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Performance analysis and flow features of swirl diffuser in a floor based air conditioning system

Mihir Kumar
mihir04219@gmail.com
Lakshmi Narain College Of Technology, Bhopal, Madhya Pradesh

Deenoo Pawar
dinupawar2011@gmail.com
Lakshmi Narain College Of Technology, Bhopal, Madhya Pradesh

Shailendra Dwivedi
shailendrakdwivedi@gmail.com
Lakshmi Narain College Of Technology, Bhopal, Madhya Pradesh

ABSTRACT

In this work, the variation in the temperature of the conditioned air and the improvement in thermal human comfort by adopting different models of floor swirl diffuser are investigated. The experiment is performed in the workshop lab of Mechanical Engineering Department, LNCT, Bhopal. The aim of this study is to find out the airflow pattern and its distribution through different swirl diffusers installed at the floor in the air-conditioned room. The experiment is based on predicting the nature and behavior of air diffused by three different types of swirl diffusers having a slot with a draft angle of 10°, 11° and 12° under different operating and flow conditions. An experiment is performed to evaluate the thermal comfort produced in a room equipped with a heat load of 1000 W load capacity with different swirl diffusers and compare their performance graphically. The graphs are plotted between temperatures of diffused air inside the room and vertical height from floor level.

Keywords— *Air swirl, Diffuser, Effectiveness, Swirling blades angle, Airflow*

1. INTRODUCTION

Air diffusers are used widely in air-conditioning systems and the air diffusion is very much influenced by the characteristics of different diffuser designs. For air supply systems in automobiles, swirling diffusers are the most popular. The method of Modeling the diffuser is critical as it has an important impact on the accuracy of the predicted airflow pattern in the car. Swirl diffusers are modular devices that are designed to mount into an access front panel space and "plug" into the air handling space. Diffusers are installed within an access front panel and can be relocated at any point on the base plate. This device delivers conditioned air to space and allows the occupant to manually control both the volume and direction of the air. The diffuser is constructed of a durable, high impact, polycarbonate material available in black or gray finish. Delivering air from the front offers added benefits, beyond rearrangement flexibility. Cool, clean air is delivered directly into the occupied zone of the space, so heat and pollutants are not continually circulated within the space as they are with an overhead system. The result is a space that has stratification of heat and pollutants, with concentrations in the lower levels of the space less than those at the upper levels of the space. Ventilation is accomplished through displacement as opposed to dilution. Commercial office spaces are progressively more reliant to new designs of Heating Ventilation and Air Conditioning (HVAC) equipment to achieve better indoor conditions measured against current industry standards. One of the most telling indoor air quality parameters when it comes to air distribution system and fresh air delivery is Air Change Effectiveness (ACE). ACE is commonly used as a measure for effective delivery of outside air by a ventilation system to the occupied space in a building.

2. REVIEW OF PAST WORK

K. Ashok Reddy (2018) performed CFD analysis on the models by varying velocities of fluid 0.25m/s, 0.5m/s & 0.75m/s to determine heat transfer coefficient, heat transfer rate. Thermal analysis is performing on all the models by considering materials of diffuser Copper alloy, Aluminum alloy 6061 and Nickel alloy to determine heat fluxes. The analysis is done in Ans 14.5. Rachamarla Pradeep Kumar et al (2017) investigated the variation in temperature of conditioned air and improvement in thermal human comfort by adopting different models of floor swirl diffuser. Different models of floor swirl diffuser having different slot angles of 60, 80 and 100are modeled in Creo 2.0 software. CFD analysis is performing on the models by varying velocities of fluid 0.25m/s, 0.5m/s & 0.75m/s to determine heat transfer coefficient, heat transfer rate. Thermal analysis is performing on all the models by considering materials of diffuser Copper alloy, Aluminum alloy 6061 and Nickel alloy to determine heat fluxes. E.T.V Dauricio (2017) focuses on the effect of swirl on important parameters of conical diffusers flows such as static pressure evolution, recirculation zones, and wall shear stress. Governing equations are solved using software based on the finite volume method. Moreover, turbulence effects are taken into account employing the k-ε RNG model with an enhanced wall treatment. The Reynolds number has been kept constant at 105, and various diffuser geometries were simulated, maintaining a high area ratio of 7 and varying the total divergence angle (16°, 24°, 40°, and 60°). Results showed that the swirl velocity component develops into a Rankine-vortex type or a forced-vortex type. In the former, the swirl is not effective to prevent boundary layer separation, and a tailpipe is recommended to allow a large-scale mixing to enhance the pressure recovery process. The analysis is done in Ansys 14.5. M Jaszczur (2016) presented the results of experimental studies of the flow of air through diffusers. Presented laboratory model is a simplification of the real system and was made in a geometric scale 1:10. Simplifying refer both to the geometry of the object and conditions of air flow. The aim of the study is to determine the actual velocity fields of air flowing out of the swirl diffuser. The results obtained for the diffuser various settings are presented. We have tested various flow rates of air. Stereo Particle Image Velocimetry (SPIV) method was used to measure all velocity vector components. The experimental results allow determining the actual penetration depth of the supply air into the room. This will allow for better definition of the conditions of ventilation in buildings.

