AN EXPERIMENTAL STUDY OF THE RADIANT CEILING COOLING SYSTEM•

دراسة تجريبية لمنظومة تبريد مشعة سقفية

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• البحث مستل من أطروحة الماجستير للطالب علي فاضل حسن / الكلية التقنية- بغداد

Abstract

Due to intensive use of energy, with a decline in natural resources, view of the CFCs damage on ozone layers and an increase in fuel costs, it has become important to find methods of reducing energy consumption and finding alternative energy sources.

Radiant cooling system is a tool used to achieve these benefits, while maintaining acceptable thermal comfort conditions.

The main aim of the present study is to test radiant cooling ceiling system in Iraq climate established in a typical office focusing on occupant's thermal comfort.

A theoretical model has been developed to predict the panel temperature, the interior surface temperatures of the office and heat output for radiant cooling panel systems.

To validate the theoretical model, an experimental work was performed on radiant panels of, (6.2X4.6X3)m full-scale office room was utilized in Technical Institute of Karbala.

The systems are constructed from 77 plates at a 0.05-mm-thick aluminum sheet with dimensions of (60×60) cm, in which the plates cover whole the ceiling and are cooled with water circulating through (1/2) inch copper tube.

The cooled water achieved from the well that manually drilling near the test room at dimensions are (15) cm in diameter and at (6) m depth.

Ventilation fan with two cooling coils that was used to maintains acceptable indoor air quality by supplying fresh, filtered air.

The study shows, the radiant cooling system can be applied for Iraqi buildings under the tropical climate and that approximately 71% energy saving can be obtained.

As the inside temperature of the walls surface has a great effect on the thermal comfort of persons in the conditioned space, the radiant system succeeded in making the differences between the mean radiant temperature and room air temperature is $2^{o}C$ and between the room air temperature and radiant surface temperature is $5^{o}C$ and the values of PMV-PPD were used to examine thermal comfort and found that the use of radiant cooling system is appropriate for creating comfortable indoor conditions.

The study shows also that effective usage of renewable energy sources that provides economic and environmental benefits.

الخلاصة:

بسبب زيادة استخدام الطاقة وزيادة في تكاليف الوقود اللازم لإنتاجها، إضافة إلى الأضرار التي تسببها مركبات وسائط التبريد على طبقة الأوزون، فقد أصبح من الضروري إيجاد طرق للحد من استخدام مركبات وسائط التبريد الحالية وإيجاد مصادر بديلة للطاقة.

ولغرض تحقيق ذلك إضافة إلى الحفاظ على ظروف الراحة الحرارية مقبولة في المحيط المكييف تستخدم منظومات

التبريد المشعة.
إن الهدف الرئيسي من هذه الدراسة هو اختبار منظومات تبريد مشعة تم نصبها في سقف غرفة مكتب تحت الظروف المناخية للعراق مع التركيز على عوامل الراحة لشاغلي المكتب. لذالك تم وضع نموذج رياضي لحساب درجة حرارة السطوح المبردة، ودرجات حرارة السطوح الداخلية للجدران والسعة الحرارية لمنظومة التبريد المشعة.
ولغرض التحقق من صحة النموذج الرياضي، تم استخدام 77 لوحة من الألمنيوم سمك 0.05 ملم بأبعاد (60 × 60) سم معلقة في سقف غرفة ذات أبعاد (6.2×3×4.6) متر في المعهد التقني كربلاء. يتم تبريد الألواح بواسطة ماء يمر في أنبوب نحاس قطر (2/1)انج مثبت على الألواح التي تغطي سقف الغرفة كله. وقد تم حفر بئر بقطر 15سم وبعمق 6متر لغرض استخدام ماء البئر في تبريد الألواح المشعة. لقد تم استخدام مروحة هواء لدفع هواء الخارجي إلى داخل الغرفة بعد تبريده بواسطة ملف تبريد يبرد بواسطة ماء البئر.
لقد أثبتت التجارب العملية التي أجريت أن الفرق بين معدل درجات الحرارة لجدران الغرفة وهواء الغرفة لا يتجاوز (2^oC)، وكذلك الفرق بين درجة حرارة هواء الغرفة والسطح المشع لا يتجاوز (5^oC). كما تم حساب قيم PMV-PPD والتي أثبتت أن المنظومة المستخدمة تحقق الظروف المناخية المريحة في الغرفة تحت الظروف المناخية لمدينة كربلاء كما أن هذه المنظومة تحقق انخفاضا في استهلاك الطاقة بمقدار حوالي 70% مقارنة بمنظومات التبريد الانظغاطية إضافة إلى الفوائد اقتصادية وبيئية.

1. Introduction

Researchers are studied theoretical and experimental radiant systems in different position and types with different energy resource, and there is ongoing research on how best to integrate and control radiant systems of different types, in different climates, and with different companion systems.

Wei-Hwa Chiang et.al [1] [2011], tested the effect of inlet water temperature and flow rate on cooling efficiency of a radiant ceiling system in Taiwan , The objective of this research is to figure out different supplying water temperature which accompanies flow rate embedded in radiant ceiling panels with a water circulatory system, they adopt five conditions of inlet water temperature 16, 18, 20, 22, and 24ºC with water flow rate at 40, 60, 80, and 100 LPM (liter per minute) to test the vertical temperature gradient inside the room and discuss the interactive effects between the parameters within one hour, The experimental values reveal that the better operative efficiency occurs as the inlet water temperature is 18 ºC and flow rate is 100 LPM without condensation of water on the radiant panel surface. The obtained results also that indicate as the inlet water temperature equals 24 and 22 ºC, the indoor air temperature only changes a little. This indicates that the difference is about 4-6 ºC between inlet water temperature and indoor air temperature.

