

Design Integration of Novel Porous Ceramic Evaporative Cooling Systems

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Abstract

The rise in demand of air conditioning is unsustainable. Concerns over the subsequent greenhouse emissions and global climate change led to a recent rise of interest by professionals in the building sector in the application of ancient cooling techniques in contemporary architecture to reduce energy demand and deliver healthy and comfortable environments. Despite the successful progress, further research is still needed to make passive cooling systems appealing to a wider audience and to address the limitations found in previous work.

This research responded to this context by focusing on the design and manufacture of a novel cooling system that combined the principles of passive cooling with the inherited use of ceramics in architecture to create an aesthetic and optimised wetted Porous Ceramic Cooling (PCC) screen that improves comfort conditions in indoor, transitional and outdoor spaces. The optimisation focused in the geometrical characteristics of the ceramic element and in the water supply method. This process was informed by a continuous performance monitoring that evaluated cooling and hydraulic capacity of several prototypes under a wide range of ambient conditions. In addition, new steady state and computational fluid dynamic (CFD) models were developed to assist in the sizing and performance prediction of PCC systems in early and late design stages.

The resulting system consisted of tubular-shaped samples arranged over a vertical plane. Water was supplied using a top-drip irrigation system and engraved water channels uniformly distributed the fluid over the ceramic surface to induce evaporation when exposed to an airstream. Experimental and computational studies confirmed the significant impact of geometry on the evaporation rate, highlighting the proportional and inversely proportional relationship of the ceramic effective surface area (ESA) and drag coefficient (Cd) respectively. Air temperature reductions up to 2°C were observed under the warmer and dryer hours of the monitored period (DBT-WBT greater than 12°C) and a competitive average water consumption of 2-4 l/m²·day was obtained. Furthermore, the developed numerical models demonstrated their effectiveness in assisting the design process of PCC systems, allowing estimate airstream velocities, achieved temperature reduction, humidity increase and water consumption prior to manufacture. It is hoped that this research will contribute to a wider application of passive evaporative cooling techniques in architecture.

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Acronyms & Symbols

a	Absorptance	
A	Area	m^2
A _{duct}	Area of duct	m^2
AH	Absolute humidity	g/kg
AST	Apparent solar time	h
BR	Blockage ratio	
β	Solar altitude	<i>degrees</i>
C	Convection loss/gain	W
C _d	Discharge coefficient	
C _{di}	Wind pressure coefficient	
C _e	Latent heat of vaporization	kJ/kgK
C _p	Specific heat of air	kJ/kgK
C _{pi}	Discharge coefficient	
d	Water expansion	m
D	Hydraulic diameter	m
	Mass diffusivity	kg/s
D _v	Diffusion coefficient for water	m^2/s
DBT	Air temperature	$^{\circ}C$
δ	Cell centre-to-wall distance	mm
	Solar declination	<i>degrees</i>
e	Vapour pressure	Pa
e _s	Saturated vapour pressure	Pa
E	Evaporative heat loss	W
	Solar irradiance	W/m^2
E _b	Beam irradiance	W/m^2
E _d	Diffuse irradiance	W/m^2
E _r	Reflected irradiance	W/m^2
ESA	Effective surface area	m^2
ε	Black globe emissivity	
Σ	Surface inclination	<i>degrees</i>
F	Force	N
F _{AHdx}	Absolute humidity correction factor	
F _d	Drag force	N
F _{DBTdx}	Air temperature correction factor	
F _{p-N}	Angle factor	
g	Gravitational force	m^2/s
GT	Goble temperature	$^{\circ}C$
h	Height	m
H	Hour angle	<i>degrees</i>
J	Diffusion flux	kg/m^2s
K _{eff}	Effective conductivity	W/mK
L	Length	m
	Latitude	<i>degrees</i>
M	Metabolic rate	W

M_a	Molecular mass of air	G
M_{evap}	Rate of evaporation	kg/s
M_v	Molecular mass of water vapour	g
MRT	Mean radiant temperature	$^{\circ}C$
n	Number of people	
NGHS	Nottingham girls high school	
OT	Operative temperature	$^{\circ}C$
θ	Angle of incidence	<i>degrees</i>
Φ	Solar azimuth	<i>degrees</i>
ρ	Density	kg/m^3
P	Prototype	
	Static pressure	Pa
P_0	Hydrostatic pressure at ground level	Pa
P_a	Air pressure	Pa
P_d	Dynamic wind pressure	Pa
P_{sat}	Saturated vapour pressure	Pa
P_v	Water Vapour pressure	Pa
P_x	Wind pressure in aperture	Pa
PCC	Porous Ceramic Cooling	
q	Airflow rate	m^3/s
R	Radiation loss/gain	W
	Molar gas constant	$J/molK$
	Species production rate	Kg/s
	Ground reflectivity	
QL	Latent cooling	kW
QS	Sensible cooling	kW
QT	Heat gains	W
S	Vector direction	
	Exposed surface area	m^2
	Body heat loss/gain	W
	Mass creation rate	Kg/s
	Shading proportion	%
S_m	Mass addition	Kg
SG	Solar gains	W
SG_c	Conductive solar gains	W
SG_d	Direct solar gains	W
ST	Surface temperature	$^{\circ}C$
t	Time	s
T	Temperature	K
	Glazing solar transmittance	
T_{conf}	Neutral temperature	$^{\circ}C$
T_{in}	Incoming air temperature	$^{\circ}C$
T_{out}	Outgoing air temperature	$^{\circ}C$
τ	Stress tensor	Pa
U	Air velocity	m/s
	Thermal transmittance	W/m^2K
U_e	External surface transmittance	W/m^2K
UHI	Urban Heat Island	

v	Velocity	m/s
V	Volume	m^3
VLC	Ventilation loss coefficient	W/K
w_d	Dry weight	kg
w_s	Saturated weight	kg
W_A	Water absorptance	$\%$
W_C	Water consumption	l
WBT	Wet bulb temperature	$^{\circ}C$
WEI	Water exploitation index	
Y	Species mass fraction	
	Normal distance	m
Y_v	Air moisture content	
γ	Surface-solar azimuth angle	<i>degrees</i>
ψ	Surface azimuth	<i>degrees</i>
z	Aperture height	m

Introduction

The use of passive cooling techniques is traditionally associated to hot and arid regions and its origins date back to ancient Egypt (Rossi and Russo, 2017). Natural ventilation, water and porous media were commonly used in everyday life to cool down fluids and environments, and from Egypt subsequently spread to the Islamic territory through the Near and Middle East, to North India, North Africa and Southern Spain (Ford et al., 2010). Through centuries ancient civilizations achieved certain grade of sophistication in the integration of passive cooling strategies by means of trial and error, mastering the use of ventilation towers to catch prevailing winds, the use of porous water jars in conjunction with air drafts to induce evaporation, and the use of water in transitional and outdoor spaces as a source of delight and thermal comfort (Bianca, 2000).

With the industrial revolution and the development of mechanical and electrical alternatives for cooling in the late 19th century and early 20th century, buildings began to be mechanically ventilated and air conditioned. This became the norm for much of the 20th century and a part of 21st-century life. Air conditioning already accounts for about 40% of power use in Mumbai, India, and about 87% of US buildings are now air conditioned (MERC, 2015, EIA, 2012). Even in Britain, air conditioning and refrigeration already account for almost 20% of total electricity use and in Europe, and energy used to cool buildings is likely to increase by 72% over the next 15 years (Henley, 2015, Pardo et al., 2012). The demand for cooling is growing exponentially, and so the concentration of *greenhouse gas (GHG)* emissions associated with the building sector. As a consequence, the global average annual surface temperature observed in the last decade was 0.9°C warmer than the pre-industrial average and is projected to increase by the end of this century in the range of 1 to 4.5 °C (Stocker, 2014). Alternatives to conventional air-conditioning are thus needed urgently, and so the commitment of professionals in the building sector.

Fortunately, a responsive attempt is now in progress, and the last decade has witnessed how passive cooling techniques became increasingly attractive methods to architects for delivering healthy and comfortable environments as an alternative to the 'default' use of mechanical cooling systems. The climatic applicability of convective and evaporative cooling techniques is high, and despite evaporative processes being limited to hot and dry climates (Givoni, 1994), it is still applicable for more than 50% of the countries in Europe and

40% the extension of China, India and the US (Salmeron et al., 2012, Xuan and Ford, 2012, Martínez de Azagra and Del Río, 2015).

Recent built examples and research in this field resulted in the development of new components and systems that integrate in building geometry through towers, stacks, walls, screens or roofs. The use of natural ventilation strategies in conjunction with misting systems, wetted pads or porous ceramics to induce evaporative cooling has spread around the world in modern architecture and demonstrated excellent results in cooling capacity, energy savings, efficiency and air quality (Santamouris and Kolokotsa, 2013). Moreover, their application in outdoor spaces as occurred in the Expo in Seville 1992 pointed up the potential for enhancing the outdoor urban environment and mitigate urban heat island effect (Guerra Macho et al., 1994). Despite the successful progress, there is still a big scope to enhance and extend the number of passive evaporative cooling options by addressing the limitations found in previous work, mostly associated to the need of electric equipment and maintenance, inefficient systems compromising water consumption, and limited design integration options, often presenting poor aesthetic solutions and restricted access to market (Martín-Gómez et al., 2009, Terrados-Cepeda et al., 2015, Ibrahim et al., 2003, Schiano-Phan, 2004, Ford et al., 2010).

At present, the significant benefits of passive cooling methods over mechanical systems are widely appreciated, and environmental design became one of the most demanded design approaches in contemporary architecture (Jankovic, 2012). Despite the growing interest, there is still a perceived knowledge and skills gap among designers and the integration of performance evaluation techniques in the early design process is not broadly considered. The benefits of integrating passive strategies from early stages are evident: delivery of a better space layout coupled with the strategy, improved efficiency and avoid of under/over-performing predictions among others. With limited time on early design stages to perform advanced simulations, designers usually lack confidence in the performance of the proposed cooling strategy, which often leads to banal assumptions on the cooling capacity of water bodies placed in the design. It is thus evident that architects need a wider range of intuitive, graphic and well enough designed tools for a successful integration during the design process, and further research is needed to extend the existing knowledge, develop new systems and provide with accessible design methods.

Aims and Objectives

In response to the current context, the main aims of this research are the design and manufacture of a novel cooling system that rediscovers the design opportunities of porous ceramics, and the development of efficient methods to size and allow advance performance evaluation of convective and evaporative cooling techniques. In the pursuit of these final outcomes, this research identified a series of objectives (sub-goals) that were addressed:

Developing a novel porous ceramic cooling (PCC) system

The developed system aimed at combining the principles of convective and evaporative cooling with the inherited use of porous ceramics in architecture by proposing an aesthetic wetted Porous Ceramic Cooling (PCC) wall to improve comfort conditions in indoor, transitional and outdoor spaces. Porous ceramics were recurrent media to induce evaporation and traditionally used for cooling liquids for beverage, food storage and for reducing ambient temperature in ancient civilizations (Fathy, 1986, Cain et al., 1976, Saraswati, 1979, Rye and Evans, 1976). Over the past decades, the ceramics became again highly appreciated and recent ceramic innovations have rediscovered the two-and-three-dimensional aesthetics of ceramics, bringing innovative design opportunities to create all sort of ceramic systems optimised for evaporative cooling (Bechthold et al., 2015). Taking ceramics as the evaporation media, the process of optimizations addressed most of the limitations found in recent applications of passive evaporative cooling in architecture and focused in the development of efficient ceramic geometries for mass production and the reduction of water consumption. For this purpose, the following stages were developed:

- a) **Design and manufacture of optimised porous ceramic prototypes that maximise cooling capacity.** The process of optimisation focused in the geometrical characteristics of the ceramic element to demonstrate the considerable impact of ceramic design on the resulting cooling performance. It is hoped that the corroboration of this remark will be appealing for designers and will promote their further implication in the design process of alternative PCC systems. The following sub-goals were addressed during this work:

- i. To develop design options that maximise ceramic surface area exposed to the airstream and evaluate its impact on the resulting cooling performance.
 - ii. To develop design options presenting low resistance to flow and demonstrate the important role of aerodynamics in the final performance of the system.
 - iii. To demonstrate the design opportunities offered by advanced fabrication techniques and the degree of performance optimization reachable using additive manufacturing technologies and parametric modelling tools.
 - iv. To demonstrate the potential for market integration by collaborating with the ceramic industry and fabricate and optimised ceramic prototype using traditional manufacturing techniques.
- b) **Design of alternative and efficient water supply methods.** The efficiency of the water supply system was approached through the use of supply options that avoid the need of additional maintenance and operational cost and allow a greater flexibility in the allocation of the system. Further, water stored within the ceramics was avoided in order to allow them to dry over non-operational hours and reduce the risk of Legionella growth. The following sub-goals were addressed during the design of the water supply system:
- i. To evaluate the impact of ceramic porosity and capillary network on the hydraulic capacity of porous ceramics and identify recommended values and fabrication methods to increase water absorptance.
 - ii. To achieve uniform water distribution over the ceramic surface using design techniques and ceramic wall treatments.
- c) **Performance evaluation of PCC systems.** Performance was assessed by determining the temperature reduction and humidity increase achieved downstream the ceramics. A range in performance as function

of ambient conditions and ceramic geometries were determined and compared against alternative passive cooling techniques. The following sub-goals were addressed during this work:

- i. To determine the cooling capacity of the system and evaluate the convective, evaporative and radiant cooling impact in the surrounding environment.
- ii. To determine the water consumption of the system and demonstrate its suitability for water stressed countries.
- iii. To identify application opportunities, area of influence and requirements in the space layout.

Developing efficient methods to size and allow advance performance evaluation of convective and evaporative cooling techniques

Evaporative cooling systems must be understood as the harmonic combination of an airstream, an evaporation media and a water supply method. Each of them should be carefully considered and efforts must be made to ease the integration and performance evaluation of every component by the designer. This part of the research focused in the development of intuitive tools and numerical models to fulfil the perceived knowledge gap in the principles of passive cooling among designers and assist in the design process of PCC systems. For this purpose, the following stages were developed:

- a) **A tool to assist in the design and evaluation of natural ventilation options.** Among other factors, the performance of a passive evaporative cooling system is determined by the airflow rate driven through the evaporation media to the space to be conditioned. When a passively driven airstream is anticipated, the driving forces must be determined and the space layout must be designed accordingly to achieve the required cooling. Despite not being the main focus of this research, design assistance on the application of natural ventilation strategies at early design stages was given through the development of a tool that allows the evaluation of common ventilation options, such as single and cross ventilation, atria and large stacks (chimneys). It is hoped that this tool will contribute to an increase in the number of professionals with expertise in designing for natural ventilation while also inform in the

design process of PCC systems. The following sub-goals were addressed during this the design process of the tool.

- b) **A tool to assist in the design and evaluation of PCC systems.** A multi-steady state calculation tool was developed to advise professionals on the design and performance prediction of PCC systems at early design stages. With this tool, it is expected to provide with an intuitive interface capable to predict temperature reduction, humidity increase and water consumption of PCC systems thanks to an extended results database previously generated from CFD studies. With this tool it is hoped to contribute to increase the number of resources available to assist in the integration of evaporative cooling systems in architecture. The following sub-goals were addressed during this the design process of the tool:

- i. To allow performance evaluation for most frequent range of environmental conditions in hot-dry climates and multiple design options.
- ii. To validate the developed tool against laboratory work and confirm the reliability of the model.

- c) **A numerical model to allow advanced performance evaluation of PCC systems.** A model defined using the commercial CFD software Ansys Fluent® was expected to allow detailed investigation of the evaporation process. It was also aimed that this model will provide with detailed information regarding the performance of any ceramic design for any context and ambient conditions, bringing the opportunity to evaluate prototypes prior to manufacture. The following sub-goals were addressed during this the design process of the numerical model:

- i. To identify the physics involved in the evaporative cooling process of PCC systems and demonstrate their integration in a CFD model.
- ii. To calibrate the developed numerical model using data obtained from laboratory work and confirm the reliability of the model.

- d) **A proposed design workflow for convective and evaporative cooling systems accessible to all users.** This was expected to

demonstrate the relation between the developed tools and how they inform each other. In addition, a workflow for the integration of these tools in the design process of PCC systems was defined and demonstrated through a series of examples, giving a hint on the multiple PCC system's design options and market opportunity.

Methodology

This thesis is structured according to the two main objectives of the work: the development of a novel porous ceramic cooling (PCC) system and efficient methods to size and allow advance performance evaluation. In first instance, the background of this research, including the fundamentals of convective and evaporative cooling techniques, built precedents and previous work is introduced in Chapters 1 and 2. Laboratory studies and fieldwork to evaluate the performance of PCC systems and inform in the design optimisation process of the ceramic geometry is presented in Chapter 3. Lastly, the development of design methods and tools to assist in the design process and integration of PCC systems in architecture are presented in Chapters 4 to 7.

The opening of this work explored the origins of passive cooling in ancient civilizations. A research on the applications and techniques used for centuries brought up the legacy of passive design. Following up, the climatic applicability of evaporative cooling in different climates was explored by looking at weather databases. Air moisture content thresholds from literature and existing applicability maps were discussed. Moreover, the existing assessments methodologies were revisited in order to generate for the first time passive cooling applicability maps for the US using advanced data processing techniques. This work helped to evaluate the market opportunity of novel evaporative cooling products.

Further, a review on the principles of convective and evaporative cooling identified the physics and parameters involved in the process of evaporation. The findings from this work were used to set the design criteria for the development of a novel PCC system: maximising effective surface area and minimising resistance to airflow. Chapter 1 ends with a description on the range of applications and benefits of using evaporative cooling techniques in indoor and outdoor spaces, focusing on their impact on human thermal comfort and their

potential to mitigate urban heat island effect. This completed the supporting statement for this thesis work.

The second part of the literature review explored the advances and contemporary application of ceramics and passive cooling techniques in architecture. Chapter 2 first evaluated the potential of ceramics as porous media to induce evaporation and explored innovative fabrication methods and integration options in architecture. Second, recent research work and built examples on passive evaporative cooling were analysed in order to determine limitations and opportunities of the developed systems, as well as the scope for future work.

In response to limited use of performance evaluation tools during the design process, the benefits of integrated design methods were identified. The existing convective and evaporative cooling tools available in the market were analysed and classified according to their level of definition. This allowed to relate each tool to a design stage of the project and identify the lack of resources to size evaporative cooling systems. To conclude, the chapter described the design approach of a novel PCC system that integrated the findings and knowledge acquired from these chapters.

After setting the background, the design & performance of PCC systems were evaluated and described in Chapter 3. First, a comparative study interrogated for the first time the technical characteristics and suitability of different ceramic geometries used in existing architectural screens and facades as evaporators. The ceramic porosity as well as the capacity to absorb water through their pores were examined following the Standard ASTM C20-00 (2015). This study determined the typical performance and characteristics of manufactured porous ceramics using traditional methods (typically extrusion and pressing), and confirmed their potential application as evaporators. This reinforced the design approach adopted in this research for the optimization of PCC systems, discarding the need of exploring ceramic chemical structures, clay composition and customised fabrication processes. Subsequently, two optimised ceramic prototypes were designed taking into consideration the physics involved in the process of evaporation. One of the prototypes was manufactured and its hydraulic characteristics was evaluated against the existing samples to first explore water supply options and water transport over the ceramic surface, and in second instance to determine the minimum water flow rate to be supplied and the time needed for saturation. In addition to the study on the hydraulic capacity

of the developed prototype, the cooling performance was also evaluated by placing the sample in an environmental chamber and exposing it to several environmental conditions. The testing procedure was designed by the author according to the objectives of this research, and a series of parametric studies were performed to evaluate both the impact of climatic conditions and the impact of the ceramic geometry and arrangement on the cooling capacity. Air temperature and surface temperature reduction, as well as the humidity increase achieved were monitored using a series of sensors distributed according to ASTM D3464-96 (2014) upstream and downstream the ceramics. These three parameters were mainly used to describe cooling performance and determine water consumption.

Moreover, the need to also interrogate the dynamic response of PCC systems during a longer period and exposed to a changing environment led to the construction of an experimental mock-up in Granada, Spain. The monitoring was conducted during the hot and dry season and the testing method used in the laboratory was adapted to the new context. A wider amplitude found in the performance results concluded the investigation.

The impossibility to manufacture and test one of the designed prototypes and the growing interest among architects in integrating performance evaluation techniques in the design process led to the second main objective of this research: the development of new resources and tools to size and predict performance of PCC systems prior to manufacture.

Chapter 4 describes the development of a new numerical model using the commercial CFD software Ansys Fluent to simulate the physics of evaporation on wetted ceramic surfaces. This advanced model allowed to extend the studies on cooling capacity of the tested samples to a wider range of environments and interrogate in more detail different aspects of the evaporation process. An initial research on previous work and existing numerical CFD models informed the design of a novel and specific model for PCC systems. The new model was validated against monitored data from the laboratory studies and was used to evaluate the performance of the developed prototypes in order to reaffirm the direct impact of geometry on the cooling capacity. A series of parametric studies exposing the ceramics to different boundary and environmental conditions interrogated the impact of time, airstream velocity, air moisture content and geometry on the achieved cooling and allowed a better understanding of the physics involved during the evaporation process.

This validated model also brought the opportunity to develop a multi-steady-state tool to assist in the design process of PCC systems. Due to the complexity of the physics involved in the evaporation process, performance prediction of evaporative systems can only be conducted using CFD studies. In order to avoid advance simulations at early design stages, the development of a simple PCC tool was approached by creating a result database that derives from an extensive parametric CFD study. This database constituted the core of a tool, which, through an intuitive interface, interpolated cooling performance for any given environment and system design. Chapter 5 introduces the parametric database and PCC Tool, describing its main functionalities, including the user interface, the tool outcomes, and the calculation methods used. With PCC Tool the user would be able to estimate, for instance, temperature reduction, humidity increase and water consumption of any customised PCC system within the range of design options allowed. The tool outcomes were compared against laboratory and fieldwork data, completing the validation process.

In parallel to the development of PPC Tool, a steady-state tool to assist in the design and evaluation of natural ventilation design options was also developed. Despite not being the main focus of this work, the redevelopment of Optivent complements the main outcomes of this research by assisting in the selection of the most suitable ventilation strategy that will drive air through the porous ceramic screen. The renovation of saturated air surrounding the ceramic surface is essential to maintain the evaporation process and, if a passively driven airstream is anticipated, the space layout must be carefully considered. Chapter 6 introduces the range of generic ventilation options used in architecture and describes the redevelopment of Optivent tool, originally conceived as a Excel spreadsheet, into a web-based interface for natural ventilation feasibility studies and PCC systems' design assistance. The tool workflow and its new functionalities, including a wider range of ventilation options available for evaluation, multizone models, updated calculation methods in line with CIBSE AM10 (2005) and thermal comfort prediction are described in this chapter.

To conclude, and in order to highlight the benefits of the developed tools, a design workflow for convective and evaporative cooling systems was introduced in Chapter 7. Through a proposed example, it was demonstrated the relation between the developed tools and how they informed each other. In addition, the proposed workflow was applied to additional scenarios, for which a climate analysis, an evaluation of the ventilation strategy, and a performance evaluation of the selected PCC system were conducted. This helped to identify, for instance,

how PCC performance can range according to the climate conditions, or how the ceramic design can impact on system's water consumption.

A final reflection on the main findings of this work, both the design and performance evaluation of PCC systems, and the advantages and limitations of the developed tools are discussed in Chapter 8. It is also pointed out recommended ways in which this research could be pushed forward, aiming to promote the practice of evaporative cooling in the field of architectural design.

1 Passive Cooling Fundamentals

The range of passive solutions applicable to architecture are vast, and so the number of built examples and lessons that can be obtained from the past (Weber and Yannas, 2014). Most of these strategies achieved certain grade of sophistication by means of trial and error during centuries and recent research has contributed to a better and detail understanding of the principles.

This chapter reviews the origins of passive cooling and the principles involved in the process of convection and evaporation in order to identify primary potential approaches for optimisation. Next, the applicability of these systems throughout the world will be determined in order to evaluate the capacity of these techniques to become alternative solutions on air conditioning systems. The last part of this chapter addresses the main benefits of passive cooling strategies in the current word and analyses in detail the impact on human thermal comfort and to mitigate urban heat island effect.

1.1 The Traditional Use of Passive Cooling Techniques in Architecture

The use of passive cooling techniques is traditionally associated to hot regions, and its origins date back to ancient Egypt, more specifically to reign of Egyptian pharaoh Amenhotep III (1387–1348 B.C.), in which a fresco in the tomb of Nebamun depicts two opposing rectangular triangles on top of a house (Figure 1.1). These are two ventilation outlets designed as wind catching devices to introduce and exhaust warm air from inside the building (Rossi and Russo, 2017). These elements were known as *malqaf* and they were optimal to catch the two prevailing winds from the sea and the mountains. Moreover, this technique was also useful in reducing sand and dust as the wind captured at higher levels contains less solid particles than wind at lower levels. The technique subsequently spread to Near and Middle East as part of the Islamic tradition and the original *malqaf* shape was transformed into a square conduit to allow wind-induced air movement from any direction. This omni-directional shaft is called *badgir* and it has been a recurrent solution in the architecture of Iran and the countries of the Gulf to naturally ventilate houses while avoiding screening of blocks (Fathy, 1986).

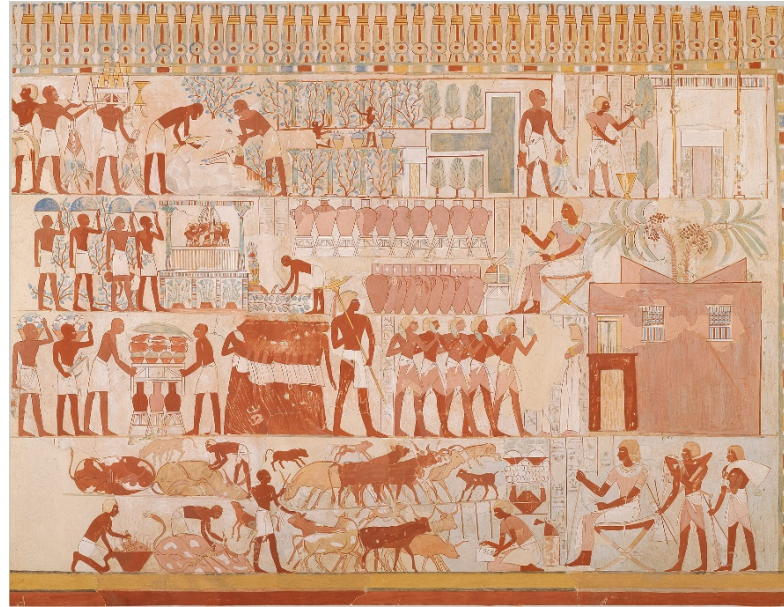


Figure 1.1 Wall painting from the tomb of Nebamun. Egyptian Dynasty 18 (about 1475 B.C.),
(Rogers, 1930, p. 1)

The earliest known reference of evaporative cooling also dates back to ancient Egypt where frescoes from about 2500 B.C. show slaves fanning porous jars of water to cool rooms for royalty. The use of the porous jars *maziaras* also settled in rural areas of Upper Egypt as elements to store the day's supply of water for drinking which was usually collected from wells or from the Nile River and its canals (Cain et al., 1976). The use of water and porous media in everyday life and in architecture to cool down liquids and the immediate surroundings were commonly used in ancient civilizations, and from Egypt also extended to the Islamic territory and North India. There is archaeological evidence on the use of pottery since ancient Mesopotamia more than 4000 years ago, in which different shapes of vessels were designated to different functions like those to drink from, eat off, prepare food in, store in, etc (Potts, 1997). This practice extended for centuries along the Mediterranean and India and it is considered the origins of the traditional vessels *matki* and *surahi* from India and Pakistan, used as water storage coolers (Saraswati, 1979), (Rye and Evans, 1976); and the ceramic water pitchers still used in rural areas of Spain and Southern France, commonly known as *botijo* or *gargoulette* respectively. Despite the use of these techniques are sharply falling today, recent studies have proven their high applicability in many regions of the world. Paradoxically, this technique still remains unknown for some of these regions (Martínez de Azagra and Del Río, 2015).



Figure 1.2 Ceramic water pitcher botijo (After Ford et al., 2010, p. 40)

Moreover, porous water jars have been eventually used in conjunction with wind catching towers and air drafts to induce evaporation and achieve lower air temperatures in rooms and houses in developing countries. Figure 1.3 is a common example of this practice, where wind catchers are oriented to the prevailing wind direction to induce air movement in the interior. When in contact with the ceramic jars, water on the ceramic surface evaporates, delivering a cool breeze of air to the internal areas at lower levels.

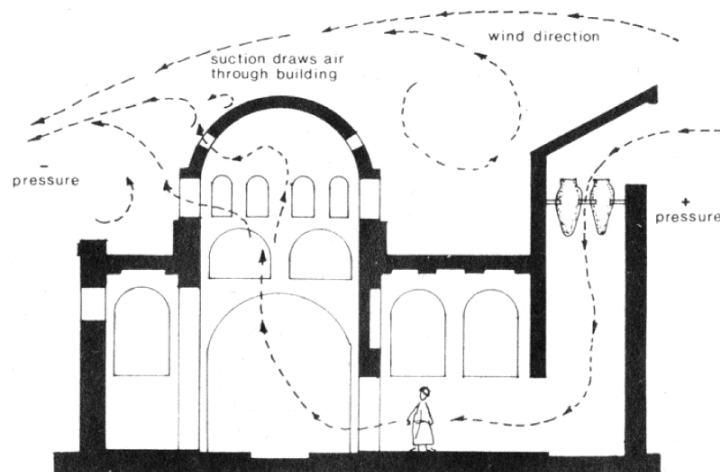


Figure 1.3 Wind catcher with cooling jars (Cain et al., 1976, p. 63)

Alternatively, the air movement can be achieved from thermal (buoyancy) forces. The air temperature differential between two adjacent or connected spaces induces air movement as the cooler (and denser) air from one space replaces the warmer (and lighter) air from the other. In addition to the physiological cooling effect of air movement, porous ceramic jars placed in the

transition contribute to further lower air temperatures through evaporation. This transition was often a carefully designed window opening as found in Muscat, Oman (Figure 1.4), or a *cool passage* in between two connected courtyards commonly found in Middle Eastern and Mediterranean houses (Figure 1.5). The combination of an air movement strategy with ceramic cooling jars became one of the most recurrent strategies to mitigate heat stress by Islamic civilization, who mastered this technique through centuries.

Evidence on the effectiveness of yard-to-yard airflows in the promotion of convective cooling through intermediate spaces is discussed by Raha Ernest using Casa de Pilatos in Seville, Spain as a laboratory, a 16th-century summer house built under the influence of Moorish tradition. The temperature differential between the two courtyards creates a constant air movement during the hot season between 0.2 and 1.5m/s in the transitional spaces (Ernest and Ford, 2012). The controlled aperture size, the scale and proportion of the spaces and the careful integration of different techniques such as shading, evapotranspiration or thermal mass led to the successfulness of this summer house building.

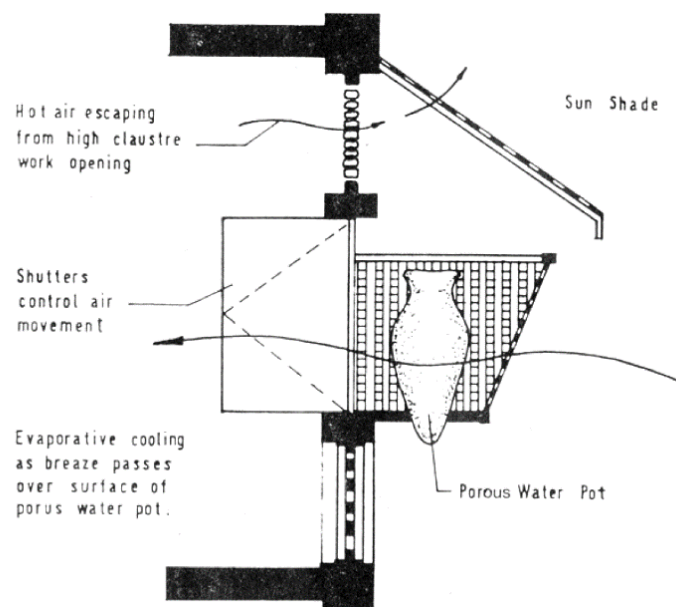


Figure 1.4 Porous ceramic cooling window in Muscat, Oman (Cain et al., 1976, p. 64)

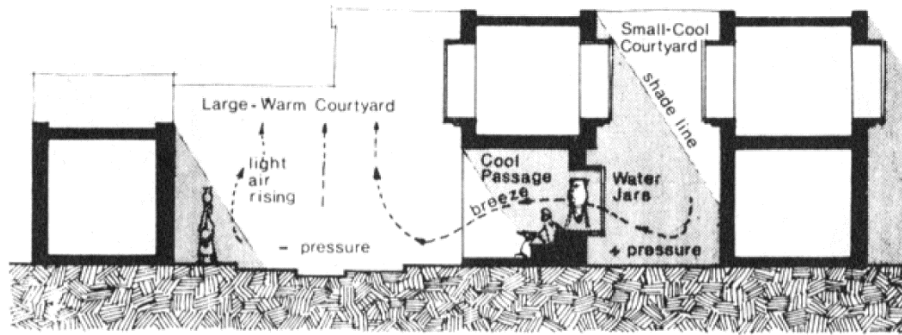


Figure 1.5 Two-courtyard Mediterranean house (Cain et al., 1976, p. 63)

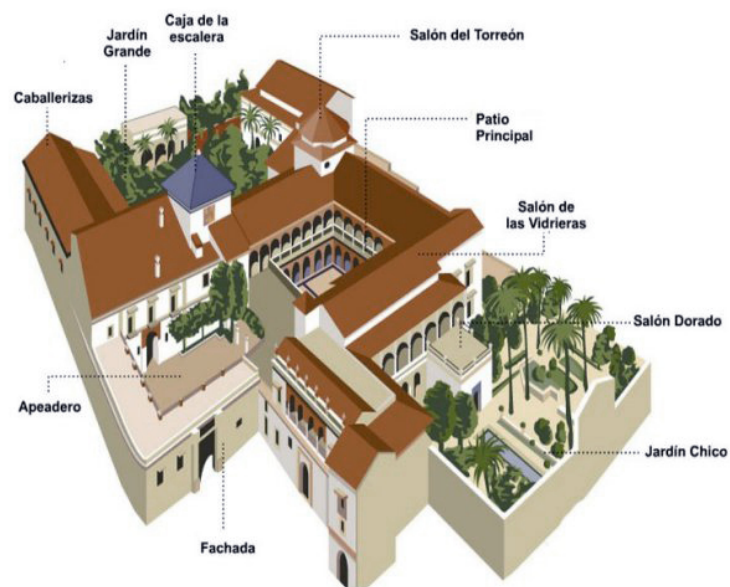


Figure 1.6 Casa Pilatos 3d view (top) and aperture on transitional passage (bottom), (Ernest and Ford, 2012, p. 243)

The presence of water has been consistent in Islam architecture and is considered as source of life and element of ritual purification. The concept of *paradise garden* was often present in courtyards with two crossing water channels that symbolize the four rivers of paradise, and many types of fountains, channels, ramps and pools induced the flow of water that served as a source of sensual pleasure, aesthetic delight and thermal comfort (Bianca, 2000). A common feature in the courtyard of medieval palaces by the twelfth century is the *salsabil*, which consist of a water spout in the back wall of an *iwan*; an inclined, carved marble slab called *shadirwan* on which water flows; a long and thin channel running through the middle of the *iwan*; and a pool in the middle of the courtyard in which the *salsabil* water flows (Tabbaa, 1986). The *shadirwan* allows the rippling of water on its carved surfaces often interpreted as a veil of water falling off a roof, and exploits evaporative cooling to delight the eye and to provide thermal relief. This technique dispersed over the thirteenth century from India to Turkey, and from Syria to Spain.

Salsabil was often combined with a wind catcher *malqaf* and porous media in its interior to maximise evaporation. Panels of wet charcoal displaced as in Figure 1.7 increased evaporation while filtered dust. Air is then directed to the *salsabil* to further reduce air temperature and increase air humidity (Fathy, 1986).

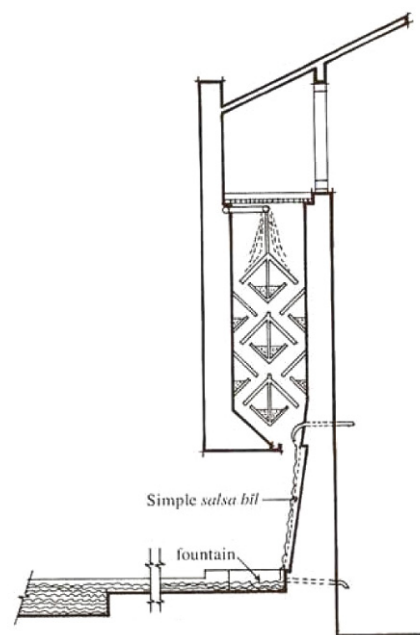


Figure 1.7 Salsabil in Red Fort, Delhi, India (left) and malqaf with wetted baffles and salsabil (right), (Fathy, 1986, p. 125)

With the arrival of Muslims in India, local and immigrant design and cooling techniques merged together to provide with new ways to cope with extreme weather conditions. During the Mughal period, most buildings were surrounded by a tailored landscape to improve the surrounding microclimate (Ali, 2013), and convective and evaporative cooling techniques couple with other passive strategies to create sophisticated systems providing comfortable and delightful spaces. The stepwells, constructed as a source of water, were used as places for meeting in the hot season (Gupta, 1984). The structure consisted of a series of broad flights of steps and covered platforms, leading down to the water. The Adalaj stepwell (Figure 1.8), dated on 1499, falls in five flights to a depth of 20 m below ground, offering a progressively cooling environment. The high thermal mass coupled to the earth, and combined with the effect of evaporative cooling achieved on the surface of the water and the porous stone walls lead to a very stable air temperature by about 8 degrees below the annual mean (Ford and Hewitt, 1996a).

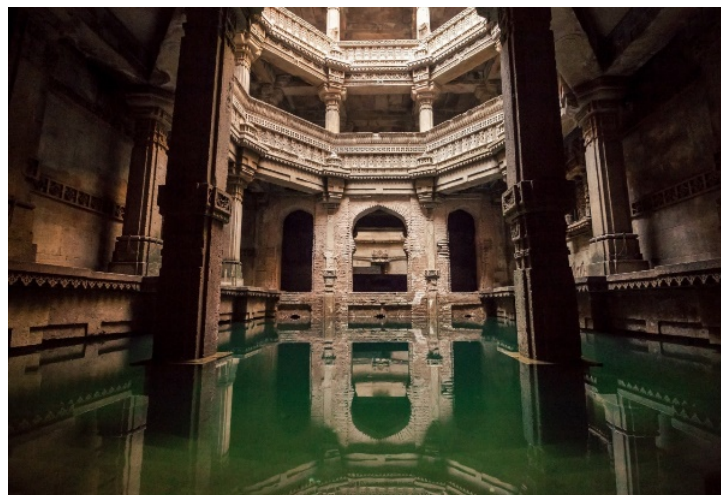
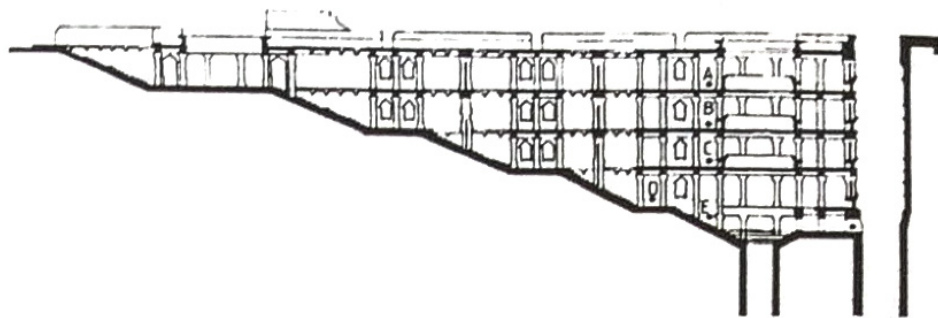


Figure 1.8 Adalaj stepwell section (Ford and Hewitt, 1996a, p. 62) and interior view, Ahmedabad, Gujarat (Krishnan, 2015, p. 1)

Another cooling method traditionally used in Indian culture and still applied due to its high availability is the use of wetted *khus* mats. The mat is usually wetted by a drip irrigation system and placed in transitional spaces exposed to an airstream that induces evaporation, reducing the surrounding air temperatures (Cain et al., 1976). It evokes the combination of air movement and porous ceramic jars in middle Eastern Mediterranean houses.

The traditional porous ceramic jars used in ancient Egypt to store water were reinvented in the 20th century in a *pot-in-pot* storage pot placed inside a bigger pot that holds water. The inner pot stores food that is kept cool due to the evaporation of water in the outer ceramic surface. The original concept has suffered multiple adaptations to different contexts like the *Janata cooler* developed by the Food & Nutrition Board of India (Roy and Khardi, 1985), where the storage pot is placed in an earthenware bowl containing water and covered with a damp cloth, or the Rolex awarded *Zeer pot* designed by Mohammed Bah Abba in Nigeria, where the smaller pot is placed inside the larger pot that holds wetted sand (Oluwasola, 2011). Experimental studies have proven that *Zeer pots* can preserve an inner temperature of 75% wet bulb temperature depression (Dutt R, 2016) and subsequently extend the shelf-life of vegetables from 5 to 20 days (Longmone, 2003). This adapted ancient technique had a major impact in developing countries, improving life quality of families with limited access to standard refrigerators.

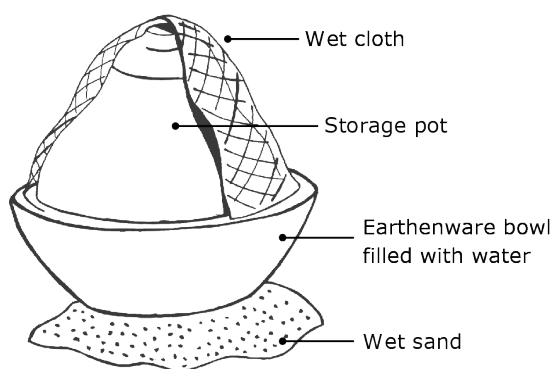


Figure 1.9 *Janata cooler* (left) and *Zeer pot* (right), (Noble, 2003, pp. 1-2)

1.2 Climatic Applicability

The applicability of convective and evaporative cooling methods has been a recurring subject in the architectural engineering science. In the last decades, empirical work derived to a range of ideal environmental conditions for the application of passive evaporative cooling (PEC) methods. During the evaporation process, sensible heat is converted into latent heat, thus reducing dry-bulb air temperature (DBT). Studies by Givoni (1994) confirmed that air temperature can be reduced by about 70% to 80% of the wet bulb temperature depression (DBT-WBT) depending on the type of the system, and, considering a typical indoor air temperature by about 3 to 5°C higher than ambient due to internal heat gains, the author concluded that PEC can be applied only in regions where WBT remains below 22°C in order to satisfy indoor comfort conditions. Likewise, Santamouris (2007a) discouraged the applicability to regions where wet bulb temperatures are too high for human thermal comfort, and now PEC methods are commonly associated to hot and dry climates. A 22°C WBT may be a reasonable numeric reference when the objective is to condition indoor spaces but, when the objective is to cool people in semi outdoor/transitional spaces as traditionally practiced, this threshold could be increased up to 24-25°C, assuming an upper limit of 26°C for environments with elevated high humidity and air velocity (Givoni, 1994). Still, this threshold is indicative and may be adapted to local thermal comfort criteria and ambient conditions.

In order to address the applicability passive cooling techniques, the designer must first evaluate the need for the strategy (demand). This need is normally associated to the number of hot hours (or days) during the application period so that, higher number of hot hours lead to a high demand of passive cooling techniques. Second, it is important to evaluate the opportunity for cooling, as this will help to determine the most suitable cooling method. For convective cooling or natural ventilation, the temperature differential between the incoming air temperature and a theoretical indoor air temperature will be indicative of the opportunity for the system. For evaporative cooling methods, the wet bulb temperature depression will do so. In both cases, a higher temperature drop will lead to higher system opportunity. The final cooling applicability should thus be informed by both parameters equally considered (applicability = demand + opportunity). Further to the above indices, additional factors may complement and contribute for a better understanding of the feasibility of the system. Water availability for evaporative cooling techniques, or daily temperature fluctuations

for night time ventilation strategies can add an extra layer of information and additional criteria to evaluate climate applicability.

Access to large databases and worldwide hourly climate data eased considerably the applicability assessment process. Most weather data is available from public sources [EnergyPlus (2017b)]; Climate.OneBuilding.Org (2017)] and are distributed by the World Meteorological Organization (NOAA, 2017). Weather data can be processed and plotted on charts and maps. Psychrometric charts, for instance, can promote rapid interpretation to support strategic decision making. Such plots can help to define both the need for cooling and the opportunity for different passive cooling strategies in a particular climate. Applicability maps, instead, allow the evaluation of passive cooling techniques at a larger geographical scale without the need of accessing multiple weather data.

In the past decade, several methods and applicability maps for evaporative cooling strategies have been published. The maps allowed to quickly evaluate the viability of a system for any region at early design stages. The downdraught cooling applicability maps for Europe (Figure 1.10) developed by Salmeron et al. (2012) followed the methodology described in Ford et al. (2010) and evaluated the potential for reducing air temperature (defined as DBT-WBT) and to reduce cooling demand in building (defined as 25-WBT). Furthermore, these parameters (opportunity) were combined with the average cooling degree-days based on 25°C (demand) for a predefined warm season from June to August in order to determine the viability of the strategy. The same rationale was applied by Xuan and Ford (2012) to develop downdraught cooling applicability maps for China (Figure 1.11) with a minor change in the design indoor air temperature from 25°C to 26°C and a new analysis period set from June to September.

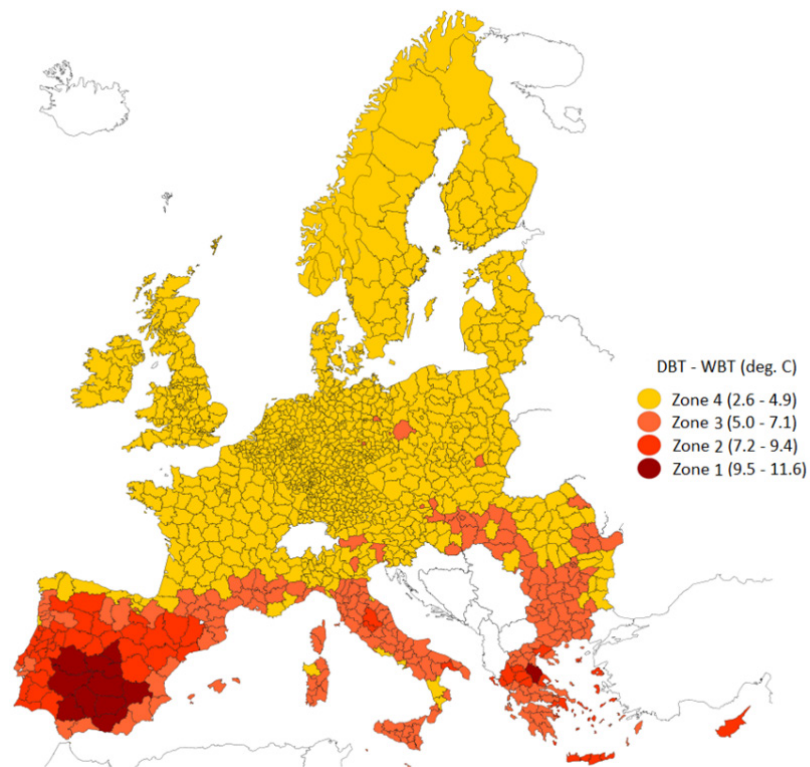


Figure 1.10 Predicted evaporative cooling systems' efficiency (DBT-WBT) in Europe (Salmeron et al., 2012, p. 261)

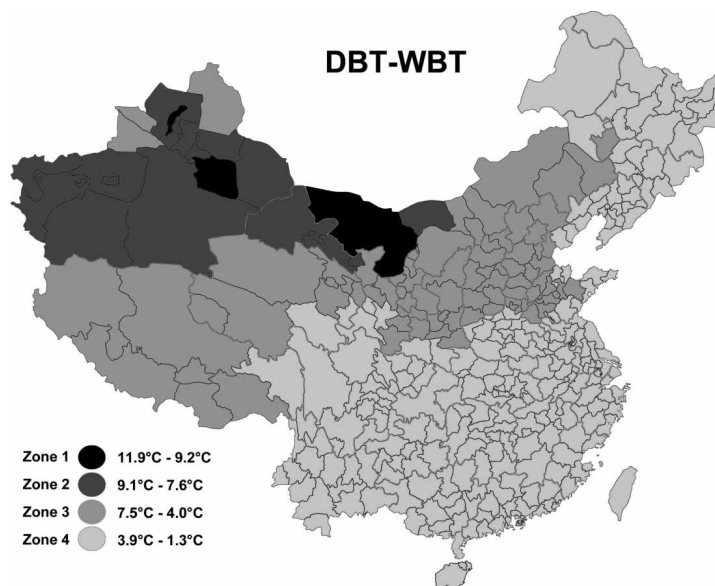


Figure 1.11 Predicted evaporative cooling systems' efficiency (DBT-WBT) in China (Xuan and Ford, 2012, p. 276)

The assessment and mapping methodology previously used has been revisited and expanded in this work to allow a full feasibility assessment of convective and evaporative cooling techniques in United States (US). The entire process was taken to a higher degree of resolution, now dealing with hourly data instead of daily average data by means of big data processing techniques (Vallejo et al., 2018).

The third generation of Typical Meteorological Year climate data (TMY3), which derives from the 1961-1990 and 1991-2005 National Solar Radiation Data Base archives, was obtained for 1020 locations in the US and post-processed to generate the applicability maps. The applicability in counties without climate data is determined using an interpolation methodology (de la Flor et al., 2008) by means of the geographical distance between the closest meteorological stations, latitude, altitude and proximity to the sea.

In addition to the generic methodology introduced, alternative methods have emerged to include other criteria: alternative applicability maps by Martínez de Azagra and Del Río (2015) prioritise the presence of surface deposits of clay and access to an improved drinking-water source to fully evaluate the potential areas for the use of water cooling pitchers botijos.

1.2.1 Cooling Demand in US

Assuming a design indoor temperature of 26°C, the need for cooling was determined by the number of hours (h) when $DBT > 26^{\circ}\text{C}$ for a theoretical warm period from June to September (presenting a maximum number of hours of 2928). The results for each county is mapped in Figure 1.12. The map suggested a higher demand in areas with lower latitudes and altitudes: the Southeast counties from Texas to Florida, Southern California and Arizona. The displayed scale responded to the following criteria: $h < 750$ (low), $750 < h < 1500$ (medium-low), $1500 < h < 2250$ (medium-high) and $h > 2250$ (high).

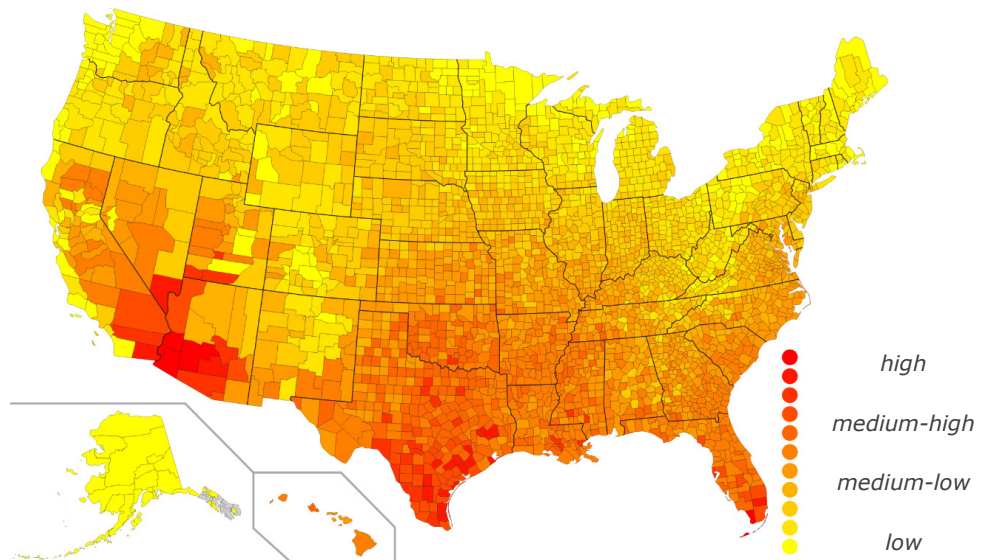


Figure 1.12 Passive evaporative cooling need. Determined from $DBT > 26^{\circ}\text{C}$ in US (Vallejo et al., 2018, p. 3)

1.2.2 Evaporative Cooling Opportunity in US

The opportunity or efficiency of an evaporative cooling method derives from the wet bulb temperature depression and responds to the question: how dry is the climate? This question was addressed in three different approaches to adapt to different contexts.

- The first approach determined $DBT-WBT$ for each hour of the analysis period and the average values are mapped in Figure 1.13. The results obtained broadly represented the humidity of the climate with no differentiation between day and night. The map also suggested a prevailing range of $DBT-WBT$ between 3°C and 6°C , with dryer areas referring to the Western counties, and highlighting an evident relation with the altitude above the sea level. The displayed scale responded to the following criteria: $\Delta T < 3$ (low), $3 < \Delta T < 6$ (medium-low), $6 < \Delta T < 9$ (medium-high) and $\Delta T > 9$ (high).
- The second approach determined $DBT-WBT$ when $DBT > 26^{\circ}\text{C}$. This index represented the maximum opportunity by mapping the wet bulb depression at the warmer hours of the day. It is indeed addressing PEC opportunity in the outdoor environment when most needed. The results mapped in Figure 1.14 extended the high opportunity also to Eastern counties and the prevailing range of $DBT-WBT$ at the warmer hours now

increased from 4°C to 8°C. The displayed scale responded to the following criteria: $\Delta T < 4$ (low), $4 < \Delta T < 8$ (medium-low), $8 < \Delta T < 12$ (medium-high) and $\Delta T > 12$ (high).

- The third approach considered the previously used design indoor temperature of 26°C to determine the wet bulb depression. As with the maps created for Europe and China, 26°C-WBT indicated the opportunity to reduce cooling demand in indoor spaces with a PEC system that theoretically could supply air at wet bulb temperature. The results mapped in Figure 1.15 suggested that PEC opportunity could be extended even in the colder and more humid regions of North-eastern US when a theoretical indoor temperature is achieved as a result of the internal and solar gains. The displayed scale responded to the following criteria: $\Delta T < 3$ (low), $3 < \Delta T < 6$ (medium-low), $6 < \Delta T < 9$ (medium-high) and $\Delta T > 9$ (high).

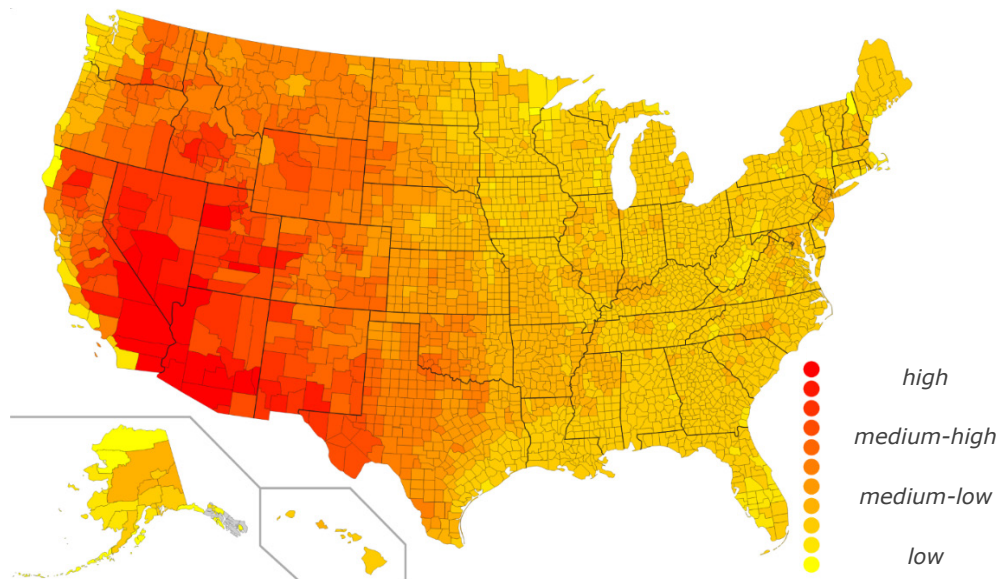


Figure 1.13 Passive evaporative cooling opportunity in US (I). Determined from DBT-WBT (Vallejo et al., 2018, p. 3)

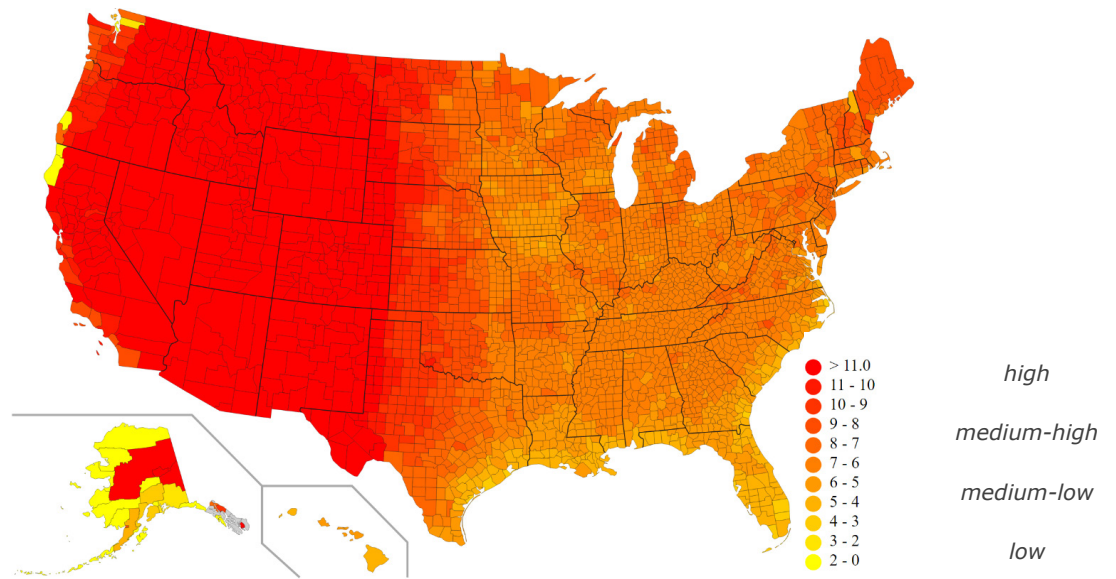


Figure 1.14 Passive evaporative cooling opportunity in US (II). Determined from DBT-WBT when DBT>26°C (Vallejo et al., 2018, p. 4)

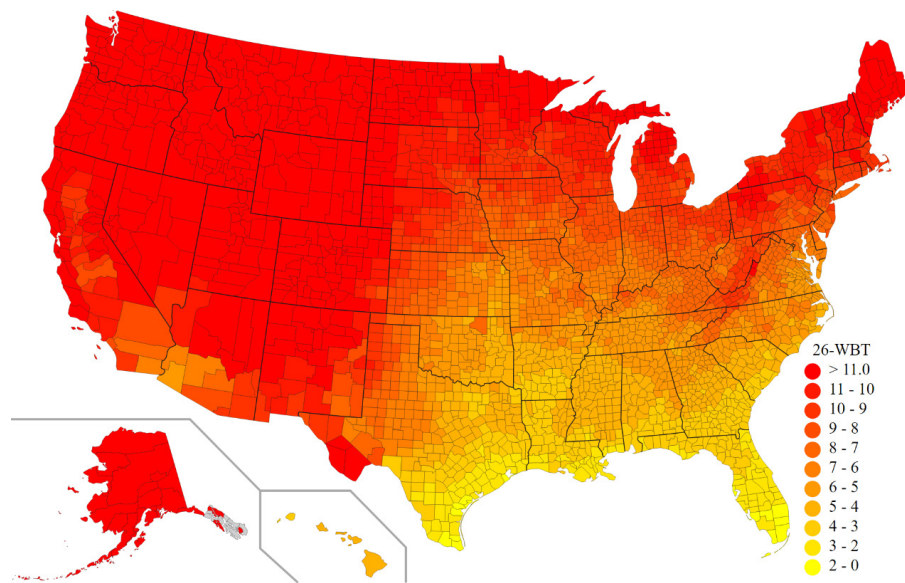


Figure 1.15 Passive evaporative cooling opportunity in US (III). Determined from 26°C-WBT (Vallejo et al., 2018, p. 4)

The above maps provided sufficient information to evaluate separately need and opportunity for passive evaporative cooling systems in early stages. As both indexes are equally important, higher number of warm hours (demand) and higher wet bulb temperature depression (opportunity) yield high applicability. It is also important to look at all of the maps for a better understanding of the viability of the system under different contexts. The maps concluded that 30% of the US counties present optimal climatic environmental conditions for the

integration of passive evaporative cooling systems in architecture. These results are satisfactory and confirm that alternative passive methods to the 'default' use of mechanical systems are very valid and present a huge potential for expansion to overcome the recurring increase in greenhouse gas emissions during the last decade (IPCC, 2014).

1.2.3 Water Availability in Europe

In some regions, water supply and consumption may become the biggest challenge of evaporative cooling techniques when applied in hot dry climates. Looking at Europe, annual precipitation averages 500 mm in southern European countries (Figure 1.16) and only a 10 % of this rain occurs during the hot period of the year (Nixon et al., 2003).

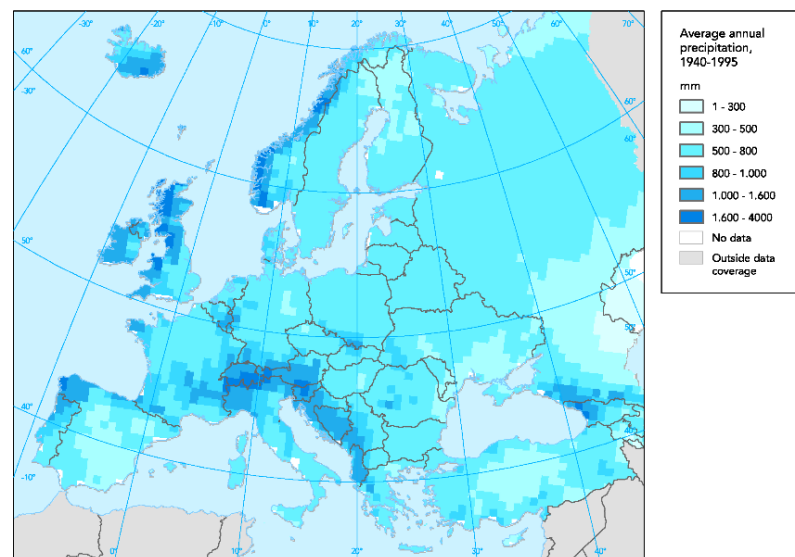


Figure 1.16 Average annual precipitation 1940-1995 (Nixon et al., 2003, p. 83)

These numbers aggravate when looking at the projected change in precipitation in the period 2071-2100 compared to the baseline period 1971-2000 (Figure 1.17). Climate change is affecting precipitation patterns in Europe, and decreasing trends in precipitation have been observed in parts of southern and central Europe. According to this prediction, annual precipitation will be reduced by 20% in 75 years' time in southern European countries, resulting in 400 mm of average annual precipitation and 20 mm of average summer precipitation (Jacob et al., 2014).

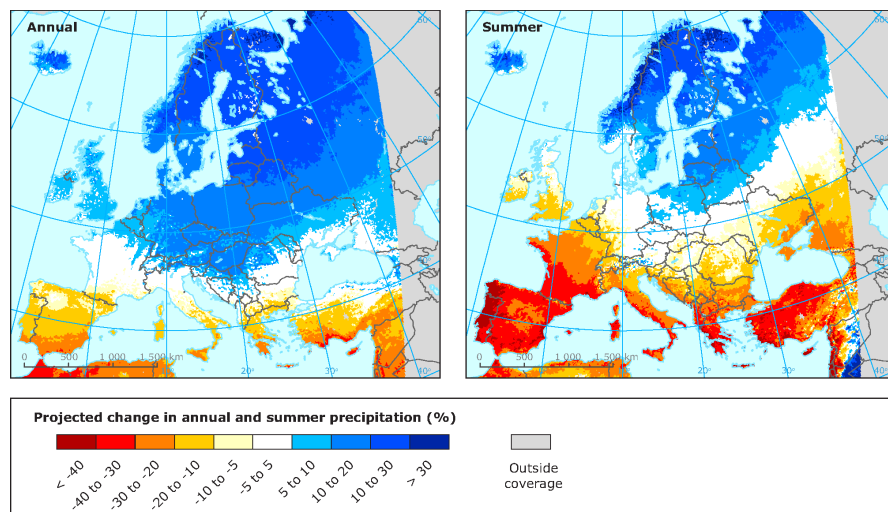


Figure 1.17 Projected change in annual and summer precipitation (%) in the period 2071-2100 compared to the baseline period 1971-2000 for the forcing scenario RCP 8.5 (Jacob et al., 2014, p. 567)

Moreover, studies conducted by Eurostat (EEA, 2009) suggest that there are five European countries (Cyprus, Italy, Malta, Belgium and Spain) which are considered to be water stressed, presenting a Water Exploitation Index (mean annual freshwater demand to freshwater resources ratio) over 20% (Figure 1.18).

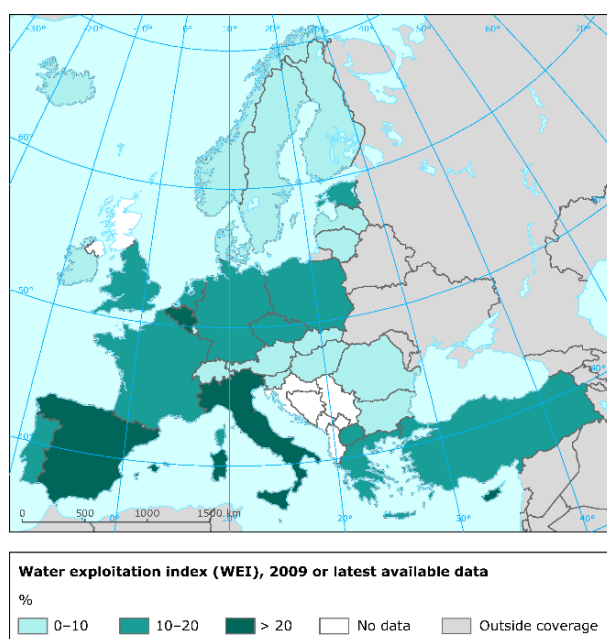


Figure 1.18 Water exploitation index (WEI) in the late for 2009 or latest available data (EEA, 2009, p. 1)

These parameters may restrict the applicability of evaporative cooling systems at different levels (size, operating hours, etc) and thus should be taken into consideration for a full feasibility study. Although there is not any benchmarking for these systems, regions presenting a high water exploitation index (WEI) should aim to minimise water consumption and/or restrict hours of operation.

1.3 Principles of Convective & Evaporative Cooling

Knowledge in the physical and chemical processes happening during the cooling process is essential for the designer to understand the opportunities (and limitations) of a cooling strategy. This section reviews the principles of convective and evaporative cooling, providing a basis for discussing the behaviour of air and water vapor in a given environment.

1.3.1 Convective Cooling

Convective cooling is the heat transfer between one place to another caused by air movement without affecting the physical state of the substances. This physical principle is also known as sensible cooling (or heat). The rate energy needed to reduce (or increase) one degree one kilogram of air is determined by its specific heat C_p , equal to 1.006 kJ/kgK for air temperatures between -5°C and 45°C (Monteith and Unsworth, 2013, p. 8). The sensible cooling (Q_s) is thus the rate of energy needed to reduce the temperature of a volume of air, keeping its moisture content constant. It is thus directly proportional to the air temperature difference between the two substances and the air volume flow rate, and can be calculated using Equation (1-1):

$$Q_s = \rho \cdot C_p \cdot q \cdot \Delta DBT \quad \text{Equation (1-1)}$$

Sensible cooling
ASHRAE (2013, p. 18.13)

where Q_s is the sensible cooling (kW), ρ is the density of the air (kg/m³), q is the air volume flow rate (m³/s) and C_p is the specific heat of air (1.006 kJ/kg°C). For a naturally ventilated space, the increment in dry bulb temperature (ΔDBT) is the air temperature difference between the incoming air temperature (cool air) and the initial (indoor) air temperature (warm air). As in normal conditions ΔDBT is typically low (1°C to 3°C) and determined by the external and internal

conditions, large air change rates are generally required to achieve significant cooling in a building (Figure 1.19).

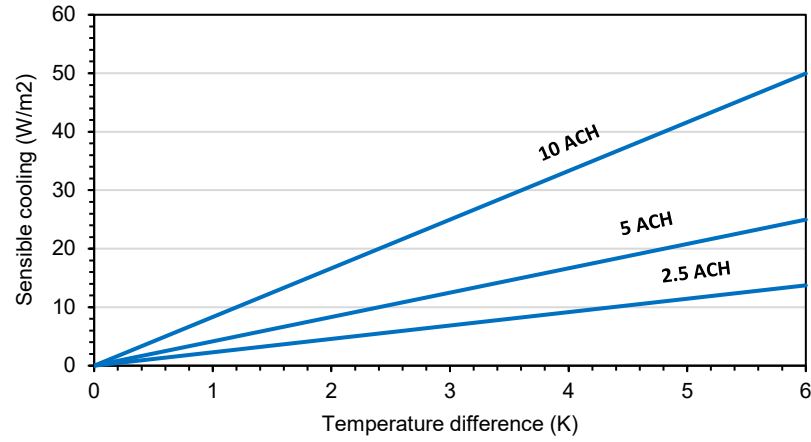


Figure 1.19 Effect of air temperature difference and air change rate on the equivalent sensible cooling (height of space taken as 2.75 m) (CIBSE AM10, 2005, p. 39)

The flow rate q is determined by the aperture area and the air velocity. Air can flow from one place to another driven by wind force or thermal (buoyancy) force. In the previous section it was discussed the traditional use of *malqafs* in the Middle East to ventilate indoor spaces. In this context, those elements benefit from a higher positive pressure and a high air velocity at the higher level of the tower. In general terms, wind creates a positive pressure on windward surfaces, and a negative pressure on leeward surfaces. The wind pressures are usually expressed in the form of a pressure coefficient C_{pi} as defined by Equation (1-2), where P_x is the wind-induced pressure in a point x in the building façade, and P_d is the dynamic wind pressure in the free stream region. Wind pressure on a building element depends on a wide range of parameters, such as wind direction and speed, and surface orientation and exposure (Etheridge, 2012).

$$C_{pi} = \frac{P_x}{P_d} \quad \text{Equation (1-2)}$$

Wind pressure coefficient
Etheridge (2012, p. 141)

From Equation (1-2), the wind pressure in an aperture exposed to the stream is directly proportional to the dynamic wind pressure, and is given by Equation

(1-3), where P_x is the aperture wind pressure (Pa), C_{pi} is the wind pressure coefficient, ρ is the density of the air (kg/m³), and U is the wind speed (m/s).

$$P_x = 0.5 \cdot C_{pi} \cdot \rho \cdot U^2 \quad \text{Equation (1-3)}$$

Aperture wind pressure
CIBSE Guide A (2015, p. 4-19)

Air velocity and direction in the urban context is dependent on the urban density, geometry and building heights among other factors. In most cases, to determine this parameter with high degree of resolution a computational fluid dynamic (CFD) simulation is needed. In general terms, there is an increase of air velocity with height relative to the topography of the location and roughness of the surrounding terrain. The wind profiles can be expressed as a power law or logarithmic function (Santamouris and Asimakopoulos, 1996) and suggest that near ground wind speeds in city areas are considerably lower than in open land areas (Figure 1.20). In high density urban forms, buildings shelter each other and thus the highest natural ventilation rates can be achieved in unobstructed apertures located above the urban canopy layer.

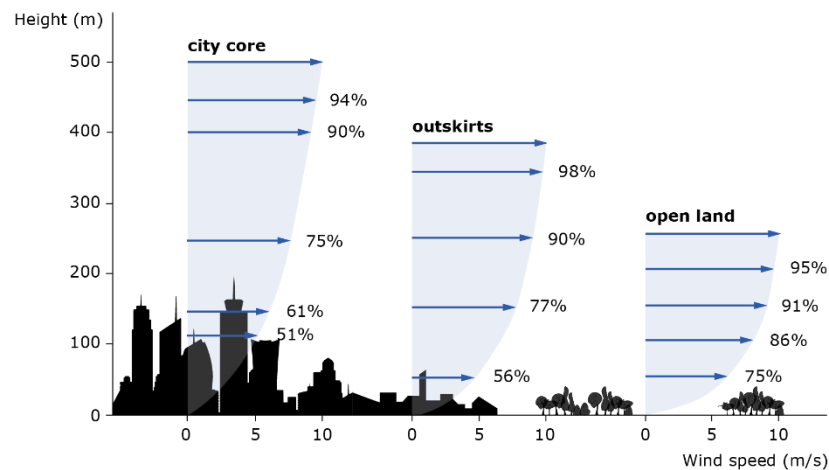


Figure 1.20 Increase of wind speed with height relative to different ground unevenness (After Baumbach, 1996, p.79)

Apart from wind forces, air can be driven in an indoor space by thermal forces. Thermal force alone (also known as buoyancy force) derives from the temperature difference between the room and the supply air temperature and is usually considered when sizing aperture areas for natural ventilation at early design stages. This represents a theoretical *worst-case* scenario. The air is

driven into the space as result of a pressure difference across the opening due to different stack pressure gradients of air temperatures (Figure 1.21).

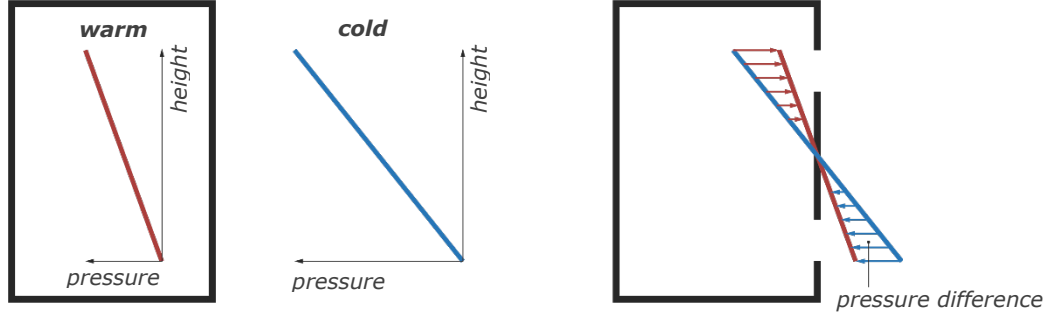


Figure 1.21 Air pressure gradient with height

The vertical pressure variation is directly proportional to the air density and height and it's given by Equation (1-4), where p is the air pressure (Pa), g is the gravitational force (m^2/s), and h is the height (m). This expression indicates that warm air presents lower air pressure than cold air at a certain height (since warm air is less dense than cold air).

$$\Delta p = \rho \cdot g \cdot h \quad \text{Equation (1-4)}$$

Vertical pressure variation
ASHRAE (2013, p. 16.7)

From the above expression, it can be determined the pressure gradient across an opening as result of the temperature difference between two connected spaces (CIBSE AM10, 2005). This pressure gradient is given by the following expression:

$$\Delta p_i = \Delta p_0 + \Delta \rho_0 g z_i \quad \text{Equation (1-5)}$$

Pressure gradient across an opening
CIBSE AM10 (2005, p. 40)

where Δp_i is the pressure gradient across the opening, Δp_0 is the hydrostatic pressure difference at ground level (Pa), $\Delta \rho_0$ is the density difference at ground

level (kg/m^3), g is the gravitational force per unit mass (m/s^2) and z_i is the height of the opening above ground level (m).

The final airflow rate achieved in a given aperture is then expressed as function of this pressure gradient and the free aperture area:

$$q_i = C_{di} A_i S_i \sqrt{\frac{2|\Delta p_i|}{\rho_0}} \quad \text{Equation (1-6)}$$

*Airflow rate across an opening
CIBSE AM10 (2005, p. 40)*

where q_i is the flow rate through the opening (m^3/s), C_{di} is the discharge coefficient, A_i is the area of the opening (m^2), S_i represents the sign of the pressure difference, Δp_i is the pressure difference (Pa) and ρ_0 is the air density (kg/m^3).

From Equation (1-5) the height difference between inlet and outlet openings has also a fundamental influence on the air flow rate achieved, and on the area of openings required to achieve the required flow rate. In buildings with high stacks, higher airflow rates will be achieved in lower levels, and this applies equally to updraught and downdraught air flow paths.

1.3.2 Evaporative Cooling

Evaporation refers to the change of state from liquid to vapour. Cooling through evaporation is a phenomenon that happens when a water body is exposed to non-saturated air and consist in the separation of water molecules from liquid structure to air. During this process, heat and mass transfer occurs, thus reducing temperature and increasing humidity of the surrounding air. The heat flows that contribute to change the physical state of the substance is also known as latent cooling (or heat). The amount of energy needed to convert a unit of mass of water from its liquid to vapour phase is given by the latent heat of vaporization of water, and is equal to 2465.56 kJ/kg, which is considerably greater than the energy needed to rise a unit of mass of air by one degree ($C_p=1.006 \text{ kJ/kg}^\circ\text{C}$). During the evaporative cooling process, the sensible heat from air is transferred to the water and becomes latent heat; in other words, air temperature is reduced to rise its humidity content (also called direct evaporative cooling). This is, of course, an idealised scenario, where no heat is gained or loss though external agents (also known as adiabatic saturation) and

all evaporation serves to cool the air and none to cool the water. However, in practice, the source of the sensible heat is also the water (as it usually gains some extra heat from circulatory friction), the surroundings (i.e porous media), and possibly solar radiation (Watt, 1986). Latent cooling is thus directly proportional to the absolute humidity increase achieved in the surrounding air and can be calculated using Equation (1-7):

$$Q_L = \rho \cdot C_e \cdot q \cdot [\Delta AH] \quad \text{Equation (1-7)}$$

Latent cooling
ASHRAE (2013, p. 18.13)

where Q_L is the latent cooling (kW), ρ is the density of the air (kg/m³), q is the air volume flow rate (m³/s), C_e is the latent heat of vaporization of water (2465.56 kJ/kg) and ΔAH is the air absolute humidity increase achieved after evaporation and equal to the mass of water evaporated per unit mass of air.

The humidity increase achieved during the evaporation process is given by the rate of evaporation $\dot{m}_{evap}$ and derives from the Fick's Law of Diffusion:

$$\dot{m}_{evap} = -\rho \cdot D_v \cdot \frac{\partial Y_v}{\partial y} \cdot S \quad \text{Equation (1-8)}$$

Fick's Law of Diffusion
Sidebotham (2015, p. 484)

The above expression postulates that the rate of evaporation, is indeed proportional to the air moisture content (Y_v), inversely proportional to the distance between the water body and a reference point in the air domain (y), proportional to the exposed surface area of the water body (S), proportional to air density (ρ) and proportional to a diffusion coefficient D_v which indicates how easy it is for molecules to migrate from the concentrated area to the less concentrated area. The diffusion coefficient of water for vapour into air is normally considered as $25.6 \cdot 10^{-6}$ m²/s (Sidebotham, 2015).

