Ceiling panels radiant heating systems

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SUMMARY
Radiant panel heating systems are widely used for different kinds of building, both commercial and residential. The aim of the publication is to identify advantages and disadvantages of heating system technical design for industrial buildings, to evade design deficiencies in future projects for practical use. It was important to prove that heating systems with low temperature of heat carrier could compensate building heat losses and provide designed comfort temperature in transient season (before and after heating season), when sharp changes of outdoor temperature are typical. The topical issue of the system is based on energy savings potential and ability of low temperature heating systems usage with renewable energy sources, such as solar systems and heat pumps.

KEYWORDS
Radiant heating ceiling panels, low temperature heating system, radiant and convective heat.

1 INTRODUCTION

Radiant heating ceiling panels with low temperature heat carrier were chosen as an object of theoretical evaluation and experimental tests. To understand system's working outlines is important for us to show percentage ratio of radiant and convection heat flow in the heated area. Proposal of destratifacators usage for warm air of upper indoor air layers of the heated area to heat the area additionally was described.

The experimental polygon is one of industrial buildings, where production of building construction prefabricated components modules are planned to be done, for production process heavy machinery transport is necessary. The building is meant for a warehouse and production areas. This plant is situated west part of Latvia territory, in Tukums.

Fig. 1a Location of a polygon

Fig. 1b Industrial plant

According to the technical design radiant heating panels were installed to compensate heat losses of the plant and heat amount required by ventilation system for afterheating of supply air. Heat carrier inlet temperature parameters were defined as 40-55^{0}C. This type of radiant panel is used for heating as there is topside insulation on the panel. Partial sensible load of a panel can be controlled by the supply water temperature. Panels are situated in different zones that could provide convenient regulation if indoor air temperature. Radiant ceiling panel surface exchange heat with room air by convection and other room surfaces through radiation.



Fig. 2a Experimental polygon Fig. 2b Heating panel Fig. 2c Heating panel

Radiant ceiling panels are made from 0.5 mm-thick sheet steel with clip profiling. Each one accommodates four precision steel tubes and the top-surface insulation. The system operates at heat carrier temperatures with input of 40-55°C and output of 30-40C°. [4]

2 MATERIALS/METHODS

While convection heat transfer between liquid and rigid body surface takes place, heat transfer with thermal conductivity occurs simultaneosly. Heat output of a panel at a given surface temperature depends upon the indoor air temperature, temperatures of the uncontrolled indoor surfaces, air movement in the heated space and other characteristics like surface emission. The natural convective heat output depends also on the altitude, and the size of the space. [1,2,5] It is assumed that the slab operates as a plate fin and loses heat from the upper surface. The total panel heat flux is the algebraic sum of the radiant and convective heat output fluxes [1]

$$q_y=q_r+q_c \qquad (1)$$

where

$$q_r = U_r \cdot (t_p - AUST) \qquad (2)$$
$$q_c = U_c \cdot (t_p - t'_a) \qquad (3)$$

when there is outdoor exposure with large fenestration area, the air temperature of the indoor air sweeping a heated floor may approach *AUST.*
Here, U_r and U_c are the radiation and convection heat transfer coefficients on the panel surface, respectively:

$$U_r = r \cdot F_t \cdot \sigma \qquad (4)$$

Linearization factor r for the radiation heat transfer may be further simplified in the practical temperature range for panel heating:

$$r = [O.O105 \cdot (t_p + AUST)/2 + 0.7955] \cdot 1O^{+8} \quad (15°C < (tP + AUST)/2 < 30°C). \ (5)$$

F_r is the simplified radiation interchange factor for $A_r/ A_u \leq 0.30$ (6)

The convective term involves altitude (h) and room size (D,) corrections [1]

$$U_c = (1 - 2.22\cdot10^{-5} \cdot h)^{2.67} \cdot (4.96/D_e)^{0.08} \cdot 2.67 \cdot (t_p - t'_a)^{0.25} \qquad (7)$$

The room size is characterized by D_e:

$$D_e = 4 * A_f/L_f \qquad (8)$$

The area-weighted average temperature of the unheated (uncontrolled) indoor surfaces, *AUST* is rather difficult to predict. It primarily depends on the area ratio of outdoor to indoor exposure, thermal properties and dimensions of the partition and wall elements, as well as the outdoor conditions.
An approximate expression was given for practical design purposes: [1]

$$AUST \approx t'_a - d \cdot z \qquad (9)$$

Here *d* is the room position index. d is 3 is for a room with two or more outdoor exposed sides. Z is a function of outdoor design temperature, t,:

$$z \approx 15/(25 + t_b) \; (t_b > -20°C). \qquad (10)$$

Table 1. demonstrates theoretically evaluated data for case study.
Table 1. Theoreticaly evaluated data

