ALTERNATIVE STUDY OF HEAT TRANSFER ON RADIANT CEILING SYSTEMS ORIENTED TO COMMISSIONING PROCESS IN REAL BUILDINGS

Fonseca Néstor. a* and Cuevas Cristian, b.
Department of Mechanical engineering,
a. Universidad Tecnológica de Pereira, Colombia
b. Universidad de Concepción, Chile
*Author for correspondence
E-mail: nfonseca@utp.edu.co

ABSTRACT

A brief general description of radiant ceiling systems and a detailed analysis of heat transfer are developed, covering such topics as materials and working principle. The most studied and documented heat flow patterns are based on natural convection and simplified radiation exchange between two surfaces. However the convective heat exchange on a radiant ceiling surface becomes a complex process, considering specially the combined effect of ceiling perforation, ventilation, fenestration systems and multiple surfaces radiation. There are too many configurations and possible combinations of these elements in the modern buildings that do not allow to completely describe the phenomenon with a correlative method. In this study the analysis of ceiling convection and radiation on the systems is performed in order to allow its use in commissioning process in real buildings.

INTRODUCTION

The system studied here consists of an air distribution system coupled to a cooled or heated ceiling surface. It takes profit of convection and long-wave radiation to supply or remove heat from a space. It also maintains acceptable indoor air quality and humidity. By operating as an air-conditioning system, a radiant ceiling system separates the task of sensible cooling or heating from those of air quality and humidity control.

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 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.

Commissioning is one of the new tools to manage the complexity of today's HVAC systems. It is actually a quality-oriented process for achieving, verifying and documenting that the performance of facility systems and assemblies meet defined objectives and criteria [1].

The definition in ASHRAE Guideline [2], is probably close to a standard or consensus definition:

"Commissioning is the process of ensuring systems are designed, installed, functionally tested and operated in conformance with the design intent. Commissioning begins with planning and includes design, construction star-up, acceptance, and training and is applied throughout the life of the building.

Furthermore, the commissioning process encompasses and coordinates the traditionally separated functions of systems documentation, equipment start-up, control system calibration, testing, balancing and performance testing"

Possibly the major reason that commissioning is needed is precisely that in many projects "commissioning" the project simply consists of turning everything on and verifying that all motors, chillers and boilers run. The problem becomes serious considering that the most of the global systems are usually not commissioned. Currently the practice is that each contractor (usually manufacturer is not the installer) does (for economic reasons) the strictly necessary for its product to be operational. Therefore despite of the sophisticated BEMS (Building Energy Management Systems) and measurements system provided in the buildings, an inadequate installation, verification and management of the individual and global system performance (according to the AS-BUILT files), produce usually the deterioration of components and global system conditions which implies an increase of the energy consumption and sub-utilization of the expensive monitoring system.

The FPT (Functional Performance Testing) information and testing procedures are viewed from a system perspective, rather than a component perspective. This is especially critical for functional performance testing and for the overall success of the system. The FPT of HVAC system means to verify that the equipment, subsystem and total system work with in harmony (including the stability and durability) to show the final function of the building air-conditioning.

The functional performance testing as a commissioning tool is devoted to the detection of a possible malfunction and its diagnosis. Active tests are mostly applied in initial commissioning, i.e. at the end of the building construction phase. Later in the building life cycle, i.e. in re-, retro- and on-going commissioning, a "passive" approach is usually preferred, in order to preserve health and comfort conditions inside the building occupancy zones.

In the frame of the program "Commissioning of Building and HVAC systems to improve energy performance Annex 40" of the International Energy Agency, some FPTs are presented. However, there is no specific information about radiant ceiling systems.

Looking at the related literature, some case studies about this system are presented [3,1] in which the influence of radiant ceiling on building commissioning is usually simplified. Therefore a FPT for radiant ceiling systems is proposed here as a tool for diagnosis in commissioning processes.

