Experimental Analysis of Radiant Ceiling Systems Coupled to its Environment

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ABSTRACT

This paper presents the results of an experimental analysis of the radiant ceiling systems in both heating and cooling modes coupled to its environment (fenestration, walls, internal loads and ventilation system). The main objective of this study is to present the results of the systems performance and measurement methods used to evaluate the influence of some important parameters on the radiant ceiling capacity and comfort conditions.

Two test chambers are adapted in a way to reproduce as good as possible the characteristics of the real offices located in Brussels. Forty-six tests were performed in which the main objective was to observe the influence of parameters such as the mass flow rate, supply water temperature and thermal load distribution, fenestration and ventilation system effects. Laboratory test results show that the influence of surfaces temperatures inside the room, especially the facade, is considerable. Therefore, the system must be evaluated together with its designed environment and not as separate HVAC equipment.

1. INTRODUCTION

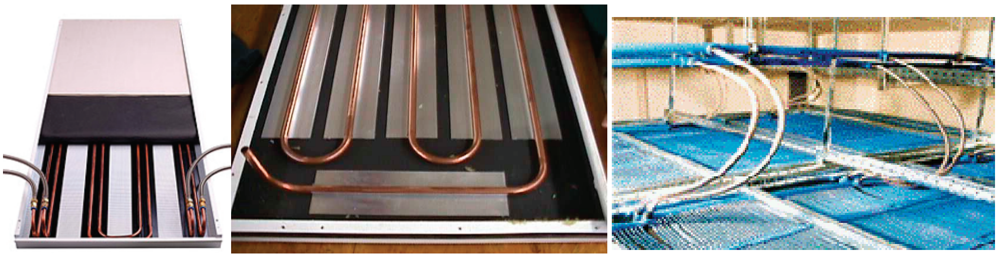
Radiant heating and cooling systems supply or extract heat from a room through the action of convective and radiative heat exchange between the room environment and heated or cooled panels situated in the ceiling. The radiation heat exchange can be calculated as a function of the room geometry and surface characteristics. The convective heat transfer is a function of air velocity and direction at the ceiling level (related to the position of the air inlet), which in turn depends on the room and diffusers geometry, the location and power of the internal heat sources and interaction with the heated or cooled facade.

The convective heat exchange (natural + forced) in a room equipped with a radiant ceiling becomes therefore a complex process. In the convective part, the existing correlations were developed from experimental measurement in specific conditions of ventilation and internal thermal loads (Alamdari and Hammond, 1983; Spitler *et al.,* 1997; Awbi and Hatton, 2000; Novoselac *et al.,* 2006). Some experimental studies were performed considering the individual influence of some of these parameters on comfort conditions: different load distributions effect (Behne, 1996), ventilation system effect (Kulpmann, 1993; Behne, 1999), facade effect (Fredriksson *et al.,* 2001). It can be only an approximation of the actual phenomenon. In the most recent works the simplifications concerning the convection treatment are implemented with CFD analysis, in which the radiation heat exchange has been studied by the computational analysis (Karadag, 2009; Corgnati *et al*., 2009; Catalina *et al*., 2009), however the experimental analysis remains too specific and facade effects are always neglected. This article summarizes the experimental investigation and analysis of two radiant ceiling systems with three different configurations. In a companion paper, a dynamic model of radiant ceiling panels in heating or cooling modes coupled to its environment is presented (fenestration, walls, internal loads and ventilation system) (Fonseca *et al*., 2010).

2. TEST BENCHES DESCRIPTION

2.1 Studied system

The system is studied here in two constructive versions, used in three configurations: copper tube and synthetic capillary tube mats (Figure 1). The first constructive version consists of a ceiling in which the copper coils are in direct contact with a smooth perforated metallic surface. The pipe-radiant panel contact must be established in such a way to get a minimum thermal contact resistance. Therefore an aluminum extrusion profile is used. A perforated plate insures suitable convective flow to improve its performance.



T1 T2 TG

Figure 1: Radiant ceilings description

For the three types of copper tube ceilings studied here, the system design is almost the same. The differences are only that the type one (T1) considers two synthetic glue layers, one between the tubes and the aluminum extrusion profile and another between this one and the metallic plate. Whereas the type two (T2) considers only one layer between the plate and the aluminum profile ensuring that the tubes are secured to the profile by pressure during its manufacturing process. Type three (T3) is similar to T2. The only difference is an additional layer of paper between the plate and the profile used as sound insulation. The second constructive version uses radiant mats consisting of numerous thin capillary tubes (Di = 2.3 mm) made of polyethylene and mounted in parallel (TG) (Figure 1). The distance between the individual small tubes is small enough to ensure that a homogeneous temperature is produced on the bottom side of the ceiling. The radiant mats in this system can be incorporated into the ceiling in three configurations: placed on the top of the metal ceiling panels with a layer of mineral wool installed above, embedded into a ceiling plaster layer, or stretched between insulation and gypsum plasterboard. The main characteristics of radiant ceiling testes are presented in Table 1.

