Effects of the Geometry of Air Outlets on Predicted Human Comfort inside Rooms: CFD vs. ADPI

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Abstract— The paper is devoted to numerically investigate the influence of the Air Supply Outlets Geometry on Human Comfort inside Living Rooms. A computational fluid dynamics model is developed to examine the air flow characteristics of a room with different supply air diffusers. The work focuses on air flow patterns, thermal behavior in the room where few number of occupants. As an input to the full-scale 3-D room model, a 2-D air supply diffuser model that supplies direction and magnitude of air flow into the room is developed. Air distribution effect on thermal comfort parameters was investigated depending on changing the air supply diffusers type, angels and velocity. Air supply diffusers locations and number were also investigated.

The pre-processor Gambit is used to create the geometric model with parametric features. Commercially available simulation software "Fluent 6.3" is incorporated to solve the differential equations governing the conservation of mass, three momentum and energy in the processing of air flow distribution. Turbulence effects of the flow are represented by the well-developed two equation turbulence model. In this work, the so-called standard k-ε turbulence model, one of the most widespread turbulence models for industrial applications, was utilized. Basic parameters included in this work are air dry bulb temperature, air velocity, relative humidity and turbulence parameters are used for numerical predictions of indoor air distribution and thermal comfort.

The thermal comfort predictions through this work were based on ADPI (Air Diffusion Performance Index),the PMV (Predicted Mean Vote) model and the PPD (Percentage People Dissatisfied) model, the PMV and PPD were estimated using Fanger's model.

Keywords—ADPI, living Room ,PMV, PPM, Thermal Comfort

I-INTRODUCTION

THE present work focuses on air flow patterns, thermal behaviours, and thermal comfort in air-conditioned living room . Air conditioning term can be defined as a process that controls the microclimate of an enclosed space. This process involves the movement of air through a space that has certain characteristics of temperature, humidity, cleanliness, pressure differential and noise level attenuation in order to satisfying a comfortable and healthy environment for the occupants.

A. Thermal comfort is a condition of mind which expresses satisfaction with the surrounding environment, most important factors influencing thermal comfort are.

Environmental factors:

- Air temperature, air speed, relative humidity, air quality, and Noise.

Other factors:

- Activity level, clothing level, and psychological factors: such as mental effort.

Achieving thermal comfort for most occupants of buildings or other enclosures is a main goal of HVAC design engineers.

B. **In-door Air Quality**

Indoor Air Quality (IAQ) deals with the content of interior air that could affect health and comfort of building occupants. The IAQ may be compromised by microbial contaminants (mold, bacteria), chemicals (such as carbon dioxide, radon), allergens, or any mass or energy stressor that can induce health effects. So, using ventilation to dilute contaminants and improve the indoor air quality in most buildings. Carbon is an indoor pollutants emitted by humans and correlates with human metabolic activity. Carbon dioxide concentration at levels that are unusually high indoors may cause occupants to grow drowsy, get headaches, or function at lower activity levels, etc... Table I is a listing of carbon dioxide air concentrations and related health effects and standards.

TABLE I
CARBON DIOXIDE AIR CONCENTRATION LEVEL STANDARDS

Carbon Dioxide Level	Health Effects	Standards or Use of Concentration	Reference
600 ppm	None	Most indoor air complaints eliminated, used as reference for air exchange for protection of children.	NIOSH [1]
800 ppm	None	Used as an indicator of ventilation inadequacy in schools and public buildings, used as reference for air exchange for protection of children.	MDPH [2]
1000 ppm	None	Used as an indicator of ventilation inadequacy concerning removal of odors from the interior of building.	ASHRAE [3]
5000 ppm	No acute (short term) or chronic (long-term) health effects	Permissible Exposure Limit (8-hour workday) / Threshold Limit Value.	ACGIH [4], OSHA [5]

C. Ventilation Principles

Ventilation is the exchange of air, typically between an indoor space and the outside. When people are present, ventilation is especially necessary to evacuate the carbon dioxide produced and renew the oxygen used up. It is also needed to remove other pollutants (smoke, chemicals, etc.) from the space. Ventilation air may be classified into natural or mechanical ventilation. In natural ventilation or gravity ventilation, uses the natural forces caused by the temperature difference inside the space to induce air circulation and removal.

D. Air Exchange Rate

The most common method to measure the ventilation rate is the air exchange rate; the air exchange rate has units of 1/time. When the time unit is hours, the air exchange rate is also called air changes per hour (ACH). The rate of ACH determines the rate at which the total volume of air in the room is cleaned by an air purification system, which is a major factor in the degree of air cleaning that can be achieved. Where it is the total volume of air flowing into a space in 1 hour divided by the volume of the space, then ACH can be expressed mathematically as,

$$ACH = 3600\,Q/V \qquad (1)$$

Where: Q = volumetric air flow rate through the room, m^3/s,
V = volume of the room, m^3

The air exchange rate may be defined for several different situations. For example, the air exchange rate for an entire space served by an air handling unit compares the amount of outside air brought into the space to the total interior volume, this the nominal air exchange rate.

E. Air Conditioning Systems

Air conditioning systems can be categorized according to the means by which the controllable cooling/heating is accomplished in the conditioned space. There are four basic systems categories:-

1- All-Air Systems; air is used to carry the energy from indoor to outdoor and vice versa.

2- All-Water Systems; water is used to carry the energy from indoor to outdoor and vice versa,

3- Air-Water Systems; air and water are used to carry the energy from indoor to outdoor and vice versa.

4-In Direct Expansion (DX) Systems; refrigerant is used to carry the energy from indoor to outdoor or vice versa [i.e. direct expansion of refrigerant, without the chilled water cooling medium].

II-Thermal Comfort

Thermal comfort is defined in the International Standard ISO 7730 [6] as ANSI/ASHRAE 55environment’’. A definition most people would agree on but also a definition, which is not easily converted into physical parameters because thermal comfort is a subjective matter. It arises from the body’s physiological state and is mainly a sensation from the nerves in the skin.

Conditions that provide thermal comfort: There are six primary factors that must be addressed when defining conditions for thermal comfort. A number of other, secondary factors affect comfort in some circumstances. The six primary factors are listed below.

1) Metabolic rate.
2) Clothing insulation.
3) Air temperature.
4) Radiant temperature
5) Air speed.
6) Humidity

Computer model method for general indoor application The method may be applied to spaces where the occupants have activity levels that result in average metabolic rates between 1.0 met and 2.0 met and where clothing is worn that provides not more than 1.5 clo of thermal insulation.

The ASHRAE thermal sensation scale, which was developed for use in quantifying people's thermal sensation, is defined as follows.

+3 hot
+2 warm
+1 slightly warm
0 neutral
-1 slightly cool
-2 cool
-3 cold

The Predicted Mean Vote (PMV) model uses heat balance principles to relate the six key factors for thermal comfort to the average response of people on the above scale (the PMV). The PPD (predicted percentage of dissatisfied) index is related to the PMV .It is based on the assumption that people voting +2, +3, -2, or -3 on the thermal sensation scale are dissatisfied, and the simplification that PPD is symmetric around a neutral PMV.”

The Air Diffusion Performance Index (ADPI)

Air Diffusion Performance Index Statistically related the space conditions of local or transverse temperature and velocities to occupants’ thermal comfort

ADPI >= 80 is considered acceptable.

