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STUDY OF OPERATIVE TEMPERATURE USING THE NOVEL DETAIL APPROACH IN DETERMINING MEAN RADIANT TEMPERATURE – COMPARISON BETWEEN WALL-MOUNTED CONVECTOR AND CONVENTIONAL RADIATOR

Summary

Most of the physical parameters that are used to assess the satisfaction with the ambient thermal condition in a mathematical way are contained within the definition of operative temperature. This temperature, which can be used as a representative of indoor thermal comfort, is a function of the air temperature, the mean radiant temperature and the relative air velocity.

In this paper, the room air, mean radian temperature and indoor air velocity were determined experimentally for wall-mounted convector and conventional radiator at controlled room conditions. The room air temperature and indoor air velocity were continuously measured at several positions and heights (0.75 m and 1.5 m) using calibrated T-type thermocouples and hot wire probes, while mean radiant temperature was calculated using the thermograms captured by the IR thermal camera and numerically computed radiation view factors. Each wall was divided into several sections with approximately similar temperatures (differences < 0.5 °C) for which view factors were determined. Thermal heat output of the tested heat emitters was derived according to EN 442-2:2014. Obtained results were analysed and conclusions about the achieved thermal comfort and related energy saving were made accordingly.

Key words: *Mean radiant temperature, operative temperature, thermal comfort, thermography, heat emitters*

1. Introduction

European and national regulations based on EPBD (Energy Performance of Buildings Directive) seek to decrease energy consumption in buildings, without decreasing the thermal comfort, to achieve goals which include emission control and reduction of primary energy consumption. Prevision and assessment of energy consumption in households are essential items in achieving these goals. Heating, ventilation and air conditioning systems (HVAC) account for over 60% of total energy consumption in buildings [1]. High energy consumption of traditional HVAC systems is based on the aspirations to achieve and maintain uniform air temperature in rooms, preferably at an interval of 2°C [2]. However, recent research has shown that the overall satisfaction with thermal comfort was accomplished in only 11% of buildings [2].

Nowadays, both new and renovated buildings have to be equipped with contemporary HVAC systems that have high performance and efficiency in order to achieve EUs 20-20-20 goals. By increasing the reliability of installed systems and measuring the relevant parameters, a major contribution to energy consumption reduction can be done. Energy savings, with maintaining the same or increasing the level of thermal comfort, are some of the main research directions in buildings energy sector.

„Thermal comfort is defined as that condition of mind which expresses satisfaction with the thermal environment“ [3]. Definition is easy to understand but difficult to describe, especially using mathematical equations, due to the large quantity of environmental and human parameters that have to be taken into account. Some of these parameters are air temperature, air velocity, wall temperature, relative air humidity, air quality, brightness level, noise level, etc.

To create an environment comfortable to man is one of the most important factors when it comes to building design. Human feeling of thermal comfort is strictly linked to metabolic heat production which includes heat flux exchanged between human bodies and their environment as well as the physiological parameter variations [4]. When it comes to measuring environmental conditions in a room, it is important to understand that humans do not feel the air temperature directly. Humans actually feel thermal energy losses from the body. For example, warm surfaces can cause a person to feel that the room air temperature is higher. On the other hand, cold walls and windows (in winter) can create a feeling of chilled air although the temperature is at a satisfactory level.

Thermal comfort/discomfort can be described with PMV and PPD indices [3] based on the 7-point ASHRAE scale of thermal condition satisfaction [5]. According to the rational approach [4], assessment of environmental thermal properties demands determination of 6 parameters: two of them are subjective (thermal isolation of worn clothes and metabolic rate) and four physical (air temperature, mean radiant temperature, air velocity and relative air humidity). These parameters, especially physical ones, have to be measured in order to assess the thermal comfort level in a mathematical way.

Measuring standards [3, 6] are based on the use of operative temperature as criteria for thermal comfort. „Operative temperature is a temperature of an enclosed black space in which a person would experience the same amount of heat losses via radiation and convection as well as in the actual room with geometrically uneven surfaces but with equal relative air humidity and air velocity“ [3]. To determine the operative temperature, mean radiant temperature has to be measured. „Mean radiant temperature is a temperature of an enclosed black space in which a person would experience the same amount of heat losses as in the actual room“ [3]. Its value depends on the position of a person in a heated/cooled environment. Position of human body takes into account view factors which are calculated using complex mathematical equations [4, 7]. Therefore, mean radiant temperature depends on the room geometry, insulation level, position of windows and the type of the installed heating/cooling system. In case of heating, mean radiant temperature can be reduced by increasing the heated surface and decreasing the surface of a heat emitter [8].

