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Investigation of the effect of radiant panel positions and water temperature on thermal comfort

Radyant panel konumlarının ve su sıcaklığının termal konfor üzerindeki etkisinin araştırılması

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Highlights

- *A numerical analysis was developed to determine thermal comfort in a standard room.*
- *The vertical temperature difference for all cases did not exceed the 3 °C limit value specified in the ASHRAE 55 standard.*
- *Local discomforts were not observed in all three cases.*

Graphical Abstract

The effect of wall mounted and ceiling radiant cooling systems on thermal comfort is investigated numerically.

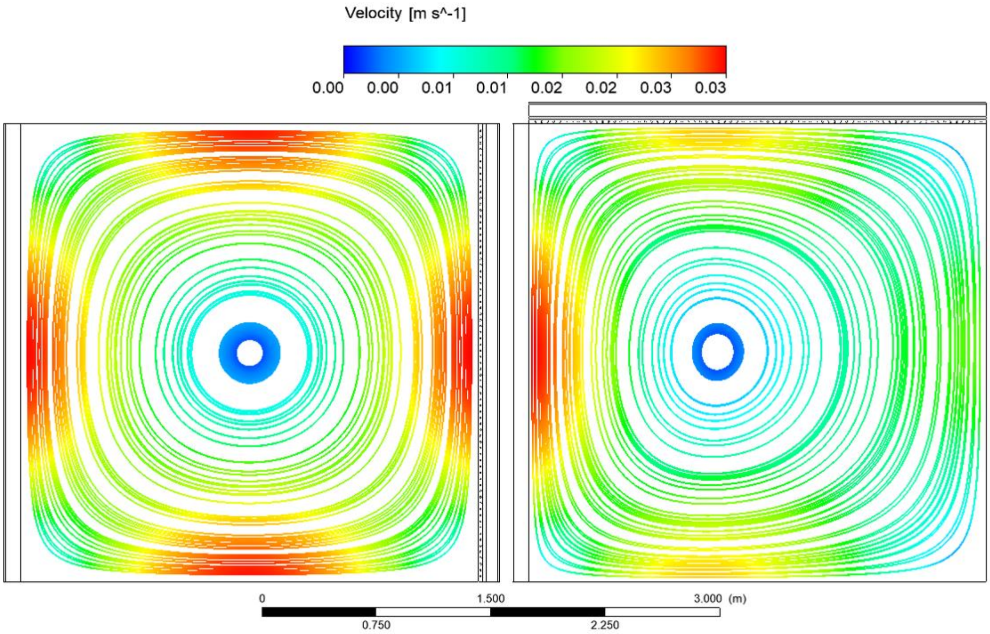

Figure. Velocity streamlines

Aim

This study aims to investigate the effect of the position of the panels and the temperature of the water on the thermal comfort in the case of the radiant wall and radiant ceiling cooling.

Design & Methodology

Thermal comfort was analyzed with Finite Volume Method using commercial software.

Originality

The evaluation of radiant panel positions in terms of thermal comfort constitutes the originality of the study.

Findings

The vertical temperature difference for all cases did not exceed the 3°C limit value specified in the ASHRAE 55 standard.

Conclusion

The radiant heat transfer rates are dominant according to the convective heat transfer rates as it was expected. Thermal comfort conditions are provided according to PMV - PPD parameters when the defined water temperature is 20 °C.

Declaration of Ethical Standards

The author(s) of this article declare that the materials and methods used in this study do not require ethical committee permission and/or legal-special permission.

Investigation of The Effect of Radiant Panel Positions and Water Temperature on Thermal Comfort

Araştırma Makalesi / Research Article

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ABSTRACT

Energy consumption and environmental pollution in the world are increasing day by day. For efficient use of energy, academicians and scientists have developed different models. One of these studies is high temperature cooling systems. Radiant panel systems are examples of high-temperature cooling systems with their performance. In this study, the effect of wall mounted and ceiling radiant cooling systems on thermal comfort is investigated numerically. Numerical results have been compared with the experimental data obtained from the literature and good agreement has been reached. The water temperature inside the radiant panels was defined as 20°C, 22°C and 24°C, respectively and the results were compared according to the PMV (Predicted Mean Vote) – PPD (Predicted Percentage of Dissatisfied) parameters. Six different conditions were investigated and the results show that the best thermal comfort is provided by 20°C of water temperature and the radiant ceiling condition.

Keywords: Thermal comfort, natural convection, PMV-PPD, radiant ceiling cooling, CFD.

Radyant Panel Konumlarının ve Su Sıcaklığının Termal Konfor Üzerindeki Etkisinin Araştırılması

ÖZ

Dünyadaki enerji tüketimi ve çevre kirliliği gün geçtikçe artmaktadır. Enerjinin verimli kullanımı için, akademisyenler ve bilim adamları farklı modeller geliştirmiştir. Bu çalışmalardan biri yüksek sıcaklıkta soğutma sistemleridir. Radyant panel sistemleri, enerji ve ekserji açısından performansları ile yüksek sıcaklık soğutma sistemlerine örnektir. Bu çalışmada, düşük ekserji duvar tipi ve tavan radyant soğutma sistemlerinin ısıl konfor üzerindeki etkisi sayısal olarak incelenmiştir. Sayısal sonuçlar literatürdeki deneysel sonuçlarla doğrulanmıştır. Radyant panellerin içerisindeki su sıcaklığı sırasıyla 20 °C, 22 °C ve 24 °C olarak tanımlanmış ve sonuçlar PMV-PPD parametrelerine göre karşılaştırılmıştır. Altı farklı koşul araştırılmış ve sonuçlar en iyi ısıl konforun 20 °C su sıcaklığı ve tavan soğutma koşuluyla sağlandığını göstermiştir.

Anahtar kelimeler: Isıl konfor, doğal taşınım, PMV-PPD, tavan radyant soğutma, HAD.