Table 1: Main characteristic of the tested radiant ceilings

Characteristic	**T1**	**T2**	**TG**
Radiant surface	On top of a steel plate. Thickness:1.1mm	On top of a steel plate. Thickness 1 mm	On top of a steel plate. Thickness 0.8 mm
L_p: Panel length	1.8 m.	1.78 m	1.78 m.
W_p: Panel width	0.6 m.	0.52 m	0.53 m.
w_t: Tube separation	145 mm.	100 mm	10 mm.
Panel surface:	1.08 m^2	3.06 m^2	0.845 m^2
Perforated area (**ρ**)	25%	13.6%	16 %
N_s: Panels in series	6	2 or 4	2 or 6 (Heating or Cooling)
N_p: Panels in parallel	1 or 3 (Heating or Cooling)	1 , 2 or 3 (Heating or Cooling)	3
Upward insulation:	20 mm mineral wool.	40 mm mineral wool.	30 mm mineral wool.
Tube-radiant surface union system	Aluminum interconnection profile	Aluminum interconnection profile	Directly placed on top of the plate
D_e	12.5 mm	12 mm	3.4 mm
D_i	10.8 mm	10 mm	1 mm

The climatic chamber used during the tests was built in such a way to reproduce as accurately as possible the structure and characteristics of a large commercial building located in Brussels according to the experiment work developed by Bourdouxhe *et al.* (1998a; 1998b).

The climatic chamber (see Figure 2) is a wood structure, strongly insulated. It is divided into two principal zones by a facade: the first, called warm/cold facade simulates the exterior climate and the second contains the equipment to be tested and the hydraulic and ventilation systems used during the test.

Figure 2 Climatic chamber: Plan and lateral views. External and internal views

The facade space is set at a temperature that can vary between -10°C and 30°C. Very low temperatures can be reached by means of two refrigeration systems placed in this space. When it is necessary to heat up the facade, an electric convector is then used (Bourdouxhe *et al.,* 1998a; 1998b).

2.2 Measurements

Internal surface temperatures (facade, mobile wall into the facade, room internal walls, floor, room side active ceiling and active ceiling surface towards the ceiling void) are measured at 5 points distributed symmetrically in each surface. Air temperatures of the corridor, facade, chambers and ceiling void are also measured at 5 positions in each space. Air temperature and globe temperatures are measured in four positions into the chamber, and measurements are vertically placed at 10 cm, 75 cm, 150 cm and 200 cm above the floor.

The reference temperature is always taken at the center of the chamber at 75 cm above the floor. A maximal vertical variation of air temperature into the chambers of 1.5K is observed during the experimental tests in heating mode. In cooling mode the variation is around 0.5K.

3. EXPERIMENTAL RESULTS AND ANALYSIS

3.1 Thermal balances

3.1.1 Radiant ceiling balance: Because of the weak water temperature difference across the supply conduits, a balance from water mass flow rate and temperatures is too imprecise. Therefore the supply and exhaust conduits “losses” or “gains” to the ceiling void must be estimated.

The total thermal power of the radiant ceiling considers also the thermal exchange with the room and the ceiling void (Figure 3):