3. ASSUMPTIONS USED IN EXPERIMENT

In the early design stage, without experimental data support, some assumptions have to be taken based on the experience and similar studies done in this field.

Following assumptions are applied in the investigation of experimental and numerical analysis.

1. Metabolic rate and preference of closing the office vary from the people to people. The standard values for metabolic rate and closing factor are to be considered. The activity of a sedentary occupant is estimated to be 1.2 met and the clothing insulation is 1.0 clo in winter and 0.5 clo in summer.
2. Ventilation effectiveness for DCV to be 1.0, as it is for the mixing ventilation.
3. The experimental setup is situated in a clean area with good air quality.
4. The acrylic sheet wooden room is closed while performing the experiment and heat transfer from room door due to leakage is neglected.
5. We have assumed that the position of the room wrt sun direction and altitude is identical for heat load calculation. So no effect has been considered.

4. EXPERIMENTAL SETUP

It consists of an acrylic sheet wooden room of size 4 ft x 4 ft x 5 ft. with different models of swirl diffuser installed at the ceiling level. The conditioned air from the air conditioner is supplied from the bottom through a duct of reducing cross-section to increase the air flow velocity through the diffuser. A heater of 1000 W is placed inside the room to provide a heat load. The heater is placed near the location Y2. A temperature sensing instrument with six thermocouple wire is placed inside the room to measure the temperature at six locations vertically at a distance of 0.6 feet. There are four exhaust vents at the top surface of the wooden block through which ventilation is carried out inside the room.

There are six locations at the floor inside the room where readings of temperature have to be noted and the variation in temperature of the air is to be studied. The various components and parts of the experimental setup are shown in the figures.

The test facility is available at the workshop of LNCT college. The test chamber has the linear dimension 4 x 4 x 5 ft. The swirl air diffuser is located at the centre of floor. At the floor o hole of diameter 280 mm and depth 280 mm is made to place the diffuser. The diffusers used are 280 mm in diameter and 280 mm height with curved slots cut at the upper surface. These curved slots are drafted through an angle of 10°, 11º and 12º. Rectangular slots of size 20 mm x 200 mm are cut on the cylindrical vertical surface of diffuser to capture the air coming out through the air- conditioner. The walls of test chamber are insulated. Four exhaust vents of size 5 x 5 cm are cut on the ceiling for ventilation. The exhaust air is exhausted directly to the surrounding. Inlet air comes from the air conditioner through a reducing cross-sectional area duct and captured in the diffuser at the floor level.

Fig. 1: 10° swirl diffuser

Fig. 2: 11° swirl diffuser

Fig. 3: 12° swirl diffuser

Fig. 4: Side view of experimental set up

5. AIR TEMPERATURE MEASUREMENT

Thermocouple type J is used to measure the air temperature at inlet, exhaust and along a measuring stand. The temperature is calculated internally in the instrument by using temperature indicator. 6 thermocouples at different heights (0.6 m, 0.12 m, 1.8 m, 2.4 m, 3.0 m, 3.6 m), are mounted on a stand to measure the air temperatures. Each thermocouple on the stand is attached by the insulated tapes the thermocouples produce an electric voltage that is converted to temperature. The reading accuracy for the thermocouples is expected to be 0.3% which can be neglected because the overall accuracy is dominated by the data processed in temperature indicator. A common reference point in the temperature indicator is used and this results in an accuracy of ± 1 K. By averaging the measured results, it is expected the accuracy can reach up to ± 0.5 K.

6. EXPERIMENTAL RESULTS

The results of experiments are presented in two parts: requirement of the air conditioner size for the particular room condition by the cooling load calculation and thermal comfort environment according to the temperature variation inside the room with different diffusers. Due to different slot angle geometry of swirl diffusers, flow pattern of diffused air inside the room changes. The air is calculated inside the room through different diffusers. As a result the temperature of air inside the room varies which we have recorded at six different locations. The variation in temperature with respects to height is tabulated for each swirl diffuser with and without heat load under two different conditions. The two conditions are when only one exhaust vent is opened and when all the exhaust vents are opened. The study helps in comparing the performance of different swirl diffusers under different operating conditions. From the graph plotted it is observed that minimum variation in temperature is obtained with 11° swirl diffuser and the uniformity in temperature is obtained when all the exhaust ports are opened.