1. INTRODUCTION

In the World, reducing fossil fuel usage and renewable energies are getting more popular due to global warming considerations. CO_2 and greenhouse gas emissions are the most critical factors in terms of their environmental effect. In 2010, 30% of the world's carbon dioxide emissions were emitted by buildings [1]. For reducing these emissions, energy consumption should be reduced in buildings. For this purpose, Net Zero Energy Buildings (NZEB) have been built in recent years. In literature, low/high valued energy and exergy definitions have been used to determine the influence of buildings [2]. Low-exergy heating and cooling systems get their energy from sustainable energy sources such as heat pumps and solar collectors [3].

Basically, thermal comfort is that condition of mind which expresses satisfaction with the thermal environment. It is based on the energy balance between the human body and the environment. It has many physiologically and psychologically parameters which are the changeable person to person. Therefore, it is difficult to satisfy everyone in the environment. Human body sweats when the environment is too hot, while the human body shivers when the environment is too cold. For this reason, thermal comfort effects the quality of life in daily life. The calculation method of thermal comfort is based on PMV-PPD approach which was developed by Fanger in 1970 [4]. In heating and cooling applications, using radiant panels provide better satisfaction than other conventional systems. Ceiling and floor radiant systems are attracting more and more attention in recent years because of their thermal comfort [5-10]. Radiant heating system can achieve better thermal comfort at a lower temperature as well as the radiant cooling system can provide better thermal comfort at higher temperature [11].

Imanari et al. [12] compared the conventional air conditioning system with the radiant ceiling panel system in terms of thermal comfort, energy consumption and cost. Radiant ceiling panel system has resulted in a very effective thermal environment in terms of cooling and saving energy. Oxizidis and Papadopoulos [13]

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compared radiant and convection systems in terms of energy consumption and thermal comfort in a test chamber. They indicated that the radiant surfaces are the best model to improve thermal comfort conditions. Catalina et al. [14] studied thermal comfort using CFD and they validated their results with experimental values. The boundary conditions were taken from the experimental data. CFD simulations showed that local discomfort occurred at the feet/ankle level zone. They have drawn attention to condensation risk while using radiant cooling panels.

Lim et al. [15] have investigated the performance and applicability of the control methods of radiant floor cooling systems experimentally and numerically according to the floor surface condensation and comfort parameters. According to the results of the test, it has been found that the floor surface temperature is 21 °C, the temperature difference between the surfaces is 6 °C and the vertical temperature difference is below 1.9 °C. Thus the results provided thermal comfort standards. Hodder et al. [16] have studied the effect of displaced ventilation with the cooled ceiling on thermal comfort. A test room with a typical chilled ceiling, an office with displacement ventilation was built and experiments were conducted involving eight female subjects. Radiant temperature asymmetry has been observed to be the highest problem affecting overall thermal comfort.

Zhao et al. [17] reviewed some applications about radiant cooling. In some large buildings such as airports and railway stations, the envelope of the building is dominated by glass facades. Therefore, the indoor thermal environment is affected by the intensity of solar radiation and high temperature of internal wall surfaces. Radiant cooling systems are strongly recommended for thermal comfort in this type of buildings. Hernández et al. [18] installed a new ventilation terminal and combined it with a radiant floor. They performed both experimental and numerical analyzes in their work. Experiments and analyzes show that the analyzes overlap with the experiments. Pipes in the floor have provided a homogeneous temperature distribution. Since the vertical temperature is lower than 2.7 °C and can be neglected, thermal comfort conditions are provided.

Dong et al. [19] combined radiant-convective heating terminal with air source heat pump (ASHP). They investigated operating characteristics and heating performances experimentally. They have also highlighted that radiant heating systems have a positive effect on the indoor thermal environment. Romani et al. [20] installed an experimental set-up in Spain which is consist of three houses like cubicles. They embedded the radiant pipes into the walls in their experiments. The ground source heat pump was also added for cooling and heating. Gao et al. [21] studied the heat loss from the human body. The heat exchange between the human body and its environment under radiant systems is discussed in detail numerically.

Chicote et al. [22] performed an experimental study on the cooling capacity of a radiant ceiling system. A Test chamber has been built with 3.6 m height and 3.6 m width. They compared the results with some other experimental studies. Cholewa et al. [23] studied an experimental study about heat transfer coefficients for the heated/cooled radiant ceiling. They noticed that when calculating the heat transfer coefficients it is very important to assume that ambient temperature inside the chamber.

This study aims to investigate the effect of the position of the panels and the temperature of the water on the thermal comfort in the case of the radiant wall and radiant ceiling cooling. In the cooling case, studies that examine the radiant panel positions and the temperature of the working fluid in the pipes are limited. The results were compared with experimental studies in the literature and found to be consistent with the study. This study clearly shows the effects of water temperatures in the radiant panel on thermal comfort.

2. MATERIAL and METHOD

2.1. 2D Model

In this study, a 2D room measuring 3 m width and 3 m height was modeled and analyzed. Room details and sections can be seen in Fig. 1. The properties of materials that are used to radiant panels are given in Table 1. Since the left and right wall properties did not change with depth, it is found that the room section was adequate. There is a size difference between the previous study [9] and this study. In this study, the room section is taken as a square to compare two different locations exactly. In addition to the previous study, the effect of the radiant panels' location on thermal comfort is also examined.

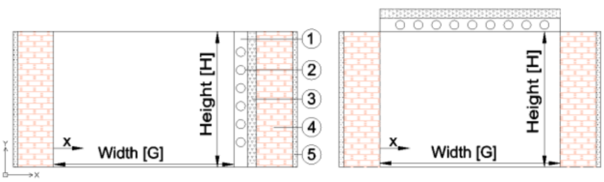


Fig. 1. 2D room model for wall-mounted (I) and ceiling radiant panel (II)

Table 1. Properties of structuring and other materials [9]

No.	Materials	Thickness [mm]	Thermal Conductivity [W/mK]
1	Drywall	30	0.37
2	Pex Pipe (Cross-linked Polyethylene)	-	0.41
3	Isolation (XPS - Extruded Polystyrene)	20	0.035
4	Brick	240	0.81
5	Cement plaster	20	0.72

