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Study of coupled transient radiation-natural convection heat transfer across rectangular cavities in the vicinity of low emissivity thin films for innovative building envelope applications

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Study of coupled transient radiation-natural convection heat transfer across rectangular cavities in the vicinity of low emissivity thin films for innovative building envelope applications

Graphical Abstract

2 new correlations for natural convection across air cavities in the vicinity of radiant barriers are proposed

Research Highlights

- Update of a transient heat transfer numerical model
- Study of natural convection within a closed air cavity
- Numerical evaluation of 8 different convection correlations
- Confrontation of simulation output to measurements
- Determination of 2 new correlations for modeling natural convection

Abstract

The coupled natural convection-radiation heat transfer across cavities is an extremely important research issue commonly encountered in many innovative building envelope applications. Extensive both experimental and numerical studies were conducted to analyze the natural convection flow coupled with radiation heat transfer across vertical and inclined rectangular cavities, proposing accurate empirical or regression correlations according to the particularities of the studied cavity problem. Nevertheless, there is a lack of correlations accurate and adapted enough to model coupled natural convection-radiation across cavities in the vicinity of low emissivity surfaces. The present work aims to cover this subject, through an exhaustive numerical analysis validated and evaluated enough with the use of experimental data. A numerical model is firstly updated to simulate infrared radiation and convective heat transfer across a sophisticated innovative building envelope prototype, while 8 different approximations have been implemented to the model in order to evaluate a variety of available correlations on the subject. Formerly the numerical approach presented here used available experimental data for the same prototype. The air's temperature experimental data served as input to the model boundaries in order to exactly impose the same initial conditions for obtaining the results. Furthermore, experimental data regarding the temperature on the sample's interfaces were used for validating the proposed theoretical model. Satisfactory enough agreements are observed between the theoretically simulated and experimentally measured temperatures and the model is afterwards evaluated to determine the most accurate approximation. Finally, the numerical study presented in this paper concluded in the

proposition of 2 new correlations to model natural convection across vertical cavities in the vicinity of low emissivity surfaces with the perspective to further support their implementation into Computational Fluid Dynamic (CFD) codes.

Keywords: natural convection, radiation, cavity, low emissivity surfaces, Milne-Eddington approximation, innovative building envelope

NOMENCLATURE

ρc_p	<i>volumetric heat capacity [J/(m³K)]</i>
c_p	<i>specific heat [J/(kg K)]</i>
k_g	<i>thermal conductivity of the air [W/(K m)]</i>
g	<i>gravitational acceleration [9.81 m/sec²]</i>
k	<i>thermal conductivity [W/K m]</i>
q	<i>thermal flux in W/m²</i>
q_r	<i>the total radiant flux in W/m² where $q_r = F^+ - F^-$</i>
F^+, F^-	<i>the incident radiant fluxes developing to the hot and the cold side of the material respectively [W/m²]</i>
R	<i>thermal Resistance [m² K/W]</i>
h_o, h_i	<i>equivalent heat transfer coefficients including convection and radiation heat transfer on the boundaries [W/(K m²)]</i>
f	<i>solid fraction volume</i>
e	<i>emissivity</i>

e_n	<i>effective emissivity</i>
T	<i>temperature [K]</i>
S	<i>surface of the composite wall [m²]</i>
t	<i>time [sec]</i>
l	<i>length of each section [m]</i>
x	<i>the spatial coordinate through the thickness of the composite wall [m]</i>
W	<i>width of the cavity [m]</i>
H	<i>height of the cavity [m]</i>
L	<i>length of the cavity [m]</i>
Gr	<i>Grashof number</i>
Nu	<i>Nusselt number</i>
Pr	<i>Prandtl number</i>
Ra	<i>Rayleigh number</i>

Greek symbols

σ	<i>Stefan-Boltzmann's constant ($5,670373 \cdot 10^{-8} \text{ W m}^{-2} \text{ K}^{-4}$)</i>
σ_{dev}	<i>standard deviation</i>
ν	<i>kinematic viscosity of the air [m²/sec]</i>
μ	<i>dynamic viscosity of the air [kg/(m·sec)]</i>
ρ	<i>density of the air [kg/m³]</i>
β	<i>absorption coefficient [m⁻¹]</i>
Δx	<i>distances between the interfaces of the finite volume [m]</i>
ΔT	<i>temperature difference on composite wall's surface [K]</i>

δx distances between the grid points [m]

φ inclination angle

ω albedo

Subscripts

conv convection

cond conduction

rad radiation

eff effective

m mean (used as a subscript for *T*)

g gas (used as a subscript for thermal conductivity *k*)

1. Introduction

Also known as global or planet warming, the concept of climate change has been introduced to illustrate the atmosphere's and ocean's average temperature increase at global level, for the last sixty years. In this context, since the building sector is assumed to be responsible for about 42% of final energy consumption, 35% of greenhouse gas emissions and 50% of material resources consumption the European Union (EU) established an institutional framework through the form of guidelines and directives that impose the adaptation of the entire building industry to new high performance targets [1], [2], [3], [4]. For this reason the label of nearly zero energy building (nZEB) is introduced to impose a high performance level for all new constructions from 2020 [4]. However, to achieve such high performance levels, researchers and thermal engineers deal with a

multifaceted problem [1], [5], [6] while solutions must anticipate the heat losses through the building envelope, integrate renewable energy resources and optimise the building systems' control and operation.

Consequently, it has been revealed that innovative building envelope design can help reduce operating costs [7]. Generally, statistics show that the building envelope (walls, roofs, windows) is responsible for approximately 25% of all building energy use [8]. Nevertheless even if the impact of the envelope on the overall building energy consumption seems limited, it has been reported that the building envelope and its components can influence up to 42% of residential use and 57% of commercial use [9]. Ergo, the development of a variety of original sophisticated materials, envelope prototypes and the interdisciplinary dimension of the produced research that covers the fields of nanotechnology, material science and architecture is a positive dimension that improves the quality of the applied thermal engineering research on the building scale. Withal there is a need to adapt the description of physical phenomena in specific configurations and implementations taking in care every prototype's particularities. This is why the last 5 years there are many research works that study in a theoretical and experimental manner the heat transfer phenomena through specific innovative envelope configurations aiming to adapt our past knowledge in the specific case studies induced by the use of innovative materials and/or sophisticated envelope configurations and prototypes.

1.1. General overview of innovative building envelope design: radiant barriers, sophisticated thin films, air layer involved envelopes (ALIEs), and functionally graded materials (FGMs)

The study conducted in the framework of this research article is fruit of a general interest regarding the conception and development of original sophisticated enough innovative building envelope prototypes with a very high insulating performance [10], [11], [12], [13]. In the past, Multilayer Thermal Insulation for building applications has been studied in an exhaustive both experimentally and numerically manner. Past research revealed, that the use of reflective materials in building applications consists an attractive option to refining buildings insulation through the reduction of the buildings' energy consumption due to the integration within the envelope of low emissivity surfaces for reducing radiant heat transfer [13]. The integration of low emissivity surfaces within the envelope makes them to work as radiant barriers that inhibit heat transfer by thermal radiation. Similarly to us, Hernández-Pérez et al [14] reported that reflective materials applied to opaque building components produce a real benefit in terms of thermal comfort and energy savings while they stay cooler than standard materials under the same conditions because of their optical properties. Additionally, Shen et al [15] experimentally investigated the efficiency of low emissivity coatings assessing the impact of different envelope material properties equipped with different coatings. Their results showed that, depending on location, season and orientation, exterior and interior surface temperatures can be respectively reduced by up to 20°C and 4.7°C using different coatings while the maximum reduction in globe temperature and mean radiant temperature was 2.3°C and

3.7°C in that order. Besides, the overall conclusion is that composite multilayer materials including layers of metallised polyester film, high strength polyester coated foils spaced with polyester wadding and closed cell foams have been used inasmuch as low emissivity surfaces are very effective in reducing radiant heat transfer [13].

On the other hand, several experimental studies revealed that there are many reasons that this kind of reflective insulation should be continuously evaluated since many samples do not have declared values due to protective foil layers applied on their surfaces. For example, Čekon et al [16] analysed materials of low-emissivity functions hollow microspheres coatings and confronted with standard facing coatings of acrylic base as a reference whereas their measured data demonstrate no differences in thermodynamic performance of the studied coating in regard of reducing heating loads especially for nocturnal conditions as well as their thermal emissivity parameters in comparison with the standard façade coating. However, Joudi et al [17] reported that we can actually overcome this kind of problems since *“the development in the coil-coating steel industries has now made it possible to allocate correct optical properties for steel clad buildings with improved thermal performance”*. Additionally, new low emissivity materials such as aluminum/high-density polyethylene (HDPE) and other functionally graded materials (FGM) [18], have been fabricated as essential components of a multifunctional building envelopes for high performance of energy efficiency and sustainability preventing air, radiation and bulk water intrusion into the wall system [19].

In every case, the general conclusion remains that the existence of reflective foils within the envelope may potentially increase the overall's envelope thermal inertia when radiant barriers are placed in the vicinity of materials with an augmented thermal mass (concrete, etc) [10], and consequently augment the time lag of the overall envelope. Conjointly, from an architectural point of view there is a need to reduce the envelope's thickness and at the same time augment the thermal mass of light structures. To anticipate to the lack of thermal inertia for such kind of applications, there exist materials that can significantly augment the thermal mass of the envelope at nearly isothermal conditions while their width is very low: the phase change materials (PCM) [20], [21], [22], [23], [24], [25]. The direct perspective of the work presented in this paper is to obtain a numerical tool that will provide assistance in conceiving and composing innovative original prototypes that include air cavities juxtaposed to PCM combined with low emissivity surfaces and classic fibrous or non-fibrous materials. In general, this kind of air, water and radiant barrier products helps enable insulation and/or PCM to perform to their installed R-values while for insulation materials embedded within the envelope it's essential that the air surrounding the envelope solution remain free of movement. Ergo, from an applied thermal engineering point of view, in most cases we deal with an embedded closed air cavity within a composite envelope.

Thence, the study presented here is just a part and the first step of an entire research axis that deals with the sophisticated conception and integration of a variety of innovative materials within building envelopes with the aim to develop original innovative envelope prototypes. Since air cavities in the vicinity of low emissivity surfaces are geometries

currently presented within such prototypes, this paper only focuses on the air cavity heat transfer problem and presents a numerical investigation coupled with experimental measurements with the scope to identify, develop and propose new correlations that meet much better the requirements of this study and update enough the numerical model that will be used in order to accelerate in the future the evaluation stage of the newly conceived prototypes forasmuch as the existence of reflective surfaces within the prototype will impose the existence of one or ideally two air cavities.

