

Universidade do Minho
Escola de Engenharia

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Numerical Modeling and Assessment
of the Thermal Environment in an
Industrial Premise of Automotive Components



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Mestrado em Engenharia Industrial

Trabalho efectuado sob a orientação de
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Abstract

Design ergonomic and thermally comfortable workplaces is a concern for nowadays industries for several reasons, such as the workers's health and satisfaction which improve their performance. Associated with the obligation by law to provide healthy working conditions, several industries have carried out rigorous thermal environment studies using, for that purpose, mathematical models that predict how (dis)comfortable their employees feel. The use of such models requires physical variables that interact with the human body, such as air velocity, temperature, and humidity at the workplaces under evaluation.

The main purpose of this work was to assess whether the numerical modeling is a suitable framework to characterize and to evaluate the physical component of the thermal environments in industrial contexts. The numerical modeling provides the behavior of the physical variables, helping engineers to study several (real or hypothetical) scenarios and therefore to design optimal thermal environments.

This approach was implemented in the context of an industrial premise for automotive components where the air velocity, temperature and humidity distributions were simulated. The results were compared with experimental values and it was seen that they fall within the simulated values range, indicating that the model is reliable. It was also concluded that the natural ventilation of the building plays a key role in the thermal environment, providing a larger air circulation and cooling the environment that goes beyond the efficiency of the artificial ventilation system installed.

An hypothetical scenario corresponding to an alternative layout was also investigated in this work to prove the usefulness of this innovative approach.

Keywords: Thermal comfort, hot thermal environments, numerical modeling, ANSYS

Resumo

Projectar postos de trabalho ergonómicos e termicamente confortáveis tem merecido grande atenção por parte das indústrias por inúmeras razões, entre elas a saúde e a satisfação dos seus trabalhadores melhorando a sua performance. Associado ao dever legislativo de proporcionar condições de trabalho saudáveis, muitas indústrias têm levado a cabo rigorosos estudos de ambiente térmico recorrendo para o efeito a modelos matemáticos que prevêem o quão (des)confortáveis os seus trabalhadores se sentem. A sua implementação requer variáveis físicas que interagem com o corpo humano, tais como a velocidade, a temperatura e a humidade do ar nos postos de trabalho sob avaliação.

O presente trabalho pretende investigar se a simulação numérica é capaz de proporcionar uma ferramenta auxiliar nos estudos de ambiente térmico. Ao prever o comportamento das variáveis físicas anteriormente referidas, o engenheiro é capaz de estudar inúmeros cenários (reais ou hipotéticos) e dessa forma projectar ambientes térmicos óptimos.

A presente abordagem foi aplicada no contexto de uma nave industrial para componentes automóveis onde foram simuladas as distribuições de velocidade, temperatura e humidade do ar. Os resultados obtidos foram comparados com valores experimentais concluindo-se que estes se enquadram dentro do intervalo de valores simulados e indicando que o modelo é válido. Concluiu-se também que a ventilação natural do edifício tem um papel fundamental no ambiente térmico estudado, proporcionando uma maior circulação de ar e de arrefecimento que vai além daquela proporcionada pelo sistema de ventilação artificial instalado.

Um cenário hipotético correspondendo a uma alteração de layout foi também investigado neste trabalho para provar a utilidade desta abordagem.

Palavras-chave: Conforto térmico, ambientes térmicos quentes, modelação numérica, ANSYS

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Roman symbols

A_c	Area of the object (m^2)	Q_{res}	Global heat flux exchanged through respiratory tract (W/m^2)
A_d	Area of skin surface (m^2)	Q_{sk}	Global heat flux exchanged through skin surface (W/m^2)
C_{res}	Global convective heat flux exchanged through respiratory tract (W/m^2)	R_{sk}	Global radiative heat flux exchanged through skin surface (W/m^2)
C_{sk}	Global convective heat flux exchanged through skin surface (W/m^2)	Re	Reynolds number
C_p	Specific heat coefficient ($J/kg \cdot ^\circ C$)	S_{cr}	Global change rate of temperature (W/m^2)
D	Mass fraction diffusion coefficient	S_{sk}	Global thermal capacity (W/m^2)
D_H	Hydraulic diameter (m)	S_f	External body forces acting on the continuum (N/kg)
E_{res}	Global evaporative heat flux exchanged through respiratory tract (W/m^2)	S_h	Heat source (W/m^3)
E_{sk}	Global evaporative heat flux exchanged through skin surface (W/m^2)	S_w	Species mass fraction source term
F	Heat flux (W/m^2)	T	Temperature ($^\circ C$)
I	Turbulent intensity	T_c	Cold surroundings temperature ($^\circ C$)
I_d	Identity matrix	T_h	Hot body temperature ($^\circ C$)
I_{cl}	Thermal resistance of clothing	U	Velocity vector (m/s^2)
L	Thermal load on the body	W	External work (W/m^2)
M	Metabolic heat generation (W/m^2)	$WBGT_i$	WBGT index computed for period i
M_w	Mass fraction of water vapor in the air	$WBGT_{ankles}$	WBGT index for ankles
P	Pressure (Pa)	$WBGT_{head}$	WBGT index for head
Q	Radiative heat transfer per unit time (W)	$WBGT_{torso}$	WBGT index for torso
		$\bar{T}_r$	Mean radiant temperature ($^\circ C$)
		$^\circ C$	Degree Celsius

f_{cl}	Clothing area factor
h	Human body's height (m)
h_c	Convective heat transfer coefficient (W/(m ² .K))
m	Human body's mass (kg)
p_a	Partial water vapor pressure (Pa)
r	Mass fraction reaction coefficient
t_a	Ambient air temperature (°C)
t_i	Exposure time to the ambient (s)
t_{cl}	Clothing surface temperature (°C)
v_{ar}	Air velocity relative to the human body (m/s)
J	Joule
W	Watt
g	Gram
kg	Kilogram

m	Meter
s	Second

Greek symbols

α	Sensitivity coefficient
κ	Thermal conductivity (W/m.K)
μ	Dynamic viscosity (kg/(m.s))
ρ	Density (kg/m ³)
ε	Emissivity coefficient

Constants

σ	Stefan-Boltzmann constant (5.67×10^{-8} W/m ² .K ⁴)
g	Gravitational acceleration (9.8 m/s ²)

1. Introduction

Nowadays industries are more than equipment, production management, and economic analysis. The **Human component** has been taken a crucial role as almost the totality of the industries still require handwork for their production. Despite that, the management of this component have often been overlooked or reprehended by the managers since it has no direct profits. In this field, one of the perspectives of the **Industrial and Human Engineering** focus on the Human component in industries aiming to guarantee a working environment as comfortable and healthy as possible, providing to the workers motivation while preventing diseases and accidents. In this way, healthy and motivated workers are more willing to produce increasing the company profits.

This document is the result of a great effort to bring together recent research on thermal environment and on numerical simulation, providing an innovative tool for the Industrial and Human Engineering practice.

1.1. Background and Motivation

Thermal comfort studies in human beings have been the subject of significant scientific developments in the recent decades, both from the theoretical point of view and from case studies in workplaces, malls, residential buildings, public spaces, among others [1, 2, 3, 4].

Situations of discomfort or thermal stress may affect human health not only psychologically (malaise, discomfort, feeling of annoyance, irritability, etc.) but also physiologically (increased workload of the heart and respiratory system, heat stroke, exhaustion, nausea, fatigue, fainting, etc.) and pathologically (aggravation of diseases) [5]. Moreover, the performance and the ability to execute simple tasks can also decrease which are reflected in absenteeism and in the lost of function and productivity.

The objective of thermal comfort studies is to establish the necessary conditions to turn a thermal environment appropriate to human occupancy [5]. According to Taleghani [6] that importance is defined by three reasons: provide satisfaction to people, control the consumption of energy, and set standards. In this way, the evaluation of the thermal conditions to which the individuals are exposed, provides crucial information about how they feel and how satisfied they are within the environment. Such information could help managers and engineers to take actions in order to ensure the thermal comfort of people, protecting their health, and design thermally efficient buildings reducing the energy waste from unnecessary or inefficient heating or cooling

systems.

Thermal studies can be carried out both in outdoor and indoor spaces. However, only in recent years the scientific community has turned its attention to thermal study of extreme indoor environments [7]. In fact, the first studies related to extreme thermal conditions were promoted by military exercises, usually associated to extreme climatic conditions [7]. Case studies carried out in expeditions and outdoor work were also an important step in the development and in the understanding of how humans feel in such environments. In this regard, with growing concern to ensure thermal comfort in indoor spaces, several studies have been carried out either in contexts where the thermal sensation of cold is prevalent, as in food industries [7] or surgical rooms [8, 9], and where the thermal sensation of hot is prevalent, as in some offices, in some classrooms, or in industries of automotive components [5, 10].

There are two main approaches to evaluate thermal environments [1, 2]: the rational approach based on studies in climatic chambers and the adaptive approach based on field studies. The Fanger's model is a rational approach to predict thermal comfort in moderate thermal environments based on four physical variables, namely air temperature, air velocity, mean radiant temperature, and relative humidity, and two personal factors, the clothing and the worker metabolic rate.

The four physical variables described in the Fanger's model are usually assessed using appropriate equipment (thermometer, anemometer, etc.) and the measurements follow international standards as ISO 7726 and ISO 9886 [11, 12]. For example, the international standard ISO 7726 recommends three points of measurement (corresponding to the head, torso, and ankles) for each workplace which are then incorporated into a single mathematical relation to predict the thermal comfort. Although such methodologies are quite simple, they can also be quite limited and very time consuming in several situations as large premises, dynamic workplaces, and situations where the physical conditions change over time. As a consequence, the conclusions drawn from a thermal study may be inaccurate or represent only a short period of time in a small number of workplaces.

Computational Fluid Dynamics, usually abbreviated as CFD, is a branch of fluid mechanics theory that uses mathematical modeling, numerical analysis, and algorithms to solve and analyze problems that involve fluid flows [13]. Computers are used to perform the calculations required to simulate the interaction of liquids and gases with surfaces defined by boundary conditions. The physical variables in thermal environments according to the Fanger's model (air temperature, air velocity, mean radiant temperature, and relative humidity) are intrinsically related to the air whose behavior can be modeled by partial differential equations (PDEs) widely studied in CFD. In this way, CFD software can be used to virtually reproduce the physical component of the thermal environment and therefore assess the thermal comfort of humans [14, 15]. This innovative methodology is particularly useful to evaluate the thermal conditions in large premises and long

periods of time by just changing the simulation parameters. Furthermore, it can also be used to test several hypothetical scenarios helping engineers to design or to redesign thermally efficient buildings and premises.

There exists an extensive literature on CFD simulation for indoor thermal environments, such as residential buildings [16, 17, 18], classrooms [19], offices [20], or surgical rooms [21, 22]. However, few studies are found in industrial contexts [23, 24, 25] where thermal conditions may be particularly severe. For this reason, extending the numerical study for the characterization and the understanding of thermal environments in indoor industrial contexts is innovative and of practical interest.

Furthermore, there are some authors who have integrated a human thermal model into CFD simulation tools [26, 27] to directly evaluate the skin surface temperature or the skin relative humidity providing more realistic and relevant data to assess the thermal sensation. Despite its importance, this issue was not addressed in this work.

1.2. Case Study Presentation

Continental A.G. is an international company founded in 1871, Hanover, and it is located worldwide in 49 countries and four continents, namely Africa, Asia, Europe, and America. This well known company engaged in the activity of automotive components production being one of the largest world producers of tires.

The vision of the company is to develop intelligent technologies for transport and mobility, providing good solutions for customers. The mission is to invent, develop, produce, and market technological solutions and generate value maintaining the highest of quality standards and providing solutions that benefit the society with faster progress and respect for the environment and protection of life. In this way, Continental accomplishes its mission efficiently, effectively and innovatively [5, 28] becoming mobility and transport safer, more comfortable, and more sustainable. Finally, the values of the company represent four main goals: trust, passion to win, freedom to act, and for one another [28].

Indústria Têxtil do Ave. S.A., also known as Continental ITA or C-ITA, is a Portuguese factory located in Lousado (see Fig. 1.1) which belongs to the family of Continental group. The factory was founded in 1948 and inaugurated in 1950 by Henrique Malheiro Dias as a consequence of the first tire factory in Portugal, Mabor [5]. C-ITA produces textile reinforcements for tires being initially used cotton as a feedstock, which was replaced in 1958 by high tenacity rayon and, years later, by synthetic fibers as polyamide and nylon. These materials provide unique features to the tire, improving security, comfort, and the best direction conditions possible which is of crucial important since the tire is the interface between the vehicle and the floor.



Figure 1.1: Continental – Indústria Têxtil do Ave. S.A. (C-ITA), located in Lousado, Vila Nova de Famalicão.

C-ITA produces around 15000 tons of textile reinforcements per year, working 24 hours a day, 7 days a week, with various work schedules: normal rotation, fixed rotation and continuous labor. It employs 172 workers and has a total area of 52000 m², of which 30000 m² are covered. The company is certified with the international standards ISO 9001 and ISO 14001 [5].

The productive process of the textile reinforcements is divided in three main sections: twisting, weaving and dipping [5]. During the twisting process, the yarns are twisted in Z or S formats, in other words, interlace them clockwise or anti-clockwise, respectively [29]. If the yarns do not have the appropriate dimensions to introduce in the twister machines, they are rewound in a process called winding. In this stage, it is also possible to remove irregularities that the yarn may contain. The goal of the twisting process is to produce a rope with better properties comparatively to the yarn, such as higher cohesion, tenacity, and strength (see Fig. 1.2). Then, in the weaving process the rope is interlaced in transverse and longitudinal direction in loom machines, obtaining the textile fabric. The textile fabric obtained is wound on a bobbin of large dimensions that become in a roll [5]. Finally, after a physical and chemical bath to the fibers, the rolls are impregnated in the rubber using ZELL machines, process called dipping. The bath is of crucial importance to confer the fibers a strong adhesion to the rubber in the tire production which otherwise would not adhere. This process can also be performed on the rope using SingleEnd machines with DC twister machines [29].

Some research projects have been carried out in C-ITA namely the work of Torrinha (2011) [30] concerning a energy audit of SingleEnd and Zell machines, the work of Oliveira (2012) [29]



Figure 1.2: Yarn (left panel) and rope (right panel).

concerning an energy optimization of the ventilation system, and the work of Guise (2014) [5] concerning an experimental thermal environment assessment. The works of Oliveira (2012) and Guise (2014) were carried in one of C-ITA's premise, called Nave 2, dedicated to the twisting process (see Fig. 1.3).



Figure 1.3: Premise Nave 2 in C-ITA in red lines.

In this premise, a huge amount of heat is constantly generated by the twisting machines increasing the air temperature. Guise (2014) has performed several experimental measurements at 29 points and implemented a Thermal Sensation Questionnaire to the workers in order to assess their thermal sensation within the environment. With these results, Guise (2014) has concluded that all the workers were uncomfortable with the thermal environment evidencing that measures have to be taken or implemented in order to protect the worker's health.

Based on the conclusions drawn by Guise (2014), the author of this work has considered of

great interest the modeling and the numerical simulation of the thermal environment in Nave 2 in order to understand the underlying phenomena and to outline engineering measures to improve the premise ventilation and, therefore, the worker's thermal sensation. This case study is tackled later in this document and was supported by the works of Oliveira (2012) and Guise (2014).

1.3. Objectives and Research Issues

The main purpose of this work was to assess whether the numerical modeling is a suitable framework to characterize and to evaluate the physical component of thermal environments in industrial premises.

To investigate this issue and draw the conclusions, the following steps were addressed in this work:

- (i) Identify the physical variables and the related physical phenomena affecting the thermal environment in industrial contexts;
- (ii) Propose a mathematical model based on partial differential equations which describes the physical component of the thermal environment;
- (iii) Investigate a real context and assess the associated physical variables by solving the proposed model with numerical procedures;
- (iv) Compare the obtained solution against experimental data collected in the premise in order to validate the model; conclude about the capability of the numerical modeling to assess the physical component of thermal environments in industrial contexts.

1.4. Organization

The document is divided in five main parts where the research topics introduced in section 1.3 were addressed.

After the introduction and motivation, a literature review on moderate and on hot thermal environments and numerical modeling is presented in Chapter 2 with the purpose to address the research topics (i) and (ii), respectively. Moreover, a brief discussion on international standards for the evaluation of thermal environments is provided as well as some theoretical context and background concerning different approaches or minor topics in order to give the reader a general overview.

Then in Chapter 3, a real context taking place in a premise of the C-ITA company is investigated following the research topic (iii). The methodology adopted to assess its thermal

environment, both experimental and numerical, is presented and discussed in this Chapter.

Research topic (iv) is addressed in Chapter 4 where the results obtained for the case study are shown, compared to the experimental ones, and discussed in order to validate the model and the numerical procedure.

The document ends with Chapter 5 where the main conclusions are drawn and the further work is proposed in order to pursue with this research.

2. Literature Review

This section provides a theoretical introduction of the addressed topics in this work both on thermal environments, in sections 2.1, 2.2, and 2.3, and on numerical modeling in section 2.4.

2.1. Environment–Human Thermal Interaction

The human body is constantly generating energy from nutrients by biochemical processes called metabolism. The energy which exceeds the body's needs is converted to heat and released to the environment to maintain the body's core temperature fairly constant [4]. In this way, some authors describe four main processes by which the human body interacts with the surrounding environment: conduction, convection, radiation, and evaporation [4].

Conduction is related to the heat transfer between the skin surface and the objects in contact with it while convection is related to the heat transfer due the ambient air movement [4]. If the skin surface temperature is higher than the temperature of the objects or the ambient air in contact with it, the heat is mainly transferred to outside the human body. Otherwise, the process occurs in the opposite direction increasing the body's heat load. On the other hand, radiation is the transmission of heat through the environment by electromagnetic waves between the body and the walls, other bodies, and objects near to it [4]. Finally, evaporation corresponds to the body's heat loss and occurs at respiratory and skin levels. Note that evaporation is a cooling mechanism which does not occur in the opposite direction, that is, heat gain, as opposed to radiation, conduction or convection, and the losses through this mode increases at high activity levels [5].

The heat balance equation describes the thermal exchanges between the human body and the surrounding environment. If a thermal balance exists the amount of heat produced by the metabolism must be equal to the energy required for the physical activity (and eventually shivering) plus the heat loss to the environment [2]. That means, the amount of energy produced by the metabolism which is not used for external work (or shivering) must be transferred to the environment such that the body's temperature remains constant (steady-state situation). Otherwise, the extra heat is stored causing the body's temperature to rise.

For steady state situations, the conduction effects can be neglected since the thermal conductivity of the air is very small. Even if a large part of the body's area is in contact with a wall or a furniture, for example, the thermal equilibrium between skin and air rapidly occurs through that object [2]. On the other hand, since the respiratory tract does not interact with any external surface, its radiative effects are not taken into account for the heat balance equation. That is, one

assume that no radiation is transferred from the respiratory tract to an external surface.

Given that, the heat balance equation for a human body can be expressed mathematically by [31]

$$M - W = Q_{sk} + Q_{res} + S = (C_{sk} + R_{sk} + E_{sk}) + (C_{res} + E_{res}) + (S_{sk} + S_{cr}), \quad (2.1)$$

where

- M – metabolic heat generation (W/m^2);
- W – external work (W/m^2);
- Q_{sk} – global heat flux exchanged through skin surface (W/m^2);
- Q_{res} – global heat flux exchanged through respiratory tract (W/m^2);
- C_{sk} – global convective heat flux exchanged through skin surface (W/m^2);
- R_{sk} – global radiative heat flux exchanged through skin surface (W/m^2);
- E_{sk} – global evaporative heat flux exchanged through skin surface (W/m^2);
- C_{res} – global convective heat flux exchanged through respiratory tract (W/m^2);
- E_{res} – global evaporative heat flux exchanged through respiratory tract (W/m^2);
- S_{sk} – global thermal capacity (W/m^2);
- S_{cr} – global change rate of temperature (W/m^2);

The area of the skin surface, A_d (m^2), can be estimated by the DuBois area formula proposed by Fanger and expressed as

$$A_d = 0.2025 m^{0.425} h^{0.725}, \quad (2.2)$$

where m is the body's mass (kg) and h is the body's height (m).

The heat balance equation was the basis for the Fanger's model to predict thermal sensation and it is totally accepted and followed by ISO 7730 for the study of the comfort conditions regardless of the region. Those topics are addressed in subsections 2.2.2 and 2.2.3, respectively.

2.2. Moderate Thermal Environments

2.2.1. Thermal Comfort

The human body of an healthy person maintains a fairly constant temperature of its internal organs at 37.0 ± 0.5 °C regardless of the environmental conditions. Due to the metabolism, it

also generates around 40 W of heat during sleep to 500 W during hard exercise [1]. Depending on the environmental conditions, the human body must transfer some of this heat outside the body or, on the other side, avoid it as longer as possible so the core temperature ranges between the healthy limits.

Those complex interactions between the outside environment and the human body are carefully controlled by the thermoregulatory system through thermoreceptors located in different parts of the body sending signals about the local temperature level and its change to the hypothalamus. In turn, the hypothalamus reacts to the signals by means of thermoregulatory functions by either inhibiting or enhancing heat production and heat loss through the increase of the skin blood flow rate (vasodilation), the decrease of the skin blood flow rate (vasoconstriction), sweating and shivering. The response of the thermoreceptors provide to the brain a sense of the skin temperature which is called thermal sensation.

In 1962, Macpherson defined six factors as those affecting thermal sensation: four physical variables including air temperature, air velocity, relative humidity, and mean radiant temperature, and two personal variables including clothing insulation and activity level (or metabolic rate) [2]. However, some authors state that it is not possible to define thermal sensation in terms of physical or psychological variables since it is also related to how people feel [32].

From the physiological point of view, the thermal comfort is achieved when there is a balance between the heat exchanged by the human body and the surrounding environment, leading to a minimization of the thermoregulation effort. In that case, the skin moisture is low and its temperature is held within narrow ranges [2]. The maintenance of a such physiological state is necessary to characterize the human comfort but it is not universally accepted that an individual holding such conditions is thermally comfortable.

In fact, many authors around the world have been defining thermal comfort in a more comprehensive way, sometimes according to their own perspectives. For example, Hesen has defined thermal comfort as a state in which there are no driving impulses to correct the environment by the behavior [2] and the ASHRAE (American Society of Heating, Refrigerating and Air-Conditioning Engineers) as the condition of the mind in which satisfaction is expressed with the thermal environment.

While the first definition assumes the comfort as a state condition depending on physical and physiological factors, both the Hense and the ASHRAE point of view also comprises the state of mind, describing thermal comfort as a synthesized feeling about the body's thermal state where the social, cultural and psychological factors are the main players of the overall process. Of course, it leaves open what condition of mind or satisfaction mean but, as state by Noël Djongyang, it correctly emphasizes that the judgment of comfort is a cognitive process involving many inputs influenced by physical, physiological, psychological, and other factors [2].

Predict the human satisfaction within a thermal environment requires analyzing a complex system of several interacting variables which lead to a subjective response in the individual. That system includes not only measurable variables, such as environment- and individual-related variables, but also psychological, social, and cultural ones, such as thermal perception and individual preferences, which are less tangible difficulting the thermal comfort assessment. According to Djongyang [2], comfort also depends on behavioral actions such as opening/closing a window, turn on/off the air-conditioner, altering clothing, and changing posture or location.

Assessing the thermal comfort of an individual subjected to certain environmental conditions is of crucial importance in order to take measures to protect the human health, specially when the exposure time is long as often occurs in industrial workplaces. In practice, the calculation of the thermal comfort is not possible since it comprises a complex system of not only measurable variables but also non-measurable ones. In this way, different philosophies and methodologies to predict the human thermal comfort have arisen, namely the rational and the adaptative approaches.

The rational approach, also known as the heat-balance approach, is supported on data collected from climate chambers, where individuals dressed in standardized clothing and doing standardized activities, are exposed to different thermal conditions. In an evaluation performed by Doherty and Arens (1988) [33], it was shown that these kinds of models, like the comfort model of Fanger [34] and the two-node model of Gagge et al. [2], are accurate for humans involved in near-sedentary activity and steady-state conditions.

On the other hand, the adaptative approach is supported by field studies of people in buildings, having the purpose of analyzing the real acceptability of thermal environments in the actual workplace, letting the subjects behave without any additional restrictions, wear their everyday clothing, and have their everyday habits. In this way, the adaptative approach collects more reliable information about how relevant parameters – which strongly depends on the context – the behavior of occupants, and their expectations – which cannot be simulated in climate chambers – interact and change the individuals thermal sensation.

2.2.2. Indices and Models

The Fanger's model is a rational approach developed in 1970 [34] to predict thermal comfort based on the six factors proposed by Macpherson in 1962 and the two-node model of Gagge et al. [2]. Fanger carried his experiments in climate chambers where people dressed in standardized clothing and completed standardized activities were exposed to different thermal environments. He then asked the participants to record how hot or cold they felt according to the seven-point ASHRAE thermal sensation scale (see Table 2.1) [35] or to adjust the temperature themselves until they felt thermally comfortable.

The model developed by Fanger combines theories of heat balance with the environment and

Table 2.1: Seven-point ASHRAE thermal sensation scale

Index	Thermal Sensation
3	Hot
2	Warm
1	Slightly warm
0	Neutral
-1	Slightly cool
-2	Cool
-3	Cold

the physiology of thermoregulation to determine a range of temperatures in which the occupants of a building will find comfortable. To be able to predict such conditions, he investigated the body's physiological processes when it is close to neutral thermal sensation and he concluded that the only physiological processes influencing heat balance in this context were the sweat rate and the mean skin temperature, and that these processes were a function of the activity level [2]. He developed a linear relationship between activity level and sweat rate and also between activity level and mean skin temperature based on the data collected in climate chambers with people wearing standardized clothing at four different activity levels (sedentary, low, medium, and high). He then substituted these two linear relationships into the heat balance equation to obtain an equation to predict the comfort range.

Based on subsequent studies, Fanger expanded its equation describing thermal comfort as the balance between the actual heat flow from the body in a given thermal environment and the heat flow required for optimum (neutral) comfort for a given activity [2]. This equation was also able to relate thermal conditions to the seven-point ASHRAE thermal sensation scale introduced before, and was called as the Predicted Mean Vote index, abbreviated as PMV, and was later incorporated into the Predicted Percentage of Dissatisfied index, also known as PPD.