2.2 CFD Modelling and Boundary Conditions

2.2.1. Governing Equations

This study assumes that the flow is 2D, steady, and the physical properties the viscosity, density, and thermal conductivity are constant. Therefore the governing equations may be written for rectangular coordinates x and y [24]. The heat transfer between the air and radiant wall panel is carried out by natural convection. The Boussinesq approach is used for a variety of natural convection problems. According to this approach, momentum equations are written and edited as follows [25]:

Conservation of mass:

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

Conservation of momentum:

$$u\frac{\partial u}{\partial x}+v\frac{\partial u}{\partial y}=-\frac{1}{\rho}\frac{\partial p}{\partial x}+\frac{\mu}{\rho}\left(\frac{\partial^2 u}{\partial x^2}+\frac{\partial^2 u}{\partial y^2}\right) \tag{2}$$

$$u\frac{\partial v}{\partial x}+v\frac{\partial v}{\partial y}=-\frac{1}{\rho}\frac{\partial p}{\partial y}+\frac{\mu}{\rho}\left(\frac{\partial^2 v}{\partial x^2}+\frac{\partial^2 v}{\partial y^2}\right)+g\beta(T-T_0) \tag{3}$$

Conservation of energy:

$$u\frac{\partial T}{\partial x}+v\frac{\partial T}{\partial y}=\alpha\left(\frac{\partial^2 T}{\partial x^2}+\frac{\partial^2 T}{\partial y^2}\right) \tag{4}$$

In natural convection, the boundary layer is not limited to the laminar region. In natural convection, the transition zone is highly dependent on the buoyancy and viscous forces. The Rayleigh number determines the transition zone. The Rayleigh number can be calculated by equation 5. The Rayleigh number of 10^9 is critical for vertical plane plates, and the values above this value are considered turbulent flow [24].

$$Ra_{x,c}=Gr\ \Pr=\frac{g\beta(T_s-T_\infty)x^3}{\nu\alpha}\approx 10^9 \tag{5}$$

In this study, the Rayleigh number was found to be approximately 1.6×10^9, and the flow was seen to be turbulent. Standard k-ε turbulence model was used as the turbulence model.

The standard k-epsilon model is the most well-known and widely used eddy viscosity model with two equations [26]. It has been observed that this model gives accurate results near the wall where the viscosity and turbulent flow are effective [27]. It has been experimentally proven to be the most suitable model for natural convection [28]. The transport equations for this model are shown below.

$$\begin{aligned}&\frac{\partial}{\partial t}(\rho k)+\frac{\partial}{\partial x}(\rho k u)=\\&\frac{\partial}{\partial x}\left[\left(\mu+\frac{\mu_t}{\sigma_k}\right)\frac{\partial k}{\partial x}\right]+G_k+G_b-\rho\varepsilon-Y_M\end{aligned} \tag{6}$$

$$\begin{aligned}&\frac{\partial}{\partial t}(\rho \varepsilon)+\frac{\partial}{\partial x}(\rho \varepsilon u)=\\&\frac{\partial}{\partial x}\left[\left(\mu+\frac{\mu_t}{\sigma_\varepsilon}\right)\frac{\partial \varepsilon}{\partial x}\right]+C_{1\varepsilon}\frac{\varepsilon}{k}(G_k+C_{3\varepsilon}G_b)-C_{2\varepsilon}\rho\frac{\varepsilon^2}{k}\end{aligned} \tag{7}$$

where

$$\mu_t=\rho C_\mu\frac{k^2}{\varepsilon} \tag{8}$$

The following equation determines heat conduction in a continuous two-dimensional regime and constant heat transfer coefficient without heat generation.

$$\frac{\partial^2 T}{\partial x^2}+\frac{\partial^2 T}{\partial y^2}=0 \tag{9}$$

Radiation heat transfer is higher than convection heat transfer in this study. Therefore, the radiation effect is considered. For applications involving optical thicknesses greater than 10, DO option can be enabled in the Radiation model. [29].

When using the Ansys-Fluent software [30], the Pressure-based model is selected as solver type in solver settings. All analyzes were made by taking the steady state into consideration. Since natural convection analyzes will be carried out in the room model, the gravitational acceleration is defined as -9.81 m/s^2 in direction y. The energy model is activated to calculate the heat transfer in the solutions. Since the heat transfer from the panels is by convection and radiation, the radiation model is also activated in addition to the energy model to calculate the heat passing through the radiation. The discrete ordinates (DO) model was used as the radiation model.

Fig. 2 shows the boundary conditions. The outdoor temperature is directly defined on the surface of the cement plaster. The outdoor temperature is defined as 33 °C for Istanbul [31]. The water temperature in the radiant panels was 20 °C, 22 °C and 24 °C, respectively. The temperatures defined in the radiant panels are defined as pipe surface temperature and the effects of the flow within the pipe are not taken into account. 72 pipes, 10 mm in diameter, were used.

2.2.2. Grid and Grid Independency

The geometry indicated in Fig. 1 is the computational domain. For mesh independence, 8 different analyzes were performed with a mesh number between 10×10^3 and 510×10^3. Fig. 3 shows the variation of the Rayleigh number depending on the mesh number. When the number of mesh is greater than 130×10^3, it is understood that the study is independent of mesh number. In this study, the mesh number was determined as 135×10^3.

