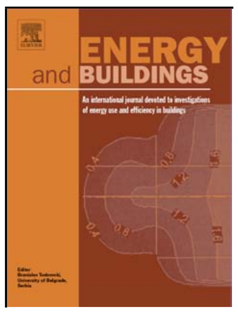

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Experimental evaluation of heat transfer coefficients between radiant ceiling and room

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Abstract

Heat transfer coefficients between radiant surfaces and room are influenced by several parameters: surfaces temperature distributions, internal gains, air movements.

Aim of this paper is to evaluate heat transfer coefficients between radiant ceiling and room in typical conditions of occupancy of an office or residential building. Internal gains were therefore simulate using heated cylinders and heat losses using cooled surfaces. Evaluations were developed by means of experimental tests in an environmental chamber.

Heat transfer coefficient may be expressed separately for radiation and convection or as one total parameter, but this choice may lead to different considerations about thermal performance of the system. Therefore, in order to perform correct evaluations, it is extremely important to use the proper reference temperature.

The obtained values confirm tendencies found in literature, indicating limitations and possibilities of radiant ceiling systems improvement.

Keywords: heat transfer coefficients, radiant ceiling, radiant heating, radiant cooling, reference temperature

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1. Introduction

Heat transfer coefficients are fundamental parameters for heating/cooling load calculations, thermal comfort analysis, dynamic thermal simulations, CFD analysis and dimensioning of radiant systems.

In a study by Loman aimed to compare thermal simulation programs [1], annual heating energy demand estimated by the software ESP and the software HTB2, varied up to 27% depending on the internal surfaces convective heat transfer algorithm used. While using the same heat transfer algorithm in both of the programs, the difference was reduced to 1%.

In literature, several values and equations of heat transfer coefficients are given between the interior building surfaces (heated, cooled or unheated/uncooled) and the space (other surfaces and air), as resumed and by Khalifa [2, 3, 4] and by Awbi and Atton [5, 6].

Usually scientists and developers of simulation software are interested in values or equations of heat transfer coefficients estimated separately for radiation and convection, while to size HVAC systems, and especially radiant systems, a combined coefficient is needed.

These different kind of coefficients requires the use of different reference temperatures, as underlined by Olesen et al. [7], because the radiant coefficient refers to the radiant heat transfer with other surfaces, the convective coefficient to the convective heat transfer with the room air and the total heat transfer coefficient to the mix of radiant and convective heat transfer.

The radiant heat transfer coefficient expresses the radiant heat exchange between a specific surface and all the other surfaces in the room. Depending mainly on parameters that are constants (Stefan-Boltzmann constant, emissivity, view factors), it can be considered constant for any kind of low temperature radiant heating system and high temperature radiant cooling system.[7].

The convective heat transfer coefficient expresses the heat exchange between a specific surface and the air boundary layer, therefore it depends on several changing parameter: air velocity, air temperature, turbulence. Several algorithms deduced from experimental tests can be used to define, from time to time, the actual value of this parameter. In the case of radiant surfaces and natural convection, high values of the convective heat transfer coefficient are possible only for: cooled ceiling, heated floor and cooled/heated walls, which are able to generate, through buoyancy forces, air movements inside the room [8]. Instead, in the case of heated ceiling and cooled floor, the convective heat transfer coefficient has low values, because little air movements are generated. Under these conditions a stable thermal stratification occurs inside the room [5].

The total heat transfer coefficient expresses the radiant and convective heat transfer between a specific surface and the room, but it cannot be calculated as the sum of the previous two coefficients because they refers to different physical phenomena, so they have different reference temperatures. While the convective heat transfer coefficient is physically defined as the surface conductance between a surface and the boundary layer, the linear radiant transfer coefficient is just a theoretic parameter, created to compare radiant heat transfer to convective heat transfer in multimode heat transfer phenomena calculations, but it has not a physical definition, since radiation do not follow a linear model [9].

In this study heat transfer coefficients for a radiant ceiling are shown. The evaluation of the coefficients was developed using a test chamber, which simulates typical condition of occupancy in an office or residential room. Values are compared to the ones found in literature, underlining the importance of the choice of the reference temperature, both during tests and in the further calculations when coefficients are used.

2. Reference temperatures

The reference temperature for the calculation of the radiant heat transfer coefficient, is the average unheated surface temperature (AUST) calculated taking into account the view factors between surfaces [10, 11]. The AUST can be also calculated as the area-weighted average temperature [10], but the calculation with view factors is more precise.

The radiant heat transfer coefficient can be calculated from the net heat transfer among the studied surfaces and the surrounding surfaces.

$$\frac{\dot{Q}_r}{A} = \sigma \sum_{j=1}^{n} F_{\varepsilon_{s-j}} \left(T_s{}^4 - T_j{}^4 \right) \quad (1)$$

$$F_{\varepsilon_{s-j}} = \frac{1}{\left(\frac{1-\varepsilon_s}{\varepsilon_s}\right) + \left(\frac{1}{F_{s-j}}\right) + \left(\frac{A_s}{A_j} \cdot \frac{1-\varepsilon_j}{\varepsilon_j}\right)} \tag{2}$$

$$h_r = \frac{\sigma \sum_{j=1}^{n} F_{\varepsilon_{s-j}} \left(T_s^{\ 4} - T_j^{\ 4}\right)}{AUST - T_s} \tag{3}$$

$$AUST = \sqrt[4]{\sum_{j=1}^{n} \left(F_{s-j} \ T_j^{\ 4}\right)} \tag{4}$$

The reference temperature for the calculation of the convective heat transfer coefficient, is the air temperature at the edge of the thermal boundary layer, beyond it the air temperature remains essentially constant. The coefficient can be calculated through several algorithms as resumed and by Khalifa [2, 3, 4] and by Awbi and Atton [5, 6].

The reference temperature for the calculation of the total heat exchange coefficient is not yet clearly defined. Since it expresses both the radiant and convective heat transfer between a studied surface and the room, the most appropriate reference would be the operative temperature (Tab. 1). It must be anyway noticed that this is just a practical solution useful to calculations, but it does not help in representing the physical phenomena occurring in the room that must be divided for radiation and convection using separate reference temperatures.

