Numerical Modeling for Radiation in Radiant Cooling Room

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Abstract

This paper presents a numerical study of the radiation and natural-convection effects in enclosure like a room with radiant cooling. The flow considered here is two-dimensional, incompressible and laminar. A numerical model, based on the finite volume method, is employed for the calculation of the governing differential equations. The SIMPLE algorithm for the pressure-velocity coupling is adopted. The matrix inversion method is used for the solution of the radiation exchange in the room. The Rayleigh number based on the cavity height is 10^5. The aspect ratio d/H (length to height ratio of the room) is varied from 0.5 to 2. Boundary conditions are uniform temperature at the cooling panel and uniform heat flux at the other sides of the room. Simulations of flow pattern and temperature distribution are performed for different locations of cooling panels.

Keywords: Finite Volume, Radiant Cooling, Matrix Inversion

1. Introduction

Radiant cooling systems have been employed in northern Europe for more than 20 years [1]. In Thailand, the system equipped with embedded pipes in the floor carrying cooling water for radiant cooling was first used at Suvarnabhumi International Airport in year 2006. It is generally known that the radiant cooling system gives better comfort than other conventional air conditioning systems and also offers calmer room environments and smaller thermal stresses for occupants during long time stays in the cooling room [2]. There are quite a few experimental studies focusing on heat transfer characteristics of cooling panels [3-5]. Basic equations for calculating heat transfer by convection, radiation and combined convection-radiation are described in ASHRAE Handbook [6]. Although the radiative heat exchange is the dominant factor in determining the thermal state in a room with radiant cooling, it should be kept in mind that the convective heat transfer also plays an important role as it influences the air temperature directly.

The present study was carried out in order to gain a better insight into air flow and heat transfer characteristics inside a room with radiant cooling. Three main configurations were studied i.e. the rooms (aspect ratio = 0.5, 1, 2) with floor cooling, wall cooling and ceiling cooling. A numerical study based on the finite volume method was performed to quantify the air temperature and velocity distribution in these rooms.

2. Problem definition and governing equations

The problem under consideration is a room filled with air of Pr = 0.71 (see Fig. 1). One wall is the cooling panel kept at a constant low temperature (T_{RCP}) and the other walls have a uniform heat flux input. The air flow in the room is assumed to be newtonian, incompressible, laminar and two-dimensional.

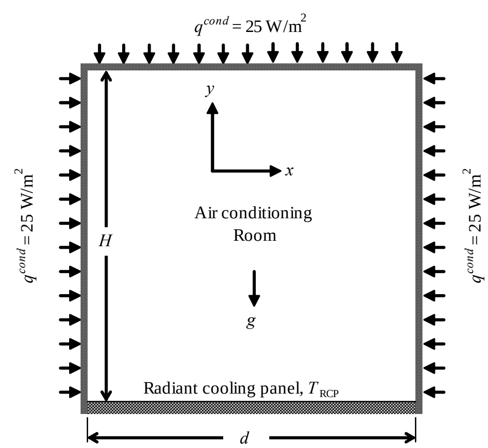


Figure 1. Problem geometry

2.1 Pure convection formulation

The governing equations for the present system can be expressed by the following transport equations:

Continuity:

$$\frac{\partial u}{\partial x}+\frac{\partial v}{\partial y}=0 \tag{1}$$

X-momentum:

$$\rho\left[u\frac{\partial u}{\partial x}+v\frac{\partial u}{\partial y}\right]=-\frac{\partial p}{\partial x}+\mu\left[\frac{\partial}{\partial x}\left(\frac{\partial u}{\partial x}\right)+\frac{\partial}{\partial y}\left(\frac{\partial u}{\partial y}\right)\right] \tag{2}$$

Y-momentum :

$$\rho\left[u\frac{\partial v}{\partial x}+v\frac{\partial v}{\partial y}\right]=-\frac{\partial p}{\partial y}+\mu\left[\frac{\partial}{\partial x}\left(\frac{\partial v}{\partial x}\right)+\frac{\partial}{\partial y}\left(\frac{\partial v}{\partial y}\right)\right]+\rho g\beta(T-T_{\infty}) \tag{3}$$

