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ASSESSMENT OF THERMAL COMFORT IN A LECTURE HALL WITH THE APPLICATION OF INSTRUMENTS FOR COMPUTATIONAL FLUID DYNAMICS

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Abstract

The examinations presented in this article aim at illustrating some possibilities of applying instruments for computational fluid dynamics to assess thermal comfort on the example of a lecture hall in the building of the Institute of Building Engineering at the University of Warmia and Mazury in Olsztyn. The obtained results have been subjected to an analysis according to the guidelines provided in the PN-EN ISO 7730:2006 norm. For the air heating system used in the hall, the distributions of velocity, temperature, as well as the PMV and PDD indices have been subjected to an analysis, with a particular focus on the optimal temperature of the air inflow.

Introduction

Thermal comfort is defined as state of satisfaction with thermal conditions of the surroundings (FANGER 1974). The main assumptions to ensure the conditions of thermal comfort are:

– a will to satisfy the need to feel content with the present thermal conditions;

– obtaining optimal conditions for proper efficiency of work.

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The subject literature distinguishes three different approaches to thermal comfort assessment (ŚLIWOWSKI 2000):

– physical – viewing the human organism in a way similar to a heat-generating and working machine;

– psychological – where the thermal comfort assessment is based on responses provided by people in questionnaires and tests;

– physiological – applying the research methods based on direct or indirect human calorimetry. Those are not very useful in thermo-neutral conditions and in small changes of metabolism, which are difficult to measure.

When assessing thermal comfort, various scale types may be applied, e.g. the Bedford scale, the ASHRAE thermal sensation scale, the Predicted Mean Vote (PMV), the assessment of subjective thermal sensation, or the assessment of local thermal discomfort (CHLUDZIŃSKA 2010).

Such a multitude of systems for thermal comfort assessment and various approaches to the problem reported in the subject literature create a need to systematize the methodology of classification and assessment for thermal conditions. To do so, some norms may be referred to and a particular attention should be paid to the PN-EN ISO 7730:2006 norm: *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*, as well as to the PN-EN 15251:2012 norm: *Indoor Environmental Input Parameters for Design and Assessment of Energy Performance of Buildings Addressing Indoor Air Quality, Thermal Environment, Lighting and Acoustics*.

This article is an attempt to assess the thermal comfort of a lecture hall in the building of the Institute of Building Engineering at the University of Warmia and Mazury, located at ul. Heweliusza 4 in Olsztyn, based on simulations performed with the use of the commercial FloVENT package that applies computational fluid dynamics (CFD). The examinations have been carried out for a design study and are an example of the correctness and efficiency assessment for the proposed conceptual solutions.

The choice Flovent package was due to the fact, that it is a software especially dedicated to the engineering issues involved in the design and optimization of heating, ventilating and air conditioning (HVAC) systems. FloVENT predicts 3D airflow, heat transfer, contamination distribution and comfort indices in and around buildings. All those parameters can be analyze in both forced and free convection ventilated systems under laminar or turbulent flow conditions. FloVENT solvers are based on Finite Volume method. (FloVent User Guide, Software Version 10.1).

Many researchers have studied the human comfort in indoor environments (RUPP et al. 2015). Different thermal comfort approaches can be found in literature: the rational (heat-balance approach) or adaptive approach (DJONGYANG et al. 2010). The impact of different factors for thermal comfort sensation can be analysed as well (FRONTCZAK, WARGOCKI 2011). The problem of human thermal comfort in the built environment becomes increasingly crucial in the face of extending time of human residing in buildings. Experimental investigation of that issue including not only research in climate chambers or in semi-controlled environments but also in the real buildings. During last 50 years CFD has been increasingly used for room air distribution and thermal comfort subject (NIELSEN 2015). Publication dedicated to the standard office room can be given as an example of numerical studies (YONGSON et al. 2007). The k-epsilon and Reynolds stress models for turbulence flow were used for the analysis of temperature and velocity distribution. What is more, to achieve the maximum comfort for the occupant, the different locations of the air conditioner blower were analyzed. For an office room, the verification of a numerical model for airflow and heat transfer were made (STAMOU, KATSIRIS 2006). The experimental results were compared with the results of the standard k-ε, the RNG k-e model and the laminar model. As a main conclusion authors has proved, that all the tested turbulent models predict correctly the main qualitative features of the air movement and the temperature distribution. Numerical and experimental investigation related to large space building can be found in many articles; case of patient hall of a hospital building (WANG et al. 2014), theatre (KAVGIC et al. 2008) or lecture hall (MUHIELDEEN et al. 2015).

Classification of the Internal Climate Quality

The PN-EN ISO 7730:2006 norm provides recommended parameters for thermal comfort together with a division into categories of thermal environment:

– Category A – characterized by a high level of the expected microclimatic conditions and the highest percentage of satisfied people;

– Category B – characterizes objects of an average quality level of the expected microclimatic conditions and an average percentage of satisfied people;

– Category C – objects of moderate, yet acceptable, quality level of the expected microclimatic for a minimal number of people.

