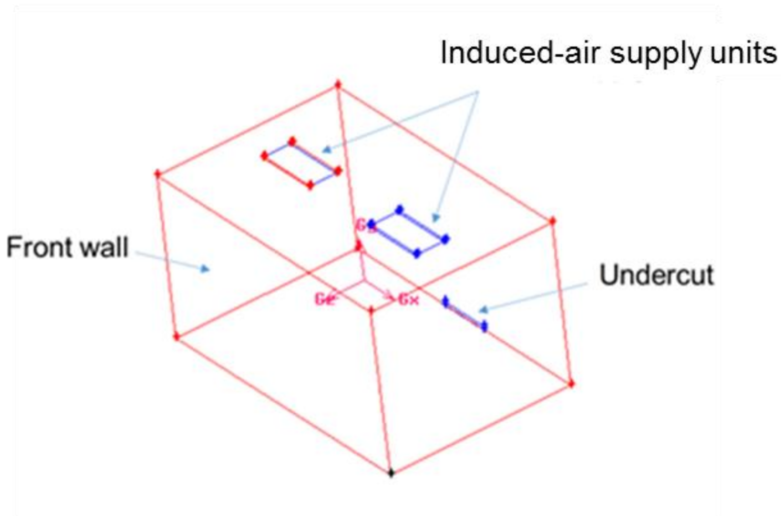

Figure 2: Computational domain for a room with two induced-air supply units

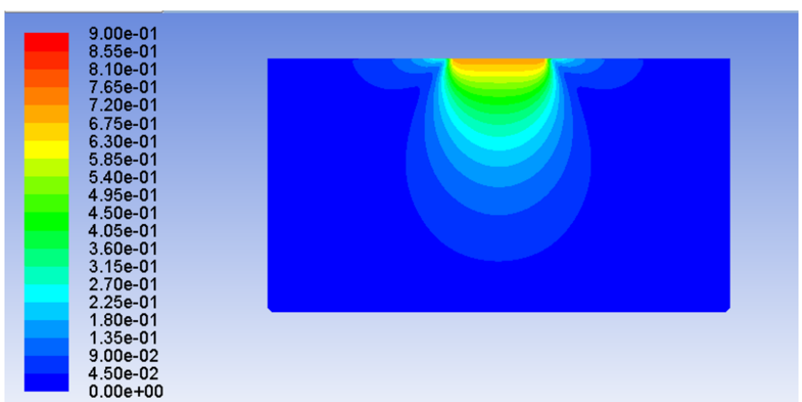

Figure 3: Air velocity (in m/s) profile of one induced-air supply unit (cooling case, middle plane)

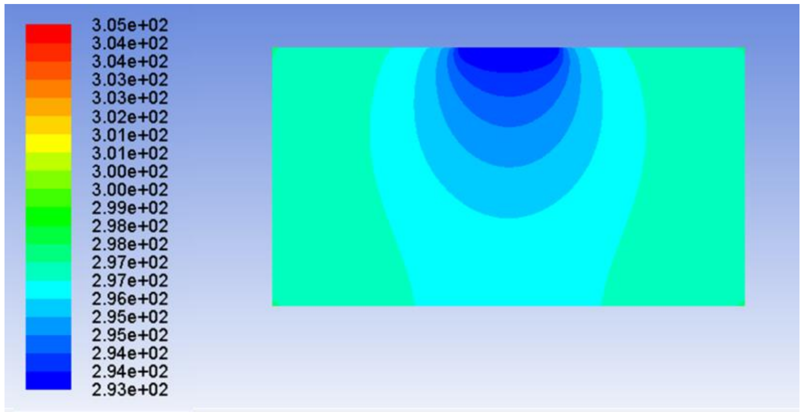

Figure 4: Air temperature (in K) profile of one induced-air supply unit (cooling case, middle plane)

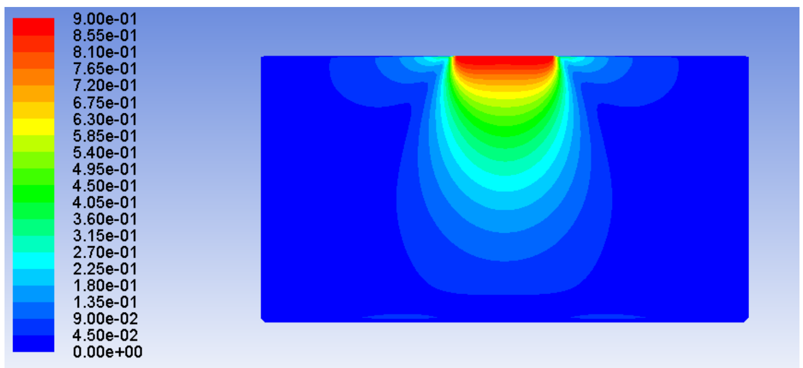

Figure 5: Air velocity (in m/s) profile of one induced-air supply unit (heating case, middle plane)