The PMV index is calculated as [2]

$$PMV = (0.303 \exp(-0.036M) + 0.028) L = \alpha L, \quad (2.3)$$

where M is the metabolic heat generation (W/m^2), α is the sensitivity coefficient, and L is the thermal load on the body defined as the difference between internal heat generation and heat loss to the environment for a person, given by

$$\begin{aligned} L = & (M - W) - 3.05 \times 10^3 [5733 - 6.99(M - W) - p_a] \\ & - 0.42 [(M - W) - 58.15] - 1.7 \times 10^5 M (5867 - p_a) \\ & - 0.0014 M (34 - t_a) - 3.96 \times 10^{-8} f_{cl} [(t_{cl} + 273)^4 - (\bar{t}_r + 273)^4] \\ & - f_{cl} h_{cl} (t_{cl} - t_a), \end{aligned} \quad (2.4)$$

where

- p_a – partial water vapor pressure (Pa);
- f_{cl} – clothing area factor (ratio of clothed/nude surface area);
- t_a – ambient air temperature ($^{\circ}\text{C}$);
- $\bar{t}_r$ – mean radiant temperature ($^{\circ}\text{C}$);
- h_c – convective heat transfer coefficient ($\text{W}/(\text{m}^2 \cdot \text{K})$);
- t_{cl} – surface temperature of clothing ($^{\circ}\text{C}$);

The clothing area factor, f_{cl} , is given by

$$f_{cl} = \begin{cases} 1.00 + 1.290I_{cl} & \text{if } I_{cl} \leq 0.078 \text{ m}^2\text{K/W} \\ 1.05 + 0.645I_{cl} & \text{if } I_{cl} > 0.078 \text{ m}^2\text{K/W} \end{cases}, \quad (2.5)$$

where I_{cl} is the thermal resistance of clothing. The convective heat transfer coefficient, h_c , is given by

$$h_c = \begin{cases} 2.38|t_{cl} - t_a|^{0.25} & \text{if } 2.38|t_{cl} - t_a|^{0.25} > 12.1\sqrt{v_{ar}} \\ 1.21\sqrt{v_{ar}} & \text{if } 2.38|t_{cl} - t_a|^{0.25} < 12.1\sqrt{v_{ar}} \end{cases}, \quad (2.6)$$

where v_{ar} is the air velocity (m/s) relative to the human body. At last, the surface temperature of clothing, t_{cl} , is given by

$$t_{cl} = 35.7 - 0.028(M - W) - I_{cl} [3.96 \times 10^{-8} f_{cl} [(t_{cl} + 273)^4 - (\bar{t}_r + 273)^4] + f_{cl} h_c (t_{cl} - t_a)]. \quad (2.7)$$

The PPD index represents the percentage of people uncomfortable with the thermal environment. Using the seven-point scale of thermal sensation, Fanger stated [36] the condition of discomfort for those who responded ± 2 and ± 3 , that is, those who felt more than slightly warm or slightly cold. Those who responded ± 1 or 0 are considered comfortable.

The relationship between the PPD index and the PMV index is given by [2]

$$PPD = 100 - 95 \exp(-0.03353PMV^4 + 0.2179PMV^2), \quad (2.8)$$

and it is graphically shown in Fig. 2.1 (retrieved from [2]).

First, it can be seen that the relationship is symmetric with respect to the thermal neutrality, that is, $PMV=0$, and second, even with a neutral thermal environment there is a small percentage of 5% of individuals who felt uncomfortable (even knowing they are dressed in a similar way and

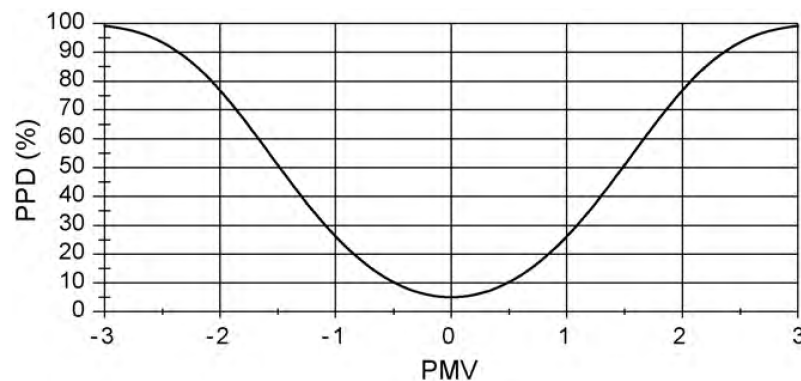


Figure 2.1: Relationship PMV versus PPD.

that the level of activity is the same). In fact, since each individual has their own physiological characteristics, it's almost impossible to guarantee the thermal comfort for all the people. Therefore, when one tries to optimize the thermal comfort, a small percentage of people dissatisfied is always expected.

The drawback of the PMV-PPD model is that it is useful only to predict steady-state comfort responses which never precisely occurs in daily life [2] and, therefore, it is only suitable for uniform thermal environments close to thermal neutrality. Years later, some authors developed models based on the transient heat balance equation which allow to predict responses of individual exposed to transient situations. For example, the two-node model developed by Gagge et al. in 1986 [37] or, more recently, the Zhang's thermal model developed by Zhang et al. in 2003 [38]. The first one divides the body into two concentric cylinders (one for the body's core and the other for the skin), each one of them with different temperatures of 37.1 °C and 33.1 °C, respectively. The Zhang's model represents the overall thermal sensation and comfort as a function of the local skin temperatures, the core temperature, and their change over time, while the local thermal sensation is calculated by a logistic function of the local skin temperature. Moreover, the PMV-PPD model becomes more suitable and accurate to predict the mean response according to the ASHRAE thermal sensation scale when the group of people under study is larger.

Despite of the drawbacks of the PMV-PPD model, it is widely used in international standards to predict thermal sensation and comfort as in ISO 7730 (2005) and in ASHRAE 55 (2004) [39].

2.2.3. Normalization

The PMV-PPD index is incorporated into the international standard ISO 7730 (2005) – Ergonomics of the thermal environment - Analytic determination and interpretation of thermal comfort using calculation of the PMV and PPD indices and local thermal comfort criteria – and is a widely used standard to evaluate the conditions for thermal comfort of people exposed to moderate

thermal environments. The standard enables the analytic determination and interpretation of the overall and local thermal comforts calculating the PMV and PPD indices and specifying the acceptable thermal environment conditions for comfort as well as those representing thermal discomfort.

The PMV-PPD model is derived for steady-state situations, the ISO 7730 states that it can also be applied with a reasonable approximation for environments with minor fluctuations of one or more variables provided that time-weighted averages of the variables during the previous 1 hour period are applied [40]. The standard also recommends that the PMV index is used only if the following conditions are satisfied:

- PMV values between -2 and $+2$;
- metabolic heat generation rate values between 46 W/m^2 and 232 W/m^2 ;
- thermal resistance of clothing between $0 \text{ m}^2\text{°C/W}$ and $0.310 \text{ m}^2\text{°C/W}$;
- ambient air temperature between 10 °C and 30 °C ;
- mean radiant temperature between 10 °C and 40 °C ;
- relative air velocity between 0 m/s and 1 m/s ;
- partial vapor pressure of water between 0 Pa and 2700 Pa .

In 1998, deDear [41] concluded that the PMV index overestimates the warmth sensations in warm naturally ventilated buildings, after gathered over 20000 individual comfort votes and the corresponding measurements of the thermal environmental conditions in approximately 160 buildings. Moreover, Humphreys has shown in 2000 [42] that the PMV index is more influenced by the mean temperature of the accommodation, that is, air temperature and the globe temperature, than the comfort votes and later, in 2002 [43], that the PMV index overestimates the warmth sensation and underestimates the cooling effect of increased air movement.

To sum up, the PMV-PPD index progressively overestimates the mean perceived warmth of warmer environments and the coolness of cooler environments [39] which can lead to the provision of unnecessary cooling or warming. The PMV index is valid for everyday prediction of the comfort vote only under several restricted conditions [43].

2.3. Hot Thermal Environments

2.3.1. Thermal Stress

According to Health and Safety Executive [44], heat stress occurs when the body's means to control its internal temperature starts to fail. In this situation, the combined contributions of

environmental factors, metabolism, and clothing to which a person may be exposed, result in an increase storage of heat in the body and, therefore, in an increase of its temperature.

As stated before, the thermal stress can affect the human health psychologically (malaise, discomfort, feeling of annoyance, irritability, etc.), physiologically (increased workload of the heart and respiratory system, heat stroke, exhaustion, nausea, fatigue, fainting, etc.), and pathologically (aggravation of diseases) [5]. The risk of those situations increases when the worker is not acclimatized and can lead to decreased performance and ability to execute, that are reflected in absenteeism and loss of productivity. The effects of the temperature increasing on an individual's health and efficiency is summed up in Fig. 2.2 (retrieved from [45]).

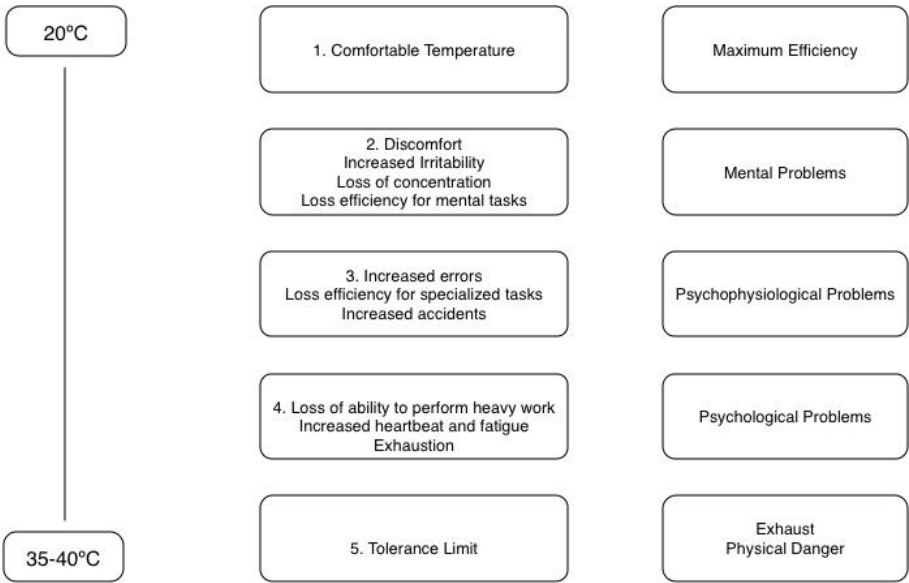


Figure 2.2: Effects of the temperature increase in human's health and efficiency.

The goals of the thermal stress assessment are the determination of the factors that cause it and the measurement of its intensity in terms of effects on health, comfort and performance [46].

2.3.2. Indices and Models

Several indices for hot thermal environments have been developed trying to better characterize the thermal conditions which lead to thermal stress on human beings. In table 2.2 (translated from [4]) are shown some of the main indices used in this context and their differences.

The WBGT index is widely used in industrial contexts for its simplicity and speed to assess thermal stress risks in hot thermal conditions. Unfortunately, in some situations the use of the WBGT index is not possible (or appropriate) namely when a high accuracy is required, the exposure period to such conditions is short, the thermal conditions are close to comfort, the values of the variables related to the risk need to be assessed, the individuals are not healthy, adapted to heat or whose clothes have a thermal resistance far from 0.6 clo [4]. Due to those limitations,

Table 2.2: Indices for hot thermal environments.

Index	Name	Developed by	Method	Type
EC	Equatorial comfort index	Webb (1960)	Correlation between temperature, pressure and air velocity with saturated and stagnated air temperature	Thermal sensation
ET	Effective temperature	Houghten et al. (1923)	Based on wet and dry bulb temperatures and wind velocity	Thermal sensation
ET*	Rectified effective temperature	Vernon and Warner (1932)	Uses globe temperature instead of dry bulb temperature to take into account the radiation effects	Thermal sensation
HSI	Heat stress index	Belding and Hatch (1955)	Based on thermal balance	Physiological effort
HU	Humidex	Masterton and Richardson (1979)	Provides an equivalent temperature depending on air temperature and relative humidity	Physiological effort
P4SR	Predicted 4 hours sweat rate	McAriel et al. (1947)	Based on the evaluation of physiological responses when exposed to certain thermal conditions for 4 hours	Physiological effort
RT	Resulting temperature	Missenard (1948)	Based on similar experiences to those for ET	Thermal sensation
SWreq	—	Vogt et al. (1981)	Based on the required sweat rate from HSI and ITS (follows ISO 7933 norm)	Physiological effort
WBGT	Wet bulb and globe temperature index	Yaglou and Minard (1957)	Based on globe and wet bulb temperatures (follows ISO/DIS 7243 1982 norms)	Physiological effort

it is recommended to use the WBGT index as a first assessment of the thermal conditions and, therefore, use another more complex and accurate index.

The WBGT index takes into account the thermal environmental conditions and the heat generated by the body due the physical activity when the individuals are dressed with regular summer clothing, in order to assess the thermal stress risk which leads to a body temperature larger than 38 °C [4]. The WBGT index is usually expressed in °C and is given by [4]

$$WBGT = 0.7t_{rw} + 0.3t_g, \quad (2.9)$$

for interior environments or exteriors environments without sun exposure, and as

$$WBGT = 0.7t_{nw} + 0.2t_g + 0.1t_a, \quad (2.10)$$

otherwise, where t_{nw} is the natural wet bulb temperature ($^{\circ}\text{C}$), t_g is the globe temperature ($^{\circ}\text{C}$), and t_a is the ambient air temperature ($^{\circ}\text{C}$).

If the worker is exposed to different thermal condition during different periods of time, the average WBGT index must be computed as

$$WBGT_{\text{average}} = \frac{\sum_{i=1}^n WBGT_i t_i}{\sum_{i=1}^n t_i}, \quad (2.11)$$

where $WBGT_i$ is the WBGT index computed for period $i = 1, \dots, n$ corresponding to an exposure of t_i units of time.

On the other hand, if the thermal conditions are not homogeneous, the index must be computed considering three different heights – head, torso, and ankles – using the following expression

$$WBGT = \frac{1}{4} (WBGT_{\text{head}} + 2WBGT_{\text{torso}} + WBGT_{\text{ankles}}). \quad (2.12)$$

If the worker is standing, the measurements to compute $WBGT_{\text{head}}$, $WBGT_{\text{torso}}$, and $WBGT_{\text{ankles}}$ must be carried out at 170 cm, 110 cm, and 10 cm from the ground, respectively. If the worker is sitting, they must be carried out at 110 cm, 60 cm, and 10 cm from the ground, respectively.

2.3.3. Normalization

The international standard ISO 7243 (1989) – Hot environments - Estimation of the heat stress on working man, based on the WBGT-index (wet bulb globe temperature) – provides guidelines to evaluate the stress on a individual exposed to hot thermal conditions using the WBGT index. The norm also suggests that the WBGT index does not apply to very short periods nor to zones of comfort, as stated before. The norm defined the upper limit values for the WBGT index depending on the individual's metabolic rate, which are given in table 2.3 [47].

The ISO 8996 (2004) [48] – Ergonomics of the thermal environment - Determination of metabolic rate – provides methods to assess the individual's metabolic rate according to: the occupation and activity (rough information and very great risk of error); group assessment tables and tables for specific activities (high error risk and accuracy of $\pm 20\%$); heart rate measurements under defined conditions (medium error risk and accuracy of $\pm 10\%$); measurements of oxygen consumption (low error risk and accuracy of $\pm 5\%$).

Table 2.3: WBGT limit values for different ranges of metabolism.

Metabolism class	Metabolism (M)		WBGT limit value	
	In relation to skin surface area (W/m^2)	Considering 1.8 m^2 of skin surface area (W)	For a person acclimatized to heat	For a person not acclimatized to heat
0	$M < 65$	$M < 117$	33	32
1	$65 < M < 130$	$117 < M < 234$	30	29
2	$130 < M < 200$	$234 < M < 360$	28	26
3	$200 < M < 260$	$360 < M < 468$	25 (or 26*)	22 (or 23*)
4	$M > 260$	$M > 468$	23 (or 25*)	18 (or 20*)

* If the air movement is felt

The ISO 9886 (2004) [11] – Ergonomics - Evaluation of thermal strain by physiological measurements – provides methods to measure thermal strain in hot thermal environments using physiological measurements as the body's core temperature, skin surface temperature, heart rate and loss of body's weight, and its interpretation.

The ISO 7933 (2004) [49] – Ergonomics of the thermal environment - Analytic determination and interpretation of heat stress using calculation of the predicted heat strain – provides methods to evaluate hot thermal conditions using the SWreq index (see table 2.2).

2.4. Numerical Modeling of Thermal Environments

As stated in subsections 2.2.2 and 2.3.2, the thermal comfort assessment requires the evaluation of the environmental conditions, that is the physical variables interacting with the human body, to which the workers are exposed. Such information is incorporated in thermal models, such as the PMV-PPD and the WBGT indices, which predict how comfortable or uncomfortable the workers feel. For instance, the PMV-PPD index requires the air temperature, air velocity, mean radiant temperature, and relative humidity while the WBGT index requires the globe temperature, wet bulb temperature, and air temperature. Notice that, in addition to the physical variables such models also take into account personal factors such as metabolism or clothing.

Whatever the index to use, the values of such physical variables are usually measured using specific equipment such as thermometer, globe thermometer, wet bulb thermometer, or anemometer. When the thermal environment comprises a large area with several workplaces, the measurement procedure can be very time consuming and the evaluation of the associated thermal environment can be inaccurate and limited as explained in section 1.1.

On the other hand, the numerical study of the physical component of thermal environments provides an alternative approach by simulating the behavior of the physical variables according to the laws of physics. In this way, one can investigate the interaction between different variables and assess their causes and their impact on the thermal environment. Moreover, hypothetical scenarios

can also be investigated using this approach which may help engineers to design comfortable and energy-efficient buildings and workplaces, as motivated in section 1.1.

Roughly speaking, the numerical modeling in the context of thermal environments comprises the following steps, represented in Fig. 2.3:

1. **Modeling** – consists in describing the physical component of the thermal environment in terms of partial differential equations (PDEs) which relate the physical variables; in this work only the air velocity, temperature, and relative humidity are considered;
2. **Discretization** – the continuous model defined before is transformed into in a set of linear arithmetic equations using numerical methods which correspond to local approximations of the PDEs;
3. **Solve** – the linear system consisting in the equations derived from the discretization step is solved using computers; the solution corresponds to the approximated distribution of the physical variables considered in the thermal environment;
4. **Validation** and **Actions** – comparisons between the numerical and experimental results are performed in order to validate the model; hypothetical scenarios may be tested as well as engineering measures in order to find an optimal thermal environment.

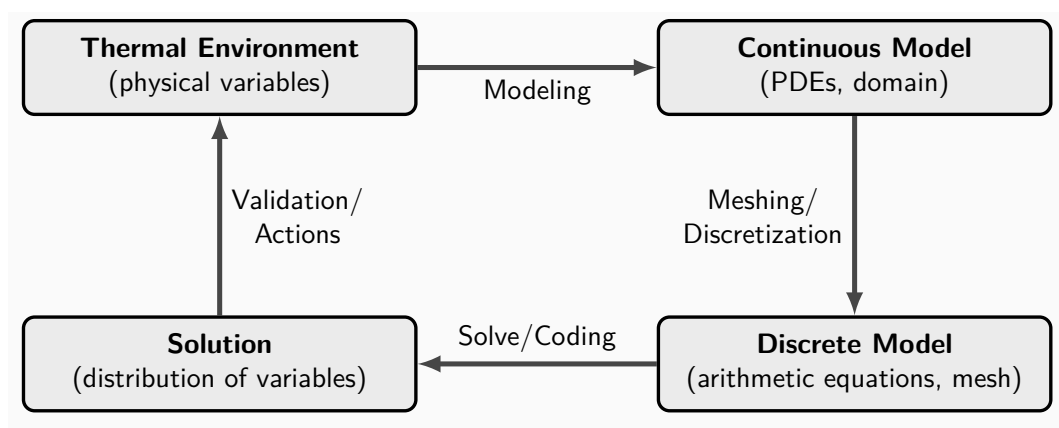


Figure 2.3: Steps of the numerical procedure.

2.4.1. Modeling

There are two main phenomena which describe the physical component of the thermal environment: heat transfer and mass transfer. The heat transfer describes how the thermal energy is transferred from one place to another while the mass transfer describes the fluid motion which may contain substances, also called species. Both phenomena are the basis of CFD problems and can modeled by partial differential equations complemented with boundary conditions.

The heat transfer in fluids occurs by conduction, convection, and radiation which can be modeled by the energy equation. Considering a domain Ω with boundary $\partial\Omega$, the general form of the steady-state energy equation writes [13]: find T such that

$$\nabla \cdot (\rho C_p U T - \kappa \nabla T) = S_h, \quad \text{in } \Omega, \quad (2.13)$$

where T is the temperature ($^{\circ}\text{C}$), ρ is the density (kg/m^3), C_p is the heat capacity ($\text{J}/(\text{kg}\cdot\text{K})$), $U = (U_1, U_2)$ is the velocity vector (m/s), κ is the thermal conduction coefficient ($\text{W}/(\text{m}\cdot\text{K})$), and S_h is a heat source (W/m^3). The fluid properties ρ , C_p , and κ can depend on the temperature itself. For example, the density of the air decreases when heated affecting the temperature distribution.

The energy equation (2.13) is presented in the form of a simple convection-diffusion equation but it may contain more terms and be written in equivalent forms. For a comprehensive literature on the subject see [13] or [50]. The divergence of the term $\rho C_p U T$ represents the convection transfer while the divergence of the term $-\kappa \nabla T$ represents the diffusion transfer, respectively. As stated before, the equation is solved with boundary conditions defined on the boundary of the domain, $\partial\Omega$, and can be of different kinds. For example, a Dirichlet condition means the temperature on the boundary is known while a Neumann condition means the heat transfer is known. On the other hand, a symmetry boundary conditions implies that there is no heat transfer through the boundary.

The source term S_h introduces into the equation energy sources as the radiation transfer between two bodies. The classical models found in the literature to take into account the radiation effects derive from the Stefan-Boltzmann law which writes: if an hot object is radiating energy to its cooler surroundings, the net radiation heat loss rate can be expressed as

$$Q = \varepsilon \sigma (T_h^4 - T_c^4) A_c, \quad (2.14)$$

where Q is the heat transfer per unit time (W), ε is the emissivity coefficient, σ is the Stefan-Boltzmann Constant ($5.67 \times 10^{-8} \text{ W}/\text{m}^2\cdot\text{K}^4$), T_h is the hot body temperature ($^{\circ}\text{C}$), T_c is the cold surroundings temperature ($^{\circ}\text{C}$), and A_c is the area of the object (m^2).

The mass transfer associated to the fluid motion is modeled by the momentum and the mass continuity equations which can be complemented with turbulence equations as the $\kappa - \epsilon$ or the $\kappa - \omega$. Considering a domain Ω with boundary $\partial\Omega$, the general form of the steady-state momentum equation, also known as Navier-Stokes equation, writes [13]: find U and P such that

$$\nabla \cdot (\rho U \otimes U - \mu \nabla U + P I_d) = \rho g + S_f, \quad \text{in } \Omega, \quad (2.15)$$

where μ is the dynamic viscosity ($\text{kg}/(\text{m}\cdot\text{s})$), P is the pressure (Pa), g is the gravity, S_f is external

body forces acting on the continuum (N/kg), and I is the identity matrix in $\mathbb{R}^{3 \times 3}$. The term ρg represents the gravitational body forces which induces natural convection. The fluid properties μ and ρ can depend on the temperature of the fluid. In this context, the term S_f is not considered since there are no external forces.

The continuity equation is given by

$$\nabla \cdot (\rho U) = 0, \quad \text{in } \Omega. \quad (2.16)$$

and may contain more terms and be written in equivalent forms as well as the momentum equation (2.15). For a comprehensive literature on the subject see [13] or [50].

As for the energy equation (2.13), the system of equations (2.15–2.16) is equipped with boundary conditions defined on the boundary of the domain, $\partial\Omega$, which can be of different kinds. For example, a velocity-inlet/velocity-outlet conditions are imposed when the velocity of the fluid on the boundary is known while a pressure-inlet/pressure-outlet conditions are imposed when the pressure of the fluid can be set. There are an extensive literature on boundary conditions which are more appropriate for different contexts.

The fluid can be compressible or incompressible but in this context the compressibility of the air is neglected since its velocity is typically below sound speed (subsonic condition).

The relative humidity can be modeled considering the mass fraction of the water vapor in the air through the solution of a convection-diffusion-reaction equation. This conservation equation takes the following general form in steady-state [50]: find W such that

$$\nabla \cdot (\rho U M_w - \rho D \nabla M_w) + \rho r M_w = S_w, \quad \text{in } \Omega, \quad (2.17)$$

where M_w is the mass fraction of water in the air, D is the mass fraction diffusion coefficient, r is the reaction coefficient, and S_w is a source term. In this context, the terms $\rho r M_w$ and S_w are not considered since the water vapor in the air does not react and is not created. This equations may contain more terms or be written in equivalent forms that can be found in the literature (see [13] or [50]).

2.4.2. Discretization and Solver

As stated before, the modeling of the thermal environment in terms of physical variables provides a continuous model consisting in a set of partial differential equations equations (2.13) and (2.15–2.17) to model the heat and mass transfer phenomena, respectively. The solution of such equations cannot be usually obtained by analytic procedures, specially when dealing with complex problems. In this way, several numerical methods have been developed and proposed to

provided approximated solutions for the underlying model [13].

Roughly speaking, the numerical procedure comprises two main steps: the discretization and the solver methods. A discretization method consists in a mathematical procedure to obtain arithmetic equations which represent local approximations of each PDE in the model. In order to do that, one has to generate a mesh for the domain considered, that is, the volume of the domain is divided in individual and small volumes called cells. For each cell of the mesh is assigned a linear arithmetic equation of each PDE in the model. All the assigned arithmetic equations in the mesh form a system of linear equations whose solution is the approximated solution of the model. The method used to solve the system of equations is called solver and usually consists in iterative procedures which successively compute a solution converging to the underlying solution. Such methods can also be equipped with other numerical tools as preconditioners, relaxation terms, and vector extrapolation in order to accelerate the solution convergence.

2.4.3. Validation and Actions

In order to check the model and the implementation of the method, one has to compare the obtained solutions against experimental data collected in the same conditions as those represented by the model.

Having the solution in hand, one can easily evaluate the thermal conditions (air temperature, velocity, and humidity) to which the worker is exposed and take actions to improve the thermal environment and the individual's thermal comfort. Such actions may include layout changes which can then be tested using the same numerical procedure in order to predict their effect on the thermal environment.

3. Methodology

In section 1.2, a case study concerning one of the industrial premises of C-ITA company, called Nave 2, was presented and motivated based on the previous studies carried out by Guise (2014) [5] and by Oliveira (2012) [29]. The background and the literature review of the addressed topics in this work were introduced in section 2, such that one is able to proceed with the methodology and the procedures adopted in this case study to numerically assess the physical component of the thermal environment in Nave 2.

The methodology presented hereby can be divided into two consecutive steps: (i) boundary measurements and (ii) numerical modeling. The first step consists in collect data which is of crucial importance to define the domain and the boundary conditions for the mathematical model introduced in subsection 2.4.1. Such measurements concern the following points:

- the geometrical dimensions of the domain and the objects or interfaces comprised by the thermal environment as walls, corridors, machines, ventilation grids, etc.;
- the values of the physical variables (air velocity, temperature, and humidity) on the boundaries and other relevant parameters to define the physical phenomena such as heat transfer coefficients and radiation emissivity coefficients (if necessary).