Néstor Fonseca Díaz [2] [2009], focuses on the experimental and theoretical analysis and modeling of the radiant ceiling systems as commissioning tool. A steady state model appears to be an appropriate tool for preliminary calculation, design and diagnosis in commissioning processes. In order to validate this model the cooling ceiling system of a commercial building in Brussels is experimentally evaluated, the results show that the average difference between simulated and measured air and surfaces temperatures is lower than ±0.5 °C. The mean temperature of water and air room are of 13 -18 °C and 37- 49 °C for water and 24-26 °C and 21-23°C for air in cooling and heating mode respectively. And the water temperature drop across the ceiling is (2-3°C). And also, commissioning test results show that the influence of surfaces temperatures inside the room, especially the facade and ventilation are significant and that the radiant ceiling system must be evaluated together with its designed environment and not as a separate HVAC equipment.

John Busch and R. Diamond [3] [2007], showed that radiant cooling systems create comfortable indoor conditions, have high potential to reduce building energy consumption and peak power demand, are economically competitive, and are not restricted to specific geographic areas in the United States.

Vangtook and Chirarattananon [4] [2006], studied radiant cooling with natural ventilation under hot and humid climate. The temperature of radiant cooled water was limited to 24ºC to avoid condensation. They found that the low heat reception capacity of the panel would limit its use only to situations when loads were low. Obviously, the system is good for dry climate and could be problematic for humid climate.

Kim et.al [5] [2005], performed field measurements on an office in Japan. They used a Computational Fluid Dynamics model to analyze the indoor environment of the office for three cases: cooling panel with all-air system for ventilation, all-air cooling system and cooling panel with natural ventilation. In all cases, the latent load is ignored. The results of this research show that more radiant heat is transferred between a cooling panel and a human than an all-air system. They also found that the cooling panel system kept the mean radiant and operative temperatures lower than the all-air system.

Conroy and Mumma [6] [2005], showed cooling ceiling systems significantly reduce the amount of air transported through the building (often only about 20% of the normal all-air system air flow rates). This results in the reduction of the fan size, energy consumption and ductwork cross-sectional dimensions.

The study of **Jeong et.al [7] [2003],** showed that radiant panel cooling systems consume 42% less energy as compared to conventional VAV systems with all air economizers.

Mumma and Badenhorst [8] [2002], indicated three reasons to employ mixing ventilation rather than displacement ventilation. First, the air movement across the cooled ceiling surface typically boosts the cooling performance by at least 5%. Second, the benefit of enhanced air quality is typically realized by a displacement system, it cannot be achieved when combined with a CRCP system. The cooled ceiling convection currents cause the room air be mixed, disturbing the otherwise stratified air, and contaminants will be mixed into rather than displaced out of the breathing zone. Third, large supply-to-room temperature differentials are used with mixed flow systems, especially if high induction ratio diffusers are used, shifting more of the sensible cooling load to the DOAS and greatly reducing the first cost of the radiant cooling system.

Niu et.al [9] [2002], proposed a 100% outdoor air displacement ventilation system consisting of a desiccant cooling system combined with a cooled ceiling. In their concept, cooling radiant ceiling panels CRCPs, and the latent and ventilation loads are meet by an auxiliary desiccant cooling system primarily treat the sensible load.

Imanari et.al [10] [1999], performed experiments to determine the effect of radiant ceiling panels on thermal comfort, compared to an all-air system, in a meeting room. An air-handling unit was used to meet the latent load and provide ventilation during the tests with the ceiling panels. The experiments were performed with human subjects who were present in the room for at least one hour and given a questionnaire to fill out about their comfort. They found that the radiant panels created a more comfortable work place than an all-air system for both genders. They also found that the mean radiant temperature of the space is lower, there is a smaller vertical temperature gradient in the space and the mean air velocity is lower. All of these factors improve the comfort of the occupants and result in a smaller percent dissatisfaction with the radiant panels. They also performed numerical simulations to determine the energy consumption of the radiant panels. From these simulations, they found that the air transport energy is reduced by 20% and the total energy consumption is reduced by 10%.

Stetiu [11] [1999], developed RADCOOL software to perform hydronic radiant cooling systems that works under the SPARK software environment. The model is based on a methodology for describing and solving the dynamic, nonlinear equations that correspond to complex physical systems as found in buildings. The model calculates loads, heat extraction rates, room air temperature and room surface temperature distributions, and can be used to evaluate issues such as thermal comfort, controls, system sizing, system configuration, and dynamic response. And the details of his experimental work was used the test room of 2.9m× 4.3m×2.85m location in California and all walls are exterior, the floor is in direct contact with the ground and the west wall has a window, the results are, when the ceiling surface temperatures variation from 20 ºC to 23 ºC, the room air temperature can be maintain at 22 ºC to 27 ºC when the ventilation air supply at 18 ºC, as well as Radiant panel cooling might save 30% of overall cooling energy for applications across a range of representative climates in north America , these predicated savings, as compared to a conventional all-air VAV system, result mainly from reductions in energy used to remove sensible heat from conditioned spaces. The same research also indicated that potential energy savings would range from approximately 17% in cool, humid regions to 42% in hot, arid regions. This range reflects the relatively large latent load in humid regions.

Niu [12] [1994], found that cooled ceiling combined with displacement ventilation can produces a thermally comfortable environment at a cooling load up to 50 W/m^2, compared with 40 W/m^2 with only displacement ventilation.

