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Experimental Study of Effects to Mixed Convection Under Real Boundary Conditions in Non-empty Room

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Abstract

Building sector uses about 40 % of all primary energy in European Union. This sector consequentially presents a great potential of improving energy efficiency, enhanced usage of renewable energy sources and reducing emission of CO_2. Beside other approaches for achieving these goals implementation o1 low-exergy systems for surface heating and cooling is very promising. The efficiency of those systems is mainly influenced by radiative and convective heat exchange on room surfaces. Methods for calculating radiative fluxes on room surfaces are well developed meanwhile empirical models for calculating convective heat exchange are mainly obtained from experimental and numerical studies in empty rooms, which are rarely empty in practice. Those empirical equations mostly do not consider real boundary conditions in practice which can effect to over or underestimation of convective heat fluxes on thermo active room surfaces.

We recognized lack of studies for mixed convection at real boundary conditions. Based on this recognition we provided a large experimental research program for mixed convection on vertical thermo active room surface in highly upgraded thermostatic chamber fully made in accordance with EN 442:1996. We were experimentally analyzing effects to mixed convection of the most common situation in real rooms which disturb temperature and velocity field near the vertical thermo active room surfaces. Thus we analyzed effects of vertical thermo active room surface's width, flow obstacles alternately on the floor and below of the ceiling and the working place which at the same time presents the flow obstacle and the additional internal heat source in the room.

Paper firstly presents theoretical background of the experimental study and main features of experimental setup. It continues with experimental results and further on it gives recommendations and limitations about using models for mixed convection on thermo active room surface in non-empty rooms.

Keywords – mixed convection; convective heat exchange, real boundary conditions; vertical thermo active room surface; low exergy heating and cooling

1. Introduction

Contemporary buildings are mainly all mechanically or hybrid ventilated. Air movement effects velocity and temperature fields at room surfaces including Thermo Active Room Surfaces (TARS). Consequently natural convection is changed into mixed convection where is dominant neither natural nor forced convection, but it is influenced by both of them at the same time [1]. Yang and Patel [2] were researching effect of buoyancy in two dimensional wall jet at a heated vertical wall. Angirasa [3] found that mixed convection of air jet which flows upward along heated vertical wall is equal to mixed convection gained by air jet flows downward along cooled vertical wall. Fisher and Pedersen [4] were made experimental research work for mixed convection along heated room surfaces in closed room with air exchange $3 \leq n \leq 12\ h^{-1}$. Cooled air has been supplying through ceiling radial diffuser. They used supply air temperature as reference temperature in their models for an average Convective Heat Transfer Coefficient (CHTC). Awbi and Hatton's [5] research was focused to mixed convection generated by two dimensional wall jet along heated room surfaces and they set equations for average CHTC on floor, ceiling and wall in form of blending equation suggested by Churchil and Usagi [6]: $(h_{mc})^C = (h_{nc})^C + (h_{fc})^C$, where they chose $C = 3.2$ in accordance with Neiswanger et al. [7] suggestion. Effects of displacement ventilation to convection along floor were studied by Novoselac et al. [8]. They also set equation of average CHTC in form of blending equation, when they used part for natural convection from study done by Awbi and Hatton [9]. Reference air temperature has to be measured 100 mm above the floor. Goldstein and Novoselac [10] were studying mixed convection downward heated vertical wall which was generated by two dimensional wall jet entering in to the room through ceiling slot diffusors. They set models for average CHTC when vertical TARS was either partially or fully thermally activated. Those two researchers also recognized that mixed convection is more intensive close to the diffusers, which is additionally confirmed in [11]. It means that mixed convection has to be studied locally [12, 13].

