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Research Article

Experimental investigation of thermal comfort performance of a radiant wall and ceiling panel system

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ABSTRACT

The number of radiant heating-cooling systems in building applications is increasing it is because of low energy consumption, low operating temperature and high values they provide in terms of working compatibility with renewable energy sources. In this study, ground source heat pump integrated radiant panels installed on the wall and suspended ceiling of the office room in Yıldız Technical University Science and Technology Application and Research Centre were experimentally examined in terms of thermal comfort according to the relevant standards. Vertical air temperature differences and mean radiant temperatures were investigated. The mean air temperature differences at 0.1 m and 1.7 m were found to be 3,9 °C 2.9 °C 3.5 °C, 3.1 °C and 3.4 °C on average for the five different stands, respectively. PMV and PPD values were found to be 0.78 and 18.9% for February 12 (Case 1), 0.36 and 8.4% for February 13 (Case 2), respectively. In the experiment carried out under the conditions of Case 2, while the comfort conditions were provided in almost all of the day, the desired comfort conditions could not be achieved in Case 1 after 11 am.

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INTRODUCTION

Water-based radiant heating and cooling systems can provide high energy efficiency, low exergy destruction and high level of comfort thanks to the possibility of using low temperature for heating and high temperature for cooling [1–5]. The proportion of radiant heating and cooling systems in commercial and residential applications is increasing [6]. Radiant heating-cooling systems are based on the formation of planar surfaces called radiant panels embedded in the floor, wall or ceiling, which can be controlled by water, air or electrical resistance, where minimum 50% of the heat transfer is realized by radiation and the rest by convection. Hydronic radiant systems working with water have been used as an alternative to conventional systems in recent years, thanks to the high comfort they provide, high energy efficiency and integrated operation with renewable energy systems. The working principle of these panels is based on the carrying out radiation and

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convection heat transfer with the surrounding air and solid surfaces. In the case of heating, the heat transfer from the panel surface by radiation can be between 70-80%, while in the case of cooling, this rate is around 60% [7].

There are many studies on radiant panel systems in the literature [8–13]. Rhee and Kim [14] conducted studies on radiant heating and cooling systems over the last 50 years in a large literature review in terms of energy performance, thermal comfort, system configuration and control. They concluded that radiant heating and cooling systems are used actively in many commercial and residential building today and these systems are clearly understood on the basis of building physics and engineering. They emphasized the need to focus on larger buildings to be built in the future and systems compatible with different climates. Myhren and Holmberg [15] conducted a numerical and experimental study to investigate thermal comfort and energy consumption on a 4.8 m x 2.4 m x 2.7 m room model. In the experiments, while for the medium temperature (55-35 °C) and high temperature (90-70 °C) cases radiator was used, for the very low temperature (35-25 °C) case radiant panel from the wall and floor were used. In their study, researchers have shown that the radiant panels can improve indoor comfort conditions by providing low vertical temperature difference and low air velocities. The use of radiant systems in buildings compared to HVAC systems was examined in the thermal comfort critical literature review by Karmann et al. [16]. As a result, there are indications that radiant systems can provide better or equal comfort than HVAC systems. Miriel et al. [17] carried out an experimental study to evaluate the energy performance of radiant panels used in cooling application in summer and winter climate conditions in a laboratory environment. According to the results, it has been seen that water radiant panels perform well in both cooling and heating in well-insulated buildings. The French conditions in which the study was conducted are less suitable for radiant cooling application than terrestrial regions, as the panel surface temperature must be kept at a minimum of 17°C to prevent condensation. Imanari et al. [18] compared the traditional air conditioning system with radiant panels embedded in the ceiling in terms of thermal comfort, energy consumption and cost. In their study, the ceiling cooling system and the classical air conditioning system were examined separately while there were people in a small office in Tokyo. As a result of the survey, it was revealed that 80% of the people found the radiant system more comfortable. It has been observed that the temperature difference in the vertical direction, which is an important parameter of thermal comfort, is less in the case of cooling from the ceiling. In addition, since a part of the sensible heat load is covered by radiant panels, the need for fresh air drawn into the space and accordingly the fan power is reduced. Thus, when a small amount of fresh air is drawn from the ceiling when cooling is done, the accumulation of cold air near the floor is eliminated. It has been observed that energy consumption is reduced by 10% when cooling from the ceiling with a radiant panel and the amortization period of the initial investment cost varies between 1 and 17 years depending on the panel prices. Gemici [19] analysed vertical air temperature differences and average radiant temperatures by analysing how thermal comfort is affected when different inlet water temperatures are applied to different wall-mounted radiant heating panel layout configurations. The average air temperature differences at 0.1 m and 1.7 m were found to be 0.14 °C, 1.11 °C and 0.73 °C, respectively. The results show that mounting radiant panels on different walls affects both thermal comfort and heating performance.

There are many numerical and experimental studies on thermal comfort performance of radiant panels in the literature [20–24]. However, it is obvious that there is a gap in the literature on a radiant wall and ceiling system with realistic space conditions integrated into the ground heat exchanger pipes placed in the ground under the building foundation during the construction phase of a large-scale building.