The research on radiant cooling systems suggests that the strategy has great potential for reduced energy consumption which varies widely and depends on climate and radiant cooling type, favorable tie-in capabilities with low-temperature and low-intensity energy sources such as (solar system, cooling tower and ground heat pump), and also improved air quality and can control the humidity, reduced space requirements, and potentially even lower first costs.

Today, in Europe and increasingly in the United States radiant ceiling panel cooling is finding new markets in tighter buildings, with controlled ventilation and dehumidification to prevent condensation problems.

2. THEORETICAL ANALYSIS

Radiant systems supply or extract heat from a room through the action of convective and radiation heat exchange between the room environments and heated or cooled panels situated in the ceiling. The radiation heat exchange can be calculated as a function of the room geometry and surface characteristics.

It is important to understand that all of the surfaces shown in Figure (1) are coupled thermally through their radiant exchange and their convective exchange with the room air. In addition, the exterior walls will transfer heat to the surroundings as well as the floor and the ceiling of inside room.

To modeling theoretical equations, heat balance on the room air environment, walls, glasses, radiant panel and uncooled roof must be taken. The system of equations were formulated as a matrix and solved by Gauss elimination method.

To achieve this aim, it was found that the following system of equations needs to be solved:

i) Energy balance on the room environment air
ii) Heat balance on the room surfaces and window glasses surfaces.
iii) Heat balance on the cooling panel.
iv) Heat balance on the uncooled roof.
v) The definition of mean radiant temperature.

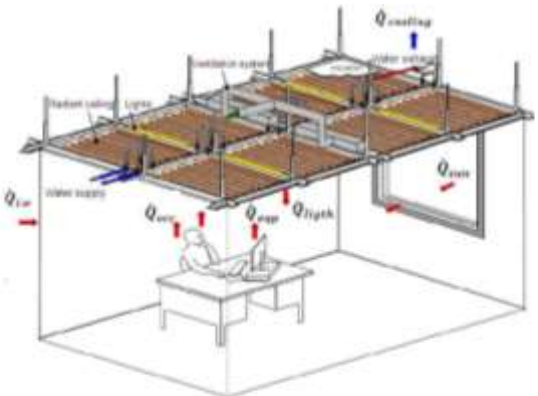

Figure (1) heat loads for a radiant ceiling

4. Room Air Heat Balance

The room air interacts with the walls and window surfaces, people and equipment and can be heated/ cooled by the air that is infiltrated or is brought in to the room for ventilation.

The auxiliary ventilation system, which is designed to provide air renewal for hygienic requirements, and also, the latent (moisture) load of the room have to be controlled by this system.

According to ASHRAE Standard 55-2004, Person needs 36 m^3/h per person in general offices.

Convection heat balance for the air in the room is

$$\sum \dot{Q}_{room-air} = 0 \ldots\ldots\ldots\ldots (1)$$

That mean,

$$\dot{Q}_{acp} + \dot{Q}_{uncp} + \sum_{i=wi}^{n=4} \dot{Q}_{awi} + \sum_{i=gi}^{n=4} \dot{Q}_{agi} + \dot{Q}_{afl} + \dot{Q}_{avent.} = 0 \ldots\ldots\ldots (2)$$

Where the heat added or removed due to air supply for ventilation is.
Equation (2) can be written as follows:

$$A_{cp}h_{cp}(T_r - T_{cp}) + A_{ucp}h_{ucp}(T_r - T_{ucp}) + \sum_{i=1}^{n=4} A_{wi}h_{wi}(T_r - T_{wi}) + \sum_{i=1}^{n=4} A_{gi}h_{g}(T_r - T_{gi}) + A_{m}h_{m}(T_r - T_{o}) + \dot{m}cp_{air}(T_r - T_{supply}) = 0 \ldots\ldots (3)$$

After simplification, the equation becomes

$$a_{1,1}T_{room_{air}} + a_{1,2}T_{w1} + a_{1,3}T_{w2} + a_{1,4}T_{w3} + a_{1,5}T_{w4} + a_{1,6}T_{g1} + a_{1,7}T_{g2} + a_{1,8}T_{g3} + a_{1,9}T_{g4} + a_{1,10}T_{panel} + a_{1,11}T_{unc} = b_1 \ldots (4)$$

3. 2 Heat Balance on Room Surfaces

Each room surface area A_i as illustrated in Figure (1) is in radiant heat exchange with all the other surfaces and is in convective heat exchange with the air in the room. The sum of these two heat flows, $\dot{Q}_{rad}$ and $\dot{Q}_{conv.}$ will, under steady state conditions, equal the conductive heat flow through the surface as shown below.

$$\dot{Q}_{cond(i)} + \dot{Q}_{conv(i)} + \sum \dot{Q}_{rad(i)} = 0 \ldots\ldots\ldots (5)$$

Where

$\sum\dot{Q}_{rad}$: Net radiation heat transfer from A_i

$\sum\dot{Q}_{conv.}$: Convection between air and surface A_i

$\sum\dot{Q}_{cond.}$: Conduction through surface A_i

Or

$$\dot{Q}_{cond.wi} + \dot{Q}_{conv.wi} + \sum_{i=wi}^{n=4,n\neq i} \dot{Q}_{rad.wi-wi} + \sum_{i=wi}^{n=4,n\neq i} \dot{Q}_{rad.wi-gi} + \dot{Q}_{rad.wi-cp} + \dot{Q}_{rad.wi-ucp} + \dot{Q}_{rad.wi-m} = 0 \ldots (6)$$

