Heat recovery with hybrid ventilation in office buildings

Reinier Daniël Scholten
1511866

28/01/2015

TU Delft
Faculty of Architecture
Department of Building Technology

Master Thesis
Subject: Heat recovery with hybrid ventilation in office buildings

Tutors:
Dr. Ir. P. van den Engel
Dr. Ing. T. Klein

External examinator:
Dr. I. Nevzgodin

Guest Supervisor:
Ir. G. Blonk, DPA Cauberg-Huygen

Acknowledgements

This thesis is the final element for acquiring the MSc degree at the TU Delft. During the entire process I was fascinated by all the different types of technology needed 'just' to ventilate an office. I would like to thank several people for their help and guidance during this research.

First of all I want to thank my tutors, Dr. Ir. P. van den Engel and Dr. Ing. T. Klein for their guidance and help during the entire process.

I would also like to thank ir. G. Blonk from DPA Cauberg-Huygen for his support and his ideas for the creation of the test setup. Also thanks to ir. R. Hadders from Vision4Energy for making a Fiwihex unit available for me and explaining all the aspects of their system.

Also many thanks to the entire graduation group for helping me to finish the long days at the office at our own table.

And finally I would like to thank my girlfriend Vivian for her great support during the research process and giving new solutions for my design problems.

Delft, January 2015

Reinier Scholten

Abstract

This master thesis examines the possibilities of creating a decentralized hybrid ventilation system with a heat recovery system. Natural ventilation is more commonly used as the main ventilation principle and has the advantage to lower the energy costs of a building. One of the downsides of the natural ventilation system is the lack of heat recovery. By combining the heat recovery abilities of a decentralized mechanical ventilation system with driving forces of natural ventilation, a new decentralized ventilation system is created. This ventilation system will be located at a typical 70's office building in the Netherlands.

The main boundary condition for a natural ventilation system is the driving force. This can be buoyancy or a wind induced principle. The wind induced ventilation principle generates the largest pressures and velocities in comparison with buoyancy based ventilation. The wind direction will have a large influence on the wind velocity at the façade. Average wind velocities of 2.4 m/s are measured during simulations for wind perpendicular to the façade.

In the simulation, the manual heat load calculations are used for the input. These simulations showed that by using two ventilation systems per office, each having a capacity of 50 m^3/h, will meet the requirements of 100 m^3/h of fresh air for the office. The Fiwihex system used for the heat exchangers will recover enough heat in order to reduce the need for additional heating to almost zero.

The final design consists of an integraded box in the façade. This box will have a wind cowl inlet and a venture shaped outlet at the outside of the façade in order to generate a sufficient air flow. Due to the small size of the box, 150x300x1200mm (d x w x h), its location on the façade can vary. The system can be used in an enclosed office area as well as an open plan office.

Due to the high efficiency of the heat exchanger and the use of wind as a driving force, energy savings of 5% of the total energy use of the building and even higher can be achieved.

List of symbols

Symbol	Description	S.I. unit
A	Area	m^2
A_b	Area bottom opening	m^2
A_e	Effective opening area	m^2
A_t	Area top opening	m^2
c	Specific heat	J/kgK
C_d	Discharge coefficient	-
C_p	Pressure coefficient	-
C_{pb}	Pressure coefficient back façade	-
C_{pf}	Pressure coefficient front façade	-
DR	Draught rate	%
g	Gravitational constant	m/s^2
h	Height difference	m
Heat	Heat load	J
h_{ref}	Reference height	m
h_x	Height	m
L	Lenght	m
L_{eq}	Equivalent lenght	m
n	Normalisation factor	-
P_0	Static reference pressure	Pa
P_d	Dynamic pressure	Pa
P_x	Static pressure at point x	Pa
ΔP	Pressure difference	Pa
Δp_{ath}	Athmospheric pressure	Pa
ΔP_{cp}	Presure difference by Pressure coefficient	Pa
$\Delta p_{in/out}$	Pressure difference as a result of air velocity	Pa
ΔP_{temp}	Temperature pressure difference	Pa
Δp_{total}	Total pressure difference	Pa
ΔP_{wind}	Wind pressure diffrence	Pa
r	Relative roughness length	-
T_a	Ambient temperature	K
T_u	Relative turbulent intensity	-
ΔT	Temperature difference	K
U	Insulation value	W/m^2K
U_h	Wind speed at height h	m/s
v	Air speed	m/s
v_{ref}	Wind velocity at reference height	m/s
v_x	Wind velocity at height x	m/s
Q	Airflow	m^3/s
z_0	Terrain roughness factor	-
ζ	Pressure resistance factor	-
λ	Friction factor	-
ε	Roughness lenght	m
ρ	Density	kg/m^3
ϕi	Internal heat load	W
$\phi loss$	Heat loss	W

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1 Introduction

A building can be ventilated by using mechanical or natural principles. Both systems have their advantages and disadvantages. In chapter 1.1 the current problems with mechanical ventilation system will be discussed. After the problem statement the research question will be discussed with the corresponding research framework in chapter 1.2 and 1.3. As the system needs to be implemented on a building this will be described in chapter 1.4. The last chapter will explain the sustainable advantages of natural over mechanical ventilation.

1.1 Problem Statement

Natural ventilation is used more often. The new office buildings created in the Netherlands have to fulfil the demands of Dutch government. The goal of the Dutch government is to create almost zero energy buildings at around 2018 (Agentschap NL, 2013). Not only in the production of energy but also in the consumption of energy. One of the main energy users in the building is the ventilation system (Pérez-Lombard, Ortiz, & Pout, 2008). In order to create a good working environment most buildings use a ventilation system that is also able to heat or cool down the building (HVAC). This HVAC system consumes about 55% of the total energy consumption of an office building (Pérez-Lombard et al., 2008). Most office buildings rely on a HVAC system according to the one-size-fits-all method. There is one HVAC unit for the entire building; clothing and behavior are the only modifications for the occupants (Brager & de Dear, 2000).
Sometimes there is a thermostat which allows the users change the temperature by +/- 3 °C. This is most of the time accomplished by increasing or decreasing the air flow. Operable windows are hardly seen in these office buildings due to the ventilation type. The HVAC system is calibrated at a predefined pressure difference between the inlet and outlet and opening a window will greatly change this difference (Raue & Prendergast, 2012). HVAC systems are also linked to the sick-building syndrome (Kumara, Rajapaksa, Perera, & Jayasinghe, 2013 & Jayasinghe, 2013). Dust and mould will create for example respiratory problems which only occur when the employees are inside the building. In order to increase the user control of the airflow small decentralized mechanical ventilation system are currently on the market (eg. climarad, Smartboxx and Troxx FSL). These systems extract air from the office and use it to preheat the colder outside air in the winter and will give the occupants the ability to set the desired temperature and airflow. However these systems still use energy in order to power the fan. The fan will create a high pressure difference in order to force the air through the system.

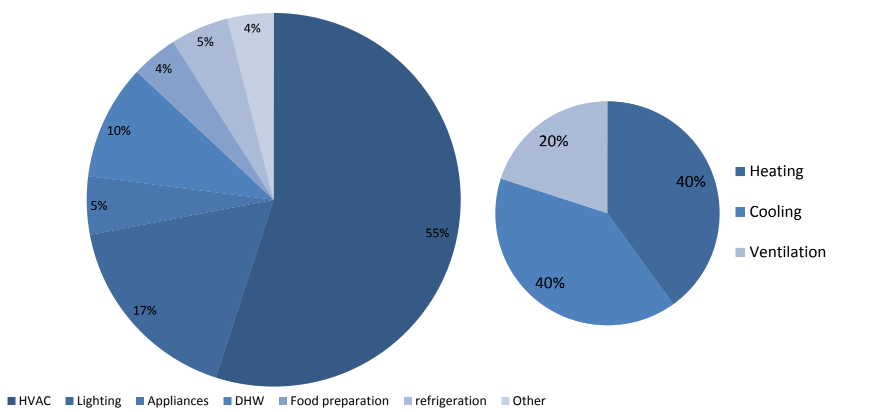


Figure 1: Left: Energy usage in office buildings (Pérez-Lombard et al., 2008), Right: Energy usage of a HVAC system. (Wood & Salib, 2013)

A good solution would be a natural ventilation system. The natural ventilation system is mostly based on pressure differences. These pressure differences can be created by the wind pressure on the façade or by the temperature difference between the office and the corridor (Kleiven, 2003). One of the advantages of a natural ventilation system is the acceptance range of the occupants. The occupants in a naturally ventilated building will accept lower and higher temperatures than in mechanically ventilated buildings (De Dear & Brager, 1998). There is also a greater manual influence of the people working in the offices.

One of the downsides of a complete naturally ventilated system is that it will not always supply enough fresh air to meet the user demands. This is a result of the driving forces like wind and sun. These forces are not always sufficient to

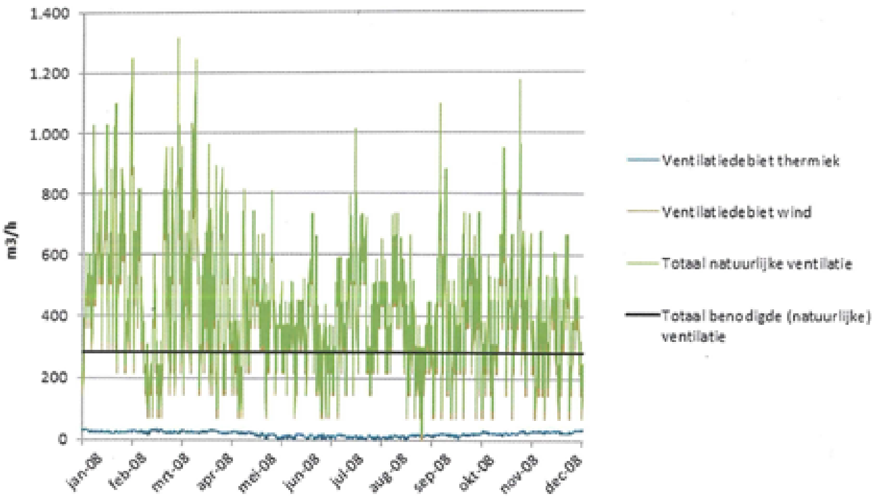


Figure 2: Air flow due to natural ventilation. (Zeiler, 2014, April).

completely ventilate a building. In order to keep the building ventilated, the natural ventilation system can be supported by a mechanical system. The combination of natural ventilation with mechanical backup is the hybrid ventilation system. This system is becoming more common but more detailed research has to be done in order to make it a good and efficient system. The hybrid ventilation principle is based on low pressure air flows next to the high pressure mechanical ventilation systems. One of the downsides of the natural or hybrid ventilation system is the increased need for heating during the colder winter months (Zeiler, 2014, April). As a result of this the energy usage increases during the winter months leading to higher costs.

1.2 Research Question

The problems created by the smaller decentralized mechanical ventilation systems, lower temperature acceptance and use of energy the entire year, and the advantages of natural/hybrid ventilation result in the following research question

How can a decentralized mechanical ventilation system be redesigned to make use of natural ventilation principles?

Scope

The scope of the research will be limited by the type of building and the amount of floors used for the system. The buildings on which the hybrid ventilation system will be applied will be a medium high office buildings of about 10 floors. This office building will be located in the Netherlands and makes use of a shading system to block the sun. This office will only be used during office hours from 9.00 to 17.00. For the weather data the location of Amsterdam is used (Liggett & Milne, 2013). The system will also be designed for the winter situation in order to make good use of the heat recovery.

Sub Questions

During the research several sub-questions will be answered;

What are the driving forces for the hybrid ventilation system?

What are the possibilities of personal adjustment in a hybrid ventilation system?

How can the quality of the air be maintained regarding external and internal pollution?

Is it possible for the system to be only wind or buoyancy based or is a combination of these possible?

How large are the energy savings compared with mechanical ventilation?

Can the system be used for single side ventilation as well as cross ventilation?

1.3 Research Framework

During the research the variables which have a direct effect on the airflow will be discussed. This will be done in the literature research. The literature research will give a better understanding of the factors influencing the airflow used for the ventilation. Also the reference projects will give examples of working principles. With this knowledge a new design will be created and tested using computational fluid dynamic models to verify the design and their influence on the airflow.

