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EXPERIMENTAL EXAMINATION OF THERMAL COMFORT PERFORMANCE OF A RADIANT WALL PANEL SYSTEM: COMPARISON BETWEEN DIFFERENT HEATING WALL CONFIGURATIONS

Zafer GEMİCİ*
*Mir Arastirma ve Gelistirme A.S., 34220, Istanbul, Turkey, zafer@mirarge.com

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Abstract: Radiant heating is a proven technology in space heating which offers many advantages to thermal comfort and energy conservation. For this reason, the usage of these systems is increasing from day to day and it has been widely investigated by the researchers. Different panel locations were examined experimentally with through the usage of a real size test chamber in accordance with pre-determined standards in this study. As a result of the research, it was become apparent that thermal comfort aspects in a room vary with the variation in placement configurations of the radiant panels. The goal was to estimate how thermal comfort is affected when varying inlet water temperatures are applied to different radiant wall heating panels' placement configurations. Vertical air temperature differences and mean radiant temperatures were investigated. Average vertical air temperatures of the locations 0.1 m and 1.7 m were found 0.14 °C, 1.11 °C and 0.73 °C respectively. The results confirm that the mounting radiant wall panels to different walls affect both thermal comfort and heating performance. Based on the experiments, the first case which is located on an exterior wall containing a window produces better results than the others.
Keywords: ISO 7730, Radiant heating system, Radiant panel arrangement, Thermal comfort.

BİR RADYANT DUVAR PANEL SİSTEMİNİN TERMAL KOMFOR PERFORMANSININ DENEYSEL OLARAK İNCELENMESİ: FARKLI ISITMA DUVAR KONFİGÜRASYONLARININ KARŞILAŞTIRILMASI

Özet: Isıl konfor ve enerji tasarrufu açısından birçok avantaj sağlayan radyan ısıtma, bir mekânın ısıtılmasında kendini kanıtlamış bir teknolojidir. Bu nedenle, bu sistemlerin kullanımı her geçen gün artmakta ve araştırmacılar tarafından daha fazla incelenmektedir. Bu çalışmada, gerçek ölçekli bir test sistemi kullanılarak farklı panel yerleşimleri deneysel olarak ilgili standartlara göre incelenmiştir. Çalışmanın sonucuna göre; radyan panellerin yerleşim konfigürasyonunun değişmesiyle ısıl konforun da değiştiği açığa kavuşmuştur. Amaç, farklı duvar tipi radyan ısıtma paneli yerleşim konfigürasyonlarına değişik giriş suyu sıcaklıkları uygulandığı zaman ısıl konforun nasıl etkilendiğini anlamaktır. Dikey hava sıcaklığı farkları ve ortalama radyan sıcaklıklar incelenmiştir. 0,1 m ve 1,7 m deki ortalama hava sıcaklığı farkları üç farklı yerleşim durumu için sırasıyla 0,14 °C, 1,11 °C ve 0,73 °C olarak bulunmuştur. Sonuçlar radyan panellerin farklı duvarlara monte edilmesinin hem ısıl konforu hem de ısıtma performansını etkilediğini göstermiştir. Deneylere göre; panellerin üzerinde cam bulunan dış duvara yerleştirildiği ilk durum diğer yerleşim durumlarından daha iyi performans göstermektedir.
Anahtar Kelimeler: ISO 7730, Radyan ısıtma sistemi, Panel yerleşimi, Isıl konfor.

NOMENCLATURE

A	Area [m^2]
c_p	Specific heat at constant pressure [J.kg^{-1}.K^{-1}]
f_{cl}	Clothing area factor
$F_{\varepsilon_{s-j}}$	Radiation interchange factor
F_{s-j}	View factor between radiant surface and j-surface
h_c	Convective heat transfer coefficient [W.m^{-2}.K^{-1}]
h_r	Radiation heat transfer coefficient [W.m^{-2}.K^{-1}]
h_{tot}	Total heat transfer coefficient [W.m^{-2}.K^{-1}]
$\dot{m}$	Mass flow rate [kg.m^{-3}]
M	Metabolic rate [W.m^{-2}]
p_a	Water vapor pressure in the air [Pa]
PMV	Predicted Mean Vote
PD	Percentage of dissatisfied [%]
PPD	Predicted percentage of dissatisfied [%]
Q_{total}	Total heat transfer [W]
Q_{loss}	Backward heat transfer [W]
q_c	Convective heat flux [W.m^{-2}]
q_{net}	Net heat flux [W.m^{-2}]
q_r	Radiation heat flux [W.m^{-2}]
T_4	Outer surface temperature of the facade [°C]
T_3	Inner surface temperature of the facade [°C]
T_a	Air temperature [°C]
T_{cl}	Clothes surface temperature [°C]
T_i	Supply water temperature [°C]
T_j	j-surface temperature [°C]
T_{mrt}	Mean radiant temperature [°C]
T_o	Return water temperature [°C]
T_{op}	Operative temperature [°C]
T_r	Mean radiant temperature [°C]
T_s	Surface temperature [°C]

T_w	Water temperature [°C]
U	Coefficient of thermal transmittance of surfaces [$W.m^{-2}.K^{-1}$]
w	Uncertainty
W	External work [$W.m^{-2}$]
WBGT	Wet bulb globe temperature
x	Independent variable
Greek Symbols	
ε	Emissivity
σ	Stefan-Boltzmann constant [$W.m^{-2}.K^{-4}$]

INTRODUCTION

Thermal comfort is one of the most important elements that have a direct effect on people's quality of life and wellness. On the other hand, thermal comfort also affects people's performance (Olesen, 2008). Recently, hydronic radiant heating and cooling panels have become a common solution in heating and cooling operations in buildings based on the advantages they offer namely thermal comfort and high energy efficiency.