In current study, an office with radiant panel on the wall and ceiling integrated with a Ground Source Heat Pump (GSHP) was installed by generating realistic space conditions at Yıldız Technical University Science and Technology Application and Research Centre. Experimental studies on thermal comfort were carried out in the full-scale office room according to the relevant standards. Air velocity, horizontal and vertical air temperature differences, mean radiant temperature, predicted mean vote (PMV) and Predicted percentage dissatisfied (PPD) which are environmental parameters that can affect thermal satisfaction, were investigated for two different cases.

The novelty of this study is experimental comparison of the thermal comfort performances of radiant wall and ceiling system integrated with GSHP systems. As a result of this comparison, the most suitable design of the GSHP integrated with radiant wall and ceiling systems can be obtained. In addition to this, according to the relevant standards, it is understood that the thermal comfort conditions of radiant systems combined with a GSHP can be provided in large buildings for different climate conditions. Thus, researchers and engineers in thermal engineering sector can decide soundly whether it is feasible to invest in it.

BASIC THEORY

Thermal comfort is defined as comfortableness sensed in the thermal atmosphere. Fanger [25] developed a mathematical model using 6 comfort parameters (air temperature, air humidity, air velocity, mean radiant temperature, activity level and clothing) and psychological theory and statistical data to define comfort conditions. With this model, developed PMV index and PPD equations. He defined PMV index with a standard scale Table 1 as a parameter

Table 1. Effect of the standard thermal sensation scale on humans

+3	Hot
+2	Warm
+1	Slightly warm
0	Natural
-1	Slightly cool
-2	Cool
-3	Cold

specifying effect of any combination of thermal ambient variables and personal variables on a crowded group of people and calculated using Eq. 1 [25, 26].

$$PMV = \left(0.303.e^{-0.036M} + 0.028\right)\begin{bmatrix}(M-W)-3.05.10^{-3}.\{5733-6.99.(M-W)-p_a\} \\ -0.42.\{(M-W)-58.15\}-1.7.10^{-5}.M.(5867-p_a) \\ -0.0014M(34-T_a)-3.96.10^{-8}f_{cl}\{(T_{cl}+273)^4-(T_r+273)^4\} \\ -f_{cl}.h_c(T_{cl}-T_a)\end{bmatrix} \tag{1}$$

$$T_{cl} = 35.7-0.028.(M-W)-I_{cl}.\begin{bmatrix}3.96.10^{-8}f_{cl}.\{(T_{cl}+273)^4-(T_r+273)^4\} \\ +f_{cl}.h_c.(T_{cl}-T_a)\end{bmatrix} \tag{2}$$

$$h = \begin{Bmatrix}2.38.|T_{cl}-T_a|^{0.25} \Leftarrow 2.38.|T_{cl}-T_a|^{0.25} > 12.1.\sqrt{V_{ar}} \\ 12.1.\sqrt{V_{ar}} \Leftarrow 2.38.|T_{cl}-T_a|^{0.25} < 12.1.\sqrt{V_{ar}}\end{Bmatrix} \tag{3}$$

$$f_{cl} = \begin{Bmatrix}1+1.29.I_{cl} \Leftarrow I_{cl} \le 0.078m^2.K/W \\ 1.05+0.645.I_{cl} \Leftarrow I_{cl} > 0.078m^2.K/W\end{Bmatrix} \tag{4}$$

Where, T_r, T_a and T_{cl} are the radiant temperature (°C), indoor air temperature (°C) and clothing surface temperature. (°C). Following, M refers the Metabolic rate (W/m^2), W refers the effective mechanical power (W/m^2) and Pa refers the water vapour partial pressure (Pa). Here the terms, F^{cl} refers clothing surface factor; I_{cl} refers clothing insulation (m^2·K/W); h_c refers convective heat transfer coefficient (W/m^2·K).

Predicted percentage dissatisfied (PPD) estimates the number of thermally dissatisfied people using PMV. PPD is used to predict the rate of thermally dissatisfied people using PMV and is calculated by Eq. 5.

$$PPD = 100-95.\exp\left(-0.03353.PMV^4-0.2179.PWV^2\right) \tag{5}$$

The desired comfort level in the space is given in Table 2 according to ISO 7730 standard A and B categories for sedentary activity.

The minimum ventilation rate required for a ventilation zone can be calculated with the Eq. 3 specified in Ashrae Standard 62.1 in order to provide indoor air quality that is acceptable to human occupants [27].

$$V_{bz} = R_p \times P_z + R_a \times A_z \tag{6}$$

Where A_z is the occupiable floor area of the ventilation zone (m^2), P_z is the number of people in the ventilation zone, R_p is the outdoor airflow rate required per person (L/s) and R_a is the outdoor airflow rate required per unit area (L/s.m^2). The minimum ventilation rate required for this test room (52.5 m^2 floor area for 4 people) is calculated as 25.75 L/s. The variables R_p and R_a are taken as 2.5 L/s and 0.3 L/s.m^2 respectively for typical office room specified in Ashrae. Currently, since fresh air conditioned at 23 degrees 50% relative humidity and 40 L/s flow rate is supplied from the rooftop air handling unit, the heating load created inside the room neglected.