Consequently, far from being a singular case study, this paper focuses on the envelope heat transfer problems across vertical low width-to-height ratio air cavities in the vicinity of low emissivity surfaces integrated in composite innovative envelope design. Suffices it to notify that the work presented here can be generalized since air layer involved envelopes (ALIEs) have gained considerable popularity in modern building design and construction, owing to their great potential in improving the building thermal performance, while the air layer mainly functions as an extra insulation layer forming a cavity [26].

Conclusively, the work presented here treats a very important issue and the findings of this work can be generalized since an intensive market's need for new facade solutions and innovative envelope design has been appeared. Thus, because of the existence of the strict institutional context briefly presented in the introduction chapter (and especially the nZEB labelling), thermal engineers are working intensively these last years on the

development and implementation of new technologies with the aim to conceive and design innovative high performance materials in order to reduce the envelope energy impacts by 80%. Since the needs of the building professionals were not limited on energy efficiency, the research on innovative envelopes has been extended to the re-invention of new functionalities with the aim to make them less costly to maintain, smarter, eco-friendly and healthier [27], implicating also air cavities within the proposed composite envelope prototypes for performance and eco-friendly purposes.

Thusly, the need to produce innovative building envelope prototypes opened building research to other fields in an interdisciplinary manner with the use of materials with nano-properties that have never been optimized for use in the building sector before. In brief, after reviewing the literature on scale in order to sketch the international research framework of the study presented in this paper, someone can conclusively assume that the contemporary produced research on this subject can be listed as follows and is developed on 3 distinct complementary axes:

- **Axis 1:** Development of sophisticated insulating products (for example multilayer thermal insulation [12], radiant barriers [13], vacuum insulation panels [28], cementitious reduced thickness lightweight and non-flammable bricks [8], development of sophisticated timberfabric structures [29]) or development of sophisticated implementation of existing materials in order to augment envelope's insulation capacity [10], [30] and thermal inertia [31], [32].
- **Axis 2:** Deposition of high rate thin film coatings on building components for efficient glazings and building surfaces (for example use of carbon films

produced by high power impulse magnetron sputtering [33], use of novel high-speed InP-based photodiodes [34], [35])

- **Axis 3:** Integration of smart coatings for efficient envelope surfaces through the use of nanotechnology achievements [36] in order to enlarge the property spectrum of façade elements with the aims to control moisture build-up in the insulation by nano clays, increase heat reflectivity, protect from fire, etc (for example use of dynamic coatings with variable electro-chromic, thermo-chromic, photo-chromic optical properties [37], [38], [39] on windows, window frames and other envelope surfaces).

Furthermore, most of the aforementioned materials are juxtaposed to integrated within composite envelope applications air cavities while a lot of them present low emissivity characteristics blocking the propagation of infrared heat transfer that is emitted from the high inertia buildings elements. Thusly, modelling heat transfer within the cavities needs a special care since the existence of smart coatings in the vicinity of the cavity may alternate the way that heat propagates due to the sophisticated thermo-physical characteristics of the cavity (for example: as we mentioned above low emissivity of the surrounding surfaces). Correspondingly, this why from now on, the study of heat transfer within air cavities in a transient regime -which is the main topic of the present paper- requires a specific analysis of the identified interactive phenomena adapted to the geometrical and physical characteristics of the investigated configuration and the combination of a variety of modelling techniques and correlations to illustrate in an optimal manner the propagation of heat.

1.2. Aims, scope and novelty of the present research article: determination of two new correlations for the natural convection problem within low width-to height air cavities in the vicinity of low emissivity surfaces

Thereupon, the main objective of this paper is to discover the most robust and accurate correlation among a selection of correlations (presented in this paper) to describe transient coupled radiation-convection heat transfer across rectangular air cavities in the vicinity of low emissivity surfaces. Before integrating the phase change problem, the most accurate manner to model transient coupled natural convection-radiation heat transfer across cavities in the vicinity of low emissivity surfaces needed to be identified in order to update the existing modelling studies. Even though many researchers were worked on convective heat transfer within cavities, very few researchers were reported about the specific problem of low width-to-height ratio air cavities in the vicinity of low emissivity surfaces. The existence of low emissivity surfaces in the boundaries of the cavity influences heat transfer propagation within the cavity and is the reason that a non updated classic heat transfer model lacks to produce robust enough results over time. Additionally, there is a lack of information regarding the existing correlations since their valid period depends on the particularity of the studied configuration.

In the past a detailed numerical approach to describe heat transfer within multilayer thermal insulation that includes a number of thin silver coated films has been developed

and presented [12], [40], [41], [13], [42], [10]. This numerical model is accurate enough to simulate this kind of applications, however it was discovered that when we simulate more than 48 hours in a transient regime the model has the tendency to slightly overestimate transient heat transfer within the studied prototype. Moreover, after implementation of the natural coupled convection-radiation correlations reported in this paper (see subchapter 4) in my model it was found that the valid period for the specific studied configuration don't exceed 12-24 hours (depending on the used approximation).

Since the studied prototype included two air gaps in the vicinity of the low emissivity foils it was considered that this lack of precision is due to the classic coupled natural convection-radiation correlations that were used for the air cavities in the vicinity of the low emissivity surfaces. For this reason, the present paper reports a brief literature review on the subject in order to present a detailed parametric study where a variety of approximations specifically for the coupled transient radiation natural convection heat transfer within the cavities have been modelled and implemented within the aforementioned numerical model. As we will discover in the following section, many researchers proposed correlations adapted to specific cavity problems [43]. Though there are similar works, in the present work a specific physical problem has been studied since the air cavity is juxtaposed to low emissivity reflecting surfaces. Finally, this study resulted in the development of two new correlations for this kind of specific air cavities that are proposed in the end of this paper after being validated with experimental data. The development of these correlations improved the robustness and the accuracy of the entire numerical model; notwithstanding the one of the updated correlations proposed in

this paper have a time valid period of at least 240 hours (10 days) as we will see in the results section.

2. State of the art and previous research

Natural convection heat transfer is an effective way of heat transport within innovative wall complexes, and in general in the vicinity of the building envelope. Furthermore, enclosures and cavities are fitted to specific geometries that can be found in building envelope applications while special care is needed to describe the participating heat transfer phenomena (convection and radiation). Albeit, the problem of natural convection in the vicinity of building envelopes is widely studied in a general manner in the literature ([27], [43], [44], [32]), there is always a need to contribute in the study of natural convection coupled with radiation inside cavities and enclosures. Every specific cavity problem impose special care regarding the description of coupled convection and radiation heat transfer, due to the state and the thermo-physical characteristics of the enclosing surfaces, the imposed gradient temperature, the eventual existence of a heat source inside the cavity, the geometry of the cavity and the nature of the investigated fluid. Thus, studying natural convection inside cavities and enclosures for building applications may generally need a multiscale research, since enclosures can be studied in nano [45], [30] micro [31], and meso-scale [46], [47] on a variety of sophisticated envelope configurations. Whereas, the heat transfer in cavities has been extensively studied in many thermal engineering applications -to mention just one example: in solar thermal receivers ([48], [49])- there are only few works that focus on the study of natural

convection in the vicinity of innovative constructive materials within building envelope complexes with the aim to investigate the thermal performance of the entire envelope system.

Henceforward, at this subchapter the literature on scale regarding theoretical, numerical and experimental investigations that focus on natural convection study is briefly reviewed, in order to explicitly position the subject of the present paper inside its scientific context.

Natural convection is one of the primary mechanisms of heat transfer in the vicinity of building envelopes while heat transfer rates depend on temperature differences between the surface and the environment and can be enhanced using natural or mechanical ventilation [43]. Dascalaki et al. [43] published a detailed review that explicitly presents a variety of correlations available in the literature for estimating the natural convection heat transfer coefficient from vertical and horizontal surfaces for building applications. Likewise, they presented information that has been exclusively taken from experiments of unconfined, natural convection flow from horizontal and vertical surfaces [43]. Besides, Palyvos [27], focused on external building envelope surfaces and reviewed the most used traditional and contemporary external convection coefficients that include also the wind velocity parameter. However, the aforementioned research works underline that an obvious lack of generality of the existing wind convection coefficient correlations exists and presents a challenge for future applied research [27], [43]. Furthermore, these

correlations even if they are principally found to describe natural convection in the vicinity of wall complexes, many times can also be applied to enclosure problems under specific conditions [27].

Withal, after reviewing the updated literature on scale, we can admit that the common general approach in natural convection problems is to first determine an applicable expression of Nusselt number for the given configuration via the other dimensionless characteristics and consequently determine the adequate to our problem convective heat transfer coefficient (h) or equivalent effective thermal conductivity coefficient (k_{eff}). In the past, as it resorts from the aforementioned reviews [43], [27] the majority of the available correlations have resulted from experimental studies while only a few among them have been developed by numerical analysis. As we know from theory one can derive a correlation for the Nusselt number measuring temperature and velocity variation data. Nonetheless, such correlations have validity only under the specific conditions of each experiment or the inputs and the boundary conditions of the numerical study. Thus, we cannot generalise the application of these empirical correlations since we are obliged to examine their adequacy before applying them under different conditions. Howbeit, contemporary research works on this subject have the tendency to study the problem only via a numerical investigation (coupling also natural convection and infrared radiation) since there is widespread acceptance that nowadays the existing CFD methods [32], techniques [50], [45] and tools are accurate and validated enough and the development of powerful hardware with a high calculation potential [51] can help us to avoid time and energy consuming high cost experimental sequences.

Thus, regarding original recent research works that focus on the natural convection study in the vicinity of building envelopes it was selected to mention here two characteristic examples that employ a pure numerical approach and significantly influenced the present work. Kîs et al [47] proposed new wall functions for natural convection including the turbulent shear stress in the correlations and compared these correlations to the natural convection wall functions used with the further aim to support their implementation into Computational Fluid Dynamic (CFD) codes. Completely, Bykalyuk et al [32], recently obtained via 2-D sophisticated CFD simulations quantitative and qualitative conclusions regarding the thermal behaviour of a scale $1:1$ thermal conductive plate placed vertically in a closed cavity by evaluating the kinetic and temperature boundary layers in the vicinity of the studied plate. The main contribution of this work is that it proved that the local convective heat transfer coefficient decreases as we vertically climb a plate, while the average value of the convective heat transfer coefficient increases with time and reaches an average value only at steady state and hence generalized correlations may generate systematic errors [32].