In the works of Guise (2014) [5] and of Oliveira (2012) [29], several experimental measurements concerning both air velocity and temperature have been carried out in order to assess the workers thermal comfort and the building energetic efficiency, respectively. Some of the values obtained by Guise (2014) and Oliveira (2012) were used to complement the experimental data in this work.

Once concluded step (i), one can proceed with step (ii) – numerical modeling – where the partial differential equations which model the physical component of the thermal environment are solved using numerical methods providing the air velocity, temperature, and humidity in the domain. The numerical procedure was introduced in section 2.4 and comprises, in practice, the following steps:

- build a CAD design corresponding to the domain and the boundaries considered;
- generate a good quality mesh for the design previously built (mesh refinements may be needed for accuracy or convergence purposes);
- set the model to be solved, the boundary conditions, and other relevant parameters to define the model as air properties or gravity;

- set the discretization methods, the iterative solver method, and other relevant parameters as relaxation factors to solve the model and to achieve a good solution convergence;
- post-process the solution to visualize the air velocity, temperature, and humidity distributions in the domain.

The following sections are dedicated to present and to comprehensively discuss the methodology described before as well as to introduce the collected experimental data.

3.1. Boundary Measurements

As said before, the experimental measurements aim to provide crucial information to define the model to be solved. Such measurements concern geometrical dimensions, boundary values, and parameters which complement the mathematical model for the physical component of the thermal environment under study.

Mathematical models based on incomplete or wrong experimental data are unreliable and may provide mistaken solutions compared to the underlying real context. Moreover, even if a high-quality mesh or a high-accurate numerical procedure is used, the solution will never converge to the real one since the errors come from the model and not from the method. On the other hand, take into account detailed geometries/boundaries and boundary values may be, in practice, unfeasible from the experimental point of view since:

- it implies an exhaustive and very time consuming experimental work;
- very accurate and expensive measurement equipment is required;
- some regions of the domain may be inaccessible or very difficult to reach.

From the computational point of view, some difficulties may also arise when applying the numerical procedure, namely:

- it is very memory and time consuming since the associated mesh should have a huge number of cells – the cell dimension has to be smaller than the detail dimension;
- the solution may converge slowly and oscillate, among other numerical issues which exponentially increase the computation time.

Given that, one should first carefully evaluate the domain comprised by the thermal environment in order to decide whether the impact of some geometrical or physical information is relevant for the air velocity, temperature, and humidity distributions. After that, some assumptions

and simplifications should be established in order to plan the experimental measurements, that is, what to measure, how to measure and when to measure, minimizing the experimental and the computational efforts and still obtaining accurate solutions.

3.1.1. Domain and Boundaries

Nave 2 is a premise with an area of around 70 m x 90 m and a height of around 6 m. The domain considered in this case study only comprises half of the Nave 2 given its large dimensions (see Fig. 3.2). In fact, such consideration significantly improves the numerical procedure since it decreases the number of cells in the mesh and therefore decreases the computation time. Moreover, assuming that the physical phenomena on the two halves of Nave 2 are not strongly correlated, an accurate solution can be still obtained if the right boundary condition is imposed on their interface (such topic is addressed in next section).

In addition to the walls of Nave 2, the following elements can be distinguished as part of the boundaries of the domain considered before:

- 9 twisting machines, as those shown in Fig. 3.1, whose dimensions are around 32 m or 35 m of length (7 and 2 of cases, respectively), 2 m of width and 2.1 of height (small differences may appear depending on the models); the twisters layout is shown in Fig. 3.2 where a region called supermarket to store the yarn and the rope is also considered;
- two types of air ventilation grids on the ceiling, as those shown in Fig. 3.3, associated to 4 main air ducts on each side of the premise and disposed as shown in Fig. 3.4; each duct has 7 grids with dimensions of 0.15 m x 0.85 m and 17 grids with dimensions 0.35 m x 0.85 m which sum to 164 grids and are disposed as shown in Fig. 3.5;
- 2 corridors which give access to another premise, as that shown in Fig. 3.6, with dimensions of 3 m x 5 m and 2.8 m x 5 m;
- a small corridor with stairs which give access to the groundfloor with dimensions of 3.1 m x 3 m;
- an interface dividing Nave 2 in half with dimensions 70 m x 6 m.

The remaining elements in Nave 2, as the anti-noise panels on the ceiling, shelves, air ducts, boxes or even human bodies, would require an exhaustive experimental work to measure and could also give rise to some numerical and computational difficulties, as stated before. Given that, it was assumed in this work that such elements do not have a relevant role in the physical phenomena and therefore were not considered as part of the boundaries of this case study.



Figure 3.1: Twister machines in Nave 2.

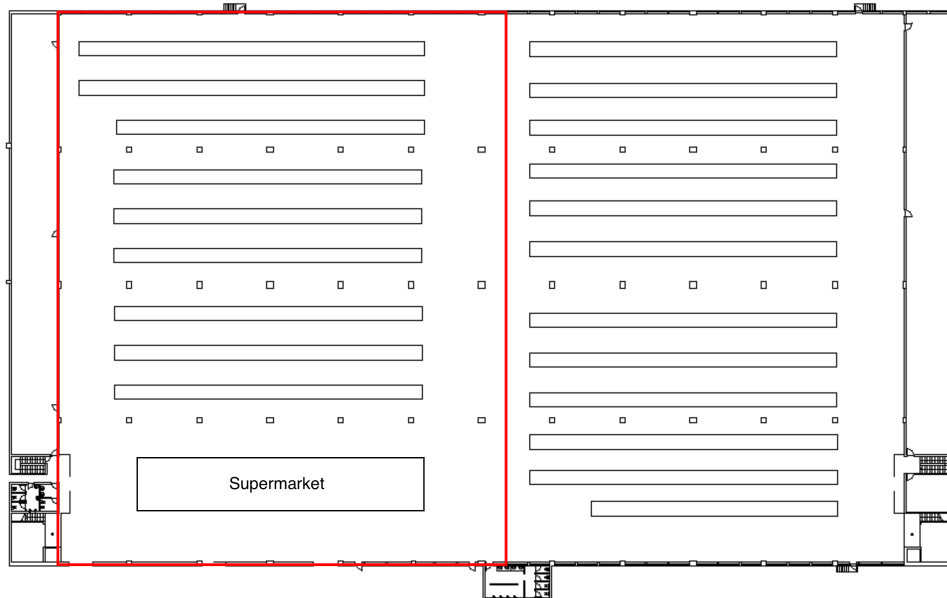


Figure 3.2: Twisters layout in Nave 2. The interior of the red rectangle corresponds to the domain considered.

Moreover, the twisters were simplified by polygons of dimensions 32 m x 2 m x 2.1 m (7 cases) or 35 m x 2 m x 2.1 m (2 cases). In the same way, the 7 grids of dimensions 0.15 m x 0.85 m belonging to each air duct were simplified by a major grid of dimensions 0.15 m x 15 m as well as the 17 grids of dimensions 0.35 m x 0.85 m were simplified by two major grids of dimensions 0.85 m x 15 m.

The boundary conditions associated to each boundary element are defined in next section.

3.1.2. Boundary Values and Parameters

The air velocity and temperature were measured on the boundaries of the domain using a digital thermo-anemometer as that shown in Fig. 3.7 (retrieved from [51]). The air humidity were measured using a Data Logger model EL-USB 2 from Lascar.



Figure 3.3: Air ventilation grids with dimensions 0.15 m x 0.85 m (left panel) and 0.35 m x 0.85 m (right panel) on the ceiling of Nave 2.

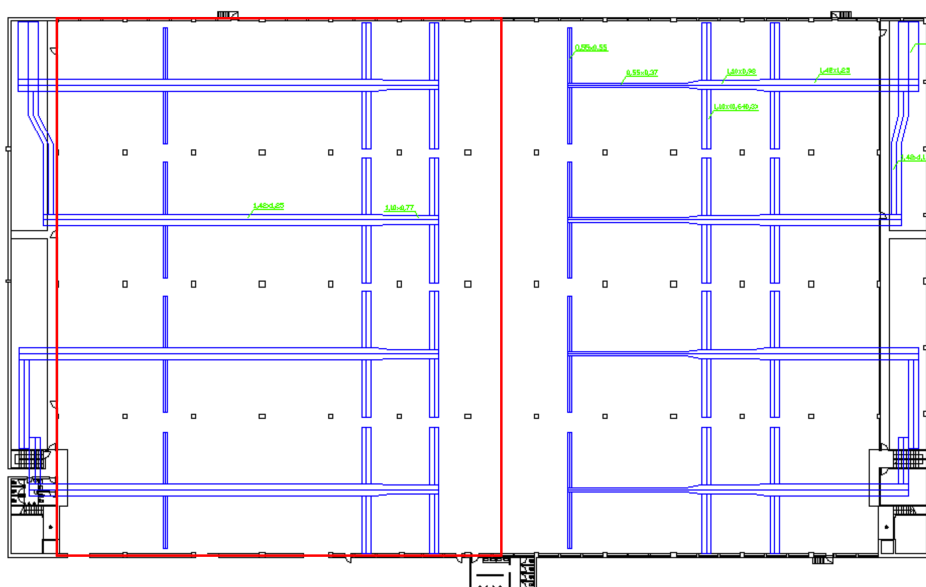


Figure 3.4: Air ducts layout in Nave 2. The interior of the red rectangle corresponds to the domain considered.

The location of the boundaries considered in Nave 2 is shown in Fig. 3.8. The air velocity, temperature, and humidity for each boundary of the domain, assigned with a symbol for the sake of simplicity, are summarized in Table 3.1.

For the air ventilation grids such values were obtained from the work of Oliveira (2012) [29] where a total air flow rate of 14.91 m³/s and of 1.24 m³/s were measured for the 0.35 m x 0.85 m and 0.15 m x 0.85 grids, respectively, of each air duct. Assuming the simplified geometry, one can compute an air velocity of 0.55 m/s for the 0.15 m x 15 m grids and of 0.58 m/s for the 0.85 m x 15 m grids whose temperature is 21.6 °C for the first case and 21.9 °C for the other, according to [29]. The measured relative humidity of the air is 89.3% for both cases [29], which corresponds to a water vapor mass fraction of 14.26 g/kg and 14.53 g/kg, respectively.

The thermo-anemometer was not appropriated to measure the temperature on the surface

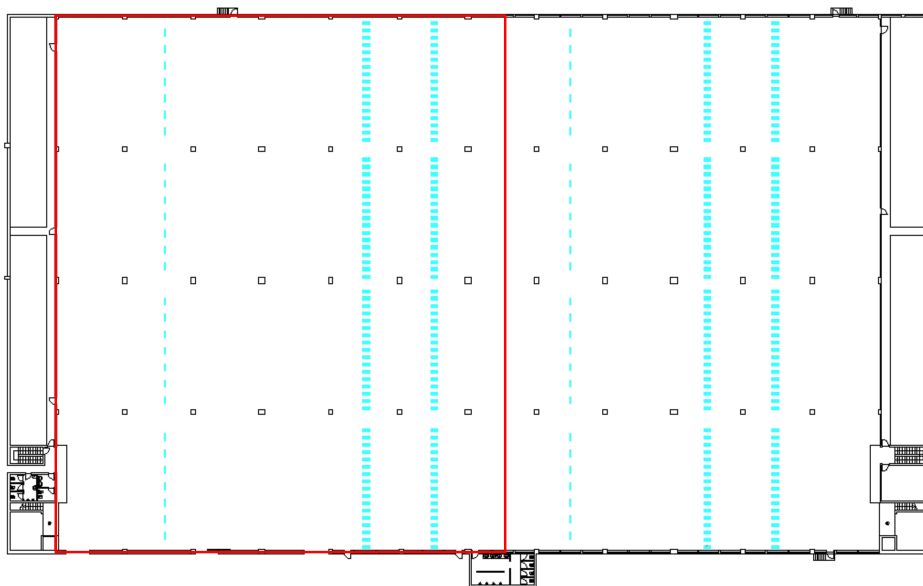


Figure 3.5: Air grids layout in Nave 2. The interior of the red rectangle corresponds to the domain considered.



Figure 3.6: Corridor and supermarket in Nave 2.

of the twister machines. To overcome the problem, the temperature of the surrounding air at several points immediately after the surface was measured and then it was predicted a mean value of approximately 35 °C. Although the twisters have their own ventilation system which extracts some of the heat produced to air ducts, the associated flow rate is very small given the large area comprised by the machine. In this way, a null air velocity was assigned.

On the corridors, the air velocity and temperature were measured and a mean value assigned. The relative humidity was retrieved from the work of Guise (2014) [5] which has measured a value of 45.0% corresponding to a mass fraction of 6.90 g/kg for one corridor and of 7.33 g/kg for the two others, as summarized in Table 3.1.

On the interface diving Nave 2 as well as on the walls and the supermarket, no temperature was measured since they comprise a huge area and their value can strongly vary. In fact, an appropriate boundary conditions which do not depend on the temperature is set later in this work



Figure 3.7: Thermo-anemometer model 8330 from TSI VelociCheck.

Table 3.1: Air velocity, temperature, and humidity assigned on each boundary.

Boundary	Symbol	Temperature (°C)	Velocity Mag. (m/s)	Water vapor (g/kg)
Twister Machines	T1 to T9	35.0	0.00	—
Supermarket	S	—	0.00	—
Air Grids 0.15 m x 15 m	V1 to V4	21.6	0.55	14.26
Air Grids 0.85 m x 15 m	V5 to V12	21.9	0.58	14.53
Corridor 3 m x 5 m	C1	21.0	1.14	6.90
Corridor 2.8 m x 5 m	C2	21.0	0.98	6.90
Corridor 3.1 m x 3 m	C3	22.0	0.10	7.33
Premise Interface	D	—	—	—
Premise Walls	W	—	0.00	—

such that those values are not required.

Finally, it was assumed standard values for the air parameters, namely a density of $\rho = 1.225 \text{ kg/m}^3$, a specific heat capacity of $C_p = 1006.43 \text{ J/(kg} \cdot ^\circ\text{C)}$, a thermal conductivity of $\kappa = 0.0242 \text{ W/(m} \cdot ^\circ\text{C)}$, and a dynamic viscosity of $\mu = 1.7894 \times 10^{-5} \text{ kg/(m} \cdot \text{s)}$.

3.2. Numerical Modeling

Having the experimental data in hands, one can now proceed with the numerical step where a solution for the physical variables of the thermal environment is computed through numerical methods. Applying such numerical methods by hand is not feasible even for a small mesh since one has to compute several arithmetic equations, solving several systems of linear equations, among other numerical procedures.

On the other hand, in the recent decades several commercial and license-free software for numerical simulation have been developed and widely applied to solve problems in almost

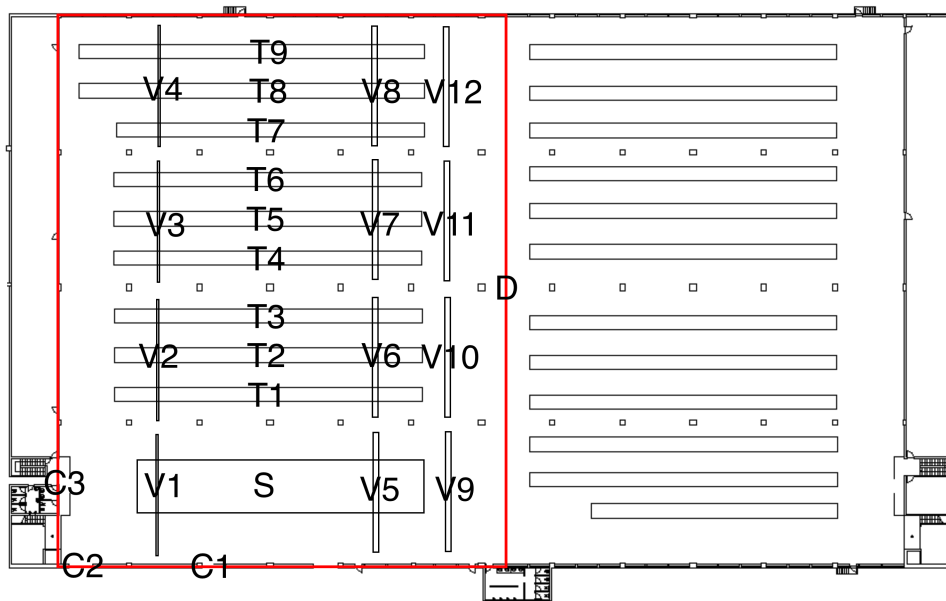


Figure 3.8: Boundaries considered in Nave 2. The interior of the red rectangle corresponds to the domain considered.

all engineering fields, such as electrical engineering, structural and fluids engineering, systems engineering, and so on. With those software one can take advantage of the potentialities of nowadays computers and solve very complex problems in a few hours or minutes which would required many years or centuries to solve by hand.

Concerning to CFD simulation software, there is commercial software as ANSYS or Comsol as well as license-free as OpenFoam or OpenFVM. The simulations in this work were carried out with ANSYS software version 15.0 (see Fig. 3.9), which includes other software to design, to mesh, and to post-process the underlying problem and whose license is provided by the University of Minho.

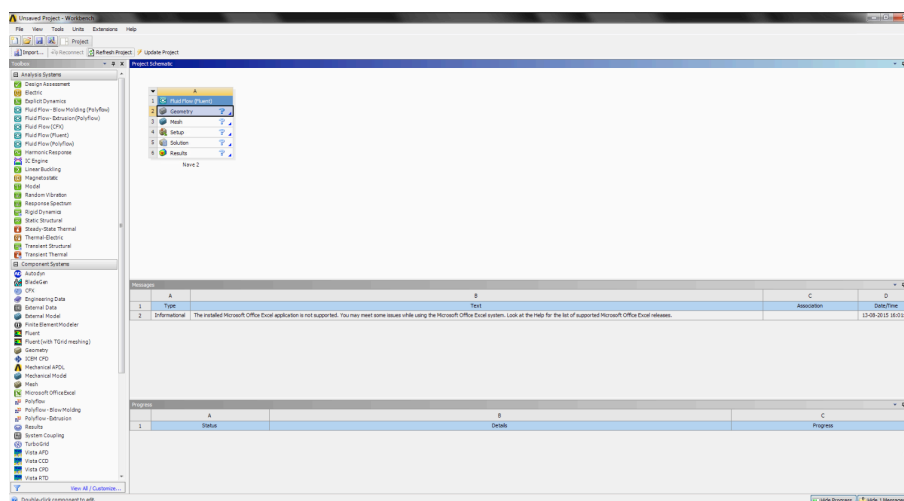


Figure 3.9: A screenshot of the ANSYS software workbench with the Fluent solver.

In the next sections, a general view over ANSYS software is given as well as a discussion on the methods, procedures, and parameters chosen.

3.2.1. CAD Design

The procedure to carry out a simulation in ANSYS software always starts with the geometry to represent the domain and the boundaries of the problem. In this case, the domain represents the volume where the air flows.

This geometry is virtually produced in a computer-aided design (CAD) system, such as the DesignModeler software integrated in ANSYS software (see Fig. 3.10), which makes easier to the user to create and to handle very complicated designs. As stated in [50], CAD models are often designed for manufacturing use and not for simulation and may comprise complex and detailed geometries which are not needed for simulation. Simulating such a fully detailed model will increase solver run times by the reasons presented before in subsection 3.1.1.

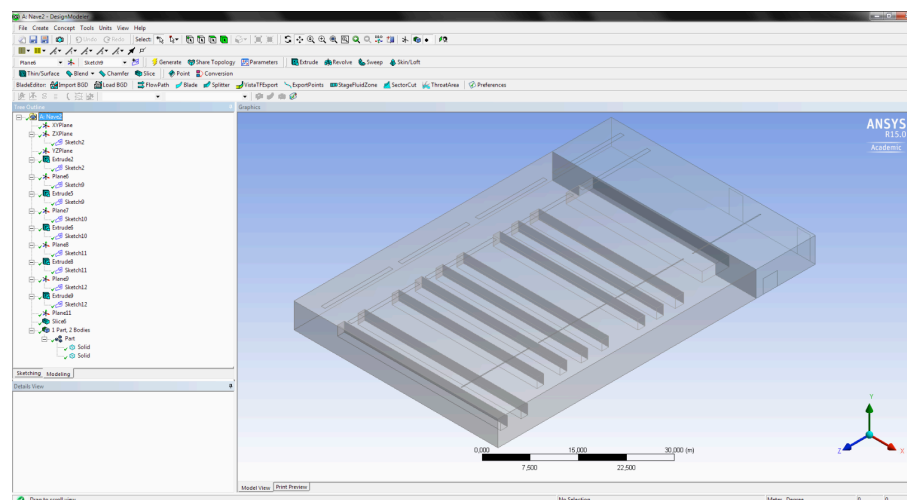


Figure 3.10: A screenshot of the ANSYS DesignModeler software environment and the produced geometry.

The produced design for the domain and boundaries considered in subsection 3.1.1 for Nave 2 is shown in Fig. 3.11.

3.2.2. Mesh

The next step to run a simulation in ANSYS software is to generate a mesh for the CAD design produced before. For that purpose, Meshing software was used which is an integrated tool in ANSYS software. This is one of the most critical steps in the numerical procedure since a good quality mesh results in more accurate solutions and faster convergence rates. On the other hand, too many cells may result in long solver runs and memory saturation while too few cells lead

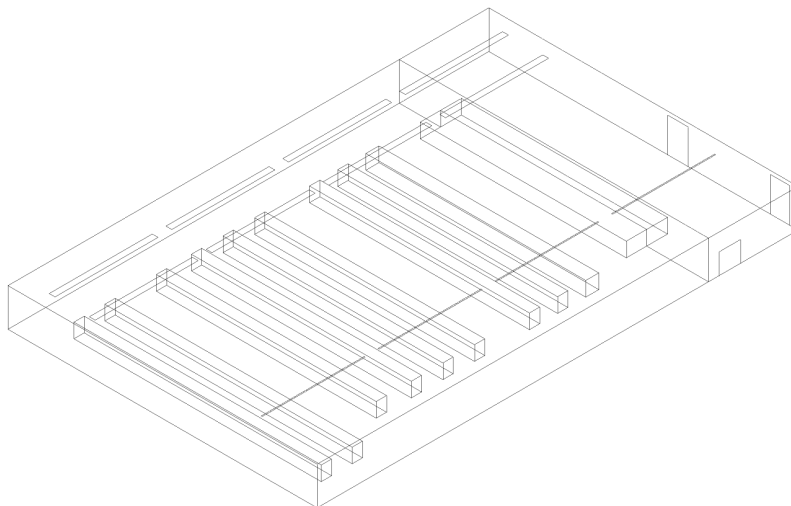


Figure 3.11: CAD design for the domain and boundaries considered in Nave 2.

to inaccurate results and slow convergence rates. As stated in [50], ANSYS Meshing technology provides a means to balance these requirements and obtain the right mesh for each simulation in the most automated way possible.

ANSYS Meshing software provides the following mesh kinds: tetrahedral, hexahedral, prismatic inflation layer, hexahedral inflation layer, hexahedral core, body fitted Cartesian, cut cell Cartesian. A comprehensive description of such kind of meshes can be seen in ANSYS documentation. In this work, the domain is meshed with approximately 16×10^6 tetrahedral cells since they can be efficiently fitted in the geometry and refined in the desired boundaries to better capture the boundary conditions.

In Fig. 3.12 the ANSYS Meshing environment before the mesh generation can be seen and Fig. 3.13 shows a partial view of the generated mesh.

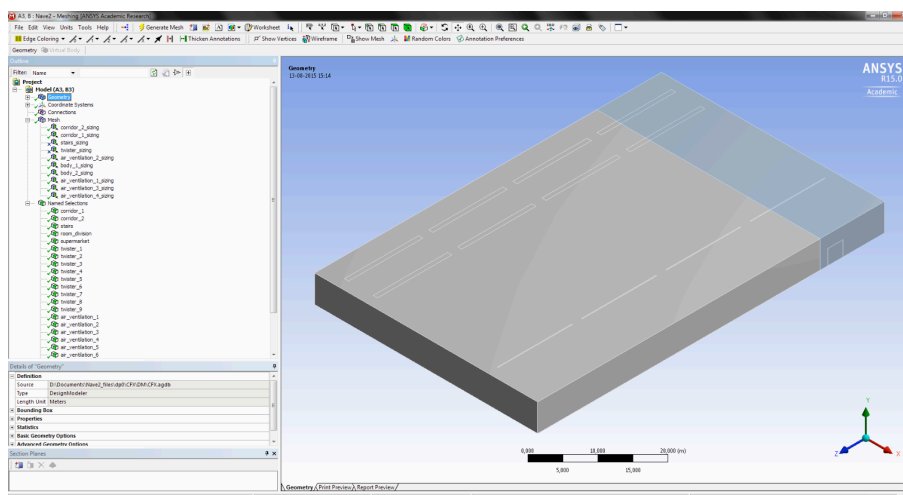


Figure 3.12: A screenshot of the ANSYS Meshing software environment with the CAD design to mesh.

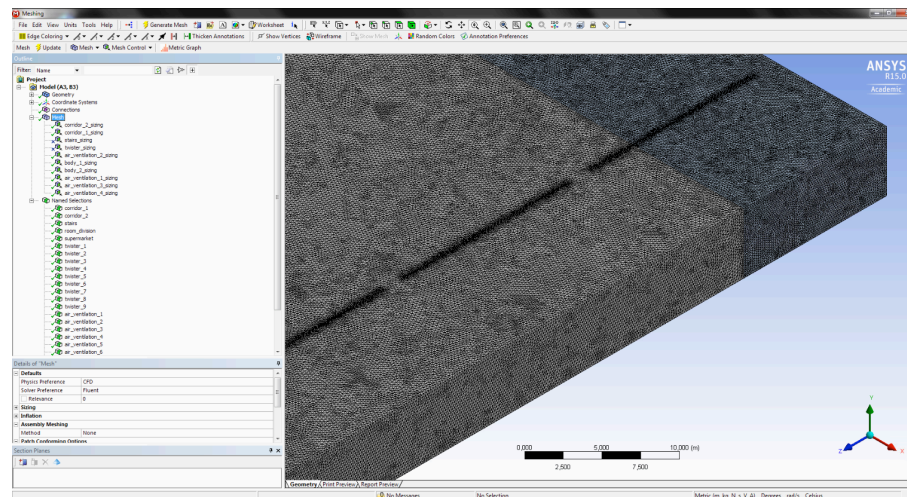


Figure 3.13: A screenshot of the ANSYS Meshing software environment with a partial view of the generated mesh.

3.2.3. Setup

Having the geometry and the mesh in hand, one proceed with the setup to run the simulation. This setup consists in setting the equations to solve, the parameters, the boundary conditions, the discretization method, the solver method, relaxation coefficients, an other features necessary to obtain a solution for the problem.

ANSYS software provides different environments which include different solvers for CFD simulation, namely Polyflow for problems concerning viscous fluids as polymer processing, CFX and Fluent for the other cases (see Fig. 3.14). The difference between CFX and Fluent relies in the way they discretize the governing equations and in their equation solution strategies [50]. Although CFX is a cell-vertex approach and Fluent a purely a cell-centered approach, both discretization techniques concern finite volume methods which have been widely used for decades to solve CFD problems [13]. In this work, the setup was carried out in Fluent solver which have been used in similar contexts and a description and discussion of the main parameters is given below.

