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Heiselberg, Per Kvols

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Experimental and Numerical Analysis of a Heat Pump Driven Chilled Ceiling and Floor Heating Systems in a Test Room

Deniz Yılmaz[*1], Ali Ozyurt[2], Ozkal Sonmez[3]

[#]*Mechanical Engineering, Arel University, Marmara University, Istanbul University*
Türkoba, Erguvan Sk. 26 / K, 34537 Tepekent / Büyükçekmece/İstanbul/Turkey

[*1]denizy.80@gmail.com

[2]ali.oyzurt1293@gmail.com

[3]ozkalsonmez@gmail.com

Abstract

Passive cooling covers the natural processes of heat dissipation and heat gain operations and provides high efficient, comfortable environment around. As commonly known, when the air heated it rises, when cooled it descends. Natural cooling and heating use this physical property of air to provide the best thermal conditions.

In this study, an air to air heat pump system used for both floor heating and chilled ceiling operations was analyzed numerically and experimentally. Main objective of this study is to obtain COP of the heat pump system from refrigerant, hydronic and air sides. In order to calculate the air side capacity, counter loads installed into the room and the system was observed whether it provides comfort conditions. The test room was constructed of 120 mm PU panels to minimize the heat loss. All the data obtained from the system is recorded on a time period of per-second based during the processes. On the other hand, CFD analysis was performed with respect to the desired conditions on ANSYS Fluent 15. Air velocity, humidity and temperature distributions are analyzed inside the room.

Key words: Chilled Ceiling, Floor Heating, Heat Pump, CFD

1. Introduction

In our day, it is enormously important to achieve as much comfortable, sanitary as cost and energy efficient environment inside the office buildings, hospitals, education and state buildings. Passive chilled beams are gradually come into use for providing required conditions in this places.

The radiant cooling effect of the chilled ceiling surface and the convective cooling effect of air seeping down through creates a comfortable indoor environment.

Having been manufactured, these devices have to be tested in accordance with some standards in Turkey as well as Europe. One of that **EN 14518:2005** has been prepared by Technical Committee CEN / TC 156. This European Standard specifies test conditions and methods for the determination of the cooling capacity of chilled beams or other similar systems with free convection i.e without forced air flow.

According to some instructions in the test standard, the test room has been designed as 5 m length and 3 m width. Surrounding walls are insulated with 120 mm PU panels to minimize the heat loss. Three chilled beam devices which are at the same properties are suspended to the ceiling with 300 mm space. The test objects were also installed symmetrically about the centre ceiling and with the longest side parallel to the longest room side. Besides, 750 W, 1000 W and 1250W resistances are representing the heat loads inside room.

The first aim when testing the chilled beams driven by heat pump, is to investigate whether the devices provide required capacity in both summer and winter conditions. To obtain capacity values, hydronic line and air side temperatures were measured respectively. Room temperature is also investigated for keeping steady state conditions and controlling the hydronic unit. To calculate the COP of heat pump unit, the refrigerant capacity that gives the exchanger for conditioning the water was also calculated. Test room set values for summer **24° C %50 RH** and for winter **22° C %50 RH**.

Figure 1. 2700x 600 mm PCB Panels

Figure 2. Ceiling is fully insulated

Heat Pump Specifications and Schematic Diagram

Heat Pump operation goes by this way; water is been cooled through the exchanger and then pumped to the water tank. Compressor turn on and off according to NTC sensor inside the tank. Another pump circulates the water inside the tank to the devices. This unit is controlled according to the NTC sensor inside the room. ±2° K offset stated to reactivate the pump on this unit. Heat Pump operation diagram is shown in ***Figure 3.***