Radiation is the most important mechanism of heat transfer in human body [9]. Radiation, both short-wave and long-wave, between a person and the environment can be monitored and modelled. The most accurate way of monitoring the mean radiant temperature is by measuring short-wave and long-wave heat flow densities from objects that are placed in human environment and by calculating its view factors (a part of radiation received from a human body in a particular direction) [10]. If the heat flow density is known, together with view factors, mean radiant temperature can be calculated considering the well-known Stefan–Boltzmann law.

Mean radiant temperature can be measured using globe thermometers together with measuring the air temperature and air velocity [11]. It can also be determined by using models such as ENVI-met [12], RayMan [13] or SOLWEIG [14]. These models are mainly used for calculation of outdoor mean radiant temperature. In that case, weather data measured at meteorological stations are used as input. Unlike external conditions where the mean radiant temperature can be higher than outdoor air temperature by more than 30 K [15], differences in a closed space are usually be very small. For that reason, climate studies in a closed space are often limited to the assumption that the mean radiant temperature is equal to the indoor air temperature [16], which is however not accurate enough.

Several devices for determining the mean radiant temperature are available on the market [17]. The most commonly used instrument is globe thermometer because of its low price and good traceability. On the other hand, this device is characterized by high response times (which results in an inability to provide successive measurements) and also, because of its spherical shape, overestimates the radiation contribution of horizontal surfaces (floor and ceiling). On the top of that, globe thermometer does not allow the assessment of variability of the mean radiant temperature in a room which is one of the main causes of thermal discomfort in enclosed spaces [18].

In literature several papers that deal with the determination of operative temperature of various heat emitters can be found. Laboratory measurements of vertical temperature gradient and operative temperature of three heating systems (radiator, underfloor heating and ventilation air supply heating) is discussed in [19]. Authors in [20] focused on thermal comfort achieved with panel, floor and wall heating system. CFD model based on laboratory measured specifications was assessed in [21] and energy savings and achieved thermal comfort obtain with ventilation radiators in [22]. In [23] authors proved that low temperature heating systems result in better thermal comfort compared to the other available on the market. Authors in [24] state that using the operative temperature, as it is not yet implemented in the prEN ISO 15316-2:2014 methodology for calculation of losses generated by heating and domestic hot water production, instead of the air temperature will create a difference in losses incurred in the generation subsystem.

In this paper, the results of measuring the operative temperature, as a representative parameter of thermal comfort, are given for two types of heat emitters. The goal was to compare operative temperatures achieved in the laboratory conditions by using two different types of heat emitters (wall-mounted convector and conventional radiator) with the same heating power and same surface walls cooling rate. The impact of newly constructed, low priced, wall-mounted convector on thermal comfort is conducted and main advantages compared to the conventional radiator are pointed out.

Operative temperature was determined at 10 different places in the test room. In contrast to the other methods (using globe thermometer in [25] as described in [26]), in this paper, operative temperature was determined by means of an IR thermal camera and numerical methods (calculate radiation view factors), respectively for each location. This represents a new approach in determining the mean radiant temperature (and consequently operative temperature) which demands using measuring instruments with shorter response times and allows bigger flexibility in latter data analysis (it can be analysed for various body shapes).

2. Methodology

2.1 Determination of the operative temperature

For each position the operative temperature (t_o) was calculated according to Eq. 1-2 based on three parameters: air temperature (t_a), mean radiant temperature (t_{mrt}) and air velocity (v_a) as described in [6]:

$$t_o = \frac{t_{mrt} + t_a}{2} \tag{1}$$

if $|t_a - t_{mrt}| \leq 4$ and $v_a < 0.2$ or:

$$t_o = (1-a)\cdot t_{mrt} + a\cdot t_a \tag{2}$$

where:

a=0.5 if $v_a < 0.2$;

a=0.6 if $0.2 < v_a < 0.6$;

a=0.7 if $0.6 < v_a < 1$.

Indoor air temperature was measured using calibrated T-type thermocouples at 10 different places in the test room. Thermocouples were set on two levels of height (five thermocouples were placed 0.75 m above the floor and five were set at 1.5 m above the floor). Positions of thermocouples marked with ordinal numbers from 1 to 10 are shown on Fig. 1. Positions of the radiator, convector and entry door are marked respectively with roman numerals I, II and III.

