Energy and Buildings 55 (2012) 738–746

Contents lists available at SciVerse ScienceDirect

Energy and Buildings

journal homepage: www.elsevier.com/locate/enbuild

Experimental and numerical study of influence of air ceiling diffusers on room air flow characteristics

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ARTICLE INFO

Article history:
Received 15 September 2012
Accepted 21 September 2012

Keywords:
Ceiling diffusers
Thermal comfort
Turbulence model
Validation
Energy consumption

ABSTRACT

This paper investigates experimentally and numerically airflow characteristics of vortex, round, and square ceiling diffuser and its effect on the thermal comfort in a ventilated room. Three different thermal comfort criteria namely; mean age of the air, ventilation effectiveness, and effective draft temperature have been used to predict the thermal comfort zone inside the room. Experimentally, a sub-scale room is set-up to measure the temperature field in the room. Numerically, unstructured grids have been used to discretize the numerical domain. Conservation equations are solved using FLUENT commercial flow solver. The code is validated by comparing the numerical results obtained from three different turbulence models with the available experimental data. The validation shows that the standard k–ε turbulence model can be used to simulate these cases successfully. After validation of the code, effect of supply air velocity on the flow and thermal field is investigated and hence the thermal comfort and energy consumption. The results show that the saved energy by vortex diffuser is 1.5 times lower than that achieved by square or round diffuser. The velocity decay coefficient is nearly same for square and round diffusers and is 2.6 times greater than that for the vortex diffuser.

1. Introduction

Nowadays, the majority of people spend up to 90% of their time indoors. Knowledge and prediction of indoor climate conditions are important for optimizing indoor climate and thermal comfort, and it is also important for energy conservation [1–3]. Indoor air quality and thermal comfort are two important aspects of indoor environmental quality that receive considerable attention.

Design conditions of HVAC as specified by ASHRAE [4] are temperature, and relative humidity should be held in the range of (20–24 °C), and 50–60%, respectively. Alongside, positive air pressure should be maintained, and all air exhausted with no recirculation is preferred [5].

The mean age of air (MAA) is one of the most important parameters describing the ventilation efficiency in a space. It is defined as the average time for air to travel from a supply outlet area to any location in a ventilated room [6–8]. Its concept is assumed to be equal zero (100% fresh) at inlet. It is obvious that the high values of MAA mean that part of the air circulates for a long time inside the room. So, the values of the MAA reflect the efficiency of ventilation system.

Spitler [9] studied the effect of the inlet velocity on air distribution in a full scale unoccupied ventilated room has dimensions of 5 m long, 3 m wide, and 3 m high. The supply and exhaust outlets are each 0.333 m wide and 1.0 m high. Nielsen [10] described experiments with wall-mounted air terminal devices. This study gave expressions for the velocity distribution close to the floor. The velocity at the floor was influenced by the flow rate to the room, the temperature difference and the type of the diffuser.

Al-Hamed [2] employed a computer program for simulating 3D room ventilation problems to predict the mean air temperatures and air velocities for a number of realistic supply air inlet and outlet locations. The k–ε turbulence model was considered. The flow field predicted by this model was validated by experimental measurements done by other investigator. His study showed that the PMV occupancy comfort response was more favorable.

Srebric and Chen [11] used a simplified method to describe flow and thermal information from eight various diffusers which were nozzle diffuser, slot (linear) diffuser, valve diffuser, displacement diffuser, round ceiling diffuser, square ceiling diffuser, vortex diffuser, and grille diffuser. The box method was suitable for most of the diffusers with an appropriate box size. The momentum method was applied well for five diffusers. Since the momentum method was simpler than the box method, the momentum method should be used, whenever it is applicable.

Zhou and Haghighat [12] developed a simplified method to define the boundary conditions at the inlet of the swirl diffuser.