NOMENCLATURE

C	(J K^{-1})	Thermal mass
E	(W m^{-2})	Emissive power
h	(W m^{-2} K^{-1})	Convection (radiation) heat transfer coefficient
$\dot{M}$	(kg s^{-1})	Mass flow rate
$\dot{Q}$	(W)	Heat flow
Nu	(-)	Nusselt Number
T	(K)	Temperature
Special characters		
ε	(-)	Emissivity
Subscripts		
ave		Average
b		Black body
conv		Convective
comb		Combined
ext		External
f		Floor or fictitious
fac		Facade (windows and external wall)
i		Internal
mr		Mean radiant
occ		Occupants
p		Panels blocks connected in parallel
RC		Radiant ceiling
rad		Radiation
res		Resultant
w		Water or wall
win		Window

DESCRIPTION OR THE RADIANT CEILING SYSTEMS

The radiant ceiling system may be heated electrically or by means of water circulating in metal or plastic pipes embedded in the ceiling. In many instances, insulation is placed behind the heat source to minimize back-loss and also as sound insulation.

Control of the heat output is achieved, in electrical system, by varying the current and in piped systems, by varying the water temperature or flow rate. The control may be linked to a room thermostat or to an external temperature sensor.

Three major types of radiant ceiling systems can be distinguished:

- The metallic ceiling panels, which are incorporated into the false ceilings. The parallel water – pipe circuits are distributed on the upper side of the panels, which form the room false ceiling. The whole system presents a low thermal inertia and the metal panel is used as a decorative element (Figure 1).
- The active slab made of concrete is relatively similar to heating floor. The propylene tubes are embedded in the lower portion of a concrete slab. The cost is low however, due to the high thermal inertia, it is difficult to control the risk of condensation.
- Another technique, similar to the previous one, uses parallel capillary tube mats made of polyethylene (internal diameter is about 2.5 mm). 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 cost is low and the thermal inertia is reduced [4]. The radiant mats in this system can be incorporated into the ceiling in three configurations: placed on 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 (Figure 2)

The metallic ceiling panels can also be used with capillary tube mats placed directly on top of the ceiling panels. Depending on the application, both copper (Figure 1) and capillary tube mats (Figure 2) used in this study are usually used with glass-wool (thermal and sound) insulation above the pipes.

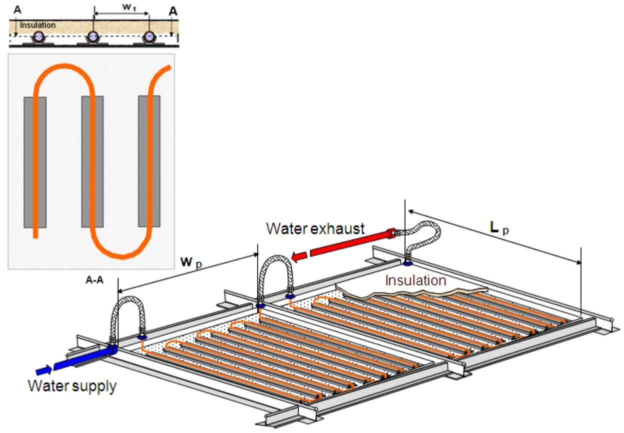

Figure 1 Copper tube secured to steel sheet radiant ceiling

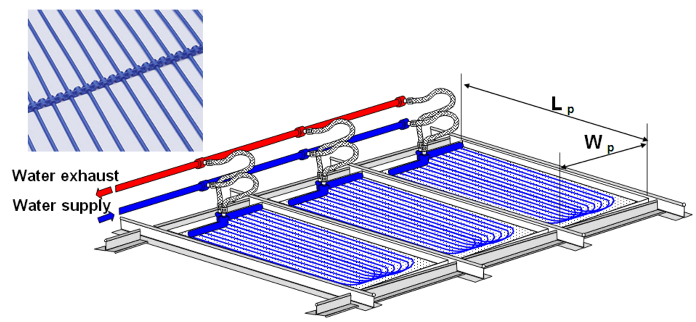

Figure 2 Synthetic capillary tube mats radiant ceiling

WORKING CONDITIONS

The radiant ceiling systems are usually mounted in the false ceiling or embedded into the ceiling and are designed to cover the sensible cooling of heating load or the room. The heating or cooling ceiling systems are connected to a closed circuit containing chilled or heated water and coupled to an air distribution system. The principle scheme of such a system is given in Figure 3.