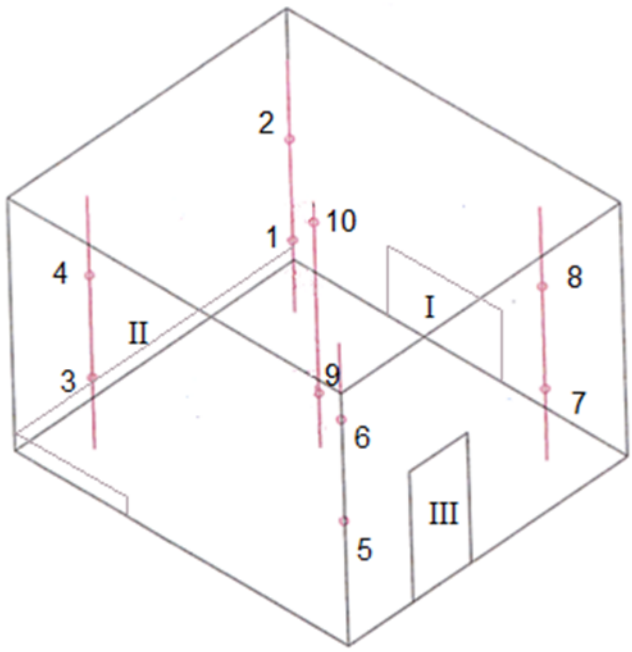

Fig. 1 Test room

In order to calculate the mean radiant temperature (Eq. 3), temperature distribution of each inner wall (in steady state condition determined in [27]) had to be determined. This was obtained from thermograms captured by the IR thermal camera.

$$t_r^4 = t_1^4 \cdot F_{p-1} + t_2^4 + F_{p-2} + \ldots + t_n^4 \cdot F_{p-n} \tag{3}$$

where:

$t_1,\ t_2,\ \ldots,\ t_n$ – inner wall surface temperature [K];

$F_{p-1},\ F_{p-2},\ \ldots,\ F_{p-n}$ – radiation view factor [-].

Radiation view factor (Fig. 2) *Fij* between two small surfaces *Ai* and *Aj* is defined as a fraction of the radiation leaving area *Ai* that is intercepted by area *Aj* and it depends on the orientation of areas and the distance between them.

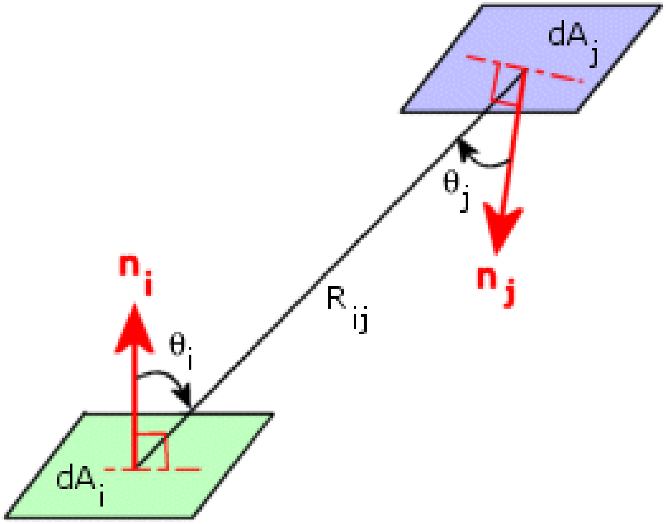

Fig. 2 Radiation view factor of infinitesimal surfaces

For two infinitesimal surfaces dAi and dAj, the view factor dFij is given by:

$$\mathrm{d}F_{ij} = \frac{\cos\theta_i \cdot \cos\theta_j}{\pi \cdot R_{ij}^2} \tag{4}$$

where:

θ_i, θ_j – angles between surface normals;

R_{ij} – distance.

For two finite areas the view factor can be calculated according to Eq. 5.

$$F_{ij} = \frac{1}{A_1} \int_{A_1} \int_{A_2} \frac{\cos\theta_i \cdot \cos\theta_j}{\pi \cdot R_{ij}^2} \mathrm{d}A_i \mathrm{d}A_j \tag{5}$$

There is a small number of analytical equations and diagramming displays for determining the view factors and this accounts only for a relatively simple configuration. Still, it is possible to divide configurations of more complex geometries into a certain number of simpler configurations in a way that their view factors can be determined from standard analytical equations or diagrams. Such a procedure is known as visual angle algebra.

In this paper, due to complexity of mentioned method, view factors were assessed numerically, based on model made in SOLIDWORKS Flow Simulation 2016. In the model each surface was divided into several sections with approximately uniform temperatures (max differences<0.5 °C) and same emission factor. Based on determined view factors and captured inner wall temperature distribution, the mean radiant temperature was calculated for 10 positions (same as air temperature), shown on Fig. 1.