The PN-EN 15251: 2012 norm provides an analogical division applying the category marking: I, II, and III. It also introduces category IV including objects that do not fall into any of the former three categories. That category should be accepted only for a part of a year. The objects in Category IV are characterized

Table 1

Categories of thermal comfort according to PN-EN ISO 7730:2006:
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

Category	Thermal Neutrality		Local Discomfort			
	PPD [%]	PMV	DR [%]	PD [%] warm or cold floors	PD [%] vertical air temperature	PD [%] asymmetric thermal radiation
A	<6	-0.2 < PMV < +0.2	< 10	<3	<10	<5
B	<10	-0.5 < PMV < +0.5	<20	<5	<10	<5
C	<15	-0.7 < PMV < +0.7	<30	<10	<15	<10

by the values PPD > 15, PMV < -0.7, or PMV > +0.7. The provided PMV (Predicted Mean Vote) index makes it possible to predict thermal sensations for any combinations of energy expenditure, clothing, and four thermal environmental parameters (air temperature, mean radiant temperature, air velocity, and air humidity) (FANGER 1974). The assessment of thermal sensation is performed with the application of a seven-degree scale, with the following values: +3, +2, +1, 0, -1, -2, and -3 where the notes mirror the sensations respectively: hot, warm, slightly warm, neutral, slightly cool, cool, and cold.

The PMV index is calculated according to formula 1 (PN-EN ISO 7730:2006):

$$\left\{\begin{array}{c} \mathrm{PMV} = (0{,}303 \cdot e^{-0{,}036 \cdot M} + 0{,}028) \cdot \\ (M-W) - 3{,}05 \cdot 10^{-3} \cdot [5733 - 6{,}99 \cdot (M-W) - p_a] - 0{,}42 \cdot [M-W) - 58{,}15 \\ -1{,}7 \cdot 10^{-5} \cdot M \cdot (5{,}867 - p_a) - 0{,}0014 \cdot M \cdot (35 - t_a) \\ -3{,}96 \cdot 10^{-8} \cdot f_{cl} \cdot [(t_{cl} + 273)^4 - (\bar{t}_r + 273)^4] - f_{cl} \cdot h_c \cdot (t_{cl} - t_a) \end{array}\right\} \quad (1)$$

where, when considering the recommended interval of values:

M – metabolic rate, $\left[\frac{\mathrm{W}}{\mathrm{m}^2}\right] = 0{,}8 \div 4{,}0$ [met],

W – external work, $\left[\frac{\mathrm{W}}{\mathrm{m}^2}\right]$,

I_{cl} – thermal insulation of the clothing $I_{cl} = 0 \div 0{,}310 \left[\frac{\mathrm{m}^2 \cdot \mathrm{K}}{\mathrm{W}}\right] = 0{,}0 \div 2{,}0$ [clo]

f_{cl} – ratio of the surface area of the clothed body to the surface area of the nude body,

t_a – air temperature $t_a = 10 \div 30$ [°C],

$\bar{t}_r$ – mean radiant temperature $\bar{t}_r = 10 \div 40$ [°C],

v_{ar} – relative air velocity $v_{ar} = 0 \div 1 \left[\frac{\mathrm{m}}{\mathrm{s}}\right]$,
p_a – partial water vapour pressure $p_a = 0 \div 2700$ [Pa],
h_c – convective heat transfer coefficient $\left[\frac{\mathrm{W}}{\mathrm{m}^2 \cdot \mathrm{K}}\right]$,
t_{cl} – clothing surface temperature [°C].

The above index identifies the predicted mean assessment of thermal comfort by a large group of people who have been influenced by the defined combination of changeable parameters of their environment. The PMV provides information on a general degree of discomfort for the whole studied group; however, the thermal sensations of people subjected to the same changeable factors in such a group are varied. Thus, the PPD (Predicted Percentage Dissatisfied) index may be applied as an assessment tool as it determines a share of people who are not satisfied with the environmental conditions. Participants who have given the notes: -2 (cool), -3 (cold), +2 (warm), or +3 (hot) are considered to be dissatisfied. The PPD index is calculated according to formula 2 (PN-EN ISO 7730:2006):

$$\mathrm{PPD} = 100 - 95 \cdot e^{-(0{,}03353 \cdot \mathrm{PMV}^4 - 0{,}2179 \cdot \mathrm{PMV}^2)} \tag{2}$$

The PPD index may be presented as a function of the PMV parameter (Fig. 1). The curve is characterized by its symmetry to the value of PMV = 0. The curve minimum is located on the level of 5% of the dissatisfied at the mean environmental assessment equal to zero.

The draught rating (DR) value may also be used as an assessment parameter for it presents a percentage of people dissatisfied with the velocity of air passing around the body and is calculated according to dependency 3 (PN-EN ISO 7730:2006):

$$\mathrm{DR} = (34 - t_{a,l}) \cdot (\bar{v}_{a,l} - 0{,}05)^{0{,}62} \cdot (0{,}37 \cdot \bar{v}_{a,l} \cdot Tu + 3{,}14) \tag{3}$$

where:
$t_{a,l}$ – air temperature $t_a = 120 \div 26$ [°C],
$\bar{v}_{a,l}$ – air velocity, if $\bar{v}_{a,l} < 0{,}05 \left[\frac{\mathrm{m}}{\mathrm{s}}\right]$, it is accepted that $\bar{v}_{a,l} = 0{,}05 \left[\frac{\mathrm{m}}{\mathrm{s}}\right]$,
Tu – turbulence intensity $Tu = 10\% \div 60\%$.