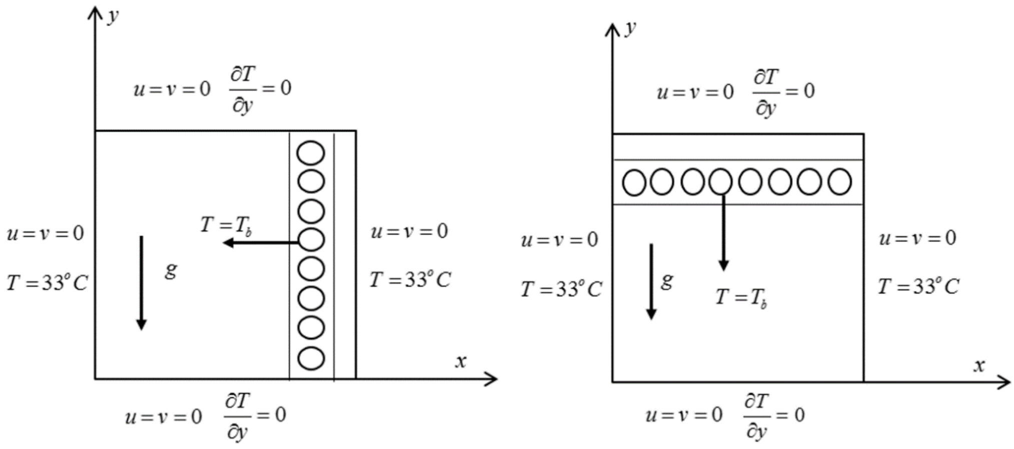


Fig. 2. Boundary conditions for both cases

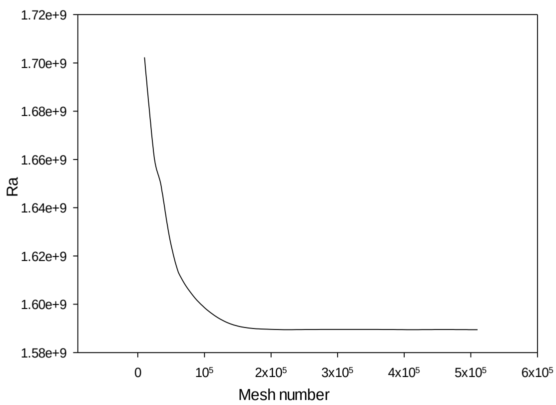


Fig. 3. Grid independency

2.3. Power Equations

Radiant heat flux can be obtained with Eq. (10) from active (heated or cooled) panel surface to other surfaces [32, 33].

$$q_r = 5x10^{-8}[(T_s + 273.15)^4 - (AUST + 273.15)^4] \qquad (10)$$

where;

$$AUST = \frac{\sum_{i=1}^{5} A_i T_i}{\sum_{i=1}^{5} A_i} \qquad (11)$$

According to the ASHRAE Standard 55 and ASHRAE Standard 138, the operative temperature can be obtained in terms of air temperature and the average of mean radiant temperature with Eq. (12) [34].

$$T_o = \frac{T_r + T_a}{2} \qquad (12)$$

The standard EN 15377–1 [35] establishes total heat flux correlation between surface temperature and the operative temperature as shown in Eq. (13).

$$q_{tot} = 8.92(T_o - T_s)^{1.1} \qquad (13)$$

There are a lot of methods to calculate the mean radiant temperature. Two of them are described in Eq. (14) and (15). In this simple equation of the mean temperature, A means that area of the surfaces, T means that the surfaces' temperature, respectively [36].

$$T_r = \frac{A_1T_1 + A_2T_2 + A_3T_3 + A_4T_4 + A_fT_f + A_cT_c}{A_1 + A_2 + A_3 + A_4 + A_f + A_c} \qquad (14)$$

The second method [36] requires a black globe thermometer as well as an airspeed sensor. Here T_g is globe temperature, D is the globe's diameter, e is the emissivity of the globe and V_a is the airspeed, respectively. In this study, Equation 14 is employed.

$$T_r = [(T_g + 273.15)^4 + \frac{1.10x10^8 V_a^{0.6}}{eD^{0.4}}(T_g - T_a)]^{1/4} - 273.15 \qquad (15)$$

2.4. Thermal Comfort Calculation Methods

Thermal comfort is a situation that expresses satisfaction with the environment in terms of thermally. In this study, local thermal comfort criteria and PMV and PPD indexes were examined using ASHRAE 55 and ISO 7730.

2.4.1. Predicted Mean Vote (PMV)

PMV is an index based on the heat balance of the human body and estimated based on the votes of a group of people.PMV can be calculated as a function of clothing insulation, metabolic rate air temperature, airspeed, mean radiant temperature and relative humidity:

$$PMV = \left(0.303\times e^{-0.036\times M} + 0.028\right)\times \begin{bmatrix} (M-W)-3.05\times10^{-3}\left\{5733-6.99\cdot(M-W)-p_a\right\} \\ -0.42\left\{(M-W)-58.15\right\}-1.7\times10^{-5}M\left(5867-p_a\right) \\ -0.0014M\left(34-T_a\right) \\ -3.96\times10^{-8}f_{cl}\left\{\left(T_{cl}+273\right)^4-\left(T_r+273\right)^4\right\} \\ -f_{cl}h\left(T_{cl}-T_a\right) \end{bmatrix} \quad (16)$$

where

$$T_{cl} = 35.7-0.028\left(M-W\right) - I_{cl}\begin{bmatrix} 3.96\times10^{-8}f_{cl}\left\{\left(T_{cl}+273\right)^4-\left(T_r+273\right)^4\right\} \\ +f_{cl}h_c\left(T_{cl}-T_a\right) \end{bmatrix} \quad (17)$$

$$h = \begin{cases} 2.38\left|T_{cl}-T_a\right|^{0.25} \Leftarrow 2.38\left|T_{cl}-T_a\right|^{0.25} > 12.1\sqrt{V_{ar}} \\ 12.1\sqrt{V_{ar}} \Leftarrow 2.38\left|T_{cl}-T_a\right|^{0.25} < 12.1\sqrt{V_{ar}} \end{cases} \quad (18)$$

$$f_{cl} = \begin{cases} 1+1.29I_{cl} \Leftarrow I_{cl} \le 0.078m^2K/W \\ 1.05+0.645I_{cl} \Leftarrow I_{cl} > 0.078m^2K/W \end{cases} \quad (19)$$

2.4.2. Predicted Percentage Dissatisfied (PPD)

PPD is an index that shows the percentage of thermally dissatisfied people who feel too hot or cold. PPD parameter can be calculated as follow:

$$PPD = 100-95\exp(-0.03353PMV^4-0.2179PMV^2) \quad (20)$$

2.4.3. Local Thermal Discomfort

PMV and PPD indexes show overall thermal comfort. However, thermal discomfort can also be caused by unwanted heating or cooling in one body part. This is called local discomfort. Some local discomfort can be summarized as high vertical temperature difference, very cold or hot floor, high radiant temperature asymmetry.