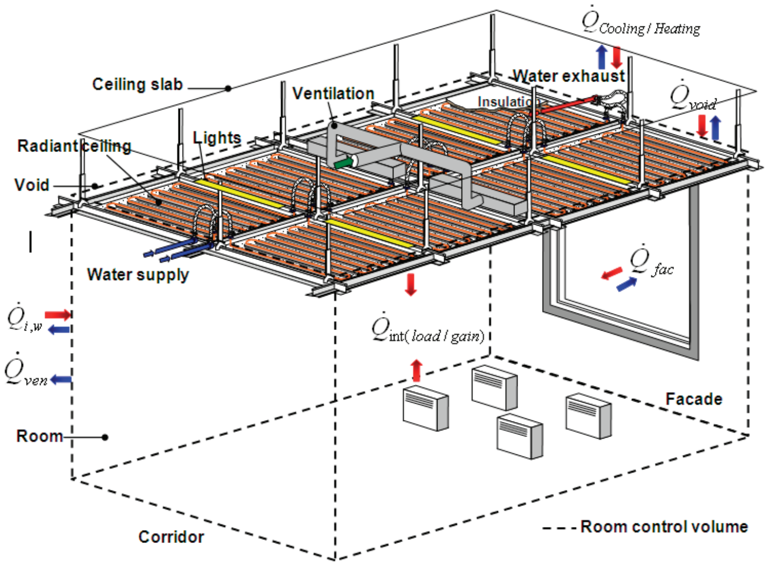


Figure 3: Control volume for the room global balance

$$\dot{Q}_{heating/cooling} = \dot{Q}_{su} + \dot{Q}_{con} \qquad [W] \tag{1}$$

where:

$\dot{Q}_{heating/cooling}$: is the total thermal power of the radiant system, [W]

$\dot{Q}_{su}$: is the total thermal power supplied or extracted by the system, [W]

$\dot{Q}_{con}$: is the conduits thermal lost (supply and return circuits) with the environment in the ceiling void, [W]

For heating mode:

$$\dot{Q}_{su} = \dot{Q}_{ER} + \dot{Q}_{cir} - \dot{Q}_{cor} \qquad [W] \tag{2}$$

Where:

$\dot{Q}_{ER}$: is the power dissipated by the electric resistance immersed into the water supply circuit, [W].

$\dot{Q}_{cir}$: is the electric power consumed by the circulators, [W].

$\dot{Q}_{cor}$: is the thermal exchange of the conduits with the environment in the corridor, [W].

The thermal test stability is such that the transients are negligible (temperature drift lower than 0.1K/h).
The total thermal power supplied to the system calculated from Eq.1 is precise and used as reference to calculate the radiant ceiling performance. As verification only:

$$\dot{Q}_{su} = \dot{M}_w \cdot c_w \cdot \left(t_{w,su} - t_{w,ex}\right) \qquad [W] \tag{3}$$

The water mass flow rate $\dot{M}_w$ is determined by counters placed inside the circuits.

In cooling mode, the electric resistance is not used. The cooling power is obtained by injection of cold water into the circuit. As well as in heating mode, the radiant ceiling power considers the thermal exchange with the room and the ceiling void.

In cooling mode the power supplied by the system can be calculated as:

$$\dot{Q}_{su} = \dot{M}_{w,in}.c_w.\left(t_{w,ex} - t_{w,su,in}\right) - \dot{Q}_{cir} + \dot{Q}_{cor} \quad \text{[W]} \tag{4}$$

where:

$\dot{M}_{w,in}$: is the water mass flow rate injected into the circuit and measured by weight, timing given water mass flow in an open loop, [kg/s].

$t_{w,su,in}$: is the water temperature injected into the circuit, [°C]

In this case, Eq. 3 can be also used to verify the estimation of the power supplied by the system.

It is important to note that for the systems tested, supply conduits thermal losses or gains into the ceiling void vary between 10% and 20% of the radiant ceiling power. The heat flow of the radiant ceiling upward to the ceiling void varies between 5% and 10% of the radiant ceiling power. This is not negligible, however in a real building this energy is not considered as a load, it is used also for heating or cooling the offices but its effect is perceived after some delayed time related to the thermal inertia of the ceiling slab connecting each level in the building (see Figure 3).

3.1.2 Room thermal balance: Figure 3 shows the control volume used to perform the room global balance and the concerned heat flows and powers.

The global thermal balances of the air in both heating and cooling modes:

$$\dot{Q}_{heating/cooling} + \dot{Q}_{void} + \dot{Q}_{i,gains} + \dot{Q}_{i,w} + \dot{Q}_{fac} + \dot{Q}_{ven} + \dot{U} = \dot{R}_{global} \quad \text{[W]} \tag{5}$$

Where the enthalpy flow rate extracted by the radiant ceiling system can be defined as follows:

$$\dot{Q}_{heating/cooling} = \dot{M}_{w,su} \cdot c_{p,w} \cdot (t_{w,su} - t_{w,ex}) \quad \text{[W]} \tag{6}$$

The internal thermal load (lighting and convectors) $\dot{Q}_{\text{int},loads}$ is measured directly by watt-meters.

