RESEARCH ARTICLE

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Numerical Study of Air Flow and Distribution of Temperature in A Room with Dislocation Ventilation and A Chilled Ceiling

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ABSTRACT

In the present study, numerical analysis has been employed to investigate the distribution of velocity and temperature of the air in space conditioned with corrugated ceiling radiant cooling panels (RCP). The model of the study is a three-dimensional room with dimensions of (5.0*4.0*3.2) m. The initial and boundary conditions of the models are set to start the simulation in which the mean cooling panel temperature (Tmp) is (15 °C), where Tmp must not be lesser than that variety to avoid the danger of water vapor condensation on the surface of panels and tubes. The air inlet temperatures Tin is taken as (15, 20, and 25) °C and inlet velocities Vin are (1 and 0.5) m/s with an outdoor air temperature of (40 °C). Ansys Fluent software is considered in this study to simulate the flow field under the influence of several factors including outdoor air temperature, mean panel temperature, air inlet velocity, and temperature. The outcomes show an improvement in the overall cooling capacity and the mixed convection heat transfer. Many of the calculated outcomes were displayed as velocity vectors and temperature contour diagrams. Numerical studies assist in using the chilled ceiling technique for cooling aims in Iraqi weather for its simplicity, easiness, and better comfort presentation.

Keywords: Chilled ceiling, mixed convection, radiant cooling panels (RCP), ventilation.

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I. INTRODUCTION

For absorbing extra thermal energy from space, systems of panel cooling utilize cold temperature-controlled indoor surfaces on the ceiling. Thermal energy is moving from the lights, apparatus, occupants, and other room surfaces in a room to the cooled sin surface in an active state. Circulating water over a circuit fixed to the panel leads to preserved temperature. If 50.0% or more of the scheme heat transference on the temperature-controlled surface happens by thermal radiation, this controlled surface is named a radiant panel [1], [2]. A scheme of radiant cooling panels can deliver comfort at a greater inside air temperature that an all-air scheme, as it depends on radiation from a cooled surface to deliver sensible cooling. Schemes of radiant cooling panels utilize water as a transference medium to join the internal radiant surface with an external heat sink. The thermal factors of water permit schemes of radiant cooling panels to [3]:

1. Eliminate a specified quantity of heat from a building and utilize less than 25.0% of the transference energy needed for a scheme of all-air to eliminate a similar quantity of heat consistent with a greater specific heat of water than air.
2. Additional easily interface with schemes of thermal energy storing.

For this reason, that schemes of radiant cooling panels can utilize great surfaces for heat exchange. The panel temperature should be a less degrees lower than the temperature of the air in the room, whereas this minor temperature variance permits an additional reduction in the electric power request of the building [2], [3].

This decreases the difficulties produced by leakage and loss of heat from ducts. The reasonably low air volume is delivered via radiant cooling panel schemes and permits the lessening of the space required for the ventilation scheme and its ductwork. Schemes of radiant cooling panels need approximately 25.0% of the building volume occupied by the usual system of air-conditioning. Floor-to-floor building height can therefore be decreased via dropping plenum height from 1 - 3 m to a quarter of this size. Otherwise, occupants of buildings can relish spaces with greater ceilings [3]. Fig. 1 displays how the transfer of heat and airflow occurs in the scheme of radiant cooling panels.

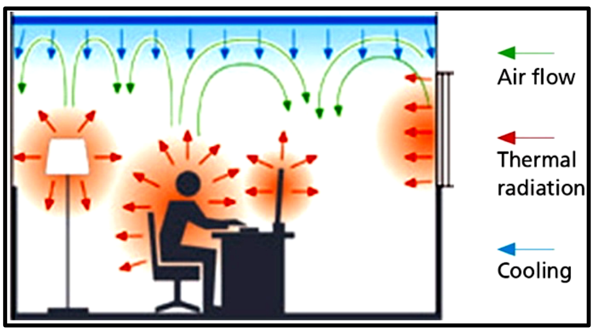

Fig. 1. Heat transfer and airflow in the RCP system.

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II. Ventilation Strategies

RCP systems deal with sensible loads and alone can't remove the latent load and may have a high condensation risk if used in water temperature region under the ambient dew point so it should be supplemented with a ventilation system that provides a very dry and fresh supply air [3], [4]. There are two ways to deliver fresh ventilation air to a space conditioned by RCP:

A. Dedicated Outdoor Air Systems (DOAS)

DOAS conditions the outdoor ventilation make-up air individually from the coming back air from the conditioned space. This method is utilized for treatment results of ventilation make-up air in greater humidity control via handling the main humidity resource in buildings (ambient humidity occurs via the ventilation air) in a straight line at its resource [3], [5]. When carried out with dedicated outdoor air systems equipment capable of controlling the space dew point temperature and interlocks used to avoid the process of the ceiling radiant cooling panels scheme for the duration of disapproving space dew point temperatures, ceiling RCPs can be applied without condensation concerns [5]. Fig. 2 shows the ceiling RCP that is integrated with the DOAS unit and also shows the reduction percentage in outdoor air needed in this hybrid system.