Nomenclature

a	Coefficient in discretized equation
c	Specific heat transfer at constant volume, J/kg.K
d	Spacing, m
E_b	Black body emissive power, W/m^2
F_{ij}	View factor from the ith element to the jth element
g	Acceleration due to gravity, m/s^2
H	Height of enclosure, m
J_i	Elemental Radiosity, W/m^2
k	Thermal conductivity of fluid, W/m.K
N_{RC}	Radiation conduction interaction parameter, $\sigma T_h^4 d / k(T_h - T_c)$
p	Pressure, Pa
Pr	Prandtl number, υ / α
q^{cond}	Elemental conductive heat flux, W/m^2
q^{conv}	Elemental convective heat flux, W/m^2
q^{rad}	Elemental radiative heat flux, W/m^2
Ra	Rayleigh number, $g\beta(T_h - T_c)H^3 / \upsilon\alpha$
T	Temperature, K
T_c	Temperature of cold wall, K
T_h	Temperature of hot wall, K
T_r	Temperature ratio, T_c/T_h
T_∞	Temperature reference, K
u,v	Velocity components, m/s
x,y	Spatial coordinates

Greek symbols

α	Thermal diffusivity, m^2/s
β	Thermal expansion coefficient, 1/K
ε	Emissivity
μ	Dynamic viscosity, kg/m.s
σ	Stefan-Boltzmann constant, W/m^2.K^4
δ_{ij}	Kronecker delta
υ	Kinematic viscosity of fluid, m^2/s

Subscripts

c	Cold
h	Hot
t	Top
b	Bottom

Energy:

$$\rho c\left(u\frac{\partial T}{\partial x} + v\frac{\partial T}{\partial y}\right) = \frac{\partial}{\partial x}\left(k\frac{\partial T}{\partial x}\right) + \frac{\partial}{\partial y}\left(k\frac{\partial T}{\partial y}\right) \tag{4}$$

The boundary conditions are the no-slip conditions at the walls, $T = T_{RCP}$ at the radiant cooling panel and constant q^{cond} equals $-k\frac{\partial T}{\partial y} + q^{rad}$ or $-k\frac{\partial T}{\partial x} + q^{rad}$ at the uniform heat flux walls.

2.2 Radiation formulation

A schematic view of the problem is shown in Fig.1. With the assumptions of gray surfaces and non-participating medium, there exists a radiative heat exchange between the walls. Radiation heat transfer can be calculated from the radiosity, J, with the view factor, F_{ij}, defined as the fraction of energy leaving the small area i that reaches the small area j. The radiosity, J_i, that leaves the small area i is expressed as

$$J_i = \varepsilon_i \sigma T_i^4 + (1-\varepsilon_i)\sum_j J_j F_{ij} \tag{5}$$

and the radiative flux along wall i is given by

$$q_i^{rad} = J_i - \sum_j J_j F_{ij} \tag{6}$$

2.3 Coupling radiation and natural-convection

The net radiative flux is evaluated from the knowledge of the temperature distribution on the surface.

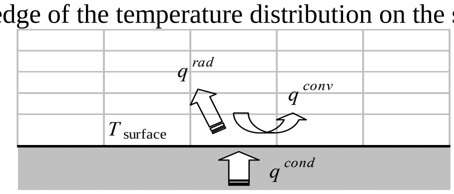


Figure 2. Thermal balance on a solid surface

Fig. 2 illustrates the thermal balance of each surface that the wall surfaces are in thermal equilibrium under the combined action of the conductive, convective and radiative contributions. This leads to

$$-k\frac{\partial T}{\partial y} + q^{rad} = q^{cond} \tag{7}$$

For the insulated walls, Eq. (7) becomes

$$-k\frac{\partial T}{\partial y} + q^{rad} = 0 \tag{8}$$

2.4 Main assumptions and problem conditions

In the present study, the numerical simulation is based on the following room conditions:

- Rayleigh number (Ra) is 10^5 (based on the height of the enclosure)
- Uniform conduction heat flux at walls is 25 W/m^2
- Surface emissivity (ε) is 0.9
- Thermal conductivity of fluid is 0.026 W/m.K
- Average air temperature is 25°C in the occupant area

3. Numerical method

3.1 Pure convection

Numerical solutions of the governing differential equations are obtained by using a finite volume method which utilizes the upwind scheme. The pressure-velocity coupling is assured by the SIMPLE algorithm [7] and the equations are solved with the tridiagonal matrix algorithm (TDMA).