If the final value of DR > 100% is obtained, the value of DR = 100% is accepted.

The norm (PN-EN ISO 7730:2006) provides also the PD (percent of dissatisfied) index as an assessing parameter as it reflects a percentage of people dissatisfied with the difference between the temperatures at the levels of their feet and their heads. It is calculated according to the formula:

$$PD = \frac{100}{1 + e^{(5,76 - 0,856 \cdot \Delta t_{a,v})}} \tag{4}$$

The above formula should be applied if the $\Delta t_{a,v}$ value of a thermal difference between the levels of the feet and the head does not exceed 8°C.

The Analysed Object

A computational simulation based on the assumptions of the computational fluid dynamics (CFD) has been carried out for a lecture hall in the building of the Institute of Building Engineering at the University of Warmia and Mazury in Olsztyn, situated at ul. Heweliusza 4. The study has been performed with the use of the commercial FloVENT application by Mentor Graphics.

The floor space of the hall is 239.9 m^2 and it capable of holding 224 people. The room is designed to contain mechanical air conditioning with cooling that is aimed at:

– providing a required amount of hygienic air for people;
– leading out excessive thermal gains from people in summertime;
– air heating.

The computational simulations have been carried out for the situation reflecting the assumptions of the design for wintertime (external temperature according to PN-EN 12831 is -22°C) and they aimed at verifying the parameters of thermal comfort (operative temperature, mean air velocity, PMV and PPD indices) and assessing the efficiency of the air heating system. The main focus was to assess the state of thermal organism as a whole.

According to the design assumptions, the amount of the air inflow and outflow is the same and is 9,000 m^3/h. The design assumes that in order to maintain the required internal temperature at 20°C in the empty room, the air inflow into the room in wintertime should be at the temperature of 23°C. When using the lecture hall, the temperature of the air inflow is to change according to the number of people in the room and the temperature of the external air.

The outlet in the room has been designed to use of seventy-two KNP-al.-P grids, sized 255×125, located under the students' chairs in the central part of the hall. The air inflow takes place in the frontal part of the hall, over the lecturer's podium (using three symmetrically placed NWP 355 rotational-centrifugal air inlets, the efficiency of each: 600 m^3/h) and in ten DUK-V 400 nozzles, the efficiency of each: 720 m^3/h, located alongside the external rear wall with windows.

The computational model of the described room has been presented in Figure 1. The inlets and outlets have been marked red. The construction of the acoustic ceiling has been marked light yellow.

Under the lecture hall, there are technical rooms with the temperature of 15^oC. The internal wall is made of a partition situated behind the lecturer's back and, partially (about a half of the length), by a vertical wall to the right from the speaking person. The temperature on the other side of those partitions is 20^oC. The remaining partitions are external. The values of heat transfer coefficients for particular constructional partitions have been accepted according to the architectonical project (*Projekt budowlany budynku Wydziału Nauk Technicznych UWM w Olsztynie*, 2008). At the back of the hall, there is technical space for multimedia services in the lecture hall as well as one external wall, in which windows are mounted.

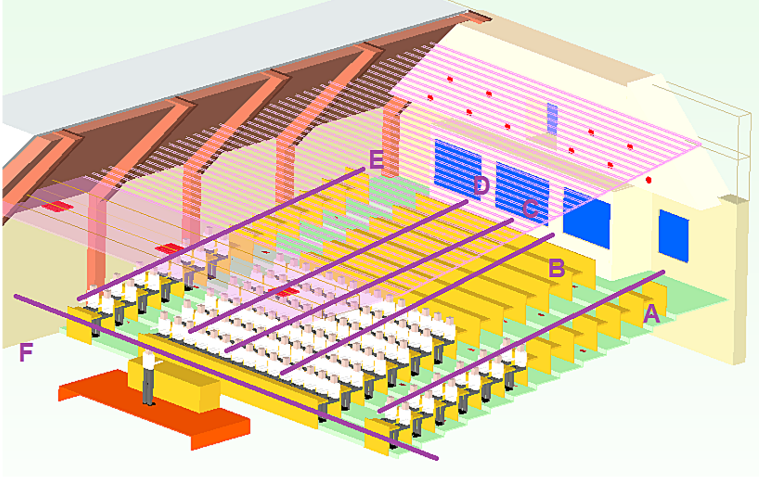

Fig. 1. Computational model of the analysed room – the half-full room case

The computational studies included two cases: when the hall is completely full and when it is filled in 50%. There is a plenty of possibilities, in which 112 people can take seats in the examined area. Because of technical CFD method limitation (steady state solution) and time needed to complete one simulation, the most popular case, when people are seating in the first rows, was analysed. Each part of the mannequin body in the model was the source of heat. Total heat amount is equal for each person 119 W. Due to the number of people in the room, the temperature of the air inflow has been changing, however, its amount has not. A grid with number of 6,860,000 cells has been used in both cases. The k-ε turbulence model has been applied to the calculations.

