International Journal of Mechanical and Industrial Engineering

Volume 1 | Issue 1 Article 13

July 2011

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Recommended Citation
Bartari, V. N. and Rajput, S. P. S. (2011) "Effect of supply air temperature on conservation of energy in air conditioning of space," *International Journal of Mechanical and Industrial Engineering*: Vol. 1 : Iss. 1 , Article 13.
DOI: 10.47893/IJMIE.2011.1012
Available at: https://www.interscience.in/ijmie/vol1/iss1/13

Effect of supply air temperature on conservation of energy in air conditioning of space

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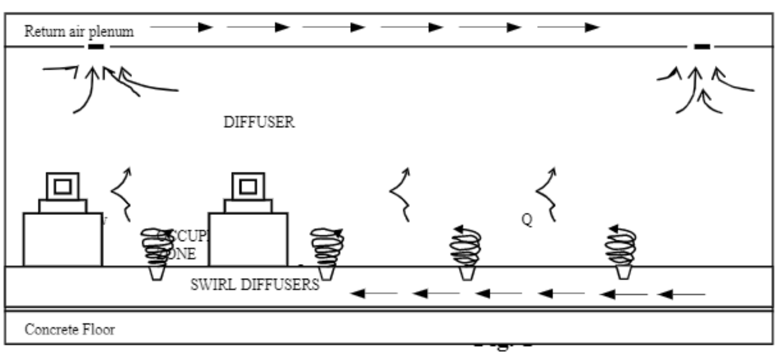
Abstract : In HVAC applications, huge amount of energy is utilized in fans and blowers to maintain the flow. In this paper energy savings associated with air distribution is discussed. In a most commonly used air distribution system, uniform thermal environment in the occupied space is established. An alternative to this method is the under floor air distribution system (UFAD) which is in its fantasy state. Thermal stratification can be established in this method due to the buoyancy flow of the air. In this paper assessment of the impact of temperature sensors in energy savings is done in UFAD system. It is observed that by the placement of temperature sensors in the occupied space, supply air temperature can be controlled while maintaining the comfort conditions. By optimal conditions of the temperature and volume flow, energy savings can be achieved due to reduction in energy requirements in refrigeration and ventilation. The comfort criteria of ASHRAE standard 55-92 is taken.

Key Words

UFAD, Temperature Stratification, Ventilation

1 Introduction

Due to the several advantages of UFAD system, there is a strong need for an improved fundamental understanding of several key performance features of UFAD system. Most of the potential performance advantages of UFAD systems over conventional air distribution systems are related to the fact that conditioned air is delivered at or near floor level and is returned at or near ceiling level[1]. Under cooling conditions, this upward movement of air in the room takes advantage of the natural buoyancy of heat gain to the space, producing a vertical temperature gradient. The purpose of this paper is to report on the assessment of impact of temperature sensor in the occupied zone in energy savings. The room air stratification is affected by the design and operating parameters. One of the crucial operating parameter is temperature of the supply air. The control and optimization of this parameter will result in energy efficient operation of the system. Fig (1) shows the concept of the UFAD system



1.1 Temperature Stratification

The gradients in the occupied zone of the space (between the floor and 1.2 m for a seated or 1.8 m for a standing occupant) increases as the room airflow is reduced resulting in a relatively large change in temperature. Fig (2) shows the temperature stratification in the occupied zone. The upper limit of 3°C is recommended by ASHRAE (ASHRAE, 1992) at the lowest room airflow rate.