As suggested by Olesen et al. [7], the operative temperature would be a convenient solution as reference for total heat transfer coefficient, also because it is used as reference for thermal comfort analysis. Furthermore, in EN Standard 12831 [12], the use of operative temperature is suggested also for heat load calculations.

The operative temperature, for comfort analysis, is express by the equation:

$$T_{op} = \frac{\left(h_{c,h} \cdot T_a\right) + \left(h_{r,h} \cdot T_{mr}\right)}{h_{c,h} + h_{r,h}} \tag{5}$$

In this equation air temperature and mean radiant temperature are measured or calculated in the reference point at 0.60 m over the floor level for a seated person and at 1.10 m for a standing person. Heat transfer coefficients for human body ($h_{c,h}$ and $h_{r,\,h}$) can be found in literature [13].

3. Experimental apparatus

A test chamber was used to carry out measurements aimed at evaluating heat transfer coefficients for a cooled/heated radiant ceiling.

It has a net floor/ceiling area of 11,61 m^2 (4,30 m × 2,70 m) and a net internal height of 2,56 m (Fig. 1); it is surrounded by another room with stable, but not controlled, temperature, while floor and ceiling are in contact with concrete slabs, through a layer of insulating boards 12 mm thick. The chamber is built in an underground level in order to get stable conditions during the measurements without controlling temperature in the surrounding room.

It is equipped with a hydronic radiant ceiling, a hydronic radiant floor and hydronic radiant panels on tree of the vertical walls, the forth wall is use to enter the room through a door and to check measurements through a window.

While a hydronic system is used as heating/cooling system, the other four radiant surfaces can be used to simulate heat losses or gains.

All the walls of the chamber are highly insulated and constructed with three layers: an internal plasterboard panel 15 mm thick (radiant panels), a second layer of insulating boards 12 mm thick, and another plasterboard panel 15 mm thick.

The internal surfaces are instrumented by a total of 14 PT1000 sensors, in order to check surface temperatures, while other three sensors are hanged to a thin wire cable in the centre of the chamber at 0.10 m, 1.10 m, 1.70 m, to measure air temperature.

Eight PT100 sensors are used to measure water temperature in the inlet and outlet of each of the four hydronic circuits, while flow meters are used to regulate water flow.

The chamber is equipped with four metallic cylinders (varnished with opaque gray) to simulate occupant loads inside the room. The power emitted by each cylinder can be regulated before any test.

Metallic cylinders were used to simulate internal gains during cooled radiant ceiling tests, while vertical walls were used to simulate heat losses during heated radiant ceiling tests.

The emissivity of the chamber surfaces has been measured using an infrared thermal imaging camera, in order to use this parameter in radiant heat transfer calculations.

4. Determination of the heat transfer coefficients

The heat flux transfer between the ceiling and the chamber was calculated subtracting from the enthalpic flux the backward heat transfer toward the upper slab:

$$\frac{\dot{Q}_{tot}}{A} = \frac{\dot{m} \cdot c_p \cdot \Delta T_w}{A} - \frac{\dot{Q}_{out}}{A} \quad (6)$$

The total heat transfer coefficient between the radiant ceiling and the room, based on operative temperature at height of 1.1 m (as expressed by (5)) was calculated as:

$$h_{tot} = \frac{\dot{Q}_{tot}/A}{T_{op} - T_s} \quad (7)$$

Since ISO Standard 7730 and 7726 and ASHRAE Standard 55 [14, 15, 16] indicate that in room with air velocity lower than 0.2 m/s, and difference between mean radiant temperature and air temperature less than 4 K, operative temperature can be calculated as adjusted air temperature (equation (8)), the total heat transfer coefficient was calculated even using this reference temperature. Results have anyway negligible differences, because operative temperature and adjusted air temperature had very close values in the developed test conditions (Tab. 2 and 3). This shows that adjusted air temperature can be reliably use in calculations in place of operative temperature for those rooms which respect limitations reported in standards.

$$T_{op} \approx T_{amr} = \frac{T_a + T_{mr}}{2} \tag{8}$$

The radiant heat flux among the ceiling and the surrounding surfaces (Q_r) was calculated through equation (1), and the radiant heat transfer coefficient (h_r) through equation (3).

The convective heat flux due to heated/cooled ceiling was calculated subtracting to the total flux the radiant flux:

$$\frac{\dot{Q}_c}{A} = \frac{\dot{Q}_{tot}}{A} - \frac{\dot{Q}_r}{A} \tag{9}$$

The convective heat transfer coefficient between radiant ceiling and room, based on air temperature, was calculated as:

$$h_c = \frac{\dot{Q}_c / A}{T_a - T_s} \tag{10}$$

Since previous experiments [5, 17] showed that in the case of a heated ceiling no thermal boundary layer is present, air temperature at three different heights (0.10 m, 1.10 m, 1.70 m) was used as reference to calculate convective heat transfer coefficients, and in order to have comparable values, the same reference temperatures were used for the cooled ceiling.

5. Experimental programme

The object of measurements was to calculate heat transfer coefficients between radiant ceiling and the room in typical occupancy conditions and typical set point values of system parameters.

In cooling conditions two different values of water flow rate were set: 200 l/h and 240 l/h, corresponding to 25 l/h and 30 l/h per circuit. For each water flow rate value three supply temperature values were set:

13 °C, 15 °C, 18 °C. Further tests were developed setting air temperature at 1.10 m to 24 °C, while supply water temperature was not controlled.

Results for each test are resumed in table 3, where the test name represents a code explaining tests conditions: C_200_13 means Cooling_200 l/h_T_{supply} 13 °C.

In heating conditions three different values of water flow rate were set: 160 l/h, 200 l/h, 240 l/h, corresponding to 20 l/h 25 l/h, 30 l/h per circuit. For each water flow rate value three supply temperature values were set: 30 °C, 35 °C, 40 °C.

Results for each test are resumed in table 5, where the test name represents a code explaining tests conditions, as shown before.