The air velocity, in order to assess convective heat loss, was measured using hot-wire air speed probes with the measuring range of 0.08-0.15 m/s. The test room (Fig. 1) is a room inside a room with internal dimensions 4x4x3, as prescribed in [27]. Inner walls are cooled with air provided by an air conditioning unit and distributed through outlets on the outer surfaces.

2.2 Determination of emitter heat output

The emitter heat output Q_{em} (Eq. 6) is calculated using the water flow rate q_m and measured inlet and outlet temperatures t_1 and t_2 (measured wit T-type of thermocouples). The temperatures are used to calculate the specific enthalpies.

$$Q_{em} = q_m \cdot (h_1 - h_2) \tag{6}$$

The water flow rate is calculated (Eq. 7) using mass of the collected water m (in the measuring vessel) and relevant time interval τ, as described in [27].

$$q_m = \frac{m}{\tau} \tag{7}$$

The scheme of the test set-up is shown on Fig. 3.

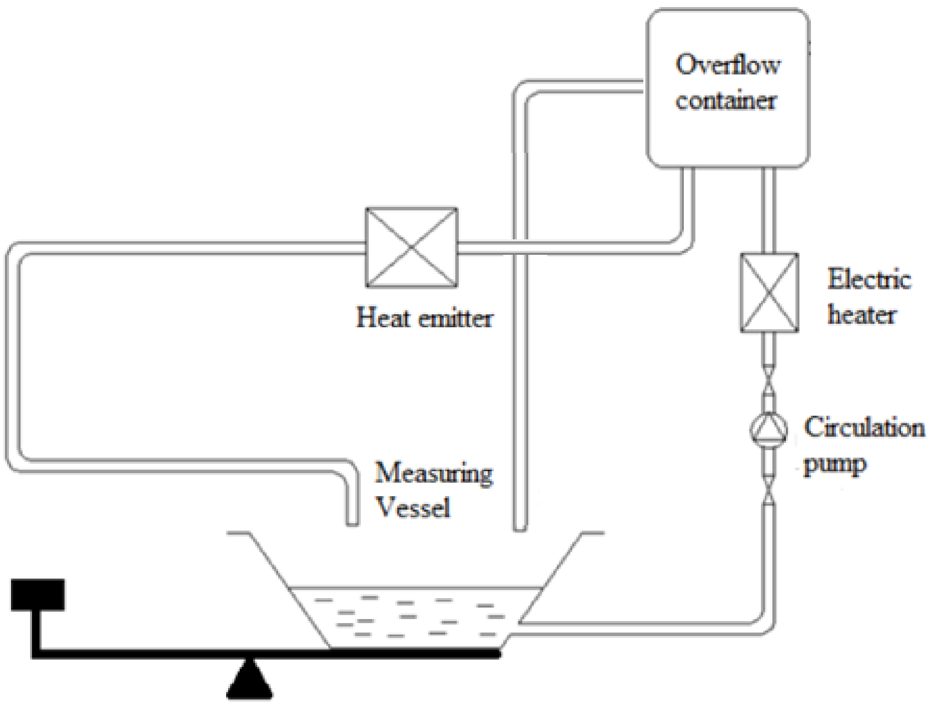

Fig. 3 Emitter heat output: Test set-up

3. Results and discussion

3.1 Tested heat emitters

In this paper two type of heat emitters (newly designed wall-mounted convector and conventional radiator) have been tested and obtained results were compared. Wall-mounted convector is made of two copper pipes (outer diameter 12 mm) connected together with brass ribs separated from one another by 8 mm. Tested radiator is a conventional aluminium radiator composed from ten 600/80 sections with the overall manufacture declared heat output of 1320 W (at 75/65/20 °C).

3.2 Heat output and indoor air temperature

Results of measured indoor air temperature and calculated emitter heat output are presented in Table 1. Comparison of indoor air temperature achieved at each position is given on Fig. 4.

Table 1 Emitter heat output and indoor air temperature

	Wall-mounted convector	Conventional radiator
Water flow rate, kg/h	43.55	124.58
Inlet temperature (t_1), °C	73.45	49.28
Outlet temperature (t_2), °C	61.19	44.92
Heat output, W	626.98	631.59
Excess temperature, °C	46.7	26.46
Average indoor air temperature, °C	20.62	20.64
Average indoor air temperature – low positions, °C	20.3	19.88
Average indoor air temperature – high positions, °C	20.94	21.4
Difference between average indoor air temperatures at high and low positions, °C	0.64	1.52
Maximum indoor air temperature, °C	21.11	21.63
Minimum indoor air temperature, °C	19.88	19.53
Difference between maximum and minimum indoor air temperature, °C	1.23	2.1