The above equation can be written in the fallowing form for the walls in the room:-

$$A_{wi} h_{wi-air}(T_{wi} - T_{air}) + A^{*}_{wi}\left[\sum_{i=1}^{n=4,n\neq 1} h_{rwi-wi}.(T_{wi} - T_{wi}) + \sum_{i=1}^{n=4,n\neq 1} h_{rwi-gi}.(T_{wi} - T_{gi}) + h_{rwi-cp}.(T_{wi} - T_{cp}) + h_{rwi-ucp}.(T_{wi} - T_{ucp}) + h_{rwi-m}.(T_{wi} - T_{o})\right] + A_{wi}[U_{Wi.}(T_{wi} - T_{outsurf\ wi})] = 0 \ldots.(7)$$

4.3 Heat Balance for the Uncooled Roof

If the cooling panel not covering the wholly ceiling, therefore the uncooled ceiling (roof) represents source of heat and transmitted heat from out to the condition space and must be considered and its equation can be expression as:

$$\dot{Q}_{cond.up} + \dot{Q}_{conv.up_{air}} + \sum_{i=1}^{n=4} \dot{Q}_{rup_{wi}} + \sum_{i=1}^{4} \dot{Q}_{rup_{gi}} + \dot{Q}_{rup_{m}} = 0 \ldots\ldots..(8)$$

Therefore

$$A_{up} \times \left[U_{up.}\left(T_{in_{up}} - T_{out_{up}}\right) + h_{conv.}\left(T_{in_{up}} - T_{room_{air}}\right) + \sum_{1}^{4} h_{r\,up_{wi}}.(T_{up} - T_{wi}) + \sum_{1}^{4} h_{r\,up_{gi}}.(T_{up} - T_{gi}) + h_{r\,up_{m}}.(T_{Uup} - T_{o})\right] = 0 \ldots..(9)$$

3.4 Comfort Equations

The objective of the heating or cooling system is to provide thermal comfort for people in the room. For economic operation, air conditioning systems should be operated at low a temperature in winter and at high a temperature in summer.

For room with radiant cooling system the Operative temperature have a strong influence on thermal comfort. The Operative temperature, which is a term combining air temperature and mean (T_{mrt}) radiant temperature, was suggested by Fanger (1967) as a measure of local thermal comfort. it is defined as “the uniform temperature of an imaginary black enclosure in which an occupant would exchange the same amount of heat by radiation plus convection as in the actual non uniform space”.

According to ASHRAE Standard 55-2004, operative temperature can be calculated by the following equation:

$$T_{operative} = A.T_{air} + (1 - A)T_{mrt} \ldots\ldots\ldots.(10)$$

Table (3.5) Value of A in Equation (9) [55]

Air speed Vr	(<0.2 m/s)	(0.2 to 0.6 m/s)	(0.6 to 1.0 m/s)
A	0.5	0.6	0.7

For radiant cooling the velocity of the air in the room must be less than 0.2 m/s, therefore the value of A is equal to 0.5.

The main radiant temperature in the room is:

$$T_{mrt} = T_1.F_{P-1} + T_2.F_{P-2} + \ldots\ldots\ldots\ldots + T_n.F_{P-n} \ldots\ldots(11)$$

where:

$T_1, T_2, \ldots\ldots\ldots. T_n$:Surface temperatures surrounding the occupant in a room.

$F_{P-1}, F_{P-2}, \ldots.., F_{P-n}$: View factor from surfaces one, two,…. to the person.

So,

$$A.T_{air} + (1 - A) \times [F_{w1_m}.T_{w1} + F_{w2_m}.T_{w2} + F_{w3_m}.T_{w3} + F_{w4_m}.T_{w4} + F_{g1_m}.T_{g1} + F_{g2_m}.T_{g2} + F_{g3_m}.T_{g3} + F_{g4_m}.T_{g4} + F_{p_m}.T_p + F_{ucp_m}.T_{ucp} = T_o \ldots\ldots(12)$$

The equations (3) , (6) ,(8) and (11) can be writing in mathematical form as:

$$\mathbf{a_{j,1}\,T_{air} + \sum\nolimits_{i=1}^{4} a_{j,i+1}\{T_{wi} + T_{gi}\} + a_{j,i+1}T_p + a_{j,i+1}T_{ucp} = b_j} \ldots\ldots(\mathbf{13})$$

Where the value of subscript (j) is equal to (1) for air heat balance equation, for heat balance equation of the four walls of the room is equal to (2 to 5), for heat balance equation of the glasses in the four walls of the room is equal to (6 to 9) for heat balance equation of the uncooled roof is equal to (10) and the equation of comfort temperature is equal to (11).

We can assemble these equations into matrices, so that the algebraic problem have be solve, can be expressed as:

[A] · [T] = [B]

$a_{1,1}, \ldots\ldots\ldots, a_{11,11}$= is a coefficients of temperature for matrix