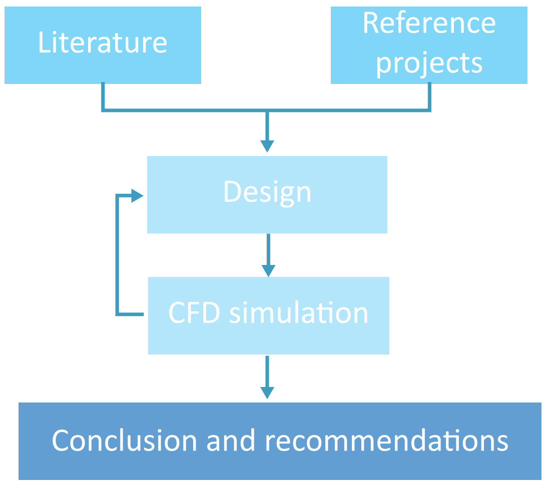


Diagram 1: Research process

Literature research

In the literature several influences will be researched using papers, magazines and books. The first factor will be the ventilation system. The principles of natural, mechanical and hybrid ventilation will be discussed as well as their driving forces. The influence of these ventilation systems on the comfort of the occupants will be discussed together with the amount of energy used in these systems and their potential energy savings. The second part of the literature research will be about the factors that influence the airflow like; building shape, location and opening sizes. The third part will be about the comfort of the occupants of the building. When do they experience draft and what are the standards for the air quality like pollution and CO2 concentration. The last part of the literature will discuss some reference projects. These projects use natural or hybrid ventilation systems to ventilate and heat/cool down the building.

Methods description

The current decentralized mechanical ventilation systems will also be analyzed to understand their principles and the airflows inside the unit. This will include the components like air filters and the heat exchanger regarding pressure loss and efficiency. With the knowledge gained during the literature research a first sketch design will be created. It will then be manually calculated in order to get a better understanding of the effects on the airflow and to create a feasible solution. In order to make optimal use of the façade, the openings used for the system will also be researched by using a computational fluid dynamics (CFD) model. This model will determine the optimal design of the openings. The CFD program used for this is ANSYS Fluent. The CFD model will also calculate the indoor airflow and mainly focusing on airspeeds and location of the airflow as it could be experienced as drought. The final design will be a total integration of all the elements, heat exchanger, air filter, façade opening and indoor openings.

Design description

In the final weeks the design which is created in combination with the CFD model will be placed into the architectural context. This will result in an optimal airflow, with the possibility of heat recovery for the hybrid ventilation, and the boundary conditions for an architectural facade design.

1.4 Building description

Typologies

For designing a new hybrid ventilation system it has to be located on a site and building. In order to keep the design available for multiple typologies of office buildings the design will be based on a hypothetical office building. This office building will be in the category which has the largest available building stock and the corresponding typology. According to a study about the building stock in 2012, the largest group of empty office buildings is built before 1979 (Bak, 2012). According to a study by van Meijel en Bouma the mostly used typology

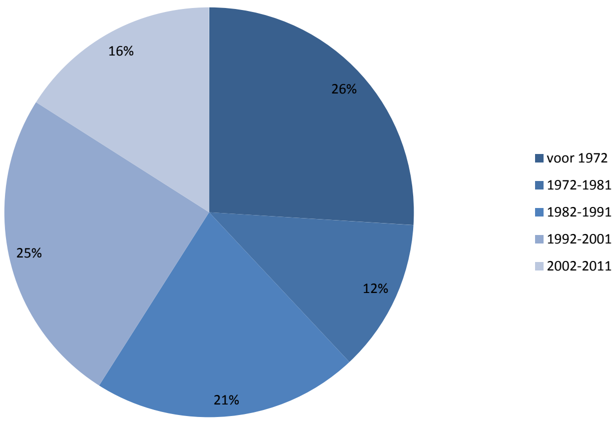


Figure 3: Building period of vacant office buildings in the Netherlands in 2012. (Bak, 2012)

in this time period are cell-based offices (van Meijel & Bouma, 2013). In these building types there is a central corridor with offices on both sides. These offices have standardized office modules with a depth of 5.4m. The corridor between the offices usually has a width of 1.8m. Later on in the seventies they started creating wider corridors and deeper offices. The width of the offices is dependent on the amount of people working together in the office but the most standard width is about 3.5m. As a result of this typology, most offices built before 1979 are rectangular shaped.

Description used office

The building used in this research will have about 10 stories of with a floor to floor height of about 3m. The dimensions of the floor plan are 75 m x 16 m. The offices are located on both sides of a 3m wide corridor resulting in an office depth of 6.5m. The width of the office is based on the grid created by the concrete columns and stand 3.6m apart. The floors consist of an in-situ casted concrete floor, supported by beams connected to the columns. The structural stability is created by a couple of solid concrete walls dividing two office cells. This results a non-load bearing façade with a lot of possibilities regarding thickness and shape.

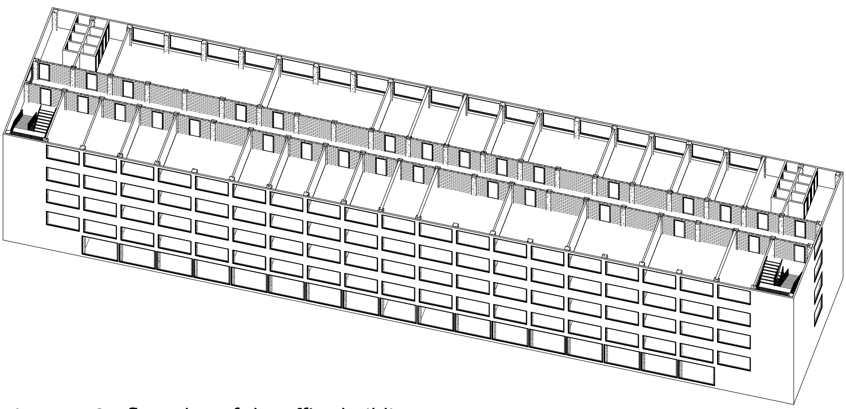


Figure 4: 3D floorplan of the office building

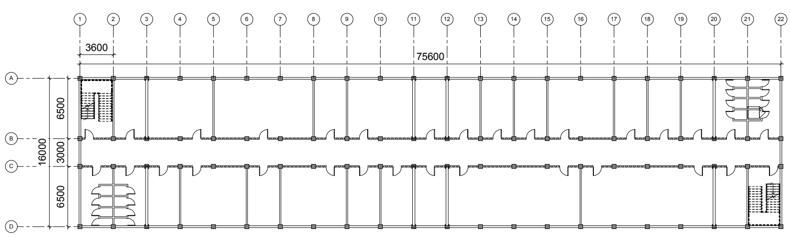


Figure 5: Floor plan

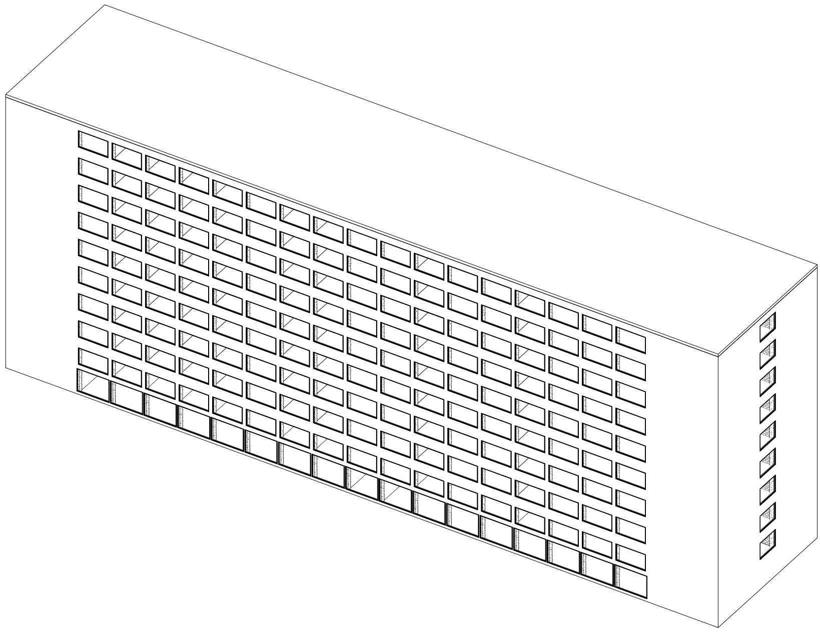


Figure 6: Outside view of the office building

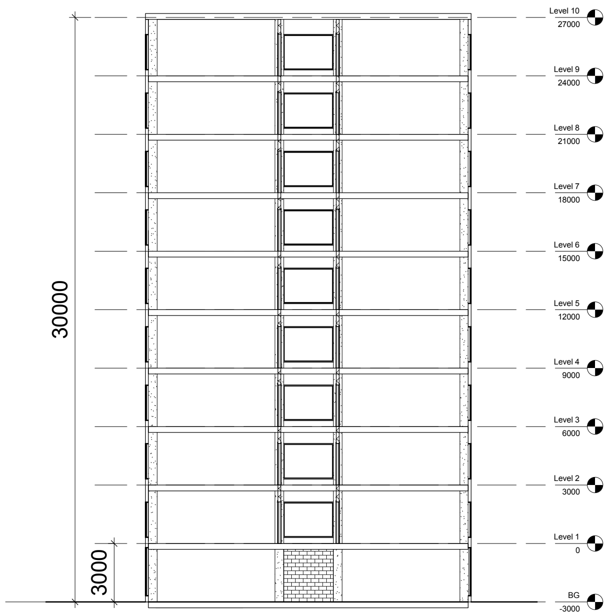


Figure 7: Cross section, grid 14

1.5 Sustainability

Office buildings are becoming more sustainable. With the introduction of certificates like BREEAM and LEED, designers are able to verify their sustainable design to one of these ratings (BREEAM-NL, 2013; U.S. Green Building Council, 2014). The certificates also encourage designers to create more sustainable buildings. One of the possibilities for designers is to decrease the amount of energy used during the operational phase of the building. This will generate lower operational costs of the building.

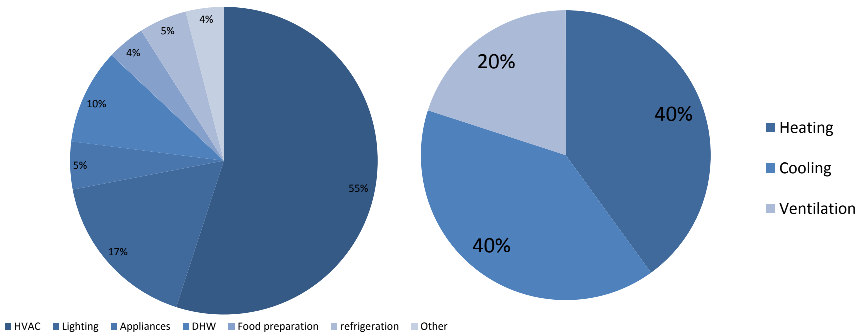


Figure 8: Left: Energy usage in office buildings (Pérez-Lombard et al., 2008), Right: Energy usage of a HVAC system. (Wood & Salib, 2013)

One of the possibilities to save money is by using a more energy efficient ventilation system. A mechanical ventilation system will use about 55% of the total energy consumption of a building (Wood & Salib, 2013). This value is an estimate as the total amount of energy needed for heating and/or cooling is based on the climate the building stands in. Out of this 55%, about the half is used to cool down the building due to internal heat loads. This part of the used energy can easily be replaced by using outdoor air as the average external temperatures in the Netherlands are most of the time lower than the average design temperature of

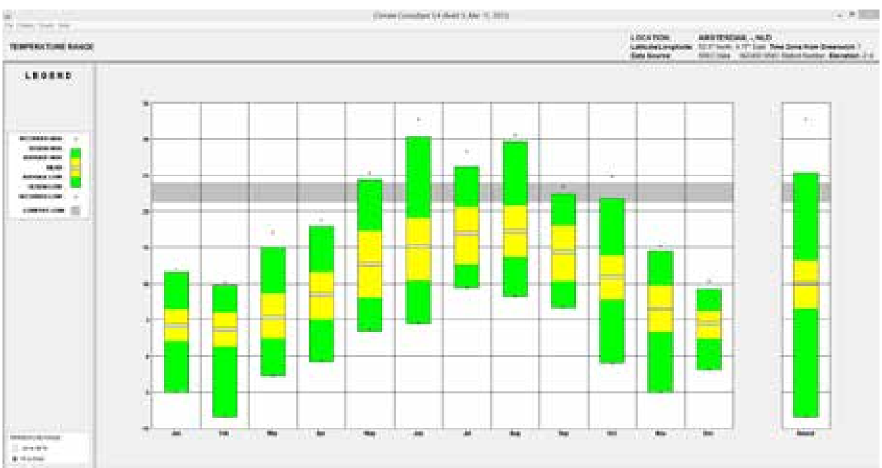


Figure 9: Temperature range in Amsterdam, The Netherlands. (Liggett & Milne, 2013)

21-23°C (Liggett & Milne, 2013) and thus using a natural ventilated system. During the winter times it is hard to use direct outdoor air to ventilate the building due to the high temperature differences between the indoor and outdoor temperature (Gratia & De Herde, 2004). As a result of this problem the air will be heated by using additional heaters.