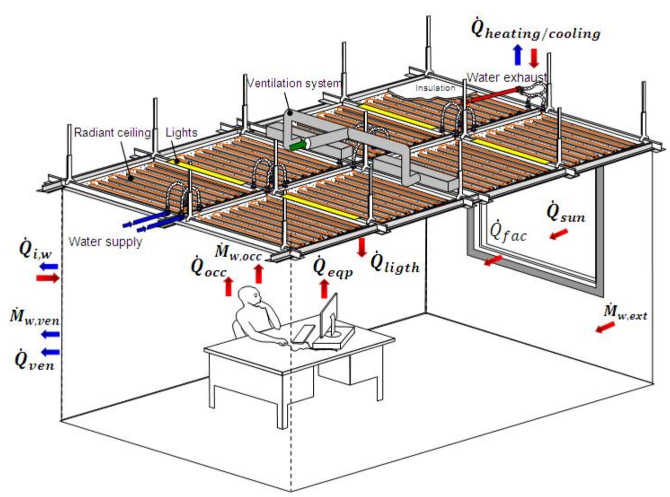

Figure 3 Sensible and latent heat loads for a radiant ceiling system

The energy and water mass balances of the office room control volume defined in Figure 3 are given by the following equations, valid in both heating and cooling modes:

$$\dot{Q}_{heating/cooling} + \dot{Q}_{fac} + \dot{Q}_{ven} + \dot{Q}_{void} + \dot{Q}_{i,w} + \dot{Q}_{occ} + \dot{Q}_{sun} + \dot{Q}_{eqp} + \dot{Q}_{lightt} = C\frac{dT}{d\tau} \quad \text{(W)} \quad (1)$$

$$\dot{M}_{w,ven} = \dot{M}_{w,occ} + \dot{M}_{w,ext} \quad (\text{kg s}^{-1}) \quad (2)$$

As the radiant elements are part of the room architecture and exposed directly to the occupant (placed above the occupancy zone) they are supposed to operate only in dry regime, as shown in Eq. 1. Therefore, the water supply temperature in the ceiling must exceed the dew point corresponding to the set point of indoor humidity ratio. Consequently the latent (moisture) load of the room can be controlled by the auxiliary ventilation system ($\dot{M}_{w,ven}$), which is also designed to provide air renewal for occupants hygienic requirements ($\dot{M}_{w,occ}$) and external air infiltrations ($\dot{M}_{w,ext}$) (Eq.2). A humidity and temperature control system can be used to avoid the condensation risk. A schematic diagram of this control system is showed in Figure 4. As long as the sensor is registering a condensation risk, either the flow to the ceiling is cut off by closing the control valve, or the water supply temperature is raised. When natural ventilation of the room is allowed, the limitation is related to the outdoor dew point.

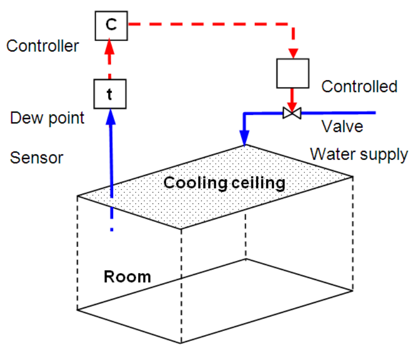

Figure 4 Simplified scheme of ceilings control system.

The water flow rate must be sufficient to maintain a turbulent regime, in order to increase the heat exchange. The water circuit should be designed to favor the parallel flow and minimize pressure drops.

The ventilation slot diffusers are usually located between the ceiling panels and above the occupancy zone. The air should be blown horizontally along the ceiling surface to increase the heat transfer coefficient and to avoid, thanks to "Coanda effect" [5] a jet fall in occupancy zone.

The contact quality (bonds between water pipes and ceiling panels) is crucial for radiant ceiling effectiveness. Identical ceiling modules (as designed) might provide completely different results only due to a bad contact quality.

In most applications, the thermal and sound insulation of the room ceiling void is recommended (in some cases required) and direct contact (cold bridges) between ceiling elements and room surfaces is prohibited. The free air circulation between rooms ceiling voids is allowed only if both rooms are equipped with the same radiant ceiling system and have identical destination (office room for example) [6].

The air velocity pattern at the occupancy zone must fulfill the comfort requirements. This means a maximum accepted average velocity in the range of 0.15-0.2 m/s with peak values limited to 0.25-0.3 m/s and a maximal allowed vertical temperature gradient of 2-3 K on the total height of the room.