All presented studies undoubtedly show, that forced air moving inside closed rooms enhances CHTC with mechanism of mixed convection comparing CHTC for natural convection boundary conditions. From that reason it makes sense for using supply air under controlled conditions in mechanically or hybrid ventilated buildings for enhanced heat transfer along TARS. In that way we can in one hand avoid the problems of contemporary systems for surface heating and the cooling of the buildings, which mainly source from low natural convection CHTC. On another hand enhanced convective heat transfer can be established at lower surface to air temperature difference which additionally allows using more environmental renewable low exergy sources for heating and for cooling. Utilization of mixed convection for low exergy heating and cooling needs reliable models for CHTC for real rooms which are usually non-empty oppositely to presented studies. Because of lack of researches of mixed convection in non-empty (real) rooms we decided for experimental study of most common situations in real rooms which disturb temperature and velocity field along the vertical TARS, similar like we did in our previous experimental study [14] for natural convection in non-empty room.

2. Theoretical background

Each heated or cooled surface exchanges its heat/cold with the room with mechanisms of convection and radiation:

$$\mathrm{q}''_{\mathrm{TARS}}(\mathrm{x},\mathrm{y}) = \mathrm{q}''_{\mathrm{k}}(\mathrm{x},\mathrm{y}) + \mathrm{q}''_{\mathrm{ir}}(\mathrm{x},\mathrm{y}) \tag{1}$$

Local CHTC can be obtained with transformation of (1) and considering Newton's low of cooling ($\mathrm{q}''_{\mathrm{k}} = \mathrm{h}(\mathrm{x},\mathrm{y}) \cdot \Delta\vartheta$):

$$\mathrm{h}(\mathrm{x},\mathrm{y}) = \frac{\mathrm{q}''_{\mathrm{TARS}}(\mathrm{x},\mathrm{y}) - \mathrm{q}''_{\mathrm{ir}}(\mathrm{x},\mathrm{y})}{\Delta\vartheta} \tag{2}$$

thus an average CHTC on limited surface can be defined as:

$$\bar{\mathrm{h}} = \frac{1}{\mathrm{A}_{\mathrm{TARS}}} \int_{\mathrm{A}_{\mathrm{TARS}}} \mathrm{h}(\mathrm{x},\mathrm{y})\, \mathrm{dA}_{\mathrm{TARS}} \tag{3}$$

Equation (1) can be rewritten in given form, if two-dimensional velocity and temperature boundary layers exist along TARS:

$$\mathrm{h}(\mathrm{x}) = \frac{\mathrm{q}''_{\mathrm{TARS}}(\mathrm{x}) - \mathrm{q}''_{\mathrm{ir}}(\mathrm{x})}{\Delta\vartheta} = \frac{\mathrm{q}''_{\mathrm{k}}(\mathrm{x})}{\Delta\vartheta} \tag{4}$$

and (3) can be consequentially redefined:

$$\bar{\mathrm{h}} = \frac{1}{\mathrm{L}} \int_0^{\mathrm{L}} \mathrm{h}(\mathrm{x})\, \mathrm{dx} \tag{5}$$

Temperature difference in (2, 4) can be considered in different ways. The most common form for natural convection or for mixed convection with dominant component of natural convection is defined:

$$\Delta\vartheta = \Delta\vartheta_{\mathrm{r}} = \vartheta_{\mathrm{TARS}} - \vartheta_{\mathrm{r}} \tag{6}$$

Goethals et al. [15] suggest that reference air temperature ($\Delta\vartheta_{\mathrm{r}}$) is measured 1 m above the floor in the middle of the room. We applied their recommendation in our experimental study. For heating we were maintaining reference air temperature 22 °C and 24.5 °C for cooling respectively.

Temperature difference ($\Delta\vartheta$) for mixed convection is defined [16]:

$$\Delta\vartheta = \frac{2\Delta\vartheta_r \Delta\vartheta_{sa}}{\Delta\vartheta_r + \Delta\vartheta_{sa}} \tag{7}$$

where $\Delta\vartheta_{sa}$ presents temperature difference between TARS and supply air.