As a result of the low operating pressures it is hard to make use of a heat recovery system in a natural ventilated system. Due to this lack of energy recovery in a naturally ventilated system the amount of energy needed to keep the room at a comfortable temperature is higher than mechanical ventilation system (Zeiler, 2014, April) (figure 10). A hybrid ventilation will use less energy compared with a completely natural or mechanical ventilated system, due to the reduction of operational time of the fans (van der Aa, 2002). But due to the lack of heat recovery and the need for additional heating elements the amount of energy used is about the same as with mechanical ventilation. The amount of energy used in a hybrid ventilation system can be reduced by using a heat recovery system. This will decrease the amount of energy needed for heating (Emmerich, 2006).

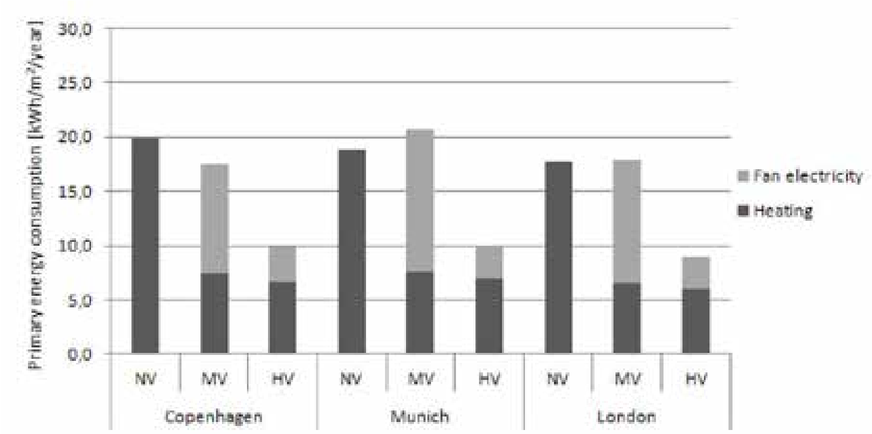


Figure 10: Primarly energy consumption of ventilation systems. (Zeiler, 2014, April)

2 Theory

A lot of research has been done about the principles of natural ventilations. Before the designs are created some theory about the topic of natural ventilation will be given. Firstly the type of ventilation systems will be discussed. This is followed by the main principles of natural ventilation in chapter 2.2. The wind is of great influence on natural ventilation and will be discussed in chapter 2.3. Chapter 2.4 and 2.5 will discuss the minimal needs for an office building and the environmental influence on the ventilation system. The last chapter explaining the theory will discuss the different types of pressure difference as well as the pressures needed for the heat exchangers.

2.1 Ventilation Systems

Natural ventilation system

Natural ventilation systems are based on pressure differences created by natural principles. These principles can be wind and sun (Etheridge, 2012). They both create pressure differences but in different ways. The principles of natural ventilation are used throughout the centuries. A simple smokestack on a house is a good example of a natural ventilation system. Air is heated in the fireplace and due to the pressure difference the air will move into the chimney. The natural

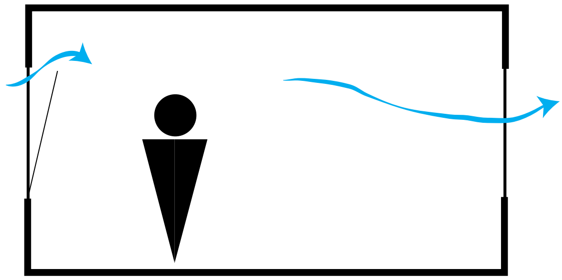


Figure 11: Natural ventilation principle, cross ventilation

ventilation system has the great advantage of not or hardly having any operational costs as it is based on natural principles which happen throughout the day. Another advantage is the ability for the occupants to manually control the airflow by for example just opening a window (van den Engel & Kurvers, 2014). This will have a direct effect on their working environment. One of the downsides of a natural ventilation system is the lack or the abundance of wind or sun. This can result in a not functioning ventilation system (Li & Delsante, 2001). An abundance of wind can result in a high air-speed through the office but this can be regulated by using gratings at the openings. When there is a lack of wind there are hardly any possible solutions that will increase the airflow in the building. Another downside of natural ventilation is heat recovery. Heat recovery is very hard to use on natural ventilated system. As a result of this the fresh air flowing into a building in the winter will need to be preheated.

Mechanical ventilation system

Mechanical systems are the most used ventilation systems in office buildings. The air is mechanically supplied into the office and also mechanically extracted. This creates a ventilation system with a very high efficiency. This mechanical ventilated system is also used for heating or cooling the office. If ventilation and heating are combined the system is called a heating, ventilation and air conditioning system (HVAC-system). Mechanically ventilated buildings are not dependant on any natural principles as the air flows from the inlet to the outlet in the office. Because it is mostly a closed system the exhaust air is returned to the HVAC unit. By returning the exhaust air to the HVAC unit heat recovery is possible in order to reduce the amount of energy needed to preheat the supply air. Mechanical ventilation systems also have some personal adjustment. Most of the time it is possible to change the temperature with +3/-3 degrees for each room. This temperature change is mostly created by increasing or decreasing the airflow. One of the downsides of this ventilation system is the lack of personal regulation. Due to the pre-calculated air flow most system will be distorted when a window is opened due to a change in pressure difference in the system. Another downside of a mechanical ventilation system is the high start-up cost and the continuous maintenance costs.

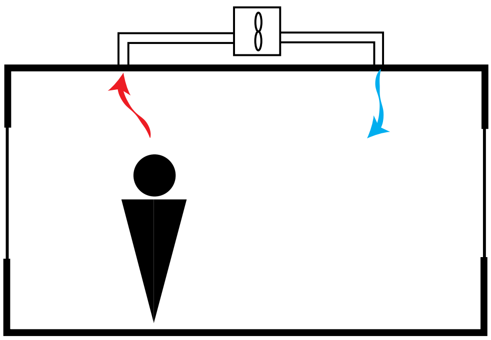


Figure 12: Mechanical ventilation

Hybrid ventilation system

A hybrid ventilation system is a combination of mechanical and natural ventilation system. These two can be combined into three different principles (Heiselberg, 2002).

The first is the use of both systems autonomous. The ventilation system will use either the mechanical or the natural system and they will never work together. This can profitable for using the mechanical ventilation during opening hours and the natural ventilation during the night. Or to switch between natural and mechanical ventilation in different seasons when there is a different demand.

The second principle is the mechanical assisted natural ventilation. In this principle the ventilation system is based on natural ventilation but when this system is not sufficient to supply enough fresh air the mechanical system will assist. It can assist the natural system by increasing the extraction rate or the supply rate in order to enhance the pressure differences used for natural ventilation.

The third one is the stack and wind assisted mechanical ventilation. This system has a continuous running mechanical ventilation system in which a stack is used as an exhaust. The mechanical system creates very small pressure differences which are increased by natural principles as a result of wind or the stack effect.

All these three systems benefit from the combination of a mechanical and a natural ventilation system. The mechanical system is most of the time subordinate to the natural system and only helps when the natural ventilation system is not able to fulfil the demand for fresh air. This is also one of the downsides as sensors have to be installed and set at the right date to activate the mechanical system. So a complete network of sensors is needed to make the hybrid system work properly. The hybrid ventilation system has the advantages of the natural ventilation system regarding the possible personal adjustments (Raue, Kurvers, & Leijten, 2014). If the designed airflow will be distorted due to for example opening a window the mechanical system starts up in order archive the designed airflow again.

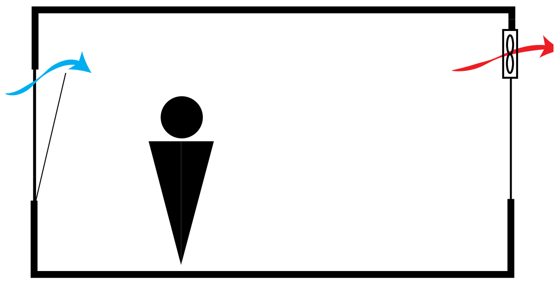


Figure 13: Hybrid ventilation

2.2 Ventilation Principles

Natural ventilation principles

Natural ventilation can be based on different principles, wind induced and buoyancy induced ventilation (Wood & Salib, 2013). Both ventilation principles are based on pressure difference but the driving force is difference. Wind induced natural ventilation is, like in the name, based on pressure difference generated by the wind flowing around the building. Buoyancy induced natural ventilation can be generated by for example solar heat. There is also a third principle, this is making use of both wind- and buoyancy induced ventilation.

Wind induced ventilation

The first principle of natural ventilation is wind induced ventilation. This type of natural ventilation is based on pressure difference generated by the wind on a building. If for example the wind is blowing against the façade of a square plan office building there will be an increase of pressure on the windward side of the building and a negative pressure on the leeward side of the building (ASHRAE., 2005). If someone would then open a window on the windward and a window on the leeward side of the building the air will flow through the office from the front to the back. This is the cross ventilation principle [see chapter 2.1]. In order to calculate the airflow and the volume of the airflow through the building the

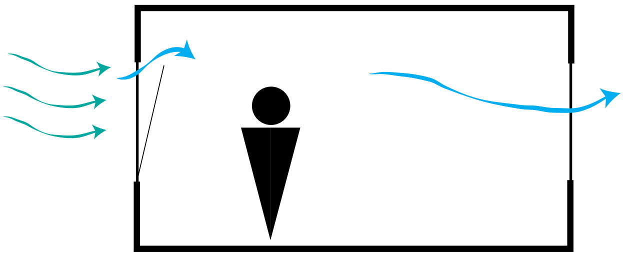


Figure 14: Wind induced ventilation principle

pressure differences between the windward and leeward side of the building have to be calculated (Price Industries Limited, 2011). This can be done with Bernoulli's equation (Germano & Roulet, 2006) [1]. This formula makes use of the wind pressure coefficient generated by the building shape, the air density and the wind speed. With this pressure difference, opening size and opening discharge coefficient the total amount of air can be calculated.

$$Q = C_d * A_e * \sqrt{\frac{2 * \Delta P}{\rho}} \qquad [1]$$

$$\Delta P = (C_{pf} - C_{pb}) * \frac{\rho * v^2}{2} \qquad [2]$$

Buoyancy induced ventilation

Buoyancy induced ventilation is the second principle for natural ventilation. Buoyancy is generated by the pressure difference as a result of the difference the inside and outside temperature (Allocca, Chen, & Glicksman, 2003). Warmer air has a lower density as a result of this an airflow will be generated from the colder air towards the warmer air. In order to calculate this airflow several parameters have to be taken into account. The first is the indoor temperature. This temperature is calculated by adding the internal heat load and the head load of the sun to the set indoor temperature. This will result in the indoor temperature including all the heat gains (Pérez-Lombard, Ortiz, & Pout). When the air inside is cooled by for example an air conditioner the amount of energy it retracts from the air has to

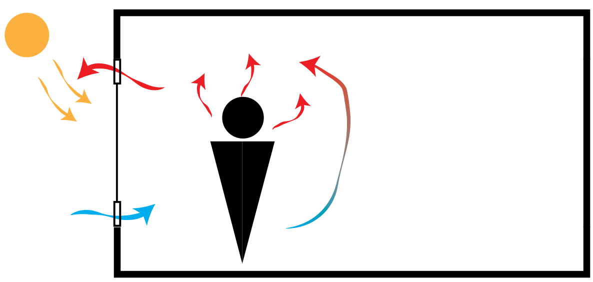


Figure 15: Buoyancy induced ventilation principle

$$A_e = \frac{A_t * A_b}{\sqrt{A_t^2 * A_b^2}}$$ [3]

$$Q = C_d * A_e \sqrt{\frac{2 * g * h * \Delta T}{T_a}}$$ [4]

$$\Delta T = \frac{\Delta Heat^{\frac{2}{3}} * T_a^{\frac{1}{3}}}{(2gh)^{\frac{1}{3}} * (\rho c_{air} C_D A_e)^{\frac{2}{3}}}$$ [5]

be implemented as a negative heat load. Another parameter of the formula is the effective area of opening combined with the corresponding discharge coefficient. The efficient opening area can be calculated by taking all the opening area's in the airflow system and then derive the effective opening area out of the formula [3] (Li, 2000). This however only works when the discharge coefficient of the inlet(s) as well as the outlet(s) are the same. In most cases this is not applicable, so each opening has to be put in the formula separately in order to take all variables into account. Another variable in the formula is the height difference between the inlet and the outlet opening. As a result of the pressure differences between hot and cold air the warmer air will rise to the top of the room. This will result in a temperature gradient going from the bottom (cold) to the top (warm) of the office. In order to get the highest efficiency in pressure differences the openings should be placed in the top and the bottom of the walls. This will create the largest pressure difference as a result of buoyancy [4,5] [6].