Panel heating and cooling systems use temperature-controlled indoor surfaces on the floor, walls and/or ceiling; surface temperature is maintained through water circulation on a circuit embedded in the panel. A temperature controlled surface is referred to as "radiant panel" if 50 % or more of the total heat transfer from the surface occurs via radiation (ASHRAE, 2008). In this respect, according to Miriel *et al.* (2002) the radiant component includes around 66 % of the total heat transfer in cooling operation and 80 % in heating operation. Radiant heating systems have been a well-known HVAC solution in the provision of thermal comfort because of their radiation capability by conditioning the room surfaces instead of conditioning the air by forced convection. Radiant systems consume less quantities of energy and provide much more comfort. It is a proven technology which carries out the requirements which are explained in related standards (ANSI/ASHRAE, 2013; EN ISO 7730, 1994).

The main parameter used in thermal comfort analysis of radiant systems, is thermal capacity which varies with design parameters such as; depth of the embedded pipe, piping type, pipe spacing, fluid supply temperature, insulation material and thickness etc. (Cholewa *et al*., 2013). Despite these aforementioned differences, heat transfer process between radiant surface and room is subject to the same physical phenomena for every type of system, providing a given heating capacity as a function of its surface temperature and the temperature characteristics of the indoor environment. According to European standard EN 15377-1 (2008), as a previous step to the system design and dimensioning process, it is therefore possible to establish a basic characteristic curve for heating, which is independent of the type of system and applicable to all surfaces. However, for more detailed and exact thermal analyses convective heat transfer and radiative heat transfer values remain necessary. In the literature, similar general curves were estimated through experimentation by several researchers (Andrés-Chicote *et al*., 2012; Fonseca, 2011; Tian *et al*., 2012; Zhang *et al*., 2013; Causone *et al*., 2009; Okamoto *et al*., 2010).

Radiant systems provide better thermal comfort levels than other HVAC systems by means of using lower supply temperatures for heating and higher supply temperatures for cooling. This allows small vertical temperature gradient, much more stable air and hence increased comfort for people (Saelens *et al*., 2011). Thus, thermal comfort should be considered in the design of the radiant systems. Djuric N. *et al.* (2007) performed a research on the optimization of parameters effecting investment and energy consumption cost as well as thermal comfort. The study that has been done by Ghaddar *et al.* (2006) is the effect of the heater's position on energy consumption in a room while providing the same level of thermal comfort. Myhren *et al.* (2008) conducted a research study as to understand the effect of different systems and their positions on the indoor climate.

Nagano and Mochida (2004) realized an experimental study by using a test room containing cooling ceilings. They investigated thermal comfort and concluded that the mean radiant temperature should be used for a supine human. Kitagawa *et al.* (1999) studied on radiant cooling system in a test chamber and determined the thermal comfort of subjects under different humidity and air movement conditions. Catalina *et al.* (2009) analysed the indoor thermal comfort using PMV utilising the results obtained from experimental and CFD studies. Memon *et al.* (2008) studied on thermal comfort for Pakistan as a subtropical region. According to their results, to feel thermally comfortable in this area, the operative temperature should be 29 - 31 °C.

One of the main reasons of thermal discomfort in an environment is thermal stratification in between the head level and the ankle level. According to the references ANSI/ASHRAE (2013) and EN ISO 7730 (1994), the allowable differences in air temperature from the ankle level to the head level should be lower than 3 °C.

In the following lines, there are some highlighted results which are given for better understanding of the level of the vertical temperature differences results of other studies related to the radiant systems.

According to Catalina et. al., nominal vertical air temperature gradients between 0.4 m and 2.1 m heights were found to be around 0.71 - 0.77 °C /m (Catalina *et al*., 2009). In Imanari's experimental study, vertical temperature difference varied between 0.3 °C and 2 °C for different radiant surface temperatures (Imanari *et al*., 1999). Maximal vertical gradient of the air temperature in the test chamber, 1.5 °C in heating mode and 0.5 °C in cooling mode was observed from the experiments by Foncesa (2011). According to the numerical studies of Sevilgen G., and Kilic, M. (2011), the temperature difference between the head and the foot region was similar, about 2 - 3 °C for all radiant wall heating cases.

As seen from the literature, the vertical air temperature differences in radiant systems varied between 0.3 °C and 3 °C for different conditions which are remained in the comfort zone according to the ISO 7730 (EN ISO 7730, 1994).

Up to now, as previously mentioned, some papers were submitted regarding radiant heating systems and their effect on thermal comfort. However, these studies did not cover radiant wall panels and not enough studies have been done in order to determine how thermal comfort is affected when changing the location of the radiant panels in the conditioned space. The reliance on the location difference of radiant wall heating in a room was studied to garner the knowledge necessary as to how thermal comfort is affected. In the previously published study of the author, heat transfer characteristics of different located panels have been examined (Koca *et al*., 2014). In this study, with the same testing infrastructures and the same arrangements, thermal comfort has been investigated by using gathered thermal comfort measurements.

It is claimed that wall configuration affects the vertical temperature differences remarkably in this study. So the experimental results of the paper may provide a substantially contribution to the literature with respect to the following goals;

- How does the radiant wall configuration affect the local discomfort?
- Additional experimental evidence for comfort performance of radiant wall systems, beyond the state of art in the existing literature.

For this purpose:

The thermal comfort rates which were investigated for different thermal conditions and heating panel configurations had been taken from the test room which was prepared with advanced test chamber measuring devices, by providing all local and general thermal comfort prerequisites in preparation of ambient conditions.