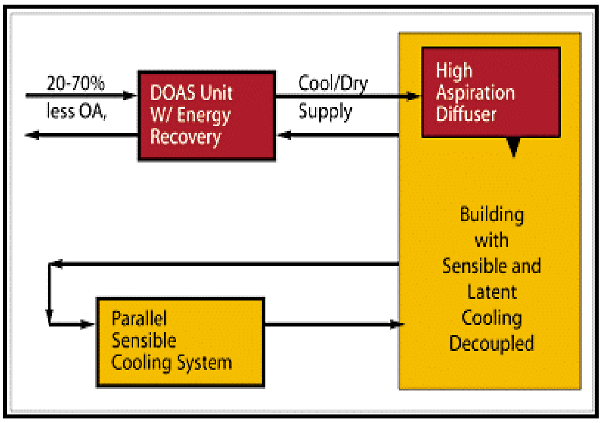

Fig. 2.Schematic of a dedicated outdoor air unit [6].

B. Displacement Ventilation (DV)

Usual "mixing" ventilation utilizes fresh air at a turbulent jet state to dilute and mix any stale contaminated air and preserve conditions at thermal comfort in a building space. In comparison displacement ventilation utilizes a fresh cold air stream at low velocity delivered close to the floor to gradually "displace" the stale air up in the direction of the ceiling from where it leaves the room. This stratifies the air in the room, with warm stale air concentrated overhead the take region and cool newer air in the space in a room where people are, usually limited to 1.8m above the floor [6]. The DV could be utilized with both ceiling or floor RCP schemes and can provide higher thermal comfort degrees and better cooling efficacy relating to the all-air conditioning scheme [7]. The theoretical configuration of the flow of air and the mechanisms of heat exchange in a room with a radiant cooling panel, and a scheme of displacement ventilation are displayed in Fig. 3.

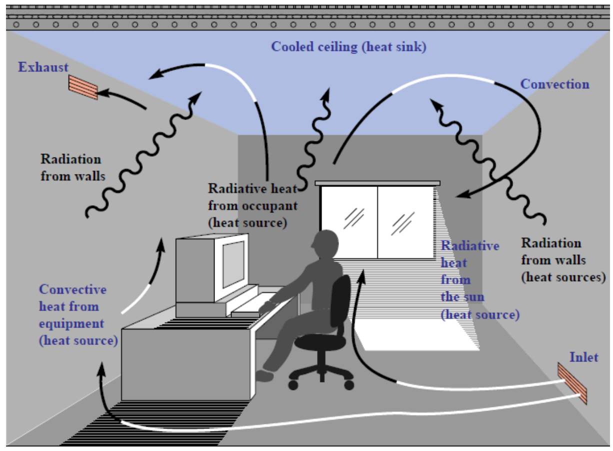

Fig. 3. Flow of air and heat exchange in a room with a cooling ceiling [3].

III. First Cost Considerations of the Panel Cooling/Dedicated Outdoor Air System

With the integration of panel cooling/ Dedicated Outdoor Air Systems matched to a usual all-air variable air volume (VAV) system, the cost matters rise and the major items are recognized as follows [8]-[11]:

1. The size of the pump is lesser as a result of the chiller size decrease.
2. The building design chiller size is dropped because of the necessity for less outdoor air and the use of entire energy (sensible and latent) recovery in the Dedicated Outdoor Air System.
3. The main price of the ductwork and related terminal units is significantly dropped since the Dedicated Outdoor Air System airflow rate is approximately 15.0 - 20.0 % that of an all-air VAV scheme.
4. Depth of plenum can be decreased in novel structure because of the lesser ductwork and removal of terminal VAV boxes. The main price reserving potential could be important as a result of dropped materials utilized for the building enclosure, i.e., interior partitions, fire protection, structural, plumbing, and vertical transference.
5. Reduction of the size of the air-handling unit.

A. Disadvantages of the RCP

1. When controls are not chosen or fixed appropriately, response time can be slow.
2. In a stand-alone panel cooling scheme, dehumidification and condensation of the panel surface might be of prime concern. Unitary dehumidifiers must be utilized, or a latent air treatment system must be presented to the indoor space.
3. Improper choice of panel cooling tube and/or incorrect sizing of cooling resource can cause non-uniform temperatures of surface or inadequate sensible cooling capacity [12].

IV. Numerical Work

With the development of computer technology, the mathematical approach has been accepted as the third approach in the scientific world, especially in the area of fluid dynamics. It is a compliment to the first two approaches in the scientific world, i.e., theoretical and experimental approaches [13]. So, these mathematical simulations permit the study of complex phenomena devoid of preserving expensive samples and problematic investigational dimensions. For the duration, the latest years, Computational Fluid Dynamics (CFD) becomes one of the main prevailing and beneficial ways of predicting the temperature distribution and air flow behaviour in environments that are conditioned with ceiling RCP. Numerical modelling of flow, transfer of heat, and associated phenomena are of considerable significance in many manufacturing implementations [14].

A. Mesh Generation

Standard Computational Fluid Dynamics approaches need a mesh that regulates the limitations of the computational field. The production of computational mesh that is appropriate for the discretized key of momentum, continuity, and energy equations has been the subject of intensive investigation. This type of problem covers an extensive range of engineering implementations.