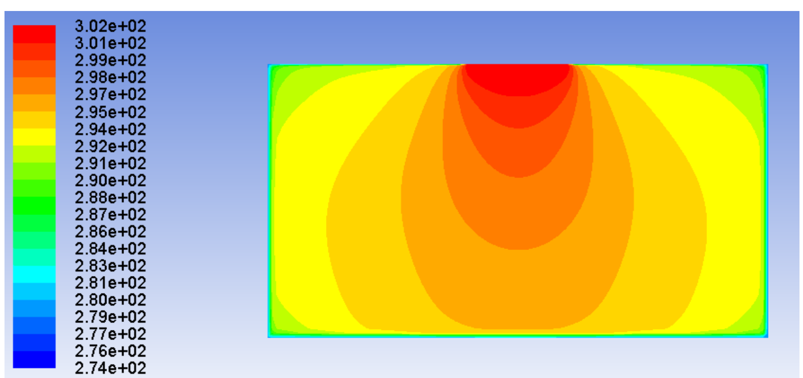

Figure 6: Air temperature (in K) profile of one induced-air supply unit (heating case, middle plane)

2.2 Reduced-order modeling and simulation

Although the previous examples demonstrate that CFD tools can be applied to simulate the air temperature field, the computation cost is high. For one CFD run, it takes from 8 hours to 24 hours depending on the mesh size. Hence it is desirable to develop a reduced-order modeling method which allows for a good compromise between the accuracy and the computation cost. The proper orthogonal decomposition (POD) was chosen as the reduced-order model to replace the above CFD simulation. The POD method was first introduced by John Lumley (Berkooz et al., 1993). In other disciplines the same method is referred to as Karhunen-Loève decomposition or principal component analysis (PCA). It has several advantages as pointed out by Berkooz et al. (1993): (1) it extracts extracting data from experiments and simulations using statistic-based method. (2) Its analytical foundations provide a clear understanding of its capabilities and limitations. The flow chart of the POD method is shown in Figure 5. In short, the method seeks to decompose a large degree of freedom system into a series of expansion as shown in Equation (3).

$$v(x,t)=\sum_{i=1}^{m}a_i(t)\varphi_i(x) \tag{1}$$

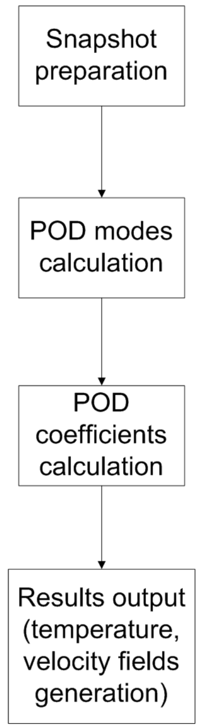

Figure 6: Flow chart of POD method

Skipping the rigorous mathematics, it turns out that the basis functions (φ) are the Eigen functions of the integral equation (Ly and Hein, 2001):

$$\int C(x,x')\varphi(x')dx'=\lambda\varphi(x) \tag{2}$$

where the kernel C is given by:

$$\int C(x,x')=\frac{1}{N}\sum_{i=1}^{N}V_i(x)V_i(x') \tag{3}$$

For the current problem, in order to obtain the POD modes for the air temperature field, a technique called "snapshot" was applied to form the matrix C by utilizing the existing CFD simulation results. Nine sets of temperature fields and velocity fields were chosen to form the snapshot. The difference between each snapshot are values for the boundary conditions (i.e., supply air velocity). A commercial software package (MathWorks, 2013) is used to calculate the eigenvalues of the kernel matrix C using singular value decomposition. If the eigenvalue is zero, indicating that it has no impact on the system anymore, then its corresponding eigenvector is neglected. The proper orthogonal decomposition provides the basis for the expansion series. The next step is to find the coefficients in the expansion (Equation 3). A method called Galerkin projection is considered to be a standard approach to obtain the coefficients. The method projects the governing equations on the modal subspace and then solves the governing equations, usually in the form of ODEs to obtain the coefficients. Figure 7 shows indoor air temperature and velocity profiles in the case of one-supply room. Since it is a reduced-order model, the resolution was downgraded to 16 by 16 by 8. The POD calculation took only ~2 minutes compared to more than 8 hours for a 3D CFD simulation. The significant speed improvement is due because POD conducts interpolation based on CFD snapshots. The validation of the POD model will be discussed in Section 4.