If any part of human body is exposed different amount of cooling or heating. Thermal dissatisfaction comes into existence and called as local discomfort. The level of discomfort under warm and cold conditions is represented by PMV and PPD values, respectively (EN ISO 7730, 1994). Vertical air temperature profile and percentage of discomfort caused by vertical temperature difference are investigated for the cases presented.

BASIC THEORY

The heat transfer amount between the panels and the test room is dependent on the surface temperatures of the walls and the air temperature of the internal environment. Total heating capacity of the wall panel is calculated by using circulated water mass flow rate and water temperature difference between the inlet and the outlet of the panels as shown in Eq. (1). $\mathrm{Q_{loss}}$ is the backward heat transfer through the radiant wall panel to the wall facade and calculated by using measured surface temperatures of wall facade layers via Eq. (2). The heat flux between the wall panel and the room is calculated subtracting from the total heat flux the backward heat transfer toward the facade of the room.

$$\mathrm{Q_{total}} = \dot{m} c_p (T_i - T_o)\ (\mathrm{W}) \tag{1}$$

$$\mathrm{Q_{loss}} = UA(T_3 - T_4)\ (\mathrm{W}) \tag{2}$$

$$q_{net} = \frac{\mathrm{Q_{total}} - \mathrm{Q_{loss}}}{A} \quad (\mathrm{W/m^2}) \tag{3}$$

The heat flux between the wall surface and the room is composed of convection and radiation heat transfer. The summation of the radiative and convective heat transfer values gives the net heat transfer amount as shown in Eq. (4).

$$q_{net} = q_r + q_c\ (\mathrm{W/m^2}) \tag{4}$$

In order to calculate the radiative heat transfer rate and the heat transfer coefficient (h_r) from Eq. (5), the view factors must be calculated by using Eq. (6).

$$q_r = \sigma \sum_{j=1}^{n} F_{\varepsilon_{s-j}} \left(T_s^4 - T_j^4\right)\ (\mathrm{W/m^2}) \tag{5}$$

$$F_{\varepsilon_{s-j}} = \frac{1}{\left[\frac{1-\varepsilon_s}{\varepsilon_s}\right] + \left(\frac{1}{F_{s-j}}\right) + \left(\frac{A_s}{A_j}\right)\left[\frac{1-\varepsilon_j}{\varepsilon_j}\right]} \tag{6}$$

By using thermal camera and thermocouples, the emissivity values of the walls were determined. For this aim, after sensing the surface temperature with thermocouples, emissivity setup value of the thermal camera was adjusted to a value in order to equalize thermal camera's measured temperature value with the thermocouple's one (Cholewa *et al.*, 2013).

And hence, the convective heat transfer could be calculated with Eq. (7), and the heat transfer coefficient (h_c) as well.

$$q_c = q_{net} - q_r\ (\mathrm{W/m^2}) \tag{7}$$

The detailed explanations, analysis and test results about the heat transfer of the panels can be seen from the previous study of the author, which was about determination of the heat transfer coefficient of the panels located in different positions (Koca *et al.*, 2014). In this study, by using the same test system, the effect of the panel locations on the comfort has been analysed in detail.

Thermal Comfort

General thermal comfort is mainly related to PMV-PPD index which can be expressed mathematically, and operative temperature which is occupant's thermal sensation temperature of his body. Fanger's PMV model (Fanger, 1972) which is very common in calculating general thermal comfort which depends on thermoregulation and heat balance theories. PMV comfort variables are metabolic rate, clothing insulation, ambient air temperature, mean radiant temperature, relative humidity and air velocity. Eq. (8) expresses PMV value (EN ISO 7730, 1994).

$$PMV = (0.303e^{-0.036M} + 0.028) \cdot A \tag{8}$$

Where

$$\begin{aligned} A = {} & (M - W) - 3.05 \cdot 10^{-3} \\ & \cdot \{5733 - 6.99 \cdot (M - W) - p_a\} - 0.42 \\ & \cdot \{(M - W) - 58.15\} - 1.7 \cdot 10^{-5} \cdot M \cdot (5867 - p_a) \\ & - 0.0014 \cdot M \cdot (34 - T_a) - 3.96 \cdot 10^{-8} \cdot f_{cl} \\ & \cdot \{(T_{cl} + 273)^4 - (T_r + 273)^4\} - f_{cl} \cdot h \cdot (T_{cl} - T_a) \end{aligned}$$

Eq. (9) expresses PPD depending PMV index value (EN ISO 7730, 1994).

$$PPD = 100 - 95 \cdot e^{(-0.03353 \cdot PMV^4 - 0.2179 \cdot PMV^2)} \quad (9)$$

Ambient air temperature and operative temperature are not equal. The operative temperature is affected by surfaces' and objects' temperatures in an indoor environment. According to Olesen (2008), the operative temperature is a practical parameter for thermal comfort analysis. Additionally, EN Standard 12831 (BS EN 12831, 2003) suggests that operative temperature can be used for heat load calculations, as well. In this study, the operative temperature was measured experimentally by using special test equipment described in the following section.

EXPERIMENTAL SETUP

The Arrangement of the Test Chamber

A real scale model test room was used to test the specimens under varied adjustable climate conditions. The thermal capabilities of the test room are listed in Table 1. The test facility consists of four different zones which are ceiling (zone-1), facade (zone-2), neighbour room (zone-3), floor (zone-4) and test zone (zone-5) represented in Figure 1. The test zones' dimensions are 6 m in depth, 4 m in width and 3 m in height. The insulation properties of the room walls were calculated according to the standard TS 825 (2008). As results of these calculations, the U values of the walls are given in Table 2. The test room contains of a window and a door whose transmittance values are 2.2 W/m^{2}K and 2.6 W/m^{2}K respectively.