The theory of evaporative cooling and the relation between the different parameters involved can be better understood with the psychrometric chart, which was first developed by Dr Willis H. Carrier in 1904 (Gatley, 2004). Figure 1.22 illustrates a direct evaporative cooling process over the psychrometric chart

and compares it to a convective cooling process. As it was stated earlier, convective cooling consists on the reduction in air temperature keeping its moisture content constant. As temperature decreases (left horizontal movement), relative humidity RH increases, but the AH is not changed. Where this horizontal movement reaches the saturation curve (dew-point temperature), RH will be 100% and if air is cooled below this point, condensation will start. In the direct evaporative cooling process, the reduction of air temperature is accompanied with an increase of humidity (both AH and RH). The status point moves up to the left, along the constant wet-bulb temperature WBT or enthalpy lines.

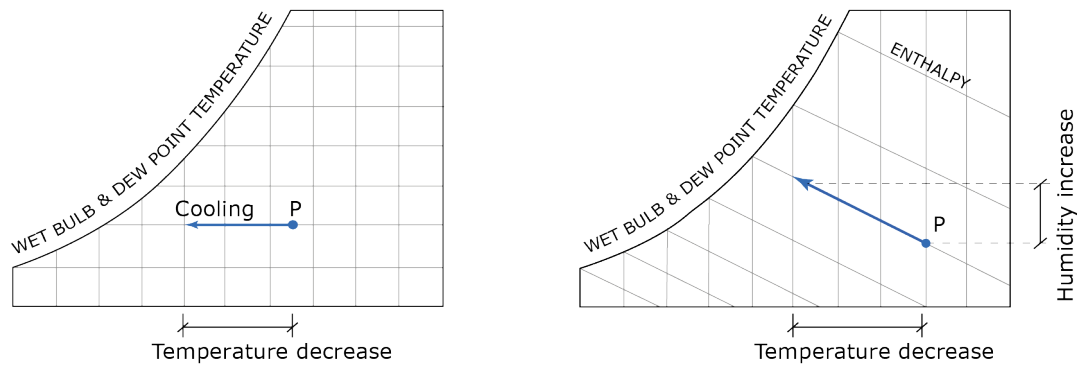


Figure 1.22 Convective cooling (left) and direct evaporative cooling process (right) explained on the psychrometric chart (After Szokolay, 2008, p. 14)

The main factors affecting the evaporation rate of an evaporative cooling system are: air humidity, air temperature, air movement, evaporation surface area and air pressure.

Air Humidity

Many parameters can be used to indicate air moisture content. Absolute humidity (AH) is the weight of moisture per kilogram of dry air and is independent from air temperature. Another indicator is relative humidity (RH), it indicates the air's degree of saturation so that, when air is fully saturated, RH is 100%. RH is inversely proportional to air temperature indicating that at constant moisture content, RH increases with lower temperatures. The ability of air to hold water vapor thus decreases with the decrease of its temperature. In this case, the potential for the evaporation of extra amount of water vapor would be

small. For an optimal performance of an evaporative cooling system, low humidity levels are thus recommended to maximise the mass transfer.

Air Temperature

Dry bulb air temperature (DBT) is the air temperature as measured by ordinary bare thermometers. As it is related to RH, a higher DBT reduces RH and thus increases the ability of air to hold extra moisture content. The wet bulb temperature indicates the temperature of the air as if it would be fully saturated and is an indicator of the lowest temperature that can be reached under current ambient conditions by the evaporation of water. This is the reason why the wet bulb temperature depression DBT-WBT determines the opportunity for the application of evaporative cooling strategies.

Water Temperature

At high temperatures, water molecules move at higher average speed. Under these conditions, it is easier to break the molecules' structure from the liquid surface thus increasing evaporation rate. Although higher water temperatures will increase evaporation rate, it will not necessarily contribute to further reduce ambient air temperature, as the sensible heat needed for the evaporation process will be also absorbed from the water, and not only from the air. Only in the case when the initial water temperature approximates the air's wet bulb temperature the adiabatic model occurs.

Air Velocity

When evaporation occurs on the water body, vapour pressure on the proximities of the surface increases to saturation, stopping the evaporation process. Constant dryer air movement is thus needed to renew the saturated air over the boundary layer to allow further evaporation. This scenario maintains or increases the evaporation rate.

Surface Area

The surface area indicates the area of the water body that is exposed to non-saturated air. It is expected that if doubled the area, doubled the evaporation rate. Thus, an increase in water surface area is desirable to maximise evaporation rate. It is for this reason that misting systems often present a high evaporation rate (Schiano-Phan, 2012). When the water body consist in a water film over a porous surface, it is important to maximise surface area while

allowing an adequate renovation of air over the boundary layer. Aerodynamic shapes presenting a low resistance to flow are thus desirable to achieve an even renovation of non-saturated air over the surface.

Air Pressure

Air pressure also has an impact on all psychrometric processes. A high air pressure will push down the water particles over the water surface and will make difficult for water to escape into the atmosphere as vapour. As seen earlier, air pressure varies with height and, at higher altitudes air pressure decreases. This implies that evaporative cooling systems are likely to present higher evaporation rates at higher altitudes.

1.4 Benefits of Passive Convective & Evaporative Cooling Techniques

One of the main objectives in environmental design is of course to achieve thermal comfort for the occupants. By reducing ambient air temperature, passive cooling brings the opportunity to achieve thermal comfort under hot conditions. At a larger scale, convective and evaporative cooling techniques can have a significant impact on the microclimate, mitigating one of the growing concerns in 21st century by architects and climatologists: the urban heat island (UHI) effect. This section reviews the fundamentals of thermal comfort and UHI, highlighting recommended techniques to maximise the benefits of the cooling strategy.

1.4.1 Human Thermal Comfort

Thermal comfort is often defined as a condition of mind which expresses satisfaction with the surrounding environment (ANSI/ASHRAE, Standard 55-2013). It involves a physiological response to the environment (human thermoregulation) and a psychological reaction by means of adaptation. As a subjective perception, the environmental conditions required for comfort are not the same for everyone and thus easy to determine. The physiological aspect of thermal comfort is the result of a constant heat exchange between the human body and the environment, and it has a direct impact on human health and the capacity to efficiently perform tasks. By biological processes, heat is generated and regulated in humans to keep a core temperature of 36.8°C. This is achieved with a balance between heat production and heat loss so that heat must be removed at a rate it is generated. The heat generated is associated to the

metabolic rate. This heat output consists in sensible and latent heat, and it is exchanged by means of vaporization from skin and lungs, radiation from the skin and convection. The equation for the heat balance between the body and its environment was first defined by Gagge (1936) as follows:

$$M = \pm S + E \pm R \pm C \quad \text{Equation (1-9)}$$

Human heat balance
Gagge (1936, p. 656)

where M is the metabolic rate, ranging from 100 W at sedentary activities to 750 W for leisure activities, S is the heat loss or gain in body heat storage, E is the evaporative heat loss, R is the radiation loss or gain, and C is the convection loss or gain.

The rate of heat exchanged with the environment will depend on four environmental conditions: air temperature, mean radiant temperature, air velocity and humidity. Lowering the temperature of the environment increases the body heat loss by convection and radiation; and lowering the ambient humidity increases the latent heat loss. In the contrary, when ambient temperature is increased above skin temperature, heat is given to the body by the environment (Jennings and Murphy, 1966).

Under hot conditions, the body increases blood flow to the skin in order to increase the output of heat. When ambient temperature is above skin temperature, sweating occurs and evaporation becomes the main source of heat loss. Sweat evaporation is a powerful mechanism to reduce skin temperature and thus heat from the core. When moisture concentration on the skin (skin wettedness) exceeds 25%, a person at sedentary activity experiences warm discomfort and unpleasantness (Winslow et al., 1937). In addition to the perception of skin moisture, skin wettedness also increases the friction between skin and fabrics, making clothing feel less pleasant. Moreover, insufficient heat loss can lead to overheating (hyperthermia).

With the Industrial Revolution, the interest in thermal comfort increased and a large number of scientific studies were performed (Parsons, 2003). In the nineteenth century, the general principles behind the establishment of physical comfort conditions were known and researchers identified the inadequacy of using air temperature alone in determining optimum conditions, pointing at the

combination of environmental parameters as the main drivers for thermal comfort (Bedford and Chrenko, 1974).

The most significant landmark in thermal comfort research was probably performed by Fanger (1970), who outlined the conditions necessary for thermal comfort through the development of his heat balance equation, as well as methods and principles for evaluating thermal environments and comfort conditions through the *predicted mean vote* PMV method. This comfort model derived from extensive laboratory and field data, focusing on physical physiological environmental parameters, becoming the most influential and widely used model throughout the world, and adopted by International Standards like ANSI/ASHRAE (Standard 55-2013) or ISO (7730:2005). In his work, Fanger identified six primary factors that must be addressed when defining conditions for thermal comfort: metabolic rate, clothing insulation, air temperature, radiant temperature, air speed and humidity.

It can be concluded that the physiological changes and comfort benefits deriving from a convective and evaporative cooling system directly relate to four of these parameters: air temperature, radiant (surface) temperature, air speed and humidity.

Air velocity

Air velocity influences convective heat exchange between people and the environment. The rate of heat exchange is increased with air velocity and, excess of heat loss improves comfort conditions under hot environments. An increase of air velocity can be achieved by an adequate ventilation strategy driven by wind force or thermal force according to the context. The offset in warm sensation created by air movement derives from the total heat transfer of the skin and is also influenced by the air and radiant temperature. When the mean radiant temperature is low and the air temperature is high, elevated air speed is less effective at increasing heat loss and a higher air speed is needed to improve comfort. Conversely, elevated air speeds are more effective when the mean radiant temperature is high and air temperature is low (Figure 1.23).

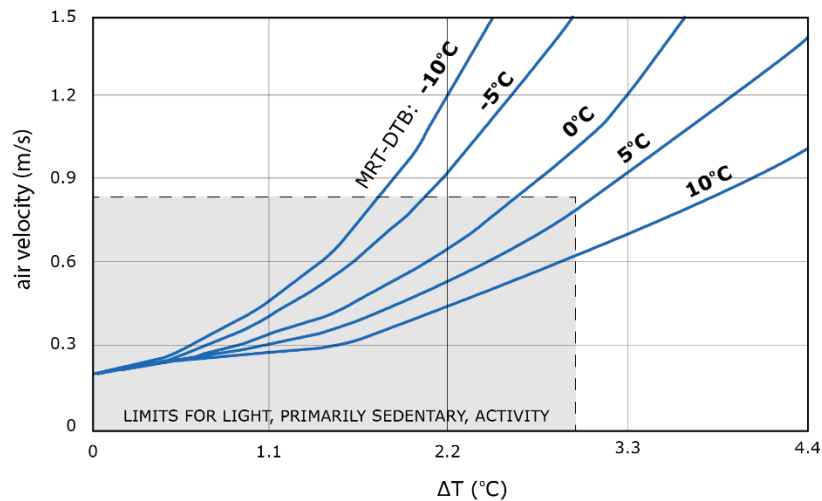


Figure 1.23 Comfort temperature rise with reference to a point at 26 °C and 0,20 m/s of air velocity. Typical summer clothing (0,5 clo) and sedentary activity (1,2 met) considered. (After ISO, 7730:2005, p. 45)

The above graph and the heat balance equation adopted by ANSI/ASHRAE (Standard 55-2013) suggest that air speed may be used to offset an increase in temperature by up to 3°C above the warm-temperature boundary at sedentary activities (1.2 met) with typical summer clothes (0.5 clo).

Humidity

Under moderate air temperatures, relative humidity has little impact on thermal sensation. Studies conducted by Nevins et al. (1975) confirmed that thermal discomfort (at 1.2 met) would not be appreciably elevated for humidity levels below 60% RH (with 0.6 clo). However, at higher levels of humidity (RH >60% at 27°C) a decrement in performance of light physical and mental tasks was noticeable. High humidity under warm temperatures reduces skin evaporation which, consequently increases skin moisture. Apart from the changes in thermal sensation produced, a discomfort is experienced as result of high moisture concentration on the skin (skin wittedness exceeding 25%), which increases the friction between skin and fabrics, making clothing feel less pleasant.

1.4.2 Urban Heat Island (UHI)

Urban heat island is a phenomenon increasingly affecting the world population. The rapid urbanization process experienced over the past six decades (Figure 1.24) support the predictions of a 66% of the world to be urban by 2050 (United Nations, 2014). The increase in urban population has direct consequences on the

total energy consumption in cities. If current trends in urban expansion continue, urban energy use will increase more than threefold from 2005 to 2050 (Creutzig et al., 2015). As consequence, the emission of greenhouse gases in un urban areas will increase, resulting in a serious environmental degradation that contributes to climate change (Figure 1.25).

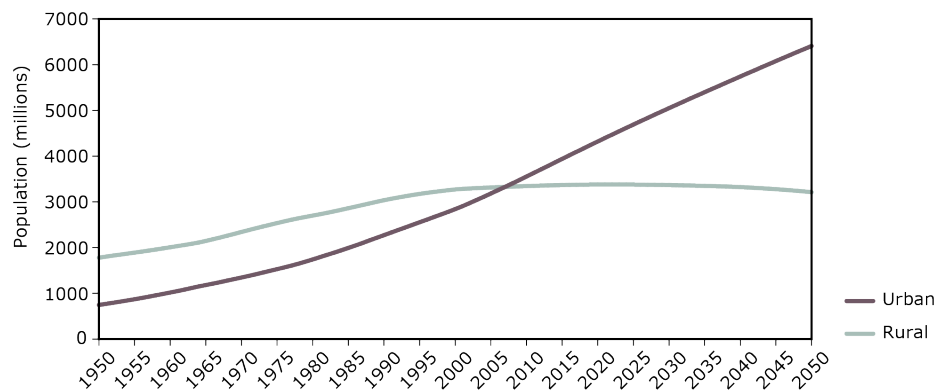


Figure 1.24 Urban and rural population of the world, 1950–2050 (United Nations, 2014, p. 7)

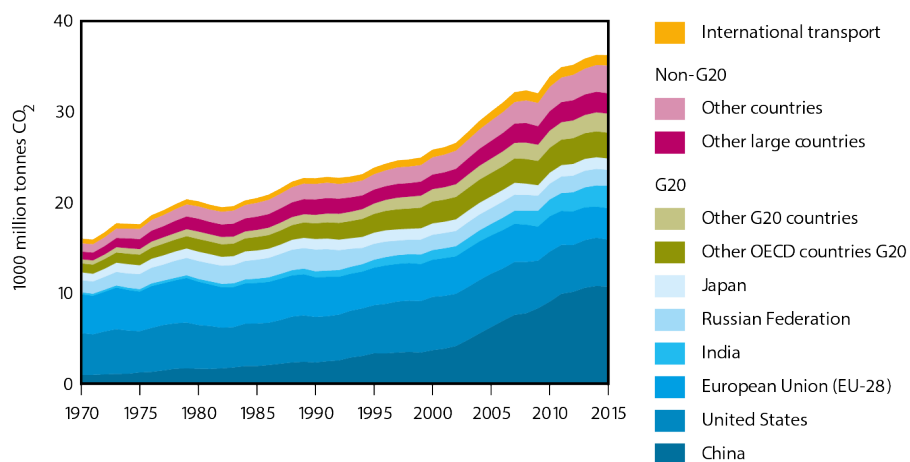


Figure 1.25 Global CO₂ emissions per region from fossil-fuel use and cement production (Olivier et al., 2016, p. 13)

In addition to the anthropogenic heat released to the urban form, extended opaque surfaces and the reduction of green areas negatively affect the thermal balance of the cities affected. The positive balance experienced in the main city capitals in the world results in higher temperatures and contributes to the so called urban heat island effect (Landsberg, 1981). Urban heat island is defined

as the increase of ambient or mean radiant temperature of a city compared to the surrounding rural zones. Its magnitude can be determined from the air temperature depression between urban and rural areas, also known as urban heat island intensity (Erell et al., 2011). Recent research conducted by Santamouris (2007b) suggest that in Europe, typical urban heat island intensities range from 2°C to 3°C at daytime, and 7°C to 8°C at night time. The studies confirmed that urban heat island is more evident during the night period, in which temperature differences between urban and rural areas can reach up to 12°C. In most cases the peaks coincide on days with clear sky and low (or null) wind velocity. The increase of air temperature in urban areas has, evidently, an impact on buildings' energy consumption. In the context of London, for instance, the increase of urban cooling load is up to 25% compared to the rural load over a year (Kolokotroni et al., 2007).

Among the vast range of existing strategies to mitigate UHI, the most recurrent techniques by experts in the field refer to the implementation of green areas, the use of *cool* materials and the application of evaporative cooling techniques. The benefits of vegetation in urban areas are highly recognised: the evapotranspiration process of plants and trees absorbs ambient heat and regulates their foliage temperature. Also, their capacity to intercept solar radiation and offer shading decreases surface temperatures, thus reducing convective and conductive heat loads. Studies confirmed that the cooling efficiency of urban trees and vegetation varies with the species composition, canopy intensity and the size and shape of the *green* area (Feyisa et al., 2014), being most useful when planted in strategic locations to shade windows, roofs, pavement in parking lots or streets (McPherson and Simpson, 1999). Other cooling techniques such as the application of cool materials in the urban fabric contribute to the reduction of surface temperatures by reducing heat absorption. In a daily cycle, materials absorb incident solar radiation and dissipate part of it to the environment by convection and radiation. The amount of heat absorbed is determined by the material's absorptance, which is strongly related to the surface colour. Cool materials are characterized by their low absorption (thus high albedo) and high infrared emittance through the use of light-coloured materials. Considering that 60% of roofs and asphalt in urban areas have low albedo, the potential benefit of replacing these surfaces with cool materials is considerable. Studies conducted by Synnefa et al. (2008) and Hashem et al. (2004) suggested that cooling demand of buildings can be reduced by 10-40% depending on the climate. Moreover, an external air temperature reduction by 1-3°C can be achieved by increasing the city's albedo (Sailor, 1995).

The evaporation of water surfaces is one of the major components of hydrological cycle which significantly affects the thermal balance of outdoor spaces. In urban environments, evaporation from concrete pavements, building rooftops, and asphalt surfaces is discontinuous and intermittent (Ramamurthy and Bou-Zeid, 2014), and the water drainage from impervious surfaces to sewage significantly reduces the actual evaporation when compared to rural areas, contributing to an increased urban heat island effect (Oke, 1987). The advantages of water bodies as a natural cooling technique are well known, but their actual thermal contribution to the urban energy balance is less clear and still requires more studies and further observations due to the large amount of external parameters and physics involved (Knozová et al., 2016). Water bodies can contribute to the mitigation of UHI through evaporation, the effect of its thermal inertia and its low reflectivity (Santamouris and Kolokotsa, 2016). The low reflectivity of water (0.06) causes a low solar reflection to other surfaces in the surroundings, avoiding in this way their warming up. The high specific heat of water (4.183 kJ/kgK) is of an order of about four times the value of most urban materials and confers the ability to store heat from the surrounding environment and consequently moderate external air temperatures. Studies by Murakawa et al. (1991) and Katayama et al. (1991) have proven that large water bodies like rivers and lakes contribute to reduce peak air temperatures in the early afternoon by up to 5°C with air velocities varying from 1 to 5m/s. However, this effect can be reversed and large water bodies in the proximities on an urban area can avoid bigger temperature drops at night, opening up a discussion on the role of large open water surfaces on mitigating urban heat island effect (Steenneveld et al., 2014). The contribution of evaporation on localised air temperature reduction is, however, more evident. This phenomenon can take place in large water bodies, wetted porous materials and water droplets. The rate of evaporation broadly depends on the climate, and more specifically, to the media and the environmental parameters that determine the microclimate. For instance, a high UHI intensity combined with low air humidity enhances evaporation. Further studies by Tang and Etzion (2004) informed that air velocity is indeed the determining parameter, suggesting that the water evaporation rate from the free exposed water bodies may be different from the evaporation rate from the rough wetted surfaces, and concluding that these last ones increase the evaporation rate with lower air velocities. This phenomenon is probably due to the different boundary layers of vapor density over the rough wetted surfaces and the exposed free water body. Evaporation thus contributes to the reduction of the surrounding air temperature and also the water and the

porous surface temperature. The studies by Guerra Macho et al. (1994) performed in a pond of 30 cm depth with a sprayer system working over four consecutive days in Seville confirmed that the water surface temperature can drop by 16°C below the external air temperature. This surface temperature reduction also applies to the porous media (if existing), and, as with *cool* materials, the potential of replacing impervious materials with wetted porous surfaces to reduce UHI effect is significant considering the extension of the urban areas, especially in high density urban layouts.

1.5 Conclusions

A review on the traditional use of convective and evaporative cooling strategies has demonstrated how ancient civilizations mastered the use of water in conjunction with additional passive cooling strategies to exploit aesthetic delight and improve thermal comfort. Only the carefully considered integration of these techniques in the architecture allowed the creation of excellent built examples like the Adalaj stepwell in Ahmedabad, Gujarat, or the Casa Pilatos in Seville, Spain, reaffirming the importance of coupling passive cooling strategies and space layout.

The applicability of passive cooling techniques is also high, with more than 50% of the countries in Europe and US, and about one third of China extension presenting optimal conditions for the application evaporative systems. The number of heat-stress hours, cooling degree hours and DBT-WBT are considered good indicators to evaluate the feasibility of these techniques, normally associated to hot/dry climates. However, most of the countries satisfying this criterion may be water stressed, pointing out water consumption and operational hours as determining factors on the overall efficiency of the system. Moreover, the mapping technique applied in the US, became a reliable source to allow architects with limited knowledge in this field to determine the feasibility of an evaporative cooling strategy or to evaluate the market opportunity of a novel evaporative cooling product.

A review on the principles of convection and evaporation highlighted the need of an airstream to maximise convective heat loss and improve evaporation rate. This airstream can be driven naturally by wind or thermal forces, and both the intensity and flow path will respond to the space configuration. Air velocity can create a physiological cooling effect on people by increasing human convective heat loss, and at the same time, can renovate saturated air in the boundaries of

a porous media to allow further evaporation. In addition, other three parameters associated to evaporative cooling will determine the resulting improvement in thermal comfort perception: dry bulb air temperature, mean radiant temperature and relative humidity, confirming the big potential of these techniques to mitigate heat stress in both indoor and outdoor spaces.

The use of water in outdoor space to induce evaporation has also been a recurrent technique to address UHI effect. Misting systems are now a popular technique to improve outdoor comfort in public spaces and terraces, and studies have confirmed that water surface temperature can drop by 16°C below the external air temperature under hot and dry conditions. This surface temperature reduction also applies to porous surfaces and, the potential of replacing impervious materials with wetted porous surfaces to reduce UHI effect is significant considering the extension of the urban areas, especially in high density urban layouts.

2 Advances in Contemporary Architecture Design

Ancient cooling techniques have been rediscovered in the last decades as an effective and less impacting alternative to air conditioning systems. Recent built examples and research work have demonstrated the great potential of passive systems to reduce energy demand while providing a good and healthy environment, and advances in design and manufacturing technologies opened up new opportunities for optimising the traditional techniques and developing novel systems.

With the growing concern of climate change and global warming, environmental design became one of the most demanded design approaches in contemporary architecture (Jankovic, 2012) and, at present, many leading industries and architectural firms have successfully integrated passive design methods and expertise during early design stages. Unfortunately, the passive design approach still has not been broadly implemented and various questions commonly emerge regarding the number of resources available to assist in design process of convective and evaporative cooling systems: where and how to obtain the required knowledge to inform the design? Are there sufficient methods to interpret specialized knowledge at early design stages?

This chapter evaluates the advances of convective and evaporative cooling techniques in architecture through the analysis of recently emerged systems and published work. Taking ceramics as main porous media to induce evaporation, the chapter explores the physical properties, integration opportunities and novel fabrication techniques to evaluate the opportunity for optimised evaporative cooling systems. Further, a review on recent built examples and previous work highlights the achievements, shortfalls and room for improvement of passive cooling techniques. The chapter finishes with a description of the resources currently available for sizing and performance prediction of convective and evaporative cooling systems in order to promote the development and integration of novel passive cooling systems in architecture.

2.1 Why Ceramics? Advances in Porous Ceramic Systems

Ceramics have been linked to the history of architecture since ancient settlements in the Middle East by 10,000BC, where the first mud bricks were sun-dried for hardening (Campbell, 2003). With the invention of fired bricks in

about 3500BC to increase their strength and durability, ceramics became the first man-made material that joined stone and wood (directly extracted from nature) for construction. One of the first brick constructions that still remains today is the great ziggurat of Ur dating back to 2100 BC. Like ancient Egyptian pyramids, ziggurats presented a pyramidal structure and were commonly used to perform religious rituals, essential to ancient Sumerian civilizations. The ziggurat of Ur-Nammu, one of the best preserved (Figure 2.1), was built using more than seven million mudbricks and 720,000 fired bricks (Woolley, 1939).

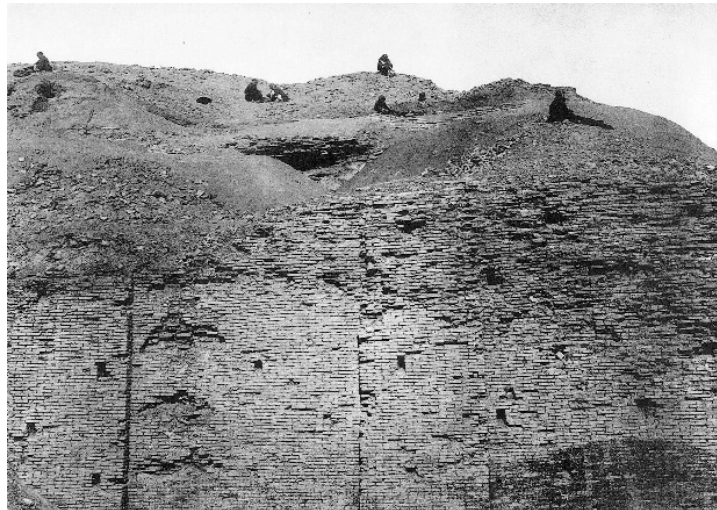
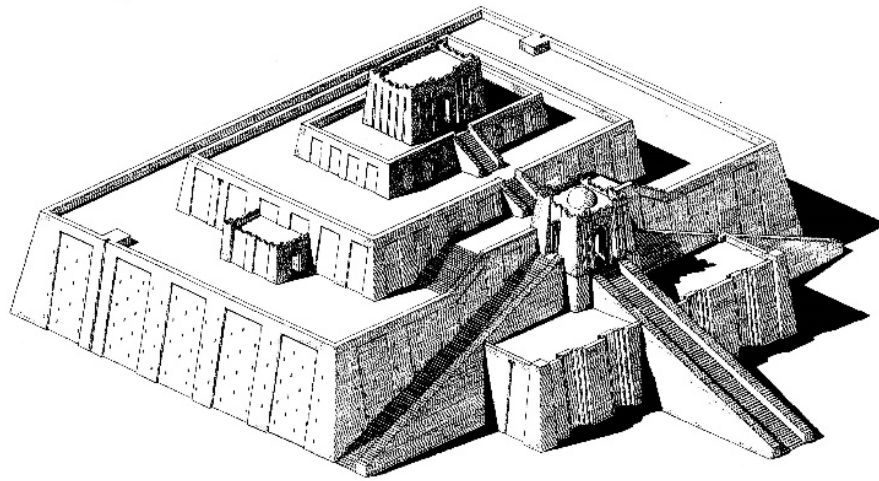


Figure 2.1 Isometric projection of the ziggurat of Ur-Nammu restored (top) and the SE side wall of the ziggurat (bottom), (Woolley, 1939, pp. 224, 256)

The application of bricks in architecture subsequently extended over the entire Roman empire, in which bricks were used for public and private buildings, and

reintroduced through centuries over the Classical, Medieval and Modern World (Hall and Cruickshank, 2015). Brick production increased massively with the industrial revolution and so its popularity in modern architecture after being used by some of the 20th century's most famous architects like Le Corbusier, F. L. Wright and Louis Khan (Pfeifer et al., 2002). Over the 20th century, ceramic application extended to roof, flooring, facades or latticework and it's also in this century when the concept of ceramic system emerged. Ceramic material systems are a step forward from the traditional decorative and protective use of ceramics. Bechthold et al. (2015) defined them as new applications for ceramics as a multi-functional, performative material system outside the enclosed building envelope that benefit from the innovations in ceramic technology. The novel systems benefit from advanced engineering structures and connections allowing freely suspended ceramic elements, first seen as ventilated façades at the TU Munich (Thomas Herzog) and the IRCAM Studio in France (Renzo Piano) in the late eighties (Jodidio, 2017). Over the past decades, ceramics became again highly appreciated and new ceramic systems emerged as a result of the collaboration between ceramic fabricators and architects. The technical knowledge from ceramists combined with the creativity of designers ended up with iconic constructions and products leading the way to the future of ceramics. In Spain and Italy, for instance, the industry of ceramic is well established and expanding, and international Ceramic Exhibitions attract every year more than 80.000 professionals worldwide (Cevisama, 2017). Recent ceramic applications and innovations rediscovered the two-and-three-dimensional aesthetics of ceramics through, for example, patterned adhered tile systems, screen façade systems, structural assemblies, daylight filters, large formats, inkjet printing, photocatalytic systems and also, evaporative cooling systems:

- Adhered tile systems. The traditional tiling was reinvented with the introduction of digital technology. New patterns generated through complex multi-variable algorithms rapidly gained the interest of designers when applied on organic curving forms. A built reference on this technique is the Santa Caterina market in Barcelona (Spain), built in 1848 and renovated in 1997 (Cumella, 2004). In this project, a colour pattern of hexagonal extruded and glazed tiles adhered with a glass fibre mesh to the curved roof. The mosaic was comprised of 120,000 tiles containing 67 unique colours mimicking the market products (Figure 2.2). Another project in which ceramic tiling was taken to its best expression is the Liling World Ceramic Art City by Archea Associati (Archea, 2014) built in 2014 in Changsha (China). The complex entirely devoted to ceramic art

and consisted in a series of concave-shaped buildings in the form of large vases clad with polychrome ceramic modules generating three-dimensional textures (Figure 2.3).

- Screen façade systems. As an evolution of ventilated facades, screen systems allow to aesthetically enrich vertical surfaces while providing with additional benefits such as shading or filtered light. Designers usually benefit from the wide range of available production methods to customise the form, which is later assembled to a substructure. One of the first examples of this practice is the Spanish pavilion designed by Alejandro Zaera for the Expo 2005 in Aichi, Japan (AZPML, 2005). The ceramic system in this project was a reinterpretation of the traditional lattice screens and consisted of about 12,400 three-dimensional hexagonal forms creating an interior and exterior face. The ceramic elements were obtained using a three-step manufacturing process: plastic pressing using a two-part mould, trimming and assembling. The pieces were glazed using a colour pattern that evoked the classic colours of Spain (Figure 2.4). A more recent example of screen system is the addition for the Lisbon Oceanarium designed by Campos Costa Arquitectos (CCA, 2010). The building was wrapped with a ceramic system consisting of about 5,000 ceramic elements that evoked fish scales, reaffirming the strong tradition of ceramic-clad buildings in Lisbon. The ceramic system transitioned from a ventilated façade in the first floor to a screen system that provided shade in the second floor using a series of forms obtained from extruding, trimming and mould-pressing manufacturing processes. The final glaze coating applied on the surface created a range of coloured shades emphasizing its visual impact in the surroundings (Figure 2.5).
- Structural assemblies. New production techniques allowing an improved strength and durability of ceramics led to structural systems that detach from the building envelope to become an architectural artefact or sculptural work. This is the case of the CCCloud project designed by Kengo Kuma and Associates (KCAA, 2010) and manufactured by Casalgrande Padana (Casalgrande, 2010). The system consisted of a 5m-tall structure comprised of 526 pressed tiles that attached to vertical steel rods. The rods were post tensioned making the ceramic element the primary supporting system of the lightweight structure. The resulting

artefact became a monument marking the entrance of the industrial ceramic district of Sassuolo, Italy (Figure 2.6).

- Daylight filters. When form follows (daylight) performance ceramic systems may present a wide range of designs informed by simulations with substantial geometrical complexity. This is the approach followed by the research initiative led by Dr Rosa Urbano involving undergraduate and postgraduate students from the Liverpool School of Architecture in collaboration with ASCER (Spanish Association of Ceramic Tile Manufacturers). In her published work (Urbano, 2013) the author demonstrated the capacity of ceramic elements to improve the distribution of daylight in buildings by means of light-control techniques, geometrical analysis and contemporary digital design tools. The sinuous cone surface was an example of daylight filter that induces multiple reflections and textures. The system consisted of casted ceramic elements differently arranged in a ceiling that mitigated daylight levels while providing a wide range of angles of incidence. While performance testing using raytracing techniques successfully informed the final design, the casting fabrication method brought a number of limitations that unfortunately made it not suitable for mass production (Urbano and Wanner, 2016).
- Large formats. For many decades the traditional dry-pressing manufacturing method used in the production of tiles limited the maximum dimensions of ceramic elements (Bechthold et al., 2015). With the integration of novel compacting techniques and a wider range of additives, the industry has been recently able to overcome this limitation, allowing larger formats over 1000x3000mm at 3-9mm thick while offering an increased strength (Refin, 2017). Large formats are ideal to cover large areas such as showrooms or reception areas and allow a wide range of surface finishes with customised colours and textures (Figure 2.8).
- Inkjet printing. This technology evokes the traditional colour printer in which the ink is distributed over a *canvas* which becomes the actual body of the tile. The *emotile* project by Ceracasa Cerámica (Emotile, 2006) first developed this technology to reproduce all types of images at resolutions over 1000 dpi, overcoming the limiting factors for ceramic inkjet printing in the past (mostly associated to the solubility of the grounded pigments).

The mass-customized production lines now implemented by most ceramic firms, allow to add a personal touch to the space by simply providing with the image to the manufacturer (Figure 2.9).

- Photocatalytic systems. Recent developments by the ceramic industry focused in sustainability. The addition of titanium dioxide (TiO_2) in the clay mixture or over the ceramic surface (traditionally used in sanitary appliances) adds photocatalytic, hydrophilic and antibacterial properties to the ceramic elements. This technology relies on the high oxidising effect of TiO_2 when exposed to natural light to react and destroy most organic compounds. Other environmental benefit from this technology relates to air purification, the reduction in NO_x and the generation of active oxygen. Further developments of this technology conducted by Casalgrande Padana in collaboration with the Japanese company TOTO Ltd. (Biosceramics, 2013) applied a selection of ever-active noble metals to maintain its antibacterial capacity even under limited illuminance levels (Figure 2.10).
- Evaporative cooling systems. The traditional use of ceramics as porous media to induce evaporation has also been reinvented by modern ceramic industry. Porous ceramic evaporators have been integrated in contemporary architecture as façade screens (Sekkei, 2011), in cavity walls (Terrados-Cepeda et al., 2015), ventilation towers (Schiano-Phan, 2004) and as ceramic sculptures in the building surroundings (Martín-Gómez et al., 2009). Specification and further description on the environmental benefits of these applications are discussed in Chapter 2.2.



Figure 2.2 Aerial view of the Santa Caterina market in Barcelona and roof tile pattern (Cumella, 2004, p. 1)



Figure 2.3 Ceramic tiling in Liling World Ceramic Art City (Bianchi, 2014, p. 1)



Figure 2.4 Spanish pavilion at the 2005 World Expo, Aichi, Japan (Cumella, 2005, p. 1)



Figure 2.5 Oceanário addition in Lisbon, Portugal. (Cumella, 2010, p. 1)

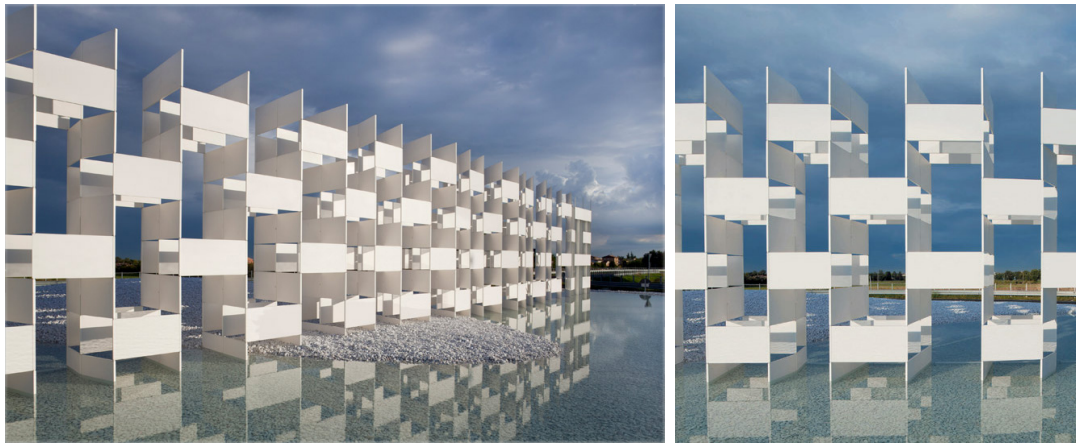


Figure 2.6 The CCCloud, commissioned by Casalgrande Padana (Casalgrande, 2010, p. 1)

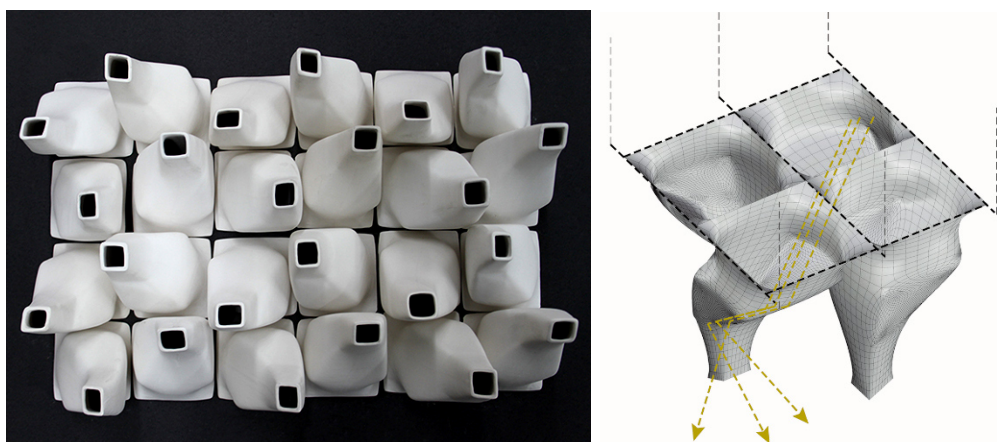


Figure 2.7 Ceramic sinuous cone surface (Urbano and Wanner, 2016, p. 10).

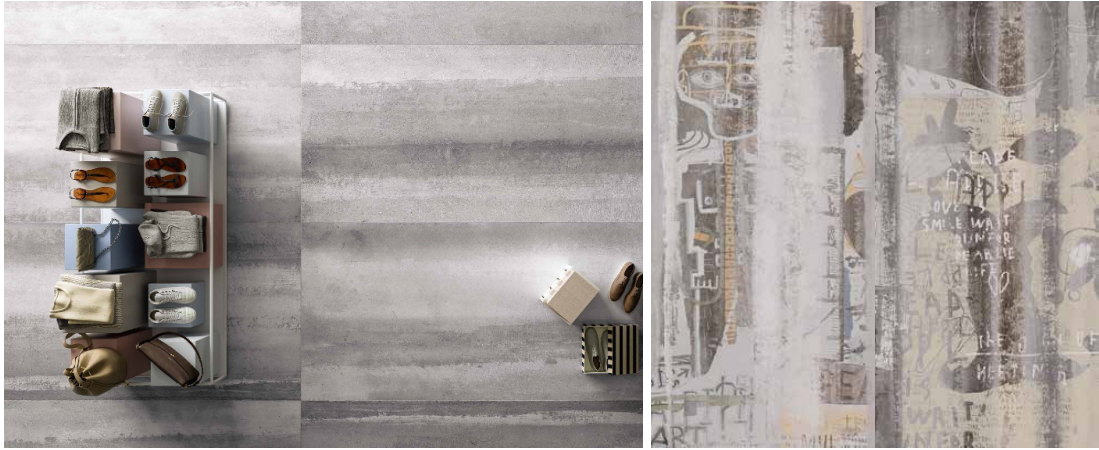


Figure 2.8 Overlay concrete line by Refin Ceramiche (Refin, 2017, p. 1)



Figure 2.9 Inkjet-printed ceramic tiles by Ceracasa Cerámica (Emotile, 2006, p. 1)

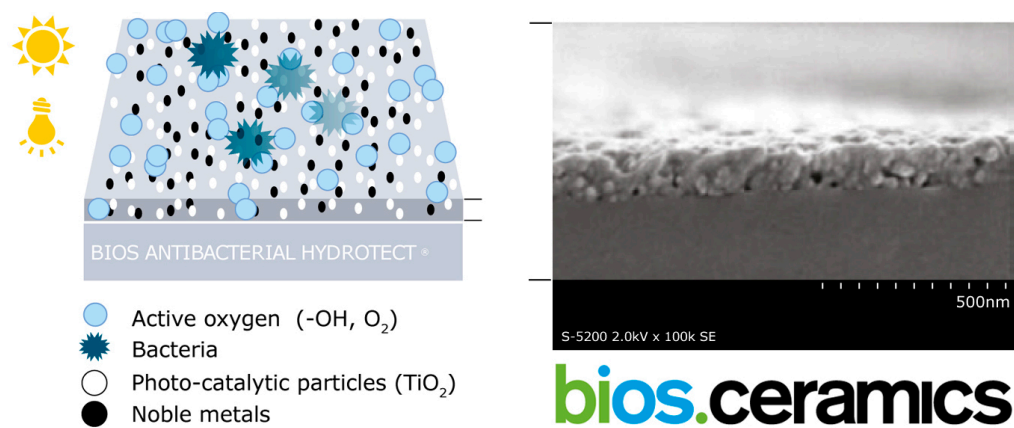


Figure 2.10 Photocatalytic principle. Biosceramics by Casalgrande Padana (Biosceramics, 2013, p. 1)

2.1.1 Ceramic Properties

Ceramics gained their popularity as construction material from their physical properties (hardness, durability and density), the wide range of applications offered and the availability of their main material component: clay. Clay is naturally formed from geological processes and is primarily composed of $\text{Al}_2\text{O}_3 + 2\text{SiO}_2 + 2\text{H}_2\text{O}$ and, in less proportion, Fe_2O_3 , which gives its characteristic red colour. For obtaining ceramic, clay first needs to undergo a four-step transformation process: blending, shaping, drying and firing. The blending process consist on the mixture of different clay materials, additives and water at different proportions that usually responds to the project specific needs. Clay mixtures are critical to the performance of the resulting ceramic, determining the porosity, strength and additional properties. With the incorporation of modern chemistry, additives can offer advanced characteristics to ceramics (Marinescu, 2007), but the design of the clay mixture often requires deep knowledge in chemistry and process engineering. Indeed, most ceramic producers have a material scientist and chemist technicians to undertake this task. Moreover, the amount of water is also a determining factor during the blending process: water will determine the level of viscosity of the mixture, which should respond to the fabrication process selected. In the other hand, the mixture water content is strongly related to the final porosity of the ceramic, as water will most evaporate during the drying and firing process. The European Commission describes the traditional manufacturing processes in modern history as follows (IPPC, 2007):

- Pressing: Die boxes are charged with a pre-set volume of clay, and pressure is usually applied from above and below.
- Extrusion: A plastic body with substantial clay content is tempered with water, and then forced to the die of the extruder usually by a powerful auger.
- Moulding: It requires a much softer mix and hands and simple wooden moulds are used to achieve the required size and shape.
- Casting: The material is mixed with water and is poured into a porous mould traditionally made of plaster. The capillary suction of the mould draws the water and then the solid body is removed from the mould.

The shaping process is followed by the drying and firing process. The first one removes most water content of the blend and a correct drying is crucial to avoid water to boil inside the blend during the subsequent firing stage. Traditionally, clay was dried naturally but in modern ceramic industry there are multiple techniques to optimise this process: hot floor dryers, chamber dryers, tunnel dryers, dehumidifying dryers, etc. The firing process is probably the most important phase of the fabrication and determines the final physical characteristics of the sample. During this process, the remaining water is evaporated and the ceramic element suffers a series of chemical reactions and internal tensions that lead to *shrinkage* and determine the final ceramic product. Shrinkage occurs when moisture is removed during drying and firing and typically the size reduction ranges from 8-12% (Pelleg, 2014). This phenomenon can often be problematic for complex geometries and in some cases, it can result in warping and cracking. It is thus recommended to scale up parts to match final desired dimensions and when needed, to use removable support structures that hold cantilevering areas. The final product can range from porous materials like terracotta, refractory products or expanded clays, to denser materials like porcelain and stoneware commonly used for architectural tile application (Morales Güeto, 2005).

Among the wide variety of ceramic products, the interest of this research focus in *terracotta*, and those products presenting certain porosity to allow water absorption and flow towards the ceramic surface to induce evaporation while maintaining adequate mechanical properties and durability. Terracotta products often present a suitable range of porosities from 7 to 24% (Hall and Hoff, 2012) determined by the blend mixture and the firing temperatures, typically ranging from 900 to 1000°C (Morales Güeto, 2005). Raising the firing temperature would allow pores to merge and coarsen, limiting the evaporative cooling application. The internal connection of the pores is known as capillary network and allow water to flow to certain directions driven by capillary forces. This connection is often determined by the manufacturing process so that, for instance, an extrusion process will create an unidirectionally oriented pore structure parallel to the extrusion direction as shown in Figure 2.11 (Kiyoshi et al., 2011), whereas a pressing process will mostly output disconnected pores that will not contribute to the transport of water (Sirota et al., 2013). This property may become a determining factor for the selection of the water supply system for evaporative cooling application and previous researches demonstrated the effectiveness of parallel connected pores to rapidly absorb water by capillary forces (He and Hoyano, 2010).

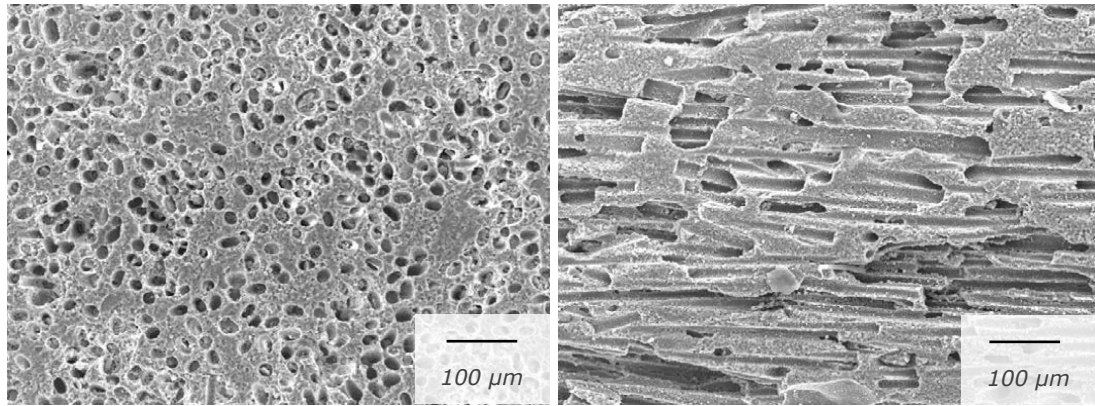


Figure 2.11 Unidirectional microstructure of porous ceramic elements prepared by extrusion method
(Kiyoshi et al., 2011, p. 6)

2.1.2 Paving the Future of Ceramics: Digital Fabrication

The examples in previous section demonstrated how digital tools contributed to the ceramic industry development over the past decades. To these applications now join new robotic techniques and 3d printing technologies to ensure the future of ceramic architecture. These innovations bring new geometrical opportunities and professionals in the ceramic sector consider them as a “rediscovery of the new three-dimensional aesthetics of ceramics” (Bechthold, 2016).

Robotics refer to the use of flexible devices capable to perform a vast range of computationally driven operations while adding additional degrees of freedom to material manipulation. In architecture, this technique is under a continuing evolution over the past 30 years and has currently achieved a high level of accuracy, flexibility, and reliability (McGee and Ponce de Leon, 2014). The published material by the robotic fabrication conference series organised by the Association for Robots in Architecture (ROB|ARCH, 2010) is an extensive catalogue of the new design and material production opportunities offered by this technique. Furthermore, recent explorations in robotic shaping, cutting, folding and bending capabilities conducted by students from the Harvard Graduate School of Design demonstrated the abilities of new designers to integrate digital modelling, performance simulation and automated fabrication in the design process. These techniques are often combined with traditional shaping processes (i.e. extrusion, rotation, etc.) to create innovative shapes widening the potential areas of application (Figure 2.12).

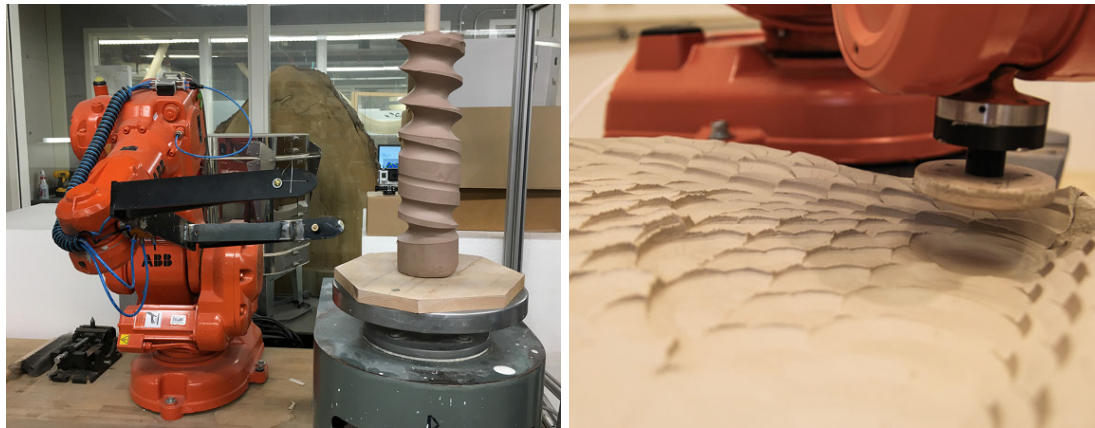


Figure 2.12 Robotic carving (left) on spiral shaped ceramics (Mishra, 2017) and robotic forming (right) of clay block (Raspall et al., 2014, p. 343)

3d printed ceramics: design opportunities and limitations.

Additive manufacturing technologies (AMT), commonly known as 3d printing, is the process of creating three dimensional objects by adding several layers of a material in succession. 3d printing has been under continuous development over the past 15 years and has been successfully integrated in aerospace, medical, energy, transportation and consumer product industries, being plastic the most used material (Core-Baillais et al., 2017). In architecture, one of the first successful application in a large scale is the construction of a two-storey 3d printed house using a blend of cement, sand and fibre by the Chinese company Windsun (Windsun, 2013). Using this technology, the company reported significant savings in construction cost by 80% and in labour costs by 60% while considerably reducing the construction time.

The use of clay as 3d printing material is continuously growing and currently represents 8% of the most used materials by industry for 3d printing (Core-Baillais et al., 2017). Ceramic 3d printing is well established in medical industry in the generation of scaffolds for bone regeneration (An et al., 2015, Sabree et al., 2015), but its application in architecture is still ongoing an extensive design exploration through prototyping. 3d printing artists, design specialists and the above-mentioned Graduate School of Design from Harvard University have demonstrated the innovative design opportunities from this technology in architecture. Through the assemblies of small-to-medium scaled 3d-printed objects, designers have been able to create all sort of screen façade systems, brick wall patterns, daylight filters and shading devices (Bechthold et al., 2011, Bechthold, 2016, Building Bytes, 2012).

Some recent acknowledged examples on the use of 3d printing technologies to improve the surrounding environment are the *cool brick* designed by Virginia San Fratello and Ronald Rael from Emerging Objects (Emerging Objects, 2015), and the *woven screen* and *thermodynamic pavilion* designed by graduate students from the Graduate School of Design from Harvard University (Friedman et al., 2014, ASCER, 2017), displayed in Figure 2.13 and Figure 2.14.

The *cool brick*, designed in 2015 and inspired by the porous ceramic cooling windows in Muscat, Oman (Figure 1.4), proposed a three-dimensional ceramic lattice-like structure to hold water in its pores and induce evaporation when exposed to the air stream. The *woven screen*, designed in 2013 and inspired by woven patterns, explored the use of deposition techniques onto non-flat surfaces to obtain weaving forms that can be used as light filtering screen façade systems. More recently, the thermodynamic pavilion exhibited in Cevisama (Cevisama, 2017), explored the optimisation of thermal mass by the use of sinusoidal textures that increase the exposed surface area of the ceramic elements assembled into a structural frame. Although the efficiency of these designs is questionable due to their apparent overcomplicated geometry and the lack of empirical studies that determine the cooling, light filtering or thermal capacity of the systems, these projects are relevant design + fabrication investigation exercises that advocate the potential of coupling digital fabrication techniques with passive design principles for improving the architecture environment.

Despite the vast design opportunities of this technology, the application of ceramic 3d printing in architecture is still under development and there is a long way to compete the cost-effectiveness of traditional ceramic production. Professor Martin Bechthold highlighted that mass production of certain 3d-printed ceramic elements is currently by about ten times more costly than a traditional extrusion method, but, assuming a quick development of this technique, the cost gap will certainly narrow over time (Bechthold, 2016). Moreover, technical limitations of this technique have been discussed by most ceramic artist: Zocca et al. (2015) claimed that more research is needed to improve surface quality and density of 3d-printed ceramic elements, and Bechthold et al. (2011) identified the drying process as a potential bottleneck for this technique. The higher amount of water content needed in the clay for 3d printing requires, in most cases, adjustable moulds for drying and increases the risk of surface dimples and break after shrinkage. To overcome these risks,

overhangs, sudden steps, undercuts and non-self-supporting ceramic elements should be avoided.

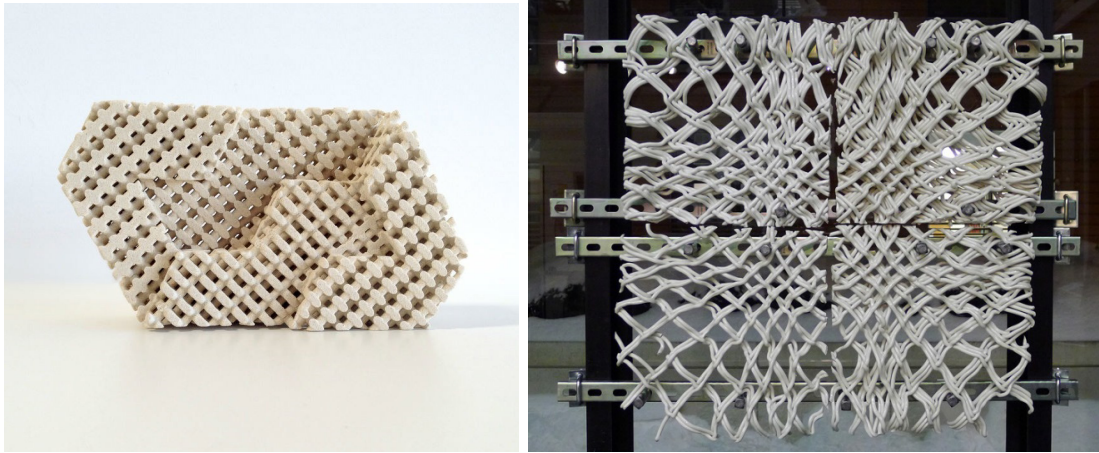


Figure 2.13 3d printed 'cool brick' (left), (Emerging Objects, 2015), and ceramic pattern (right) assembled on a building façade (Friedman et al., 2014, p. 268)

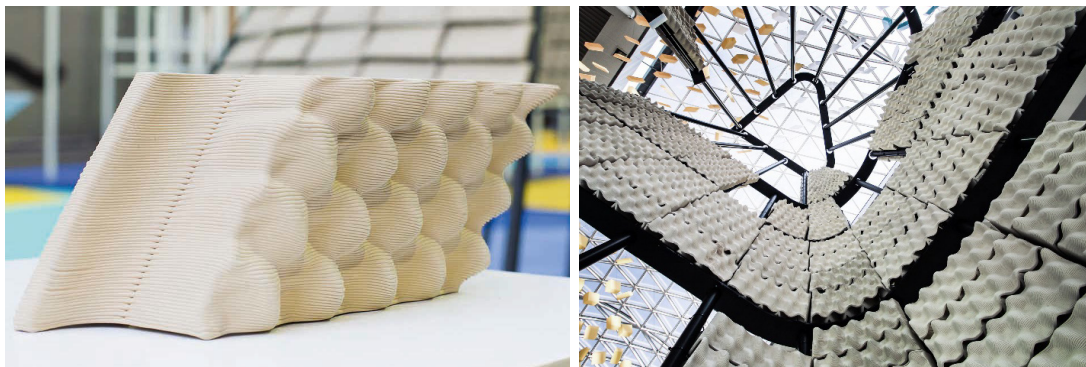


Figure 2.14 Textured 3d printed pavilion by researchers and students from the Material Processes and Systems (MaP+S) Group at the Harvard University Graduate School of Design (ASCER, 2017, p. 10).

2.2 Advances in Convective & Evaporative Cooling Techniques

The use of mechanical and electrical alternatives for cooling in architecture became the norm for much of the 20th century and a part of 21st-century life. Santamouris (1992) confirmed the extended use of these systems has increased energy demand by about 100 kWh/m² in Greece, which could likely represent most Mediterranean countries, contributing to the subsequent increase in CO₂

concentrations in the atmosphere (NASA, 2017). Fortunately, designers have currently raised their concern on the negative effect of mechanical systems associated to poor air quality and the increase of energy demand and a wide range of passive design alternatives have emerged in the past decades.

2.2.1 Built Examples & Experimental Studies

Research, experimental studies and built precedents contributed to expand the knowledge and expertise in this field, and the applications of contemporary convective and evaporative cooling techniques in architecture include a wide range of components and systems that integrate in building geometry through towers and stacks, walls, screens and roofs. Classified by geometrical integration, some of the most relevant applications are discussed in this section.

Towers & stacks

Passive Downdraught Evaporative Cooling (PDEC) in stacks rely on the principles of natural ventilation to drive air into the occupied zones. When coupled with an evaporative cooling method, a downdraught flow path is achieved (under still air conditions) due to an increase in air density when warm air is cooled at the top of the tower. The downdraught flow path can also be accentuated by capturing the prevailing wind using wind catchers. The evaporation is achieved by incorporating water curtains, porous media or misting systems.

The earliest precedent of PDEC application in modern architecture dates back to 1986, when Cunningham and Thompson (1986) built an experimental building in Tucson, Arizona to evaluate the efficiency of this technique. In this building, two side towers defined the ventilation strategy, in which the driving force was determined by the solar chimney at the opposing end of the PDEC tower. The cooling method consisted in wetted pads arranged close to the intake, which induced the evaporation of water contained within the porous surface when exposed to the airstream. Analysis on the monitored data by Givoni (1992) confirmed the efficiency of the system, in which under a peak ambient dry bulb air temperature of 40.5°C and wet bulb temperature of 21.3°C, the air exited the tower at 23.8°C at a rate of 30 ACH, maintaining an average indoor air temperature of 24.6°C (Figure 2.15). Further studies on PDEC performed in the 90s by Givoni and Al-Hemiddi (1995) and inspired by the traditional use of wet charcoal in the Egyptian *malqaf* (Figure 1.7), explored the use of water curtains as the source of evaporation. Monitored data from a real scale application in

Riyadh demonstrated that under ambient dry bulb air temperature of about 45°C, the indoor air room temperature remained at 29°C.

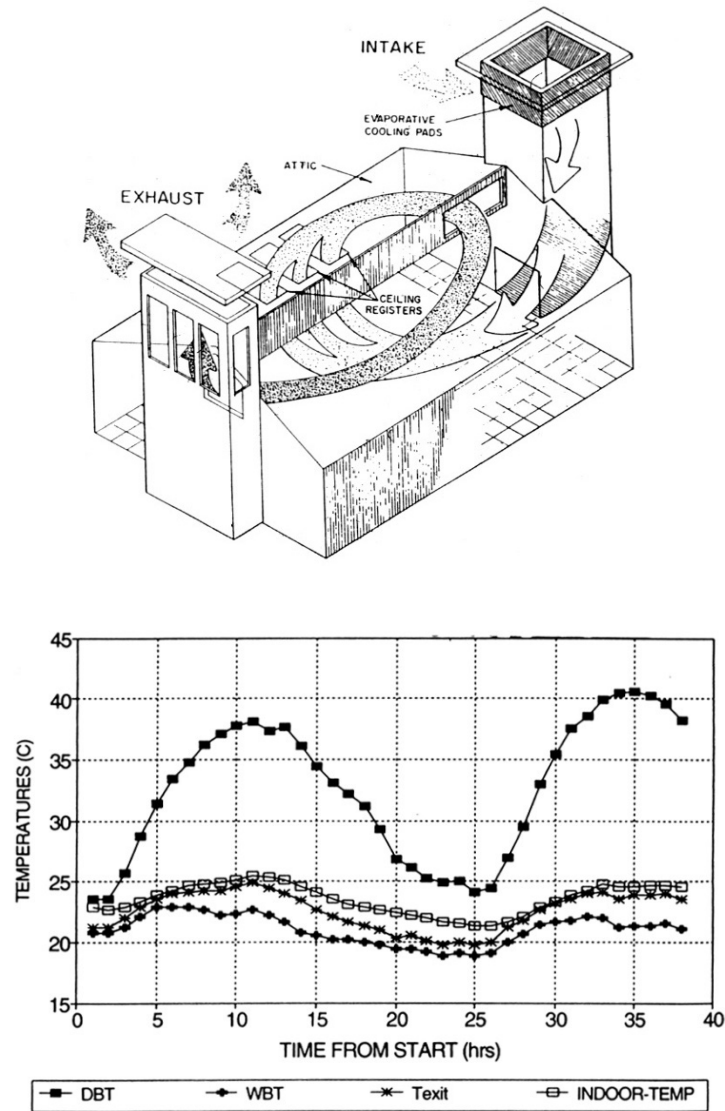


Figure 2.15 Experimental building in Tucson, Arizona. 3d diagram (top) and (bottom) monitored temperature data (Ford et al., 2010, p. 37)

Newer techniques incorporated micronisers to spray smaller water droplets with a diameter well below 100µm to increase the evaporation rate. Misting systems increase the number of water droplets while reducing their diameter when compared to water shower systems, which subsequently increases the exposed surface area for evaporation. The drop diameter should theoretically be small to, during the evaporation process, minimise the heat extracted from the water and maximise the heat extracted from air. Moreover, the relative movement of the

drop falling within the stack also accelerates the cooling process. The evaporative cooling towers designed for the Expo in Seville 1992 (Alvarez-Dominguez et al., 1991) served as precedent for most PDEC applications in architecture (Figure 2.16). In order to mitigate the hot and dry climate of Seville during the summer season, the design implemented vernacular passive cooling strategies (shading, high reflectivity, vegetation, water bodies, etc) and novel techniques like PDEC systems to improve thermal comfort and condition the open spaces of the exposition. A known example is the *rotonda bioclimática*, which combined different systems to integrate the principle of evaporative cooling. The plaza was 900 m² and design and technique were coupled together to maximise the effectivity of the strategy: a mechanical system supplied precooled water from the ground, and several micronisers placed at the top of the central tower, on the trees and at the side entrances sprayed misted water to cool the environment. The evaporative cooling method was coupled with a downdraught ventilation strategy achieved by the large stack at the centre of the plaza and assisted by a fan that supplied 3.3m³/s into the space. Monitored data from August 1989 confirmed that air temperature inside the *rotonda* (1m above ground level) remained 10°C below ambient temperature while relative humidity increased from 20% to 60% (Guerra Macho et al., 1994).

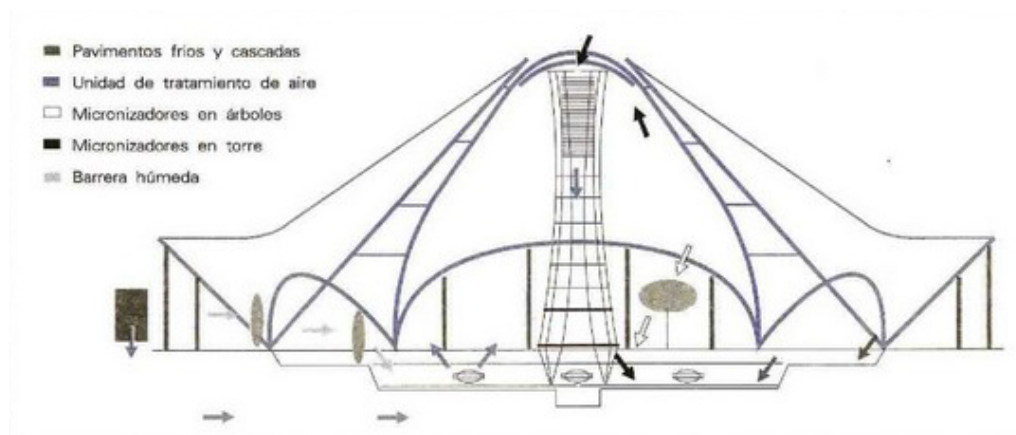


Figure 2.16 The Rotonda Bioclimática and passive cooling strategies-2d diagram (Alvarez-Dominguez et al., 1991, pp. 55,56)

The application of cooling towers in Sevilla served as precedent of the subsequent studies performed by Alhemiddi (1991) in Southern California and Pearlmutter et al. (1996) in Southern Israel using this technique. In the latter case, the authors used a laboratory model serving a small glazed courtyard to demonstrate reductions in delivered air temperature over ambient of 10-15°C and a peak cooling potential of 100-120 kW (Figure 2.17). Other studies (Bowman et al., 1997, Bowman et al., 2000) confirmed an average 10-15°C temperature reduction in systems applied in Southern Europe and the Middle East, highlighting the potential for substantial cooling of semi-enclosed spaces in arid regions with a modest consumption of water and energy. The above-mentioned precedents led to the application of PDEC systems in a large number of buildings in US, Europe and India, and further details on their cooling performance and system configuration are given by Ford and Hewitt (1996b) and Ford et al. (2010).

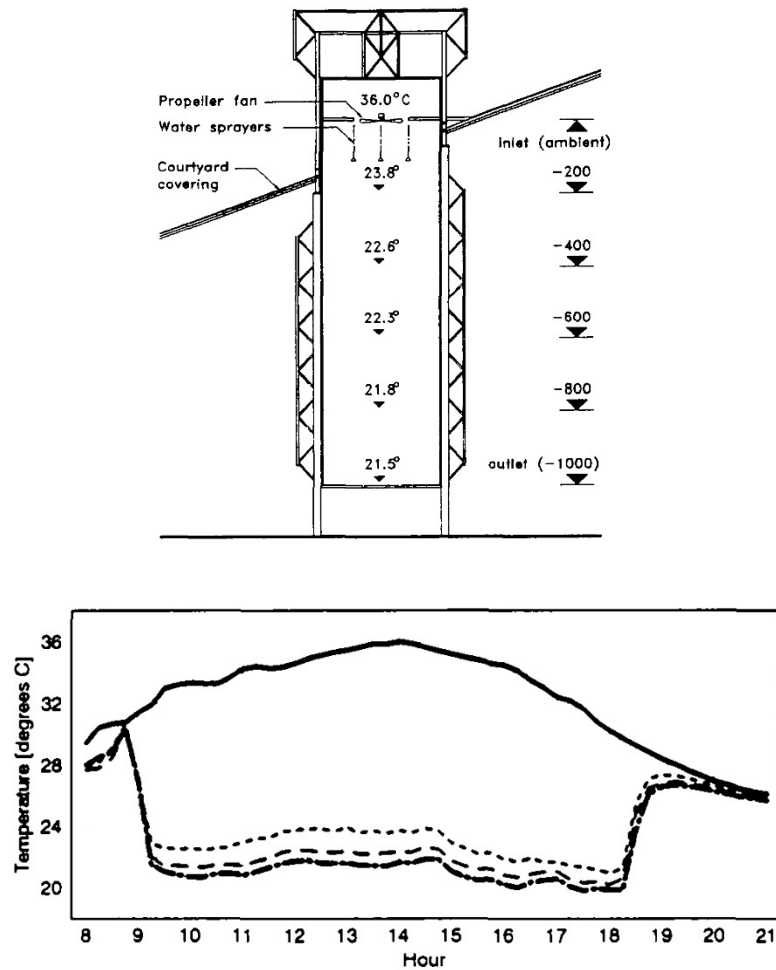


Figure 2.17 Schematic section of evaporative cooling tower showing typical temperature profile during summer daytime operation (Pearlmutter et al., 1996, p. 194)

The first PDEC application in residential buildings in Europe was designed and built by students from the University of Nottingham for the 2010 Solar Decathlon Europe event in Madrid, Spain (Ford et al., 2012). The proposed two-storey house presented a double-height space to improve natural ventilation for convective cooling at night time, and induce a downdraught air movement at daytime. A misting system with eight micronisers placed at the top of the double-height space cooled the air drawn through the roof during the warmer hours of the day and the average daily water consumption was reported to be 40 litres for five hours of operation. The hot and dry weather conditions recorded during the monitoring period allowed the PDEC system to reduce ambient temperature by 12-14°C (with no significant air stratification observed) while relative humidity increased from 15% to 40-65% during operating hours (Figure 2.18). The work also contributed with a numerical model to allow performance

prediction of similar systems and focused future work on the control of airflow rates drawn by the inlets in order to improve the efficiency of the system.

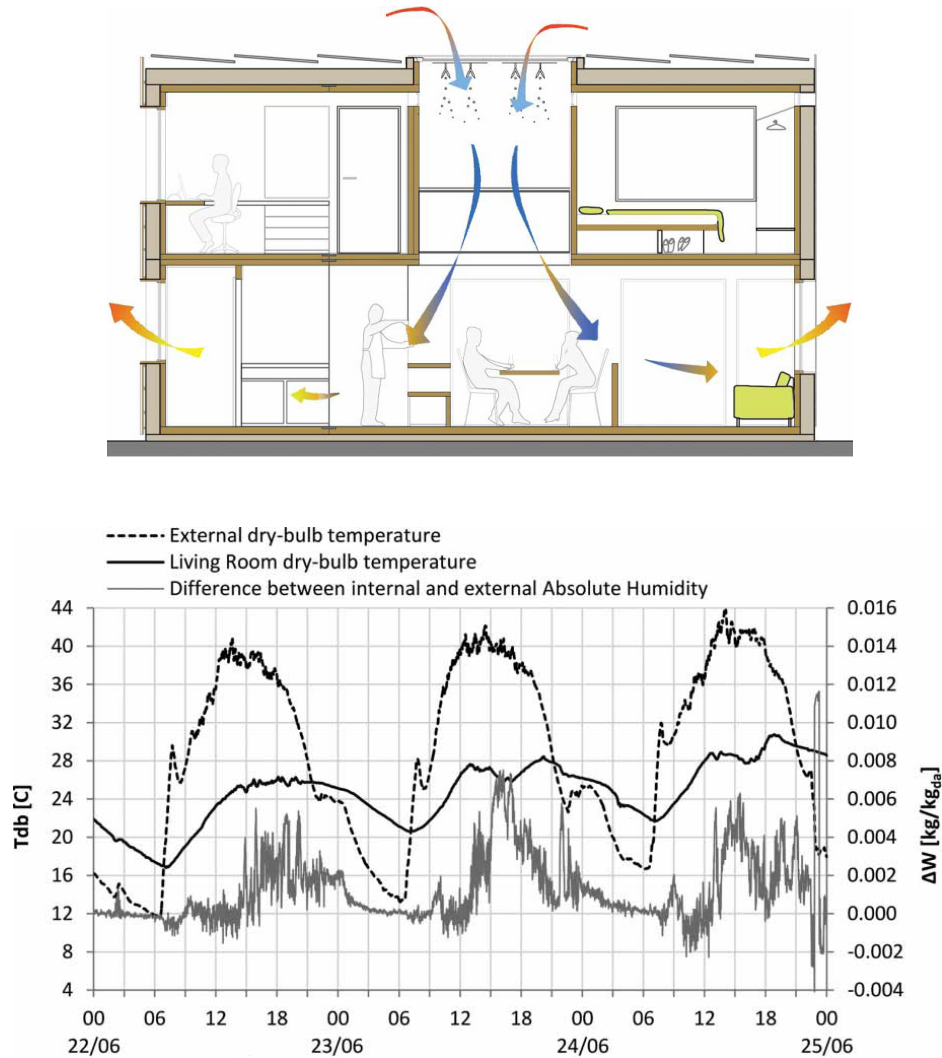


Figure 2.18 House section showing the summer daytime airflow path and external and internal dry-bulb temperatures monitored during 22nd-24th June 2010 (Ford et al., 2012, pp. 293, 297)

More recent studies performed by Chiesa et al. (2017), explored the application of PDEC systems in existing residential buildings in Northern Italy. The system comprised a tower constituted by Ø400mm PVC tubes and a misting system with micronisers at the top of them (Figure 2.19). The evaporation of water inside the columns created a downdraught air movement that supplied the cooled air through horizontal ducts into the adjacent apartments of 40-60m² of floor area. Monitored data collected from a testing tower confirmed an ambient air

temperature deduction of 8 to 11°C inside the columns while the mean water consumption resulted in 0.65 l/h per column. The research contributed with numerical models that estimated an indoor mean air temperature reduction of one degree for a mean supply airflow rate of 2 ACH, and highlighted the easy integration of PDEC systems on the existing residential building stock of Northern Italy.

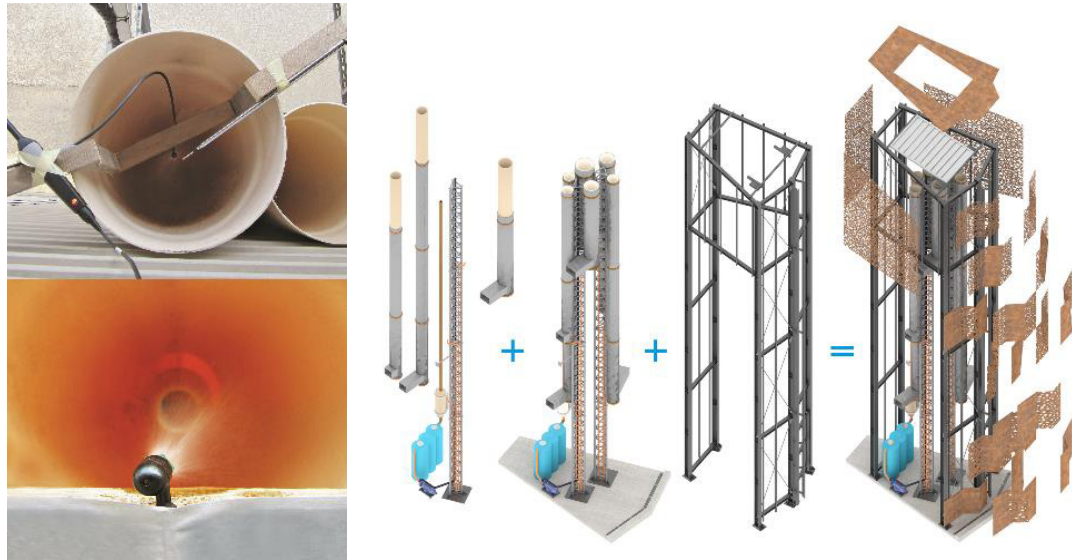


Figure 2.19 PDEC system mock-up and proposed application on the residential building stock of Northern Italy (Chiesa et al., 2017, pp. 603, 608)

The experience and knowledge gained over 30 years of experimental studies and applications have confirmed the potential of PDEC techniques in architecture. Reported performance from these projects ranged between 70–95% of the incoming air's wet-bulb temperature and confirmed the average PDEC cooling efficiency suggested by Givoni (1994), who stated that the delivered air temperature from the PDEC tower is by about 80% of the wet bulb ambient air temperature depression.

Walls

The integration of evaporative cooling methods in walls is often accomplished with a porous element which induces the evaporation of water contained within its surface and pores while offering certain mechanical characteristics. Generally, water is supplied by a dripping or misting system, which enters the pores and

distributes over the entire surface. When exposed to non-saturated air, evaporation takes place by absorbing heat from the surrounding air and the porous element. As consequence, apart from the cooled air provided by the system, the occupants can benefit from a low mean radiant temperature when the space is coupled with the porous system.

In the last decades, porous evaporative cooling systems have been implemented in many of the Mediterranean greenhouse structures, representing by about 20% of the total surface area of greenhouses worldwide (Baille, 2001). In Southern Spain, for instance, there are more than 30,000ha of farms and greenhouses that are evolving to improve their air tightness and integrate pad-fan systems in their structures (Valera and Molina-Aiz, 2008). Pad-fan systems consist in a permeable wall made of wetted cellulose pads coupled with a cross-ventilation strategy driven by fans placed at the opposing wall of the pads. The geometry of the pads play an important role on the system's efficiency and studies conducted by Franco et al. (2011);(2014) concluded that the pad geometry (length and width of the undulation of the sheets and the angles of incidence) has a direct impact on the pressure drop produced. Monitored data from a greenhouse in Almeria, Spain, presented a mean ambient temperature reduction of 8°C under air velocities of 1-1.5 m/s with average water consumption of 146.3 l/m² per eight hours of operation per day (Franco et al., 2010). Further studies conducted by Dayioğlu and Sillelia (2015) in Ankara, Turkey explored the air temperature gradient from the cellulose pads and monitored data presented ambient air temperature reductions of 7, 4 and 5°C at 1, 4 and 7 m respectively (Figure 2.20).

