Heritage Science

RESEARCH ARTICLE

Microclimate numerical simulation to obtain the minimum safe distances between a painted wood panel and the inner face of an exterior wall

Santi Ferrer[1], Gema Campo-Francés[2], Cristina Ruiz-Recasens[2*] and Marta Oriola-Folch[2]

Abstract

This study provides a detailed understanding of the heat fluxes and temperatures that take place in the *channel* between the inner face of an exterior wall and the back of a painted wood panel hung on it. This is performed by means of a numerical simulation with a 2 dimensional CFD software. Distributions of temperatures, heat fluxes, and other parameters are quantified for 56 cases where the classical equations—Raithby-Hollands and similar—cannot be applied as these require vertical isothermal plates or isofluxes. Studied scenarios include different panel heights, *channel* widths, and room heights. Combining these data with outside temperature (− 3 °C) and heating air supply temperature (20 °C), to provide a nearly constant 19.6 °C in the room except in the channel between panel and wall, and with two values of specific humidity in the room, we provide for every studied case, advised distances, for these conditions, between the panel and the wall.

Keywords: Museum microclimates, CFD model, Numerical simulation, Wall condensation, Easel painting condensation, Thermal convection

Introduction

If we place an easel painting or a painted wood panel in the interior face of an exterior wall (either touching the wall or very close to it) both act as thermal insulators. So, in historical buildings with non-thermally insulated exterior walls and low outside temperatures, the inside wall face and the painting back temperatures can drop some degrees below the room temperature and consequently the relative humidity (RH) in these places can rise to unacceptable levels for the proper conservation of paintings [1]. Neuhaus [2] states that in this case, spacers shall be placed between paintings and wall in order to provide insulation at the painting back but required distances are not provided.

In Padfield previous study [3] temperature and RH have been measured on site and a minimum distance of 20 mm has been recommended between a 35 cm high framed painting and the inner face of a window facing north. Although broader distances have been suggested for bigger artefacts, specific values have not been calculated for this or other scenarios.

The aim of this research is not to set the exhibition room climatic conditions in historical buildings, which today in Europe are established by the CEN recommendations but to calculate the temperatures and heat flux distributions in the *channel* between a cold exterior wall and painted wood panels of different sizes placed at varying distances from the wall and at two different relative humidity values, to extend the general recommendation given in the Padfield study. Performing measurements on site requires large number of tests in different and very accurate environment conditions. The differential

*Correspondence: cruiz@ub.edu
[2] Arts and Conservation Department, Fine Arts Faculty, University of Barcelona, C/Pau Gargallo 4, 08028 Barcelona, Spain
Full list of author information is available at the end of the article

equations that model heat transfer in fluids take into account three fundamental laws as the mass, momentum and energy conservation, and thus theoretically, they can solve the heat transfer simulation between the fluid and the boundaries in whatever complex geometry. However, as they are in not linear partial derivatives, they can be solved analytically only in a few particular cases through dimensional analysis and simplifications.

Heat transfer between two parallel vertical isothermal and isoflux plates through the air by natural convection has been well parametrized by Raithby-Hollands [4], Bar-Cohen [5], Cadafalch-Oliva [6], Spalding [7, 8], and others, but as the exterior wall and the wood panel are neither isothermal and nor isoflux plates, their equations cannot be applied.

Instead, at present, CFD tools offer the possibility to solve numerically differential equations that model heat transfer in our selected geometry and materials with enough accuracy and at a reasonable cost. The commercial 2D fluid mechanics software EasyCFD_G developed and tested by A. Gameiro of the Coimbra University [9–14], is used in this research to extend the only existing single recommendation, up to now, of minimum 20 mm for the distance between a "cold" external wall in historical buildings and a painting hung on it.

This tool allows us to use the (k-ω) or the (k-ε) equations, where k is for turbulent kinetic energy, ε for its dissipation rate, ω for the turbulent kinetic energy frequency of dissipation, according to the distance from each mesh element to the boundaries and to quantify some magnitudes, raised in the Grau-Bové and Strlic article [15] such as the turbulence and the thermal fluxes in each mesh element, distinguishing between the contributions by convection and conduction.

These simulations show important vertical temperature gradients at the wall and the wood panel back when the room heating air supply (20 °C) provides a nearly constant value (19.6 °C ± 0.3 K) around the room (except in corners and in the space between the back of the panel and the wall).

The specific humidity (SH) is considered constant in all the room. The possible water buffer action from the boundary surfaces (walls and panel) is not considered, as the simulations are done for a steady state and this allows to simplify the problem and to get an easier understanding of the heat fluxes involved. Two SH scenarios are considered (7.5 g/kg and 8.7 g/kg) which correspond to 52% RH and 60% RH both at 19.6 °C.

From this point on, *Wall* refers to the inner face of the exterior wall, *Back* to the wood panel surface facing the *channel* and *Front* to the wood panel surface facing the room. Also the space between the inner side of the exterior wall and the back of the panel is named *channel*.

Materials and methods

Simulation description

We have built 8 different 2-dimensional CFD "models", each of them composed by a vertical cross-section of a room 3 m wide with a 2 cm thick wood panel hanging at different distances (*"s")* from the *Wall*. From this point on, the word *panel* will be used with the same meaning as *wood panel*.

All the models also have the same following features: heating system by hot air forced convection, distance from the panel centre to the floor (1.5 m) and boundary conditions. The difference among the 8 models lies in the following variables: different panel height *"h"* (2 m, 1 m and 0.5 m), different room height *"ht"* (3 m and 6 m) to quantify the effect of the distance travelled by the air along the cooled inner face of the *Wall* before entering in the *channel*, one model without the panel (for comparison) and another model in a room 6 m high with a cornice 15 cm high and 5 cm wide protruding from the inner side of the outside wall at 30 cm above the wood panel.

For each of the 8 described models, there are different "cases" (one for each distance *"s"* from panel to wall), see Table 1. Combining all the mentioned variables, we get 45 basic cases. All basic cases and models are listed in Table 1. See sketch in Fig. 1.

Doing it we have simulated 45 basic cases where in a heated room of different heights, painted wood panels of different dimensions (models) are hung at different short distances on the inner face of an exterior wall.

In these simulations we got air velocities (u horizontal and v vertical), temperatures (Tw wall, Tb back and Ta air), heat fluxes and turbulence intensity (Tu) for each of