Besides ensuring the heating or cooling of a building, the operation of a radiant ceiling system has also to prevent or minimize two side-effects associated with the presence of the radiant surface in the building (prevention of these adverse side-effects limits the heating or cooling power of the system). The first side-effect is the deterioration of comfort conditions due to the asymmetrical character of the radiant exchange in a room with a radiant surface. The second side-effect is the condensation risk in cooling mode.

RADIANT CEILING HEAT TRANSFER

The radiant ceiling (RC) can be represented as a fin. The heat exchange of the system considers the convective resistances on the water side, conduction through the tube shell and union system (tube-ceiling surface) or through a plaster layer and convective-radiative resistances from the tube and radiant ceiling surfaces to the cavity (between tubes and insulation) and the room. The fin effectiveness, the mixed convection close to the radiant ceiling surface (generated by the ventilation system) and the panel perforations influence must be considered.

The connection between the panel surface and the tubes is therefore a critical factor. Poor connections (higher thermal contact resistance) provide only limited heat exchange between the tubes and the panel, resulting in increased temperature differences between the panel surface and the cooling or heating fluid. Each one of these parameters will be considered at the simulation model developed in this study and described in [7].

Room-Radiant Ceiling convection ($h_{RC\ room\ conv}$)

The convective heat exchange inside the room equipped with a radiant ceiling becomes a complex process due to the combined effect of ventilation, ceiling perforations, internal thermal load and cooled or heated facade. Existing correlations were developed from experimental measurements in specific conditions of ventilation and internal thermal loads [8, 9, 10, 11]. Experimental studies were performed considering the individual influence of some parameters on comfort conditions: load distributions [12]), ventilation [13, 5] and facade [14].

According to ASHRAE [15] only natural convection (NC) should be considered on the radiant ceiling surface. However, among others to make sure that the cooling ceiling system is operating only in dry regime, moisture has usually to be removed from the room through a mechanical ventilation system which generates some air movement. Consequently the convection heat transfer coefficient needs to be corrected by an improvement factor (that is also including the effect of the perforation and fenestration).

Therefore the combined effects of natural and forced convection at ceiling surface must be considered using a modified version of the Yuge, [16] method developed originally for mixed convection on a sphere in transverse flow (Figure 5).

The air velocity on the radiant ceiling (u_∞) and the characteristic length of the radian ceiling in forced convection ($L_{RC,FC}$) (distance of the jet detachment) are usually defined from diffuser manufacturer's catalogue.

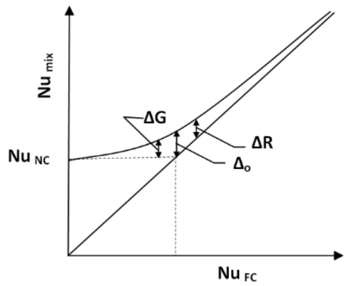

Figure 5 Combined convective heat transfer in traverse flow.

The effect of buoyancy on heat transfer in a forced flow is strongly influenced by the direction of the buoyancy force relative to that of the flow. For a perpendicular direction (transverse flow) caused by ventilation system, buoyancy acts to enhance the rate of heat transfer associated with pure forced convection. Finally, we get that the convective heat transfer coefficient on the radiant ceiling surface in combined (Forced convection FC and natural convection NC) regime is:

$$h_{RC,room\,conv} = \frac{k_a}{L_{c,RC}} Nu_{RC,room\,comb} \quad (\mathrm{W\ m^{-2}\ K^{-1}}) \qquad (3)$$

With:

$$Nu_{RC,room\,comb} = \left| \begin{array}{l} For: Nu_{RC,room\,FC} > Nu_{RC,room\,NC} ....; Nu_{RC,room\,comb} = Nu_{RC,room\,FC} + \Delta R \\ For: Nu_{RC,room\,FC} < Nu_{RC,room\,NC} ....; Nu_{RC,room\,comb} = Nu_{RC,room\,NC} + \Delta G \\ For: Nu_{RC,room\,FC} = Nu_{RC,room\,NC} ....; Nu_{RC,room\,comb} = Nu_{RC,room\,NC} + \Delta_o \end{array} \right.$$

$$\Delta R = \Delta_o \exp\left[-n * (Nu_{RC,room\,FC} - Nu_{RC,room\,NC} \right]$$

$$\Delta G = \Delta_o \exp\left[-m * (Nu_{RC,room\,NC} - Nu_{RC,room\,FC} \right]$$

$$m = \frac{7 + 0.011 Nu_{,room\,NC}}{1 + 0.1 Nu_{cc,room\,NC}} \quad n = \frac{0.993}{2 + 0.2 Nu_{cc,room\,NC}}$$