Combining buoyancy- and wind induced ventilation

Most of the time both driving forces occur together rather than independent (Li & Delsante, 2001). In order to make full use of natural ventilation both systems have to be taken care of in the design. If the ventilation system is using both buoyancy and wind induced ventilation a combination of the pressure differences can be calculated. As mentioned in the paragraph about the wind induced natural ventilation, this system is based on pressure differences created by the wind. The buoyancy induced ventilation principle is based on pressure difference as a result of the temperature difference of the air. These pressures can be calculated for each floor and can be combined in order to calculate the pressure difference based on the combination of both principles. By combining these system it is also possible that both principles work against each other, resulting in a lower pressure differences or a complete malfunctioning system (Allocca et al., 2003). A system based on both systems will be able to ventilate almost throughout the year as there are hardly any days where there will be no wind or sun.

$$\Delta P_{Tver} = g * h * (\Delta \rho_{\text{top/bottom}})$$ [6]

2.3 Wind

Wind pressure on the facades

If the wind is blowing onto a building it will create pressure difference due to the airflow around the building. Several studies have been done in order to explain where and how large these pressure differences are (ASHRAE, 2005; Ernest, Bauman, & Arens, 1992; Phaff, 1977; Van Moeseke, Gratia, Reiter, & De Herde, 2005). The wind pressure on the façade is known as the pressure coefficient (Cp). This Cp value can be calculated with the following formula [7, 8] (Costola, Blocken,

$$C_p = \frac{P_x - P_0}{P_d} \qquad [7]$$

$$P_d = \frac{\rho - U_h^2}{2} \qquad [8]$$

& Hensen, 2009). In this formula the static wind pressure, the static reference pressure [8] and wind speed are taken into account. This formula gives one value for the entire façade. Tests and analytical studies have shown that this is not the case on the facades and the pressure coefficient calculated with formula [8] can only be used as a rough estimate. Actually the airflow at the facades of the building goes into several different directions and that there are different pressure coefficients over the entire façade (Jensen True, 2003). As a result of this the research of Davenport and Hui from 1982 is still used as a guide for the local pressure coefficients on tall buildings. This research is until today used in the

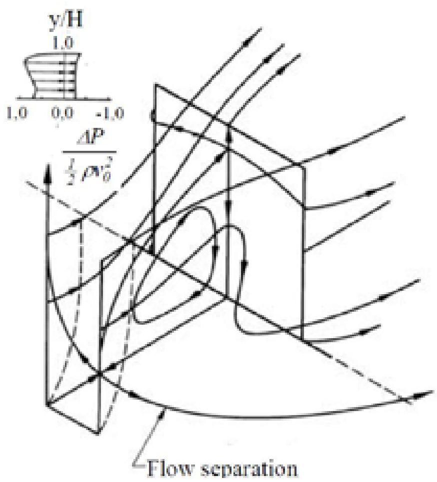


Figure 16: Flow separation when the wind hits a high building. (Jensen True, 2003)

ASHRAE handbook (ASHRAE., 2005). There is however a large difference between the methods used for determining the Cp values. The study of Costola, Blocken and Hensen shows that there can be quite a difference between the wind tunnel testing and the CP calculation programs like CpCalc+ and the Cp Generator. The difference can be explained by the large complexity in all the parameters in the formula's and the simplification of this data to be suitable for the calculation (Costola et al., 2009).

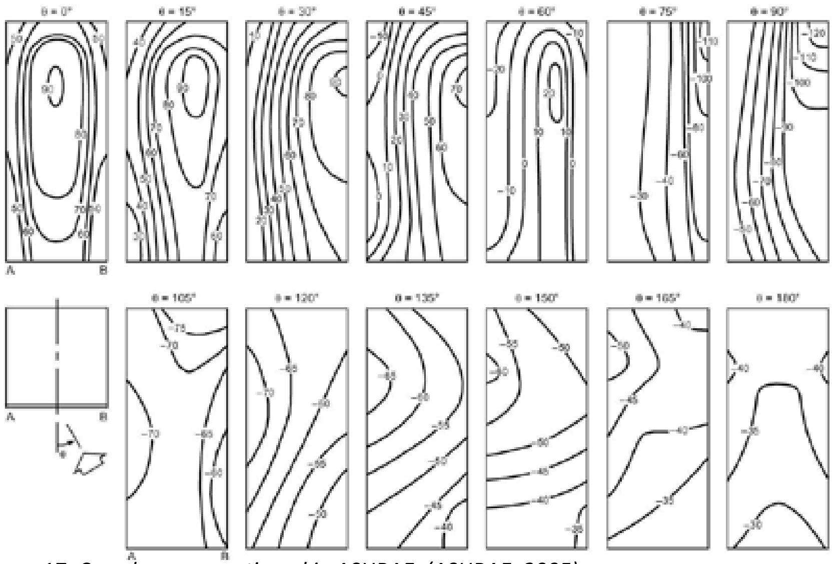


Figure 17: Cp values as mentioned in ASHRAE. (ASHRAE, 2005)

Wind flow around buildings

Due to the diversion of the airflow around the building, vortexes will be created at the front and back of the building (figure 16). As a result of these vortexes the wind speeds at these locations is different. Several studies have been done in order to simulate and test these wind speeds (Bottema, 1993; Mochida et al., 2002; Stathopoulos & Baskaran, 1996; Summers, Hanson, & Wilson, 1986). These studies are looking at the turbulence models and the velocities they create. As a result of these prediction models there are factors given to describe the relation between the main wind speed and the wind speed in the vortexes. There are however some differences between the measured data and the computational data (figure 18) . According to Strathopoulos and Baskaran these can be a result of small scale phenomenoms in the environment which are very hard to put into the computer model.

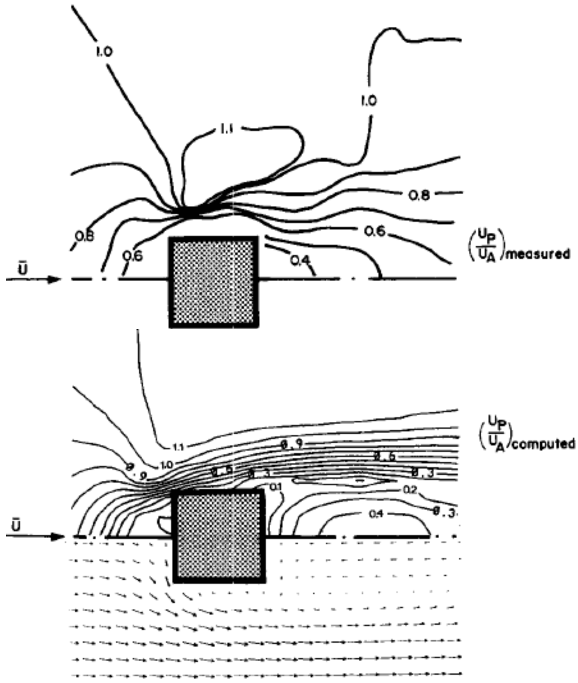


Figure 18: Velocity field around building. Relation between the local wind velocity and the main wind velocity. (Stathopoulos & Baskaran, 1996)

Building Shape

Another variable on the wind pressure is the shape of the building. Buildings with large flat surfaces facing the wind will have a larger pressure differences between the windward and the leeward side of the building (Zhou, Kijewski, & Kareem, 2003). For the data given in ASHRAE, the height of the high-rise building should be larger than 3 times the width (H > 3W)(ASHRAE., 2005). Circular buildings would for example create less drag and genially guide the airflow around the building. This creates a completely different pattern for the Cp values. Most of the values can be derived from the data and tables given by the ASHRAE handbook (figure 17) But if the floor plan of the building is not square or rectangular, the data given by the handbook will not be applicable. In order to use data for these building simulation programs or wind tunnel testing is needed.

2.4 User Needs and Demands

Fresh air supply

The main goal of ventilation is providing fresh air. According to the European building regulations/standards the minimum amount of fresh air for a person in an office is 6.5 dm^3/s (Nederlands normalisatie instituut, 2013). This will result in an air supply of roughly 25 m^3/h per person. This person will take up about 20 m^2 in the office (Nederlands normalisatie instituut, 2013). As a result of this the ventilation needed for purely fresh air,is 1.25 m^3 per square meter of office floor. This is only for fresh air. If there are any pollutants in the air or a large source of pollutants in the office the amount of ventilation needs to be increased. Studies have shown that increasing the ventilation rate in an office will result in a decreast amount of dissatisfied people. (Wargocki, Wyon, Sundell, Clausen, & Fanger, 2000). So it is preferred to get a higher amount of fresh air per person.

Pollution

Fresh outdoor air will enter the building through vents in the façade. As a result of this all the pollutions in the outdoor air might be able to enter the building. When the air is not filtered all this polluted air will enter the building and decrease the air quality for the occupants. An example of the outdoor polluters is the traffic around a building. This traffic will create particular matter which can be harmful for the occupants of the building. There are also a lot of possible polluters inside the building (Spengler & Chen, 2000). A few examples are leaking water pipes, ventilation filters and particles created by office equipment.

But also the occupants themselves are sources of pollution as they exhale CO_2 gas. Increased amounts of CO_2 gas in the air can lead to irritations to the respiratory system. If an occupant of the building has health issues which are related to their workspace it is called the sick building syndrome. Another source of pollution is inside the office furniture and the interior of the office. These products can create volatile organic compounds (VOC) (Kumara, Rajapaksa, Perera, & Jayasinghe, 2011). These VOC's can evaporate at room temperature and can create eye, nose, throat and skin irritations.

The sick building syndrome is a term, given to health issues related to the direct work environment. Some of these issues can be irritation to the eyes, bad smell in the office and headache. These symptoms will disappear when people leave the building (Bluyssen, 2009). One of the solutions can be to increase the amount of fresh air for the occupants. A study of Wargocky (Wargocki et al., 2000) shows that an increase of the ventilation rate (L/s per person) of about 3 will result in less health problems regarding throat and mouth dryness, thinking problems and personal sensation.

Air speeds

The amount of air will have a great influence on the performance of the office employees. But a too large air flow will be experienced as negative. The sensation of air flowing to fast around a person is called draft. Every person will have a different experience about the airspeed when they call it draft but in general an air speed larger than 0.2 m/s will be experienced as draft (Arens, Turner, Zhang, & Paliaga, 2009). The formula used to calculate the draught rate was create in the 1970 by Fanger and is still used today (see formula [9]) (Zeiler, Smelt, & Boxem, 2013). In this formula he uses the average air speed, air temperature and the standard deviation of fluctuations in air speed. In the research about the thermal comfort in Dutch energy efficient office buildings they stated that in an office a draught rate of 15% is recommend. The draught experienced by the people in the offices is also dependent on the air temperature. Arens' research states that even if the air movement is larger than 0.2 m/s the people in the offices even want

$$DR = (34 - T_a) * (\bar{v} - 0.05)^{0.62} * (37 * \bar{v} * T_u + 3.14) \quad [9]$$

increased air speed during high operational temperatures (Arens et al., 2009). The opposite is happening with colder operational temperatures when the occupants want less air movement.

Personal adjustment

Thermal comfort in an office building is a state of mind. This states Fountain, Brager and de Dear in their paper about expectations of indoor climate control

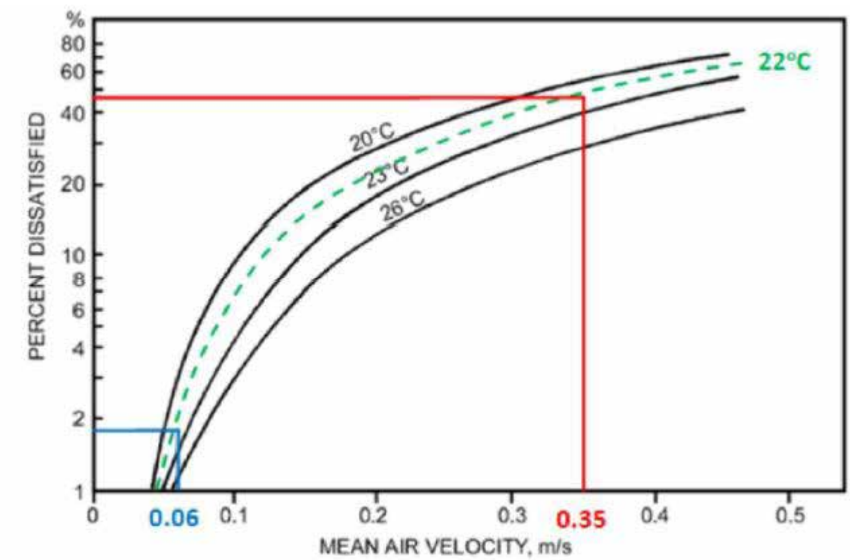


Figure 19: Relation between the air velocity and the amount of people dissatisfied.