Name of position	**Units**	**Value**	**Formula**
Building parameters			
Heating load of indoor space	Q, W	500 000	
Perimeter of heated space	L_f (m)	340	
Total floor area in heated space	A_f (m^2)	6 000	
Total surface area in heated space	A_U (m^2)	8 720	
Altitude of location above sea level	h (m)	40	
Heating systems parameters			
Single panel surface	A_p (m^2)	108.00	
Heating panel load	Q_y (W/m2)	534.19	
Panel surface area in heated space	A_R (m^2)	936.0	
Indoor design temperature of heated space	t'_a °C	15	
Outdoor design temperature	t_b (°C)	-20	
Effective panel surface temperature	t_p (°C)	35	
Room position index	d	3	
Emissitivy	σ (W/m^2K^4)	0.0000000567	
Evaluated values			
Function of outdoor design temperature	z (°C)	3.0	[10]
Average temperature of unheated indoor surfaces	AUST (°C)	6.0	[9]
Linearization factor for radiation heat transfer:	r (°C^3)	101 131 002	[5]
Simplified radiation interchange factor	F_r	1.000	[6]
Area size characterized value	D_e	70.588	[8]

Convection heat transfer coefficient on panel surface	U_c ($W/m^2 K$)	4.56	[7]
Radiation heat transfer coefficient on panel surface	U_r (W/m^2K)	5.734	[4]
Radiant heat flux on panel surface	q_R (W/m^2)	166	[2]
Convective heat flux on panel surface	q_c (W/m^2)	91	[3]
Total heat flux	q_y (W/ m^2)	257	[1]

3 RESULTS

All the values of for theoretical evaluations such as heat losses of the building, heat carrier supply temperature and as also constructive parameters are shown in table 1. The characteristic values and correspondent coefficients of the experimental polygon are shown, based on formulas mentioned in the theoretical part of the case study the evaluation has been made and results are presented in the table below.

Fig. 3 Experimental polygon photo and thermo camera pictures

Pictures shown on Fig.3 allow getting inside surfaces temperature data of the building to estimate calculations. Gained results can serve as material that will assess availiable variety of heat energy consumption decrease in terms of heat carrier in transmission from heat channel to room where temperature is controlled. Difference between effective panel surface temperature and indoor design temperature of a heated space mostly effects on the convective heat flux. Percentage ratio of the convective heat flux is in range from 58.91% to 63.63% with heat carrier supply temperature range from 37 to 55 °C. The blue graph curve on the Fig.4 presents the range of percentage for convective heat flux part in the whole heat flow. The higher the difference is the higher the percentage is. The green graph curve on the Fig.4 presents the range of percentage for radiant heat flux part in the whole heat flow.

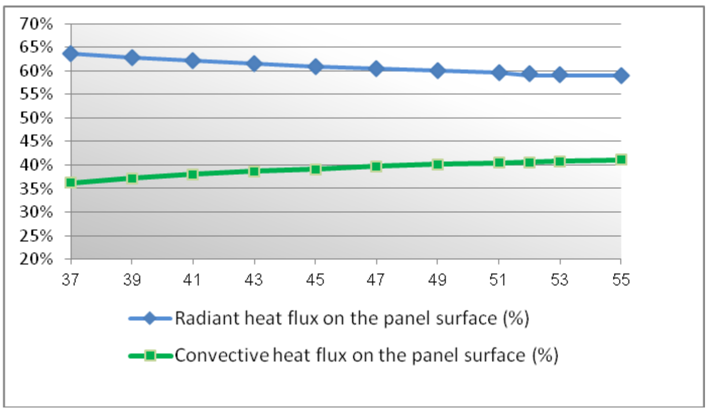

Fig.4 Range of percentage for radiant and convective heat flux

In practice radiant and convective heat fluxes have not been measured in details, but smoke tests have been performed. As it can be seen on pic.2 warm air mass has been concentrated under the roof. Lightened roofing farms allow air to spread across all the building, not concentrating in one separate section of roofing. But the “warm air pillow” effect still negatively influences on heat balance of the building.

Fig.5 Smoke tests

To improve the situation destratificators are proposed to be used. These devices are designated for destratification of the air for industrial areas with high ceiling height to lower useless consumption of heat. Blasting grid is design allow to change air deflation angle in dependence on installation height and required comfort level. Destratificator design is aimed achieving maximal effectively of the system, by aspirating air from areas under the roof and direct it to the work zone.

Measurement instrument was installed in the industrial plant for period of a week. Black ball thermometer measured radiant temperature, while the installed thermometer was put to measure indoor air temperatures (channel one and two) and relative air humidity. Graph 3 demonstrates values that give common view on the heating balance (relative humidity – right Y ax, temperature values – left Y ax) of the building and outdoor air temperature changes during one day.