Moreover, in 1985, Babus'Haq et al [52] published an original research article that focus on natural convection across cavities, where experimental measurements and theoretical predictions of steady-state heat transfer to or from horizontal single or double pipelines enclosed in horizontal circular or rectangular enclosures have been collated. Even if this research article is limited to steady state, this work is very important since it constitute the first report that concentrates the research findings regarding natural convection across

cavities in a synthetic explicit manner. The experimental studies on natural convective heat transfer across horizontal circular or rectangular enclosures around single or double pipe systems presented in this paper suggested that the steady-state rate of free-convective heat transfer is a function of Grashof number (Gr), Prandtl number (Pr), and some other geometrical parameters in addition to the diameter ratio, linking the convective heat transfer coefficient to the dimensionless numbers of free convection and to the specific geometry [52]. Although the optimal configurations of the pipelines to achieve maximum thermal resistances of the air-filled cavities are identified, they underlined the necessity to fully establish the exact nature of the empirical correlation that includes the geometrical characteristics of the cavity [52]. From 1985 till today many research works studied numerically and experimentally natural convective heat transfer coefficient in one, two and three dimensional enclosures, while a variety of accurate correlations are reviewed by Khalifa [46]. However there exists a lack of review articles that especially focus in natural convection within air cavities.

Contrariwise, the contemporary numerical era of applied thermal engineering conducted to a remarkable production of sophisticated correlations adapted to specific geometries of enclosures under a variety of initial and boundary conditions. These works are in accordance with classic experimental studies, enriched the accuracy of numerical models and updated our literature by adapting each correlation to a specific problematic for a variety of applications however the time validity period of the proposed correlations is not mentioned. For example, Baïri [53] studied transient thermal characteristics of airborne electronic equipment with discrete hot bands in closed air-filled square cavities

with the aim to correct sizing of the airborne electronic circuits. Mezrhab et al [54] developed a numerical model to study combined natural-convection and radiation heat-transfer in a cavity having a square body at its centre and found that the radiation exchange homogenizes the temperature inside the cavity and produces an increase in the average Nusselt number, while the average Nusselt-number increases with increasing surface emissivity, especially at high Rayleigh numbers. Mezrhab et al [54] results are in accordance to experimental results presented in a classic reference on this field published in 1989 by Rahman et al [55], where their measurements of local heat transfer coefficients in correlation with flow visualization experiments for orthogonal mixed convection in a partial enclosure with uniformly heated side walls and adiabatic top and bottom walls are presented and discussed. Conjointly, Li and Tong [56] after reviewing studies about natural convective heat transfer coefficients on surfaces in two- and three-dimensional enclosures, conducted a detailed CFD analysis and updated a classic correlation to describe natural convective heat transfer in the inclined rectangular cavities with low width-to-height ratios.

Besides, there is plenty of literature on convection heat transfer coefficients; however there is not so much literature that couples natural convection with radiation processes. Reciprocally, there are two very important recent works that manage to study in a consolidated manner natural convection and thermal radiation within cavities.

Montiel Gonzalez et al [57] published a detailed numerical approach on natural

convection and surface thermal radiation in an open cavity receiver considering large temperature differences and variable fluid properties for Rayleigh number (Ra) values in the range of 10^4 – 10^6 . To simultaneously treat natural convection and thermal radiation they introduced the concept of two distinguished *Nusselt* numbers respectively for convection and radiation. Hence, they initially calculated the average convective *Nusselt* number within the cavity by integrating the local *Nusselt* number (obtained employing the Fourier equation) over the length of the studied isothermal wall. Then they calculated a radiative *Nusselt* number that they define as the ratio between the radiative heat flux at the hot wall and the heat flux due to conduction only. Afterwards, they calculated the average radiative *Nusselt* number to finally identify the total average *Nusselt* number by summing the average convective *Nusselt* number and the average radiative *Nusselt* number. Finally, as a summary of their calculations, they proposed a correlation for the total average *Nusselt* number that includes natural convection and thermal radiation and is adequate for laminar convection within an open cavity. Their correlation is radiation sensitive and produces accurate results especially for large temperature differences where the radiative heat transfer is more important than convective heat transfer.

Similarly, Shati et al [58] employed the concept of a new dimensionless group RCn which is the ratio of convection heat transfer to surface radiation via average convective (Nu_c) and radiative (Nu_r) *Nusselt* numbers to study the problem of turbulent natural convection with the interaction of surface thermal radiation in square and rectangular enclosures. They conducted at the same time a parametric study analysing configurations with different fluids, enclosure sizes, aspect ratios and cold and hot boundary

temperatures under variable fluid properties. Finally they concluded by providing a palette of correlations to predict the relation between convection and radiation heat transfer as a function of temperature ratio in the rectangular enclosure determining also a variety of average deviations for each imposed regime.

Finally, many researchers instead of employing correlations, they used the Lattice Boltzmann Method LBM for modelling natural convection within cavities [50], [45], [31], [59], [60], [61]. Instead of numerically solving the conservation equations of macroscopic properties (i.e., mass, momentum, and energy equations), in the aforementioned works we can find LBM applications where the fluid is modelled via fictive particles, and such particles perform consecutive propagation and collision processes over a discrete lattice mesh [62], [63], [64], [65].

As to conclude this short review, we can assume that there is a rich literature on available techniques and robust validated correlations to model natural convection within enclosures and cavities. Even if most of the available correlations have been extracted from experiments in controlled environments (and non in situ measurements) in small and simplified test enclosures and laboratory test cells or limited numerical studies, it has been observed that there is a good agreement between the available correlations [43]. Additionally, the contemporary numerical era of applied thermal engineering enriched the literature on scale and updated the available correlations. Moreover, we have obtained robust enough correlations that we can apply in sophisticated configurations for building

applications to describe natural convection within enclosures and cavities taking under consideration the inward surface characteristics, the dimension and geometry of the cavity and the fluid's characteristics. However, there is a lack of literature regarding adequate correlations to model coupled laminar natural convection-radiation across closed cavities in the vicinity of low emissivity surfaces and thin radiant barrier films while the time validity period of the proposed correlations is rarely mentioned and this is the main novelty of the present research paper. Thus, before identifying which correlation fits more to the specific physical and geometrical problem that is studied here a number of simulations had to be performed in order to confront the simulated data to experimental data. Furthermore, in this paper two new updated convective heat transfer correlations that produced results that perfectly fitted to available experimental data for the prototype under consideration and can be adapted for cavities in the vicinity of low emissivity surfaces are proposed. Hereinafter, the general numerical formulation of the problem will be presented.

3. Materials, methods and main numerical formulation of the problem

The main tool employed in the framework of the research presented in the present paper is an original “*in-house*” numerical model that is based on detailed analytical and numerical modeling of the heat transfer phenomena that occur in a precise sophisticated composite wall prototype that includes a radiant barrier composed by low emissive thin films and optically thin insulation materials and put between two air cavities (Figure 1). The sample employed to conduct the present study follows the general pattern of such

kind of applications shown in figure 1 (for further details on precise geometry of the sample see also figure 3), where two non-ventilated air cavities are placed between a wood panel and low emissivity films in the vicinity of a hot and a cold ambiance. The aim of such applications is to block heat transfer from the hot to the cold ambiance.

Figure 1: This paper focus on the transient coupled radiation-natural convection heat transfer across cavities in the vicinity of low emissivity thin films employed on innovative building envelope applications. The detail A is presented in figure 2 while the control volume formulation for the spatial discretization of the numerical solution is discussed below.

Throughout our past research works, we managed to merge classic and novel modeling techniques in common software in order to obtain an easy adaptive to a variety of configurations, formal solutions and sophisticated materials numerical tool, with the objective to simulate transient heat transfer through originally conceived innovative

building envelope prototypes (more details on different configurations that have been simulated with success are provided here: [12], [13], [42], [40], [11]).

The goal of our past research was to evaluate this numerical model in order to assure its accuracy and robustness and hence accelerate research on composite wall's performance forecasting by avoiding experimental tests that need expensive time and energy consuming calibration operations. However, in the past we didn't focus the study on the cavity natural convection-radiation issues since the domain of interest was mainly the heat transfer phenomena within multilayer thermal insulation (MTI). However, the flexibility of this numerical model was the reason this model has been used for a variety of envelope applications [42], [10], since the geometry of the sample can be easily modified without needing to transform the spatio-temporal discretization while the same accuracy of the outputs is conserved.

Hence, the part of the numerical model regarding the insulation cluster of the wall prototype as well as the boundary conditions are not modified, while this paper presents the part of the code regarding coupled radiation-natural convection within the air cavities that is explicitly updated: a total of 8 distinct different convection-radiation schemes regarding natural convection within the juxtaposed cavities have been inserted and evaluated with the aim to choose the most robust and accurate correlations.

Regarding the transient heat transfer modeling within the insulation component of the envelope [66], [67], [68] the Milne-Eddington approximation is always employed to

describe in detail the radiant heat transfer phenomena through the low emissivity thin films of the prototype [69], [70], [71], [72], [73], [74], [75], [76].

Furthermore in order to calculate the attenuation of the forward and backward radiation fluxes the albedo of scattering is took into consideration using the absorption coefficient β (function of the solid fraction volume f), the emissivity of the fibrous material e and the radius of the fiber r_f as it is described in detail elsewhere [69], [13], [41], [12], [11], [40], [10], [42], [75].

By the bye, the spatial discretization follows a control volume pattern since this kind of discretization is more adequate to represent the physical dimension of the studied problem according to Patankar [77] and Anderson [78]. In this paper heat transfer within air cavities in the vicinity of low emissivity surfaces is studied. Hence, the existence of multiple radiant barriers within the insulation material [66], [67], [68] appointed to also divide the simulation domain in radiation blocks (see Figure 2), in order to impose the needed local boundary conditions in the limits of each radiation block for the Milne-Eddington approximation [73], [69], [75], [13], [42] as we can see on Figure 2.

Figure 2: Detail A (see figure 1) of the composite wall under study including one cavity in the vicinity of the hot boundary, both optically thin fibrous (with different radius of fibers) and non-fibrous materials as well as low emissivity surfaces within the complex and especially in the vicinity of the cavities. The entire complex is divided in N control volumes and N radiation sections, for describing numerically the heat transfer phenomena.

Another question raised in the past consisted on how to calculate the thermal conductivity in the conjunction of the different materials. This problem was overthrown by calculating the thermal conductivity in the interface of each control volume by merging Daryabeigi's [75] and Patankar's model [77] that were revealed to be adequate enough to such kind of applications and provided results that perfectly fitted to the experimental sequences [12], [13].