6.1. Experimental readings: 1

6.2. Condition 1: When exhaust all vent is opened

Initial room temperature = 32°C

Room temperature with load 1000 W without A.C = 38°C

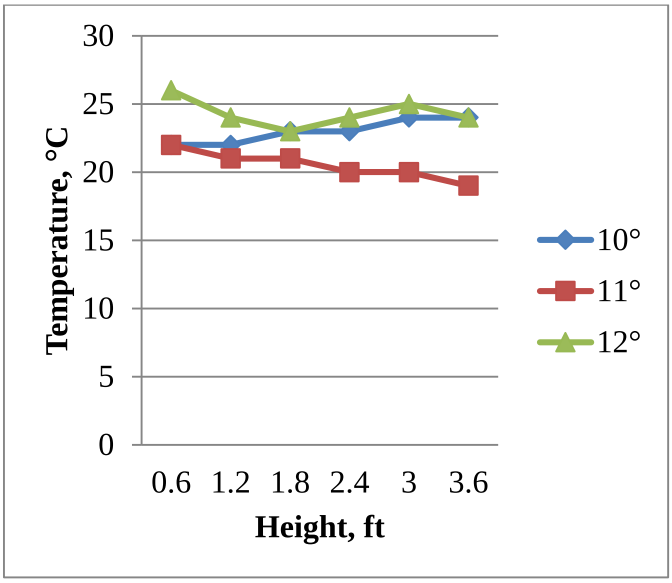

Fig. 5: Variation in temperature vs. height at location Y1 in condition 1 without load

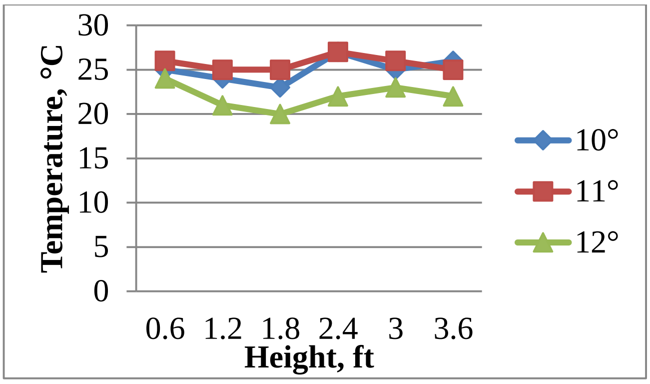

Fig. 6: Variation in temperature vs. height at location Y1 in condition 1 with load 1000 W

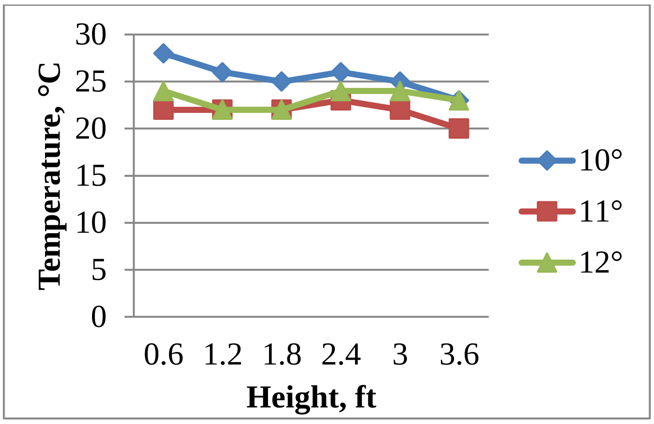

Fig. 7: Variation in temperature vs. height at location Y2 in condition 1 without load

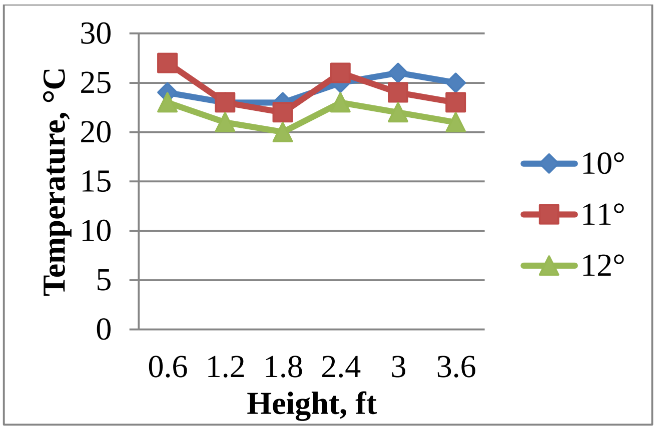

Fig. 8: Variation in temperature vs. height at location Y2 in condition 1 with load 1000 W