During cooled radiant ceiling tests, four cylinders were used to simulate occupants loads inside the room, while during heated radiant ceiling tests, vertical panels were used to simulate walls/windows transmission heat losses.

6. Results

6.1 Cooled radiant ceiling results

The data elaborated for the cooled ceiling (Fig. 2), shows that the radiant heat transfer coefficient can be considered constant for this technology (~ 5.6 W m^{-2} K^{-1}). These data are very close to the ones calculated by Olesen et al. [7] for a cooled radiant floor (Tab. 2).

This can be clearly explained analyzing equation (3): all the parameters of the algorithm are in fact constants, excluding temperatures. Since surface temperatures inside the chamber are very low, within the range 18 °C – 33 °C, they cannot influence significantly the linear radiant heat transfer coefficient value.

Air temperature in the chamber shows a little thermal stratification between 1.10 m and 1.70 m (average ΔT_a = 0.6 °C), while under 1.10 m air temperature can be considered constant. This explains why convective heat transfer coefficient is a little bit lower at the maximum height. The maximum reported difference is 0.8 W m^{-2} K^{-1}, while the average value is 0.6 W m^{-2} K^{-1}. Analyzing data reported in figure 2, it is possible to notice that the convective heat transfer coefficient value does not vary a lot among different tests. The convective heat transfer value depends in fact on the temperature difference between the cooled ceiling surface and room air, which also does not vary a lot among tests, as shown in table 3.

The values of the convective heat transfer coefficient measured during the tests fairly correspond to the values calculated with some of the equations proposed in literature [2, 3], but are higher then other values calculated with different equations (Tab. 2).

The convective heat transfer coefficient shows anyway lower values than the radiant ones, it means that in condition of free convection, radiant heat transfer is dominating.

The total heat transfer coefficient shows a wider range with an average value of about 13 W m^{-2} K^{-1}. These values are significantly higher than ones typically shown in literature (~ 11 W m^{-2} K^{-1}) [18], but also in other experiments about cooling floor by Olesen et al. [7], higher values of total heat transfer coefficient, compared to values from literature, were also noticed.

This difference is probably due to the choice of the reference temperature. Both in these and in Olesen's experiments, operative temperature was used as reference, but air temperature is often used as reference by designers.

If in previous tests air temperature at 1.10 m in the centre of the chamber was used to calculate total heat transfer coefficient, a lower average value would be obtained of about 10.3 W m^{-2} K^{-1}, much closer to the ones reported in literature. It is not anyway clearly specified which reference temperature was used for these literature values

Reference temperature is commonly expressed in literature as room temperature and this allows the designer to assume it as an air temperature or as an operative temperature. Sensible errors may occur if the temperature chosen by the designer for his calculations is not the same used during evaluations of the total heat transfer coefficient.

It is extremely important to specify in reports about heat transfer coefficients which reference temperature was used, and designers should use the same temperature together with the coefficients in their calculations.

As stated in previous paragraphs, the most appropriate temperature for evaluations of total heat transfer coefficient is operative temperature. This value should furthermore be used to set controls of radiant systems.

Average values of heat transfer coefficients derived from measurements are compared to literature values in table 2. For linear radiant heat transfer coefficient and total heat transfer coefficient values proposed by Olesen are considered as reference [7, 18]. For convective heat transfer coefficient equations resumed by Khalifa are considered as reference [2, 3], due to this choice a maximum and a minimum value are reported.

6.2 Heated radiant ceiling results

The data elaborated for the heated ceiling (Fig. 3), shows the same values of the radiant heat transfer coefficient, as the cooled radiant ceiling (~ 5.6 W m^{-2} K^{-1}). Surface temperatures inside the chamber have values in the range 20 °C – 33 °C, still too low to influence significantly linear radiant heat transfer coefficient calculations.

The heated ceiling has convective heat transfer coefficient values very similar to the ones presented by Awbi and Atton [5], in a previous detailed research about natural convection from heated room surfaces (Tab. 4). These values are very low, in fact measurements reveal stable thermal stratification during the tests (ΔT_a = 1.1 °C – 1.8 °C). Under these conditions little air movements occur inside the chamber, with consequent low heat transfer by convection from the ceiling to the air.
For a heated surface facing downward, the tendency of the air to ascend is impeded by the surface itself. Only few horizontal air movements along the ceiling are generated, and the convective heat transfer is very limited. On the contrary for a cooled ceiling, air close to the cool surface is driven by buoyancy forces downward, while it is replaced by ascending warm air from the ambient. Since air movements are much stronger than in previous case, convective heat transfer is much more effective, as shown by reported data.

The total heat transfer coefficient has values similar to the ones reported in literature (~ 5.8 W m^{-2} K^{-1}) [18], because operative temperature and air temperature at 1.10 m have very close values.

Due to the reasons explained in the previous paragraphs, the operative temperature must be anyway considered as the best reference temperature.

Radiant heat transfer coefficient and total heat transfer coefficient show similar values, but since these values derive from different reference temperatures they cannot directly be compared. The first one is in fact calculated from the radiant heat balance, while the second one is a parameter defined to have global description of the heat transfer between the radiant ceiling and the room, but it does not refer to a specific physical heat transfer type.

Average values of heat transfer coefficients derived from measurements are compared to literature values in table 4. For linear radiant heat transfer coefficient and total heat transfer coefficient values proposed by Olesen are considered as reference [7, 18]. For convective heat transfer coefficient the equation proposed by Awbi and Hatton [5] is considered as reference and used to calculate the coefficient.

7. Discussions

Data reported in this paper and data existing in literature [5, 6] demonstrate that convective heat transfer coefficient, depending on several parameters, is the most difficult factor to evaluate.

As expected, cooled ceiling showed higher values of this coefficient than heated ceiling. This is due to the higher air movement generated by buoyancy forces. A little error in the evaluations could be explained by the fact that during cooled ceiling tests, heated cylinders were used to generate loads, while during heated ceiling tests cylinders were not working. Heat plums generated by cylinders could increase cooling heat transfer coefficient, anyway no evaluations of this phenomenon was found in literature.