(Fountain, Brager, & de Dear, 1996) . What someone experiences as a comfortable temperature doesn't have to be comfortable for other people. The state of mind of the people will have a large influence on their temperature acceptance. In a free-running building (the indoor temperature is in relation with the outdoor temperature) there will be a larger acceptance than in buildings with a HVAC system. One of the features of free-running buildings is the ability to adjust the system to the personal needs of the occupants. These adjustments can be made by for example opening a door, opening a window or putting on a fan in order to adjust the temperature. The influence of each of these adjustments is different but it makes the people change their state of mind regarding the indoor environment (Raja, Nicol, McCartney, & Humphreys, 2001). The effects of the personal adjustments are also the greatest when used in free running buildings due to the direct relation with the outdoor temperatures.

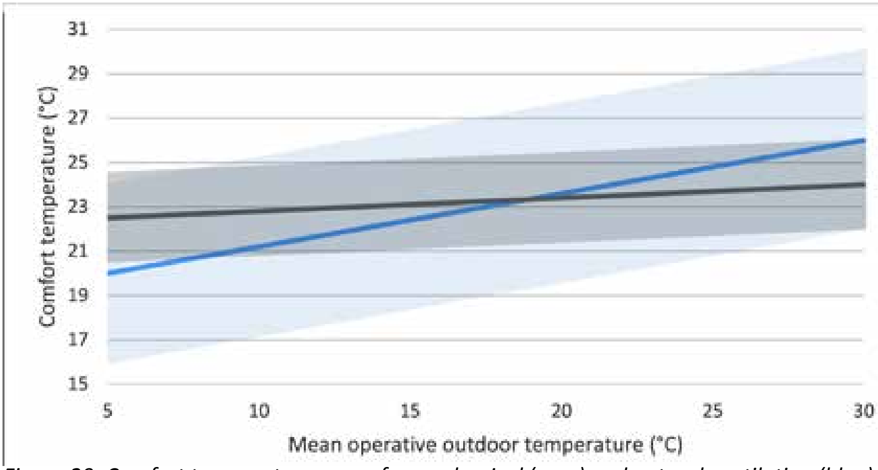


Figure 20: Comfort temperature range for mechanical (grey) and natural ventilation (blue). (Fountain, Brager, & de Dear, 1996)

2.5 Environmental Influences

The environment plays an important role in naturally ventilated buildings. The environment can have different influences on the building. These influences are; the location, the orientation and external pollution (Spengler & Chen, 2000). These variables can have an influence on the amount of ventilation but also on the quality of the air entering the building or the comfort levels inside the offices.

Location

Every location is different. There might be some other tall buildings in its direct environment or it could be located in an open field. This is of great influence on the wind pressure and the wind speeds on the façade (ASHRAE, 2005; Van Moeseke, Gratia, Reiter, & De Herde, 2005). The influence of the terrain on the wind speeds is known as the surface roughness length (table 1). This factor will define the effect

Terrain description	z_0 (mm)
Very smooth, ice or mud	0.01
Calm open sea	0.20
Blown sea	0.50
Snow surface	3.00
Lawn grass	8.00
Rough pasture	10.00
Fallow field	30.00
Crops	50.00
Few trees	100.00
Many trees, hedges, few buildings	250.00
Forest and woodlands	500.00
Suburbs	1500.00
Centers of cities with tall buildings	3000.00

Table 2: Surface roughness lengths. (Van Moeseke, Gratia, Reiter, & De Herde, 2005)

$$V_x = V_{ref} \frac{\ln\left(\frac{h_x}{z_0}\right)}{\ln\left(\frac{h_{ref}}{z_0}\right)} \qquad [10]$$

of the surface on the drag for the wind (Wieringa, 1986). A smooth open grass field will have a smaller roughness factor than a residential area. As a result of the higher roughness factor in dense city centres, the wind surface layer will become higher and gives lower wind speeds at higher attitudes (Manwell, McGowan, & Rogers, 2009) [10]. This has to be taken into account when designing natural ventilation systems for high-rise buildings in a dense area. The wind speeds will change with the height of the building.

Orientation

The second role of the environment is the orientation. This can be split into two parts, the sun and the wind direction. The sun is most of the time needed for generating buoyancy in buoyancy induced ventilation system. If for example a solar chimney is needed for the natural ventilation system this has to be oriented into the sun during the needed time period (Gratia & De Herde, 2007). As the sun will follow a given and standard path around the hemisphere the shading created by the environment has to be taken into account. Large buildings or trees casting shadows over the building can result in disabling the buoyancy induced ventilation system. Also the wind direction has to be taken into account. Most of the time there is a wind chart available of the location (Liggett & Milne, 2013) of the building but the wind chart is generated with data collected at 10m altitude and in an open plane. As a result of the environment the wind can come out of one direction when this isn't the main wind direction.

Pollution

The third part of the influence of the environment is the external pollution. Pollution might decrease the comfort level when using natural ventilated system. Looking at the environment possible pollution sources are busy roads and industry (Spengler & Chen, 2000). These create fine particular matter which can have a negative effect on the health of the occupants of the building when they enter the building together with the fresh air.

2.6 Ventilation Openings

Opening categories

There are different sizes of openings which allow air to enter or exit the building. Between these sizes there are large differences regarding the Reynolds number (turbulent or laminar air flows) and the discharge coefficient. According to Etheridge (Etheridge, 2012) the openings can be divided into 4 different categories.

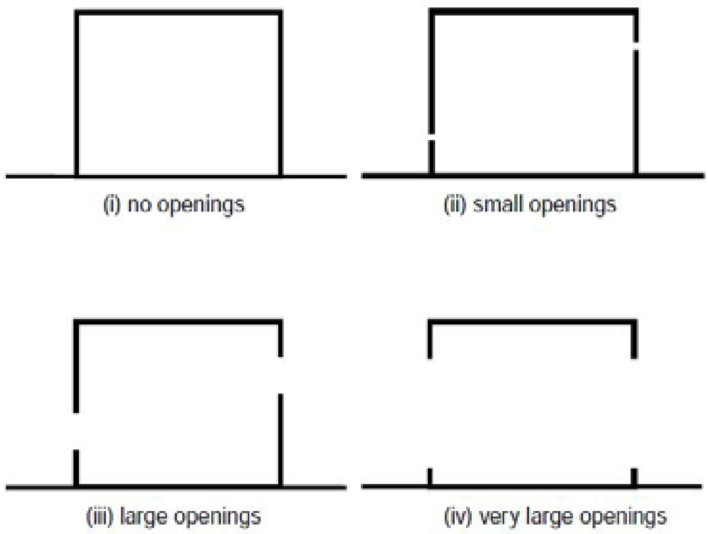


Figure 21: Types of ventilation openings. (Etheridge, 2012)

Firstly there are no openings at all. The ventilation in these buildings is created by air flowing through small cracks in the building. As a result of these very small openings it is very hard to take these openings into account in a computational fluid dynamics (CFD) program.

The second opening category is the small opening. Small openings are not large enough to distort the pressure levels on the façade and will only change the pressure within a few diameters of the opening. As a result of this the pressure differences as a result of buoyancy can be used with the corresponding still-air Cd in order to calculate the airflow. The wind will have great influences on these pressure levels and need to be taken into account.

The third types of openings are large openings. The main characteristic is the influence of the opening on the airflow. With large opening it is not unidirectional anymore and the still-air Cd value is no longer applicable. Also the large opening size has a large effect on the pressure levels on the façade in range of the opening.

The volumetric flow rate can be calculated by the following formula.

The last category of opening Etheridge mentioned is two very large windows. They have about the same properties as the large openings but they only affect the airflow on the entire façade. As a result of their size the wind flow through these openings is usually very large, even with relatively low pressure differences.

Opening	Typical opening area, A (m^2)	A/A_w (%) $A_w = 10\ m^2$	External shape
Adventitious	<0.005	<0.05	often flush, but not always
Vents, small windows	0.005 to 0.2	0.05 to 2	often flush, but not always
Chimneys, stacks	0.05 to 5	not relevant	usually not flush
Large open windows	0.5 to 5	5 to 50	often not flush

Table 3: Classification of the openings. (Etheridge, 2012)

The windows can be placed into one of these categories by looking at the typical opening area of the window. The first category will have area lower than 0.005 m2. The second category 0.005 – 0.2 m², the third 0.5 – 5 and the last category has opening areas larger than 5 m² (table 2).

Discharge coefficient

In a naturally ventilated office the fresh air is most of the times entering the building by openings in the façade. These openings can vary from an open window to a small vent integrated in the window frame. All the different types of openings have an independent discharge coefficient. This discharge coefficient (Cd) can be calculated by using the following formula [11] (Etheridge, 2012): In this formula

$$C_p = \frac{Q}{A}\sqrt{\frac{\rho}{2 \bullet \Delta p}} \qquad [11]$$

the pressure difference across the opening, the opening size and the volumetric flow rate are the main variables. Pressure difference can be generated by wind or buoyancy. For the wind the pressure difference can be calculated by using the Cp value and for buoyancy the pressure difference will be generated by the different densities on both sides of the openings. The given formula however assumes that throughout the whole opening there will be a constant pressure with a laminar flow. A turbulent flow (determined by the Reynolds number) will generate fluctuations in the pressure difference and the volumetric flow rate. This can be a result of changing wind pressures. Also the size of the opening affects the Reynolds number and the influence on the flow rate.

Air flow

As stated in chapter 2.4 draft can be experienced as uncomfortable and needs to be reduced or redirected in order to reduce the amount of unsatisfied people as a result of draft. Depending on the type of grille and the wind speed the flow pattern of the air flow can be calculated. Several studies have determined the relation between for example the opening of an open-able window and the airflow (Heiselberg, Svidt, & Nielsen, 2001) and the reduction of draft created by ventilation openings (van den Engel, 1995). These studies give some good examples on the effect caused by draft in an office building.

2.7 Pressure Difference

In order to generate airflow through the heat exchanger, there needs to be a pressure difference between the inlet and outlet. As a result of this pressure difference the air will flow from the high-pressure area (inlet) towards the low-pressure area (outlet).

$$\Delta P_{total} = \Delta P_{ath} + \Delta P_{cp} + \Delta P_{temp} + \Delta P_{in/out} \qquad [12]$$

Total pressure difference.

The total pressure difference (ΔP_{total}) between the inlet and outlet can be calculated by adding up the pressure difference create by multiple factors, see formula [12]. There will be no difference in atmospheric pressure (Δ_{Path}) as it is at the same location at about the same height of the building.

The next factor is the pressure created by the wind pressure on the façade (ΔP_{Cp}). This pressure is a result of the wind speed and the corresponding Cp value for the given wind direction and façade direction, as can be seen in chapter 2.3. This value has no influence on the pressure difference due to the very small difference in Cp value between the inlet and outlet on the façade.

There will be a pressure difference as a result of the temperature difference (ΔP_{temp}) between the floor and the ceiling. According to the formula (formula [6]) given in chapter 2.2, this will result in a pressure difference of 0.4 Pa at a height difference between the inlet and outlet of 2m.

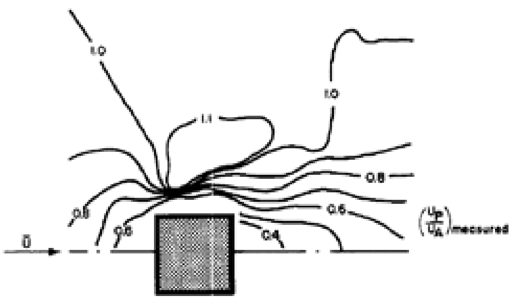


Figure 22: Wind velocity factor around buildings (Stathopoulos & Baskaran, 1996)

The pressure difference created by the velocity ($\Delta P_{in/out}$) is a result of the shape of the inlet and outlet of the ventilation system. The wind speed at the façade will be a result of the vortex pattern in the recirculation zone (Bottema, 1993) and the corresponding normalisation of the main wind speed, see chapter 2.3. A wind cowl on the inlet will direct wind into the system with no loss of energy. So the dynamic pressure created of the façade with a normalisation factor of 0.4 (Stathopoulos & Baskaran, 1996) will, according to formula [13] , be 3.4 Pa. As a result of the wind cowl at the exhaust of the ventilation system there will be a dynamic pressure as a result of the passing wind. The cowl used to create this pressure will have to be able to adapt to the flow direction of the wind on the façade. This cowl will have a

$$P_{dyn} = \left(\frac{1}{2} * \rho * v^2\right) * n \qquad [13]$$

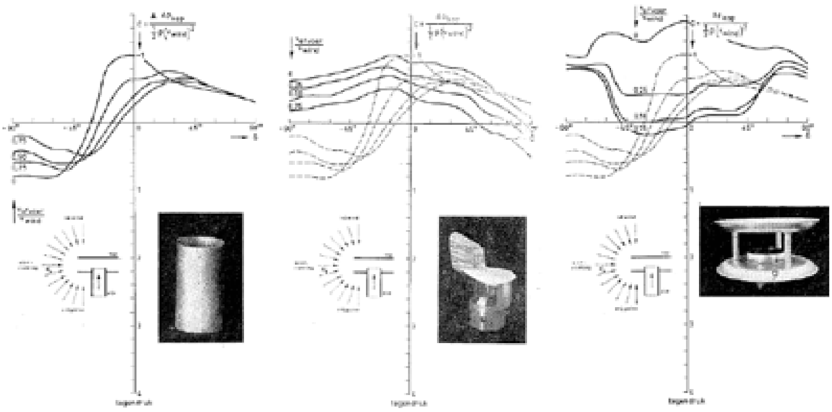


Figure 23: Wind cowl characteristic values. (de Gids & den Ouden, 1986)

characteristic value of -0.8 (de Gids & den Ouden, 1986). In combination with the normalisation factor of to the main wind speed, the pressure difference create by the exhaust will be 2.89 Pa. If a bend tube is used and the wind is perpendicular descending with the shape of the exhaust, the pressure drop will only be 1.27 Pa due to the characteristic value of -0.35.