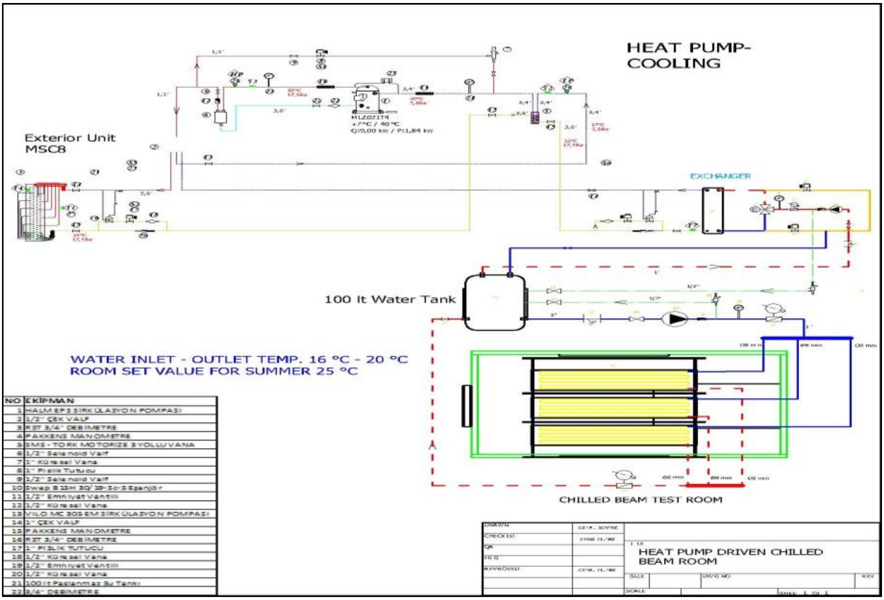

Figure 3. Heat Pump and Hydronic Unit Schematic Diagram

Determining the cooling load

Three different loads are come into account when calculating the cooling load. These loads are simulated by 3 electrical resistances **(*Figure 4*)** with same capacities and 2 of them are placed on sides of the room and one in the middle of the room.

Figure 4. Counter loads

Chilled beam devices first have to meet the cooling requirement in summer condition. When heat gain is calculated, a typical insulated wall was taken into account, shown in ***Figure 5***. Specific heat gain passing through the walls was approximately calculated (Table 1). Enviroment conditions were taken **34° C for Istanbul in summer season**. In addition, miscallenous factors that contribute heat gain (e.g. lighting, sun energy, people, electronic equipments) were added and total heat gain was calculated to simulate those inside the test room. Also a humidifier was placed to provide the latent heat generated by human activation and environment. Simulated heat gain table is shown following.

HEAT GAIN TABLE							
Structure Elements		Thickness of the Structure element d (m)	Conductivity λ (W/m.K)	d/λ,1/α (m2.K/W)	Coefficient of Heat Transfer U (W/m^2.K)	Surfaces of Heat Loss A (m2)	Heat Loss A X U (W/K)
Wall Elements	$1/\alpha_i$			0,13			
	Plaster	0,02	0,87	0,023			
	Horizontal coring brick	0,19	0,45	0,42			
	Insulation material	0,06	0,04	1,5			
	Plaster	0,005	0,87	-			
	$1/\alpha_e$			0,04			
Sunlight energy through windows							33
Latent Heat (Humidity)							8
Other loads In the room							20
TOTAL				2,11	0,47	70	94
						ΔT	10
						HEAT GAIN (W)	940

Table 1. Heat Gain table with respect to insulated wall.

The only heat loss occured is through the room panels to the enviroment. Test room panels consist of **120 mm Polyurethane foam** of the coefficient of heat transfer is **0,03 W/m.K.** Heat loss calculations were assumed that through the 6 sides of the room and depicted in ***Figure 6***. Enviroment conditions are based on the temperature of the day when test operations run out.

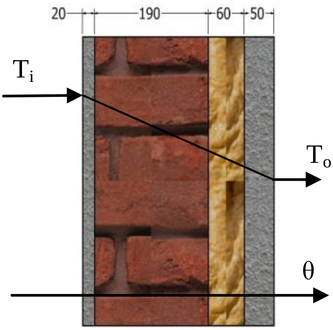

Figure 5. Wall section

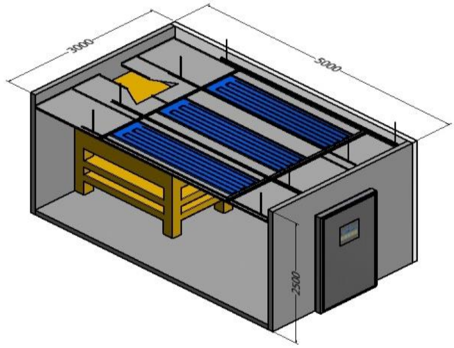

Figure 6.Test Room

Hence, enviroment temperature is **13°C** at that day. As we said before, room summer set value is **24°C %50 RH**. Heat transfer through the panels according to these informations;

A; Heat transfer area	55 m^2
U; Coefficient of Insulation	0,03 W/m.K
ΔT; Temperature differance	24-13 = 11° K
Q_P ; Total Heat Loss (U.A.(ΔT)/l)	**165 W**

Test Methods

The cooling capacity of the test object was determined from measurements of the cooling water flow rate and cooling water temperature rise under steady state condition. The cooling capacity was presented as a function of the temperature differance between the referance air on coil temperature and the mean cooling water temperature.

Measurements

The measurements shall be carried out at steady state conditions with at least 3 different ΔT: 6° ± 1° K, 8°± 1° K, 10° ± 1° K for each flow rate.

The measurements were carried out with at least 20 recordings under steady state conditions. For the internal heat supply method the temperature of the test room's inner walls, floor and ceiling shall be controlled and maintained uniform at any level necessary to maintain a maximum temperature differance between these surfaces and the air on coil temperature of less than 1,0 K during the measurements. The probes placed on the beams are shown in ***Figure 7***

Also a humidifier was monitored and controlled along the process to calculate the latent heat shown ***Figure 8***.