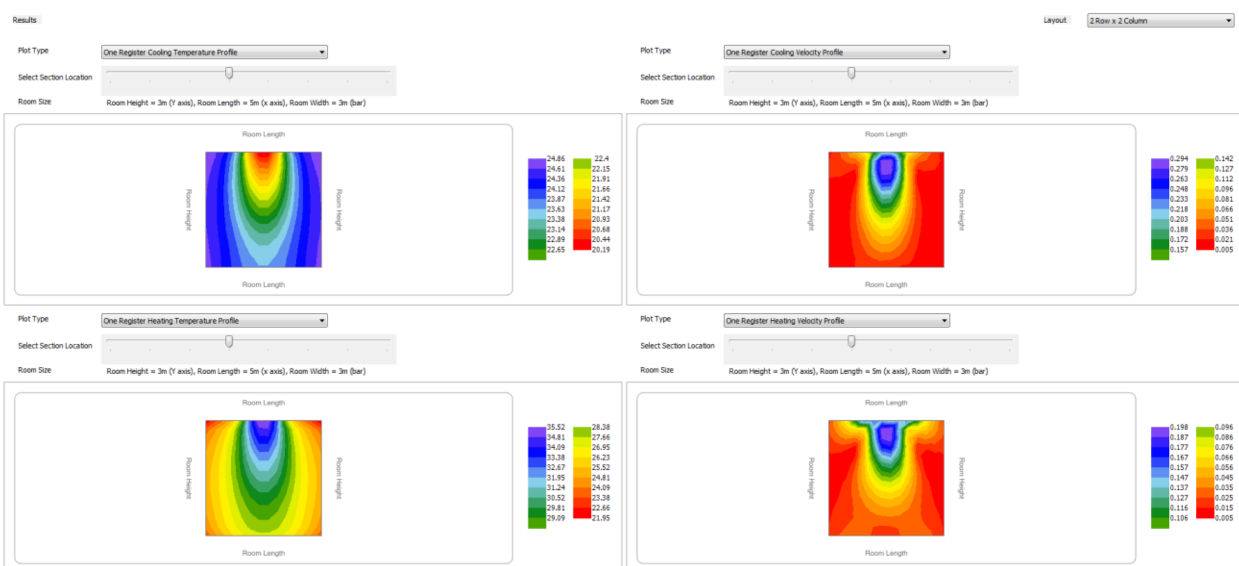

Figure 7: Sample POD results of temperature and velocity profile for the mid-plane (left top: temperature profile during cooling; right top: velocity profile during cooling; left bottom: temperature profile during heating; right bottom: velocity profile during heating)

3. PMV AND PPD FIELD SIMULATION

Although air temperature is an important index for thermal sensation, occupants' thermal comfort evaluation requires more parameters such as humidity and air velocity. Therefore, the PMV and PPD in the rooms are then simulated by the software tool. The PMV is a seven-scale system to describe occupant's thermal sensation from very hot (+3) to very cold (-3). The PPD is its derivative to statistically show what percentage of occupants feel uncomfortable at different PMV scale. Table 1 and Figure 8 describe the PMV and PPD metrics, respectively. To calculate the PMV, Equation 6 is used. Although the equation includes many parameters, most of them can be obtained from the ISO standard 7730 (2005).

$$\begin{aligned} PMV = {} & [0.303e^{(-0.036M)} + 0.028]\cdot\{(M-W) - 3.05\times10^{-3}\cdot \\ & [5733 - 6.99\cdot(M-W) - p_a] - 0.42\cdot[(M-W) - 58.15] \\ & -1.7\times10^{-5}\cdot M\cdot(5867 - p_a) - 0.0014\cdot M\cdot(34 - t_a) - \\ & 3.96\times10^{-8}\cdot f_{cl}\cdot[(t_{cl}+273)^4 - (t_r+273)^4] - f_{cl}\cdot h_c\cdot(t_{cl} - t_a)\} \end{aligned} \tag{4}$$

$$PPD = 100 - 95 - e(-0.03353\cdot PMV^4 - 0.2179\cdot PMV^2) \tag{5}$$

Table 1: PMV scale (ISO 7730, 2005)