The first step in Fluent setup is to set the equations to solve in "Models" tab which, in this case, are the energy equation (for the heat transport), the $\kappa - \epsilon$ turbulence equation (momentum and continuity equations with $\kappa - \epsilon$ turbulence model, for the mass transport), the surface-to-surface equation (for the radiation transfer), and the species transport equation (for water vapor transport).

Then, in "Materials" tab the air properties and the species (water vapor) are defined and the Boussinesq approximation is set in order to consider the natural convection phenomena due the temperature effect on the air density.

One should next define the boundary conditions and the corresponding parameters in

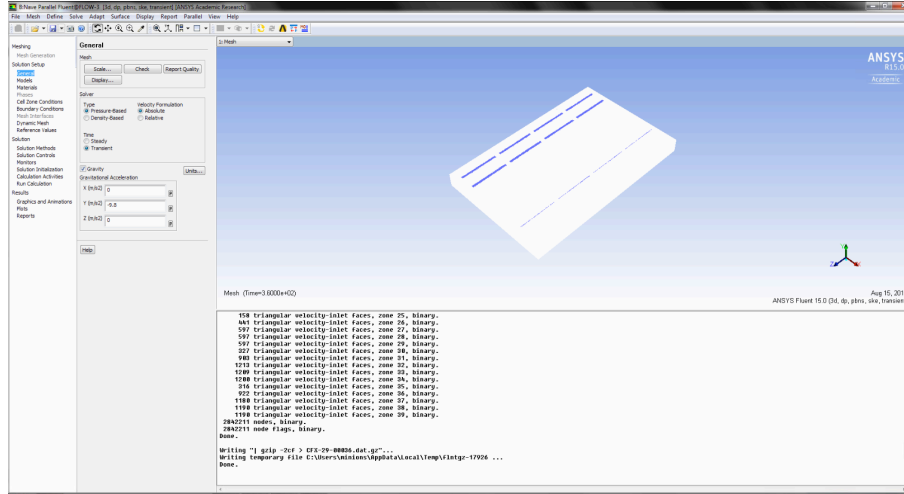


Figure 3.14: A screenshot of the ANSYS Fluent software environment.

"Boundary Conditions" tab which are given in Table 3.2 for this case study. The parameters, when applied, are the following: T stands for the air temperature, U stands for the air velocity, P stands for the gauge pressure, M_w stands for the water vapor mass fraction, F stands for the heat flux, D_H stands for the hydraulic diameter which for rectangular boundaries can be computed by

$$D_H = \frac{2ab}{a+b}, \quad (3.1)$$

where a and b are the dimensions of the boundary, Re is the Reynolds number given by

$$Re = \frac{\rho U D_H}{\mu} = \frac{U D_H}{\nu}, \quad (3.2)$$

and I is the turbulent intensity which can be approximated by

$$I = 0.16 Re^{-1/8}. \quad (3.3)$$

To assign the values of Re and I for the interface D , the mean velocity based on the air flow rate on the boundaries C1, C2, C3, and V1 to V12 was computed .

The discretization methods and the solver are set in "Solution Methods" tab. The default and simplest solver is SIMPLE which has lead, in this case, to divergent solutions due the large skewness of the mesh (ratio between the largest and smallest cells). Given that, several methods were tested in order to optimize the solution convergence, namely the Pressure-Implicit with Splitting of Operators, also known as PISO. This method is based on a more accurate relation to perform the corrections for pressure and velocity [50].

As stated in [50], one of the limitations of the SIMPLE and SIMPLEC algorithms is that new velocities and corresponding fluxes do not satisfy the momentum balance after the pressure-correction equation is solved. In this way, the calculation must be repeated until the balance is

Table 3.2: Boundary conditions and the corresponding parameters.

Boundary	Boundary condition	Parameters		
T1 to T9	Wall	$T = 35\text{ }^{\circ}\text{C}$		
S	Wall	$F = 0\text{ W/m}^2$		
V1 to V4	Velocity-inlet	$T = 21.6\text{ }^{\circ}\text{C}$ $U = 0.55\text{ m/s}$	$D_H = 0.30\text{ m}$ $Re = 1.13 \times 10^4$	$I = 4.98\%$ $M_w = 0.014$
V5 to V12	Velocity-inlet	$T = 21.9\text{ }^{\circ}\text{C}$ $U = 0.58\text{ m/s}$	$D_H = 1.61\text{ m}$ $Re = 6.39 \times 10^4$	$I = 4.01\%$ $M_w = 0.015$
C1	Velocity-inlet	$T = 21\text{ }^{\circ}\text{C}$ $U = 1.14\text{ m/s}$	$D_H = 3.75\text{ m}$ $Re = 2.93 \times 10^5$	$I = 3.31\%$ $M_w = 0.007$
C2	Velocity-inlet	$T = 21\text{ }^{\circ}\text{C}$ $U = 0.98\text{ m/s}$	$D_H = 3.59\text{ m}$ $Re = 2.41 \times 10^5$	$I = 3.40\%$ $M_w = 0.007$
C3	Velocity-inlet	$T = 22\text{ }^{\circ}\text{C}$ $U = 0.10\text{ m/s}$	$D_H = 3.05\text{ m}$ $Re = 2.09 \times 10^4$	$I = 4.61\%$ $M_w = 0.007$
D	Pressure-outlet	$T = 30\text{ }^{\circ}\text{C}^*$ $U = 0.03\text{ m/s}^{**}$	$D_H = 54.78\text{ m}$ $Re = 1.14 \times 10^5$	$I = 3.73\%$
W	Wall	$F = 0\text{ W/m}^2$		

* Backflow temperature, ** Only used to predict Re

satisfied. By performing two additional corrections – neighbor correction and skewness correction – PISO algorithm allows to improve the efficiency of this calculation [50]. In fact, with PISO solver a reasonable solution convergence was achieved using a skewness correction factor of 5.0 and a neighbor correction factor of 2.0.

The discretization methods were the following: least squares cell based for the spacial discretization, second-order for the pressure, second-order upwind for the momentum, first-order upwind for both the turbulent kinetic energy and turbulent dissipation rate, and second-order upwind both for the energy and for the species transport. The relaxation factors were the default values set by Fluent.

Finally, one has to run the simulation. Once the simulation has finished, the solution of the problem (air velocity, temperature, humidity, pressure, and other model variables) is ready to be post-processed in ANSYS Post-CFD software.

4. Results and Analysis

This section is dedicated to present and to discuss the results obtained for the simulations carried out in the context of the thermal environment in Nave 2. Two different cases were addressed as follows.

The first case corresponds to the current layout introduced and discussed previously in Chapter 3. With the results obtained from this case, one can validate the model for the physical component of the thermal environment in Nave 2. Moreover, one can analyze the physical variables in all domain and understand the underlying physical phenomena leading to such thermal environment.

The second case concerns an alternative layout where the location of the twister machines is changed in order to seek for an optimal thermal environment with the air temperature and humidity as low as possible and the air velocity as large as possible. Of course, an uncountable number of alternative scenarios could be tested, which was not possible to perform in this work. Instead, this case shows that the numerical modeling can also be used to test different scenarios and perform thermal evaluations while designing workplaces. The numerical procedures were applied as in the first one considering a different CAD design.

Since the solution is associated to a three-dimensional geometry, five vertical planes were created aligned to the boundaries of interest as well as one horizontal plane at 1.0m of height in order to better visualize the workplace conditions. The vertical planes named P_1 to P_4 (see Fig. 4.1) are aligned to the air inlets C2, V1 to V4, C1, and V5 to V8 while the plane P_5 corresponds to the premise interface D.

4.1. Current Layout

In Fig. 4.2 the air velocity magnitude on the vertical and on the horizontal planes are shown. Additional figures are provided in Annex I of this document.

The first conclusion to be draw from the results obtained is that the air flow coming from corridors C1 and C2 has a significant impact in the air movement in Nave 2. In fact, these air inlets, which are originated from the natural ventilation of the building itself and not from an installed ventilation system, input a larger air flow in the premise than the air ventilation grids on the ceiling. In this way, their presence cannot be neglected when modeling and analyzing the thermal environment in Nave 2.

Moreover, according to the results obtained, the air ventilation coming from grids V1 to V4

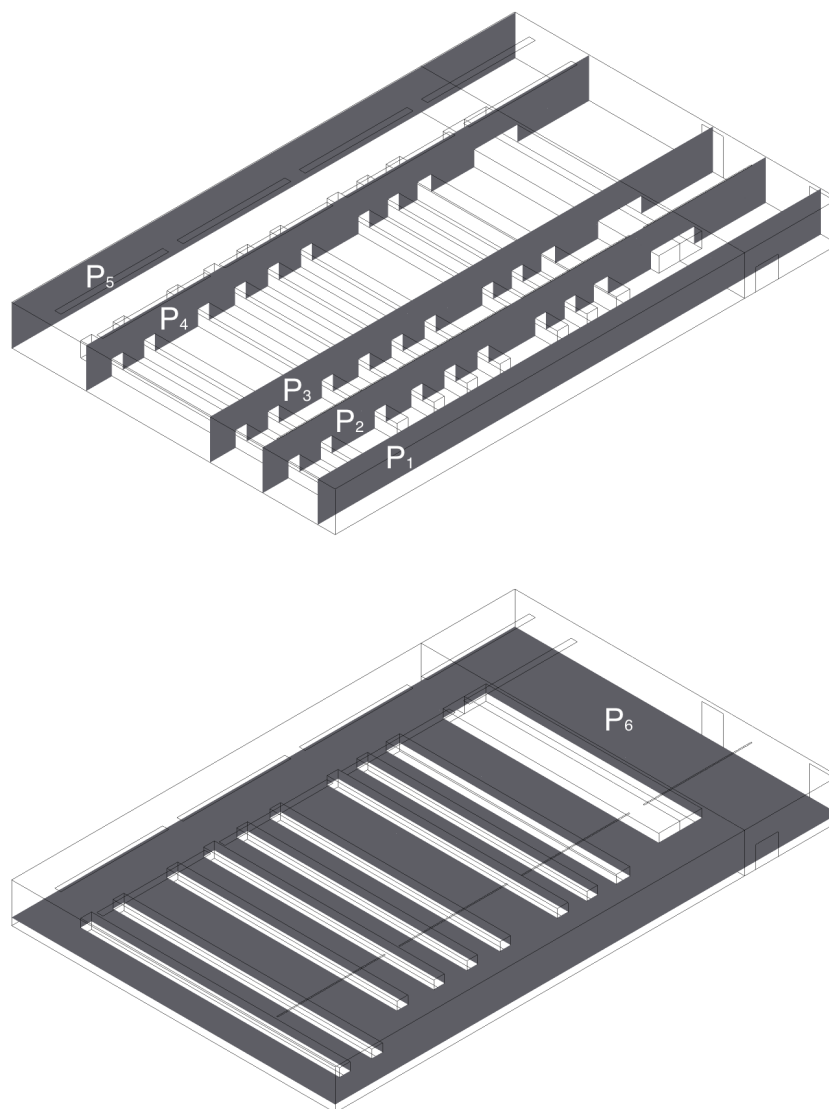


Figure 4.1: Vertical planes P_1 , P_2 , P_3 , P_4 , and P_5 (top panel), and horizontal plane P_6 (bottom panel) to visualize the solution.

seems to have a low impact on the local air velocity compared to grids V5 to V12, as can be seen in planes P_2 and P_4 , respectively. In fact, despite of their comparable velocities (0.55 m/s and 0.58 m/s, respectively), the air flow rate is much lower for the first case.

Concerning the air circulation in Nave 2, one can conclude that the critical zones correspond to the corridors between the twister machines achieving velocities not larger than 0.30 m/s. As a consequence, it is expected an increment of the air temperature in these regions, as is shown later in this document.

To better visualize and analyze the effects of the air inlets on Nave 2 individually, the streamlines of the flow coming from corridors C1 and C2 and from air ventilation grids V1 to V4 and V5 to V12 are shown in Fig. 4.3, top and bottom panels, and in Fig. 4.4, top and bottom

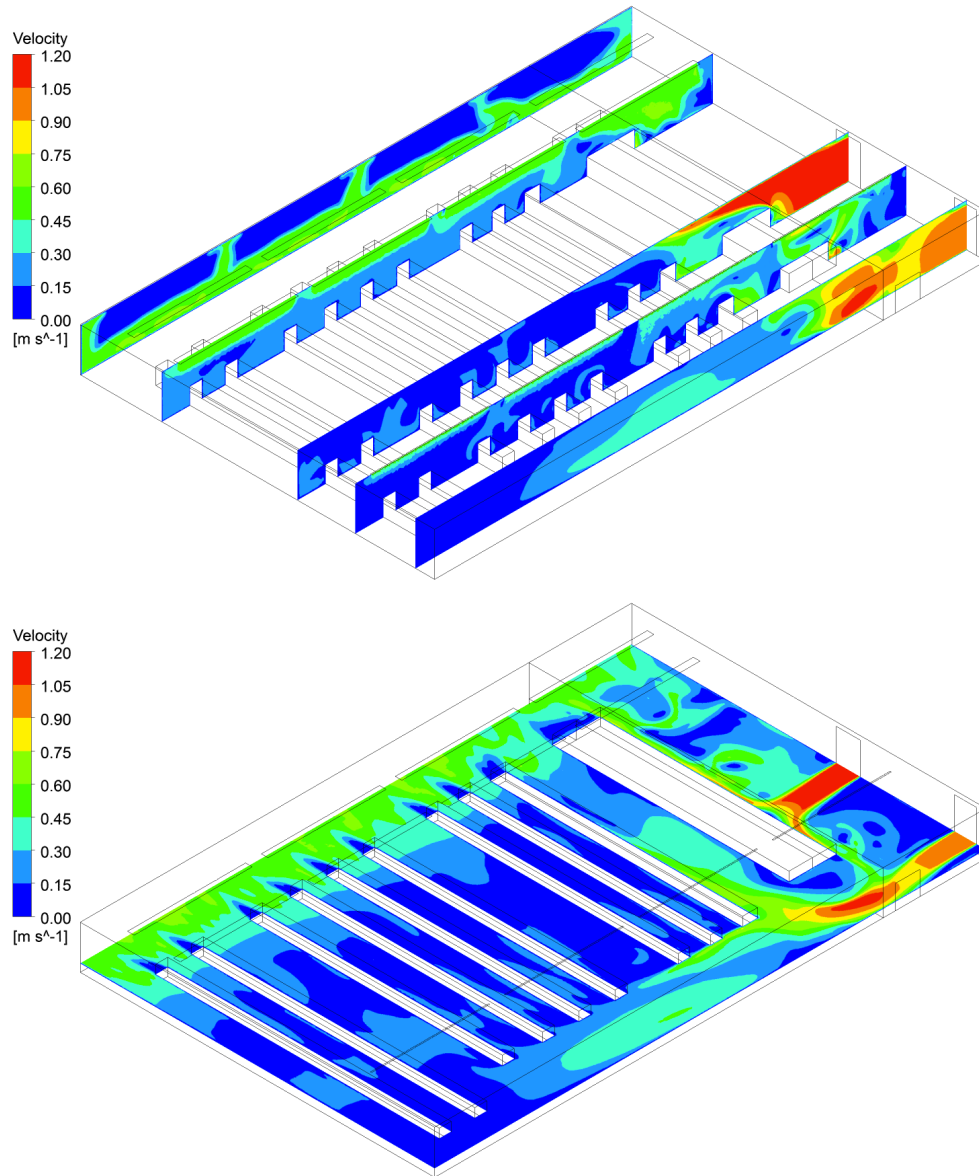


Figure 4.2: Air velocity magnitude contours on the vertical planes (top panel) and on the horizontal plane (bottom panel).

panels, respectively. By definition, the streamlines are a family of curves that are instantaneously tangent to the velocity vector of the flow. These lines show the direction a massless fluid element will travel in at any point in time. In this context, the streamlines show individually which way the air coming from each inlet flows.

Fig. 4.3 shows that corridors C1 and C2 have, in fact, a huge impact and the air coming from both achieve all the regions of the premise. Of course, such impact vanishes when approaching the opposite wall on Nave 2 since the air flow spreads and loses momentum throughout the domain. Concerning to air ventilation grids V1 to V4, one can observe in Fig. 4.4, top panel, their low impact on the local velocity. On the other hand, one can conclude that, although the effect of the air ventilation grids V5 to V12 shown in Fig. 4.4, is locally significant they should be more efficient if

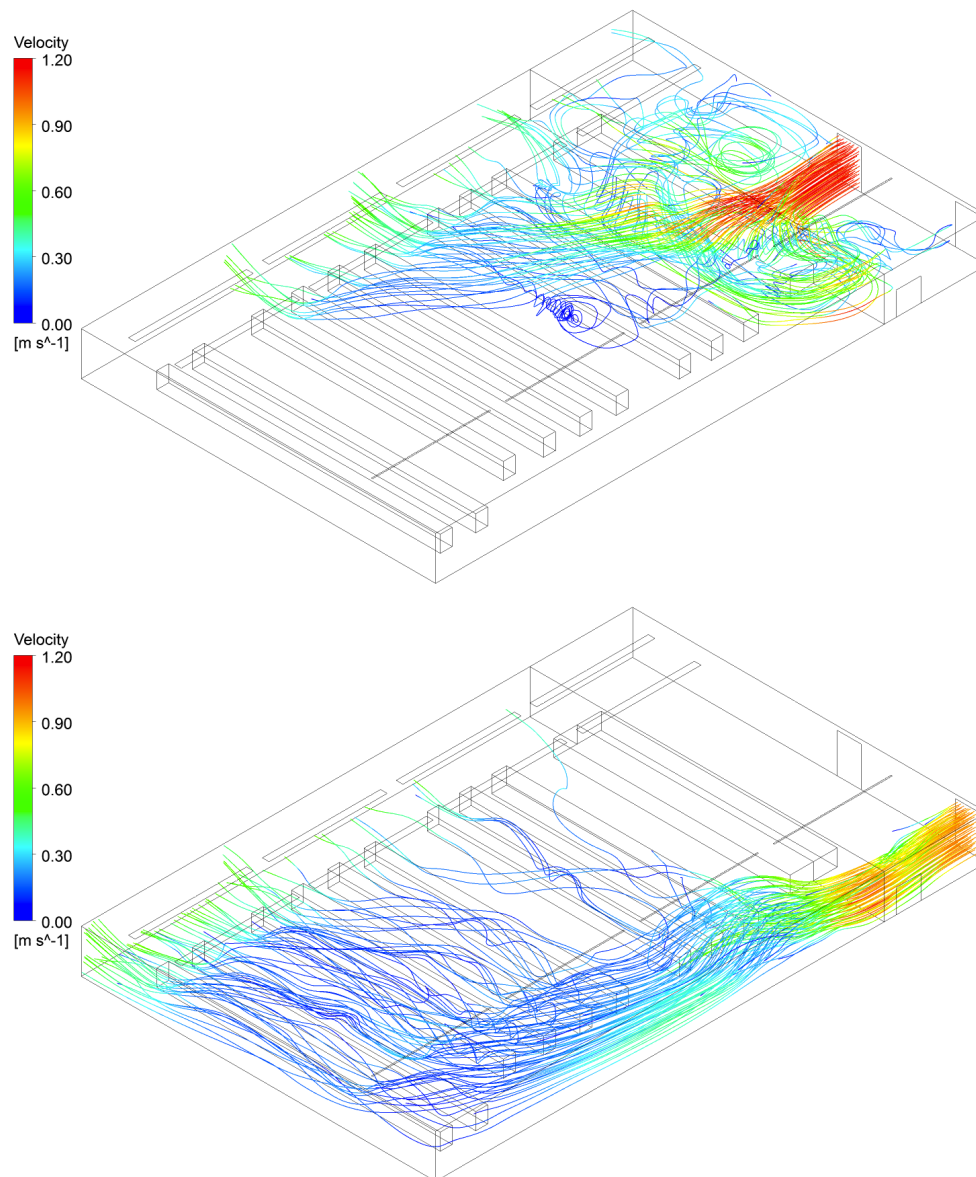


Figure 4.3: Streamlines starting from corridor C1 (top panel) and from corridor C2 (bottom panel).

located on the inner zones of the ceiling above the twister machines.

Although the pressure is not a relevant variable for the thermal environment, it is part of the Navier-Stokes equations which model the air flow and may be important to analyze where the air is speeding up or slowing down. In other contexts, the pressure analysis is of crucial importance such as polymer processing affecting the melting point of the polymer or flows in pipes since a pressure drops indicates frictional forces which are caused by the resistance to flow acting on the fluid.

As seen in Fig. 4.5, the relative pressure for the considered thermal environment is almost constant all over the premise. The supermarket, placed in front of corridor C1, represents an obstacle to the air movement forcing it to flow through a narrow zone above it. As a consequence, the air speeds up and the pressure increases (red zones). On the other hand, the blue zones

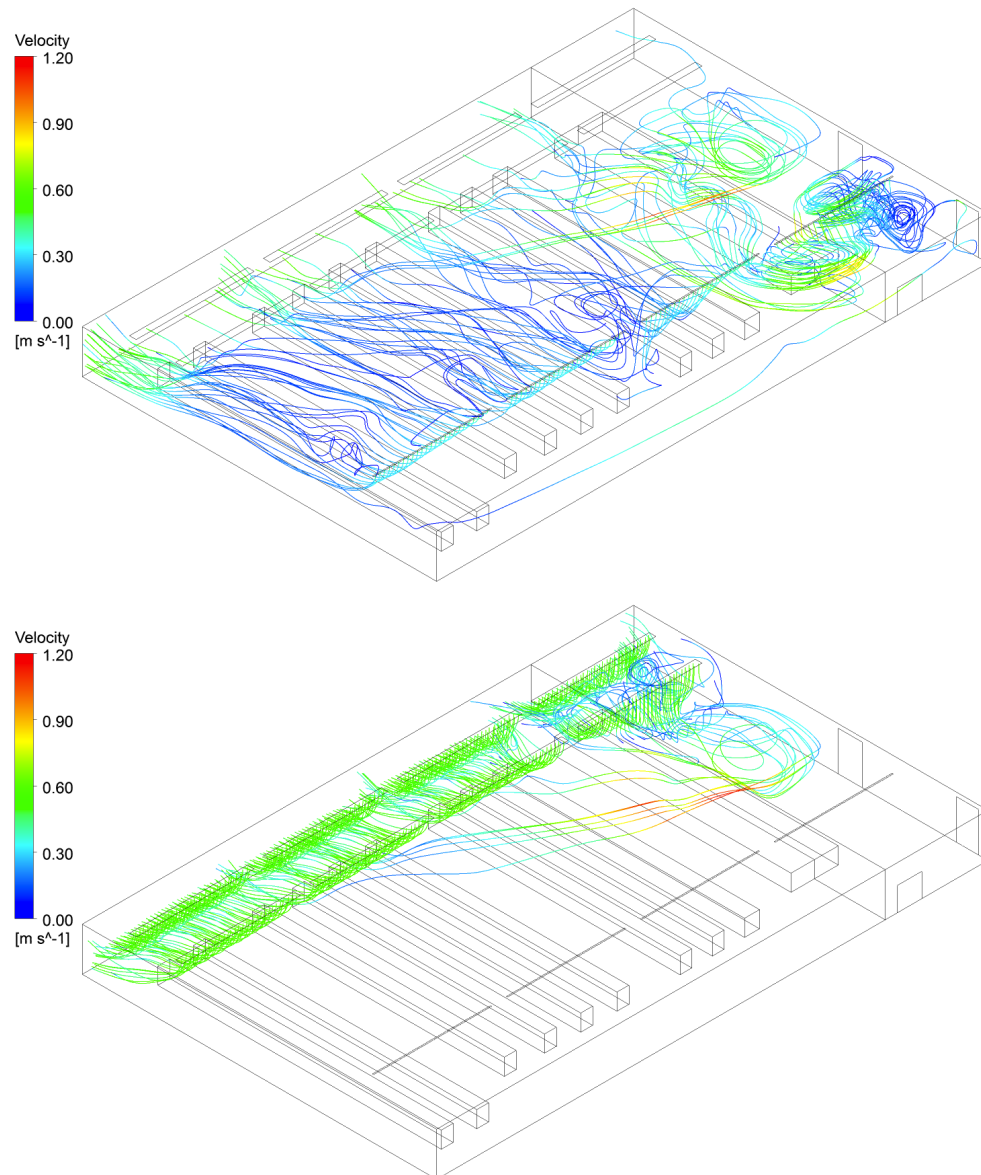


Figure 4.4: Streamlines starting from air ventilation grids V1 to V4 (top panel) and from air ventilation grids V5 to V12 (bottom panel).

correspond to negative pressure values and, therefore, indicate that the air velocity is slowing down. This phenomenon is caused by the air recirculating seen in Fig. 4.3 causing a pressure drop at its very center.

The temperature distribution on the planes considered in Nave 2 is shown in Fig. 4.6. Notice that the scale was set from 21 °C to 31 °C although the maximum value is 35 °C on the twister machines. One can observe that the air flow coming from corridors C1 and C2 have a huge impact on the local temperature while the inner zones between the twister machines achieve temperatures up to 31 °C.