For all the equations, underrelaxations have been used to achieve the convergence of u, v, p, T and q^{rad} (Table 1). The initial low convective flux at boundary has been used to prevent the exceeding surface temperatures.

Table 1. Relaxation factor used in simulation

Configuration	Relaxation factor				
	u	v	p	T	q^{rad}
Floor cooling	0.2	0.2	0.3	0.2	0.01
Ceiling cooling	0.5	0.5	0.5	0.5	0.01
Wall cooling	0.5	0.5	0.5	0.5	0.01

3.2 Combined convection and radiation

After the solutions of the pure convection were obtained, the surface temperatures on the constant heat flux walls are established by using linear approximation. The radiosities are obtained by solving the radiosity equations for an enclosure with the matrix inversion method as reported in [8]. The view factors are evaluated using Hottel's crossed-string method [9]. The matrix of radiosity equations is solved by Gaussian elimination technique. Once the radiative heat fluxes at the walls are defined, the next iteration for the convection equations is performed. The whole process is then repeated until the temperature solutions converged within the 0.05% residual range [10]. The computation flow chart is shown in Fig. 3.

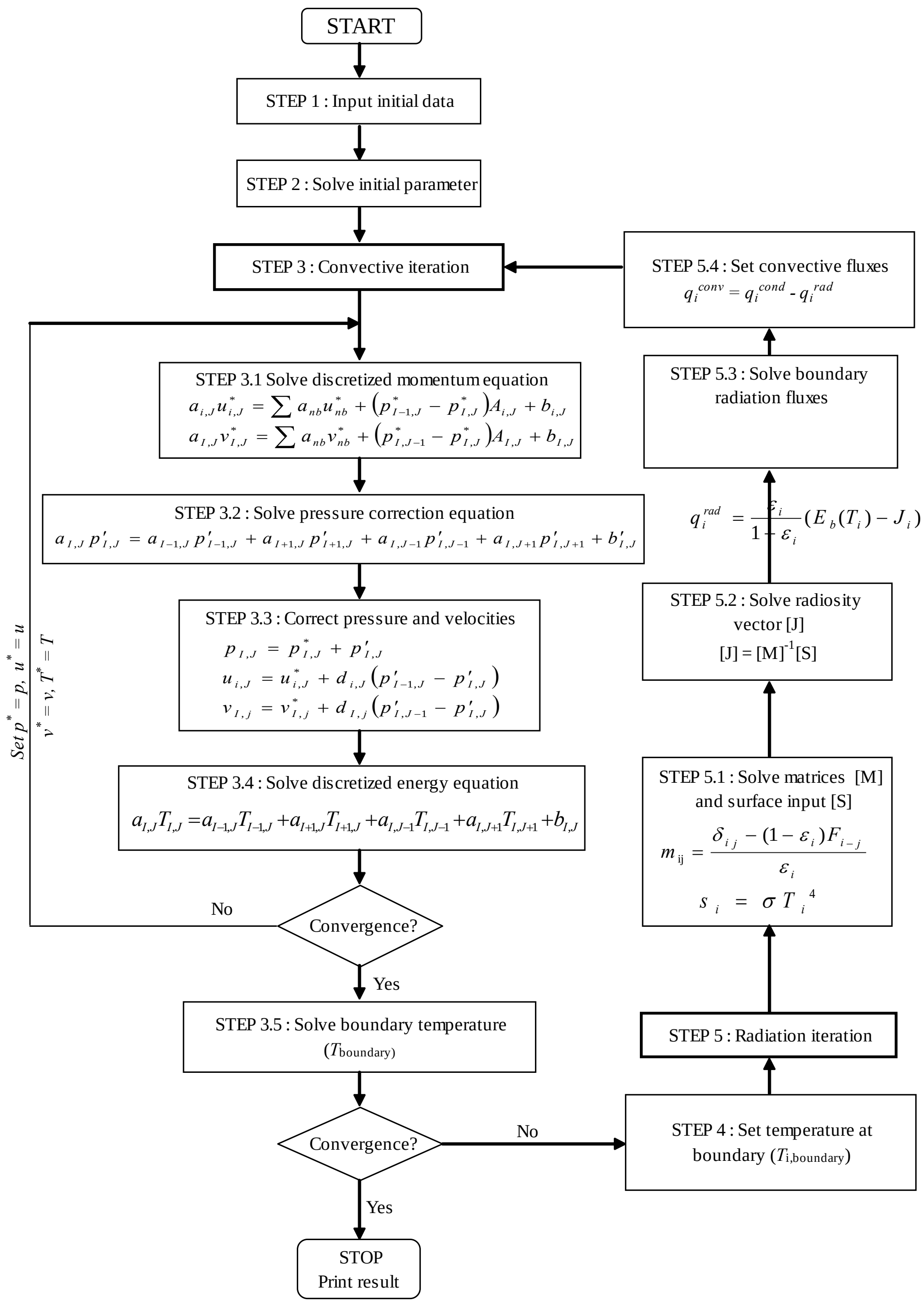


Figure 3. Combined convection and radiation computation flow chart

4. Computer code validation

The computer code is validated with a benchmark problem of a square cavity with the end walls kept at different temperatures and the top and bottom walls are insulated. Table 2 shows the average radiation Nusselt-number comparisons of the hot left-end wall between the present study and Balaji and Venkateshan [11] and Mezrhab et al. [12]. It can be seen that the results agree well with the benchmark values.