Effective draft temperature ΘΘ= (tx-tc) –0.07(Vx-30).

% of points where -3<= ΘΘ<= +2 =ADPI.

Velocity below 70 fpm

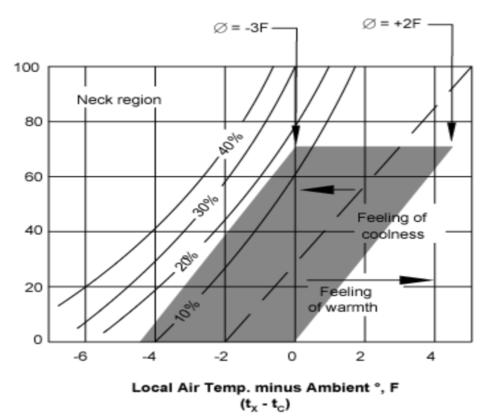

Figure1: Local air temperature minus ambient temperature, [8]

The grey area represents -3<= ΘΘ<= +2
You can vary temperature or velocity to maintain comfort
15 fpm = 1 °F

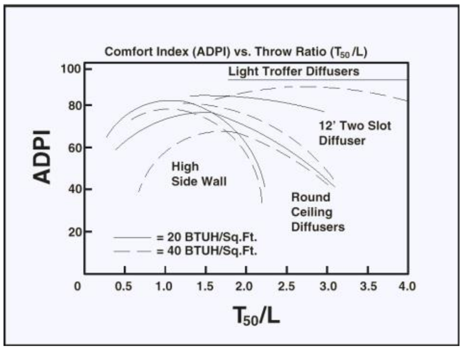

Figure 2 A: Comfort index (ADPI) vs. throw ratio (T50/L),[8]

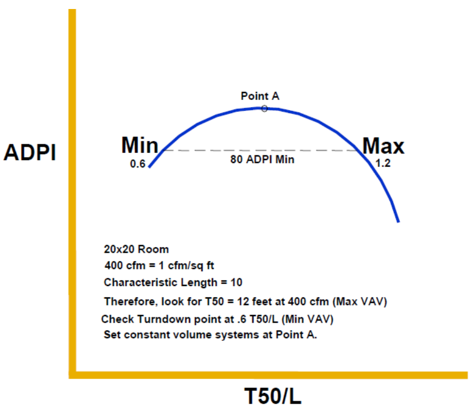

Figure 2 B: Comfort index (ADPI) vs. throw ratio (T50/L),[8]

TABLE 2

Recommended ADPI Ranges for Outlets. [8]

Outlet	T_{50}/L	Range	Calculated	T_{50} & L	Data		
Sidewall Grilles	1.3-2.0	L T_{50}	10 13-20	15 20-30	20 26-40	25 33-50	30 39-60
Ceiling Diffusers Round Pattern	0.6-1.2	L T_{50}	5 3-6	10 6-12	15 9-18	20 12-24	25 15-30
Ceiling Diffusers Cross Pattern	1.0-2.0	L T_{50}	5 5-10	10 10-20	15 15-30	20 20-40	25 25-50
Slot Diffusers	0.5-3.3	L T_{50}	5 3-18	10 5-33	15 9-50	20 10-66	25 15-83
Light Troffer Diffusers	1.0-5.0	L T_{50}	4 4-20	6 6-30	8 8-40	10 10-50	12 12-60
Sill & Floor Grilles All Types	0.7-1.7	L T_{50}	5 4-9	10 7-17	15 11-26	20 14-34	25 18-13

The ratio (Tv/L) can be used to predict the ADPI and expected occupant comfort for a number of different types of air supply outlets for a range of cooling loads in the space. Also, the obtainable ADPI for various ceiling outlet type diffuser with certain outlets capable of obtaining a higher predict resulting ADPI others.

In the current ASHRAE Handbook Chapters on room air distribution, ADPI, using the ratio of Tv/L, is the recommended method to predict the percentage of points within a space that meet the effective draft temperature criteria for comfort. The source research to predict the ADPI for a space using the ratio of Tv/L was conducted in the late 1960.

The available types of outlets have increased over time and are not reflected in the current tables included in the ASHRAE Handbook. Also, the manufacturing methods for many of the outlets have changed since the original testing that occurred in the 60's and may change the performance characteristics of these devices. The current ADPI table presented in the ASHRAE Handbook Fundamentals Chapter 20 (table 3) and in the ASHRAE Applications Chapter 56 (table 4) are outdated and need to be revised. The results of the research presented in this project would validate the current Tv/L relationships and add new products to the current table.

A high percentage of people are comfortable in office occupations where the effective draft temperature is between -3 and +2°F, and the air velocity is less than 70 fpm, when calculated with the current ADPI draft temperature equation. This equation assumes that 30 fpm is an ideal 'neutral' airspeed. With the loads present at the time of the development of the ADPI calculations, average room airspeeds were in the 30 fpm range, in cooling. With today's significantly reduced room loads, room average airspeeds are much lower. As a result, the ADPI calculation tends to be overly sensitive to slight temperature variations in cooling, and is not useable in heating mode as an evaluation tool.

If measurements of air velocity and temperature are taken throughout the occupied zone of a space, the ADPI is the percentage of those measurements that meet the specifications for acceptable draft temperature and velocity. The most comfortable conditions are identified with high ADPI values (greater than 80%).Miller and Nevins 1969, 1970, 1972, Miller 1971; Miller and Nash 1971; Nevins and Ward 1968;Nevins and Miller 1972. Current ADPI application is for cooling only with overhead mixing air distribution systems.

The ADPI Selection Guide relates the isothermal terminal velocity (Tv) and the characteristic room length (L) of an air outlet over a range of space uniform heat loads. The selection guide lists the value of Tv/L at which the ADPI is a maximum for various loads, and a range of Tv/L for which ADPI is greater than a specified value.

The throw (Tv) of a jet is the distance from the outlet to a point where the maximum velocity in the air stream cross section has been reduced to a selected value. When estimating ADPI, the selected terminal velocity is usually 50 fpm (0.25m/s). Manufacturers' performance data for various diffusers lists the throw distance to a terminal velocity of 50 fm (T50) for isothermal conditions without boundary interferences.
The characteristic room length (L) is the distance from the diffuser to the nearest boundary in the principle horizontal direction of the air flow. This boundary can be a wall surface, or in the case where air collides with the air from a neighboring diffuser, one-half the distance between diffusers plus the vertical distance to the occupied zone.

III-Assessment and Validation*

An experimental investigation on a real air-conditioned lecture room was done. This investigation aims to validate the used computational fluid dynamics code, the results from both investigations, experimental and numerical, will be compared. Flow parameters like velocity and temperature have been measured at relatively important places on a plane perpendicular to a grill in the supply duct. The space configuration and the measuring instruments used are described. In addition, the experimental locations are described in details. Furthermore, the experimental procedure and test precautions are discussed briefly.

A. Description of the lecture room configuration

Room Geometry

The room under investigation is a real lecture room "dissuasion room at building number 17" Faculty of Engineering, Cairo University, which has main dimensions as shown in the following figure 3. Conditioned air is supplied to the room through four air conditioners with outside dimensions as shown in figure 4.