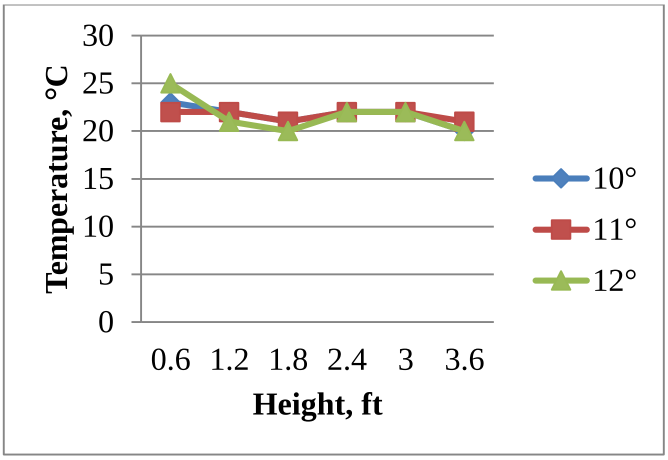

Fig. 9: Variation in temperature vs. height at location Y3 in condition 1 without load

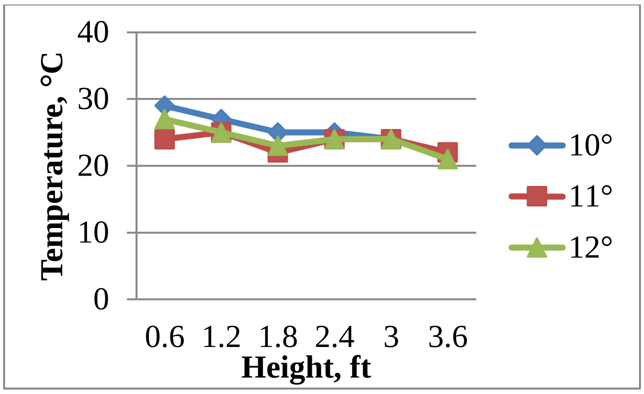

Fig. 10: Variation in temperature vs. height at location Y3 in condition 1 with load 1000 W

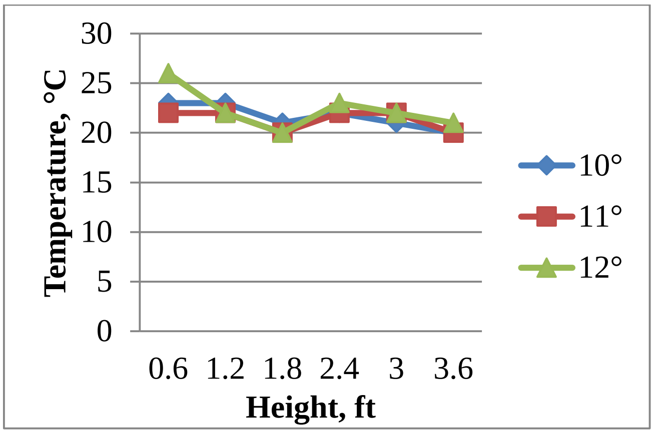

Fig. 11: Variation in temperature vs. height at location X1 in condition 1 without load

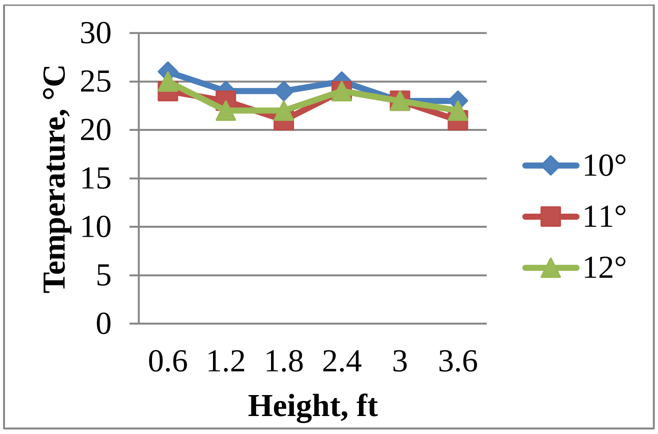

Fig. 12: Variation in temperature vs. height at location X1 in condition 1 with load 1000 W

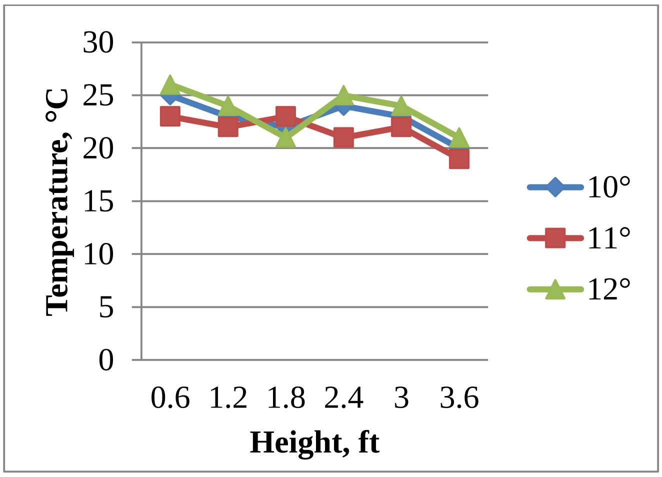

Fig. 13: Variation in temperature vs. height at location X2 in condition 1 without load