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http://dx.doi.org/10.1016/j.enbuild.2012.09.027

Nomenclature

A	inlet area [m^2]
C	concentration [m^3 of vapor/m^3 of moisture air]
C_d	discharge coefficient
C_p	pressure coefficient
d	neck diameter of diffuser [m]
E	ventilation effectiveness
g	gravity acceleration [m/s^2]
k	turbulent kinetic energy [m^2/s^2]
K	velocity decay coefficient
P	pressure [Pa]
Re	Reynolds number
RH	relative humidity [%]
Sc_{eff}	effective Schmidt number
t, T	temperature [°C, K]
t_o	temperature of a reference point [°C]
V	velocity [m/s]
x	distance [m]
x, y, z	co-ordinate system
y	vertical location [m]

Greek symbols

β	thermal expansion coefficient [1/°C]
Γ	diffusion coefficient
ε	turbulent dissipation rate [m^2/s^3]
θ	swirling angle [°]
λ	thermal conductivity [W/m °C]
μ	dynamic viscosity [Pa s]
ρ	air density [kg/m^3]
ϕ	diameter [m]

Abbreviations

3D	three dimensions
CFD	computational fluid dynamics
EDT	effective draft temperature
HVAC	heating, ventilation and air conditioning
LDA	laser Doppler anemometry
MAA	mean age of the air
PIV	particle image velocimetry
PMV	predicted mean vote
PTV	particle tracking velocimetry
RNG	re-normalization group
T.C	thermocouple

Subscripts

d	digital manometer
e	exit
m	mean
o	inlet conditions
s	supply

With this method a round diffuser was divided into six triangular sectors with equal air discharge rate, while various air throw orientation angles were assigned to each sector. Comparisons between smoke airflow visualization and CFD predictions demonstrated the effectiveness of the current simplified modeling method of swirl diffuser. Profiles of temperature and air velocity from measurements were also presented to validate the CFD simulation results. This validation showed that the simplified method can be used to model the diffusers.

Einberga and Hagstrom [13] discussed the modeling results from CFD simulation of a multi-cone air diffuser for industrial spaces. CFD simulations were compared systematically with data from experimental measurements while air velocity was measured by ultrasonic sensors. The outcomes of this study showed that CFD simulation with a standard k–ε model accurately predicted non-isothermal airflow around the diffuser.

The majority of previous studies in indoor airflows were conducted in reduced scale rooms and the numerical simulations were compared to the reduced scale experiments. There was a lack of experimental data available to validate mathematical models. The previous studies were not agreed on the suitable turbulence model that more valid. Also, there was not a clear vision about the diffusers energy consumption. As a result, some fundamental issues remain unsolved related to the predictability of existing mathematical models for low Reynolds number flows.

The important goal of this study is to improve the analysis of real life environments. Thus, particular emphasis is placed on the complexity of the diffuser geometry where the three-dimensional (3D) flow occurs. Evaluation of the performance of three ceiling diffusers is performed and the more suitable turbulence model for these cases is specified. Also, the region of the thermal comfort is predicted. The diffusers energy consumption is determined. This investigation of the air diffusers characteristics and its impact on the indoor airflows is conducted through two approaches: experimental measurements and numerical simulations.

2. Experimental work

2.1. Experimental set-up

The general layout of the experimental set-up is shown in Fig. 1. It is consists of two main parts namely: air conditioning unit and test rig (model room).

The air conditioning unit is a simple vapor compression refrigeration cycle which used to deliver cold air to the test room. A fan pulls atmospheric air and passing it over evaporator. Cubic glassed model room is fabricated from Perspex has 1 m long. The round supply duct has 15 cm diameter that runs from the air conditioning to the inlet of the test room.

The supply duct is fitted with discharge control gate in the form of the vortex diffuser which vanes can turn for changing the air volume flow rate supplied to the room. The airflow enters the room from an opening at its ceiling of 15 cm diameter in which tested diffuser is installed.

The temperatures of floor, ceiling of the room, and air are measured by means of copper-constantan thermocouples (T-type). A sliding arm having fixed thermocouple at its terminal end is installed at the corner of the room, it can move along three dimensions to measure the temperature inside the room (Fig. 1). Thermocouple is fixed just before the outlet diffuser to measure the inlet temperature. Alongside, other thermocouples are mounted on the four walls, floor, and the roof. All thermocouples are connected to switches followed by digital thermometer. The thermocouples readings are measured from digital thermometer which has uncertainty of ±0.2 °C of received readings according to its manual data.

Digital manometer is used to measure the air velocity (V_d) before the diffuser and hence, the air flow rate can be measured. The error of the measured value is about ±0.1 m/s. The pressures drop a cross orifice meter is measured by using inclined water manometer having error about ±0.1 mm H_2O. The calibrations of the instruments should be carried out in this type of tests. T-type thermocouples are calibrated by a direct comparison method, where the readings of the thermocouples are compared with those of standard mercury-in-glass thermometers. The calibration process has been performed in temperature variation range of 14–97 °C for heating mode, and 95–17 °C for cooling mode. Also, other instruments are calibrated by using this methodology.