During heated ceiling tests little air movements occurred in the test room; only cooled walls generated cool air downdraught, balanced by warmer air lifting up to the ceiling. In the steady-state condition, a stable thermal stratification developed inside the room, with warm still air close to the heated ceiling and low convective heat transfer.

Data elaborated in this work shows little correlation between convective heat transfer coefficient and temperature difference (Fig. 4), while in literature higher dependence is shown [5].

Convective heat transfer coefficient is nevertheless the parameter which mainly influences low temperature heating and high temperature cooling radiant system performance, because, as demonstrated, radiant heat transfer coefficient may be considered constant. The different thermal power between a cooling ceiling and a heating ceiling mainly depends on the different convective capacity of the two systems.

The linear radiant heat transfer coefficient can be generally considered constant for any kind of low temperature heating and high temperature cooling systems (floor, ceiling and wall), as deduced from results shown in figure 5 and from literature results [7].This is not true for high temperature radiant heating systems and in general in any situation in which a high temperature source is present. A particular case is when solar radiation enters the room, heating surfaces or enhancing cooling radiant system capacity.

The sun is a special kind of radiant source, since solar spectrum encloses ultraviolet, visible and infrared radiation and the maximum radiant energy is in the visible field. Glass surfaces are then opaque to long wave radiation and therefore solar radiation entering in rooms is mainly constituted by visible radiation. All these elements underline the particularity of the problem, because many materials exhibit different behaviour to long wave and short wave radiation.

Cooling system thermal power improvement cannot therefore be easily explained through the enhancement of the radiant heat transfer coefficient, but it would be better to express it through the total heat transfer coefficient enhancement. Detailed analysis should be developed, because some parameters like radiant surface colour are much more relevant in this case than emissivity.

Measurements reported in this paper show higher values of total heat transfer coefficient respect to literatures ones, when operative temperature was different from air temperature. This demonstrates the importance to use the same reference temperature during tests.

In building simulation software calcualtions it is preferable to separate radiation from convection. The radiant heat transfer coefficient can be calculated by the software but also a constant value can be used to accelerate calculations.

Convective heat transfer values should be calculated through the use of proper algorithms reported in literature [5, 6, 17, 19].

8. Conclusions

Values of heat transfer coefficients for radiant ceiling obtained during measurements confirm values reported in literature. In the present study a little higher average value of total heat transfer coefficients is found of about 13.2 W m^{-2} K^{-1}, for cooled ceiling, compared to 11 W m^{-2} K^{-1} from literature. The difference may be caused by the used reference temperature, which is not specified for values presented in literature.

For a heated radiant ceiling a total heat transfer coefficient of about 5.8 W m^{-2} K^{-1} is measured, which is similar to the literature value of 6 W m^{-2} K^{-1}.

The present work underlines the fundamental role of reference temperature in any calculation of heat transfer coefficients. For the total heat transfer coefficient, the most useful reference temperature is the operative temperature at 1.10 m for standing persons and at 0.6 m for sitting persons. The value at 1.10 m can be considered as the reference in all the situations in which no specific information about occupants is given.

The same reference temperature is used for heat and cooling load calculations and for thermal comfort analysis.

Radiant heat transfer coefficient can be considered constant, approximately about 5.6 W m^{-2} K^{-1}, for low temperature heating and high temperature cooling ceilings, while convective coefficient varies significantly depending on the surface temperature. Cooled ceiling has higher values of the convective coefficient (~ 4.4 W m^{-2} K^{-1}) then heated ceiling (~ 0.3 W m^{-2} K^{-1}). Algorithms reported in literature [5, 6, 17, 19] should be used for their correct evaluation.

Since radiant heat transfer coefficient is constant, heating and cooling capacity enhancement of a radiant system strongly depends on the possibility to increase convective heat transfer. For example it may be increased by introducing surface roughness to enhance turbulence. The most suitable solution is anyway to couple radiant system to ventilation system, creating air movements close to the radiant surface and consequently improving convective heat transfer.

9. Nomenclature

A	Area	m^2
$AUST$	Average unheated surface temperature	K
c_p	Specific heat	J kg^{-1} K^{-1}
$F_{\varepsilon\, s\text{-}j}$	Radiation interchange factor [10]	–
$F_{s\text{-}j}$	View factor between radiant surface and j-surface	–
h_c	Convective heat transfer coefficient	W m^{-2} K^{-1}
$h_{c,h}$	Convective heat transfer coefficient of human body	W m^{-2} K^{-1}
h_r	Radiant heat transfer coefficient	W m^{-2} K^{-1}
$h_{r,h}$	Radiant heat transfer coefficient of human body	W m^{-2} K^{-1}
h_{tot}	Total heat transfer coefficient	W m^{-2} K^{-1}
$\dot{m}$	Mass flow rate	Kg s^{-1}
Q_c	Convective heat flux	W
Q_{out}	Backward heat transfer	W
Q_r	Radiant heat flux	W
Q_{tot}	Total heat flux	W
T_a	Air temperature	K
T_{amr}	Adjusted air temperature	K
T_{mr}	Mean radiant temperature	K
T_{op}	Operative temperature	K
T_s	Radiant surface temperature	K
T_j	j-surface temperature	K
T_{ref}	Reference temperature	K
ΔT_w	Temperature difference between water inlet and outlet	K
ε	Emissivity	–
σ	Stefan-Boltzmann constant	W m^{-2} K^{-4}

Subscript

a	Air
a 0.1	Air at 0.10 m
a 1.1	Air at 1.10 m
a 1.7	Air at 1.70 m
c	Convective
h	Human body
j	j-surface
r	Radiant
s	Radiant surface
tot	Total

Acknowledgments

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Figure 1. The environmental chamber

Figure 2. Heat transfer coefficients for a cooled radiant ceiling

Figure 3. Heat transfer coefficients for a heated radiant ceiling

Figure 4. Convective heat transfer coefficients for a radiant ceiling

Figure 5. Radiant heat transfer coefficients for a radiant ceiling

Table 1. Heat transfer coefficients and reference temperatures

Table 2. Comparison between literature and measured heat transfer coefficients for a cooled ceiling

