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Ceiling mounted radiant panels – calculations of heat output in heating and cooling application

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Abstract. This paper describes calculations of heat exchange between a radiant heating and cooling ceiling panel and the surrounding space. The calculations were created on the basis of empirical equations published in literature and experimental research of radiant ceiling panels carried out at the Institute of Environmental Engineering of the Poznan University of Technology. Changes in heating or cooling output vs. air and panel surface temperature difference were investigated. Based on the calculation results, the influence of the colour, gloss and shape of the panels surface on the heating or cooling capacity was indicated. Performed calculations show that the proportion of radiation in the whole heat exchange for the matte black ceiling panel reach 50% for cooling and 75% for heating mode for the considered temperatures. Comparing two rectangular, square and circular panels with the same area, it has been shown that the shape of the surface affects the heat exchange due to the difference in convectional flows in the boundary layers for different cases. Possible explanations of the discrepancies between the theoretical model and experimental results were presented.

1 Introduction

In recent years radiant heating and cooling system have gained popularity. The reason is good cooperation with renewable energy sources, like heat pumps. Radiant ceiling cooling is also characterized by high thermal comfort, hence it is a common solution in new energy-efficient buildings. With a large surface area, radiant systems have a significant influence on the sensible temperature, which is the arithmetic mean of the air temperature and the mean radiation temperature of all surrounding barriers. Using radiant surface systems, the same sensible temperature can be achieved at a lower air temperature compared to air and convectional heating. This generates significant energy, economic and environmental benefits per annum [1].

Designing a heating or cooling system using radiation must be dovetailed with appropriate precision. Any underestimation and subsequent replacement of system components can be very costly. An additional complication is that the performance of radiation systems strongly depends on indoor conditions as well as the texture and other properties of the heat exchange surface. All this aspects should be taken into account during designing process.

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This paper discusses the methods of determining energy performance of ceiling mounted radiant heating and cooling panels supplied by water. The aim of the study is to build a model of heat exchange of the panel. The basis are the results of experiments carried out at the Institute of Environmental Engineering at Poznan University of Technology. Model is created to analyse the influence of various factors on the power of heating and cooling panels.

2 Analytical model

2.1 Model description

The heat transfer model consists of two parts: convection and radiation. The free convection model creating is about choosing the appropriate equation to calculate the Nusselt number. A distinction must be made between heating and cooling options, because the convection heat flux differs significantly for those two cases. One can find a great number of free convection equations in the open literature [2–4]. The most credible equations are presented in monographs (eg. [5]). The radiant heat flux is analogous to heat exchange between a body and the surrounding space. In such case the influence of emissivity of surrounding space is negligible [6]. Only the emissivity of the panel is taken into account. Equations used in the model are shown below Eq. (1–3).

$$\mathrm{Nu_C} = 0.835 \cdot \mathrm{C_t} \cdot \mathrm{Ra}^{0.25} \tag{1}$$

$$\mathrm{Nu_H} = \frac{0.527}{\left(1+\left(\frac{1.9}{\mathrm{Pr}}\right)^{0.9}\right)^{\frac{2}{9}}} \mathrm{Ra}^{0.2} \tag{2}$$

$$\mathrm{q_R} = \sigma\varepsilon_s \left[\left(\frac{T_1}{100}\right)^4 - \left(\frac{T_2}{100}\right)^4\right] \tag{3}$$

where: Nu_C, Nu_H [-] – Nusselt number for the cooling and heating mode, C_t [-] – equation constant, for air equal 0.515, Ra [-] – Rayleigh number, Pr [-] – Prandtl number, σ [W/(m^2·K^4)] – Stefan–Boltzmann constant, ε_s [-] – panel surface emissivity.

The thermo-physical properties of air and water have been determined using approximation equations [7]. The reference air temperature is the average temperature of the boundary layer. The reference water temperature is the mean temperature between inlet and outlet water temperature. The characteristic length used to calculate the convectional heat flux is the hydraulic radius of the panel surface [5].

$$\mathrm{L} = \frac{\mathrm{A}}{\mathrm{P}} \tag{4}$$

where: L [m] – characteristic length, A [m^2] – panel surface area, P [m] – panel surface perimeter.