Both conditions (low air velocity and high air temperature) in the corridors between the twister machines, increase the worker's thermal discomfort within the environment which results

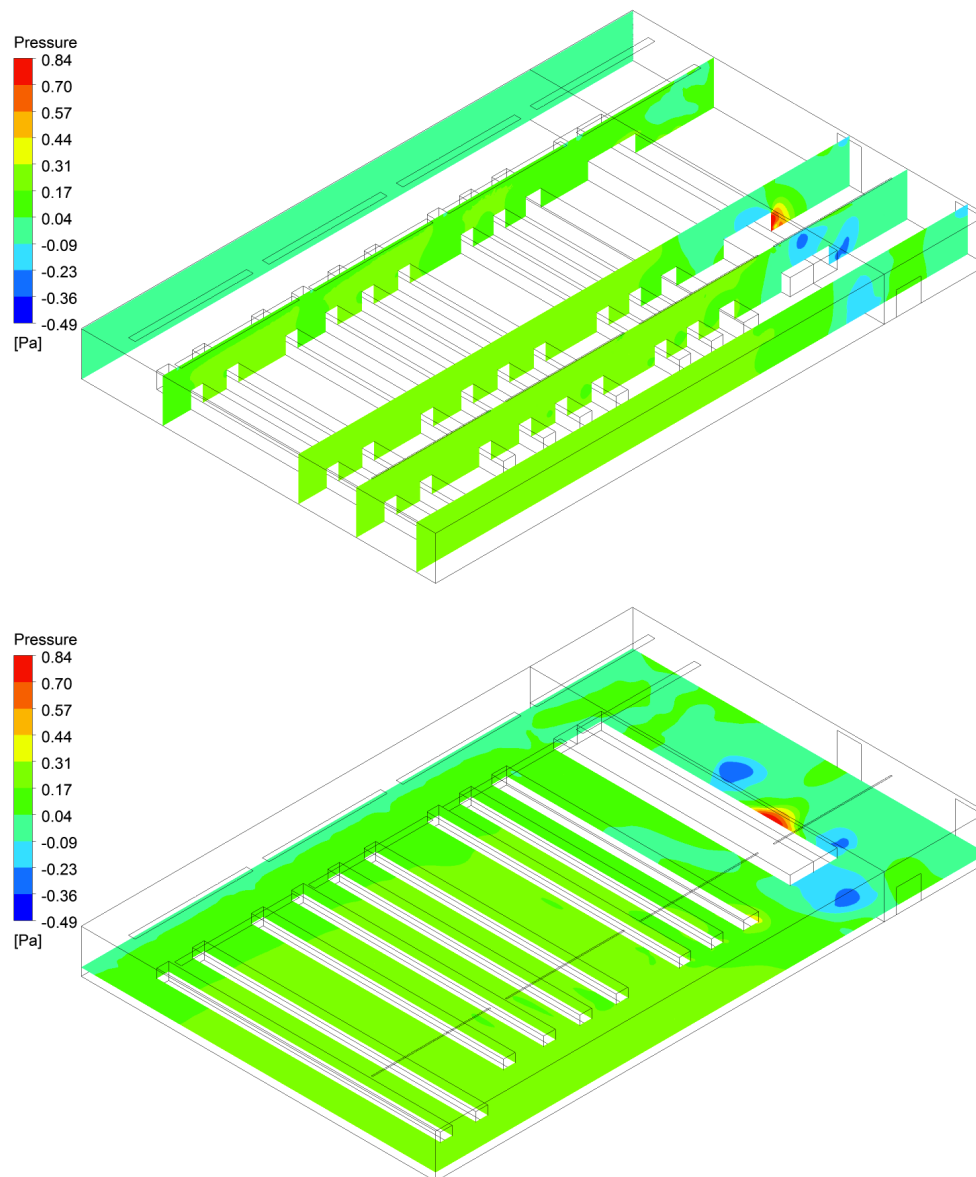


Figure 4.5: Air pressure contours on the vertical planes (top panel) and on the horizontal plane (bottom panel).

in large values of PMV-PPD and WBGT indices, as concluded by Guise (2014) [5].

In order to validate the model, Fig. 4.7 shows the mean values for the air temperature in three different zones, Z_1 , Z_2 , and Z_3 , obtained by Guise (2014) [5].

As can be seen, for zone Z_1 the numerical temperature ranges from 21 °C to 24 °C while the experimental mean temperature is 22.5 °C. In the same way, in zone Z_2 the temperatures range mainly from 26 °C to 29 °C while the experimental mean temperature is 26.9 °C. Finally, in zone Z_3 the temperatures range mainly from 28 °C to 30 °C while the experimental mean temperature is 29.3 °C.

The comparison with the results obtained by Guise (2014) have shown that the experimental

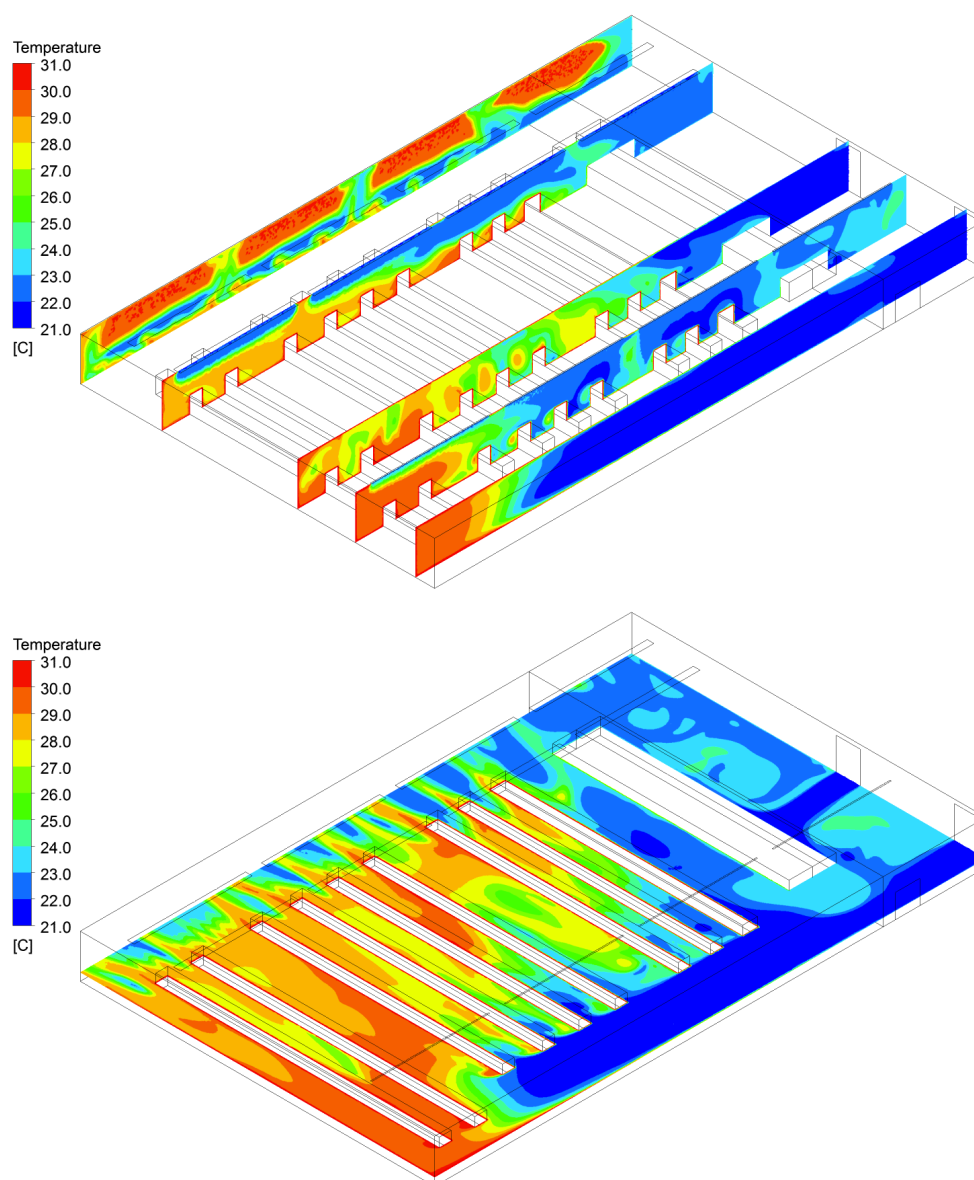


Figure 4.6: Air temperature contours on the vertical planes (top panel) and on the horizontal plane (bottom panel).

mean values for each zone always fall in the range of the temperatures obtained for the numerical values. Although a more precise validation should be performed in the future (considering pointwise values, for example), the reasonable correspondence between the numerical and the experimental results obtained indicate that the model is reliable. Even if point-to-point comparison were performed, comparable results are still expected or can be obtained after some adjustments in the model, that is, geometry and boundary conditions.

In Fig. 4.8 the water vapor mass fraction on the vertical and on the horizontal planes are shown. Additional figures are provided in Annex I of this document. The results show that the water vapor mass fraction in Nave 2 ranges mainly between 0.0062 and 0.0079 at 1.0m of height although a very humid air comes from the ventilation grids. Such result was expected since, as

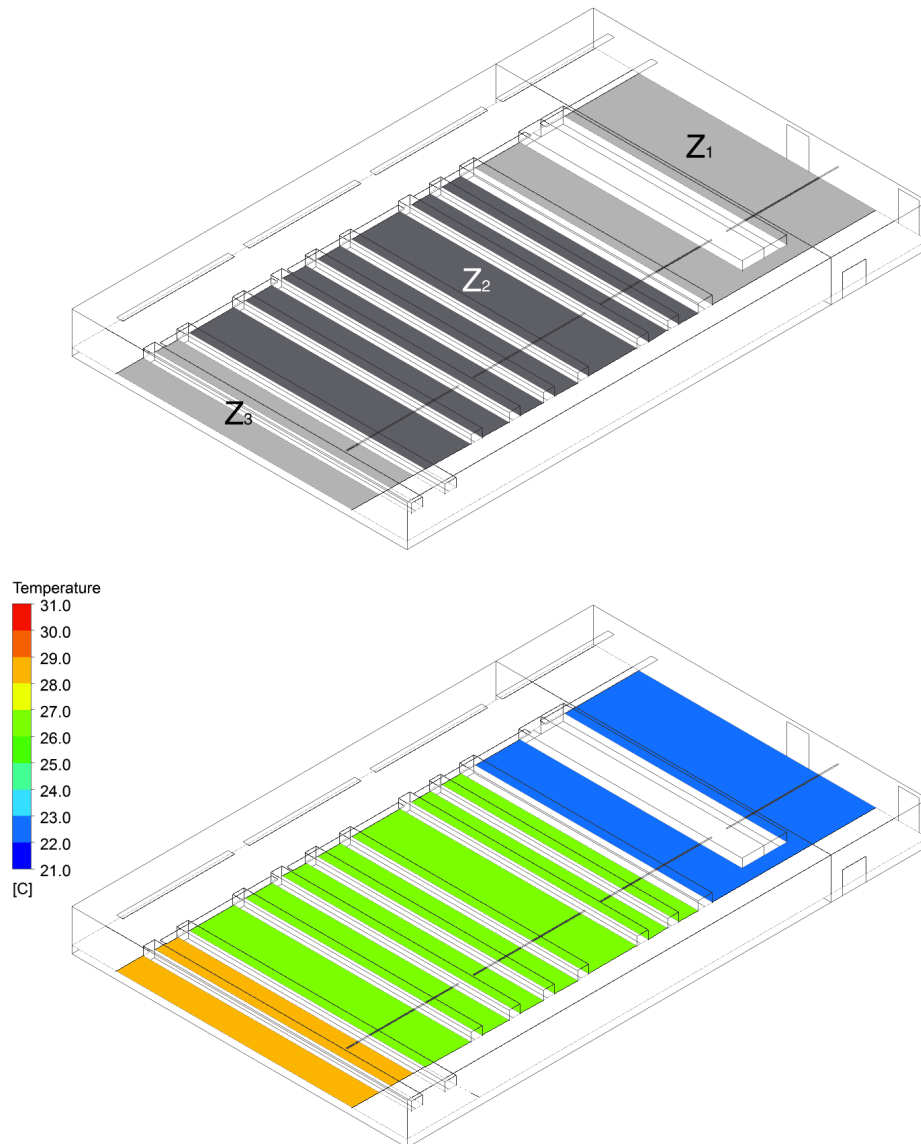


Figure 4.7: Zones Z_1 , Z_2 , and Z_3 , and the associated mean air temperature obtained by Guise (2014).

stated before, the air coming from the ventilation grids does not have a significant impact at 1.0m of height. With experimental results obtained by Guise (2014) [5], was computed a mean water vapor mass fraction of 0.0072 on plane P6 which falls between the range of the numerical results obtained on the same plane.

In order to analyze what it would be if there was no air flow coming from the corridors, a simulation was carried out considering a velocity-inlet boundary condition on boundaries C1, C2, and C3 with a null air velocity and a null heat flux. The velocity magnitude contours are shown in Fig. 4.9 and the temperature contours are shown in Fig. 4.10 for the vertical and the horizontal planes defined before.

The results show that at 1.0m of height the velocity magnitude is lower compared to the

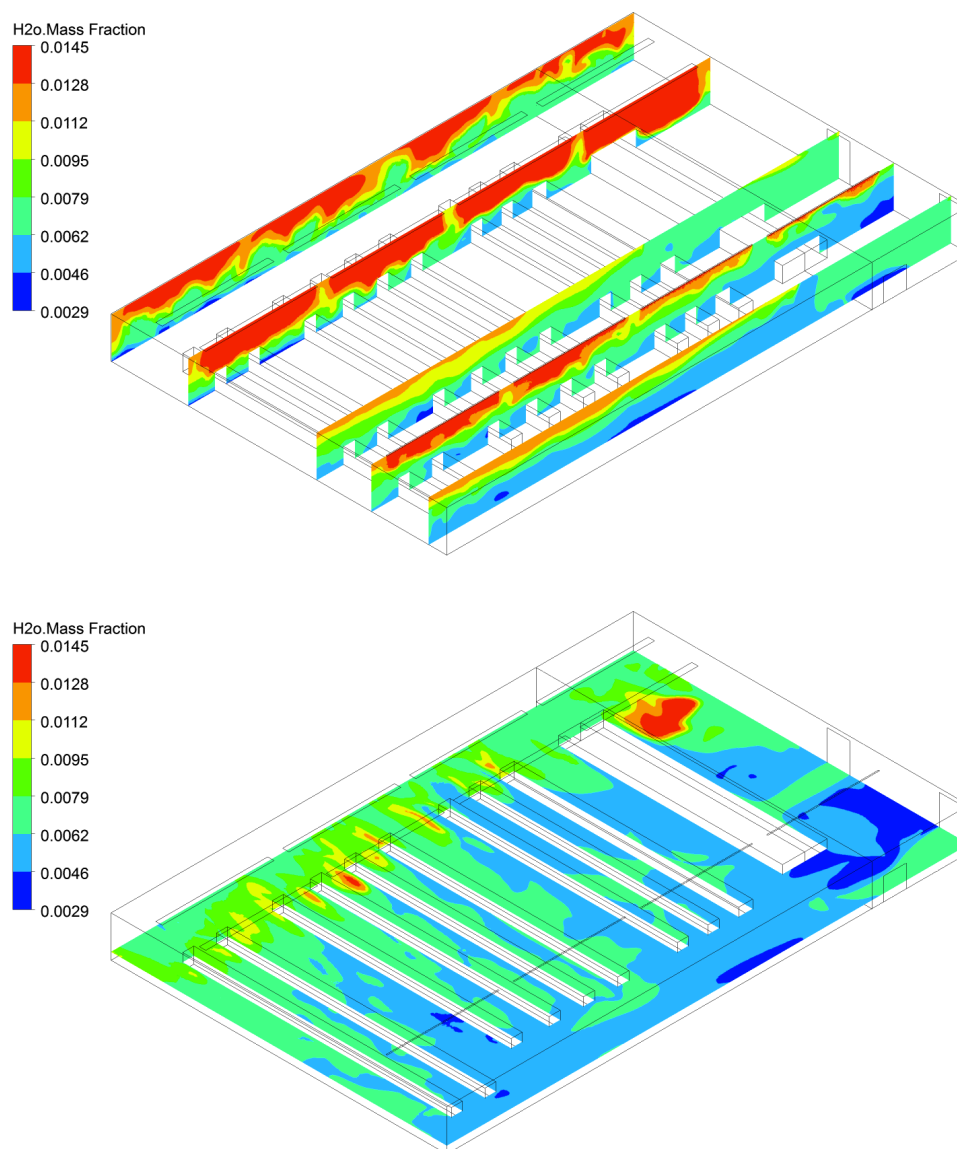


Figure 4.8: Water vapor mass fraction contours on the vertical planes (top panel) and on the horizontal plane (bottom panel).

previous simulation. In fact, only the regions beneath air ventilation grids V5 to V12 achieve velocities up to 0.57 m/s. What concerns to the temperature, the numerical results show values ranging from 27 °C to 35 °C between the twister machines. Due the larger air flow beneath air ventilation grids V5 to V12, the temperature decreases ranging from 21 °C to 25 °C in this region.

When compared to the current situation (first simulation), these results indicate that the fresh air flow coming from the corridors have a crucial role in the thermal environment of Nave 2 maintaining a larger air flow and, therefore, cooling the environment. Even assuming that these results may be contain a small error, they certainly explain the reasons and what would happen to the thermal environment if there was no natural ventilation of the building.

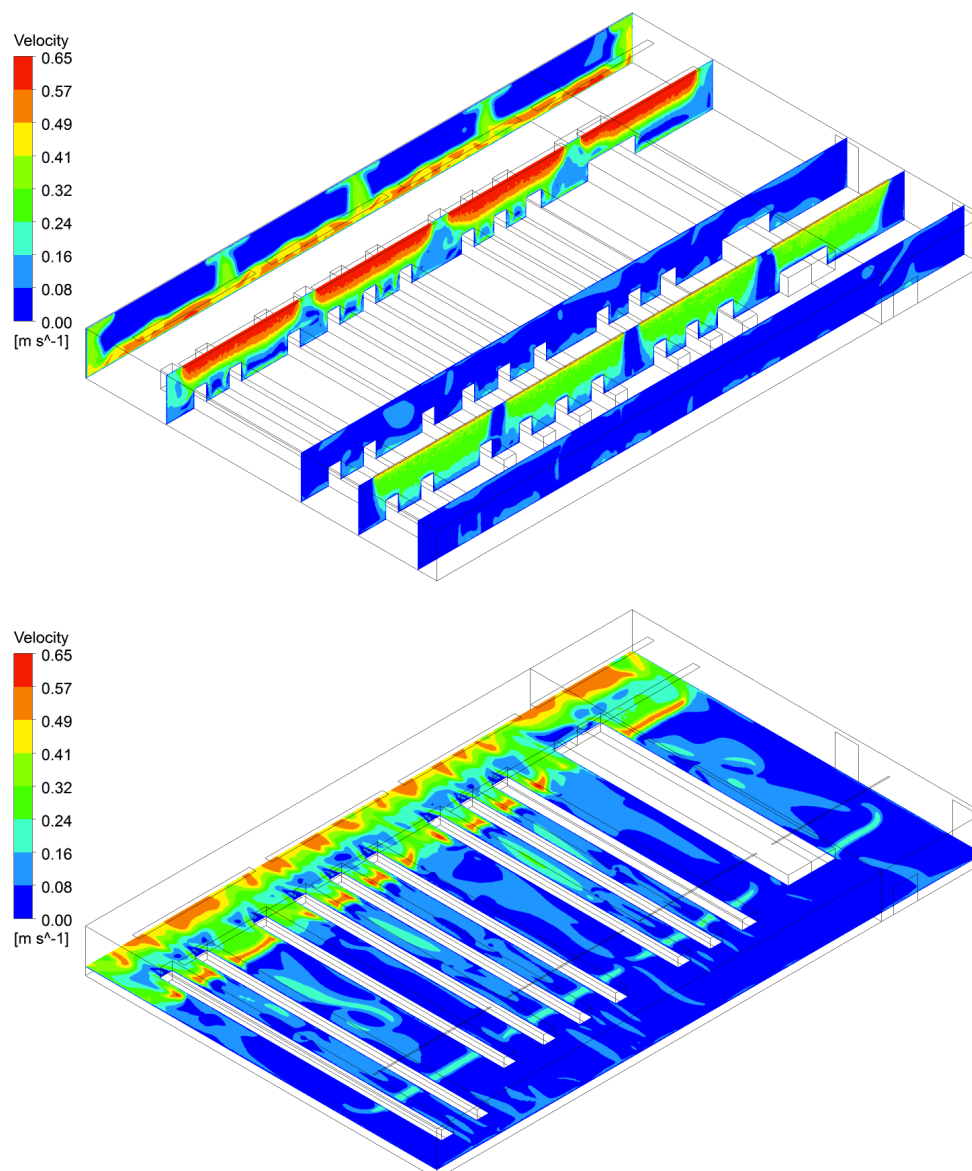


Figure 4.9: Air velocity magnitude contours on the vertical planes (top panel) and on the horizontal plane (bottom panel).

4.2. Alternative Layout

In the search for an optimal thermal environment, one can test through the numerical procedures presented before, hypothetical scenarios changing the layout or the boundary values imposed. Such layout or boundary values variations corresponds to hypothetical conditions but must represent possible scenarios in practice, that is, there is no practical interest in studying scenarios that can never exist in practice due structural, economic, or organizational reasons. That implies one should evaluate very carefully whether it can be done before assess which improvements can be achieved.

For example, it was concluded that the air ventilation in Nave 2 has a very low air flow rate

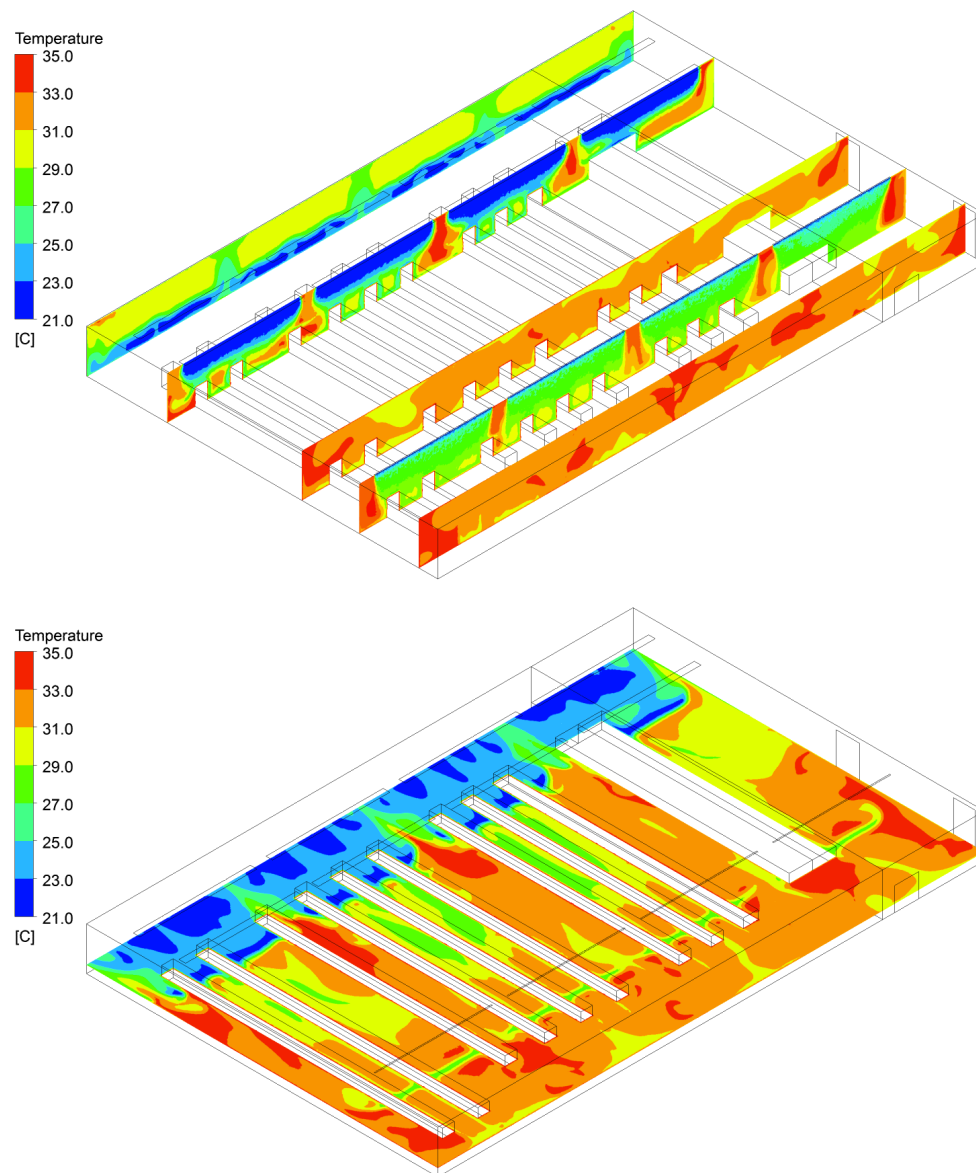


Figure 4.10: Air temperature contours on the vertical planes (top panel) and on the horizontal plane (bottom panel).

and distribution for such large heat production by the twisters machines. In this way, the air is not efficiently cooled exposing the workers to critical temperatures. One could consider changing the air ventilation layout or increasing the power of the ventilation system (hypothetical scenarios). Before studying numerically such scenarios, a structural analysis should be carried out in order to evaluate whether the air ventilation grids and ducts may change and how much its power can be increased (which may imply technological upgrades in the climatic chambers), as well as an economic analysis to assess how much it costs to the company.

Such analysis is not in the scope of this work but the author highly recommend that it should be performed in further work since the air ventilation in Nave 2 is a critical component. Instead, in the following hypothetical situation the twister machines arrangement have changed which, in fact,

is a possible scenario since other layout modifications have been performed in the past. Fig. 4.11 shows the CAD design of the alternative layout proposed in this work. The idea is to arrange the twister machines in the direction of the air coming from corridors C1 and C2 trying, in this way, to improve the air circulation between them and therefore reducing the air temperature.

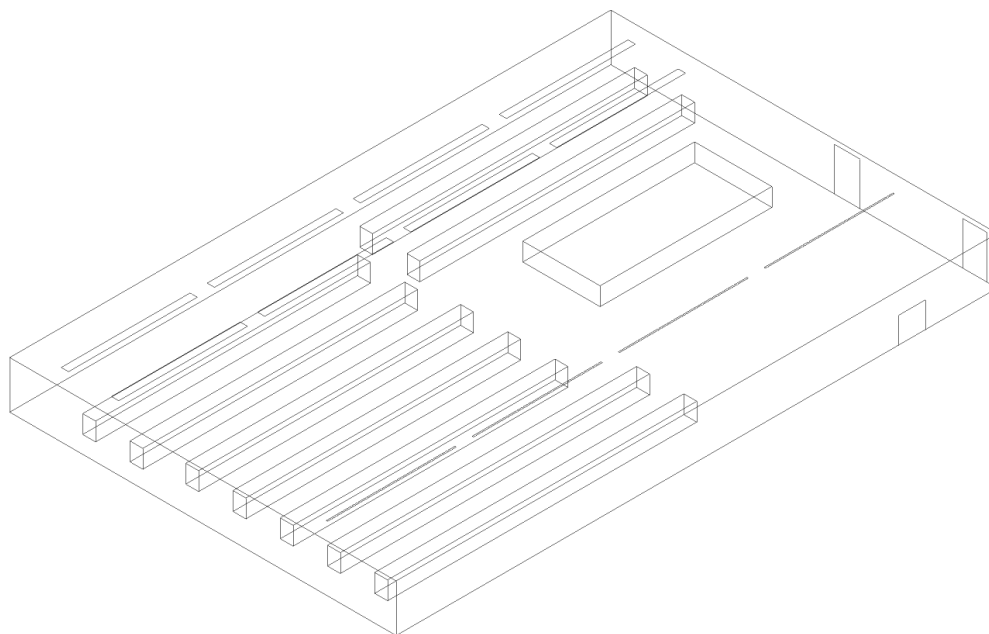


Figure 4.11: CAD design for the alternative layout.

As in the last case, five vertical planes and one horizontal plane at 1.0m of height, as shown in Fig. 4.12, have been considered in order to visualize the solution contours.

The boundary conditions are the same and the simulation has been carried as in the previous case. In Fig. 4.13 the air velocity contours on the planes considered are shown and in Fig. 4.14 and Fig. 4.15 the streamlines associated to the corridors C1 and C2 and to the air ventilation grids V1 to V4 and V5 to V12 are shown, respectively. Additional figures are provided in Annex II of this document.