B. Three-Dimensional Mesh Generation

Mesh generation of solid geometry and three-dimensional models are more durable to be divided into the following steps containing two main matters for additional controlling of the mesh. This might include the followings:

C. Volume Mesh Generation

As far as the panel surface has meshed, now volume mesh can be created. Building the mesh wants fine cells in an area close to the surface of the panel. Conversely, utilizing this part size in the entire domain would cause a huge number of parts. That is why it was decided to utilize a fine mesh in the zone near the surface of the panel and utilize coarse meshes as the space from the surface produced.

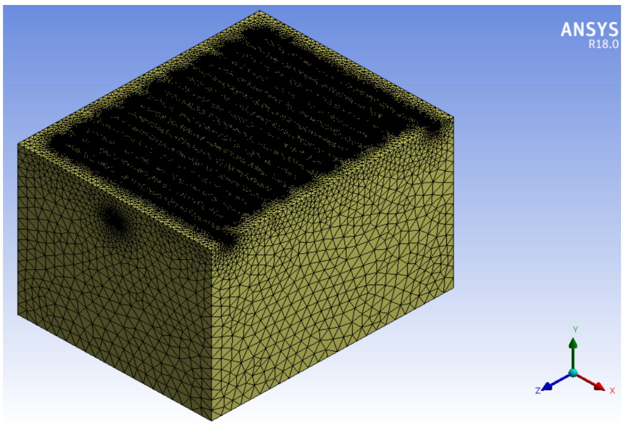

Fig. 4. Room model mesh.

V. Main Governing Equations

The equations of conservation for momentum, continuity, and energy equations and turbulent model are expressed as [15]

A. Conservation of Mass (Continuity)

The variation in mass flow inside a control volume must be identical to the mass flow out over the control volume surfaces, as mass cannot be destroyed or created. This can be exposed arithmetically for an incompressible fluid as:

$$\partial u/\partial x+\partial v/\partial y+\partial w/\partial z=0 \qquad (1)$$

Where:
u: velocity corresponding to the x
v: velocity corresponding to the y
w: velocity corresponding to the z

B. Navier-Stokes Equations (Momentum)

The equations of momentum that govern the fluid flow are resulting from the 2nd of motion of Newton (the momentum conservation). The equations are named Navier-Stokes equations, and for an incompressible fluid, as follows:

$$\frac{\partial(\rho uu)}{\partial x}+\frac{\partial(\rho vu)}{\partial y}+\frac{\partial(\rho wu)}{\partial z}=-\frac{\partial p}{\partial x}+\mu[\frac{\partial^2 u}{\partial x^2}+\frac{\partial^2 u}{\partial y^2}+\frac{\partial^2 u}{\partial z^2}] \qquad (2)$$

$$\frac{\partial(\rho uv)}{\partial x}+\frac{\partial(\rho vv)}{\partial y}+\frac{\partial(\rho wv)}{\partial z}=-\frac{\partial p}{\partial y}+\mu\left[\frac{\partial^2 v}{\partial x^2}+\frac{\partial^2 v}{\partial y^2}+\frac{\partial^2 v}{\partial z^2}\right]+S_{bj} \qquad (3)$$

$$\frac{\partial(\rho uw)}{\partial x}+\frac{\partial(\rho vw)}{\partial y}+\frac{\partial(\rho ww)}{\partial z}=-\frac{\partial p}{\partial z}+\mu[\frac{\partial^2 w}{\partial x^2}+\frac{\partial^2 w}{\partial y^2}+\frac{\partial^2 w}{\partial z^2}] \qquad (4)$$

Where:
u: velocity corresponding to the *x*
v: velocity corresponding to the *y*
w: velocity corresponding to the *z*
x, *y* and *z* are the matching directions.
S_{bj} is the buoyancy resource or sink term.

C. Equation of Energy

The subsequent equation signifies the transference of heat inside the flow domain.

$$\rho\frac{\partial}{\partial x}(uT)+\rho\frac{\partial}{\partial y}(vT)+\rho\frac{\partial}{\partial z}(wT)=\frac{\partial}{\partial x}\left(\Gamma_{eff.h}\frac{\partial T}{\partial x}\right)+\frac{\partial}{\partial y}\left(\Gamma_{eff.h}\frac{\partial T}{\partial y}\right)+\frac{\partial}{\partial z}\left(\Gamma_{eff.h}\frac{\partial T}{\partial z}\right)+S_T \qquad (5)$$

D. Assumptions

Numerous expectations were utilized to resolve the current cases of heat transfer and flow, which are incompressible, steady, and Newtonian flow.

E. Solution

The group of governing equations (1) to (5) for the corrugated radiant ceiling panel cases were resolved by utilizing ANSYS FLUENT 18 [16].

VI. Numerical Cases Setup in Ansys Fluent 18:

Overall Setup:

- Absolute velocity function.
- Pressure-Based resolver kind.
- Steady-state for time consideration.

A. Materials Setup

It is very significant to identify the model construction parts' features of materials, and the features of the medium fluid that is the air, so the solution will be more correct and more convergent. Table I displays air properties.