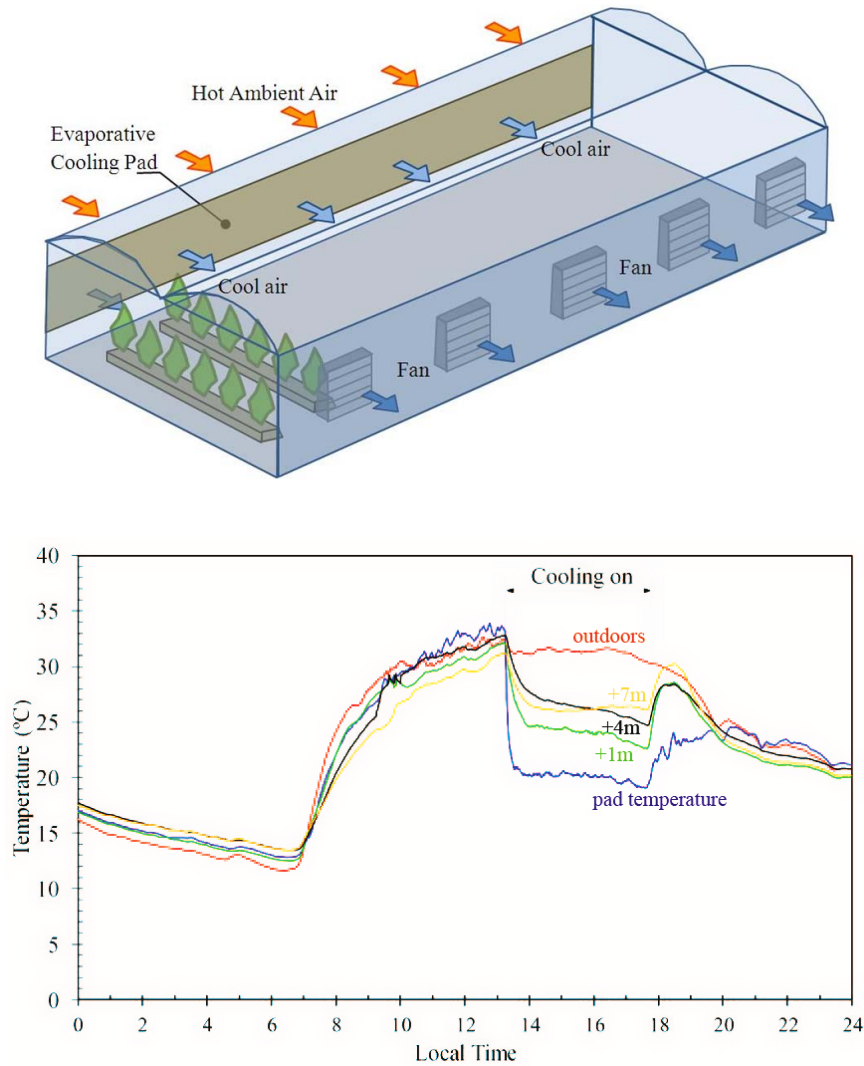


Figure 2.20 Greenhouse with fan-pad system 3d diagram (Franco et al., 2014, p. 1430), and ambient air temperature reductions from monitored data in Ankara, Turkey (Dayioğlu and Sillelia, 2015, p. 139)

Alternative building applications using porous ceramic elements in cavity walls were explored by Ibrahim et al. (2003). The author crafted porous ceramic evaporators that consisted in closed, hollow, rectangular units of 315x165x35mm and 5mm wall thickness containing water in their interior. His experiments evaluated the impact of different firing temperatures on the cooling achieved when exposed to controlled air temperatures, humidities and air velocities inside an environmental chamber. Monitored data confirmed that higher porosities yielded to consistently better cooling, and typical air temperature drops of 6-8°C accompanied by about 30% of relative humidity increase were achieved under mean air velocities of 0.08-0.10 m/s. Further studies explored the evaporation rate of multiple evaporators arranged in a stack and exposed to a downwards fan-driven air movement. The collected data

suggested that the arrangement of the evaporators was critical and the configuration that allows greater air mass flow over the ceramic surfaces will yield to higher cooling. Using a single row arrangement in a stack resulted, for instance, in mean air temperature drops of 2°C behind the evaporators (Figure 2.21).

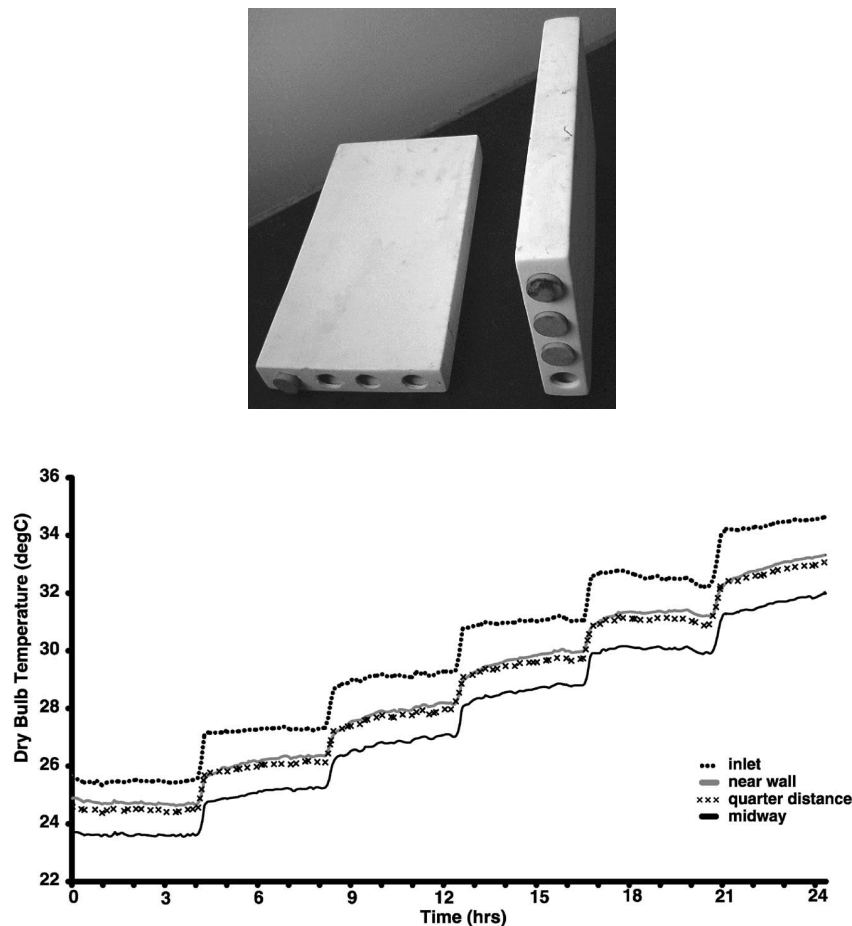


Figure 2.21 Porous ceramic evaporators prototype and typical duct air temperature drop under a single row arrangement in a stack (Ibrahim et al., 2003, p. 945)

The integration of porous ceramic evaporators within the building envelope was further explored by Schiano-Phan (2004). Her work involved the development of a new crafted ceramic prototype also storing water and the monitoring of a wall mock-up in which the ceramic elements were placed in a row. The collected data suggested that the peak air temperature could be reduced by about 6-7°C whilst the resulting U-value of the wall was reduced from 1.3 to 0.3 W/m²K. The integration of the ceramic evaporators in cavity walls and ventilation towers coupled with a downdraught air ventilation strategy were further explored to

determine the system's applicability in the residential building stock to reduce annual cooling loads (Figure 2.22). The author concluded that energy savings up to 1.5 MWh/yr could be achieved in a typical apartment and CO₂ savings of 18.5 Tonnes/year could be achieved for the entire building. Extending the application to a larger scale, the system could potentially meet the entire cooling loads of 70% of the building stock of Seville. Although the energy saving potential was evident, there were still barriers to overcome for a successful market penetration, such as a dissemination program and the need of a more cost-efficient manufacturing process.

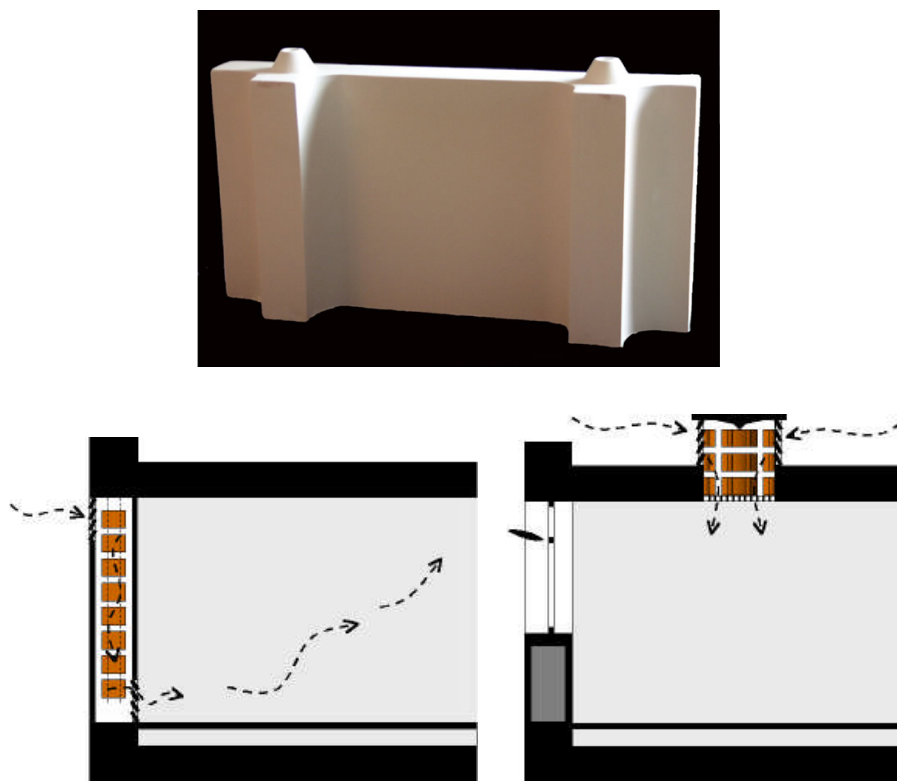


Figure 2.22 Stacked evaporator prototype and potential integration in the existing building stock (Schiano-Phan, 2004, p. 3)

Other studies conducted by Naticchia et al. (2010) explored the application of porous elements in standard ventilated facades to reduce wall surface temperatures and conduction gains. The system consisted in the integration of a misting system within the cavity wall that sprayed water on a porous (calcium silicate) insulation layer. Monitored data from a wall mock-up suggested that the evaporative wall could reduce external surface temperature and air cavity temperature during the operating hours by 5 and 10°C respectively when

compared to a standard cavity wall. Further studies also confirmed that the evaporative cooling system reduced conductive heat fluxes by $3\text{--}4\text{W/m}^2$ (Figure 2.23). The work contributed with a novel application of indirect evaporative cooling and, although the reduction in conductive gains could have also been achieved with a lower solar absorptance and thermal transmittance of the wall, the low surface temperature recorded on the outer layer opened up further research options on the mitigation of urban heat island effect by lowering mean radiant temperatures. Further studies by Chen (2011) developed theoretical models that contributed to determine radiative, convective and conductive heat exchanges of the porous plate with the adjacent environment.

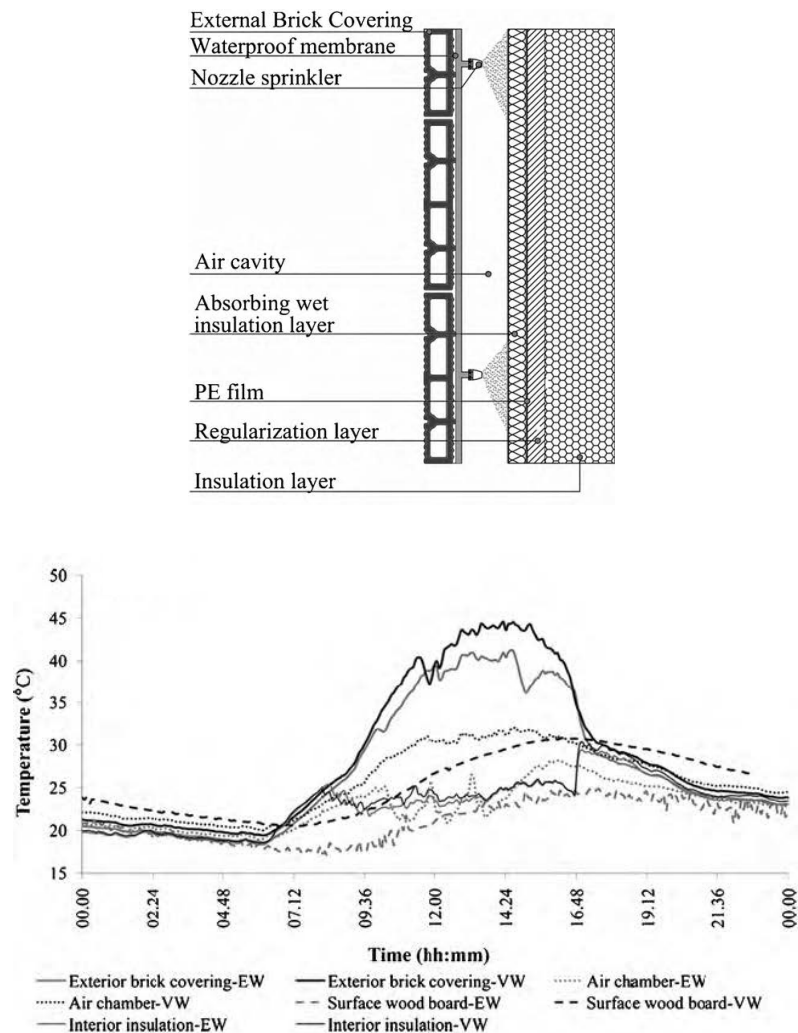


Figure 2.23 Functional model of the evaporative façade and air temperature reduction achieved during the testing period (Naticchia et al., 2010, pp. 1928, 1935)

A large scale wall-application of evaporative cooling systems using porous ceramic elements was achieved in the Patio 2.12 prototype, designed and built for the 2012 Solar Decathlon Europe event in Madrid, Spain (Terrados and Moreno, 2014). The project, inspired by the traditional central courtyard as thermal buffer and the ancient wetted vessels to cool air through evaporation, organised the spaces around a central *patio* and covered the entire envelope with porous ceramic tiles wetted by an irrigation system integrated in the cavity walls (Figure 2.24). As result, the internal spaces benefitted from both convective and radiative cooling: the cooled air from the air cavity was driven into the occupied spaces by small fans placed at the base of the walls; and the low surface temperatures of the wetted ceramics were exposed to the central *patio*. Monitored data and dynamic thermal simulations reported that the cavity air temperature remained between 5-10°C lower than ambient air while the occupied spaces presented an indoor air temperature between 1-3°C lower than ambient air during the operating hours. Although the indoor air temperature reduction achieved by the evaporative cooling system is small, the project served as a good example to demonstrate the flexibility offered by porous ceramics to integrate evaporative cooling methods in residential buildings.



Figure 2.24 Interior and exterior view of Patio 2.12, Madrid, September 2012 (Terrados and Moreno, 2014, pp. 71, 78)

Screens

The use of evaporative cooling systems as screens or permeable walls mostly rely on a porous media presenting certain architectural aesthetics coupled with a cross or stack ventilation to induce evaporation. Ceramics have been the preferable material by most recent studies and applications due to their durability and versatility.

One of the first modern studies on evaporative cooling screens used, however, nylon lines to create water films exposed to the airstream. This study was conducted by Giabaklou and Ballinger (1996) with the objective to minimise the growing cooling demand in Pakistan by applying this technique in openings and balconies of low-rise multi-storey buildings. When water dropped vertically over the filaments, fieldwork studies conducted in New South Wales (Australia) confirmed a 6.6 to 9.9°C ambient air temperature drop under a mean air velocity of 0.28m/s (Figure 2.25). The author contributed with equations and charts to enable the calculation of the saturating efficiency and opened up new research options on the application constraints and the creation of numerical models for performance prediction.

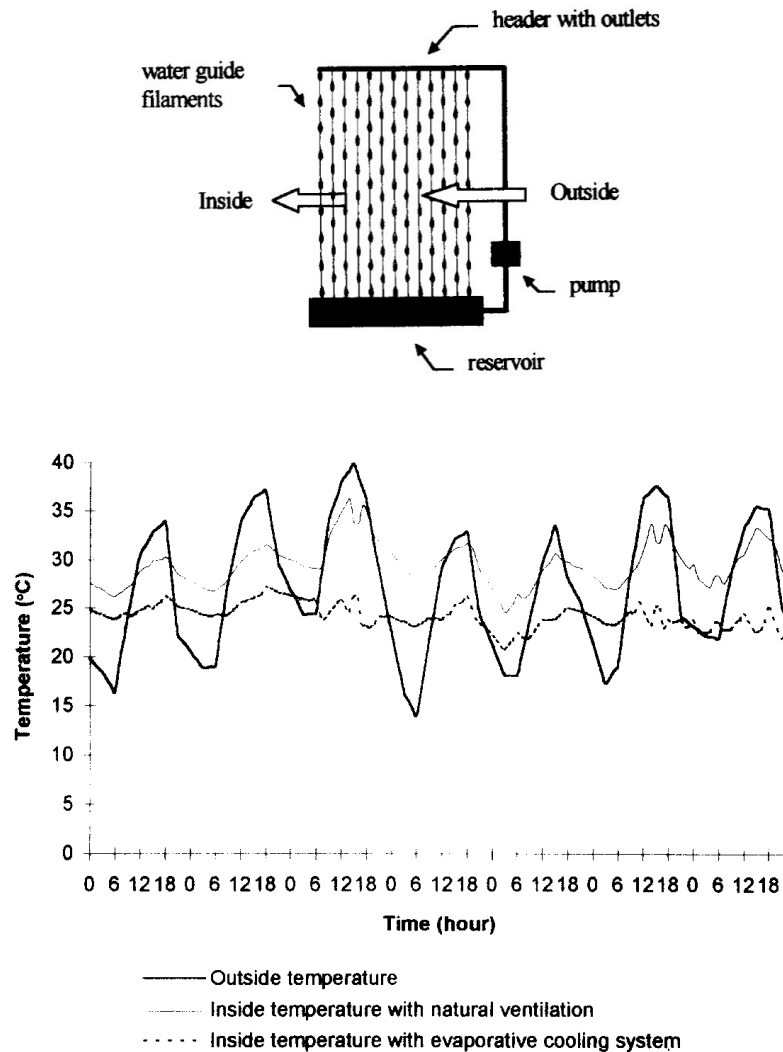


Figure 2.25 Evaporative cooling system diagram and impact of natural ventilation and evaporative cooling system on the indoor air temperature (Giabaklou and Ballinger, 1996, pp. 506, 504)

One of the most influencing and inspiring ceramic screen system at larger scale was the Spanish pavilion in Expo Zaragoza 2008 designed by Francisco Mangado. The project combined large ceramic columns and water to evoke a forest that created a cooler microclimate around the building. The vertical columns were covered by custom ceramic shapes and a misting system with micronisers arranged inside the columns kept the porous surface wet (Figure 2.26). Evaporation took place inside the columns as fan-driven air flowed downwards the inner ceramic surface, and the delivered air at ground level reduced the surrounding air temperature by about 4 to 7°C. Monitored data revealed that the ceramic surface temperature ranged from 25.2°C to 32.5°C when the ambient temperature and relative humidity was 34.9°C and 33.6% (Martin-Gomez and Mambrilla-Herrero, 2011). The results obtained from post-

occupancy evaluation concluded that a considerable local cooling effect was obtained and visitors could benefit from improved comfort conditions during the summer season. Despite the challenges faced in the application of an evaporative cooling system in a large project and not being fully passive, this example fulfilled the objectives of architectural integration of PCC systems and set a precedent for future applications.

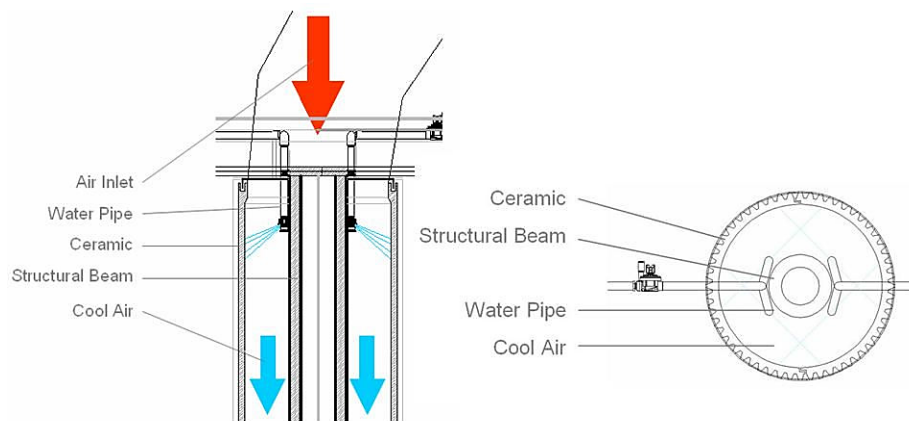
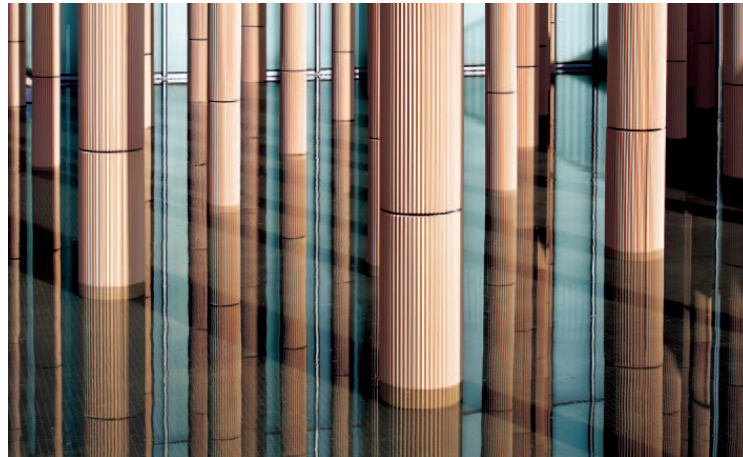


Figure 2.26 Porous ceramic columns in Expo Zaragoza 2008 and pillar section (Lanceta et al., 2007, p. 4)

A more recent example of a large-scale project integration is the Bioskin façade of Sony Osaki Building designed by Nikken Sekkei in Tokyo, 2011. Inspired by the traditional Japanese screens, Bioskin is a PCC system functioning as a handrail of the balcony on the northeast façade of the building that evaporates rainwater from porous ceramic pipes (Figure 2.27). The architects stated that ceramic surface temperature could be lowered by 10°C on the hottest day in the summer and that Bioskin could help reduce the temperature in the surrounding walkways and the entrance hall by 2°C (Sekkei, 2011). However, the integration

of PCC in a double skin façade has certain limitations. Firstly, occupants cannot benefit from the 2°C temperature drop in the surroundings for a long time as the immediate space is a narrow walkway in between the double skin façade. Secondly, although the temperature near the outer glass surface is also 1-2°C lower, the reduction on the building's total energy consumption is estimated by about 3% (RIBAJ, 2015). Lastly, rainwater only provides 50% of the water needed and mains supply is required. Despite water availability being not an issue in Japan (McDonald et al., 2014), large systems may become a major challenge when applied in water stressed countries. Despite the small impact on the thermal performance of the building, Bioskin's main contribution is the reduction of *heat island* effect. A recently published thermal image confirmed the cooling effect of the North-East ceramic facade relative to neighbouring buildings (Figure 2.28). The image confirmed that if similar facades were installed on several buildings in a neighbourhood, the cumulative effect would significantly reduce surrounding mean radiant temperatures at urban scale.



Figure 2.27 Bioskin NE façade. Sony Osaki Building, Japan (Nikken, 2011, p. 1)

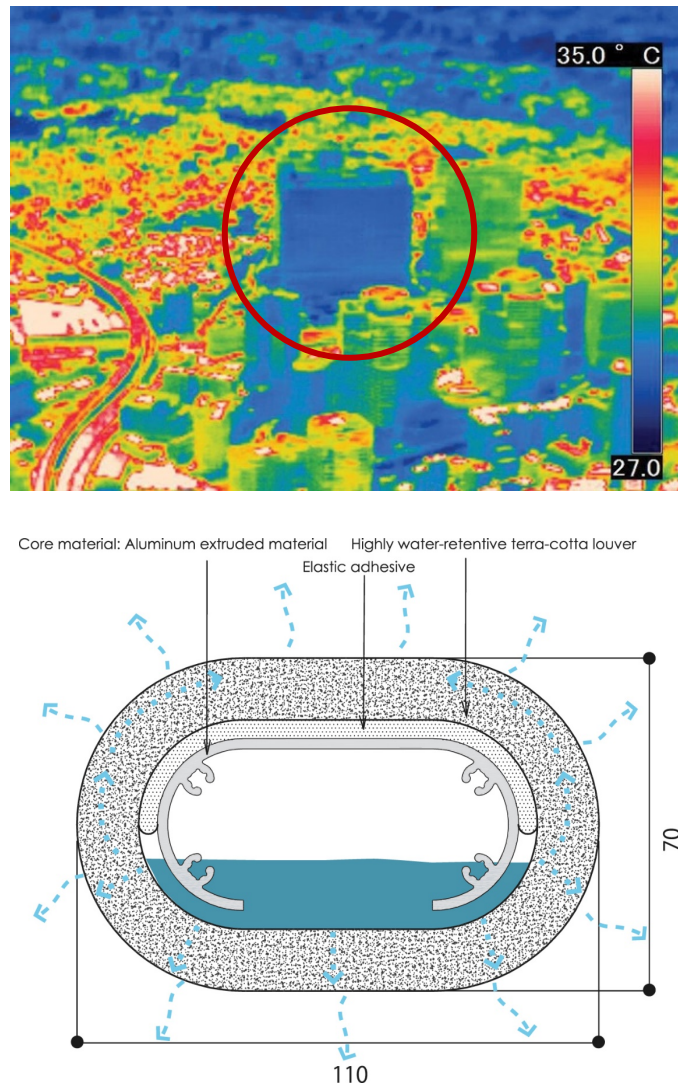


Figure 2.28 Aerial thermal image of the NE façade of Sony Osaki Building in Tokyo, Japan (RIBA, 2015) and Bioskin porous ceramic pipe section (Sekkei, 2011, p. 98)

Other studies undertaken by He and Hoyano (2010, 2011) in the Tokyo Institute of Technology, Japan, explored the application of ceramic evaporative screens in outdoor spaces as a strategy to mitigate urban heat island effect (Figure 2.29). Apart from monitored data from an experimental mock-up, this work contributed with the development of an optimised ceramic element with high soaking-up ability that allowed water to uniformly distribute over the ceramic surface by capillary forces. The extruded ceramic elements presented an extremely thin (between 5 and 50 μm) and unidirectional porous network presenting a 47% of porosity capable to absorb water upwards to a level higher than 1m.

The experimental mock-up consisted in 88 ceramic-pipe-wall distributed in four lines and connected to a water tank in its base. Monitored data confirmed that ceramics' surface temperatures dropped up to ambient wet bulb temperatures

and over 2°C air temperature reduction was recorded downwards the ceramic wall. The authors concluded that the system's cooling capacity was strongly dependent on air velocity and the estimated water consumption for 24 hours of operation was 1 l/day per ceramic element (Figure 2.30). Further studies from the same authors explored the impact of the ceramic pipe's arrangement on the system's cooling capacity and discovered that the first two ceramic lines contributed to more than 90% of the total cooling achieved. This confirmed that a larger number of ceramic lines does not necessary increase cooling capacity due to the obstruction to airflow created by the evaporators (He and Hoyano, 2011).

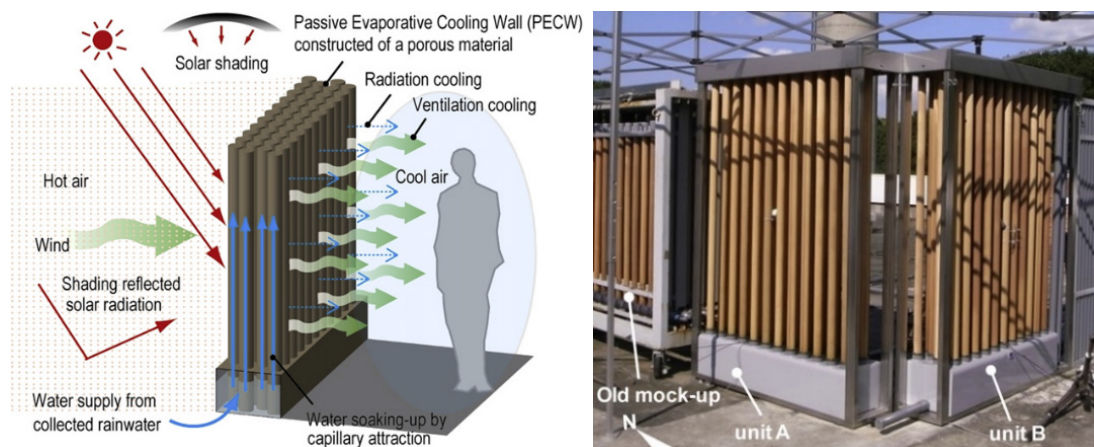


Figure 2.29 Evaporative cooling wall concept (He and Hoyano, 2010, p. 462) and experimental mock-up (He and Hoyano, 2011, p. 105)

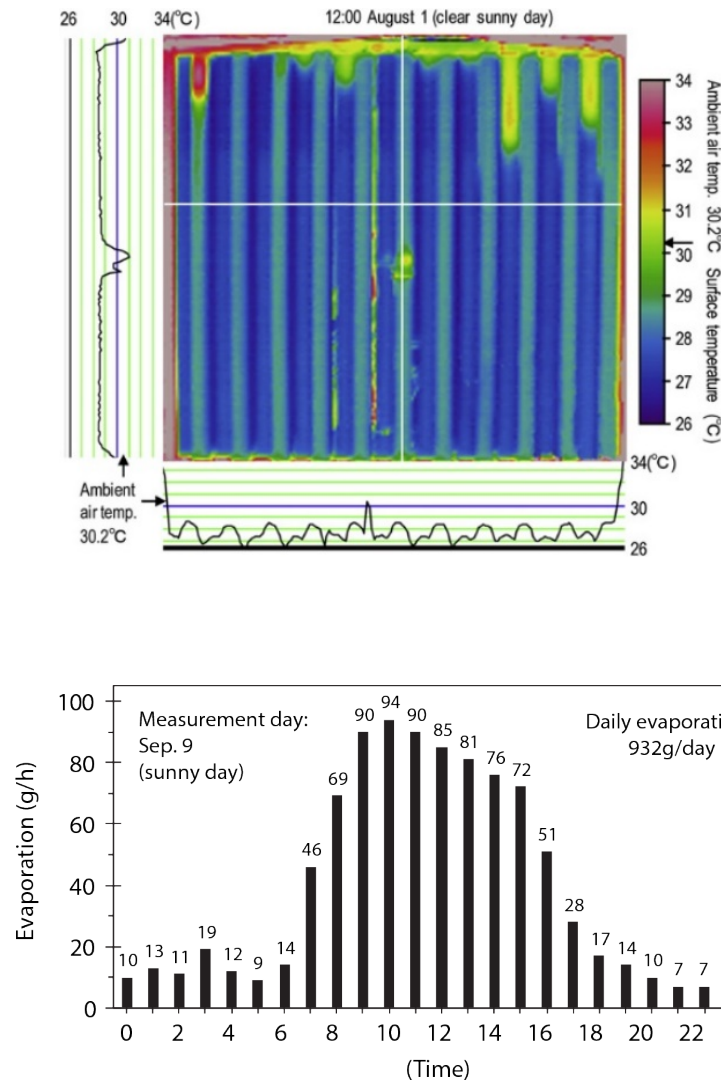


Figure 2.30 Daily evaporation rate profile for a ceramic pipe and wall surface temperature obtained from a thermal imaging camera (He and Hoyano, 2010, pp. 468, 469)

On the same year, a ceramic-screen cooling system named *Ecooler* and designed by the Israeli firm StudioKahn (2010) was introduced into market. The system, inspired by the traditional *mashrabiya*, consisted in hollow ceramic tiles filled with water and connected between each other conforming a vertical or horizontal surface (Figure 2.31). The ceramic element was crafted by using plaster moulds and burnt at high temperatures. The designers stated that each sample contained approximately 550 ml of water, and on average, half of this was emptied per day. This number suggested that the system may consume by about 2.4 litres/m² per day. In spite of the low water consumption reported, it's performance has not been rigorously tested in a laboratory environment and indications on the actual cooling capacity was only given by the manufacturer.

Still, the strength of this work relies on the modularity and flexibility of the system that ease the integration in both indoor and outdoor spaces.

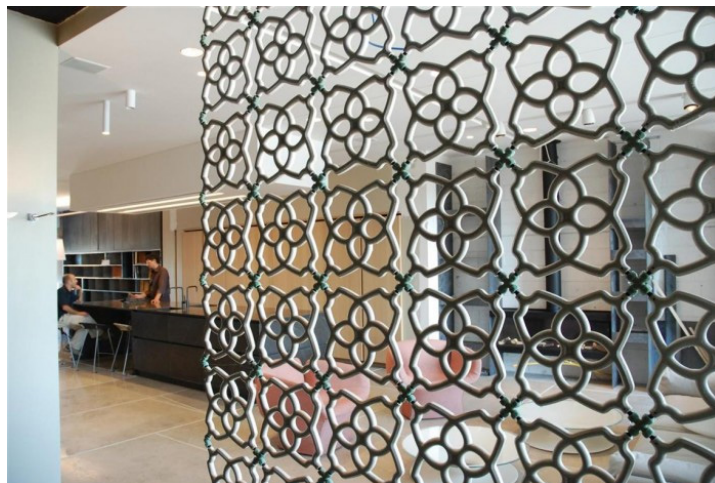
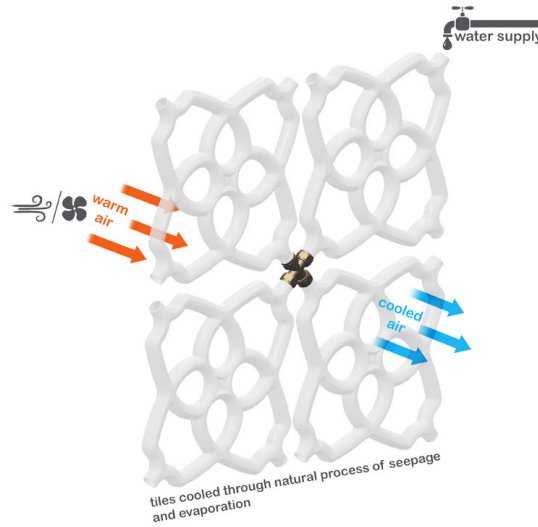


Figure 2.31 Ecooler concept and screen application in indoor spaces (StudioKahn, 2010, p. 1)

Roofs

The application of evaporative cooling methods in roofs mostly rely in indirect cooling systems. As a result, indoor spaces benefit from a reduced roof surface temperature derived from the latent heat absorbed during the evaporation process taking place at the outer face. One of the earliest studies on this technique was performed by Houghten et al. (1940), who evaluated the cooling capacity of a water layer formed on the roof by spray and an open roof pond. The collected experimental data concluded that both methods are equally effective and Sutton (1950) reported a 20°C outer surface temperature

reduction for a 50mm pond depth can be achieved as result of the evaporation process. Numerical models for cooling performance prediction of roof ponds were developed by Sodha et al. (1981) and more recent studies from Ben Cheikh and Bouchair (2004) and Spanaki et al. (2014) explored the efficiency of these systems when coupled with a radiative night cooling strategy. Monitored data from an experimental mock-up confirmed the effect of evaporation and also the high thermal capacity of the water pond: a 3.5°C peak indoor air temperature reduction and a lower indoor air temperature fluctuation was observed when compared to a building without a roof pond (Figure 2.32).

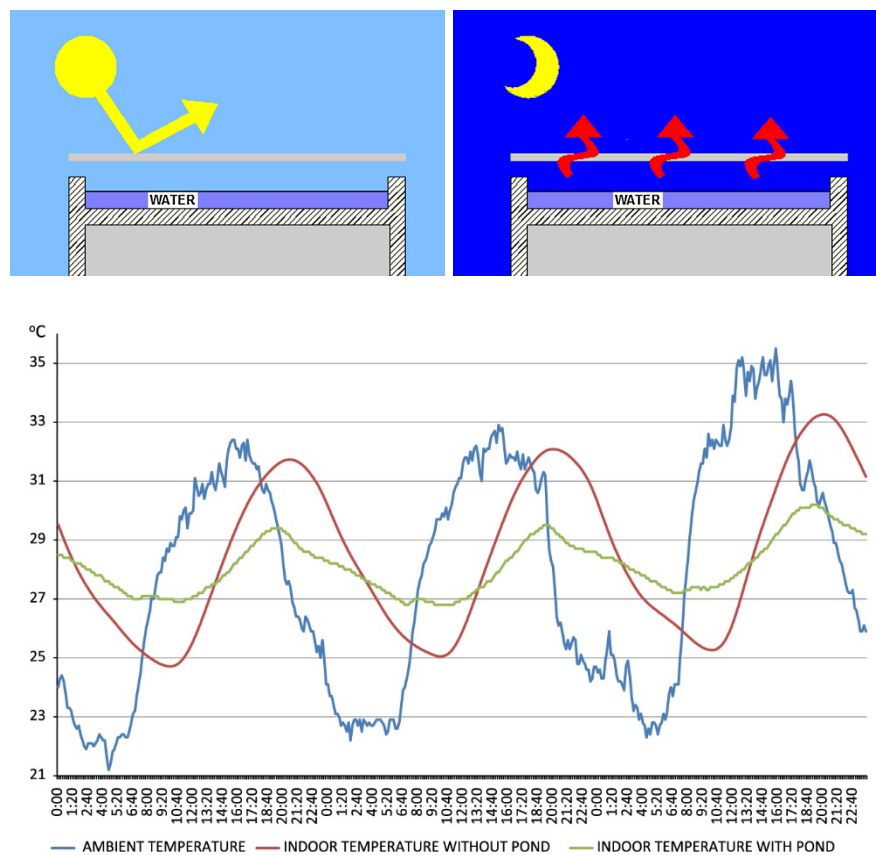


Figure 2.32 Ventilated pond with aluminium layer diagram and monitored ambient and indoor air temperatures from an experimental mock-up (Spanaki et al., 2014, pp. 274, 280)

2.2.2 Scope for Future Applications

The described systems and published data on the performance of these systems allowed the understanding and evaluation of the current stage of development of

convective and evaporative cooling systems. Most relevant outcomes from previous built examples and work are summarised in Table 2.1.

Table 2.1 Recent evaporative cooling systems and achievements for building application

	Author, Year	Cooling method	Zone	ΔDBT^1	ΔH^1	ΔTS^2	Water usage
<i>Towers</i>							
Experimental Building (Tucson, Arizona)	Cunningham and Thompson (1986)	Wetted pads	Indoor	-16.7°C	-	-	-
Shower Cooling Tower (Riyadh, Saudi Arabia)	Givoni and Al-Hemiddi (1995)	Water curtains	Indoor	-16°C	-	-	-
Rotonda Bioclimática (Seville, Spain)	Guerra Macho et al. (1994)	Misting system	Outdoor	-10°C	20 to 60%	-	-
Cool Towel (Negev Highlands, Israel)	Pearlmutter et al. (1996)	Misting system	Indoor	-10 to -15°C	-	-	0.5 to 2 m ³ /day
Nottingham H.O.U.S.E. (Madrid, Spain)	Ford et al. (2012)	Misting system	Indoor	-12 to -14°C	25 to 50%	-	40 l/day
PDEC Housing Stock (Turin, Italy)	Chiesa et al. (2017)	Misting system	Indoor	-8 to -11°C		-	0.65 l/h per column

Walls

Greenhouse (Almeria, Spain)	Franco et al. (2010)	Pad-fan system	Indoor	-8°C	-	-	146 l/m ² ·day
Greenhouse (Ankara, Turkey)	Dayioğlu and Sillelia (2015)	Pad-fan system	Indoor	-5 to -7°C	10 to 38%	-	-
Ceramic evaporators (Nottingham, UK)	Ibrahim et al. (2003)	Porous ceramics (building envelope)	Indoor	-6 to -8°C	30%	-	-

Evapcool (Sevilla, Spain)	Schiano-Phan (2004)	Porous ceramics (cavity wall)	Indoor	-5 to -7°C	-	-	5 l/day per person
Water-Evaporative Walls (Ancona, Italy)	Naticchia et al. (2010)	Porous ceramics (ventilated façade)	Outdoor	-5 to -10°C	-	-	-
Patio 2.12 (Madrid, Spain)	Terrados and Moreno (2014)	Porous ceramics (cavity wall)	Indoor	-5 to -10°C	-	-	-

Screens

Cooling Screen (New South Wales, Australia)	Giabaklou and Ballinger (1996)	Nylon lines (balconies)	Outdoors	-6.6°C	-	-	-
Spanish Pavilion (Zaragoza, Spain)	Martin-Gomez and Mambrilla-Herrero (2011)	Porous ceramics	Outdoors	-4 to -7°C	-	-	-
Bioskin (Tokio, Japan)	Sekkei (2011)	Porous ceramics	Outdoors	-2°C	-	-10°C	-
Evaporative Ceramic Screen (Tokyo, Japan)	He and Hoyano (2010, 2011)	Porous ceramics	Outdoors	-2°C	3 g/kg	-3 to -5°C	1 l/day per ceramic element
Ecooler (Ramot HaShavim, Israel)	StudioKahn (2010)	Porous ceramics	Indoor/Outdoor	-	-	-	2.4 l/m ² ·day

Roofs

Roof Pond (Crete, Greece)	Spanaki et al. (2014)	Water pond	Indoors	-3.5°C	-	-8 to -13°C	-
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¹ Achieved air temperature reduction and humidity increase in the surroundings of the system.

² Achieved surface temperature reduction of water or porous media.

Passive evaporative cooling has been applied in the past decades in conjunction with natural ventilation strategies for both indoor and outdoor spaces. Compared to other applications, evaporative cooling systems on large towers/stacks presented higher temperature drops by about 8 to 16°C, close to wet bulb

temperatures. These low temperatures are achieved by systems that maximise water surface exposure in semi-enclosed stacks usually excluded from internal/solar gains, radiant heat, facilitating the temperature drop of small air volumes. The subsequent temperature drop achieved in the adjacent occupied space is generally smaller and depends on the room dimensions, heat gains and the airflow rates exiting the tower. Collected data from ceramic screen cooling systems integrated the effect of the above mentioned internal conditions and lower air temperature drops by about 2 to 7°C were observed in the surroundings of the system. The associated humidity increase of the evaporation process can reach up to 60% as seen in *Rotonda Bioclimática* (Seville), and is strongly related to the size and efficiency of the system, which in the case of porous ceramic systems, does not necessarily increase with the number of porous elements, but with also a lower resistance to airflow around the ceramics. Moreover, the wide range of applications have contributed to different outcomes, validating the use of convective evaporative cooling systems for indoor space conditioning and building energy savings, localised improved indoor and outdoor comfort, and the mitigation of urban heat island effect.

The room for improvement is still high, and several components can be revisited or complemented with further research for system optimisation:

- Many of the developed evaporative cooling systems require electric equipment and maintenance. This is the case of the misting systems, requiring pumps and small air compressors demanding approximately 3.5kWh of electricity per day for an 8-nozzle system (Ford et al., 2012); and the pad/ceramic-fan systems in greenhouse structures, the *Spanish Pavilion in Zaragoza* or *Patio 2.12* (Franco et al., 2014, Martín-Gómez et al., 2009, Terrados-Cepeda et al., 2015). In the latter case replacing fan-driven ventilation options with natural ventilation strategies can create attractive and fully passive cooling alternatives to mechanical systems.
- Oversized porous surface area and ineffective ceramic arrangement to airstream have compromised water consumption and the overall efficiency of these systems (Ibrahim et al., 2003). Studies from He and Hoyano (2011) confirmed that the first two ceramic lines contributed to more than 90% of the total cooling achieved and further research and data is needed on additional ceramic arrangements and shapes in order to assist in the design process of novel systems.

- Developed evaporative systems using cast prototypes presented limited access to manufacturing industry, restricting their expansion to market (Schiano-Phan, 2004). Optimised ceramic prototypes designed in collaboration with ceramic manufacturers will extend the application of porous ceramic cooling systems in architecture while ceramic design guidelines can be provided to guarantee industrial manufacture.
- Alternative and more efficient water supply methods must be considered. Pumps and air compressors used to wet the porous surfaces of the *Spanish Pavilion* or the *PDEC Housing Stock Project* (Martín-Gómez et al., 2009, Chiesa et al., 2017) can be replaced with electricity-free systems as seen in *greenhouse structures* (Franco et al., 2014) and the *evaporative ceramic screen prototype* by He and Hoyano (2010).
- More resources are needed to widespread the knowledge acquired from these precedents. Research findings and databases must contain sufficient information for evaluating the applicability of each system in different contexts, identifying the opportunities and limitations and highlighting the space layout requirements for a successful integration.

2.3 Advances in Design Integration Methods

Progress and innovation is engraved in our culture. Professionals adapt workflows and integrate new techniques to remain in the leading industry. With the growing concern on climate change and the extreme weather conditions experienced in the last years (NASA, 2016, MetOffice, 2016), environmental design became one of the most demanded design approaches in contemporary architecture (Jankovic, 2012). Educators, sustainable building certifications and local authorities & regulations have contributed to the successful integration of passive design principles in architecture, but a higher contribution is expected from architects and designers. Despite the growing interest in building performance, the integration of performance evaluation techniques in the design process is not broadly considered and often is delegated to the specialist at later design stages or largely restricted to the analysis of a single design solution (Anderson, 2014). As result, the question shared by many initiated designers still remains: How to deliver climate-responsive architecture?

Environmental design rationale builds up from seeing buildings as living beings that breathe, feel hot and cold and need to respond to weather conditions.

Alexandros N. Tombazis pointed out that young architects need to move from the static traditional approach of architectural design to a dynamic approach that considers building behaviour and understands the laws of physics (Tombazis, 2009). This knowledge is indeed crucial and needed at early design stages, and a multidisciplinary collaboration and communication between other specialists such as environmental and acoustic scientists, landscape architects, interior designers, lighting designers and many other disciplines that take part in building projects must be achieved in order to solve all relevant issues simultaneously (Lopez Barnett and Browning, 1995).

The integrated design approach also benefits from maximised efficiency and performance while minimising effort and cost. One of the first references on the level of influence and project costs of design decisions dates back to the 70s by Boyd C. Paulson (1976) and Boehm (1981). The Integrated Project Delivery (IPD) curves generated by the authors are based on a self-evident observation: an architectural project becomes more difficult to change the more developed it becomes (Figure 2.33). Furthermore, HOK Chairman Patrick MacLeamy stated in 2004 the need of shifting design effort forward in the project in order to reduce the cost of design changes (CURT, 2004). Although MacLeamy's position is acknowledged, the real interest is the alternative (and more inspiring) interpretation that these curves offer: the need to shift the design effort forward in order to add value to the project and maximise the design impact on building performance.

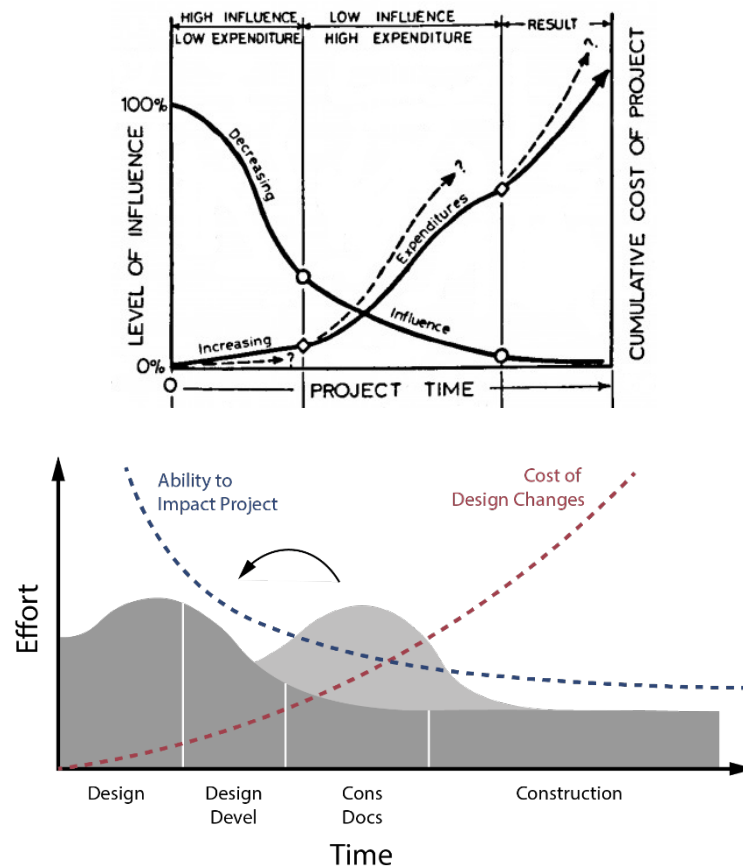


Figure 2.33 Paulson's IPD curve (Boyd C. Paulson, 1976, p. 588) and MacLeamy's IPD curve (After CURT, 2004, p. 4)

There are a wide range of resources currently available to swift the design curve forward. These includes books, manuals, video tutorials, online courses, support forums and simulation/analysis tools. With the rapid increase of computer power in the last years, complex multi-physics simulations can now be performed inhouse in a matter of minutes. Access to affordable subscription pricing or free and open source advance simulation software contributed to an increase in the number of practices that explore *performance* to obtain *form*. Despite the advantages of the current market scenario, not all of the available resources and simulation tools are intuitive for novel users, which often can lead to a dangerous scenario if appealing to the saying *there's no power without control*. Effective design for natural ventilation, daylighting, orientation, etc. requires early design decisions; thus, architects need intuitive, graphic and well enough designed tools for a successful integration during the design process. In addition, designers must meet certain minimum skills to test design options, including a basic understanding and knowledge on the limitations of the tool, and the capacity to abstract particular conditions. It is for this reason that simulation

tools can be classified according to the level of abstraction and complexity, leading to the so-called steady-state tools, dynamic tools and computational fluid dynamic (CFD) tools. This classification is of course proportional to the resolution of the digital model, the number of user inputs required, and the time consumption for a successful application. This classification also makes the application of these tools ideal for different design stages; while steady-state tools can provide with easy to interpret and quick design inputs to assist in decision making, dynamic and CFD tools provide with detailed daily or seasonal performance and visualizations of energy flows over a two-or-three-dimensional domain that help to refine the design at later and more advanced design stages. The contribution of design tools can be very valuable and they can inform on many aspects of site and building planning; for example, orientation, solar access, daylight, thermal comfort, energy use, etc.

2.3.1 Integrating Convective and Evaporative Cooling Techniques in Design

Convective and evaporative cooling techniques must be addressed at early design stages. Often, these systems require large volumes of air and controlled environments that conditions the space geometry. Moreover, their efficiency will be determined by climatic and internal conditions that should be taken into consideration sometime during the design process. The complex nature of these systems rises a common question among designers: how to integrate them in design? The answers to this question are perhaps arguable, but there are common actions that must be accomplished for a successful integration:

Feasibility Studies

These assessments evaluate the applicability of the system in a reduced time period with high reliability. The studies are based on climate data processing and visualization. Climate data provides with weather-averaged annual hourly values for each environmental parameter that describes the local climate. This robust piece of data can be easily interpreted and must be used in first instance to highlight the need and potential efficiency of the system. Hourly external air and wet bulb temperatures can be interrogated to evaluate the need for cooling and the opportunity for convective and evaporative cooling systems for the analysed period as described in Chapter 1.2 . In both indices, higher values would lead to higher applicability (Figure 2.34). Climate data can be manipulated using spreadsheets and data processing tools like the commercial software Microsoft Excel (Microsoft, 2016) or Apple Numbers (Apple, 2017). In addition, alternative

climate analysis tools are also available to guide novel designers in the climate evaluation process. Climate Consultant, developed by the Department of Architecture and Urban Design of University of the University of California, Los Angeles, records more than ten years of existence helping designers interpreting climate data and providing with customised building design guidelines (UCLA, 2017). Climate Tool, developed by Dr Petra Liedl from the School of Architecture at the University of Texas at Austin, US, allows a visual representation of most climatic parameters while highlighting the challenges and potentials of each climate (ClimateTool, 2016). The recently developed software package Ladybug Tools, highly supported by industry and academics, performs detailed analysis of climate data allowing the creation of customized, interactive visualizations (Ladybug, 2017).

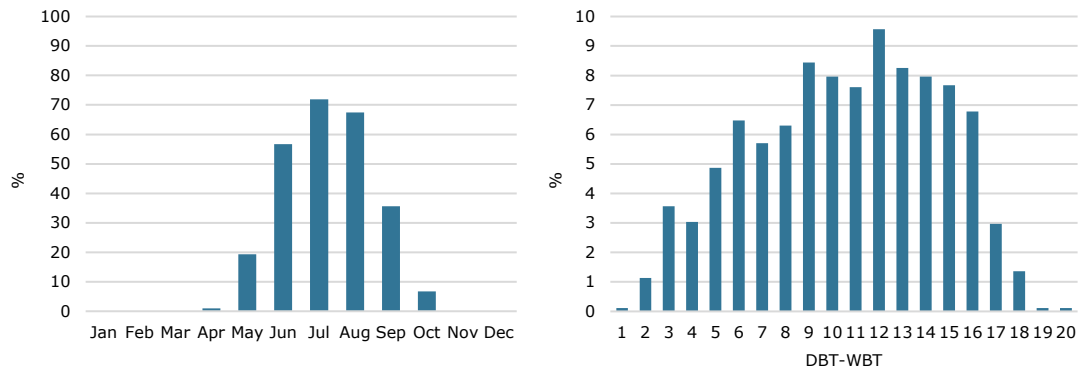


Figure 2.34 Frequency of occupancy hours (8am-6pm) when ambient air temperature is higher than 26°C (demand) and frequency of wet bulb temperature depression (opportunity) during the cooling season (May-September) at occupancy hours (8am-6pm) for the climate of Granada, Spain

Sizing the system

Most design processes start with simple pre-sizing methods based on rules of thumb and manual calculations. This applies to mechanical, electrical, and plumbing (MEP), structural engineering and many other fields in the building sector. When applied to convective and evaporative cooling systems, steady-state tools can provide with valuable information regarding the aperture areas needed for a natural ventilation strategy, or the dimensions required for an evaporative cooling system while estimating the performance. Steady-state tools

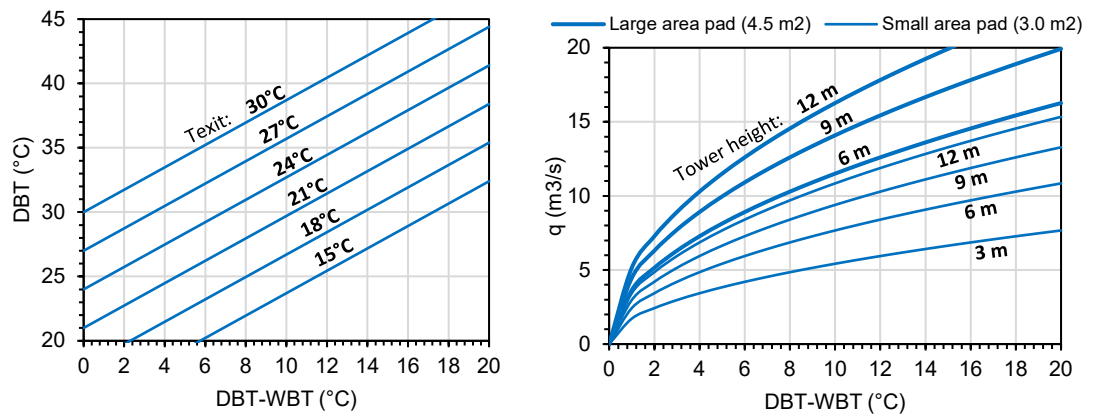


Figure 2.36 Exiting evaporative cool tower air temperature as a function of wet-bulb depression and outdoor dry-bulb air temperature, and recommended cool tower height and wetted pad area as a function of required airflow rate and wet-bulb depression (Kwok and Grondzik, 2007, pp. 153-154)

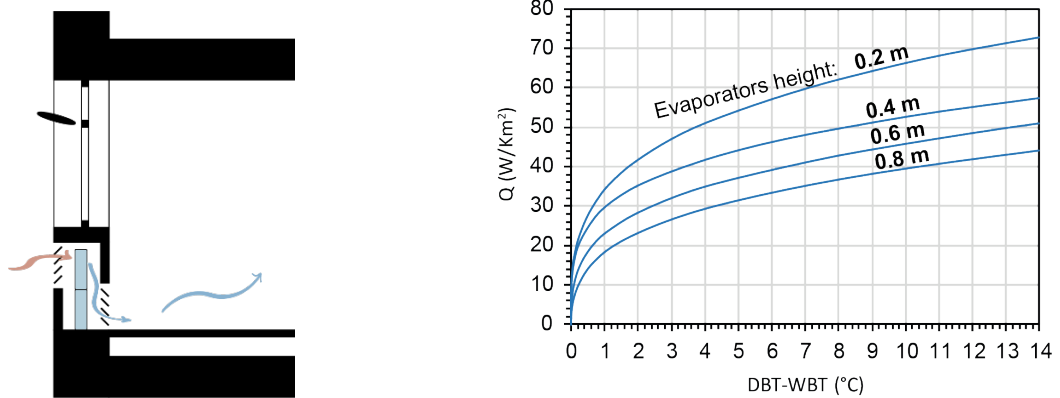


Figure 2.37 Specific cooling capacity of Evapcool systems per meter width and per meter height as a function of wet bulb temperature depression (Schiano-Phan, 2004, pp. 3-4)

Refining the System

As building design increases in complexity, the passive cooling system requires a comprehensive level of definition. The system can be refined using dynamic thermal simulation tools, able to evaluate building performance on an hourly basis and inform on achieved airflow rates and cooling delivered at different time periods and seasons throughout the year. Dynamic thermal simulations take into account internal conditions (in addition to the climatic conditions), and consider most heat exchange processes involved in building physics: convection, conduction, radiation and, partially, evaporation. Further, commercial software

like IES (IES, 2017) and ESDL Tas (EDSL, 2017) have recently integrated advanced & dynamic daylight-responding user controls to achieve a higher level of definition in thermal modelling. Others, like the extensively used and validated EnergyPlus simulation engine (EnergyPlus, 2017a) allow the customization of user inputs through the also open-sourced interfaces OpenStudio (OpenStudio, 2017) and Honeybee (Honeybee, 2017). The evaluation of convective cooling systems (natural ventilation) using dynamic simulations is especially effective as the buoyancy forces driving air into the zones are estimated for each hour of the analysis period while responding to the dynamic behaviour of the building. Wind force is also considered in most dynamic simulation software, but its integration in the calculation process is often simplified by means of typical wind pressure coefficients determined from the relation between the surfaces' normals of the building envelope, the meteorological wind speed and direction, and the terrain type (CIBSE AM10, 2005, p. 54). Although solution outcomes like air stratification and distribution within a three-dimensional domain are restricted, processed results can provide with valuable and detailed information regarding the dynamic performance of the natural ventilation strategy, like, for instance, peak, average and time of occurrence of airflow rates and airflow directions of simple or complex building geometries presenting one or multiple apertures. The integration of evaporative cooling systems is, however, limited, and the estimated cooling capacity or impact on building performance is strongly conditioned by the user inputs. The nature of the software does not conceive air temperature and vapour pressure gradients around the evaporators, and this may result in inaccurate models and misleading solutions. Still, there have been some attempts by several authors to integrate evaporative cooling systems in dynamic thermal models. Kang and Strand (2009) used Givoni's steady state model (Givoni, 1994) to predict exit air temperatures and air volume flow rates of a PDEC tower with pads for each hour of the analysis period (using climate data) and successfully input the resulting values as additional cooling supplied in the dynamic thermal model. Earlier, Schiano-Phan (2004) used a similar methodology using monitored data from the Evapcool system mock-up. This integration allowed the author to predict an annual energy saving of 1.5 MWh for a typical apartment from the building stock of Seville. Provided these simulation outputs are robust enough, the dynamic evaluation of evaporative cooling systems is still limited to systems that have been previously tested and experimental data is accessible for dynamic model integration.

When a deeper investigation is required, designers can opt for computational fluid dynamic (CFD) simulations. CFD tools allow the simulation of multi-physics

and multi-fluid interaction in a two-or-three-dimensional domain. It is for this reason that CFD have been applied to many fields involving prototyping design, such as mechanical engineering, electronics cooling, turbomachinery, etc. The nature of the software allows the user to practically solve any problem that could occur in reality, but the reliability of the results is strongly related to the skills of the user, its expertise and capacity to identify the physics involved and implement them in the software. In architecture, CFD has been used to visualise air movement in both indoor and outdoor environments. When applied to an urban scale model, designers can predict, for instance, air velocities and directions at pedestrian level, or accurately obtain wind pressure coefficients over the building envelopes for a detail calculation of building infiltration or wind-driven natural ventilation airflow rates. If applied to indoor environments, CFD allows the visualization of air stratification and the identification of typical airflow paths driven, in most cases, by thermal forces. Illustrations like Figure 2.38 can be useful to identify poorly ventilated zones or overheated areas in large space volumes. CFD tools can also be used to simulate evaporation processes with high level of accuracy, and many authors have successfully validated CFD models for difference evaporative cooling systems and techniques (Chapter 4.1.2). There are a wide range of CFD tools available in the market, among which are worth mentioning the commercial software Ansys Fluent (Ansys, 2017) and Comsol CFD (Comsol, 2017) for their broad integration in industry, being extensively validated and the extensive suite of products offered; Autodesk CFD (Autodesk, 2017) for its intuitive interface and easy integration with other Autodesk products such as AutoCAD and Revit; and OpenFOAM (OpenFOAM, 2017) as the open source alternative, also extensively validated and recently integrated with parametric modelling tools like Grasshopper for Rhino (Grasshopper, 2017) with the use of the plugin Butterfly (Butterfly, 2017).

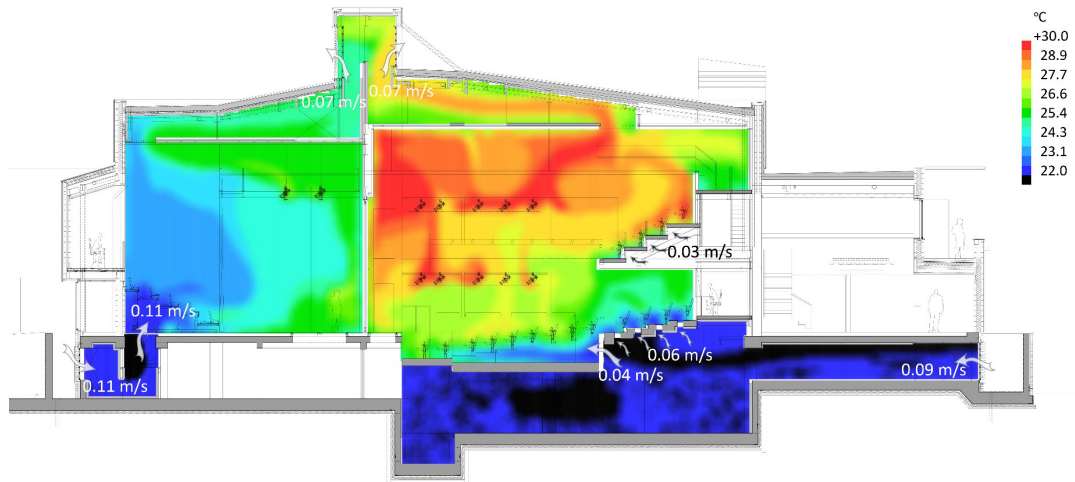


Figure 2.38 The Space. Nottingham Girls High School Theatre. Two-dimensional CFD simulation illustrating indoor air temperature distribution for a typical occupied warm day at noon. (Vallejo et al., 2015b, p. 8)

2.4 Design & Process Optimization of PCC Systems

The design and optimization of novel PCC systems for architecture is infrequent today. The default application of air conditioning systems in current practices interferes with the investment on alternative and innovative environmental friendly systems. Moreover, the need of continuous expertise inputs during the design process conflicts with the limited resources available, making it very difficult to enlarge the list on contemporary passively cooled buildings though evaporation.

A PCC system must be understood as the harmonic combination of an airstream, a ceramic evaporator, a water supply and space layout. Each of them should be carefully considered and efforts must be made to ease the integration and performance evaluation of every component by the inexperienced designer. In this research, the development of a novel PCC system includes a guided design process based on the principles of evaporation and convective heat transfer, monitored data and the provision of adequate performance prediction tools. Each PCC system design component (airstream design, porous ceramic design and PCC system design) is next introduced and further developed in the subsequent chapters.

2.4.1 Airstream Design

Most evaporative cooling applications are accompanied by a ventilation strategy. Precedents have demonstrated a wide range of examples: from draught air

movements driven by wind forces in evaporative cooling towers, to simple cross or stack ventilation driven by wind or buoyancy forces in the Muscatese screens and cool passages in Mediterranean houses. Although the airstream is not strictly required for evaporation to take place, it can considerably increase the evaporation rate and is strongly related to the efficiency of the system. Its major contribution, apart from the pressure generated on the water surface inducing the separation of water particles, is the capacity to replace saturated air with dryer air in the proximities of the evaporation surface, allowing further evaporation. Moreover, the ventilation strategy and airflow rates achieved will determine the boundaries of the conditioned space or, in other words, the area of influence of the PCC system.

Despite the recurrent use of electric fans in contemporary examples of PCC application to deliver the cooled air into the occupied zone, ancient techniques have demonstrated that fully passive systems are achievable. In the pursuit of developing a fully passive PCC system, the consideration of the principles of natural ventilation in the design of evaporative cooling systems is required, and so certain understanding on external and internal conditions and space geometry in order to propose the adequate ventilation strategy for the system. As previously suggested, the final objective is to achieve a constant airstream driving air through the evaporative surface using natural forces during the operational hours. In order to achieve this, and to estimate upstream air velocities, methods for natural ventilation feasibility studies and system sizing are reviewed in Chapter 6.

2.4.2 Porous Ceramic Design

The use of porous surfaces as elements to induce evaporation have been recurrent in the past. Among the good availability, durability and physical properties that characterise ceramic elements, precedents have also demonstrated the great flexibility offered for building integration. Emerging digital fabrication methods allow designers to create authentic pieces of art integrated in architecture that, at the same time, can improve the environmental conditions of a space, including the promotion of evaporation. The now frequent collaboration between manufacturers and designers bring new opportunities to create customised ceramic elements, making the ceramic design process an important stage of PCC system integration.

Shape can be optimised to enhance evaporation: the increase of surface area per unit of volume, and the reduction of resistance to airflow will considerably increase evaporation rate. Moreover, form and fabrication method determine water transport characteristics over the ceramic body. As seen in Chapters 1.1 and 2.2, ceramics may present a superficial or interstitial water flow that distributes water over the surface. The interstitial water transport can be optimised by means of special additives, controlled firing temperatures and manufacturing processes that improve the porosity and capillary network of the ceramic element. Superficial water transport can be optimised by geometrical forms that respond to the water supply method and facilitate the uniform distribution over the surface, in most cases, exploiting the gravitational force. Despite previous research have demonstrated the potential of special clay mixtures to increase the hydraulic capacity of ceramics (He and Hoyano, 2010, 2011), the chemical structure and customised fabrication processes are not explored in this thesis due to the deep expertise required in the area of chemistry and industrial manufacturing. Instead, superficial water transport is investigated through geometrical design options without any restriction on the clay mixture composition.

2.4.3 PCC System Design

In parallel to the airstream and ceramic design, a successful PCC design must address the system's integration in architecture, the water supply method, the arrangement of the ceramic elements and, finally, the size and location of the system. With the initial intention of emphasizing the artistic expression of ceramics, the system developed in this research can be described as a screen application with an aesthetical expression. Further, water supply is needed to keep the porous surface wet, and it can be achieved using different techniques, for instance, spraying systems, filled ceramic vessels or irrigation systems (Chapter 2.2). Efforts are put to explore and adopt a top irrigation system as the main water supply method, exploiting the gravitation force as a main driver to uniformly distribute water over the ceramic surface. The water flowrate, supply points and temperature are parameters to be considered in the design process. As most direct evaporative cooling systems, a collection and recirculation of water may be required, and a correct maintenance must be guaranteed to avoid the such well known issue of Legionnaire's growth (CIBSE TM13, 2013). Ceramic arrangement, as well as geometry, should contribute to the uniform water distribution over the porous surfaces. The size of the system and number of ceramic elements needed can vary depending on the user expectations, cooling

required and limitations in the space layout. Access to monitored data and sizing methodologies and tools are needed to assist in this process, and is further explored in Chapters 4, 6 and 5.

Finally, the ceramic system must be strategically located to condition the desired space while exposed to the airstream. If a buoyancy driven airstream is anticipated, the connection of two adjacent spaces presenting different air temperatures will drive the cool air to the warmer space, making the boundary between both spaces a desired location to place the PCC system. To maximise efficiency, the space layout and area of influence should also be carefully designed: excess of heat gains, exposure to direct solar radiation and poor user accessibility can lead to poor performance. These recommendations make the system suitable for semi-outdoor spaces, and ideally applicable to transitional spaces. Some application examples of PCC systems in architecture are reception spaces in atria, facades, or circulation areas around a central courtyard (Figure 2.39).

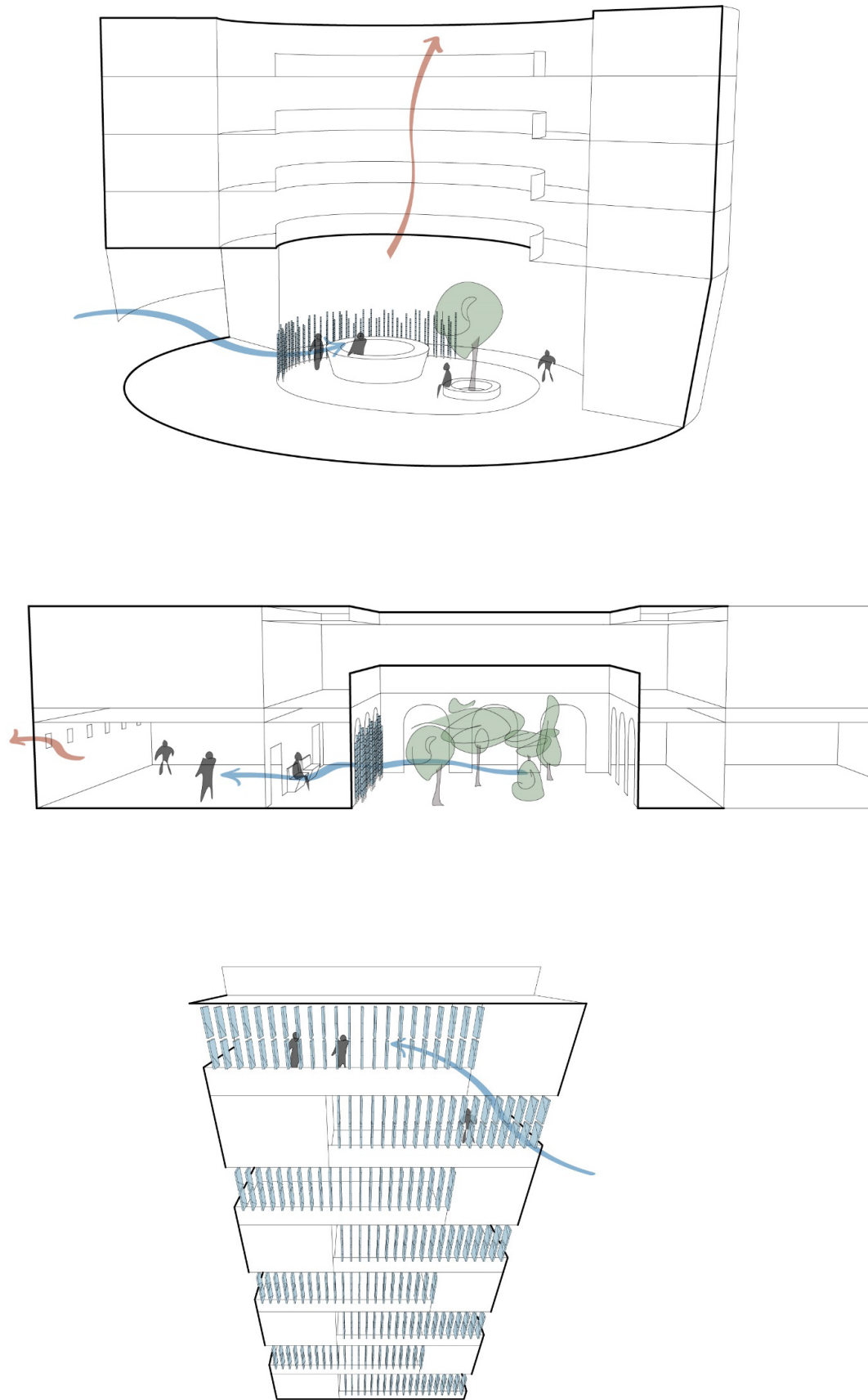


Figure 2.39 Application examples of PCC systems in architecture

2.5 Conclusions

A review on contemporary applications of ceramics and evaporative cooling systems confirmed the wide range of integration options in architecture. Advances in manufacturing techniques and emerging technologies enhanced and added new properties to ceramics (strength, durability photocatalyst capacity, etc.), and also allowed the materialization of complex geometries using additive manufacturing technologies. Despite the vast design opportunities of this technology demonstrated with several examples like the *cool brick* (Emerging Objects, 2015) or the *woven screen* (Friedman et al., 2014), there is still a long way to reduce the cost gap (currently by about ten times more costly than traditional methods) and improve surface quality and density (in order to minimise the risk of surface dimples and break after shrinkage) in order to compete with traditional manufacturing methods. Among the vast range of final products that can be obtained from clay, *terracotta* products present a typical range of porosities from 7 to 24% that make them suitable to absorb water and induce evaporation on their surface. Moreover, as the overall capacity to absorb water is also determined by the capillary network, ceramic design and fabrication method must be taken into account for the selection of the water supply system.

Evaporative cooling systems have been successfully integrated in contemporary architecture in towers/stacks, walls, screens and roofs. The main benefits from these techniques include indoor space conditioning and building energy savings, localised improved indoor and outdoor comfort, and the mitigation of urban heat island effect. Built examples and research work conducted in the past 50 years demonstrated air temperature drops by about 8 to 16°C in stacks/towers (70–95% of the incoming air's wet-bulb temperature) and by about 2 to 7°C in the surroundings of porous walls & screen systems. The exposure of the system to the conditioned space was found to be indicative of the cold air dispersion within the space and the area of influence of the system: reduced internal/solar gains and radiant heat will facilitate the temperature drop of small air volumes. Despite the successful progress, further research is still needed to make passive cooling systems appealing to a wider audience and to address the limitations found in previous work:

- Optimisation. A large number of the alternative evaporative cooling systems developed in the past decades require electric equipment and maintenance: pressure pumps are essential in misting systems and other applications are often assisted by fans to drive air through wet porous

media (Martín-Gómez et al., 2009, Terrados-Cepeda et al., 2015). In addition, oversized porous surface area and ineffective arrangement to airstream have compromised water consumption and the overall efficiency of these systems (Ibrahim et al., 2003).

- Limited access to market. Developed evaporative systems using cast prototypes presented limited access to manufacturing industry, restricting their expansion to market (Schiano-Phan, 2004).
- Alternative building integration options. Evaporative cooling systems have been successfully integrated in building geometry through towers, stacks, walls, screens or roofs (Ford et al., 2010), but the design opportunities offered are limited, lacking flexibility and often, presenting poor aesthetic properties.
- More resources are needed to fulfil the widespread knowledge gap in the principles of ventilation and evaporation among designers, and to assist in the design and integration of novel cooling systems in early design stages.

In addition, more resources to widespread the acquired knowledge is needed to successfully integrate novel systems in architecture. Access to knowledge and data interpretation is perhaps one of the main obstacles for most designers in order to shift the *design effort curve* forward. Fortunately, last decade has witnessed a growing interest in building performance, leading to a wide range of simulation tools accessible to designers. Tools and resources must adapt to the resolution of the model, thus simple pre-sizing methods based on rules of thumb and manual calculations are also needed to start the design process. While the physics of natural ventilation have been integrated in simple spreadsheet tools like the CIBSE AM10 Design Tool (CIBSE AM10, 2005) or the Multi-cell Ventilation Tools (Etheridge, 2012b), the tools to assist in the design and sizing of evaporative cooling systems rely on experimental data in form of tabulated data or diagrams that are limited to certain systems and dimensions (Kwok and Grondzik, 2007, Schiano-Phan, 2004), opening up a new research opportunities to contribute in the design of novel evaporative cooling systems.

3 Development & Performance Investigation of PCC Systems

Ceramic have been recurrent material to induce evaporation on porous surfaces. Built examples and previous work introduced in Chapter 2.2 demonstrated their capacity to reduce the surrounding air temperature by 2 to 7°C and surface temperatures up to ambient wet bulb temperature. Moreover, a wide range of ceramic designs have been proposed and successfully integrated in walls and screen elements. Despite the competitive performance of these systems, the geometrical characteristics of the ceramics used often derived from the assembly system, building integration constraints or additional features required (i.e. light filtering, shading, etc), and rarely responded to an optimization criterion based on the principles of evaporation.

This chapter describes the design process of a novel evaporative cooling system that combines the principles of convective and evaporative cooling with the inherited use of porous ceramics in architecture. The development of a Porous Ceramic Cooling (PCC) wall involved in first instance the design and manufacture of optimised porous ceramic prototypes that maximise cooling capacity and, in second instance, the design of alternative and efficient water supply methods that avoid the need of additional maintenance and operational cost. The process of optimisation focused in the geometrical characteristics of the ceramic element to demonstrate the considerable impact of ceramic design on the resulting cooling performance and hydraulic capacity.

This chapter also reaffirms the design criteria described in Chapter 2.4 by performing for the first time a comparative study of hydraulic and cooling capacity between different manufactured ceramic geometries used in architecture and the developed prototypes.

Hydraulic capacity studies explored water supply options and water transport over the ceramic surface. These included laboratory testing to determine porosity, as well as unidirectional and omnidirectional water absorption and distribution under different ambient conditions, ceramic manufacturing techniques and water temperatures.

Studies on cooling capacity explored the temperature reduction and humidity increase obtained downstream the ceramic samples. Parametric studies were

conducted to evaluate the impact of ceramic geometry on the cooling achieved and to identify recommended ceramic arrangements for the design of a larger scale system. The studies determined both evaporative and radiant cooling components of the system. Further investigations described the performance of an experimental mock-up exposed to the hot-dry ambient conditions of Granada, Spain, in order to determine the dynamic behaviour of the developed system.

3.1 Design Criteria

The process of evaporation can be explained as the separation of water molecules from a liquid structure to the air. As seen in Chapter 1.3.2, the amount of evaporated water depends of the surface area of water exposed to the airstream among other factors. Many of the emerged systems in contemporary application have applied this principle to increase evaporation, being misting systems the most extended application due to the increased surface area of water droplets achieved. On ceramic cooling systems, the water surface area is limited to the ceramic geometry, and optimised designs should aim to increase the ceramic surface area per unit of volume while minimising its resistance to flow.

Ceramic design can also improve evaporation rates by improving air flow on its boundary layer. When evaporation takes place on the ceramic surface, the vapour pressure on the proximities of the surface increases to saturation, stopping the evaporation process. Constant dryer air movement is needed to renew the saturated air over the boundary layer to allow evaporation. Air flow over the ceramic surface also responds to design, and aerodynamic shapes must be considered during the design process of optimised porous ceramic evaporators.

3.1.1 The Prototype P1

Prototype 1 (P1) is an ambitious design exercise that maximised the ceramic surface area per unit of volume, integrated water channels on his body and presented a low drag coefficient to improve airflow over its boundary layer. It relied on Algorithms-Aided Design (AAD) tools which allowed the creation of complex forms and associative models to explore multiple configurations through parametric design (Tedeschi, 2014).