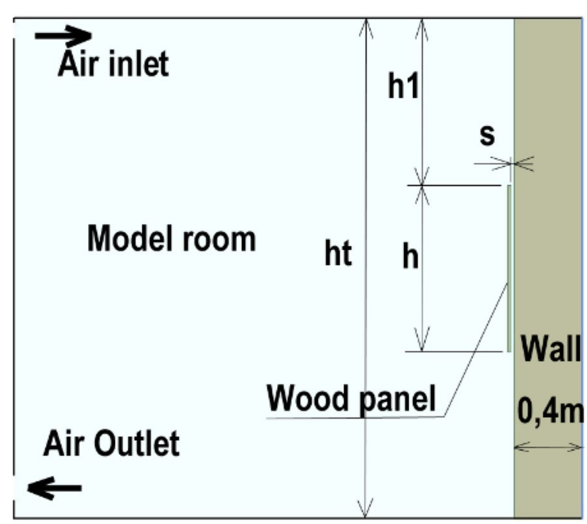

Fig. 1 Sketch room/panel case 12 ht = 3 m, h = 1 m, h1 = 1 m, s = 20 mm

Table 1 Summary of the 45 basic cases data

Panel height	*Channel* width	–	Distance from panel to the ceiling	Room wall height	Mesh cells	Cell rows in *Channel* Width	Total iterations	Initial time step subrelax factor	Next time step subrelax factor	Next time step subrelax factor	Class	Model	Case
h	s	s/h	h1	ht	–	–	–	–	–	–	–	–	–
m	mm	Ratio	m	m	nº	nº	nº	–	–	–	–	–	–
2	40	0.020	0.5	3	70,847	16	2968	0.1	0.02	–	1	1	1
	30	0.015			65,714	12	2471	0.1	0.02	–	1		2
	25	0.0125			68,442	10	3023	0.1	0.02	–	1		3
	20	0.010			64,526	8	4971	0.1	0.02	–	1		4
	15	0.0075			11,8332	11/12	3495	0.1	0.02	–	1		5
	10	0.0050			11,0785	8	1775	0.1	–	–	1		6
	5	0.0025			17,8393	6	6971	0.1	–	–	1		7
	3	0.0015			12,1399	5/9	2925	0.1	0.02	–	1		8
	2.5	0.0013			13,1171	5/7	1257	0.1	–	–	1		9
	2	0.0010			11,6523	4/6	1213	0.1	–	–	1		10
	0	0.000			56,490	0	2644	0.1	0.02	–	1		11
1	20	0.020	1	3	47,763	8	2342	0.1	0.02	–	1	2	12
	15	0.015			46,880	7	3686	0.1	0.02	–	1		13
	12.5	0.0125			76,151	11	2119	0.1	–	–	1		14
	10	0.010			76,889	9	3012	0.1	0.02	–	1		15
	7.50	0.0075			102253	12	4599	0.1	0.02	–	1		16
	5	0.0050			99,777	8	3748	0.1	0.02	–	1		17
	2.5	0.0025			91,920	4	2380	0.1	–	–	1		18
	1.5	0.0015			15,4030	4/5	1535	0.1	0.02	–	1		19
	0	0.000			43,943	0	2517	0.1	–	–	1		20
0.5	15	0.030	1.25	3	75,821	14	2508	0.1	–	–	1	3	21
	10	0.020			74,855	10	2969	0.1	–	–	1		22
	7.5	0.015			75,385	8	3111	0.1	–	–	1		23
	6.25	0.0125			73,701	7	3392	0.1	0.02	–	1		24
	5	0.010			72,294	5	2567	0.1	–	–	1		25
	3.75	0.0075			96,530	6	2577	0.1	–	–	1		26
	2.5	0.0050			97,771	4/5	2615	0.1	–	–	1		27
	1.25	0.0025			16,1379	5	1149	0.1	–	–	1		28
	0	0.000			68,316	0	2597	0.1	–	–	1		29

Table 1 (continued)

Panel height	*Channel* width	–	Distance from panel to the ceiling	Room wall height	Mesh cells	Cell rows in *Channel* Width	Total iterations	Initial time step subrelax factor	Next time step subrelax factor	Next time step subrelax factor	Class	Model	Case
h	s	s/h	h1	ht	–	–	–	–	–	–	–	–	–
m	mm	Ratio	m	m	nº	nº	nº	–	–	–	–	–	–
2	40	0.020	3.50	6	86,321	16	2189	0.1	0.02	0.01	2	4	30
	20	0.010			72,953	8	6939	0.1	0.02	0.01	2		31
	10	0.005			12,5094	8	1772	0.1	–	–	2		32
	5	0.0025			13,9375	9/15	2352	0.1	–	–	2		33
	3	0.0015			14,4616	6/9	3101	0.1	0.02	0.01	2		34
	2	0.0010			18,2261	5	2474	0.1	0.02	–	2		35
	0	0.000			77,784	0	4113	0.1	0.02	0.01	2		36
2	40 *	0.02	3.5	6	10,2612	19	2825	0.1	0.02	0.007	4	5	37
1	20	0.020	4	6	94,336	16	2327	0.1	0.02	0.01	2	6	38
	10	0.010			83,360	9	5953	0.1	0.02	–	2		39
	5	0.005			70,471	4	2141	0.1	–	–	2		40
	4.2	0.0021			17,1693	9/11	2888	0.1	0.02	0.01	2		41
	3.5	0.0018			17,5508	7/9	2413	0.1	0.02	–	2		42
	2.5	0.0013			17,3025	6	4470	0.1	0.02	–	2		43
	0	0.000			73,057	0	4285	0.1	0.02	–	2		44
Only wall 3 m height model													
0	0	0.000	0.0	3	22,494	–	1157	0.1	–	–	3	7	45

Advection scheme: hybrid for all the cases. Maximum residual allowed: 5.0E−06

*In this case (37) there is a 15 × 5 cm cornice, 30 cm above the panel all the CFD mesh elements in the room (between 43943 and 182261). Thus, we have exported the temperatures and heat fluxes distribution from 100 points located in three vertical layers along the panel height: along the *Back*, along the *Front* and along the *Wall*. Knowing these temperatures distribution and the SH in the room, we have calculated the RH values in these same 100 points for the two SH scenarios.

Besides, we also got the air velocities and turbulence intensity along the *channel* that helps to improve the understanding of the heat and mass transfer through the *channel*. Comparing the exported values, we have seen that in addition to the classical non-dimensional number *s/h* (*channel* width/wood panel height) there are other parameters (as the distances from the ceiling to the panel or the possibility of eddies at the *channel* inlet and outlet) that can modify the temperature and the heat flux distribution in the *channel*.

Heat transfer parameters and room conditions

Walls and wood panel heat conduction

The room floor, the roof and the inner wall are adiabatic. The exterior wall is limestone 41 cm thick, and includes a 1 cm mortar layer. The exterior wall parameters are: thermal conductivity $k_h = 1.8$ W/m K, density $\rho = 2095$ kg/m^3, and specific heat or fluid heat capacity $cp = 903$ J/kg K, all including the correction factor for the mortar layer. The parameters used for the wood panel are: $k_h = 0.2$ W/m K, $\rho = 700$ kg/m^3, and $c_p = 1600$ J/kg K.

Although as stated in the Fourier equation $\rho c_p \frac{\partial T}{\partial t} = 0$, with T for temperature and t for time, the ρ and c_p values are not required for heat fluxes through solid materials in steady conditions (see Additional file 1: Appendix S2.1 point D), we use them in the calculations as they are required for the transient approach, with a virtual time factor, used to get a more stable convergence (see Additional file 1: Appendix S2.2) As in this study we are not concerned with the real time needed to reach the steady condition, the temperature and heat flux obtained are applicable to other exterior walls and hung panels provided with correlated conductivity materials and widths (same $width * k_h$).

Outdoor air temperature set for all the simulations is − 3 °C and outdoor *heat convection boundary film coefficient* has been considered to be 25 W/m^{2}K as recommended for building heat load calculations under normal wind conditions.

Heating by air convection

In order to maintain temperature in the room at 19.6 °C ± 0.3 K (except for the four corners and a narrow area near the exterior wall and the floor) we have simulated a forced convection system blowing air into the room at 20 °C and at 1 m/s for a room 3 m high. In this 2D simulation, the air enters the room through a grille 0.1 m high, located on the upper corner of the opposite wall from the wood panel. The air direction is shifted by the grille − 15 °C from the horizontal axis to counteract part of the Coanda effect. The air is exhausted by an identical grille near the floor level in conservative conditions. Done this way, we achieve a very homogeneous temperature distribution without stratification.

Air supplied in models with rooms 6 m high is blown at 2 m/s and 20 °C (instead of 1 m/s) to compensate heat losses through the exterior wall which is twice the area of the wall 3 m high and to achieve the 19.6 °C ± 0.3 K in the room.

Inside the room the air conduction heat coefficient used in the boundary layer laminar flow is 0.0255 W/m K.

SH scenarios

Two different SH scenarios, both within the range of proper museum conditions, have been set out to calculate the RH values in the different areas of the room, according to the temperatures computed in the CFD models and cases. In this way the *channel* microclimate was correctly quantified. The conditions for both scenarios are the following:

Scenario SH1: 52% RH in a room heated at 19.6 °C with a Dew Point of 9.7 °C and a SH = 7.5 g water/kg air, which corresponds to high occupation (5 m^2/person) + natural infiltration at 1 Volume/h with outside air at − 3 °C and 90% RH.

Scenario SH2: 60% RH in a room heated at 19.6 °C with a Dew Point of 11.9 °C and a SH = 8.7 g water/kg air, which corresponds to middle occupation (8 m^2/person) + natural infiltration at 0.5 Volumes/h with outside air at − 3 °C and 90% RH.

RH criteria for the proper conservation of the artistic assets

To guarantee the proper conservation of the artistic assets, the following two criteria have been set: 1) RH in the panel *Back* must be < 66%; 2) If RH in the *Wall* = 100%, at least "*s*" ≥ 2 cm.