Table 3. Measured and calculated parameters for a cooled radiant ceiling

Table 4. Comparison between literature and measured heat transfer coefficients for a heated ceiling

Table 5. Measured and calculated parameters for a heated radiant ceiling

Figure 1
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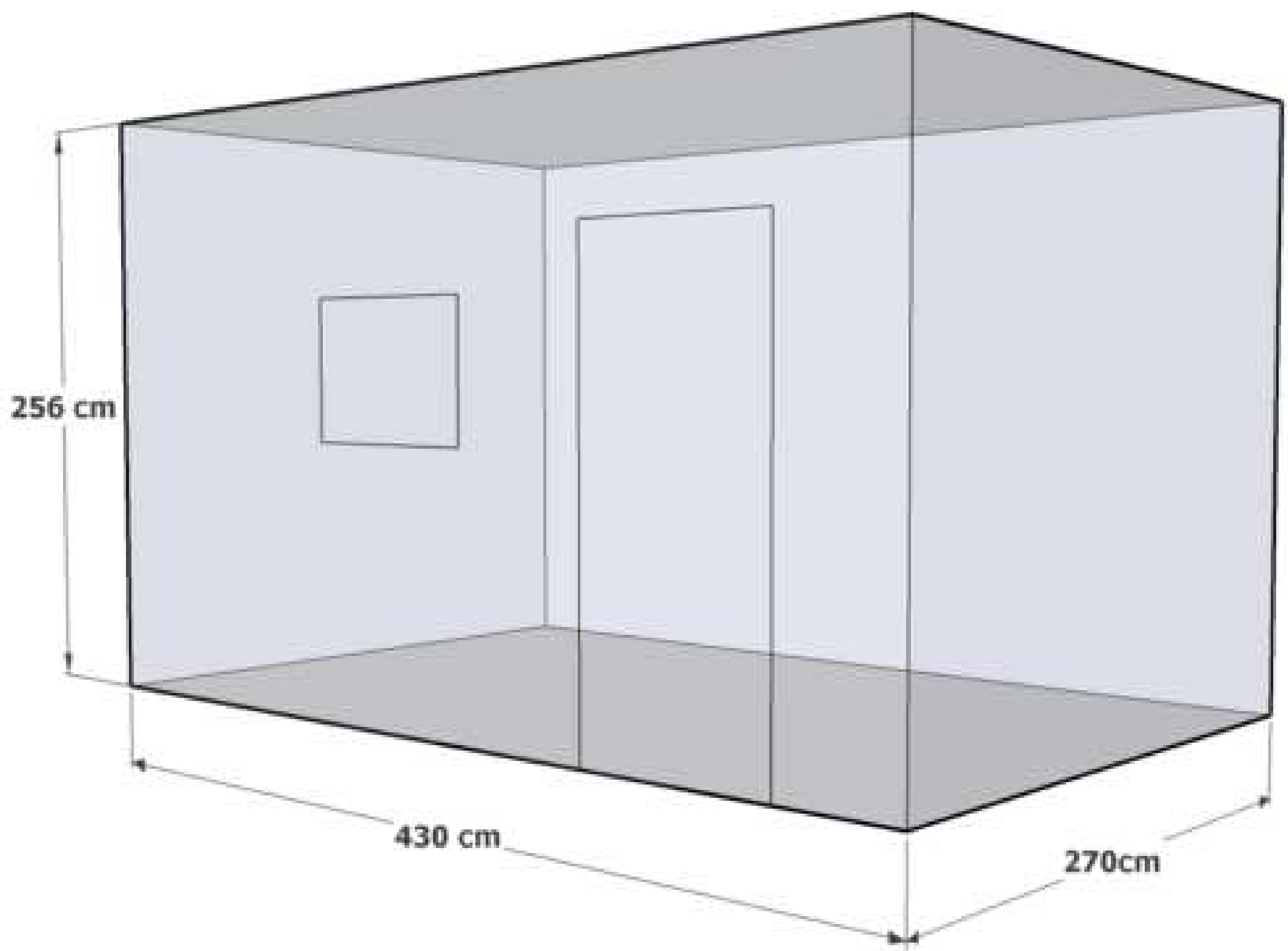