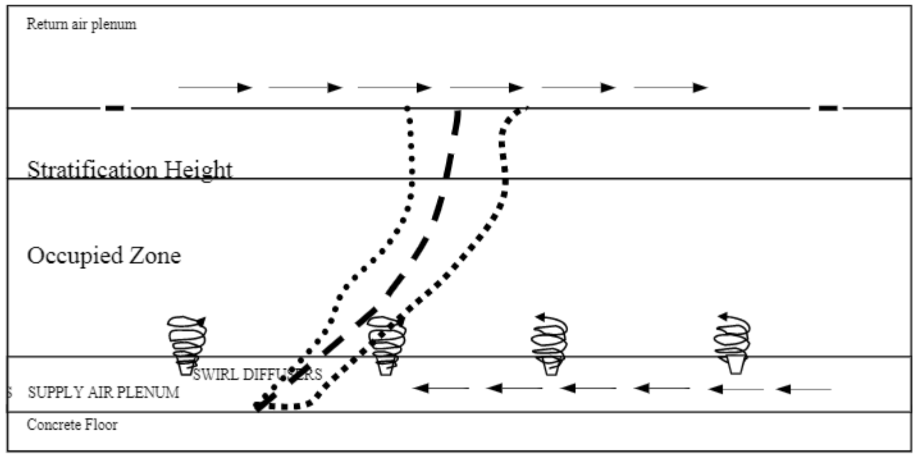


Fig. 2

2 CFD Simulation

Computational fluid dynamics (CFD) simulation is increasingly being used to guide the development of ventilation strategies in system designs and optimization. This is attributable due to the availability of powerful affordable desk-top computers which are able to solve the complex nonlinear equations in CFD models within an acceptable time frame[2].

The aim of this work has been to simulate the air flow in a room to capture the temperature contours and temperature profiles for different values of crucial operating parameter [7]. The CFD model used in the simulation is a commonly used and validated model by comparison with analytical predictions of these flows (e.g. Linden et al. (1990), Hunt and Linden (2000, 2001, 2005)) and experimental data taken from their laboratory measurements. The CFD offers a convenient means for predicting, for example, the detailed spatial variation of flow parameters such as air speed and temperature which may be crucial for comfort analysis and detailed design in more complex spaces.

The simulation results will be of use to study how the temperature variations are taking place with the change in supply air temperature. If temperature sensors are suitably placed in the occupied space and the AHU, suitable thermal stratification in the room can be maintained while also maintaining the comfort. Energy savings can thus be achieved due to the reduced power usage in fans and blower due to stratified environment in the room.

2.1 Geometrical modeling

For CFD simulation a 2D modeling was made [3]. The geometry used in the CFD simulations comprised two openings at floor level. The flow was driven by the heat load of the room. The room size was 4m(length) X 3m(height). Two openings at ceiling are provided for the exit of room air.

2.2 Grid generation

The commercial mesh generating software GAMBIT is used for generating meshes. For the purpose of mesh independent analysis, three independent mesh densities (a) 70500, (b) 162000 and (c) 200000 are used. Negligible differences are found in the results. Final mesh with 162000 cells of structured elements was used for generation of simulation plots Fig (3)

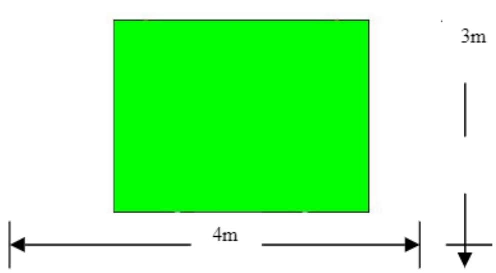


Fig. 3 Room model with mesh

2.3 Governing equations

The code solves the following conservation equations for mass, momentum and energy for a steady, incompressible turbulent flow:

$$\frac{\partial u_j}{\partial x_j} = 0$$

$$\frac{\partial}{\partial x_j}(\rho u_j u_i) = \frac{\partial}{\partial x_j}\left(-p_0\delta_{ij} + (\mu + \mu_t)\left(\frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i}\right)\right) + \rho g_i$$

$$\frac{\partial}{\partial x_j}(\rho u_j He) - \frac{\partial}{\partial x_j}\left(\left(\frac{\lambda}{C_P} + \frac{\mu_t}{\sigma_{He}}\right)\left(\frac{\partial He}{\partial x_j}\right)\right) = 0$$

The Boussinesq approximation is applied in which the density in the momentum equation is written as

$$\rho = \rho_0[1 - \beta(T - T_0)].$$

The turbulent viscosity, μ_t is determined using a two equation k-ε model [4][5]. Based on the literature available the Standard k-ε model was employed. The transport equations for turbulent kinetic energy k and turbulent energy dissipation ε in their standard form for this model were used.