Although, the research are prepared in 3D, the results for this article are presented in the form of two-dimensional planes with specified location or for more precise analysis of numerical values, in a six measure lines (Fig. 1). The first five measure lines are situated in the audience area. They start at the location of the lowest seats for students in the hall and end at the backrests of the last seats, located at the highest position. The parameters have been analysed for the height of about 1.5 m above the floor. When looking with the direction of the lecturer's sight, line *A* is located in the mid-width a two-person row of seats situated on the right from the aisle. Analogically, line *E* is in the mid-width of a two-person row of seat situated on the left (along the external wall). Measure line C is located in the axis of symmetry of the hall, in the mid-width of a 14-person row of seats. Measuring lines *B* and *D* are placed symmetrically, to the left and to the right to line *C*, in the distance of about 2 m from it (in 1/4 and in 3/4 of the width of the central 14-person row of seats). Measure line *F* is located across the lecture hall (in the central part of the lecturer's podium), at the height of about 1.5 m from it.

Results of the Simulations

The first of basic questions to arise when analysing the results is the relevance of the conditioned air amount accepted in the assumptions of the design. The PN-EN 15251:200 norm defines the way to identify a conditioned airflow according to formula 5:

$$q_{tot} = n \cdot q_p + A \cdot q_B \tag{5}$$

where:
n – number of people in the room,
q_p – airflow per person $\left[\frac{1}{\mathrm{s}}, \text{person}\right]$,
A – room surface [m^2],
q_B – airflow for building emissions pollutions $\left[\frac{1}{\mathrm{s}}, \mathrm{m}^2\right]$.

When assuming that the room is characterized by a low level of pollution, the conditioned airflow must be no less than 2,480 $\frac{1}{\mathrm{s}}$ to comply with the requirements of Category I. With the assumed amount of 9,000 m^3/h, the above requirement is met.

According to the PN-EN ISO 7730:2006 norm, the designing criteria for the analysis of temperature distribution describe the optimal operative temperature as 22°C for the heating season. For objects in Category *A*, the accepted deviation is ±1°C, for objects in Category *B* it is ±2°C, and for object in Category *C*, the temperature deviation may be ±3°C. The FloVENT application makes it possible to separately analyse the environment temperature and the mean radiant temperature in the room.

The results presented in Figure 2 allow for a clear observation of a decrease in the environment temperature (*a*) and the mean radiant temperature (*b*) along the internal partitions (yellow and green areas). The walls of the technical space at the end of the lecture hall are marked grey. Presented in the figure is also a fragment of the external area (temperature: -22°C, navy blue) alongside the external wall with windows.

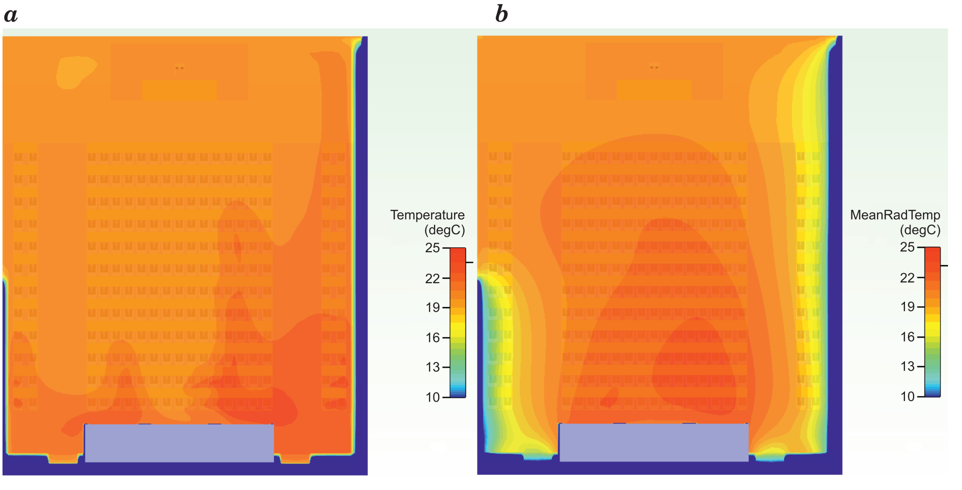

Fig. 2. The environment temperature (*a*) and the mean radiant temperature (*b*) – the results for the room filled with students in 100%, bird's eye view, distribution in a plane about 1.5 m over the floor for the highest-located seats

The operative temperature is what humans experience thermally in a space. The operative temperature is defined as a uniform temperature of a radiantly black enclosure in which an occupant would exchange the same amount of heat by radiation and convection as in the actual non uniform environment. It's expressed as the dependency between the environment temperature (t_a) and the mean radiant temperature (t_r), has been calculated for the room according to guidelines of the PN-EN ISO 7726 norm: *Ergonomics of the Thermal Environment – Instruments for Measuring Physical Quantities* based on the formula 6:

$$t_o = \frac{h_c t_a + h_r t_r}{h_c + h_r} \tag{6}$$

where:

h_r – radiative heat transfer coefficient – 4.6 $\left[\frac{\mathrm{W}}{\mathrm{m^2K}}\right]$,

h_c – convective heat transfer coefficient – 50.0 $\left[\frac{\mathrm{W}}{\mathrm{m^2K}}\right]$.