2.4.4. ASHRAE 55 and ISO 7730 Thermal Comfort Criteria

The thermal comfort criteria that are evaluated by standards can be seen in Table 2.

Table 2. Thermal Comfort Criteria

Parameter	Limited Value
PMV	-0.5 < PMV < 0.5
PPD	PPD < 10

3. RESULTS AND DISCUSSION

When the water temperature is 20 °C, the radiant wall and ceiling's temperature contours are shown in Fig. 4. It has been observed that the temperature distribution is homogeneous for both cases. The average air temperature values of the room are very close to each other.

When the water temperature is 20 °C, the radiant wall and ceiling's velocity streamlines are shown in Fig. 5. It is seen that the streamlines are close to each other, and the

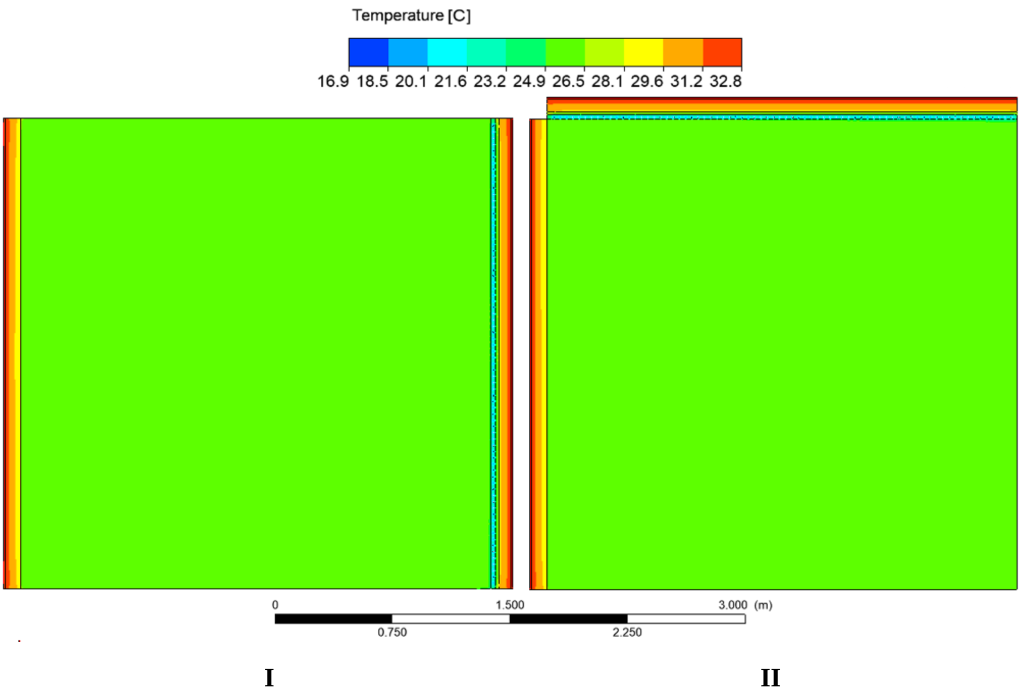
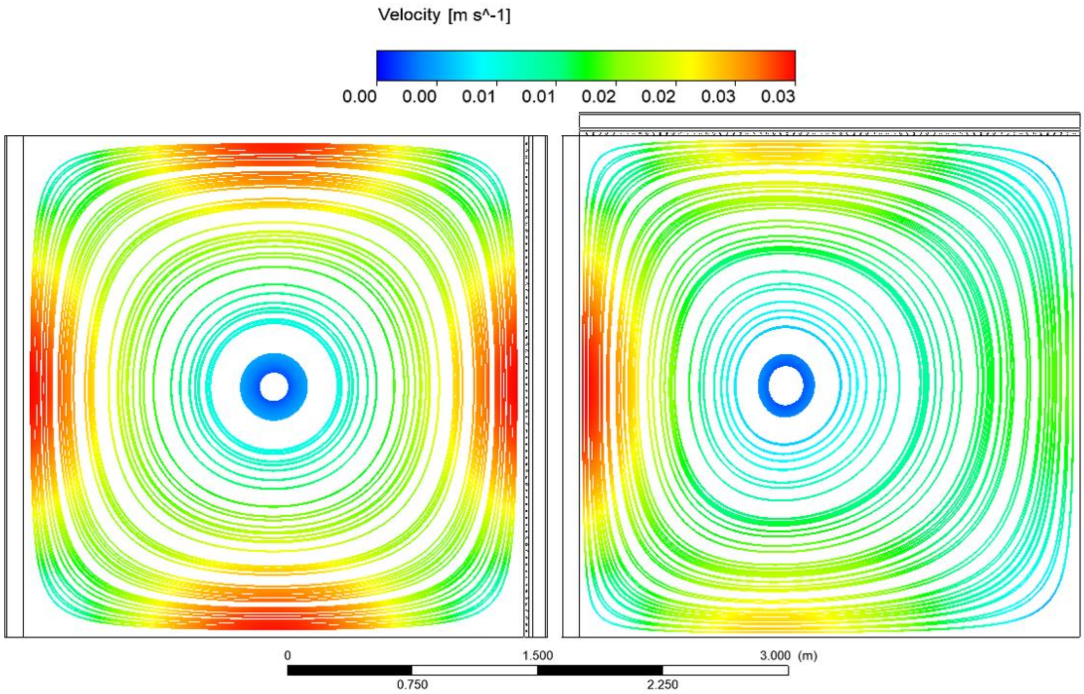


Fig. 4. Temperature contours of the wall panel (I) and the ceiling (II) cases for a water temperature of 20 °C

velocity increases in the areas that appear in red. In the case of cooling from the ceiling, there is a faster air movement in the hot wall area on the left side. In contrast, in other regions, the air movement is slower due to the adiabatic boundary conditions. In the radiant wall case, the hot air at the left side of the room goes upper, then it cools at the right side of the room with radiant cooling and goes down because of the buoyancy effect.