B. Boundary Conditions

The boundary conditions for the model room are significant in attaining a correct result. Table II shows the momentum of boundary conditions for the entire factors in a room of model in detail. Table III shows the thermal boundary conditions.

TABLE I: PROPERTIES OF AIR

Factor	Unit	Value		
		Piecewise-Linear Profile		
		Point	T_a (°C)	ρ
Density (ρ)	($kg.m^{-3}$)	1	15	1.2256
		2	20	1.2043
		3	25	1.18435
		4	30	1.1644
		5	40	1.1272
Viscosity (μ)	($Kg.m^{-1}.s^{-1}$)	1.7894×10^{-5}		
Thermal Conductivity (k)	($W.m^{-1}.K^{-1}$)	0.0242		
Specific Heat (C_p)	($J.kg^{-1}.K^{-1}$)	1006.43		
Thermal Expansion Coefficient (β)	K^{-1}	3.47×10^{-3}		

TABLE II: MOMENTUM BOUNDARY CONDITIONS

Part	Type	Momentum Conditions	
		Wall Motion	Shear Condition
Panel	Wall	Stationary	No Slipping
Floor and Top Insulator	Wall	Stationary	No Slipping
Side Walls	Wall	Stationary	No Slipping
Supply Grill	Velocity Inlet	- Velocity Magnitude = (0.5 and 1 m/s), [constant].	
Return Grill	Pressure Outlet	- Gauge Pressure = (0 pascal), [costant]. - Backflow Direction specification Way: (Normal to Boundary).	

TABLE III: THERMAL BOUNDARY CONDITIONS

Part	Type	Thermal Conditions
Panel	Wall	- Temperature (T_{mp}=15 °C). - Rate generation of Heat = 0 (W/m^3). - Thickness of Wall = 0.0005 (m).
Floor and Top Insulator	Wall	- Heat Flux = 0 (W/m^2). - Rate generation of Heat = 0 (W/m^3). - Thickness of Wall = 0.04 (m).
Side Walls	Wall	- Convection Heat Transfer Coefficient = 5 ($W/m^2.K$). - Free Stream Temperature = 40 °C. - Heat generation Rate = 0 (W/m^3).
Supply Grill	Velocity Inlet	Temperature = 15, 20and 25 °C.

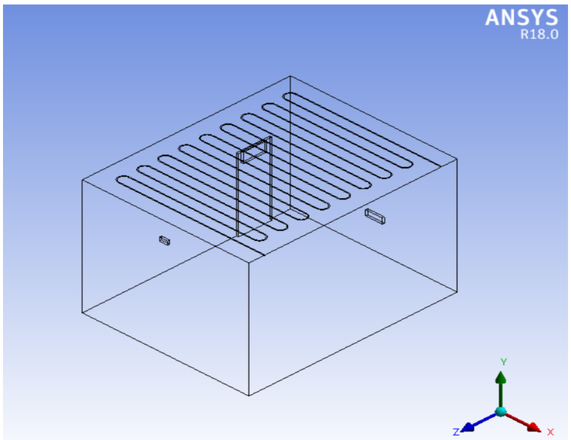

Fig. 5. Boundary conditions of the room model.

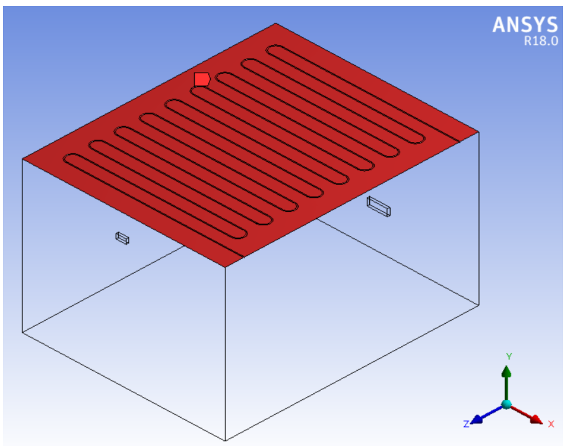

Fig. 6.Boundary conditions of the room model.

VII. NUMERICAL RESULTS AND DISCUSSION

A model size that has the following dimensions (L=5m×W=4m×H=3.2m), was taken in this study. Several computational runs were achieved at different mean ambient temperature, panel temperature, temperature, and velocity of inlet air. One method of exposing data sketchily is to display slices (plane) of the flow in which the (x=2 m) coordinate is held constant. All schematic outcomes are planned on the plane presented in Fig. 7. All slices, sections, temperature contours, and velocity vectors for the numerical work were presented.

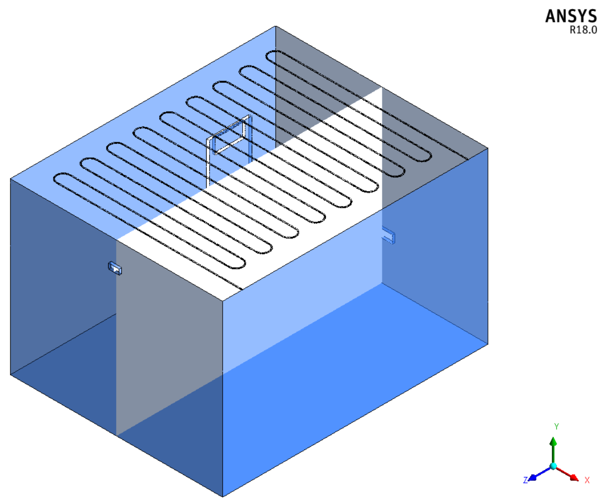

Fig. 7. Description of contour plot section.