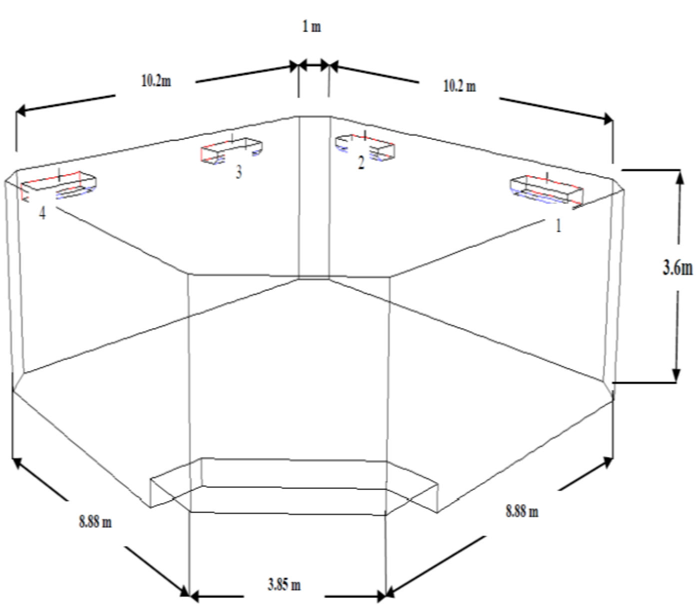

Figure 3: lecture room configuration

***the Assessment and Validation part was published before in AIAA conference 2012 paper number4097with mesh size500,000 ;but in this paper the mesh size increased to about 4,500,000,and this change in mesh size gave us a little change in the results.**

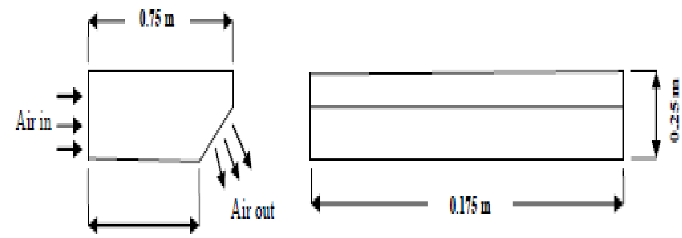

Figure 4: air conditioner configuration

B. Measuring Locations

One place was chosen to perform measurements a plane passing with a supply grill. A vertical plane perpendicular to the supply grill of air conditioner number 3 to show the decay in inlet air velocity and temperature variation downstream. This plane was taken to pass with a supply grill of air conditioner number 3. Measuring points is selected at each 20 cm on this plane. Temperature and velocity are measured at 110 points in this plane. The layout of these measuring points is shown in figure 5.

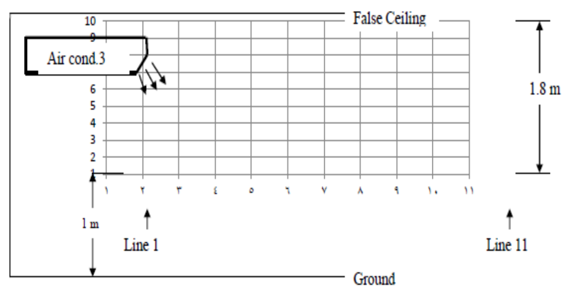

Figure 5 A: Lines of measurements near supply grill.

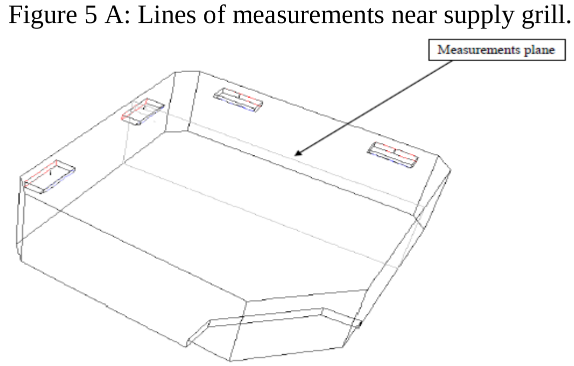

Figure 5B: configuration of measurements plane.

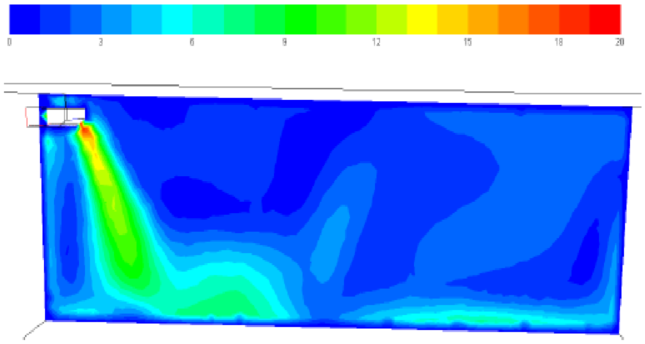

Figure 5 C: predicted velocity contours in the measurements plane

TABLE 3
MEASUREMENTS LINES COORDINATES AT SUPPLY GRILL

No.	X (m)	Y (m)		Z (m)
		From	**To**	
1	0.25	1	2.8	-11.5
2	0.4	1	2.8	-11.4
3	0.55	1	2.8	-11.3
4	0.65	1	2.8	-11.2
5	0.8	1	2.8	-11.1
6	0.95	1	2.8	-10.9
7	1.05	1	2.8	-10.75
8	1.2	1	2.8	-10.6
9	1.4	1	2.8	-10.45
10	1.5	1	2.8	-10.3
11	1.75	1	2.8	-10

C. ASSESSMENT OF CFD MODELING VALIDATION:

The typical validation procedure in CFD, as well as other fields, involves graphical comparisons of computational results and the corresponding available experimental data. If the computational results "generally agree" with the experimental data, the computational results are declared "validated".

D. RESULTS AND DISCUSSION

Temperature and mean velocity values downstream the supply duct is compared below.

***i. Temperature measurements*:-**

Figure 6 shows comparisons between measured and predicted air temperature profiles downstream the supply grill at line 5.

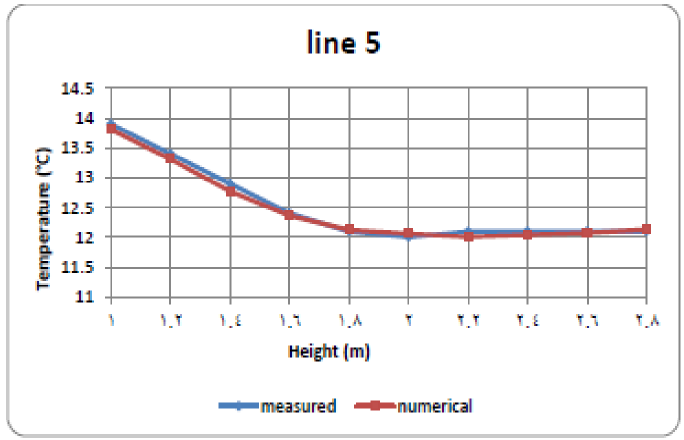

Figure 6: Comparisons between measured and predicted air temperature profiles downstream the supply grill at line 5.

The assumptions were suggested in the numerical model to represent the air supply grille, gave a good agreement with the measured results. The measured values are not equal the numerical ones due to the limited measuring instrument resolution.

ii. Velocity measurements

Figure 7 shows comparisons between measured and predicted air velocity downstream the supply grill at line 5.