T= is vector containing the temperatures

b= is vector containing the constant coefficients

So, the matrix equation when fully expanded looks like:

$$\begin{pmatrix} a_{1,1} & a_{1,2} & a_{1,3} & a_{1,4} & a_{1,5} & a_{1,6} & a_{1,7} & a_{1,8} & a_{1,9} & a_{1,10} & a_{1,11} \\ a_{2,1} & a_{2,2} & a_{2,3} & a_{2,4} & a_{2,5} & a_{2,6} & a_{2,7} & a_{2,8} & a_{2,9} & a_{2,10} & a_{2,11} \\ a_{3,1} & a_{3,2} & a_{3,3} & a_{3,4} & a_{3,5} & a_{3,6} & a_{3,7} & a_{3,8} & a_{3,9} & a_{3,10} & a_{3,11} \\ a_{4,1} & a_{4,2} & a_{4,3} & a_{4,4} & a_{4,5} & a_{4,6} & a_{4,7} & a_{4,8} & a_{4,9} & a_{4,10} & a_{4,11} \\ a_{5,1} & a_{5,2} & a_{5,3} & a_{5,4} & a_{5,5} & a_{5,6} & a_{5,7} & a_{5,8} & a_{5,9} & a_{5,10} & a_{5,11} \\ a_{6,1} & a_{6,2} & a_{6,3} & a_{6,4} & a_{6,5} & a_{6,6} & a_{6,7} & a_{6,8} & a_{6,9} & a_{6,10} & a_{6,11} \\ a_{7,1} & a_{7,2} & a_{7,3} & a_{7,4} & a_{7,5} & a_{7,6} & a_{7,7} & a_{7,8} & a_{7,9} & a_{7,10} & a_{7,11} \\ a_{8,1} & a_{8,2} & a_{8,3} & a_{8,4} & a_{8,5} & a_{8,6} & a_{8,7} & a_{8,8} & a_{8,9} & a_{8,10} & a_{8,11} \\ a_{9,1} & a_{9,2} & a_{9,3} & a_{9,4} & a_{9,5} & a_{9,6} & a_{9,7} & a_{9,8} & a_{9,9} & a_{9,10} & a_{9,11} \\ a_{10,1} & a_{10,2} & a_{10,3} & a_{10,4} & a_{10,5} & a_{10,6} & a_{10,7} & a_{10,8} & a_{10,9} & a_{10,10} & a_{10,11} \\ a_{11,1} & a_{11,2} & a_{11,3} & a_{11,4} & a_{11,5} & a_{11,6} & a_{11,7} & a_{11,8} & a_{11,9} & a_{11.10} & a_{11,11} \end{pmatrix} \begin{pmatrix} T_{air} \\ T_{w1} \\ T_{w2} \\ T_{w3} \\ T_{w4} \\ T_{g1} \\ T_{g2} \\ T_{g3} \\ T_{g4} \\ T_{cp} \\ T_{ucp} \end{pmatrix} = \begin{pmatrix} b_1 \\ b_2 \\ b_3 \\ b_4 \\ b_5 \\ b_6 \\ b_7 \\ b_8 \\ b_9 \\ b_{10} \\ b_{11} \end{pmatrix}$$

The values of the coefficient of the matrix are shown in appendix (1).

4. Cooling Load on The Cooling Panel

The panel was installed in the ceiling and exposed to every things in the room, so, the advance system (radiant panel) absorbed the heat from walls, windows, and from internal heat such as (equipment ,persons and lighting by radiation and also, by convection with room air , so the cooling load on the panel can be expressed as:

$$\dot{Q}_{cooling} = \dot{Q}_{inload} + \dot{Q}_{vent} + \sum_{i=1}^{n=4} A_{wi} h_{con.wi_{air}} (T_{air} - T_{wi}) + \sum_{i=1}^{n=4} A_{gi} h_{con.gi_{air}} (T_{air} - T_{gi}) + A_{unp} h_{con.ucp} (T_{air} - T_{ucp})$$

5. Experimental Work

Full-scale experimental office room was chosen in the Mechanical Laboratory Department of Mechanical Engineering in Karbala Technical Institute was chosen for an experimental work to test the radiant cooling panels as air-conditioning system in Iraq climate. The details and specifications of the test room Figure (2) are listed below:

Test room dimensions are 6.20 m length× 4.60 m width × 3.10m height and locate in Karbala city- Iraq where Lat: 32.59, Lon: 44.01faced north. Two walls and roof of test room are exterior and others exposed to un conditioning

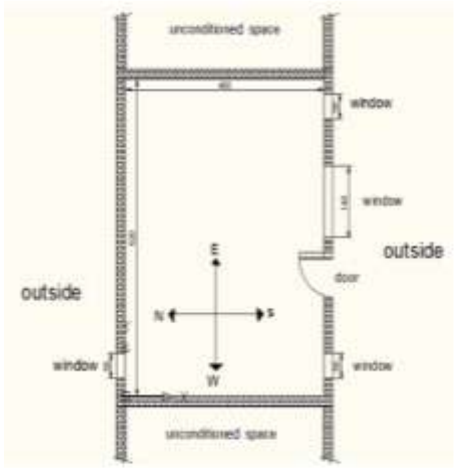

Figure (2) view of test

rooms, and the floor is in direct contact with the ground as shown in Figure (2). The wall construction is from 8 cm poured concrete of walls plastered with cement mortar on both surfaces.

- Number of window is four, one is large and others are small, the large one is of dimension 140cm ×100cm, and others have the same dimension of 50cm×50cm.All windows are with steel sash at single glazing 3.2 mm glass width with shading.
- Door construction is from two plates with 25 cm insulation between them of 1.2m length and 2.25m height.
- The radiant panels constructed of several numbers of plates, (79 plates) with dimension (60×60) cm, are mounted at 2.5m height from the floor level.