Finally, since the aim of this paper is to confront the results to experimental sequences in order to evaluate the accuracy of the examined convection schemes, h_0 and h_i Newtonian

boundary conditions have been defined using the concept of equivalent heat transfer coefficients respectively including convective and radiant heat transfer in both hot and cold limits of the composite complex (Figure 3), while by 0 is denoted the outer limit of the complex (warm limit) and by L the second outer limit (cold limit) (Figure 3). In the boundaries the following classic equations to calculate the adimensionnal convection characteristics [79], [74], [76] have been used :

$$Nu = \frac{h_{\text{conv}} \cdot L_c}{k_g} \Rightarrow h_{\text{conv}} = \frac{k_g \cdot Nu}{L_c} \quad (1)$$

$$Gr = \frac{g \cdot \frac{1}{T_m} \cdot (|T_B - T_\infty|) \cdot L_c^3}{\nu^2} \quad (2)$$

$$Pr = \frac{\mu \cdot c_{p_{\text{air}}}}{k_g} \quad (3)$$

$$Ra = Gr \cdot Pr \quad (4)$$

where :

$$T_B = T_{B_1} \text{ or } T_{B_{N-1}}$$

$$T_\infty = T_1 \text{ or } T_N$$

$$T_m = \frac{T_B + T_\infty}{2}$$

T_m is the average temperature in the boundary layer (film temperature) and L_c is the characteristic length of the geometry.

Figure 3 : A schematic representation of the model domain, the sample's geometry and characteristics and its boundary conditions (samples dimension: length 1.91m x height 1.96m).

Then, according to the flow the following equations have been declined. For laminar flow we have [13]:

$$(Ra < 10^9): Nu = 0.68 + \left(\frac{0.670 \cdot Ra^{1/4}}{\left(1 + \left(\frac{0.492}{Pr} \right)^{9/16} \right)^{4/9}} \right) \quad (5)$$

while for turbulent flow we have [13]:

$$(Ra > 10^9): Nu = \left\{ 0.825 + \frac{0.387 \cdot Ra^{1/6}}{\left(1 + \left(\frac{0.492}{Pr} \right)^{9/16} \right)^{8/27}} \right\}^2 \quad (6)$$

Based on the aforementioned calculations, the coefficients h_{conv} in the boundaries are calculated from the combination of equations (1), (5) or (6), while [11]:

$$h_{rad} = \frac{\sigma \cdot e_n \cdot (T_B^4 - T_\infty^4)}{(T_B - T_\infty)} \quad (7)$$

and finally

$$h_0 \text{ or } h_i = h_{conv} + h_{rad} \quad (8)$$

4. Theory and Calculation: different approaches and approximations to model coupled radiation-natural convection across the two cavities of the prototype

Let's focus now on the modelization of natural convection and radiation within the cavities, which is the main novel production and originality of the present paper. After reviewing the literature this paper focus on the following 8 approaches that were revealed to be the most accurate after confrontation of the simulated data to the experimental sequences. Furthermore, these correlations have been adapted to the studied configuration and to the physical particularities of this problem (the existence of a low emissivity thin

film in the vicinity of the cavities) in order to reach a maximum accuracy and propose updated correlations at the discussion section of this paper. Ergo, regarding the modelization of the coupled radiation-natural convection heat transfer the following 8 different approaches have been simulated.

The thermal properties of the air are calculated for all approaches employing the following correlations proposed by the Worcester Polytechnic Institute [80] as follows:

$$v(T) = -1.1555 \cdot 10^{-14} \cdot T_m^3 + 9.5728 \cdot 10^{-11} \cdot T_m^2 + 3.7604 \cdot 10^{-8} \cdot T_m - 3.448410^{-6} \quad (9)$$

$$\rho(T) = 360.77819 \cdot T_m^{-1.00336} \quad (10)$$

$$c_{p_{air}}(T) = 1.9327 \cdot 10^{-10} \cdot T_m^4 - 7.9999 \cdot 10^{-7} \cdot T_m^3 + 1.1407 \cdot 10^{-3} \cdot T_m^2 - 4.4890 \cdot 10^{-1} \cdot T_m + 1.0575 \cdot 10^3 \quad (11)$$

$$\mu(T_m) = v(T_m) \cdot \rho(T_m) \quad (12)$$

4.1. Approximation 1 (*Simulation N°1*)

This approach approximates that for the equivalent thermal conductivity of the air we have:

$$k_{eff} = k_{conv} + k_{rad} \quad (13)$$

where k_{conv} and k_{rad} respectively represent the convective and radiative heat transfer within the cavity. Thus, before calculating the Nusselt number, we need to calculate the Grashof, Prandtl and Rayleigh number as follows [79]:

$$Gr = \frac{g \cdot \frac{1}{T_m} \cdot (|T_1 - T_2|) \cdot W^3}{\nu^2} \quad (14)$$

$$Pr = \frac{\mu \cdot c_{p,air}}{k_g} \quad (15)$$

$$Ra = Gr \cdot Pr \quad (16)$$

where T_1 and T_2 are the surface temperatures in the limits of the cavity, T_m is the mean air temperature within the enclosure and W is the width of the cavity, while k_{cond} (or k_g) can be calculated as follows according to [80]:

$$k_g = 1.5207 \cdot 10^{-11} \cdot T_m^3 - 4.8574 \cdot 10^{-8} \cdot T_m^2 + 1.018 \cdot 10^{-4} \cdot T_m - 3.9333 \cdot 10^{-4} \quad (17)$$

Then the Ra number is employed to calculate three times the Nu as follows in order to select the maximum value according to Kakatsios [79]:

$$Nu(1) = 0.0605 \cdot Ra^{\frac{1}{3}} \quad (18)$$

$$Nu(2) = \left\{ 1 + \left[\frac{0.104 \cdot Ra^{0.293}}{1 + \left(\frac{6310}{Ra} \right)^{1.36}} \right]^3 \right\}^{\frac{1}{3}} \quad (19)$$

$$Nu(3) = 0.242 \left(\frac{Ra}{\frac{H}{W}} \right)^{0.272} \quad (20)$$

where H is the height of the cavity. Then the “*convective component*” of the air’s equivalent thermal conductivity that is an input in the equation (13) is calculated using the greater value of Nu that resulted from the equations (18)-(20) as follows [74]:

$$k_{\text{conv}} = k_g \cdot \text{Nu} \quad (21)$$

Finally, the “*radiative component*” of the air’s equivalent thermal conductivity is calculated as follows [74]:

$$k_{\text{rad}} = \frac{\sigma \cdot e_n \cdot W \cdot (T_1^4 - T_2^4)}{(T_1 - T_2)} \quad (22)$$

where e_n is the effective emissivity and is calculated as follows in correlation of the emissivities of the cavity’s vertical surfaces:

$$e_n = \frac{1}{\left(\frac{1}{e_1} + \frac{1}{e_2} - 1 \right)} \quad (23)$$

4.2. Approximation 2 (*Simulation N°2*)

The natural convection is modelled as above (approximation 1) via the equations (13)-(21). Since the width of the cavities is very small (2 cm) we can employ the following equation according to Duffie et al [81] in order to calculate the “*radiative component*” of the air’s equivalent thermal conductivity :

$$k_{\text{rad}} = 4 \cdot \sigma \cdot e_n \cdot W \cdot (T_m)^3 \quad (24)$$

where e_n is calculated using the equation (23).

4.3. Approximation 3 (*Simulation N°3*)

Firstly, the k_g is calculated using the equation (17) mentioned above. Then according to

Schuh [82] and Rotari [83], we have to conduct the following analysis and define the limits between “*pseudo-conduction*” and natural convection within the cavity. Thus, we have to initially calculate the Ra number as follows:

$$Ra = 500 \cdot H/W \quad (25)$$

Then if $Ra \leq 2 \cdot 10^3$ we have “*pseudo-conduction*” so $k_{conv}=k_g$. In our case we are in the following range of Ra values $2 \cdot 10^3 < Ra \leq 2 \cdot 10^5$ and according to De Graaf et al [84], Schuh [82] and Rotari [83] the following correlation is employed to calculate the “*convective component*” of the air’s effective thermal conductivity [84] :

$$k_{conv} = 0.197 \cdot Ra^{1/4} \left(\frac{H}{W} \right)^{-1/9} \cdot k_g \quad (26)$$

Then in order to calculate the “*radiative component*” of the air’s equivalent thermal conductivity the equation (24) of the aforementioned 2nd approximation is used.

4.4. Approximation 4 (Simulation N°4)

In this approximation, the aforementioned approximation 3 for the calculation of the “*convective component*” of the air’s equivalent thermal conductivity, and the approximation 1 for the calculation of the “*radiative component*” have been combined. Hence the equations (22) and (26) have been respectively employed.

4.5. Approximation 5 (Simulation N°5)

In this approximation regarding the “*convective component*” of the air’s equivalent

thermal conductivity the following correlations proposed by Isachenko et al [85], Prahlad et al [86] and Mikheev [87] are used after calculating the Ra number :

$$\begin{aligned} \text{if } Ra \leq 1850 \quad k_{\text{conv}} &= k_g \\ \text{if } 1850 \leq Ra \leq 10^6 \quad k_{\text{conv}} &= k_g \cdot 0.105 \cdot Ra^{0.3} \\ \text{if } 10^6 \leq Ra \quad k_{\text{conv}} &= k_g \cdot 0.4 \cdot Ra^{0.2} \end{aligned} \quad (27)$$

The “*radiative component*” has been modelled via the equation (22).

4.6. Approximation 6 (*Simulation N°6*)

In this approximation after calculating the Ra number the following correlations proposed by Mc Adams [88] are employed for the “*convective component*” of the air’s equivalent thermal conductivity :

$$\begin{aligned} \text{if } Ra < 10000 \quad k_{\text{conv}} &= k_g \\ \text{if } 10000 < Ra < 10^9 \quad k_{\text{conv}} &= k_g \cdot 0.59 \cdot Ra^{0.25} \\ \text{if } 10^9 < Ra \quad k_{\text{conv}} &= k_g \cdot 0.13 \cdot Ra^{0.33} \end{aligned} \quad (28)$$

Similarly to the approximation 5, the “*radiative component*” is modelled via the equation (22).

4.7. Approximation 7: new updated correlation (*Simulation N°7*)

In this approximation the Gr number is firstly calculated via the equation (22). Then in order to calculate the “*convective component*” of the air’s equivalent thermal conductivity an adapted to our case version of the following correlation form proposed by Versteegh et al [89] to calculate the Nu number is proposed:

$$Nu = A \cdot (Gr \cdot Pr^2)^{1/C} \quad (29)$$

Within the medium range of Grashof numbers in the study by Versteeg et al [89], their constants $A=0.142$ and $C=3.0$ best fit to their problem. So their equation takes the form:

$$Nu = 0.142 \cdot (Gr \cdot Pr^2)^{1/3} \quad (30)$$

However, Kîs et al [47] had initially modified the constants A and C of the equation (29) employing the same pattern and proposing the following form according to the physical and geometrical particularities of their problem ([47], pp 2635):

$$Nu = 0.186 \cdot (Gr \cdot Pr^2)^{1/3.2} \quad (31)$$

Once again, within the medium range of Grashof numbers in the aforementioned study by Kîs et al [47], their constants $A=0.186$ and $C=3.2$ seem appropriate. Nonetheless, in the case presented in this paper, the medium range of Grashof numbers guided me to propose the following correlation using the regression analysis employed by Li et al [56] that meets much better the requirement of this study, since there is not heat source within the cavity and consequently we have no turbulences. So the correlation used in this approximation that is proposed here is the following:

$$Nu = 0.1427 \cdot (Gr \cdot Pr^2)^{1/3.7} \quad (32)$$

Similarly to the approximations 5 and 6, the “*radiative component*” is modelled via the equation (22).