The first conclusion to be drawn from the results obtained is that the air velocity distribution does not improve between the twister machines for this arrangement. As in the previous case, a large air flow comes from corridors C1 and C2 but the air velocity vanishes in the opposite side of the premise. Although the twister machines arrangement aligned to the corridors eases the air circulation between, that does not improve significantly the air circulation between them due the low air flow rate at these distances. Once again, improve the air ventilation coming from the ventilation grids on the ceiling is of crucial importance in order to achieve lower temperatures in the premise.

The air temperature distribution is shown in Fig. 4.16. As expected due the low air circulation,

the regions close to the opposite side of the corridors achieve critical values of temperature up to 35 °C. Moreover, the air coming from corridor C2 goes through all the twister machines on the left side of the premise before leaving the considered domain. In this way, the air is successively heated as it passes through one twister machine to another rising its temperature to critical levels. On the other hand, the regions on the right side of the premise (supermarket side) are relatively cooled due the large air flow and the fewer number of operating twister machines.

The water vapor mass fraction in the air is shown in Fig. 4.17.

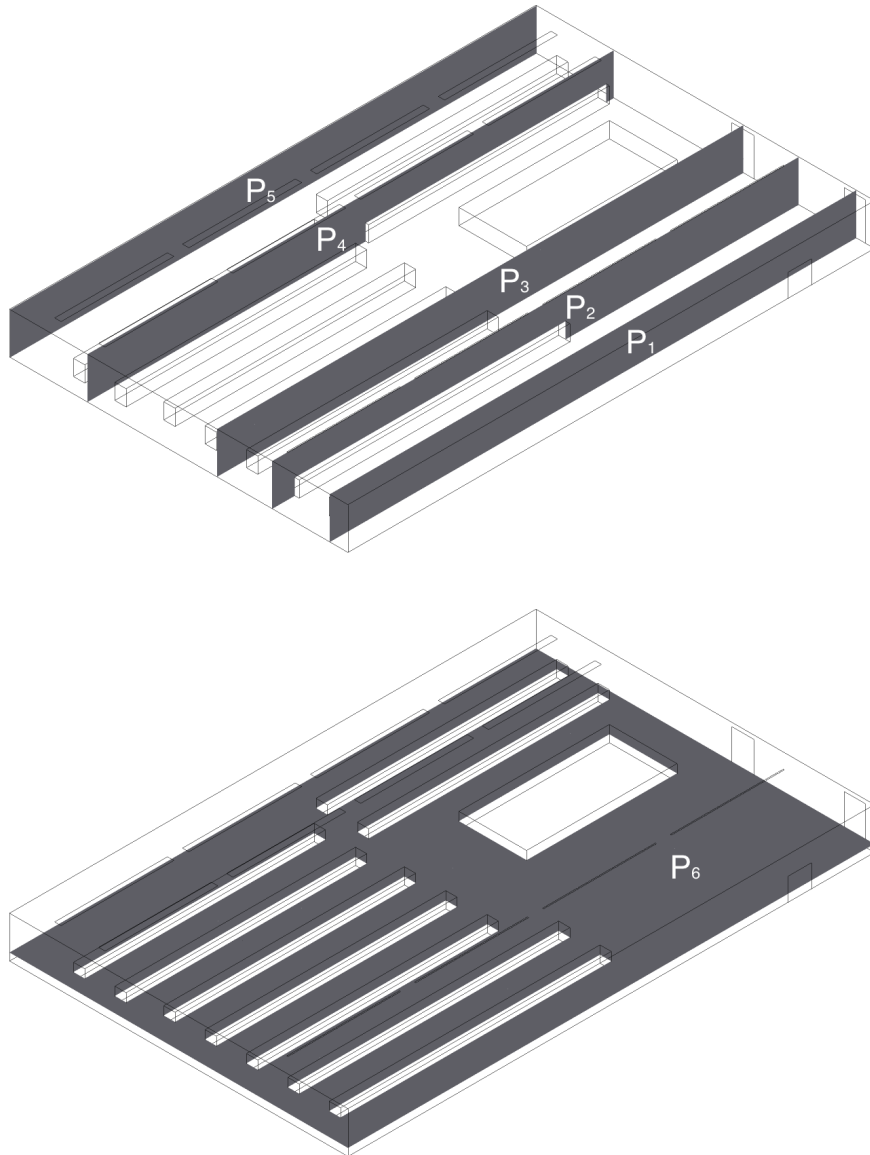


Figure 4.12: Planes P₁, P₂, P₃, P₄, and P₅ (top panel), and P₆ (bottom panel) to visualize the solution.

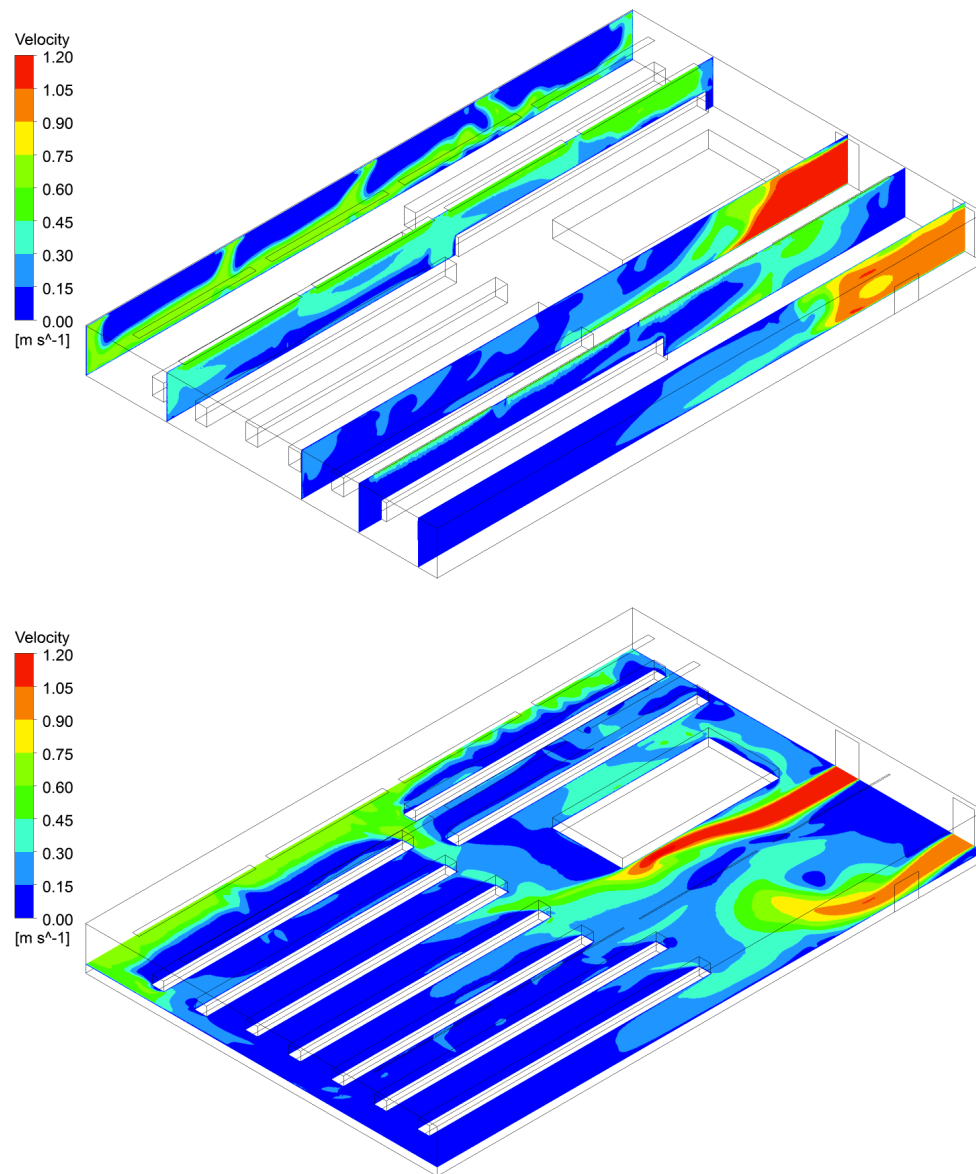


Figure 4.13: Air velocity magnitude contours on the vertical planes (top panel) and on the horizontal plane (bottom panel).

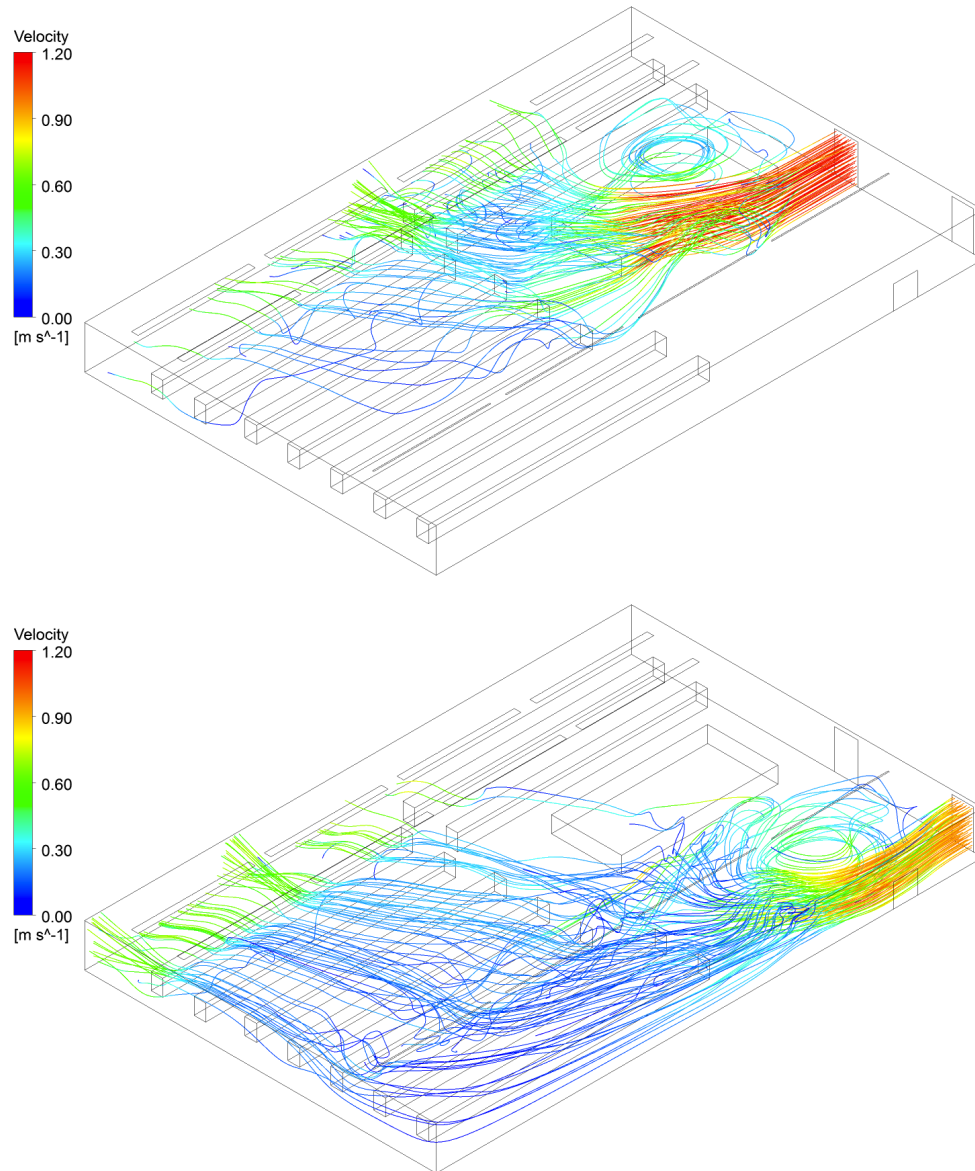


Figure 4.14: Streamlines starting from corridor C1 (top panel) and from corridor C2 (bottom panel).

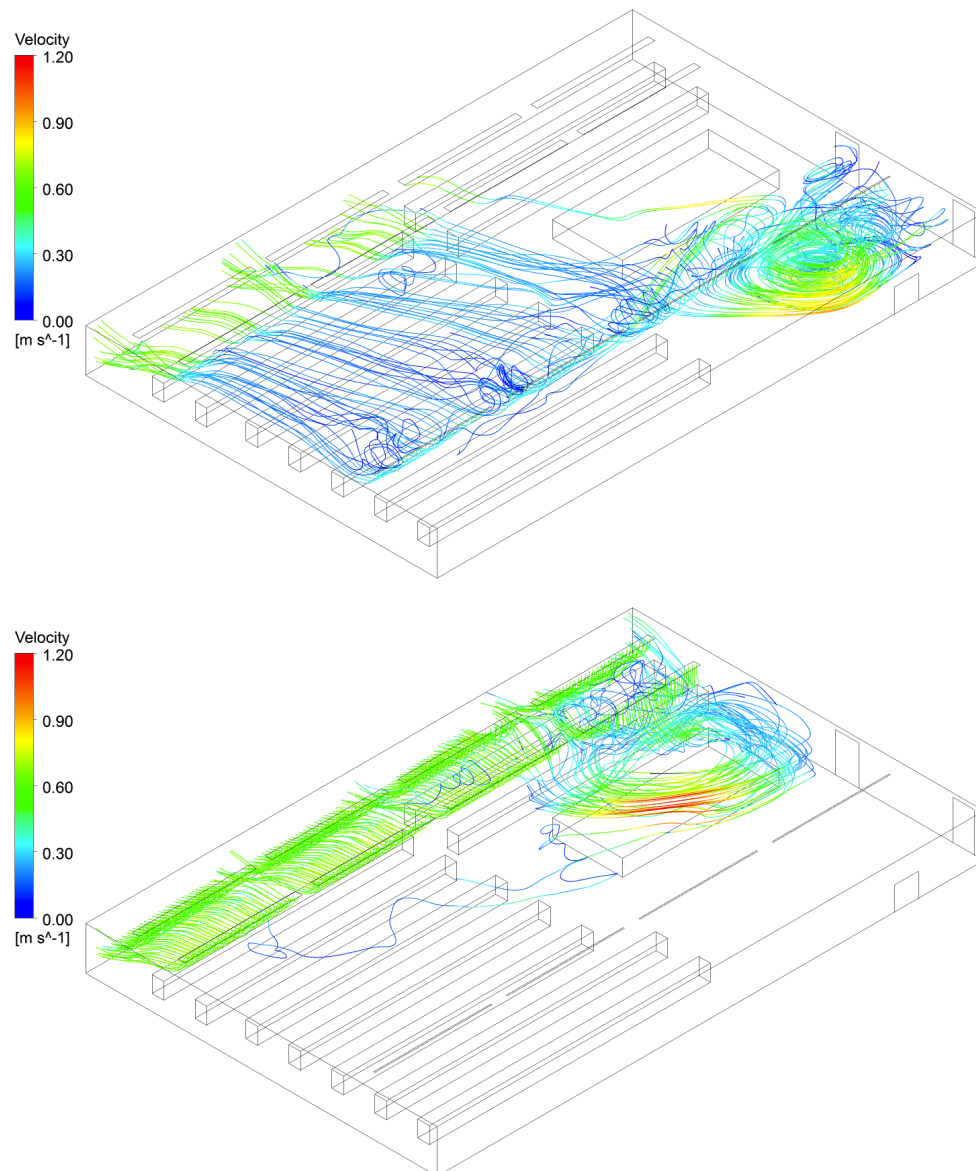


Figure 4.15: Streamlines starting from air ventilation grids V1 to V4 (top panel) and from air ventilation grids V5 to V12 (bottom panel).

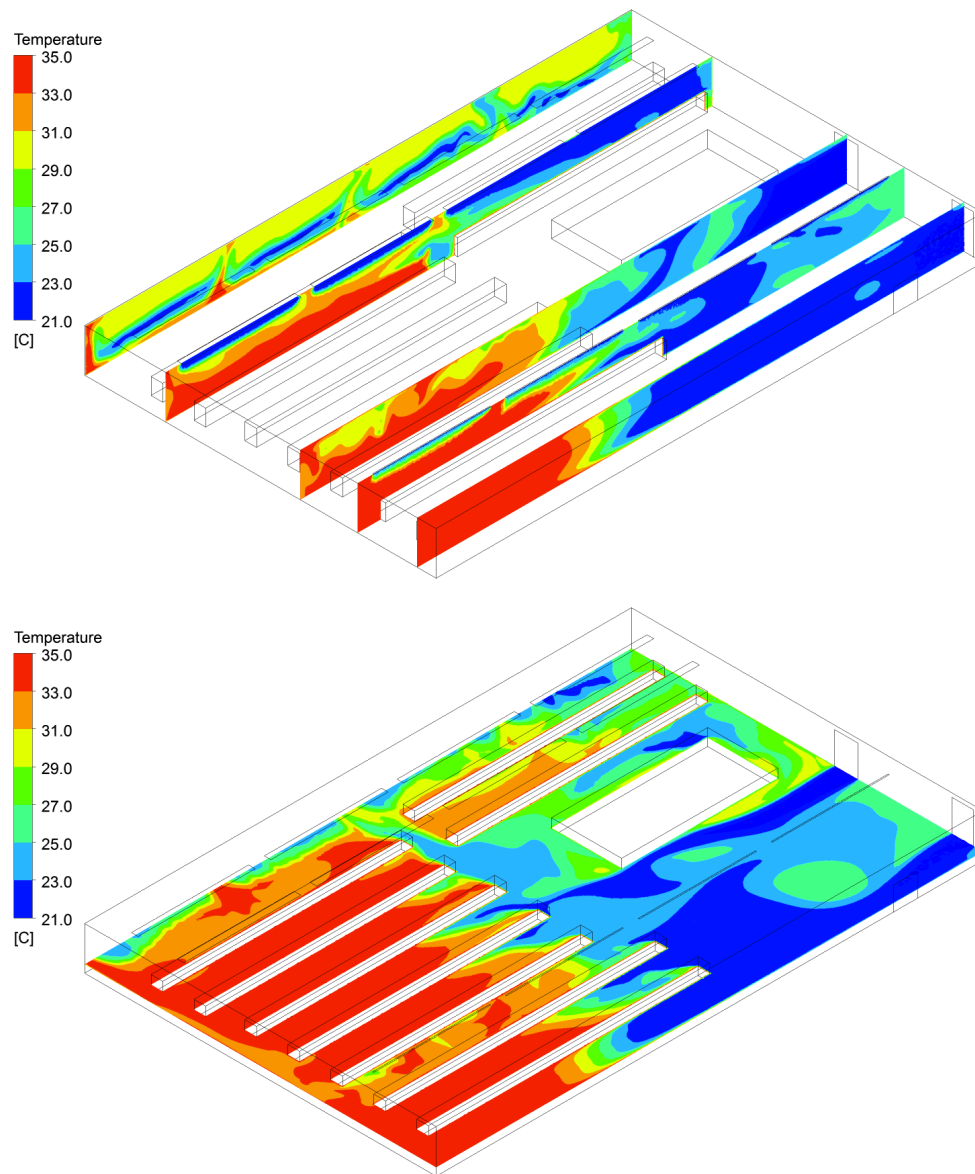


Figure 4.16: Air temperature contours on the vertical planes (top panel) and on the horizontal plane (bottom panel).

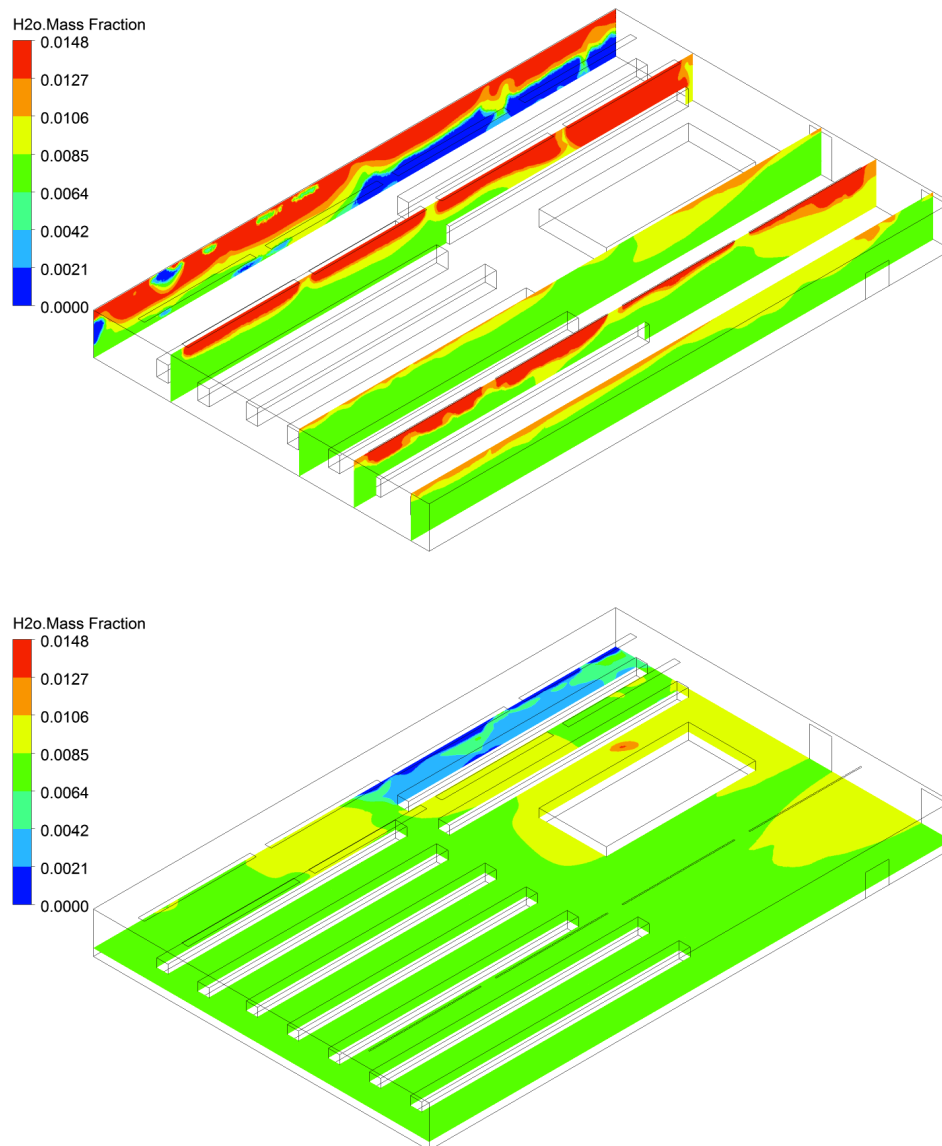


Figure 4.17: Water vapor mass fraction contours on the vertical planes (top panel) and on the horizontal plane (bottom panel).

5. Conclusions and Further Work

The main goal of this work was to assess whether the numerical modeling is a suitable framework to characterize and to evaluate the physical component of the thermal environments in industrial premises. For this purpose, a computational tool for simulation of CFD problems was studied and applied in the context of thermal environments, in particular, the one of the twisting premise of C-ITA company located in Lousado, Portugal. Several simulations were carried out in order to test both the current layout and an hypothetical one in the search for an optimal thermal environment. The analysis of the results obtained were already presented and now this documents ends with the conclusions and the further work.

5.1. Conclusions

The very first conclusion to be draw from this work is that the physical component of a thermal environment can be virtually assessed using numerical procedures in CFD simulation tools as Ansys. Indeed, the physical variables considered in thermal environments, such as air temperature, air velocity or humidity, follow the same physical laws as those in any other problem concerning fluid flows and heat transfer. In this way, using CFD standard models as Navier-Stokes equations, convection-diffusion equation, and radiation equation, among others, and using computational methodologies and numerical procedures, one can model and assess the physical component of a thermal environment. Of course, one still has the need to assess other components with experimental measurements depending on the comfort model considered, as clothing or metabolism for the Fanger's model. Having all the variables in hand, one can assess the worker's thermal comfort and conclude about the need to implement appropriate measures in order to improve the thermal environment.

Concerning to the twisting premise, it was possible to conclude that the natural ventilation of the building has a crucial importance in this environment cooling the air and maintaining relative low temperatures. Simulating the case were no natural ventilation was present, one can notice larger temperature values which should increase the thermal discomfort of the workers. Moreover, one can also notice that the ventilation system installed on the ceiling is not well designed for such heat generation by the twister machines. Its efficiency could be increased changing the location of the air ventilation grids to inner zones on the ceiling or increasing its power. Such scenarios should be analyzed in detail and previously tested using the same numerical procedures in order to assess the improvements which can be obtained.

In the search for an optimal thermal environment minimizing the temperature and increasing the air circulation in the premise, an hypothetical scenario was simulated changing the orientation of the twister machines. The results obtained evidence larger temperature ranges compared to the current situation and therefore a worst thermal environment. Of course, such situation is not desirable but it proves the interest of using numerical procedures to predict the thermal conditions in different situations and therefore to find the optimal layout.

5.2. Further Work

The results obtained for the twisting premise should be validated using point-to-point experimental measurements in order to well validate the model. If considerable errors were obtained, then one can conclude that relevant boundaries in the premise were not taken into account or some of the considerations done in this work lead to inaccurate results. In that case, a more exhaustive experimental work should be done in order to better define the boundaries and the associated physical values as velocity, temperature or relative humidity.

Regarding to the improvement of the thermal environment in the twisting premise, different scenarios should be tested concerning changes both in the twister machines layout and in the installed ventilation system. Before carry out the simulations, one should analyze carefully if such scenarios are possible from the structural, productive, and economic points of view.

It is also of great interest the development of a framework which inputs the simulated physical variables as well as other factors (as clothing and metabolism) and provides the PMV/PPD or WGBT indices, for example, not at a particular point but for the whole domain. In this way, one should be able to assess the worker's thermal sensation and find improvements for the thermal environment.