  The fallowing parameters used in this experiment are:
- The water flow rate on header pipe supply to the panels was 0.9 L/s.
- The temperature of water supply to the panels was19.5^{o}Cand return was 23^{o}C.
- Operating time was 24 hours starting from 10 to 31 August 2011.
- Ventilation : dedicated 100% of outdoor fresh air at rate 120m^3/h
- The water flow rate on the each ventilation cooling coils is 0.15 L /s
- Outdoor humidity approximately is 15% and winds is NW at 10 to 20 mph.
- Internal loads of the test room were (2 persons with 100W each, and equipment with 300W).
- Total solar radiation on horizontal surfaces which are obtained directly from Iraqi Meteorological Organization are shown in figure (3), which indicate that the differences between year's values are too small, and then any of these values can be used.

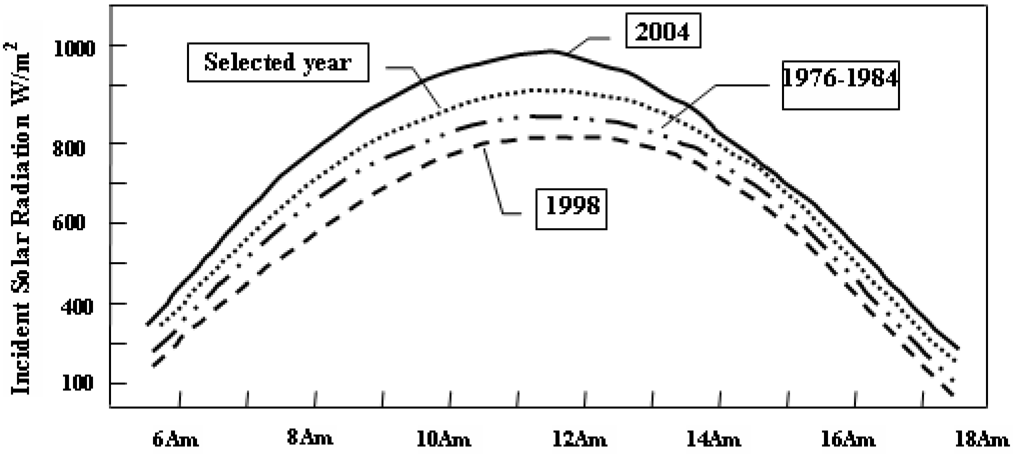

Figure (3) Hourly mean value of solar radiation on horizontal surface for different year

6. Results and Discussion

The amount of ventilation air flow rate is $120m^3/h$ passing through two coils before entering the room test, 100% of ventilation air, so, the supply air temperature is influenced by outdoor temperature and cooling coils temperature.

Figure (4) represents compression between the theoretical and experimental average room air temperature that be measured at 1.1m above the floor, in 17 August 2011**,**for each hour of daily time.

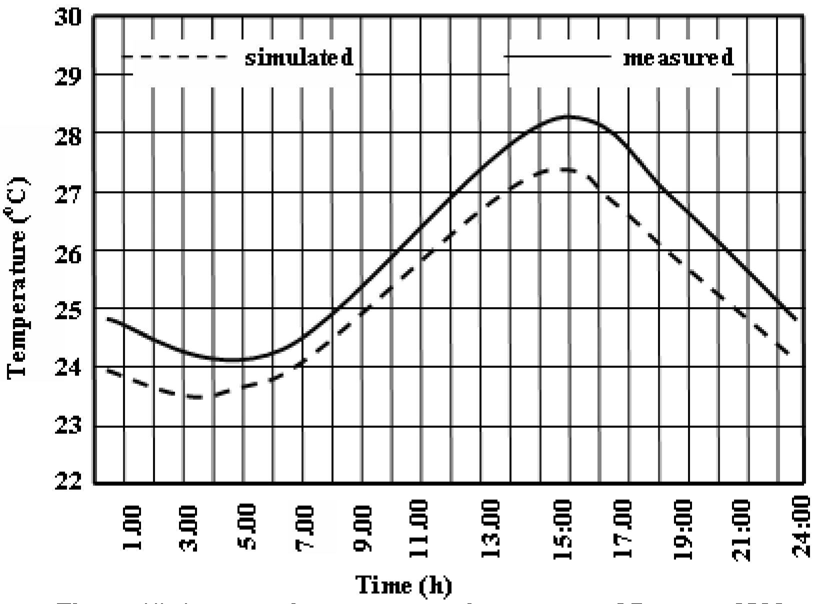

Figure (4) Average air temperature in test room, 17 august 2011

It can see that from 6:00 PM to 6:00 AM the temperature started to degrease because the decrease in external load (sun and outdoor supply air) and after 6:00 AM the temperature become to rising according to the increase in external load until rich to the maximum temperature at peak in 15:00 PM, instead of 3:00 PM that mean, The thermal storage capacity of the radiant cooling systems helps to shift the peak cooling load to later hours. Because of the energy transport, this cooling system has the potential to interact with thermal energy storage (TES).

Figure (5) represents the theoretical and experimental average room air temperature at 1.1m above the floor for each hour, in continuous seven days in August 2011, and the maximum