Table 2. Average radiation Nusselt-number comparisons on the hot left-end wall

Ra	Radiation Nusselt number		Boundary
	Benchmark	Present study	conditions
5×10^4	0.0387 [11]	0.0384	$\varepsilon_h = \varepsilon_c = 0.1$, $\varepsilon_t = \varepsilon_b = 0.9$, $T_r = 0.85$, $N_{RC} = 1.5$
1×10^6	1.18 [12]	1.19	$\varepsilon_{\text{all surface}} = 0.9$, $T_h - T_c = 20$, $N_{RC} = 30$

5. Results and discussion

The mathematical model developed in Section 2 is used to investigate the mutual radiation natural-convection interaction in an air conditioning room.

In general, the configuration for an air conditioning room is initially specified by the architect. Then the proper type of air conditioning for the room is selected by the engineer. In the present work, three main configurations of air conditioning rooms are selected, namely, a square room (aspect ratio = 1), a hall or plaza (aspect ratio = 2) and a high rise hall (aspect ratio = 0.5). Different locations of cooling panels i.e. floor cooling, wall cooling and ceiling cooling, are then specified for each room configuration.

5.1 Square room (aspect ratio = 1)

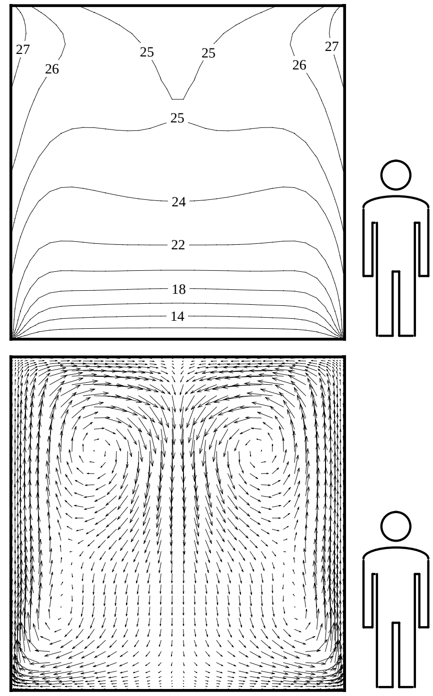
This section studies the air conditioning room that has square size such as bedroom or personal room by the established code.

Fig. 4 shows temperature and velocity profiles for the rooms with floor cooling at T_{RCP} = 10°C, ceiling cooling at T_{RCP} = 13°C and wall cooling at T_{RCP} = 13°C.