The first criterion corresponds to the Smithsonian Institution standard for RH in exhibition rooms released in 2007 [16] which established the range from 37% to 53% and was extended by Erhardt et al. [17] to 30% < RH < 60% after the tests performed on different art materials such as wood, pigments, paint layers, gesso, etc. Also the present ASHRAE conditions [16, 18, 19] can be met with two options for class A (this class foresees small risk of mechanical damage to high vulnerability artifacts and no mechanical risk to most artifacts, paintings, photographs, and books). The first one is RH from 35 to 65% with the set point at 50%, allowing ± 10% seasonal set point variations and ± 5% short term fluctuations. This option allows to extend the RH up to 65%. The second one is RH within the range of 40–60% with 50% as a permanent set point for the whole year and ± 10% short term fluctuations.

Our second criterion—a minimum separation of 2 cm between the panel *Back* and the *Wall* when this later is at 100% RH—is under review and it is considered only as a temporary solution in the case that it is impossible to change the painting location or to improve the exterior wall isolation, as it usually happens in some historical buildings.

Actually others standards as EN 15757 [20], EN15759-1 [21] and EN 15759-2 [22] are developed to deal with climate at historical buildings, churches and chapels where wood panels and easel paintings have been submitted to important RH variations along the centuries and have been adapted to the climate conditions showing visible effects such as cracks and flakes or not visible as plastic strains. In these cases, changing the historical climate conditions for others theoretically better may lead to a more endangered condition of the weak artwork.

These standards allow the seasonal RH set point to move in a broader range (safe band), according to the computed mean average (CMA) calculated after 13 months of measured RH inside values and the conservators criteria about the exposed or stored artwork condition, as Camuffo [23, 24], Bratasz [25, 26], Broström et al. [27], Neuhaus and Schellen [28], Schijndel [29] and others have explained.

Today the room RH control in exhibitions and museums varies according to the institutions economic saving policy and energetic goals, the condition and history of exposed or stored objects, the wall materials and air tightness, the importance of the visitors comfort, and other factors.

In order to simplify, in our research we have used, as RH threshold, the maximum allowed value in the ASHRAE standard for the A class. It was not possible to give a universal solution to all the real cases, as this involves studying the endless combinations of temperature and RH possible conditions, the different isolation and thermal inertia of the walls, the unlike building shapes, the varying RH buffers, etc. Our goal is to improve the understanding of heat fluxes, RH and temperatures that take place at the back of wood paintings on exhibition hung in peripheral walls in order to provide conservators with the knowledge to be applied in cases with similar conditions.

Complementary cases

After the analysis of the 45 basic cases, only 3 of them complied the RH values to avoid damages in the artistic asset. Thus, we have extended the simulations to other complementary cases and by a trial and error procedure we have found the distances required, from the panel to the *Wall*, to comply with the required RH in each layer. In this way we got 9 complementary cases, all of them in the same conditions as before, but not exceeding the 66% RH in both SH scenarios. In the cases of water condensation in the *Wall* a minimum of 20 mm from *Back* to *Wall* is maintained. Also we have extended the simulations to 2 complementary cases by adding 60 mm of expanded polystyrene (EPS) insulation to the exterior wall and to 2 other complementary cases (*channel* width 8 mm and 20 mm) adding an aluminium protection layer, 0.5 mm thick, at the *Back* which improves greatly the *Back* temperatures. As the 8 mm *channel* width case has a minimum *Wall* temperature of 7.3 °C (RH = 100%) we cannot recommend it and the 20 mm minimum *channel* width is recommended for the Wood panel with Aluminium protection. For the isolated wall composition see Table 2.

Like this, we have extended the general recommendation [3] that states a minimum of 20 mm between a painting and an exterior wall, to more detailed specifications according the different situations. In addition we quantified and clarified the heat transfer modes between the *Back* and the *Wall* in these circumstances.

There are other possible variables to take in account, as possible different panel thickness, obstacles to the air flow as cradling, other room or outside temperatures, other inside SH, other geometries, etc., but we have focused in our variables in order to understand and quantify the temperatures, RH and heat fluxes in different conditions, to provide to the conservators the knowledge to understand and deal with the particular conditions they would find.

Calculation procedure. Equations used

The equations used are the conservation of mass (continuity) equation, the conservation of momentum (Navier-Stockes) equations, the conservation of energy (transport equation for enthalpy) equation and the heat transfer through solids (wall and wood panel) equation which it is the Fourier equation but without internal heat production.

Table 2 Isolated wall parameters for two complementary cases

Material	Layer	Width cm	Thermal conductivity W/m K
Concrete block	Outside	25	1
EPS insulant	Internal	6	0.038
Ceramic brick	Inside	10	0.6

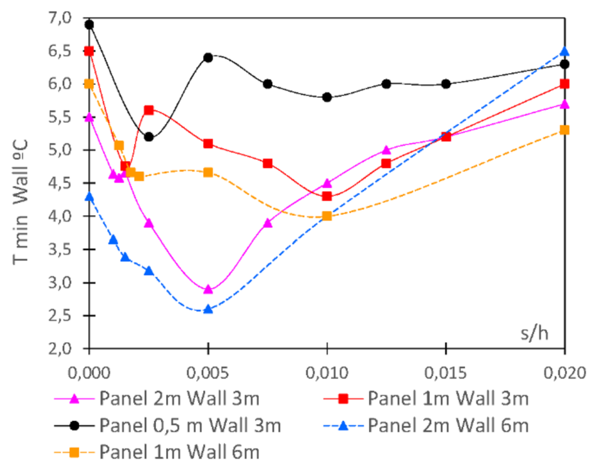

Fig. 2 Wall minimum temperature (basic cases)

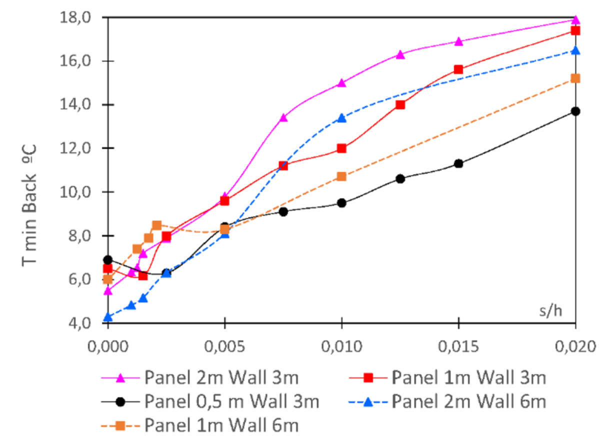

Fig. 3 Minimum back temperature (basic cases)

All these equations are developed in Additional file 1: Appendix S2.1

The procedure used to solve these equations in a steady flow is shown in Additional file 1: Appendix S2.2

The boundary and initial conditions used to solve these equations are shown in Additional file 1: Appendix S2.3

Results

Temperature and heat flux values in 100 points of the CFD simulation, distributed in the vertical axis along the interface layers—*Wall*/*channel* air, *channel* air/*Back*, and *Front*/room air—have been exported for each case.

Also for each of the cases, the temperature and the vertical component (v) of the air velocities have been exported from 20 points distributed horizontally along the *channel* inlet between the *Wall* and the panel to compute *Ti,* the *channel* inlet air mean temperature (vertical velocity weighed).

For the cases of the model where the wood panel is 50 cm high, the turbulence intensity *(Tu)* has also been exported in order to compute the turbulence intensity average *(Turb)* at different heights along the *channel*. Doing it in this way, the computed heat fluxes have been assigned to the different convection modes (laminar or turbulent) in the detailed analysis of "Heat fluxes and minimum temperatures in the *Wall* and the Panel *Back*" section.

In Figs. 2 and 3 the minimum temperatures in the *Wall* and the *Back* can be seen for all models and cases.