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Annexes

Annex I - Simulated Physical Variables for the Current Layout

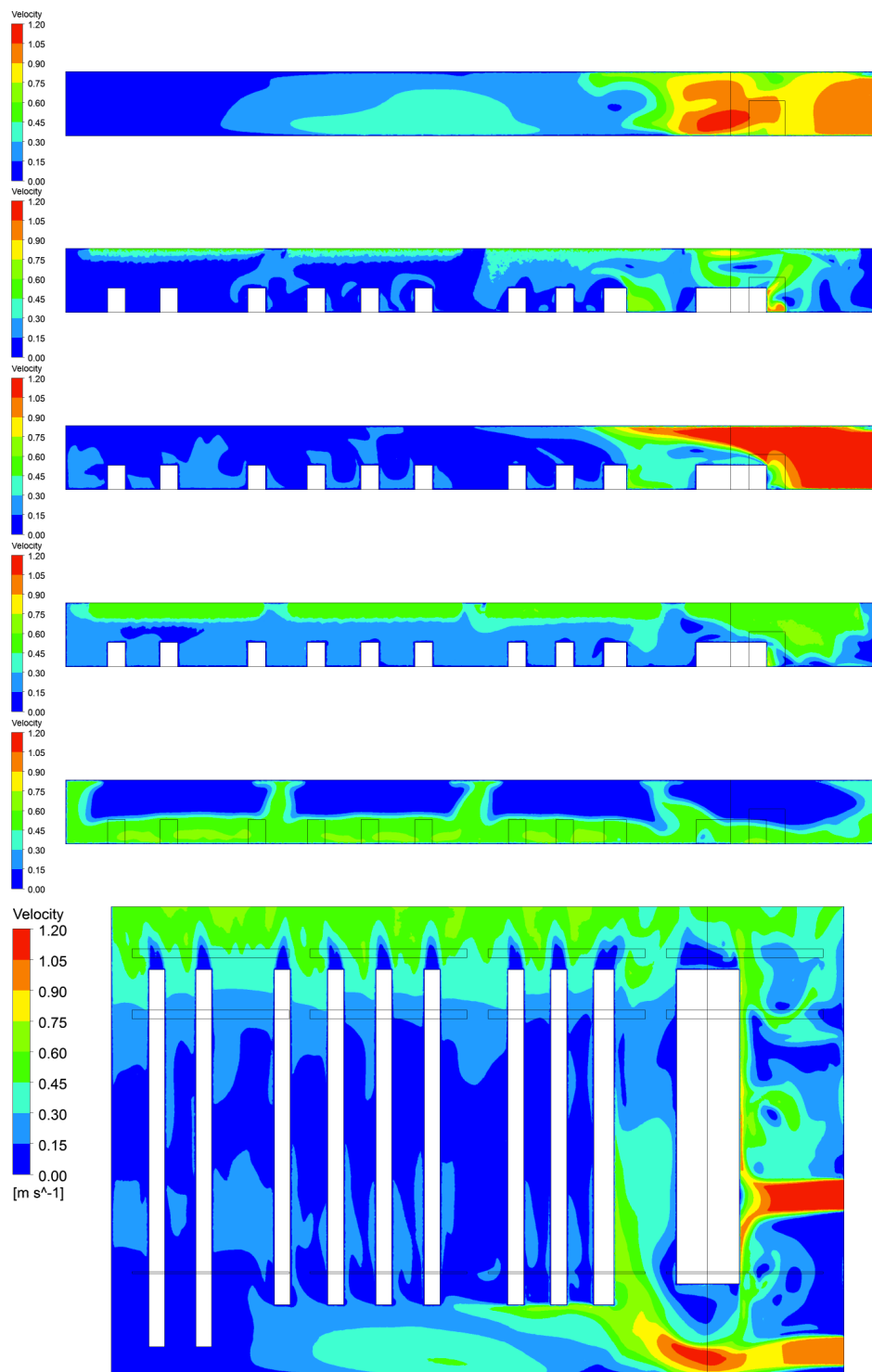


Figure I.1: Air velocity magnitude on the vertical planes P₁, P₂, P₃, P₄, and P₅ and on the horizontal plane P₆ from top to bottom, respectively.

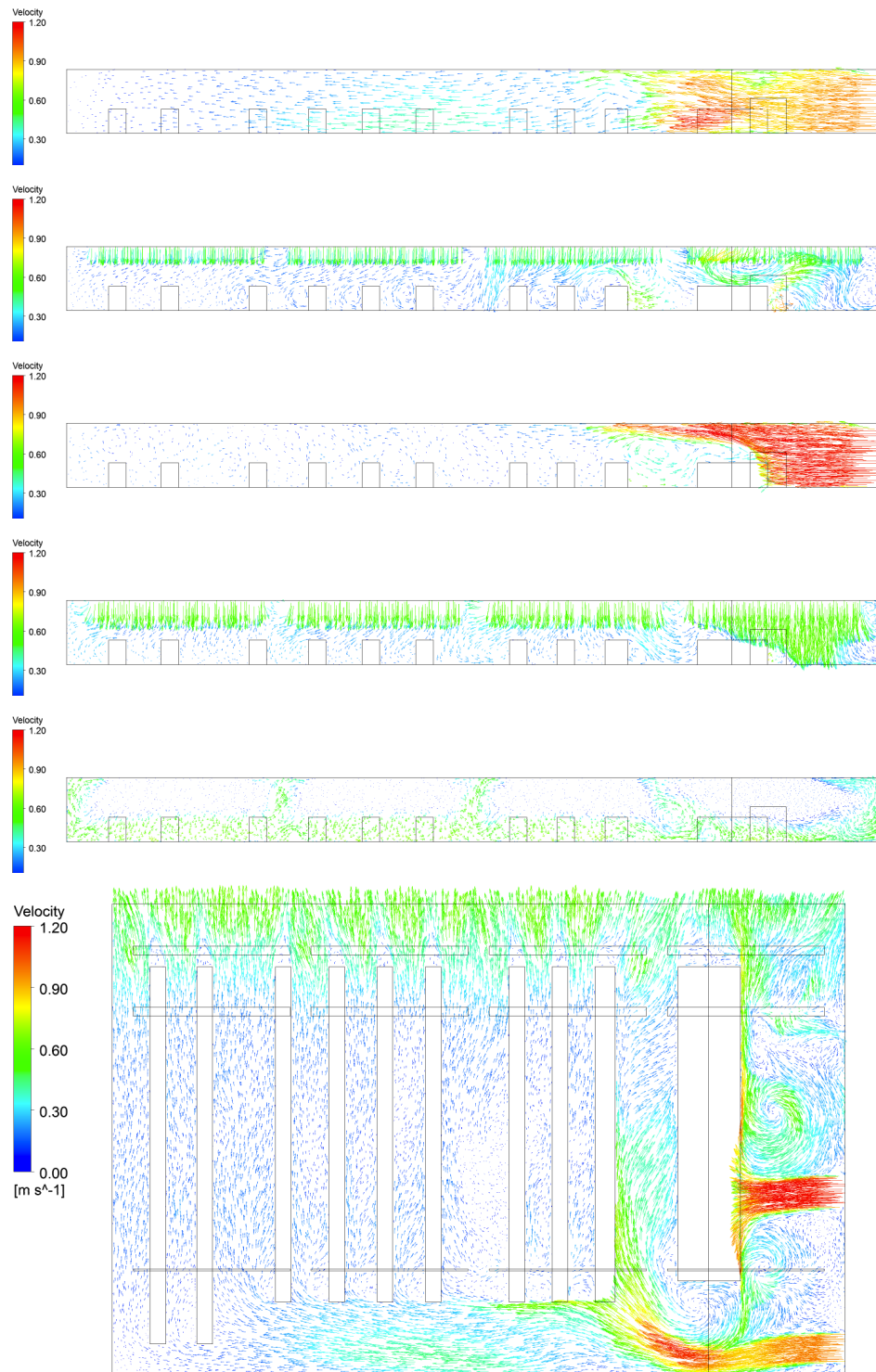


Figure I.2: Air velocity vectors on the vertical planes P₁, P₂, P₃, P₄, and P₅ and on the horizontal plane P₆ from top to bottom, respectively.

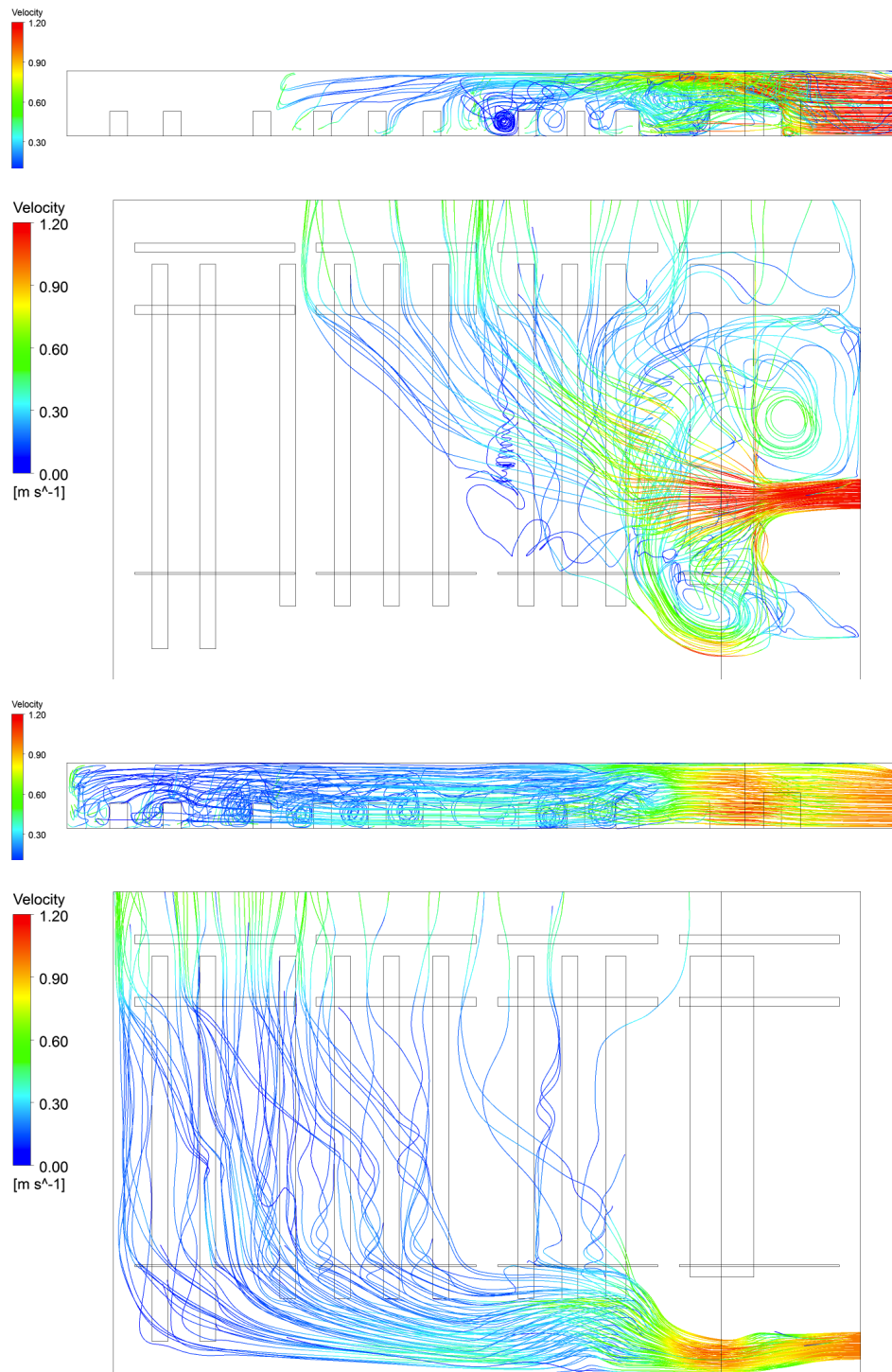


Figure I.3: Streamlines coming from corridors C1 in a vertical and horizontal view and from corridor C2 in a vertical and horizontal view from top to bottom, respectively.

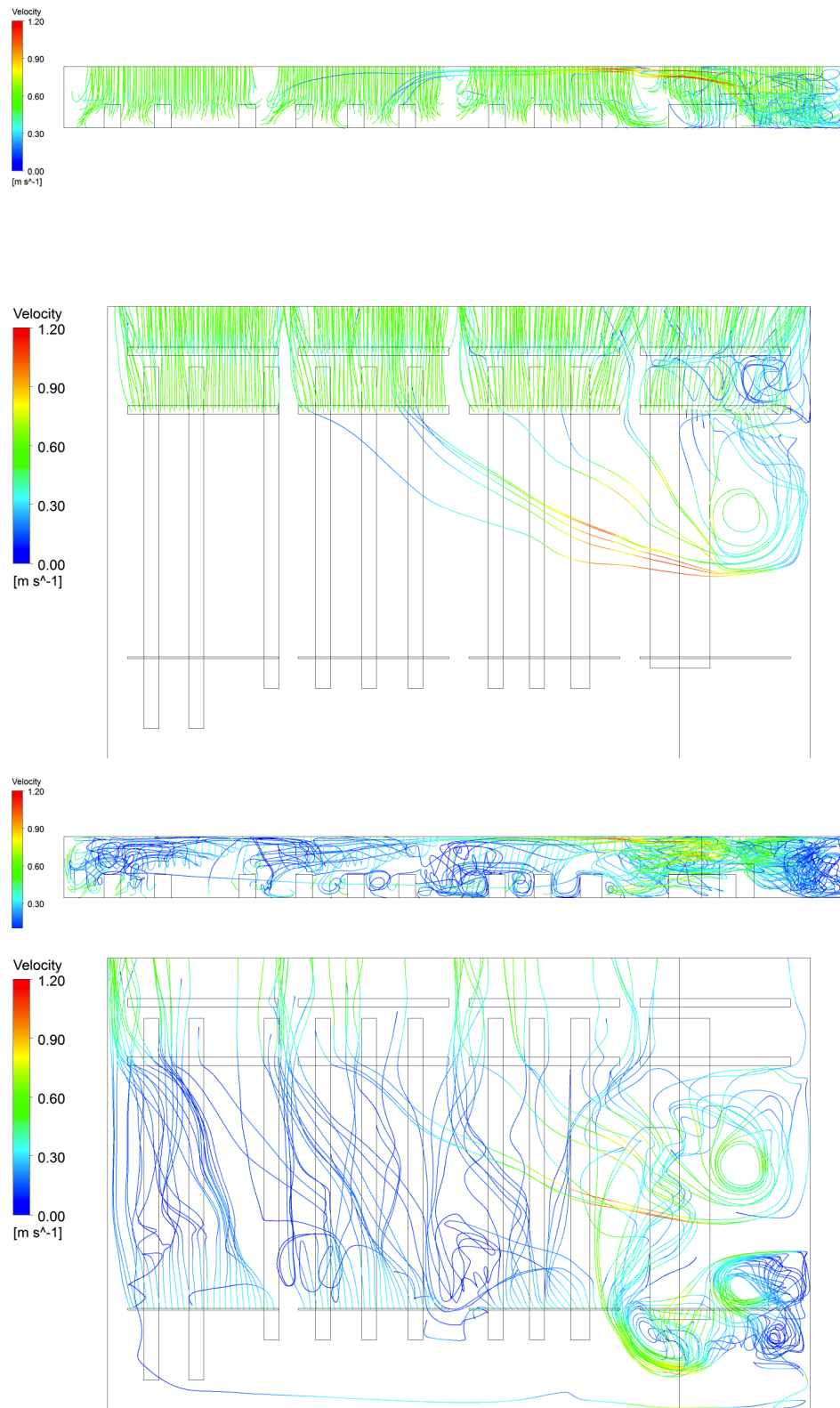


Figure I.4: Streamlines coming from air ventilation grids v1 to V4 in a vertical and horizontal view and from air ventilation grids V5 to v12 in a vertical and horizontal view from top to bottom, respectively.

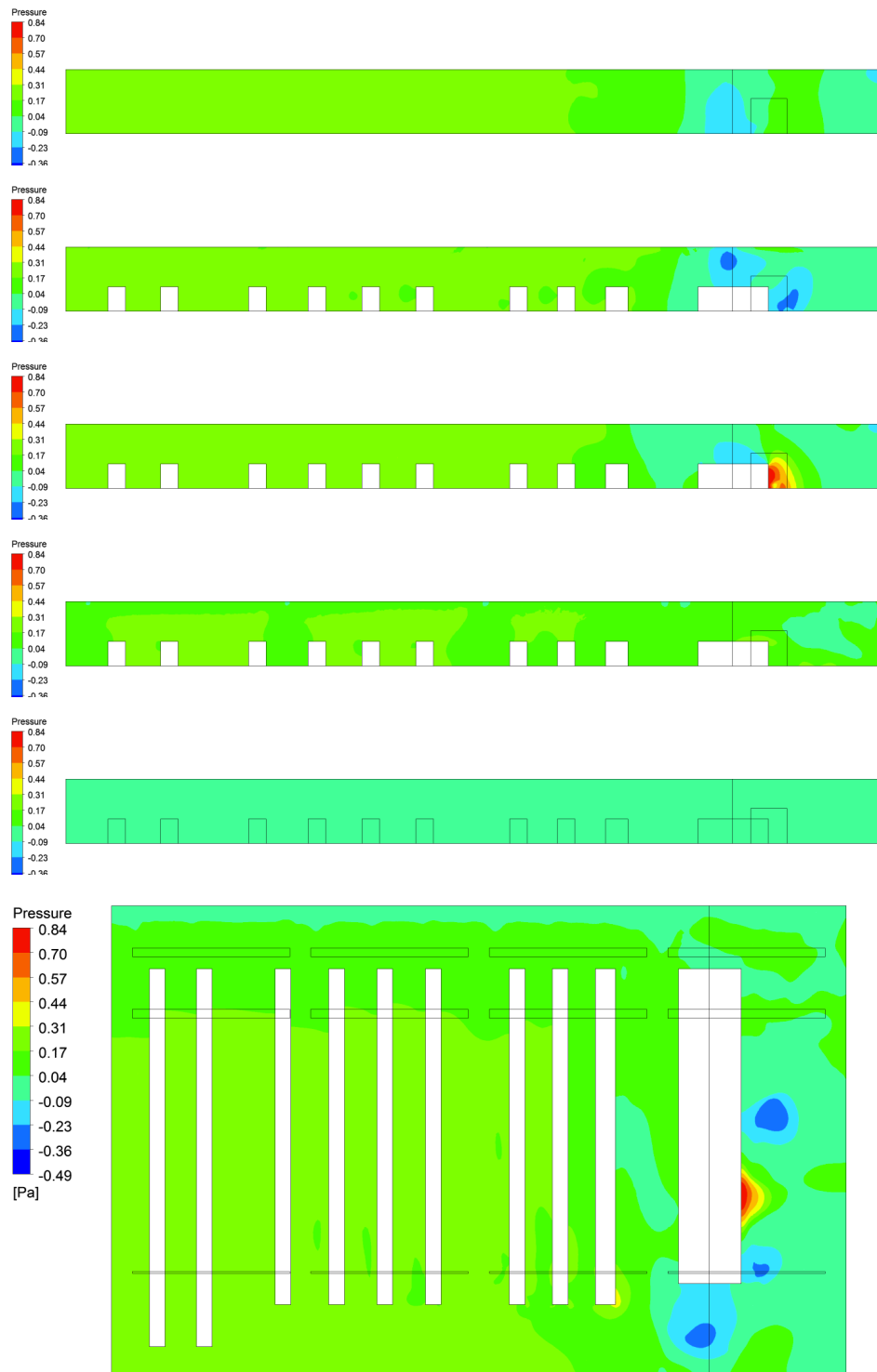


Figure I.5: Air pressure contours on the vertical planes P₁, P₂, P₃, P₄, and P₅ and on the horizontal plane P₆ from top to bottom, respectively.

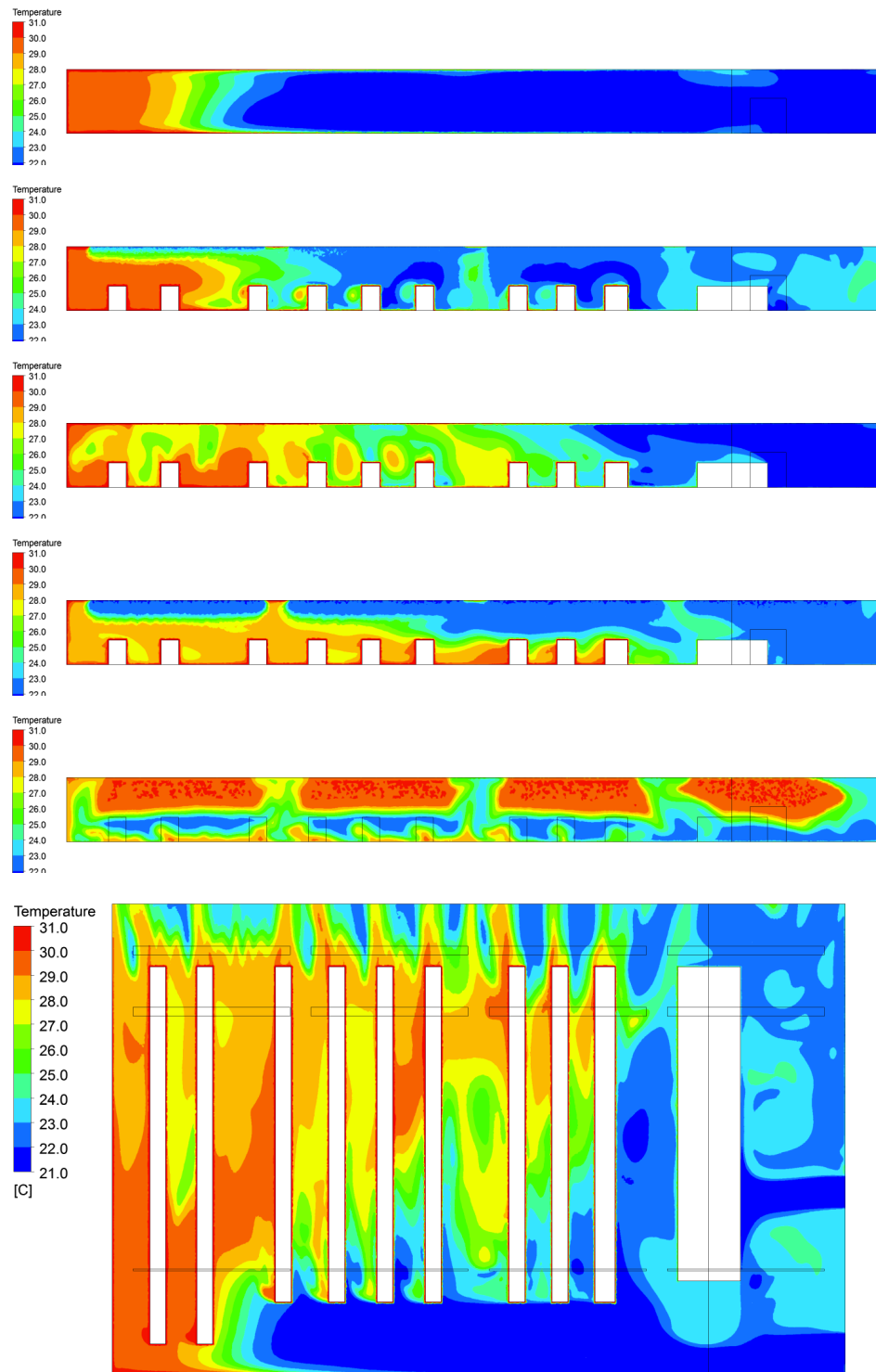


Figure I.6: Air temperature contours on the vertical planes P₁, P₂, P₃, P₄, and P₅ and on the horizontal plane P₆ from top to bottom, respectively.

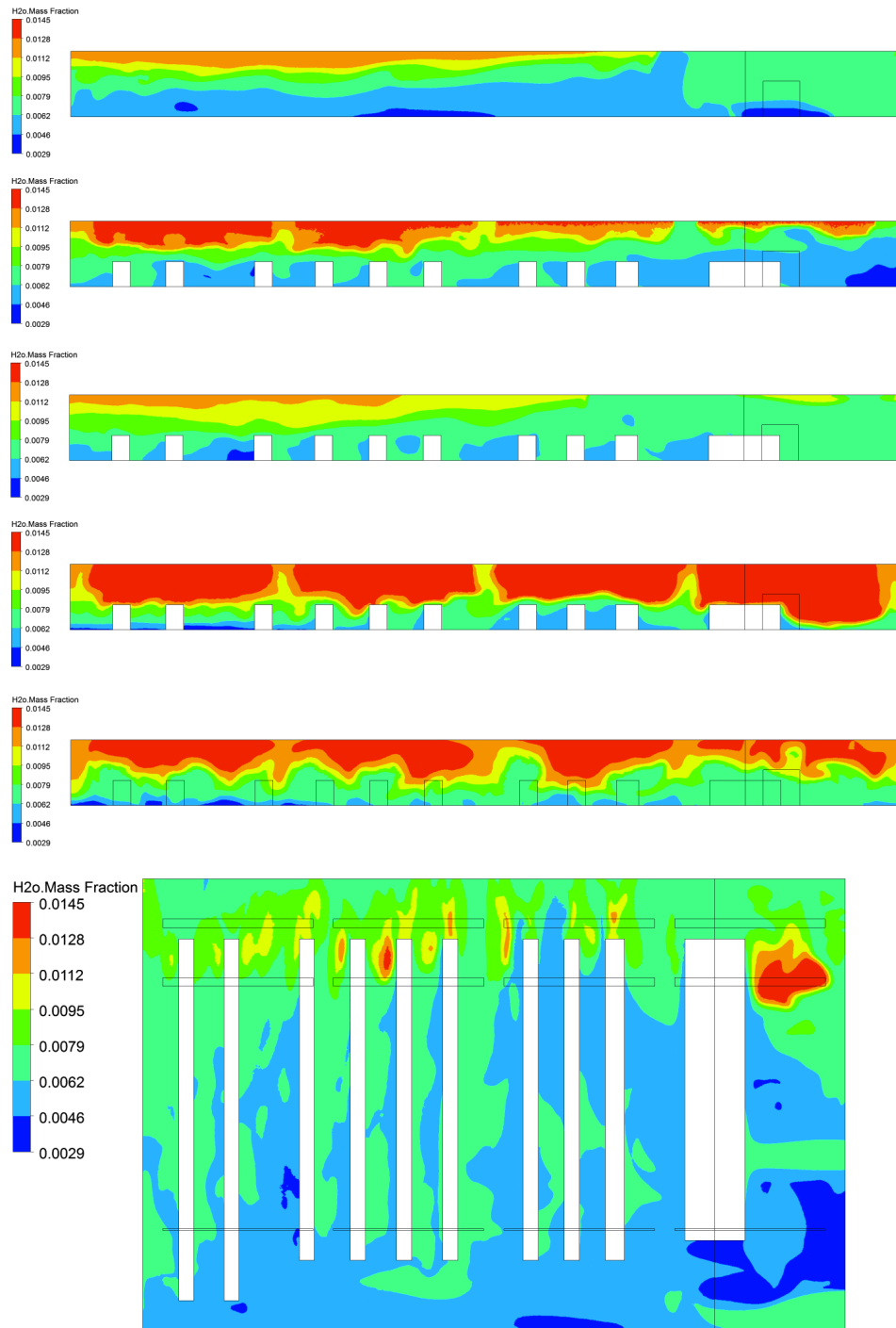


Figure I.6: Water vapor mass fraction contours on the vertical planes P₁, P₂, P₃, P₄, and P₅ and on the horizontal plane P₆ from top to bottom, respectively.