4.8. Approximation 8: new updated correlation (*Simulation N°8*)

This approximation is based on a very recent approach introduced by Li et al [56] who studied the natural convection flow within inclined cavities with low width-to-height ratios via 3-D CFD numerical simulations. They employed a classic regression analysis to correlate the Nusselt number (Nu) to the modified Rayleigh number ($Ra \times \sin\theta$) introducing an adjusting factor f and an adjusted aspect ratio (fH/L ; see also Figure 4). The adjusting factor f is introduced in order to analyse the impact of cavity width W on the natural convective heat transfer in the cavity and is calculated as follows:

$$f = 1/W \cdot \sqrt{(H^2 + W^2)} \quad (33)$$

Figure 4: 3-D geometry and coordinate system for the natural convection flow in narrow cavity: a) setup for an inclined cavity (Li et al [56]), b) setup for a vertical cavity (present study).

Once the adjusting factor f is calculated they calculated the adjusted aspect ratio (fH/L) that perfectly fits to their case (see Figure 5a), since the air cavity is inclined. However in

the present case study, the air cavity is vertical. So, to adapt the equation 33 to this case the adjusting factor f is calculated using the equation 33 putting the H in the denominator instead of W since I deal with a vertical cavity problem, and then the aspect ratio fW/L that best fits to the geometry of our problem is identified. Then the Nu correlations proposed by Li et al [56] are modified using the regression analysis that they employed. Thus, the medium range of Rayleigh numbers guided me to propose the following correlation that meet much better the requirement of this study ($\theta=90^\circ$, $\sin\theta=1$):

$$10^3 < Ra \cdot \sin \theta \quad Nu = 0.0009461(Ra \cdot \sin \theta)^{0.45} \cdot \left(f \cdot \frac{W}{L}\right)^{-0.88} \quad (34)$$

Finally, similarly to the approximations 5, 6 and 7, the “*radiative component*” is modelled via the equation (22).

In figure 5 we have in a synthetic manner the flowchart regarding the solution of the equations with emphasis on the 8 different natural convection approaches integrated within the model especially for this study.

Figure 5: Flowchart of the solution of the equations with integration of the 8 different approaches.

5. Sample description, Results and Discussion

The aim of the present paper is to evaluate the accuracy of the simulated data and discover which approach describes better transient coupled radiative-natural convective heat transfer across vertical cavities in the vicinity of low emissivity surfaces. In this study, a classic configuration as it is presented in Figure 3 is simulated. This configuration has been extensively studied experimentally employing a calibrated guarded hot box in accordance to the *EN ISO 8990* international standard [13], [11]. The composition and the thermo-physical characteristics of the studied configuration are tabulated in the following table (Table 1).

Table 1: Materials and thermo-physical characteristics of the sample.

MATERIAL	ρc_p [J/(m ³ ·K)]	k [W/(K·m)]	e	$N^\bullet$ and thickness
wood (white pine)	524800	0.12	0.7	2 (2.1cm)
polyester	39600	0.023 (for T_{init} and then calculated in each timestep)	0.69	5 (6.1mm)
polyester wadding	45822	0.024 (for T_{init} and calculated in each timestep)	0.33	2 (2.5mm)
silver coated reflective foils	-	-	0.15	7

Effective thermal conductivities and volumetric heat capacities of the wood, the polyester and the polyester wadding insulation composites were measured using the patented Hot Disk TPS 1500 Thermal Constants Analyzer test system with an average disk resolution of 3.65% during the experiment sessions while the detailed measuring procedure is presented elsewhere [13]. Emissivity of each insulation material and reflective foil was measured using the Emissometer Model AE manufactured by Devices & Services Company. The detector portion of the device was heated to 355 K so that the sample to be measured did not have to be heated, whilst the emittance of the unknown aluminium foil

was measured after the detector had been calibrated with two standards of known emittance, which were maintained at the same temperature as the unknown (for details on the measuring process see also [13]). Determinately, the overall experimental characterisation of the sample has been done developing a calibrated guarded hot box as the measurement procedures for specific structures are specified in the standard NF EN ISO 8990 (1996) and several international and European standards (i.e., ISO 12567-1 2000) whereas the calibration of the guarded hot box has been done using a sample of Effisol (the detailed experimental procedure of the sample's characterization is presented here: [40], [11], [13]). A large specimen panel of $1.91\text{ m} \times 1.96\text{ m}$ according to the geometry illustrated in figure 3 has been tested.

Henceforward, the numerical model is set up in order to represent the geometry of the aforementioned sample, while the initial temperature conditions were the same that have been imposed during the experimental sequences. The reconstructions of initial air temperature input profiles conditions in the cold and the hot boundaries are illustrated on figures 6a and 6b.

Figure 6: Reconstruction of initial air temperature input profiles according to the available experimental data (data source: [13]) a) hot boundary, b) cold boundary.

For the simulation temperature outputs, the scores produced are the *Bias score* (standard mean error) and the Root Mean Square Error (*RMSE*) as follows [90], [91], [92], [93], [94] :

$$RMSE = \sqrt{\frac{\sum_{i=1}^N (M_i - O_i)^2}{N}} \quad (35)$$

$$BIAS = \frac{1}{N} \sum_{i=1}^N (M_i - O_i) = \bar{M} - \bar{O} \quad (36)$$

where the M represents the simulated data and the O the experimental measurements. In order to obtain valuable insights regarding the accuracy of the predictive capacity of each different approximation integrated in the numerical model and confronted to experimental measurements, the accuracy of the numerical model outputs is measured via the *RMSE* and *Bias score*, since these scores are used –among other applications- for complex statistical evaluation of weather prediction numerical models [90], [92], [94]. Ergo, the respective *RMSEs* and *Bias scores* corresponding to each approximation regarding the temperature gradient as well as the temperatures in the vicinity of the sample are tabulated in table 2 and illustrated in figure 7, while the model's ΔT outputs for each approximation are presented in Figures 8 and 9.

As it results from the table 2 and the figures 7, 8 and 9 there is a close agreement between measured and model-predicted values for all the different approximations. However the approximations n°5, n°7 and n°8, produced the best *RMSE* and *Bias Scores* for both boundaries and temperature gradients. Furthermore, the least square line equations of the 3 selected configurations and the R^2 , confirm the accuracy of the simulated results (Figures 8ii, 8iv, 8vi, 8viii, 9ii, 9iv, 9vi, 9viii). Finally, we can assume that the best approximation for this case, whereas we deal with vertical cavities in the vicinity of low emissivity surfaces is the approximation n°8.

Table 2: Summarized results of the numerical model's evaluation after the integration of the aforementioned 8 different approximations

	<i>RMSE</i> ΔT	<i>Bias</i> ΔT	<i>RMSE</i> T_{cold}	<i>Bias</i> T_{cold}	<i>RMSE</i> T_{hot}	<i>Bias</i> T_{hot}
Approximation n°1	0.062849	0.05071	0.00843	-0.00318	0.05943	0.04750
Approximation n°2	0.062361	0.05001	0.00837	-0.00315	0.05898	0.04693
Approximation n°3	0.083998	-0.07513	0.008799	0.004211	0.07789	-0.07032
Approximation n°4	0.082420	-0.07451	0.008800	0.004213	0.07933	-0.07100
Approximation n°5	0.039461	-0.00647	0.00789	0.00012	0.03493	-0.00563
Approximation n°6	0.070850	0.05763	0.00917	-0.00389	0.06530	0.05460
Approximation n°7	0.036075	0.00150	0.00798	-0.00030	0.03460	9.38263E-06
Approximation n°8	0.035672	0.00024	0.00796	-0.00021	0.03426	2.6369E-05

Figure 7: Comparison of RMSE and Bias score for the 8 different approximations.

Besides, after examining figures 8 and 9 regarding the point-to-point confrontation of the simulated and experimental data, it is observed that the less adapted to this case approximations have the tendency to overestimate or underestimate (in comparison to the experimental reference data) the temperature gradients as time passes. In this study 4 days (real time) are simulated and it is observed that all the approximations have a satisfactory accurate response the first 12-24 hours (depending on the approximation), however most approximations have the tendency to need readjustment after this time valid period since even if the simulated data follow the pattern of the experimental measurements, the data need to be readjusted to obtain the correct temperature values. More precisely, approximations n°1, n°2, and n°6 overestimate the temperature gradient picks after 24 hours (real time) while the approximations n°3, n°4 underestimate the same picks. Nonetheless, approximations n°5, n°7 and n°8 are the best adapted to this study approximations, since they produced data that follow the pattern and the absolute

temperature experimental values even after simulating 96 hours, while the approximation n°8 slightly provides the most valuable results. Regarding the radiation schemes we observe not a real influence since the exchanging surfaces are very close each other (very low width of the cavity) within a limited temperature range (for example approximations 1 and 2).

Figure 8: Comparison of predicted and measured temperature gradients in the boundaries of the sample: i) point-to point confrontation for the Approximation n°1, ii) least square line equation and the square of the linear correlation coefficient for the Approximation n°1, iii) point-to point confrontation for the Approximation n°2, iv) least square line equation and the square of the linear correlation coefficient for the Approximation n°2, v) point-to point confrontation for the Approximation n°3, vi) least square line equation and the square of the linear correlation coefficient for the Approximation n°3, vii) point-to point confrontation for the Approximation n°4, viii) least square line equation and the square of the linear correlation coefficient for the Approximation n°4.

Figure 9: Comparison of predicted and measured temperature gradients in the boundaries of the sample: i) point-to point confrontation for the Approximation n°5, ii) least square line equation and the square of the linear correlation coefficient for the Approximation n°5, iii) point-to point confrontation for the Approximation n°6, iv) least square line equation and the square of the linear correlation coefficient for the Approximation n°6, v) point-to point confrontation for the Approximation n°7, vi) least square line equation and the square of the linear correlation coefficient for the Approximation n°7, vii) point-to point confrontation for the Approximation n°8, viii) least square line equation and the square of the linear correlation coefficient for the Approximation n°8.