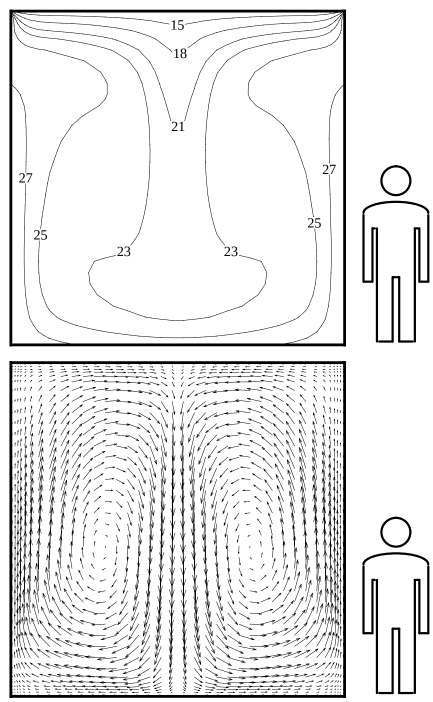
It can be seen that, for the floor and ceiling cooling rooms, the flow fields are similar with two counter-rotating zones which are symmetric about the vertical midplane of the cavity. The symmetric appearance also occurs with the temperature distribution.

On the contrary, the flow field and temperature contour of the room with wall cooling are asymmetric and the flow field contains only a single-cell clockwise flow pattern.

Also from Fig. 4, it is found that the floor cooling room requires lower panel temperature than those of the ceiling and wall cooling rooms in order to keep the average room temperature not exceeding 25 °C. This is due to the cool air closed to the floor does not rise up which leads to low air movement and less convective heat transfer (see Fig. 5).

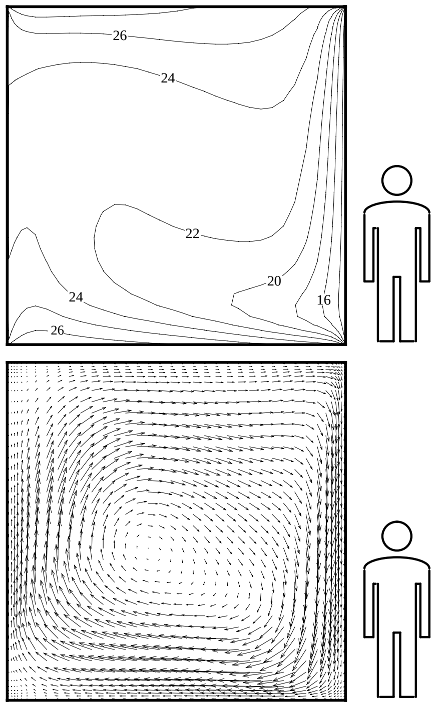


(a)



(b)

Figure 4. Temperature and velocity profiles for the square room (a) Floor cooling at T_{RCP} = 10°C (b) Ceiling cooling at T_{RCP} = 13°C (c) Wall cooling at T_{RCP} = 13°C



(c)

Figure 4. Temperature and velocity profiles for the square room (a) Floor cooling at T_{RCP} = 10°C (b) Ceiling cooling at T_{RCP} = 13°C (c) Wall cooling at T_{RCP} = 13°C (Cont'd)

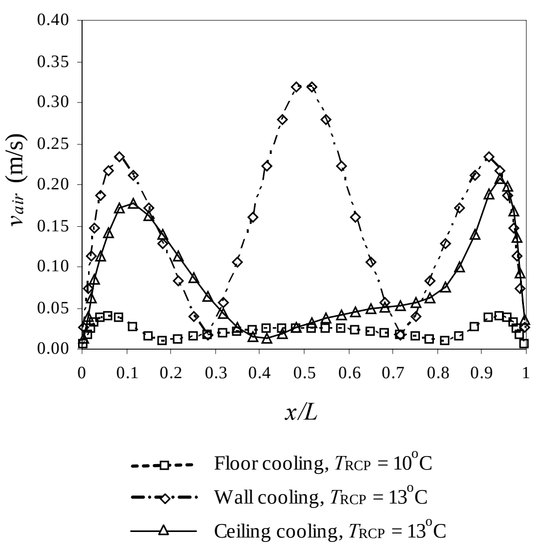


Figure 5. Absolute velocity at half height for the square room

5.2 Hall or plaza (aspect ratio = 2)

Hall or plaza is often a wide and high room. In this section, the air conditioning room of hall or plaza type with the aspect ratio of 2 is considered.