This prototype also aimed to use 3d printing technologies to create a physical model that would allow further experimental studies to evaluate its performance for evaporative cooling applications. This exercise itself was an exploration on the opportunities offered by new design techniques and construction methods to take optimised design concepts a step further.

Concept design

The design of P1 can be described as a three-dimensional aerodynamic shape with an external envelope delimited by a spiral (Figure 3.1). It is based on an aerofoil-shaped section that was translated over the normal direction and scaled using multiple factors. A spiral-shaped channel was then applied from the top through the ceramic envelop and to the bottom of the delimited volume. This channel was designed to allow a uniform distribution and absorption of water over the ceramic surface, while the aerodynamic shape guaranteed low airflow resistance over the boundary layer. The engraved channels on the ceramic material also contributed to increase ceramic surface area.

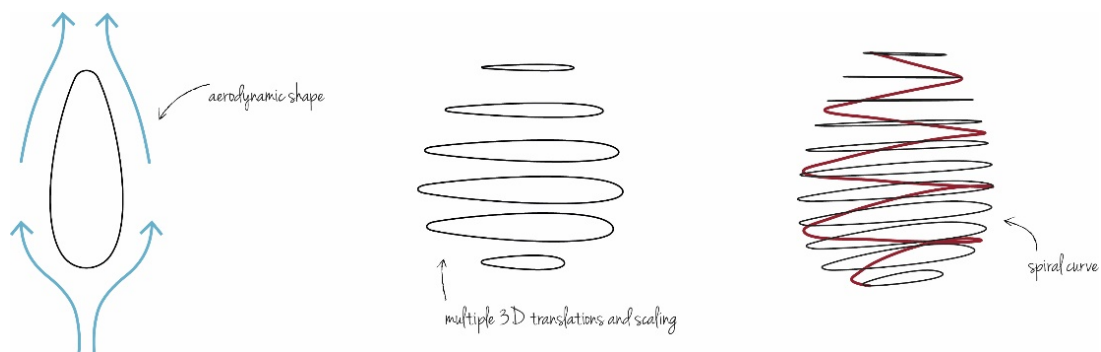


Figure 3.1 P1 design concept

Technical data

P1 was originally designed to be 3d printable with commercial 3d printers, limiting its dimensions to 200x200x50 mm (Figure 3.2). The achieved surface area is 0.253 m² and the equivalent effective surface area to volume ratio (ESA:V) is 234 m⁻¹. The aerodynamic shape of P1 resulted in a drag coefficient (Cd) of 0.52, calculated using the methodology described in Chapter 4.3. Its evaporative cooling capacity was explored through computational models and described in Chapter 4.4.2.



Figure 3.2 P1 Design drawings

Fabrication

A series of technical and economical limitations were found during the manufacturing process, preventing its fabrication. Despite the vast design opportunities 3d printing technology, the fabrication of P1 (and more complex geometries) was restricted by the nature of the ceramic material. The high amount of water content needed in the clay for 3d printing required, for this and most cases, adjustable moulds for drying that increased the risk of surface dimples and break after shrinkage during the firing process.

3.1.2 The Prototype P2

Following to the unsuccessful attempt to manufacture a physical model of P1, the Prototype P2 adopted a modest design approach. P2 adopted an extrusion method as the driving design parameter and applied wall treatments to increase its surface area and engrave water channels.

Concept design

The design of P2 consisted of an extruded circle generating a tubular shaped ceramic. The tubular shape contributed to reduce resistance to airflow over its boundary layer. A spiral shaped channel was also applied on the surface to uniformly distribute water from top to bottom of the ceramic surface. Multiple spiral angles and channels were designed to evaluate the hydraulic capacity of the different design options. By using auxiliary vertical structures, attached P2 pieces can define vertical columns conforming decorative ceramic screens coupled with a top drip irrigation system (Figure 3.3).

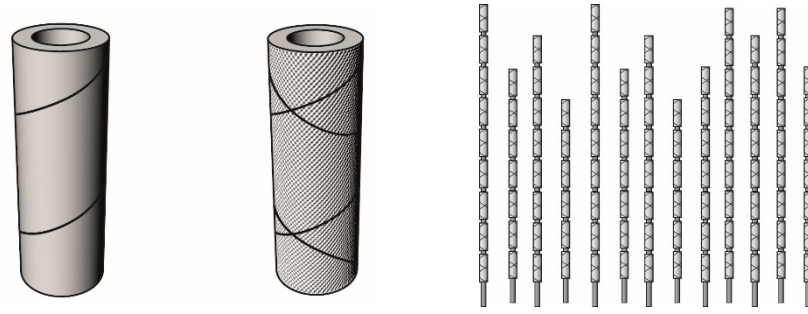


Figure 3.3 P2 ceramic design (left) and wall system (right)

Technical data

The design of P2 allowed different heights due to the extrusion manufacturing method. The final design presented a 64 mm external diameter and 220 mm high (Figure 3.4). The achieved surface area is 0.117 m² and the equivalent effective surface area to volume ratio (ESA:V) is 67 m⁻¹. The tubular shape of P2 presented a drag coefficient of 0.56. Its evaporative cooling capacity was also explored through computational models and described in Chapter 4.4.2

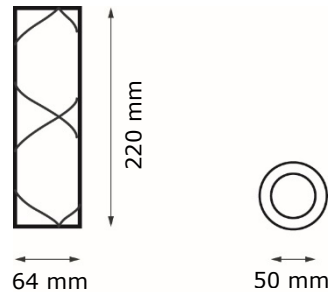


Figure 3.4 P2 dimensions

Manufacturing process

P2 was manufactured in April 2016 by Ceramica Decorativa Ltd in Spain. A circular mould was used to extrude the clay and generate the tubular shape. Prior to the drying process, the samples were placed over a textured surface and rolled over to increase external surface area and engrave the water channels. During this process, and due to the low water content of the clay mixture, it was observed that spikier textures were needed to texturize the ceramic surface. A drying process followed up for 24 hours and the samples were subsequently fired at 1050°C in one of the available ovens (Figure 3.5). Multiple samples we made

with different spiral angles and number of channels in order to determine the achieved the ceramic hydraulic capacity.



Figure 3.5 P2 manufacturing process

Room for improvement

Although the geometrical shape of Prototype 2 does not present a complex geometry, the manufactured samples presented good performances as ceramic evaporators (Chapter 3.3). The engraved channels succeed in transporting water uniformly over the ceramic surface, and the tubular shape presented uniform air distribution on its boundary layer (Chapter 3.2.2). The room for improvement for P2 resides in his surface area. A study was performed to evaluate the impact of a bumpy texture on the increment in surface area achieved considering different cell sizes and depths. Figure 3.6 confirms that textured patterns could considerably increase tubular-shaped ceramics' surface area by 20-30% (assuming a maximum texture depth of 3mm).

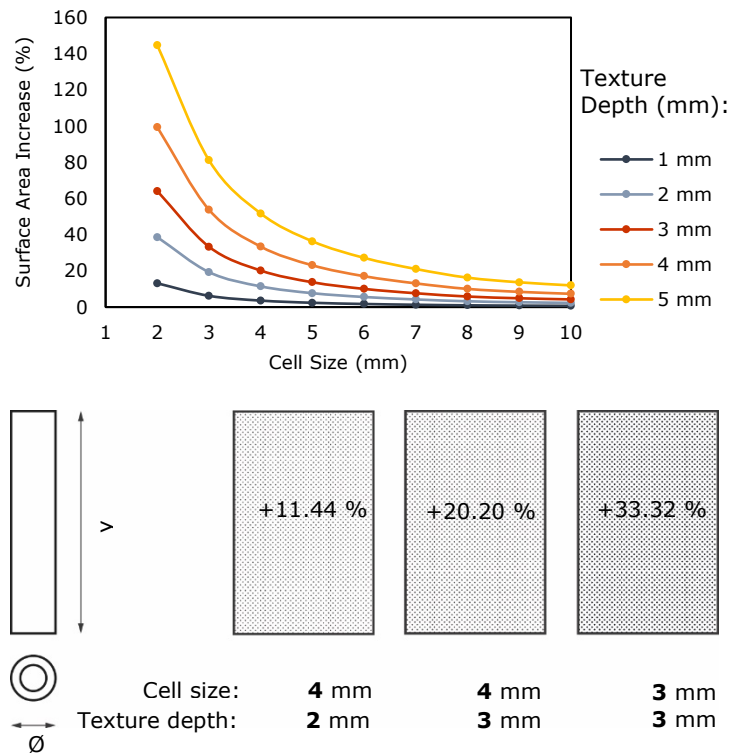


Figure 3.6 Impact of textured patterns on surface area

3.2 Hydraulic Capacity

Ceramic hydraulic capacity can be defined as the capacity of the ceramic sample to absorb and transport water through its pores. This property is essential to maintain a uniform water film on the ceramic boundary layer for evaporation. Ceramic porosity, internal capillary network and water supply methods are determining factors on the achieved water distribution, leading to a series of laboratory studies that evaluated the impact of each component.

Precedents have demonstrated that water can transport in the ceramic body in four different manners:

- Omnidirectional (outwards): The first evidences of using porous ceramics and water for cooling were found in Islamic culture, where water jars were placed on *muscatese* windows to promote evaporation (Schiano-Phan, 2004). The ceramic containers were filled up with water and exposed to airstream to cool down indoor and semi outdoor spaces. In these jars, water transports through porous network in all directions to

reach the external surface, where the evaporation takes place. The advantage of this method is the achievement of a uniform water distribution over the ceramic external layer. The limitations of the system are related to crafted fabrication techniques, limiting the design options and market expansion. The method, however, extended to North Africa, Spain and India and presented different applications like food storage, ambient conditioning and drinking water refrigeration (Cain et al., 1976).

- Unidirectional (Upwards): This method entirely relies on the capillarity network of porous ceramics. Water can transport from a base of water through the pores driven by capillarity forces to reach the external surface of ceramic. The velocity of water transport depends on the size of pores and its internal structure (Hall and Hoff, 2012), which is directly related to the manufacturing method. Extruded elements present a more prominent unidirectional network, whereas casting or pressed methods present more irregular structures depending on pressure and forces gradients applied during the manufacturing process. The hydraulic capacity of a ceramic evaporator using unidirectional water absorption methods thus depends on the specifications of the ceramic, clay composition, firing temperature and manufacturing process. Previous studies using high porous ceramics proved the velocity of absorption is reduced with the ceramic height, and rarely expands beyond 1 m high (He and Hoyano, 2010).
- Unidirectional (downwards): Based on the same principles, this method relies on a top-drip irrigation system. Water is supplied to the top of the ceramic and then absorbed through the capillary network. This method is broadly applied in greenhouses, where the top-drip irrigation system supplies water to cellulosic evaporative pads (Barzegar et al., 2012).
- Omnidirectional (inwards): This principle is based on the supply of water directly to the ceramic surface to achieve a uniform water distribution. It has been a recurrent method to wet the ceramic elements of the *Expo Zaragoza, Patio 2.12* or the *PDEC Housing Stock* through the use of misting systems (Chapter 2.2).

This section describes the laboratory studies conducted in order to evaluate advantages and disadvantages of unidirectional and omnidirectional water transport in ceramics. The results were used to identify effective water supply methods for conventional and optimised ceramic samples.

3.2.1 Unidirectional Water Transport Studies

The unidirectional water transport studies explored the capacity of ceramics to absorb water on the vertical direction using two different water supply methods: a water base and a top-drip irrigation system. The study involved the determination of the porosity of the ceramics, which were later related to the achieved hydraulic capacity. The selected samples have been integrated in architecture as double-skin façades, lattice walls or shelters, and were provided by Ceramica Decorative Ltd. They present different geometries, porosity and manufacturing processes (Figure 3.7).

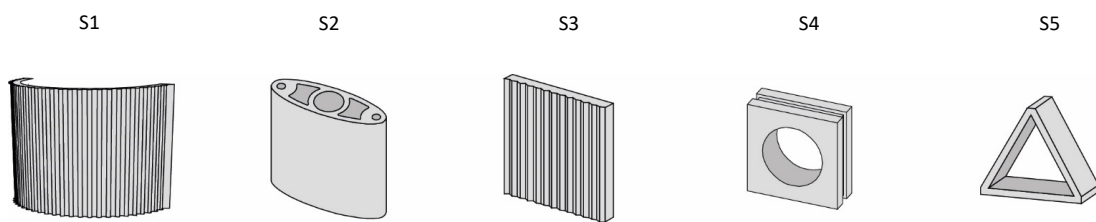


Figure 3.7 Porous ceramic samples used for hydraulic capacity test

Objectives

This experimental work aimed to obtain a broad understanding of the water absorptance of ceramic elements presenting typical ranges of porosity and manufacturing methods already integrated in architecture. The results aimed to identify the range in performance determined by these samples and optimised samples developed in previous work. The studies include:

- i. Measure of the porosity of the above-mentioned samples.
- ii. Evaluation of their unidirectional water absorptance, both upwards and inwards. The study aims to evaluate their speed of absorption and the time needed to keep their surface fully wetted. This data is crucial to size water supply systems and determine the required waterflow rates.

Methodology

In first instance, the porosity of each ceramic sample was determined following the Standard Test Method ASTM C20-00 (2015). The porosity expresses the relationship of the weight of water absorbed to the weight of the dry sample. In practice, the result is often reported as water absorption (WA):

$$WA (\%) = 100 \left(\frac{w_s - w_d}{w_d} \right) \quad \text{Equation (3-1)}$$

Water absorption
ASTM C20-00 (2015, p. 2)

where w_s is the dry state weight and w_d the saturated weight.

Commonly used methods to calculate porosity involve penetration of liquid into the pore. The most widely used techniques are gravimetric, based on weighting the material in a dry and a saturated state. The measurement of the dry weight was determined before saturation.

An electric oven was used to dry the samples. Two samples presenting the same design/geometry were introduced each time in the oven at 105°C and the dry weight was measured after 12 and 24 hours. This process was repeated for each sample, and a digital scale was used to weight the dry samples.

Next, a tea urn with 20 litres of volume capacity was used to immerse the samples in water. Two samples presenting the same design/geometry were introduced and entirely covered with water each time. The water was heated to boiling when connected to mains power supply. The samples remained in boiling water for 2 hours and then cooled to room temperature for 12 hours as suggested by the Standard. The rationale is that during heating to boiling, air originally within the pores expands and is flushed from the pore system by water vapour and steam generated within the sample. The digital scale was again used to weight the saturated samples (Figure 3.8).

The calculation of the porosity of each sample was done after both dry and saturated weights were determined using Equation (3-1).

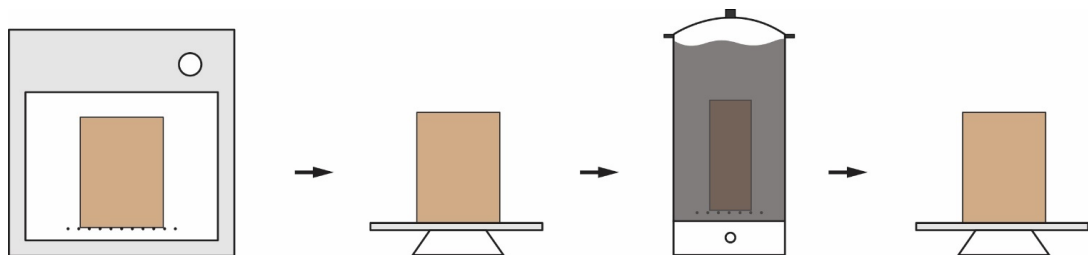


Figure 3.8 Methodology for porosity calculation

In second instance, a study explored the hydraulic potential of the ceramics by measuring the water penetration at different times with two different water supply systems (downwards and upwards). The measurements were done with a Testo 606-1 moisture meter (measurement range: 0-50%; accuracy: $\pm 1\%$) and the results were complemented with thermal images.

Two ceramic samples of each design/geometry were placed on a shelving unit and a water supply system released water to the top of the first sample and to the bottom to the second sample. The water penetrated downwards and upwards respectively. Each sample was placed on a plastic tray with a stainless-steel trivet to avoid any leakage in the system. The water was supplied through 50 mm tubes connected to a 5L plastic water tank placed at the top of the shelving unit. A peristaltic pump provided with pressure the dripping system and delivered a flow rate of 0.15-0.20 ml/s. Multiple 10 mm-distance measurements were taken with the moisture meter every 15 minutes for 4 hours (Figure 3.9).



Figure 3.9 Methodology for hydraulic capacity estimation

Results

The porosities of the ceramic samples were estimated using Equation (3-1) and are summarised in Table 3.1. Porosity ranged from 7.5 to 16.1 %. The results confirmed that the manufacturing method does not necessarily have a direct implication on the total porosity, as S2 (extruded) presents half the porosity of

S5, also extruded; being instead mostly dependant on firing temperatures and the drying process.

Table 3.1 Weight and porosity of porous ceramic samples

	dry weight (g)	saturated weight (g)	Porosity
S1	2836	3243	14.4
S2	2831	3043	7.5
S3	2068	2308	11.6
S4	1616	1840	13.9
S5	1530	1776	16.1

The studies on water absorptance presented different results for downwards and upwards unidirectional irrigation methods. The penetration of water varied in time, and a typical water penetration profile is shown in Figure 3.10.

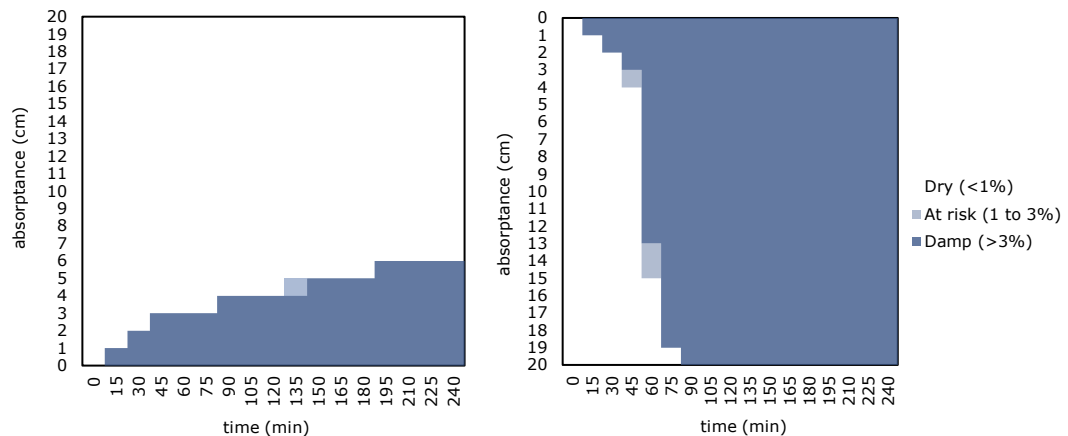


Figure 3.10 Typical upwards and downwards water penetration profile

The results presented a poor performance on upward absorptances when compared to the previous work conducted by He and Hoyano (2010), who suggested that optimised ceramic samples are capable to absorb water (upwards) to a level higher than 100 cm. After achieving a peak absorption rate of 3 cm in less than one hour, the rate dropped up to 1 cm per hour. After a four-hour test, water rarely expanded beyond 6 cm. For downward absorptances, water penetration followed a similar profile, penetrating 3 cm in the first hour. This indicated that gravitational force had a minimal impact on the

hydraulic capacity of the ceramics. Interestingly, when the absorption rate dropped after 1 hour, the ceramics were not able to absorb the delivered flow rate and water fell over the side surfaces of the ceramics. This explains the high damp levels recorded in the figure.

Differences in penetration speeds were also spotted over the ceramics. Upward absorptance profiles for all the ceramic samples are shown in Figure 3.11. The graph illustrates relatively small differences below 0.5 cm between each sample during the testing period, in exception of sample S4, which presented the worst performance. This confirmed that the main driving parameter for water penetration is the pore arrangement and capillaries, as S4 was the only non-extruded sample, presenting a non-unidirectional capillary network.

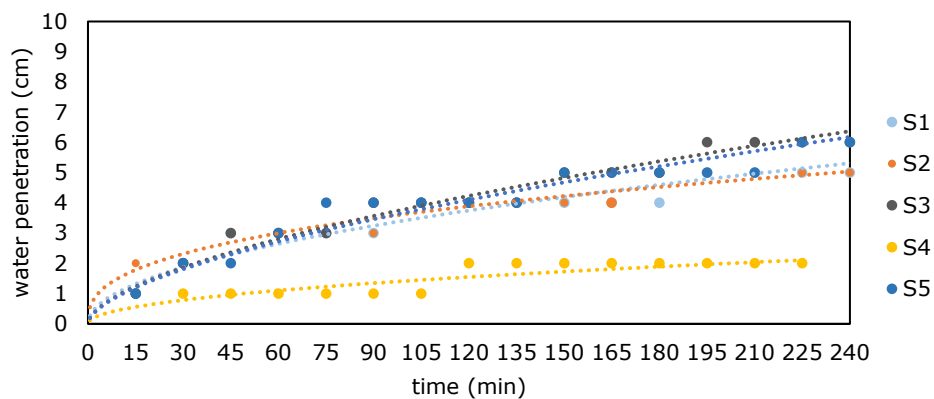


Figure 3.11 Water penetration in ceramic samples (upwards direction)

Conclusions

The results suggested that the porosities from manufactured ceramic elements used in architectural ceramic systems typically range from 5 to 16%. These values do not differ from the porosities associated to traditional fired bricks and terracotta products, typically ranging from 7 to 24% (Hall and Hoff, 2012). However, previous studies successfully manufactured high soaking-up ceramics presenting porosities up to 50% (He and Hoyano, 2010), highlighting the room for improvement in this field.

Interesting findings were also found during the hydraulic studies. First, it was observed a poor water absorption rate in the tested ceramics, presenting an average velocity of 0.7 mm/min. This value was reduced to 0.2 mm/min after the first 30 min. A total water expansion below 10 cm in four hours made the

unidirectional water absorption method not feasible for an evaporative cooling system using conventional ceramics.

The downward absorption method also presented similar velocities during the first hour, suggesting that gravitational force is not contributing to the absorption speed, being the capillary network the main driving force.

In addition, the top irrigation test suggested a potential good performance of omnidirectional water supply methods, as water expanded in all directions when slid over the ceramic surface. This observation led to the omnidirectional water transport studies described in next section.

3.2.2 Omnidirectional Water Transport Studies

The omnidirectional water transport studies explored the capacity of P2 to absorb water uniformly over its ceramic surface using a top-drip irrigation system combined with the engraved water channels. The channel networks collect the water from the top of the ceramic prototype and deliver it to the bottom, and empirical work was conducted to validate this method for different channel arrangements, environmental conditions and water temperature.

The results were used to identify the relation between the angle and the number of channels, and the percentage of wetted surface per unit of time. The data was also used to estimate the absorptance rate and hydraulic potential of the sample, determining factors of the water flow rate needed for the irrigation system.

A set of channels we engraved over the ceramic surface of P2 conforming one or two spirals with different angles to deliver water uniformly from top to bottom. The channels were engraved prior to the firing process and four different spiral arrangements were studied. The samples presented one and two spirals starting and ending at the same point, and two different spiral angles of 45 and 53 degrees as shown in Figure 3.12.

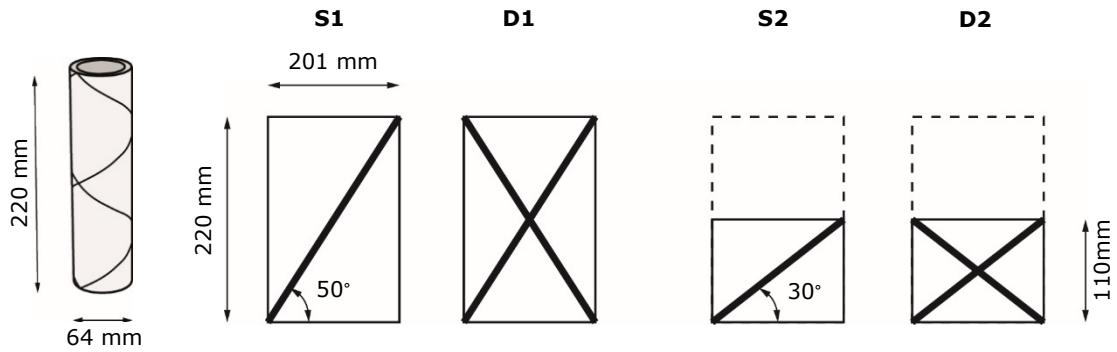


Figure 3.12 P2 samples and spiral arrangements

Objectives

The empirical work undertaken using P2 and its different spiral arrangements aimed to confirm the suitability of this water supply method for an efficient decorative cooling wall. The objectives of this study are described as follows:

- i. To confirm the effectiveness of the spirals, capable to collect water from the top and deliver it to the bottom throughout the spiral-shaped channel.
- ii. To achieve a uniform distribution over the ceramic surface.
- iii. To understand the impact of different environmental conditions (air temperature and relative humidity) on the hydraulic capacity of the system.
- iv. To understand the impact of water temperature on the hydraulic capacity of the system.
- v. To estimate the percentage of wetted surface achieved per unit of time for different spiral arrangements.
- vi. To estimate water absorptance and minimum water flow rate needed for the irrigation system.

Methodology

The work undertaken started with the definition of the dry, saturated weight, and the porosity of P2. Secondly, the samples were exposed to a top drip irrigation system and water penetration was measured over time. The results were post-processed to estimate water absorptance per unit of time for the different spiral arrangements. This process was repeated to evaluate the impact of environmental conditions and water temperature on the hydraulic capacity of the samples.

Three sets of parametric studies were performed to address the described objectives. The first set of parametric studies evaluated hydraulic capacity of the

four different spiral arrangements under constant air temperature, relative humidity (DBT-WBT=9°C), and a water temperature set to 20°C (Figure 3.13).

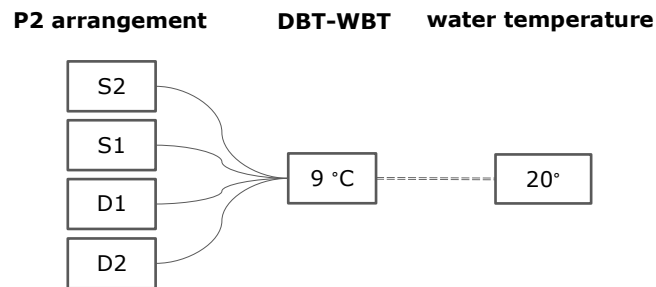


Figure 3.13 First set parametric studies

The second set of parametric studies evaluated the impact of four different environmental conditions presenting a dry bulb-to-wet bulb temperature depression of 8, 9, 10 and 11°C. The water temperature during the test was set 20°C (Figure 3.14).

The third set of parametric studies evaluated the impact of water temperature on hydraulic capacity. Water is supplied at 20,30 and 40°C on the ceramic Prototype P2 under constant environmental conditions of DBT-WBT=8°C.

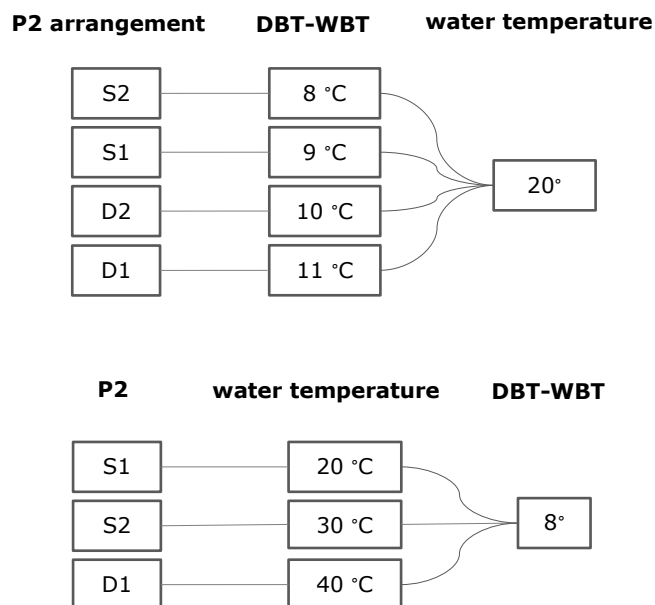


Figure 3.14 Second (top) and third (bottom) set parametric studies

Firstly, the porosity from the new ceramic samples is estimated using Equation (3-1) after weighting them under dry and saturated conditions. The ceramic samples were later exposed to a top-drip irrigation system delivering a mean flow rate of 0.02 ml/s. The spiral channels distributed water along the ceramics' surfaces, which was subsequently absorbed by the porous material. The same laboratory setup described in 3.2.1 was used to undertake these studies.

Water expansion over the surface was measured perpendicularly to the channel direction every 30 minutes over a period of time of 120 minutes for the first set of parametric studies, and every 5 minutes over a period of 20 min for the second and third set of parametric studies (Figure 3.15).

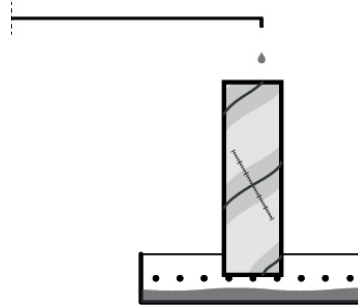


Figure 3.15 Experimental set-up for P3 hydraulic capacity

Results

The porosity studies reported that prototype P2 presented an average porosity of 15%, in line with the range of porosities of the samples studies in 3.2.1 . A summary of the collected data is shown in Table 3.2:

Table 3.2 P2 weights and porosity

	height (m)	volume (m3)	dry weight (g)	saturated weight (g)	porosity (%)
S1	0.22	0.00027	521	600	15.09
S2	0.13	0.00016	305	351	15.02
D1	0.22	0.00027	510	587	15.07
D2	0.11	0.00014	262	301	14.81

Different spiral arrangements were tested under a constant water flow rate of 0.2 ml/s, a constant DBT-WBT of 9°C, and constant water temperature of 20°C. The water expansion perpendicularly to the channel direction over the time is summarised in Table 3.3:

Table 3.3 First set studies results

	time (min)	absorption (mm)	ceramic weight (g)	ΔP (g)
S1	0	0	521	0
	30	20	551	30
	60	32	563	42
	90	45	572	51
	120	saturated	584	63

S2	0	0	300	0
	30	22	322	22
	60	33	335	35
	90	51	341	41
	120	saturated	348	48

D1	0	0	511	0
	30	21	540	29
	60	32	556	45
	90	50	568	57
	120	saturated	574	63

D2	0	0	262	0
	30	21	286	24
	60	32	293	31
	90	50	300	38
	120	saturated	304	42

The results of the first set of parametric studies confirmed the efficiency to the spiral-shaped channels as water distributors over the surface. Water expanded along the length of the spiral in all directions as shown in Figure 3.16. Moreover, it was observed that 100% of the ceramic surface was wetted before 120 minutes. The velocity of absorption, however, presented an identical profile as the previously tested ceramic samples, presenting a peak absorption rate during the first 20-30 min.

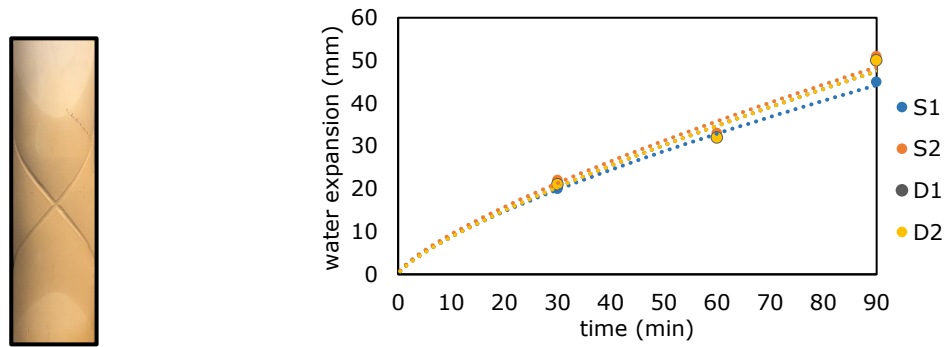


Figure 3.16 Photograph of water expansion in P2-S2 (left) and water expansion over the time for P2 samples (right)

After 90 minutes, water expanded between 40 and 50 mm in all directions (Figure 3.17). The results also suggested the minimal impact of the spiral angle on the absorption speed, as the observed difference remained below 1 mm after 60 minutes. The equal expansion of water above and below the channel confirmed the secondary role of gravity force in the ceramic water absorption.

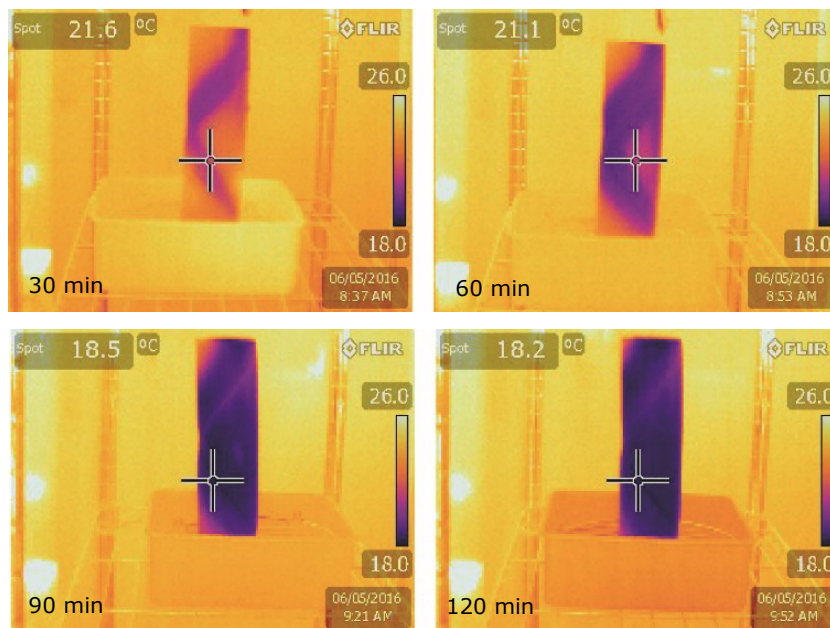


Figure 3.17 Thermal images of ceramic water absorptance in time

The methodology was repeated for the second test of parametric studies, testing the hydraulic capacity of P2 samples under four different environmental conditions, with DBT = 30,32,34 and 36°C, and a RH of 48,38,41, and 44%

(DBT-WBT = 8,9, 11 and 12°C respectively). The results obtained are summarised in Table 3.4.

Table 3.4 Second set studies results

DBT-WBT (°C)	time (min)	absorption (mm)	weight (g)	ΔP (g)
8	0		307	22
	5	8		
	10	11		
	15	15		
	20	17	329	
9	0		521	23
	5	8		
	10	12		
	15	15		
	20	17	544	
10	0		263	18
	5	7		
	10	11		
	15	14		
	20	17	281	
11	0		512	26
	5	7		
	10	10		
	15	13		
	20	16	538	

The ceramic samples presented a similar hydraulic capacity and absorptance profile independently of the environmental conditions. The four samples achieved an average water expansion of 17 mm in 20 minutes' time, following the same progression as obtained in the first set of parametric studied. The maximum difference in water expansion observed during the analysis period was below 1 mm (Figure 3.18).

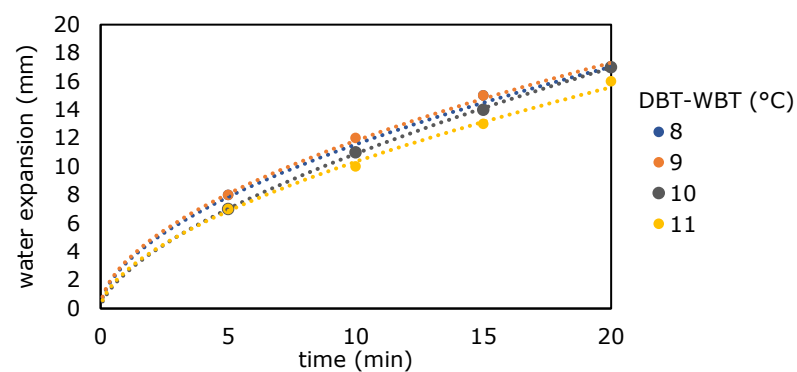


Figure 3.18 Water expansion in time under different DBT-WBT conditions

The same methodology was used to test different supplied water temperatures from the irrigation system. The water absorptance of ceramics were tested with water at 20,30 and 30°C under a constant DBT-WBT of 9 °C. The results obtained are summarised in Table 3.5.

Table 3.5 Third set studies results

H2O T (°C)	time (min)	absorption (mm)	ceramic weight (g)	ΔP (g)
20	0		520	10
	5	8		
	10	12		
	15	14		
	20	16	530	
30	0		301	20
	5	8		
	10	12		
	15	14		
	20	16	321	
40	0	0	511	37
	5	7		
	10	9		
	15	11		
	20	13	548	

The ceramic samples presented the same hydraulic capacity and absorptance profile as the previous studies when water temperature was between 20 and 30°C. At 40 °C, the hydraulic capacity of the sample presented a small reduction between 2 and 3 mm in the first 20 minutes (Figure 3.19).

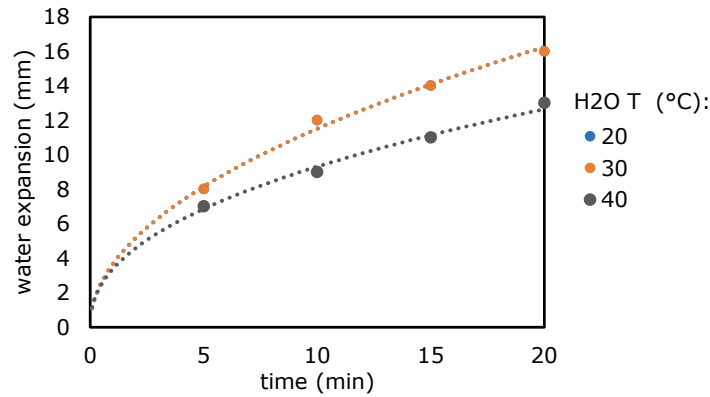


Figure 3.19 Water expansion in time under different water temperatures

Conclusions

The hydraulic studies conducted with prototype P2 presented similar water absorption rate (0.6 mm/min) to the previously tested ceramic samples, presenting a peak absorption rate during the first 20-30 min. This absorption rate desponded to an average measured porosity of 15% and unidirectional capillary network, also equivalent to the samples S1, S3 and S5 tested in 3.2.1.

The omnidirectional irrigation system presented a good water distribution over the ceramic surface, achieving 100% of surface saturation before 120 minutes of analysis period for a 0.02 ml/s of supplied water flowrate. It was also observed that the ceramic surface absorbed water from the channel without any visible impact of gravitational force.

Variations in air temperature and relative humidity presented a neglected impact on the hydraulic capacity of P2, and a reduction of water absorption rate by 0.15mm/min was observed only with water temperatures above 40°C. As high-water temperatures from mains water supply or water wells are considered an extraordinary scenario, it can be concluded that the impact of typical ambient conditions and water temperatures on the water absorption rate of porous ceramic *terracotta* elements presenting an unidirectional capillary network can be neglected.

The average water expansion of the tested prototypes can be expressed in mm as function of time (in minutes) as shown in Equation (3-2) and Figure 3.20.

$$d \text{ (mm)} = 2.7034T^{0.618}$$

Equation (3-2)

Average P2 water expansion

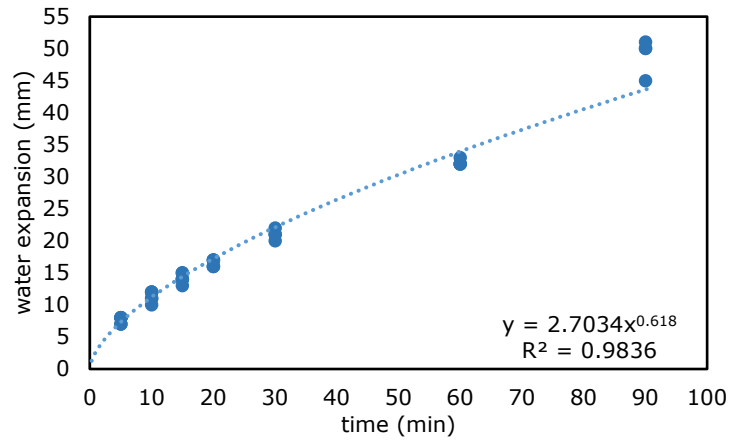


Figure 3.20 Water expansion in time under moderate hot-dry conditions

This data can also be expressed as the percentage of wetted prototype P2 surface area. The percentage of P2 wetted surface area depends on the diameter of the sample, the angle and number of spirals, and the water expansion in mm (Figure 3.21).

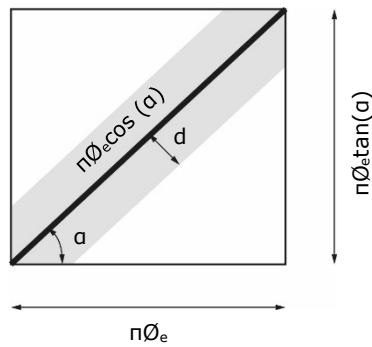


Figure 3.21 P2 geometric values

Using this geometrical relation and Equation (3-2), the percentage of wetted surface area per unit of time can be obtained for one Equation (3-3) and two spirals Equation (3-4).

$$WA (\%) = \frac{5.4068T^{0.618}}{\pi\phi_e \cos(\alpha)\tan(\alpha)} - \frac{7.3084T^{1.236}}{\pi^2\phi_e^2 \tan(\alpha)^2} - \frac{7.3084T^{1.236}}{\pi^2\phi_e^2} \quad \text{Equation (3-3)}$$

*P2 wetted surface area
(1 spiral)*

$$WA (\%) = \frac{10.8136T^{0.618}}{\pi\phi_e \cos(\alpha)\tan(\alpha)} - \frac{14.6168T^{1.236}}{\pi^2\phi_e^2 \tan(\alpha)^2} - \frac{14.6168T^{1.236}}{\pi^2\phi_e^2} - \frac{29.2336T^{1.236}}{\pi^2\phi_e^2 \tan(\alpha)} \quad \text{Equation (3-4)}$$

*P2 wetted surface area
(2 spirals)*

For a diameter of 64 mm, as in P2, the percentage of wetted surface achieved over the time is shown in Figure 3.22 and Figure 3.23:

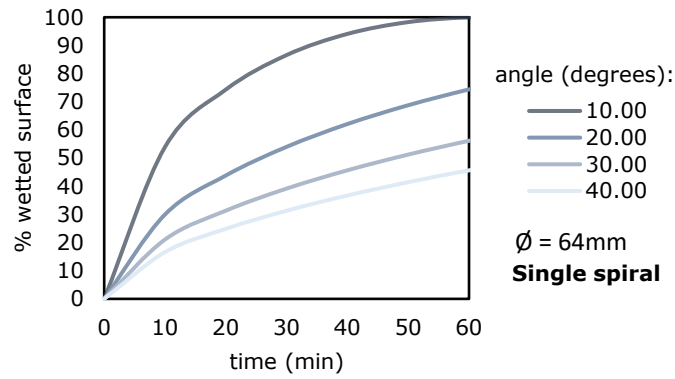


Figure 3.22 Percentage of wetted surface in time (1 spiral)

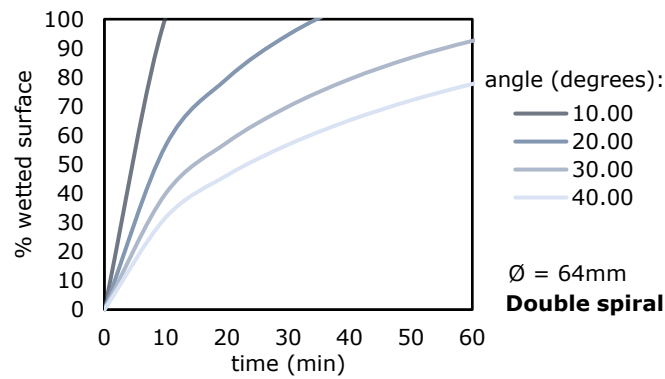


Figure 3.23 Percentage of wetted surface in time (2 spirals)

This ratio can be used for future predictions on the time needed to fully wet the surface of a tubular-shaped design. Double spirals with angles below 40 degrees can wet more than 80% of the total surface area in less than 60 minutes.

Moreover, the total amount of water absorbed by the ceramic, is also proportional to the percentage of wetted surface, its volume and density. Assuming an average porosity of 15%, a density of 1900 kg/m³, and an omnidirectional water transport, the amount of absorbed water per ceramic sample with one and two completed spirals is:

$$P_{H2O} = WA0.15V\rho \quad \text{Equation (3-5)}$$

Water absorbed in P2

In Equation (3-5), V can be expressed as function of the diameter and spiral angle:

$$V = \frac{\pi^2 \phi_e \tan(\alpha) (\phi_e^2 - \phi_i^2)}{4} \quad \text{Equation (3-6)}$$

P2 volume

These equations were adapted to prototype P2 dimensions, and the amount of water absorbed per meter of ceramic element in time is shown in Figure 3.24 and Figure 3.25.

This absorbed water must be the minimum flow rate delivered to 1m of ceramic column to ensure the predicted percentage of wetted surface is achieved within the predicted time. In this example, a recommended average flow rate of 0.35 and 0.45 l/h must be delivered with the top irrigation system for single and double spiral design respectively.

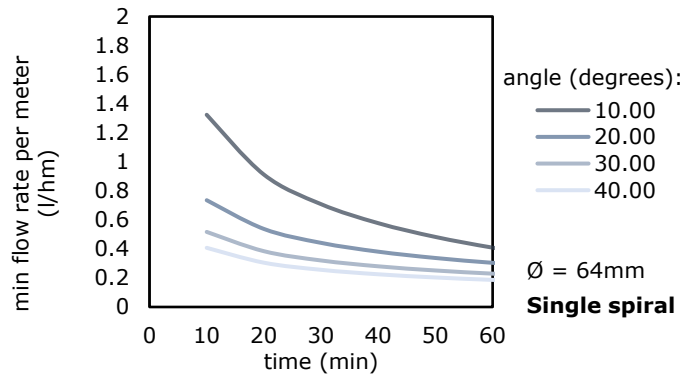


Figure 3.24 Velocity of water absorption (single spiral)

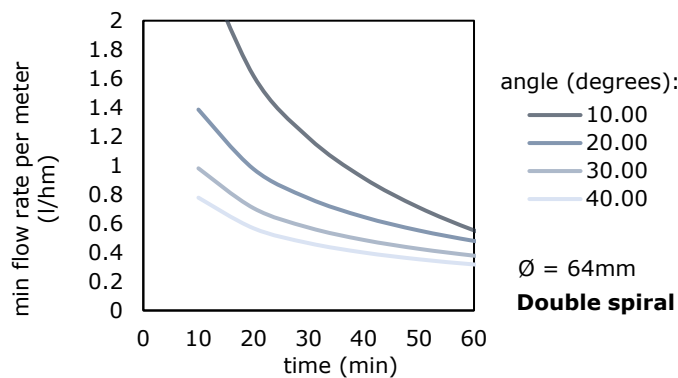


Figure 3.25 Velocity of water absorption (double spiral)

3.3 Cooling Capacity

The cooling capacity refers to the equivalent cooling load and the capacity of the ceramic system to reduce ambient air temperature (and increase ambient moisture content) mainly by evaporation. Apart from the evaporative component, convective and radiant cooling also contribute to the final temperature reduction. Chapter 1.3 introduced the different factors determining the cooling capacity of the ceramic samples (geometry, ambient conditions, air velocity), and each of them were evaluated with a series of laboratory studies conducted under controlled environments.

The studies were structured in order to determine in first instance the evaporative cooling capacity of the samples from monitored air temperature reduction and humidity increase achieved downstream the ceramics; second, the radiant cooling capacity from monitored surface temperatures and black globe

temperatures; and lastly, the dynamic performance of a PCC system mockup from continuous monitored air temperature reduction and humidity increase in a semi-outdoor space located in Granada, Spain.

3.3.1 Evaporative Cooling Studies

The laboratory studies described in this section aimed to monitor the cooling capacity of the ceramic samples introduced in 3.2.1 and the prototype P2 under controlled environmental conditions. A series of objectives were set along the progression of this work:

- i. To evaluate and compare the cooling capacity of the mentioned ceramic samples under different environmental conditions.
- ii. To test the cooling capacity of prototype P2 under the same environmental conditions and confirm the increment in performance associated to the optimised geometrical design.
- iii. To evaluate the impact of different P2 arrangements to assist in the design process of large-scale systems.
- iv. To determine the water consumption of the prototype P2 and demonstrate its suitability for water stressed countries.

These results were post-processed to allow the validation of the numerical model and the performance prediction tool developed in Chapters 4 and 5.

Methodology

Two parametric studies were conducted to address the impact of different ceramic geometries and P2 arrangements on achieved cooling capacity. The first set of studies tested the ceramic samples introduced in 3.2.1 and the prototype P2 under constant dry bulb temperature, relative humidity and air velocity (Figure 3.26). Air temperature was set to 32°C and relative humidity to 46%. The resulting DBT-WBT was 8.7°C. Air velocity was set to 0.5 m/s.

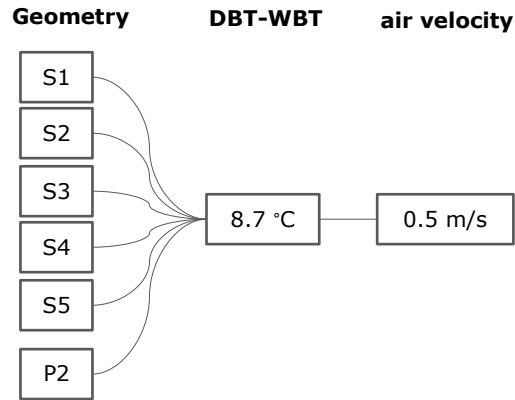


Figure 3.26 First set of parametric studies

The second set of studies addressed the impact of P2 arrangements for large-scale applications. The studies explored different exposed ceramic surface areas within a cross section of 500x500 mm. Three different arrangements presented two (A2), three (A3), and four (A4) ceramic columns of 440 mm high, equivalent to a blockage ratio BR (opaque to free area) of 25, 40 and 50%. The ceramic agreements and distances between columns are shown in Figure 3.27.

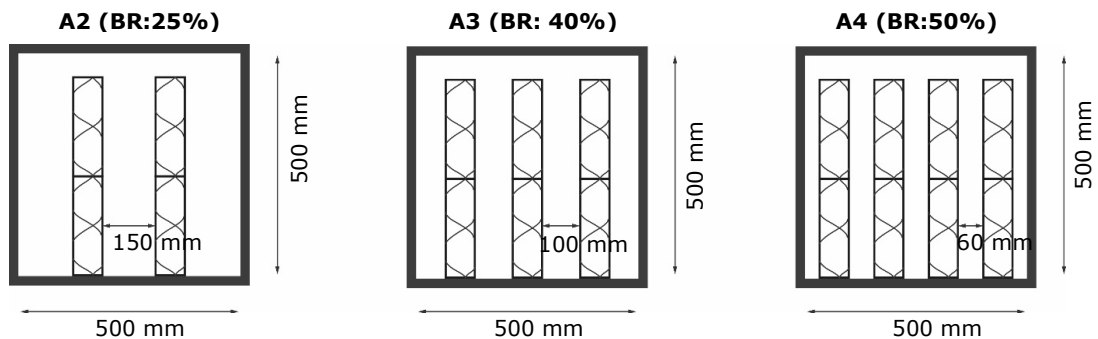


Figure 3.27 Ceramics' arrangement

In this study, temperature reduction, relative humidity and the equivalent cooling loads were obtained and compared between the three different arrangements under four different DBT-WBTs (8, 9, 10, and 11°C) and an air velocity of 0.5 m/s (Figure 3.28).

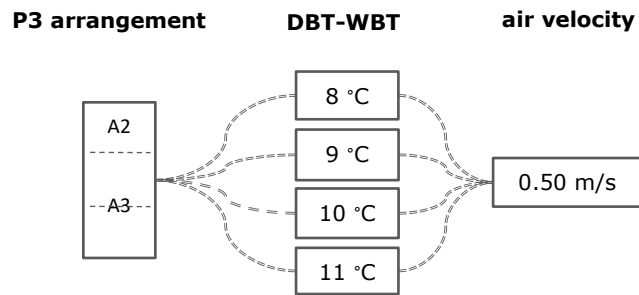


Figure 3.28 Second set of parametric studies

Both set of studies were performed in the environmental chamber available in the Marmont Building in the Department of Architecture and Built Environment in the University of Nottingham. The chamber allowed the control of temperature and relative humidity of air and kept them constant over the analysis period. The room was 3 meters long by 3 meters wide, with a total air volume of 22 m³. The temperature and humidity sensors and the mechanical system were calibrated prior to the studies on the 10th of May 2016.

A 1800x500x500 mm insulated box was placed in the chamber to create a controlled micro-environment where the ceramic samples were placed. The front side of the box had a cone-shape finishing with a honeycomb fabric sheet to allow better air distribution within the box.

Air was supplied with a portable fan at the front of the insulated box, and hit the saturated ceramic sample located at 960 mm distance from the fan (Figure 3.29 and Figure 3.30).

The evaporative process took place when the wetted surface of the ceramic was exposed to the airstream, and a reduction in temperature (and humidity increase) was achieved downstream the ceramic. Temperature on the leeward of the ceramic surface were continuously monitored with nine Resistance Temperature Detectors (RTDs) located at 400 mm from the surface of the ceramic and arranged over the duct section as described in Standard Test Method ASTM D3464-96 (2014). Relative humidity was also measured using a Temperature and Humidity probes located at 500 mm from the ceramic surface.

At the front, DBT and RH were recorded with a second temperature and humidity probe, located at 200 mm from the ceramic surface, and representing the initial conditions prior to the evaporation process.

The temperature and humidity sensors were connected to a system which recorded data every 5 seconds during 20 min of analysis period.

Air velocity was recorded manually at 9 different points as described in ASTM D3464-96 (2014) at 200 mm from the ceramic surface at the beginning of each analysis. Barometric pressure was also taken at the beginning of the test using digital readers.

At the beginning of the experiment, the ceramic samples were saturated and weighted, and at the end, the samples were weighted again to determine the evaporation rate, water consumption and its relation with the effective cooling achieved in order to obtain latent-to-sensible heat ratio.

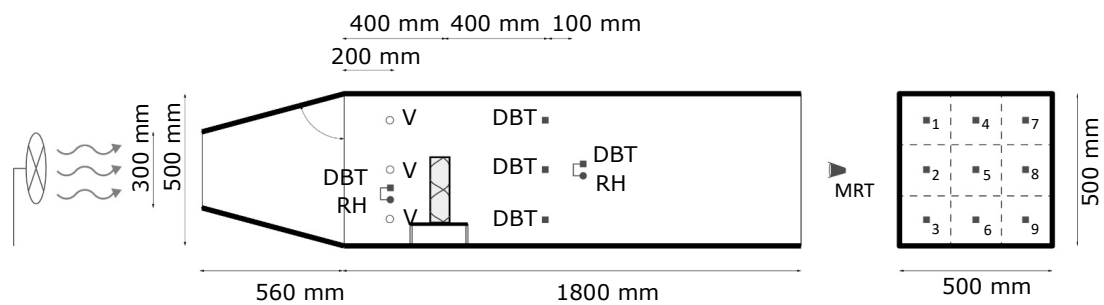


Figure 3.29 Test set-up section



Figure 3.30 Chamber box set-up

During the experimental work, the following parameters were monitored:

- Initial dry bulb temperature (T_i): Representing the initial air temperature of the test, recorded at 200 mm from the ceramic surface. Measured in $^{\circ}\text{C}$.
- Initial Relative Humidity (RHi): Representing the initial relative humidity of the test, recorded at 200 mm from the ceramic surface. Measured in %.
- Barometric pressure (Pa): Recorded at the beginning of the experiment and assumed to remain constant over the duration of each set of experiments. Measured in Pa.
- Initial saturated ceramic weight (Ws). Measured in g.
- Ceramic surface area (Ace). Measured in m^2 .
- Air velocity (v): Representing the wind conditions and assumed to remain constant over the duration of each test. Measured in m/s.
- Surface temperature: Recorded using thermal imaging camera. Measured in $^{\circ}\text{C}$.
- Final air temperature (Tout): Representing the reduced air temperature as result of the evaporation process. Measured in $^{\circ}\text{C}$.

- Final relative humidity (RH_{out}): representing the final humidity levels achieved after the evaporation process. Measured in %.

These data were used to obtain the psychometric parameters described below and needed to evaluate the cooling potential of the ceramic. The cooling capacity of the direct evaporative system was given by the equivalent convective cooling expression per ceramic unit area:

$$Q_c (W/m^2) = \frac{\rho A_{duct} v C_p [T_{in} - T_{out}]}{A_{ce}} \quad \text{Equation (3-7)}$$

Sensible cooling per ceramic surface area

where C_p is the specific heat of moist air, deriving from the ratio of dry air and water vapour content in the air and defined using the following expression:

$$C_{p (moist)} = C_{p (air)} + C_{p (vapour)} \cdot AH \quad \text{Equation (3-8)}$$

Specific heat of moist air

The specific heat of dry air derived from the ratio of C_p and C_v and the energy associated with the vibration of its molecules. The theoretical ratio C_p/C_v for diatomic molecules like oxygen, which are major constituents of air is 7/5, and since molecular weight of the air is 28.96 kg/kmol, C_p of dry air can be expressed as:

$$C_{p (air)} = \left(\frac{7}{5}\right) \left(\frac{8.314}{28.96}\right) = 1.005 \text{ kJ/kg}^\circ\text{C} \quad \text{Equation (3.9)}$$

Specific heat of dry air

Monteith and Unsworth (2013, p. 8)

Water vapour has six degrees of freedom and thus the specific heat ratio is given by $C_p/C_v = 1.334$. As its molecular weight is 18.01 g/mol (Adkins, 1983), its specific heat capacity is given as:

$$C_{p(vapour)} = 4 \left(\frac{8.314}{18.01} \right) = 1.846 \text{ kJ/kg}^\circ\text{C} \quad \text{Equation (3.10)}$$

Specific heat of water vapour
Monteith and Unsworth (2013, p. 8)

The density of humid air can be calculated with the Ideal Gas Law (Monteith and Unsworth, 2013),

$$\rho_{moist} = \left(\frac{P_a M_a + P_v M_v}{RT} \right) \quad \text{Equation (3-11)}$$

Moist air density
Monteith and Unsworth (2013, p. 5)

The vapour pressure (Pv) was first defined by Tetens (1930) as the pressure exhibited by vapour above a liquid surface. Vapour pressure is often written as e and is measured in Pa. When vapour is said to be saturated, the pressure is called the saturated vapour pressure (Psat). Saturated vapour pressure is often written as e_s , and can be approximated with the expression:

$$P_{sat} = 6.1078 \left(\frac{7.5T}{T+237.3} \right) \quad \text{Equation (3-12)}$$

Saturated vapour pressure
Monteith and Unsworth (2013, p. 13)

From the relation $e_s - e$, vapour pressure can be estimated with the expression:

$$P_v = \left(\frac{RH \cdot P_{sat}}{100} \right) \quad \text{Equation (3-13)}$$

Vapour pressure of moist air

The deficit $e_s - e$ is an indicator of evaporation potential of ambient air, where a higher $e_s - e$ depression yields to higher evaporative cooling potential.

Another indicator of vapour content in air is AH, and its relation to Pv is defined as:

$$AH = 0.622 \frac{P_v}{P_d}$$

Equation (3-14)

*Absolute humidity as function of Pv
Monteith and Unsworth (2013, p. 16)*

Wet bulb temperature is the temperature of air as if it were saturated, and it can be estimated by the relation between RH and T as described by Stull (2011),

$$WBT = T \tan[0.151977(RH + 8.313659)^{1/2}] + \tan(T + RH) - \tan(RH - 1.676331) + 0.00391838(RH)^{3/2} \tan(0.023101RH) - 4.686035$$

Equation (3-15)

*WBT from RH and DBT
Stull (2011, p. 2267)*

Each test was monitored during 20 min, and the first 5 min were allowed for the box to couple with the environmental chamber climatic conditions. During the next 15 min the wetted ceramic was placed inside the insulated tunnel and exposed to air movement to promote evaporation. The typical outcome profile obtained is shown in Figure 3.31. The graph shows readings of the nine RTD sensors and the SDI12 probe (DBT and RH) located at the leeward of the ceramic surface, and readings of the SDI12 probe (DBT and RH) on the windward of the ceramic surface. There is an evident temperature drop and an RH increase after placing the ceramic in the tunnel. These conditions remained fairly constant during the rest of the time before the ceramic being removed.

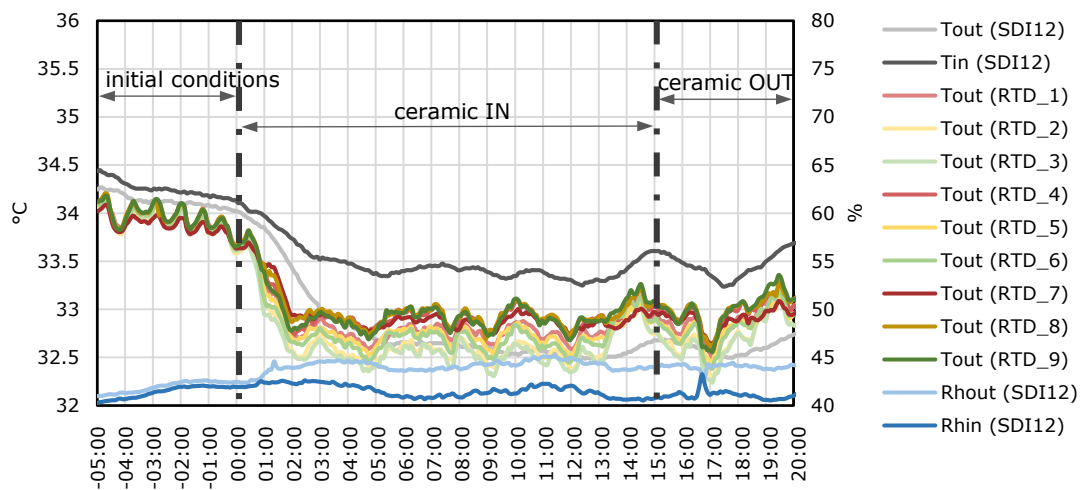


Figure 3.31 Recorded data from sample P3_A4. DBT-WBT=10C

Impact of geometry

The first set of parametric studies addressed the impact of geometry on the cooling performance of the system. All samples were fully saturated prior to the beginning of the test and then exposed to an environment with DBT of 32°C and RH of 46%, which corresponded to a DBT-WBT of 8.7°C. The average air velocity inside the tunnel is 0.5 m/s. This value derived from 9 spot readings taken at the beginning of each iteration as shown in Figure 3.32.

0.35	0.24	0.83
0.32	0.40	0.81
0.34	0.32	0.85

Figure 3.32 Air velocities in m/s recorded over the cross section of the insulated tunnel

The results obtained are shown in

Table 3.7. Overall, they presented small variations in temperature and humidity. The maximum temperature drop recorded was 1.09°C and RH increase reached 4.64%, corresponding to the ceramic Sample 1 (S1), which presented the highest effective surface area. The equivalent cooling loads per surface area achieved presented competitive numbers (290 to 524 W/m²) if compared to the 100-200 W/m² achieved by Ibrahim et al. (2003).

Moreover, the cooling capacity achieved was found to vary with different ceramic geometries. The effective surface area and drag coefficient of the ceramic elements were obtained to explore the relation between them and the cooling capacity achieved. Effective surface area was obtained by measuring the surface area exposed to the environment. Drag coefficient was obtained as described in Chapter 4.3. Table 3.6 shows the ESA and Cds of the tested samples.

Table 3.6 Effective surface area and drag coefficient of ceramic samples

	ESA:V (m ⁻¹)	ESA (m ²)	C _d	ESA/C _d (m ²)
S1	106.70	0.30	0.60	0.51
S2	68.46	0.11	0.45	0.24
S3	103.79	0.13	1.02	0.13
s4	60.59	0.05	1.01	0.05
S5	157.09	0.13	0.73	0.18
P2	160.28	0.04	0.50	0.09

Table 3.7 Summarised results obtained from 1st set parametric studies

	DBT (°C)	RH (%)	WBT (°C)	DBT-WBT (°C)	ΔT (°C)	ΔRH (%)	ΔAH (g/kg)	Qc (W)	Qc:ESA (W/m ²)
s1	31.34	45	22.79	8.78	1.09	4.64	0.40	159	524.22
s2	31.13	46	22.94	8.48	0.30	1.40	0.24	43	401.01
s3	31.30	45	22.97	8.73	0.27	0.96	0.37	38	290.14
s4	31.25	46	23.02	8.69	0.13	0.96	0.23	19	222.79
s5	31.35	45	23.00	8.71	0.29	1.23	0.31	42	324.75
p2	31.02	46	22.84	8.45	0.15	0.36	0.15	21	496.74

The results suggested that higher exposed surfaces areas tended to present higher cooling capacities. However, this statement was not always met for all of the tested samples (Figure 3.33). Instead, the ESA/C_d factor presented a proportional relation with the cooling capacity of the samples, suggesting that the drag coefficient is also a determining factor on the cooling achieved (Figure 3.34).

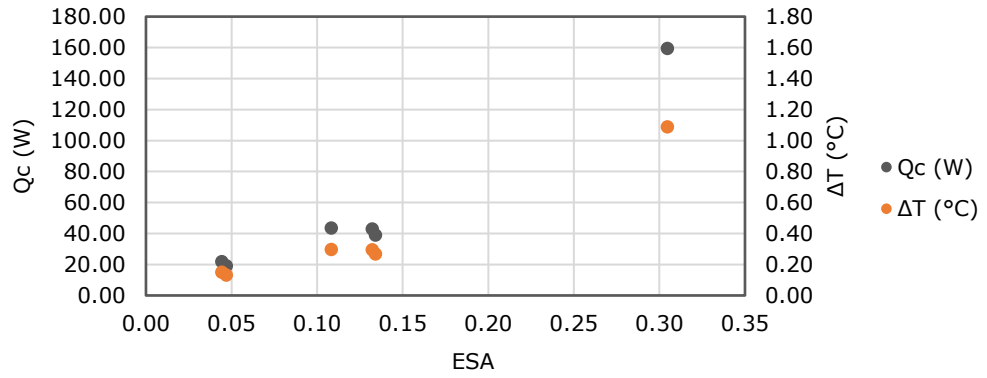


Figure 3.33 Relation between equivalent convective cooling and ESA

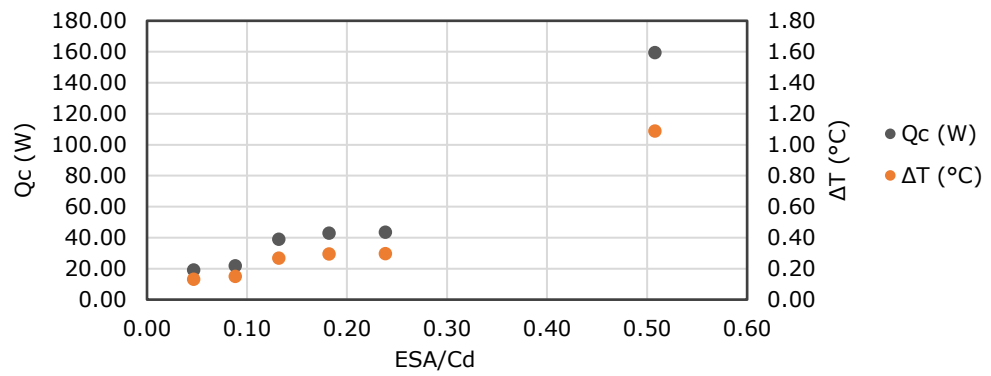


Figure 3.34 Relation between equivalent convective cooling and ESA:Cd

For the evaluation of water consumption, the ceramic sample was weighted before and after the test. The decrement of weight was an indicator of total evaporated water and latent cooling during the test. The data was then compared with variations in absolute humidity and relative humidity during the experimental work. The variations in ceramic weights and the equivalent evaporated water are shown in Table 3.8. In this table, the second column shows the decrement in weight of each ceramic sample measured in grams, and the fifth column the evaporation rate given in grams per minute per ceramic surface area, which was calculated by dividing the decrement in weight by the duration of the test process.

Table 3.8 Amount of evaporated water

	ΔM (g)	ΔRH (%)	ΔAH (g/kg)	Evaporation rate (g/min)	Evaporation rate (g/m ² min)
s1	8	4.64	0.40	0.53	1.75
s2	13	1.40	0.24	0.87	8.01
s3	13	0.96	0.37	0.87	6.46
s4	2	0.96	0.23	0.13	1.56
s5	9	1.23	0.31	0.60	4.55
p3	5	0.36	0.15	0.33	7.58

The results suggested a considerable variation in evaporation rate per surface area under the same environmental conditions. This reaffirmed the impact of the geometry and aerodynamics on the amount of water released from the ceramic surface. The increment of absolute humidity measured during the experimental work also confirmed the relation between air moisture content (given as AH and RH) and the effective surface area to drag coefficient ratio (ESA/Cd), shown in Figure 3.35. It can be observed that higher ESA/Cd implied higher absolute humidities downstream the ceramics, and vice versa.

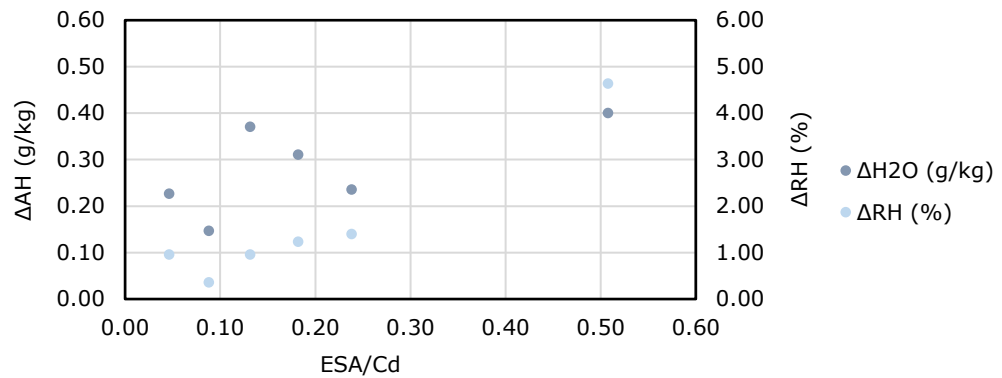


Figure 3.35 Increment of AH and relative humidity to ESA/Cd

Impact of P2 arrangement

The third set of experimental work addressed the impact of different arrangements of the ceramic prototype P2. Two (A2), three (A3) and four (A4) columns were located in the insulated tunnel exposed to four different DBT-WBT and an air velocity of 0.5 m/s.

The results from these studies are summarised in Table 3.9, suggesting that the achieved temperature reduction ranged from 0.5 to 2.0°C, and varied according to the P2 arrangements and environmental conditions. The ambient-to-saturated vapour pressure (e-es) calculated from Equation (3-12) and Equation (3-13) was used as an indicator to express the potential for evaporation.

Table 3.9 Summarised results from 3rd set of parametric studies

	DBT (°C)	RH (%)	WBT (°C)	DBT-WBT (°C)	v (m/s)	ΔT (°C)	ΔRH (%)	ΔH₂O (g/kg)	Qc (W)	Qc:ESA (W/m²)	es-e (mb)
P3 (A2)	29.75	46	23.41	8.30	0.53	0.20	0.23	0.11	31	178	22.40
P3 (A2)	31.80	43	22.92	8.88	0.53	0.40	0.35	0.29	62	355	26.76
P3 (A2)	33.96	40	25.00	10.20	0.53	0.65	1.21	0.61	69	3975	32.87
P3 (A2)	35.87	37	24.54	11.33	0.53	0.70	3.63	0.98	86	493	38.33
P3 (A3)	29.93	47	21.76	8.18	0.43	0.66	3.39	0.29	83	315	22.28
P3 (A3)	32.05	43	22.84	9.21	0.43	0.95	4.11	0.35	120	454	26.88
P3 (A3)	33.89	40	23.63	10.27	0.43	1.15	5.31	0.77	143	544	31.70
P3 (A3)	35.72	38	24.77	10.95	0.43	1.25	2.65	0.98	155	588	35.96
P3 (A4)	29.81	45	21.29	8.52	0.53	0.94	5.00	0.61	148	422	22.97
P3 (A4)	32.11	43	22.81	9.30	0.53	1.31	4.83	0.36	204	582	27.18
P3 (A4)	34.10	40	23.89	10.21	0.53	1.36	4.04	0.52	211	601	31.83
P3 (A4)	36.01	36	24.59	11.41	0.53	1.53	3.43	1.36	237	674	37.55

The obtained data confirmed a proportional increase of cooling capacity of the system with the ambient-to-saturated vapour pressure. This also implied that higher DBT-WBT yielded to higher temperature reductions, and thus to a more effective system. Increasing the number of ceramic elements per 0.25 m² of cross section also presented a proportional increase in their cooling capacity. Figure 3.36 and Figure 3.37 relate the achieved temperature reduction (ΔT) and

the achieved cooling capacity per square meter of effective surface area ($Q_c:ESA$) with the ambient-to-saturated vapour pressure ($e-es$) to for the three different arrangements.

As expected, the graphs suggested a better performance in the arrangement A4, due to higher exposed surface area. The results also suggested that a 2 degrees' temperature reduction is only achievable under hotter and dryer conditions ($DBT-WBT$ higher than $12^{\circ}C$). The cooling capacity per ceramic surface area presents, once again, competitive results as discussed in the previous section.

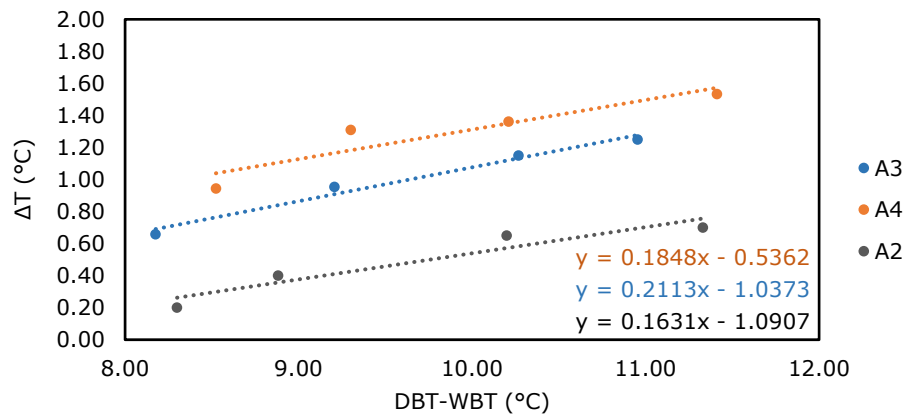


Figure 3.36 Relation between $DBT-WBT$ and increment in air temperature for different column arrangements

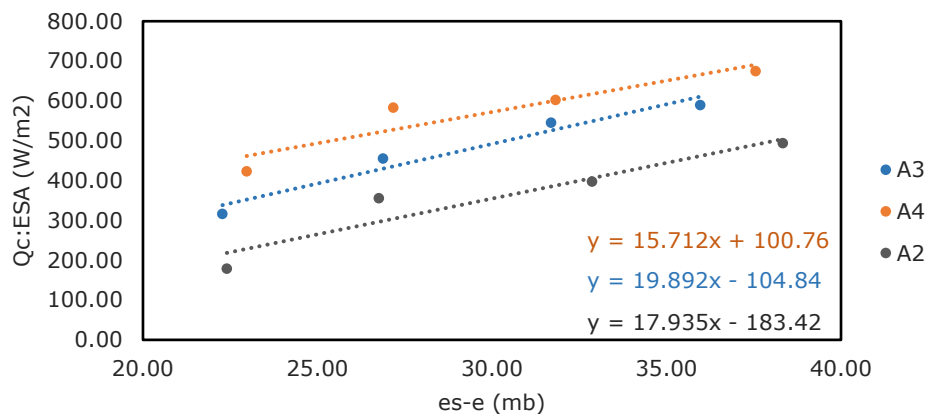


Figure 3.37 Relation between $e-es$ and cooling capacity per ceramic surface area for different ceramic arrangements

Water consumption for the system was also estimated following the same methodology. The results of water consumption and increase of absolute humidity are summarised in Table 3.10.

Table 3.10 Evaporated water data

	DBT-WBT (°C)	ΔM (g)	ΔRH (%)	ΔH₂O (g/kg)	Evaporation rate (g/min)	Evaporation rate (g/m²min)
P3 (A2)	8.30	16	0.23	0.11	1.07	6.06
P3 (A2)	8.88	18	0.35	0.29	1.20	6.82
P3 (A2)	10.20	21	1.21	0.61	1.40	7.95
P3 (A2)	11.33	23	3.63	0.98	1.53	8.71
P3 (A3)	8.18	16	3.39	0.29	1.07	4.04
P3 (A3)	9.21	24	4.11	0.35	1.60	6.06
P3 (A3)	10.27	26	5.31	0.77	1.73	6.57
P3 (A3)	10.95	28	2.65	0.98	1.87	7.07
P3 (A4)	8.52	38	5.00	0.61	2.53	7.20
P3 (A4)	9.30	34	4.83	0.36	2.27	6.44
P3 (A4)	10.21	38	4.04	0.52	2.53	7.20
P3 (A4)	11.41	38	3.43	1.36	2.53	7.20

As discussed before, cooling capacity increases under hotter and dryer conditions, implying both sensible and latent heat. The increase in absolute humidity in air during the studies performed ranged from 0.5 to 1.2 grams per kilo. This water content increased proportionally with the DBT-WBT, and also with the blockage ratio BR (Figure 3.38).

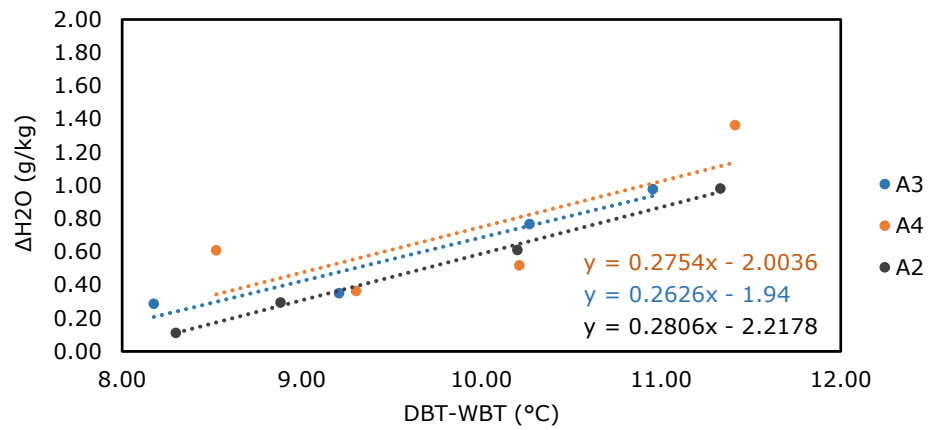


Figure 3.38 Relation between DBT-WBT and increment in absolute humidity for different column arrangements

Surface temperatures were also recorded using a thermal imaging camera during the I process (Figure 3.39). The recorded data is summarised in Table 4.11.