PMV scale	Thermal sensation
+3	hot
+2	warm
+1	slightly warm
0	neutral
-1	slightly cool

-2	cool
-3	cold

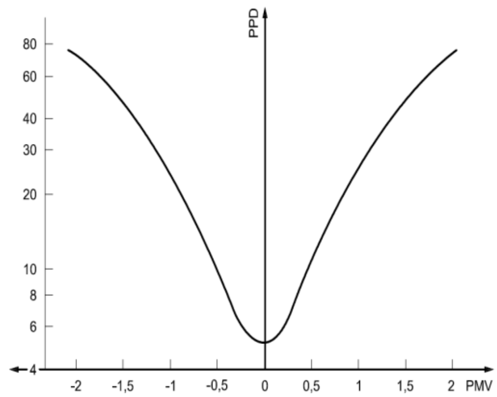

Figure 8: PPD as a function of PMV (ISO 7730, 2005)

Based on the POD outputs of temperature and velocity distributions, the average air temperature and velocity are calculated. Humidity ratio simulation was not included in the model (will be included in the future study) and therefore it is obtained from field measurements. The mean radiation temperature is calculated by using room surface temperature as well as view angles of the center point in the room to individual surface. Figure 9 and 10 show the PMV and PPD prediction for the room based on one typical weather day in Beijing, China. The typical weather days are defined based on the coldest and hottest days in one year.