EXPERIMENTAL FACILITY

The experimental system is established in Yıldız Technical University Science and Technology Application and Research Centre building. The experimental system, whose flow chart is shown in Figure 1, consists of radiant panels placed on the ceiling and walls of a space integrated with a GSHP with a heating capacity of 10.5 kW and a cooling capacity of 8.5 kW. The system is basically established in 3 separate sections. The first of these is the test room at the entrance of the building, where comfort measurements are made, with radiant panels on the wall and ceiling. While there are 6 radiant panels of 2 m x 0.6 m in the south-west façade of the room, there are 9 radiant panels of the same dimensions on the north-east façade. There are a total of 29 radiant panels in the suspended ceiling, measuring 1 m x 0.6 m, consisting of similar building layers. The radiant panels can be controlled separately by the ball valves on the collector group on the suspended ceiling and the panels on the ceiling and walls. The second part is the heat pump group installed in the mechanical room of the building. The

Table 2. Suggested values of ISO 7730 for category A, B and C (EN ISO 7730, 1994)

Category	PPD	PMV	Operative temperature (°C)	Vertical temperature difference (°C)	PD% Caused by warm and cold wall
A	<6	-0.2<PMV<0.2	21-23	<2	10
B	<10	-0.5<PMV<0.5	20-24	<3	10
C	<15	-0.7<PMV<0.7	19-25	<5	15

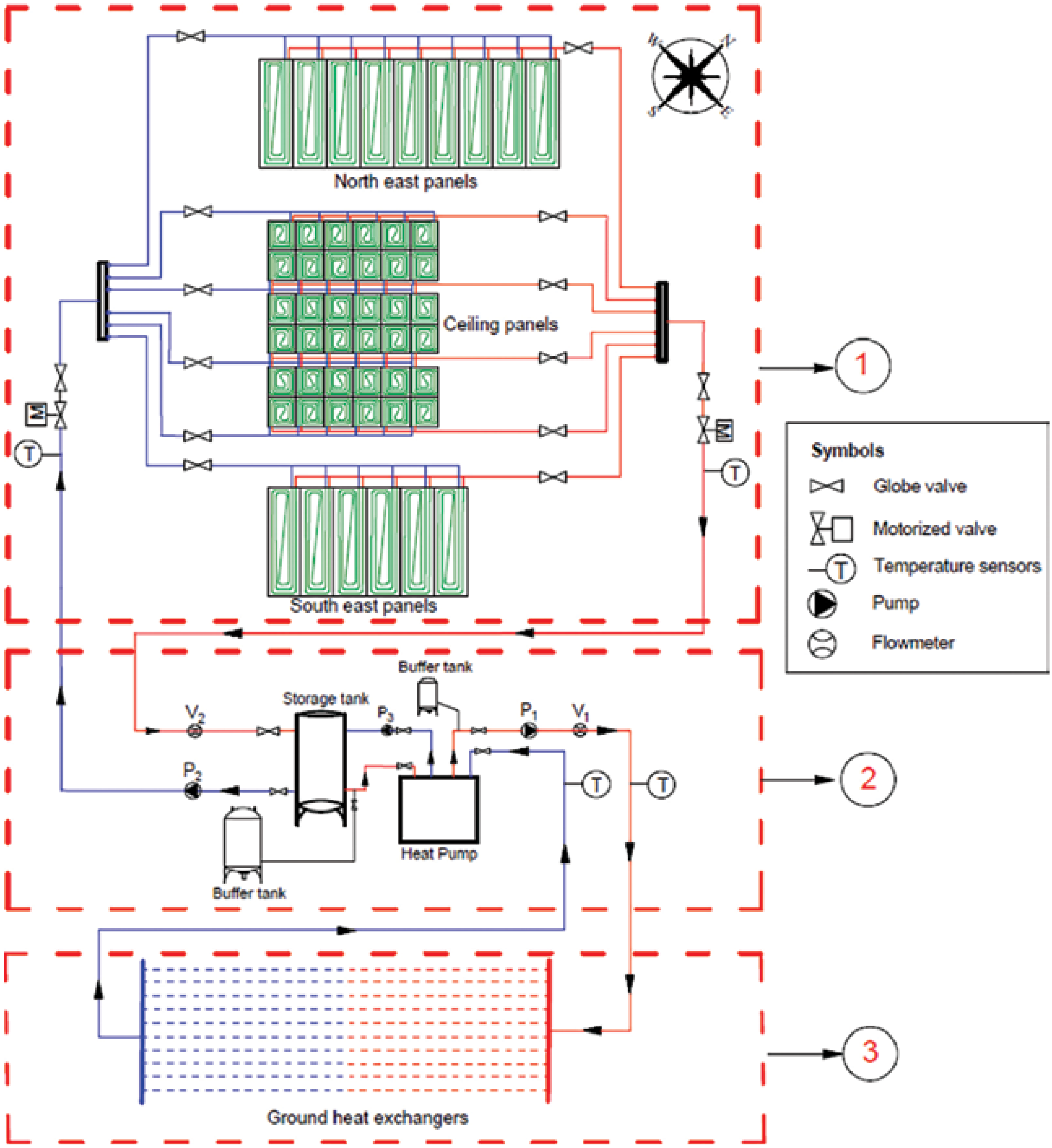

Figure 1. The flow diagram of hydraulic system.

main equipment in the heat pump group are; heat pump main block, storage (accumulation) tank, balancing tank and water circulation pumps. Parallel and horizontal ground heat exchangers placed in the ground under the foundation of the building where the system is located form the third part.