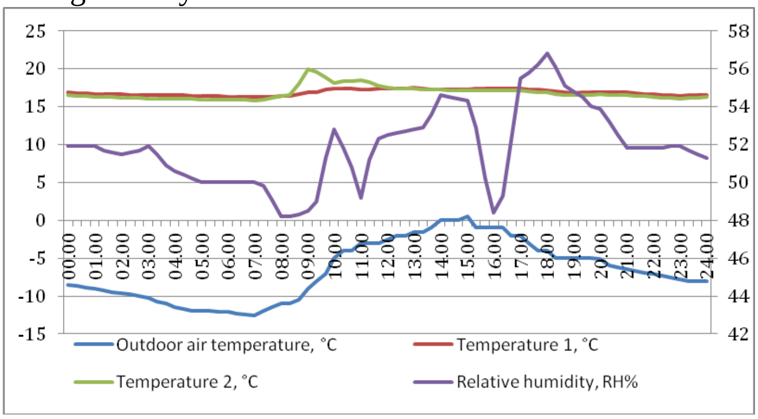

Fig.6 Measured values for one day

4 DISCUSSION

The graph indicates that during only one day the outdoor air temperature ranged from -13°C to 0°C and heating system with heat carrier temperature 45°C successfully managed to provide indoor air designed temperature. These conditions allow using heating source economically, which provides a positive effect.   Temperature 2. curve on the fig. 8 increases during period of time from 9:00 to 11:00, which is explained by sunshine activity, fenestration location facilitated to this effect. Fig. 6 curve demonstrates outdoor air changes during this period of time, sharp changes of outdoor temperature are typical of transient season. In Latvia in March and October, as well as partly in April, September and November traditional heating system with high heat carrier inlet temperature mostly does not work on a full capacity, it is switched on and off periodically, which is not effective, as system is heated and cooled down completely. As mentioned period of the year is about 30% of all heating season, economical effect could be impressive. Curve on the fig. 7 shows system suitability to the technical design requirements, despite sharp changes of outdoor air in short periods of time. As it was mentioned above sharp increase on temperature 2 can be explained by sunshine activity.

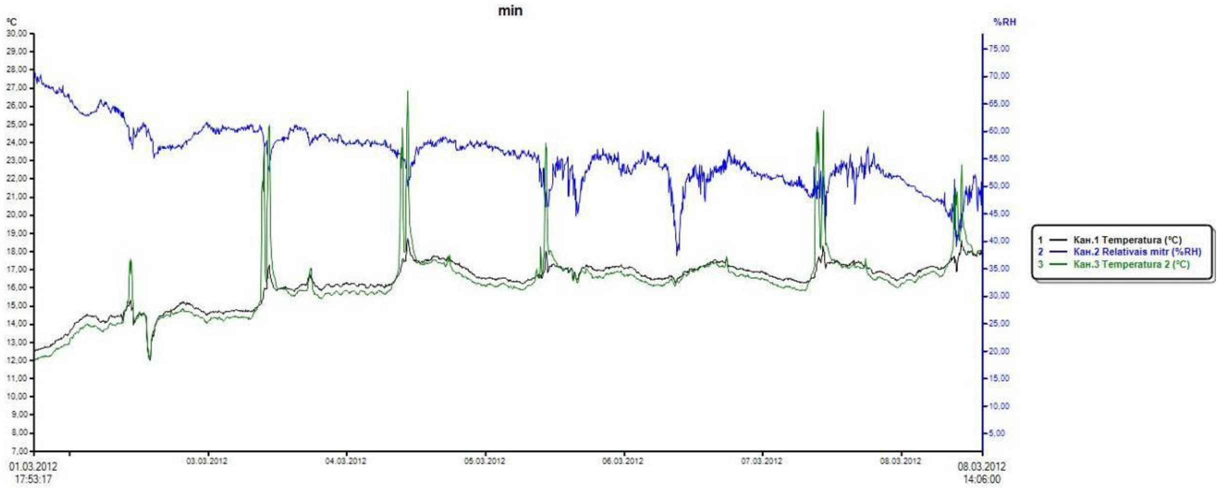

Fig.7. Indoor air temperature and relative humidity measurement in period of experiment

5 CONCLUSIONS

In conclusion it must be mentioned that having experimental data, we managed to estimate theoretically radiant and convective percentage ratio. The temperature difference between the heating panel surface temperature and the temperature of the building envelope is the main value influencing on the range of percentage for radiant heat flux. Percentage ratio of the radiant heat flux is in range from 36.37 to 41.09% with heat carrier supply temperature range from 37 to 55 °C. The temperature difference between the building envelope and indoor air temperature is the main value influencing on the range of percentage for convective heat flux. Percentage ratio of the convective heat flux is in range from 58.91% to 63.63% with heat carrier supply temperature range from 37 to 55 °C. Smoke tests showed “warm air pillow” effect spreading across all the building due to lightened roofing farms, destratificators are proposed to be used to improve the situation. Taking into the account that sharp changes of outdoor temperature are typical of transient season, economical effect of low temperature heating system’s usage could be impressive. The positive effect is possibility of using natural resources as an energy source, as for example ground heat pump could provide required temperature for heat carrier inlet without additional heat up.

ACKNOWLEDGEMENT

Acknowledge assistance was made by Riga Technical University.

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