Table 1. Capabilities of the test room

	Ceiling (Zone 1)	Facade Room (Zone 2)	Neighbour Room (Zone 3)	Floor (Zone 4)
Temperature Range	-10 ˚C / +40 ˚C	-10 ˚C / +40 ˚C	+0 ˚C / +30 ˚C	+0 ˚C / +30 ˚C
Temperature Tolerance	± 0.5 ˚C	± 0.5 ˚C	± 0.5 ˚C	± 0.5 ˚C
Humidity Range	n/a	%35 / %85 RH	n/a	n/a
Humidity Control Steps	n/a	%1	n/a	n/a
Humidity Tolerance	n/a	± % 0.5 RH	n/a	n/a
Air Velocity	n/a	0.5 – 5 m/s	n/a	n/a

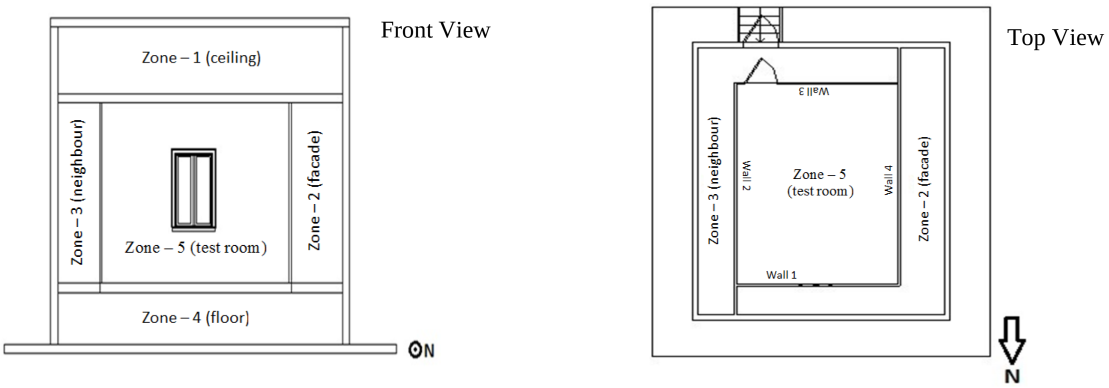


Figure 1. General view of the climatic test room

Table 2. Thermal transmittance coefficients according to TS825 (2008)

Surfaces	U (W/m^{2}K)
Ceiling	0.3
Floor	0.4
Wall 1	0.4
Wall 2	0.8
Wall 3	0.8
Wall 4	0.4

Hydraolic Circuit and Radiant Panel

A versatile water circulation system was used to supply different heat amount to the testing room by means of the radiant panels by adjusting any needed temperature and mass flow rate of the circulated water. The detailed circuit of the hydraulic system can be seen in Figure 2. The temperature condition of the circulated water was adjusted with a chiller and two electric heaters. By using four and three way valves the inlet temperature and the mass flow rate of the water can be supplied automatically. To measure volumetric flow rate, an electromagnetic flow meter was used.

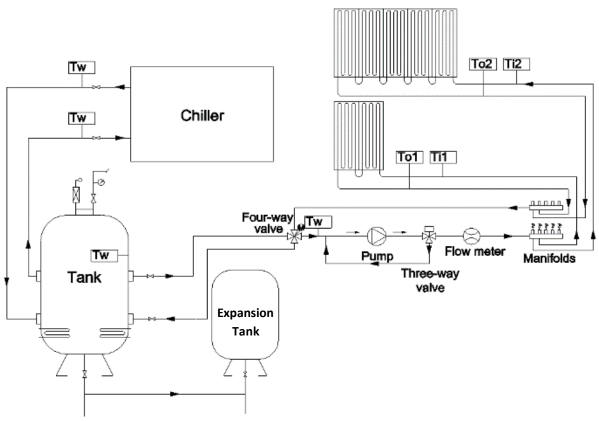


Figure 2. Hydraulic system (Koca *et al*., 2014)

The cross-section of the panels used in the tests is shown in Figure 3. The layers of the panels are insulation, aluminium foil, serpentine and drywall. The thicknesses of EPS (expanded polystyrene) insulation, aluminium foil and drywall are 30 mm, 0.1 mm and 15 mm respectively. The serpentine heating pipe is placed into the grooves on the drywall. The aluminium foil is wrapped on the insulation including the grooves. Then, the PEX (cross-linked polyethylene) pipes are located into this foil coated grooves. The outer diameter of the pipe is 10.1 mm while the spacing is 150 mm.

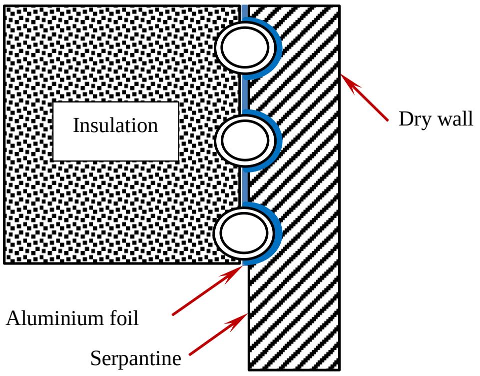


Figure 3. Cross-section of the panel

The Measurements Equipment

To measure the air temperatures in different locations in the test room, eight K-type thermocouples were used. The locations of these thermocouples are shown in Figure 4. Two different locations (location a and location b) and four sensors for each location in vertical direction were chosen to get more accurate mean air temperature. When measuring air temperature, the sensors were shielded to ensure that only air temperature is sensed rather than both air and radiant temperature together.