The system is activated by adjusting the panel return temperature according to the heating or cooling mode from the heat pump control unit in the mechanical room. The cold or hot water produced by the heat pump is stored in the accumulation tank, the motorized valves at the collector inlets are opened with the help of the

Table 3. Instruments used for measurement of test room parameters

Instruments	Range	Accuracies	Measured values
Temperature sensor	0 ... 100 %RH -20 ... +70 °C 0 … 10000 ppm	±2 %RH (5 ... 90 %RH)3) ±0.5 °C	Humidity, temperature and CO_2
Testo Turbulance sensor	0 … 5 m/sn	±(0.03 m/sn + 4%)	Velocity
Testo black globe temp. sensor	0 … +120 °C	Class 1	Globe temperature
T-Type temp.	-20…+200	±0.3 °C	Air and surface temperature
RTD	-200….+800	±0.15+0.0002 °C	Water temperature

Figure 2. The view of test room.

thermostat on the wall, depending on the cooling or heating need in the test room, and water circulation is carried out in the radiant panels. In addition, with the temperature sensor located on the panel return line, the water return temperature is adjusted from the thermostat in the room, and the motorized valve is opened and water flow is provided.

The office room (Fig. 2) where the experiments were carried out was transformed into a fully equipped experiment room in order to be able to carry out the comfort measurements completely. In the experiments, as seen in Figure 2c, considering the EN 14240 standard, a total of 4 cylinders with 120 W heat emitting were positioned to create a heat load in the room. A total of 51 thermocouples were used for surface and air temperature measurements, and their positions in the room are given in detail in Figure 3. For vertical temperature distribution, 5 stands were used and 5 thermocouples at different heights were placed on each. In order to prevent these thermocouples from being affected by radiation, they are covered with a cylindrical radiation shield. Two PT100s were used to measure the water temperatures entering and leaving the wall and ceiling panels. Testo brand comfort kit shown in Figure 2b was used for PMV and PPD measurements, which are the most important parameters of this study, and its technical details are given in Table 3.

There are radiant panels on the wall and ceiling of the office to be heated and cooled, and ground heat exchangers buried in the ground under the building foundation (Fig. 4). The place where the connection between these two circuits takes place is the heat pump room where the water source heat pump is located. There is a water source heat pump belonging to the ground line in the heat pump room. The view of the whole system as a group in the heat pump

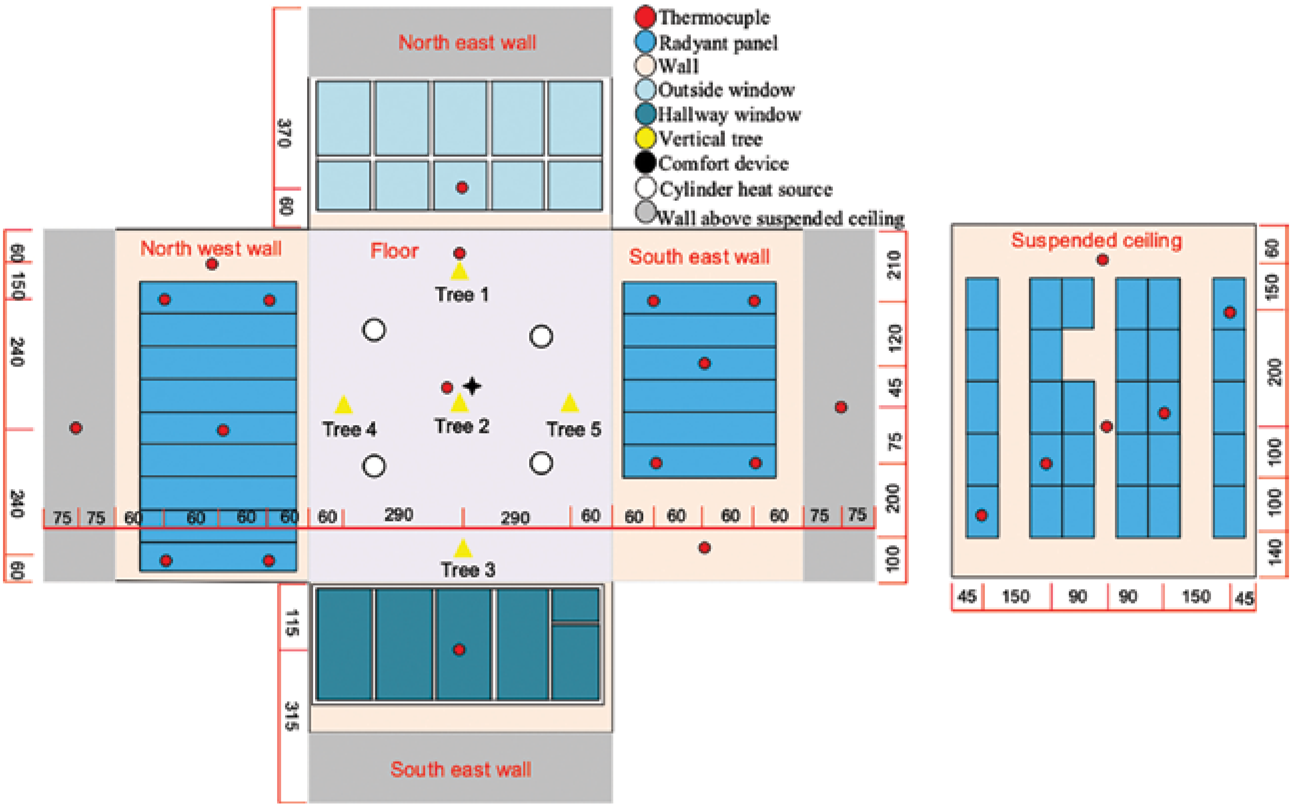

Figure 3. Indoor measurement instrument layout.