Figures 4, 5 and 6 show the averaged heat fluxes for all models and cases. Figure 4 displays the total heat flux received by the *Wall*, Fig. 5 the heat flux supplied by the *Back* and Fig. 6 the heat flux supplied directly by the air entering the *channel*. Figure 7 shows heat flux ratios

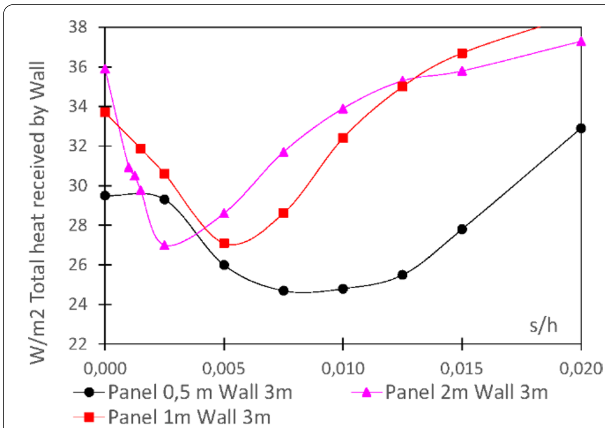

Fig. 4 Heat flux received in the wall (cases room 3 m high)

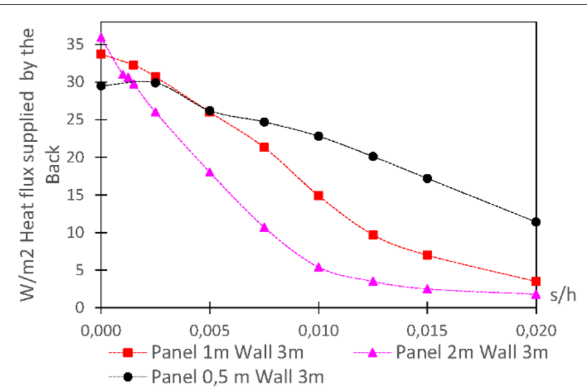

Fig. 5 Heat flux supplied by back (cases room 3 m high)

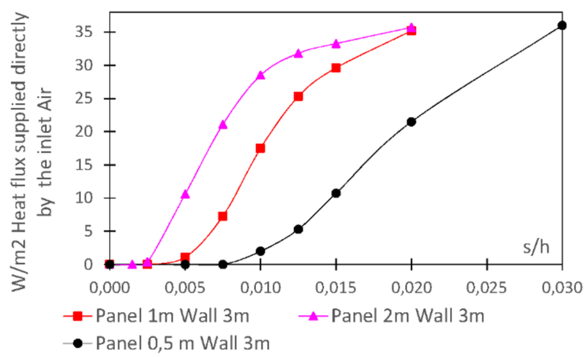

Fig. 6 Heat flux supplied by inlet air (cases room 3 m high)

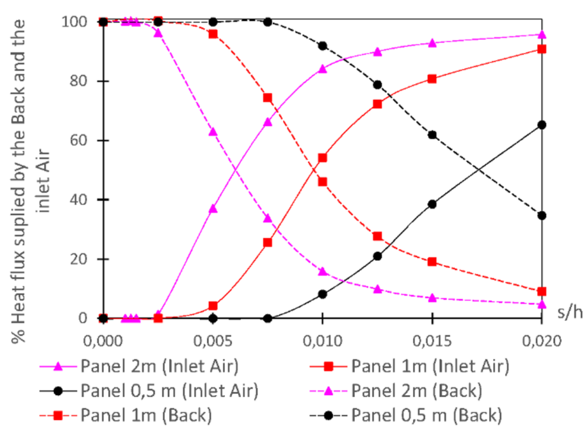

Fig. 7 Heat flux back and inlet air % (cases room 3 m high)

between *Back* and *Wall* (total) and also between Inlet air and *Wall* (total). In all these graphics heat fluxes and temperatures are plotted versus the *"s/h"* adimensional number.

Table 3 presents all these values for the 45 main cases together with other parameters such as dimensions, plus the inlet air eddy, if it appears. In addition, RH in the *Wall*, the *Back* and the *Front* are included for the scenarios SH1 and SH2.

For complementary information, see Additional file 1: Table S0.1 in Additional file 1: Appendix S0 with more detailed results for each layer (*Wall*, *Back* and *Front*).

Discussion

Heat fluxes and minimum temperatures in the *Wall* and the Panel *Back*

In Additional file 1: Appendix S1 there is a detailed analysis of the heat fluxes, turbulence and temperatures at different heights in the *channel* between *Back* and *Wall* for all the cases in a 50 cm high panel. It allows to understand the involved phenomena and temperatures distributions in the *channel*.

We can see how the *Wall* temperature decreases as the panel separates from the *Wall,* due to the laminar flow close to the *Wall,* and thus the heat flux is only by conduction. Added to the effects of the eddy locations in the *channel* inlet and *channel* outlet, the temperature attain a minimum at $s=1.25$ mm. As the *channel* width continues increasing at $s=2.5$ mm the outlet eddy change its direction and the *Wall* is heated enough to compensate the conduction heat flow reduction and the *Wall* temperature attain a relative maximum.

As the *channel* width continues increasing at $s=5$ mm, it appears another temperature minimum. If the *channel* width continues increasing and attain $s=7.5$ mm, the heat flux to the *Wall* increases, as inlet air heat flux increases more than the *Back* heat flux decreases (due to the increasing of inlet air flow and turbulence). This raises the *Wall* temperatures.

From this point the *Back* and *Wall* temperatures increase, approaching the air room temperatures. It is quantified in Additional file 1: Table S1.1 and explained with detail in Additional file 1: Appendix S1.

Figure 2 and Table 3 show how the two *Wall* temperature minimums discussed in Additional file 1: Appendix S1 for a 50 cm high panel also appear for panels with $h=1$ m (4.8 °C and 4.3 °C) and for panels with $h=2$ m (4.58 °C and 2.9 °C) in a room 3 m high and for panel with $h=1$ m (4 °C and 4.6 °C) in a room 6 m high. As $h1$ (distance from the panel upper border to the ceiling) increases, the difference between the first minimum *Wall* temperature and the intermediate maximum decreases. The intermediate maximum nearly disappears for $h=2$ m and $h1=3$ m. As h increases, the second minimum is displaced to lower *s/h* values (to the left side of the curve in Fig. 2).

The explanation is that as $h1$ increases, more cooled is the air descending close to the *Wall* and the *Ti* (average air inlet *channel* temperature) is lower. Thus, the air layer cooled by the *Wall* is wider and more air mass is cooled. This can be seen in the Table 3, column T_i *(Channel inlet air mean temperature, vertical velocity weighed).*

However, the phenomenon is not that simple and other factors shall also be considered.

The air mass flow going into the channel

It can be seen in the special case for $h=2$ m, $s=40$ mm, where the *minimum Wall temperature* for $ht=6$ m, is higher (6.5 °C) than for $ht=3$ m (5.7 °C). For $ht=6$ m, the air mean velocity in the *channel* inlet is 0.6 m/s (at 16.4 °C as average) whereas for $ht=3$ m it is 0.3 m/s (at 18 °C average). Thus, being the air mass twice for $ht=6$ m and the air temperature only 1.6 K lower, the inlet air thermal energy level is higher for $ht=6$ m than for $ht=3$ m and it provides enough heat to raise the *minimum Wall*