Figure 2
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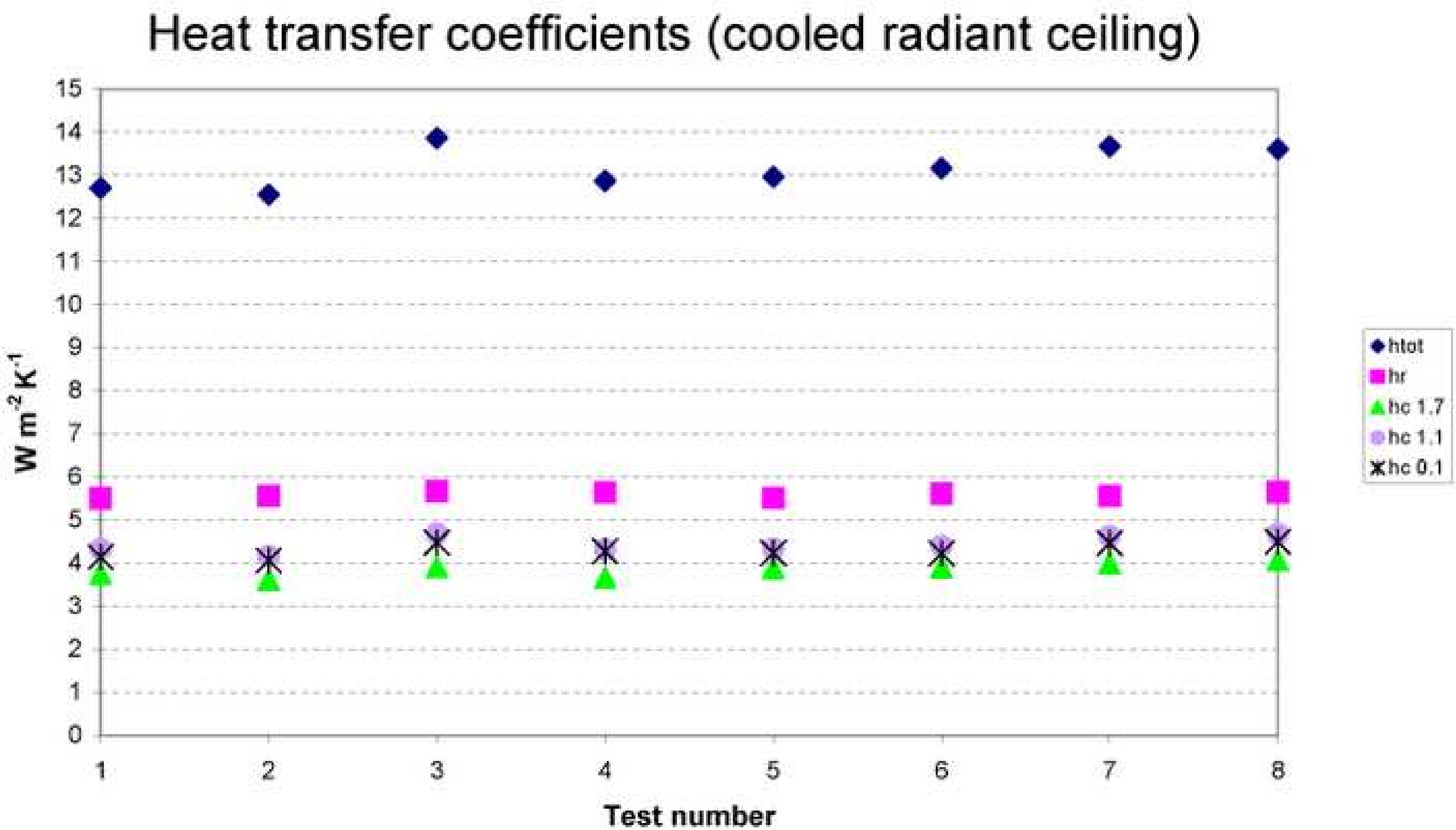

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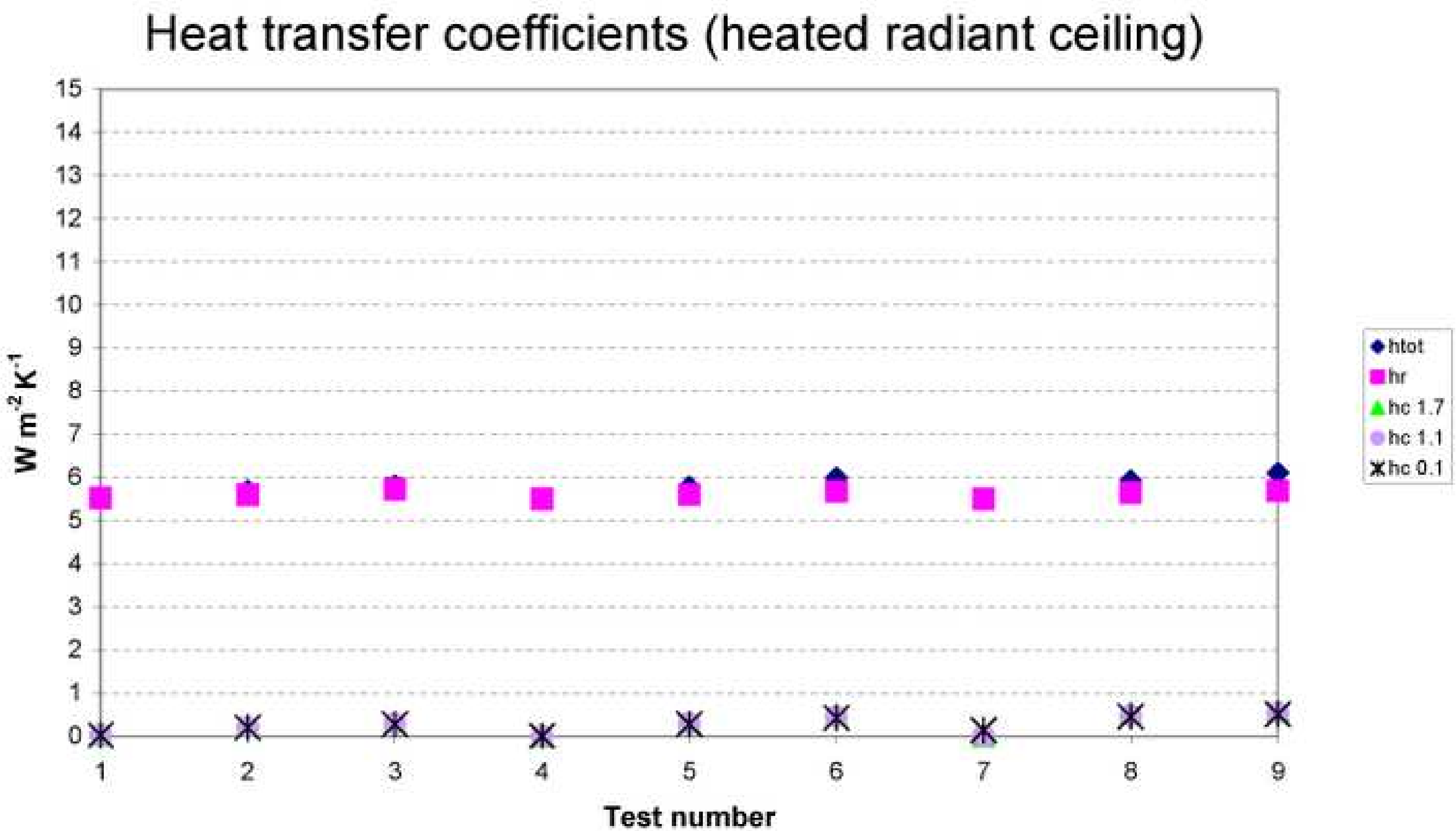