The governing equation were discretised using hybrid differencing, except the mass equation where central differencing was used, and solved on a collocated grid. Pressure and velocity are coupled using the SIMPLE technique. The following under relaxation factors were used: Momentum=0.7, Turbulent kinetic energy=0.8, Turbulent dissipation rate=0.8, turbulent viscosity=1 and energy=1.

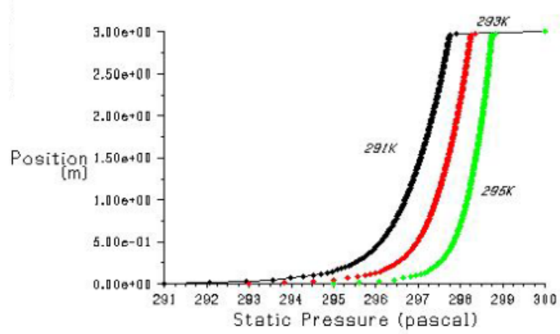
2.4 Boundary Conditions

In CFD model, the openings for the supply air inlet was modelled as mass flow inlet at 0.015Kg/s. The boundary condition at exit was modelled as exhaust fan. All the walls are modelled as adiabatic walls at temperature of 27ºC. Three points in the model at x=1m, x=1.5m and x=2m were set to capture vertical temperature profiles at these locations.

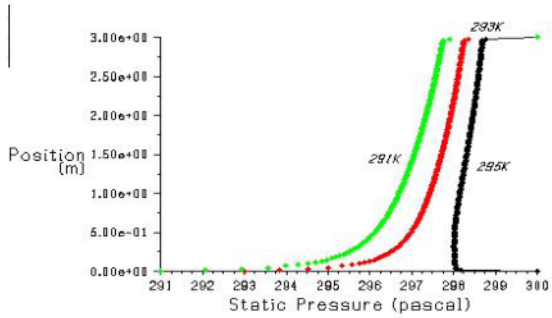
3 Results and Discussion

Convergence was taken to be when the residual fell blow the set criteria for the velocity, energy, momentum and kinetic energy and dissipation[6].

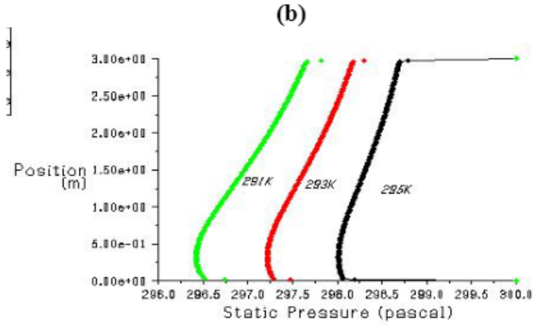
A typical temperature patterns are shown in Fig (4) (a)-(c) in which the temperature variations at three locations at x=1m , x=1.5m and x=2m from the left corner are compared for different supply air temperatures of the air.



(a)



(b)



(c)

Fig. 4 Temperature Plots at different SAT conditions (a) x=1m, (b) x=1.5m and (c) x=2m

CFD predictions of the temperature variations with height show that the thermal stratification is established in the room. ASHRAE conditions of thermal comfort are also fulfilled for the supply temperatures of 20°C(293K) in an acceptable form as the temperature difference of about 3°C is achieved within the human occupied space of about 1.8m.

It is clear from the predictions of vertical temperatures for different supply air conditions of 18°C(291K), 20°C(293K) and 22°C(295K) that the thermal stratification with comfort can be maintained at the SAT of 20°C(293K) while the other conditions of the room are fixed. By acquiring the data through temperature sensors placed in the occupied space, the supply air temperature of the air can be suitably controlled in order to provide thermal stratification while maintaining the comfort.

As the stratification in the space with comfort conditions can be established while acquiring temperatures of the occupied space through temperature sensors and suitably controlling the SAT. The savings in energy can thus be achieved as due to stratified environment within the occupied space the air can be supplied at comparatively higher temperature and it would be possible to return the air at higher temperature. As less quantity of air will be handled by AHU at comparatively higher temperatures, savings in energy in fans and blowers and in refrigeration will then become possible.

Nomenclature

ρ - density

V - volume flow rate

Cp - specific heat

ε - turbulent dissipation

μ - viscosity

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