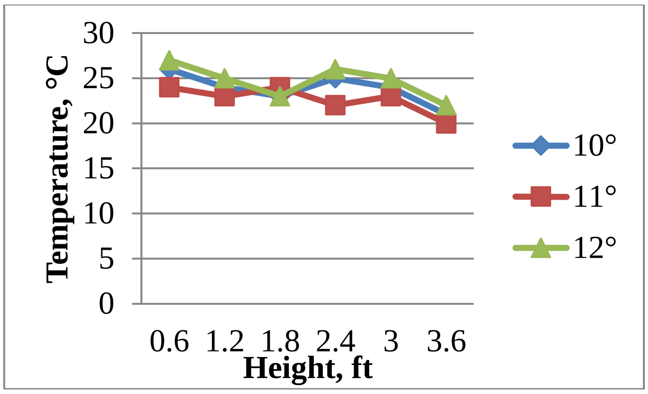

Fig. 14: Variation in temperature vs. height at location X2 in condition 1 with load 1000 W

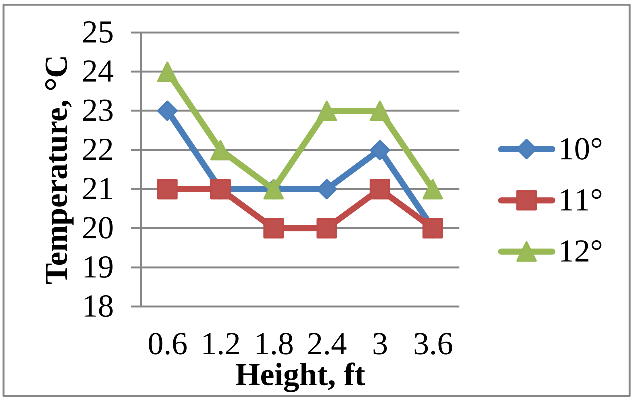

Fig. 15: Variation in temperature vs. height at location X3 in condition 1 without load

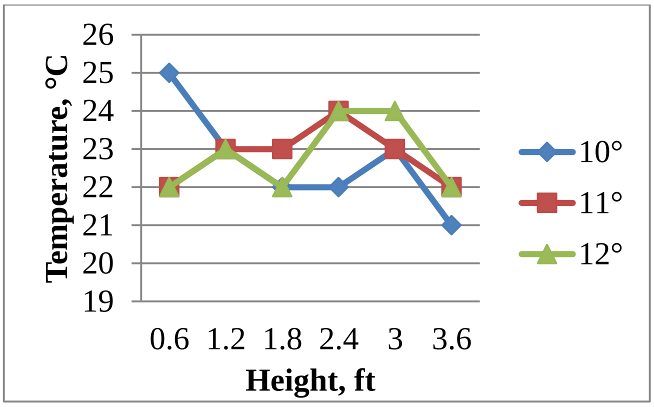

Fig. 16: Variation in temperature vs. height at location X3 in condition 1 with load 1000 W

6.3 Experimental readings: 2

6.3.1 Condition 2: When one exhaust port is opened

Initial room temperature = 32°C
Room temperature with load 1000 W without A.C = 38°C

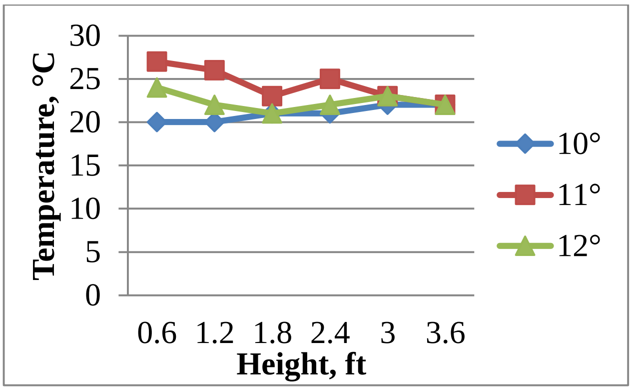

Fig. 17: Variation in temperature vs. height at location Y1 in condition 2 without load

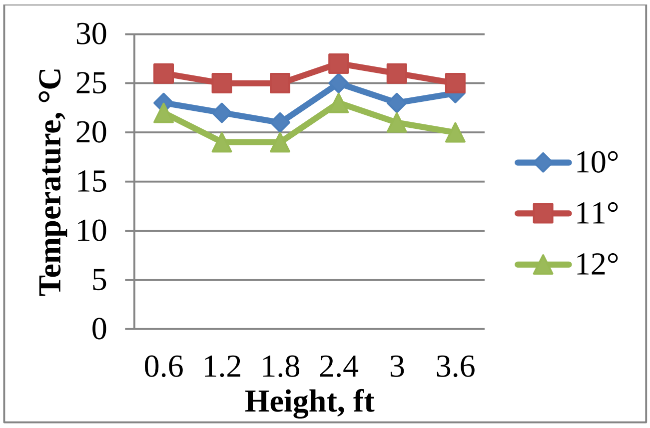

Fig. 18: Variation in temperature vs. height at location Y1 in condition 2 with 1000 W load