The indoor air humidity was sensed by two relative humidity transducers located in the same place with the thermocouples, which has ±3.5 % of uncertainty. It can be seen from Figure 4 that the surface temperature sensors (K type thermocouples) were located in the middle of the related surface. In addition to all thermocouples, to see the temperature gradients through the facade wall, four thermocouples were located as shown in the cross-section (-A- detail) drawn in the Figure 4. The height of these thermocouples' location from the floor is 1.5 m. Finally, the last thermocouple was located in the middle of the window. During the tests, to see the thermal map of the surfaces, a thermal camera was used.

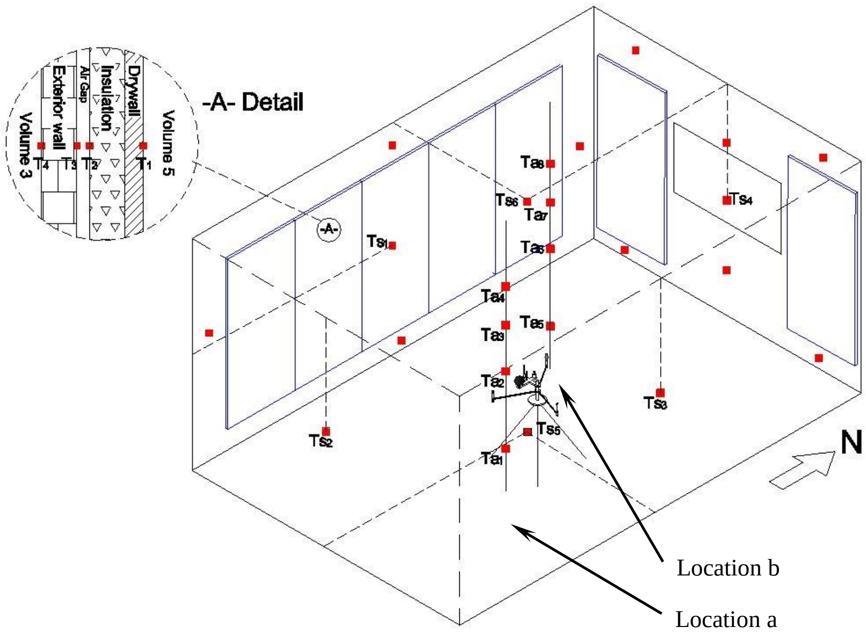


Figure 4. Measurement equipment in the test room

Four PT100 sensors were used to measure the inlet and outlet water temperatures of the two hydronic circuits, while electromagnetic flow meter, with a relative uncertainty of ±0.5 %, was used to regulate the water flow.

Some of thermal comfort parameters were measured with thermal comfort measuring equipment which has four module slots; each module includes three input sockets. Operative temperature, air velocity, radiant temperature asymmetry, air temperature, humidity, surface temperature, WBGT and dry heat loss were measured with transducers connected to input sockets. These instruments were used to evaluate thermal comfort conditions attached to a tripod located in the centre of the room at 1.1 m height as shown in Figure 5.

Figure 5. Thermal comfort measurement tool

To collect measured data and set the needed values and control the system in real time, a PXI connected to PC running LabVIEW software on was used during the tests.

EXPERIMENTS

As mentioned in the preceding lines, the aim of this study is to understand the effect of the location of the radiant panels on thermal comfort. For this aim, a real scale testing room and the same configuration as the previous study (Koca *et al*., 2014) were used in the analysis.

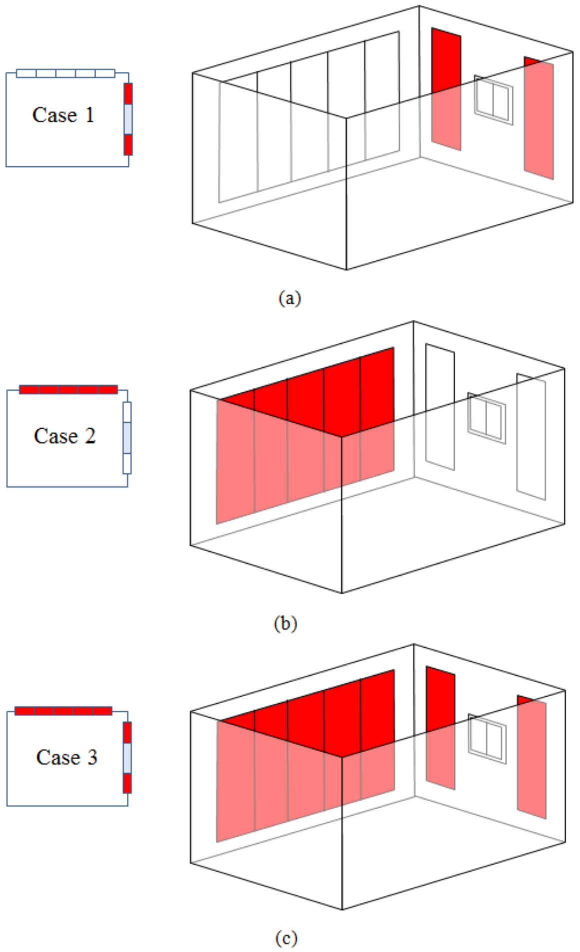


Figure 6. Different arrangements of the wall panels (a) Case 1 (b) Case 2 (c) Case 3

Seven heating wall type panels which are in 2.2 m x 1 m dimensions were used in the tests. Three different arrangements of these panels shown in Figure 6 were examined during the tests.

In all tests, the water flow rate fixed at a value of 0.08 m^3/h while the water inlet temperature was changed by 2 °C from 30 °C to 42 °C. During these tests, the facade temperature was fixed at 3 °C and the neighbour room and floor temperature was fixed at 20 °C as to get a constant heating load for all these three cases.

