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Experimental Determination of the Optimum Radiant to Convective Heat Transfer Split for Hybrid Heating System

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Abstract

In this study, radiant to convective heat transfer split of a hybrid heating system investigated experimentally in order to obtain energy-optimum split while operative temperature was kept constant. Operative temperature is fundamental metric of thermal comfort temperature and it is a function of mean radiant temperature and dry-bulb air temperature. Experiments were conducted in a special test chamber that was set up according to ANSI/ASHRAE Standard 138. Interior surface temperature of each wall and air temperature of the test chamber can be controlled independently. Electric fan heater and radiant floor heating were hybridized with different capacities. To obtain optimum radiant/convective split; operative temperature were kept constant for a data set, radiant and convective systems were simultaneously operated with different capacities. According to the results, optimum radiant/convective split interval is obtained between 0.65 and 0.75.

Keywords - hybrid heating hystems; load-sharing HVAC systems; radiant-convective split; thermal comfort, operative temperature

1. Introduction

HVAC loads constitute an important proportion of total building energy consumption. Moreover, indoor thermal comfort requirements become more and more important for modern life and reducing adverse effects of energy and HVAC systems become a real necessity. Sustainable, comfortable and low exergy HVAC systems designs can be possible with integration of different sub systems [1] [2].

Studies shows that, decoupling sensible and latent HVAC loads and assign them relevant sub-systems is a good approach to obtain next generation low-exergy systems. Decoupling the loads and supply them from relevant sub-system enable to maximize thermal comfort and minimize power consumption [3] [4] [2]. Thus, next generation HVAC systems will probably be more complex in order to achieve multidimensional design goals such as thermal comfort, low energy consumption and low exergy destruction.

ASHRAE Handbook of HVAC Systems and Equipment defines hybrid HVAC systems as below [4]:

"*Panel systems may be combined either with a central forced-air system of one-zone, constant-temperature, constant-volume design, or with dual-duct, reheat, multizone or variable-volume systems, decentralized convective systems, or in-space fan-coil units. These combined systems are called hybrid (load-sharing) HVAC systems.*"

As can be seen, scope of the definition is very wide, in this study, radiant floor heating combined with in-space electric fan heater.

Kilkis et al. designed a hybrid system for The Ankara Museum of Ethnography (Turkey) in 1995. They investigated the optimum radiant to convective heat transfer split for hybrid HVAC systems theoretically. According to their results, optimum radiant/convective split for the hybrid HVAC system is 0.6 [3] [5]. In a single story adobe house, radiant/heating and cooling panels, convective ventilation and dehumidification systems were coupled within the scope of ASHRAE RP-1140 [6]. In 2006, Kilkis developed a composite (hybrid) radiant wall panel (CRWP) which is able to heating via both radiant and forced-convective heat transfer simultaneously and can be directly and completely supplied by low-enthalpy energy resources. According to the study, CRWP system provides increase of rational exergy management model efficiency up to 80% [7] [2]. In 2013, researchers from ETH-Zürich and Future Cities Laboratory (Singapore) established the BubbleZERO laboratory which is included combination of radiant and convective systems which can be operated simultaneously with a special control system [8] [9]. None of the above-mentioned studies, except which were done by Kilkis et al. in 1995 [3] and by Kilkis in 1999 [5], not investigated radiant/convective split of the combined systems. These two studies investigate radiant/convective split theoretically. In this study we focused on investigate the energy-optimum radiant/convective split for hybrid heating system experimentally.

Another important studies, from the thermal comfort aspect, was investigated from Shukuya [10] [11] [12]. Results of the studies give strong cues for importance of the hybrid HVAC systems and optimization of the radiant/convective split in order to maintain thermal comfort properly. According to the studies, which is given in Fig. 1, minimum exergy consumption within human body occurs on a specific resultant of mean radiant temperature and air temperature. In other words, minimum exergy consumption within human body occurs on an optimum radiant/convective split. Hybrid HVAC systems are able to control both of these temperatures separately and they can also control this split properly. In this study, we aimed to obtain a power-optimum split and assess our optimum split according to minimum exergy consumption within human body.