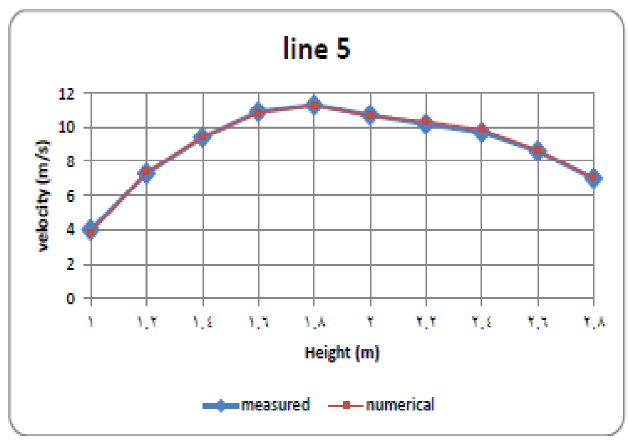

Figure 7: Comparisons between measured and predicted air velocity downstream the supply grill at line 5.

E. SUMMERY

The measured air temperatures and velocities were compared against the predicted results. Fair agreement can be found between the simulated and measured results. For the measuring points, the average velocity and temperature prediction errors were calculated equal to 0.04 m/s and 0.5 °C (1.8%), respectively. These values verify the satisfactory performance of the CFD model, taking into account the accuracy of the measurement. And most of the predicted air temperatures and velocities were overestimated. Generally the calculations yielded the same trends as the measurements. Curves of measured temperature is displaced from the predicted ones, this could be due to errors in specifying the boundary conditions. All comparisons carried out and shown in this validation gave a direct conclusion of the numerical model capability to predict the air flow characteristics within acceptable deviation from the measured values.

IV-RESULTS AND DISCUSSION

A computational fluid dynamics model is developed to examine the air flow characteristics of a room with different supply air diffusers. This Part is devoted to numerically investigate the influence of location and number of air supply and extracts openings on air flow properties in a typical Living room. The work focuses on air flow patterns, thermal behavior in the room where few number of occupants. As an input to the full-scale 3-D room model, a 2-D air supply diffuser model that supplies direction and magnitude of air flow into the room is developed. Air distribution effect on thermal comfort parameters was investigated depending on changing the air supply diffusers type, angels and velocity. Air supply diffusers locations and number were also investigated.

The pre-processor Gambit is used to create the geometric model with parametric features. Commercially available simulation software “Fluent 6.3” is incorporated to solve the differential equations governing the conservation of mass, three momentum and energy in the processing of air flow distribution. Turbulence effects of the flow are represented by the well-developed two equation turbulence model. In this work, the so-called standard k-ε turbulence model, one of the most widespread turbulence models for industrial applications, was utilized. Basic parameters included in this work are air dry bulb temperature, air velocity, relative humidity and turbulence parameters are used for numerical predictions of indoor air distribution and thermal comfort.

The thermal comfort predictions through this work were based on ADPI (Air Diffusion Performance Index)the PMV (Predicted Mean Vote) model and the PPD (Percentage People Dissatisfied) model, the PMV and PPD were estimated using Fanger’s model.

Air Diffusion Performance Index (ADPI)
Extensive studies have resulted in relationships between local temperatures, velocities and comfort reactions. On the basis of the temperature and velocity at a specific point, an effective draft temperature can be calculated for that location. The draft temperature is calculated by the equation:

$\Theta ed = (Tx - Tc) - 8(Vx - 0.15)$ where:

Θ = draft temperature **°C**

Tx = local temperature **°C**

Tc = control temperature **°C**

Vx = local velocity **m/s**

Research indicates that a high percentage of people are comfortable when the effective draft temperature difference is between -3 °F [-2 °C] and +2 °F [+1 °C] and the air velocity is less than 70 fpm [0.36 m/s]. This comfort zone is illustrated as the shaded area in the follwing Figure

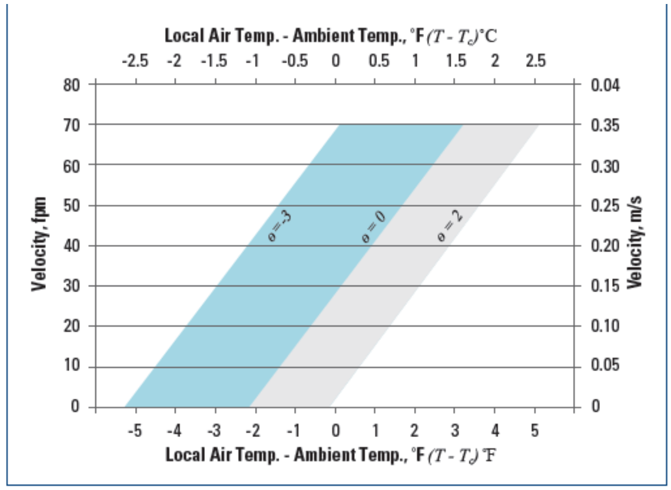

Figure 8:Cofort zone using ADPI

Using this draft temperature as our criteria, the quality of room air diffusion can be determined based on the Air Diffusion Performance Index (ADPI).

ADPI is defined as the percentage of locations in the occupied space which meet the comfort criteria based on velocity and temperature measurements taken at a given number of uniformly distributed points. This ADPI value has proven to be a valid measure of an air diffusion system.

The ADPI rating of an air diffusion system depends on a number of factors :

- Outlet type
- Room dimensions and diffuser layout
- Room load
- Outlet throw

When properly selected, most outlets can achieve an acceptable ADPI rating.

The higher the ADPI rating, the higher the quality of room air diffusion within the space, Generally an ADPI of 80 is considered acceptable.

Through extensive testing, relationships have been developed between ADPI and the ratio of throw over characteristic length (T/L). Throw is the isothermal throw at a selected terminal velocity taken from catalog performance charts. The characteristic length is the distance from the outlet to the nearest boundary as in ASHREA Fundamentals chapter 20 table

We will choose 12points at the comfort zone and using its temperature, and velocity we will calculate the effective drat temperature using CFD simulation, then we will calculate the percentage of this points which inside the range of [-2 °C] and [+1 °C] this percentage will represent the ADPI.

Also using the air outlets manufacturer data and each case configuration we will calculate the ratio T_{50}/L and from it we will find the ADPI for each case.

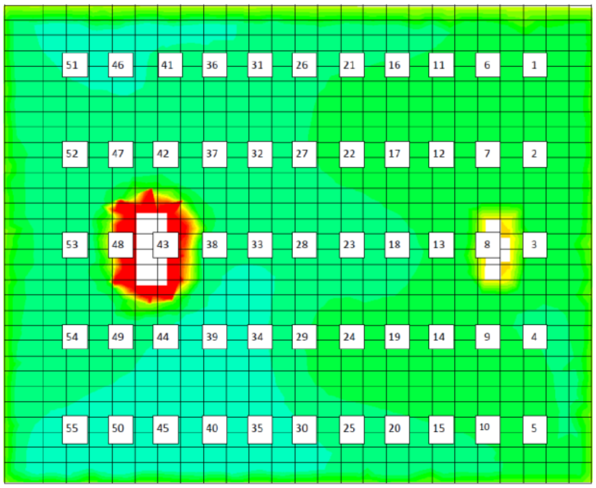

Figure 9: location of the selected points

The present work is concerned with the air flow patterns in a simple room due to changing the supply diffuser shape and angles. The room main dimensions are 4 m width, 5 m length and 3 m height as shown in figure. A person setting on a sofa is modeled, and a television is added in front of him. The air is supplied to the room with different conditions as shown later in the boundary conditions section.