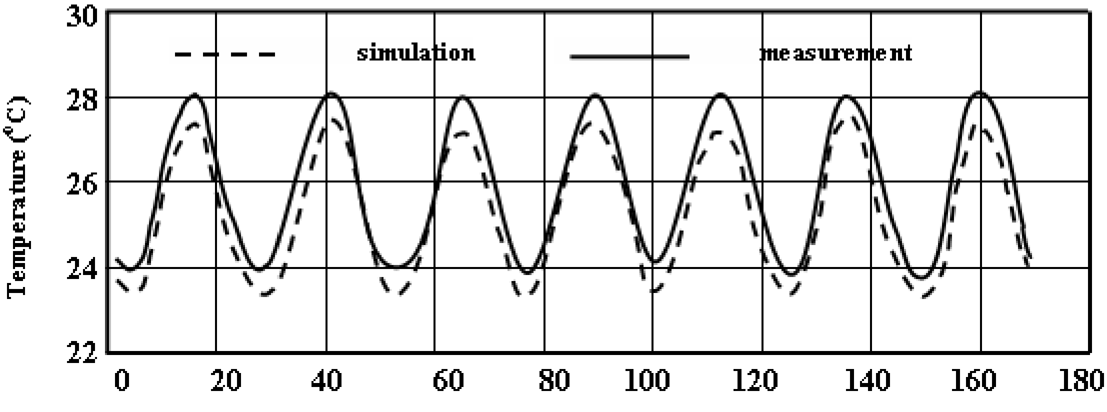

Figure (5) Measured and simulated air temperature in test room, 16-22 august 2011

deviation between the theoretical and experimental result is within 1°C.

The maximum temperature differences between the mean radiant temperature and room air temperature is 2°C, as shown in Figure (6). Also from this figure, the temperature of border testing room (T_{mrt}) is affected by room load. From 6:00 PM to 6:00 AM the mean radiant temperature started to decreasing because the decreasing the room load and after 6:00 AM the temperature become to rising according to the increase in external load, until rich to the maximum temperature at 5:00 PM, the reason for that is thermal storage inside the walls and it cause the peak load shifting to 5:00 PM.

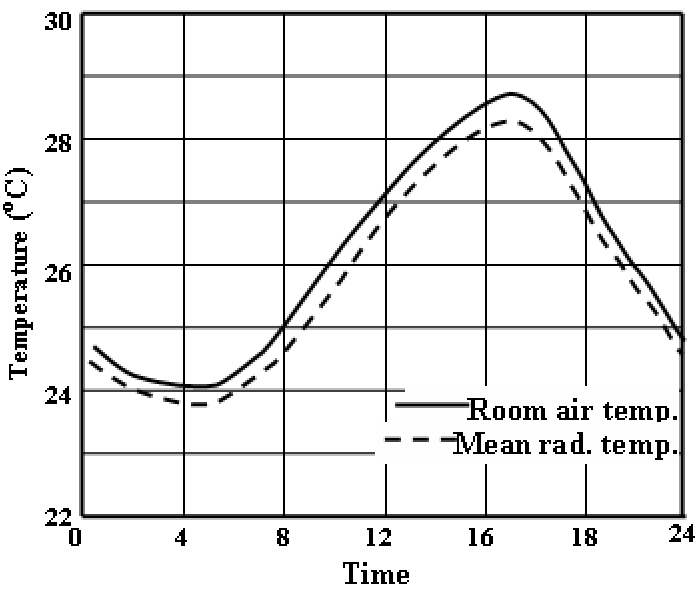

Figure (6) Experimental mean radiant and air temperatures of the test room.

As the inside temperature of the walls surface has a great effect on the thermal comfort of persons in the conditioned space, the radiant system succeeded in making the inside surface temperature of the walls were very closed and the temperature difference between the other areas of surrounding room and room air was little. Therefore, no local thermal discomfort inside the testing room and the thermal environment of the testing room was maintained in good condition. Figure (7) represents the measured mean radiant and air temperature of the test room form 16 to 22 August 2011.

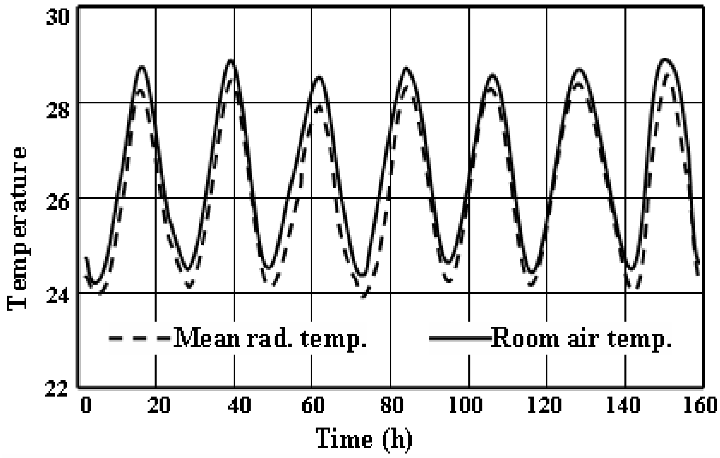

Figure (7) measured mean radiant and air temperatures of the test room, 16-22 August 2011

Figure (8) represents the theoretical and experimental average surface temperature of whole cooling ceiling panel system in test room for each hour, in continuous seven days in August 2011.

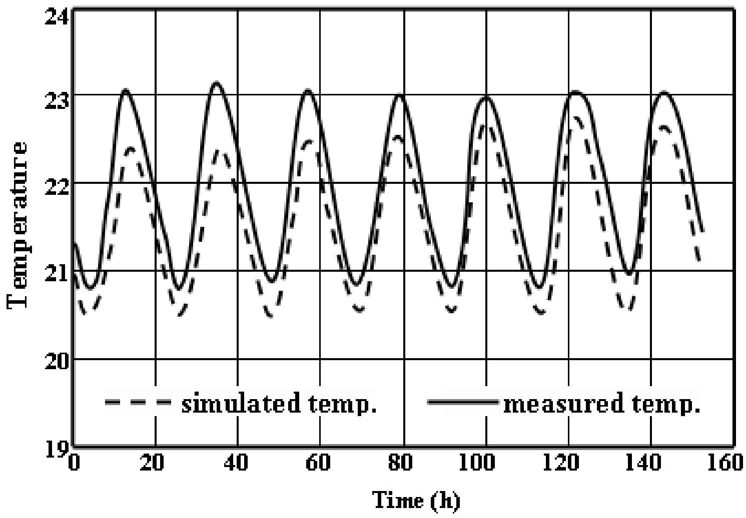

Figure (8) Surface temperature of cooling panel in test room, 16-22 August 2012

From these figure it can
be seen that when the radiant cooling panel, cooled with 19.5^{o}C water temperature can maintain the panel surface temperature of the radiant panel between 21^{o}C and 23^{o}C which is below the room air temperature with 5^{o}C.