The flow field of the rooms with floor, ceiling and wall cooling consist respectively of two counterrotating regions which are symmetric about the vertical midplane of the room. The temperature distributions are also symmetric for these three cases.

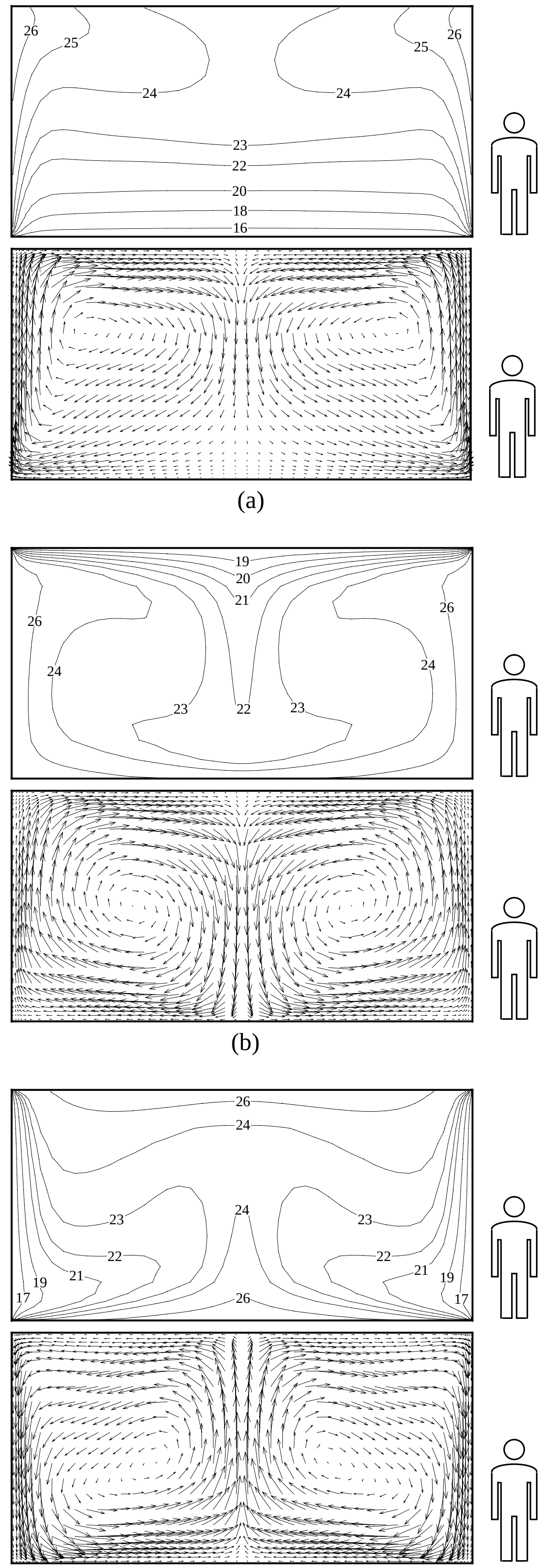


Figure 6. Temperature and velocity profiles for the hall (a) Floor cooling at T_{RCP} = 15°C (b) Ceiling cooling at T_{RCP} = 18°C (c) Wall cooling at T_{RCP} = 15°C

Fig. 6 shows temperature distribution and velocity vectors for the rooms with floor cooling room at T_{RCP} = 15°C, ceiling cooling room at T_{RCP} = 18°C and wall cooling room at T_{RCP} = 15°C.

In this case, the floor and wall cooling rooms require lower panel temperature than that of the ceiling room due to less convective heat transfer in the case of floor cooling panel and the far distance from the middle of the room in the case of cooling wall panel.