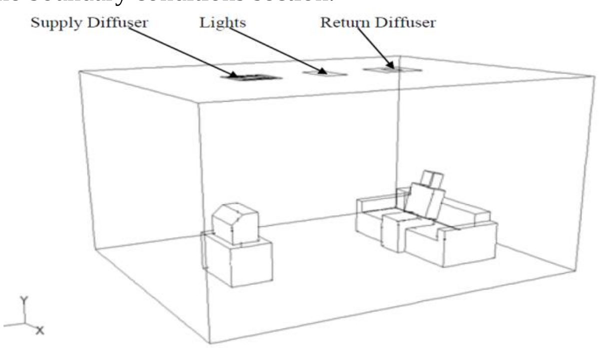

Figure 10:CASE GEOMETRY

Boundary Conditions:

This section will present the various boundary conditions assumed for all the studied cases herein described in the next chapter.

Walls

The room walls were set in the solver to be a constant temperature surface and its temperature was assumed to be 30 ºC. The walls material was simulated to be gypsum which matches the real configuration walls properties.

Interior

The room furniture and non-heat dissipating equipment in the room was set to be a zero wall- heat fluxes.

Lights heat load

Lighting fixtures are mounted at ceiling; the light heat flux is set to be 555 w/m^2 for an area of 0.18 m^2

Body as a heat source

The body is treated as a wall at a constant temperature, and it is set according to the following Figure , where the skin temperature is a function of the metabolic rate in Met (1 Met = 58 W/m^2). As it has been assumed that the occupant's metabolic rate is 116 W/m^2 (2 Met), this is equivalent to 32.5°C skin temperature, and the body is assumed to have zero diffusive flux.

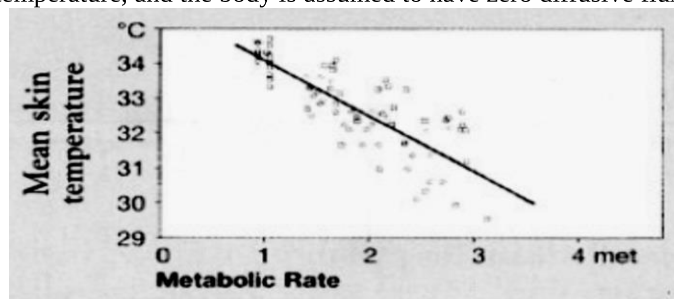

Figure 11:*Mean skin temperature as a function in activity level*

Television heat load

Television is placed on a table in the middle of the room, the heat load is assumed to be 200 W which corresponds to a heat flux of 200 w/m^2 for an area of 1 m^2.

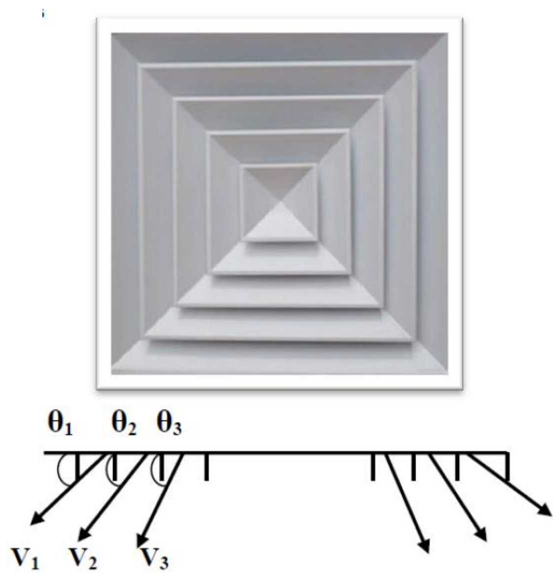

Figure 12:SUPPLY DIFFUSER FOR CASE 1,2,AND 3

TABLE 4
Inlet Air Conditions

Case	Mesh Size	Inlet Air Conditions		Supply descriptions		
		V_S, m/s	T_S, K	θ_1	θ_2	θ_3
1	4283110	0.9	285.8	15	30	60
2	4283110	0.65	285.8	30	60	90
3	4283110	0.65	285.8	30	60	90
4	4400600	0.65	285.8	30	60	90
5	4322200	0.65	285.8	-	-	-
6	4320115	0.65	285.8	-	-	-
7	5122350	0.65	285.8	-	-	-

CASE (1) SQUARE DIFFUSER (60-30-15˚):

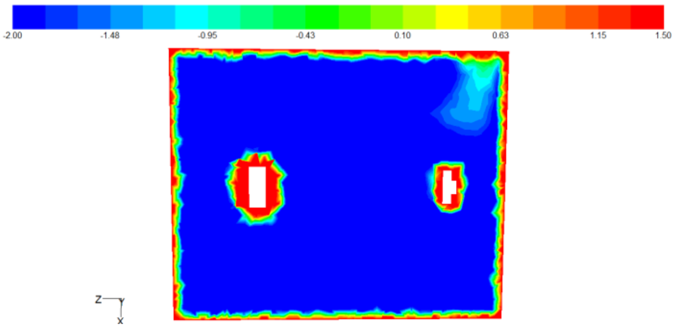

Figure 13: Effective Draft Temperature contours in a horizontal plane at Y=1

TABLE 5

Results From CFD Simulation For Case 1

Location	Edt (Θ) °C	Evaluation	Location	Edt (Θ) °C	Evaluation	Location	Edt (Θ) °C	Evaluation
1	-1.35	Accepted	20	-1.85	Accepted	39	-1.85	Accepted
2	-1.85	Accepted	21	-1.85	Accepted	40	-1.85	Accepted
3	+1.1	Not-Accepted	22	-1.9	Accepted	41	-1.9	Accepted
4	-1.9	Accepted	23	-1.8	Accepted	42	-1.8	Accepted
5	-1.8	Accepted	24	-1.9	Accepted	43	+1.5	Not-Accepted
6	-1.9	Accepted	25	-1.85	Accepted	44	-1.85	Accepted
7	-1.85	Accepted	26	-1.85	Accepted	45	-1.85	Accepted
8	+1.35	Not-Accepted	27	-1.85	Accepted	46	-1.85	Accepted
9	-1.85	Accepted	28	-1.9	Accepted	47	-1.9	Accepted
10	-1.85	Accepted	29	-1.8	Accepted	48	+1.5	Not-Accepted
11	-1.9	Accepted	30	-1.9	Accepted	49	-1.9	Accepted
12	-1.8	Accepted	31	-1.85	Accepted	50	-1.85	Accepted
13	-1.9	Accepted	32	-1.85	Accepted	51	-1.85	Accepted
14	-1.85	Accepted	33	-1.85	Accepted	52	-1.85	Accepted
15	-1.85	Accepted	34	-1.9	Accepted	53	-1.9	Accepted
16	-1.85	Accepted	35	-1.8	Accepted	54	-1.8	Accepted
17	-1.9	Accepted	36	-1.9	Accepted	55	-1.9	Accepted
18	-1.8	Accepted	37	-1.85	Accepted			
19	-1.85	Accepted	38	+1.2	Not-Accepted			