All the generated data collected with 1 minute interval and the records started after getting steady state conditions for testing room air temperature which took around 4 or 5 hours.

RESULTS

Heat Flux and Uncertainly Analysis

Collected data of all three cases was analysed and calculated for heat transfer characterisation of the panels under different location scenarios in the previous study (Koca *et al*., 2014). The needed values for comfort analysis were shown in Table 3. The thermal capacities of the panels depending on the difference of operational temperature and surface temperature for winter conditions can be seen from Figure 7 (Koca *et al*., 2014).

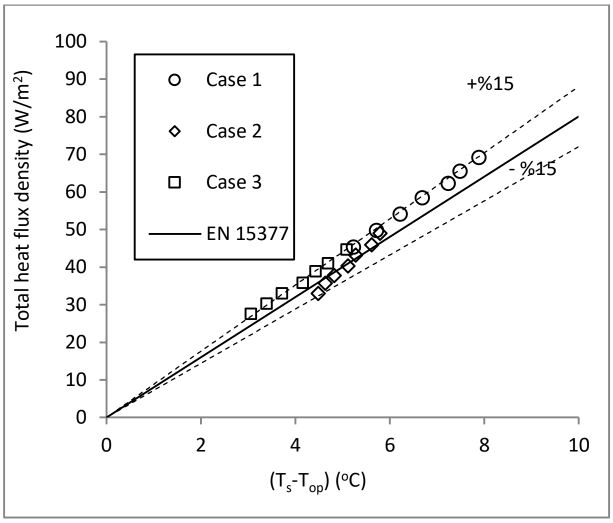


Figure 7. Total heating capacity for a radiant heated wall panel system (Koca *et al*., 2014)

According to the analysis, the most suitable formula with EN 15377 (EN 15377-1, 2008) for the heating capacity of the panels can be written as follows (Koca *et al*., 2014):

$$q = 8\left(T_s - T_{op}\right) \tag{10}$$

In the calculated and measured properties, uncertainty analysis was calculated in detail. It is well known that there are a lot of error sources in the experiments (ISO, 1995). By using Eq. (11), total uncertainty of any calculated parameters can be found.

$$w_R = \pm\left[\left(\frac{\partial R}{\partial x_1}w_1\right)^2 + \left(\frac{\partial R}{\partial x_2}w_2\right)^2 + \cdots + \left(\frac{\partial R}{\partial x_n}w_n\right)^2\right]^{1/2} \tag{11}$$

While R is a calculated value such as total heating capacity, x is one of the independent variable of it, such as temperature and w is the uncertainty of this independent variable.

All the uncertainty analysis results calculated by using Eq. (11) could be seen in the previous study (Koca *et al*., 2014).

Table 3. Measured and calculated parameters for a heated radiant wall (Koca *et al*., 2014)

Cases	Test Descriptor	1	2	3	4	5	6	7
Case 1	T_w (°C)	31.16	32.96	34.65	36.48	38.17	40.02	41.59
	T_s (°C)	22.31	23.42	24.42	25.42	26.42	27.29	28.27
	T_a (°C)	17.05	17.65	18.15	18.67	19.13	19.76	20.34
	T_{op} (°C)	17.08	17.7	18.2	18.73	19.18	19.8	20.39
	q_r (W/m^2)	28.05	31.14	34.13	37.16	39.87	42.27	44.94
	q_c (W/m^2)	16.16	17.81	19.56	21.22	23.23	24.14	25.57
	q_{net} (W/m^2)	44.22	48.95	53.69	58.38	63.10	66.41	70.51
Case 2	T_w (°C)	31.89	33.52	35.08	36.81	38.41	40.24	41.83
	T_s (°C)	23.35	24.58	25.01	26.3	27.21	28.03	29.03
	T_a (°C)	19.15	20.06	20.42	21.1	21.72	22.13	22.8
	T_{op} (°C)	19.31	20.21	20.31	21.21	21.83	22.29	22.94
	q_r (W/m^2)	21.21	23.50	25.59	27.69	29.93	31.91	34.26
	q_c (W/m^2)	9.47	9.48	10.28	10.95	12.14	12.97	14.07
	q_{net} (W/m^2)	30.68	32.98	35.86	38.64	42.06	44.89	48.33
Case 3	T_w (°C)	30.91	32.76	34.67	36.41	38.42	40.2	42.18
	T_s (°C)	24.8	25.97	27.2	28.02	29.63	30.26	31.84
	T_a (°C)	21.59	22.39	23.29	23.66	24.95	25.34	26.51
	T_{op} (°C)	21.74	22.58	23.48	23.86	25.2	25.57	26.75
	q_r (W/m^2)	16.52	18.47	20.89	23.19	25.76	28.10	29.84
	q_c (W/m^2)	8.13	8.32	9.33	10.33	11.60	12.54	13.29
	q_{net} (W/m^2)	24.65	26.79	30.22	33.52	37.36	40.63	43.13

Thermal Comfort

The desired thermal environment for the real sized test room is category B as defined in ISO 7730 (EN ISO 7730, 1994). Table 4 shows the recommended limits of ISO 7730 for category B.

PMV and PPD

Figure 8 shows the PMV and PPD for all three cases. For the calculations, metabolic rate and clothing value are assumed 1.2 and 0.8 respectively As shown in Figure 8, thermal comfort criteria for category B was obtained using different supply water temperatures in different cases. Since different amount of serially combined panels used in different cases, air temperature in steady state conditions varies with water temperature. In Case 1 and Case 2, thermal comfort criteria could not be obtained for supply water temperatures under 40 °C and 32 °C respectively, since head load was not handled. In Case 3 which was conditioned with 7 panels, when water temperature was above 36 °C, thermal comfort conditions were not provided due to the high air temperatures.