The characteristic length ($L_{c,RC}$) has to be experimentally identified due to the fact that, in modern buildings, there are too many different configurations and possible combinations of ventilation systems, thermal load types and distributions, as well as facade effects. In this study, for a flat plate in transverse flow, the adaptation of the coefficients *m* and *n* was performed.

In this study the analysis of ceiling convection is performed by considering that is not possible to get a general correlation law which covers all the possible combinations of a real case.

Room-Ceiling radiation ($h_{RC\ room\ rad}$)

In order to analyze the internal radiant exchanges, each surface of the enclosure can be characterized by its uniform radiosity and irradiation. The net radiative heat flux of the ceiling surface can be evaluated from Eq. 4 and Eq. 5 from radiosities (J_i), emissivities (ε_i), areas (A_i), view factors ($F_{i,j}$) and black body emissive powers (E_{bi}):

$$\dot{Q}_{rad,i} = \sum_{j=1}^{N} A_i . F_{i,j} (J_i - J_j) \quad (\mathrm{W}) \qquad (4)$$

$$\frac{E_{b,i} - J_i}{\dfrac{1-\varepsilon_i}{\varepsilon_i . A_i}} = \sum_{j=1}^{N} A_i . F_{i,j} (J_i - J_j) \quad (\mathrm{W}) \qquad (5)$$

The view factors can be calculated for the surfaces considered in Figure 6, according to the typical experimental test conditions.

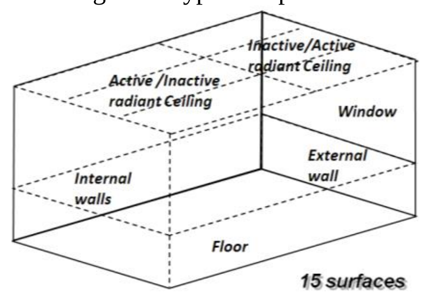

Figure 6 Room radiation surfaces

The net radiant heat flux at the ceiling surface can be determined by solving the unknown J_i. This method assumes that the surface temperatures are uniform and known.

According to the method proposed by Davies [17], the heat transfer coefficient for each surface can be calculated using the transformation "delta to star" to obtain a linearized radiative heat transfer coefficient. For a parallelepiped enclosure (six surfaces) the delta and star networks transformation can be represented in Figure 7.

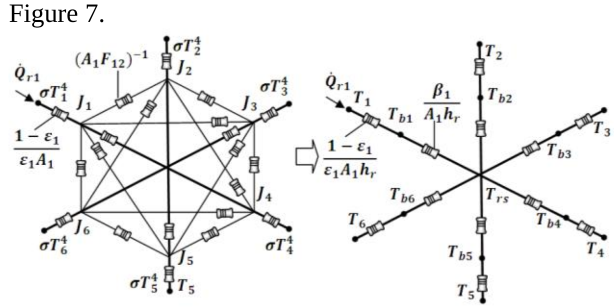

Figure 7 Delta and star networks of a parallelepiped enclosure.

Where:
T_{rs}: The radiant star temperature, (K)
T_{bi}: Black-body equivalent temperature of the surface i, (K)
B_i: Delta-star transform of the surface i, (-)

And:

$$T_{rs} = \frac{\sum T_i . S_i}{\sum S_i} \quad (\mathrm{K}) \qquad (6)$$

With: $\dfrac{1}{S_i} = \dfrac{1-\varepsilon_i}{\varepsilon_i . A_i . h_{ri}} + \dfrac{\beta_i}{A_i . h_{ri}}$

S_i is the total physical conductance between the surface node T_i and the radiant node T_{rs}.

If the room is represented by three surfaces (radiant ceiling, facade and internal walls) the "delta to star" transformation can be presented in Figure 8.