Surface temperature $\vartheta_{\mathrm{TARS}}$ has to be measured locally for Neumann boundary condition and just in one point on observed surface for Dirichlet boundary condition, which is defined as:

$$\frac{\partial\vartheta_{\mathrm{TARS}}}{\partial \mathrm{x}} = \frac{\partial\vartheta_{\mathrm{TARS}}}{\partial \mathrm{y}} = 0 \tag{8}$$

Although we used Dirichlet boundary condition, we measured surface temperature in the same locations as local surface heat fluxes densities $\mathrm{q}''_{\mathrm{TARS}}(\mathrm{x},\mathrm{y})$.

3. Experimental setup

We provided our experimental study of mixed convection in thermostatic chamber made fully in accordance with EN 442-2:1996 [17] and additionally highly upgraded to meet requirements for our research program (Fig. 1, Fig. 2). Convection was being analyzed on vertical TARS with dimensions 1.25 m × 2.51 m. Surface heat fluxes and surface temperatures were being measured by heat flux sensors with integrated temperature sensors (PT-100) bonded in three vertical columns on vertical TARS (Fig. 1). Space between sensors on surface of TARS was filled with polypropylene with equal thermal conductivity and thickness as heat flux sensors to meet Dirichlet boundary condition. Finally the whole surface of TARS was coated with a special coating with emissivity of only 0,143 [18] for minimizing radiative heat exchange between TARS and other surfaces in chamber. Existing radiation was anyway considered during data analyzing.

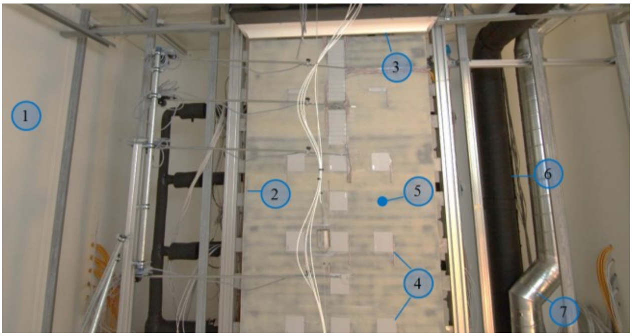

Fig. 1 Vertical TARS during installation of experimental setup into the thermostatic chamber: 1 – thermostatic chamber's wall, 2 – vertical TARS, 3 – slot diffusor, 4 – heat flux sensors with integrated temperature sensors, 5 – polypropylene, 6 – supply air duct, 7 – exhaust air duct

Fig. 2 Interior of thermostatic chamber with coordinate system

Vertical TARS split interior of thermostatic chamber in two compartments, lager one with dimensions 4 m × 3 m and height of 2.51 m presented small office room for one occupant. The two-dimensional wall jet which was generating mixed convection downward along vertical TARS during our research was being supplied in to the room through linear slot diffusor installed in the ceiling directly at the top of TARS.

The whole experimental setup is in all details showed and presented in one of our previous papers [19].

4. Results

We needed to found out at the beginning of our research program if we can consider temperature and velocity boundary conditions along the vertical TARS as two-dimensional and consequentially local surface heat flux and CHTC as one dimensional variables. For this purpose we bonded heat flux sensors on TARS in three columns with distance ±280 mm between them and separately established different combinations of boundary conditions (temperature differences, supply air velocities) for observing if any difference in CHTC exists at the same distance from the supply air diffusor and at different distances from vertical symmetry axis of TARS (z = - 280 mm, z = 0 mm, z = 280 mm). Results in form of local CHTC are shown in Fig. 3.

Presented results evidently show, that no significant differences in local CHTC exist along of TARS at different distances from the vertical symmetry axis of TARS. This founding allowed us to study mixed convection as one-dimensional variable in further analyzes of effects to mixed convection in real non-empty rooms.