Figure 7. Probes

Figure 8. Hygrostat

Results

The cooling capacity of the beams were calculated from the equation:

$$P = \dot{m} \cdot c_p \cdot \Delta T_w \qquad (1)$$

The specific cooling capacity of the chilled beams was calculated from the equation:

$$P_L = P / L \qquad (2)$$

The specific cooling capacity was plotted in a diagram as a function of the temperature differance (ΔT) between the on coil temperature (referance room temperature) and the mean cooling water. This shows that as temperature difference rises, we may obtain more cooling capacity from the beams.

Results of Measurements		Measurement Points		
Number		1	2	3
Date		15.1.2016	15.1.2016	15.1.2016
Cooling Water flow rate	$\dot{m}$	0,1307 kg/s		
Water Inlet	T_{in}	15,5° C	17,1 ° C	19 ° C
Water Outlet	T_{out}	17,2 ° C	18,5 ° C	19,9 ° C
Globe	T_g	25,3° C	25,1° C	25,4° C
Air - 1,7 m	$T_{a\ 1,7}$	25,6° C	25,4° C	25° C
Air - 1,1 m	$T_{a\ 1,1}$	25,9° C	25,4° C	24,8° C
Air - 0,1 m	$T_{a\ 0,1}$	25,2° C	24,7° C	24,5° C
Surface Inside Ceiling	$T_{ceiling\ 1}$	24,9° C	24,9° C	24,8° C
Air on coil	T_{ac}	19,8° C	20,6° C	21,2° C
Air Temperature 0,75 m below beam	T_{air}	18,4° C	19° C	19,5° C
Main air Velocity 0,75 m below beam	V_{air}	0,23 m/s	0,18 m/s	0,15 m/s
Heat Loss During Steady State	Q_p	165 W		
Total Cooling Capacity	P	928 W	764 W	491 W
Heating Capacity (Counter Loads)	P	1250 W	1000 W	750 W

Table 2. Measurements of the Beams

Cooling capacity of the beams was calculated in accordance with the standards and measurements are shown in ***Table 2***. Also specific capacities needed for the beams are shown ***Table 3***. All these values are drawn in ***Figure 9***.

Accordingly, the chilled beam devices estabilished in a test room, performed for cooling in summer conditions. The needed heat gain was provided by counter loads placed in three differrent points in the room. Tests were sccomplished when the room temperature stays in steady state conditions in 5 minutes and measured 20 times during this condition.

Calculations on Measurements		Measuring point		
No.		1	2	3
Reference Temperature	T_{ac}	25,3 ° C	25,1 ° C	25,4 ° C
Water temperature rise	ΔT_w	1,7 K	1,4 K	0,9 K
reference - water mean	ΔT	9,8 K	8 K	6,4 K
specific; test room area	P_{ht}	-61,8 W/m^2	-50,9 W/m2	-32,7 W/m2
specific; total lenght	P_{lt}	-343,7 W/m	-282,9 W/m	-181,85 W/m
specific; active lenght	P_l	-386 W/m	-318,3 W/m	-204,5 W/m
total	P	-928 W	-764 W	-491 W
Heat transfer test room periphery	P_t	-160 W		
Heat balance	ΔP	-162 W	-76 W	-99 W

Table 3. Specific and total capacities

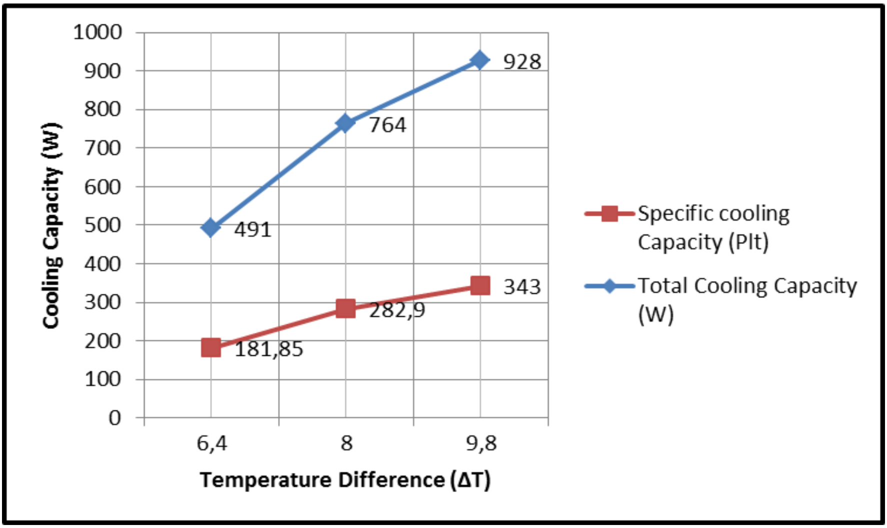

Figure 9. Cooling Capacity Range with respect to temperature difference

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