The total combined pressure difference created by the inlet and outlet cowls can be added to each other resulting is a generated pressure difference of 6.29 Pa.

Pressure loss over the heat exchanger.

According to data given by ir. R.H. Hadders (Hadders, 2014), the pressure loss over the heat exchanger is 4 Pa at a flow of 100 m3/h and 10 Pa at 300 m3/h. This will result in a pressure loss coefficient (ζ , formula [14]) of 4.37 with a velocity of 1.23 m/s.

$$\zeta = \frac{\Delta P * 2}{\rho * v^2} \qquad [14]$$

3 Reference projects

In order to take a look at the current decentralized mechanical ventilation systems 3 systems will be discussed; Fiwihex, Smartbox and Climarad. The last chapter will go in detail in order to explain the principles and system of the hybrid ventilated Bang & Olufsen Headquarter in Denmark. These reference projects combined with the theory will lead to a sketch design in chapter 3.5.

3.1 Fiwihex

Figure 24: Fiwihex system air to air heat transfer.

The fine wire heat exchanger sytems (fiwihex) exists of a 'cloth' created by very thin copper wires. In the cloth the copper wires can be divided into chambers in order to create a extended flat plate heat exchanger. The most commonly used type of the fiwihex is when the copper wires are connected by copper tubes with water running through them. Due to the conductive properties of copper and the large surface area for the air, through the copper cloth, the efficiency of this system is very high. According to Vision4Energy, producer Fiwihex, the fiwihex system has an efficiency of about 85 % at temperature differences of 2 degrees (Vision4Energy, 2014). Also the pressure difference over the heat exchanger is very low. According to mail contact with the director of Vision4Energy, R. Hadders, the pressure loss is 4 Pa at 100 m3/h and 10 Pa at 300 m3/h (Hadders, 2014). The advantage of the fiwihex is the low pressure loss and the small size of the panels. The Fiwihex system is mostly used in greenhouses where the water flowing through the tubes is stored in underground aquifers for later use or to serve as a heating source for local housing. The downside of the fine copper cloth is that it is sensitive for pollution so a filter in front of the heat exchanger is recommended to reduce the amount of service needed. This filter will than increase the pressure drop significantly.

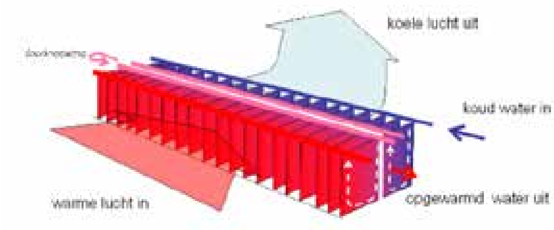


Figure 25: Fiwihex system principle for air to liquid heat transfer. (de Zwart, van Noort, & Bakker, 2008)

3.2 Smartbox

Figure 26: Concept model of the Smartbox. (Smartfacade, 2006)

The Smartbox is a concept idea created by the smart façade team. This team exist of a combination of architectural companies, Cepezed, contractors, Jan van der vlucht en zn BV, Level energy technology bv, manufactures, glaverbel westland BV and research institutes, TNO, ECN. The main component of the Smartbox system is the cross flow heat exchanger. This heat exchanger has a relatively high efficiency, about 92% (Smartfacade, 2006). Only a small portion of the total pressure drop is created by the heat exchanger. With an indoor temperature of 20 °C, an outside temperature of 4 °C and an air flow of 150m3/h, the pressure drop will be about 26 Pa. (Recair, 2014). The other components are the heat pump, humidifier, evaporator, condenser and the fan. All the components are placed in a large box of 1120x350x400 mm, w x h x d. The box will be placed on the ceiling between two floors. This will allow the system to blow the fresh air under the ceiling for maximum mixing and distance. The air is then extracted near the window. In order to keep the same humidity levels the heat exchanger is able to change the flow direction. The condensate moisture will then re-evaporated by the airflow in order to humidify the airflow. One of the advantages of the smartbox is the ability for the users to not only stop the changing or flow directions but also to completely stop the heat exchanging by bypassing the heat exchanger. During the summer the outdoor air will be able to provide sufficient cooling energy in order to cool down the office.

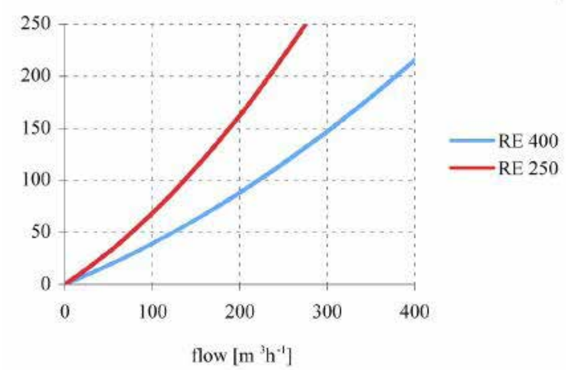


Figure 27: Relation between the air flow and the pressure loss. (Smartfacade, 2006)

3.3 Climarad

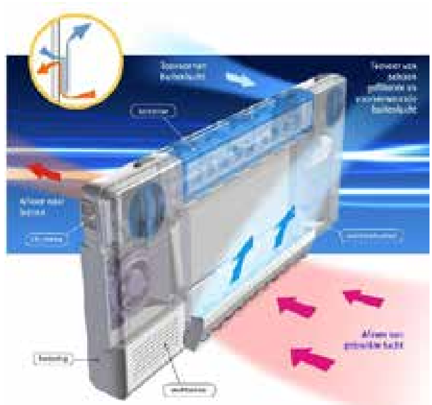


Figure 28: Principle of the ClimaRad system. (Climarad BV, 2014 a)

Climarad has created a couple of decentralized ventilation systems. Their system is based on a flat radiator based heat exchanger. This system is combined with a radiator which is able to provide the additional heating. The ventilation system extracts air from the floor of the office. The heat from the system is than exchanged by a honeycomb heat exchanger made of polypropylene. Due to the thickness of the material and the large surface area of the honeycomb the efficiency of the heat exchanger is about 85% (Climarad BV, 2014 a). The standard Climarad ventilation unit has dimensions of 1135x512x104 mm, w x h x d. This system does have a maximum ventilation rate of 125 m3/h. Climarad also has a larger system called the Maxibox. This system does have a capacity of 300 m3/h but at the cost of a pressure difference of 200 Pa (Climarad BV, 2014 b). The smaller Minibox system has a pressure drop of 50 Pa with a capacity of 125 m3/h (Climarad BV, 2014 c). All the climarad systems are computer guided and triggered by CO2- and humidity levels. There is a small manual override which changes the airflow volume.

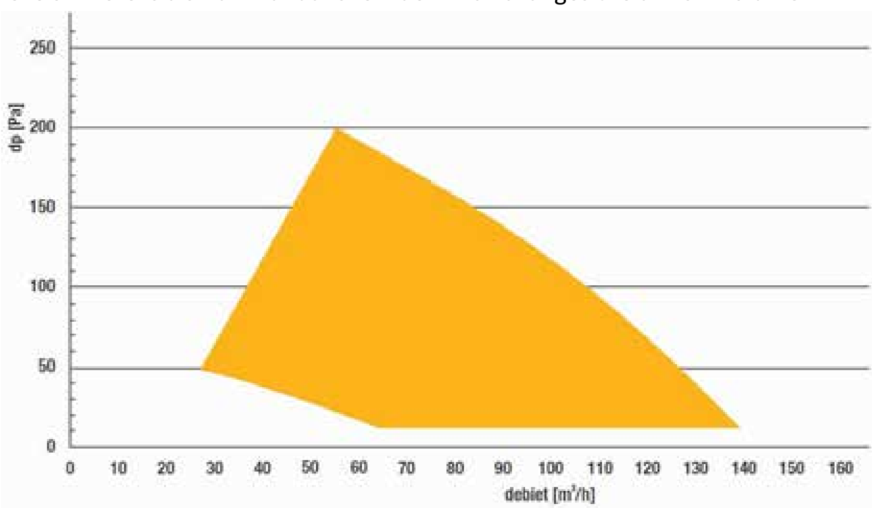


Figure 29: Relation between the air flow and the presure loss in the ClimaRad Minibox system. (Climarad BV, 2014 c)

3.4 Bang & Olufsen Head Quarter

The Bang and Olufsen (B&O) head quarter is located in Streuer, Denmark designed

Figure 30: B&O Headquarter, Streuer, Denmark. (Hendriksen, Brohus, Frier, & Heiselberg, 2002)

by KHR Architects. It is an office building which was finished in 1998 (Hendriksen, Brohus, Frier, & Heiselberg, 2002). For the design of the building B&O demanded an office building which had a minimum of installations. So the building had to be as simple as possible but with a high quality of indoor comfort. The building is oriented north – south so it has a 'cold' and a 'warm' façade. Its structure exists of precast concrete elements with a complete glass north façade and a closed south façade (Hendriksen et al., 2002).

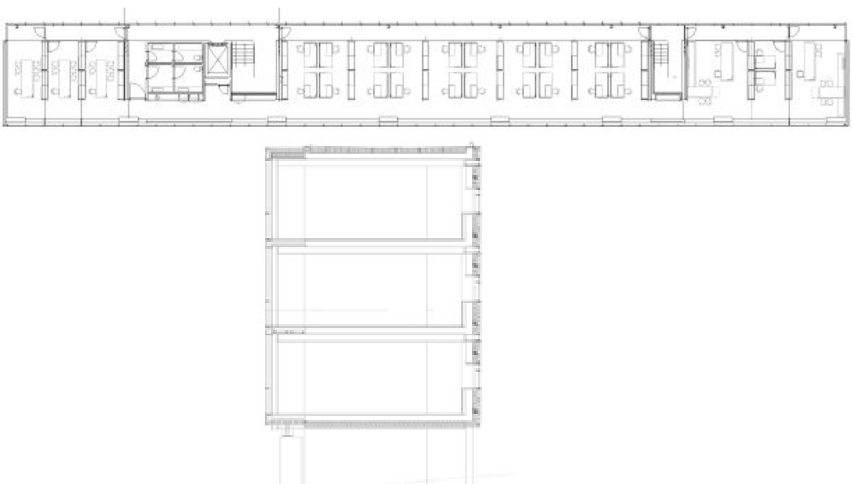
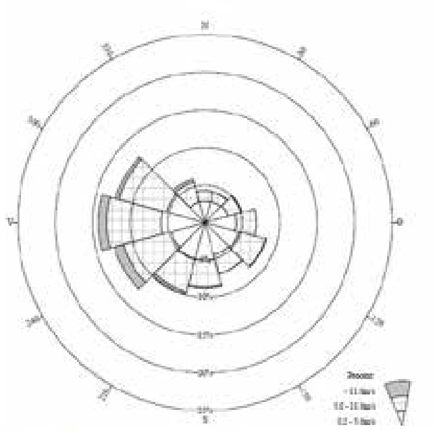


Figure 31: Floorplan and section of B&O Headquarter. (Hendriksen, Brohus, Frier, & Heiselberg, 2002)



	N	30	60	Ø	120	150	S	210	240	V	300	330	Ialt
%	4.3	4.2	5.1	6.9	8.4	7.1	8.7	9.9	12.5	14.2	12.3	6.0	99.7
% 0.2-5.0m/s	2.8	2.9	3.1	4.2	4.7	4.5	5.0	5.1	4.7	5.9	5.3	2.8	51.2
% 5.0-11.0m/s	1.4	1.2	1.9	2.7	3.6	2.6	3.5	4.4	6.6	7.3	6.5	2.8	44.6
% > 11.0m/s	0.0	0.0	0.1	0.0	0.1	0.0	0.2	0.4	1.3	1.1	0.4	0.4	4.0
Middel hastighed	4.5	4.3	4.7	4.6	5.0	4.5	4.9	5.3	6.4	6.1	5.7	5.8	5.3
Største hastighed	15.2	13.8	15.7	15.0	13.6	12.5	16.4	18.2	24.0	23.1	25.0	22.3	25.0

Totalt antal observationer = 87234 Kilde: DMI
Vindstille defineret som hastighed <= 0.2m/s
Antal observationer med vindstille/varierende vind: 223 = 0.3%

Table 4: Weather data at Streuer, Denmark. (Hendriksen, Brohus, Frier, & Heiselberg, 2002)

The ventilation system of the building is based on a stack ventilation system with a mechanical fan as backup. The stack used in the ventilation system is located in the middle of the building and acts as the stairwell for the connections between the floors. The air inlet is regulated by computer system (BEMS system) which opens the inlets at floor level of the north façade. The air will enter the building through small windows on the north façade and goes through a small grille on the floor near the façade. This way the air comes directly from the outside to the inside of the building. The windows in the south façade will be open-able for additional ventilation during the summer. The exhaust of the stairwell also uses a special design in order to use the driving forces of the wind to extract air from the building. If these driving forces are not sufficient for the fresh air supply an additional fan is used to create a lower pressure in the stairwell to extract the air from the offices. During the winter months the rooms are heated by radiators fuelled by a combined heat and power plant.