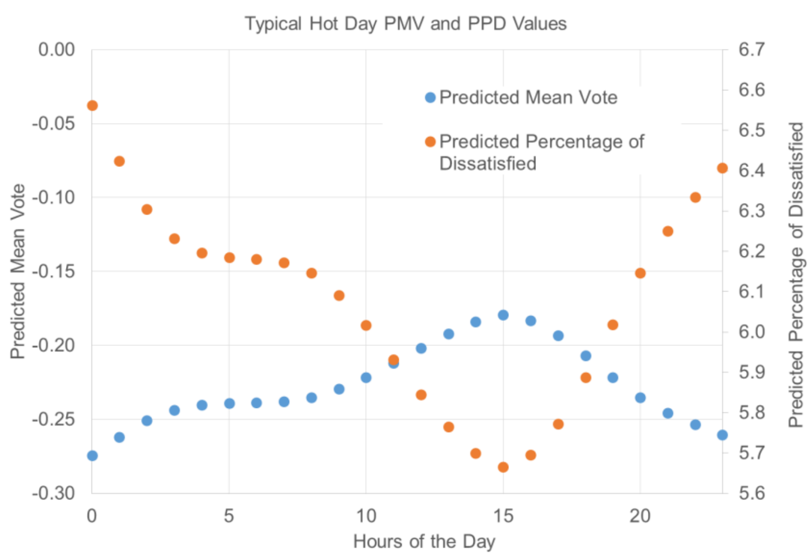

Figure 9: Predicted PMV and PPD for typical hot day

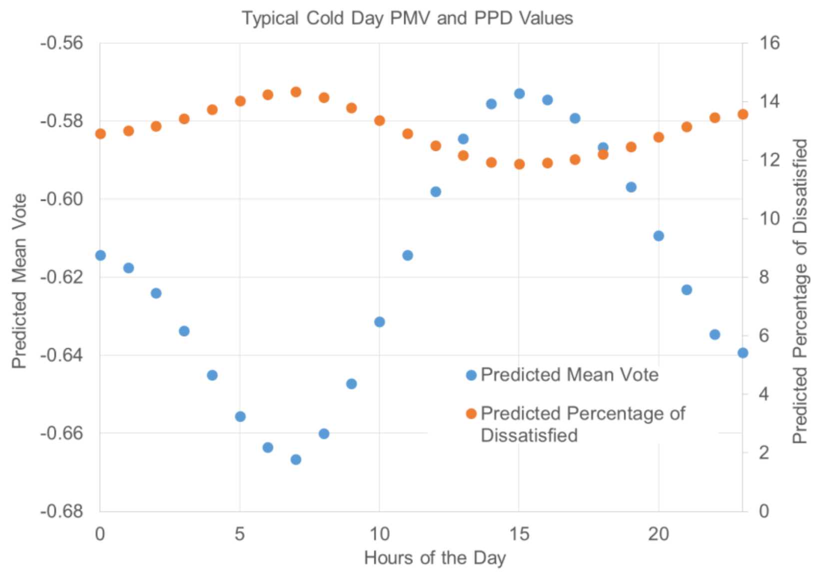

Figure 10: Predicted PMV and PPD for a typical cold day

3. VALIDATION OF POD CALCULATIONS

To validate the prediction of temperature field using POD, field measurements are set up in a 45-person office setting. The office is equipped with multiple induced-air units. The number of units installed is determined based on manufacturer's recommendation that one unit covers 15 m^2 area which is also aligned with the CFD and POD calculations. The units are installed at a height of 2.7 m and are uniformly distributed in the room area. Two poles, 2.6 m high each, are used to measure the vertical temperature distribution of indoor air. Figure 11 shows a picture of these poles. The thermocouples are inserted inside the black sphere. The real poles used in the experiment may or may not have black sphere depending on whether mean radiation temperature is measured or not (Ling, 2011). The temperature distribution is measured by the four thermocouples tied on each pole. Those thermocouples are located at the heights of 2.6 m, 1.5 m, 0.8 m and 0.2 m from the floor, respectively. The numbers represent the air temperature at the supply outlet, occupants' head position, body position and foot position, respectively. One pole is in the middle of the room and the other is located near windows (solar radiation source). The measurement temperature at 12:00 P.M. is compared with POD simulation data. This is because the assumption of 800 W/m^2 heat load matches better with the solar radiation at noon. Table 2 shows the comparison of field measurements and simulation data. The maximum discrepancy is 1.6 K and the average deviation is 0.46K.

Table 2: Temperature profile comparison between POD predictions and measured values

TC locations	**Measurements**	**Simulation**
Pole in the middle of the room		
TC @ 0.2 m	21.0°C	22.6°C
TC @ 0.8 m	22.5°C	23.0°C
TC @ 1.5 m	24.8°C	25.3°C
TC @ 2.6 m	30.0°C	29.2°C
Pole near the window		
TC @ 0.2 m	22.0°C	22.5°C
TC @ 0.8 m	22.2°C	23.0°C
TC @ 1.5 m	24.0°C	24.7°C
TC @ 2.6 m	31.0°C	30.9°C

Figure 11: Sample of thermocouples on the pole (when mean radiation temperature measurment is not required, black spheres may be removed and the inside thermocouples may be exposed)

From Table 2, the largest discrepancies between measurement and calculations are observed for the lower thermocouples. The deviations mainly come from two sources: field equipment operation and grid resolution of POD. The induced-air unit is supposed to provide supply air velocity at around 0.7 to 0.9 m/s, however, in the real operation, the unit only provides air velocity around 0.5 m/s. Although the issue was been fully resolved on-site later, the updated data was not available during manuscript preparation. The second reason is that, due to the nature of being a reduced-order order, POD method has much less resolution compared to a full CFD simulation. One cell has a rough height of 0.17 m (the room has 2.7m in height and divided into 16 cells in that direction) and therefore causes deviation when compared with thermocouple readings at 0.2m and 0.8 m, respectively. Nevertheless, the POD method is able to deliver reasonable accuracy at much less computation cost (~ 2 minutes in POD vs.~ 8 hours full CFD simulation).

4. CONCLUSIONS

A novel induction based supply air unit is introduced in the paper. The thermal comfort evaluation of the unit is the main focus. An office equipped with induced-air unit is modeled using a commercial 3D CFD software package. To speed up the calculation time, proper orthogonal decomposition method is used. The method requires a one-time limited number of CFD snapshot at the beginning. With the help of these snapshots, the POD method is capable of interpolating air temperature profiles of the space with other supply air velocities. The interpolation reduces the computation cost to only 2 minutes. The accuracy of the POD method is evaluated by comparing its results against field measurement. The comparison shows an average deviation of only 0.46 K. A software tool is developed using POD method and used to predict thermal comfort indices such as PMV and PPD for an office setting. On both coldest and hottest day in Beijing, the induced-air unit successfully maintains an average PMV value of -0.2 and -0.6, respectively. The power saving potentials for the entire heat pump system will be evaluated in the future.

NOMENCLATURE

a	POD coefficient	(-)	**Subscripts**	
A	area	m^2	a	air
f_{cl}	Clothing factor	(-)	c	convective

h	Heat transfer coefficient	W/K	cl	clothing
I	Solar incidence	W	d	diffusive
M	metabolic rate	W	D	direct
Pa	Water vapor partial pressure	Pa	g	glass
t	air temperature	°C	i	inside
v	Velocity or temperature	m/s or °C	o	operative/outside
W	work output	W	r	radiative
α	absorptivity	(-)		
φ	POD basis	(-)		

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