ADPI = (50/55) x100 =90.9%

Case (2) Square Diffuser (60-30-15˚):

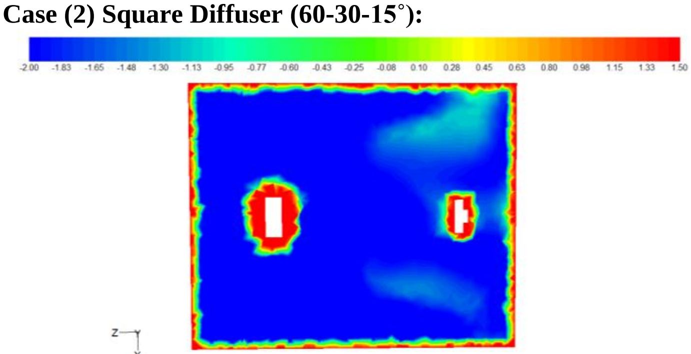

Figure 14:Effective Draft Temperature contours in a horizontal plane at Y=1m

TABLE 6

Results From CFD Simulation For Case 2

Location	Edt (Θ) °C	Evaluation	Location	Edt (Θ) °C	Evaluation	Location	Edt (Θ) °C	Evaluation
1	-1.35	Accepted	20	-1.85	Accepted	39	-1.85	Accepted
2	-1.85	Accepted	21	-1.85	Accepted	40	-1.85	Accepted
3	+1.1	Not-Accepted	22	-1.9	Accepted	41	-1.9	Accepted
4	-1.9	Accepted	23	-1.8	Accepted	42	-1.8	Accepted
5	-1.8	Accepted	24	-1.9	Accepted	43	+1.5	Not-Accepted
6	-1.35	Accepted	25	-1.85	Accepted	44	-1.85	Accepted
7	-1.35	Accepted	26	-1.85	Accepted	45	-1.85	Accepted
8	+1.35	Not-Accepted	27	-1.85	Accepted	46	-1.85	Accepted
9	-1.85	Accepted	28	-1.9	Accepted	47	-1.9	Accepted
10	-1.85	Accepted	29	-1.8	Accepted	48	+1.5	Not-Accepted
11	-1.9	Accepted	30	-1.9	Accepted	49	-1.9	Accepted
12	-1.8	Accepted	31	-1.85	Accepted	50	-1.85	Accepted
13	-1.9	Accepted	32	-1.85	Accepted	51	-1.85	Accepted
14	-1.85	Accepted	33	-1.85	Accepted	52	-1.85	Accepted
15	-1.85	Accepted	34	-1.9	Accepted	53	-1.9	Accepted
16	-1.85	Accepted	35	-1.8	Accepted	54	-1.8	Accepted
17	-1.9	Accepted	36	-1.9	Accepted	55	-1.9	Accepted
18	-1.8	Accepted	37	-1.85	Accepted			
19	-1.85	Accepted	38	+1.2	Not-Accepted			

ADPI = (50/55) x100 =90.9%

Case (3) Square Diffuser (90-60-30˚)

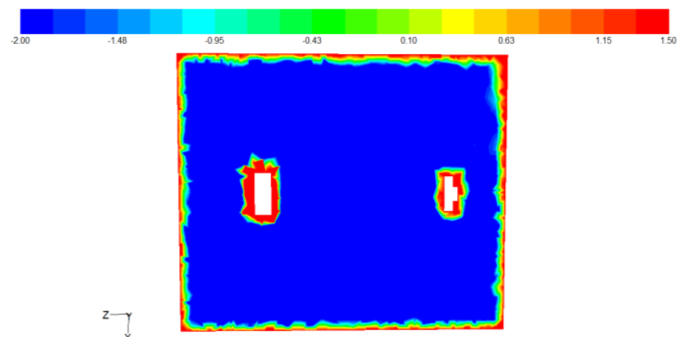

Figure 15:velocity Effective Draft Temperature contours in a horizontal plane at Y=1m

TABLE 7

Results From CFD Simulation For Case 3

Location	Edt (Θ) °C	Evaluation	Location	Edt (Θ) °C	Evaluation	Location	Edt (Θ) °C	Evaluation
1	-1.8	Accepted	20	-1.85	Accepted	39	-1.85	Accepted
2	-1.85	Accepted	21	-1.85	Accepted	40	-1.85	Accepted
3	+1.1	Not-Accepted	22	-1.9	Accepted	41	-1.9	Accepted
4	-1.9	Accepted	23	-1.8	Accepted	42	-1.8	Accepted
5	-1.8	Accepted	24	-1.9	Accepted	43	+1.5	Not-Accepted
6	-1.9	Accepted	25	-1.85	Accepted	44	-1.85	Accepted
7	-1.85	Accepted	26	-1.85	Accepted	45	-1.85	Accepted
8	+1.35	Not-Accepted	27	-2	Accepted	46	-1.85	Accepted
9	-1.85	Accepted	28	-2	Accepted	47	-1.9	Accepted
10	-1.85	Accepted	29	-2	Accepted	48	+1.5	Not-Accepted
11	-2	Accepted	30	-2	Accepted	49	-1.9	Accepted
12	-1.8	Accepted	31	-2	Accepted	50	-1.85	Accepted
13	-1.9	Accepted	32	-1.85	Accepted	51	-1.85	Accepted
14	-2	Accepted	33	-1.85	Accepted	52	-1.85	Accepted
15	-2	Accepted	34	-1.9	Accepted	53	-1.9	Accepted
16	-2	Accepted	35	-1.8	Accepted	54	-1.8	Accepted
17	-2	Accepted	36	-1.9	Accepted	55	-1.9	Accepted
18	-1.8	Accepted	37	-1.85	Accepted			
19	-1.85	Accepted	38	+1.2	Not-Accepted			

ADPI = (50/55) x100 =90.9%

Case (4) Circular Diffusers (90-60-30˚)