An analysis of the operative temperature distribution for the fully filled room (Fig. 3) has shown that its mean value is 21.38°C. This makes it very close to the normative optimal value. A closer look at the local distribution of the parameter makes it possible to identify areas with the largest diversification. As for measure lines *B*, *C*, and *D*, in the central part of the audience, the criteria defined in Category *B* of the norm are met. Line *D* is located in the area characterized by the highest operative temperature values that locally are coming close to almost 24°C. It may also be noticed that in the frontal area of the lecture hall, near the lecturer's podium, a lower operative temperature occurs – reaching about 20°C (measuring line *F*). This results from the fact that this area is used by a lecturer only; hence thermal gains from humans are lower. A negative influence of the low mean radiant temperature, resulting from the closeness of the external partitions, is visible in the area that is wholly covered by measure line *E*, partially (for the distance below 5.5 m a wall separates the hall from a heated corridor) by the measure line *A*, and in the section of about 3 m by the measure line *F*. The lowest recorded value of the operative temperature is 19.09°C.

For the computational case in which a partial load of the lecture hall has been analysed (students seated in the front rows – the distance below 5.5 m), the mean value of the operative temperature is 21.42°C (Fig. 4). Similarly as in the case of Figure 3, a negative influence of the neighbouring external partitions is visible (measure lines *A*, *E*, and *F*). It is even more visible and falls down to the minimal value of 18.84°C. What is noteworthy is a very visible influence of thermal gains from people in the central part of the hall (measure lines *B*, *C*, and *D*). The increase to about 24°C in the zone of the students seated in places that are the most far away from the lecturer's podium is also noticeable.

Obtaining the presented level of temperature distribution requires an inflow of conditioned air at the temperature of 18.6°C for the maximum number of students in the lecture hall and at the temperature of 21.0°C for the half-full room.

As for the maximum mean airflow velocity, the PN-EN ISO 7730:2006 norm indicates that the value should not exceed 0.10 m/s for rooms in

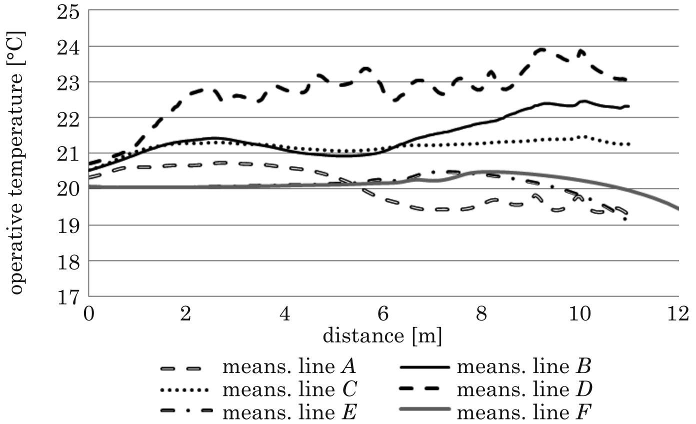

Fig. 3. The distribution of the operative temperature in the analysed measure lines – the full room case

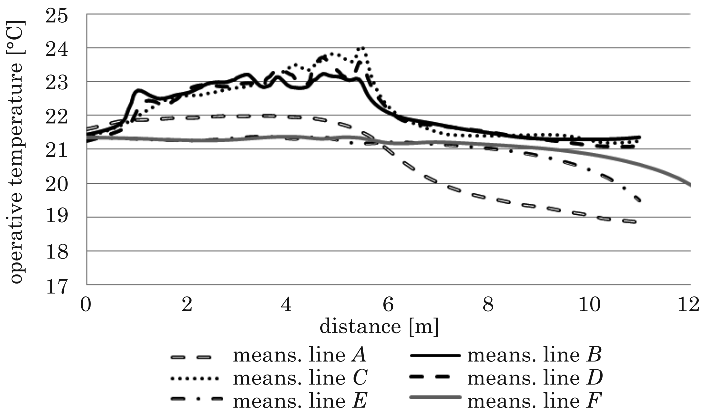

Fig. 4. The distribution of the operative temperature in the analysed measure lines – the half-full room case

Category *A*, 0.16 m/s for Category *B*, and the threshold value of 0.21 m/s is accepted for Category *C*.

An analysis of the results for the case in which the room is filled in 100% (Fig. 5) reveals that maximum mean airflow velocities significantly exceed the normative level of values for rooms in Category *C*. Locally, the recorded maximum velocity value is 0.46 m/s and in a huge majority of the results its level is above 0.21 m/s. Measure line *A* is an exception here, as its recorded maximum velocity value is 0.17 m/s.

When considering the design case in which the room is filled in 50% (Fig. 6), it is visible that the distribution of velocities in the studied area where measure line *A* is located is analogical to the case in which the room is filled in 100%.