Figure 4. GSHP equipment.

room is shown in Figure 4 As can be seen in Figure 4, all equipment on a heat pump is shown by numbering, and in Table 4 the model, brand and specification of all components are given.

UNCERTAINTY ANALYSIS

The uncertainties of the thermal comfort indexes PMV and PPD are calculated indirectly with the Eq. 7 developed by Kline and McClintock [28] as the combined uncertainties

Table 4. The technical specifications of equipment on GSHP system

Equipment	Brand/model	Specification
1. Heat pump	Restherma/ IP11SS	**Heating:** Capacity: 10.5 kW, Power: 2.1 kW, COP: 5, Operating temp. range:-5/+45 °C, Max. supply temp.: 55 °C
		Cooling: Capacity: 8.5 kW, Power: 1.98 kW, EER:4.29, Operating temp. range:+10/+43°C, Min. outlet temp.: 7 °C
2. Flow switch	Ayvaz-AK 100	
3. All pumps	**3a:** Grundfos/MAGNA3 25-100	**3a:** Max. flow rate: 78.5 m^3/h, Max. head: 18 m, Max. pressure: 16 bar, operating temp. range: -10/110 °C
	3bGrundfos/Alpha2	**3b** ve **3c**: Max. flow rate: 4.8 m^3/h, Max. head: 5.8 m, Max. pressure: 10 bar, operating temp. range: 2/110 °C
4. Thermometer	Pakkens/TE100DB1,	Measurement range: -30/+60 °C
5. Manometer	Pakkens/MG063DRM1,	Measurement range: 0-10 bar
6. Accumulation tank	Resboyler/KAT	Capacity: 100 lt, Test pressure: 13 kg/cm^2, Operating pressure: 10 kg/cm^2
7. Expansion tank	Reflex/15P1125,	Operating temp.: -10/120 °C, Max. operating temp.: -10/70 °C, Operating pressure: 6 bar, Volume: 23L
8. RTD	Tekon/PT100,	Measurement range: 0/100 °C
9. Flowmeter	Bass/FMPV,	Operating temp.: -10/70 °C, Max. operating pressure: 10 bar
10. Rotameter	ZYIA/LZM-25T	Operating range: 5/35 LPM

of the parameters such as room air temperature, radiant temperature, air velocity etc. with devices listed in Table 3.

$$W_R = \left[\left(\frac{\partial R}{\partial x_1}w_1\right)^2 + \left(\frac{\partial R}{\partial x_1}w_1\right)^2 + \left(\frac{\partial R}{\partial x_1}w_1\right)^2 + \ldots + \left(\frac{\partial R}{\partial x_n}w_n\right)^2\right]^{1/2} \quad (7)$$

Where, W_R is the total uncertainty of the data R, R is the function of the independent variables x_1, x_2, x_3,…… x_n and U_i is the total uncertainty of the independent variable x_i of the data R (for i=1 to n). The total average estimated uncertainty of the PMV and PPD are found approximately ±8% and ±7.5% respectively the times between 09:00-18:00 for the first day (Case 1) of experiments. Calculations of the uncertainty of the PMV and PPD for day 2 (Case 2) experiment are found approximately ±11% and ±8.8% respectively.

RESULTS AND DISCUSSION

In the previous section, detailed information was given about the measuring instruments used in the test room. The changes in surface temperatures, water inlet and outlet temperatures and air temperature measurements in the test room are given in Figure 5 for two cases at 12th and 13th February. The heat pump water outlet temperature is set to a maximum of 55 ^{0}C on February 12, which is Case 1, and 50 ^{0}C for Case 2 on February 13, from the control panel in the engine room which is located at basement of the building.

The heat pump, which was started the day before, keeps the water in the storage tank at the set temperature continuously. Experiments are carried out at 6 am by opening the motorized valves on the suspended ceiling through the thermostat in the test room. Although there are 51 thermocouples in the test chamber, important parts are given in Figure 5 so that the temperatures can be seen clearly.

Radiant and air temperatures are measured in the centre of the room at 1.1 m height for 2 cases and operative temperature in working hours are given in Figure 6. Heating and cooling systems are usually controlled by thermostats with air temperature adjustment, since they are simpler and lower cost. When the system is started and stabilized, there is an average of 1 ^{0}C difference between radiant temperature and air temperature. This shows that the radiant and air temperature largely cover the temperature values of the EN ISO 7730 standard. Taking the operating temperature, which is the average of the air and radiant temperature, as the reference temperature is provide more accurate to comfort standards. Jia et al. [29] found temperature difference as 0.6 and 0.2 K in radiant ceiling panel and radiant slab applications with low heat load (30 W/m^2).