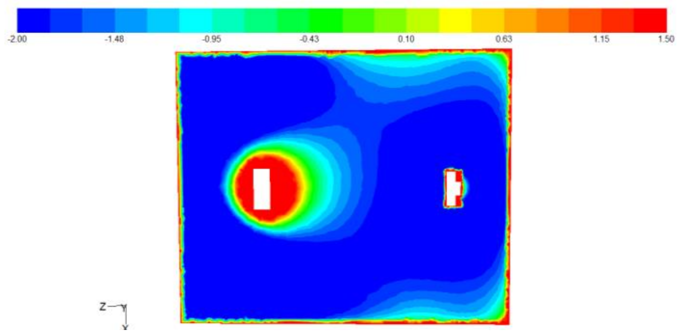

Figure 16:Effective Draft Temperature contours in a horizontal plane at Y=1m

TABLE 8

Results From CFD Simulation For Case 4

Location	Edt (Θ) °C	Evaluation	Location	Edt (Θ) °C	Evaluation	Location	Edt (Θ) °C	Evaluation
1	-1.35	Accepted	20	-2.0	Accepted	39	-1.85	Accepted
2	-1.85	Accepted	21	-1.85	Accepted	40	-1.85	Accepted
3	-1.8	Accepted	22	-2.0	Accepted	41	-1.9	Accepted
4	-1.9	Accepted	23	-1.9	Accepted	42	-1.8	Accepted
5	-1.8	Accepted	24	-2.0	Accepted	43	+1.5	Not-Accepted
6	-1.9	Accepted	25	-1.85	Accepted	44	-1.85	Accepted
7	-1.85	Accepted	26	-1.85	Accepted	45	-1.85	Accepted
8	+1.35	Not-Accepted	27	-1.85	Accepted	46	-1.85	Accepted
9	-1.85	Accepted	28	-1.0	Accepted	47	-1.9	Accepted
10	-1.85	Accepted	29	-1.8	Accepted	48	+1.5	Not-Accepted
11	-1.9	Accepted	30	-1.9	Accepted	49	-1.9	Accepted
12	-1.8	Accepted	31	-1.85	Accepted	50	-1.85	Accepted
13	-1.9	Accepted	32	-1.85	Accepted	51	-1.85	Accepted
14	-2.0	Accepted	33	0.1	Accepted	52	-1.85	Accepted
15	-1.85	Accepted	34	-1.9	Accepted	53	-1.9	Accepted
16	-2.0	Accepted	35	-1.8	Accepted	54	-1.8	Accepted
17	-1.9	Accepted	36	-1.9	Accepted	55	-1.9	Accepted
18	-2.0	Accepted	37	-1.85	Accepted			
19	-1.85	Accepted	38	+1.35	Not-Accepted			

ADPI = (51/55) x100 =92.7%

CASE (5) (SWIRL FLOW)

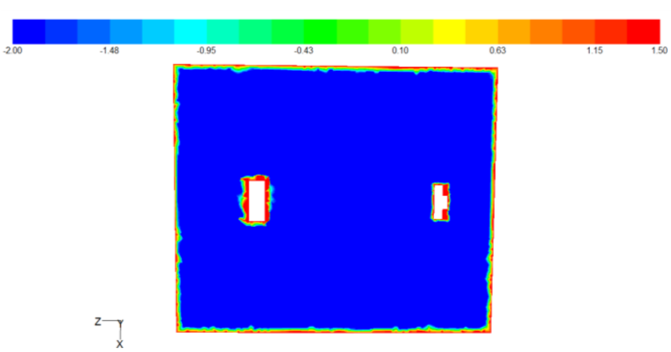

Figure 17:Effective Draft Temperature contours in a horizontal plane at Y=1m

TABLE 9

Results From CFD Simulation For Case5

Location	Edt (Θ) °C	Evaluation	Location	Edt (Θ) °C	Evaluation	Location	Edt (Θ) °C	Evaluation
1	-1.35	Accepted	20	-2.0	Accepted	39	-1.85	Accepted
2	-1.85	Accepted	21	-1.85	Accepted	40	-1.85	Accepted
3	-1.8	Accepted	22	-2.0	Accepted	41	-1.9	Accepted
4	-1.9	Accepted	23	-1.9	Accepted	42	-1.8	Accepted
5	-1.8	Accepted	24	-2.0	Accepted	43	+1.5	Not-Accepted
6	-1.9	Accepted	25	-1.85	Accepted	44	-1.85	Accepted
7	-1.85	Accepted	26	-1.85	Accepted	45	-1.85	Accepted
8	+1.35	Not-Accepted	27	-1.85	Accepted	46	-1.85	Accepted
9	-1.85	Accepted	28	-1.85	Accepted	47	-1.9	Accepted
10	-2.0	Accepted	29	-1.8	Accepted	48	+1.5	Not-Accepted
11	-2.0	Accepted	30	-1.9	Accepted	49	-1.9	Accepted
12	-2.0	Accepted	31	-1.85	Accepted	50	-1.85	Accepted
13	-2.0	Accepted	32	-1.85	Accepted	51	-1.85	Accepted
14	-2.0	Accepted	33	-2.0	Accepted	52	-1.85	Accepted
15	-1.85	Accepted	34	-2.0	Accepted	53	-1.9	Accepted
16	-2.0	Accepted	35	-2.0	Accepted	54	-1.8	Accepted
17	-1.9	Accepted	36	-1.9	Accepted	55	-1.9	Accepted
18	-2.0	Accepted	37	-1.85	Accepted			
19	-1.85	Accepted	38	-1.85	Accepted			

ADPI = (52/55) x100 =94.5%

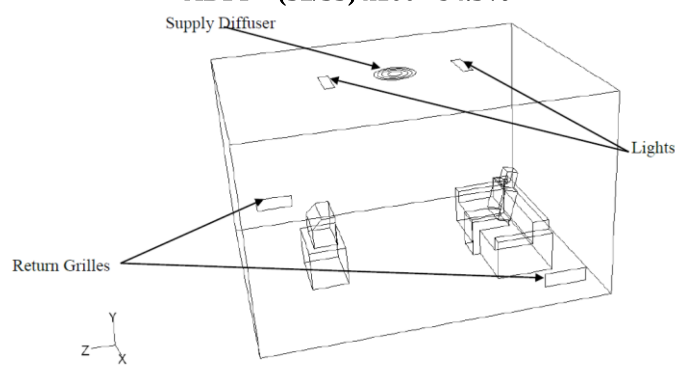

Figure 18:Room Configuration for case 6

Case (6) Side Return

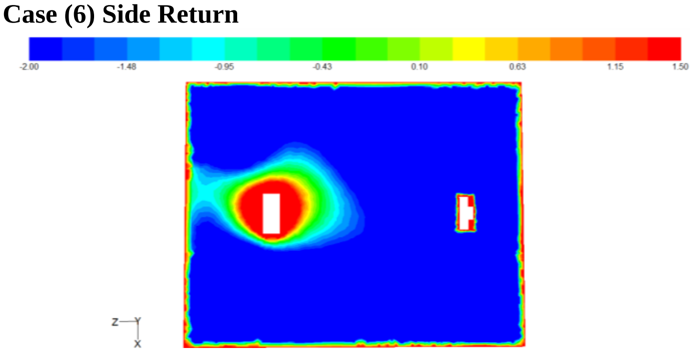

Figure 19:Effective Draft Temperature contours in a horizontal plane at Y=1m

TABLE 10

Results From CFD Simulation For Case 6

Location	Edt (Θ) °C	Evaluation	Location	Edt (Θ) °C	Evaluation	Location	Edt (Θ) °C	Evaluation
1	-1.35	Accepted	20	-2.0	Accepted	39	-1.85	Accepted
2	-1.85	Accepted	21	-1.85	Accepted	40	-1.85	Accepted
3	-1.8	Accepted	22	-2.0	Accepted	41	-1.9	Accepted
4	-1.9	Accepted	23	-1.9	Accepted	42	-1.8	Accepted
5	-1.8	Accepted	24	-2.0	Accepted	43	+1.5	Not-Accepted
6	-1.9	Accepted	25	-1.85	Accepted	44	-1.85	Accepted
7	-1.85	Accepted	26	-1.85	Accepted	45	-1.85	Accepted
8	+1.35	Not-Accepted	27	-1.85	Accepted	46	-1.85	Accepted
9	-1.85	Accepted	28	-1.2	Accepted	47	-1.9	Accepted
10	-1.9	Accepted	29	-1.8	Accepted	48	+1.5	Not-Accepted
11	-2.0	Accepted	30	-1.9	Accepted	49	-1.9	Accepted
12	-2.0	Accepted	31	-1.85	Accepted	50	-1.85	Accepted
13	-1.9	Accepted	32	-1.85	Accepted	51	-1.85	Accepted
14	-2.0	Accepted	33	0.1	Accepted	52	-1.85	Accepted
15	-1.85	Accepted	34	-1.9	Accepted	53	-1.9	Accepted
16	-2.0	Accepted	35	-1.8	Accepted	54	-1.8	Accepted
17	-1.9	Accepted	36	-1.9	Accepted	55	-1.9	Accepted
18	-2.0	Accepted	37	-1.85	Accepted			
19	-1.85	Accepted	38	+1.35	Not-Accepted			