The panel surface is treated as isothermal. The heat exchange is assumed to take place in steady state conditions. The air humidity influence is neglected.

2.2 Model validation

Experimental studies carried out at PUT were based on a ceiling mounted hydronic radiant panel in heating and cooling mode [1, 8]. The panel surface was covered with black matt paint, hence the emissivity can be assumed equal 0.95. A comparison between the experimental results and the calculation model were done. The relative difference between the calculated and the experimental results was derived as follows:

$$\Delta h = \frac{h_{tot,calc} - h_{tot,exp}}{h_{tot,exp}} \cdot 100\% \quad (5)$$

The results of heat transfer coefficient obtained with the theoretical model are similar with the experiments for temperature difference greater than 4°C in heating and 6°C in cooling mode. A compliance within 10% for majority of the results is achieved. The largest discrepancies appear for low ΔT, where the accuracy of the experimental results was the lowest. The presented calculation model is accurately enough to can be treated as representative.

Table 1. Comparison of the experimental and analytical heat transfer coefficient for heating and cooling mode.

$h_{tot,exp}$	$h_{tot,calc}$	Δh	Nu	L	Ra	ΔT	T_s	T_a
[W/(m²·K)]		[%]	[-]	[m]	[-]	[°C]	[°C]	[°C]
Ceiling cooling								
5.6	9.94	78%	22.13	0.14	7.00E+06	2.39	27.18	29.57
7.33	10.15	38%	23.54	0.14	9.00E+06	2.96	26.22	29.18
8.77	10.55	20%	25.75	0.14	1.29E+07	4.25	25.61	29.86
10.09	11.05	9%	28.82	0.14	2.02E+07	6.40	23.71	30.11
10.6	11.49	8%	31.74	0.14	2.97E+07	8.88	21.29	30.17
11.23	11.78	5%	33.81	0.14	3.82E+07	10.78	19.21	29.99
11.19	12.10	8%	35.95	0.14	4.88E+07	13.13	17.13	30.27
11.17	12.35	10%	37.81	0.14	5.97E+07	13.66	15.70	29.36
Ceiling heating								
6.22	7.29	17%	9.99	0.14	9.60E+06	2.249	22.59	20.34
7.14	7.60	7%	11.3	0.14	1.77E+07	4.44	24.90	20.46
7.77	7.83	1%	12.07	0.14	2.47E+07	6.65	27.43	20.78
7.66	7.97	4%	12.54	0.14	2.99E+07	8.39	29.20	20.81
8.11	8.13	0%	12.96	0.14	3.52E+07	10.44	31.40	20.96
8.34	8.28	-1%	13.37	0.14	4.12E+07	12.86	33.76	20.9
8.28	8.39	1%	13.58	0.14	4.45E+07	14.54	35.71	21.17
8.26	8.52	3%	13.87	0.14	4.95E+07	16.97	38.08	21.11
8.29	8.62	4%	14.15	0.14	5.47E+07	19.39	40.10	20.71
8.49	8.72	3%	14.27	0.14	5.70E+07	21.14	42.12	20.99

3 Impact analysis

3.1 The influence of surface covering

The theoretical model was created to made an analysis of the influence of various shape and surface covering on the heat output of the radiant panel. Three surface coverings have been compared: black matt, green chromatic and polished aluminium. The emissivity of those

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surfaces are 0.95, 0.7 and 0.05 respectively. The panel surface in this analysis is a square of an area of 1 m^2. Obtained results are shown in Table 2 and Fig. 1–2.

The results of calculations show that with the same temperature differences, different values of heat transfer coefficient are obtained. The thermal output of the polished aluminium panel is at the average level of 49% of black matt panels output for cooling and 24% for heating. The power of the green panel is equal to 86% of the power of the matt black panel for cooling and 79% for heating mode.