They suggested to use conventional temperature control systems for radiant ceiling panel application where the temperature difference is low as it is lower cost and more simple system. In order to reach desired comfort levels loads are raised to 55-65 W/m^2 level. As seen when higher loads are experienced temperature difference higher and this makes impossible to use of air temperature controlled systems. These systems requires longer time to stabilise therefore it is not possible to use air temperature controlled systems. Instead, radiant and operative temperature based MPC, PMV and air temperature estimation based load estimation models are preferred.

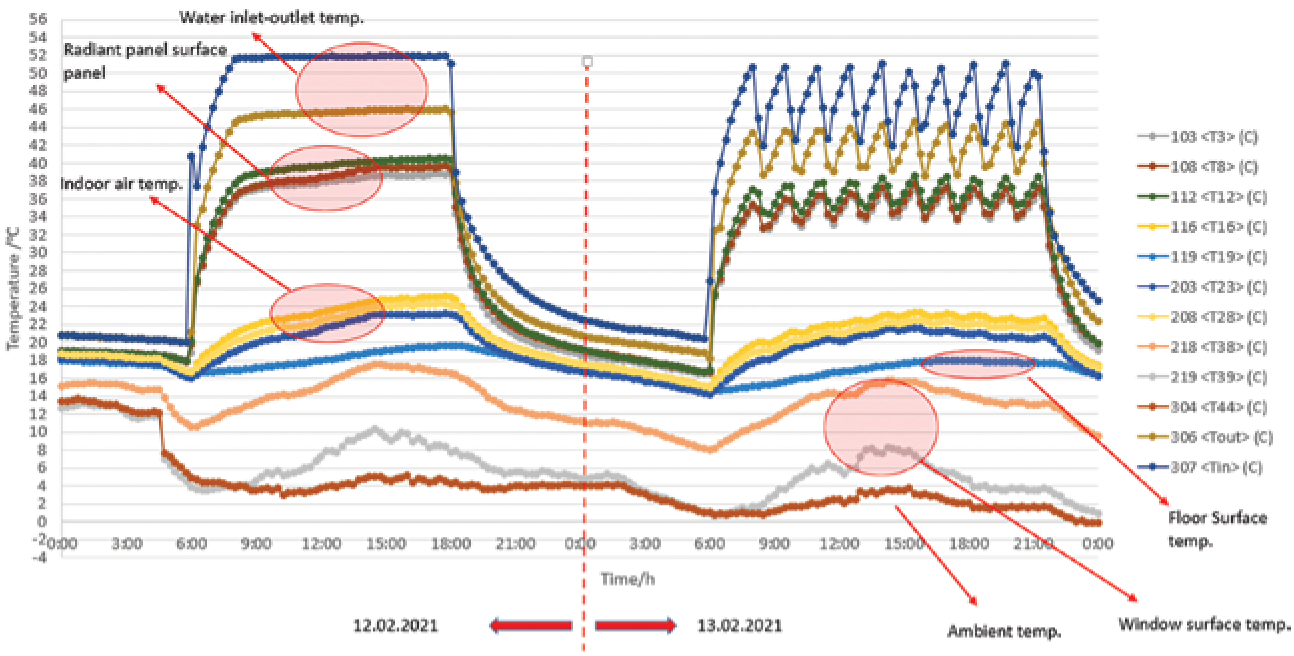

Figure 5. Temperature measurements in the test room for two days.