Table 3 Comparison among models and cases in SH1 and SH2

Panel height	Channel width		Distance from panel to the ceiling	Room wall height	Minimum temperature			RH at minimum temperature in scenario SH1 (52% RH in the room)*			RH at minimum temperature in scenario SH2 (60% RH in the room)**			Average heat flux Wall = Back + inlet Air				Ti ***	Air mean vertical velocity at the channel inlet.	Channel inlet Eddy
H	*s*	*s/h*	*h1*	*ht*	*Wall*	*Back*	*Front*	*Wall*	*Back*	*Front*	*Wall*	*Back*	*Front*	*Wall*	*Back*	Inlet Air	Inlet Air/ *Wall*	*Channel* Inlet	Inlet	Inlet
m	mm	ratio	m	m	°C	°C	°C	%	%	%	%	%	%	W/m²	W/m²	W/m²	%	°C	m/s	Y/N
Only wall 3 m height model																				
0	0	0.000	0.0	3	12.6	–	–	82.5	–	–	95.2			66.3		66.3	100	–	–	–
2	40	0.020	0.5	3	5.7	17.9	18.1	100	**58.3**	**57.6**	100	67.3	66.5	37.3	1.8	35.7	95.8	19.1	0.35	Not
	30	0.015			5.2	16.9	17.3	100	**62.2**	**60.6**	100	71.8	70.0	35.8	2.5	33.27	92.9	18.9	0.33	Not
	25	0.013			5.0	16.3	16.6	100	**64.7**	**63.4**	100	74.7	73.2	35.3	3.5	31.8	90	18.8	0.33	Not
	20	0.010			4.5	15.0	15.8	100	70.4	66.8	100	81.3	77.1	33.9	5.4	28.5	84.2	18.4	0.32	Not
	15	0.008			3.9	13.4	15.0	100	78.2	70.4	100	90.3	81.3	31.7	10.7	21.1	66.3	17.8	0.27	Not
	10	0.005			**2.9**	9.8	11.8	100	99.6	87.0	100	100	100	28.6	18	10.6	37.1	16.8	0.19	Not
	5	0.003			3.9	7.9	10.1	100	100	97.6	100	100	100	27	26	0.4	1.4	9.1	0.06	Not
	3	0.0015			4.65	7.2	9.7	100	100	100	100	.100	100	29.8	29.8	0	0	7.1	0.02	Not
	2.5	0.0013			**4.58**	6.5	9.1	100	100	100	100	100	100	30.5	30.6	−0.08	0	6.7	0.02	Not
	2	0.0010			4.64	6.3	9.1	100	100	100	100	100	100	30.9	31	−0.08	0	7.0	0.01	Not
	0	0.000			5.5	5.5	8.1	100	100	100	100	100	100	35.9	35.9	0	0			
1	20	0.020	1.0	3	6.0	17.4	18.1	100	**60.2**	**57.6**	100	69.5	66.5	38.7	3.5	35.2	90.9	17.3	0.36	Not
	15	0.015			5.2	15.6	16.8	100	67.7	**62.6**	100	78.2	72.3	36.7	7	29.6	80.8	16.6	0.30	Not
	12.5	0.013			4.8	14.0	15.5	100	75.2	68.1	100	86.8	78.7	35	9.7	25.3	72.3	16.0	0.27.	Not
	10	0.010			**4.3**	12.0	14.1	100	85.9	74.7	100	99.1	86.2	32.4	14.9	17.5	54.2	15.2	0.22	Not
	7.50	0.008			4.8	11.2	13.7	100	90.6	76.7	100	100	88.5	28.6	21.3	7.3	25.6	15.0	0.14	Yes
	5	0.005			5.1	9.6	12.3	100	100	84.2	100	100	97.2	27.1	26	1.1	4.3	14.1	0.07	Yes
	2.5	0.003			5.6	8.0	11.3	100	100	90.0	100	100	100	30.6	30.7	0	0	10.5	0.02	Yes
	1.5	0.0015			**4.8**	6.2	9.5	100	100	100	100	100	100	31.9	32.3	−0.4	0	7.9	0.01	Yes
	0	0.000			6.5	6.5	9.5	100	100	100	100	100	100	33.7	33.7	0	0			
0.5	15	0.030	1.25	3	7.5	.16.3	17.2	100	**64.7**	**61.0**	100	74.7	70.4	41.5	5.4	36	87	16.1	0.34	Not
	10	0.020			6.3	13.7	15.3	100	76.7	69.0	100	88.5	79.7	32.9	11.4	21.5	65.3	14.2	0.25	Not
	7.5	0.015			6.0	11.3	13.5	100	90.0	77.7	100	100	89.7	27.8	17.2	10.7	38.5	12.9	0.18	Yes
	6.25	0.013			6.0	10.6	12.9	100	94.3	80.9	100	100	93.3	25.5	20.1	5.3	21	12.5	0.13	Yes
	5	0.010			**5.8**	9.5	12.0	100	100	85.9	100	100	99.1	24.8	22.8	2	8.2	11.4	0.09	Yes
	3.75	0.008			6.0	9.1	11.9	100	100	86.4	100	100	99.8	24.7	24.7	0.0	0	10.0	0.05	Yes
	2.5	0.005			6.4	8.4	11.2	100	100	90.6	100	100	100	26	26.2	0	0	8.3	0.02	Yes
	1.25	0.003			**5.2**	6.3	9.5	100	100	100	100	100	100	29.3	29.9	−0.6	−2	6.4	0.01	Yes

Table 3 (continued)

Panel height	*Channel* width		Distance from panel to the ceiling	Room wall height	Minimum temperature			RH at minimum temperature in scenario SH1 (52% RH in the room)*			RH at minimum temperature in scenario SH2 (60% RH in the room)**			Average heat flux *Wall* = *Back* + inlet Air				*Ti* ***	Air mean vertical velocity at the *channel* inlet.	*Channel* inlet Eddy
H	*s*	*s/h*	*h1*	*ht*	*Wall*	*Back*	*Front*	*Wall*	*Back*	*Front*	*Wall*	*Back*	*Front*	*Wall*	*Back*	Inlet Air	Inlet Air/*Wall*	*Channel* Inlet	Inlet	Inlet
m	mm	ratio	m	m	°C	°C	°C	%	%	%	%	%	%	W/m²	W/m²	W/m²	%	°C	m/s	Y/N
	0	0.000			6.9	6.9	9.8	*100*	*100*	99.6	*100*	*100*	*100*	29.5	29.5	0	0			
2	40	0.020	3.50	6	6.5	16.5	16.8	*100*	**63.9**	**62.6**	*100*	3.7	72.3	37.8	1.7	36.1	95	16.4	0.63	Not
	20	0.010			4.0	13.4	13.9	*100*	78.2	75.7	*100*	90.3	87.4	30.8	5.8	25	81	15.2	0.40	Yes
	10	0.005			***2.6***	8.1	9.6	*100*	*100*	*100*	*100*	*100*	*100*	24.2	15.9	8.2	34	15.0	0.17	Yes
	5	0.0025			3.2	6.3	8.2	*100*	*100*	*100*	*100*	*100*	*100*	24.9	24.2	0.7	0.03	12.0	0.05	Yes
	3	0.0015			3.4	5.2	7.1	*100*	*100*	*100*	*100*	*100*	*100*	26.9	26.8	0.05	0.00	9.7	0.00	Yes
	2	0.0010			3.7	4.8	6.8	*100*	*100*	*100*	*100*	*100*	*100*	28.1	28.0	0.02	0.01	9.6	0.01	Yes
	0	0.000			4.3	4.3	6.3	*100*	*100*	*100*	*100*	*100*	*100*	33.4	33.4	0	0			
1	20	0.020	4.00	6	5.3	15.2	15.9	*100*	69.5	66.4	*100*	80.2	76.6	34.2	5.6	28.7	83	14.6	0.42	Yes
	10	0.010			***4.0***	10.7	12.6	*100*	93.7	82.5	*100*	*100*	95.2	27.4	13.9	13.5	49.	13.7	0.27	Yes
	5	0.005			4.66	8.3	10.8	*100*	*100*	93.1	*100*	*100*	*100*	25.2	24.2	1.0	4	10.4	0.07	Yes
	4.2	0.0021			***4.60***	8.5	11.1	*100*	*100*	91.2	*100*	*100*	*100*	26.1	25.7	0.4	0	11.2	0.05	Yes
	3.5	0.0018			4.66	7.9	10.7	*100*	*100*	93.7	*100*	*100*	*100*	26.3	26.2	0.06	0	9.0	0.04	Yes
	2.5	0.0013			5.1	7.4	10.3	*100*	*100*	96.3	*100*	*100*	*100*	27.8	27.7	0.01	0	8.5	0.02	Yes
	0	0.000			6.0	6.0	9.0	*100*	*100*	*100*	*100*	*100*	*100*	32.2	32.2	0	0	–	–	–
2	40 ****	0.020	4.00	6	3.6	13.5	14.1	*100*	77.7	74.7	*100*	89.7	86.2	27.9	3.5	24.4	87.3	16.8	0.16	Yes

*SH1 = 7.5 g water/kg air; dew point = 9.7 °C; room at 19.6 °C and 52% RH; outside at − 3 °C

**SH2 = 8.7 g water/kg air; dew point = 11.9 °C; room at 19.6 °C and 60% RH; outside at − 3 °C

****Ti* = *Channel* inlet air mean temperature, vertical velocity weighed

****In this case (37) there is a 15 × 5 cm cornice, 30 cm above the panel

Bolditalic: minimum *Wall* temperature; italic: condensation; bold: RH < 66%

temperature up to 6.5 °C. When *"s"* diminishes, as in the case of $h=2$ m and $s=20$ mm, the *minimum Wall temperature* is 4 °C for $ht=6$ m (inlet velocity 0.4 m/s at 15.2 °C) and 4.5 °C for $ht=3$ m (inlet velocity 0.32 m/s at 18.4 °C). In this case, the air velocities are closer and the inlet air thermal energy level is lower for $ht=6$ m than for $ht=3$ m. An intermediate case is for $h=2$ m, $s=30$ mm where the *minimum Wall temperatures* are the same for $ht=6$ m and for $ht=3$ m (5.2 °C).