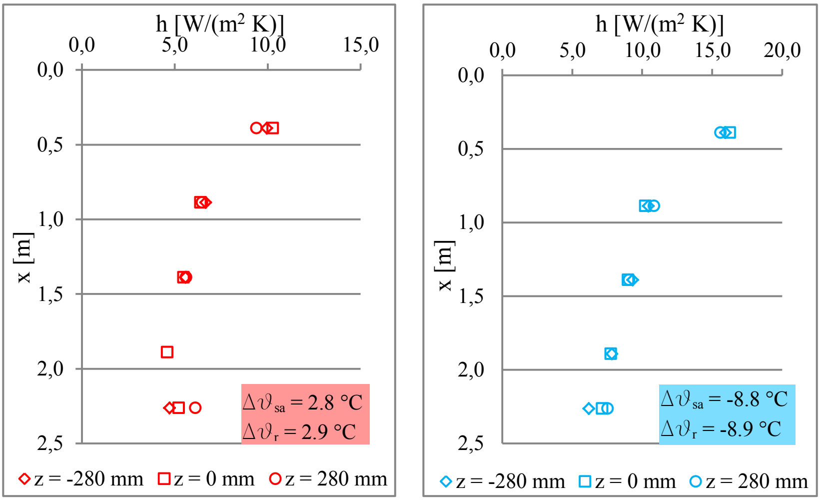

Fig. 3: Comparison of local CHTC for mixed convection at different widths on heated TARS at v_{sa} = 2 m/s (left) and on cooled TARS at v_{sa} = 4 m/s (right)

Room with the flow obstacles

Many different objects are usually in rooms which can act as flow obstacles for free air movement inside of the rooms. Convection generally strongly dependents on flows field which can be interrupted by flow obstacles. With purpose to find out any effects to mixed convection on vertical TARS as a consequence of disturbed temperature and velocity fields we make comparison of local convective heat fluxes for empty room and for the room with flow obstacle of height 0.62 m alternately mounted on the floor and then below of the ceiling in both cases 0.56 m away of the TARS as seen in Fig. 4.

Fig. 4: Flow obstacles parallel to TARS alternately on floor and below of the ceiling

Results for comparison are presented in Fig. 5 and Fig. 6. We can recognize difference between empty room and room with flow obstacle only for mixed convection on cooled TARS for flow obstacle on the floor at higher temperature differences and higher supply air velocity (Fig. 5). But difference of local CHTC in this case never exceeds 11,7 % comparing to local CHTC in empty room.

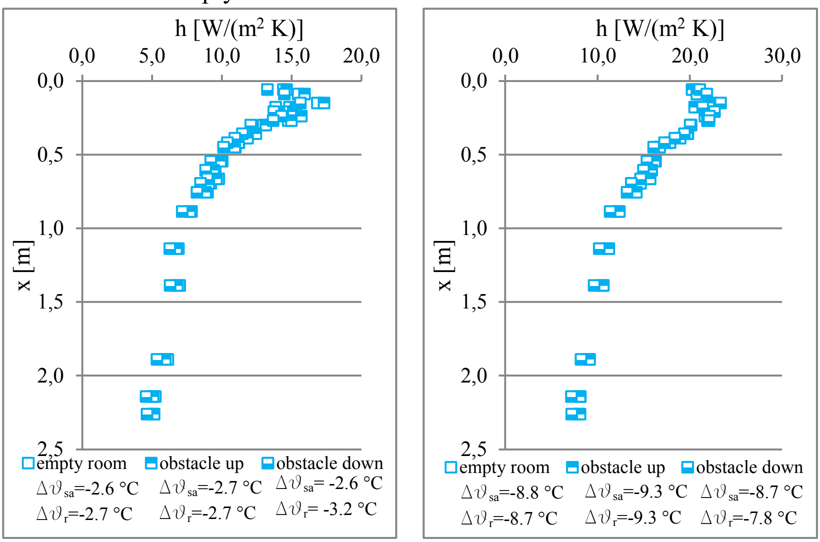

Fig. 5: Comparison of local CHTC for mixed convection on cooled TARS at $v_{sa} = 2$ m/s (left) and $v_{sa} = 4$ m/s (right) for empty room and for the room alternately with flow obstacle on floor and bellow of the ceiling