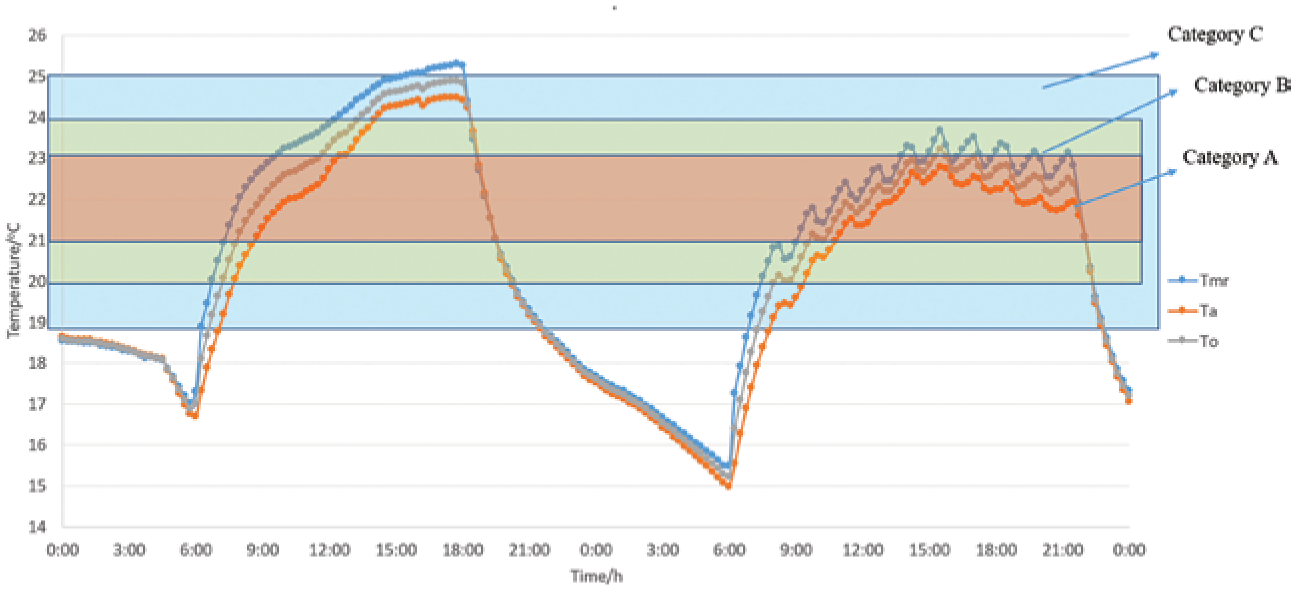

Figure 6. Variation of mean radiant temperature, operative temperature and air temperature in category A, B and C.

Figure 7 and Figure 8 illustrations the variation of PMW, PPD for 2 different cases. Comfort measurements were made at the centre of the room and at a height of 1.1 meters from the floor. The aim of the experiment is to determine the comfort values between 09:00-18:00, which are office hours. For this purpose, hot water flow was started in the panel at 06:00 with the thermostat in the room. According to the EN ISO 7730 standard, the comfort level in the office is determined as -0.5<PMV<0.5 PMV and category B with 10% PPD value. Looking at Graph 7, the desired PMV and PPD values were reached within an hour, while the comfort level deteriorated after 11. When the Case 2 graph (Fig. 8), where the water temperature is set at 50 ^{0}C, is examined, the desired. The discomfort situation deteriorated in Figure

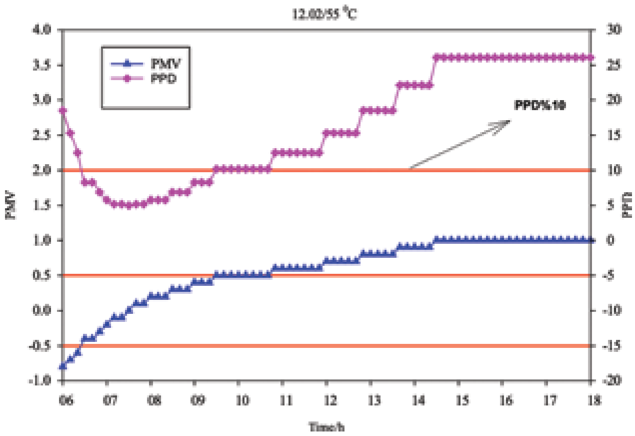

Figure 7. Changes of PPD and PMV for Case 1.

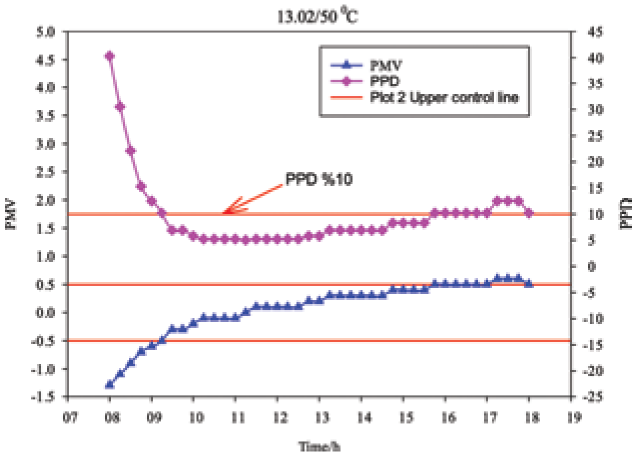

Figure 8. Changes of PPD and PMV for Case 2.

7 is not because of the inadequacy of the heating capacity of heat pump, but due to the lack of any control mechanism on the water supplied or return to the panels. It is possible to keep it within the comfort range standards by using a proportional valve sensitive to indoor air or radiant temperature instead of the on-off motorized valve on the collector.

As shown in Figure 3, the testing room was equipped with thermocouples in five different locations, in a vertical manner, in order to measure the vertical temperature distribution. The thermocouples were placed on stands, numbered 1, 2, and 3, the thermocouples being placed at locations 0.1m, 0.6 m, 1.1 m, 1.7 m and 2.5 m (the distance from the ankle to the ground being 0.1m, from the knees 0.6 m, from the head distance to the ground while seated 1.1m, from the head distance to the ground while standing 1.7 m, from the ceiling, another thermocouple being 0.3m below the ceiling, respectively). The thermocouples, that are placed on stands, which are numbered 4 and 5, are

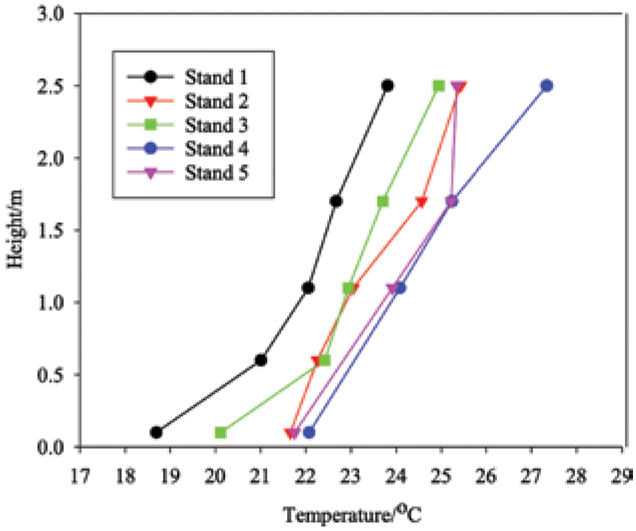

Figure 9. Air temperature distribution in vertical direction.

placed next to the wall, 0.1 m, 1.1m, 1.7 m, and 2.5 m apart from one another, respectively.