Fig. 6 and Fig. 7 show the variation of air temperature values with respect to the dimensionless width and height



I II

Fig. 5. Velocity streamlines of the wall panel (I) and the ceiling (II) cases for a water temperature of 20 °C

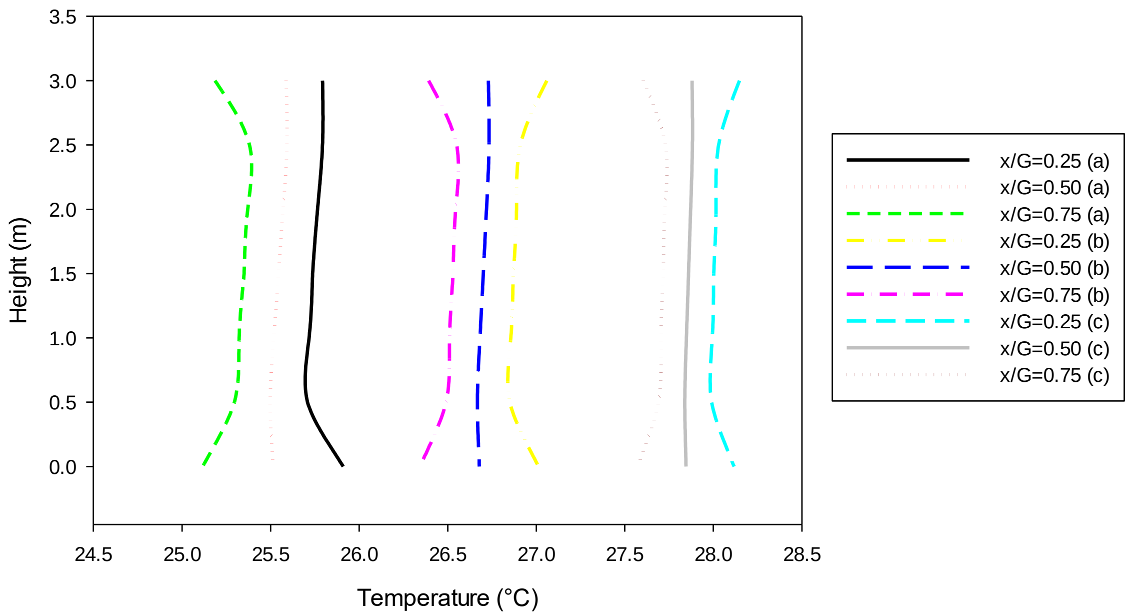


Fig. 6. Change of air temperature according to height when the water temperature is 20 °C (a), 22 °C (b) and 24 °C (c) for radiant wall

in case of radiant ceiling and wall cases, where x is the distance from the wall, G is the width of the room. The vertical temperature difference must not exceed 3 °C according to ASHRAE 55 standard to avoid local discomfort [37]. The vertical air temperature difference values for all cases were found to comply with this standard.

Fig. 8 and Fig. 9 show the variation of air velocity values with respect to the dimensionless width and height in case of radiant ceiling cases, where x is the distance from the wall, G is the width of the room. To ensure that the airflow rate does not create local discomfort, it must be a maximum of 0.18 m/s according to the ASHRAE 55 standard [37]. It has been found that for all cases the air velocity values comply with this standard.

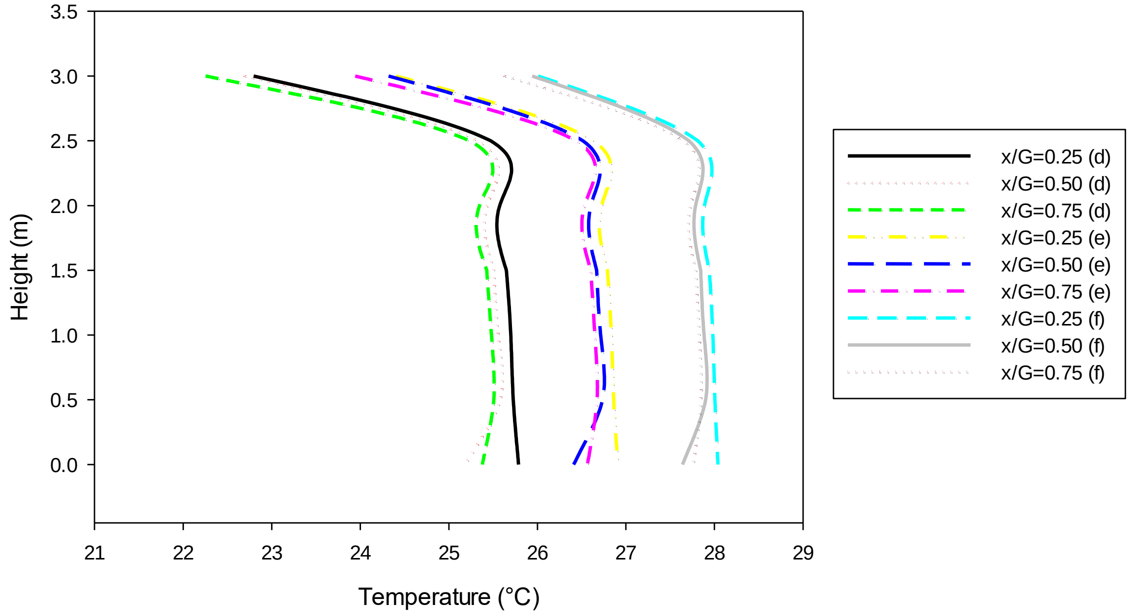


Fig. 7. Change of air temperature according to height when the water temperature is 20 °C (d), 22 °C (e) and 24 °C (f) for the radiant ceiling