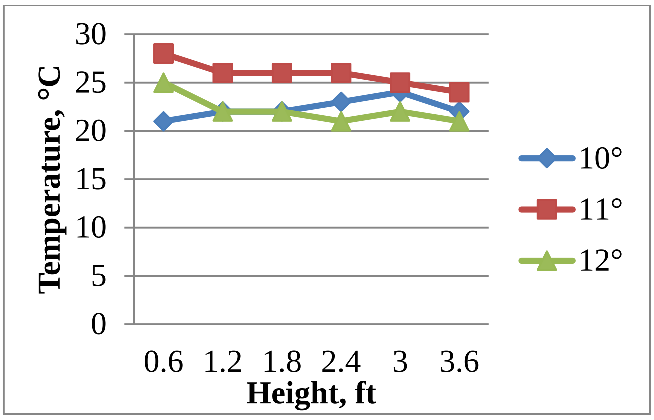

Fig. 19: Variation in temperature vs. height at location Y2 in condition 2 without load

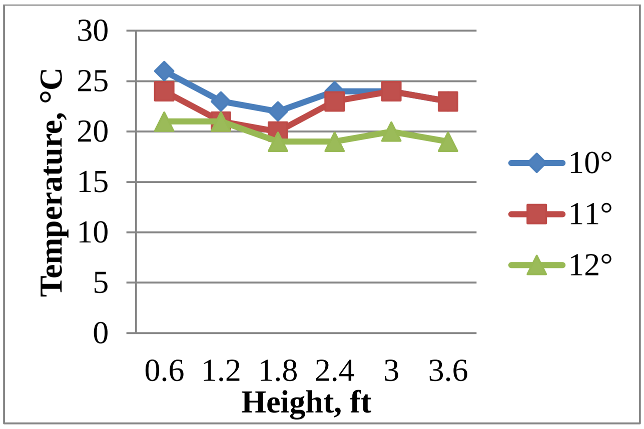

Fig. 20: Variation in temperature vs. height at location Y2 in condition 2 with 1000 W load

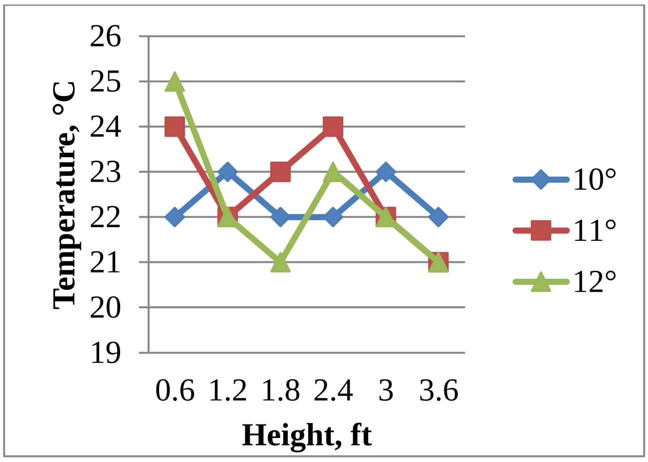

Fig. 21: Variation in temperature vs. height at location Y3 in condition 2 without load

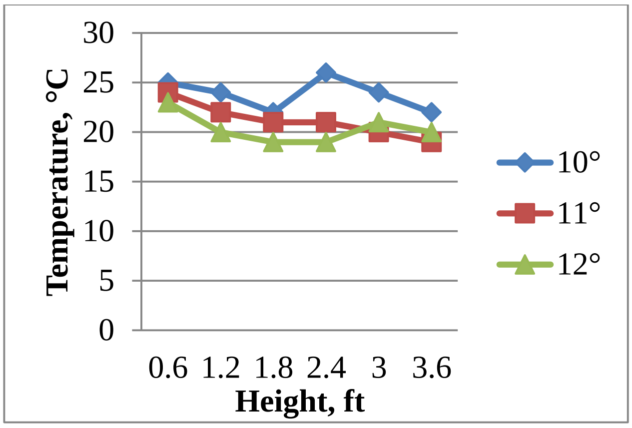

Fig. 22: Variation in temperature vs. height at location Y3 in condition 2 with 1000 W load