As expected, the temperatures on the stand near the window were the lowest, followed by the stands near the corridor and in the middle. Stands close to the wall were at almost the same temperature and were the highest values. When we look at the vertical temperature difference, which is another parameter of the comfort evaluation, the difference between 0.1 to 1.7 m that should be maximum 3 ^{0}C on the window side has been exceeded with 0.9 ^{0}C at average of 3.9 ^{0}C. In other stands, these values were 2.9 ^{0}C 3.5 ^{0}C, 3.1 ^{0}C and 3.4 ^{0}C on average, respectively. This shows that wall and ceiling heating cannot provide sufficient comfort in vertical temperature distribution. The investigation of the vertical temperature distribution in 3-hour periods is given in Table 3. Looking at the details, the vertical temperature distribution during office working hours has almost never fallen below 3 ^{0}C. The most important reason for this can be wall and ceiling heating instead of underfloor heating.

Uninsulated walls, roofs, floors, cold windows, or equipment emitting high heat etc. cause asymmetrical or non-uniform thermal radiation in an enclosures, which can affect the comfort of occupants in office. As seen in Figure 3, the air temperature distribution in the room could not achieve a uniform distribution due to the fact that the there is a low window surface temperature and high panel surface temperatures when the heat pump actively work between 6 and 18:00 o clock. In Figure 10, the temperature difference changes are shown by measuring the temperatures on the stands located at the window, middle (reference) and wall in the room. When the temperature values measured according to the reference temperature (middle of the room) are examined with the window and the heated wall, it is seen that there is an average difference of 1.7 ^{0}C and 0.7 ^{0}C,

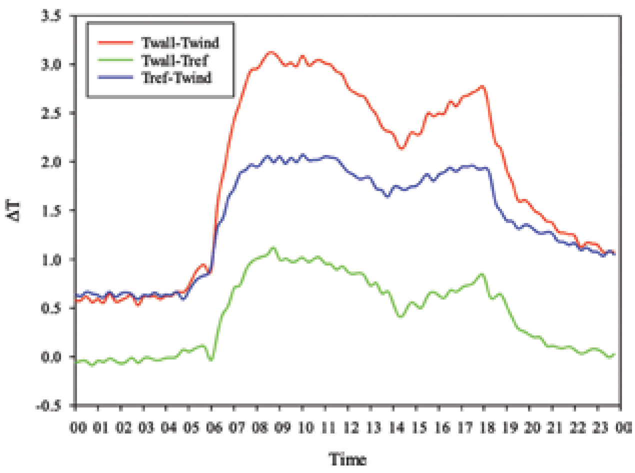

Figure 10. Variation of temperature differences between different points of the room.

respectively. This temperature difference between the wall and the window goes up to 2.5 ^{0}C on average. When the reference temperature differences in Figure 10 is examined, it is obvious that temperature values recommended for comfort limit in the standards cannot be achieved by the window.

CONCLUSION

The use of a radiant panel heating system with an integrated GSHP system for comfort was evaluated for its practical application in commercial buildings in the cold winter in Yıldız Technical University Science and Technology Application and Research Centre.

In the measurements made with the comfort device in the centre of the room, the PMV and PPD values for the first case started to deteriorate at 11 am, while the category B comfort values were provided all day in Case 2. The vertical temperature distribution was measured experimentally in different parts of the room and it was seen that it did not meet the recommended values in ISO 7730 at almost any point. Considering the comfort at different points in the room, it is provided all day in the centre of the room. Especially in Case 2, while the fact that it is 1.7 ^{0}C lower than the reference temperature by the window is insufficient to provide the same comfort.

Although 2 cases are compared, the comfort in the room can be provided in a large part, the room consists of highly glazed surface and the absence of floor heating cause it to not provide the comfort conditions near the window and in the vertical direction. Since the ratio of window on the building surfaces has a significant effect on the thermal comfort, indoor lighting and energy consumption in the office, the window to wall ratio (WWR) should be carefully designed. The high WWR (>0.9) and the lack of direct sunlight in the office room where the experiments are carried out have an undercooling effect, especially in winter days when the ambient temperature are low. As a result, it is difficult to obtain a uniform thermal comfort and temperature in the room. In order to overcome the problem, investigating the optimum WWR and covering the window surface with the appropriate low emissivity films in will make a great contribution to ensuring uniform thermal comfort inside.

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AUTHORSHIP CONTRIBUTIONS

Authors equally contributed to this work.

DATA AVAILABILITY STATEMENT

No new data were created in this study. The published publication includes all graphics collected or developed during the study.

CONFLICT OF INTEREST

The author declared no potential conflicts of interest with respect to the research, authorship, and/or publication of this article.

ETHICS

There are no ethical issues with the publication of this manuscript.

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