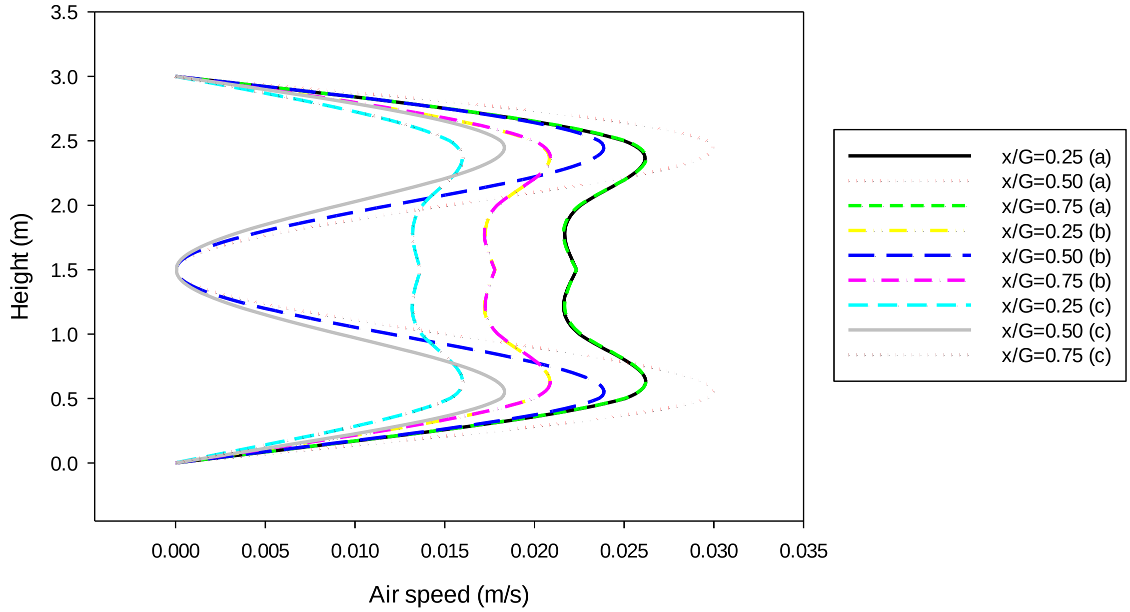


Fig. 8. Change of air speed according to height when the water temperature is 20 °C (a), 22 °C (b) and 24 °C (c) for radiant wall

As the water temperature increases, the temperature inside the room increases for all cases. When PMV-PPD values were calculated, clothing insulation values and metabolic rate were assumed as 0.50 clo and 1.2 met respectively. Relative humidity was assumed as 50% according to the ASHRAE 55 standard.

All properties for both cases are summarized in Table 3. Three different water temperatures, which are supposed to provide the thermal comfort standard of PMV values, were used in the calculations. When the water temperature is reduced, it is observed that the PMV value will increase in the negative direction and the PMV value

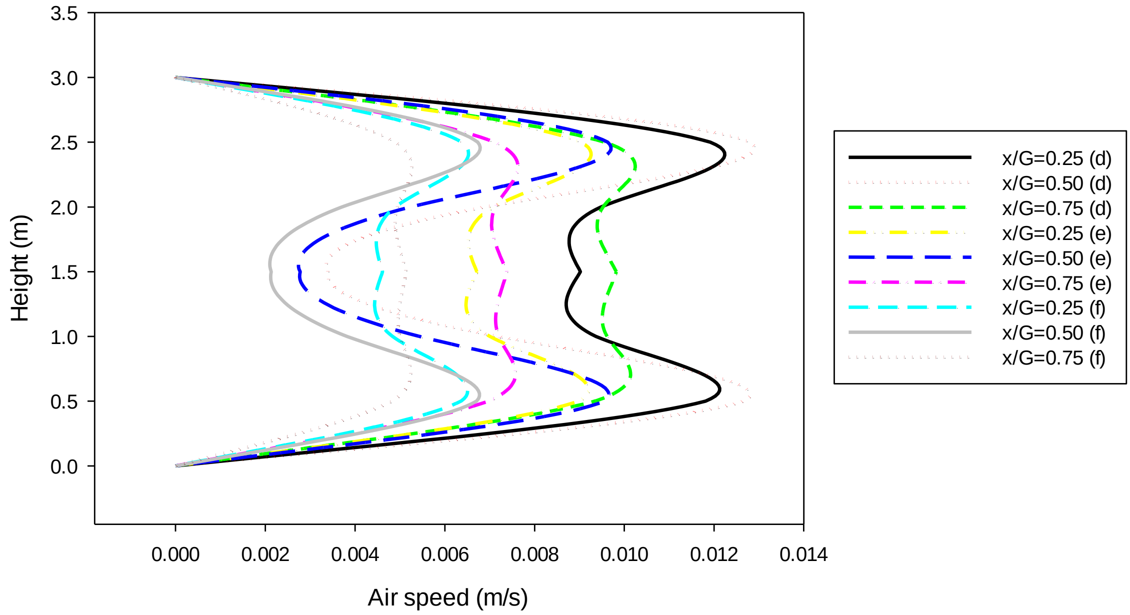


Fig. 9. Change of air speed according to height when the water temperature is 20 °C (d), 22 °C (e) and 24 °C (f) for radiant ceiling

Table 3. Comparison of all cases in terms of temperatures, speeds, heat fluxes and thermal comfort parameters

Location	**Top**			**Right**		
Cases [°C]	**20**	**22**	**24**	**20**	**22**	**24**
T_s [°C]	22.6	24.8	25.9	22.8	24.4	26
T_a [°C]	25.5	26.7	27.8	25.5	26.7	27.8
T_{op} [°C]	25.4	26.6	27.8	25.5	26.6	27.8
T_r [°C]	25.4	26.6	27.8	25.5	26.6	27.8
V_a [m/s]	6.78E-03	5.11E-03	3.57E-03	1.55E-02	1.23E-02	9.43E-03
q_r [W/m^2]	14.8	9.7	10.5	14.2	12	9.9
q_c [W/m^2]	13.6	8	8.2	12.8	10.1	7.7
q_{tot} [W/m^2]	28.4	17.7	18.7	27	22.1	17.6
PMV	0.27	0.64	1	0.29	0.64	1
PPD [%]	6.51	13.6	26.12	6.75	13.6	26.13

will increase in the positive direction when the water temperature is increased. When these three different temperature conditions are examined, it is understood that when the temperature of the water is 20 °C, it will have better thermal comfort than the other conditions. When the ceiling and wall combinations were examined, the cooling from the upper side provided better thermal comfort than the wall cooling.