Table 2. Heat transfer coefficient in the function of temperature differences for various surface emissivity.

ε = 0.95	**ε = 0.7**	**ε = 0.05**	**Temperatures**		**Temperature difference**
h_{tot}	**h_{tot}**	**h_{tot}**	**T_s**	**T_a**	**$\Delta T = \lvert T_s - T_a \rvert$**
[W/(m²·K)]	[W/(m²·K)]	[W/(m²·K)]	[°C]	[°C]	[°C]
Ceiling heating					
6.77	5.30	1.49	25	20	5
7.07	5.56	1.66	30	20	10
7.31	5.76	1.76	35	20	15
7.52	5.94	1.83	40	20	20
7.72	6.10	1.89	45	20	25
7.92	6.26	1.93	50	20	30
8.11	6.41	1.98	55	20	35
Ceiling cooling					
9.18	7.68	3.77	24	26	2
9.83	8.34	4.47	22	26	4
10.27	8.79	4.96	20	26	6
10.61	9.16	5.37	18	26	8
10.91	9.47	5.72	16	26	10
11.18	9.75	6.04	14	26	12
11.43	10.02	6.34	12	26	14

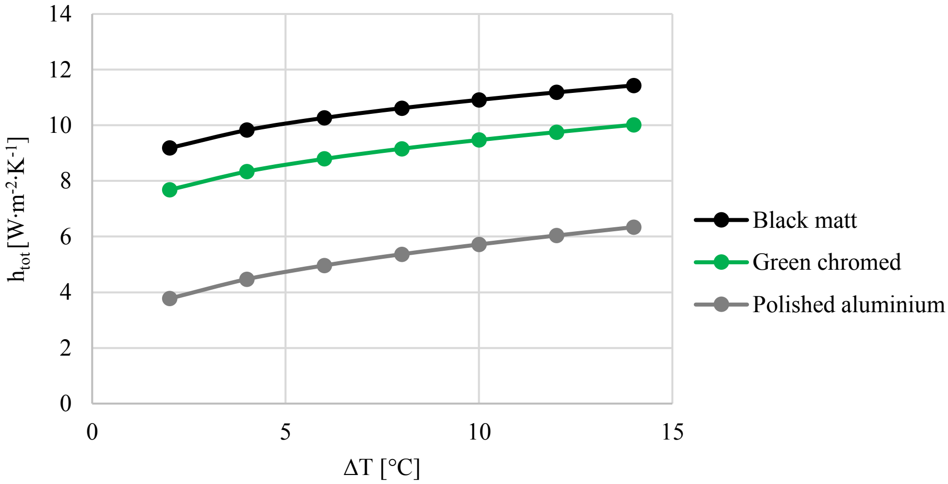

Fig. 1. Heat transfer coefficients as a function of temperature difference depending on surface coverage – ceiling cooling.