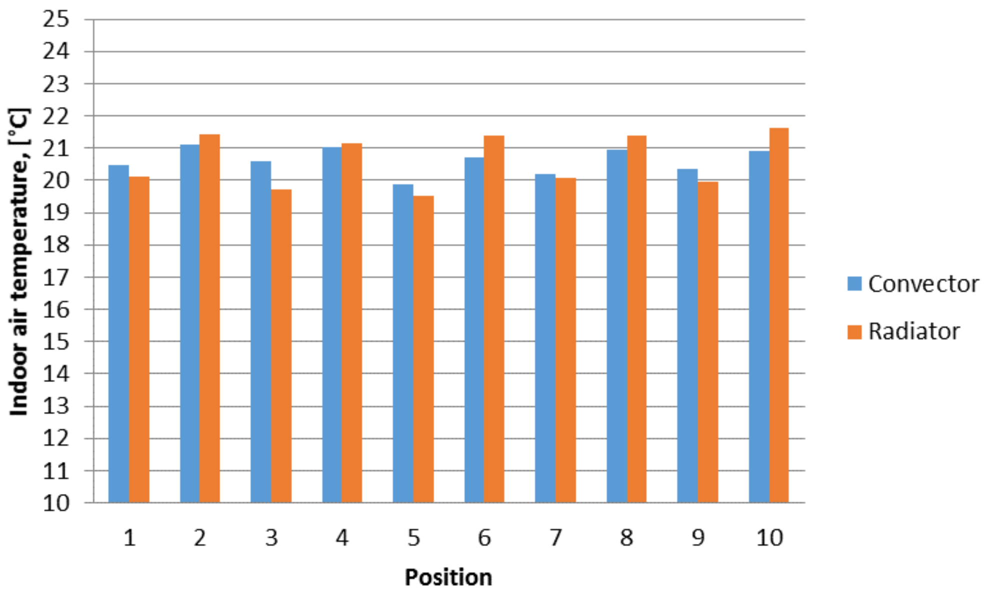

Fig. 4 Indoor air temperature at 10 positions

For measurements to be valid and comparison justified, approximately equal heating outputs from both heat emitters had to be achieved (achieved difference 0.7%). This results in achieving practically the same averaged indoor air temperature (difference below 0.1%). Since the convector has a considerably lower nominal heating power, it had to be operated at a higher temperature regime (excess temperature of 46.7 °C for convector and 26.46 °C for radiator). Results from Table 1 show that the average difference between the at higher and lower positions temperature sensors in case of the radiator is more than 1 °C, whereas in case of convector is 0.64 °C. It can be concluded that the convector, although operating on higher temperature regime, provides smaller temperature stratification which contributes to more uniform vertical temperature distribution and potentially better overall thermal comfort.

Maximum indoor air temperature was measured at position 10 (21.63 °C) in case of radiator and at position 2 (21.11 °C) in case of convector. Minimum indoor air temperature was measured in point 5 (19.88 °C for convector and 19.5 °C for radiator) in case of both heat emitters. From all these data it can be concluded that warm air lifting is more expressed in case of the radiator.

3.3 Mean radiant temperature

Results of measured wall temperature and computed mean radiant temperature is given in Table 2.

Table 2 Average wall and mean radiant temperature

	Wall-mounted convector	Conventional radiator
Heat output, W	626.98	631.59
Excess temperature, °C	46.7	26.46
Average wall temperature, °C	20.39	19.34
Average wall temperature – opposite to door entry, °C	21.91	18.92
Average wall temperature – left to door entry, °C	19.94	18.3
Average wall temperature – right to door entry, °C	20.3	21.04
Average wall temperature – door entry, °C	19.42	19.1
Average mean radiant temperature, °C	19.48	18.78
Average mean radiant temperature – low positions, °C	19.29	18.53
Average mean radiant temperature – high positions, °C	19.67	19.02
Difference between average mean radiant temperatures at high and low positions, °C	0.38	0.49

In case of the convector, a more uniform wall temperature distribution compared to radiator is noticed. This positively affects thermal comfort but on the other hand, due to average higher wall temperature, increases the transmission heat losses through vertical walls. Average mean radiant temperature is higher in case of the convector by 0.7 °C which only confirms the previous assertion that convector provides higher average temperatures of surrounding walls. Some representative thermograms are shown on Fig. 4-7.