Furthermore, for each approximation the k_{eff} values within the air cavities are also calculated as shown in Figure 10, 11 and 12 and tabulated in table 3. The equivalent thermal conductivity value k_{eff} is very important, since it is a value directly linked to the overall coupled heat transfer within the cavity. As we can observe in figures 10, 11a, 11b, 11c and table 3 the mean k_{eff} values range between 0.048321 - $0.063563 \text{ W m}^{-1} \cdot \text{K}^{-1}$ within the cavity in the vicinity of the hot chamber. As it is shown in figures 10, 12a, 12b, 12c and tabulated in table 3 the mean k_{eff} values are calculated between

$0.044890\text{-}0.059793\text{ Wm}^{-1}\cdot\text{K}^{-1}$ within the cavity in the vicinity of the cold chamber. The standard deviation for the mean k_{eff} values within both air cavities is calculated as follows and tabulated in table 2:

$$\sigma_{\text{dev}} = \sqrt{\frac{1}{N} \sum_{i=1}^N (k_{\text{eff}_i} - \overline{k_{\text{eff}}})^2} \quad (37)$$

Table 3: Summarized results of the mean k_{eff} coefficient and corresponding standard deviation for both cavities for each approximation.

	k_{eff} air cavity 1 ($\text{WK}^{-1}\text{m}^{-1}$)	Standard deviation (air cavity 1)	k_{eff} air cavity2 ($\text{WK}^{-1}\text{m}^{-1}$)	Standard deviation (air cavity 2)
Approximation n°1	0.048552	0.000114347	0.045854	7.42194E-05
Approximation n°2	0.048598	0.000116571	0.045944	7.81899E-05
Approximation n°3	0.063562	0.000103167	0.059792	6.07833E-05
Approximation n°4	0.063563	0.000103241	0.059793	5.11398E-05
Approximation n°5	0.053821	0.000133655	0.052297	0.000187802
Approximation n°6	0.048321	0.000101352	0.044890	6.07833E-05
Approximation n°7	0.053232	0.000221027	0.052008	0.000257809
Approximation n°8	0.052787	0.000296797	0.051578	0.000168744

Figure 10: Comparison of the k_{eff} values within the two closed air cavities.

Figure 11: Comparison of the estimated k_{eff} coefficients within the cavities according to the different approximations: a) cavity in the vicinity of the hot chamber: approximations n°1 and n°2, b) cavity in the vicinity of the hot chamber: approximations n°3 and n°4, c) cavity in the vicinity of the hot chamber: approximations n°1, n°5, n°6, n°7, n°8.

Figure 12: Comparison of the estimated k_{eff} coefficients within the cavities according to the different approximations: a) cavity in the vicinity of the cold chamber: approximations n°1 and n°2, b) cavity in the vicinity of the cold chamber: approximations n°3 and n°4, c) cavity in the vicinity of the hot chamber: approximations n°1, n°5, n°6, n°7, n°8.

Literatim, figure 11a shows a close agreement between the calculated k_{eff} values provided by approximations n°1 and n°2 since the convection scheme is the same for these two approximations ; figure 11b shows identical calculated k_{eff} values for the approximations n°3 and n°4 since the convection scheme is the same for these two approximations ; while figure 11c shows that the best -adapted to this case- approximations (n°5, n°7 and

n°8) provide similar calculated k_{eff} values. The same conclusions can be extracted from figures 12a, 12b and 12c regarding the cavity in the vicinity of the cold boundary.

Finally in order to examine the time validity period of the best fitted to this case correlation, 240 hours of real time (10 days) have been simulated. Figures 13, 14 and 15 confirms that the best fitted approximation is the new updated correlation proposed in this paper which has a time validity period of at least 10 days. As we can observe even if simulation n°3 underestimates heat transfer during steady state, no significant difference is observed since we are within the error limit (between 96-144 hours the mean value of measurements has been considered since the system reached a full steady state regime); however in transient regime we observe that only the approximation n°8 tends to correctly simulate the post-steady state local temperature picks (Figures 14a and 14b).

Figure 13: Comparison of predicted and measured temperature gradients in the boundaries of the sample after simulating 240 hours of real time (10 days): i), least square line equation and the square of the linear correlation coefficient for the Approximation n°3, ii) least square line equation and the square of the linear correlation coefficient for the Approximation n°8.

Figure 14: Comparison of predicted and measured temperature gradients in the boundaries of the sample after simulating 240 hours of real time (10 days): a) point-to-point confrontation for the Approximation n°8, b) least square line equation and the square of the linear correlation coefficient for the Approximation n°3.

Finally, in comparison to the worst approximation (approximation n°3) the new updated

correlations provide very low RMSE and bias score as we can observe in Figure 15. More precisely, approximation n°3 produced a *Bias score* = -0.07758 and a *RMSE* = 0.08689 whilst the new updated correlation (approximation n°8) produced a *Bias score* = 0.00014 and a *RMSE* = 0.039914.

Figure 15: Comparison of RMSE and Bias score for approximations n°3 and n°8 after simulating 240 hours (10 days). The Bias score for the approximation n°8 is 0.00014 and for this reason it is not observable in this figure.

Henceforward, throughout the study presented in this paper, the Nu correlations proposed by Kîs et al [47] and Li et al [56] have been adapted to this case as follows, via a classic regression analysis employed by Li et al [56]. The new correlations that are proposed here meet much better the requirement of this study regarding air cavities in the vicinity of low emissivity surfaces. The first proposed correlation links the Nusselt number to the Grashof and Prandtl numbers:

$$Nu = 0.1427 \cdot (Gr \cdot Pr^2)^{1/3.7} \quad (38)$$

Additionally, the second correlation links the Nusselt number to the Rayleigh number:

$$\text{for } 10^3 < Ra \cdot \sin \vartheta \quad Nu = 94.606 \cdot 10^{-5} \cdot (Ra \cdot \sin \vartheta)^{0.45} \cdot \left(f \cdot \frac{W}{L}\right)^{-0.88} \quad (39)$$

Both equations can be used to model with accuracy natural convection within vertical air cavities in the vicinity of low emissivity surfaces and can be implemented in CFD calculations while approximation 8 presents a time validity period of at least 10 days. The aforementioned correlations are robust and accurate enough to describe natural convection across low width-to-height ratios in the building scale (for a cavity width up to *5cm*) in the vicinity of low emissivity surfaces.

6. Conclusions and Perspectives

In this work, the coupled with radiation natural convection heat transfer within vertical cavities in the vicinity of low emissivity surfaces is studied. An experimentally validated (via a calibrated guarded hot box apparatus) numerical model for modelling combined radiation/conduction heat transfer within innovative radiant barrier insulation solutions currently used on building applications, which consists of high-porous, semi-transparent optically thin materials and silver coated low emissivity foils has been updated, while 8 different approximations to model coupled natural convection-radiation heat transfer within the cavities have been implemented after explicitly reviewing the literature on scale regarding this subject. Then the geometry and the thermophysical characteristics of

an innovative envelope prototype (experimentally tested in the past [11]) have been introduced to the model while 8 different simulation sequences have been performed simulating 96 hours (real time). Afterwards, the simulated data were confronted point-to-point to available experimental data for the same prototype (under exactly the same conditions). Subsequently, a selection of the best adapted to this case approximations has been made according to an explicit statistical procedure of evaluation. Furthermore, two updated novel correlations that are valid for the convection flow within vertical cavities in the vicinity of low emissivity surfaces are proposed. The novelty and originality of this paper is resumed as follows:

- proposition of two new updated correlations for modelling natural convection across low-width-to-height ratio air cavities in the vicinity of low emissivity surfaces ($0.05 < e < 0.3$), since there is lack of available literature on this specific subject
- the new updated correlations have a time valid period of at least 10 days of real time (240 hours), whilst the available correlations after implementation into the model revealed to hardly have a time valid period of maximum 24 hours
- the new updated correlations model with accuracy the local temperature picks impose by the initial temperature conditions

Far from being a singular case study, this paper proposes 2 updated correlations specifically adapted to natural convection low width-to-height ratios at the building scale cavity problems integrated within innovative building envelope prototypes. The large time validity period of the proposed equations makes them adequate for integration into

dynamic building codes.

Thence, because the heat transfer problem is much complicated, and the perspective of this study is to extend this research axis towards phase change materials (PCM) in order to conceive innovative envelope prototypes combining radiant barriers and PCM as it is explained in the introduction chapter, it is necessary to further develop the presented numerical model by considering phase change and realizing multiple tests in adequate experimental platforms including PCM in the studied complexes, varying at the same time the parameters that influence the efficiency of the proposed sophisticated prototypes. However before conducting this study and implement new physics within the numerical model it was important to identify and propose the most appropriate correlation adapted to the particularity of coupled transient heat transfer within air cavities in the vicinity of low emissivity surfaces. Finally, another interesting perspective is the implementation of the new updated correlations into CFD codes in order to conduct a parametric study on the different aspect ratios, with the aim to generalize their application.

References

- [1] J. Kurnitski, K. Kuusk, T. Tark, A. Uutar, T. Kalamees, E. Pikas, Energy and investment intensity of integrated renovation and 2030 cost optimal savings, *Energy Build.* 75 (2014) 51–59. doi:10.1016/j.enbuild.2014.01.044.
- [2] E. Pikas, M. Thalfeldt, J. Kurnitski, Cost optimal and nearly zero energy building solutions for office buildings, *Energy Build.* 74 (2014) 30–42. doi:10.1016/j.enbuild.2014.01.039.
- [3] E. Pikas, M. Thalfeldt, J. Kurnitski, R. Liias, Extra cost analyses of two apartment buildings for achieving nearly zero and low energy buildings, *Energy*. 84 (2015) 623–633. doi:10.1016/j.energy.2015.03.026.
- [4] L.E. Mavromatidis, A review on hybrid optimization algorithms to coalesce computational morphogenesis with interactive energy consumption forecasting, *Energy Build.* 106 (2015) 192–202. doi:10.1016/j.enbuild.2015.07.003.