Figure 3.39 Surface temperatures recorded for DBT-WBT of 9,10 and 11°C; and different ceramic arrangements

The readings suggested that surface temperatures ranged between three to five degrees above wet bulb temperatures, in line with the studies conducted by Givoni (1994), and responding to the expression in Equation (3-16). The direct relation with DBT-WBT confirmed that warmer and dryer environmental conditions increases convective and radiant cooling capacity, the latter addressed in Chapter 3.3.2.

$$ST = DBT - 0.8 \cdot (DBT - WBT) + 3 \quad \text{Equation (3-16)}$$

Monitored surface temperatures of P2

Table 3.11 Recorded surface temperatures for P2

	DBT-WBT (°C)	DBT (°C)	WBT (°C)	ST (°C)	DBT-ST (°C)	WBT-ST (°C)
P2 (A2)	8.30	29.75	23.41	26.4	3.35	2.99
P2 (A2)	8.88	31.80	22.92	27.9	3.90	4.98
P2 (A2)	10.20	33.96	25.00	28.9	5.06	3.90
P2 (A2)	11.33	35.87	24.54	28.6	7.27	4.06
P2 (A3)	8.18	29.93	21.76	26.5	3.43	4.74
P2 (A3)	9.21	32.05	22.84	27.5	4.55	4.66
P2 (A3)	10.27	33.89	23.63	28.7	5.19	5.07
P2 (A3)	10.95	35.72	24.77	28.9	6.82	4.13
P2 (A4)	8.52	29.81	21.29	26.1	3.71	4.81
P2 (A4)	9.30	32.11	22.81	27.5	4.61	4.69
P2 (A4)	10.21	34.10	23.89	28.4	5.70	4.51
P3 (A4)	11.41	36.01	24.59	29.6	6.41	5.01

Conclusions

The experimental work conducted suggested a relation between the geometry and cooling loads, temperature reduction and humidity increase downstream the ceramics under different environmental parameters. The first set of parametric studies confirmed the important role of effective surface area and aerodynamics (drag coefficient) on the achieved cooling capacity. Higher ESA/Cd yielded to an increase in air moisture content. The variation of evaporation rates ranged from 1.5 to 8 grams of water per square meter of ceramic surface area and, although the achieved air temperature reductions were small (below 1.09°C), the equivalent sensible and cooling load was proved to be significant if compared with previous work (Chapter 2.2). Results also confirmed that each sample presents different sensible-to-latent cooling ratio.

The second set of empirical studies focused on the performance of the ceramic prototype P2. Different ceramic arrangements were tested under different environmental conditions to address variations in cooling capacity of the system. A polynomial relation was obtained from the results, confirming the proportional growth in cooling performance with the number of columns.

Results also confirmed that air temperature reductions of 2 degrees are only achievable under warmer and dryer conditions (DBT-WBT greater than 12°C).

Figure 3.36 and Figure 3.37 can be used to predict air temperature reduction and cooling loads for different P2 arrangements as a function of DBT-WBT.

The evaporation rate also increases proportionally to DBT-WBT and Figure 3.38 can be used to predict average water consumption of the system. Using this graph, it can be concluded that one-meter-high ceramic column will require a minimum top-drip irrigation flow rate from top between 0.07 and 0.1 l/h to meet its evaporation rate.

In addition, the recorded ceramic surface temperatures during the experimental work ranged from 3 to 5°C higher than wet bulb temperature. The increase of surface temperature is proportional to DBT-WBT, suggesting that sensible cooling is also higher under warmer and dryer environmental conditions. The low surface temperature registered also reaffirmed the potential for radiant cooling of the system. This is further explored in 3.3.2.

3.3.2 Radiant Cooling Studies

The radiant cooling studies aimed to determine the mean radiant temperature achieved downstream the ceramics as result of the low surface temperatures observed in Chapter 3.3.1. Radiant fluxes of porous ceramic systems were determined using the back-globe temperature measuring method under controlled environments. The main objective was to assess the full impact of PCC systems on user thermal comfort, including the radiant component, needed to obtain the Operative Temperature (OT). OT is a simplified measure of human thermal comfort adopted by ASHRAE and ISO standards and derived from air temperature, mean radiant temperature and air velocity. In environments with low air velocities, the relative influence of MRT on human thermal perception can be up to 50%, confirming the relevance of this component.

The results obtained from the testing process were processed in order to meet the following objectives:

- To find a numerical relation between ambient conditions (DBT-WBT) and the resulting mean radiant temperature (MRT).
- To identify the MRT at different distances from the PCC system.
- To validate the MRT calculation methods described in the next section.

Measuring Method

Mean radiant temperature (MRT) of a given space can be determined from the black globe temperature (GT), ambient air temperature (DBT) and velocity (v). GT can be measured by a black-globe thermometer, which consist of a black globe presenting an external dark surface (generally by means of a layer of matt black paint), and a temperature sensor placed inside the globe. When exposed to radiant fluxes, the globe tends towards a thermal balance under the effect of exchanges by radiation and also convection. The globe temperature at thermal balance allows MRT to be determined using Equation (3-17), expression adopted by the International Standards ISO (7726:2001) and ASHRAE (2013).

$$MRT = \left[(GT + 273)^4 + \frac{1.10 \times 10^8 v^{0.6}}{\varepsilon D^{0.4}} (GT - DBT) \right]^{1/4} - 273$$

Equation (3-17)

Mean radiant temperature
ISO (7726:2001, p. 16)

where MRT is the mean radiant temperature in °C, GT is the globe temperature in °C, DBT is the air temperature in °C, v is the air velocity in m/s, ε is the emissivity of the black globe, and D is the diameter of the black globe in meters.

Among other considerations, it is important to allow 20 to 30 min for the black-globe thermometer to respond to the environmental conditions. Also, it must be accepted that this method is an approximation for the assessment of MRT, in which many other parameters have an influence on the final result: globe diameter and emissivity, temperature sensor sensitivity, air velocity, etc.

For the completion of this test, the following parameters were monitored:

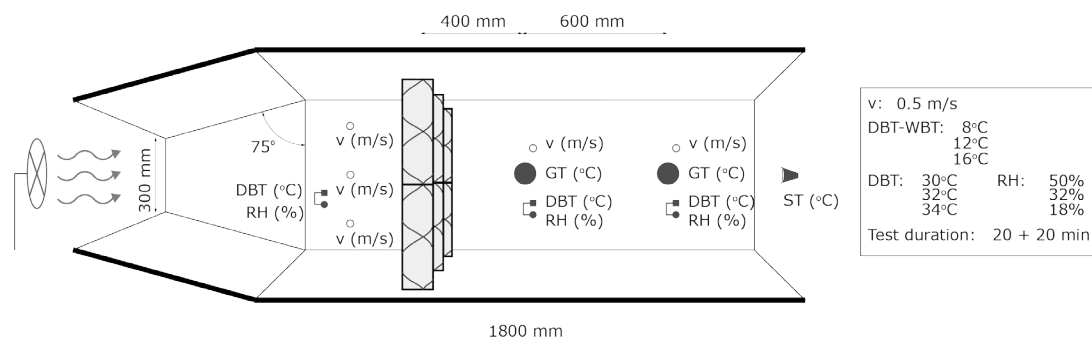
- Dry Bulb Air Temperature (DBT) upstream and downstream the ceramic samples.
- Relative Humidity (RH) upstream the ceramic samples.
- Air velocity (v) upstream and downstream the ceramic samples.
- Black-Globe Temperature (GT) downstream the ceramic sample.
- Surface temperature (ST) of the ceramic samples.

The equipment used for the monitoring process were:

- Thermal anemometer (v)
- Extech Thermal Stress Meter HT200 (DBT, RH, GT)
- Thermal Imaging Camera (ST)

Air was supplied at 0.5 m/s with a portable fan at the front of the insulated box and hit the saturated prototype P2 presenting a blockage ratio of 50% located at 960 mm distance from the fan (Figure 3.40 and Figure 3.41).

- i. Set ambient DBT and RH at 30°C and 50% in the climate chamber. Record the barometric pressure using the chamber reader.
- ii. Monitor DBT and RH during 20 minutes until chamber air conditions stabilises (initial conditions prior to the evaporation process).
- iii. Turn the electric fan on to drive an air velocity of 0.5 m/s.
- iv. Wet the ceramic samples and place them inside the insulated box. Take a thermal image of the ceramic surfaces at initial conditions.
- v. Monitor (manually) air velocity at 200 mm upstream the ceramic sample according to Standard Test Method ASTM D3464-96 (2014)
- vi. Monitor DBT, RH, v and GT (manually) at 400 mm downstream the ceramic sample every 5 minutes during 20 minutes, until GT stabilises. Take a thermal image of the ceramic surface at the final conditions.
- vii. Repeat step 6, monitoring (manually) DBT, RH, v and Gt at 1000 mm downstream the ceramic surface.
- viii. Repeat steps 1 to 7, setting the climate chamber ambient DBT and RH to 32°C and 32%.
- ix. Repeat steps 1 to 7, setting the climate chamber ambient DBT and RH to 34°C and 25%.



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Table 3.13 Monitored data for DBT-WBT: 12K

DBT-WBT: 12K (DBT:32°C; RH:32%) // ΔD: 0.4m						ΔD: 1m				
	Initial Conditions	5 min	10 min	15 min	20 min	Initial Conditions	5 min	10 min	15 min	20 min
DBT	31.20	30.00	30.00	30.00	30.00	31.20	30.20	30.10	30.10	30.10
RH	29.60	33.50	33.50	33.20	33.70	30.30	32.30	32.50	31.90	32.50
GT	31.10	29.50	29.50	29.50	29.40	31.10	29.80	29.70	29.70	29.70
v	0.70	0.45	0.45	0.44	0.43	0.52	0.47	0.49	0.48	0.48
ST		24.50	24.20	24.00	24.30		24.10	23.80	23.80	23.10
WBT	19.18					19.34				
DBT-WBT	12.02					11.86				
MRT					28.14					28.81
DBT-MRT					1.86					1.29

Table 3.14 Monitored data for DBT-WBT: 14K

DBT-WBT: 14K (DBT:34°C; RH:25%) // ΔD: 0.4m						ΔD: 1m				
	Initial Conditions	5 min	10 min	15 min	20 min	Initial Conditions	5 min	10 min	15 min	20 min
DBT	34.20	32.90	32.90	32.90	32.80	34.20	33.00	33.00	33.00	33.10
RH	24.00	30.60	31.00	31.10	30.70	26.50	29.90	30.60	30.40	30.40
GT	34.10	32.30	32.30	32.20	32.10	34.00	32.60	32.60	32.60	32.60
v	0.61	0.30	0.32	0.36	0.45	0.70	0.53	0.45	0.54	0.48
ST		26.30	25.60	26.20	25.70		26.60	26.30	26.30	26.20
WBT	19.94					20.63				
DBT-WBT	14.26					13.57				
MRT					30.62					31.51
DBT-MRT					2.18					1.59

3.3.3 Fieldwork

An experimental mock-up was done in August 2016 in order to evaluate the performance of a large scale evaporative cooling system under the hot dry climatologic conditions of the city of Granada, Spain. The proposed mock-up consisted in a ceramic screen made of P2 ceramic elements and derived from previous findings from the empirical work, which determined the required water flow rate, temperature and effective ceramic surface area.

The main objective was to observe the dynamic response of the system over a 24-hour cycle and one week analysis period through the continuous monitoring of the achieved temperature reduction, humidity increase, and ceramic surface temperature. Moreover, it was intended to identify a wider range in performance through the comparison of fieldwork data with laboratory data from Chapter 1173.3.1. This section describes the climatic conditions of the selected location, space configuration, and system set-up, concluding with the main findings of the study.

Climate description

The experimental mock-up took place in the city of Granada, an urban area in the South of Spain with latitude 37.18 (Figure 3.42). The city is classified as a Continental dry climate by the Köppen-Geiger climate classification system, presenting very severe hot and dry summers influenced by African winds. According to averaged weather data recorded from Granada Airport weather station, the hot season starts in June with mean max temperatures above 30°C and extends until mid-September when mean maximum temperature drops below this value (Figure 3.43).

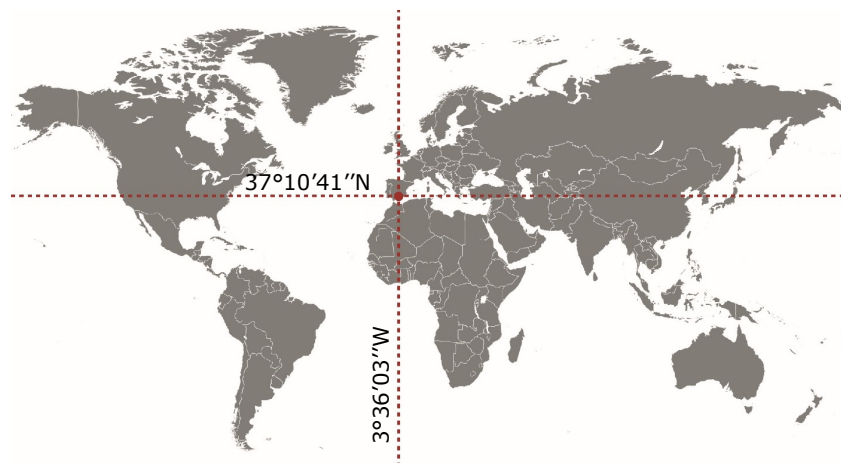


Figure 3.42 Geographical location of Granada, Spain

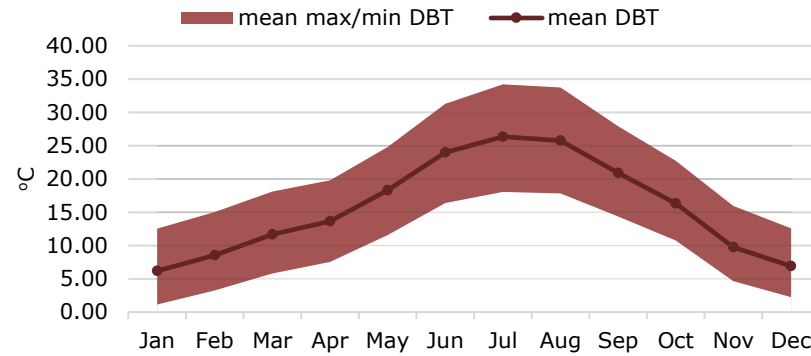


Figure 3.43 Monthly average DBT (Granada)

August is the warmest month in this season, presenting mean maximum temperatures of 38°C with potential peaks of 45°C normally occurring at 3pm, and a day-to-night temperature fluctuation of 15 degrees. RH remains below 35% during the daytime, equivalent to a water vapour content of 10g/kg remaining constant over the day. The wet bulb temperature averages 19°C at the warmest hour, below the threshold of 25°C recommended by Givoni (1994) and 24°C from Santamouris and Asimakopoulos (1996). The equivalent DBT-WBT depression under these conditions is 15°C, making evaporative cooling systems highly applicable in this location. Moreover, these dry conditions are met for 100% of the hours above 30°C during the warm season, making it an excellent environment to evaluate the cooling capacity of porous ceramic wall mock up (Figure 3.44 and Figure 3.45).

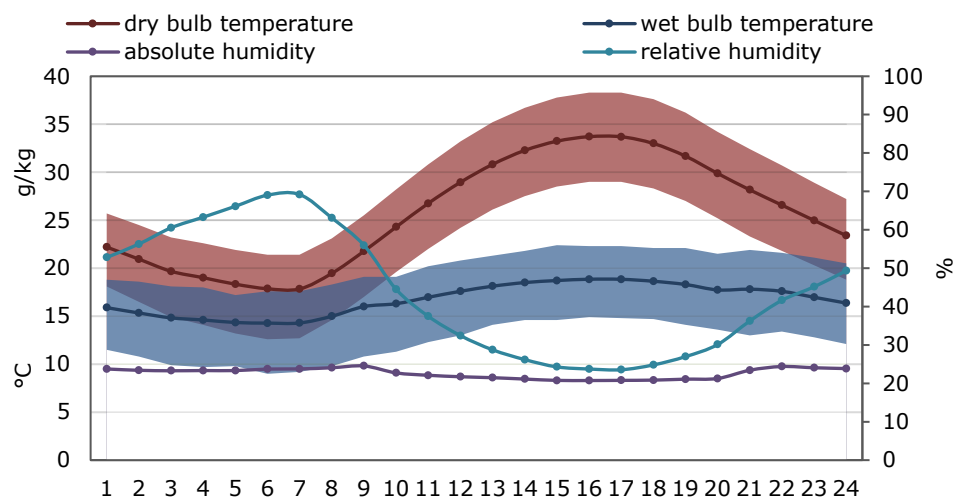


Figure 3.44 Daily average climate conditions (August, Granada)

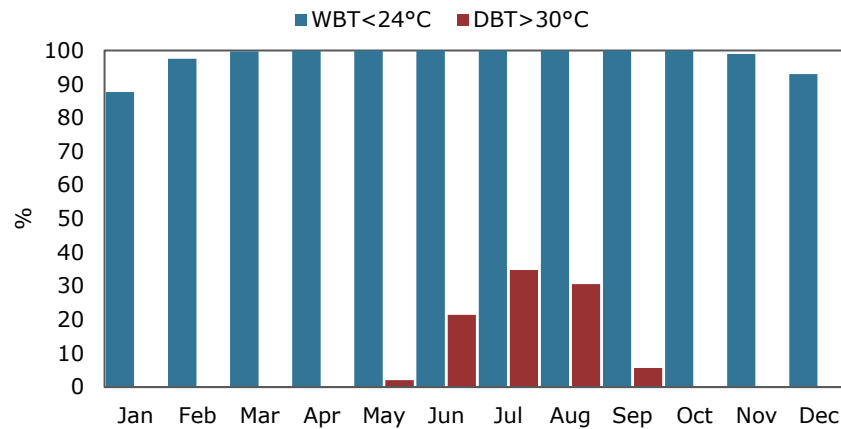


Figure 3.45 Frequency of WBT < 24°C and DBT > 30°C (Granada)

Site

The site was selected according to the ease of access and space alteration capabilities. It settled in a small industrial area in the outskirts of the city, juts by the North border of the city centre. The area covered 2600 m² with three 2-storey buildings around a central patio for trucks loading area. The buildings are NNW-SSE and NEE-SWW oriented and considerably exposed to solar radiation and wind flow due to the small number of obstructions in the surroundings and the low density of edification (Figure 3.46).

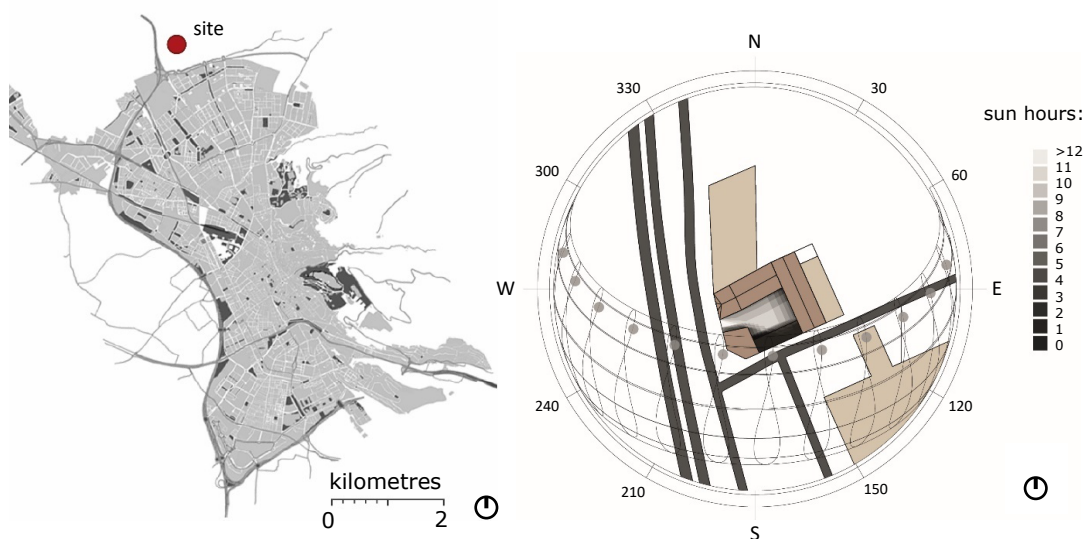


Figure 3.46 Site location (left) and estimated sun hours on August 9am-9pm, Granada (right)

Among the different space configurations that can be found on site (both indoor and outdoor), the selected location for the PCC analysis responded to a semi-outdoor space, protected from solar radiation and strong winds, and connecting the external patio with an indoor space of 160 m². The space was a double high room currently empty and the total free volume is 960 m³. This space connected to the patio through a side wall opening of 3m width and 3m high SWW oriented. A 10m long and 6m wide pergola covered the immediate surroundings, reducing solar access on the wall opening to only 2 hours in the late evening (Figure 3.47 and Figure 3.48). This strategic location transitioning the semi outdoor space and the patio benefits from a natural air movement driven by the temperature difference between both, needed to induce evaporation. Moreover, the lack of internal gains (occupancy and electrical equipment and lighting) in the semi-outdoor space avoided any uncontrolled alteration of the ambient air temperatures registered.

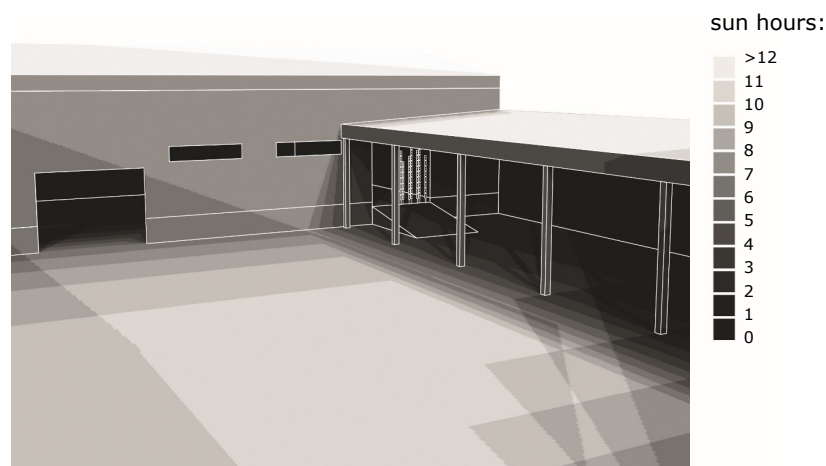


Figure 3.47 Site perspective view. Sun hours (August 9am-9pm, Granada)

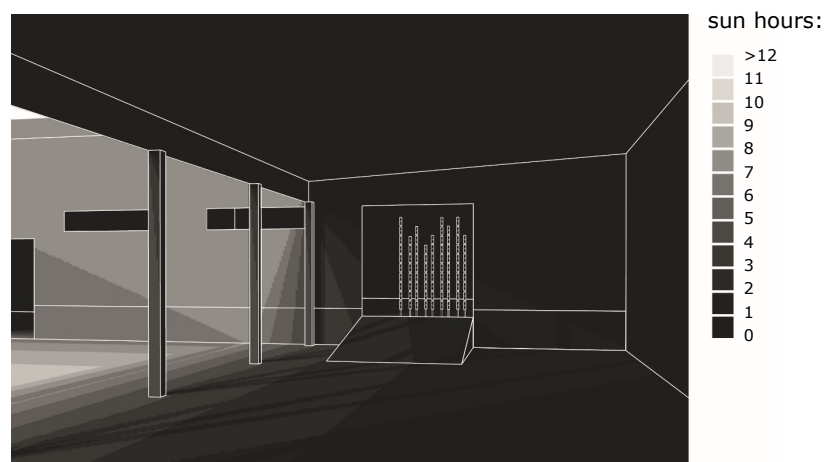


Figure 3.48 PCC wall perspective view. Sun hours (August 9am-9pm, Granada)

System set-up

The PCC wall mock-up consisted of three main parts: the water supply system, the bearing structure and the water collector ceramics.

The water supply system consisted of a main 12 mm diameter polystyrene pipe connected to the water mains system of the building, located at 6m distance from the wall aperture. A series of 6 mm pipes connected to the main line delivered the water to the top of the ceramic structure. The provision of water was done through a drip irrigation system with drippers delivering a water flow rate of 1.25 l/h.

The bearing structure of the ceramic consisted of a vertical steel column covered by the ceramic elements. It is formed by vertical tubes of 30mm diameter and 2500mm height, with a 200x200 mm rectangular plate at the bottom attached to the ground using a rotatory tool and four steel screws at each corner. The attachment of the ceramics to the structure is done using double-end bolts M8 90mm under each ceramic element. The water collection at the bottom of each column was solved using a funnel-shaped plastic element attached to the steel column and connected to individual 1 litre water collectors. This was used to estimate the released water during the test Figure 3.49.

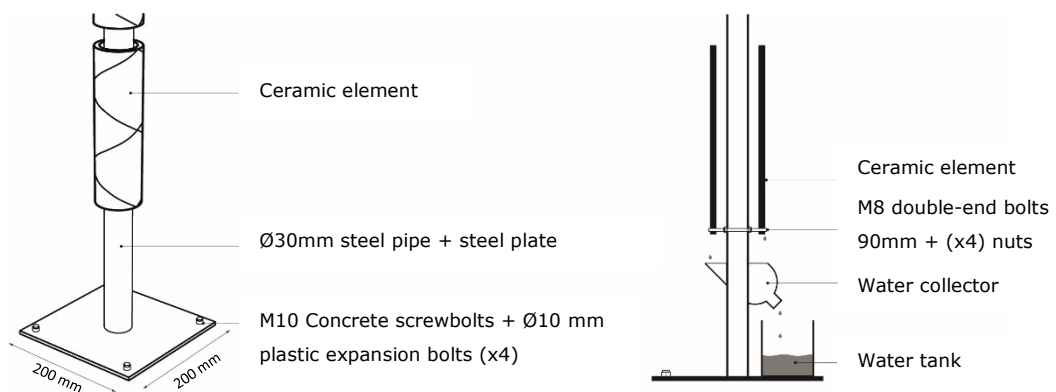


Figure 3.49 Ceramic wall structure (left) and water collector (right)

The ceramics columns were disposed along the SWW wall aperture covering 4m². Nine ceramic columns were raised on two parallel and 400mm-distanced rows of 2000mm long. The distance between columns was also 400mm, equivalent to a blockage ratio (BR) of 25%. The columns extended to 2 meters high, presenting 10 ceramic elements of 220mm high in each column and allowing 10mm

distance between each sample to avoid potential cracks during the study (Figure 3.50).

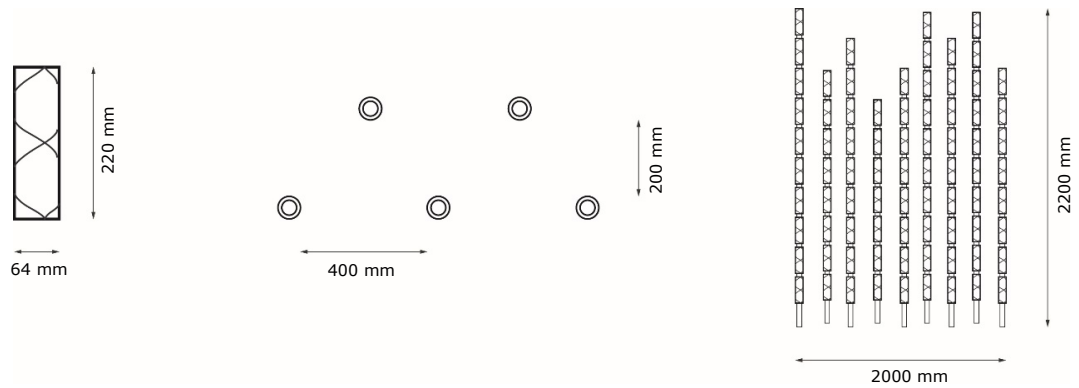


Figure 3.50 Ceramic wall arrangement

The ceramic wall was arranged in a transitional line between a semi outdoor space and the external patio, and air was driven mechanically to assist air movement and achieve 0.25m/s. The existing aperture was 3x3 m and the ceramic wall covered 70% of the surface area to allow access to people from the left side. Water was delivered from mains and temperature was measured using 12 RTD temperature sensors connected to a monitoring system. The arrangement of the RTD sensors was as follows: three sensors vertically aligned at 0,75, 1,5 and 2,25 m above the floor level at 400mm, 2000mm and 5000mm downstream the ceramic wall. A SDI12 DBT and RH sensor monitored the external conditions of the environment, constantly protected from sun. In addition, a digital anemometer recorded air velocities during the experiment. An additional DBT and RH sensor recorded the temperature reduction and HR increase achieved at 1m downstream the ceramics to evaluate the sensible and latent heat released from the ceramics (Figure 3.51). The monitoring period was 7 days, with measurements taken every 15 minutes from 10 to 6pm.

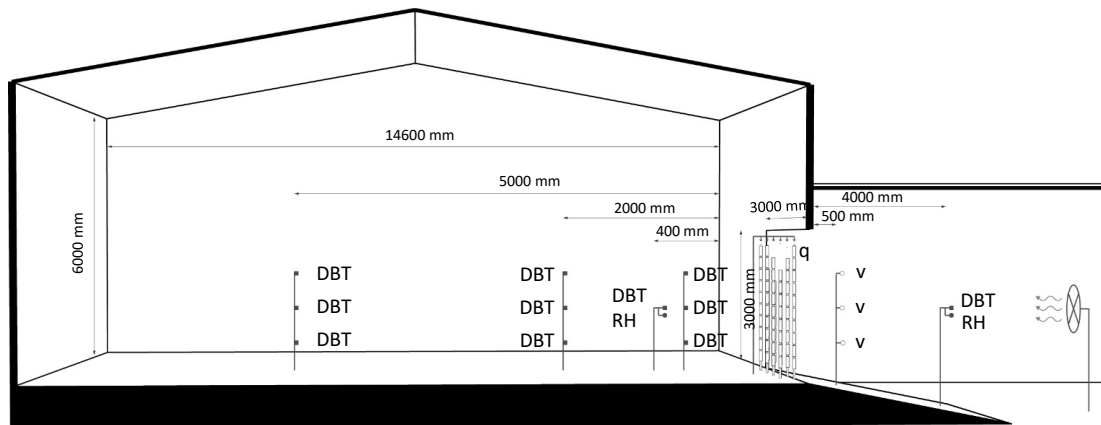


Figure 3.51 Fieldwork set-up

Results & conclusions

The experimental mock-up conducted in Granada aimed to evaluate the dynamic behaviour of a PCC System exposed to daily variations in environmental conditions. Measured ambient air temperature during the operational period ranged from 24°C to 34°C, relative humidity ranged from 30% to 50% and air velocity ranged from 0m/s to 0.6m/s. A 2000x2200mm ceramic system presenting a 25% blockage ratio was built and placed on a transitional space, naturally ventilated, shaded and wind sheltered. Temperature and humidity on the windward and leeward of the ceramic surface were monitored using temperature and humidity sensors at 400mm, 2000mm and 5000mm distance from the ceramic walls (Figure 3.52).



Figure 3.52 PCC System Mock-up

The collected data was post-processed and the resulting temperature reduction achieved for every interval was correlated to the data obtained from laboratory studies (Figure 3.53). The measurements confirmed an expected wider amplitude of $\pm 0.3^{\circ}\text{C}$ in the temperature reduction achieved when compared to

laboratory results, which is mainly caused by air velocity variations. Moreover, these temperatures followed the same linear profile found in the conducted laboratory analysis, presenting higher temperature reductions with hotter and dryer conditions, and achieving temperature reductions up to 2°C downstream the ceramics.

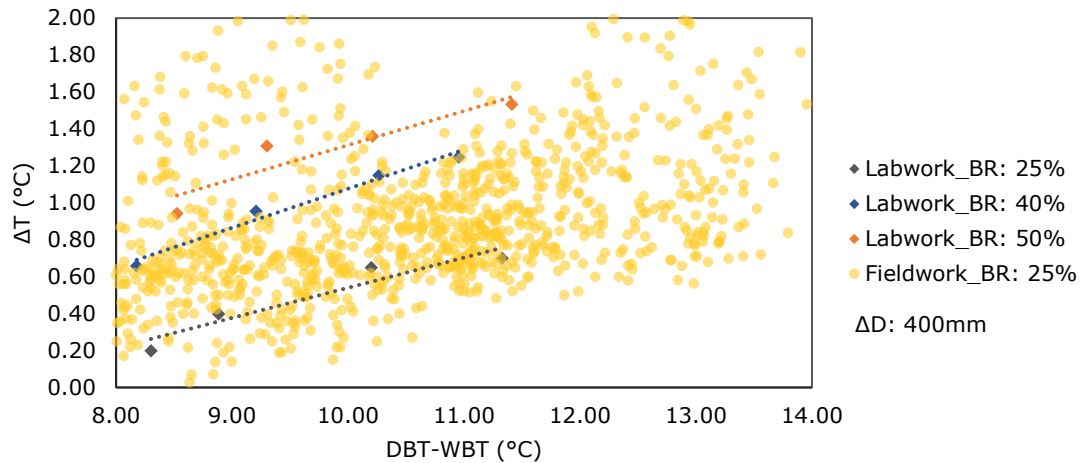


Figure 3.53 Laboratory & Fieldwork data: monitored temperature reduction achieved

3.4 Conclusions

This chapter described the design and performance evaluation of optimised PCC systems. The prototype P1 was first designed presenting a high effective surface area to volume ratio ($\text{ESA:V} = 234 \text{ m}^{-1}$) and a low drag coefficient ($C_d=0.52$) due to its aerofoil shape. Unfortunately, a series of technical and economical limitations were found during the fabrication process that prevented its fabrication, confirming that digital fabrication techniques have long way to compete the cost-effectiveness of traditional ceramic production techniques. The second prototype P2 demonstrated that optimised ceramic designs can be manufactured using traditional mass-production methods. It consisted in tubular-shaped ceramics providing a low drag coefficient ($C_d = 0.50$) to reduce resistance to flow over its boundary layer and engraved spiral-shaped channels on its surface to allow a uniform water distribution throughout the ceramic surface.

These spirals allowed an even water distribution over the ceramic surface by transporting water from top to bottom of bottom. The water supply system consisted in a top-drip irrigation system and the conducted hydraulic capacity studies confirmed that ceramic surfaces can absorb water in all directions

without any visible impact of gravitational force, air and water temperature. The proposed omnidirectional water transport on the ceramic surface was proved to be more efficient than unidirectional methods, which presented a poor water absorptance rate of 0.2 to 0.7 mm/min. Laboratory studies demonstrated that the total unidirectional water expansion after four hours of analysis period was below 10cm, making this method not feasible for an evaporative cooling system using conventional ceramics. Instead, further studies using omnidirectional absorption methods in prototype P2 confirmed that water expansion on the ceramic surface can be expressed as function of time: $d = 2.7034 \cdot T^{0.618}$, where d is the expanded distance of water in mm and t is the time of absorption in minutes (for extruded ceramic elements with average porosity of 15% and density of 1900 kg/m³); and engraved spirals with angles below 40 degrees can wet more than 80% of the total surface area in less than 60 minutes. Moreover, the nature of the system allowed the ceramics to fully dry over non-operational hours, reducing the risk of Legionella growth.

Cooling capacity studies confirmed the significant impact of geometry on the evaporation rate, highlighting the proportional and inversely proportional relationship of the ceramic effective surface area (ESA) and drag coefficient (C_d) respectively. Further studies evaluating the cooling capacity of different arrangements of prototype P2 confirmed a linear relation between DBT-WBT and the achieved temperature depression and humidity increase downstream the ceramics. The collected data confirmed that air temperature reductions of 2 degrees are only achievable under warmer and dryer conditions (DBT-WBT greater than 12°C), and estimated water consumption for prototype P2 ranged from 2-4 l/m²·day (considering 8 operational hours per day). This number is comparable to the average water consumption of one misting nozzle (4 l/day) commonly used in public outdoor spaces across Mediterranean countries, making PCC systems a competitive alternative requiring lower maintenance cost while improving the architectural quality of the space.

These results were compared against monitored data from an experimental mock-up in Granada, Spain (37.18N), in which temperature reductions up to 2°C downstream the ceramics were also observed under the hotter and dryer hours of the monitoring period. The data collected from fieldwork allowed to identify a wider amplitude of $\pm 0.3^\circ\text{C}$ in the achieved air temperature reduction when

compared to laboratory results, mostly associated to uncontrolled air velocity fluctuations.

Moreover, the observed ceramic surface temperatures during the experimental studies ranged from 3 to 5°C above ambient wet bulb temperatures. Further studies conducted using an infrared thermal camera and a black-globe thermometer confirmed that warmer and dryer ambient temperatures lead to higher mean radiant temperature depressions. The results also suggested that the radiant intensities obtained are similar to the convective and evaporative cooling components, achieving temperature reductions up to 2°C when ambient DBT-WBT is over 12°C. However, a small area of influence of radiant cooling was identified from the studies, as the temperature depression beyond 2 meters' distance downstream the ceramics hardly exceeded 1°C. In spite of this, the mean radiant temperature reduction obtained pointed up the importance of surface temperatures as a determining factor in the evaluation of thermal comfort in the proximities of the system.

4 Numerical Model for Performance Evaluation

Computational Fluid Dynamic techniques are very powerful and spans a wide range of industrial and non-industrial application areas (Versteeg and Malalasekera, 2007). Many studies (Barzegar et al., 2012) have been done to simulate evaporative cooling under different scenarios, including cooling pads, water ponds, misting systems and water films. However, examples on the prediction of performance of Porous Ceramic Cooling systems are very limited.

The main objective of this chapter was to establish a valid numerical model using the commercial CDF software Ansys Fluent to simulate the physics of evaporation on wetted ceramic surfaces, taking into account all the physical and chemical processes involved. It was also aimed that this model will provide with detailed information regarding the performance of any ceramic design for any context and ambient conditions, bringing the opportunity to evaluate prototypes prior to manufacture and assist during the design process of PCC systems.

Secondary objectives involved the performance prediction of the ceramic prototypes P1 and P2, in which design considerations to maximise cooling capacity were applied. This exercise aimed to reaffirm the direct impact of geometry (ESA and Cd) on the cooling capacity of the ceramic prototypes, and to evaluate the achieved increment in performance of the optimised prototype P1 when compared to other samples not designed for this purpose. Moreover, the computational work aimed to interrogate in more detail prototype P2 performance under different boundary and environmental conditions, allowing a better understanding of the physics involved during the evaporation process.

To address these objectives, this chapter introduces to the process of CFD simulation, describing each of the steps constituting the simulation workflow. A review on previous work and developed numerical models of passive evaporative cooling systems informed the definition of a CFD model for PCC systems, involving phase change and mass particle transfer in a transient model. The obtained CFD model was used to structure a parametric study that responded the main objectives initially set.

4.1 Simulation of Evaporative Cooling Processes

Numerical models of evaporative cooling processes can provide with detailed information regarding the performance of the systems. Moreover, the opportunity to evaluate the efficiency of a prototype prior to manufacture makes this practice appealing and supportive in the design process. Further to the specific model outputs explored in this chapter, CFD tools can provide with a wide range of results, including data visualization of pressure gradients over the ceramic elements, air humidity and temperature distribution downstream the ceramics, quantification of the rate of evaporation and water consumption, and the equivalent sensible and latent cooling provided.

Despite the numerous advantages of this practice, the application of these tools and the creation of customized models is limited to users with expertise in the field. Design considerations, simulation processes and model examples of evaporative cooling processes are explored in this section.

4.1.1 CFD Workflow

Most commercial CFD packages are structured around three main elements to ease the problem-solving process of fluid flows: pre-processor, solver and post processor (Figure 4.1). As described by Versteeg and Malalasekera (2007), the pre-processor process consists of the input of the flow problem through the definition of the fluid domain, the domain subdivision (mesh) and the specification of fluid properties and boundary conditions. In the simulation of evaporative cooling processes, at least two fluids must be defined: water and moist air, and the number of fluid domains required may vary according to evaporative cooling system used. The solver uses the *finite volume method* to ensure the conservation of relevant properties (momentum, mass, energy, etc.) for each finite size cell (Varga, 1962). This is achieved through the definition of the governing equations that embed the physics (and chemistry) present in the problem, and a discretization phase that converts the resulting integral equations into a system of algebraic equations. The solution of the algebraic equations is then achieved by an iterative method for both steady state and transient models. In transient simulations, where the relevant properties are time-dependent, the discretization process is augmented over the defined time step. This type of simulation is particularly effective in evaporative cooling models to track water particles in the domain. The post-processor stage allows the user to read and visualise the results. Most commercial CFD packages

include outstanding graphics capabilities for data visualization (vector, contour, surface plots, particle tracking, animations, etc.), and numerical operations such as surface or volume integrals to generate alphanumeric reports.

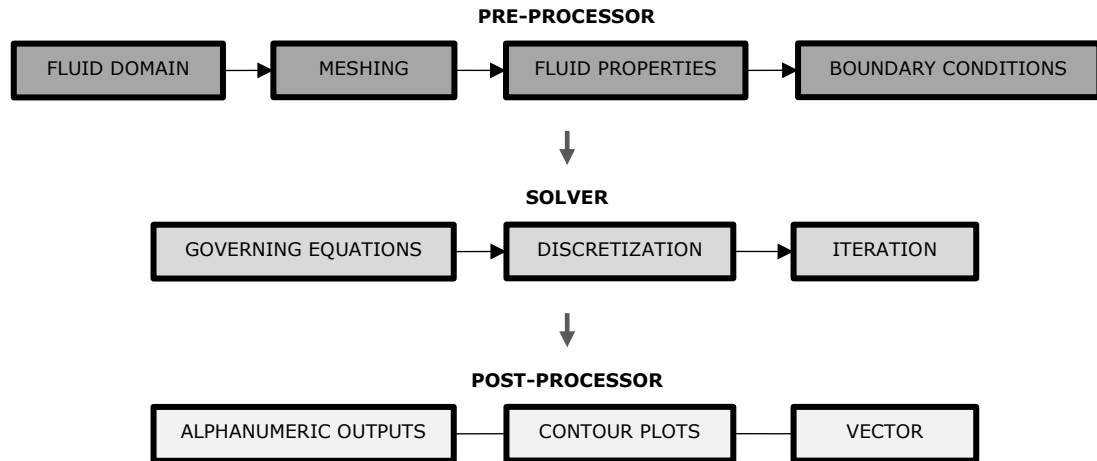


Figure 4.1 CFD simulation workflow

4.1.2 Passive Evaporative Cooling Models

The growth of computer processing power in the past two decades allowed the performance of high calculation-intensive tasks (Roser and Ritchie, 2018). CFD simulation is now accessible to a wider audience and complex numerical models involving multiple physics and chemical processes in large (single or multi-phase) domains allow the reproduction of almost any natural process happening in everyday life. The simulation of evaporative cooling processes has also been further developed, and in the last decade multiple approaches have been proposed and validated for most application systems, some of which are described in this section. The preparation and definition of evaporative cooling models increase in complexity and special attention must be paid to the following aspects:

- **Physics involved.** Underlying the physics is perhaps the most complex stage of the process and the results obtained are at best when all physics (and chemistry) involved are embedded in the model. In most cases this requires certain level of expertise in a number of areas. Apart from fluid flow, the evaporative process involves a phase change and mass transfer from water in liquid state to vapour (latent heat of vaporization). In

addition, convective and conductive (if present) heat transfer must be considered, and radiative heat exchange may be neglected in certain scenarios. Moreover, the solution requirements should also determine whether a steady state or transient simulation is needed. Despite some turbulent flows can be statistically stationary, transient simulations are particularly effective to track water particles, temperature gradients or evaporation rates over the time.

- Fluid domains. The domain definition sets the boundaries of the different fluids present in the numerical model. In evaporative cooling models, moist air will be the main fluid and its domain should be equivalent to the conditioned space. The thermal properties of moist air need to be defined and must vary in density and moisture content to take into account buoyancy driven airflows. The presence of water and its interaction with moist air must also be carefully considered. Evaporation can take place on large water bodies, small water particles or over porous surfaces. Additional fluid domains (leading to multi-phase simulations), *Discrete Phase Models* or *Eulerian Wall Films* (ANSYS®, Release 17.2b, Chapter 16, 19) can be used to integrate large or small volumes of water within the domain.
- Meshing. The quality of the mesh is a critical aspect of the overall simulation since it has a direct impact on the accuracy of the model, results convergence and solver run time. It can be evaluated by looking at the average mesh skewness and orthogonality, which can range from 0 to 1. Skewness determines how close to ideal (equilateral or equiangular) a face or cell is and for an excellent mesh quality the average skewness must range from 0 to 0.25. Orthogonality evaluates the normal vectors for each cell face and for best mesh quality the average orthogonality must range from 0.75 to 1 (ANSYS®, Release 17.2a). In addition, evaporative cooling systems where evaporation takes place on wetted surfaces require velocity gradients to be well captured at the walls. In order to achieve this, a mesh inflation to resolve the boundary layers is crucial in the meshing process.
- Turbulence models. Turbulence flows are commonly observed in everyday phenomena and are characterised by the velocity and viscosity of the fluid. They can be predicted by the Reynolds number constant, which calculates the balance between kinetic energy and viscous damping of fluid flow, and several turbulence models are now available to allow the evaluation of different flow regimes (ANSYS®, Release 17.2b, Chapter 4).

The $k-\varepsilon$ model is the most widely used for heat transfer and industrial applications. It solves for two variables: k , the turbulent kinetic energy, and ε , the rate of dissipation of kinetic energy, and rapidly popularised due to its good convergence rate and relatively low processing requirements. However, alternative models may offer better performance and convergence rate for certain scenarios. For instance, studies have proved that the turbulence model $k-\omega$ is sustainably more accurate than the standard $k-\varepsilon$ in the near wall layers (Menter, 1993), making it more suitable for evaporation processes taking place on wetted surfaces.

Cooling Pads

Numerical models for evaporative cooling pads emerged thanks to the rapid implementation of greenhouse structures in the Mediterranean (Chapter 2.2) and the consequent need of system optimization. The large concentration of greenhouses in Almeria, Spain led to Franco et al. (2010) to develop a numerical model for the optimization of the geometry of the pads by exploring its resistance to flow, directly related to savings in fan energy consumption (commonly needed to drive air into the greenhouses). The authors contributed with the definition of the volume of water retained in the pads in the numerical model, as well as with valuable information regarding the impact of draft and porosity on the performance of the tested pads (Figure 4.2). Other authors (Osorio-Saraz et al., 2012), characterized the pads in terms of specific mass, porosity, average diameter and surface area, and implemented a species mass transfer by defining the water diffusion coefficient (speed at which water molecules are transported to air) from Incropera and DeWitt (1985). The results obtained helped to identify the best performing pad under the tested environmental conditions and concluded that the system efficiency tended to be greater for thinner pads.

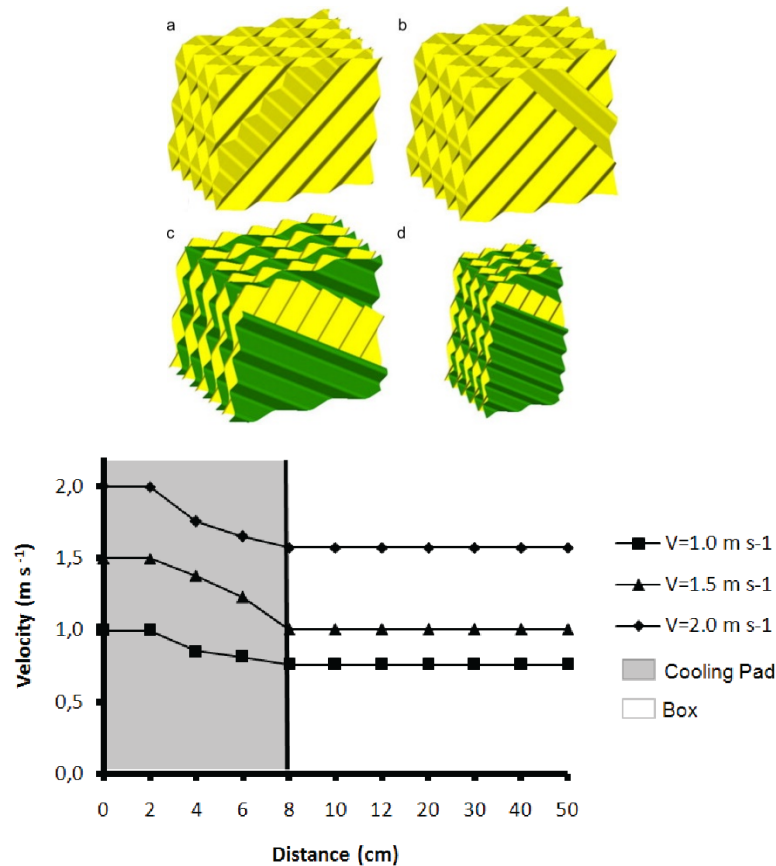


Figure 4.2 Evaporative pads analysed (Franco et al., 2011p. 221) and air velocity distribution downstream the cooling pads (Osorio-Saraz et al., 2012p. 215)

Cooling Ponds

Cooling ponds are, perhaps the most frequent and simple application of evaporation in outdoor spaces. The rate of evaporation and cooling capacity is difficult to predict and only possible by means of finite element calculations. Ramadan et al. (2014) developed a numerical model that determined water evaporation on a pond surface. The model required a multi-zone model consisting of three domains for the vapour zone, the air zone and an opaque concrete surface. The vapour zone consisted in a thin volume representing the boundary layer just above the water body in which the *Species Transport Equations* were applied (ANSYS®, Release 17.2b, Chapter 7). The air zone represented the immediate outdoor air volume exposed to a wind-driven airstream. Results were validated against Li (2005) and allowed the visualization of humidity contours and temperature distribution within the domain (Figure 4.3). Recent work by Blázquez et al. (2017) also proposed a multi-component flow model with species transport in which a mixture of ideal gases (dry air and water vapour) were considered. Thermodynamic properties of each component

and the diffusivity of water vapour in air were temperature-dependent and also defined according to Incropera and DeWitt (1985). Buoyancy forces were modelled using the *Boussinesq* approximation (ANSYS®, Release 17.2b, Chapter 4) and results were successfully validated against four experimental tests with different geometries and scales in stationary conditions.

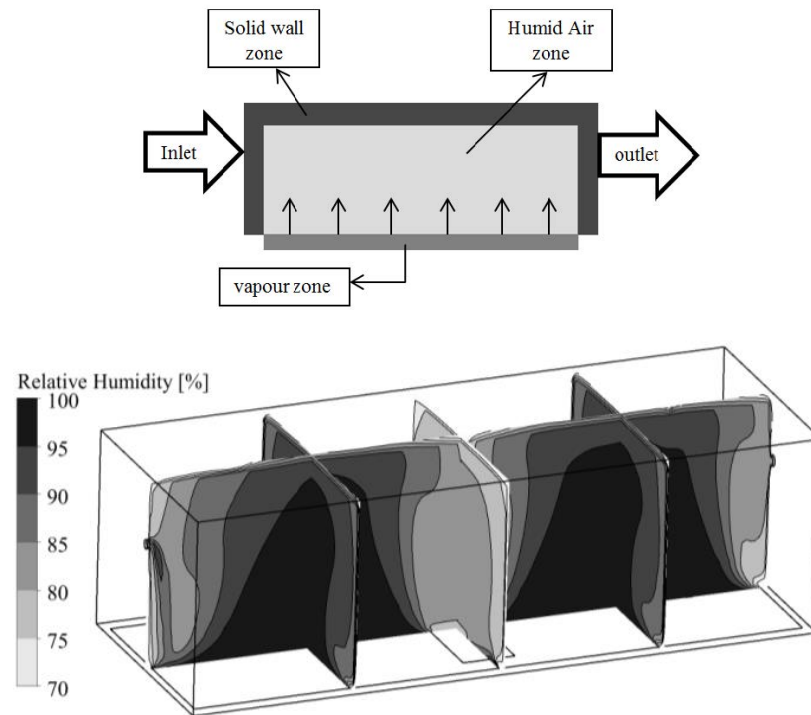


Figure 4.3 Schematic diagram of the three-domain model of a pond and relative humidity contours within the domain (Ramadan et al., 2014, pp. 2, 4)

Mist Systems

The numerical model of misting evaporation consists on the evaporation of small droplets/water particles (secondary phase) dispersed in the air as a continuous fluid phase (primary phase). With the rapid increase in the application of water-based cooling systems to improve outdoor comfort in public spaces (Nishimura et al., 1998), a strong interest emerged in determining advanced numerical models for performance evaluation. Several authors have contributed in the past two decades to the definition of a mist system numerical model (Cook et al., 2000, Erell et al., 2006, Li and Wang, 2006, Montazeri et al., 2015) resulting in a two-phase system in which droplets were discrete phase entities, with their corresponding trajectories and heat and mass transfer to/from them (Figure 4.4). The discrete phase trajectory was calculated using a *Lagrangian* formulation that includes inertia, aerodynamic drag and the force of gravity

(Zhang and Chen, 2007). In the models, water droplets were injected in the domain in downstream direction applying the *atomizer model* provided by ANSYS® (Release 17.2b, p. 447), and the size of the droplets usually ranged from 50 to 600µm distributed using the *Rosin-Rammler Diameter Distribution Method* (ANSYS®, Release 17.2c, Chapter 16.12.1.5). The discrete phase models also implemented a species mass transfer formulation for the prediction of air moisture increase.

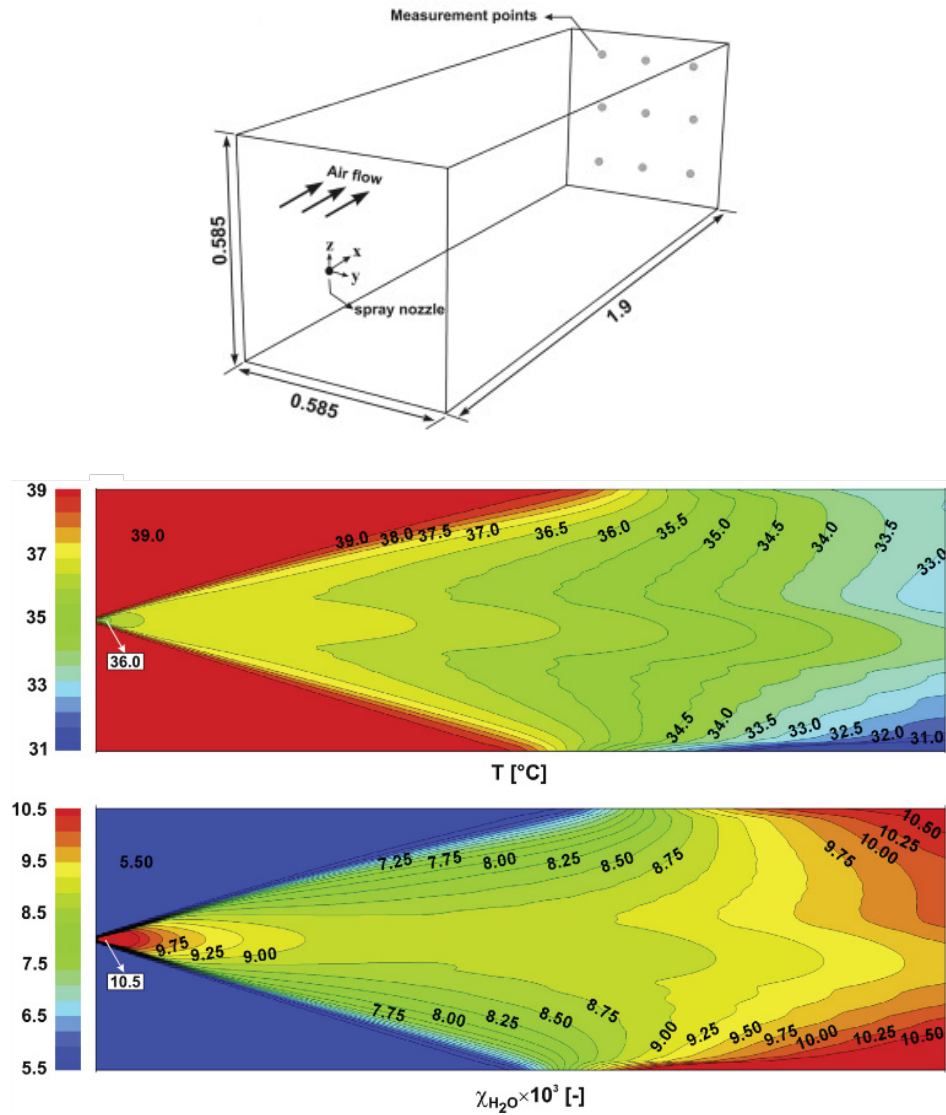


Figure 4.4 Computational domain (units in metres) and temperature and vapour mass fraction distribution in cross-section of the numerical model developed by Montazeri et al. (2015, pp. 131, 136)

Water films

Water films are widely adopted in process industry for high heat and mass transfer efficiency (Ambrosini et al., 1995). The numerical model developed by Ambrosini et al. (2001) characterised the evaporation of water films on metallic walls of nuclear reactors for cooling purposes. The liquid film, with a given thickness, was formulated according to the *Nusselt theory* (Nusselt, 1943), presenting a purely conductive thermal behaviour and neglected velocity. This is particularly appropriate when water particles are treated as a continuous entity, simplifying the calculation process. The method accounted for species mass transfer with molecular mass diffusivity obtained using *Chapman-Enskog* relationships (Chapman et al., 1970). Similar approaches have also been applied by automotive and aeronautic companies to simulate surface evaporation and also, the inverse process of condensation. The published example by ANSYS® (Release 15.0) introduced a numerical model to analyse the de-fogging process of windshield panes in vehicles. The model used the *Eulerian Wall Film* method, which applied a virtual film with variable thickness that flows parallel and in contact to the pane surface without the need of further meshing processes and momentum calculations. This model was also coupled with a species transport model to allow the liquid film to evaporate and increase the moisture content of the air. The results from the transient simulation illustrated the distribution of the film thickness over the pane at different times periods (Figure 4.5).

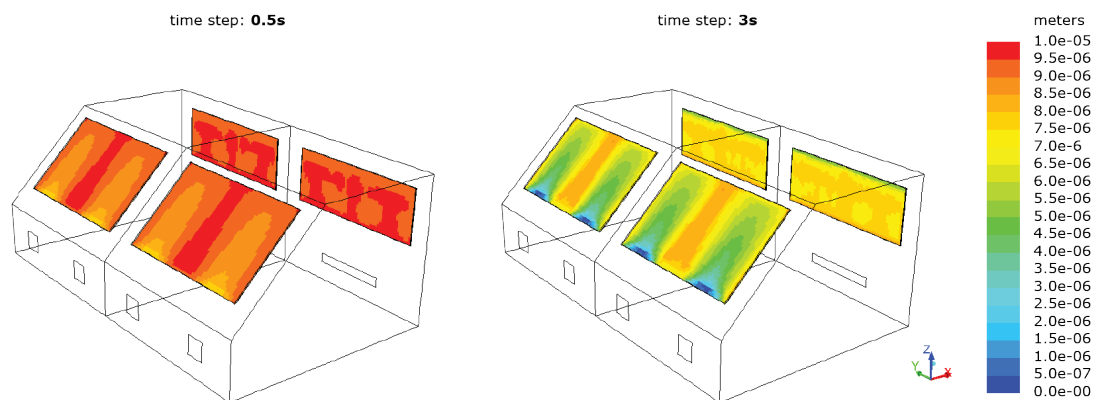


Figure 4.5 Distribution of film thickness over the windshield pane at different times periods
(ANSYS®, Release 15.0, p.17)

The latter models are equivalent to the process of evaporation that takes place on the surface of wetted porous materials and were used as a reference to define a numerical model for PCC Systems in next section.

4.2 The Numerical Model for Evaporative PCC Systems

The numerical model of PCC systems is defined by the governing equations that integrate the physics involved in the evaporation process, the boundary conditions determining the space configuration and material properties and meshing. Each of them is described in detail in this section.

4.2.1 Governing Equations

The governing equations for the fluid flow represent mathematical statements of the conservation laws of physics are those related to continuity, momentum, energy and species. These equations are written as follows:

- Mass conservation equation:

$$S_m = \frac{\partial \rho}{\partial t} + \nabla(\rho \vec{v}) \quad \text{Equation (4-1)}$$

Mass conservation
ANSYS® (Release 17.2b, p. 2)

- Momentum conservation equation:

$$\frac{\partial}{\partial t}(\rho \vec{v}) + \nabla(\rho \vec{v} \vec{v}) = -\nabla p + \nabla(\vec{\tau}) + \rho \vec{g} + \vec{F} \quad \text{Equation (4-2)}$$

Momentum conservation
ANSYS® (Release 17.2b, p. 3)

- Energy equation:

$$\frac{\partial}{\partial t}(\rho E) + \nabla(\vec{v}(\rho E + p)) = \nabla \left(k_{eff} \nabla T - \sum_j h_j \vec{J}_j + (\vec{\tau}_{eff} \vec{v}) \right) + S_h \quad \text{Equation (4-3)}$$

Energy conservation
ANSYS® (Release
17.2b, p. 15)

- Species transport equation:

$$\frac{\partial}{\partial t}(\rho Y_i) + \nabla(\rho \vec{v} Y_i) = -\nabla \vec{J}_i + R_i + S_i \quad \text{Equation (4-4)}$$

Species transport

ANSYS® (Release 17.2b, p. 189)

In the above equations, ∇ represents the partial derivative of a quantity with respect to all directions in the Cartesian coordinate system. In Equation (4-1), S_m is the mass added to the continuous phase from the dispersed second phase. In Equation (4-2), p is the static pressure; $\bar{\tau}$ is the stress tensor and $\rho \vec{g}$ and $\vec{F}$ are the gravitational body force and external body forces respectively. In Equation (4-3), k_{eff} is the effective conductivity, and $\vec{J}_j$ is the diffusion flux of species. In Equation (4-4), R_i is the net rate of production of species by chemical reaction and S_i is the rate of creation by addition from the dispersed phase (ANSYS®, Release 17.2b, Chapters 1, 7).

In Equation Equation (4-3), the diffusion flux of species $\vec{J}_j$ uses *Fick's Law* to model mass diffusion (ANSYS®, Release 17.2b, Chapter 7.1.1.1), which can be written as:

$$\vec{J}_j = -\rho D_{i,m} \nabla Y_i - D_{T,i} \frac{\nabla T}{T} \quad \text{Equation (4-5)}$$

Fick's Law

ANSYS® (Release 17.2b, p. 190)

where $D_{i,m}$ is the mass diffusion coefficient. *User Defined Functions* (UDF) were applied to define the mass diffusivity as a function of temperature and pressure, obtained from Incropera and DeWitt (1985) and written as:

$$D_{i,m} = \left(\frac{0.926}{P} \right) \left(\frac{T^{2.5}}{T + 245} \right) \quad \text{Equation (4-6)}$$

H₂O mass diffusivity

Incropera and DeWitt (1985, p. 937)

The Eulerian Wall Film (EWF) model was also used to emulate a thin water film on the surface of the ceramics. The EWF model was coupled with the mixture species transport model to consider phase changes between water (liquid) and the gas species (vapour). The rate of phase change is governed by:

$$\dot{m}_{phase} = \frac{(\rho D / \delta)}{\rho D / \delta + C_{phase}} C_{phase} (y_{sat} - y_i) \quad \text{Equation (4-7)}$$

Eulerian wall film model
ANSYS® (Release 17.2b, p. 645)

where ρ is the density of the gas mixture, D is the mass diffusivity of the vapour species, δ is the cell centre-to-wall distance, C_{phase} is the phase change constant, and y_i and y_{sat} represent the cell-centre mass fraction of the vapour species and saturation species mass fraction respectively (ANSYS®, Release 17.2b, Chapter 19.2.1.6.).

The turbulent flow was modelled from the *SST k- ω* turbulence model. The shear stress transport (SST) formulation applies the *k- ω* formulation in the boundary layers and the *k- ϵ* behaviour in the free-stream, presenting very good behaviour in adverse pressure gradients and separating flow (Menter et al., 2003).

4.2.2 Meshing

The meshing component from the commercial software Ansys Fluent was used to generate a mesh of tetrahedrons and hexahedrons with advanced sizing function of proximity and curvature. The minimum face size for all models was set to 0.2 mm and the maximum to 60 mm. An inflation method with 10 maximum layers and smooth transition was also applied over the ceramic walls in order to capture air velocity and air temperature gradients in the proximities of the ceramic surface.

The obtained meshes presented a good quality with an average orthogonality of 0.75 and skewness of 0.30. The total number of nodes varied according to the complexity of the ceramic shapes tested, ranging, for instance, from 230,000 to 1,000,000 nodes for Prototype 1 (Figure 4.6).

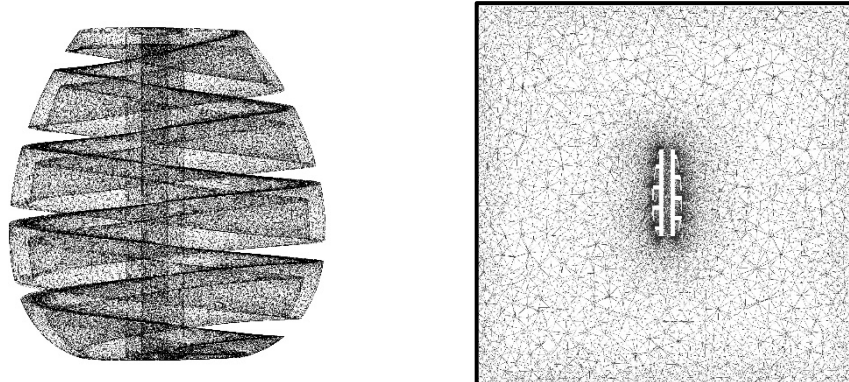


Figure 4.6 P1 surface and cross section mesh

4.2.3 Boundary Conditions

The fluid domain consisted in a 1000x1000x2000 mm rectangular prism with water-vapour (solid + liquid mixture) as main fluid material. One of the faces of the volume was defined as inlet and air velocity, temperature and humidity were set according to the parametric study performed. The opposite face of the volume was defined as outlet, where Gauge pressure was set to 0 Pa. The side walls of the fluid domain are solid stationary walls with wall temperature constant and equal to the provided air temperature. The opaque ceramic volume is placed at 500 mm on the normal direction of the inlet surface. The ceramic material was defined by its density (1900kg/m³), specific heat (840 J/kgK), and thermal conductivity (0.72 W/mK). The ceramics' wall temperatures were constant and set according to the performed parametric studies (Chapter 4.3). Over the ceramic walls, a water-liquid film was applied with a constant temperature also set according to the parametric studies. The film height was determined from the retained water volume-to-effective surface area ratio of the ceramics. The film heights for each ceramic sample are shown in Table 4.1

The model is a transient simulation with a time step defined from Equation (4-8). The number of iterations per time step were set to 20. Convergence was achieved for each timestep, requiring less than 20 iterations per step.

$$\Delta t = \frac{\text{typical cell size}}{\text{characteristic flow velocity}} \quad \text{Equation (4-8)}$$

*Recommended time step for transient simulations
Versteeg and Malalasekera (2007)*

Overall, the numerical model can be described as follows: water film over the ceramic surface evaporates when exposed to air movement from the inlet normal direction. When increased water vapour pressure, a phase change takes place over the ceramic surface and mass transfer occurs from the water liquid film to the water vapour fluid domain (Figure 4.7).

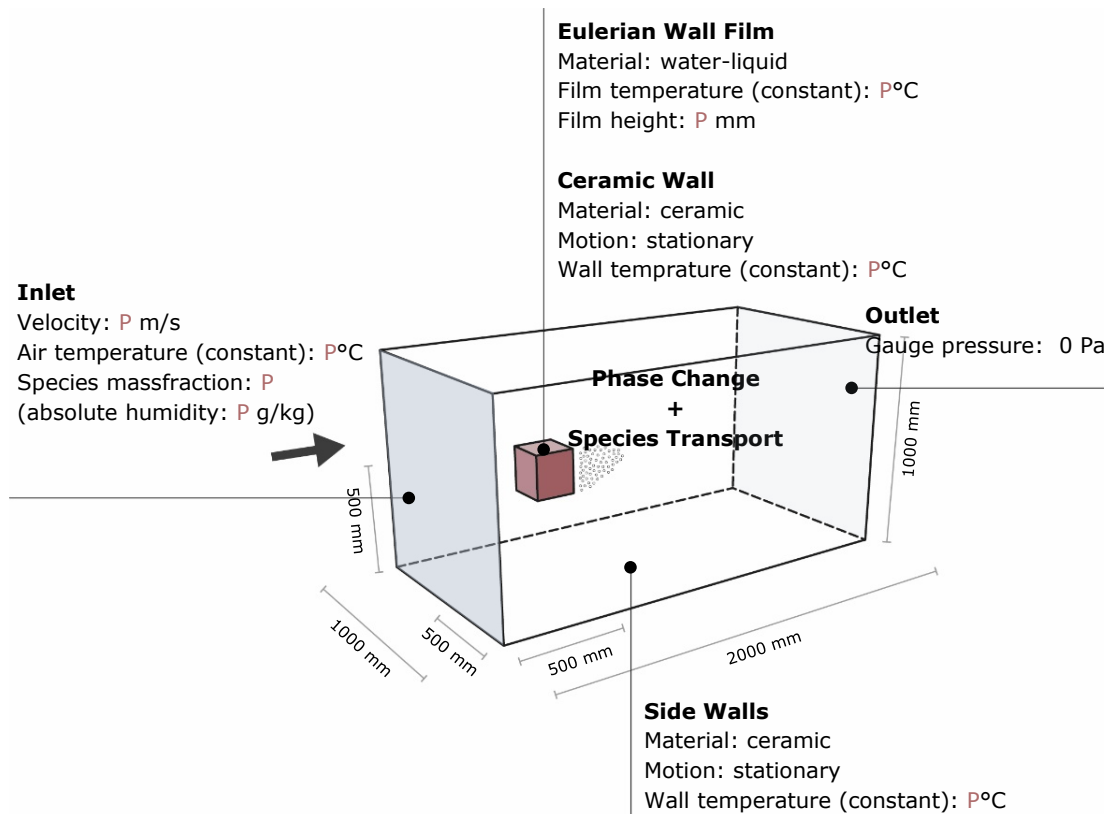


Figure 4.7 Boundary conditions for CFD studies, where P is determined from the parametric study performed

4.3 Parametric Studies

In order to interrogate in detail performance prediction of prototypes P1 and P2 under different environmental conditions, the computational work was structured in two sets of parametric studies.

The first set of parametric studies addressed the impact of environmental variables (DBT-WBT, DBT-H₂O, v and t) on the temperature reduction and evaporation rate achieved at different distances from the rear ceramic surface. The studies were performed with Prototype P2 and the reference conditions were set to DBT-WBT=10°C, DBT-H₂O=10°C, v =0.50m/s and t =4s. Different values

for each parameter were then changed as shown in Figure 4.8 to allow comparative studies.

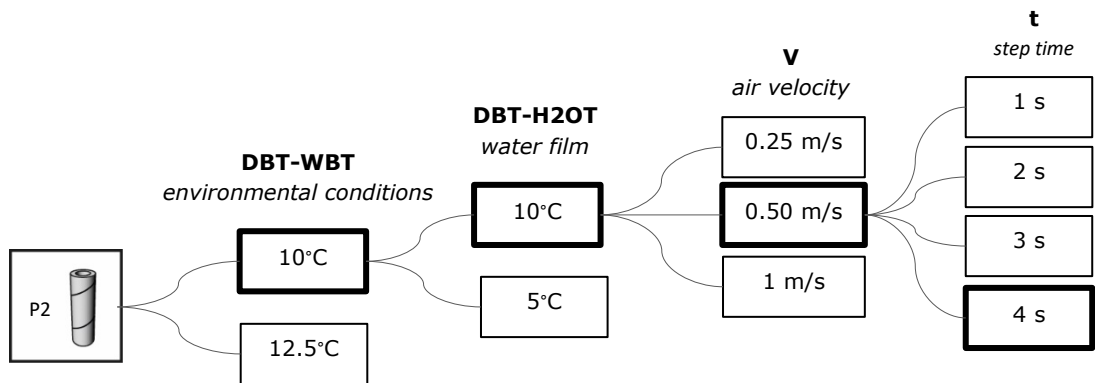


Figure 4.8 Parametric studies performed with P1 using CDF simulation tools

The second set of parametric studies evaluated the impact of geometry on the temperature reduction and evaporation rate achieved at different distances downstream ceramic surfaces. Presenting the same volume, the samples P1, P3, S2, S4 and S5 were exposed to the reference environmental conditions as shown in Figure 4.9.

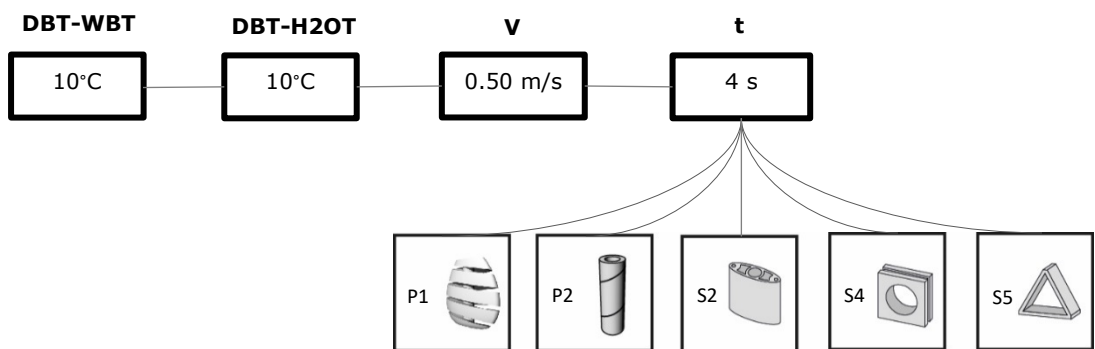


Figure 4.9 Parametric studies performed using CFD simulation tools

To allow an impartial comparison between the ceramic samples, their geometry was scaled to a unit of volume. After scaling the samples, the ceramic still preserve their effective surface area-to-volume ration (ESA:V).

The drag coefficient was used to quantify the drag or resistance of the ceramic shapes in the fluid environment. It was calculated using the following expression:

$$C_d = \frac{F_d}{0.5\rho Av^2} \quad \text{Equation (4-9)}$$

Drag coefficient
McCormick (1979, p. 24)

where F_d is the drag force [N], ρ is the fluid density [kg/m³], A is the projected area on the streamline direction [m²], and v is the air velocity [m/s].

A summary with technical information on the ceramic samples is shown in Table 4.1.

Table 4.1 Properties of ceramic samples (digital models)

	Surface area (m²)	ESA (m²)	Volume (m³)	ESA:V (m⁻¹)	Density (kg/m³)	Cd	ESA/Cd (m²)	Film height (mm)
P1	0.253	0.235	0.001	234.879	1900	0.52	0.454	1
P2	0.117	0.067	0.001	67.445	1900	0.50	0.135	5
S2	0.188	0.080	0.001	79.750	1900	0.45	0.176	2
S4	0.102	0.056	0.001	55.628	1900	1.13	0.049	5
S5	0.153	0.148	0.001	148.314	1900	1.12	0.132	2

4.4 Performance Evaluation

Results were post-processed using contour, surface plots, and numerical operations to generate alphanumerical reports. Air temperature reduction and humidity increase were obtained from each mesh node located in the downstream direction from the ceramic centroid.

4.4.1 Impact of Environmental Conditions on Cooling Performance

Impact of time

A transient simulation allowed observations on the dynamic behaviour of the model. Variation in air temperature and humidity were observed at different time

steps (1, 2, 3 and 4 s). The results allowed to evaluate the dynamic response of the system and to set an analysis time step for the rest of parametric studies performed.

The achieved air temperature reduction and humidity increase downstream P2 presented minimal variations in $t=3s$ and $t=4s$. When t was set to $2s$, small variations in air temperature and absolute humidity were observed starting at 0.5 m from the rear ceramic surface. The poor results obtained with $t=1s$ are due to the low air velocity (0.5 m/s), so that air only translated 0.5 within the domain (Figure 4.10 and Figure 4.11).

The results confirmed that a reliable time step to observe variations in air temperature and humidity within the fluid domain depends on air velocity and the distance from the rear surface where the data is being analysed. This relation can be expressed as follows:

$$t[s] = \frac{d[m]}{v[m/s]} \quad \text{Equation (4-10)}$$

Warmup simulation time

Following to the results obtained, the time step was set to $4s$ on the rest of parametric studies.

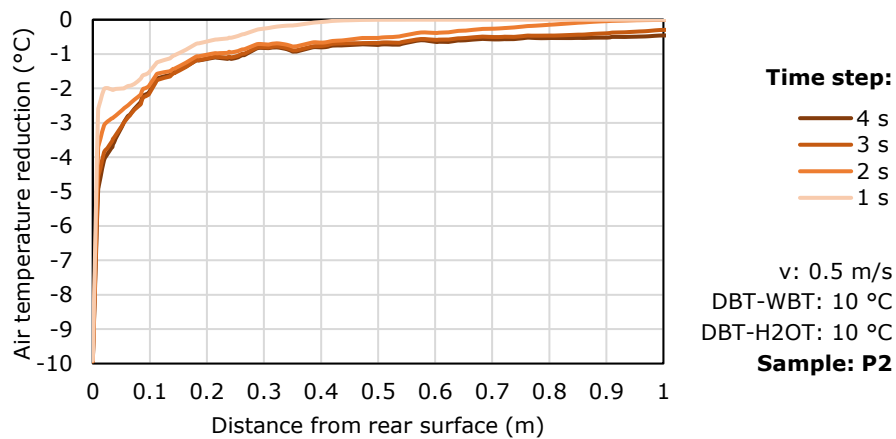


Figure 4.10 Impact of time on air temperature reduction

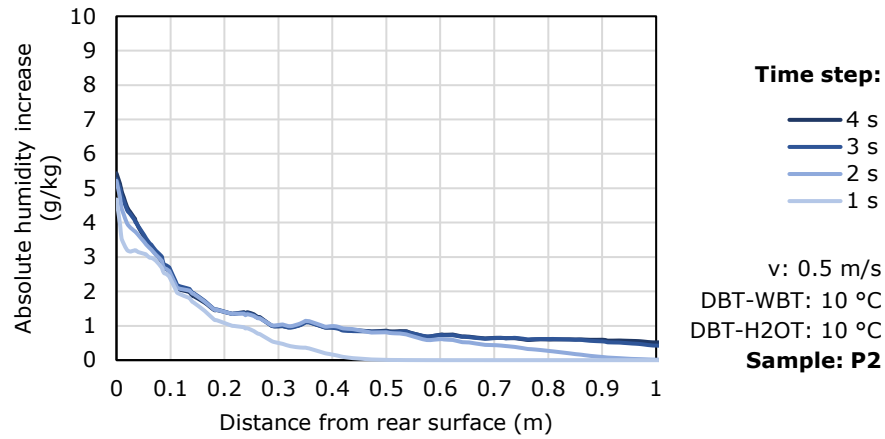


Figure 4.11 Impact of time on absolute humidity increase

Impact of air velocity

Following to the time step studies, different air velocities (0.25, 0.5 and 1m/s) were tested to evaluate their impact on air temperature reduction and humidity increase at different distances from the rear surface of P2.