A. Velocity Vectors

The plots of the velocity vectors are presented in this section for all the simulated cases. Two inlet air velocities were considered, i.e. (0.5, 1 m/s). The volumetric flow rate of air is calculated depending on ventilation rates only, the buoyant force does not influence the velocity jet due to the relatively small source grill which increases the throw of the air jet. The air throw reaches the reverse wall and the air circulates efficiently in the room and preventing the jet from travelling downward along the floor and also preventing the formation of stationary regions beneath the inlet source. In this type of air-conditioning system, the volume flow rate should be within the ventilation limits. The general features of the flow in these cases are highlighted by the velocity vector plot displayed in Fig. 8-13. The velocity vector plots for forced convection with (u_a =0.5 m/s) are displayed. The effect of air velocity with the location of the side wall grill to the cooled ceiling panel can show clearly how the jet attaches to the ceiling directly after entering the room due to the circulated air which comes from beneath the supply grill. As velocity increased gradually for the cases of u_a =1m/s the buoyant influence decreases on both the velocity jet and the momentum forces overcome. The buoyancy gives better air circulation as shown in Fig. 8-10. The air will flow and circulate downward, and that is because of the higher density of cold air under the panel so it will be heavier than the warmer air below it that has a lower density (i.e. the buoyancy effect) as shown in the velocity flow pattern.

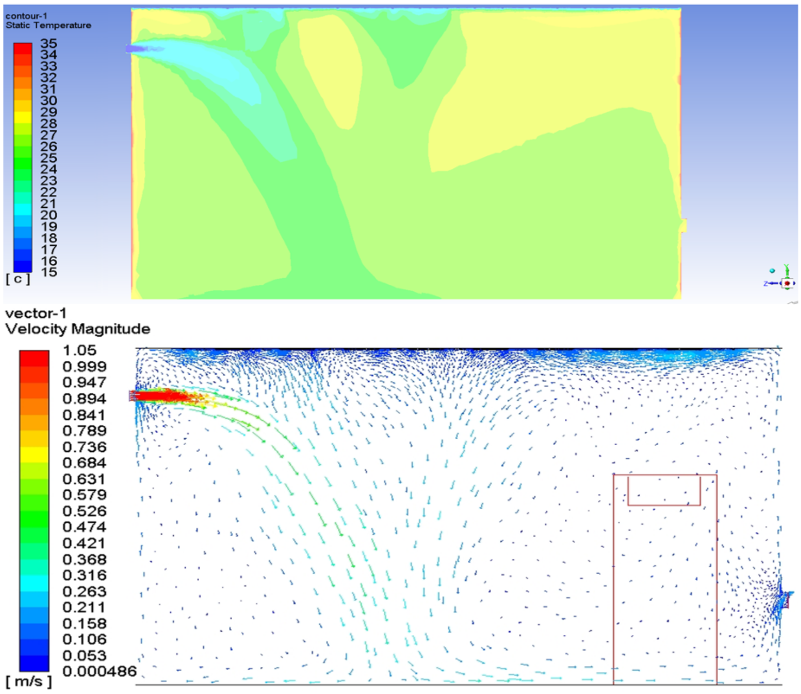

Fig. 8.Temperatures and Velocity Distribution at Tin=15 °C, Vin=1 m/s.

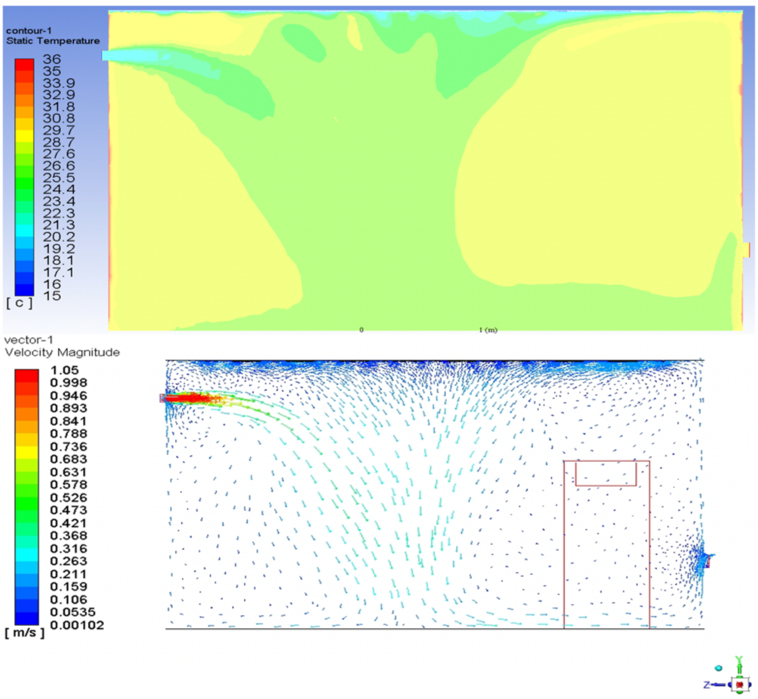

Fig. 9.Temperatures & Velocity Distribution at T_{in}=20 °C, Vin=1 m/s.