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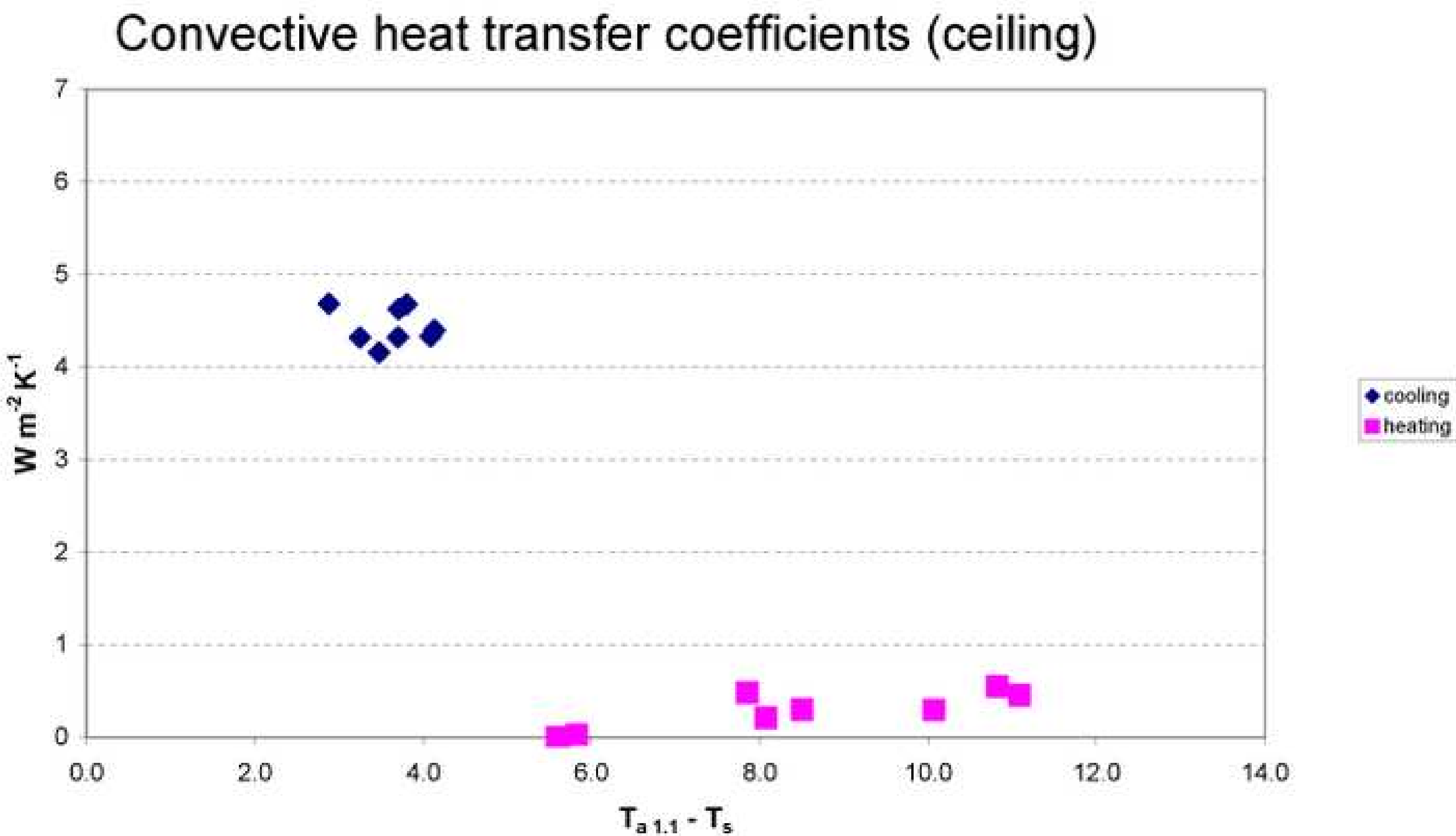

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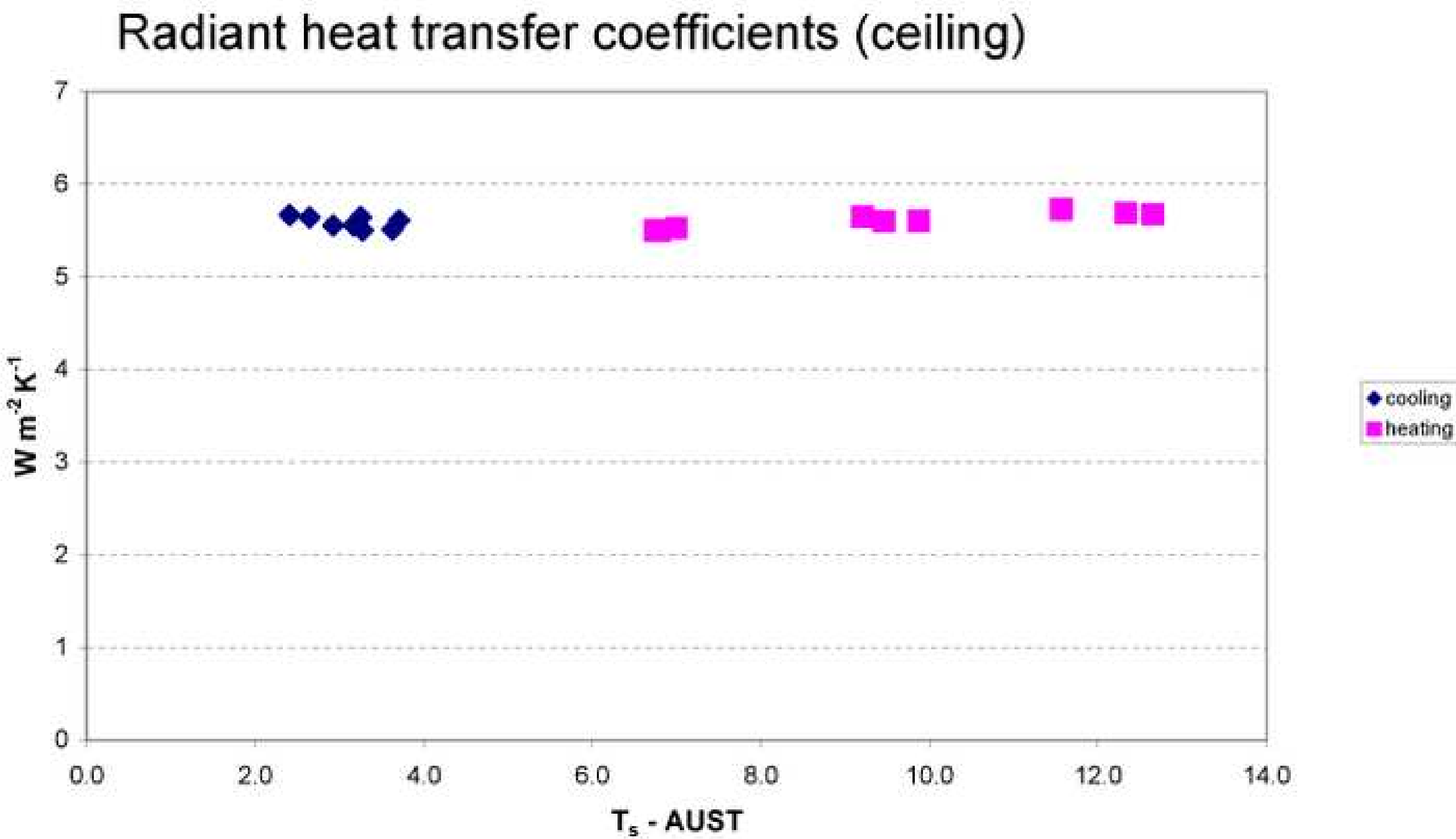