The heat gain through internal walls is estimated as a function of the measured surface temperature of each wall and floor surfaces of the enclosure. The internal heat transfer coefficient is estimated at 8 W/m^{2}K:

$$\dot{Q}_{i,w} = h_{i,w} \cdot A_s \cdot (t_{s\_w} - t_{res,room,center})\text{[W]} \tag{7}$$

The ceiling void heat gain is estimated as a function of the measured ceiling upward surface temperature (in contact with the air in the ceiling void) and the average water temperature:

$$\dot{Q}_{void} = A_{ceiling} \cdot U \cdot \left[t_{s,ceiling} - \frac{(t_{w,su} + t_{w,ex})}{2} \right]\text{[W]} \tag{8}$$

The external heat gain through the facade $\dot{Q}_{fac}$ is measured directly by watt-meters.

The enthalpy flow rate extracted by the ventilation system can be defined as:

$$\dot{Q}_{ven} = \dot{M}_a \cdot c_{p,a} \cdot (t_{a,ex} - t_{a,su}) \quad \text{[W]} \tag{9}$$

In Eq. 5, $\dot{U}$ is the internal energy variation of the control volume. It can be calculated as:

$$\dot{U} = C \cdot \frac{dT_a}{d\tau} \tag{10}$$

With:

C : Global thermal mass of all components included in the control volume. [J K^{-1}]. In general, it is estimated as the product of the mass and the average specific heat (cooling ceiling, water and air).

$$C = (m_1 \cdot c_1 + m_2 \cdot c_2 .....m_n \cdot c_n) \tag{11}$$

$\frac{dT_a}{d\tau}$: is the air temperature variation (supposed to represent the state variable, hypothesis used as better estimation), [K s^{-1}].

The differential $dT_a/d\tau$ is calculated using the method shown in Cuevas and Lebrun (2002). Initial and final temperatures are determined by averaging 5 points at the beginning and 5 other points at the end of each sampling stabilized period, in such a way to define a period of 60 minutes.

$\dot{R}_{global}$: is the residual of the thermal balance, [W].

The residuals allow to compare the thermal balances (air and water sides). They give an idea about the precision of the results considering that if the measurements do not have error, the residuals must be zero.

3.2 Test results

3.2.1 Copper radiant ceiling type 1 (T1): These results allow to determine the radiant ceiling emission in heating (H) and cooling modes(C), according to the tests listed in Table 2.

Table 2: T1: Tests description

Test	T1H1	T1H2	T1C1	T1C2	T1C3
ventilation	ON	OFF	ON	OFF	ON
Heating	X	X	-	-	-
Cooling	-	-	X	X	X
Effective area m^2	6.5	6.5	19.4	19.4	19.4
Without thermal load	X	X	-	-	-
Convective thermal loads	-	-	X	X	-
Convective and radiative thermal loads	-	-	-	-	X

Table 3: T1: Thermal balance results

Test	T1H1	T1H2	T1C1	T1C1	T1C1
$\dot{Q}_{cooling}$ (W)	-	-	1484	1502	1368
$\dot{Q}_{heating}$ (W)	806	670	-	-	-
$\dot{Q}_{ven}$ (W)	156	-	289	-	269
$\dot{Q}_{\text{int},\frac{gains}{loads}}$ (W)	46	42.4	1748	1570	1716
$\dot{Q}_{fac}$ (W)	750	797	290	237	216
$\dot{U}$ (W)	-9.8	-5.2	-7.2	6.3	4.5
$\dot{Q}_{void}$ (W)	32.5	26	54.3	30.5	50.2
$\dot{Q}_{i,w}$ (W)	90.4	108.6	-317	-315	-312.6
$\dot{R}_{global}$ (W)	-5.9	-7.2	-4.9	26.8	37.1

The estimated values of each term of the thermal balance are shown in Table 3. From experimental results it is observed that in heating mode an average of 84% of the total room heat losses are compensated by the radiant ceiling system. In cooling mode an average of 74% of the room cooling load is compensated by the cooling ceiling and 14% by the ventilation system. In heating and cooling mode, the test results are presented in Table 4.

Table 4: T1: Measurements and AU calculated values

Test	ventilation	Thermal loads	$t_{w\,su}$ [°C]	$t_{w\,ex}$ [°C]	$t_{ref\,centre}$ [°C]	$\Delta T_{,Ln}$ [K]	$\dot{M}_w$ [kg/s]	AU_{centre} [W/K]	U_{centre} [W/m²K]	$\dot{Q}_{cooling}$ [W]	$\dot{Q}_{heating}$ [W]
T1H1	ON	Without	35.91	32.6	21.88	11.78	0.06798	68.4	10.5	-	806
T1H2	OFF	Without	36.92	34.27	21.65	14.27	0.06798	46.92	7.2	-	670
T1C1	ON	Convective	16.34	18.31	25.65	8.18	0.1998	181.4	9.3	1484	-
T1C2	OFF	Convective	16.32	18.28	26.21	8.86	0.1998	169.4	8.7	1502	-
T1C3	ON	Conv+Rad	15.47	17.29	24.21	7.71	0.1998	177.3	9.1	1368	-