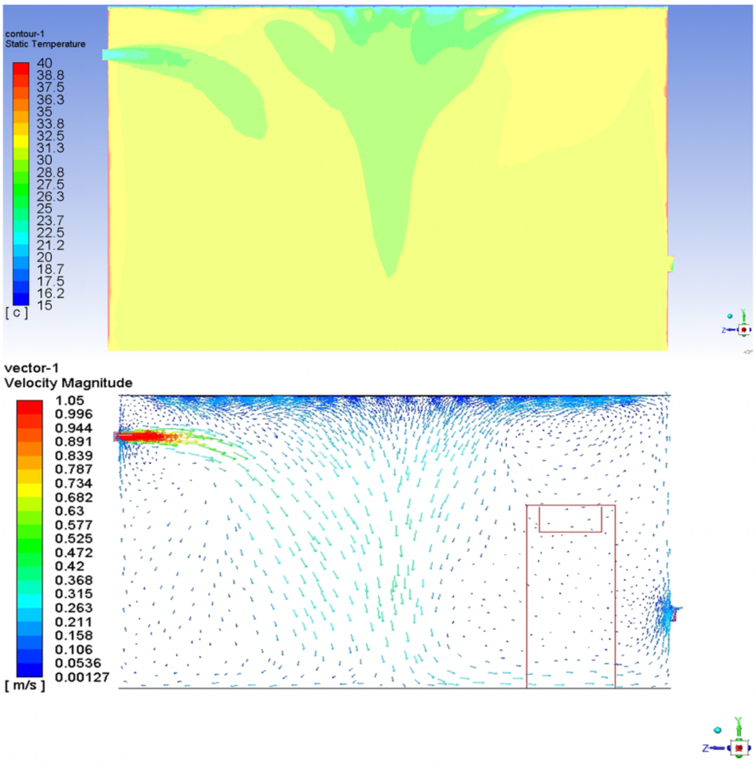

Fig. 10. Temperatures & Velocity Distribution at Tin=25 °C, Vin=1 m/s.

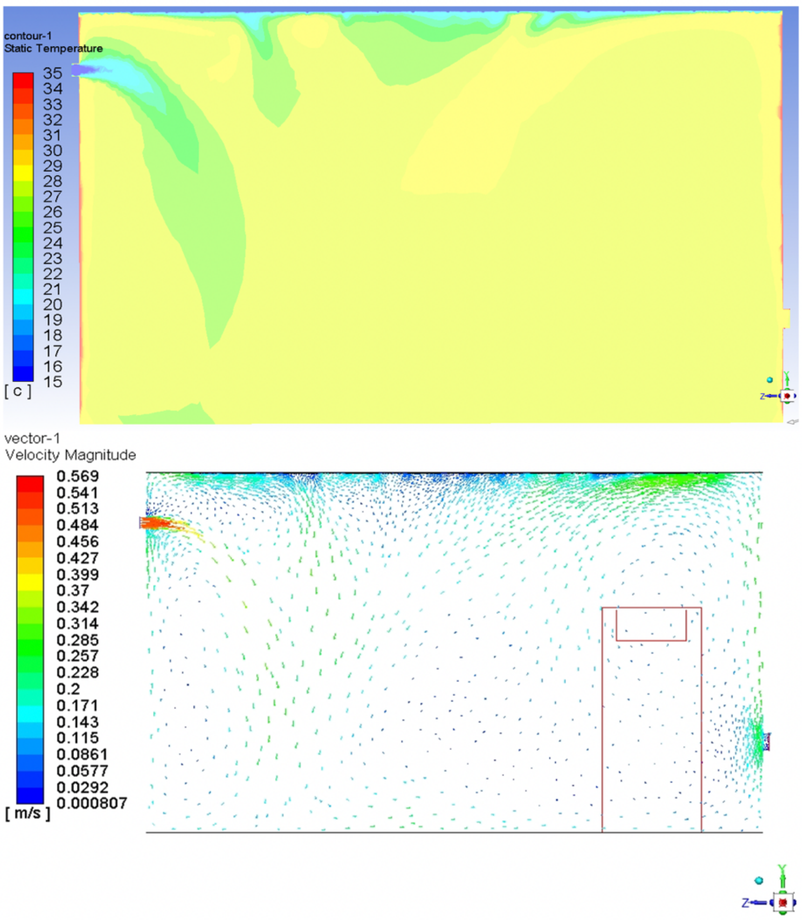

Fig. 11. Temperatures & Velocity Distribution at Tin=15 °C, Vin=1 m/s.