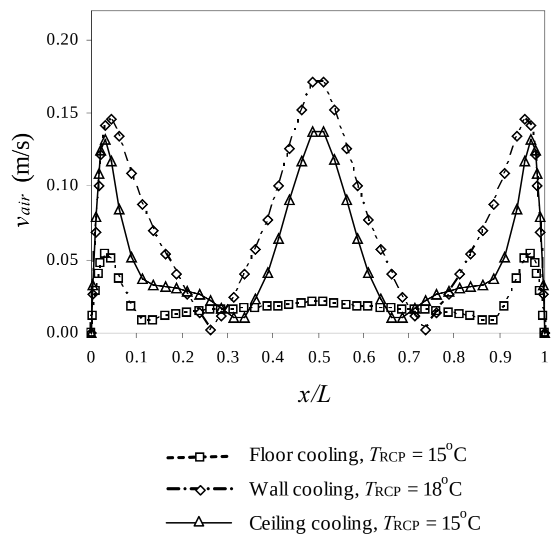


Figure 7. Absolute velocity at half height for the hall

Similar to the square room case, the absolute air velocity at the half height distance of the room with floor cooling is less than those of the rooms with ceiling and wall cooling (Fig. 7). It is found that the ceiling cooling established the greatest air movement along the half height distance of the room.

5.3 High rise hall (aspect ratio = 0.5)

It can be easily observed that only the floor cooling panel type is appropriate for the high rise hall, as the wall and ceiling cooling is not practical to construct. By using the same boundary conditions as in the previous cases, we discover that the area of the cooling floor panel is not enough to cool the air volume in the room. Thus, in this case, it is assumed that the end walls are adjacent to other air conditioning rooms and free from heat flux input (considered as adiabatic walls). The ceiling is kept at a uniform temperature of 38°C.

From Fig. 8, it can be seen that the flow filed consists of four counterrotating regions which are symmetric about both the vertical and horizontal midplanes of the room. The temperature distribution is also symmetric about both planes.

It is found that floor temperature at 18°C can keep the average room temperature not exceeding 25°C for the occupant zone (below ¼ height of room in this case).

5.4 Heat transfer configuration

The parameter which is used to describe heat transfer characteristics of each case is the percentage of radiative heat exchange on the radiant cooling panel. It can be

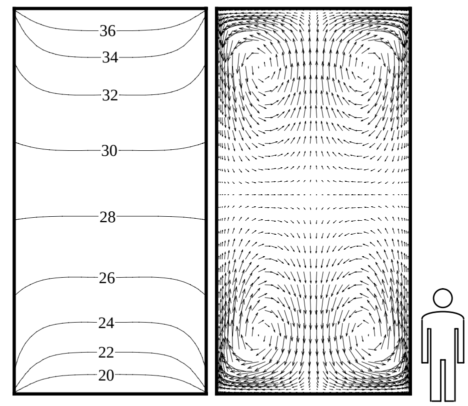


Figure 8. Temperature and velocity profiles for the high rise hall with the floor cooling at T_{RCP} = 18°C

observed that more than 90% of the heat transfer amount is the radiative heat exchange while less than 10% is from the convective contribution (see Table 3). These results are in the same trend with the experimental data of ASHVE laboratory [13]. It is evident that the radiative heat exchange in enclosure is significantly influenced by the heat condition.

Table 3. Comparison of radiation percentages for different cooling configurations

Cooling configuration	Percentage of the area covered by cooling panel	Percentage of radiation
Square room		
Floor	25%	95.75%
Ceiling	25%	94.34%
Wall	25%	95.31%
Hall or plaza		
Floor	33%	95.16%
Ceiling	33%	92.24%
Wall	33%	89.20%
High rise hall		
Floor	16.7%	87.32%

6. Conclusion

In the present study, a computer code is developed for calculation of the combined natural convection and radiation in a room with radiant cooling. Within the specified room configuration range in this paper, the following conclusions can be drawn:

- The radiative heat exchange is the dominant factor in determining the thermal state in a room with radiant cooling.
- The floor cooling room establishes the weekest air movement while the ceiling cooling room gives the strongest air flow.
- In order to maintain the average room temperature at 25°C for the square and hall rooms, the lowest panel temperature (T_{RCP}) is required in the case of floor cooling room while the highest panel temperature can be applied in the case of ceiling cooling room.
- The ceiling cooling room is considered the best configuration for radiant cooling room due to the high panel temperature which can save energy from chilled water generation [14]. In addition, it

also provides the greatest air movement that offer more comfort for the occupants in the room.

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