It is not possible to determine which case has better results in terms of thermal comfort with the PMV-PPD values; since the temperature of the indoor air in the experiment varies in all cases. In previous sections, it was mentioned that operative temperature is used as reference temperature. Thus, operative temperatures at which thermal comfort is achieved for category B (PMV = - 0.5 for heating) is compared. As shown in Figure 9, Case 1 has a better result than others. PMV = - 0.5 is reached when operative temperatures were at 19.37 °C, 19.65 °C and 19.86 °C respectively.

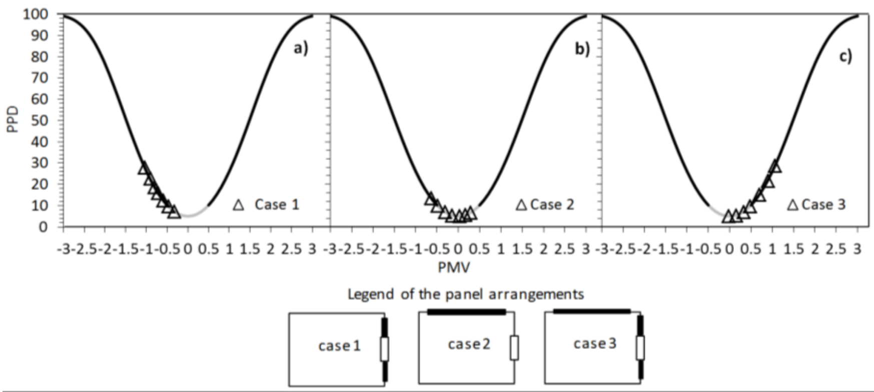


Figure 8: Comparison of PMV values for different panel configurations a) Case1, b) Case 2, c) Case3

Table 4. Recommended factors of ISO 7730 for category B (EN ISO 7730, 1994)

PPD	PMV	Operative temperature (°C)	Vertical temperature difference (°C)	PD% Caused by warm and cold wall	DR %
<10	-0.5<PMV<0.5	20-24	<5	<10	<20

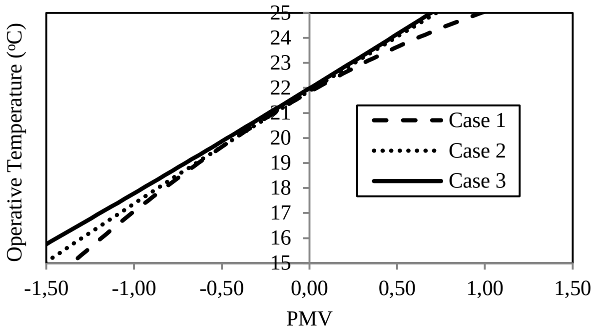
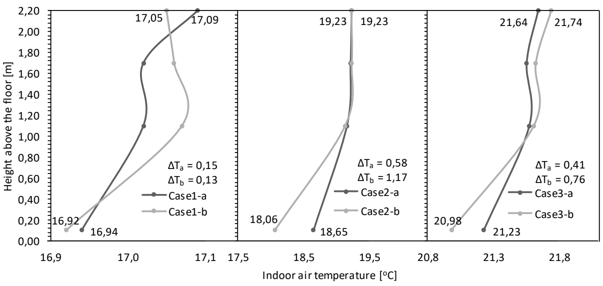


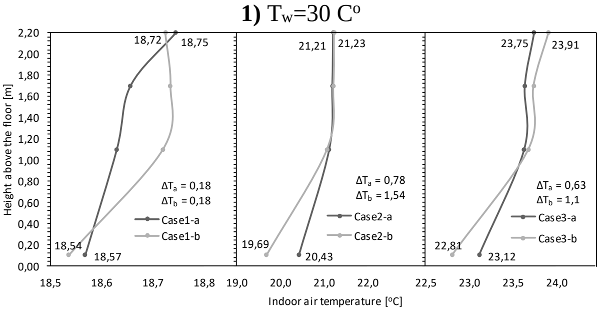
Figure 9. Comparison of operative temperature based on PMV values

Vertical Air Temperature Profile and % PD

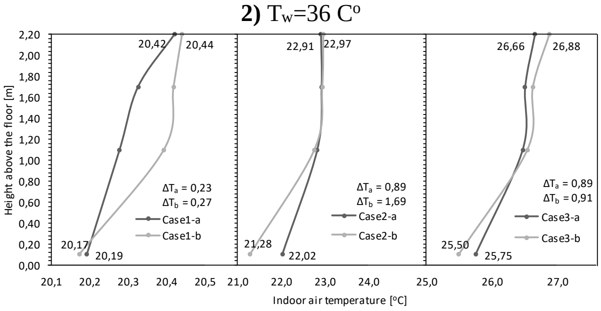
A high vertical air temperature difference can cause discomfort. In Figure 10 vertical indoor temperature distributions is presented for three different water supply temperatures. The figures 'a' and 'b' indicate "Location a" and "Location b" shown in Figure 4 that temperatures were measured. The vertical air temperature difference between 0.1 m and 1.7 m is less than 1.5 °C. In the experiments, the average temperature differences in the vertical direction are obtained and are 0.14 °C, 1.11 °C and 0.73 °C respectively. These results show that Case 1 created a smaller vertical temperature difference than Case 2 and Case 3. The obtained experimental data is more than satisfied, the maximum allowable temperature difference between head and feet is 3 °C, regardless of the operative temperature.