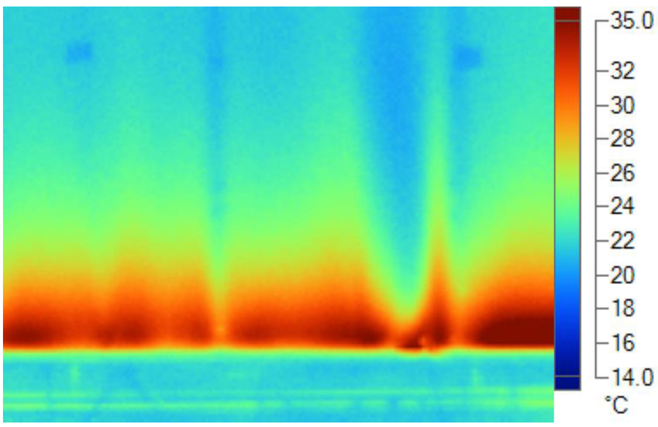

Fig. 5 Wall temperature distribution heated by convector (section of the wall opposite to door entry)

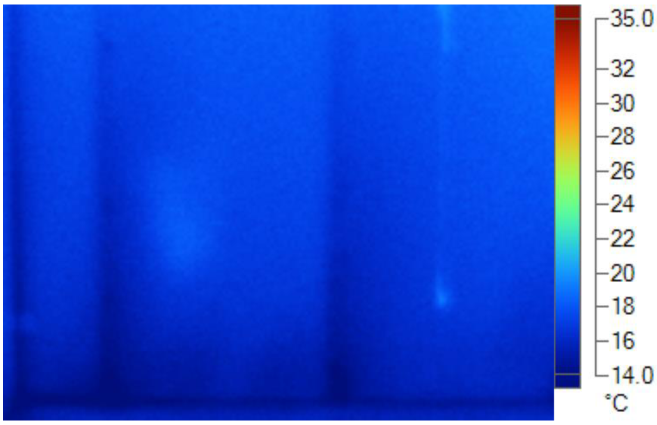

Fig. 6 Wall temperature distribution heated by convector (section of the wall left to door entry)

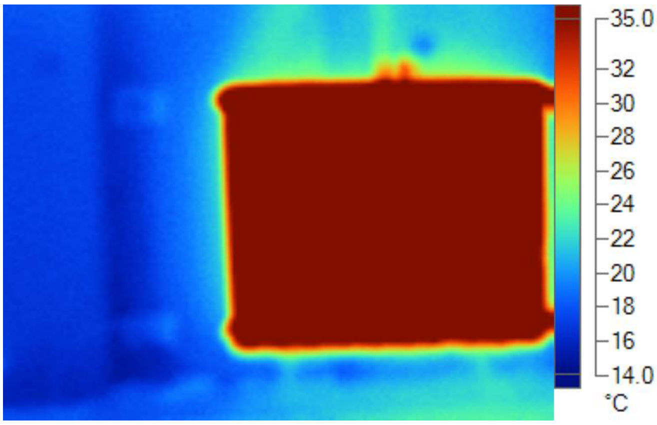

Fig. 7 Wall temperature distribution heated by radiator (section of the wall right to door entry)

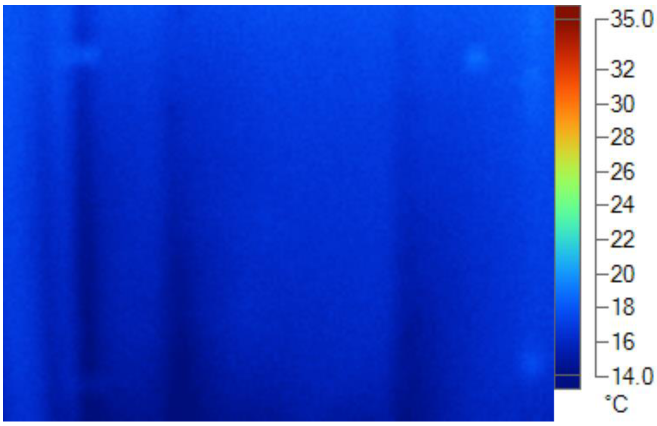

Fig. 8 Wall temperature distribution heated by radiator (section of the wall left to door entry)

3.4 Operative temperature

Since the absolute difference between mean radiant and indoor air temperature is less than 4 °C and measured air velocity was less than 0.15 m/s at each of the 10 positions, calculation of operative temperature was carried out using the equation 1. Results of computed operative temperature are given in Table 3. Comparison of the operative temperature achieved at each position is given on Fig. 7.