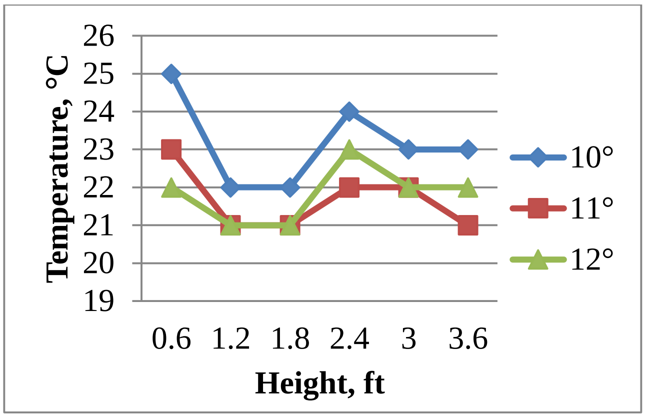

Fig. 23: Variation in temperature vs. height at location X1 in condition 2 without load

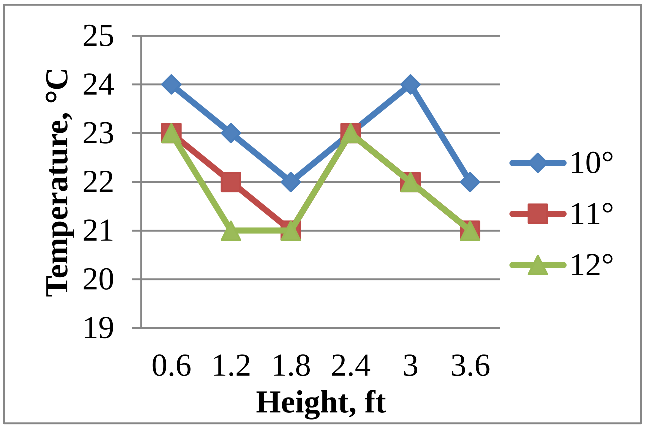

Fig. 24: Variation in temperature vs. height at location X1 in condition 2 with 1000 W load

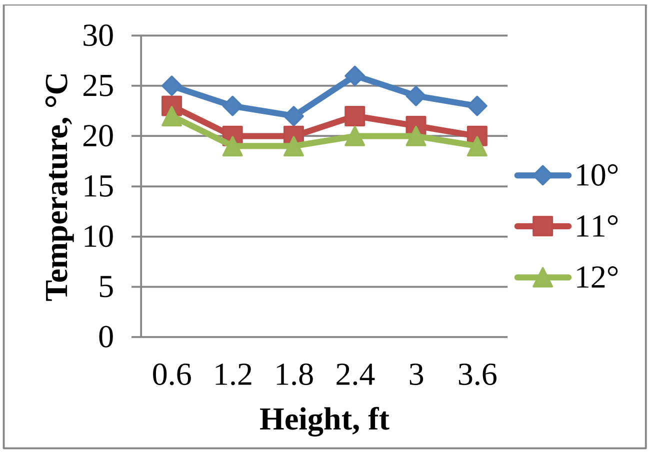

Fig. 25: Variation in temperature vs. height at location X2 in condition 2 without load

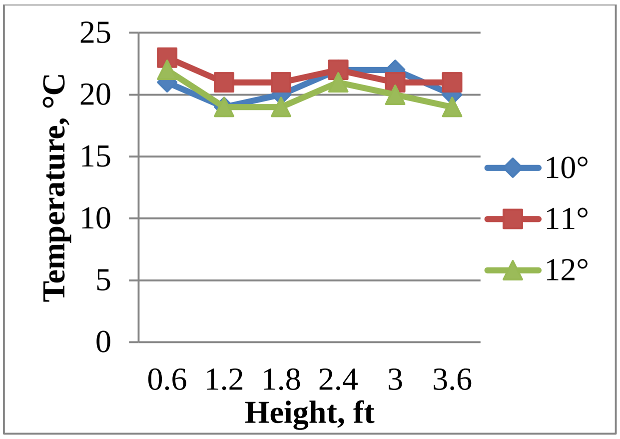

Fig. 26: Variation in temperature vs. height at location X2 in condition 2 with 1000 W load

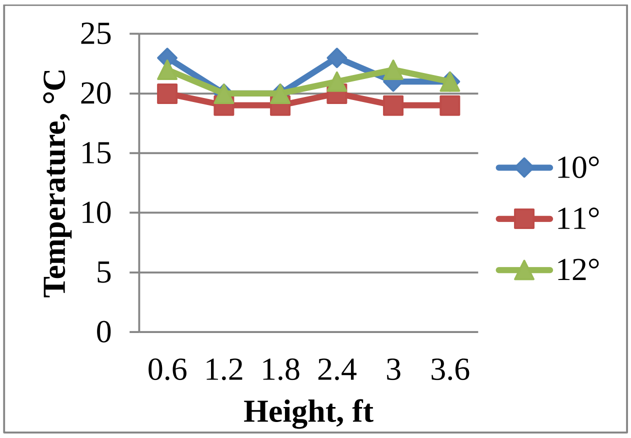

Fig. 27: Variation in temperature vs. height at location X3 in condition 2 without load