3.1. Validation

The numerical solutions were compared to different studies. A full agreement has been reached. Fig. 10 presents convective heat flux as a function of the difference between air temperature and surface temperature. Fig. 11 illustrates radiant heat flux as a function of the difference between radiant temperature and surface temperature. Numerical results obtained in this study were compared to studies investigated by Chicote et al. [22] and Cholewa et al. [23], respectively. Almost the same heat flux values were provided for temperature differences. Fig. 12 demonstrates total heat flux as a function of the difference between operative temperature and surface temperature. Data obtained from numerical analysis in this study and also from Chicote et al. [22] and Cholewa et al. [23] are presented. Heat flux increases with growing temperature differences.

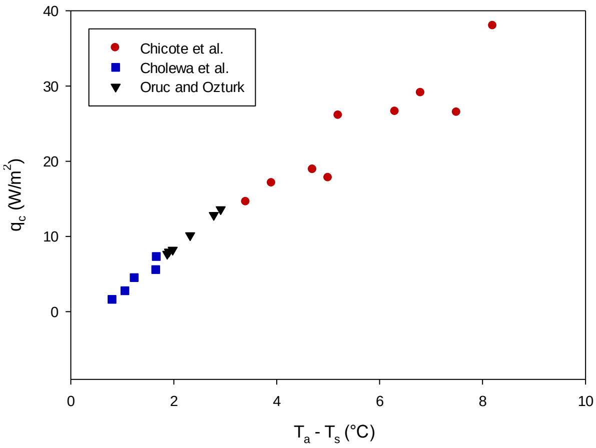


Fig. 10. Comparison of convective heat flux with literature

Fig. 12 demonstrates total heat flux as a function of the difference between operative temperature and surface temperature. Data obtained from numerical analysis in this study and also from Chicote et al. [22] and Cholewa et al. [23] are presented. Heat flux increases with growing temperature differences.

6. CONCLUSION

In this study, a numerical analysis was developed to determine thermal comfort and radiant cooling in a standard room. Thermal comfort conditions are provided according to PMV - PPD parameters when the defined water temperature is 20 °C. The vertical temperature difference for all cases did not exceed the 3 °C limit value specified in the ASHRAE 55 standard. The limit speed specified in the same standard is not exceeded. For these reasons, local discomforts were not observed in all three cases.

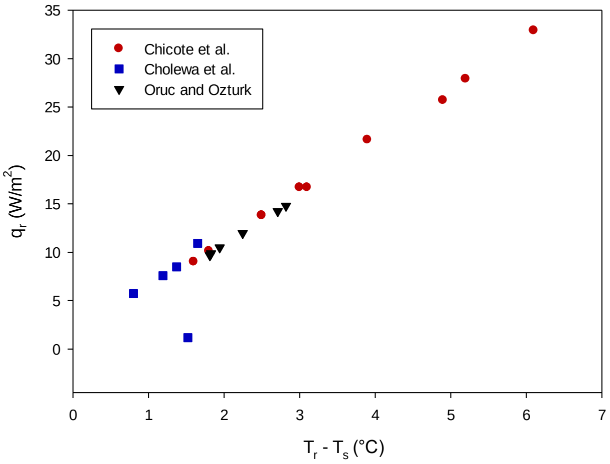


Fig. 11. Comparison of radiant heat flux with literature

Some real and experimental data have been obtained from the literature, it was seen that full compliance with the acquired data is achieved.

For all cases, the radiant heat transfer rates are dominant according to the convective heat transfer rates as it was expected. As different configurations of the radiant systems have more influence on the thermal comfort, further studies can be done according to the thermal comfort parameters.

DECLARATION OF ETHICAL STANDARDS

The author(s) of this article declare that the materials and methods used in this study do not require ethical committee permission and/or legal-special permission.

AUTHORS' CONTRIBUTIONS

Onur ORUÇ: Performed the simulations, did the literature research and wrote the manuscript.

Merve ÖZTÜRK: Performed the simulations, did the literature research and wrote the manuscript.

CONFLICT OF INTEREST

There is no conflict of interest in this study.

NOMENCLATURE

A	Area
α	Thermal diffusivity coefficient
β	Volumetric thermal expansion coefficient
$C_{1\varepsilon}$	k-ε turbulence model constant
$C_{2\varepsilon}$	k-ε turbulence model constant
$C_{3\varepsilon}$	k-ε turbulence model constant

C_μ	k-ε turbulence model dynamic viscosity constant
ε	Turbulence dissipation rate
f_{cl}	Body surface factor
G	Width of the room
G_b	The generation of turbulent kinetic energy due to buoyancy
G_k	The generation of turbulent kinetic energy due to the mean velocity gradients
Gr	Grasshoff number
g	Gravitational acceleration
h	Heat transfer coefficient
I_{cl}	Insulation of clothing
k	Turbulence kinetic energy, heat conduction coefficient
L	Characteristic length
M	Metabolic rate
μ	Dynamic viscosity
μ_t	Turbulence dynamic viscosity
ν	Kinematic viscosity
p	Pressure
p_a	Water vapor partial pressure
Pr	Prandtl number
Ra	Rayleigh number
ρ	Density
σ	Stefan-Boltzmann constant
σ_k, σ_ε	Prandtl numbers for k-ε
T	Temperature
T_a	Air temperature
T_b	Surface temperature of the pipes
T_{cl}	Clothes surface temperature
T_r	Mean radiant temperature
T_0	Working temperature
u, v, w	Average velocity components of fluid
u_i	Instant velocity
V_{ar}	Relative air velocity
W	Effective mechanical power
x, y, z	Cartesian coordinates
Y_M	The contribution of the fluctuating dilatation in compressible turbulence to the overall dissipation rate

ABBREVIATIONS LIST

ASHRAE	American Society of Heating Refrigerating and Air-Conditioning Engineers
AUST	Area-weighted temperature of all indoor surfaces
CFD	Computational Fluid Dynamics
DO	Discrete Ordinates
ISO	International Standarts Organization
PMV	Predicted Mean Vote
PPD	Predicted Percentage of Dissatisfied

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