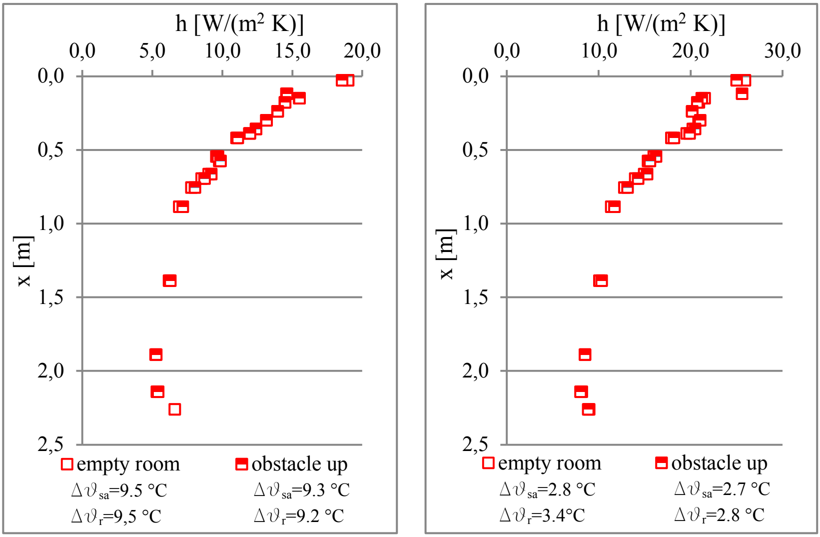

Fig. 6: Comparison of local CHTC for mixed convection on heated TARS at v_{sa} = 2 m/s (left) and v_{sa} = 4 m/s (right) for empty room and for the room with flow obstacle bellow of the ceiling

Room with a working place

Fig. 7 Room with a working place

For this part of research work a typical working place was placed in the middle of the room and it was simulated with a desk, heated dummy (123 W), computer with monitor (74 W), and fluorescent lamp (107 W) below of the ceiling (Fig. 7). We were unable to measure reference room temperature in the middle of the occupied zone because of such configuration and placement of working place. Then we were consequently maintaining the same TARS temperatures as for the empty room. Higher room air temperatures expectedly appeared in the room due to additional internal heat gains. It has recognizable effect to convective heat fluxes at small temperature differences (ϑ_r) and at low supply air velocities (Fig. 8) therefore at boundary conditions where component of natural convection is considerable. Influence of additional heat sources to mixed convection decreases at higher supply air velocities and therefor at stronger component of forced convection (Fig. 9).

In practice internal heat gains are considered during designing of cooling systems and consequentially they do not contribute to higher indoor temperatures, which was the reason for higher CHTC in our research at boundary conditions where the component of natural convection is important.

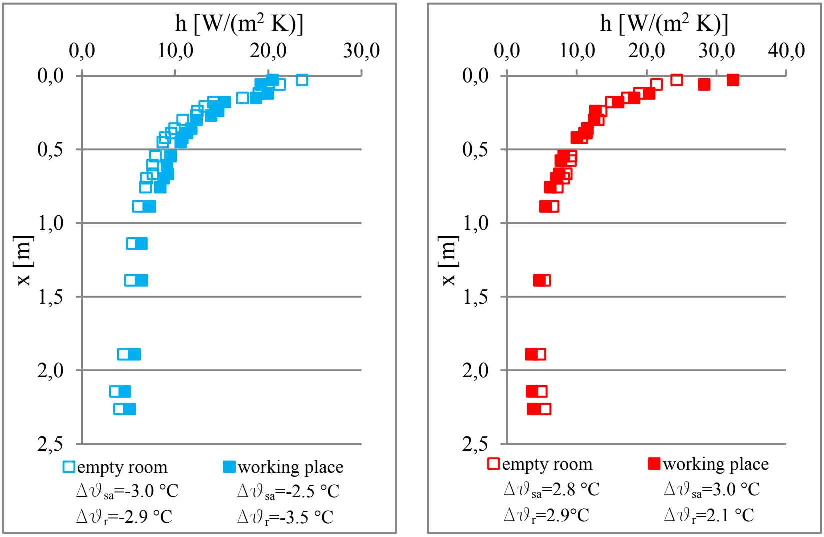

Fig. 8: Comparison of local CHTC for mixed convection at v_{sa} = 2 m/s on cooled (left) and on heated TARS (right) for empty room and for the room with the working place