The air turbulence in the channel

For small values of *"s"* (*channel* width), the air flux is laminar and the heat transfer from the *Back* is only by conduction. This heat flux diminishes lineally as *"s"* increases, but as *"s"* increases more inlet air goes into the *channel* and the inlet air becomes the heat flux mean supplier to the *Wall.* As the *channel* air flow increases the heat flux turns to be turbulent and the addition of the both effects results in the curve "*s/h" versus heat flux* with a minimum in the zone where the inlet air heat flux starts (*s/h* between 0.003 and 0.01). This phenomenon can be observed in the three curves of Fig. 4 which are, in turn, composed by the curves of Figs. 5 and 6. The minimum in these curves moves to the right side of the chart as the panel height decreases. In order to further clarify this, Fig. 7 gathers these curves expressed as a percentage.

Also, as shown in Additional file 1: Appendix S1, when the *channel* is very narrow, air turbulence decreases as the air goes down inside the *channel*. In these cases, the air flux becomes laminar, still if it has been turbulent just near the *channel* inlet.

Depending on Temperatures, *"s"* and *"h"* parameters, the minimum temperature occurs at mid-height or at the lower zone of the *channel*. (See Additional file 1: Appendix S0, Table S0.1).

The eddy in the channel inlet

The widening of the air layer descending close to the *Wall* due to large values of *h1* and the narrowing of the *channel* can form an air eddy in the *channel* inlet, as more mass of the cooled air descending close to the *Wall* collides with the top edge of the panel. In this way the air going down close to the *Wall* gets mixed with the air of the room that enters the *channel.* This smooths the temperature differences in the *channel* inlet between the *Back* and the *Wall.* The developed eddy is shown in Figs. 8, 9 and 10.

Table 3 shows in which cases an eddy is developed at the inlet.

The eddy in the channel outlet

Usually one or more air eddies appear in the *channel* outlet. Depending on the position of these eddies, heated air of the room around the outlet can get mixed with the cooled air of the *channel*. This may cause heating of the *Wall* just under the *channel*. For example, in the case of a panel 50 cm high and a *channel* width of 2.5 mm (case 27), the *Wall* under the panel is heated enough by the outlet eddy to send heat flux by conduction through the *Wall* in vertical direction, raising the minimum temperature in the *Wall* (see Fig. 11 for s = 2.5 mm). In this same case, this factor explains the maximum of 6.4 °C in *the "Wall minimum temperature"* compared to the two neighbouring minimums of 5.2 °C that occur when $s=1.25$ mm where the eddy is too weak to heat the *Wall* enough and 5.8 °C when $s=5$ mm where the heated air of the room is not sent to the *Wall* near the *channel* outlet and, instead, the cooled *channel* air flows into the room (see Table 3 and Fig. 11 with s = 5 mm).

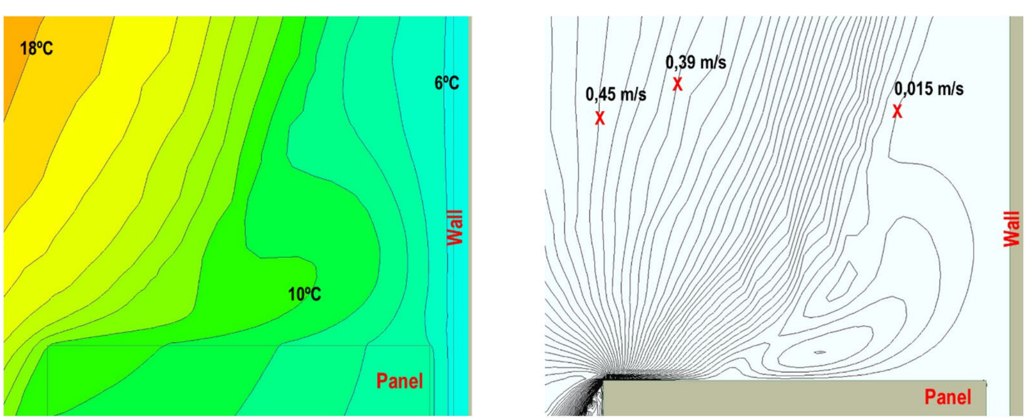

Fig. 8 *Channel* inlet temperature and velocity isolines. Case 28. Panel h = 50 cm. s = 1.25 mm. h1 = 1.25 m. Values at inlet: Tw = 5.7 °C, Tb = 6.7 °C. T average = 6.4 °C. Air channel vertical mean velocity = 0.01 m/s. Δ v between isolines = 0.015 m/s. Inlet eddy

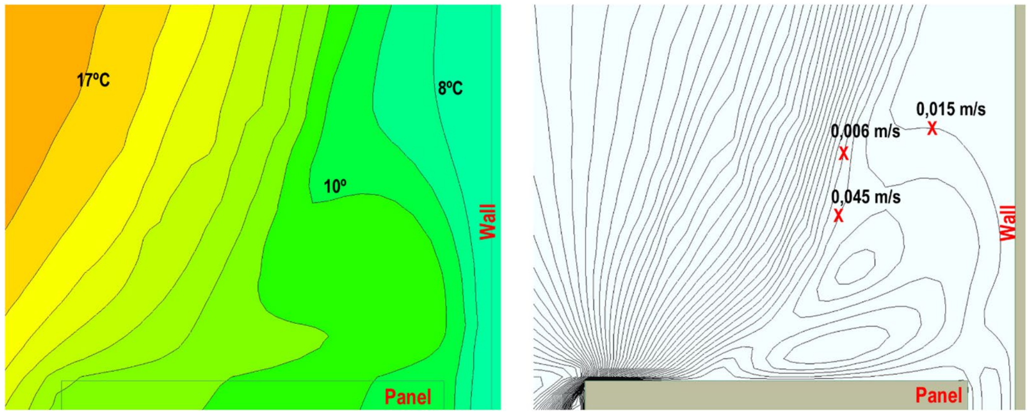

Fig. 9 *Channel* inlet temperature and velocity isolines. Case 27. Panel h = 50 cm. s = 2.5 mm. h1 = 1.25 m. Values at inlet: Tw = 6.8 °C, Tb = 8.6 °C. T average = 8.3 °C. Air channel vertical mean velocity = 0.02 m/s. Δ v between isolines = 0.015 m/s. Inlet eddy

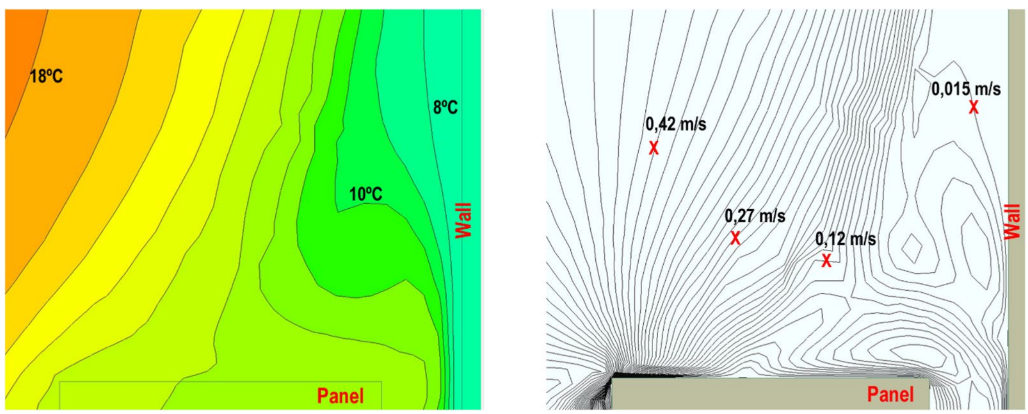

Fig. 10 *Channel* inlet temperature and velocity isolines. Case 25. Panel h = 50 cm. s = 5 mm. h1 = 1.25 m. Values at inlet: Tw = 7.8 °C, Tb = 11.2 °C. T average = 11.4 °C. Air channel vertical mean velocity = 0.09 m/s. Δ v between isolines = 0.015 m/s. Inlet eddy