Figure (8) shown the theoretical panel temperature agree well with the measurements, also with those of other researchers such as **Stetiu** in California in 1999[11], **Xuemin Sui** in China 2006 [13] and **Akbulut** in Yildiz Technical University in Istanbul 2011[14].

Thermal comfort is treated as the PMV-PPD to testing thermal comfort of the test room for each hour of daily time as shown in Figure (9, 10) respectively.

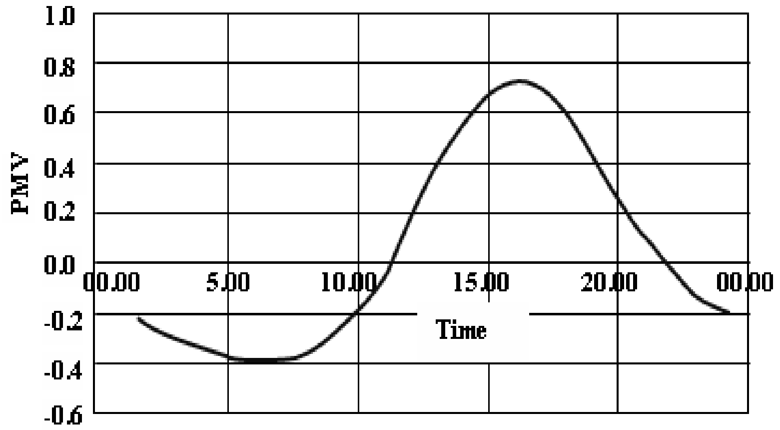

Figure (9) Experimental estimated of PMV in test room, on 20 August 2011

From 6:00 PM to 6:00 AM The PMV (0.6clo, 1.0met) started to decrease because decreasing in both the mean radiant temperature and room air temperature and after 6:00 AM the PMV begin to rising according to increasing in both the mean radiant temperature and room air temperature, The maximum value of PMV is 0.73 occurs at peak load in 5:00 PM and the minimum value of PMV is -0.44 and according to the thermal sensation ASHRAE scale the test room in comfortable sensation for each hour of daily time.

The PPD is function of PMV, and from PPD has been conclude only 16% of occupants are dissatisfied at peak load.

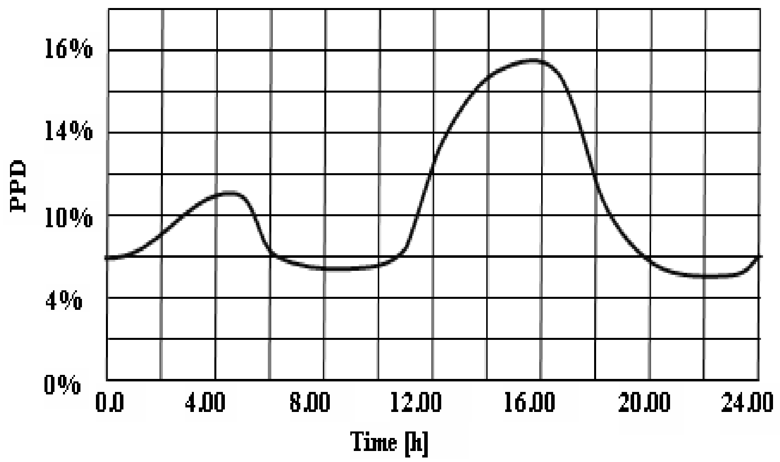

Figure (10) Experimental estimated of ppd in test room, on 20 August

7. CONCLUSIONS

From the analytical model results several conclusions can be drawn, which are validated by the experimental results and observations, the following conclusions are deduced:-

1. As the inside temperature of the walls surface (T_{mrt}) has a great effect on the thermal comfort of persons in the conditioned space, the radiant system succeeded in making the differences between the mean radiant temperature and room air temperature is 2^oC,Therefore, no local thermal discomfort inside the testing room.
2. Values of PMV-PPD are used to examine thermal comfort and it are between (-0.44 to 0.73) that mean, the use of radiant cooling system is appropriate for creating comfortable indoor conditions, furthermore little noise is created in space.
3. There is a 5^oC temperature differential between the room air temperature and radiant surface temperature.
4. The thermal storage capacity of the radiant cooling systems helps to shift the peak cooling load to 5:00 PM instead of 3:00 PM, The benefit from that is reducing energy consumption, because the officers are off, at peak load time.
5. Effective usage of renewable energy sources of country will provide economic and environmental benefits in terms of sustainable energy usage.
6. Experimental results reveal that energy saving can be obtained approximately 71% using radiant cooling system instead of the use of air conditioner, but the initial cost is approximately two times more than air conditioner.
7. The radiant cooling system can be applied to Iraqi buildings under the tropical climate and consumes less energy compared with the use of air conditioner.
8. The main factors currently limiting the use of system are lack of real technical of product and high price of this system.

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