- [5] A. Franco, F. Fantozzi, Experimental analysis of a self consumption strategy for residential building: The integration of PV system and geothermal heat pump, *Renew. Energy*. 86 (2016) 1075–1085. doi:10.1016/j.renene.2015.09.030.
- [6] S.K. Pal, K. Alanne, J. Jokisalo, K. Siren, Energy performance and economic viability of advanced window technologies for a new Finnish townhouse concept, *Appl. Energy*. 162 (2016) 11–20. doi:10.1016/j.apenergy.2015.10.056.
- [7] S. Mirrahimi, M.F. Mohamed, L.C. Haw, N.L.N. Ibrahim, W.F.M. Yusoff, A. Aflaki, The effect of building envelope on the thermal comfort and energy saving for high-rise buildings in hot-humid climate, *Renew. Sustain. Energy Rev.* 53 (2016) 1508–1519. doi:10.1016/j.rser.2015.09.055.
- [8] R. Ruparathna, K. Hewage, R. Sadiq, Improving the energy efficiency of the existing building stock: A critical review of commercial and institutional buildings, *Renew. Sustain. Energy Rev.* 53 (2016) 1032–1045. doi:10.1016/j.rser.2015.09.084.
- [9] N. Lepkova, D. Zubka, R.L. Jensen, *Global Sustainable Communities Handbook*, Elsevier, 2014. doi:10.1016/B978-0-12-397914-8.00010-2.
- [10] L.E. Mavromatidis, M. EL Mankibi, P. Michel, M. Santamouris, Numerical estimation of time lags and decrement factors for wall complexes including Multilayer Thermal Insulation, in two different climatic zones, *Appl. Energy*. 92 (2012) 480–491. doi:10.1016/j.apenergy.2011.10.007.
- [11] L. Mavromatidis, A. Bykalyuk, M. El Mankibi, P. Michel, M. Santamouris, Numerical Investigation of a Wall's Optimum Multilayer Thermal Insulation Position, in: *12th Conf. IBPSA*, 2011: pp. 980–987. http://www.ibpsa.org/proceedings/BS2011/P_1369.pdf.
- [12] L.E. Mavromatidis, M. El Mankibi, P. Michel, A. Bykalyuk, M. Santamouris, Guidelines to study numerically and experimentally reflective insulation systems as applied to buildings, *Adv. Build. Energy Res.* 6 (2012) 2–35. doi:10.1080/17512549.2012.671495.
- [13] L. Mavromatidis, P. Michel, M. El Mankibi, M. Santamouris, Study on transient heat transfer through multilayer thermal insulation: Numerical analysis and experimental investigation, *Build. Simul.* 3 (2010) 279–294. doi:10.1007/s12273-010-0018-z.
- [14] I. Hernández-Pérez, G. Álvarez, J. Xamán, I. Zavala-Guillén, J. Arce, E. Simá, Thermal performance of reflective materials applied to exterior building components—A review, *Energy Build.* 80 (2014) 81–105. doi:10.1016/j.enbuild.2014.05.008.
- [15] H. Shen, H. Tan, A. Tzempelikos, The effect of reflective coatings on building surface temperatures, indoor environment and energy consumption—An experimental study, *Energy Build.* 43 (2011) 573–580. doi:10.1016/j.enbuild.2010.10.024.
- [16] M. Čekon, M. Kalousek, J. Hraška, R. Ingeli, Spectral optical properties and

- thermodynamic performance of reflective coatings in a mild climate zone, *Energy Build.* 77 (2014) 343–354. doi:10.1016/j.enbuild.2014.04.005.
- [17] A. Joudi, H. Svedung, M. Cehlin, M. Rönnelid, Reflective coatings for interior and exterior of buildings and improving thermal performance, *Appl. Energy.* 103 (2013) 562–570. doi:10.1016/j.apenergy.2012.10.019.
- [18] T. Henriksen, S. Lo, U. Knaack, An innovative approach to manufacture thin-walled glass fibre reinforced concrete for tomorrow's architectural buildings envelopes with complex geometries, *J. Build. Eng.* 4 (2015) 189–199. doi:10.1016/j.job.2015.09.006.
- [19] F.L. Chen, X. He, H.M. Yin, Manufacture and multi-physical characterization of aluminum/high-density polyethylene functionally graded materials for green energy building envelope applications, *Energy Build.* 116 (2016) 307–317. doi:10.1016/j.enbuild.2015.11.001.
- [20] L. Li, G. Wang, C. Guo, Influence of intumescent flame retardant on thermal and flame retardancy of eutectic mixed paraffin/polypropylene form-stable phase change materials, *Appl. Energy.* 162 (2016) 428–434. doi:10.1016/j.apenergy.2015.10.103.
- [21] S.N. Gunasekara, R. Pan, J.N. Chiu, V. Martin, Polyols as phase change materials for surplus thermal energy storage, *Appl. Energy.* 162 (2015) 1439–1452. doi:10.1016/j.apenergy.2015.03.064.
- [22] K. Pielichowska, M. Nowak, P. Szatkowski, B. Macherzyńska, The influence of chain extender on properties of polyurethane-based phase change materials modified with graphene, *Appl. Energy.* 162 (2016) 1024–1033. doi:10.1016/j.apenergy.2015.10.174.
- [23] R. Fukahori, T. Nomura, C. Zhu, N. Sheng, N. Okinaka, T. Akiyama, Thermal analysis of Al–Si alloys as high-temperature phase-change material and their corrosion properties with ceramic materials, *Appl. Energy.* 163 (2016) 1–8. doi:10.1016/j.apenergy.2015.10.164.
- [24] X. Sun, Q. Zhang, M. a. Medina, K.O. Lee, Experimental observations on the heat transfer enhancement caused by natural convection during melting of solid–liquid phase change materials (PCMs), *Appl. Energy.* 162 (2015) 1453–1461. doi:10.1016/j.apenergy.2015.03.078.
- [25] M. Mu, P.A.M. Basheer, W. Sha, Y. Bai, T. McNally, Shape stabilised phase change materials based on a high melt viscosity HDPE and paraffin waxes, *Appl. Energy.* 162 (2016) 68–82. doi:10.1016/j.apenergy.2015.10.030.
- [26] T. Zhang, Y. Tan, H. Yang, X. Zhang, The application of air layers in building envelopes: A review, *Appl. Energy.* 165 (2016) 707–734. doi:10.1016/j.apenergy.2015.12.108.
- [27] J.A. Palyvos, A survey of wind convection coefficient correlations for building envelope energy systems' modeling, *Appl. Therm. Eng.* 28 (2008) 801–808. doi:10.1016/j.applthermaleng.2007.12.005.

- [28] T. Voellinger, A. Bassi, M. Heitel, Facilitating the incorporation of VIP into precast concrete sandwich panels, *Energy Build.* 85 (2014) 666–671. doi:10.1016/j.enbuild.2014.05.038.
- [29] M. Sistaninia, M. Hudert, L. Humbert, Y. Weinand, Experimental and numerical study on structural behavior of a single timber Textile Module, *Eng. Struct.* 46 (2013) 557–568. doi:http://dx.doi.org/10.1016/j.engstruct.2012.08.024.
- [30] H.J. Xu, C.Y. Zhao, Z.G. Xu, Analytical considerations of slip flow and heat transfer through microfoams in mini/microchannels with asymmetric wall heat fluxes, *Appl. Therm. Eng.* 93 (2016) 15–26. doi:10.1016/j.applthermaleng.2015.09.068.
- [31] J. Miranda Fuentes, K. Johannes, F. Kuznik, M. Cosnier, J. Virgone, Melting with convection and radiation in a participating phase change material, *Appl. Energy.* 109 (2013) 454–461. doi:10.1016/j.apenergy.2012.11.031.
- [32] A. Bykalyuk, F. Kuznik, K. Johannes, Studying the evolution of both thermal and kinetic boundary layers in the vicinity of a vertical conductive gypsum plate under dynamic time-depending conditions at the building scale, *Energy Build.* 86 (2015) 898–908. doi:10.1016/j.enbuild.2014.11.010.
- [33] R. Ganesan, D.G. McCulloch, N.A. Marks, M.D. Tucker, J.G. Partridge, M.M.M. Bilek, et al., Synthesis of highly tetrahedral amorphous carbon by mixed-mode HiPIMS sputtering, *J. Phys. D: Appl. Phys.* 48 (2015) 442001. doi:10.1088/0022-3727/48/44/442001.
- [34] J.-W. Shi, K.-L. Chi, C.-Y. Li, J.-M. Wun, Dynamic Analysis of High-Efficiency InP-Based Photodiode for 40 Gbit/s Optical Interconnect Across a Wide Optical Window (0.85 to 1.55 μm), *J. Light. Technol.* 33 (2015) 921–927. doi:10.1109/JLT.2014.2364627.
- [35] L. Vanhoutteghem, G.C.J. Skarving, C.A. Hviid, S. Svendsen, Impact of façade window design on energy, daylighting and thermal comfort in nearly zero-energy houses, *Energy Build.* 102 (2015) 149–156. doi:10.1016/j.enbuild.2015.05.018.
- [36] C. Tummeltshammer, A. Taylor, A.J. Kenyon, I. Papakonstantinou, Losses in luminescent solar concentrators unveiled, *Sol. Energy Mater. Sol. Cells.* 144 (2016) 40–47. doi:10.1016/j.solmat.2015.08.008.
- [37] J.N. Bullock, Y. Xu, D. Benson, H.M. Branz, Wide GAP a-SiC:H alloys for novel photovoltaic-electrochromic window coatings, in: *Mater. Res. Soc. Symp. - Proc.*, Materials Research Society, 1995: pp. 589–594.
- [38] C. Bechinger, J.N. Bullock, J.G. Zhang, C.E. Tracy, D.K. Benson, S.K. Deb, et al., Low-voltage electrochromic device for photovoltaic-powered smart windows, *J. Appl. Phys.* 80 (1996) 1226–1232.
- [39] J.M. Walls, A. Abbas, G.D. West, J.W. Bowers, P.J.M. Isherwood, P.M. Kaminski, et al., The Effect of Annealing Treatments on Close Spaced Sublimated Cadmium Telluride Thin Film Solar Cells, *MRS Proc.* 1493 (2013) mrsf12-1493-e04-02. doi:10.1557/opl.2012.1704.

- [40] L.E. Mavromatidis, A. Bykalyuk, M. El Mankibi, P. Michel, M. Santamouris, Numerical estimation of air gaps' influence on the insulating performance of multilayer thermal insulation, *Build. Environ.* 49 (2012) 227–237. doi:10.1016/j.buildenv.2011.09.029.
- [41] P. Marin, X. Marsault, L.E. Mavromatidis, R. Saleri, F. Torres, Ec-co-gen: An evolutionary simulation assisted design tool for energy rating of buildings in early design stage to optimize the building form, *Proc. BS 2013 13th Conf. Int. Build. Perform. Simul. Assoc.* (2013) 2201–2208.
- [42] L.E. Mavromatidis, A. Bykalyuk, H. Lequay, Development of polynomial regression models for composite dynamic envelopes' thermal performance forecasting, *Appl. Energy*. 104 (2013) 379–391. doi:10.1016/j.apenergy.2012.10.045.
- [43] E. Dascalaki, M. Santamouris, C. a. Balaras, D.N. Asimakopoulos, Natural convection heat transfer coefficients from vertical and horizontal surfaces for building applications, *Energy Build.* 20 (1994) 243–249. doi:10.1016/0378-7788(94)90027-2.
- [44] A.-J.N. Khalifa, Natural convective heat transfer coefficient – a review, *Energy Convers. Manag.* 42 (2001) 505–517. doi:10.1016/S0196-8904(00)00043-1.
- [45] J.H. Lienhard, Heat Transfer, *J. Heat Transfer*. 82 (2010) 198. doi:10.1115/1.3246887.
- [46] A.-J.N. Khalifa, Natural convective heat transfer coefficient – a review II. Surfaces in two- and three-dimensional enclosures, *Energy Convers. Manag.* 42 (2001) 505–517. doi:10.1016/S0196-8904(00)00043-1.
- [47] P. Kiš, H. Herwig, The near wall physics and wall functions for turbulent natural convection, *Int. J. Heat Mass Transf.* 55 (2012) 2625–2635. doi:10.1016/j.ijheatmasstransfer.2011.12.031.
- [48] F. Corvaro, M. Paroncini, A numerical and experimental analysis on the natural convective heat transfer of a small heating strip located on the floor of a square cavity, *Appl. Therm. Eng.* 28 (2008) 25–35. doi:10.1016/j.applthermaleng.2007.03.018.
- [49] S.E. Ahmed, H.F. Oztop, K. Al-Salem, Natural convection coupled with radiation heat transfer in an inclined porous cavity with corner heater, *Comput. Fluids*. 102 (2014) 74–84. doi:10.1016/j.compfluid.2014.06.024.
- [50] Y. Wei, H.-S. Dou, Z. Wang, Y. Qian, W. Yan, Simulations of natural convection heat transfer in an enclosure at different Rayleigh number using lattice Boltzmann method, *Comput. Fluids*. 124 (2015) 30–38. doi:10.1016/j.compfluid.2015.09.004.
- [51] C. Obrecht, P. Asinari, F. Kuznik, J.-J. Roux, Thermal link-wise artificial compressibility method: GPU implementation and validation of a double-population model, *Comput. Math. with Appl.* (2015). doi:10.1016/j.camwa.2015.05.022.