B. Temperature Contours

To clarify the effect of temperature on the air density and flow the temperature distribution shown in Fig. 8-13, the temperature contours for (T_o=40 °C) only, this because in the Iraqi summer climates, the present design of the corrugated ceiling radiant cooling panels (RCP) achieve the desired comfort. Therefore, the concentration will be on the forced convection case. In this part, the cases that are inside the ASHRAE standard of thermal comfort will be exposed. Fig. 8-13. display the temperature contours, the influence of rising the inlet temperature ($T_{a\,i}$= 15, 20 and 25 ºC) is obvious in these figures. Where the trends of contour lines are analogous nonetheless temperature in the occupied zone is augmented and tends to be unstable. Mean panel temperature plays a major role in this analysis (T_{mp}= 15 ºC). A better comfort level was achieved at 15 ºC. Nonetheless, the temperature of the panel should not be lesser than these varieties because of the risk of water vapour condensation involved in the air. The effect of changing the ambient temperature (T_o) is always significant on thermal comfort. The ambient temperature was taken in this analysis (T_o= 40 ºC). Fig. 8-10 show temperature contours with ($T_{a\,i}$= 15, 20 and 25 ºC) inlet temperature and 1 m/s inlet velocity, the comfort achieved until T_o=40 ºC. When decreasing the inlet temperature to 20º C and 15º C with the same velocity above, comfort is achieved until T_o=40 ºC and better distribution for 20 ºC. When changing the inlet velocity to 0.5m/s as shown in Fig. 11-13, better comfort was noticed and temperature distribution in the model room was up to To=40 ºC where uncomfortable areas appear especially the middle areas of the room, Fig. 14 and 15 show the mean room temperature and the cooling load of the room for Vin=0.5, 1 m/s and ($T_{a\,i}$= 15, 20 and 25 ºC) which not exceed T= 25 to 30 ºC and cooling load 0.7 to 2.5 tons.

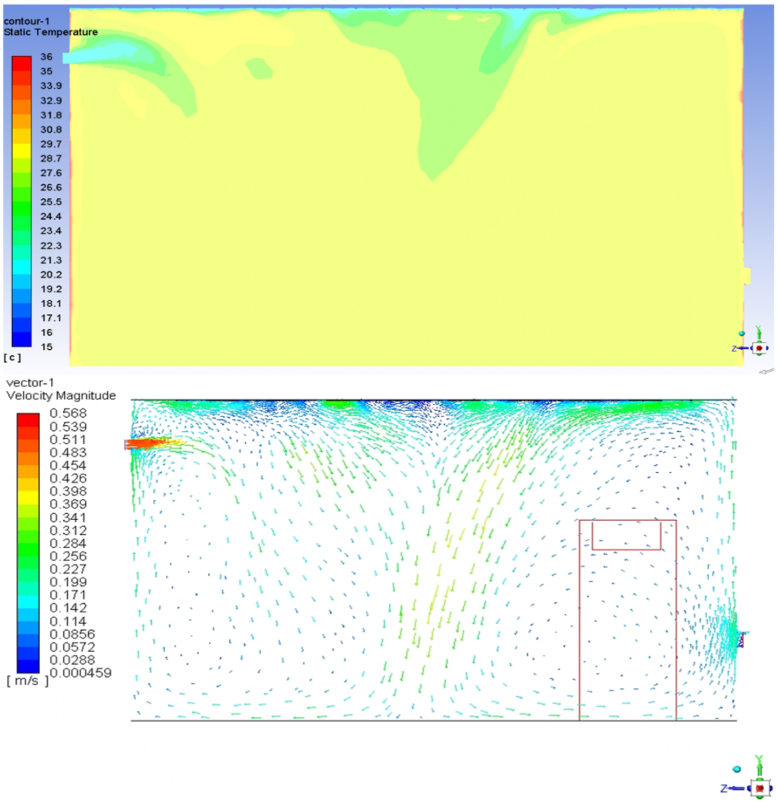

Fig. 12.Temperatures & Velocity Distribution at Tin=20 °C, Vin=0.5 m/s.

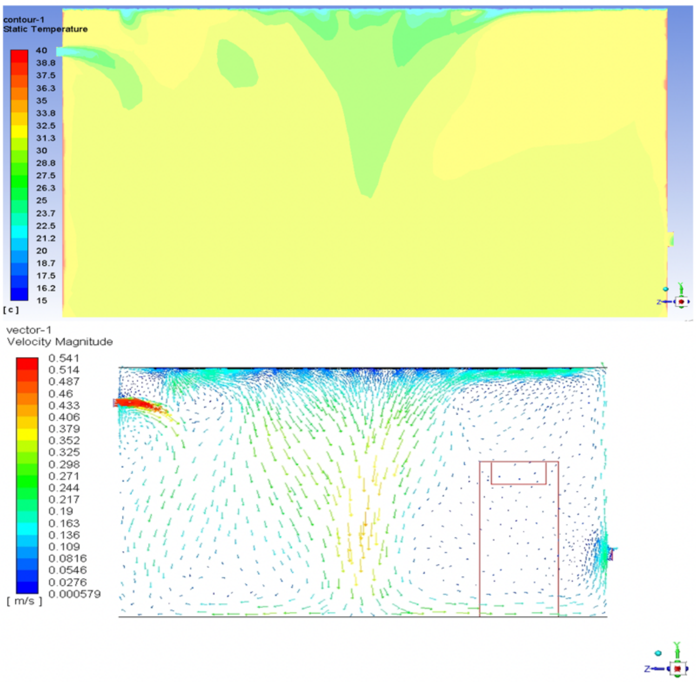

Fig. 13. Temperatures & Velocity Distribution at Tin=25 °C, Vin=0.5 m/s.