In the experimental domain considered, it is observed that, in heating mode, the influence of ventilation system on A.U values is significant. The ventilation plays a beneficial role in the thermal performance of the radiant ceiling. An increase of the thermal exchange with the active panels, of about 30 % in heating mode is observed. In cooling mode the enhancement is apparently of the order of 6 %, however this result must be taken with a certain caution, because it can be due to the measurement uncertainty. It is important considering that in most of the related literature about radiant ceiling systems only the effect of natural convection on the ceiling surface is considered (Refet, 2009; Corgnati *et al.,* 2009; Catalina *et al.,* 2009 ; ASHRAE, 2004).

3.2.2 Copper radiant ceiling type 2 (T2): These results allow to determine the radiant ceiling emission in both heating and cooling modes. Following the same methodology used to type 1, it is observed that in heating mode an average of 95% of heating loads of the room are supplied by the radiant ceiling system. In cooling mode, an average of 78% of the room cooling loads is compensated by the cooling ceiling system and 22% by the ventilation system.

As with the radiant ceiling types T1, the ventilation plays a beneficial role in the thermal performance of the radiant ceiling. In heating mode the air supply permits an increase of about 32% of the thermal exchange with the active ceiling T2. In cooling mode the apparently enhancement is around 5%.

In general, the global heat transfer coefficient of the radiant ceiling system in heating mode is 10% higher than in cooling mode (the comparison is performed when the ventilation system is active).

In cooling mode there is a reduction of 8% of the heat transfer coefficient when the active panels are placed close to the corridor, due to the "contact interruption" between the radiant ceiling and the facade . For the same reason, if the active ceiling is placed only close to the facade there is an increase of the heat transfer coefficient of 20% due to the higher air velocity in this zone and the temperature gradient.

3.2.3 Capillary tube mats: type "G" (TG): The tests performed in the big office of the climatic chamber, allow to determine the radiant ceiling emission in heating and cooling mode, with and without ventilation and without thermal loads (except the lightings).

It is observed that in heating mode, an average of 95% of heating loads of the room is supplied by the radiant panels. In cooling mode, an average of 67% of the room cooling loads is extracted by the cooling ceiling system and 19% by the ventilation system. For capillary tube mats as for copper tubes ceiling, ventilation plays a beneficial role. In heating mode the air supply permits an increase of about 11% of the thermal exchange with the active plates. In general for this system the global heat transfer coefficient is 9% higher in heating than in cooling mode (the comparison is performed when the ventilation system is active). For similar test conditions, one observes a reduction of the global heat transfer coefficients when the capillary mats (TG) are used with respect to the copper ceiling systems (T1 and T2). Then, for the test performed (in heating or cooling mode) with and without ventilation an average reduction in A.U value of 23% and 15% respectively is observed. This is due to the increase of the thermal contact resistance for capillary tubes mats considering the additional air layer between the tubes and the ceiling metal panel. In general, all these results are considered during the modeling approach (Fonseca N. *et al*. 2009).

4. CONCLUSIONS

In the experimental domain considered, it is observed that, in heating mode, the influence of ventilation system on A.U values is considerable. An increase of the thermal exchange with the active panels of about 30% in heating mode is observed. In cooling mode the enhancement is apparently in the order of 6%, however this result must be taken with a certain caution because it can be due to the measurement uncertainty. The influence of surfaces temperatures inside the room, especially the facade, is considerable. Then, applicability of a certain heat sources concentration which is closely related to the actual conditions must be taken into account and the radiant ceiling must therefore be evaluated together with its designed environment and not as separate HVAC equipment.
For the test conditions presented in this study, the global heat transfer coefficient of the radiant ceiling system in heating mode is always 10% higher than in cooling mode (comparison made with the ventilation system active). It is also observed that, a "contact interruption" between the radiant ceiling and the facade (e.g. an inactive ceiling zone) can produce a reduction of 8% of the cooling capacity.
Due to the increase of the thermal contact resistance for capillary tubes mats (TG) (including the additional air layer between the tubes and the ceiling metal panel) for similar test conditions, the global heat transfer coefficient on the ceiling surface is lower than the radiant ceiling systems with copper tubes (T1 and T2). Therefore, for the test performed (heating or cooling mode) with and without ventilation an average reduction of 23% and 15% respectively is observed.

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