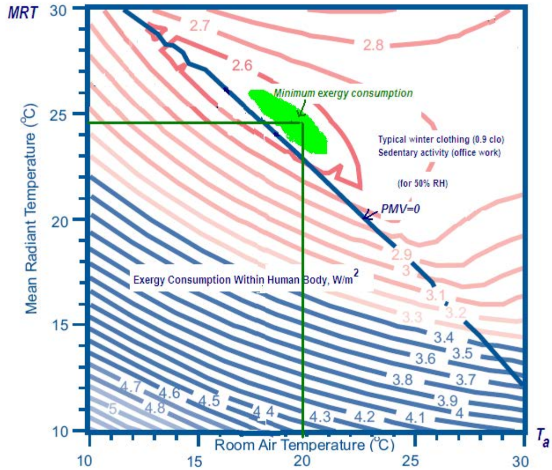

Fig. 1 Minimum exergy consumption within human body [10] [11] [12]

Hybrid HVAC systems are complex and design of them requires a holistic approach, which is consider thermal comfort, efficiency, low energy and exergy consumption, economy and sustainability together. Moreover each of this parameters is multidimensional, for example to maintain thermal comfort, designers have to examine relative humidity, air velocity, temperature homogeneity and other required parameters simultaneously. This study aimed to point out an optimum radiant to convective split ratio for hybrid heating systems. This results has strong cue for initial point of further iterative multi-objective optimization.

2. Theory

PR is the radiant heat transfer to total heat transfer split, it can be calculated with (1) [3].

$$PR = \frac{q_r}{q_r + q_c} = \frac{\begin{matrix} Sensible\ load\ assigned \\ to\ panel\ system \end{matrix}}{Total\ sensible\ load} \tag{1}$$

The specific power consumption ratio, *SPC*, is the ratio of power consumption of auxiliaries such as fans, pumps, motors to total sensible load satisfied by the system [3].

$$SPC = \frac{\begin{matrix} Power\ required \\ to\ run\ auxiliaries \end{matrix}}{\begin{matrix} Sensible\ load \\ satisfied\ by\ the\ system \end{matrix}} \tag{2}$$

Operative temperature, T_o, is the average of mean radiant temperature and dry-bulb air temperature. Equation of the operative temperature is given in (3) [13] [14].

$$T_o = \frac{T_{mr} + T_a}{2} \tag{3}$$

In (3), air temperature can be measured by a proper sensor and there are several ways to calculate mean radiant temperature. In a prior study we have compare different calculation methods in the test chamber and results of methods were consistent [15]. In this study mean radiant temperature, T_{mr}, was calculated by using (4). Here T_{gl} is temperature measured by the black globe sensor, V_a is air velocity, e is the the black globe surface emittance (0.95) and D is diameter of the black globe (0.15m) [16].

$$T_{mr} = \left[\left(T_{gl} + 273\right)^4 + \frac{1{,}10 \times 10^8 V_a^{0{,}6}}{eD^{0{,}4}} \left(T_{gl} - T_a\right) \right]^{1/4} - 273 \tag{4}$$

3. Experimental Setup

In this study, experiments were conducted in a special test chamber, which were established according to the ANSI/ASHRAE Standard 138 with some minor differences. These differences effects only standard panel performance rating. Enumeration of the test chamber walls and schematics of the integrated hydronic heating/cooling pipes of the surfaces are given in Fig. 2.

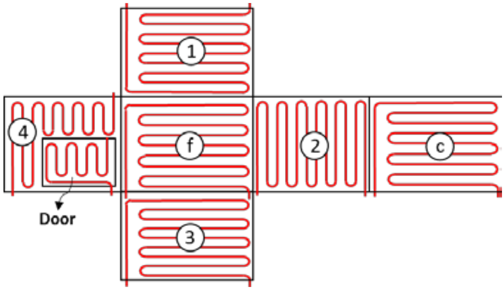

Fig. 2 Enumeration and hydronic pipe schematics of the test chamber walls

Interior surfaces of the each wall are coated with aluminum sheets that have 0.9 thermal emissivity. Thermal resistance of insulation for all walls, except door, is 2.7 m^2K/W, thermal resistance of door insulation is 2 m^2K/W.

Each wall, floor and ceiling designed as a dedicated zone, in order to control their interior surface temperatures independently. Inlet water temperature of each zone is controlled with 3 automatic control valves. Totally 18 on/off controlled with 10 second positioning time automatic control valves are used for zoning in the facility. Chilled water demand supplied by an 8 kW air source heat pump and stored at, 200 liter, cold water tank. Hot water demand supplied by 3 kW electric boiler, which have also 200 liter water capacity. Fig. 3. shows a view of experimental setup.

80 calibrated K-Type thermo-couple were attached to the interior surfaces of the chamber in order to measure temperature distribution and average surface temperature of each surface.

A hot wire anemometer and a black globe sensor were located in the center of volume to measure required parameters for mean radiant temperature. Vertical air temperature sensors, hot wire anemometer and black globe sensor can be seen in Fig. 4.