Table 3 Operative temperature

	Wall-mounted convector	Conventional radiator
Heat output, W	626.98	631.59
Excess temperature, °C	46.7	26.46
Average indoor air temperature, °C	20.62	20.64
Average mean radiant temperature, °C	19.48	18.78
Average operative temperature, °C	20.05	19.71
Average operative temperature – low positions, °C	19.79	19.2
Average operative temperature – high positions, °C	20.31	20.21
Difference between average operative at high and low positions, °C	0.51	1.01
Maximum operative temperature, °C	20.56	20.34
Minimum operative temperature, °C	19.33	18.87
Difference between maximum and minimum operative temperature, °C	1.23	1.47

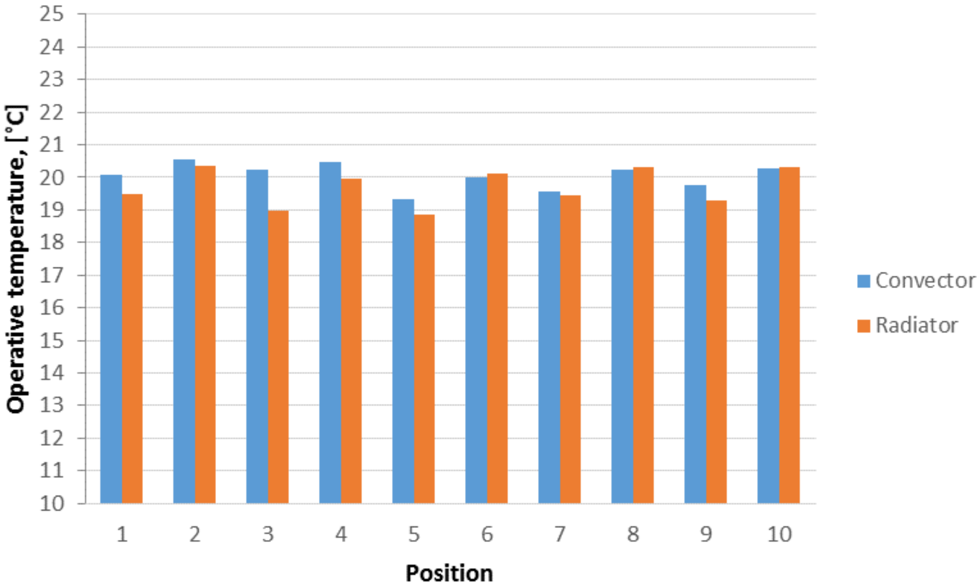

Fig. 9 Operative temperature at 10 positions

Results from Table 3 show that the average difference between average operative temperature at higher and lower positions in case of the radiator is 1.01 °C, whereas in case of convector is 0.51 °C. As it was presumed before, convector gives better overall thermal comfort due to more uniform operative temperature distribution and higher average operative temperature (by 0.34 °C). Maximum operative temperature was computed for position 2 (20.34°C) in case of radiator and at position 2 (20.56°C) and position 4 (20.47 °C) in case of convector as well. Minimal operative temperature was calculated for position 5 in case of both emitters.

4. Conclusion

In this paper, thermal comfort achieved by two types of heat emitters (newly designed wall-mounted convector and conventional radiator) was analysed in laboratory conditions, through the comparison of vertical temperature distribution and determined operative temperature.

It was concluded that the vertical indoor air temperature distribution is more uniform if heated by the convector despite the higher heating regime. The same rule applies for the operative temperature distribution as well. In same conditions (heat output and inner wall cooling rate) convector achieved 0.34 °C higher average operative temperature. In spite of the increased transmission heat losses (through walls) due to higher wall temperatures achieved by the convector, it is expected that the total room heat losses are greater in case of radiator. This is primarily consequence of pronounced temperature stratification effect (rising of warm air to the celling). Also, the obtained results indicates that ventilation heat losses can be lower when using convector as it is allow operation with lower air temperatures at fixed operative temperature i.e. same level of thermal comfort.

REFERENCES

[1] Pérez-Lombard, L, Ortiz, J, Pout, C. A review on buildings energy consumption information. Energy and Buildings 2008; 40: 394–398. https://doi.org/10.1016/j.enbuild.2007.03.007

[2] Hoyt, T, Arens, E, Zhang, H. Extending air temperature setpoints: simulated energy savings and design considerations for new and retrofit buildings. Building and Environment 2015; 88: 89–96. https://doi.org/10.1016/j.buildenv.2014.09.010

[3] EN ISO 7730:2005 - Ergonomics of the Thermal Environment – Analytical Determination and Interpretation of Thermal Comfort Using Calculation of the PMV and PPD Indices and Local Thermal Comfort Criteria. International Standard Organization, Geneva.