- [52] R.F. Babus'Haq, S.D. Probert, M.J. Shilston, Natural convection across cavities: Design advice, *Appl. Energy*. 20 (1985) 161–188. doi:10.1016/0306-2619(85)90022-4.
- [53] A. Baidi, Transient thermal characteristics of airborne electronic equipment with discrete hot bands in square cavities, *Appl. Energy*. 85 (2008) 951–967. doi:10.1016/j.apenergy.2008.02.009.
- [54] A. Mezrhab, H. Bouali, H. Amaoui, M. Bouzidi, Computation of combined natural-convection and radiation heat-transfer in a cavity having a square body at its center, *Appl. Energy*. 83 (2006) 1004–1023. doi:10.1016/j.apenergy.2005.09.006.
- [55] V.P. Carey, Experimental measurements of orthogonal mixed convection in a partial enclosure, 33 (1990) 1307–1319.
- [56] H. Li, S. Tong, Natural convective heat transfer in the inclined rectangular cavities with low width-to-height ratios, *Int. J. Heat Mass Transf.* 93 (2016) 398–407. doi:10.1016/j.ijheatmasstransfer.2015.10.027.
- [57] M. Montiel Gonzalez, J. Hinojosa Palafox, C.A. Estrada, Numerical study of heat transfer by natural convection and surface thermal radiation in an open cavity receiver, *Sol. Energy*. 86 (2012) 1118–1128. doi:10.1016/j.solener.2012.01.005.
- [58] A.K.A. Shati, S.G. Blakey, S.B.M. Beck, A dimensionless solution to radiation and turbulent natural convection in square and rectangular enclosures, *J. Eng. Sci. Technol.* 7 (2012) 257–279.
- [59] S. Arun, A. Satheesh, Analysis of flow behaviour in a two sided lid driven cavity using lattice boltzmann technique, *Alexandria Eng. J.* 54 (2015) 795–806. doi:10.1016/j.aej.2015.06.005.
- [60] Q. Ren, C.L. Chan, Numerical study of double-diffusive convection in a vertical cavity with Soret and Dufour effects by lattice Boltzmann method on GPU, *Int. J. Heat Mass Transf.* 93 (2016) 538–553. doi:10.1016/j.ijheatmasstransfer.2015.10.031.
- [61] M. Sheikholeslami, M.G. Bandy, H.R. Ashorynejad, Lattice Boltzmann Method for simulation of magnetic field effect on hydrothermal behavior of nanofluid in a cubic cavity, *Phys. A Stat. Mech. Its Appl.* 432 (2015) 58–70. doi:10.1016/j.physa.2015.03.009.
- [62] B. Blais, J.-M. Tuckey, D. Vidal, F. Bertrand, A conservative lattice Boltzmann model for the volume-averaged Navier–Stokes equations based on a novel collision operator, *J. Comput. Phys.* 294 (2015) 258–273. doi:10.1016/j.jcp.2015.03.036.
- [63] A. Gupta, M. Sbragaglia, A. Scagliarini, Hybrid Lattice Boltzmann/Finite Difference simulations of viscoelastic multicomponent flows in confined geometries, *J. Comput. Phys.* 291 (2015) 177–197. doi:10.1016/j.jcp.2015.03.006.
- [64] H. Liu, H. Wang, S. Liu, C. Hu, Y. Ding, J. Zhang, Lattice Boltzmann method for

- the Saint–Venant equations, *J. Hydrol.* 524 (2015) 411–416.
doi:10.1016/j.jhydrol.2015.03.002.
- [65] R. Huang, H. Wu, Phase interface effects in the total enthalpy-based lattice Boltzmann model for solid–liquid phase change, *J. Comput. Phys.* 294 (2015) 346–362. doi:10.1016/j.jcp.2015.03.064.
- [66] E. Palik, Palik-I, *Handbook of Optical Constants, Vol.1* (AP, 1985)(ISBN 0125444206), (1998).
- [67] E. Palik, Palik-II, *Handbook of Optical Constants, Vol.2* (AP, 1991)(ISBN 0125444222), Elsevier Ltd, San Diego California, n.d.
- [68] E. Palik, Palik-III, *Handbook of Optical Constants, Vol.3* (AP, 1998)(ISBN 0125444230), Academic Press, San Diego California, 1998.
- [69] N. Du, J. Fan, H. Wu, W. Sun, International Journal of Heat and Mass Transfer Optimal porosity distribution of fibrous insulation, *Int. J. Heat Mass Transf.* 52 (2009) 4350–4357. doi:10.1016/j.ijheatmasstransfer.2009.03.067.
- [70] C. Shi, T. Nakajima, Journal of Quantitative Spectroscopy & Radiative Transfer The Eddington approximation calculation of radiation flux in the atmosphere – ocean system, *J. Quant. Spectrosc. Radiat. Transf.* 157 (2015) 34–41.
doi:10.1016/j.jqsrt.2015.02.007.
- [71] A.R. Degheidy, M. Sallah, M.T. Attia, International Journal of Heat and Mass Transfer Radiative heat transfer in variable refractive index slab media using Pomraning – Eddington approximation, *Int. J. Heat Mass Transf.* 70 (2014) 688–692. doi:10.1016/j.ijheatmasstransfer.2013.09.038.
- [72] B. Subasilar, Analytical approaches to the delta-Eddington model of the radiative transfer through vertically inhomogeneous optical depths, 32 (2008) 514–534.
doi:10.1016/j.apm.2007.02.005.
- [73] D. Bai, X.J. Fan, On the combined heat transfer in the multilayer non-gray porous fibrous insulation, 104 (2007) 326–341. doi:10.1016/j.jqsrt.2006.09.003.
- [74] F.P. Incropera, D.P. DeWitt, T.L. Bergman, A.S. Lavine, *Fundamentals of Heat and Mass Transfer*, John Wiley & Sons, 2007.
- [75] K. Daryabeigi, T. Conference, *Heat Transfer in High-Temperature Fibrous Insulation*, AIAA. (2002).
- [76] L.K. Mathews, R. Viscanta, L. Incropera, Combined conduction radiation Heat Transfer Porous Materials Heated by Intense Solar Radiation, *J. Sol. Energy Eng.* 107 (1985) 29–34.
- [77] S. Patankar, *Numerical Heat Transfer and Fluid Flow*, Taylor & Francis, 1980, ISBN 0-89116-522-3.
- [78] J. Anderson JD, *Computational fluid dynamics. The basics with applications*, McGraw Hill, New York, 1995.
- [79] X. Kakatsios, *Principles of Heat and Mass Transfer (In Greek)*, SIMEON, Athens,

2007.

- [80] S. Worcester Polytechnic Institute, Handbook of fire protection Engineering Society of Fire Protection Engineering., Quincy MA, Boston, 1995.
- [81] J.A. Duffie, W.A. Beckman, Solar Engineering of Thermal Processes, Wiley Interscience Publication, New York, 1980.
- [82] H. Schuh, Heat Transfer in Structures, Pergamon Press, Oxford, 1965.
- [83] D. Rotari, Matériaux réfléchissants : Echanges radiatifs GLO. (In French), Lyon, 2008.
- [84] J.G.A. De Graaf, E.F.M. Van Der Held, The relation between the heat transfer and the convection phenomena in enclosed plane air layers, Appl. Sci. Res. Sect. A. 3 (1953) 393–409.
- [85] V.P. Isachenko, V.A. Osipova, A.S. Sukomel, Heat Transfer, Mir, Moscow, 1977.
- [86] B. Prahlad, R.D. Kale, K.V.C. Rao, V.M.K. Sastri, Effective Thermal Conductivity of Reflective Multiple Thermal Insulation, Exp. Therm. Fluid Sci. 2 (1989) 375–383.
- [87] M. Mikheev, Heat Transfer, Fountas / Greek Edition, Athens, 1997.
- [88] W.H. Mc Adams, Heat Transmission, Third Edit, Mc Graw Hill, New York, 1954.
- [89] T.A.M. Versteegh, F.T.M. Nieuwstadt, A direct numerical simulation of natural convection between two infinite vertical differentially heated walls scaling laws and wall functions, Int. J. Heat Mass Transf. 42 (1999) 3673–3693. doi:10.1016/S0017-9310(99)00037-X.
- [90] D.S. Wilks, Statistical methods in the atmospheric sciences, Academic Press, New York, 1995.
- [91] C.D. Schunn, D. Wallach, Evaluating goodness-of-fit in comparison of models to data, in: W. Tack (Ed.), Psychol. Der Kognition Reden Vor. Anlässlich Der Emeritierung von Werner Tack, University of Saarland Press, Saarbrücken, Germany, 2005: pp. 115–54.
- [92] E. Mavromatidis, T.I. Lekas, G. Kallos, Analysis of a two-layer cloud system with RAMS model and comparison to airborne observations, Environ. Fluid Mech. 7 (2007) 537–68.
- [93] S. Robert, H. Pashler, How persuasive is a good fit? A comment on theory testing, Psychol. Rev. 107 (2000) 358–67.
- [94] A. Papadopoulos, P. Katsafados, Verification of operational weather forecasts from the POSEIDON system across the Eastern Mediterranean, Nat. Hazards Earth Syst. Sci. 9 (2009) 1299–306.