Two different type portable electric fan heaters used for forced convection heating in the test chamber. First type of the heater has 800 W heating capacity and 110 W fan power, second type of the heater has also 800 W heating capacity and 10W fan power.

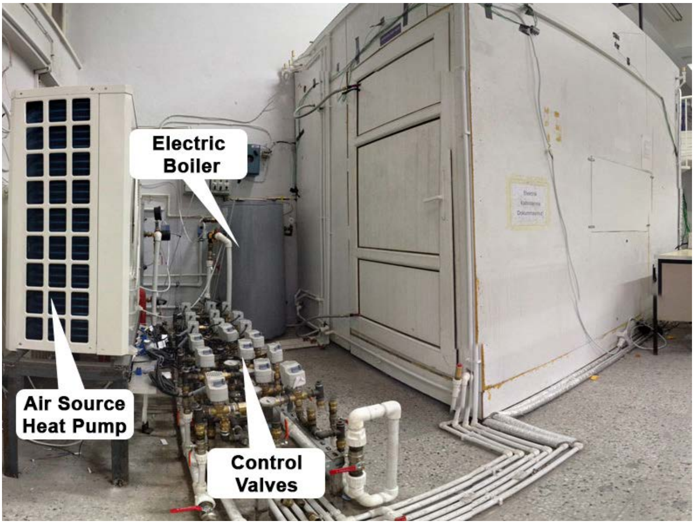

Fig. 3 A view of experimental setup

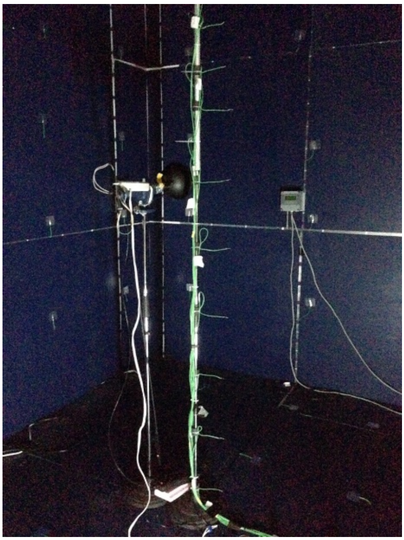

Fig. 4 Black globe, hot-wire anemometer and vertical temperature sensors

4. Experiments

In this study, stand-alone radiant floor heating, stand-alone electric fan heating experiments were conducted. Then radiant floor heating system and electric fan heater were operated together with different capacities in order to obtain different radiant/convective split for hybrid heating system. In order to analyze effects of fan capacity, experiments were repeated with second type, low fan power, of electric fan heater. Experiments, which were hybridized, second type fan heater and radiant floor heating is named as hybrid heating system. Table 1 shows control system set points for each surface.

Table 1. Set-points for experiments

Set-Points of Interior Surface Temperatures [°C]																				
Radiant							**Hybrid**							**Convective**						
1	**2**	**3**	**4**	**C**	**F**	**T_a**	**1**	**2**	**3**	**4**	**C**	**F**	**T_a**	**1**	**2**	**3**	**4**	**C**	**F**	**T_a**
18	18	18	15	18	21	-	20	20	20	17	20	23	22	15	15	15	15	15	15	18
20	20	20	17	20	23	-	20	20	20	17	20	23	22	17	17	17	17	17	17	20
22	22	22	19	22	25	-	22	22	22	19	22	25	24	19	19	19	19	19	19	22
18	18	18	15	18	21	-	24	24	24	21	24	27	26	19	19	19	18	19	19	22
20	20	20	17	20	23	-	26	26	26	23	26	29	28	21	21	21	20	21	21	24
18	18	18	15	18	21	-	28	28	28	25	28	31	30	23	23	23	22	23	23	26
20	20	20	17	20	23	-	20	20	20	17	20	22	24	25	25	25	24	25	25	28
22	22	22	19	22	25	-	22	22	22	19	22	24	26	27	27	27	26	27	27	30
24	24	24	21	24	27	-	24	24	24	21	24	26	28	27	27	27	26	27	27	30
24	24	24	21	24	27	-	26	26	26	23	26	28	32	29	29	29	28	29	29	32
26	26	26	23	26	29	-	22	22	22	19	22	26	24							
28	28	28	25	28	31	-	24	24	24	21	24	28	26							
28	28	28	25	28	31	-	26	26	26	23	26	30	28							
30	30	30	27	30	33	-	28	28	28	25	28	32	30							
32	32	32	39	32	35	-														

All data is the average of the measurements for 30 minutes period that were taken after system reached to the steady state conditions.