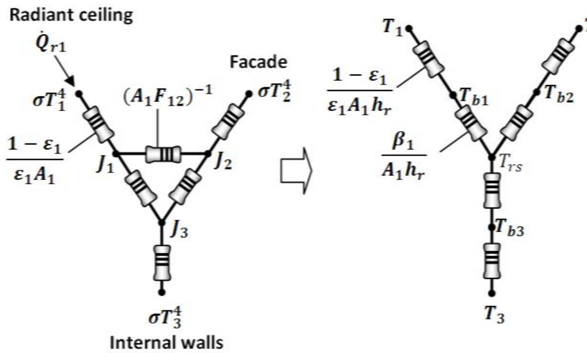

Figure 8 Delta and star networks of three rectangular sides of indefinite length.

Where:

$$\beta_1 = \frac{A_1.A_2.F_{2,3}}{A_1.F_{1,2}.A_2.F_{2,3} + A_2.F_{2,3}.A_3.F_{3,1} + A_3.F_{3,1}.A_1.F_{1,2}}, etc \quad (-) \ (7)$$

$$T_{rs} = \frac{T_1.S_1 + T_2.S_2 + T_3.S_3}{S_1 + S_2 + S_3} \quad (K) \quad (8)$$

and

$$S_j = \frac{A_j.h_r}{\left(\frac{1-\varepsilon_j}{\varepsilon_j}\right) + \beta_j} \quad (W/K) \quad (9)$$

As the radiosities and temperatures are known and assuming linearization of the heat transfer coefficient, it can be calculated as:

$$h_{ri} = \frac{\dot{Q}_{rad,i}}{A_i * (T_{rs} - T_i)} \quad (W\ m\text{-}2\ K\text{-}1) \ (10)$$

This coefficient depends therefore on the surface and radiant star temperature definition, emissivities and view factors between the room surfaces and the radiant ceiling.

Several methods have been developed to simplify this calculation. For example in the "mean radiant temperature" method (MRT), the thermal radiation interchange inside an indoor space is modeled by assumption that the surfaces radiate to a fictitious, finite surface (representing the room walls including the facade and the floor) that gives about the same heat flux as the real multisurface case [18]. The MRT equation for the radiant ceiling surface may be written as:

$$\dot{Q}_{RC,room,rad} = A_{RC,effec} * \sigma * F_{r,room} * ((t_{RC,ave} + 273{,}15)^4 - (t_{mr,room} + 273{,}15)^4) \quad (W) \ (11)$$

$$t_{mr,room} = \frac{\sum_{j=RC,ave}^{n} A_j.\varepsilon_j.t_j}{\sum_{j=RC,ave}^{n} A_j.\varepsilon_j} \quad (K) \quad (12)$$

When the surface emittances of the enclosure are nearly equal and the surfaces directly exposed to the panel are marginally unheated or uncooled, the fictitious temperature $t_{mr,room}$ become the area-weighted average uncooled or unheated temperature (AUST) widely used in the related literature [19, 20, 15] but it has to be considered that there is usually an important temperature difference between the facade and the room surfaces and this can be a source or error of the model .

As a better approximation, the mean radiant temperature of uncooled or unheated surfaces can be calculated from measurements of the resultant and air temperatures, by correcting the mean radiant temperature of the room (Eq. 13) as the radiant ceiling "sees" an environment which excludes its own influence [6].

$$t_{mr,room} = \left[2 * t_{res,room} - t_{a,room} - \frac{A_{RC,s}}{A_{room\,f,s}} * t_{RC,ave}\right] * \frac{1}{1 - \frac{A_{RC,s}}{A_{room\,f,s}}} \quad (°C)(13)$$

Eq. (13) is applicable only if: $|t_{mr\ room} - t_{a\ room}| < 4$ K [13].

The radiation exchange factor ($F_{r\ room}$) for any two diffuse, gray surfaces that form an enclosure can be calculated from Eq. (14) :

$$F_{r,room} = \frac{1}{\frac{1}{F_{RC,f}} + \frac{1}{\varepsilon_{RC}} - 1 + \frac{A_{RC,s}}{A_{room\,f,s}} * \left[\frac{1}{\varepsilon_{f,room}} - 1\right]} \quad (-) \ (14)$$

Where $F_{RC,f}$ is the radiation view factor from ceiling to a room fictitious surface giving an equivalent heat transfer, as in the real multi-surface case (1.0 for flat ceiling ASHRAE, 2009).