Annex II - Simulated Physical Variables for the Alternative Layout

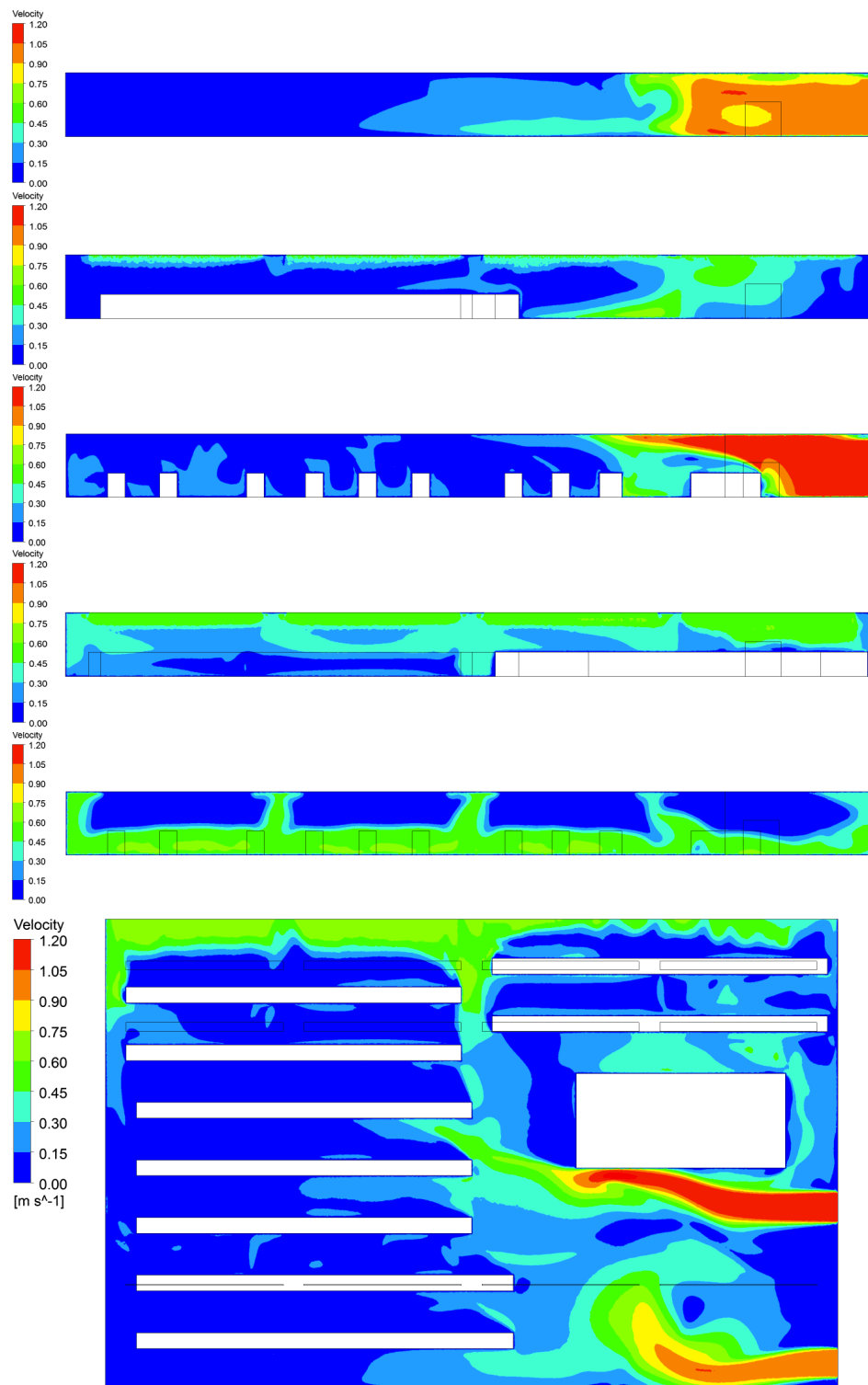


Figure II.1: Air velocity magnitude on the vertical planes P_1 , P_2 , P_3 , P_4 , and P_5 and on the horizontal plane P_6 from top to bottom, respectively.

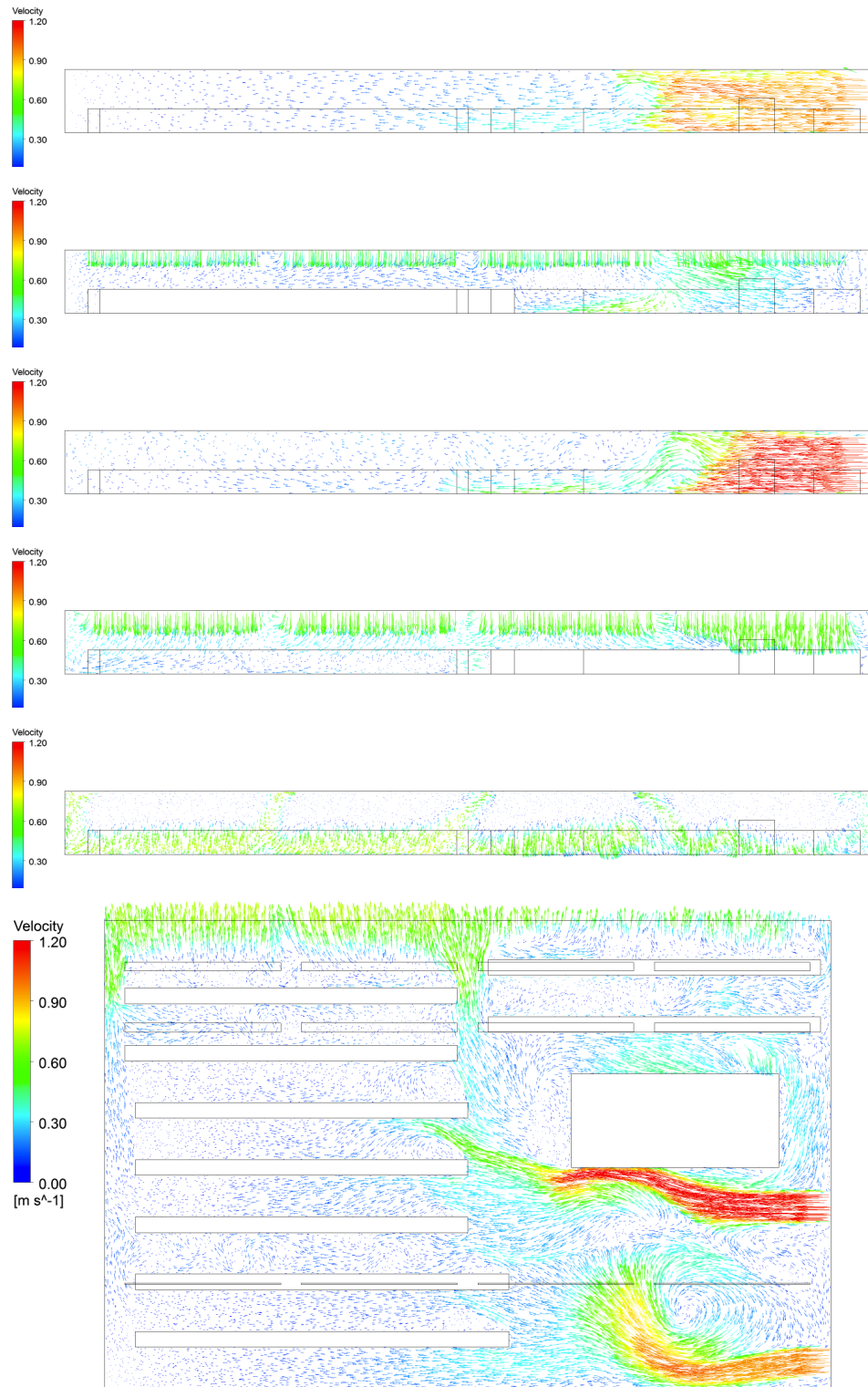


Figure II.2: Air velocity vectors on the vertical planes P₁, P₂, P₃, P₄, and P₅ and on the horizontal plane P₆ from top to bottom, respectively.

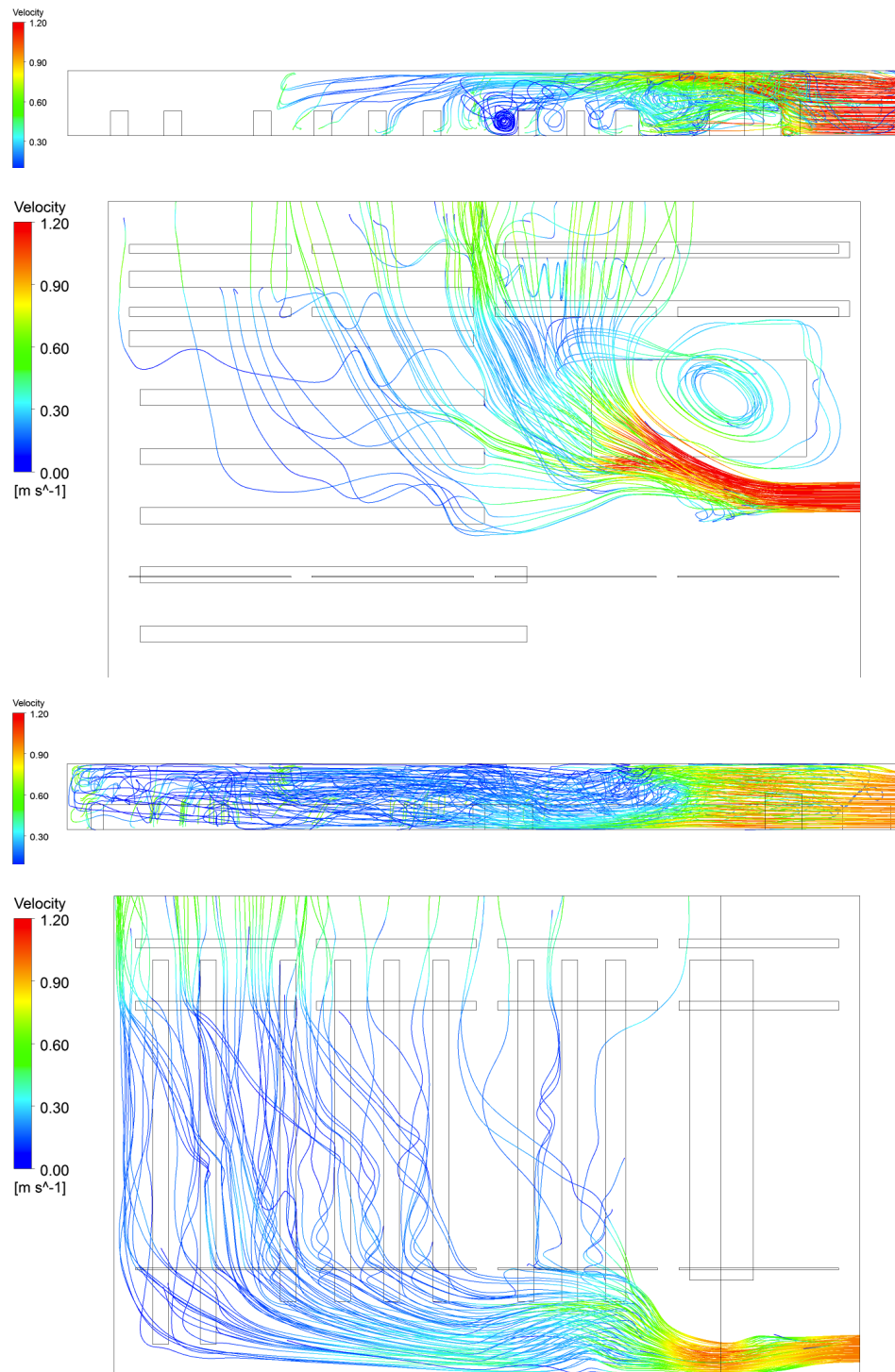


Figure II.3: Streamlines coming from corridors C1 in a vertical and horizontal view and from corridor C2 in a vertical and horizontal view from top to bottom, respectively.

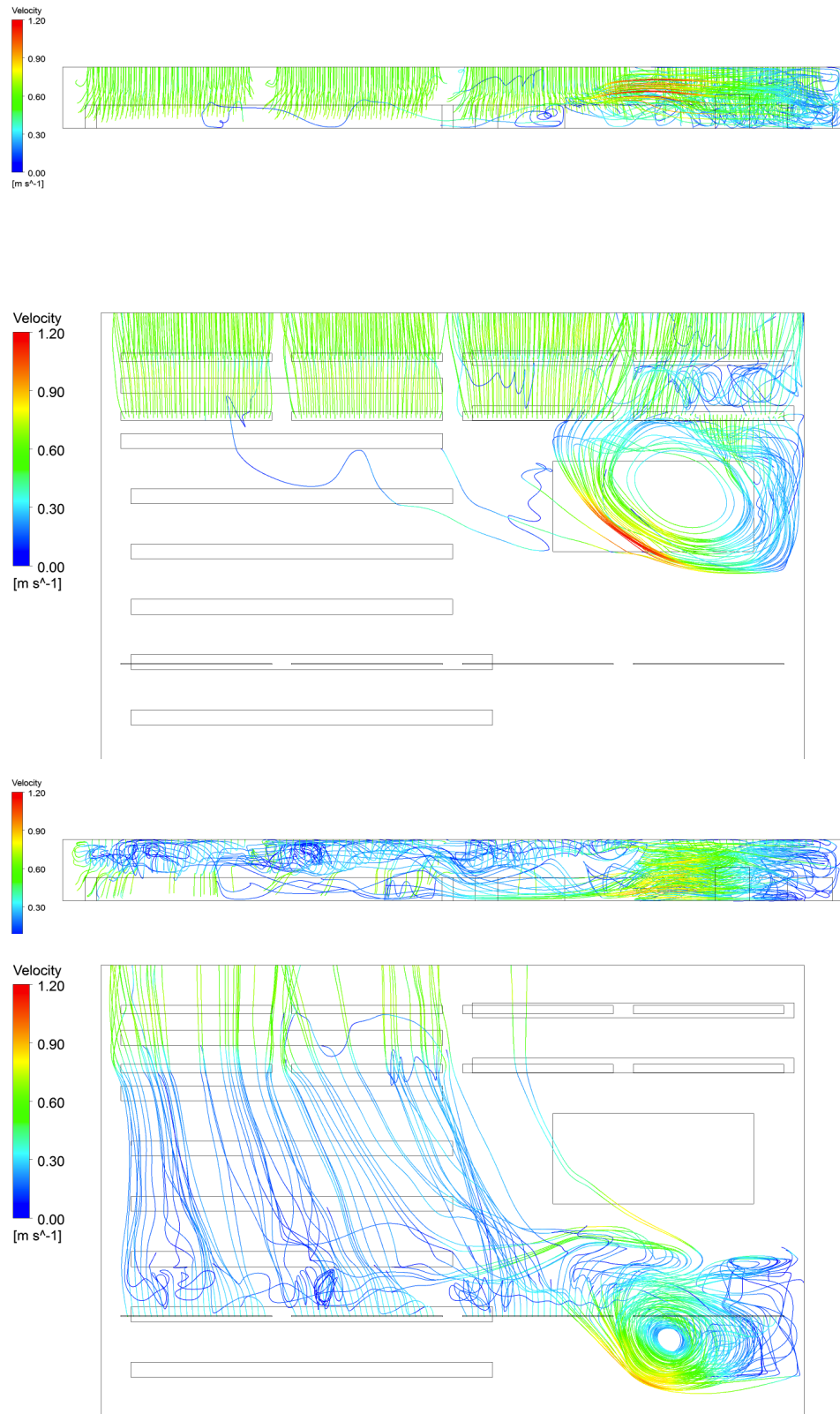


Figure II.4: Streamlines coming from air ventilation grids v3 to V4 in a vertical and horizontal view and from air ventilation grids V5 to v32 in a vertical and horizontal view from top to bottom, respectively.

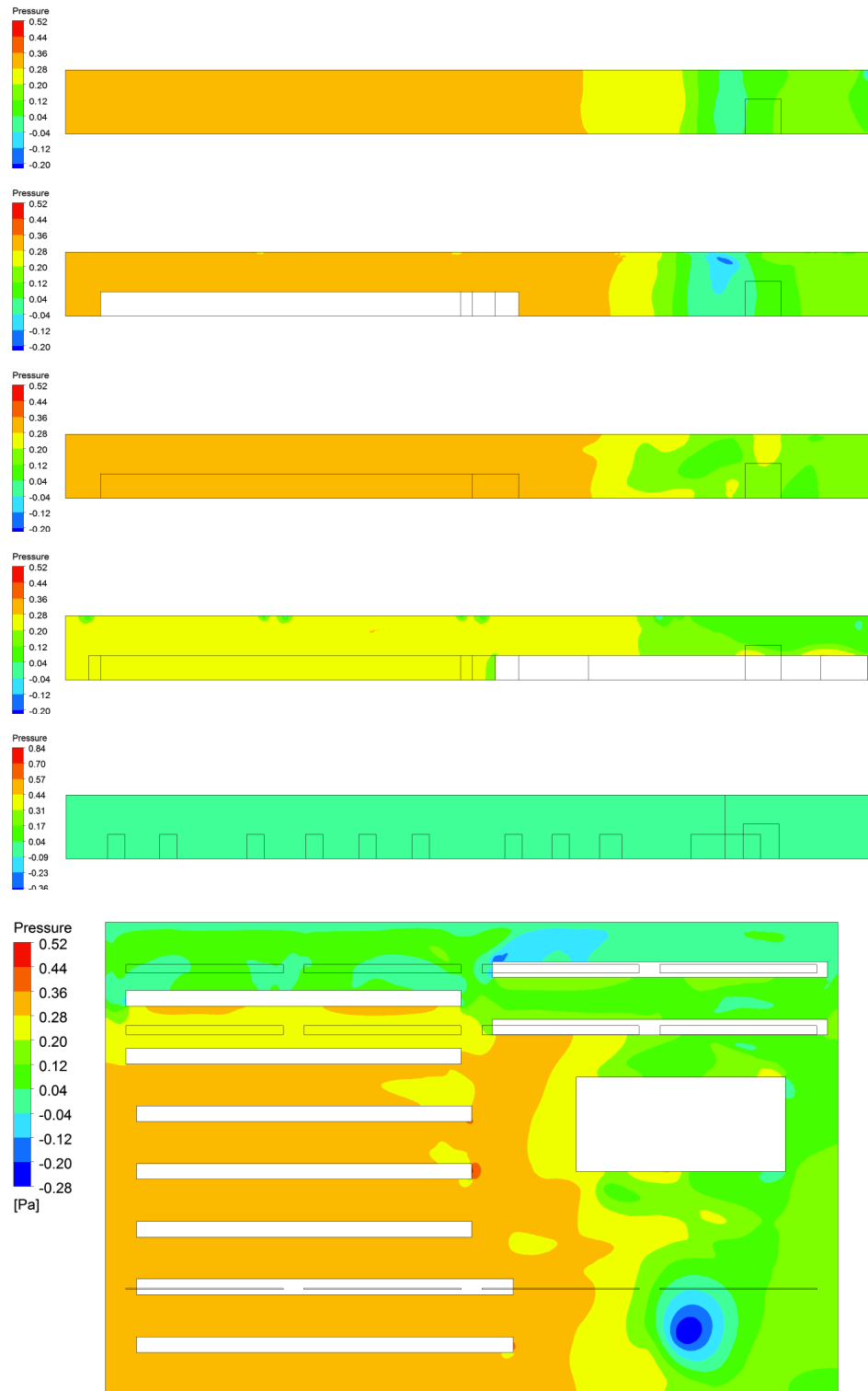


Figure II.5: Air pressure contours on the vertical planes P₁, P₂, P₃, P₄, and P₅ and on the horizontal plane P₆ from top to bottom, respectively.

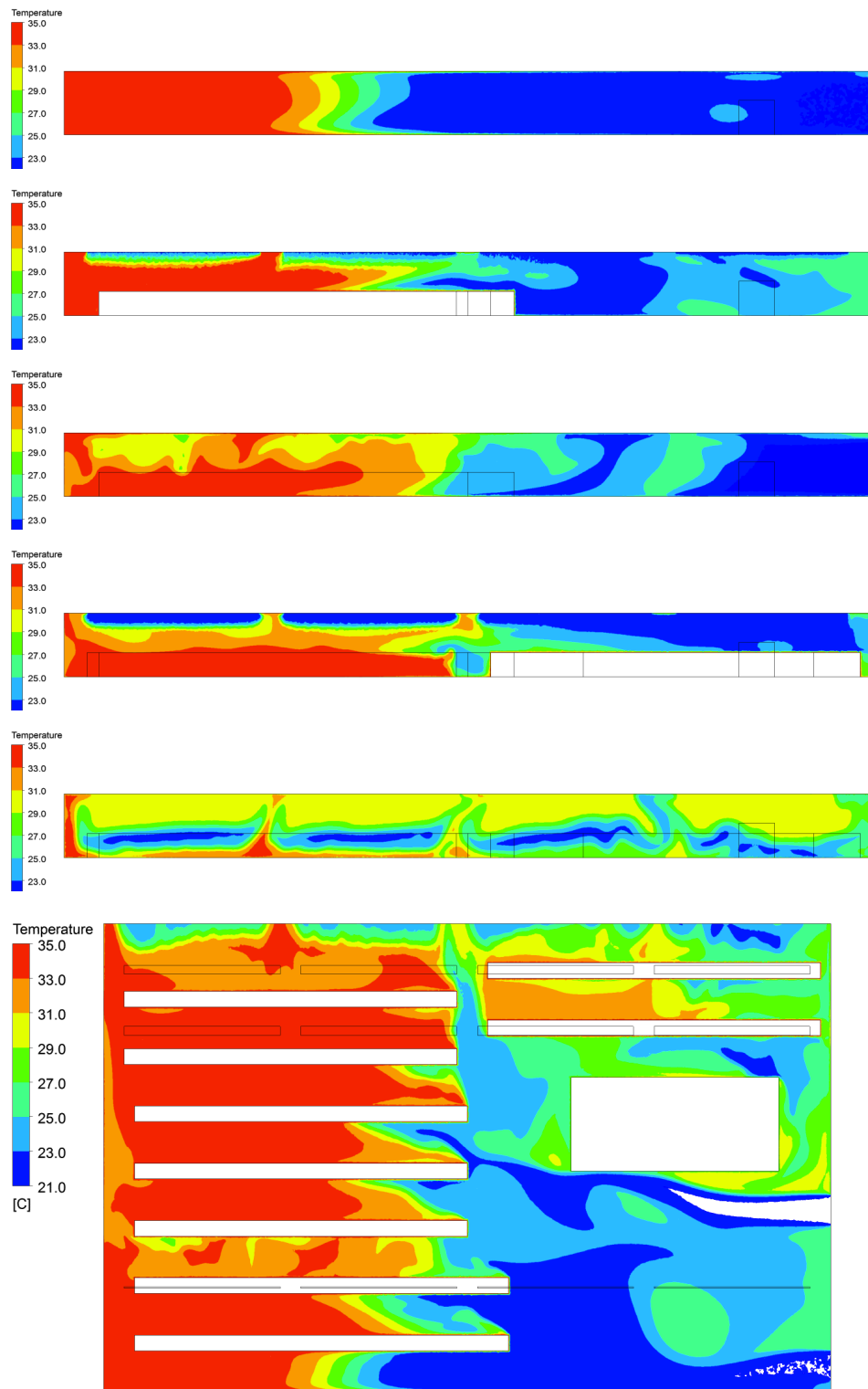


Figure II.6: Air temperature contours on the vertical planes P₁, P₂, P₃, P₄, and P₅ and on the horizontal plane P₆ from top to bottom, respectively.

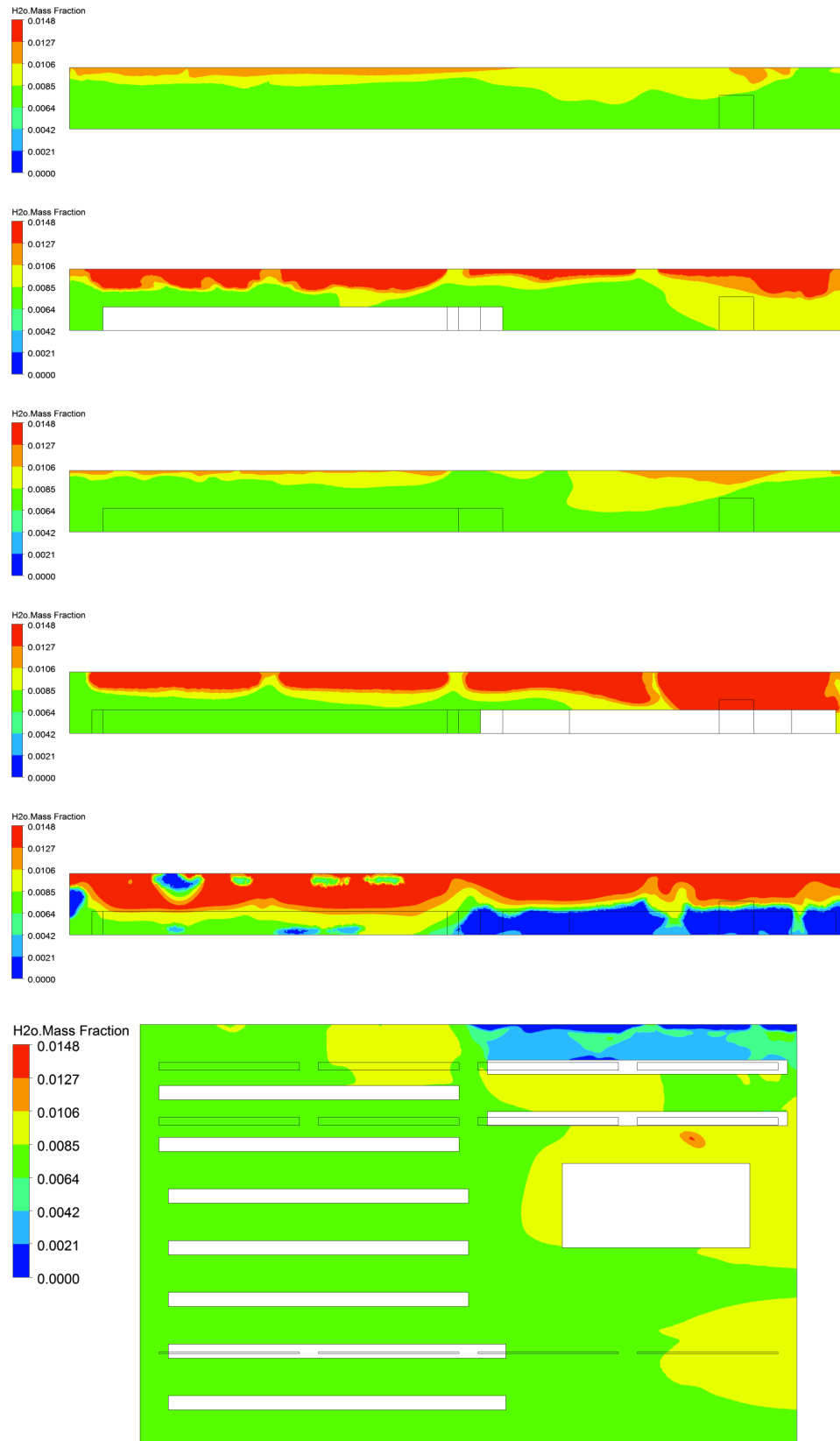


Figure I.6: Water vapor mass fraction contours on the vertical planes P₁, P₂, P₃, P₄, and P₅ and on the horizontal plane P₆ from top to bottom, respectively.