Cornices over the panel

In the case of a cornice protruding from the *Wall* (*s* = 40 mm, case 37) it can be seen that the minimum temperatures in each layer are at least 3 °C lower that in the case with a straight *Wall*, due to the air flux deviation at the cornice and the eddy developed below it, that reduces the entrance of hot air from the room to the *channel*. So, the mean air temperature, vertical velocity weighed at the *channel* inlet, is reduced from 0.62 to 0.16 m/s (compared to the same geometry without cornice as it is the case 1) and still if the mean air temperature at this point is increased from 16.4 to 16.8 °C, the mean heat flux received by the *Wall* in the *channel* drops from 36.1 to 27.9 W/m^2 (see Table 3).

In order to compensate the *Back* and *Wall* temperature reduction and the heat flux reduction which occurs in this case, and to get the correct temperatures and RH values in the *Back*, it is necessary to increase the *channel* width from 40 to 90 mm in scenario SH1 and from 85 to 140 mm in scenario SH2 (see Tables 4 and 5).

RH values in the different cases

Table 3 also shows the maximum RH values of the significant layers (*Wall*, *Back* and *Front*) for scenarios SH1 and SH2. In both scenarios condensation appears on the *Wall* and, in many cases, RH values in the *Back* are above 66%, which is the limit set for the proper conservation of paintings.

Minimum required distances from a painted wood panel to the exterior wall

From the results of the 3 basic cases, that comply with the two conservation criteria set in Sect. 2.3, and the 13 complementary cases, we got a set of 8 cases for the SH1 scenario (room RH of 52% at 19.6 °C) and other set of 8 cases for the SH2 scenario (room RH of 60% at

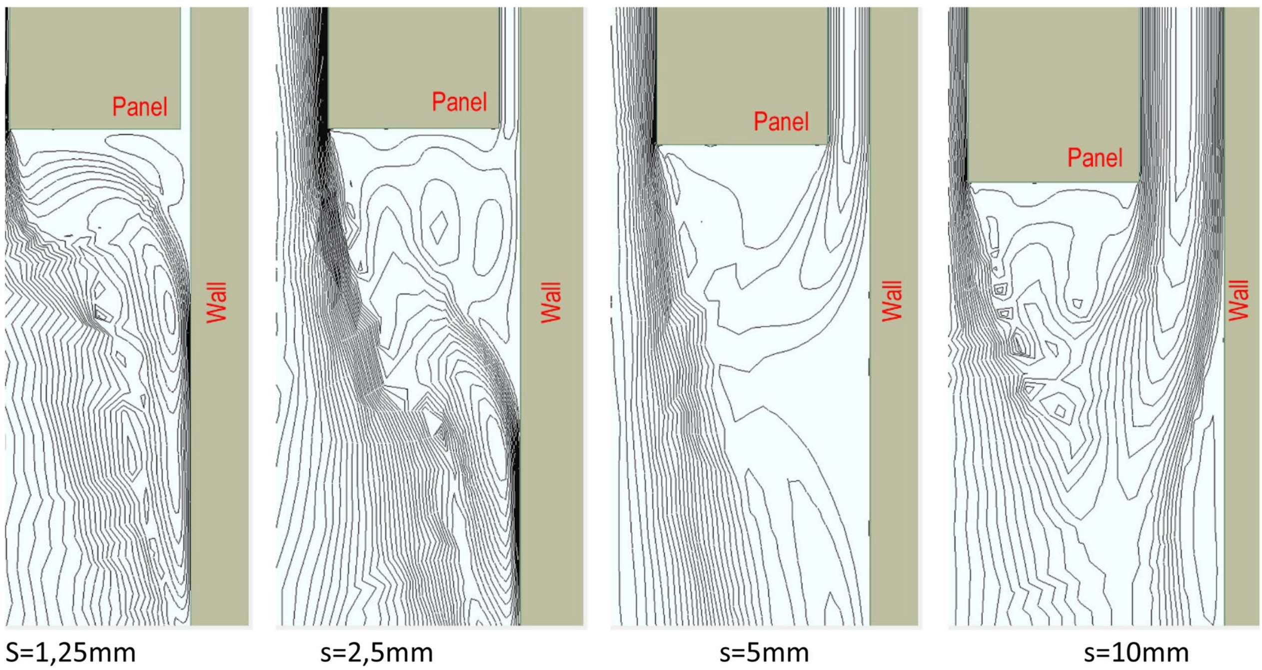

Fig. 11 Velocity isolines and eddys in *channel* outlet for panels 50 cm height and different widths

Table 4 Minimum "s" distances in scenario SH1

Panel height	Minimum *Channel* width		Distance from panel to ceiling	Room wall height	Minimum temperature			RH at minimum Temp. in scenario SH1 (52% RH in the room)		
	s	s/h	h1	ht	*Wall*	*Back*	*Front*	*Wall*	*Back*	*Front*
m	mm	ratio	m	m	°C	°C	°C	%	%	%
2	25	0.013	0.5	3	5.0	16.3	16.6	100	64.3	63.1
1	20	0.020	1.0	3	6.0	17.4	18.1	100	59.9	57.2
0.5	20	0.040	1.25	3	7.6	17.5	18.1	100	59.5	57.2
2	40	0.020	3.50	6	6.5	16.5	16.8	100	63.5	62.2
1	30	0.030	4.00	6	6.5	16.8	17.3	100	62.3	60.3
2	90*	0.045	3.50	6	5.1	16.2	16.3	100	64.7	64.3
Isolated wall 2 m height with 60 mm EPS										
2	20	0.010	0.50	3	13.7	17.2	18	76.2	60.7	57.6
Back protected with Aluminium layer 0.5 mm width										
2	20	0.010	0.5	3	5.3	19.4	18.9	100	52.0	54.4

If there is condensation in the wall (100% RH), the minimum required gap between the Wall and the painting to avoid damages is 20 mm

Room width 3 m. Outside wall: limestone (1.8 W/m * K), width 40 cm

SH = 7.5 g water/kg air; Dew Point = 9.7 °C; Room at 19.6 °C and 52% RH; Outside at − 3 °C

*In this case there is a 15 × 5 cm cornice, 30 cm above the panel

19.6 °C) which comply with these conservation criteria. From these 16 cases we got the correct *channel* widths for all the models analysed.

Table 4 presents the 8 recommended minimum *channel* widths for the scenario SH1 and Table 5 the same figures for SH2.

Table 5 Minimum "s" distances in scenario SH2

	Minimum *Channel* width		Distance from panel to ceiling	Room wall height	Minimum temperature			RH at minimum temp. in scenario SH2 (60% RH in room)		
h	s	s/h	h1	ht	*Wall*	*Back*	*Front*	*Wall*	*Back*	*Front*
m	mm	ratio	m	m	°C	°C	°C	%	%	%
2	50	0.025	0.5	3	5.8	18.4	18.5	100	65.2	64.8
1	35	0.035	1.0	3	6.8	18.8	19.0	100	63.6	62.8
0.5	30	0.060	1.25	3	7.9	18.7	18.9	100	64.0	63.2
2	85	0.043	3.5	6	7.0	18.6	18.6	100	64.4	64.4
1	65	0.065	4.00	6	7.4	18.5	18.6	100	64.8	64.4
2	140*	0.07	3.50	6	6.0	18.3	18.4	100	65.6	65.2
Isolated wall 2 m height with 60 mm EPS										
2	30	0.015	0.50	3	14.2	18.5	18.6	85.7	64.8	64.4
Back protected with Aluminium layer 0.5 mm width										
2	20	0.010	0.5	3	5.3	19.4	18.9	100	61.2	63.2

If there is condensation in the wall (100% RH), the minimum required gap between the wall and the painting to avoid damages is 20 mm

Room width 3 m. Outside wall: limestone (1.8 W/m * K), width 40 cm

SH = 8.7 g water/kg air; dew point = 11.9 °C; room at 19.6 °C and 60% RH; outside at − 3 °C

*In this case there is a 15 × 5 cm cornice, 30 cm above the panel

It shall be taken in account that changes of the *Wall* geometry, as cornices protruding above the painted wood panel, or other obstacles to the air flux can modify the recommended minimum separation distances, increasing them.