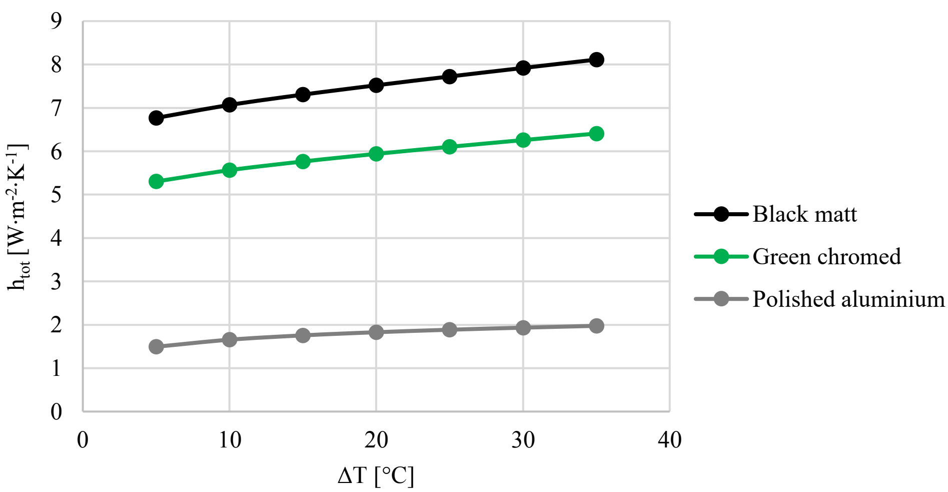

Fig. 2. Heat transfer coefficients as a function of temperature difference for various surface coverage – ceiling heating.

3.2 The influence of panels shape

The free convectional heat flow depends on the characteristic length of the heat exchange surface. In the analytical heat transfer model, the characteristic length is defined as the hydraulic radius of the panel surface. Calculation were carried out, to determine the qualitative and quantitative differences in heat output for 4 panel shapes: rectangular with a side length ratio 1:10 and 1:2, square and round. Calculations were made for the surface area of 1 m^2 and the emissivity equal 0.95. Results are presented in Table 3.

In both cooling and heating operations, the highest performance is achieved by a rectangular panel with a side length ratio of 1:10. In both cases, the circular shape is the least effective. The heat flux density of the round panel constitutes 93% for cooling and 92% for heating of the maximal heat flux density achieved by the 1:10 rectangular panel. Heat transfer coefficients for rectangular shapes with a side length ratio of 1:2, square and round are similar. The relative discrepancies between the square and rectangular area of 1:2 do not exceed 1%, whereas the difference between circular and rectangular area of 1:2 is less than 2.3%.

Table 3. Results of heat transfer coefficient as a function of temperature differences for various surface shapes.

Rectan. 1:10	**Rectan. 1:2**	**Square**	**Round**	**Temperatures**		**Temperature difference**
h_{tot}	h_{tot}	h_{tot}	h_{tot}	T_s	T_a	$\Delta T = \lvert T_s - T_a \rvert$
[W/(m²·K)]	[W/(m²·K)]	[W/(m²·K)]	[W/(m²·K)]	[°C]	[°C]	[°C]
Ceiling heating						
7.64	7.27	7.23	7.15	25	20	5
8.05	7.63	7.58	7.50	30	20	10
8.35	7.90	7.85	7.76	35	20	15
8.60	8.14	8.09	7.99	40	20	20
8.83	8.35	8.30	8.20	45	20	25
9.05	8.56	8.51	8.41	50	20	30
9.26	8.76	8.71	8.60	55	20	35
Ceiling cooling						
9.70	9.23	9.17	9.06	24	26	2
10.45	9.88	9.81	9.68	22	26	4
10.96	10.33	10.25	10.10	20	26	6
11.37	10.68	10.60	10.44	18	26	8
11.72	10.98	10.90	10.73	16	26	10
12.04	11.26	11.16	10.98	14	26	12
12.33	11.51	11.41	11.22	12	26	14

4 Discussion and conclusions

The calculations have shown that the heating and cooling capacity increases with the temperature difference. The percentage of radiation in heat exchange for the analysed range ΔT = 2–14°C for cooling constitutes between 47% and 62% of the total energy absorbed by the matt black panel. When we limit the emission of the panel, we significantly reduce the density of the heat stream possible to obtain. Analytical model proofs that while replacing a matt black panel with a surface temperature of 22°C with a polished aluminium panel, its surface would have to be at a temperature of 18.7°C (at an air temperature of 26°C). Assuming an air temperature of 24°C and a relative humidity of 50%, the dew point temperature is 12.8°C. While maintaining a certain safety margin, the surface temperature of the panel should not drop below 14°C. In this case, the maximum heat flux density for a matt black panel would be 109 W/m^2 and that for a polished aluminium panel would be only 59 W/m^2. It is 53% of the higher result. In the case of heating, the differences are even greater.