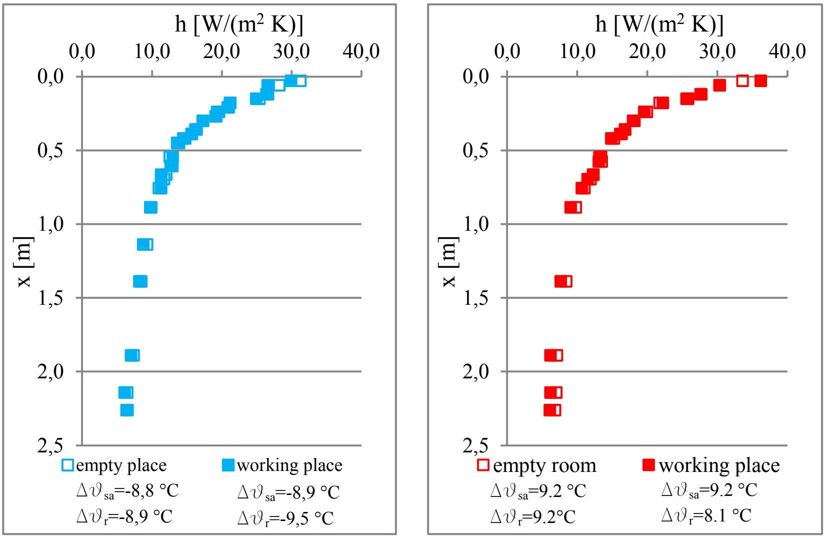

Fig. 9: Comparison of local CHTC for mixed convection at v_{sa} = 4 m/s on cooled (left) and on the heated TARS (right) for empty room and for the room with working place

5. Conclusions

Experimental research study about effects of flow obstacles and presence of working place to mixed convection on vertical TARS in real size room is presented in this paper. We can figure out based on experimental results presented in this paper that results and founding sourced from studies for mixed convection generated by two dimensional walls yet along vertical TARS made in empty room can also be used for non-empty rooms with flow obstacles and working places. Differences in local CHTC are noticeable in one hand only for boundary conditions where the natural convection component is either stronger or comparable to forced convection component. Combined systems for ventilation and surface heating and cooling suggested in this study can be advantageous in another hand to contemporary systems for surface heating and cooling which base on radiative and natural convection heat exchange if component of forced convection is dominant in mixed convection heat transfer.

Authors of this paper suggest additional experimental research work to evaluate other possible effects to mixed convection over vertical TARS, which can be e.g. due to different distance of supply air diffusor from the vertical TARS, variable thermo active height of TARS, position of exhaust air diffusor etc. One of the aims of all suggested researches of mixed convection in real rooms should be how to use as less as possible energy for ventilating, heating and cooling and utilization as much as possible environmental sources for these tasks.

Nomenclature

A	[m^2]	area
CHTC	[$W/(m^2 K)$]	convective heat transfer coefficient
h	[$W/(m^2 K)$]	convective heat transfer coefficient
$\overline{h}$	[$W/(m^2 K)$]	average convective heat transfer coefficient
L	[m]	characteristics length
n	[h^{-1}]	number of air exchanges in room
q"	[W/m^2]	heat flux density
ϑ	[°C]	temperature in Celsius
x	[m]	vertical distance from the ceiling
y	[m]	perpendicular distance away from TARS
z	[m]	lateral distance away from the vertical symmetry axis of TARS

C	exponent of blending equation
fc	forced convection
ir	infrared radiation
k	convection
mc	mixed convection
nc	natural convection
r	room
sa	supply air
TARS	thermo active room surfaces

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