Attention: If the outside air temperature increases over − 3 °C and the 90% RH is maintained, the SH of the infiltration and ventilations will rise. Therefore the inside room SH and RH will be increased and in this case the recommended *channel* width shall be increased according the foreseen *channel* temperatures and the new inside room SH value.

Conclusions

By means of a 2 dimensional CFD simulation of a heated room (19.6 °C) with a painted wood panel hung in the inside face of a non-isolated exterior wall (− 3 °C in the outside face), we have obtained the temperatures, air velocities, and heat fluxes distribution in the *channel* between the wood panel and the exterior wall, for different wood panel heights, different *channel* widths, and different distances to the ceiling.

Based on the temperature data results in the wood panel *Back* and in the *Wall*, RH values have been obtained at the same locations and for two different SH scenarios (7.5 g water/kg air and 8.7 g water/kg air) corresponding to a room at 52% RH and 60% RH respectively, which are within the normal range of conditions for a museum. From these values, we have established, for the foreseen SH scenarios, the minimum distances between the panel *Back* and the *Wall* required to avoid damages in the painted wood panel (RH < 66% in the panel *Back*) and a minimum of 20 mm in the case there is condensation in the *Wall*.

In addition, factors that affect temperatures distribution and heat fluxes in the *channel* have been analysed for series of discrete values in the normal range of use. In the case of a wood panel 50 cm high, these have been discussed in further detail in order to help understand the heat transfer modes involved in the situation.

The study of these factors that modify temperature distribution in both sides of the *channel*, such as the distance to the ceiling, the *channel* width, the possible eddy near the *channel* inlet or outlet, can be used as guidelines to be applied to other similar situations like other model dimensions or possible obstacles to the air flux in the *Wall* over the wood panel. As examples of this, the effects of a cornice in the *Wall* or the effects of an aluminium protection layer in the *Back* of the panel or an EPS 60 mm isolation in the *Wall* have been quantified to compare with cases without these peculiarities.

This simulation carried out with a commercial 2 dimensional CFD allows to extend the standard 20 mm recommendations for the distances between cold exterior walls and wood panel paintings to a more detailed recommendations for paintings hung on the walls in similar conditions. These recommendations do not entail neither financial costs nor climate variations in the museums and exhibition rooms and conservators can easily apply them.

For the studied cases, Table 3, *Back* and *Wall* minimum temperatures and maximum RH values, and Additional file 1: Table S0, *Back* and *Wall* minimum temperatures at different heights, show how different these values are from those in the room or near other walls without panels hung on them. When conservators advise how to protect artistic wood panels hung in a poorly insulated peripheral wall, this fact shall be taken into account in addition to the recommendations in the EN standards [20–22].

Supplementary information

Supplementary information accompanies this paper at https://doi.org/10.1186/s40494-020-00376-1.

Additional file 1. Appendices.

Abbreviations

Wall: Inner face of the exterior wall; *Back*: Wood panel surface facing the *Wall*; *Front*: Wood panel surface facing the room; *channel*: Space between *Back* and *Wall*; *s*: Distance between wall and wood panel, *channel* width (m); *h*: Wood panel height (m); *s/h*: Ratio *channel* width/wood panel height; *ht*: Room height (m); *h1*: Distance from ceiling to wood panel upper border (m); $\boldsymbol{V}$: Velocity vector (m/s); $\mathbf{g}$: Gravity acceleration vector (m/s^2); $\boldsymbol{I}$: Buoyancy force vector $= -\mathbf{g}(\rho - \rho_0) = \rho_0 \mathbf{g} \beta (T - T_0)$ if $\rho \Delta T \ll 1$(N); *u*: Velocity horizontal component (m/s); *v*: Velocity vertical component (m/s); *x*: Horizontal coordinate m. Origin $x = 0$ at inner wall (left side). (In Fig. 1); *y*: Vertical coordinate m. Origin $y = 0$ at floor level. (In Fig. 1); *t*: Time (s); *P*: Pressure (Pa); *P'*: Pressure correction (Pa); *T*: Temperature (K); T_0: Reference temperature (K); *Tw*: Inner face temperature of the exterior wall (°C); *Tb*: Panel back Temperature (°C); *Ti*: Air *channel* inlet average temperature, vertical velocity weighed (°C); *Tu*: Turbulence intensity (%) = 100 RMS ($\mathbf{v}'/\mathbf{v}$), being $\mathbf{v}'$ fluctuations around the average velocity $\mathbf{v}$; *Turb*: Turbulence intensity average at *y* height, velocity weighed in 20 *channel* points at the same height (%); *H*: Enthalpy (J); *k*: Kinetic energy (m^2/s^2); k_h: Thermal conductivity (W/m K); c_p: Specific heat or Fluid heat capacity (J/kg K); δ_{ij}: Delta Kronecker ($i = j$, $\delta_{ij} = 1$ $i \neq j$, $\delta_{ij} = 0$); *Pr*: Prandtl number $Pr = \mu c_{p/} k_h$; φ: Generic property (velocity, temperature, and others); ρ: Density (kg/m^3); ρ_0: Reference density at T_0 (kg/m^3); *v*: Kinematic viscosity (m^2/s) $= \mu/\rho$; *vt*: Computed turbulent kinematic viscosity (m^2/s); *μ*: Dynamic viscosity N $s/m^2 = kg/(m\ s)$; μ_t: Turbulent viscosity N $s/m^2 = kg/(m\ s)$; λ: 2° coefficient of viscosity (Lamé); $\boldsymbol{\varepsilon}$: Turbulent kinetic energy dissipation rate (m^2/s^3); *ω*: Turbulent kinetic energy frequency dissipation (s^{-1}); *β*: Thermal expansion coefficient for perfect gases = 1/Tunits1/K; $\boldsymbol{\Gamma} = \mu + \mu_t$: units N $s/m^2 = kg/(m\ s)$; $\boldsymbol{\Gamma}_h$: Heat diffusion coefficient $= \boldsymbol{\Gamma}_h = \left(\frac{\mu}{Pr} + \frac{\mu_t}{Pr_t}\right) c_p$; $\boldsymbol{\Phi}$: Magnitude to be computed in each cell; RH: Relative humidity (%); SH: Specific or absolute humidity (kg_{water}/kg_{air}); α_1, α_2, β_1, β_2, σ_{k1}, σ_{k2}, $\sigma_{\omega1}$, $\sigma_{\omega2}$, σ_β, a_1: Turbulence closure constants; *Rr*: Normalized Euclidian residual (general); *Rm*: Normalized Euclidian residual for mass.

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Authors' contributions

The order of authors in this paper has been set applying the *sequence-determines-credit* approach (SDC). SF analyzed and interpreted the data resulting from software and carried out the calculation procedure and was a major contributor in writing the manuscript. GC-F designed the models and scenarios, provided conservation literature background and contributed in the discussion of results. CR-R collaborated in the analysis of data performing the graphics, images and tables and prepared the manuscript for final publication. MOF participated in the discussion of the results from the conservation point of view and contributed significantly in the writing process. All authors read and approved the final manuscript.

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Availability of data and materials

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Competing interests

The authors declare that they have no competing interests.

Author details

[1] Independent Researcher, Domènec Oristrell, 55. esc. C, 2º-3ª, 08173 Sant Cugat Del Vallès, Spain. [2] Arts and Conservation Department, Fine Arts Faculty, University of Barcelona, C/Pau Gargallo 4, 08028 Barcelona, Spain.

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