4.1. Outer Temperature Variation Around the Test Chamber

The test chamber is in a laboratory that is about 20 times larger than the test chamber. There are not any ventilating and air conditioning system and temperature control in the laboratory, thus laboratory dry-bulb air temperature, T_{lab}, is not constant. Thus, energy consumption of each experiment was adjusted proportionally for reference 16°C that is the average laboratory temperature, $T_{lab\text{-}ref}$, during the experiments [17]. Equation (5) is shows the proportional adjustment equation. Here, q_j, is the actual energy consumption and q_{adj} is adjusted energy consumption of the jth experiment. $T_{lab\text{-}j}$ and $T_{i\text{-}j}$ are, respectively, laboratory air temperature and test chamber air temperature of the jth test.

$$q_{adj} = q_j \frac{(T_{i-j} - T_{lab-ref})}{(T_{i-j} - T_{lab-j})} \tag{5}$$

5. Experimental Results

Radiant/convective split and specific power consumption diagram is given in Fig. 5. PR-SPC values and fitted second order polynomial curves is given in the diagram. Each trough each curve operative temperature is in the given interval. R^2 values of the each curve is higher than 0.95.

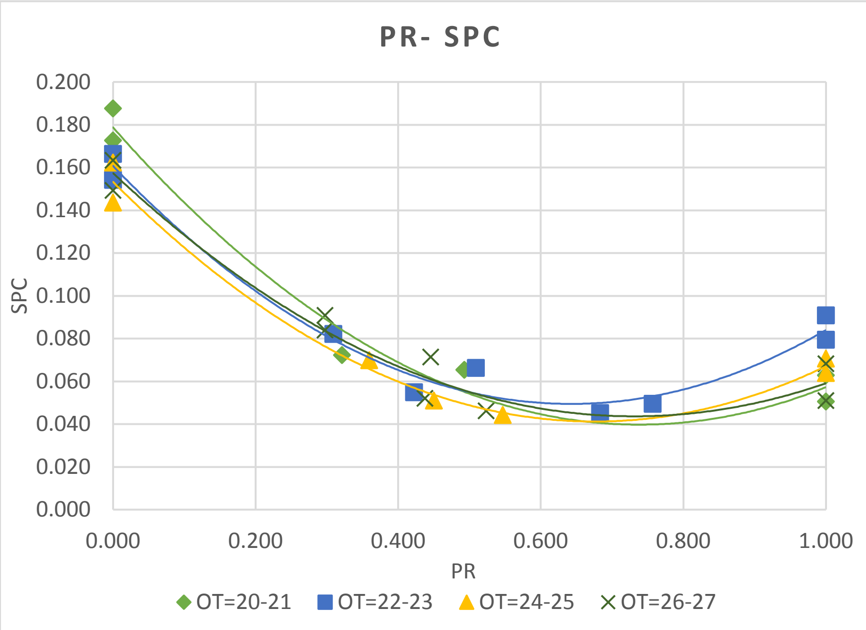

Fig. 5 PR-SPC optimisation diagram

Optimum PR is obtained between 0.65 and 0.75 and these values are consistent with the theoretical study of Kilkis et al. [3] [5]. While the hybrid heating system operated on the optimal PR, required fan capacity reduced with respect to stand-alone electric fan heating system and required pump capacity also reduced with respect to stand-alone floor heating system.

Optimum PR that obtained, is also provides minimum exergy consumption within human body according to the Fig. 1. Thus, optimum PR provides not only minimum power consumption but also maximum thermal comfort.

6. Discusscion and Conclusion

In this study, optimum radiant/convective split of the hybrid heating system was investigated experimentally to point out the benefits of hybrid systems. Optimum radiant/convective heat transfer split was obtained between 0.65 and 0.75. This result is compatible with previous theoretical studies [3] [5].

According to the results, decoupling of the latent and sensible loads and supplied them by relevant dedicated system is enable to supply all of the HVAC loads from renewable sources and /or waste heat.

In this study relative humidity wasn’t controlled and effects of the relative humidity was not investigated. It should be investigated in further studies.

Although systems are compared while they maintained same operative temperature, other thermal comfort parameters such as vertical temperature distribution etc. should also be analyzed more detailed according to ANSI/ASHRAE Standard 55 and/or ISO Standard 7730.

Supplying experimental studies with numerical and simulation based studies is necessary to analyze temperature homogeneity, velocity distribution and other related parameters for more location.

Acknowledgment

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