During a one year monitoring project the ventilation rates, CO2 concentrations and temperatures were recorded. These data were analysed in order to check the designed ventilation system with the reality. One of the results of this study was that the system worked perfectly during the summer months but had complications during the winter months. During the winter months the occupants experienced a cold draught from the north façade as well as a lack of ventilation in some of the rooms due to their orientation. Also the noise produced by the assisting fans was unacceptable. These fans also directly started when one of the sensors detected

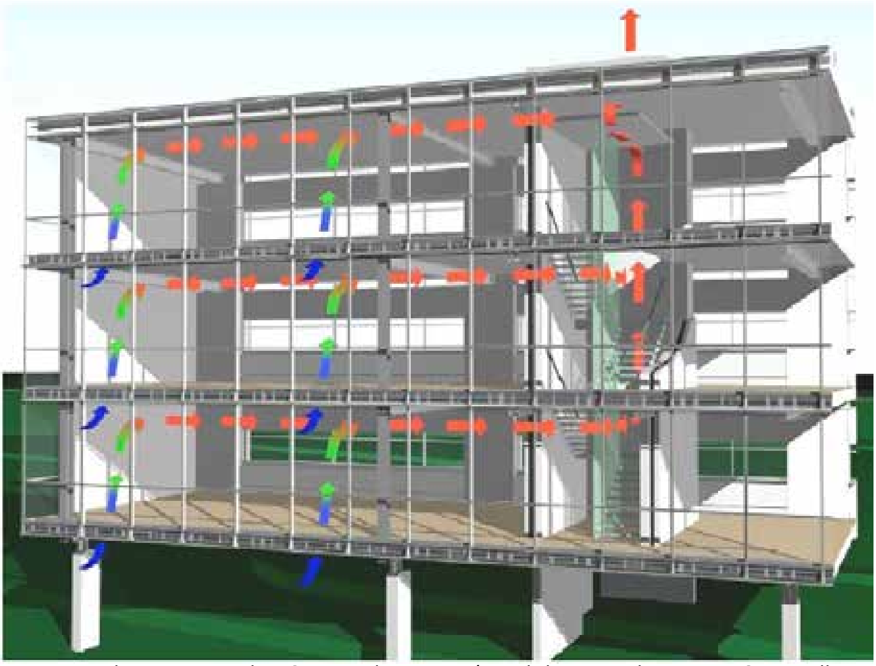


Figure 32: Climate principle B&O Headquarter. (Hendriksen, Brohus, Frier, & Heiselberg, 2002).Air enters the building at grilles in the facade and floor. The air is extracted in the stairwell as a result of the stack effect. At the top of the staircase is a mechanical ventilation to increase the airflow if the natural principles are not sufficient

an insufficient air flow. As the system was not calibrated correctly it did not wait a couple of minutes before starting up. Due to this the fans started more often than calculated. Also due to the large glass façade on the north façade and the inlet of cold outdoor air the energy performance regarding heating was 42% higher than accepted by the Danish building code.

	Percentage of monitoring period (8760 hours)
Hybrid ventilation potentially overruled due to low outdoor temperatures (<5°C) Adjusted during monitoring period	25%
Hybrid ventilation potentially overruled due to strong wind (>11 m/s) Adjusted during monitoring period	1%
Hybrid ventilation running in night cooling mode	4%

Control constraints of the hybrid ventilation system and period of night cooling mode

	Hours with temperatures above 26°C during the working period	Hours with temperatures above 27°C during the working period
DS 474	100	25
1st Floor	90	26
2nd Floor	88	27
3rd Floor	226	100
Outdoor air	17	14
Outdoor air according to Danish Design Reference Year (CPH.DRY)	21	13

Table 5: Temperatures exceeding the set standards. (Hendriksen, Brohus, Frier, & Heiselberg, 2002)

During the summer periods the ventilation also cooled down the building during the daytime in combination with night ventilation when there was no-one in the building. As a result of the neutral pressure plane in the building the third floor did not experience enough pressure difference in order to ventilate correctly. This resulted in exceeding the Danish 474 standard, regarding the amount of hours with temperatures above the 26°C and the 27°C, by respectively 126 and 75 hours.

Figure 33: Grilles in the floor for the air inlet and the regulation of the airflow. (Hendriksen, Brohus, Frier, & Heiselberg, 2002)

Conclusion

The B&O building makes perfect use of the hybrid ventilation system regarding the need for fresh air. If the heating system is also included, it preforms poorly, due to the need for additional radiators for heating. During the design the pressure difference between the three floors was not correctly taken into account resulting in exceeding the Danish standards.

3.5 Sketch Design

The first sketch design is a combination of the vertical ventilation system from ClimaRad and the heat exchanger from Smartbox. The vertical ventilation system allows the air to be introduced in the office at the floor or at the ceiling without creating an air flow at working level (1.2 m). The vertical shaft contains a large vertical flat plate heat exchanger. Extruded air from the office will flow through this heat exchanger to preheat the fresh outdoor air for the office. At the top and the bottom of the system are two valves. These valves can be turned manually to bypass the heat exchanger. Air will enter the office directly without going through the heat exchanger.

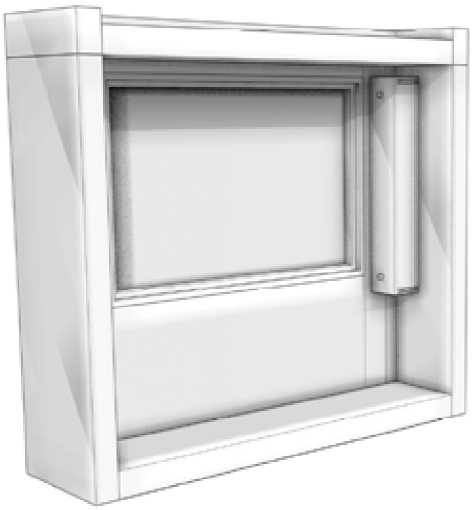


Figure 34: Inside view of the facade with the decentralized ventilation system in the corner

The inlet and exhaust on the façade are designed to make optimal use of the wind pressures on the façade. The exhaust at the bottom will act as a wind cowl. This cowl creates a lower air pressure inside the exhaust and extracting the air from the room. The grille used at the top is a grille with vertical plates which are adjustable. The occupants can change the airflow by changing the position of these plates and letting more fresh air enter the building.

When the wind is blowing from the wrong angle or is not strong enough to create a sufficient airflow in the office the mechanical system starts. In the exhaust and the inlet pipe there are two fans. These fans will boost the airflow making use of the same principles.

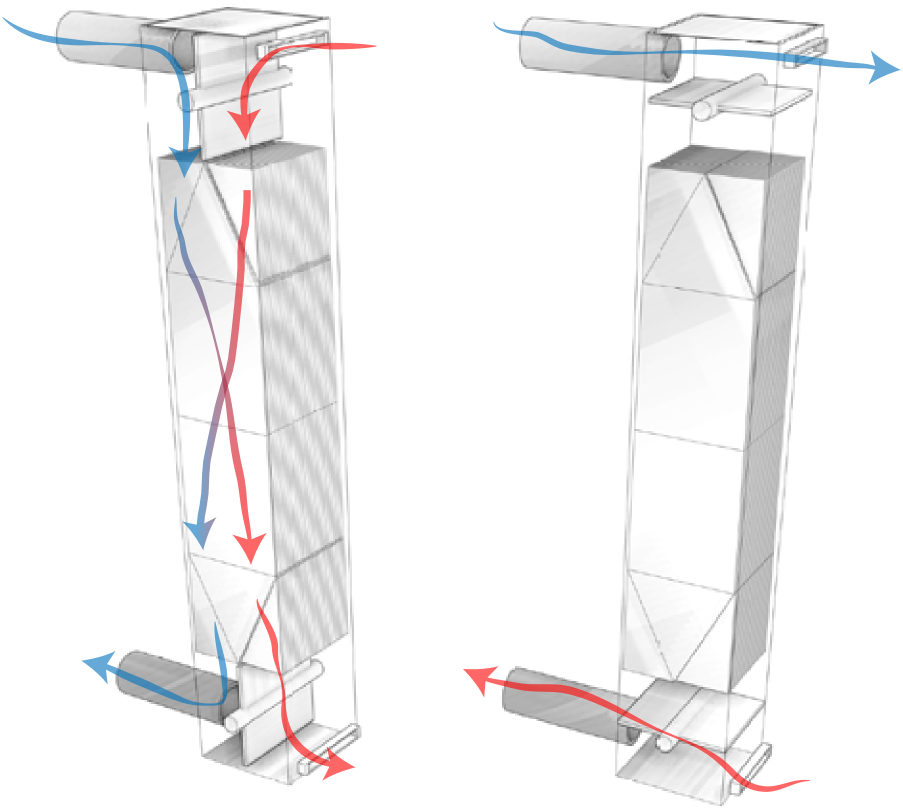


Figure 35: Open view of the ventilation system with the air flow during the winter (left) and summer (right)

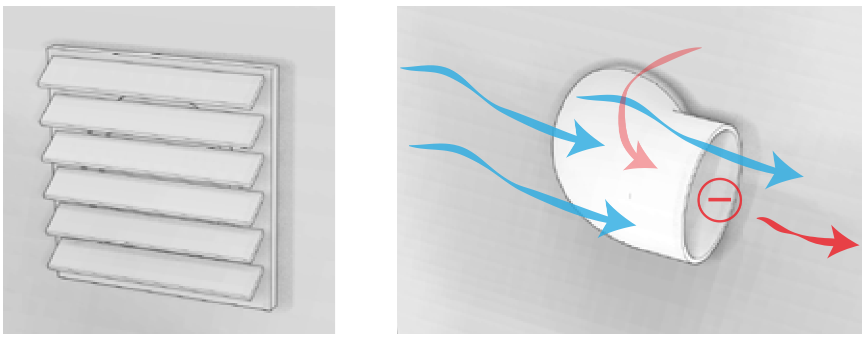


Figure 36: Facade openings; inlet (left), exhoust with principle (right)

4 Simulations

By using manual calculations and CFD-simulations the air flow around buildings and proposed sketch design will be tested. In the first chapter the cooling load and minimal ventilation will be calculated. In chapters 4.2 until 4.6 the airflow around a building will be simulated and compared with the literature. In the last chapter the possibilities of the inlet and outlet will be tested using the CFD program.

In order to understand the velocities and pressure differences in for the natural ventilation system manual calculations will be made. These calculations will be checked later on with a fluid dynamic model in the Ansys fluent program.

4.1 Manual Calculation

Internal heat load

The internal heat load of the office will increase the temperature in the office. Due to this rise in temperature the pressure difference between the floor and ceiling area will also increase due to buoyancy. In order to calculate the rise in temperature the internal heat load needs to be calculated. According to Hosni and Wilkins the average for heat gain will be 0.81 W/ft2 or 8.683 W/m2 (Hosni & Wilkins, 2000). Load factor for this average is 2, heavy, regarding to 1 workstation for every 5.75 m2 or 63 ft2. This data is based on a multiple researches from before the year 2000 (Hosni, Jones, Sipes, & Xu, 1996; Wilkins & McGaffin, 1994). However, the power consumption and the heat load of pc became larger during the following years(Duška, Lukes, Barták, Drkal, & Hensen, 2007). The heat load of computers for example increased from 65 W to 145 W. This is for example a high-end computer which will hardly be used in any standard office. So the value used in the calculations will be set at 70W. Also the heat loads for the computer screens (15 W) and the desktop printer (30 W) were retrieved from Hosni and Wilkins.

Heat load interior							
Occupants		#				total	
	People	4	#	100	W	400	W
Office equipment							
	computers	4	#	70	W	280	W
	Computer screen	8	#	15	W	120	W
	Printer	1	#	30	W	30	W
Lighting							
	fixtures	21.6	m^2	4	W/m^2	86.4	W
					Total (ϕ_i)	916.4	W

Table 6: Internal heat load.