[4] Fanger, P. O. Thermal Comfort. Danish Technical Press, Copenhagen 1970

[5] ASHRAE, 2010. ANSI/ASHRAE Standard 55-2010. Thermal Environmental Conditions for Human Occupancy

[6] EN ISO 7726:2001 Ergonomics of the thermal environment. Instruments for measuring physical quantities

[7] Burt J.E Terjung W.H, O'Rourke P.A. View-factors leading to the simulation of human heat stress and radiant exchange. Theoretical and Applied Climatology 1982; 321–331. https://doi.org/10.1007/BF02324673

[8] Kalmar F, Kalmar T. Interrelation between mean radiant temperature and room geometry. Energy and Buildings 2012; 1-8.

[9] Huizenga C, Hui Z, Arens E. A model of human physiology and comfort for assessing complex thermal environments. Building and Environment 2001; 36: 691–699. https://doi.org/10.1016/S0360-1323(00)00061-5

[10] Thorsson, S., Lindberg, F., Eliasson, I., Holmer, B. Different methods for estimating the mean radiant temperature in an outdoor urban setting. International Journal of Climatology 2007; 27: 1983–1993. https://doi.org/10.1002/joc.1537

[11] Thorsson, S, Lindberg F, Björklund, J, Holmer B, Rayner D.P. Potential changes in outdoor thermal comfort conditions in Gothenburg, Sweden due to climate change: the influence of urban geometry. International Journal of Climatology 2011; 31: 324–335. https://doi.org/10.1002/joc.2231

[12] Bruse M, Fleer H. Simulating surface-plant-air interactions inside urban environments with a three dimensional numerical model. Environmental Modelling & Software 1998; 13: 373–384. https://doi.org/10.1016/S1364-8152(98)00042-5

[13] Matzarakis A, Rutz F, Mayer H, Modelling radiation fluxes in simple and complex environments: basics of the RayMan model. Inernational Journal of Biometeorology 2009; 54: 131–139. https://doi.org/10.1007/s00484-009-0261-0

[14] Lindberg F, Grimmond C.S.B, Nature of vegetation and building morphology characteristics across a city: influence on shadow patterns and mean radiant temperatures in London. Urban Ecosystem 2011; 14: 617–634. https://doi.org/10.1007/s11252-011-0184-5

[15] Mayer H, Holst J, Dostal P, Imbery F, Schindler D. Human thermal comfort in summer within an urban street canyon in Central Europe. Meteorologische Zeitschrift 2008; 17:241. https://doi.org/10.1127/0941-2948/2008/0285

[16] Kantor N, Unger J. The most problematic variable in the course of human biometeorological comfort assessment - the mean radiant temperature. Central European Journal of Geosciences 2011; 3: 90-100. https://doi.org/10.2478/s13533-011-0010-x

[17] Chen X. Using radiant temperature sensor to measure room thermal environment. Technical Papers of 1999 Annual Meeting of SHASE; 1999.

[18] Olesen BW, Fanger PO, Jemsen PB, Nielsen OJ. Comfort limits for man exposed to asymmetric thermal radiation. In: Proceedings of CIB Commission W45 (human requirements) symposium: thermal comfort and moderate heat stress. Building Research Station. London: HMSO; 1973. 133-48

[19] Maivel M, Kurnitski J, Lahesoo K. Vertical temperature gradient and operative temperature with three heating systems in a low-energy building. Sustainable Cities and Buildings 2015

[20] Myhren, J. A. Flow patterns and thermal comfort in a room with panel, floor and wall heating, Journal of Energy and Building 2008; 524-536

[21] Myhren, J. A. Comfort temperatures and operative temperatures in an office with different heating methods. Proceedings of Healthy Buildings Conference, Vol. 2: Indoor Climate 2006; 47-52

[22] Myhren, J. A. Energy savings and thermal comfort with ventilation-radiators – a dynamic heating and ventilation system. Proceedings of Clima 07, Well-Being Indoors 2007; 110

[23] Hesaraki A, Holmberg S. Energy performance of low temperature heating systems in five new built Swedish dwellings: A case study using simulations and on-site measurements, Building Environment 2013; 64: 85–93. https://doi.org/10.1016/j.buildenv.2013.02.009

[24] M. Maivel, J. Kurnitski, Radiator and floor heating operative temperature and temperature variation corrections for EN 15316-2 heat emission standard, Energy and Buildings, 2015. 204-214.

[25] Kalmár F. Summer operative temperatures in free running existing buildings with high glazed ratio of the facades. Journal of Building Engineering 2016; 1-7.

[26] Kalmár T, Kalmár F. Comfort and energy analysis of heating up. International Review of Applied Sciences and Engineering 2010; 1: 35–43. https://doi.org/10.1556/IRASE.1.2010.1-2.6

[27] EN 442-2:2014 Radiators and convectors – Part 2: Test methods and rating

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