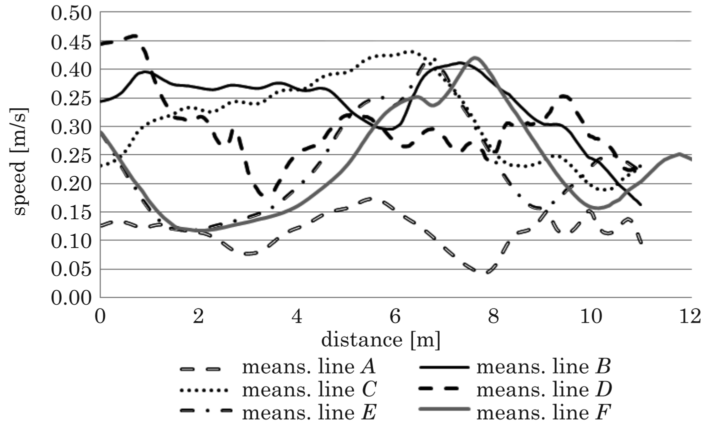

Fig. 5. The distribution of the mean airflow velocity in the analysed measure lines – the full room case

In the front rows of the audience, much higher (about two times) values of the mean airflow velocity occur than in the part of the hall that is not filled with people (the distance no higher than 5.5 m for lines *B*÷*E*). This poses the question whether in the proposed system of conditioned air division (the air outflow by grids mounted in the steps under the students' chairs) those are the location of chair seats, their opened worktops, and students sitting on the chairs that play key roles in the way the air flows.

Figure 7 reflects the results of the distribution for the mean airflow velocity when there are no people in the hall and the seats and worktops are closed. A maximal limitation of elements that may disturb the airflow results in a significant decrease of velocity. As for the case of extreme measure lines (*A* and *E*) in the audience area, as well as for the results in the frontal area of the hall where the lecturer's podium is placed (measure line *F*), the mean velocity value is 0.09 m/s. Locally, in the first row, included into measuring line *A*, some maximum velocity values of 0.18 m/s have been recorded. Measure lines *B*, *C*, and *D*, which run along the central area of the audience, are of totally different characteristics. In the distance of about 3.5 m (the third row), the maximum values of the mean airflow velocity have been reached at the level of about 0.27 m/s. There is a decrease in the velocity visible for the higher rows, down to the levels of about 0.02 m/s near the last rows at the technical space of the lecture hall.

An analysis of the PMV and PPD indices has been performed with the assumption that the metabolic rate of the people in the hall is 70 W/m^2 and the value of thermal insulation for daily wear clothing is 0.110 (m^{2}K)/W (PN-EN ISO 7730:2006). Figure 8, presented below, reflects the distribution of the PMV parameter in the central area of the hall, in a vertical plane.

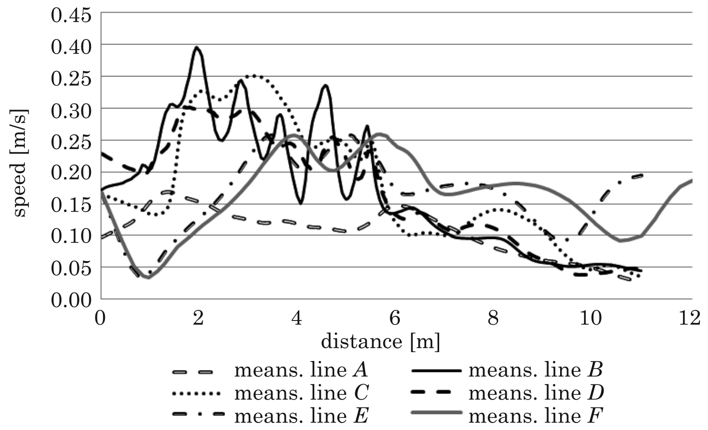

Fig. 6. The distribution of the mean airflow velocity in the analysed measure lines – the half-full room case

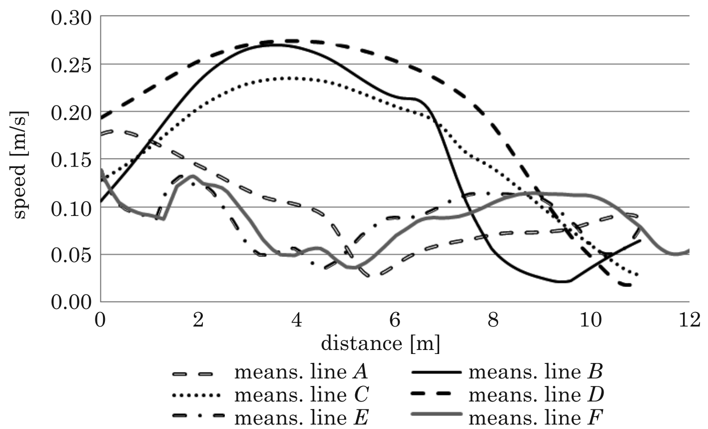

Fig. 7. The distribution of the mean airflow velocity in the analysed measure lines – the case with no people in the room