ADPI = (51/55) x100 =92.7%

Case (7) Side Supply

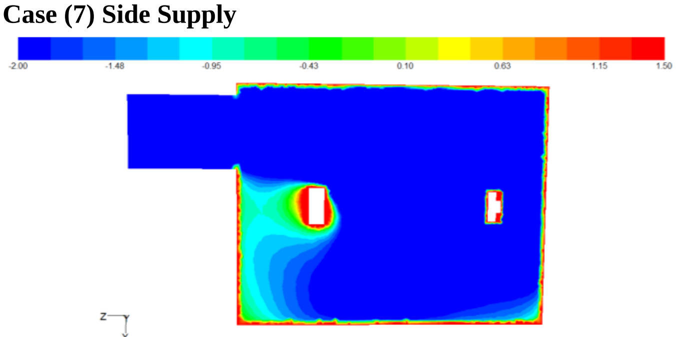

Figure 20:Effective Draft Temperature contours in a horizontal plane at Y=1m

Table 11

Results From CFD Simulation For Case 7

Location	Edt (Θ) °C	Evaluation	Location	Edt (Θ) °C	Evaluation	Location	Edt (Θ) °C	Evaluation
1	-1.35	Accepted	20	-2.0	Accepted	39	-1.85	Accepted
2	-1.85	Accepted	21	-1.85	Accepted	40	-1.85	Accepted
3	-1.8	Accepted	22	-2.0	Accepted	41	-1.9	Accepted
4	-1.9	Accepted	23	-1.9	Accepted	42	-1.8	Accepted
5	-1.8	Accepted	24	-2.0	Accepted	43	+1.5	Not-Accepted
6	-1.9	Accepted	25	-1.85	Accepted	44	-1.85	Accepted
7	-1.85	Accepted	26	-1.85	Accepted	45	-1.85	Accepted
8	+1.35	Not-Accepted	27	-1.85	Accepted	46	-1.85	Accepted
9	-1.85	Accepted	28	-1.85	Accepted	47	-1.9	Accepted
10	-2.0	Accepted	29	-1.8	Accepted	48	+1.5	Not-Accepted
11	-2.0	Accepted	30	-1.9	Accepted	49	-0.85	Accepted
12	-2.0	Accepted	31	-1.85	Accepted	50	-0.85	Accepted
13	-2.0	Accepted	32	-1.85	Accepted	51	-1.85	Accepted
14	-2.0	Accepted	33	-2.0	Accepted	52	-1.85	Accepted
15	-1.85	Accepted	34	-2.0	Accepted	53	-1.0	Accepted
16	-2.0	Accepted	35	-2.0	Accepted	54	-0.95	Accepted
17	-1.9	Accepted	36	-1.9	Accepted	55	-0.85	Accepted
18	-2.0	Accepted	37	-1.85	Accepted			
19	-1.85	Accepted	38	-1.85	Accepted			

ADPI = (52/55) x100 =94.5%

Comparison between CFD Results and ASHRAE Fundamentals chapter20 table 3 ADPI results.

FIGURE 21:ASHREA FUNDAMENTALS CHAPTER 20 TABLE 3

TABLE 12

T50/L VS. ADPI for the cases under study

case	T50/L	ADPI
1	2.0	90.9
2	1.8	90.9
3	2.00	90.9
4	1.1	92.7
5	1.1	94.5
6	1.5	92.7
7	2.5	94.5

V-CONCLUSIONS

From the previous Results and discussion part and according to the results obtained using the numerical investigation, the following conclusions can expressed:

-Generally the experimental work, performed to examine the validity and applicability of the current CFD model, showed a good agreement with the predicted results.

-The Comparison between the current CFD work and the experimental work which made by ASHRAE to predict the ADPI Values for different air outlets geometries shows a good agreement between the CFD and ADPI values.

-Using CFD to predict the ADPI values for the different air outlets geometries which is not found in ASHREA tables.

-Comparing between ADPI and FUNGER model to predict the thermal comfort in living rooms.

VI-REFERENCES

[1] NIOSH, "Guidance for Indoor Air Quality Investigations ", National Institute for Occupational Safety and Health, Hazards Evaluations and Technical Assistance Branch, Division of Surveillance, Hazard Evaluation and Field Studies Cincinnati, OH, 1987.

[2] MDPH, " Indoor Air Quality Assessment ", Massachusetts Department of Public Health, Bureau of Environmental Health Assessment, Boston, MA, 2000.

[3]ASHRAE, " ASHRAE Standard 62-2010: ventilation for acceptable indoor air quality ", American Society of Heating, Refrigerating and Air Conditioning Engineers, Atlanta, USA, 2004.

[4] ACGIH, "Guide to Occupational Exposures-1999 ", American Conference of Governmental Industrial Hygienists, Cincinnati, OH, 1999.

[5] OSHA, "Limits for Air Contaminants ", Occupational Safety and Health Administration, Code of Federal Regulations, 29 C.F.R. 1910.1000 Table Z-1-A, 1997

[6] ISO 7730:2005, Ergonomics of the thermal environment— Analytical determination and Interpretation of thermal comfort

Terminal Device	Room Load, Btu/h·ft^2	T_{50}/L for Maximum ADPI	Maximum ADPI	For ADPI Greater than	Range of T_{50}/L
High sidewall grilles	80	1.8	68	—	—
	60	1.8	72	70	1.5–2.2
	40	1.6	78	70	1.2–2.3
	20	1.5	85	80	1.0–1.9
Circular ceiling diffusers	80	0.8	76	70	0.7–1.3
	60	0.8	83	80	0.7–1.2
	40	0.8	88	80	0.5–1.5
	20	0.8	93	90	0.7–1.3
Sill grille, straight vanes	80	1.7	61	60	1.5–1.7
	60	1.7	72	70	1.4–1.7
	40	1.3	86	80	1.2–1.8
	20	0.9	95	90	0.8–1.3
Sill grille, spread vanes	80	0.7	94	90	0.6–1.5
	60	0.7	94	80	0.6–1.7
	40	0.7	94	—	—
	20	0.7	94	—	—
Ceiling slot diffusers (for T_{100}/L)	80	0.3	85	80	0.3–0.7
	60	0.3	88	80	0.3–0.8
	40	0.3	91	80	0.3–1.1
	20	0.3	92	80	0.3–1.5
Light troffer diffusers	60	2.5	86	80	<3.8
	40	1.0	92	90	<3.0
	20	1.0	95	90	<4.5
Perforated, louvered ceiling diffusers	11–50	2.0	96	90	1.4–2.7
				80	1.0–3.4

using calculation of the PMV and PPD indices and local thermal comfort criteria.

[7] ANSI/ASHRAE 55-2013, Thermal Environmental Conditions for Human Occupancy.

[8] Titus company (http://www.titus-hvac.com) ASHRAE-Chicago Air Distribution Selection Basics June 8, 2010