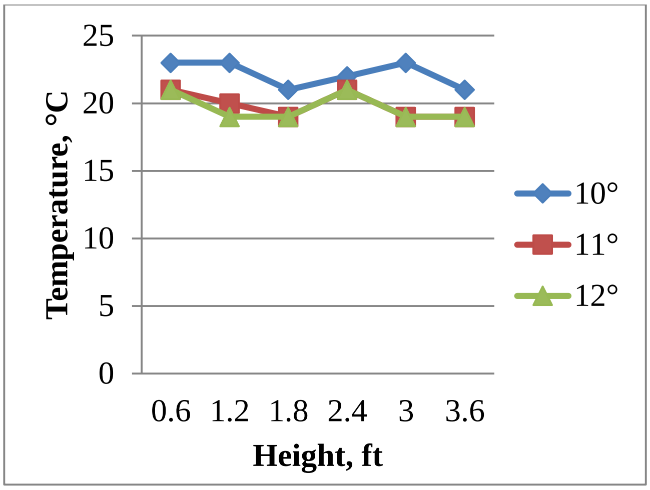

Fig. 28: Variation in temperature vs. height at location X3 in condition 2 with 1000 W load

7. CONCLUSIONS

The results from this dissertation work show that floor supply displacement ventilation using swirl diffuser can improve indoor air quality because the containment concentration in the breathing zone is lower than that of the mixing system. It helps us in comparing the performance of three different types of swirl diffuser under different operating and flow conditions.

Due to swirl action produced more unidirectional flow was created, the slow recirculation at the occupant zone was eliminated for ceiling supply ventilation and the risk of cross-contamination can be effectively reduced. The system with the swirl diffusers can provide a better comfort level than that with the perforated panels due to the mixing by the diffusers.

The maximum variation in temperature is obtained at location Y2. This happens due to the presence of heater of load capacity 1000 W near location Y2. As we move away from the heat source variation in temperature reduces and we obtained an almost uniform temperature at the upper region of the experimental setup. We have compared the performance of different swirl diffuser models having a slot with draft angle 10°, 11° and 12° under different operating conditions and the best performance is obtained with 11° swirl diffuser. We have also compared the performance of diffuser when only one exhaust vent is opened and when all the four exhaust vents are opened.

This study helps in selecting optimum models for floor swirl diffuser under different operating conditions. We can improve the air change effectiveness and human comfort by varying the slot design angle of the diffuser. It will result in better mixing of air inside the room and the variation in temperature of air from floor height will be reduced. We can achieve better human comfort and proper ventilation by using floor swirl diffuser.

8. REFERENCES

[1] K. Ashok Reddy, Thermal Analysis of Swirl Diffuser at Eight Degree Slot Angle Using Ansys with Different Materials, International Journal of Mechanical Engineering and Technology, 9(6), PP: 1093–1097.
[2] Rachamarla Pradeep Kumar, P. Saritha, J. Pradeep Kumar (2017), Design and Performance of Under Floor Swirl Diffuser at Different Operating Angles Using FEA, International Journal & Magazine of Engineering, Technology, Management and Research, 4(11), 102-107.
[3] E.T.V Dauricio (2017), Numerical Analysis of Swirl Effects on Conical Diffuser Flows, Journal of Aerospace Technology and Management, 6(2), 112-119.
[4] M Jaszczur (2016), Experimental analysis of the velocity field of the air flowing through the swirl diffusers, 7th European Thermal-Sciences Conference, 1-9.
[5] Dinesh Singh, Pankaj Mishra (2015), Swirl Diffuser Design and Performance Characteristics for Air Flow in Air Conditioning of an Automobile-A Review, International Journal of Innovative Research & Development, 4(8), 67-75.
[6] Rahul Pandey, Anil Kumar Rao, Vinay Yadav (2015), Comparison Between Simulation and Experimental Technique Inside an Air-Conditioned Room to Analyse the Performance and Flow Characteristics of Ceiling Swirl Diffuser under Different Operating and Flow Parameters, International Journal of Engineering Sciences & Research Technology, 4(11), 424-431.
[7] Omprakash Ahirwar, Neeraj Kumar Ahirwar, Ishwar Singh (2014), Experimental and Simulation-based comparative analysis of UFAD and OHAD in Air-conditioning of room, International Journal of Scientific & Engineering Research, 5(9), 640-649.
[8] Suraj, V.N. Barataria, Performance and Flow Characteristics of Floor Swirl Diffuser under Different Operating and Flow Parameters, International Journal of Mechanical Engineering and Technology, 4(4), 154-165.
[9] Ramazan (2013), CFD Simulation of Swirling Effect in S-Shaped Diffusing Duct by Swirl Angle of 10˚, IOSR Journal of Mechanical and Civil Engineering, 6(2), 11-19.
[10] P. K. Sinha, A.K. Das and B. Majumdar, Numerical Investigation of Flow through Annular Curved Diffuser, International Journal of Mechanical Engineering and Technology, 2(2), 1-13.