In the case of the maximal load of the hall, the mean value of the PMV indices is -0.43 while in the case of loading the room with students partially, the mean PMV value is -0.26. There is a clearly visible trend to decrease the value of the parameter in the areas characterized by the lowest values of the mean radiant temperature. As for loading the room with students completely, the minimum PMV value is -0.87 and it has been recorded at the edge of measure line *E*. As for the case of seats in the hall taken in half, the minimum value of the PMV indices is -0.75 and it is also located by the lowest row in the area of line *E*. The highest values of the indices (for both cases) have been noticed in the central area of the hall, by the highest rows that are character-

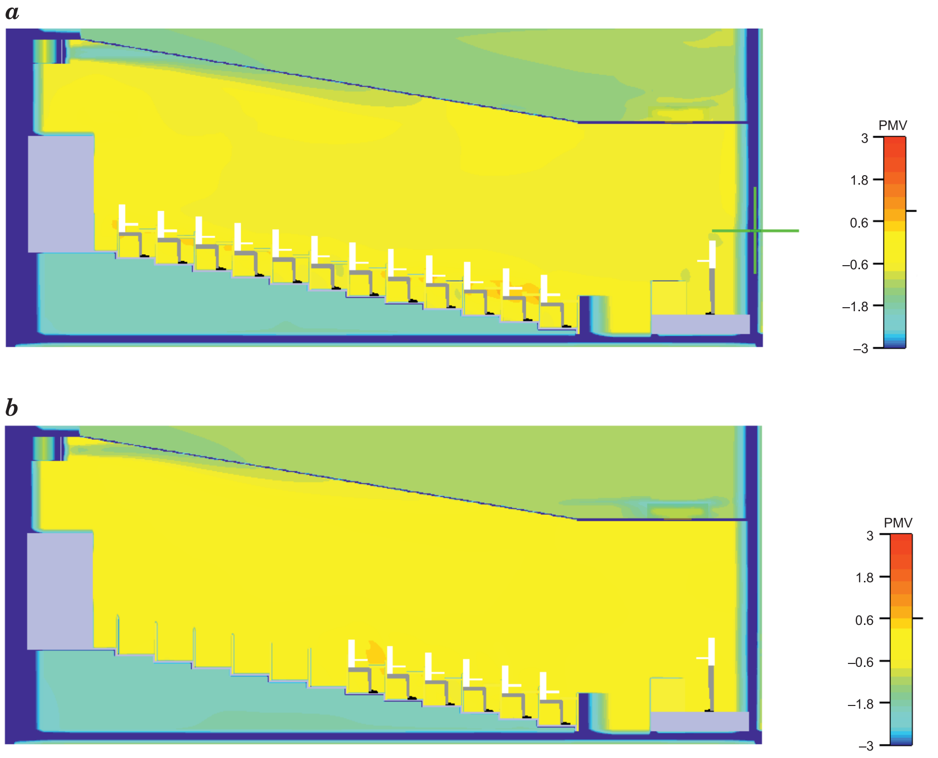

Fig. 8. The value of the PMV index in a vertical plane, in the axis of symmetry of the lecturer's hall: *a* – the room filled in 100%, *b* – the room filled in 50%

ized by the lowest airflow velocities. The maximum PMV value is 0.07 for case (*a*) and 0.15 for case (*b*).

An analysis of the PPD indices indicates that as for the case of 100% load of the lecture hall, the mean PPD value is 10.21; however, in the case of using the seats in the hall by the students in 50%, the value is at the level of 7.10. In both design cases, the minimum values of the PPD indices are about 5.00. As for case (*a*), the maximum values are PPD=20.90, in the zone of the highest rows in the area of measure line E. When half of the seats are taken by students in the front rows, the highest levels of the PPD indices, at the level of about 16.95, have been recorded in measure lines *A* and *B* that run along the walls in the area of the highest rows.

Table 2, presented below, contains a summary of the obtained results for the parameters of local discomfort. The criteria for rooms in Category *C* are met for the discussed design case in which 100% of seats in the hall are used. An analysis of the case in which a half of the seats is taken by students