$A_{RC,s}$, $A_{room,f,s}$ are the areas of radiant ceiling and fictitious room surface (other than the ceiling).

ε_{RC} and $\varepsilon_{f,room}$ are emissivities of the ceiling (model parameter) and of the fictitious surface (0.98) [21].

The radiation heat transfer coefficient can be expressed as follows:

$$h_{RC,room,rad} = \sigma * F_{r,room} * \frac{(t_{RC,ave} + 273.15)^4 - (t_{mr,room} + 273.15)^4}{t_{RC,ave} - t_{mr,room}} \quad (W\ m^{-2}\ K^{-1}) \quad (15)$$

The radiative heat transfer coefficient can also be linearized according to

$$\dot{Q}_{rad,i} = h_{r,i} * A_i * (t_i - t_f) \quad (W) \quad (16)$$

$$h_{r,i} = \varepsilon_i * \sigma_i * (T_i - T_f)(T_i^2 - T_f^2) \quad (W\ m^{-2}\ K^{-1}) \ (17)$$

If the difference between T_i and T_f is small (only in some applications), the following approximation can be considered during the commissioning process:

$$h_{r,i} = 4 * \varepsilon_i * \sigma_i * \overline{T}^3 \quad (W\ m^{-2}\ K^{-1}) \quad (18)$$

$$\overline{T} = (T_i + T_f)/2 \quad (K) \quad (19)$$

The temperature T_f is actually the mean radiant temperature viewed by the surface i.

In the previous cases (two and three surfaces), the result of $h_{r,i}$ coefficient linearization is exact. However, for a parallelepiped enclosure of six surfaces, there is an error of about 6 % [17]. In this study, the radiation heat transfer coefficient of the radiant ceiling is calculated considering the star temperature for three surfaces: facade, internal walls and radiant ceiling as the most representative and useful case for commissioning process.

A comparison is established hereafter between simplified and detailed methods to calculate the radiant heat flow from the ceiling surface: For example for a radiant ceiling in copper tube tested in this study, if it is calculated following the detailed method (Eq. 4 and 5) and the laboratory experimental results in which all surface temperatures were measured, an average difference of 3.71 % with respect to the simplified method (Eq. 11) is obtained (see Figure 9) (the same result is found for synthetic capillary tube mats tested with all the radiant ceiling active and reference temperature placed at the room center).

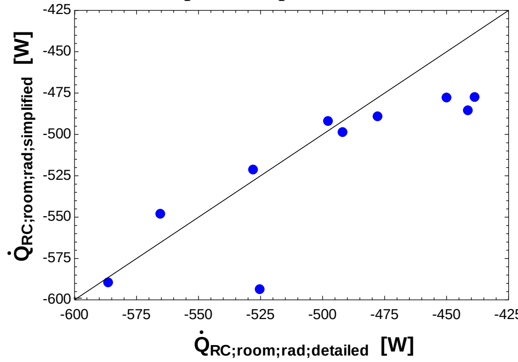

Figure 9 Comparison between detailed and simplified methods for room radiant heat exchange

In general the differences between simplified and detailed methods shown in Figure 9 are due to the air temperature stratification. For simplified method, the measurements of resultant and air temperatures at 75 (cm) from the floor are used. For the point which is outside the range in figure 9, the stratification was particularly important (1.5 (K)).

If the detailed method is used, it is important to take into account that the surface temperature uncertainties could be significant, specially glazing surface [22] and the global uncertainty increases with the number of measured variables. This is a typical difficulty in the commissioning process.

CONCLUSION

With the regard to the previous studies the main difference is the detailed treatment of convection and radiation heat exchange in this study (considering the influence of ventilation, facade and perforation effects) which is usually neglected or too simplified. The convective and radiative heat exchange on a radiant ceiling surface is a really complex process, considering specially the combined effect of ventilation and fenestration systems. In the convective part, a correlative method cannot describe completely the heat transfer processes, considering that in modern buildings there are too many configurations and possible combinations of these systems. In the radiation part an alternative method is presented that allows simplifier the modelling in order to use less measuring point during the commissioning process without loss too much accuracy of the modelling results.