Small variations in temperature and humidity were generally observed. An air velocity (0.25 m/s) presented the lowest air temperature reduction and humidity increase at distances beyond 0.5m from the rear ceramic surface. Interestingly, an air velocity of 1 m/s presented slightly lower temperature reductions when compared to 0.5 m/s, which presented the best results. The similar humidity levels obtained confirmed that the lower temperatures achieved with $v=1\text{m/s}$ are due to sensible heat transfer and indicated that 1m/s is certainly high for the cooling capacity of the prototype P2 (Figure 4.12 and Figure 4.13).

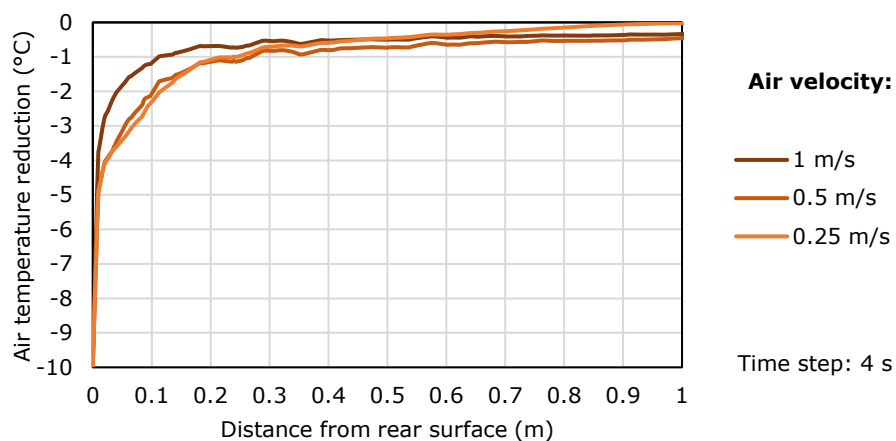


Figure 4.12 Impact of air velocity on air temperature reduction

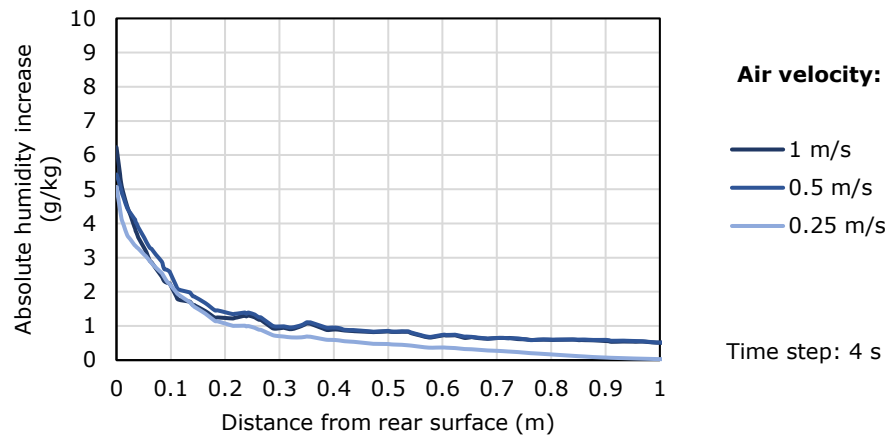


Figure 4.13 Impact of air velocity on absolute humidity increase

Impact of DBT-H₂O

In order to evaluate the impact of water temperature on sensible and latent heat exchange two different dry bulb to water temperature depressions were tested (DBT-H₂O_T = 10 and 5°C).

The results obtained showed significant differences in air temperatures at closer distances to the rear ceramic surface. At 0.5m distance, the temperature reduction ranged from 0.5 to 1°C. However, variations in absolute humidity obtained are considerably smaller (Figure 4.14 and Figure 4.15). Figure 4.15 also shows higher evaporation rate from warmer water (DBT-H₂O_T=5°C). Warmer water temperatures cause molecules to move faster facilitating their separation from the liquid's molecular structure.

This study confirmed that although higher water temperatures will increase the evaporation rate and latent heat, lower water temperatures must be considered in the design of a PCC wall due to the considerable impact on sensible heat.

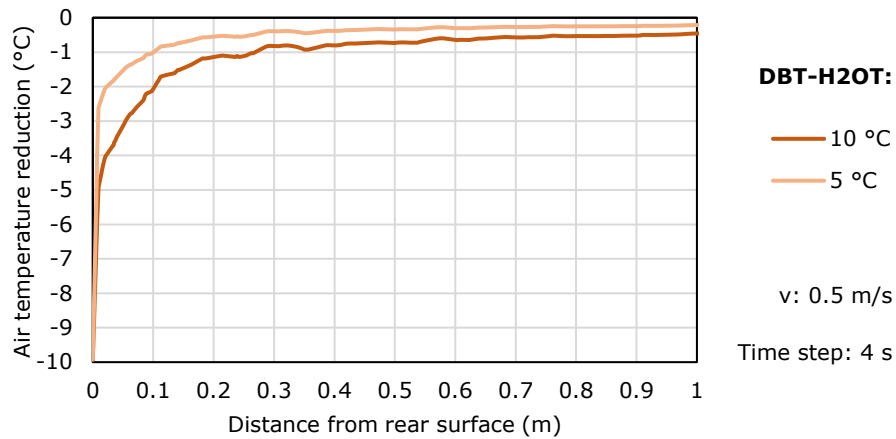


Figure 4.14 Impact of DBT-H2OT on air temperature reduction

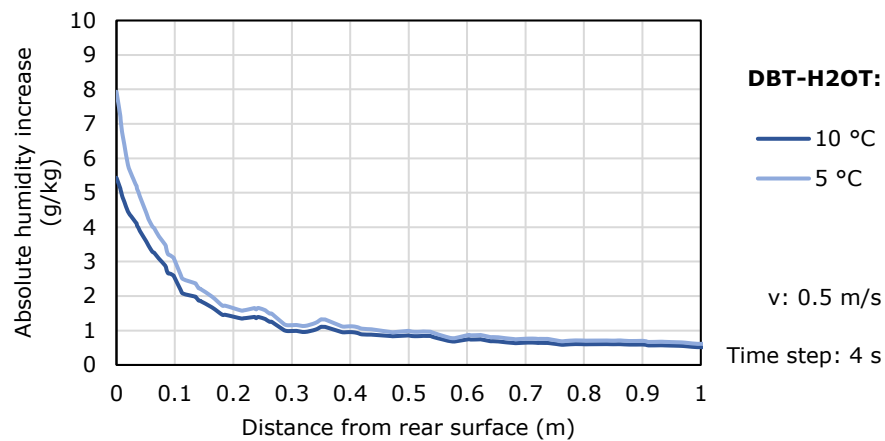


Figure 4.15 Impact of DBT-H2OT on absolute humidity increase

Impact of DBT-WBT

In order to evaluate the impact of different ambient-to-saturated pressure depression on evaporation rates, a parametric study was performed setting dry bulb-to-wet bulb temperature depression (DBT-WBT) to 10 and 12.5°C. A DBT-WBT of 10°C corresponded to an air temperature of 32°C, a relative humidity of 42% and wet bulb temperature of 22°C; whereas a DBT-WBT of 12.5°C corresponded to an air temperature of 35°C, a relative humidity of 34% and wet bulb temperature of 22.5°C.

The results again confirmed a better performance under warmer and dryer conditions. Figure 4.16 and Figure 4.17 also demonstrated that greater air

temperature reductions were associated to a greater evaporation rate (humidity increase downstream the ceramics), confirming a higher latent-to-sensible cooling ration.

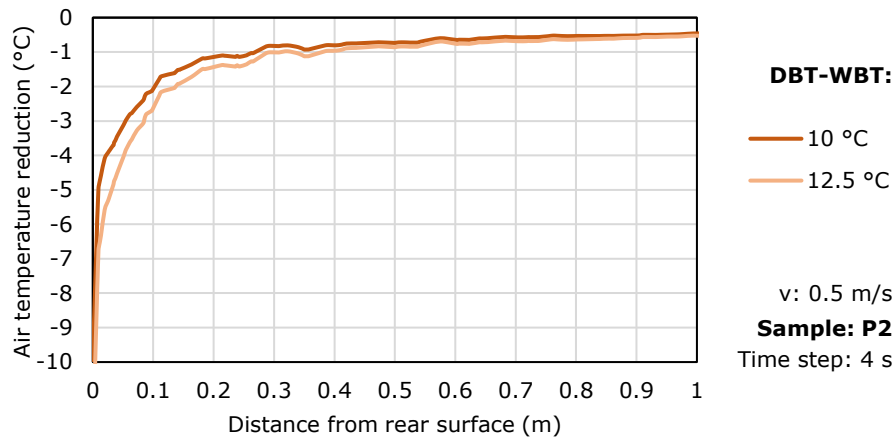


Figure 4.16 Impact of DBT-WBT on air temperature reduction

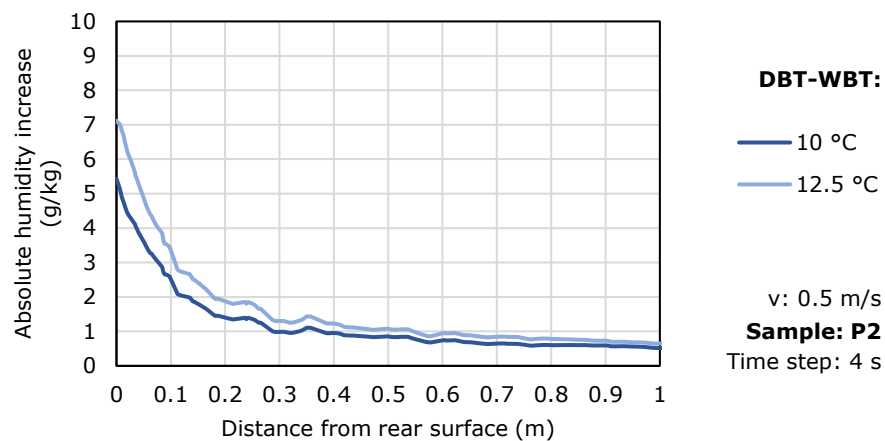


Figure 4.17 Impact of DBT-WBT on absolute humidity increase

4.4.2 Impact of Geometry on Cooling Performance

The last parametric study performed responded to the impact of geometry on air temperature reduction and humidity increase at different distances from the rear ceramic surface.

This study allowed the performance evaluation of prototype P1 and its comparison with alternative ceramic shapes, aiming to confirm a considerable air temperature reduction and evaporation rate associated to shape optimisation.

Figure 4.18 shows the variations in air temperature achieved at different distances from the rear ceramic surface. These temperature variations ranged from 0.5 to 1.5°C at 0.5m distance. With constant boundary and environmental conditions, the temperature variations achieved were strictly related to ceramic geometry and confirmed that the cooling capacity of a porous ceramic cooling systems is proportional to this factor.

Similar behaviour was found in Figure 4.19, where it can be observed that absolute humidity can increase up to 1g/kg with prototype P2. Contour plots of air temperature, humidity and air velocity distribution over a horizontal plane is shown in Figure 4.20.

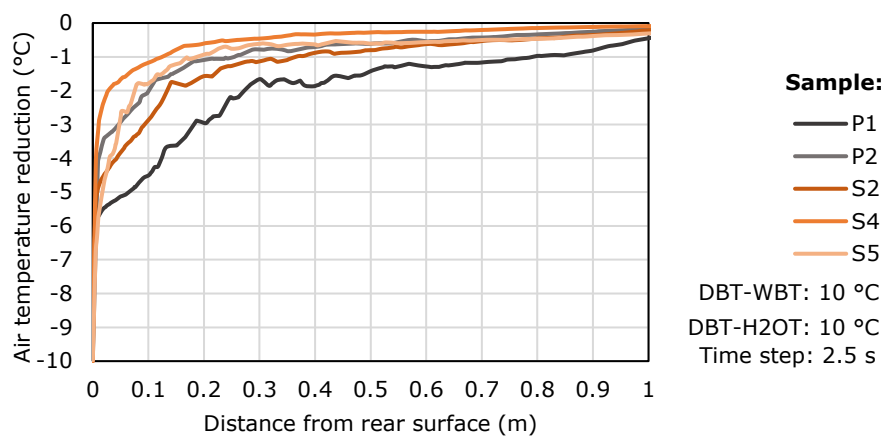


Figure 4.18 Impact of geometry on air temperature reduction

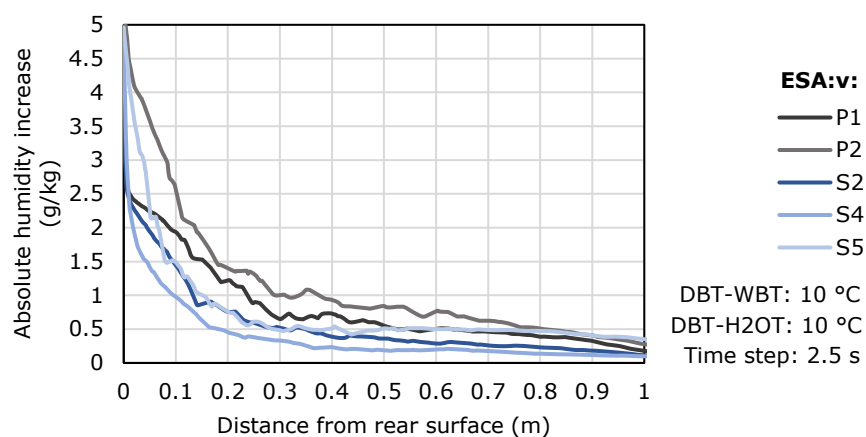


Figure 4.19 Impact of geometry on absolute humidity increase

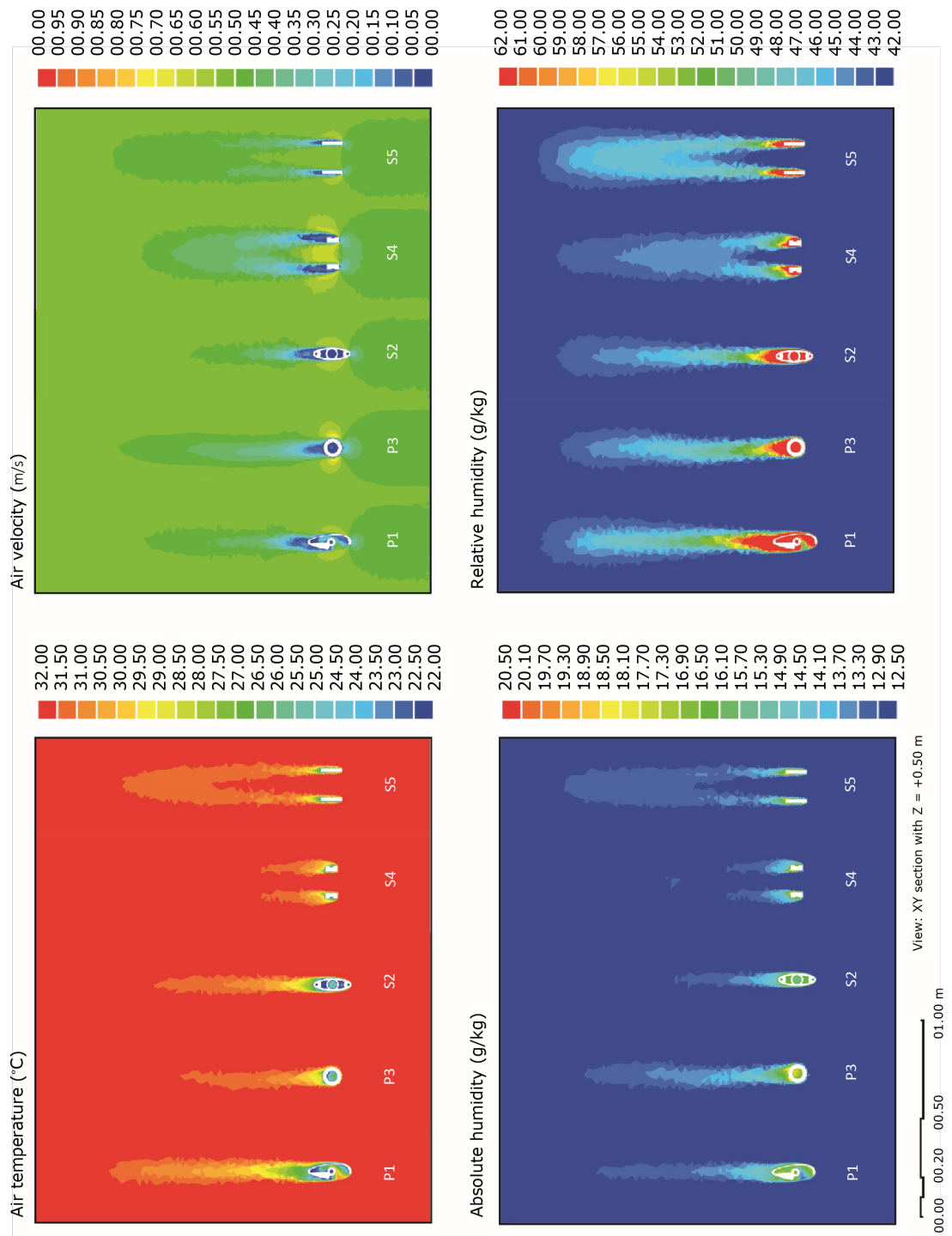


Figure 4.20 Impact of geometry on air temperature reduction, humidity increase and air velocity variations

4.5 Conclusions

CFD simulations are robust tools that help designers to predict the performance of ceramic systems prior to manufacture. Appropriately used, they can help assist in the design process of novel ceramic prototypes and systems. The described governing equations and boundary conditions in 4.2 proved to be valid for the simulation of evaporation with decorative Porous Ceramic Cooling Systems, and parametric studies allowed to interrogate in detail the impact of different boundaries and environmental conditions on temperature reduction and humidity increase.

The first set of parametric studies allowed the definition of sensible-to-latent heat ratios for different boundary conditions. Sensible heat proved to have a considerable impact on temperature reduction and, in order to enhance the cooling performance of a ceramic cooling wall, it is recommended to avoid high air velocities (above 1m/s) and warm water (and surface) temperatures. In the contrary, higher evaporation rates were observed with warmer water temperatures, strategy that can be used to passively increase humidity levels in a dry space.

Moreover, the achieved temperature reduction for the tested samples ranged from 0.5 to 1.5°C at 0.5m downstream the ceramic samples. This reduction remained below 0.5°C at longer distances, suggesting a limited area of influence of PCC systems under optimal environmental conditions. This reaffirmed the localised cooling effect of the system, which must be placed on areas accessible to users to maximise benefit.

The impact of geometry on the ceramic cooling performance was proved to be significant. Sharing the same volume, the ceramic samples presented different temperature and humidity profiles. The Prototype P1 presented the best performance among the analysed samples, achieving up to 1°C lower air temperature, and supporting the original design criteria described in Chapter 3.1. The highest effective surface area-to-drag coefficient ratio (ESA/Cd) thus resulted in the best cooling performance. Moreover, the ESA/Cd was discovered to be proportional to the cooling performance of each ceramic sample, gradually decreasing as shown in Figure 4.21.

		ESA:V (m ⁻¹)	ESA (m ²)	Cd	ESA/Cd (m ²)
best performance	 P1	234	0.23	0.52	0.45
	 S2	79	0.08	0.45	0.17
	 P2	67	0.07	0.50	0.13
	 S5	148	0.15	1.12	0.13
worst performance	 S4	55	0.06	1.13	0.04

Figure 4.21 Ceramic performance and ESA/Cd correlation

5 An Early Design Stage PCC System Calculation Tool

The integration of environmental strategies on early design stages has gradually increased in the past decade. The wide range of simple tools available in the market allowed architects to test their designs within minutes without the need of expertise knowledge in the field. Most architects are now capable to predict through the use of rules of thumb or steady-state tools solar access, natural ventilation flow rates or daylight levels in any designed space. The benefits from this practice are multiple: confidence on the design, good position for further refinement of the strategy and better building performance.

Despite the wide range of tools available in most fields, access to simple and intuitive tools addressing evaporation processes is limited due to the complexity of the physics involved, which unfortunately, often leads to banal assumptions on the cooling capacity of water bodies placed in the design. The existing tools to size passive evaporative cooling system are limited to available fieldwork data and are discussed in Chapter 2.3.1

The development of a steady state calculation tool to predict the performance of porous ceramic cooling systems (PCC Tool) aimed to add a new resource to the environmental tool repertoire, providing with knowledge on the feasibility and performance of these systems without the need of computationally intensive analysis. To fulfil this task, the PCC Tool addressed in first place the climate suitability (wet-bulb temperature depression and water consumption), and in second place the cooling capacity and improvement in thermal comfort conditions (dry bulb air temperature, mean radiant temperature, air humidity and air velocity).

The tool is based on a database generated from an extensive parametric CFD analysis subsequently validated. Through its functionalities, the user would be able to estimate temperature reduction, humidity increase and water consumption of a PCC system exposed to any environmental conditions and for different ceramic shapes and arrangements with a limited number of inputs and time consumption.

This chapter describes the parametric database constituting the core of the tool, the user interface, the tool outcomes, and the calculation methods embedded in

the program. Lastly, laboratory and fieldwork data were compared against PCC tool outcomes to complete the validation process.

PCC Tool is accessible through: <http://naturalcooling.co.uk/PCCT.xlsx>

5.1 Parametric Database

A parametric CDF database becomes the core of PCC Tool. The numerical model used for the studies is in line with the model described in Chapter 4.2, consisting in a transient simulation with water phase change taking place on the ceramic surface, and consequently mass transfer from liquid to vapour in a rectangular prism (1000x1000x2000mm) domain. Four variables were considered to define the parametric matrix, allowing users to predict the PCC system performance under different environmental conditions, ceramic shapes and arrangements.

Wet-bulb temperature depression (DBT-WBT). This parameter is one of the main indicators of the efficiency of a passive evaporative cooling system. DBT-WBT indicates the achieved temperature reduction when air is saturated, being at the same time proportional to the amount of water liquid that can phase-change to water vapor. In Europe, the average DBT-WBT during the hot season ranges from 2°C to 12°C, reaching peaks over 16°C at the warmest hours of the day. Following the applicability method described by Alvarez et al. (1997), zones presenting an average DBT-WBT below 5°C are characterized for their low applicability of passive evaporative cooling methods. An average DBT-WBT of 5°C to 9.5°C suggest a medium applicability; and above 9.5°C the evaporative system is highly applicable. In order to cover most of the spectrum of DBT-WBT typically occurring in Europe, the parametric database incorporates data corresponding to four environmental conditions associated to the above mentioned three levels of applicability: DBT-WBT 4, 8, 12 and 16°C.

Air velocity (v). Air velocity has a direct implication on the airflow pattern over the ceramic and the adjacent space. Air movement and kinetic forces on wetted ceramic surfaces separates water molecules and the variation of this parameter alters the evaporation rate. Assuming that the system is applied on transitional spaces to improve semi-outdoor/indoor thermal comfort conditions (Chapter 2.4) and willing to avoid excess of hot air streams, desired air velocities should not exceed 1m/s, value at which air movement might be disruptive (i.e. blowing papers). The range of air velocities considered for the parametric study goes

from 0.25 m/s (perceptible air movement) to 1m/s, and four values are included in the database: 0.25, 0.50, 0.75 and 1 m/s.

Ceramic shapes. The geometry of the ceramic element defines the surface area exposed to the airstream and the resistance to airflow. Both parameters are equally important in PCC systems. Higher effective surface areas lead to higher evaporation rates, and lower flow resistances also improve the cooling capacity of the system as seen in Chapter 3.3. Despite the drag coefficient is not an absolute constant and can change with the ceramic size or air velocity, C_d is still a valid indicator of the aerodynamics of the ceramic element and its suitability for evaporation purposes. Typical drag coefficients of simple geometries range from 0.05 for aerofoils to values beyond 1.5 for rectangular shaped elements. The parametric database includes the cooling performance of circular, triangular, quadrangular and rectangular 1000 mm extruded shapes, covering a range of drag coefficients from 0.56 to 1.37 with Reynold Number approximately $2 \cdot 10^3$ as shown in Figure 5.1.

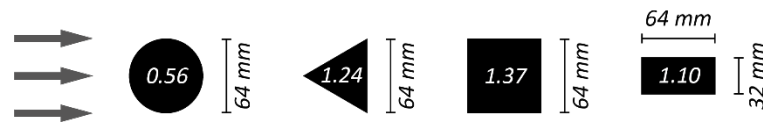


Figure 5.1 Drag coefficients for single extruded shapes with Reynold Number approximately $2 \cdot 10^3$

Blockage ratio (BR). This parameter is the opaque to free area ratio and is commonly used to describe obstructions in HVAC systems. It is directly proportional to the number of columns facing the airstream, and it can be calculated by dividing area of the projected ceramic shape over the vertical surface perpendicular to the airstream, by the total wall (inlet) area. The number of ceramics and distance between them also has an impact on the airflow pattern (Reynolds Number), and it may alternate the drag coefficient characteristic from each shape, and consequently, the cooling performance. Moreover, the BR is directly proportional to the effective ceramic surface area (ESA), the other main indicator of the effectiveness of the system. The parametric database includes four blockage ratios: 15, 25, 40 and 50%, equivalent to 2, 4, 6 and 8 columns per lineal meter of ceramic wall for the shapes with C_d equal to 0.56, 1.24, 1.37 and 4, 8, 12 and 16 columns for the shape with C_d 1.10.

Overall, the parametric study consisted of sixteen environmental conditions: four DBT-WBT (8, 10, 12, and 16°C) under four air velocities (0.25, 0.50, 0.75 and 1 m/s). Four ceramic shapes presenting different airflow resistance (C_d : 0.56,

1.24, 1.37, and 1.10) were analysed for four different ceramic arrangements (BR: 15, 25, 40 and 50%). In total, the study involved 256 CFD simulations (Figure 5.2).

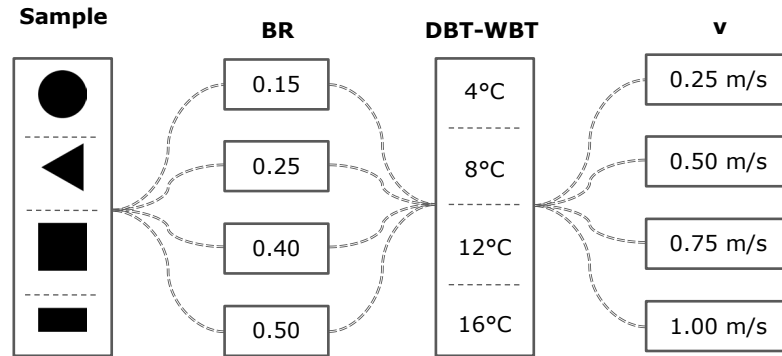


Figure 5.2 CFD Parametric study performed for PCC Tool database

The resulting database contains, from each simulation, the dry bulb air temperature reduction (ΔDBT) and absolute humidity increase (ΔAH) achieved at each node of the simulation mesh downstream direction from the centroid of the ceramic element distanced every 0.02 m from 0 to 1.4m, and a cross section area-weighted average ΔDBT and ΔAH at multiple distances (0.05, 0.1, 0.2, 0.4, 0.6, 0.8, 1, 1.2 and 1.4 m) downstream the ceramics. The fluid domain volume-average ΔDBT and ΔAH increase downstream the ceramics completes the collection of data from each scenario (Figure 5.3).

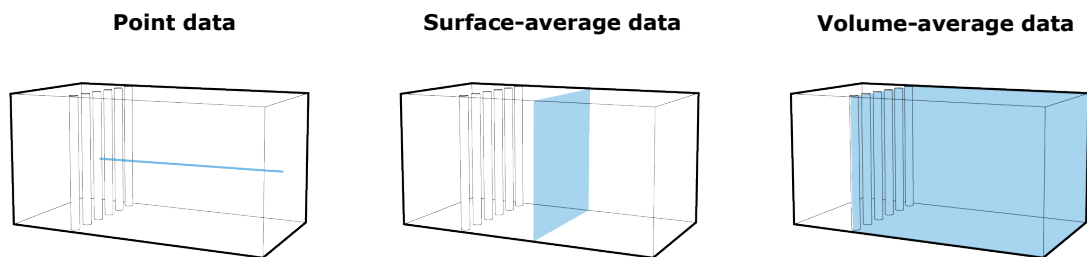


Figure 5.3 Point, surface and volume averaged collected ΔDBT and ΔAH data

5.1.1 Methodology

The tool itself is not a solving program, and it could be better described as an optimised database management system. It is a multi-steady state model that

generates a series of scenarios on a daily basis, collects data from the parametric database, and through an interpolation process obtains the final outcomes. It is thus a five-step process that provides with immediate results that respond to the user inputs.

User inputs. The user inputs include climatic and geometrical ceramic data. Climatic data is given for a typical 24 hours' profile when the system is expected to be in operation, and it is defined by means of DBT, AH and v . This allows the user to test the efficiency of a PCC system at almost any particular location. The geometry of the system is defined by the ceramic shape and the number of ceramic elements, also allowing a wide range of geometries to be evaluated.

Scenarios. The tool creates a series of scenarios for each particular hour of operation (typically presenting different DBT-WBT) for the defined ceramic system. This information, which consist on a DBT-WBT, v , Cd and BR value, is stored and later used to retrieve results from the database and perform the interpolation process.

Access to database. The tool accesses the database to obtain the results corresponding to the nearest environmental conditions and system geometries to the user defined scenarios. Typically, it will include data for the nearest up and nearest down DBT-WBT, v and BR.

Tool interpolation process. A series of interpolations are performed between the nearest values obtained from the parametric database in a logical order to obtain the final output. A first interpolation is done between the nearest v ; secondly, between the nearest DBT-WBT: and lastly between the nearest BR. This process is repeated for each hour of operation (Figure 5.4).

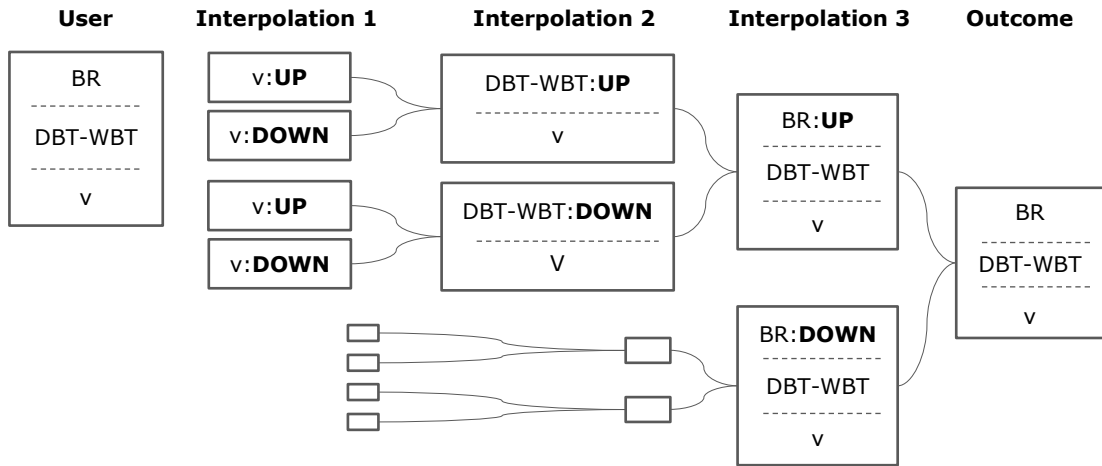


Figure 5.4 Interpolation process between the nearest v , DBT-WBT and BR

Results. The results are plotted in a graphical way and contain hourly and daily (cumulative) data for the given operating time. This allows to address thermal comfort conditions at different times downstream the ceramics, and alternatively, the efficiency of the system and overall performance (cooling capacity and water consumption) for the same operating time.

Despite the extensive database and the number of interpolations needed, the workflow remains simple, and the whole process is automated and running under an intuitive and graphical user interface.

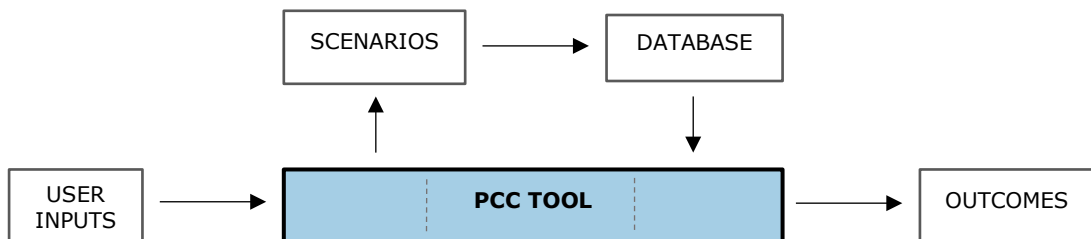


Figure 5.5 PCC Tool workflow

5.1.2 Outcomes

The tool provides with the information needed to assist designers in the sizing process of PCC systems, including overall performance and potential limitations deriving from the local context (i.e. water availability). The quick response also allows comparative studies and becomes a self-educational tool for the designer,

who can explore the impact of different parameters on the overall performance of the system.

To address the improvement in thermal comfort conditions downstream the ceramics, the tool outputs a cross section area-weighted average ΔDBT and ΔAH at any given distance between 0 and 1.40 meters from the ceramics.

Mean radiant temperature (MRT) is also given at the selected distance from the ceramics, being proportional to the view angle to the ceramic wall (Chapter 5.3.4) and the ceramic surface temperatures achieved.

To address the overall cooling capacity and efficiency of the system, the tool outputs the average sensible and latent cooling achieved over the operating time. This differentiation is important to understand the role of evaporation in a cooling process where both convective and evaporative cooling contribute to the achieved temperature reduction. The cooling capacity, given in W/m^2 , also allows a comparison with alternative passive and mechanical cooling systems in the pursuit of the most suitable system for the given context.

In addition, PCC Tool provides with an estimated water consumption during the operational hours. It is indeed, one of the most relevant outputs as high values can make designers reconsider the size or even the feasibility of the system. Water availability is often limited in hot-dry climates in Europe (Chapter 0) and small changes in the system geometry and BR can considerably change the final water consumption per day. Indeed, the range of values for water consumption (l/day) that could be obtained from different ceramic systems is approximately two orders of magnitude bigger than the range of ΔDBT achieved downstream the ceramics, confirming the importance of this parameter when exploring the feasibility of the system.

5.2 User Interface

As described in the introduction of this chapter, PCC Tool is oriented to architects and designers and it does not require any expertise knowledge in the field of evaporation. Its usability is associated to early design stages and the limited time for testing design concepts define the context of this tool, which seeks for an intuitive graphical interface requiring the minimum number of inputs. It is achieved by structuring the tool in three main bodies: Microclimatic Data, Ceramic System Data and Evaporative Cooling Performance. The first two

modules contain the user inputs that define the environmental conditions and the geometry of the ceramic system. The last module outputs cooling performance and sufficient data to assist designers on the decision-making process.

5.2.1 Microclimatic Data

This first module collects the data needed to generate the environmental conditions of each scenario on a 24-hour profile. The user should have previous knowledge on the climatic conditions of the site where the PCC system is intended to be placed. Ideally, the climatic data given by the user should respond to a typical day of the season where the PCC system will be in operation. In case the user is not familiar with the climate, monthly and hourly weather-averaged data is available from several sources (Chapter 1.2).

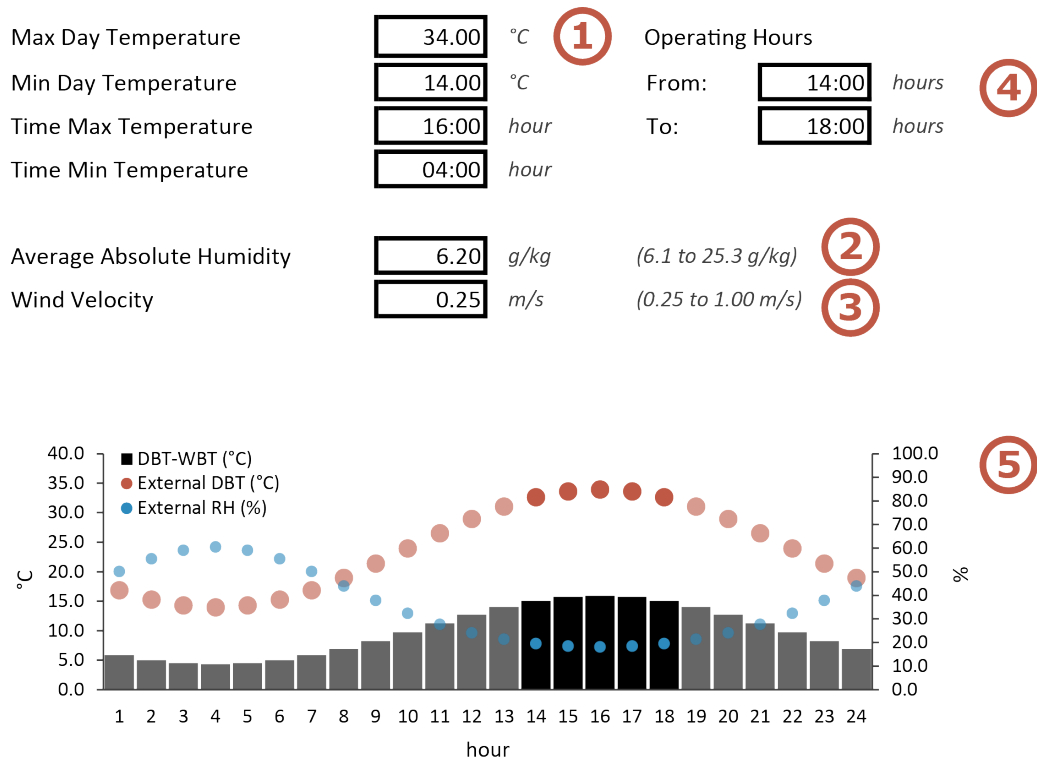


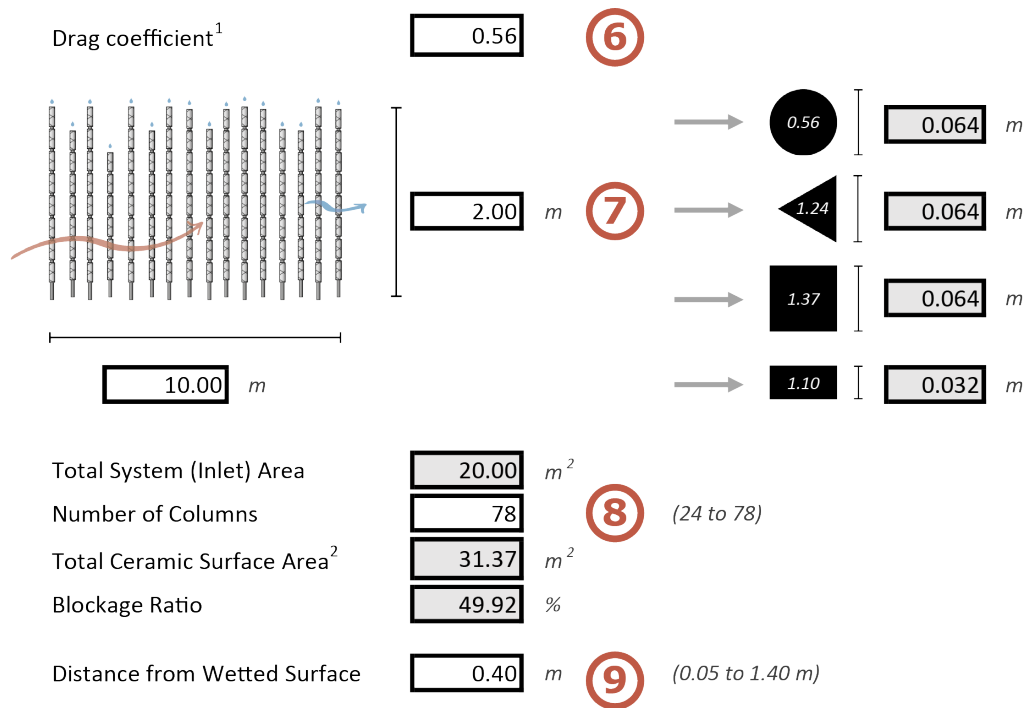
Figure 5.6 Microclimatic Data PCC Tool inputs

1. A 24-hour air temperature profile given in °C is obtained from the maximum and minimum DBT of a typical day (usually during the warm season), and the time of occurrence (refer to Chapter 5.3 for calculation methods).

2. Unless extreme weather conditions, absolute humidity (amount of water vapour in air) typically remains constant during the day, and the variation in relative humidity (RH) occurring during the day mainly responds to the air temperature fluctuation. In most climates, the AH fluctuation observed during a day rarely exceeds 1 g/kg, and major variations in AH are usually observed at seasonal levels. This reduces the number of inputs required to define moisture content, and a range of values are also suggested by the tool in order to fit within the range of DBT-WBT (4 to 16°C) included in the database. Moreover, the resulting RH is plotted over a 24-h profile.
3. As a multi steady-state tool (not dynamic), air velocity on the windward side of the ceramic elements is considered constant during the operating hours and thus for the estimation of cooling performance of the system. This value given in m/s, should have been previously estimated by the user and should respond to the adopted ventilation strategy and space layout. Exposure to wind or adjacent spaces with high stacks may lead to higher air velocities (>0.5 m/s), whereas cross ventilated indoor spaces (in absence of wind) lead to lower air velocities (<0.5 m/s). The range of air velocities allowed and included in the parametric database ranges from 0.5 to 1 m/s.
4. The operating hours indicates the period of time for which the PCC system is in operation. The final outcomes are based on the selected period and the rest of hours are not considered. To maximise efficiency and minimise water consumption, the user may operate the system for only the warmest hours of the day.
5. DBT, RH and DBT-WBT are plotted on a 24-hour chart to provide with an overall view of the climatic conditions selected. The chart itself is the first indicator of the applicability of a PCC system and assists in the selection of the operating hours. Plotted DTBs suggest the period of time when a cooling system would be desirable, and DBT-WBT indicates the suitability of an evaporative system. As discussed in Chapter 1.2, an average DBT-WBT $>9.5^{\circ}\text{C}$ is commonly associated to high applicability and will imply better evaporative cooling performance. The highlighted hours in the chart are associated to the operating hours selected by the user.

5.2.2 Ceramic System Data

The second module collects the required data to define the geometry of the system. The inputs determine the resulting ceramic surface area, and an iterative process is recommended to find the optimal number of columns, shape and system dimensions that best suits the initial user requirements and meets the local context restrictions.



¹ The Drag Coefficients are calculated for single extruded shapes (displayed) with Reynolds Number approximately $2 \cdot 10^3$. These values may change for different Blockage Ratios, Air Velocities and Ceramic Sizes.

² Uniform roughness and smooth finish are assumed for the ceramic surface. Under these conditions, the roughness effects are negligible and are not considered in the calculations.

Figure 5.7 Ceramic System Data PCC Tool inputs

- The ceramic shape is defined by the tool with the drag coefficient (Cd). Cd a valid indicator of the aerodynamics of the ceramic element and it has a direct impact on the evaporation rate achieved (Chapter4.3). Under the same airflow (turbulent) model -Reynolds Number-, a lower Cd yields to higher cooling capacities, whereas a higher Cd yields to higher flow resistances and lower cooling capacities. The images displayed represent the cross sections of the extruded ceramics used in the CFD simulations that generated the parametric database, and the Cd values correspond to one ceramic sample with the shape and dimensions shown under a flow model with Reynolds Number approximately $2 \cdot 10^3$ (air density: 1.225

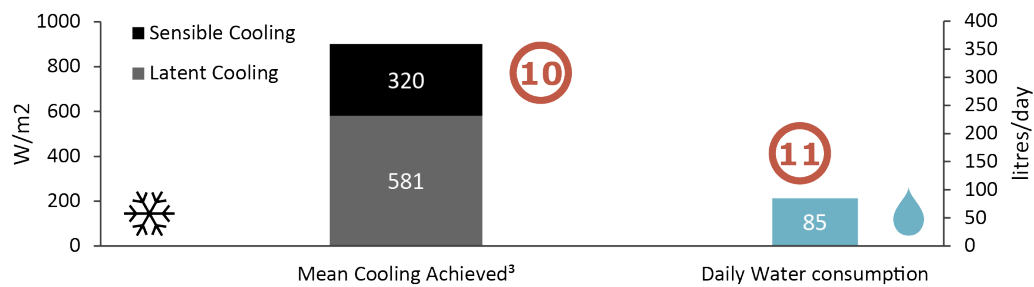
kg/m³, air velocity: 0.5 m/s, air dynamic viscosity: $1.79 \cdot 10^{-5}$ Kg/m·s). A change in the cross-section dimensions will alter the airflow profile (Reynolds Number) and thus the Cd. As the prediction of air flow is computationally intensive, no other sizes are considered in the tool.

7. The size of the ceramic system, assumed as a permeable vertical screen, is defined by the width and the height in meters (m). These dimensions have direct impact on the blockage ratio (BR), and the high of the system is considered equal to the height of the ceramic columns. The system dimensions have a considerable impact on water consumption and specially, on the mean radiant temperature depression (Δ MRT) downstream the ceramics. Bigger systems yield to higher water consumption and higher mean radiant temperature reduction.
8. This parameter defines the number of vertical columns placed along the width of the wall. The columns are assumed equally distanced from each other and the number of columns is a direct indicator of the total ceramic surface area and blockage ratio (BR). The BR included in the parametric database range from 15 to 50%, and these limits were used to indicate the minimum and maximum number of columns allowed to satisfy these values. For the calculation of ceramic surface area, as well as Cd, a uniform surface roughness has been considered on the ceramic wall, and thus any variation on waviness height over the surface has been neglected. The PCC system is oriented to manufactured ceramics through an extrusion process, and this method typically produces smooth surfaces with average roughness height (R_A) of 0.8 to 3.2 μ m (DeGarmo, 2003, p. 939). To be in line with experimental work conducted and the manufactured prototype P2 (Chapter3.1.2), the R_A considered for the ceramic elements was set to 0.8 μ m, which has an equivalent non-dimensional sand-grain roughness (K_s^+) of 0.27 (Levin et al., 2014). Surface roughness can have a considerable impact on Cd and heat and mass transfer on walls presenting a rough regime ($K_s^+ > 90$), but under smooth regimes with $K_s^+ \leq 2.25$ (ANSYS®, Release 17.2c, pp. 315-318), the roughness effects are negligible and have not been considered in the calculations.
9. Air temperature and water vapour distribution downstream the PCC system is a function of the airflow pattern and turbulence model. It is rarely constant, and as seen in Chapter4.4, Δ DBT and Δ AH decrease with bigger distances from the wetted surfaces. The final Δ DBT and Δ AH shown in the outcomes, refer to this distance, and so the Δ MRT, which is

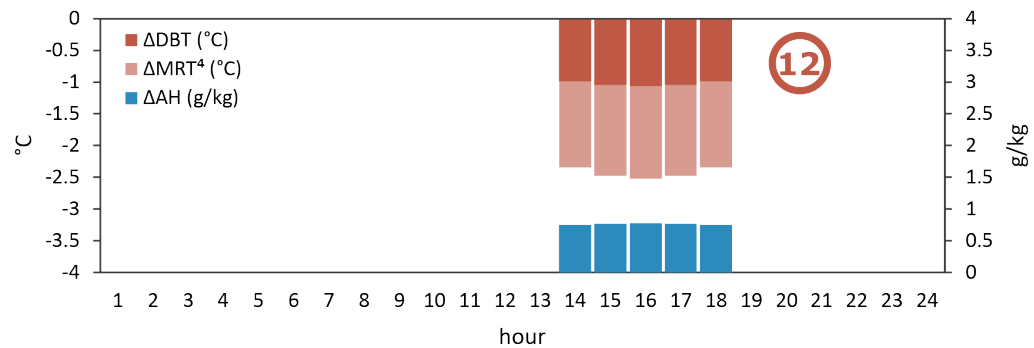
calculated from angle factors. This parameter is given in meters (m) and the minimum and maximum values considered in the parametric database ranges from 0 to 1.40 meters.

5.2.3 Cooling Performance

The third module outputs the cooling performance of the system. The results are given on daily and hourly basis, allowing first to evaluate the feasibility and capacity of the system; and second to analyse in detail the improvement in thermal comfort conditions under different scenarios.



³ Mean Sensible + Latent cooling per unit area of ceramic system.



⁴ Mean Radiant Temperature (MRT) estimated for a standing person from Ceramic Surface Temperatures and Angle View Factors at the given downstream distance from the ceramic system (EN ISO 7726:2001).

Figure 5.8 Evaporative Cooling Performance PCC Tool outputs

10. The mean cooling achieved refers to the average sensible and latent cooling achieved during the operating hours of the system. The sensible cooling refers to the amount of energy needed to cool down the environment the archived ΔDBT. Latent cooling refers to the amount of energy needed to increase air moisture content the achieved ΔAH. Sensible and latent cooling are thus proportional to the ΔDBT and ΔAH achieved downstream the ceramics respectively, and the results in W/m²

are given per unit area of ceramic system. The sensible cooling achieved and water consumption best describe the cooling capacity of the system and allow the comparison with alternative cooling methods.

11. Water consumption derives from the ΔAH achieved downstream the ceramics during the operating hours and it is a cumulative value given in litres/day. It is indeed, one of the most relevant outputs as high values can make designers reconsider the size or even the feasibility of the system.
12. This chart plots ΔDBT , ΔMRT and ΔAH achieved during each hour of operation. All the results refer to the selected downstream distance from the ceramics and multiple iterations will give a broad understanding of the area of influence of the system and thus the recommended distance between the user and the ceramics.

5.3 Calculation Methods

This section describes the calculation methods used by PCC tool to postprocess the parametric database and to generate the results described in Chapter 5.1.2. The CFD model and its governing equations is described in Chapter 4.2.

5.3.1 Air Temperature Reduction and Absolute Humidity Increase

ΔDBT and ΔAH refer to the air temperature reduction and absolute humidity increase achieved at a given distance downstream the ceramics. ΔDBT can be expressed as the difference between the initial/windward (DBT_{IN}) and the final/leeward (DBT_{OUT}) air temperature and it is measured in °C Equation (5-1). Similarly, ΔAH is expressed as the difference between the final/leeward (AH_{OUT}) and initial/windward (AH_{IN}) absolute humidity and it is measured in g/kg Equation (5-2).

$$\Delta DBT = DBT_{IN} - DBT_{OUT} \quad \text{Equation (5-1)}$$

Air temperature reduction

$$\Delta AH = AH_{OUT} - AH_{IN} \quad \text{Equation (5-2)}$$

Air humidity increase

where DBT_{IN} and AH_{IN} are user inputs and DBT_{OUT} and AH_{OUT} derive from the parametric database. For the calculation of ΔDBT and ΔAH , the tool first obtains the associated air temperature reduction and humidity increase at a point at the given distance downstream from the ceramic centroid $\Delta DBT_{(x,y,z)}$, $\Delta AH_{(x,y,z)}$ (Figure 5.9) through the first two steps of the iteration method described in Chapter 5.1.1 (v:UP, v:DOWN, DBT-WBT:UP, DBT-WBT:DOWN). The interpolation between the upper and lower values from the database is a linear interpolation computed by the spreadsheet software Microsoft Excel.

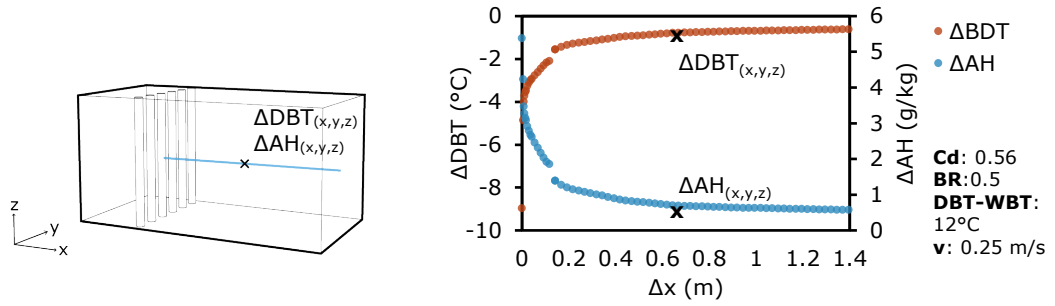


Figure 5.9 Example of point-in-domain (left) and its associated ΔDBT and ΔAH (right) downstream from the ceramic centroid obtained from the parametric database

Following up, the tool extrapolates the point-in-domain $\Delta DBT_{(x,y,z)}$, $\Delta AH_{(x,y,z)}$ onto a surface averaged ΔDBT_{yz} and ΔAH_{yz} over a cross section of the domain at the same given distance. This step is conducted to consider the permeability of the ceramic screen and it can be calculated with the expressions:

$$\Delta DBT_{yz} = \Delta DBT_{(x,y,z)} \cdot BR \cdot F_{DBTdx} \quad \text{Equation (5-3)}$$

Surface averaged ΔDBT_{yz}

$$\Delta AH_{yz} = \Delta AH_{(x,y,z)} \cdot BR \cdot F_{AHdx} \quad \text{Equation (5-4)}$$

Surface averaged ΔAH_{yz}

where (yz) and ΔAH_{yz} are the surface-averaged air temperature reduction and absolute humidity increase, $\Delta DBT_{(x,y,z)}$ and $\Delta AH_{(x,y,z)}$ are the point-in-domain air temperature reduction and absolute humidity increase, BR is the blockage ratio and F_{DBTdx} and F_{AHdx} are correction factors.

The correction factors increase the accuracy of Equation (5-3) and Equation (5-4) by considering the non-uniform temperature and humidity distribution on the given cross section. This temperature and humidity distribution over the two dimensions varies with distance downstream the ceramics and is specific for each ceramic shape, blockage ratio and air velocity, in other words, with different Reynolds Number (Figure 5.10).

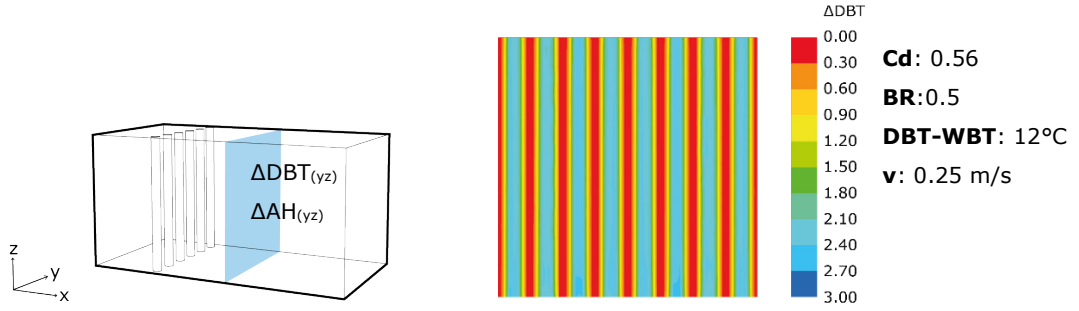


Figure 5.10 Example of two-dimensional cross section downstream the ceramics (left) and its associated ΔDBT distribution at 0.20 m downstream distance from the ceramics (right)

F_{DBTdx} and F_{AHdx} thus derive from the parametric database and is calculated at 0.05, 0.10, 0.20, 0.40, 0.60, 0.80, 1.00, 1.20 and 1.40 m downstream distance from the ceramics and for each Blockage Ratio included in the database by using the expressions:

$$F_{DBTdx} = \frac{\Delta DBT_{yz}}{\Delta DBT_{x,y,z} \cdot BR} \quad \text{Equation (5-5)}$$

Correction factor for $\Delta DBT(yz)$

$$F_{AHdx} = \frac{\Delta AH_{yz}}{\Delta AH_{x,y,z} \cdot BR} \quad \text{Equation (5-6)}$$

Correction factor for $\Delta AH(yz)$

where $\Delta DBT(yz)$ and $\Delta AH(yz)$ are directly obtained from the CFD software Ansys Fluent with the Area-Weighted Average feature, which is the summation of the product of the selected field variable (DBT, AH) and facet area by the total area of the surface (ANSYS®, Release 17.2b, p. 748):

$$\phi = \frac{1}{A} \sum_{i=1}^n \phi_i |A_i|$$

Equation (5-7)

Area-weighted calculation
ANSYS® (Release 17.2b, p. 748)

where Φ is the average-weighted variable (DBT, AH) and A is the surface area.

The final F_{DBTDx} and F_{AHDx} values associated to the distance given by the user are adjusted through a 2-step iteration process: first between the nearest calculated distances, and then to the nearest air velocities (Δx :UP, Δx :DOWN, Δv :UP, Δv :UP). A typical profile of F_{DBTDx} and F_{AHDx} can be seen on Figure 5.11. The charts confirm an irregular distribution closer to the ceramics' surfaces, whereas at distances beyond 1m the temperature and humidity tend to a uniform distribution.

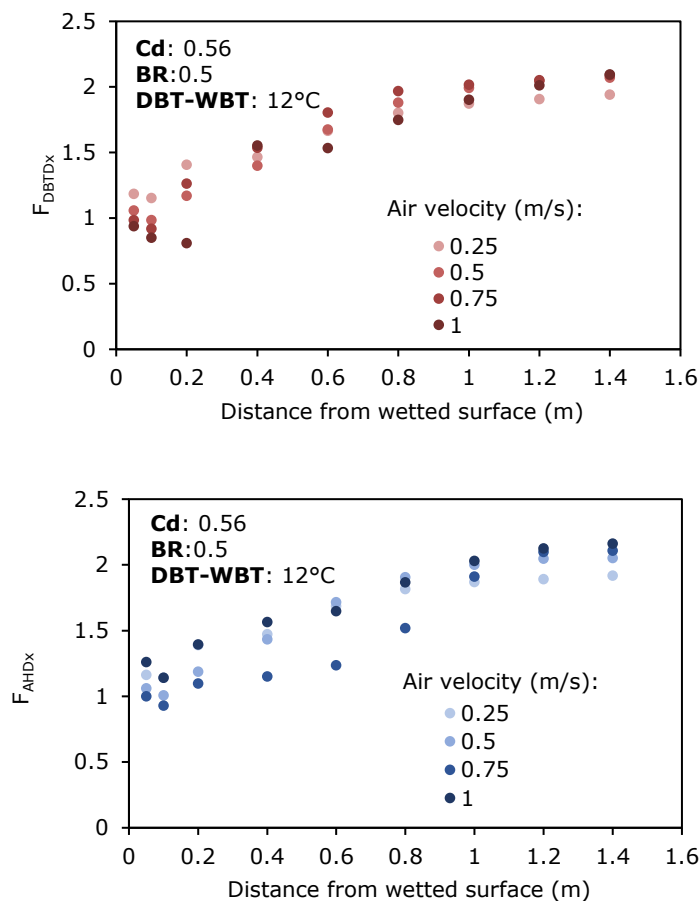


Figure 5.11 Correction factors F_{DBTDx} (top) and F_{AHDx} (bottom) as a function of the downstream distance from ceramics and air velocity for the given scenarios

5.3.2 Sensible and Latent Cooling

The sensible cooling (Q_s) is the amount of energy needed to reduce the temperature of a volume of air, keeping its moisture content constant. It is thus directly proportional to the air temperature reduction achieved downstream the ceramics and can be calculated using the following expression (ASHRAE, 2013, p. 18.13):

$$Q_s = \rho \cdot C_p \cdot q [DBT_{V-IN} - DBT_{V-OUT}] \quad \text{Equation (5-8)}$$

Sensible cooling
ASHRAE (2013, p. 18.13)

where ρ is the density of the moist air (kg/m^3), q is the air volume flow rate (m^3/s) and C_p is the specific heat of air ($1.006 \text{ kJ/kg}^\circ\text{C}$). DBT_{V-IN} is the initial/windward air temperature and DBT_{V-OUT} is the air temperature downstream the PCC system. The air volume flow rate q is the amount of air delivered per unit of time and is directly proportional to the air velocity (m/s) and the (inlet) area of the ceramic wall (m^2). In the computational model used (Chapter 4.2), q can be expressed as:

$$q = A_{duct} \cdot v \quad \text{Equation (5-9)}$$

Air volume flow rate

where A_{duct} is the cross-section area of the fluid domain (m^2) and v is the initial (inlet) air velocity (m/s).

The latent cooling (Q_L) is the amount of energy needed to increase the moisture content of a volume of air, keeping its air temperature constant. It is thus directly proportional to the absolute humidity increase achieved downstream the ceramics and can be calculated using the following expression (ASHRAE, 2013, p. 18.13):

$$Q_L = \rho \cdot C_e \cdot q[AH_{V-OUT} - AH_{V-IN}] \quad \text{Equation (5-10)}$$

Latent cooling
ASHRAE (2013, p. 18.13)

where ρ is the density of the moist air (kg/m^3), q is the air volume flow rate (m^3/s), and C_e is the latent heat of vaporization of water (2465.56 kJ/kg). AH_{V-IN} is the initial/windward absolute humidity and AH_{V-OUT} is the absolute humidity downstream the PCC system. The air volume flow rate q is the amount of air delivered per unit of time and it is obtained from Equation (5-9).

The air temperature DBT_{V-OUT} and absolute humidity AH_{V-OUT} downstream the PCC system are obtained from the CFD software Ansys Fluent using the Volume-Weighted Average feature, which is the summation of the product of the selected variable (DBT, AH) and cell volume by the total volume of the fluid domain.

$$\Phi_V = \frac{1}{V} \sum_{i=1}^n \phi_i |V_i| \quad \text{Equation (5-11)}$$

Volume-weighted calculation (I)
ANSYS® (Release 17.2b, p. 753)

where Φ_V is the volume-weighted variable (DBT, AH), and V is the fluid domain volume (m^3). The above equation is equivalent to:

$$\phi_V = \frac{\phi_i \cdot V_i + \phi_1 \cdot V_1 + \phi_2 \cdot V_2 + \phi_3 \cdot V_3 + \dots + \phi_n \cdot V_n}{V} \quad \text{Equation (5-12)}$$

*Volume-weighted
calculation (II)*

The Volume-Weighted Average Φ_V calculation from Ansys Fluent refers to the entire fluid domain (V), and the known initial/windward volume-weighted variable Φ_{V-IN} must be subtracted in order to obtain the volume-weighted variable downstream the ceramic Φ_{V-OUT} (Figure 5.12):

$$\Phi_{V-OUT} = \frac{\Phi_V \cdot V - \Phi_{V-IN} \cdot V_{IN}}{V_{OUT}}$$

Equation (5-13)

*Volume-weighted calculation
downstream the ceramics*

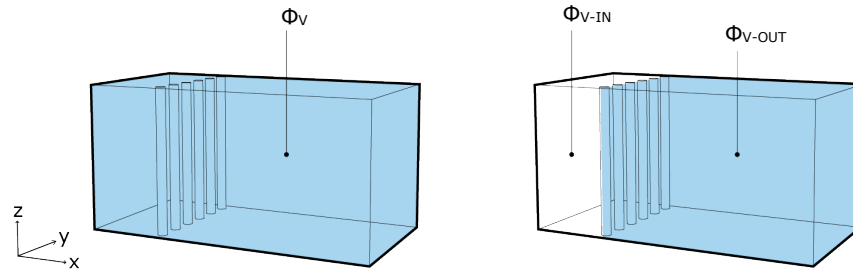


Figure 5.12 Volume weighted variable (DBT, AH) as function of the entire fluid domain (left) and windward + leeward domains (right)

5.3.3 Water Consumption

The water consumption (W_C) per time of PCC system operation is equivalent to the total amount of water particles transferred from liquid to vapour state. It is thus proportional to the absolute humidity increase achieved downstream the ceramics and it can be calculated from the following expression:

$$W_C = [AH_{V-OUT} - AH_{V-IN}] \rho \cdot q \cdot t$$

Equation (5-14)

Water consumption

where ρ is the moist air density (kg/m³), q is the air volume flow rate (m³/s), and t is the time for which the system is in operation (s).

5.3.4 Mean Radiant Temperature

The mean radiant temperature (MRT) is defined by International Standards (ISO, 7726:2001) and (ANSI/ASHRAE, Standard 55-2013) as the uniform temperature of an imaginary enclosure in which radiant heat transfer from the human body is equal to the radiant heat transfer in the actual non-uniform enclosure. It describes the radiative exchanges between man and his surrounding

environment, and it can be measured by instruments or calculated from theoretical models.

Traditionally, measuring methods allow the calculation of MRT by using globe thermometers, two-sphere radiometers or constant-air-temperature sensors as described in Chapter 3.3.2. The calculation methods, however, normally require knowledge on surface temperatures and the relative position of the surrounding surfaces with respect to the user, although more recent studies present alternative methods to estimate MRT from radiant intensities (Palmer and Chapman, 2000). Fanger et al. (1985) first defined the calculation of MRT from the definition of the surface temperatures and the angle factor between the user and the surrounding surfaces, assuming all surfaces of the room black (as most building materials present high emittance ϵ):

$$MRT^4 = T_1^4 \cdot F_{p \rightarrow 1} + T_2^4 \cdot F_{p \rightarrow 2} + \dots + T_N^4 \cdot F_{p \rightarrow N} \quad \text{Equation (5-15)}$$

*MRT from surrounding
surface temperatures
ISO (7726:2001, p. 20)*

where MRT is the mean radiant temperature (K), T_N is the surface temperature (K), and $F_{p \rightarrow N}$ is the angle factor between the person and the surface N. Environments presenting small surface temperature differences ($T_N < 10^\circ\text{C}$) allow the simplification of Equation (5-15) to the following linear form:

$$MRT = T_1 \cdot F_{p \rightarrow 1} + T_2 \cdot F_{p \rightarrow 2} + \dots + T_N \cdot F_{p \rightarrow N} \quad \text{Equation (5-16)}$$

*MRT calculation for small
surface temperature differences
ISO (7726:2001, p. 20)*

In other words, MRT can be calculated as the mean value of the surrounding surface temperatures weighted according to the respective angle factors. To estimate the impact of PCC systems on MRT on early design stages, Equation (5-16) can be further simplified by only considering the mean ceramic system surface temperature (T_{PCCSM}), and assuming that the adjacent surface temperatures equal to the dry bulb air temperature downstream the ceramics

(DBT_{OUT}). As the sum of the angle factors is unity, the corresponding angle factor for the adjacent surfaces DBT_{OUT} will be $1 - F_{p \rightarrow PCCS}$:

$$MRT = T_{PCCSM} \cdot F_{p \rightarrow PCCS} + DBT_{OUT}(1 - F_{p \rightarrow PCCS}) \quad \text{Equation (5-17)}$$

Simplified MRT calculation

Empirical work in Chapter 3.3 confirmed that the monitored ceramic surface temperatures during the evaporative cooling process are in line with Givoni (1994) studies, and thus the PCC System surface temperature (T_{PCCS}) can be expressed as a function of dry bulb (DBT) and wet bulb (WBT) air temperatures:

$$T_{PCCS} = DBT - 0.8(DBT - WBT) + 3 \quad \text{Equation (5-18)}$$

Monitored surface temperatures of P2

The above expression can be further refined to obtain the mean temperature of the vertical surface defined by the PCC System according to the system blockage ratio (BR):

$$T_{PCCSM} = DBT - 0.8BR(DBT - WBT) + 3BR \quad \text{Equation (5-19)}$$

*Mean surface temperature
for ceramic screens*

Furthermore, by combining Equation (5-17) and Equation (5-19), we can express the mean radiant temperature depression (DBT-MRT) as a function of DBT-WBT, DBT_{OUT} and the corresponding angle factor:

$$\begin{aligned} DBT - MRT = & 0.8BR \cdot F_{p \rightarrow PCCS}(DBT - WBT) - 3BR \\ & \cdot F_{p \rightarrow PCCS} - (DBT - DBT_{OUT}) \cdot F_{p \rightarrow PCCS} \\ & + DBT - DBT_{OUT} \end{aligned} \quad \begin{array}{l} \text{Equation (5-20)} \\ \text{MRT for ceramic screens} \end{array}$$

The angle factor corresponding to the PCC system can be calculated from Equation (5-21) and Figure 5.13, from (ISO, 7726:2001):

$$F_{p \rightarrow PCCS} = F_{max}(1 - e^{(a/c)/\tau})(1 - e^{(b/c)/\gamma}) \quad \text{Equation (5-21)}$$

Angle factor calculation
ISO (7726:2001, p. 27)

where: $\tau = A + B(a/c)$

$$\gamma = C + D(b/c) + E(a/c)$$

The constants F_{max} , A , B , C , D and E derive from the position of the person (standing or seated) and the surface arrangement (horizontal or vertical) and are shown in Table 5.1.

Table 5.1 Constants for angle factor calculation (ISO, 7726:2001).

		Fmax	A	B	C	D	E
STANDING	VERTICAL	0.12	1.24186	0.1673	0.61648	0.08165	0.05128
	HORIZONTAL	0.116	1.59512	0.12788	1.22643	0.04621	0.04434
SEATED	VERTICAL	0.118	1.2159	0.1689	0.71739	0.08733	0.05217
	HORIZONTAL	0.116	1.39569	0.13021	0.95093	0.07967	0.05458

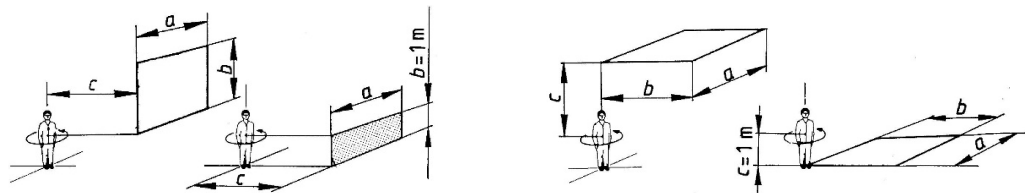


Figure 5.13 Angle factor between a standing person and a vertical rectangle (ISO, 7726:2001, pp. 23-26)

From Equation (5-20) it can be estimated the relation between the DBT-MRT, DBT-WBT, and the distance from the PCC System for a given blockage ratio and dimensions. This relation is shown in Figure 5.13 and the obtained MRT refers to a horizontal plane at 1m high.

5.4 PCC Tool Validation

The tool outcomes were validated using laboratory data. The tubular shaped samples presenting blockage ratios of 25, 40, and 50% were used for the validation, and the predicted temperatures at 0.40m distance from the ceramic wall for a wide range of environmental conditions were compared against the monitored data. Following the same methodology, the water consumption for each hour of operation was also compared.

5.4.1 Laboratory Work vs Computational Work

PCC Tool outcomes have been validated using monitored data from the conducted laboratory work (Chapter 3.3). The tubular shaped samples ($C_d=0.53$) presenting blockage ratios of 25,40, and 50% were used for the validation of the predicted ΔDBT , ΔRH and ΔMRT at 0.40m downstream the ceramics, and the predicted water consumption per square meter of PCC system (inlet) and per hour of operation.

A 24-hour DBT-WBT daily profile was generated covering the whole spectrum of environmental conditions included in the parametric database (DBT-WBT= $4\sim 16^\circ\text{C}$). A PCC System emulating the laboratory model presenting the same wall (inlet) dimensions and number of columns were defined in the tool. The associated results for each BR considered in the PCC Tool were compared against the monitored data under three DBT-WBT ($8,9,10,11^\circ\text{C}$).

The predicted air temperature reduction by PCC Tool and the monitored data from laboratory work both presented a linear relation with DBT-WBT at a constant air velocity of 0.50 m/s (Figure 5.14). Monitored and computational data also presented a narrow spectrum of temperature reduction below 2°C , with a minimum and maximum deviation of 0.05 and 0.35°C respectively. Considering the accuracy $\pm 0.30^\circ\text{C}$ of the resistance temperature detectors (RTDs) used for the collection of data (Table 5.2), the predicted results are in line with monitored data and within the range of acceptable values.

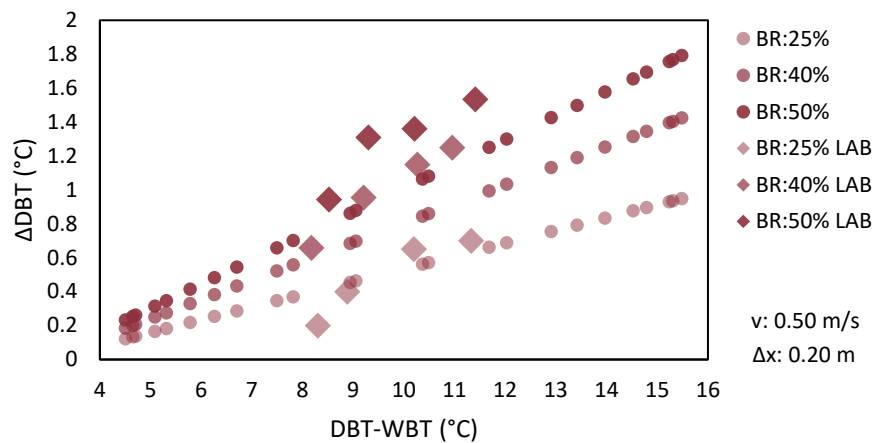


Figure 5.14 Predicted vs monitored (LAB) air temperature reduction downstream the PCC system under different environmental conditions

Table 5.2 RTD class B specifications

	Unit	Range	Resolution	Accuracy
RTD Class B (DBT)	°C	-50~500	0.01	±0.3 (-50~500°C)

The predicted absolute humidity increase downstream the ceramics also presented a linear relation with DBT-WBT at constant air velocity as observed with monitored data (Figure 5.15). Although predicted and monitored lines present slightly different slopes, the data spectrum observed for both methods for a DBT-WBT from 8 to 11°C is narrow, ranging from 0.5 to 1.2g/kg for computational data, and 0.1 to 1.3g/kg for monitored data. The minimum and maximum deviation observed is 0.1 and 0.4g/kg respectively. It is worth mentioning that the sensor used (Table 5.3) outputs relative humidity values and the accuracy of the corresponding absolute humidity has been affected by the measurements of air temperature and barometric pressure, both parameters needed to conduct psychrometric calculations. The predicted results are still within the range of acceptable values.

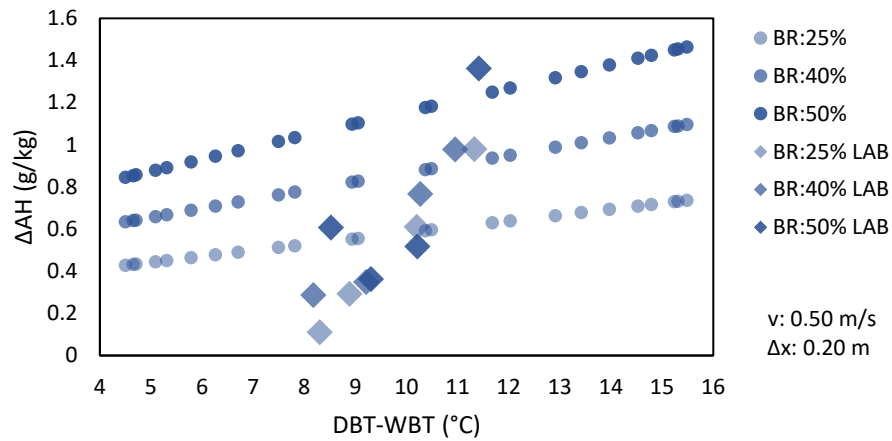


Figure 5.15 Predicted vs monitored (LAB) absolute humidity increase downstream the PCC system under different environmental conditions

Table 5.3 Air Temperature + Relative Humidity sensor specifications

	Unit	Range	Resolution	Accuracy
Campbell Scientific CS215-L (DBT)	°C	-40~70	0.01	±0.4 (5~40°C)
Campbell Scientific CS215-L (RH)	%	0~100	0.03	±4% (at 25°C)

Water consumption is perhaps one of the crucial outputs of PCC Tool. The predicted water consumption per hour of operation and per square meter of PCC system (inlet) was compared against monitored data in Figure 5.16. Both predicted and monitored data presented a linear relation with DBT-WBT at constant air velocity, and the water consumption data spectrum associated to the tested environmental conditions ranged from 0.4 to 0.8 l/h for monitored data, and 0.4 to 1.1 l/h for predicted data. Both models presented a minimum and maximum deviation of 0.01 and 0.3l/h respectively. Monitored water consumption was obtained by weighting the saturated ceramic samples at the beginning and at the end of a 15min test (Chapter 3.3), and the accuracy assumed of these values derived from the scale specifications (Table 5.4) and the time of operation.

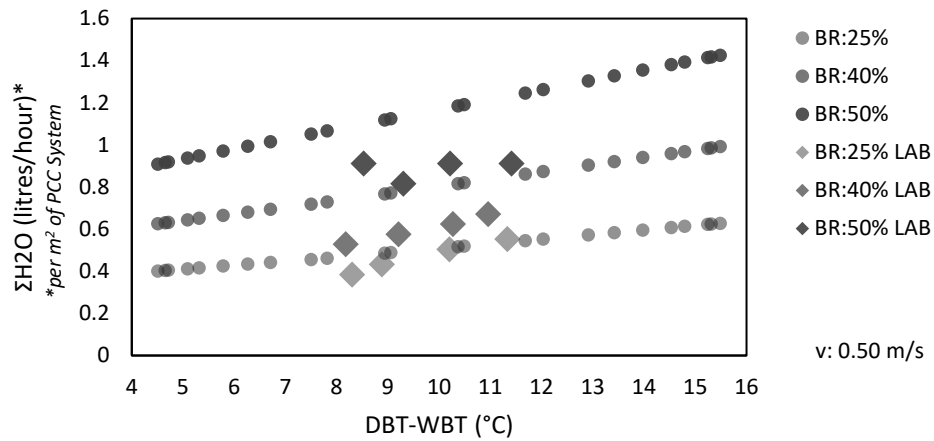


Figure 5.16 Predicted vs monitored (LAB) water consumption per square meter of PCC system

Table 5.4 Scale specifications

	Unit	Range	Resolution	Accuracy
RTD Class B (DBT)	g	-50~500	0.01	±0.3 (-50~500°C)

Mean Radiant Temperature reduction downstream the ceramics also depends on the distance from the ceramic system, and more specifically, the angle factor. The monitored black globe temperature and the associated MRT from laboratory studies (Chapter 3.3.2) for a 500x500 mm domain and a 50 mm globe, were compared against the predicted MRT for an equivalent environment at human scale presenting the same ceramic wall height/width-to-globe diameter ratio (Figure 5.17). The comparison was made for three different environmental conditions (DBT-WBT= 8,12,14°C) and using tubular-shaped ceramic elements presenting a blockage ratio of 50%. The predicted Mean Radiant Temperature reduction downstream the ceramics using the calculation method described in 5.3.4 presented a power-law relation with distance from ceramics (Δx), and was in line with the results obtained for the equivalent environment used during for conducting the laboratory work (Figure 5.18). The minimum and maximum deviation observed was 0.1 and 0.3°C respectively. Both measurement & calculation methods described in Chapters 3.3.2 and 5.3.4 present certain limitations that must be considered when interpreting this data. In the measuring method, the use of the black globe thermometer to estimate MRT is an approximation due to the difference in shape between person and globe. Moreover, the accuracy of measuring MRT using a black globe can vary to a great extent according to instrumentation used (Table 5.5), and the values for

the other characteristics of the environment (v , DBT). In the calculation methods, mean values of angle view factors have been used for standing persons where the location but not the orientation of the person is known (ISO, 7726:2001). Overall, the predicted MRT values responded reasonably to the environmental conditions given, remaining within the range of acceptable values for early stage performance prediction purposes.