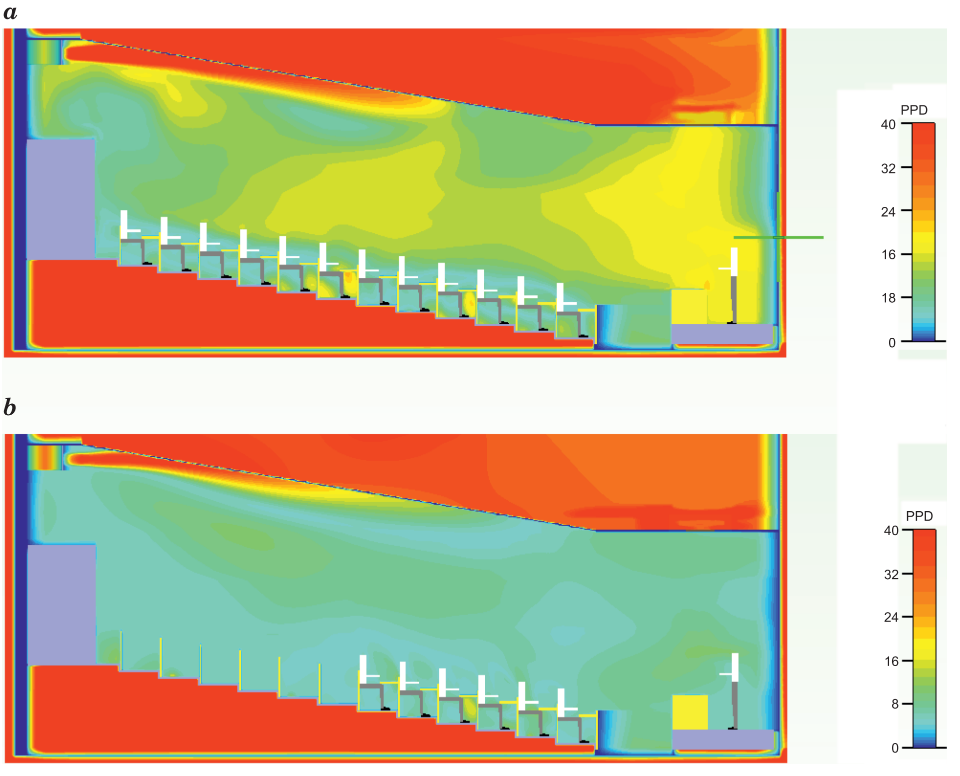

Fig. 9. The value of the PPD index in a vertical plane, in the axis of symmetry of the lecturer's hall: *a* – the room filled in 100%, *b* – the room filled in 50%

Table 2

The discomfort parameters for the analysed model

Specification	DR		PD	
	mean	maximum	mean	maximum
The lecture hall illed in 100%	15.12	24.53	0.67	5.32
The lecture hall filled in 50%	8.01	19.14	0.44	2.84

indicates that, maximally, about 3% of the users may express their dissatisfaction with the vertical temperature difference between the levels of their feet and their heads, however, about 10% of the users may complain about a discomfort caused by air drafts.

Conclusions

The obtained results of the analyses indicate that, according to the PN-EN ISO 7730:2006 norm, the requirements for Categories $A \div C$ are met in the studied room. The conducted computational simulations show that its concept of leading the conditioned air should be reanalysed at the design stage. In the proposed system, taking seats by students and opening the worktops significantly disturbs the airflow between the inlets and outlets. Maintaining a proper airflow velocity is not going to be possible even if adjusting the air inflow temperature to the thermal requirements is ensured by the measuring and driving sub-system in the automated system of air heating (the air temperature depending on the temperature of the air outflow and the external air). In the discussed cases, the analysis of PMV and PPD indices distribution has been presented for the air inflow temperature selected by the method of successive approximations.

It should be remembered, however, that methods of computational fluid dynamics have got their limitations. The presented analyses reflect situations that are deprived of dynamics. None of the elements in the model has changed its parameters and location in time. Also, the external parameters outside the lecture hall and location of people in the room have both been considered as unchangeable. Moreover, the presented study does not include an analysis of humidity distribution. However, CFD models should be validated and in optimal case, compared with experimental investigation of the distributions of velocity, temperature and PMV and PDD indices.

The example of the analysis presented in this article includes the designing phase for an object. It provides a possibility to verify its correctness of conceptual assumptions and to avoid mistakes that may turn out to be severe and their removal may be very time-and-money-consuming or, in some cases, just impossible.

Comment

Analyzed lecture hall is part of a new building. Initial version of the following article was being prepared while the object was still not in use. At present, the object is fully used. On 14 January 2016 among users of the analyzed lecture hall, the survey has been carried out. There were 107 participants of the survey (52 women 55 men). The average age was 21.8 years. Most of the respondents were characterized by a mediocre body build (sevenfold scale where 0 is average value). No one had chosen the „very fat” option. Only three people have declared that they are corpulent and five, that they are very thin. During the test everyone was sitting (metabolic rate about 70 W/m^2, 1.2 met). 35% of participants has been wearing clothes with high thermal insulation (clo 1.10; panties, stockings, blouse, long skirt, jacket, shoes). 28% of respondents has been wearing clothes with thermal insulation clo 1.00. During examination, occupants were staying in the lecture hall from 30 minutes. The average indoor air temperature was 20.4°C. Respondents were sitting evenly in all rows. Between the next seated person there was an empty seat. According to sevenfold scale where 0 is the optimal value (PN-EN 15251:2012): 9% chose the value -3 – cold; 21% pick the -2 value – cool; 46% chose the value -1 – slightly cool; 18% chose the value 0 – neutral; 3% indicated option +1 – slightly warm; 1% chose the value +2 – warm and 2 respondents pick the +3 value – hot.

Surveys, as well as CFD simulations, indicate that users complain about too low temperature in analyzed lecture hall. Experimental results are even more disadvantageous; during numerical investigation. In worst case about 10% of users should be dissatisfied because of thermal conditions meanwhile questionnaire shows 30% dissatisfied.

However, the precise, experimental investigation of the distributions of velocity, temperature and humidity are required for CFD model validation.

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