1) T_W=30 C°



2) T_W=36 C°



3) T_W=42 ºC

Figure 10. Indoor air temperature distribution in vertical direction

PD % is the percentage of people dissatisfied by the vertical temperature difference which is estimated by a function given in ISO 7730. PD can be determined by using Eq. (12).

$$\mathrm{PD} = \frac{100}{1+\exp(5.76-0.856\,\Delta \mathrm{T_{a,v}})} \tag{12}$$

In Figure 11, percentage dissatisfied is introduced for three cases. It can be easily recognised that Case 1 offers a better temperature distribution. The vertical temperature difference increases with the increase of the supply water temperature and air temperature. In Case 1, radiant wall panels reduce the air temperature difference caused by the window. However, in Case 3 which is also containing panels on window side, vertical air temperature difference was not satisfying as well as Case1 because of the high air temperature.

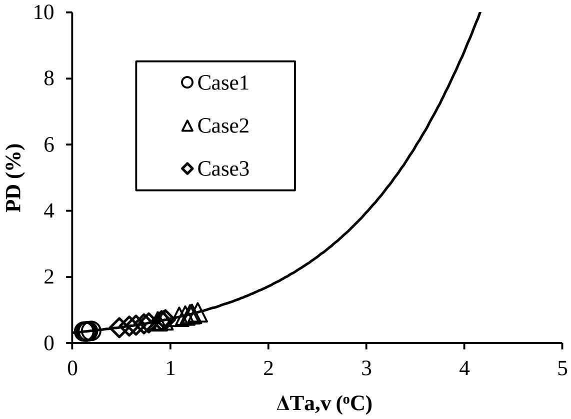


Figure 11. Local discomfort caused by vertical air temperature difference

Mean Radiant Temperature

Panel heating systems provide an acceptable thermal environment by controlling surface temperatures as well as indoor air temperature in an occupied space. With a properly designed system, occupants should be unaware that the environment is being heated. The mean radiant temperature (T_{mrt}) has a strong influence on human thermal comfort. When the temperature of surfaces comprising of building (particularly outdoor exposed walls with extensive fenestration) deviates excessively from the ambient temperature, convective systems sometimes have difficulty counteracting the discomfort caused by these cold or hot surfaces. Heating panels neutralize these deficiencies and minimize radiation losses or gains by the human body.

Mean radiant temperature was determined by Eq. (13) which is used for operative temperature calculations normally (ISO 7726, 2002).

$$\mathrm{T_{op}} = \frac{(\mathrm{h_c T_a}) + (\mathrm{h_r T_{mrt}})}{\mathrm{h_c + h_r}} \tag{13}$$

In this equation, the air temperature and operative temperature are measured experimentally. $\mathrm{h_c}$ and $\mathrm{h_r}$ values are used from the previous study (Koca *et al.*, 2014) which are about heat transfer coefficients of the same cases.

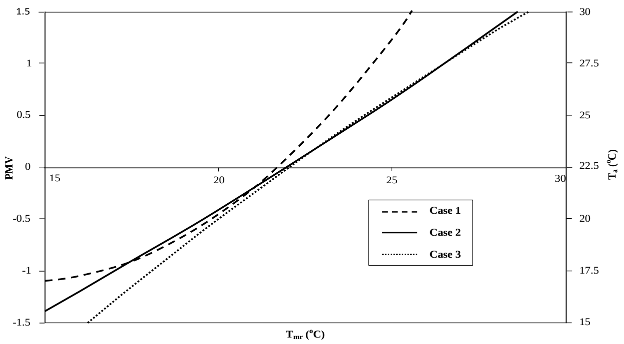


Figure 12. The effect of mean radiant temperature on PMV value

As shown in Figure 12 the effect of mean radiant temperature on PMV value is presented. While the thermal environmental conditions are ideal for human body (PMV = 0), mean radiant temperatures are measured at 21.71 °C, 21.97 °C and 22.06 °C respectively.

CONCLUSION

The thermal comfort was studied using the PMV index scale. The parameters to calculate PMV index were taken from experiments. Metabolic rate and clothing value were assumed to be 1.2 and 0.8 respectively. PMV-PPD diagram shows that thermal comfort is achieved at different air temperatures. In Case 1, comfort conditions were achieved at lower operative temperature, when the three cases were compared at the ideal thermal environment.

As it is claimed, wall configuration affects the vertical temperature differences (0.14 °C, 1.11 °C and 0.73 °C for Case 1, Case 2 and Case 3 respectively) remarkably. The vertical air temperature difference between 0.1 m and 1.7 m was less than recommended values in ISO 7730 in all cases. Furthermore, room air temperature is highly uniform in Case 1 which is less than 0.3 °C.

For PMV calculations, the mean radiant temperature calculated by using experimental data and radiosity method using view factors of the surfaces.

Although there is not much difference between these three cases, better results were obtained in terms of thermal comfort and local discomfort in Case 1 since negative effect of window is reduced by the radiant wall panels. Thus, it is recommended that the locations of the mounting radiant wall panels should be on the exterior walls with window.

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Zafer GEMİCİ received his B.S. degree in 1997, M.Sc. degree in 1999 and Ph.D. degree in 2007 in mechanical engineering from Istanbul Technical University. Currently he is Deputy General Manager and Research and Technology Development Coordinator of Mir Research & Development Inc. He has been participated in or managed more than 30 projects in various engineering disciplines. His main specialities are Management, Technology Management, IP Management, Project Management, Product Development, Heat and Mass Transfer, Fluid Mechanics and CFD (Computational Fluid Dynamics), Energy systems and Energy Management, Heating, Ventilating, Refrigeration and Air Conditioning, Sanitary Systems (pipes and fittings), Water Management etc. Moreover he is the author of 28 papers in international journals and congresses. He is the inventor of 14 international, 26 national patents. He is also a board member of Istanbul University Technopark and vice president of International Technology Union. Additionally, he has been a lecturer in Istanbul University since 2004.