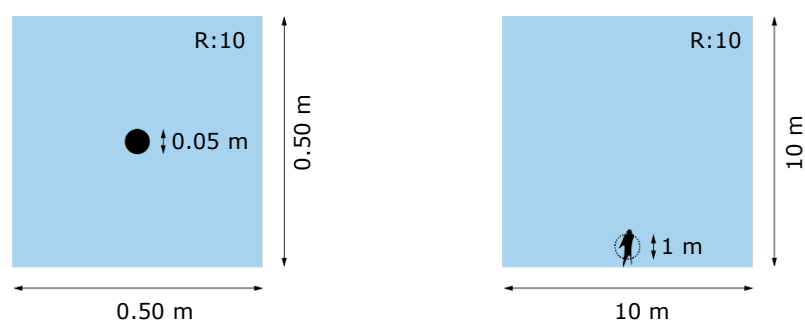


Figure 5.17 Black Globe to insulated box ratio (left) and Human to PCC system ratio (right) used for calculation model calibration

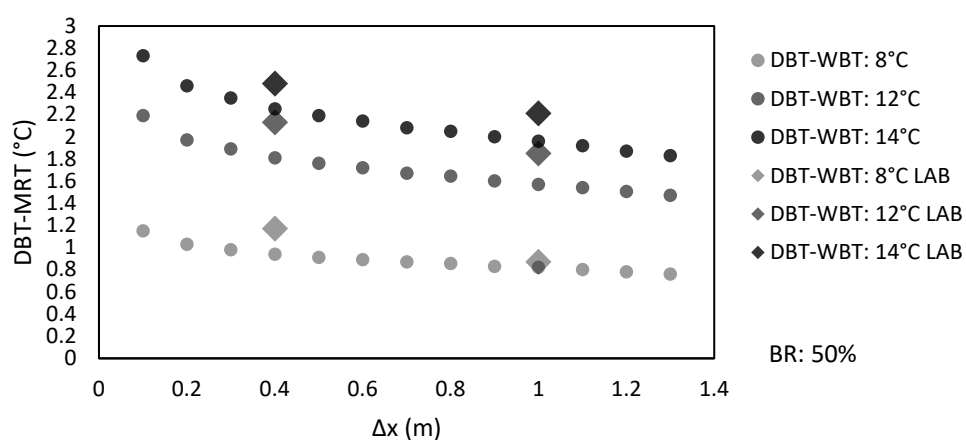


Figure 5.18 Predicted vs monitored (LAB) mean radiant temperature depression downstream the PCC system under different environmental conditions

Table 5.5 Extech® HT200 Heat Stress Meter specifications

	Unit	Range	Resolution	Accuracy
Black Globe Temperature (GT)	°C	0~80.0	0.1	±0.6 (15~40°C)
Air Temperature (DBT)	°C	0~50.0	0.1	±0.8 (15~40°C)
Relative Humidity (RH)	%	1~99	0.1	±3% (20~80%)

5.4.2 Tool Limitations & Further Development

The extrapolation of reality into a theoretical model to be manageable at early design stages is often achieved by prioritising the physics involved in the dynamic context and defining a series of reasonable assumptions that will lead to an equivalent steady-state model. These assumptions, not necessarily reduce the accuracy of the model, and they are mostly tailored according to the required outcomes needed to perform feasibility studies and assist in the design process.

PCC Tool is a multi steady-state tool which focuses on the cooling method for the application of PCC systems. The outcomes given by the tool refer to the microscale associated to people and their thermal perception of an environment within the area of influence of the system. To fully assess the feasibility of a PCC system, apart from the cooling method, an additional study on the airflow strategy and driving forces needs to be explored in parallel. The split between cooling method and ventilation strategy allows more flexibility in the definition of design options and quicker response by the tool while minimising the number of inputs. The space layout in PCC Tool is thus dimensionless and not strictly needed to predict cooling performance on the immediate surroundings of the ceramic walls. This approach leads to the first assumption considered in PCC Tool, where the predicted cooling performance does not respond to the downstream space energy balance. Instead, it relies on a constant (and positive) ΔDBT that will continuously drive air onto the ceramic wall (Figure 5.19). In a real scenario, the internal space temperature and thus ΔDBT will be affected by the cooling input of the system, internal gains and other parameters affecting the space energy balance. A change in ΔDBT will alter the velocity of the air driven by thermal forces and moving towards the ceramic system, which at the same time will influence on the final cooling achieved. However, the small area of influence of PCC systems will have a limited impact on the average temperature of the bigger adjacent indoor zone. A constant ΔDBT and v is thus a reasonable simplification of a theoretical model for early stage performance prediction and valid for most scenarios where PCC system can be applied.

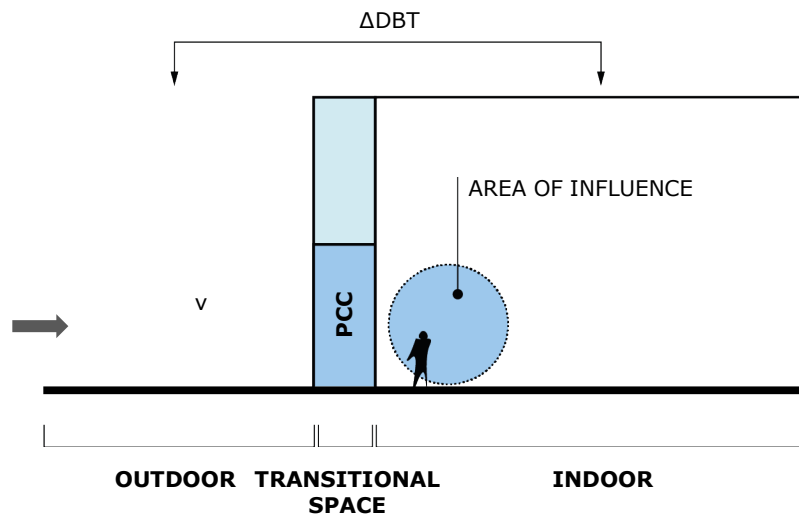


Figure 5.19 PCC Tool: dimensionless space & constant ΔDBT assumption

The parametric database, also considered the 'core' of the tool, becomes itself a potential limitation of PCC Tool due to a reasonable limited range of environmental conditions and porous ceramic designs considered. Although the database is vast and responds to most common scenarios, it is apparent that there is room for further increase. Nevertheless, the environmental conditions (DBT-WBT) included in the database cover more than 75% of the summer conditions in Europe (Chapter 1.2). A major limitation could be found on the ceramic shape and dimensions included in the tool. A change of size of a given ceramic sample will alter its drag coefficient and thus its resistance to flow. This means, for instance, that the cooling performance will be different for two systems made of different ceramic sizes but presenting the same blockage ratio and ceramic surface area. This prevents the interpolation of data and results between different ceramic sizes and limits the number of ceramic designs that can be tested by PCC Tool.

Among the above-mentioned tool restrictions, there are a number of assumptions also adopted that constitute the theoretical model:

- PCC Tool does not size the water supply (top irrigation) system (Chapter 5.2.1). Instead, it assumes a fully saturated ceramic surface during the entire operational period. This is equivalent to evaporated water particles immediately replaced by a continuous and homogeneous water distribution system over the ceramic surface.

- Wind direction is assumed perpendicular to the ceramic wall with a homogenous distribution on vertical direction.
- Ceramic walls are not exposed to direct solar radiation nor any other heat source that may alter its surface temperature.
- PCC Tool considers smooth ceramic surfaces with average roughness height (R_A) of $0.8\ \mu\text{m}$, typically achieved through an extrusion manufacturing process.

The approach for future development may thus address some of the restrictions commented in this section in order to access a wider audience and allow a wider range of ceramic samples and extreme environmental conditions to be tested. In order of priority, the areas to be improved in PCC tool are listed as follows:

- Integrating a water supply system design option to provide with the required water flow rates for the given ceramic shape and the number of engraved water channels on the ceramic surface.
- An increased parametric database, including different ceramic sizes and alternative ceramic shapes.
- The addition of a second ceramic wall parallel to the main one, as a strategy to increase the exposed ceramic surface area.

5.5 Conclusions

The multi-steady-state tool presented in this chapter allowed the prediction of cooling capacity and water consumption of a PCC systems with a reduced number of inputs and accepting a wide range of ceramic shapes and arrangements. The tool, as an interface, is a successful approach to manage the large amount of data generated by the CFD studies in an efficient manner and provided with sufficient data to assess the applicability of PCC systems. By using the tool, designers could predict, for instance, that a $30\ \text{m}^3$ PCC system built with tubular ceramic samples arranged in the vertical plane and presenting a blockage ratio of 50%, can reduce the surrounding air temperature by 2 degrees in the peak hours of the day ($\text{DBT-WBT} \geq 14^\circ\text{C}$). The water consumption for one hour of operation is estimated 72 litres, half the average household water consumption in Europe, and the mean radiant temperature (MRT) depression achieved downstream the ceramic surface is 1.8°C at 1m distance. By coupling the air temperature and mean radiant temperature reduction achieved with the psychological cooling effect of air movement, the user perception of the

environment can be considerably improved, making the PCC systems a good alternative to minimise local thermal discomfort under hot and dry conditions.

The tool outcomes were also validated, and both predicted temperature reduction and water consumption were in line with laboratory data. It is hoped that PCC Tool will raise awareness and knowledge on the benefits and limitations of this system and ease the integration of evaporative cooling systems in early design stages.

6 Design & Integration of Natural Ventilation Strategies

The last decade has witnessed how passive cooling techniques became increasingly attractive methods to architects for delivering healthy and comfortable environments as an alternative to the 'default' use of mechanical cooling systems. Indeed, a good selection and design of a naturally ventilated strategy can meet solar and internal heat loads in the UK (daily average of around 30 to 40 W/m²), overcoming overheating problems in most building types (CIBSE AM10, 2005). In warmer climates, when ambient air temperature frequently exceeds 30°C, alternative cooling methods may be needed. If an evaporative cooling system is to be applied, the design and size of a natural ventilation strategy coupled with the evaporation media is still necessary to drive the cooled air into the space. In the case of the developed PCC system using P2, the renovation of saturated air surrounding the ceramic surface is essential to maintain the evaporation process and, if a passively driven airstream is anticipated, the space layout must be carefully considered.

Compared to mechanical systems, the natural ventilation design process requires more attention, as high volumes of fresh air are delivered at low air velocities, requiring bigger supply, air circulation and outlet areas, and having a major impact on the design of the space layout. Inappropriate space dimensions or proportions, small aperture areas and obstruction in the airflow path can lead to a poor performance of the strategy. It is thus essential to address the design and application of natural ventilation strategies in combination with the evaporative system (if existing) from early design stages and along the design process. As any other environmental strategy to be applied in design, this involves feasibility studies, pre-sizing the system using simple rules of thumbs, components design and detailed performance analysis.

Emergent technologies and the development of advanced dynamic thermal simulation and CFD tools allowed to analyse in detail the performance of a natural ventilation strategy. Thermal simulations allow advanced natural ventilation studies, including operational and aperture area control, thermal and wind force consideration, air stratification and multi-zones models. In the other hand, CFD tools allow (among a wider range of features) further analysis of airflow paths and pressure and air velocity distribution within a volume of air.

The physics involved in natural ventilation for convective cooling are indeed manageable with manual calculations and some of the expressions have been successfully implemented in simple spreadsheet tools and applied on early design stages as discussed in Chapter 2.3.1. These spreadsheets can be an advantage for experienced designers to speed up the sizing process, but at the same time, it may be insufficient for less experienced designers. The strong engineering component of these tools are often an obstacle for most architects and designers.

The use of simple calculations to design a natural ventilation strategies on early design stages has been proved advantageous for the successful performance of the strategy (Ford, 2003), and more accessible tools for designers with limited knowledge are needed to successfully design naturally ventilated spaces.

During the development of this thesis, an opportunity to redevelop an existing spreadsheet tool named Optivent appeared. The creation of a more intuitive tool for architects to evaluate the feasibility of a natural ventilation strategy and, at the same time, inform PCC tool on the naturally driven airflow achieved through the ceramics was considered of great interest.

Optivent was developed originally by Dr Camilo Diaz and Prof. Brian Ford as an in-house tool for use by a practice of consultants in the UK. It was applied in evaluating options, airflow path strategies and preliminary sizing of vent opening areas in a series of buildings in Europe, North America, India and China. Over the last ten years, the original tool has also been applied by post-graduate students at the University of Nottingham in the evaluation of their design projects (Vallejo et al., 2015a).

Over this time, the original tool presented a number of limitations and potential improvements that have been addressed during the redevelopment of the tool now named Optivent 2. These limitations were related to the software platform used (Optivent was a spreadsheet tool), the number of strategies that could be evaluated (Optivent was only applicable to single-cell models with a maximum of two apertures) and the output provided (Optivent only informed on the airflow rates achieved with no consideration of indoor thermal comfort). Apart from maintaining the same essence and aim, Optivent 2 resulted in a new tool developed from the ground up that presents a new web-based platform and interface, more ventilation strategies to be evaluated (including multi-cell models), new calculation methods and additional outputs that also inform on the thermal comfort conditions of the ventilated space.

This chapter introduces the most common component design and natural ventilation options at early design stages and describes the process of integrating the principles of natural ventilation in a new and intuitive steady-state natural ventilation design tool to allow the evaluation of a wide range of generic airflow strategies, support strategic decisions regarding the feasibility of natural ventilation and inform in the design process of PCC systems. The tool workflow and calculation methods are described, and a validation process was undertaken with a fieldwork conducted in a naturally ventilated theatre in the UK (Appendix A3).

Optivent 2 tool is accessible through: <http://naturalcooling.co.uk/optivent.html>

6.1 Component Design & Strategies

The success of a natural ventilation strategy can be measured by comparing the airflow rate achieved by the system, and the airflow rate required to remove certain amount of heat. The amount of heat to be removed derives from the heat generated in the given space and the established comfort criteria; whereas the achieved airflow rates depends on the environmental conditions and the driving forces involved. Thermal force alone (also known as buoyancy force) derives from the temperature difference between the room and the supply air temperature and is usually the only force considered when sizing aperture areas for natural ventilation in early design stages, representing a theoretical worst-case scenario.

6.1.1 Ventilation Strategies

For naturally driven ventilation the air flow path may be essentially either an *updraught* or a *downdraught* strategy. An *updraught* airflow strategy usually makes provision for low level inlet openings and high level outlet openings. Sometimes referred to as the *stack* effect, this relies on the fact that warm air rises and is replaced by relatively cooler (denser) air. This flow of air up through the building will continue as long as the low and high level vents remain open, and as long as there is a positive temperature difference between outside and inside the building. The following section explores a number of different *updraught* airflow strategies.

Single-Sided Ventilation

Possibly the simplest *updraught* airflow strategy is to provide openings on one side of a single room, to allow cooler outside air to enter at low level and for the internal warmer air to rise and leave the space from the upper part of the opening. The air flow rate induced by this strategy can be enhanced by providing a low-level *supply* opening and a separate high-level *exhaust* opening. This strategy has been found to be effective for room depths of about twice the room height (CIBSE AM10, 2005, p. 15), although this depends on occupant density. While this may be satisfactory for a low-density office space, it is unlikely to be satisfactory for the much higher occupant density of a school classroom, for example, so the strategy must of course account for internal heat gains from occupants, equipment etc.

Cross Ventilation

Cross-ventilation will generally be a more effective strategy than single-sided ventilation, because cross-ventilation takes advantage of wind pressure differences across the building (Passe and Battaglia, 2015, p. 151). Generally, cross ventilation has been found to be effective for building depths up to five times the room height (CIBSE AM10, 2005, p. 16), although the achievable air flow rate will be significantly influenced by the area of inlets and outlets and the height difference between the mid-point of these openings. The air volume flow rate is directly proportional to the effective area of the ventilation openings, and that air flow rate increases in proportion to the square root of the *stack* height.

Atrium/Concourse

Single sided and cross ventilation are limited by the floor to ceiling height and depth of the space considered, and so for many non-domestic buildings air flow strategies which link multiple floors within a building can have significant advantages (Passe and Battaglia, 2015, p. 153). The design of a central atrium or concourse can significantly enhance air flow rates through multi-floor buildings. In the case of an open office arrangement, (where the total volume will be limited by fire compartmentation), the space can be treated as a single cell, while recognising the role of the openings at each floor level in achieving an even distribution of air. It is also important to ensure that the final exhaust is significantly above roof level to avoid undesired flow reversal in top levels, where wind force could create negative pressures higher than the positive pressure from thermal force. Alternatively, the atrium/concourse and the adjacent floors

may be enclosed, in which case the linked spaces are treated as a multi-cell arrangement.

Perimeter Stack

A further option for promoting airflow through multi-floor buildings is to provide a perimeter stack or ventilated façade to enhance both buoyancy and wind driven ventilation. Under buoyancy conditions, it is assumed that the air temperature within the stack is higher than in the occupied floors, in most cases increasing the pressure difference across the openings. Depending on orientation and latitude, this can be enhanced by encouraging solar gain within the stack, sometimes called solar chimneys (CIBSE AM10, 2005, p. 17).

6.2 Optivent 2: A Steady-State Natural Ventilation Tool

A primary objective during the development of Optivent 2 was to assist on the definition of a reliable ventilation strategy that will achieve the ventilation flowrates required to provide a healthy environment and avoid the risk of overheating. In addition to this main objective, there was an intention to allow a wide range of generic airflow strategies that can be explicitly evaluated, extend the geographic applicability of the tool, and incorporate a user-friendly graphic interface accessible from any device in a web-based platform.

6.2.1 Design Tool Workflow

Despite many engineer-oriented natural ventilation design guidelines, the tool targets architects and designers with no need of previous knowledge in the field. This means the main inputs refer to the building layout (aperture areas, stack heights, etc) and the achieved airflow rates become the main output for a later user's evaluation. This approach leads to a quick input process, understanding that in most cases the user is more familiar with the building geometry than with the volumes of air required. To assist on the decision-making process and the evaluation of the feasibility of a ventilation strategy, additional information on the building context (internal and external conditions) is also processed to allow a comparison between air volumes achieved and air volumes required for fresh air or for cooling before considering any strategy as valid.

The methodology is thus a five-step process where the user is asked to select a ventilation strategy, define the building layout and identify internal gains and

solar exposure in order to estimate the airflow rates required to provide fresh air and for removing excess heat. Finally, a set of outcomes including airflow rates, air velocities and thermal comfort prediction are plotted in comparative charts to give the user sufficient information to evaluate the feasibility of the selected strategy. The entire process takes only a few minutes, inviting the user to explore further strategies or different space layouts for a better understanding of the impact of different factors (stack height, aperture areas, internal-external temperatures) affecting the airflow rates achieved.

Natural ventilation strategies

Most of the space arrangements available in the tool are in line with CIBSE AM10 (2005) document and already described in 6.1.1. The tool covers single sided and cross ventilation through a single aperture or two open vents in an isolated space (single cell), a single-cell building with multiple inlets and a stack or atrium connected to wide open occupied spaces, and a multi-cell building with multiple inlets and occupied zones connected to a well-defined chimney, defining an additional zone. A downdraught scenario is also available for this last scenario to allow feasibility studies on the application of direct evaporative cooling systems. In total, the tool integrates four scenarios plus two sub-scenarios for each of the main ones as indicated in Figure 6.1.

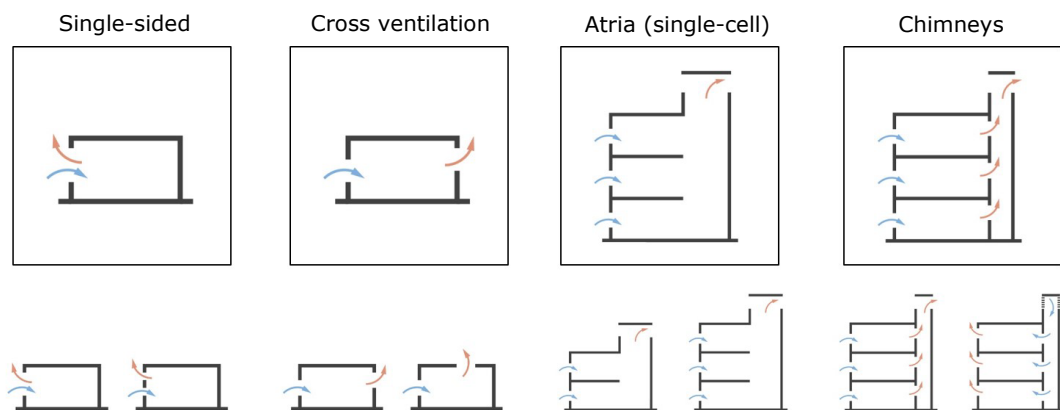


Figure 6.1 Optivent 2: Ventilation strategies

Project location

The first set of inputs in Optivent 2 refer to the geographical location of the space to be tested. Although there is not a direct influence of this information on the airflow rates achieved (except from the wind velocity), it is strongly related to the prediction of comfort and the estimation of solar gains, needed to

evaluate the feasibility of the strategy. Each of the user inputs numbered in Figure 6.2 is described below:

1. The latitude, month and hour of the analysis indicate the solar azimuth and altitude for the selected time in the analysis. The solar position with respect with the space's surface azimuths are used to estimate solar gains (direct and conductive) in the zone. If the objective of the natural ventilation strategy is to minimise overheating risk, it is recommended to select a daytime when solar gains are expected to be higher.
2. Prevailing mean outdoor temperature was introduced by ANSI/ASHRAE (Standard 55-2013) as an input variable for the adaptive comfort model, which is used in Optivent 2 to evaluate overheating problem in the analysed zone. It is based on the arithmetic average of the mean daily outdoor temperatures over no fewer than 7 and no more than 30 sequential days prior to the day in question. This information can be obtained from weather data or on-site monitored data if available.
3. Wind speed is a required user input to estimate airflow rates achieved in the selected zone driven by wind force. It is thus considered in the Buoyancy + Wind calculations and wind direction is assumed perpendicular to the inlets. It is recommended to consider reduced wind velocities as result of the obstructions generated by the complexity of the urban layout. To assist with this input, a terrain data factor can be applied on the meteorological wind speed (typically recorded by unobstructed weather stations at 10 m high) in order to obtain more realistic wind pressure coefficients. The terrain coefficients are obtained from CIBSE AM10 (2005, Table 4.12, p.55) and the calculation methods are described in 6.2.2.
4. The inlet (surface) azimuth is used to estimate solar gains in cells for the current sun position (given by latitude, month and time). From this input, the tool associates a surface azimuth to each building element (vertical and horizontal) conforming the space.

Latitude (decimal degrees) ①

Month

Hour ②

Prevailing mean outdoor temperature (°C) ②

Prevailing mean outdoor temperature was introduced in ASHRAE Standard 55-2010 as an input variable for the adaptive model. It is based on the arithmetic average of the mean daily outdoor temperatures over no fewer than 7 and no more than 30 sequential days prior to the day in question.

Meteorological Wind Speed (m/s) ③

Terrain data ③

Wind speed is only considered in Buoyancy + Wind calculations. Wind direction is assumed perpendicular to the surface of the inlets. It is recommended to consider reduced wind velocities according to the complexity of the urban layout.

Inlet (surface) Azimuth ④

The ASHRAE Clear Sky Model (2005) initially developed by J. L. Threlkeld in 1962, is used to estimate hourly clear sky beam and diffuse irradiance for any month of the year. This model is applicable to the United States continent and its applicability outside the USA has never been demonstrated. Optivent 2.0 extends the Clear Sky application to both northern and southern hemispheres. Note that the Clear Sky model has been updated in later versions of ASHRAE H-F.

The inlet (surface) azimuth is used to estimate solar gains in cells for the current sun position (given by latitude, month and time).

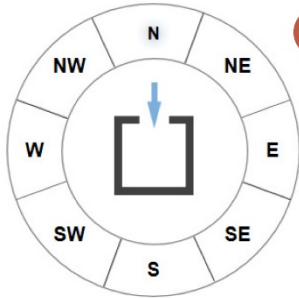


Figure 6.2 Optivent 2: Project location data input

Airflow data input

This section collects the necessary data for the estimation of airflow rates achieved in the selected zone and strategy. The majority of the user inputs are shown over diagrams and images that help understanding the meaning of each value. The diagrams also provide with a simple record of input assumptions and aperture heights, areas, and temperatures are displayed as shown in Figure 6.3. Each numbered input is described below:

5. Buoyancy driven ventilation rates depend on the difference between the internal and external air temperature among other factors. It is recommended to assume a ΔDBT between 1°C and 3°C for day-time ventilation and between 2°C and 4°C for night-time ventilation. A higher temperature difference would lead to higher ventilation rates, but to guarantee a good ventilation performance under adverse conditions, small temperature differences are strongly recommended, and considered *good practice* for preliminary aperture sizing.
6. Aperture areas and stack heights are main indicators (as well as ΔDBT) of the airflow rates achieved by buoyancy forces. Indeed, aperture areas are directly proportional to the natural ventilation rates achieved (doubled the area, doubled the flow rates). The areas given on the diagram refer to construction areas, and the percentage of free area can be indicated in a different input. The heights associated to each aperture refer to the distance between the ground floor level and the centroid of the opening.

7. Effectives apertures indicate the percentage of effective area considered for the estimation of natural ventilation flow rates. This parameter is given as percentage (of construction area) and must be specified for each aperture in the model. It is clearly related to the way the window opens and the surrounding head, sill and jamb details. Some indicative areas are suggested by the tool for typical aperture types. Reference should be made to actuator opening details from manufacturer's data.

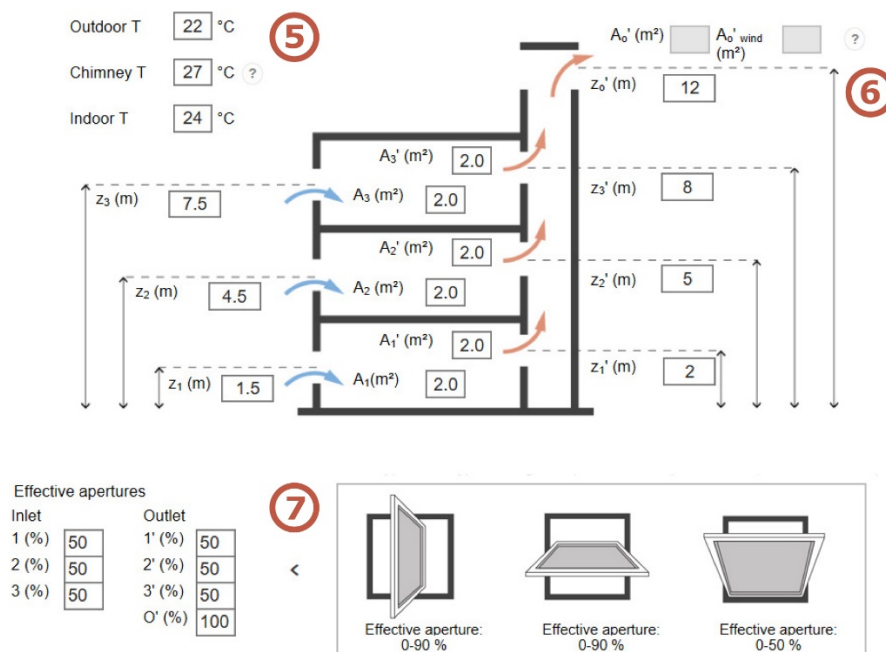


Figure 6.3 Optivent 2: Airflow data input

Building geometry & solar gains

The definition of the space layout and the construction thermal properties are required for the estimation of internal and solar gains, and thus the ventilation rates required to remove the heat generated in the zone. Each numbered input in Figure 6.4 is described below:

8. Space height, width and length will determine the floor area and the space volume. The floor area is needed to estimate total internal gains, whereas the space volume is used to obtain the equivalent air changes per hour (ACH) required and achieved in the zone. Height, width and length also determine the surrounding wall's surface area which, together with their exposure to sun and surface azimuth, determine the amount of

conductive solar gains through the vertical elements. In addition, the building geometry and the graphs displayed do not necessarily represent the entire building, and they could represent a single room or an interior bay of a larger open space.

9. The thermal properties are required for those elements exposed to sun and the estimation of solar gains. For the glazing elements, the solar transmittance factor (0-1) and the shading proportion (%) determine the direct solar gains. The solar transmittance factor is the fraction of solar radiation transmitted by the glazing, expressed as a number between 0 and 1, where 1 indicates the maximum possible solar heat gain, and 0 no solar heat gain. The shading proportion is the proportion of the glazing that is in the shade, expressed as percentage, where 100% indicates full shading and 0% no shading. This is used to represent external shading. For opaque elements, the external surface absorptance (0-1), the construction U-value ($\text{W/m}^2\text{K}$) and its external surface transmittance ($\text{W/m}^2\text{K}$) are used to estimate the conductive solar gains. The surface absorptance is the proportion of solar radiation absorbed by a surface, expressed as a number between 0 and 1. Standardised values suggest 0.9 for dark coloured materials and 0.5 for light coloured materials. The external surface transmittance is the thermal transmittance of the external surface exposed to solar radiation expressed in $\text{W/m}^2\text{K}$ and it is dependent on the thickness of the construction layer and thermal conductivity of the material.

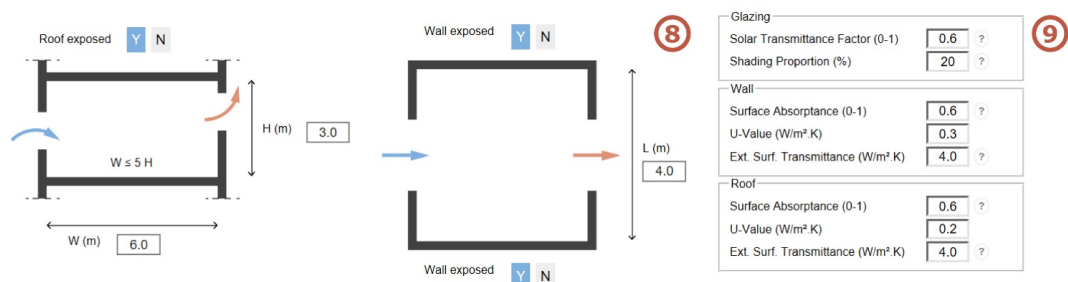


Figure 6.4 Optivent 2: Building geometry & solar gains

Internal conditions

The calculation of internal gains is used to estimate the amount of air required to remove the heat generated in the zone (required for cooling). The internal gains refer only to a single cell and for multi-cell scenarios they are assumed to be the same for each cell. Each numbered input in Figure 6.5

10. The number of people per zone is used to calculate the minimum ventilation required for the supply of fresh air (10 l/s). The heat generated for the indicated metabolic activity is used for the estimation of air required for cooling.
11. Equipment and lighting sensible gains (W/m²) also determine the required air for cooling and they may change according to the building type. Benchmark allowances for internal heat gains can be obtained from CIBSE Guide A (2015, Table 6.2, p.6-3).

Occupants		Internal Gains	
<i>The number of people is used to calculate the minimum ventilation required for the supply of fresh air (10 l/s).</i>		equipment W/m ²	15.0
at rest (76W)	0 people	lighting W/m ²	10.0
office work (85W)	2 people		
walking (100W)	0 people		
exercising (120W)	0 people		

Figure 6.5 Optivent 2: Internal conditions

Results

The calculation process outputs airflow rates driven by buoyancy and driven by buoyancy + wind. Buoyancy driven airflow rates would represent a worst-case scenario and must be considered for feasibility assessment. Buoyancy + wind driven airflow rates will inform the user about the impact (and potential benefits) of wind force in the chosen ventilation strategy. Three charts are plotted to provide a general overview on the performance of the strategy. The results and the user inputs are summarised in one A4 page which can be downloaded in PDF format for future data comparison, revisions, presentations, etc. Each numbered input in Figure 6.6 is described below:

12. The first graph indicates the airflow rates achieved in the given scenario and compares them with the airflow rates required for the supply of fresh

air and for removing the total heat gains generated within the space (required for cooling). The results are given in m^3/s , ACH, and Wh/m^2 by means of the space volume and indoor-outdoor temperature difference.

13. The second chart plots air velocities in m/s at the inlets and outlets, together with the average room air velocity. The graphs also relate the air velocities achieved with the physiological equivalent temperature reduction determined by the operative temperature expression (CIBSE Guide A, 2015, p5-6). Under hot conditions, air movement is a valid strategy to improve comfort conditions of occupants and must also be considered when assessing the benefits of a natural ventilation strategy.
14. The third chart predicts comfort conditions within the occupied space based on the adaptive comfort method from ANSI/ASHRAE (Standard 55-2013). The chart plots the operative temperature for each zone over the predicted comfort bands referring to 80% and 90% of user acceptability limits.

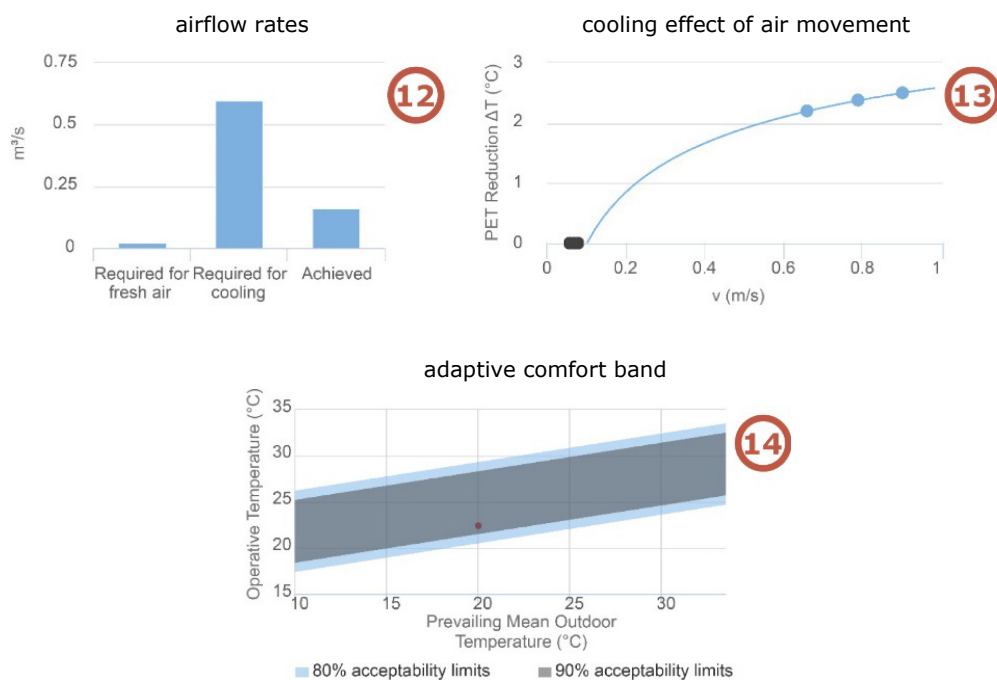


Figure 6.6 Optivent 2: Results

6.2.2 Calculation Methods

The calculation methods used in Optivent 2 implement the laws of physics involved in air movement to determine the achieved airflow rate, solar geometry

to determine solar gains, and a comfort theory to evaluate overheating risk. Each of these methods is described in this section.

Natural ventilation flow rates

The principle of mass conservation is applied in each envelope flow model Equation (6-1), where q_i is the airflow rate for each aperture (m^3/s). The airflow rate through each opening is expressed as a relationship between the pressure difference across the opening and the specified effective aperture area Equation (6-2) and Equation (6-3), where Cd_i is the discharge coefficient, A_i is the effective aperture area (m^2), S_i indicates the flow direction (-1 or +1), Δp_i is the pressure difference across the opening (Pa), Δp_0 is the external-to-internal hydrostatic pressure difference at ground level (Pa), $\Delta \rho_0$ is the density difference at ground level ($\text{kg}\cdot\text{m}^{-3}$), g is the gravitational force per unit mass ($\text{m}\cdot\text{s}^{-2}$), z_i is the height of the opening above ground level (m), U is the air velocity perpendicular to the aperture (m/s) and Cp_i is the corresponding wind pressure coefficient. All these equations are in line with CIBSE AM10 (2005).

$$\sum q_i = 0 \quad \text{Equation (6-1)}$$

*Principle of mass conservation
CIBSE AM10 (2005, p. 40)*

$$q_i = Cd_i \cdot A_i \cdot S_i \sqrt{\frac{2|\Delta p_i|}{\rho_0}} \quad \text{Equation (6-2)}$$

*Airflow rate through an opening
CIBSE AM10 (2005, p. 40)*

$$\Delta p_i = \Delta p_0 - \Delta \rho_0 \cdot g \cdot z_i + 0.5 \cdot \rho_0 \cdot U^2 \cdot Cp_i \quad \text{Equation (6-3)}$$

*Pressure difference across an opening
CIBSE AM10 (2005, p. 41)*

In order to reduce the user inputs, the discharge coefficients and wind pressure coefficients have been set to default values optimised for each airflow model. Table 6.1 and Table 6.2 show the coefficients used for each aperture according to the selected scenario, surface azimuth and the wind direction.

Table 6.1 Discharge coefficients used for each scenario

	Inlets	Outlets
Single-sided (single aperture)	0.25	0.25
Single-sided (two apertures)	0.61	0.61
Cross ventilation	0.61	0.61
Atria	0.61	0.61
Chimneys	0.61	0.80

Table 6.2 Wind pressure coefficients for different aperture orientations

	Windward	Leeward	Roof
C_{pi}	0.6	-0.1	-0.4

One particular feature from Optivent 2 is its calculation process, which responds to an implicit method for determining the results. David Etheridge denotes this method when the above equations are used to calculate flow rates for given openings (Etheridge, 2012b). The opposite method, the explicit method, is where the magnitude and direction of the flow rates are specified, and the areas of the openings are calculated. This last method is commonly used in engineering design guidelines; however, the implicit method is more attractive to designers, who are more familiar with the building geometry rather than with the volumes of air required.

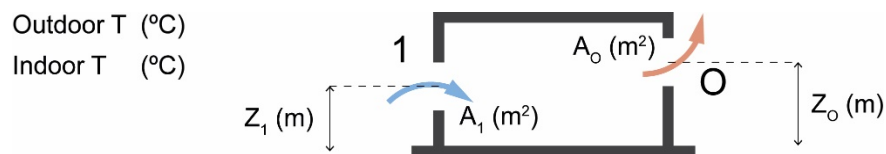


Figure 6.7 Example for airflow rates calculation (single-cell scenario)

Therefore, for a simple cross ventilated model as shown in Figure 6.7, four steps are followed in order to determine the airflow rates achieved.

- i. Definition of the density difference ($\Delta\rho_0$) using the following approximation Equation (6-4), where ρ_0 is the reference density taken as $1.20 \text{ kg}\cdot\text{m}^{-3}$, ΔDBT is the internal-to-external temperature difference ($^{\circ}\text{C}$) and T_E is the external temperature ($^{\circ}\text{C}$).

$$\frac{\Delta\rho_0}{\rho_0} = \frac{\Delta DBT}{T_E + 273}$$

Equation (6-4)

Air density difference

CIBSE AM10 (2005, p. 55)

- ii. Determining the external-to-internal pressure difference at ground level (Δp_0) by defining the pressure difference across each aperture as a function of Δp_0 using Equation (6-3), substituting the obtained Δp_i in Equation (6-2) to determine the airflow rates in each aperture as a function of Δp_0 , and finally identifying the Δp_0 that satisfies Equation (6-1). The resulting expression is given by Equation (6-5). Note that in absence of wind (buoyancy alone) U can be considered as 0.

$$\Delta p_0 = \frac{\Delta\rho_0 \cdot g \cdot (z_1 \cdot A_1^2 + z_0 \cdot A_0^2) - 0.5 \cdot \rho_0 \cdot U^2 \cdot (Cp_1 \cdot A_1^2 + Cp_0 \cdot A_0^2)}{A_1^2 + A_0^2}$$

Equation (6-5)

*Air pressure
difference at
ground level*

- iii. Calculating the pressure difference across each aperture (Δp_i) from Equation (6-3) for the given aperture height, wind velocity, reference density (taken as $1.20 \text{ kg}\cdot\text{m}^{-3}$) and the previously calculated density difference ($\Delta\rho_0$) and external-to-internal pressure difference at ground level (Δp_0).
- iv. Calculating the airflow rates achieved in each opening from Equation (6-2) for the given aperture area, reference density and the previously calculated pressure difference across each aperture (Δp_i).

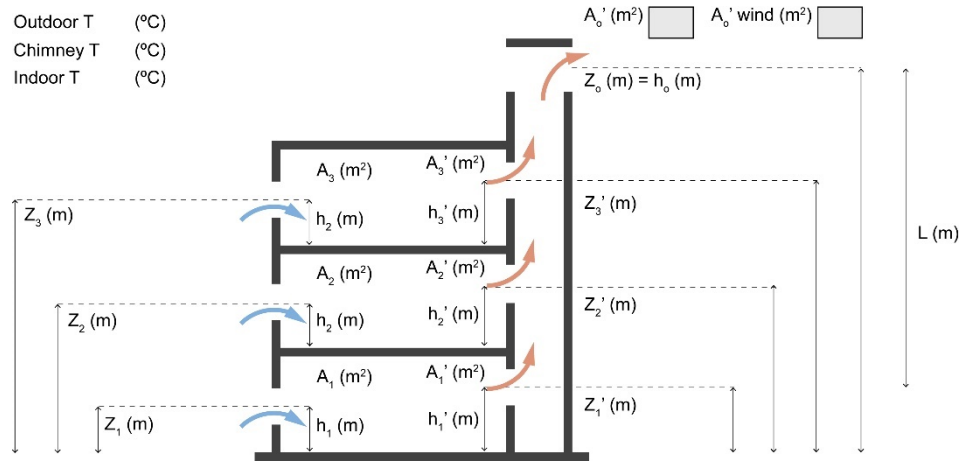


Figure 6.8 Example for airflow rates calculation (multi-cell scenario + multiple apertures)

In zones with multiple apertures (Figure 6.8), the pressure differences across the openings are shown in Equation (6-6).

$$\Delta P_1 = P_E - P_1 - \rho_E \cdot g \cdot z_1 + \rho_I \cdot g \cdot h_1 + 0.5 \cdot \rho_E \cdot U^2 \cdot Cp_1 \quad \text{Equation (6-6)}$$

*Air pressure
difference across
multiple openings*

$$\Delta P_2 = P_E - P_2 - \rho_E \cdot g \cdot z_2 + \rho_I \cdot g \cdot h_2 + 0.5 \cdot \rho_E \cdot U^2 \cdot Cp_2$$

$$\Delta P_3 = P_E - P_3 - \rho_E \cdot g \cdot z_3 + \rho_I \cdot g \cdot h_3 + 0.5 \cdot \rho_E \cdot U^2 \cdot Cp_3$$

$$\Delta P_O = P_O - P_E - \rho_O \cdot g \cdot h_O + \rho_E \cdot g \cdot z_O - 0.5 \cdot \rho_E \cdot U^2 \cdot Cp_O \\ + \rho_E \cdot g \cdot L - \rho_O \cdot g \cdot L$$

$$\Delta P_{1'} = P_1 - P_O + \rho_O \cdot g \cdot z_{1'} - \rho_I \cdot g \cdot h_{1'}$$

$$\Delta P_{2'} = P_2 - P_O + \rho_O \cdot g \cdot z_{2'} - \rho_I \cdot g \cdot h_{2'}$$

$$\Delta P_{3'} = P_3 - P_O + \rho_O \cdot g \cdot z_{3'} - \rho_I \cdot g \cdot h_{3'}$$

This set of expressions, require an iterative procedure to obtain the solution. Spreadsheets can often perform the iteration with no major complications but, to

avoid tentative iterations by the user seeking for the aperture areas and heights that satisfy the selected flow pattern, a new relationship between pressure drops was established. This condition is affecting the position of the neutral plane, which has been set at a height between the top inlets and outlet. As a result, the tool estimates the outlet area required to satisfy the selected flow pattern without the need of an iterative procedure. Moreover, this approach also avoids undesired flow reversal, which unfortunately is commonly happening in many poorly naturally ventilated buildings, reflecting the lack of integrated design (Figure 6.9). Therefore, apart from the principle of mass conservation Equation (6-1), the relationships established between pressure drops at the apertures and airflow rates in multi-cell scenarios are displayed in Equation (6-7).

$$\Delta P_0 = \Delta P_3 + \Delta P_{3'}, \quad \text{Equation (6-7)}$$

*Relationship between air
pressure drops*

$$A_1^2 \cdot \Delta P_1 = A_{1'}^2 \cdot \Delta P_{1'},$$

$$A_2^2 \cdot \Delta P_2 = A_{2'}^2 \cdot \Delta P_{2'},$$

$$A_3^2 \cdot \Delta P_3 = A_{3'}^2 \cdot \Delta P_{3'},$$

The first expression sets the neutral plane between to top apertures, whereas the others satisfy the mass conservation at each zone. Finally, Optivent 2 outputs the ventilation flow rates for each aperture, and the outlet area required to satisfy the airflow path shown in the diagram.

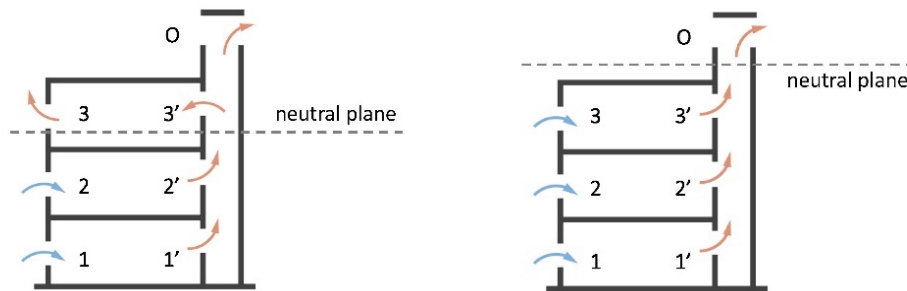


Figure 6.9 Neutral plane position and impact on airflow pattern

Solar gains

The ASHRAE Clear Sky Model (ASHRAE, 2005, Chapter 31) initially developed by J. L. Threlkeld (Threlkeld, 1962) is used to estimate hourly clear sky beam and diffuse irradiance for any month of the year. This model is applicable to the United States continent and its applicability outside the USA has never been demonstrated by the authors. Although the Clear Sky model usually overestimates the amount of solar radiation available and it has been updated in later versions integrating climate data (ASHRAE, 2013), is the initial model the one implemented in Optivent 2 as it does not require a climatic database and a preliminary approximation is sufficient to estimate the airflow rates required for cooling at early design stages. The application of the Clear Sky model has been also extended to both northern and southern hemispheres as described by Energy Plus Documentation (2017).

Nine steps are thus followed in order to estimate the incident solar radiation on a vertical or horizontal surface:

- i. Define the hour angle H (degrees), which determines the angular displacement of the sun east or west of the local meridian, for the given apparent solar time (AST) using Equation (6-8):

$$H = 15(AST - 12) \quad \text{Equation (6-8)}$$

Sun hour angle
ASHRAE (2005, p. 31.15)

- ii. Estimate the solar altitude β (degrees) from Equation (6-9) which relates solar altitude angle β to the latitude L , the previously calculated hour angle H and the solar declination δ (the angle between the earth/sun line and the equatorial plane), which can be obtained from Table 6.4. The trigonometric calculations must be performed in radians.

$$\sin \beta = \cos L \cdot \cos \delta \cdot \cos H + \sin L \cdot \sin \delta \quad \text{Equation (6-9)}$$

Solar altitude
ASHRAE (2005, p. 31.15)

- iii. Obtain the solar azimuth angle Φ (degrees) from its sine and cosine, given the expressions in Equation (6-10).

$$\sin \Phi = \sin H \cdot \cos \delta / \cos \beta \quad \text{Equation (6-10)}$$

Solar azimuth
ASHRAE (2005, p. 31.15)

$$\cos \Phi = (\cos H \cdot \cos \delta \cdot \sin L - \sin \delta \cdot \cos L) / \cos \beta$$

- iv. Obtain the surface-solar azimuth angle γ , which is the angular difference between the solar azimuth Φ and the surface azimuth ψ , which defines the surface orientation in degrees according to Table 6.3. Values of γ greater than 90° or less than -90° indicate that the surface is in the shade.

$$\gamma = \Phi - \psi \quad \text{Equation (6-11)}$$

Surface-solar azimuth angle
ASHRAE (2005, p. 31.15)

Table 6.3 Surface Orientations and Azimuths (degrees) measured from South

	N	NE	E	SE	S	SW	W	NW
Surface azimuth ψ	180°	-135°	-90°	-45°	0°	45°	90°	135°

- v. Obtain the angle of incidence θ , which is the angle between the line normal to the irradiated surface and the earth-sun line from Equation (6-12), where Σ is the surface inclination (0° for horizontal surfaces, and 90° for vertical surfaces).

$$\cos \theta = \cos \beta \cos \gamma \sin \Sigma + \sin \beta \cos \Sigma \quad \text{Equation (6-12)}$$

Angle of incidence
ASHRAE (2005, p. 31.16)

- vi. The beam (irradiance) component E_b (W/m²) on the surface is given by Equation (6-13) by means of the apparent solar irradiation A (W/m²), the atmospheric extinction coefficient B , the solar altitude β and the angle of incidence θ . A and B can be obtained from Table 6.4.

$$E_b = \cos \theta \left(\frac{A}{\exp\left(\frac{B}{\sin \beta}\right)} \right) \quad \text{Equation (6-13)}$$

Beam irradiance
ASHRAE (2005, p. 31.16)

- vii. The diffuse (irradiance) component E_d (W/m²) is given by Equation (6-14) by means of the surface inclination Σ , the beam irradiance component E_b , the angle of incidence θ C value obtained from Equation (6-4).

$$E_d = C \cdot E_b \frac{(1 + \cos \Sigma)}{2 \cdot \cos \theta} \quad \text{Equation (6-14)}$$

Diffuse irradiance
ASHRAE (2005, p. 31.16)

- viii. The ground reflected irradiance component E_r (W/m²) for surfaces at all orientations is given by Equation (6-15), where R is the ground reflectivity, taken as 0.2 (representing a typical mixture of ground surfaces).

$$E_r = \left(\frac{E_b}{\cos \theta} \right) (C + \sin \beta) \cdot R \left(\frac{1 - \cos \Sigma}{2} \right) \quad \text{Equation (6-15)}$$

Ground reflected irradiance
ASHRAE (2005, p. 31.16)

- ix. The total irradiance E (W/m²) on the given surface is finally obtained from the sum of the direct, diffuse and reflected components Equation (6-16).

$$E = E_b + E_d + E_r \quad \text{Equation (6-16)}$$

Total irradiance
ASHRAE (2005, p. 31.15)

Table 6.4 Extra-terrestrial solar irradiance and data coefficients. This data refers to the 21st day of each month during the base year of 1964 (Energy Plus Documentation, 2017)

	I _o (W/m ²)	ET (minutes)	δ (degrees)	A (W/m ²)	B	C
Jan	1416	-11.2	-20.0	1202	0.141	0.103
Feb	1401	-13.9	-10.8	1187	0.142	0.104
Mar	1381	-7.5	0.0	1164	0.149	0.109
Apr	1356	1.1	11.6	1130	0.164	0.120
May	1336	3.3	20.0	1106	0.177	0.130
Jun	1336	-1.4	23.45	1092	0.185	0.137
Jul	1336	-6.2	20.6	1093	0.186	0.138
Aug	1338	-2.4	12.3	1107	0.182	0.134
Sep	1359	7.5	0.0	1136	0.165	0.121
Oct	1380	15.4	-10.5	1166	0.152	0.111
Nov	1405	13.8	-19.8	1190	0.144	0.106
Dec	1417	1.6	-23.45	1204	0.141	0.103

The calculation of solar gains in the zone includes a direct component through transparent elements and a conductive component through opaque elements. By default, Optivent 2 considers the construction area of the inlets as transparent elements, and are exposed to irradiation. The solar exposure of the opaque building elements is given by the user in *building geometry and solar conditions* section through a YES/NO input.

The calculation of direct solar gains SG_d (W) through transparent elements is calculated with Equation (6-17), where E is the total irradiance on the given surface (W/m²), T is the glazing solar transmittance (0-1) defined by the user, S is the shading proportion (%) defined by the user, A_i is the construction area of the opening (m²) and a is the room absorptance, taken as 0.8.

$$SG_d = E \cdot T \cdot A_i \cdot a \cdot \left(\frac{1 - S}{100} \right) \quad \text{Equation (6-17)}$$

Direct solar gains
CIBSE Guide A (2015, p. 5-63)

The calculation of conductive solar gains SG_c (W) derive from the definition of the sol-air temperature (ASHRAE, 2013, p. 18.22), an equivalent outdoor air temperature that, in the absence of all the irradiation over the surface, gives the same rate of heat entry through the building element. It is calculated for each exposed element with Equation (6-18), where E is the total irradiance on the given surface (W/m²), U is the thermal transmittance of the building element

(W/m²K), a is the external surface absorptance (0-1) defined by the user, A_i is the surface area (m²) and U_E is the external surface transmittance, equivalent to the thermal transmittance of the external surface (layer) exposed to solar radiation expressed in W/m²K and it is dependent on the thickness and thermal conductivity of the material.

$$SG_c = \frac{E \cdot U \cdot a \cdot A_i}{U_E} \quad \text{Equation (6-18)}$$

Conductive solar gains

The total amount to solar gains SG (W) in the zone would be the sum of direct and conductive solar gains through each exposed building element of the zone Equation (6-19).

$$SG = \sum SG_d + SG_c \quad \text{Equation (6-19)}$$

Total solar gains

Ventilation required for fresh air

Natural ventilation design guidelines and standards like CIBSE AM10 (2005), CIBSE Guide A (2015, Chapter 4) and BS EN (13779:2007) suggest the need of a minimum supply of fresh air between 8-10 l/s per person to dilute carbon dioxide, occupation contaminants and give a feeling of freshness. A flowrate of 10l/s per person is considered by Optivent 2 to estimate the ventilation required for fresh air q_{min} (m³/s) with Equation (6-20), where n is the number of people in the zone. Any ventilation strategy should provide at least this amount of air during the occupancy time.

$$q_{min} = n \cdot 0.01 \quad \text{Equation (6-20)}$$

Minimum ventilation rate (fresh air)

Ventilation required for cooling

The ventilation required for cooling q_c is the volume of air needed per unit of time to remove the total amount of internal and solar heat gains Q_T (W) generated in the zone through convection. The calculation of the ventilation required for cooling derives from the sensible cooling expression from ASHRAE (2013, p. 18.13), where ρ is the air density taken as $1.20 \text{ kg}\cdot\text{m}^{-3}$, C_p is the specific heat of air ($1006 \text{ J/kg}\cdot^\circ\text{C}$) and ΔDBT the inside-to-outside temperature difference defined by the user.

$$q_c = \frac{Q_T}{\rho \cdot C_p \cdot \Delta DBT} \quad \text{Equation (6-21)}$$

Ventilation required for cooling
ASHRAE (2013, p. 18.13)

Psychological cooling effect of air movement

The human thermal perception of a given environment is the result of the joint effect of air temperature, mean radiant temperature and air velocity of a given environment. Increased air movement can provide an enhanced perception of thermal comfort due to the psychological cooling effect of air movement, which derives from the operative temperature expression OT ($^\circ\text{C}$) given in CIBSE Guide A (2015, p. 1.3) and is shown in Equation (6-22).

$$OT = \frac{DBT \cdot \sqrt{10 \cdot v} + MRT}{1 + \sqrt{10 \cdot v}} \quad \text{Equation (6-22)}$$

Operative temperature
CIBSE Guide A (2015, p. 1.3)

where DBT is the air temperature in $^\circ\text{C}$, MRT is the mean radiant temperature in $^\circ\text{C}$ and v is the average room temperature in m/s . To estimate the average room air velocity (v_{av}), the airflow rates q (m^3/s) achieved in the zone are divided by the cross-sectional area of the room (m^2), as shown in Equation (6-23), where L and H are the room length and height given by the user in meters. This approximation is considered as valid for feasibility studies and comfort evaluation on early design stages.

$$v_{av} = \frac{q}{L \cdot H}$$

Equation (6-23)

Average room air velocity

Thermal comfort prediction

The ASHRAE Standard 55 Adaptive model (ANSI/ASHRAE, Standard 55-2013) is used for the prediction of comfort and overheating risk in the naturally ventilated space. The range of acceptable temperatures in an indoor space depends on the running mean outdoor temperature given by the user (T_o) and the required percentage of user satisfaction. A neutral temperature (T_{conf}) is first calculated from Equation (6-24), and the limits of acceptable comfort zones are obtained by adding $\pm 2.5^\circ\text{C}$ for 90% of user satisfaction and $\pm 3^\circ\text{C}$ for 80% of user satisfaction. This methodology and graphic interface is also in line with the online tool CBE Thermal Comfort Tool developed by Tyler Hoyt, Stefano Schiavon, Alberto Piccioli, Dustin Moon and Kyle Steinfeld at Center for the Built Environment CBE, University of California Berkeley (Tyler et al., 2013).

$$T_{conf} = 0.31 \cdot T_o + 17.8$$

Equation (6-24)

Neutral temperature

ANSI/ASHRAE (Standard 55-2013, p. 12)

A predicted indoor temperature, deriving from the heat gains and ventilation losses in the analysed zones are thus compared against the calculated comfort band. The calculation of indoor temperatures over short time intervals is computationally intensive and difficultly performed with simple calculations. However, a series of assumptions and simplifications (6.2.3) were adopted in order to get a useful indication of the mean indoor temperature over a short period of time based on the conservation of energy in the space. The predicted indoor temperature, is given by Equation (6-25), where DBT_{OUT} is the external temperature in $^\circ\text{C}$, Q_T is the total amount of internal and solar heat gains (W) and VLC a ventilation loss coefficient, which has been defined as the ventilation convectance per degree temperature ($\text{W}/^\circ\text{C}$), as is calculated from Equation (6-26), where q is the airflow rate achieved in the analysed zone (m^3/s), ρ is the air density taken as $1.20 \text{ kg}\cdot\text{m}^{-3}$ and C_p is the specific heat of air ($1006 \text{ J/kg}^\circ\text{C}$). This method for indoor temperature prediction is limited as no conduction losses

through fabric nor infiltration has been consider; however, it has been considered sufficient to meet the objective of obtaining an indication of potential overheating issues in the zones.

$$DBT_{IN} = DBT_{OUT} + \frac{Q_T}{VLC} \quad \text{Equation (6-25)}$$

Predicted indoor temperature

$$VLC = q \cdot \rho \cdot C_p \quad \text{Equation (6-26)}$$

Ventilation loss convectance

6.2.3 Optivent 2 Limitations, Assumptions & Further Development

A series of assumptions have been taken along the development of Optivent 2 in order to simplify the calculation process while still achieving an acceptable accuracy. These assumptions also minimise the number of user inputs required, which is crucial for a tool that aims to perform feasibility studies in early design stages:

- No air temperature stratification has been considered in the zones, and therefore temperatures within the space are assumed to be the same at any given height.
- A hypothetical internal-to-external temperature is considered to predict natural ventilation flow rates in the zones. In northern Europe, generally indoor temperatures are higher than external temperatures as result of internal and solar gains, thus driving air into the indoor space. An indoor-outdoor temperature difference is suggested for daytime and night time ventilation strategies in order to obtain reliable results. Natural ventilation flow rates are directly proportional to temperature differences, and thus small values are recommended to not overestimate the performance of the selected ventilation strategy. Optivent 2 suggest a ΔDBT between 1°C and 3°C for day-time ventilation and between 2°C and 4°C for night-time ventilation, although different values can be defined by the user.
- The ASHRAE Clear Sky Model (ASHRAE, 2005) used for solar radiation calculation does not integrate climate data and usually over estimates the amount of solar radiation hitting vertical and horizontal surfaces. The solar gains calculated by Optivent 2 are thus the maximum gains that

could be achieved if clear sky-sunny conditions at the given month and time.

- The dynamic behaviour of the physics involved in the building energy balance are not considered in this tool, and thus not considered for the prediction of the internal temperature. Potential heat absorbed by building elements with high thermal capacity and conductive heat losses are not considered.
- Deriving from 4, surfaces temperatures are not calculated and thus mean radiant temperatures MTR are considered equal to air temperature. From this assumption, the predicted operative temperature OT_{IN} which is compared against the predicted comfort zone, also equals the predicted dry bulb indoor air temperature DBT_{IN} .

The approach for a further development of Optivent 2 may address some of the restrictions commented in this section and include some new features in order to access a wider audience and minimise the need of alternative thermal calculation tools to evaluate overheating risk. Some of the areas to be improved are listed as follows:

- The consideration of air humidity levels and the evaluation of natural ventilation as a strategy to reduce indoor air moisture content. This can be useful in humid climates and zones with high amount of latent gains, usually characterised by a high density of occupation.
- The possibility to set single building elements and apertures as wind-sheltered. This will allow more accurate studies of ventilation strategies by having certain control on wind forces.
- A further development on the calculation of the zone's energy balance and the inclusion of conductive heat exchange and, potentially, thermal mass capacity though the use of correction factors derived from dynamic thermal simulations applied to the predicted indoor temperature.

6.3 Optivent 2 Validation

Optivent 2 integrates the principles of natural ventilation using the numerical expressions included in ASHRAE (2013) and CIBSE AM10 (2005). Apart from these expressions being extensively validated (Etheridge, 2012b), the original tool was successfully applied in a series of buildings in Europe, North America, India and China. This included the design of the Stock Exchange in Malta with the architects Architecture Project (Ford and Diaz, 2003), the Centre for

Sustainable Energy Technologies, Ningbo, China by Mario Cucinella Architects (Cucinella, 2008) and the Lawrence Convention Centre, Pittsburgh, USA by Rafael Vinoly Architects (Vinoly, 2003). More recently, Optivent was used to support early stage design of the new Performing Arts Building for Nottingham Girls High School, UK with Marsh Growchowski Architects (Vallejo et al., 2015b). The development of this project was followed closely from beginning to completion, and field measurements were made to confirm that the achieved air flow rates were in line with design stage predictions, aiming to reaffirm the validity and reliability of the tool. The results obtained proved that the measured incoming and outgoing airflow were mostly balanced and in line with the calculated predictions at design stage using Optivent 2, presenting an average performance gap of 15%, results that were considered as satisfactory assuming the steady-state nature of the tool (Appendix A3).

6.4 Conclusions

Optivent was originally developed with the objective to evaluate natural ventilation options and assist in the design of the apertures. The redeveloped Optivent 2 extended the original capabilities with a more intuitive interface, providing with more efficient methods to size and allow preliminary performance evaluation of convective cooling techniques. The steady-state tool demonstrated that multiple design options including updraught and downdraught strategies for single and multi-cell models can be tested in a matter of minutes with a reduced number of inputs. Moreover, the outcomes from the tool have been proved to be reliable and the monitoring of the natural ventilation performance in the NGHS theatre during a winter and summer day (Appendix A3) proved that the measured incoming and outgoing airflow were mostly balanced and in line with the calculated predictions at design stage using Optivent 2, reaffirming the value of integrating environmental design approaches and the high reliability of early stage design tools.

Since 2016, the new Optivent 2 has been continuously introduced and well received by postgraduate students from several MSc courses in the UK and future objectives target the promotion of Optivent 2 in architecture and engineering practices. It is hoped that this tool will contribute to an increase in the number of professionals with expertise in designing for natural ventilation, and that this will lead to wider application of natural ventilation and passive cooling solutions in architecture.

The tool also became a good companion to the developed PCC tool by informing on the naturally driven airflow rates at the ceramic screen and assisting in the design of the space layout in order to achieve the above mentioned air movement.

7 PCC System Design Workflow

Previous chapters have introduced numerical models to evaluate different aspects of the cooling process. Each of the models has been successfully validated and contributed to facilitate the integration of convective and evaporative cooling techniques at early design stages. This chapter describes the usage of the steady-state models PCC Tool and Optivent 2 in a design workflow for PCC systems. The design process of PCC systems must include a climate analysis to understand the applicability of the system, a study on the suitability of the ventilation strategy that drives air into the ceramic wall, and a performance evaluation of the PCC system for the given context (ambient conditions and airflow rates). While the climate analysis can be performed with any data processing tool, the evaluation of natural ventilation and PCC performance can be addressed using Optivent 2 and PCC tool. The resulting workflow is illustrated in Figure 7.1.

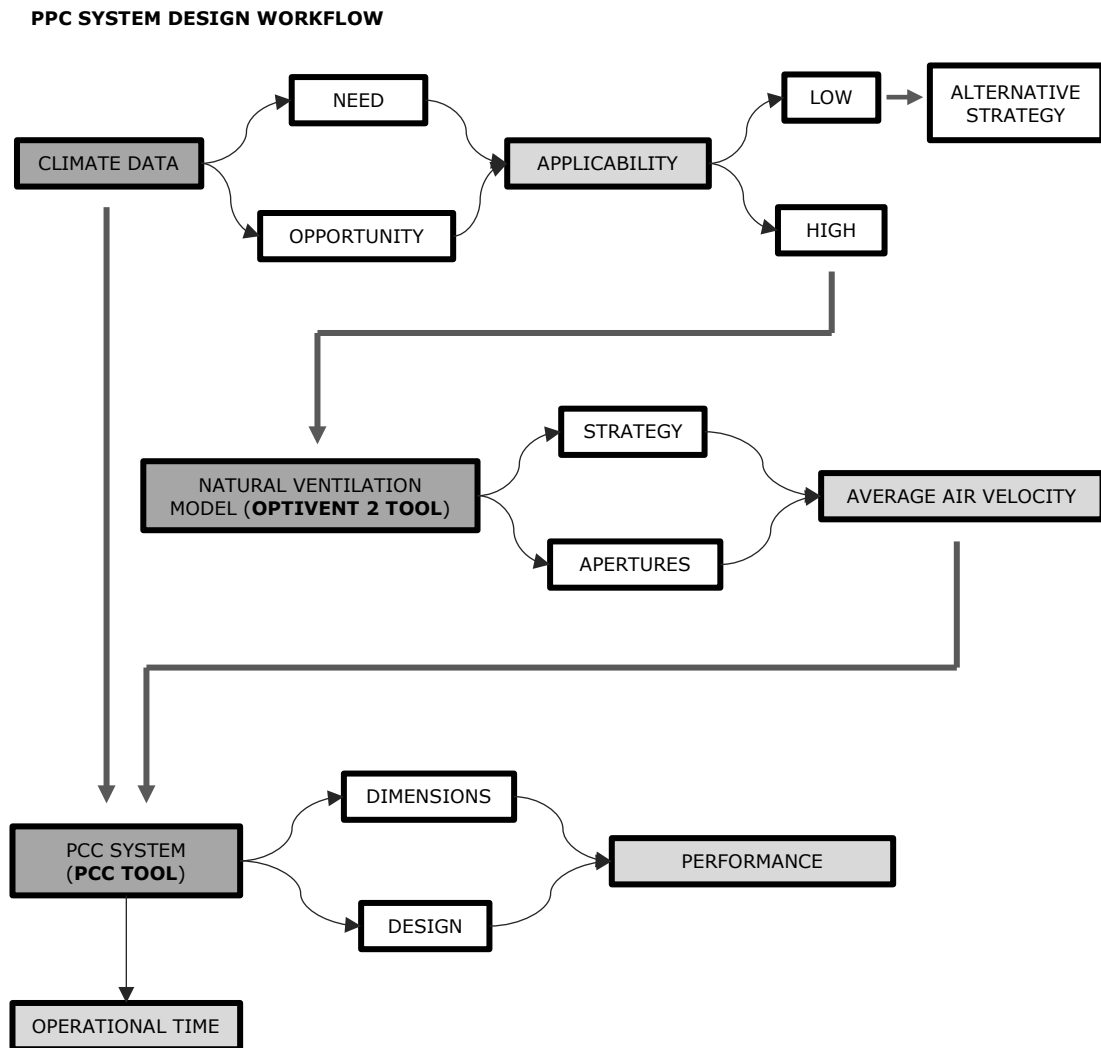


Figure 7.1 PCC Systems design process

By using an example, this chapter demonstrates the design process and the usefulness of the developed models to size PCC systems. The proposed example is described as follows: in the climatic context of Granada, previously introduced in Chapter 3.3.3, an indoor space coupled with an outdoor space by a natural ventilation strategy is given. The indoor space is a large volume part of a public building in which is intended to locate a PCC system to locally condition a cross sectional area of 8 m^2 . The space is ventilated using a cross-stack ventilation strategy in which three apertures of total area 5 m^2 drive air in, and an 8 m long tower-stack exhausts the air at the top, intentionally designed to achieve sufficient flow rates to provide fresh air to the occupants irrespectively to the external wind conditions (Figure 7.2). It is required to size the PCC system, predict the achieved temperature reduction and more importantly, to estimate

water consumption during the daily operational hours, also to be suggested by the designer. This study must also interrogate the suitability/feasibility of the system.

To overcome this exercise, the previously introduced design workflow was applied, evaluating in first instance the appropriateness of the climate; second, the ventilation strategy and the average flow rates/air velocities achieved by using Optivent 2, and last, the size and performance of the system by using PCC tool. Further, a parametric study on the impact of the system's design and climate was conducted in order to point up the range in performance that can be achieved and the impact of some of the parameters involved in the evaporation process.

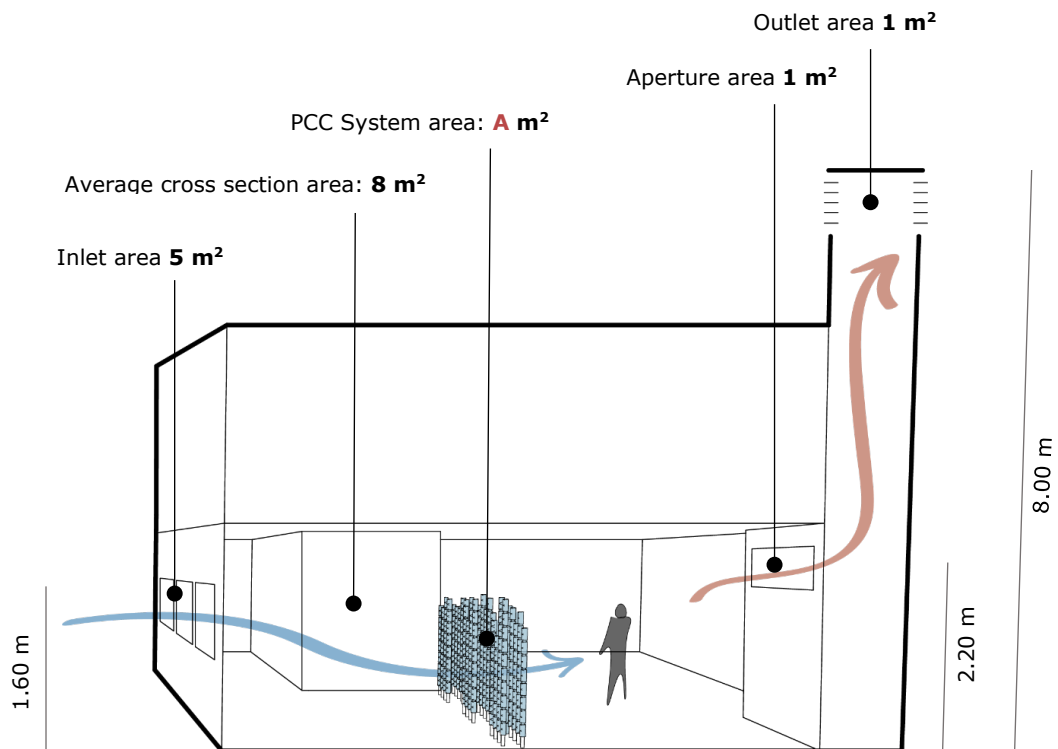


Figure 7.2 Space layout

7.1 Climate Suitability

Climate analysis is a crucial stage of the environmental design process to understand the applicability of any passive strategy and can be conducted by interrogating weather data. Most weather data available derive from public sources and can be accessed for free. They contain hourly data for most common environmental parameters (DBT, WBT, RH, v, etc.), representing a typical meteorological year. The climate data of Granada is provided by the Spanish Meteorological National Institute (AEMet, 2018) and obtained from EnergyPlus (2017b). In order to evaluate the applicability of evaporative cooling systems, DBT and RH are the two main parameters needed to conduct the study. In line with the applicability methods described in Chapter 1.2, two indices are determined. The first index evaluates the need of cooling in the climate and can be determined by identifying the number of hours when the external air temperature is warm enough to create thermal stress, that is to say, when $DBT > 26^{\circ}\text{C}$ as described by Givoni (1994). Additional thresholds could be also determined by using international comfort models, for instance, EN15251 (2007). The second index evaluates the opportunity of evaporative cooling systems, which can be determined from the wet-bulb depression DBT-WBT. Wet bulb temperature can be calculated from air temperature and relative humidity using Equation (3-15).

Back to the example, frequency studies were performed to determine the number of hours when $DBT > 26^{\circ}\text{C}$ and DBT-WBT. The first index was estimated for each month and averaged to obtain daily mean hours as shown in Figure 7.3. The graph illustrates the average number of hours per day when $DBT > 26^{\circ}\text{C}$ for each month of the year and provides with useful information regarding the need of a cooling system. The displayed scale responds the following criteria: $n < 2$ (low), $2 < n < 4$ (medium-low), $4 < n < 6$ (medium-high) and $n > 6$ (high). Overall, the results confirm that the need of cooling in Granada is high during the summer season (June to August), presenting air temperatures above 26°C for more than 8 hours per day. The use of cooling systems could also be extended to the months of May and September for about 2-6 hours per day.

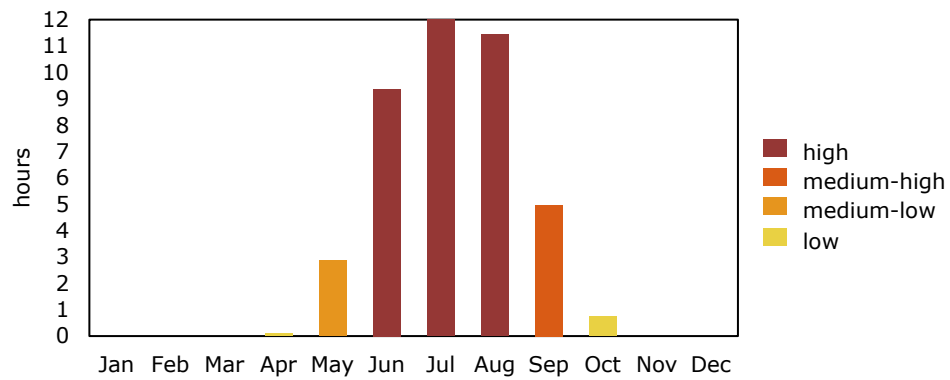


Figure 7.3 Cooling NEED, determined from the average number of hours per day when Dry Bulb Temperature is higher or equal than 26°C

The opportunity for using evaporative cooling systems was determined from the second index, (DBT-WBT), responding to the question: how dry is the climate? The index was estimated for each hour of the month and averaged in order to obtain the data illustrated in Figure 7.4. The displayed scale responds the following criteria: $i < 3$ (low), $3 < i < 6$ (medium-low), $6 < i < 9$ (medium-high) and $i > 9$ (high), in line with Chapter 1.2. The results suggest high applicability in the months of July and August, and medium-high in June, presenting monthly average values by about 9°C. This opportunity decreases in mid-season, with average values from 3 to 6°C, and remains low in the winter season, where ambient air conditions are closer to saturation.

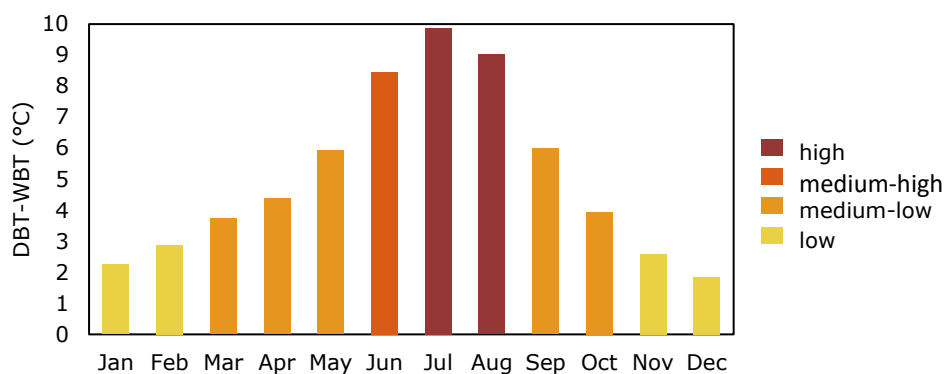


Figure 7.4 Cooling OPPORTUNITY, determined from the average Dry Bulb-to-Wet Bulb Temperature Depression

The above two indices confirmed the high applicability of evaporative cooling systems during the summer period in Granada due to the high estimated demand (9-12 hours per day) and opportunity (ambient air temperature could potentially be reduced by about 9°C). These figures strongly support the

application of a PCC system to locally reduce ambient air temperature in this context.

7.2 Natural Ventilation Strategy (Optivent 2)

PCC systems perform best when saturated air on the boundaries of the ceramics are constantly renovated with dryer air to allow further evaporation. This constant air renovation can be achieved passively using natural ventilation. The incoming air flow rates and velocities can be estimated using steady-state calculation methods as discussed in Chapter 6.2. Optivent 2 was designed to assist during the process, informing at the same time on the feasibility of a given ventilation strategy. Progressing with the exercise, a multi-zone + chimney scenario was used to determine these values (Figure 7.5). Despite the tool does not offer scenarios for every single design option, the existing ones can be applied to multiple variations in the design. In this exercise, the interest lays on the performance of the 3rd level, so that aperture areas and stack heights for the lower levels are set to a minimum allowed value of 0.01m² for the aperture areas and 0.01m for the stack heights. Aperture areas and stack heights for the 3rd floor were input according to the described space layout in the exercise. In addition, internal and external temperatures were defined for the prediction of buoyancy-driven ventilation flow rates. The indoor temperature refers to the average temperature of the room, and no air stratification is considered. The resulting cooling effect of the PCC system can also be neglected at this stage when determining the indoor temperature, assuming a relatively small area of influence compared to the total space volume (Chapter 4.4). A typical scenario with indoor air temperature of 2°C above the external air and 3°C below the chimney air was then considered at this stage. This hypothesis is based on the assumption that internal gains and solar gains (when available) contribute to keep indoor temperature warmer than external air, generating the airflow path illustrated in the diagram. This assumption is commonly used at early design stages and may represent a hypothetical worst-case scenario due to the low temperature differences considered between the spaces. Finally, internal and solar gains were not considered in the calculation as the final objective is to obtain an air flow rate and estimated air velocity driven by the defined air temperature differences, aperture areas and stack heights, and not to evaluate comfort conditions.