Because the average temperature between the panel and the surrounding surfaces increases, the radiant heat exchange is more intense. For the maximum temperature difference between the ceiling panel and air at 20°C of 25°C, the matt black panel reaches a heat transfer density of 193 W/m^2, the polished aluminium panel 47 W/m^2. It is 24% of the black panel power. Presented examples show that it is important to take into account at the design stage all the features of radiant systems. In particular those that affect the radiant heat transfer, i.e. texture, gloss, surface colour or possible metallic additions to the varnishes.

The second issue discussed during the sensitivity analysis was the influence of panel shape on heat exchange. The rectangular panel with a ratio of 1:2 sides length, square and circular panels achieved very similar values of heating and cooling power. Discrepancies of 2.3% are within the limits of the accuracy of the used equations. It is not possible to determine which of the three is the most efficient. Certainly, the differences do not matter in engineering applications. Hence it was proved that a rectangular panel with a ratio of 1:10 sides' length is characterized by heat transfer coefficients higher by 7–10% (in relation to circular). Such difference cannot be negliged.

The shape influence of the heat exchange consist of differences in variations in convectional moves of the fluid. Three zones of the boundary layer can be distinguished for horizontal plates. Two of them are symmetrical in relation to the axis of symmetry, on both sides of the board. Air velocity vectors in both zones are parallel to the surface of the plate and the fluid particles move from the edge of the plate to the centre. The third zone forms a streak of fluid, the vectors of fluid velocity are perpendicular to the plate surface [Fig. 2. a, b]. The thickness of individual zones and the intensity of heat transfer in these zones strongly depends on the shape of the plate, its size and the temperature difference between the incoming fluid and the plate [6, 7].

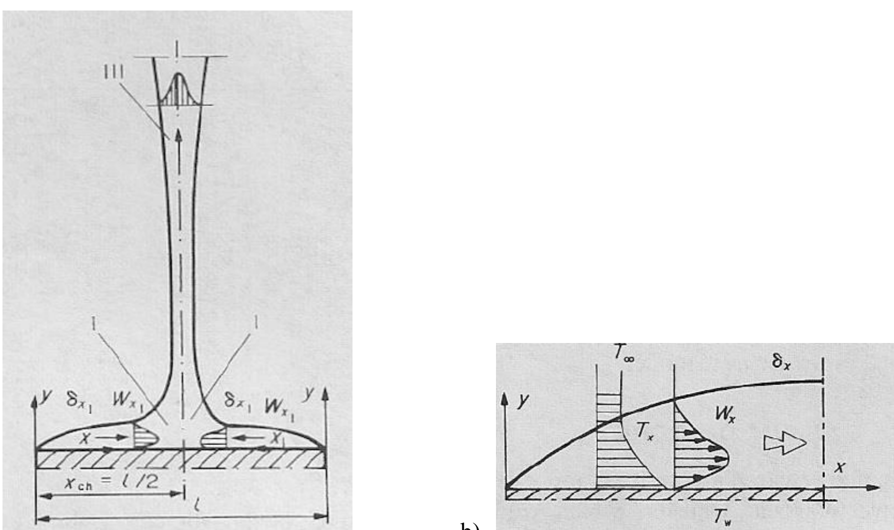

Fig. 3. Model of the convection heat transfer from a horizontal plate: (a) Three boundary-layer regions, (b) Region I [9].

As the fluid flows along the surface, it exchanges heat with the plate and its temperature is approaching the plate temperature. The intensity of heat exchange reduce and the fluid velocity increases with moving towards the centre. During heat exchange on the surface of the long and slim panel, the fluid flows onto the surface of the panel mainly from the long side, intensively exchanges the heat over a short section and forms a relatively slim chimney illustrated by Zone III. Most of the panel surfaces actively participate in heat exchange.

In the case of a circular or square plate, the fluid must pass much longer distance to the centre of the plate and there is a substantial chance that it will receive enough heat on the way to break from the slab and create zone III. In this case, the central part of the panel may not participate at all or only to a negligible extent in the heat exchange.

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