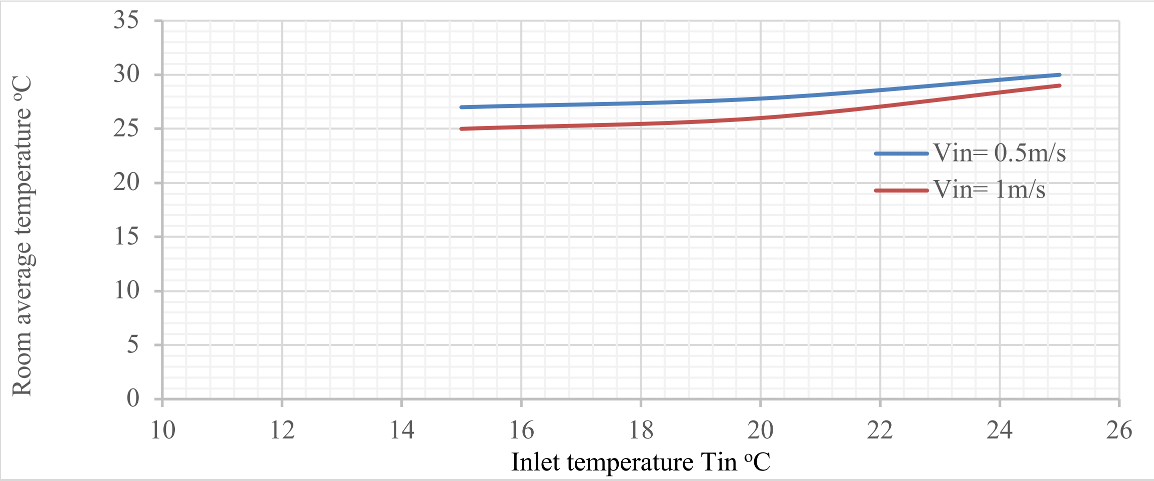

Fig. 14.Effect of inlet air temperature on room average temperature.

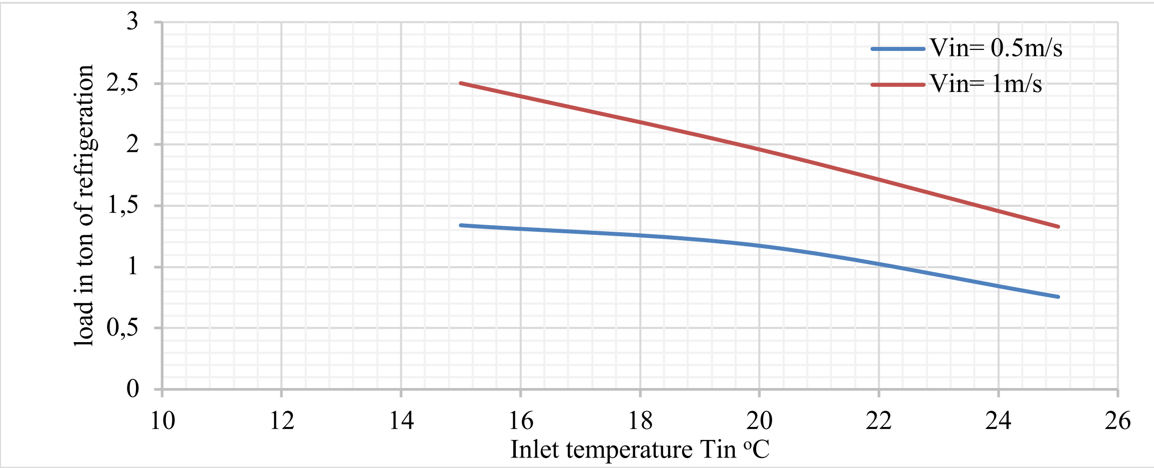

Fig. 15. Effect of inlet air temperature on load in a ton of refrigeration.

VIII. Conclusions

The subsequent conclusions can be drawn from the present work:

1) The outcomes presented the improvement that has happened in overall cooling capacity and the mixed convection heat transfer.
2) The velocity vector plots for forced convection with (u_a =0.5 m/s) are displayed. The effect of air velocity with the location of the side wall grill to the cooled ceiling panel can show clearly how the jet attaches to the ceiling directly after entering the room due to the circulated air which comes from beneath the supply grill.
3) As velocity increased gradually for the cases of u_a =1m/s the buoyant influence decreases on the velocity jet and the momentum force overcomes the buoyancy giving better circulation of air.
4) The effect of changing the ambient temperature (T_o) is always significant on thermal comfort. The ambient temperature was taken in this analysis (T_o= 40 °C).
5) Temperature contours with ($T_{a\,i}$= 15, 20 and 25 °C) inlet temperature and 1 m/s inlet velocity, the comfort achieved until T_o=40 °C.
6) When changing the inlet velocity to 0.5m/s better comfort was noticed and temperature distribution in the model room was up to T_o=40 °C where uncomfortable areas appear especially the middle areas of the room.

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