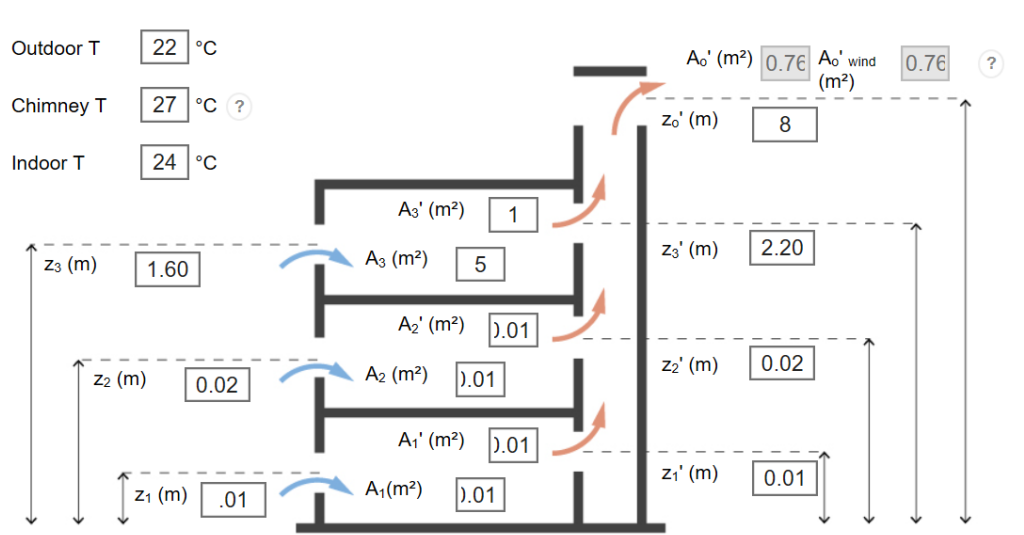


Figure 7.5 Optivent 2 model for the evaluation of buoyancy-driven airflow rates and average room air velocities

The results obtained and shown in Figure 7.6 indicated an airflow rate driving into the space of $0.91 \text{ m}^3/\text{s}$, which gives an air velocity of 0.12 m/s if considering the average cross-sectional area of the space layout (Figure 7.2). As the air movement is driven by buoyancy alone for a hypothetical worst-case scenario, it can be assumed that this value is the minimum air velocity of an airstream constantly present in the space. This air velocity is later used to estimate the evaporative cooling capacity of the PCC System.

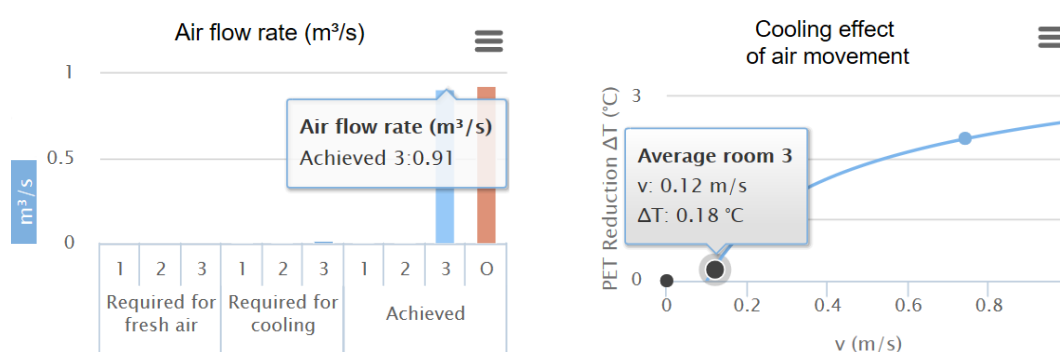


Figure 7.6 Optivent 2 results: buoyancy-driven air flow rates achieved and average air velocity in the room

7.3 PCC Sizing (PCC Tool)

The evaporative cooling performance of PCC systems depends on a larger number of factors when compared to the natural ventilation study. These include, apart from the environmental variables (DBT, RH and v), the geometrical characteristics of the ceramic elements, the screen's blockage ratio and the airflow regime. The developed PPC tool is used to take into account these parameters and respond to the initial request of the exercise. First, the environmental conditions of the model are given by defining ambient air temperature and absolute humidity for a typical operational day. The information can be obtained from climate data, and the mean maximum/minimum air temperatures and average absolute humidity registered in the month of July are given to the tool (mean DBT: 18°C at 6am and 34°C at 4pm; mean absolute humidity: 8g/kg). With the mentioned inputs Figure 7.7 is obtained. The graph illustrates the daily temperature fluctuation, RH and wet bulb depression of a typical day in July in Granada. The diagram facilitates the definition of the operational hours, which must rely on the need and opportunity of the system. Taking into account the warmer hours of the day and higher wet bulb temperature depressions, the operational period was defined from 1pm to 8pm, hours in which $DBT > 30^{\circ}\text{C}$ and a $DBT - WBT > 10^{\circ}\text{C}$. This decision may also rely on additional factors such a particular comfort expectation or water availability. In addition, the previously calculated air velocity is input in the PCC Tool to define a constant air velocity upstream the PCC system.

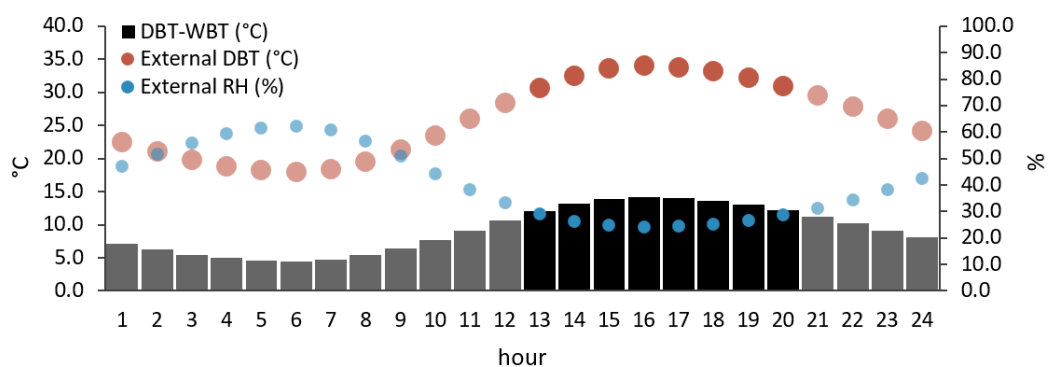


Figure 7.7 PCC Tool: operational time and typical daily external air temperature and relative humidity fluctuation in July, Granada

Next, the geometrical characteristics of the ceramic wall are defined. The screen dimensions are 3x2m as described in the exercise, and several ceramic shapes

(drag coefficient) and number of columns can be parametrically tested for a better understanding of the system performance. For this exercise, tubular-shaped ceramics ($c_d=0.56$) presenting a blockage ratio of 50% were selected. A user distance of 0.40m downstream the ceramic screen was selected to visualise the results. The results could also be tested for different user distances (Figure 7.8).

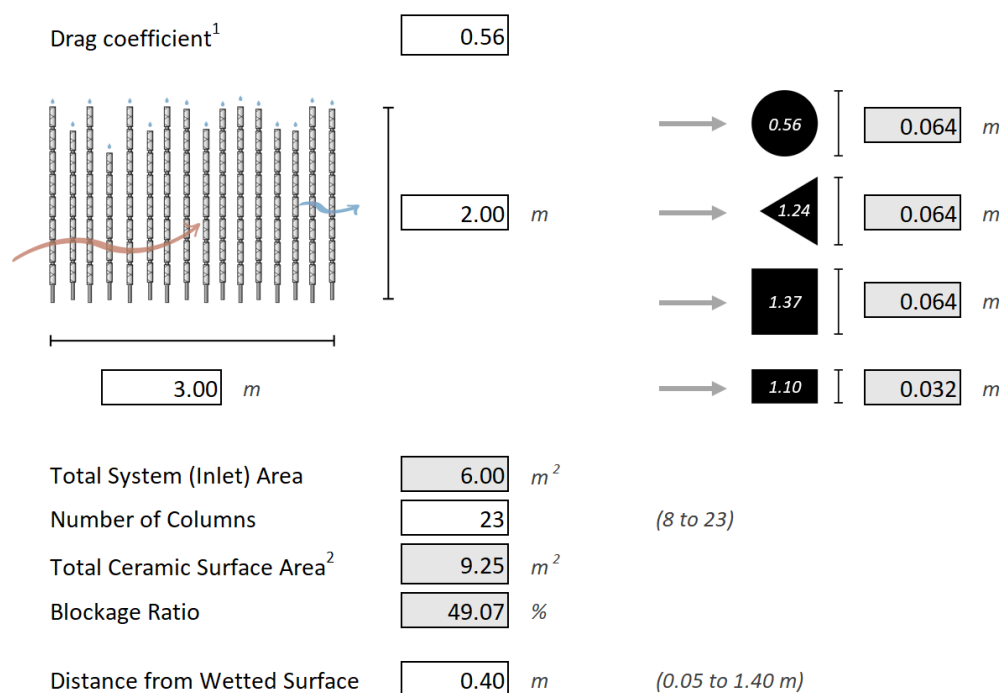
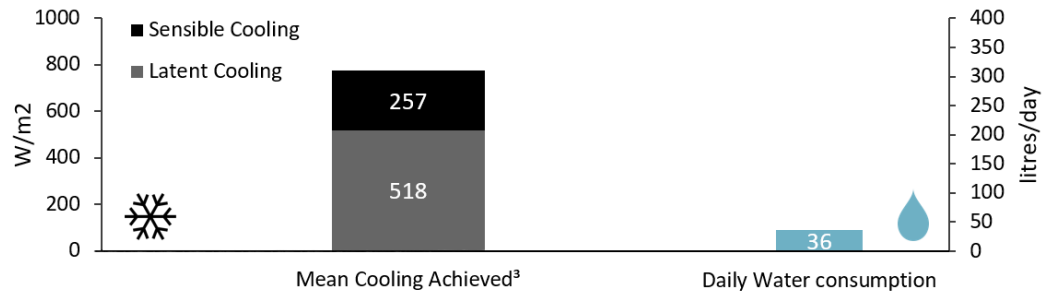


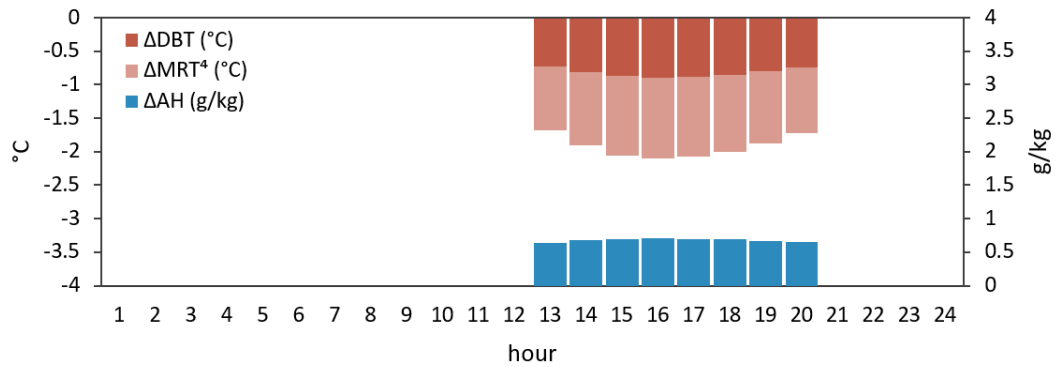
Figure 7.8 PCC Tool: user inputs in ceramic system design data

The first set of outputs obtained after defining the context and geometry indicated the provided sensible and latent cooling of the system in Watts per unit area of ceramic screen. The estimated consumption taking into account the hours of operation is 36 litres per day (Figure 7.9). The predicted air temperature reduction ranges from 0.8 to 1°C, and the additional radiant cooling provided ranges from 1 to 1.4°C for the given user distance. In addition to the cooling provided, the average increase of air moisture content is estimated to 0.6 g/kg. These values can be interrogated for each hour of operation from Figure 7.10.



³ Mean Sensible + Latent cooling per unit area of ceramic system.

Figure 7.9 PCC Tool: cooling performance and daily water consumption of the ceramic system in July, Granada



⁴ Mean Radiant Temperature (MRT) estimated for a standing person from Ceramic Surface Temperatures and Angle View Factors at the given downstream distance from the ceramic system (EN ISO 7726:2001).

Figure 7.10PCC Tool: air and mean radiant temperature reduction + absolute humidity increase for each hour of operation of the ceramic system for a typical day in July, Granada

7.3.1 PCC System Performing Range

The results obtained above were determined by multiple parameters involved in the evaporation process. This section explored the impact of some of these parameters associated to the system's design and climate on the cooling performance achieved. Using the previously introduced context, alternative design options varying the blockage ratio (number of ceramic columns) and the geometrical shape (drag coefficient) were tested to identify the best and worst performing option. Firstly, blockage ratios of 50, 40, 25 and 15% (equivalent to 23, 19, 12 and 8 vertical columns) were applied using the tubular-shaped ceramics. Secondly, rectangular, triangular and squared-shaped ceramics were tested with the initial 50% blockage ratio used in the conducted example. Lastly, the performance of the tubular-shaped ceramics was again interrogated for two

additional geographical locations presenting different levels of air humidity and temperature. The selected cities, Granada (Spain), Bologna (Italy) and London (UK), broadly represent the range of latitudes and climatic zones existing in Europe, giving a general understanding of the applicability and performance of PCC systems in Europe.

The results from the first parametric study, which altered the blockage ratios (BR) from the previous example, indicated a linear relation between BR and the air, mean radiant temperature depression and water consumption. As it can be expected, higher blockage ratios (or, what is the same, higher number of columns) yielded to higher water consumptions and temperature reductions (Figure 7.11). From the illustration bellow it can also be observed that the radiant component is a more determining factor when increased the number of columns, suggesting that, a 50% BR is not only the best performing, but the most efficient design option. However, it will be the role of the designer to determine an upper limit for this parameter, possibly indicated by the water consumption of the system, aesthetics, or an airflow obstruction due to a large number or a wrong arrangement of the ceramic columns.

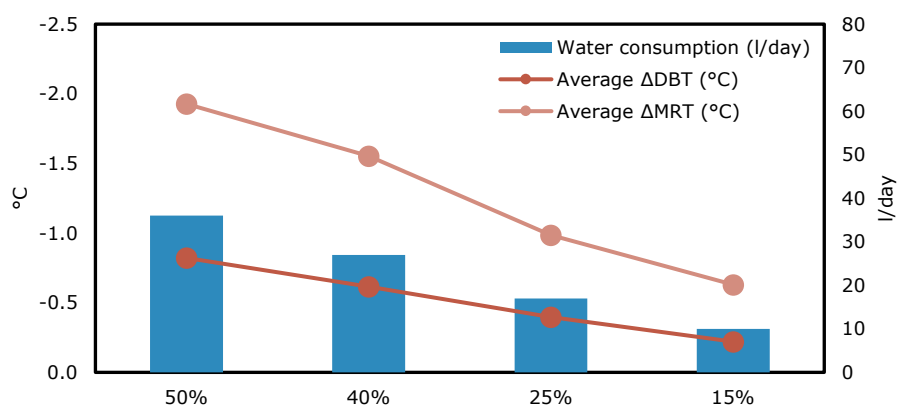


Figure 7.11 Impact of blockage ratio on PCC system performance

The second parametric study explored the performance of alternative ceramic shapes under the same context of Granada. The results illustrated in Figure 7.12 reaffirmed once again the impact of ESA and Cd on the achieved cooling capacity. Given the same blockage ratio, water consumption ranged from 25 up to double (45 litres per day), whereas the achieved air temperature reduction ranged from -0.4 to -1.0 $^{\circ}C$, being the triangular and squared shapes the worst performing, and the rectangular and circular shapes the best ones. It is worth mentioning that, despite both presenting a similar air and mean temperature

reduction, it is the circular shape (presenting lower resistance to flow) the one consuming less water and therefore, the most efficient. Contrarily, the squared shape presented the worst performance (lowest temperature reduction and highest water consumption) due to the lowest ESA/Cd associated to this shape.

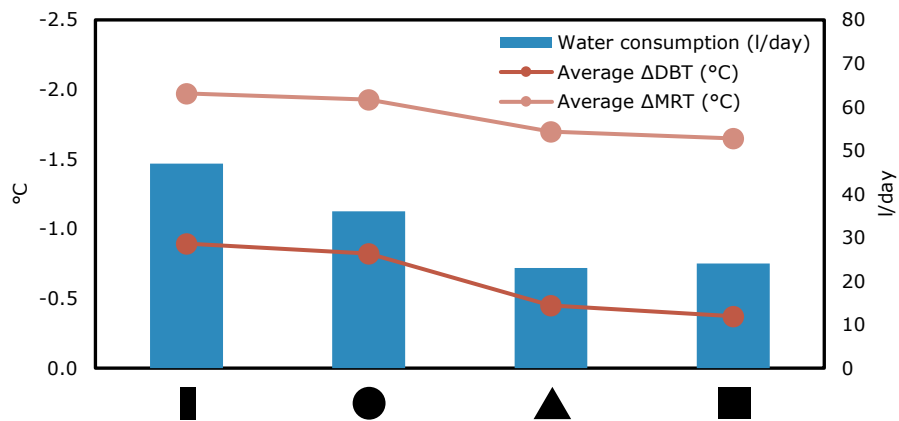


Figure 7.12 Impact of geometrical shape on PCC system performance

The third study explored the impact of climate on the performance of PCC systems. Focusing in Europe, three cities presenting different climatic characteristics were chosen for the study. Granada, Bologna and London broadly represent the range of latitudes and climatic zones existing in Europe, giving a general understanding of the applicability and performance of PCC systems in the continent. Granada presents optimal conditions for the application of evaporative cooling systems, presenting a summer season classified as hot and dry according to the Köppen Climate Classification (Kottek et al., 2006). Bologna is considered as a humid subtropical climate, with relatively hot summers under the influence of moist and humid maritime airflows. On the contrary, the climate of London presents mid-warm summers with an intermediate amount of air moisture. The climatic differences between these cities are more evident when air temperature and humidity are represented over the psychrometric chart. By looking at Figure 7.13, it can be anticipated that Granada presents the warmest and driest summer, Bologna presents the most humid, and London the coldest of the three.

The study repeated the calculation process using PCC Tool, modifying the climatic conditions of a typical hot day in July for each of the city as shown in Table 7.1.

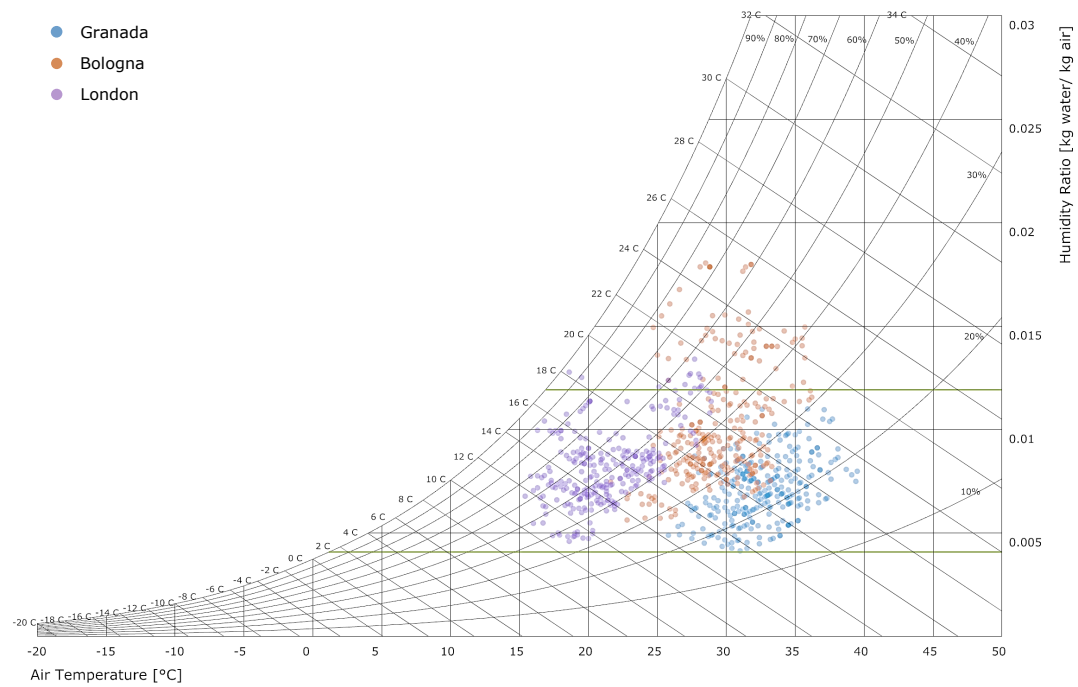


Figure 7.13 Air temperature and humidity in July (noon to 8pm) of Granada, Bologna and London plotted in the psychrometric chart

Table 7.1 Averaged day temperatures and humidity In July of Granada, Bologna and London

	Latitude (°)	Max Day Temp (°C)	Min Day Temp (°C)	Average AH (g/kg)
Granada	37.18	34	18	8
Bologna	44.49	30	20	11
London	51.50	22	15	9

Given the same PCC system, the average performance is considerably different in each of the climates. While the daily water consumption presents small differences between the three, the variance in air temperature depression and, specially, mean radiant temperature drop is more evident (Figure 7.14). As expected, Granada presented the best results (-1°C ΔDBT and -2°C ΔMRT) while London presented the worst performance. It is interesting to see how the higher moisture content in Bologna (despite having a similar ambient air temperature to Granada) entailed a slightly lower air temperature drop and up to -1°C drop less in the ceramic surface temperature. On the other hand, air and mean radiant

temperature drops in London remained below 0.5°C. Despite presenting lower air moisture content than Bologna, the city of London presented a considerably lower ambient air temperature (-8°C less) when compared to the Italian city, subsequently achieving the lowest DBT-WBT between the three climates.

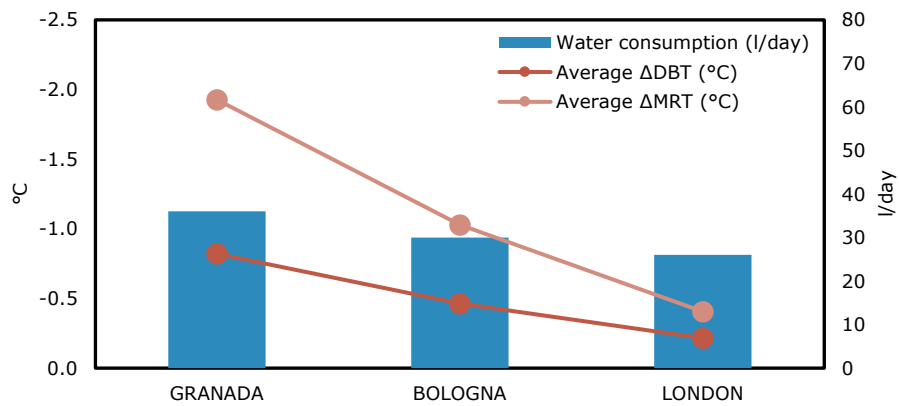


Figure 7.14 Impact of climate on PCC system performance

This comparison highlighted the important role of the climate on the performance of PCC systems. Both low moisture content and high ambient air temperatures are equally needed to maximise the efficiency of the system. It is also important to highlight that by only analysing one typical (averaged) day in July, this study did not aim to suggest that PCC systems are not applicable in Bologna, London and other similar climates, but to point up the need to identify and quantify the number of dryer and warmer days throughout the summer season when the system may be applicable and will perform best.

7.4 Conclusions

This chapter introduced a design workflow for the integration of steady-state tools in the design process of PCC systems. The workflow was successfully applied to a proposed example and demonstrated its adaptability to any context and space layout. Each of the steps in this process informs the following one, and alternatively, they can be used independently to meet additional design objectives.

The use of Optivent 2 during the design process provided with, apart from the required airflow rates and air velocities, additional information regarding the suitability of the proposed ventilation strategy. With this exercise it was also demonstrated the applicability of Optivent 2 and PCC Tool to any space layout

and context, including indoor and outdoor spaces without the need of complex CFD simulations.

The results concluded that, a 6m² screen wall made of 23 ceramic columns can reduce the user temperature perception in the surroundings by about 1.2°C from 1pm to 8pm, hours presenting the highest applicability in a typical day in July, Granada. During this time, the estimated water consumption was 36 litres, a significant small amount equivalent to two toilet flushes. In addition, the predicted airflow rate of 0.93 m³/s driven by a cross-stack ventilation allowed the estimation of the sensible cooling load of system: 257W/m².

Further, a series of comparative studies informed about the multiple design options and their impact on the system performance. The results obtained reaffirmed the findings from previous chapters: higher blockage ratios, ESA/Cd and DBT-WBT yield to better performance. Within the range of design options tested, it was also observed that a good design solution can perform up to three times better than a less efficient option.

Through a design example, the workflow has proved to be quick and a valid method to assist in the design process of convective and evaporative cooling systems. Moreover, the small number of inputs and the limited time required made this workflow ideal for early design stages. It is hoped that this workflow will help to identify benefits and limitations of passive cooling systems for any context and contribute with knowledge dissemination among professionals in the building sector.

8 Discussion and Conclusions

This research addressed the application and design optimization process of convective and evaporative cooling systems in architecture, responding to the exponential increase of global cooling demand and the subsequent growing interest in passive cooling techniques to deliver healthy and comfortable environments. The developed system combines the principles of convective and evaporative cooling with the inherited use of porous ceramics in architecture by proposing an aesthetic wetted Porous Ceramic Cooling (PCC) wall to improve comfort conditions in indoor, transitional and outdoor spaces.

The benefits of convective and evaporative passive cooling systems are now widely appreciated and their use date back to ancient civilizations, who mastered these techniques through centuries and demonstrated the importance and strong potential of coupling both passive cooling strategies to maximise thermal delight. Further, the strong potential of convective and evaporative cooling techniques to improve air quality and minimise greenhouse gas emissions derives from the high applicability of these systems in countries with highest consumption of energy for cooling. The number of heat-stress hours, cooling degree hours and DBT-WBT are considered good indicators to evaluate the feasibility of these techniques, and recent studies concluded that more than 50% of the countries in Europe, most located in latitudes below 48°, present medium to high applicability of evaporative cooling systems, while about one third of China extension present optimal conditions for the application evaporative systems. In the US, 50% of the counties are eligible for the application of PEC methods and more than 70% of the counties could overcome overheating problems in buildings with a good natural ventilation strategy and without the need of mechanical systems. Moreover, the mapping technique developed and applied in the US, became a reliable source to allow architects with limited knowledge in this field to determine the feasibility of an evaporative cooling strategy or to evaluate the market opportunity of a novel evaporative cooling product.

In the last decades, evaporative cooling also became a highly demanded technique to improve outdoor comfort and mitigate urban heat island effect. The increase in urban population had direct consequences on urban heat island intensities, which aggravated outdoor heat stress and increased cooling demand in buildings. Among the most recurring techniques by environmental designers to mitigate UHI and improve outdoor comfort in hot/dry environments is the use

of evaporation by increasing the presence of water bodies. Misting systems are now a popular technique to improve outdoor comfort in public spaces and terraces (Lozano, 2012), and studies have confirmed that water surface temperature can drop by 16°C below the external air temperature under hot and dry conditions (Guerra Macho et al., 1994). This surface temperature reduction also applies to porous surfaces and, the potential of replacing impervious materials with wetted porous surfaces to reduce UHI effect is significant considering the extension of the urban areas, especially in high density urban layouts.

For this and other reasons ceramics are considered excellent media to induce evaporation. The internal connection of the pores allows water absorption and flow towards the ceramic surface creating a thin water film that evaporates when exposed to the airstream. Design-wise, ceramics became again highly appreciated and new ceramic systems have emerged as a result of the collaboration between ceramic fabricators and architects. Moreover, recent ceramic innovations have rediscovered the two-and-three-dimensional aesthetics of ceramics and 3d printing technologies have demonstrated the innovative design opportunities to create through parametric modelling techniques all sort of ceramic systems optimised for evaporative cooling.

The development of a novel Porous Ceramic Cooling (PCC) system involved the exploration of the geometrical impact of the porous ceramic elements, the fabrication of a ceramic prototype, the evaluation of the hydraulic capacity, cooling performance of ceramic samples in laboratory-controlled environments and the fabrication of a PCC system mockup for continuous performance monitoring.

The first part of this research confirmed that porous ceramic cooling systems can be optimised through design and are suitable for large-scale production, demonstrated with the manufacture of the tubular-shaped Prototype P2 using a traditional extrusion method. Optimised ceramic designs must increase their surface area per unit of volume while minimise their resistance to flow. Cooling capacity studies confirmed the proportional and inversely proportional relationship of the ceramic effective surface area (ESA) and drag coefficient (Cd) respectively. Further studies evaluating the cooling capacity of different arrangements of prototype P2 confirmed a linear relation between DBT-WBT and the achieved temperature depression and humidity increase downstream the ceramics. The collected data confirmed that air temperature reductions of 2

degrees are only achievable under warmer and dryer conditions (DBT-WBT greater than 12°C). These results were compared against monitored data from an experimental mock-up in Granada, Spain (37.18N), in which temperature reductions up to 2°C downstream the ceramics were also observed under the hotter and dryer hours of the monitoring period. The data collected from fieldwork allowed to identify a wider amplitude of $\pm 0.3^\circ\text{C}$ in the achieved air temperature reduction when compared to laboratory results, mostly associated to uncontrolled air velocity fluctuations.

Moreover, the observed ceramic surface temperatures during the experimental studies ranged from 3 to 5°C above ambient wet bulb temperatures. Further studies conducted using an infrared thermal camera and a black-globe thermometer confirmed that warmer and dryer ambient temperatures lead to higher mean radiant temperature depressions. In other words, DBT-MRT downstream the ceramics is proportional to ambient DBT-WBT. The significant mean radiant temperature reduction obtained during the studies, points up the importance of surface temperatures as a determining factor in the evaluation of thermal comfort in the proximities of the system.

Studies on the hydraulic capacity of the developed ceramic system estimated water consumption for prototype P2, which ranged from 2-4 l/m²·day (considering 8 operational hours per day). This number is comparable to the average water consumption of one misting nozzle (4 l/day) commonly used in public outdoor spaces across Mediterranean countries, making PCC systems a competitive alternative requiring lower maintenance cost while improving the architectural quality of the space. An omnidirectional and even distribution of water over the manufactured Prototype 2 was achieved thanks to the parallel connection of its pores, a drip irrigation system placed at the top of the tubular sample, and engraved channels on its porous surface that transported water from top to bottom. The chosen irrigation system avoided the need of additional maintenance and operational cost and allowed a greater flexibility in the allocation of the ceramic screen. Further, the nature of the system allowed the ceramics to fully dry over non-operational hours, reducing the risk of *Legionella* growth. Conducted studies on the hydraulic capacity of several ceramic samples suggested that the main driving force of water absorptance in ceramics is the capillary network. It was observed that ceramic surfaces can absorb water in all directions without any visible impact of gravitational force, air and water temperature. Furthermore, water expansion on the ceramic surface of P2 can be expressed as function of time: $d = 2.7034 \cdot T^{0.618}$, where d is the expanded

distance of water in mm and t is the time of absorption in minutes (for extruded ceramic elements with average porosity of 15% and density of 1900 kg/m³), suggesting, for instance, that engraved spirals presenting angles below 40 degrees can wet more than 80% of the total surface area in less than 60 minutes. The above expression can be used for predicting the time and the flow rate required saturate the ceramic sample, easing the sizing process of the irrigation system.

Following to the design, manufacture and testing of a novel PCC system, the research focused in the development of efficient methods to size and allow advance performance evaluation of convective and evaporative cooling techniques. The developed tools and methods to assist in the design and performance prediction of PCC systems allowed each of the components constituting the system to be evaluated: airstream, evaporation media and water supply method. This contribution responded to the growing interest in performance evaluation among architects/designers and to need of more intuitive, graphic and well enough designed tools for a successful integration of passive cooling strategies during the design process.

A valid numerical model using the commercial CFD software Ansys Fluent® was defined to simulate the physics of evaporation on wetted ceramic surfaces. The model is a transient simulation with water phase change taking place on the ceramic surface, and consequently mass transfer from liquid to vapour state within a defined fluid domain. The described governing equations and boundary conditions in Chapter 4.2.1 have proved to be suitable for the simulation of the evaporation process in PCC systems and results were validated against laboratory data. The model can provide with detailed information regarding the performance of any ceramic shape, bringing the opportunity to evaluate prototypes prior to manufacture and making this model appealing and supportive in the design process.

Support on early design stages was also provided with the development of PCC Tool and Optivent 2, two intuitive interfaces that allowed the sizing of PCC systems and the evaluation of natural ventilation options.

PCC Tool is a multi-steady-state model developed to assist designers in the sizing and performance prediction of PCC systems in early design stages with minimum user inputs and time consumption. By using the tool, designers can estimate temperature reduction, humidity increase and water consumption of a PCC system exposed to any environment and for any ceramic shape and

arrangement within the range of parameters previously analysed during the computational work. The tool outcomes were validated, and both predicted temperature reduction and water consumption were in line with laboratory data. PCC Tool can be accessed through the following link:

<http://naturalcooling.co.uk/PCCT.xlsx>

Optivent 2 integrated the principles of natural ventilation in an intuitive interface, informing on the feasibility of a given ventilation strategy. This steady-state tool allowed the evaluation of single and multi-zone models, including single and cross ventilation, atria and large stacks (chimneys), extending its application to a wide range of contexts. The tool demonstrated that multiple design options can be tested in a matter of minutes with a reduced number of inputs and can be accessible through the following link: <http://naturalcooling.co.uk/optivent.html>.

Since 2016, the redeveloped Optivent 2 has been continuously introduced and well received by postgraduate students from several MSc courses in the UK and future objectives target the promotion of Optivent 2 in architecture and engineering practices. The reliability of the tool was proved through its application in multiple buildings throughout the past ten years and was recently validated against field data collected for the commissioning of the natural ventilation system of the Nottingham Girls High School Theatre, project that was followed closely from beginning to completion. The resulting data and the actual natural ventilation performance of the theatre allowed the gap in performance between the predicted and actual flow rates to be evaluated, and the reliability of the natural ventilation design tool Optivent 2 to be determined. The obtained data indicated an average performance gap of 15%, results that were considered as satisfactory assuming the nature of the tool, reaffirming the value of integrating environmental design approaches and the value of early stage design tools.

A final design workflow integrating the numerical models developed in this research was proposed to ease the design process of convective and evaporative cooling systems in design. The design workflow included a climate analysis to understand the applicability of the system, an evaluation of the ventilation strategy that drives air into the ceramic wall, and the performance evaluation of PCC systems for a given context. Through a design example, the workflow was proved to be quick and valid for most scenarios, including indoor and outdoor spaces for any location in the world. Comparative studies also informed on the performing range of PCC systems, concluding that a good design solution can perform up to three times better than a less efficient option. Overall, the small

number of inputs and the limited time required made this workflow ideal for early design stages.

The developed system proved to be a good alternative to minimise local thermal discomfort by reducing ambient air and mean radiant temperature downstream the ceramic system, while the developed workflow proved to be quick and a valid method to assist in the design process of convective and evaporative cooling systems.

It is hoped that this work will contribute to an increase in the number of professionals in the building sector with expertise in designing for convective and evaporative cooling, and that this will lead to wider application of passive cooling solutions in architecture. It is also hoped that the developed tools and design workflow will contribute to a knowledge dissemination and a reduction in the carbon emissions and maintenance costs associated with mechanical ventilation and air conditioning.

Finally, it is intended to introduce the developed PCC system into market for a commercial use. The manufacturing limitations overcome with P2 and the interest shown by ceramic product managers in Spain and Portugal open up an opportunity for mass production in the near future.

8.1 Limitations of This Work

Despite the proved usefulness of the proposed workflow and design tools to promote the use of convective and evaporative systems in architecture, the applicability of PCC systems in indoor, transitional or outdoor spaces can be compromised by a series of limitations. The main limitation is related to potential high expectations on these systems. PCC systems should not be understood as a technique to condition medium to large spaces. The minor temperature reduction achieved downstream the ceramics and, more importantly, the limited area of influence observed during the conducted laboratory work and fieldwork, confirmed that its contribution to the reduction of cooling loads in spaces is not significant. In other words, PCC systems are suitable to mitigate the surrounding environment and not to condition medium-to-large spaces. It is, however, a good alternative to improve thermal comfort conditions of users located in the proximities of the system, or an efficient method to mitigate urban heat island effect when large ceramic surface areas are saturated in the urban context under hot and dry ambient conditions.

Other limitations found in this research are further described below:

1. **The developed PCC system is limited to vertical screen (wall) configurations.** The vertical ceramic arrangement explored in this research derives from the water supply method chosen for the system. The verticality of the system allows water supplied at the top to be distributed to bottom thanks to the capillary network and the gravitational force. This supply method allows the system to be a fully passive cooling strategy, considerably reducing operational cost. However, the restricted position of the ceramics limits the design options and possibly the integration of the system in complex space layouts.
2. **PCC systems present a small area of influence.** Monitored data from the optimised prototype 2 under different laboratory conditions confirmed that temperature reductions of 2°C are only achievable under warmer and dryer conditions (DBT-WBT greater than 12°C). In addition, fieldwork data and computational CFD studies confirmed the limited area of influence of PCC systems: temperature depression beyond 2 meters' distance downstream the ceramics rarely exceeds 1°C under optimal environmental conditions. This reaffirms the localised cooling effect of the system, which must be placed on areas accessible to users to maximise benefit.
3. **Ceramic 3d printing in architecture presents technical limitations and is still under development.** An optimised ceramic prototype (P1) was first designed presenting a high *effective surface area to volume ratio* ($ESA:V = 234 \text{ m}^{-1}$) and a low *drag coefficient* ($C_d=0.52$) due to its aerofoil shape (Chapter 3.1.1). The excellent predicted performance of the prototype (Chapter 4.4.2) contrasted with a series of technical and economical limitations found during the fabrication process, which prevented its fabrication. The high amount of water content needed in the clay for 3d printing required, for this and most cases, adjustable moulds for drying that increased the risk of surface dimples and break after shrinkage. This experience reaffirmed the fact that, despite the vast design opportunities of digital fabrication techniques, there is a long way to compete the cost-effectiveness of traditional ceramic production.
4. **PCC Tool is limited to the range of environmental scenarios previously analysed in the computational CFD model.** PCC Tool is

not a solving program, and it could be better described as an optimised database management system that collects data from the parametric database generated from the computational model, and, through an interpolation process, obtains the final outcomes. The parametric study consists of sixteen environmental conditions: four DBT-WBT (8,10,12, and 14°C) under four air velocities (0.25, 0.50, 0.75 and 1 m/s). Four ceramic shapes presenting different flow resistance (drag coefficient: 0.56, 1.10, 1.24, and 1.37) were analysed for four different arrangements (blockage ratio: 25,30,40 and 50%). Despite covering most frequent range of environmental conditions and allowing a considerable amount of design options, the tool does not allow performance evaluation of alternative geometries, ceramic sizes and ceramic arrangements.

8.2 Recommendations for Further Work

The evolution and limitations found in this research determined further areas to explore in the field of design and integration of convective and evaporative cooling systems in architecture. The identified areas for further work are described below:

1. **Investigation on optimised ceramic clay properties to maximise evaporation.** Including and exploration of different clay compositions, additives and the definition of optimised shaping and firing configurations for the optimisation of the ceramic elements, targeting enhanced capillary network, density, durability, etc. This opens up new design opportunities of PCC systems, including alternative water supply methods and innovative ceramic arrangements.
2. **Further fieldwork studies of PCC systems under different space layouts,** including indoor, outdoor and transitional spaces. This data will potentially identify variations in the area of influence of the system related to the space layout and context, and expectantly will reaffirm the efficacy of PCC systems to improve local thermal comfort and the reliability of steady-state design tools for the integration of convective and evaporative cooling strategies.

3. **More capabilities of PCC Tool**, including an extended parametric database to include a wider range of environmental conditions and air velocities to be tested, new geometrical shapes and ceramic arrangements.
4. **Validation of the proposed PCC system design workflow.**
Despite PCC Tool and Optivent 2 were successfully validated against fieldwork and laboratory work, the proposed design workflow that integrates the developed numerical models has not been validated. Fieldwork data from an application example will validate and reaffirm the reliability of the proposed design workflow for the successful integration of convective and evaporative cooling strategies in design.
5. **Dissemination activities.** It is hoped that this research will contribute to an increase in the number of professionals with expertise in the field of convective and evaporative cooling, and that this will lead to a wider number of passive cooling solutions integrated in future building projects. The process of dissemination has already started through a series of workshops delivered in different postgraduate courses in environmental design in the UK and Italy. However, the next dissemination stage must target professionals in the building sector through participation in business conferences and the offering of training workshops.

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Appendix A1. Equipment Used in Laboratory and Fieldwork Studies

The following equipment was used to set-up the test rig and undertake the laboratory studies in Sustainable Research Building in the Department of Architecture and Built Environment in the University of Nottingham, and the fieldwork studies in Granada, Spain.

Measuring the porosity

- Induction oven (x1). 230V/50-60Hz,1000W.
- Digital scale (x1). Accuracy: 0.1g.
- Tea urn (x1). Dimensions: 320x320x520mm. Volume capacity: 20L. Material: stainless steel. 220-240V, 50Hz, 2200W.
- Stainless steel trivet (x2). Dimensions: 500x280x20mm. Material: stainless steel.
- Metallised polyester laminated to polyethylene foam 2mm.
- Heat resistant gloves for contact heat up to 250°C.

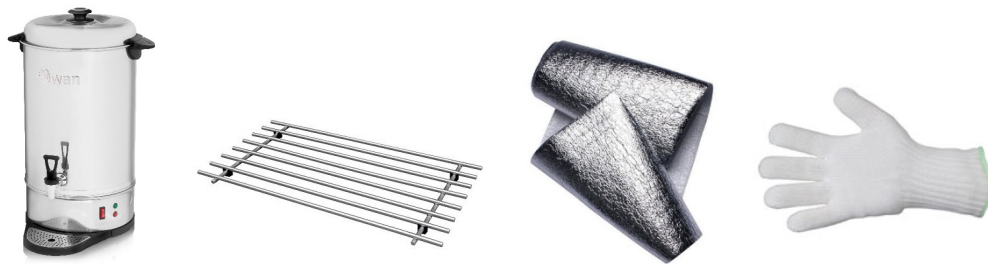


Figure 8.1. Part of the equipment used to measure ceramic porosity in the Research Laboratory

Defining the hydraulic potential

- Heavy duty shelving (x1). Dimensions: 915x360x1520 mm. Maximum weight: 55 kg per shelf. Material: chrome.
- Water tank + Tap (x1). Dimensions: 273x150x150 mm. Volume capacity: 5L. Material: plastic. Outlet: ½ inch BSP female.

- Plastic tray (x2). Dimensions: 77x520x434 mm. Material: plastic.
- Stainless steel trivet (x2). Dimensions: 500x280x20mm. Material: stainless steel.
- Irrigation tubes. Dimensions: 4Ø mm and 13Ø mm. Material: plastic.
- Drippers (x4). Material: plastic.
- Digital moisture meter (x1). Measurement range: 0-50 %. Accuracy: $\pm 1\%$. Model: Testo 606-1.
- Thermal camera (x1). Object temperature range: -20°C to $+250^{\circ}\text{C}$. Model: Flir E4.



Figure 8.2. Part of the equipment used to evaluate the ceramics' hydraulic capacity in the Research Laboratory

Measuring the cooling capacity

- Environmental chamber (x1). Calibrated on the 10th of May 2016.
- RTD temperature sensors (x12). Accuracy: $\pm 0.3^{\circ}$ at 0°C . Model: Omega's RTD Class B Pt100.
- SDI12 RH +DBT sensors (x2). Accuracy $\pm 2\%$ RH and $\pm 0.3^{\circ}\text{C}$ at 25°C . Model: Campbell Scientific CS215-L.
- Thermal anemometer (x1). Measuring range: 0 to 20m/s. Accuracy: $\pm 0.03\text{m/s}$. Model: Testo 425.
- Thermal camera (x1). Object temperature range: -20°C to $+250^{\circ}\text{C}$. Model: Flir E4.

- Data logger with 15 analog inputs +12 digital channels (x1). Model: DataTaker DT85.
- Plastic tray (x1). Dimensions: 77x520x434 mm. Material: plastic.
- Stainless steel trivet (x1). Dimensions: 500x280x20mm. Material: stainless steel.
- Portable fan - 8 inch (x1). 220-240V, 50Hz, 35W. Polypropylene blades. Three speed settings.
- Insulation board (x2). Dimensions: 1200x700x50 mm, 1200x600x50 mm, 600x700x50 mm, 600x600x50 mm. Material: polyisocyanurate. Thermal conductivity: 0.022 W/mK.



Figure 8.3. Part of the equipment used to evaluate the ceramics' cooling capacity in the Research Laboratory

Fieldwork

- Lever ball tap 1/2" (x1).
- Plastic irrigation tubes. Dimensions: 13Ø and 4Ø mm.
- 8x micro-drippers (x10). Material: plastic.
- Porous ceramic prototype P2 (x80). Dimensions: Ø64 and 23 mm high.
- RTD temperature sensors (x12). Accuracy: $\pm 0.3^\circ$ at 0°C . Model: Omega's RTD Class B Pt100.
- SDI12 RH +DBT sensors (x2). Accuracy $\pm 2\%$ RH and $\pm 0.3^\circ\text{C}$ at 25°C . Model: Campbell Scientific CS215-L.
- Thermal anemometer (x1). Measuring range: 0 to 20m/s. Accuracy: $\pm 0.03\text{m/s}$. Model: Testo 425.
- Data logger with 15 analog inputs +12 digital channels (x1). Model: DataTaker DT85.
- Portable fan - 8 inch (x1). 220-240V, 50Hz, 35W. Polypropylene blades. Three speed settings.
- Rotatory tool 8V (x1). Model: Dremel 8050-N/18 Micro 8V.



Figure 8.4. Part of the equipment used to undertake fieldwork studies in Granada, Spain.

Appendix A2. Access to Optivent 2 and PCC Tool

Optivent 2 was developed in 2015 as a tool to assist the decision-making process at early design stage, regarding the feasibility of natural ventilation options. As a web-based tool, it is accessible from any portable device and was designed to support the latest versions of web browsers. The tool may not be fully functional in older versions of IE. The latest version of Optivent (v2.1) was recently released in Fourth Quarter 2017, including several improvements, bug fixes, and a new status of free-licensed product.

Optivent 2.1 can be accessed from:

<http://naturalcooling.co.uk/optivent.html>

Acknowledgements to Dr. Pablo Aparicio (Dipl. Eng, MSc, PhD), who wrote Optivent 2 in HTML.

PCC Tool was developed in 2017 as a tool to allow performance prediction of PCC systems exposed to a wide range of ambient conditions at early design stages. The tool was compiled in a spreadsheet using Microsoft Excel 2016 and is compatible with older Excel versions and mobile devices using Microsoft Excel App. The tool may not be fully functional in another spreadsheet software. PCC Tool is also a free-licensed product.

PCC Tool can be accessed from:

<http://naturalcooling.co.uk/PCCT.xlsx>

Appendix A3. Optivent 2 Validation: The NGHS Theatre

The design of naturally ventilated theatres had become commonplace by the 19th century (Reid, 1844). With the development of mechanical and electrical services in the late 19th century and early 20th century, theatres began to be mechanically ventilated and air conditioned. This became the norm for much of the 20th century, and the refurbishment and conversion from mechanical to natural ventilation has been the focus of recent work (Short and Cook, 2005). Recently a number of new naturally ventilated theatres have been built in the UK including the Hull Truck Theatre, and the Everyman Theatre, Liverpool, but the literature on design guidance is very limited.

The Nottingham Girls High School Theatre in Nottingham UK, is another recent example of embedding natural ventilation strategies at early design stages while simultaneously meeting complex acoustic, structural and environmental requirements. This design approach had a significant influence on the geometry and final layout of the building. Designed by Marsh Grochowski Architects, the building occupies a corner site with a road and housing to the east, an arboretum to the south, connections to school buildings to the west and an access road to the north (Figure 8.5). Both the architect and the client were keen to adopt natural ventilation in auditorium and stage area for both environmental benefits and life-cycle cost savings.

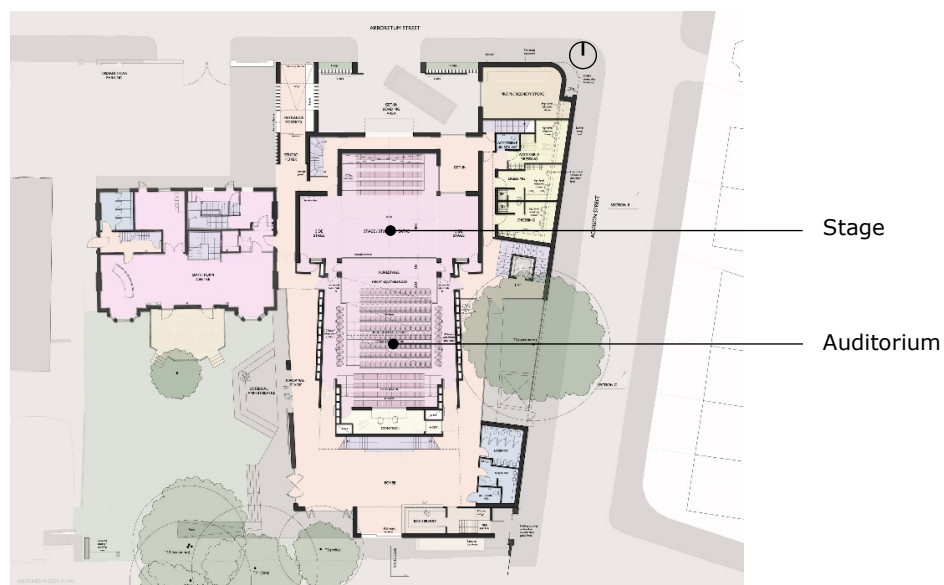


Figure 8.5 NGHS Site & Ground Floor plan. Source: Marsh Grochowski Architects

The brief required a 334 seat multi-purpose auditorium, foyer, dressing rooms, back of house areas and music rehearsal/theory rooms. The main auditorium was also to be divisible into a *studio* theatre space with bleacher seating for 64 people and a main 'hall' for 270 people (Figure 8.6). Further, the main hall was to be capable of having both flat floor and tiered seating arrangements.



Figure 8.6 NGHS Theatre west main entrance, north and east facades, auditorium and foyer
(Grochowski, 2015, p. 1)

Among the natural ventilation requirements, a minimum fresh air supply of 8-10 litres per person per second was to be available at all times that the different spaces were occupied. Also, a much larger air volume flow rate would be required to remove the internal heat gains from performers, audience, lighting and equipment. Moreover, potential acoustic issues required the risk of disturbance from external traffic noise had to be addressed. Also, since both classical orchestral music and amplified music would take place within the theatre at different times, efforts would need to be made to minimise the risk of disturbance to neighbouring residences at these times.

A successful ventilation strategy would therefore have to be capable of removing excess heat at the warmest time of the year while simultaneously providing

sufficient acoustic attenuation to avoid disturbance to neighbouring residences and avoiding noise from vehicles on the adjacent roads penetrating the auditorium. This flow of air would have to be achieved without the assistance of the wind, driven by buoyancy alone.

Discussions with the architect led to the definition of a strategy to introduce air primarily from the clean and quiet area on the south and west side adjacent to the arboretum, below the floor of the auditorium via a large plenum which would also provide space for moveable wagons and floor equipment. A series of supply points strategically located underneath the seats, on the side walls, front and back of the space were proposed to achieve an even distribution of the air to all parts of the auditorium, and to the upper seating levels, to avoid the risk of creating stagnant air pockets. In the stage, the air was introduced from a north opening at ground level and evenly distributed through the bleachers seating located at the back of the stage. Two main exhausts at high level above the auditorium and stage were designed for the air flow to be promoted due to buoyancy, but also to function successfully when the wind was blowing, irrespective of the direction of the wind (Figure 8.7).

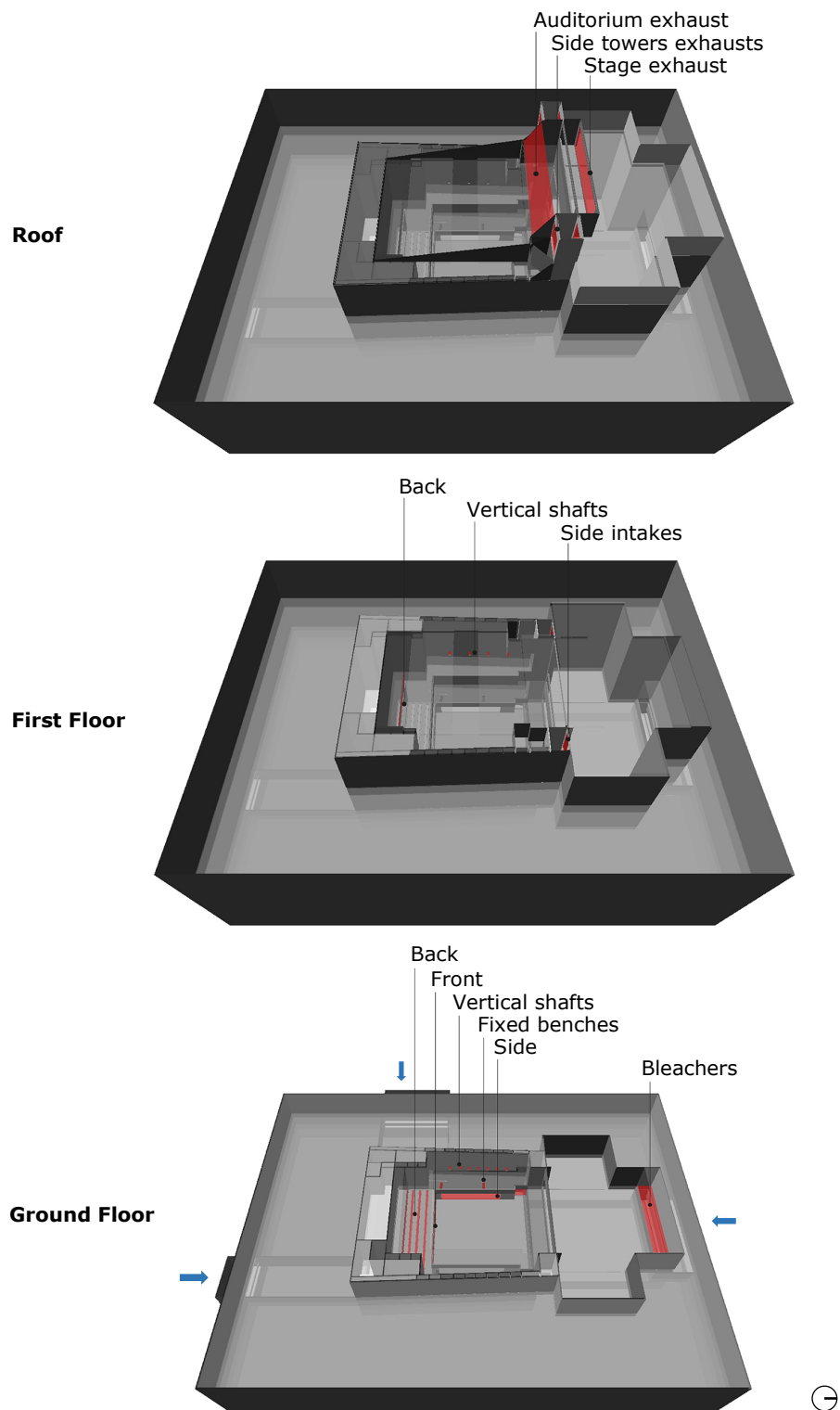


Figure 8.7 Auditorium and stage air supply points

It was also anticipated that both supply and exhaust air paths would have to include sufficient acoustic attenuation to prevent noise breaking in or breaking out of the auditorium, which, after a series of detailed acoustic calculations, it implied a 50% reduction of the free area of the inlet and outlet areas.

The development of this project was followed closely from beginning to completion. The work conducted as a natural ventilation consultant involved feasibility studies and aperture sizing, design optimization through dynamic thermal simulations and CFD simulations, component design and control system design (Vallejo et al., 2015b).

At early design stages, a simple steady state calculation of the airflow rates required to remove internal heat gains was based on a conservative assumption of a 1-2°C temperature difference between inside and outside the building. These initial calculations were made using Optivent 2 and indicated the approximate free area of openings (inlets and outlets) required and enabled the architect to ensure that this area of openings could be provided.

After building completion, a series of field data were collected for the commissioning of the natural ventilation system. Part of this data and the actual natural ventilation performance of the theatre has been compared with the initial calculation outputs from Optivent 2 as a validation exercise for the tool. The field studies consisted in:

- Determining the actual maximum aperture free area in plenum inlets and roof exhausts downstream the acoustic attenuators.
- Monitoring natural ventilation air velocities at each aperture in order to determine the airflow rates achieved driven by buoyancy alone under a typical cold and warm day in Nottingham.
- Monitoring external and internal air temperatures to determine the average Δ DBT during the measuring period.

To validate the tool, airflow rates were determined on site under different external and internal conditions and related to the effective aperture area and Δ DBT during the measuring period. The same conditions were applied to Optivent 2 to evaluate the validity of the results for exploring the feasibility of a natural ventilation strategy and for conducting a preliminary sizing of the required aperture areas for a successful integration in design and performance.

Fieldwork data

Free areas downstream the acoustic attenuators in the plenum inlets (south, west and north) and roof exhausts (auditorium and stage) were measured on site using a standard measuring tape. The free areas are shown in Table 8.1 and compared to the maximum free areas of the motorised louvers placed just before the acoustic attenuators, which regulate the effective aperture area according to the need of natural ventilation and internal and external environmental conditions.

Table 8.1 Site measured free aperture areas and maximum motorised louvers free aperture areas

	Site measured ¹	Vengen Louvers spec ²
South Inlet	1.28 m ²	3.63 m ²
West Inlet	2.98 m ²	9.22 m ²
North Inlet	5.79 m ²	5.31 m ²
Auditorium Exhaust	4.81 m ²	6.28 m ²
Stage Exhaust	4.79 m ²	9.10 m ²

¹ Measured downstream the acoustic insulators.

² Maximum motorised louvers free area.
Equivalent to 43% of the construction area.

The measurements taken on a typical cold day in Nottingham correspond to the 5th December 2016. The sky was overcast at daytime and no wind was present during the fieldwork. The external temperature during the measuring period remained relatively constant with a maximum air temperature of 4.5°C at noon and a minimum of 3°C in the evening. Internal temperatures also presented very small fluctuations and the plenum, auditorium and roof void temperatures averaged 14.5°C, 16.5°C and 17.5°C respectively (Figure 8.8). These conditions became an excellent opportunity to conduct the study, as the achieved airflow rates can be associated to a constant Δ DBT during the measuring period. During this time, both auditorium and stage were unoccupied.

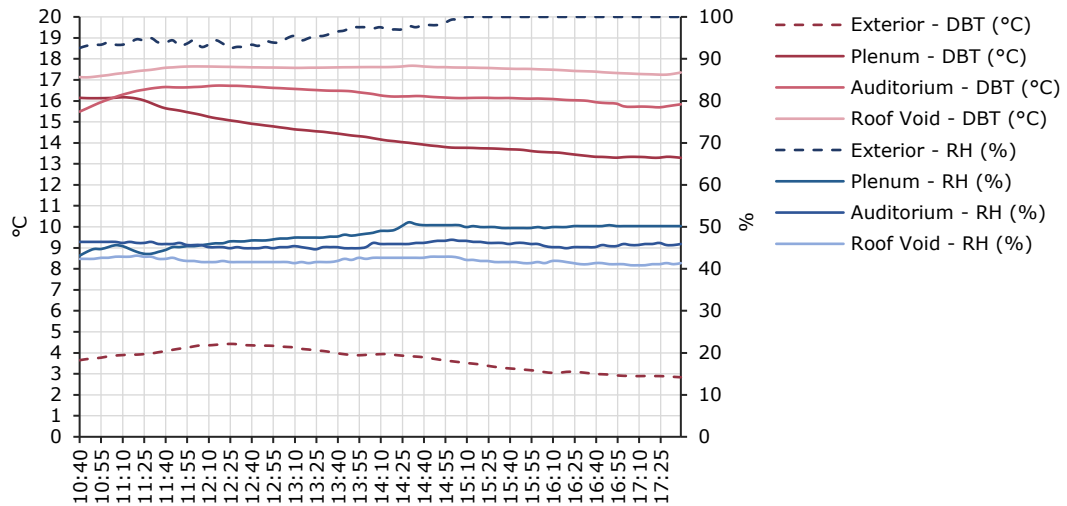


Figure 8.8 NGHS Theatre external and internal DBT and RH monitored on the 5th December 2016

To monitor air velocities on the large aperture areas downstream the acoustic attenuators it was necessary to record several velocities as the airflow cannot be assumed to be uniform due to the obstructions generated by the louvers and the acoustic attenuators. A rectangular grid in line with ASTM D3464-96 (2014) was used to capture air velocity gradients over the apertures, and a thermal anemometer Testo Smart Probe 405i with ± 0.1 m/s accuracy (0 to 2 m/s) was used to obtain the data.

For this study, apertures were manually opened 15% of the louvers' free area, and the high temperature difference during the measuring period (12.4°C) drove the air inside the auditorium and stage. The recorded air velocities ranged from 0 to 1m/s and typical values remained between 0.2 and 0.4 m/s. Figure 8.9 shows an even distribution of air velocities on the apertures despite some localised peak values, and an air velocity gradient with height can also be observed in most of the apertures. The shaded grey areas from the image represent obstructions within the apertures, in most cases (except the north inlet) generated by the acoustic attenuators.

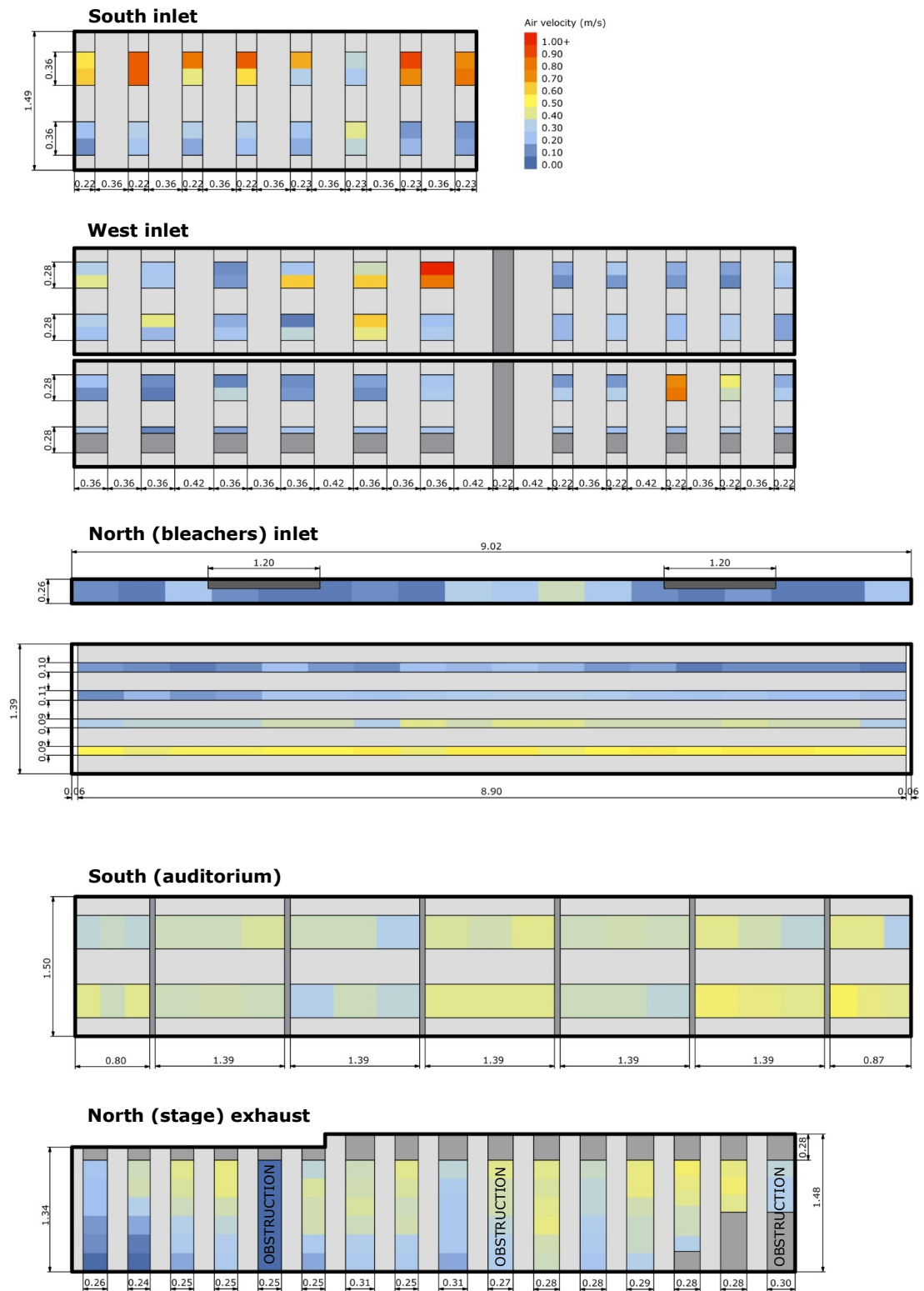


Figure 8.9 Monitored air velocities on NGHS Theatre plenum inlets and roof outlets on the 5th December 2016

The air flow rates achieved during the monitoring period were determined from the recorded air velocity and the area of the opening Equation (8-1), where q is the airflow rate in m^3/s , v is the monitored velocity in m/s , and A is the free aperture area downstream the acoustic attenuators in m^2 .

$$q = v \cdot A$$

Equation (8-1)

Ventilation flowrate from monitored air velocity

The total flow rates IN (through the plenum inlets) and OUT (through the roof exhausts) achieved are shown in Figure 8.10. Ideally, the volume of air coming inside the theatre should be equal to the exhausted volume of air. The results obtained suggested a 10% deviation ($0.33\text{m}^3/\text{s}$) between total airflow IN and OUT, which, considering the number potential external agents that may have slightly influenced on the results and, at the same time, difficult to control during the measuring period (i.e. opening doors by staff members, punctual wind loads, etc) it is considered within the range of acceptable values. The mean airflow rate driven by buoyancy forces under an average ΔDBT of 12.34°C and effective aperture areas as shown in Table 8.2 was found to be $3.05\text{ m}^3/\text{s}$. This would be sufficient to provide sufficient fresh air to a fully occupied auditorium and stage (334 people) in a typical cold/winter day.

Table 8.2 NGHS Theatre effective aperture areas (15% of the louvers' free area) during the monitoring period on the 5th December 2016

	Effective aperture area ¹	
South Inlet	0.54 m^2	IN (2.72 m^2)
West Inlet	1.38 m^2	
North Inlet	0.80 m^2	
Auditorium Exhaust	0.94 m^2	OUT (2.31 m^2)
Stage Exhaust	1.37 m^2	

¹ 15% of the louvers' free area

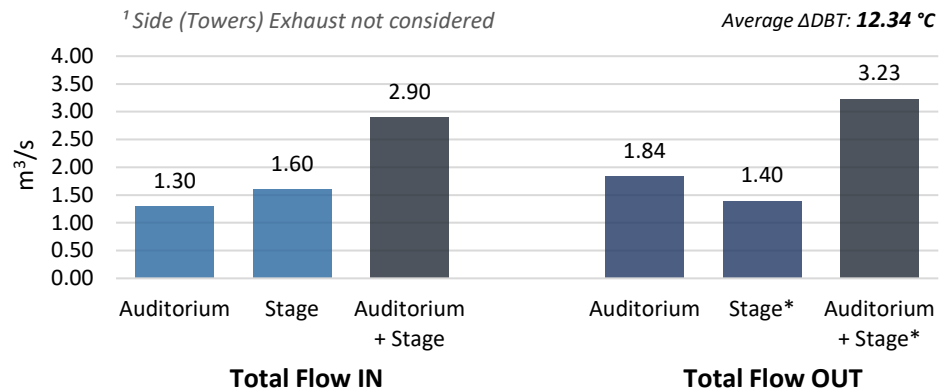


Figure 8.10 NGHS Theatre typical airflow rates achieved with 20% louvers' free area and Δ DBT: 12.34°C

The measurements taken on a typical warm day in Nottingham correspond to the 20th June 2017. The sky was partially covered at daytime and some wind was registered from East direction between 1-3 m/s during the fieldwork. The external temperature during the measuring period remained relatively constant with a maximum air temperature of 20°C at 2pm and a minimum of 18.5°C in the evening. Internal temperatures also presented very small fluctuations and the plenum, auditorium and roof void temperatures averaged 20°C, 23°C and 24°C respectively (Figure 8.11). The stage was constantly occupied for rehearsals and an evening concert for 150 people took place in the evening. As a result, an increase in the roof void internal temperatures can be observed during the evening concert.

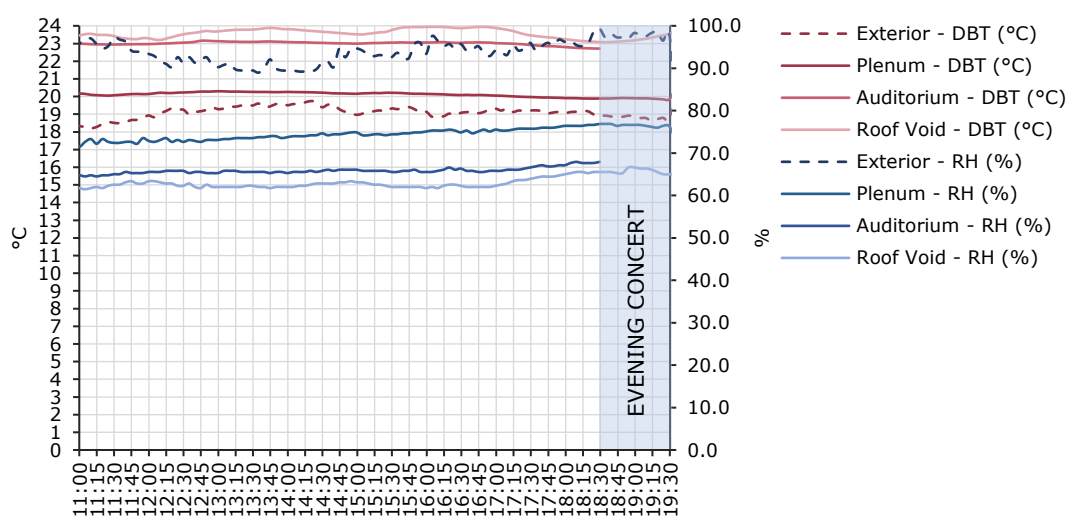


Figure 8.11 NGHS Theatre external and internal DBT and RH monitored on the 20th June 2017

The same methodology was applied to monitor air velocities in the apertures (inlets and outlets) downstream the acoustic attenuators. During this study, the openings responded to the control algorithm, which defined the free opening area according to the season of the year, the external conditions, internal temperatures, CO₂ levels in auditorium and stage, and the occupancy pattern. During the measuring period, apertures opened 40% of the louvers' free area, and the mean temperature difference achieved (2.98°C) drove the air inside the auditorium and stage. The recorded air velocities ranged from 0 to 1m/s and typical values remained between 0.2 and 0.4 m/s. Figure 8.12 shows more irregular distribution of air velocities on plenum inlets when compared to the measurements taken during the cold season, which could have been influenced by wind loads, new obstructions identified at the bleacher seats by the orchestra, and the constant movement of people during the rehearsals. On the contrary, a more even distribution of air was found in the roof outlets, which were less exposed to external agents and further obstructions.

The total flow rates IN (through the plenum inlets) and OUT (through the roof exhausts) achieved were calculated using the expression from Equation (8-1), and are shown in Figure 8.13. The results obtained suggest that the volume of air coming inside the theatre equals the exhausted volume of air at the roof outlets. The mean airflow rate driven by buoyancy forces under an average Δ DBT of 2.98°C and effective aperture areas as shown in Table 8.3 was found to be 3.25 m³/s. This airflow rate was sufficient to provide fresh air and to remove the heat generated by 150 people who were occupying the auditorium during the measuring time. Although there was a little imbalance in airflow rates between auditorium and stage, both spaces were connected as a single space for the evening concert. The concentration of people at the stage during the rehearsals and potential wind forces may have led to this imbalance, very difficult to address.

Table 8.3 NGHS Theatre effective aperture areas (40% of the louvers' free area) during the monitoring period on the 6th June 2017

	Effective aperture area ¹		
South Inlet	1.28 m ²	IN (6.38 m ²)	
West Inlet	2.98 m ²		
North Inlet	2.12 m ²		
Auditorium Exhaust	2.51 m ²	OUT (6.15 m ²)	¹ 40% of the louvers' free area
Stage Exhaust	3.64 m ²		

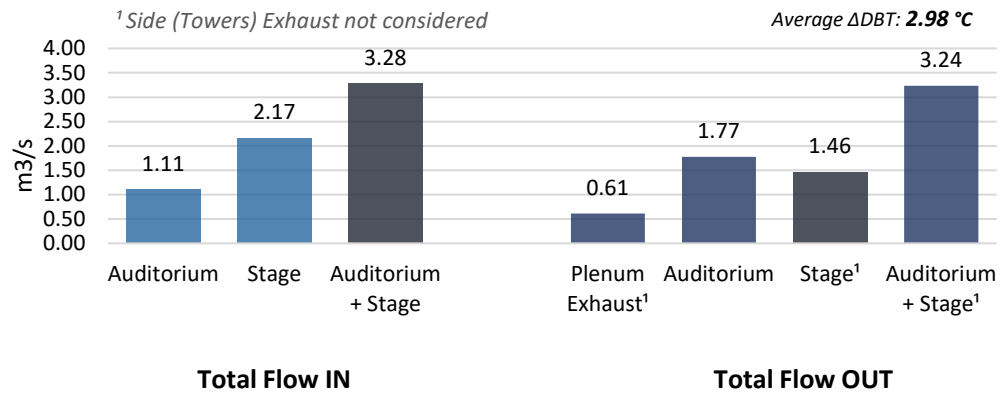


Figure 8.13 NGHS Theatre typical airflow rates achieved with 40% louvers' free area and ΔDBT : 2.98°C

Optivent 2 validation

To validate the tool, the same external and external conditions found during fieldwork were applied to Optivent 2 to evaluate the validity of the results for exploring the feasibility of a natural ventilation strategy and for conducting a preliminary sizing of the required aperture areas for a successful integration in design and performance. Two runs were done for the tool, representing the winter and the summer conditions. A summary of the calculation inputs is shown in Table 8.4. The model was simplified as a single cell model with a main inlet and outlet at lower and higher level respectively (Figure 8.14 and Figure 8.15). The predicted airflow rates achieved were, in both cases, sufficient so supply fresh air to a full occupied theatre in winter, and to remove the heat generated by 150 people in summer to minimise overheating risk.

Table 8.4 Optivent 2 inputs for data validation

	Winter data	Summer data
Latitude	53°	53°
Month	Dec	Jun
Hour	12pm	6pm
Prevailing mean outdoor temp.	5.6	14.3
Meteorological wind speed	N/A	N/A
Terrain data	N/A	N/A
Inlet azimuth	North	North
Outdoor DBT	4.00°C	19.00°C
Indoor DBT	16.34°C	22.00°C
Inlet (free) area	2.72m ²	6.38m ²
Outlet (free) area	2.31m ²	6.15m ²
Stack height	11m	11.09m
Space width	30m	30m
Space height	11m	11m
Space length	12m	12m
Building elements exposed	NO	NO
Solar gains	N/A	N/A
Occupancy gains	334	150
Equipment gains	N/A	N/A
Lighting gains	N/A	N/A

The predicted airflow rates driven by buoyancy alone were finally compared to the monitored data for the same external and internal conditions. The results are shown in Figure 8.16 and suggested that the predicted data by Optivent 2 is in line with the monitored data. It was found a 10% performance gap in the winter day, whereas during the summer day the performance gap resulted 20%. These results can be considered as satisfactory assuming the context of Optivent 2: a steady-state tool to assess the feasibility of natural ventilation strategies in early design stages. At later design stages, advance modelling and dynamic simulations will help to refine the design of the natural ventilation system, as well as minimise the performance gap between predicted and actual data.

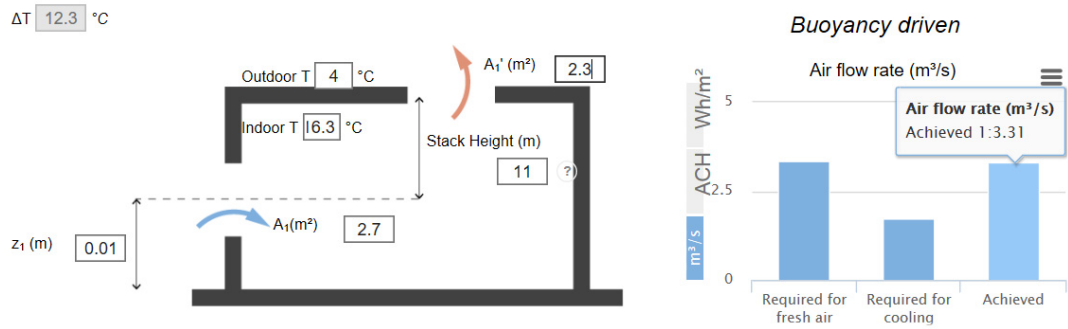


Figure 8.14 Optivent 2 winter model for data validation

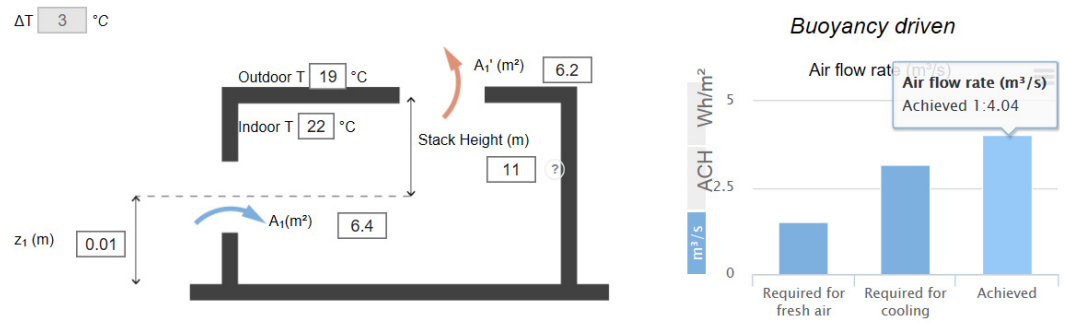


Figure 8.15 Optivent 2 summer model for data validation

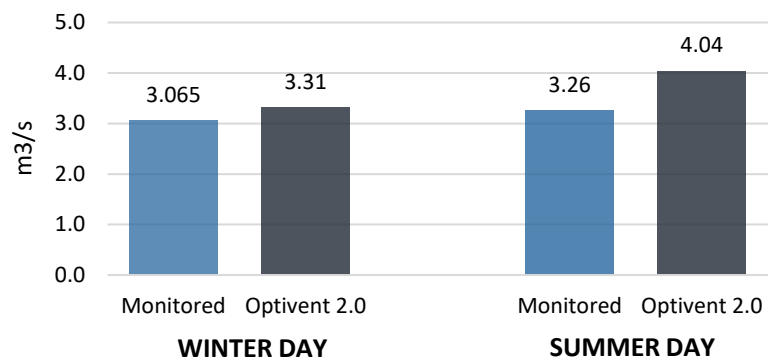


Figure 8.16 Optivent 2 data validation: comparison between predicted and monitored airflow rates for a typical winter and summer day