Table 1

Heat transfer equation	Heat transfer coefficient	Reference temperature (T_{ref})
$\frac{\dot{Q}_r}{A} = h_r\left(T_{ref} - T_s\right)$	h_r	$AUST$
$\frac{\dot{Q}_c}{A} = h_c\left(T_{ref} - T_s\right)$	h_c	T_a
$\frac{\dot{Q}_{tot}}{A} = h_{tot}\left(T_{ref} - T_s\right)$	h_{tot}	T_{op}

Table 2
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Literature values [W m^{-2} K^{-1}]			Measurements [W m^{-2} K^{-1}]		
h_c	h_r	h_{tot}	h_c	h_r	h_{tot}
3.1 - 4.4	5.5	11.0	4.4	5.6	13.3

Table 3

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	1 C_200_13	2 C_200_15	3 C_200_18	4 C_200_24	5 C_240_13	6 C_240_15	7 C_240_18	8 C_240_24
T_c	18.8	19.9	22.0	21.6	18.7	20.4	22.8	21.2
T_{sw}	21.2	22.0	23.5	23.4	21.2	23.0	24.9	23.4
T_{op}	21.8	22.6	24.1	24.1	21.9	23.7	25.7	24.2
T_{aw}	21.8	22.7	24.2	24.1	22.0	23.8	25.7	24.2
$T_{a\,1.7}$	23.1	23.9	25.4	25.4	23.3	25.1	27.1	25.6
$T_{a\,1.1}$	22.5	23.4	24.8	24.8	22.8	24.6	26.5	25.0
$T_{a\,0.1}$	22.7	23.5	25.0	24.9	22.9	24.7	26.6	25.2
Q_{tot}	38	34	30	32	42	43	39	40
h_{tot}	12.7	12.5	13.9	12.9	13.0	13.2	13.7	13.6
Q_r	22	20	16	18	24	25	22	22
h_r	5.5	5.6	5.7	5.6	5.5	5.6	5.6	5.6
Q_c	16	14	13	14	18	18	17	18
$h_{c\,1.7}$	3.7	3.6	3.9	3.7	3.9	3.9	4.0	4.1
$h_{c\,1.1}$	4.3	4.2	4.7	4.3	4.3	4.4	4.6	4.7
$h_{c\,0.1}$	4.1	4.1	4.5	4.3	4.2	4.2	4.5	4.5

Table 4

Literature values [W m^{-2} K^{-1}]			Measurements [W m^{-2} K^{-1}]		
h_c	h_r	h_{tot}	h_c	h_r	h_{tot}
0.5	5.5	6.0	0.3	5.6	5.8

Table 5

	1 H_160_30	2 H_160_35	3 H_160_40	4 H_200_30	5 H_200_35	6 H_200_40	7 H_240_30	8 H_240_35	9 H_240_40
T_c	26.1	29.1	32.8	26.5	29.4	32.6	26.5	29.8	32.6
T_{sw}	20.9	22.0	24.1	21.4	22.0	23.1	21.4	22.9	23.3
T_{op}	20.6	21.5	23.5	21.2	21.5	22.4	21.1	22.5	22.6
T_{asw}	20.6	21.5	23.4	21.2	21.4	22.3	21.1	22.4	22.5
$T_{a\,1.7}$	21.0	21.8	23.7	21.7	21.7	22.5	21.6	22.8	22.7
$T_{a\,1.1}$	20.3	21.0	22.7	20.9	20.9	21.5	20.8	22.0	21.7
$T_{a\,0.1}$	19.9	20.4	22.0	20.6	20.3	20.7	20.5	21.5	20.9
Q_{tot}	30	43	54	29	46	61	29	44	61
h_{tot}	5.5	5.7	5.8	5.5	5.8	6.0	5.5	5.9	6.1
Q_r	30	41	51	29	43	56	29	40	55
h_r	5.5	5.6	5.7	5.5	5.6	5.7	5.5	5.6	5.7
Q_c	0	2	3	0	3	5	0	4	6
$h_{c\,1.7}$	0.0	0.2	0.3	0.0	0.3	0.5	0.0	0.5	0.6
$h_{c\,1.1}$	0.0	0.2	0.3	0.0	0.3	0.5	0.0	0.5	0.5
$h_{c\,0.1}$	0.0	0.2	0.3	0.0	0.3	0.4	0.1	0.5	0.5