REFERENCES

[1] International Energy Agency (IEA). 2003. Commissioning of Building and HVAC systems for improve energy performance. Annex 40 Final report.

[2] ASHRAE Guideline 2-2005. Engineering analysis of experimental data. American Society of Heating, Refrigerating, and Air-Conditioning Engineers, Inc., Atlanta, USA.

[3] AuditAC 2006. Successful case Studies in AuditAC. Technical Guides for owner/manager of HVAC systems. Vol 10.

[4] Miriel J., Serres L. and Trombe A. 2002. Radiant ceiling panel heating-cooling systems: experimental and simulated study of the performances, thermal comfort and energy consumptions. Applied Thermal Engineering, Vol. 22; pp. 1861-1873.

[5] Behne M. 1999. Indoor air quality in rooms with cooled ceiling. Mixing ventilation or rather displacement ventilation?. Energy and Building. Vol 30; pp. 155-166.

[6] Ternoveanu A, Hannay, C, Qingping, W. 1999. Preliminary Analysis on a Research Project for Cooling Ceilings- synthesis of available information. Laboratoire de thermodynamique applique Université de Liège.

[7] Fonseca N., Lebrun J. André P. 2010 a. Experimental study and modeling of cooling ceiling systems using steady- state analysis. International Journal of Refrigeration.vol 33, issue 4.

[8] Alamdari F. and Hammond P. 1983. Improved data correlations for bouyancy-driven convection in rooms. Building Services Engineering research and Techonology. Vol 4. N° 3; pp.106-112.

[9] Spitler J., Pedersen C. and Fisher D. Interior convective heat transfer in buildings with large ventilative flow rates. ASHRAE Transsactions 97 (1): 505-515.

[10] Awbi H. and Hatton A. 2000. Mixed convection from heated room surfaces. Energy and Buildings. Vol 32; pp. 153-166.

[11] Novoselac A., Burley B., and Srebric J. 2006. New convection correlation for cooled ceiling panles in room with mexed and stratified airflow.HVAC/R Reseach. Vol 12. Number 2; pp. 279 -294.

[12] Behne M. 1996. Is there a risk of draft in rooms with cooled ceilings. Measurement of air velocities and turbulences ; ASHRAE Transactions: Symposia – SD 96-4-5; pp. 744-751.

[13] Kulpmann R. 1993 Thermal comfort and air quality in rooms with cooled ceilings – results of scientific investigations; ASHRAE Transactions: Symposia – DE 93 – 2 – 2; pp. 488-502.

[14] Fredriksson J., Sandberg M., Moshfegh B.2001. Experimental investigation of the velocity fiel and airflow pattern generated by cooling ceiling beams. Building and Environment. Vol 36. pp. 891-899.

[15] ASHRAE HANDBOOK-HVAC Systems and Equipment. 2004. Chapter 6. Atlanta: American Society of Heating, Air-Conditioning and Refrigeration Engineers, Inc.

[16] Yuge T. 1960. Experiments on heat transfer from spheres including combined natural and forced convection. Trans. ASME Journal of Heat Transfer 82 (1960)3 pp. 214-220.

[17] Davies M. 2004. Building heat transfer. John Wiley and Sons, Ltd Editors. University of Liverpool.UK.Chapter 4-6.

[18] Walton G.1980. A new algorithm for radiant interchange in room loads calculations. ASHRAE Transaction: 86 (2) 190-208.

[19] Kilkis B. 1995. Coolp: A computed program for the design and analysis of ceiling cooling panles. ASHRAE Transaction: Symposia. SD 95-4-4. pp. 705-710.

[20] Jae-Weon Jeon and Mumma S. 2004. Simplified cooling capacity estimation model for top insulated metal ceiling radiant cooling panels. Applied Thermal Engineering. Vol.24; pp. 2055-2072.

[21] ASHRAE HANDBOOK-HVAC Systems and Equipment. 2004. Chapter 6. Atlanta: American Society of Heating, Air-Conditioning and Refrigeration Engineers, Inc.

[22] Fissore A. and Fonseca N. 2007. Experimental study of the thermal balance of a window, design description. Buildong and Environment. Vol. 47; pp. 3309-3321.