The total internal heat load for the office (ϕi) is 916.4 W. This value is excluding the heat gained by solar radiation. This value will also not be used, because of the boundary condition that there will be sun shading in front of the window.

$$\phi_{loss} = A * U * \Delta T \quad [15]$$

$$\phi_{res} = \phi_i - \phi_{loss} \quad [16]$$

Heat Loss

The Internal heat load doesn't have to be fully cooled down by the ventilation system. Some of the energy will be lost by the walls, floors and windows. In order to calculate the heat loss the following formula (formula [15]) will be used. This formula uses the wall area, the heat transfer coefficient or "U-value" of the cross section is gathered from (CIBSE, 2004). For the temperature difference between the in- and outside the design indoor temperature is set at 20 °C, the temperature in the adjacent rooms and hallway 15 °C and the outside temperature -5 °C. The combination of the heat losses through the walls is 739.5 W (ϕ_{wall}).

Heat loss											
						T outside	Design T				
						-5	20	°C			
		A		U		dT					
Wall	interior	18	m2	1.72	W/m2K	5	K			154.8	W
	interior	18	m2	1.72	W/m2K	5	K			154.8	W
	interior back	10.8	m2	1.72	W/m2K	5	K			92.9	W
	outside	4.95	m2	0.25	W/m2K	25	K			30.9	W
	outside window	5.85	m2	1	W/m2K	25	K			146.3	W
	floor	21.6	m3	0.74	W/m2K	5	K			79.9	W
	ceiling	21.6	m4	0.74	W/m2K	5	K			79.9	W
								Total (ϕ_{loss})		739.5	W

Table 7: Heat loss by walls, floors and ceiling.

Cooling by ventilation

In order to cool down the resulting heat load in the office (ϕres) ,176.9 W (formula [16]), the inlet temperature of the fresh air needs to be cooled down below the design temperature of 20 °C. The needed temperature difference can be calculated by formula [17]. The amount of fresh air is a result of the minimum amount of air needed in an office. Dutch regulations state that in an office environment, each employee needs a minimum of 25 m^3/h. This results for the used office with 4 people in a minimum fresh air supply Q_{vent} of 100 m^3/h or 0.028 m^3/s. According to formula [17] this will result in a temperature difference of 5 degrees. This temperature difference can be achived with an inlet temperature of 17.5 °C and an outlet temperature of 22.5 °C.

$$\Delta T = \frac{\phi_{res}}{Q * \rho * c} \qquad [17]$$

	Total heat load (ϕ_{res})			176.9	W				
						T in	T out		
	Temperature diff inlet/exhoust			5	K	17.5	22.5		
	Cooling by ventilation			0.027	m3/s				
				99.0	m3/h				
			Fresh air					Total	
Minimum ventilation	People	4 #	25	m3/h		0.007	m3/s	0.028	m3/s
		w (m)	h (m)	A (m2)		Amount			
	Inlet	0.15	0.15	0.0225	m2	1	#		
	Determining air need			0.028	m3/s				
	speed			1.23	m/s				

Table 8: Cooling by ventilation.

Pressure resistance of the system

Another pressure resistance in the system next to the heat exchanger are the ducts themselves. Due to the surface roughness of the ducts and the turbulent resistance of the bends, a larger presser is needed in the system. The total pressure loss due to the roughness and the bends can be calculated with the D'Arcy-Weisbach equation [18](Recknagel, Sprenger, & Schramek, 1995). This

$$\Delta P = \lambda * \frac{l_{rep}}{d} * \frac{\rho}{2} * v^2 \qquad [18]$$

formula uses a friction factor (λ) and the representative length of all the ducts (l_{rep}). The friction factor can be derived from the Reynolds number [19] and the Moody-diagram. The designed system has a Reynolds number of 10328. Another

$$\mathrm{Re} = \frac{\rho * v * \mathrm{d}_h}{\mu} \qquad [19]$$

parameter needed for the moody diagram is the relative pipe roughness. This parameter gives the relation between the roughness length of the surface and the

$$r = \frac{\varepsilon}{d} \qquad [20]$$

diameter of the duct [20]. This dimensionless value is 2.5e-5 for the system with an inside diameter of 0.1 m and a surface roughness of 2.5e-6 m (table 9). If the Reynolds number and the relative pipe roughness are used in the Moody-diagram it results in a friction factor (λ) of 0.018 (figure 37).

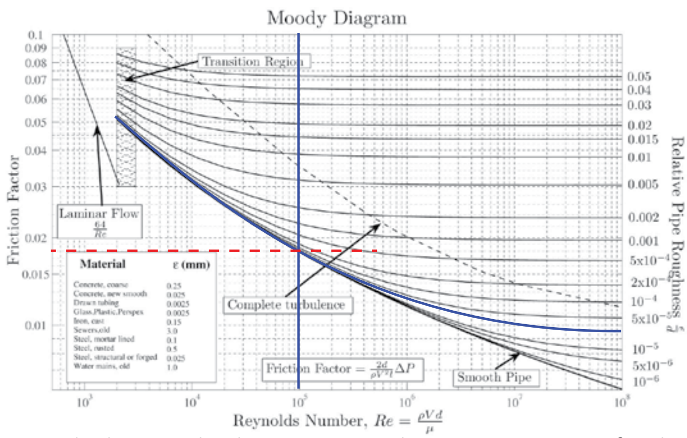


Figure 37: Moody digram. Blue lines represent the input parameters for the ventilation system.

For the calculation of the representative length of the ducts, all the bends have to be taken into account. With the use of the pressure resistance of all the seperate elements (VDL Klima BV, 2003), an equivalent length can be calculated [21-22]. As can been seen in table 10 the total pressure resistance coefficient of the system is 2.76. If this is implemented in formula [21], the equivalent length of the curves, inlets and outlets is 15.33 m resulting in a total representative length of 17.33 m. If these values are implemented in the D'Arcy-Weisbach equation [18], the total pressure loss of the system will be 3.71 Pa. This is the amount of pressure that is needed in the beginning of the system in order to overcome the pressure resistance of all the separate elements.

$$l_{rep} = l + l_{eq} \qquad [21]$$

$$l_{eq} = \zeta * d / \lambda \qquad [22]$$

inside diameter	d	0.1	m		name	#
air velocity	v	1.4	m/s		curve	
surface roughness	ε	2.5E-06	m		Inlet	
density	ρ	1.215	kg/m3			
dynamic viscosity air	μ	1.65E-05	Pas		total	
Reynolds number	Re	10328	-			
relative pipe roughness	r	2.5E-05	-			
friction factor	λ	0.018	Moody-diagram			

Table 9: Determining the input parameters for the friction factor using the Moody-diagram.

Individual pressure resistance				
name	#	ζ	ζ total	
curve r/d = 1	5	0.35	1.75	
Inlet office	1	1	1	
Outlet office (α=0.9)	1	0.01	0.01	
total			2.76	
l	2.00	m		
leq (ζ*d/λ)	15.33	m		
l rep	17.33	m		
D'Arcy-Weisbach equation				
total pressure loss	3.71	Pa		

Table 10: Total pressure resistance of the system.

4.2 Ansys Modeling

Ansys fluent is a computational fluid design (CFD) program for calculating and visualizing the airflow around a building. By using Ansys, the manual calculations will be checked in order to verify the results. Ansys is also able to give a more detailed picture of the airflows at the leeward side of the building.

Wind direction

According to the manual calculations the wind speeds at the leeward side of the building will be a factor 0.4 of the main wind velocity. This is only when the wind is perpendicular to the front façade. By using Ansys the wind speeds at 3 other directions were calculated. Not only at the leeward side of the building but also at the windward side of the building. The directions used in the calculations are at angles of 0 (perpendicular to facade), 30, 60 and 90 (parallel to façade) degrees. In order to do a preliminary check about the data the velocities at an angle of 0 degrees calculated by Ansys will be compared to the velocity factors given by Stathopoulos and Baskaran (Stathopoulos & Baskaran, 1996). These values were about the same so it is assumed that the model input was correct and results for the other 3 wind speeds will be interpreted as correct.

4.3 Input

Model

For the calculation in Ansys a three dimensional model is created. This model is made from the combination of an 'environment box' and a building. The box for the environment will be used to create the three-dimensional mesh and to create the planes for the inlet and outlet of the air. It represents a sort of wind tunnel for the building. The six planes of this box will have 1 solid face (ground), 1 inlet face (wind inlet) and 4 outlet planes. The outlet planes allow the program to create an open environment for the wind calculated. The building in the middle of the environment box has the same dimensions as given in chapter 1.4, 75 x 16 x 30 m. The building is placed parallel to the inlet face firstly in order to simulate the wind blowing at 0 degree angle onto the façade. For the other 3 calculations the building is rotated around its centre point in steps of 30 degrees.

Mesh

For the Ansys mesh a triangulated mesh is used. The mesh size will be larger at the faces of the environment box (mesh size of 5 m) and will be very fine (mesh size of 0.15 m) at the façade for the calculation. So in total 8 different types of models are created for the calculation of the 4 angles and the 2 facades.

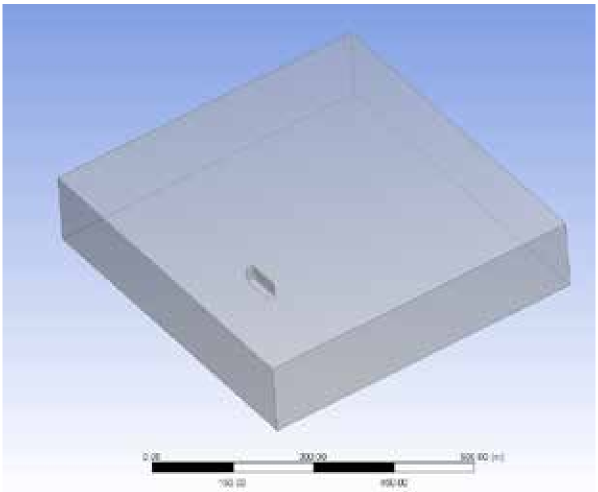


Figure 38: Ansys geometry input

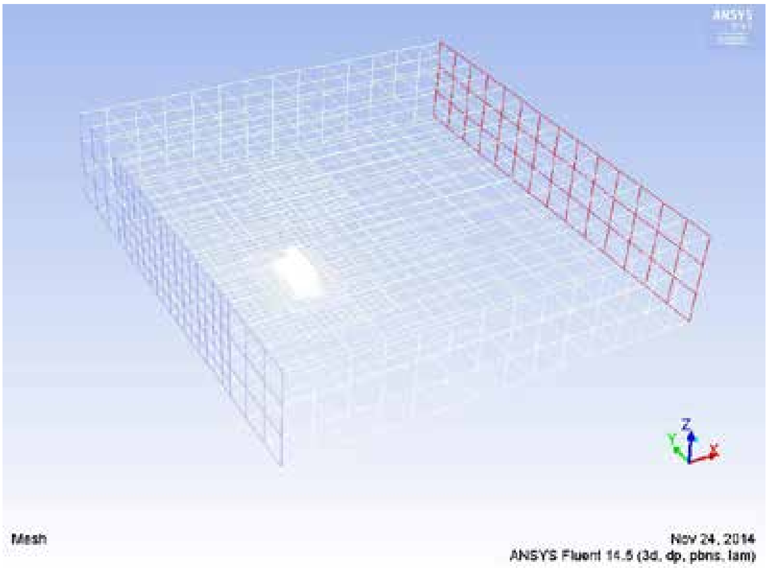


Figure 39: Ansys Mesh input with the higher mesh density near the building. Inlet (blue), Outlet (red) and solid (white).

Model setup data

For the calculation the boundary condition for the inlet of the environment box will be set to a ‘velocity inlet’ with a wind speed of 6 m/s in the x-direction. The 4 outlet faces will be set to ‘pressure outlet’ with no additional settings so the only force factor will be the inlet.

Results

For the collected data 5 lines were placed perpendicular to the façade with a length of 1m at heights of 5, 10, 15, 20 and 25 m. The values measured at these lines will be used for the further research. Also wind direction vectors will be plotted in order to compare the calculated directions with the ones given in the literature (chapter 2.3). Also contour plots will be used to visualize the velocities and dynamic pressures.

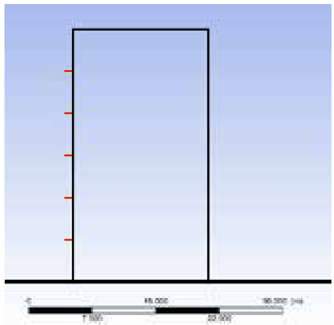


Figure 40: Measurement lines at heights of 5, 10, 15, 20 and 25 m.

4.4 Ansys Results

Interpretation of the data

If the data from Ansys are compared to the literature the values at 1 m from the façade will have no large differences. In the graphs created at the 5 lines at the façade (see figure 41) the wind speed at the façade is 0. This is a result of the boundary conditions of the façade as at the solid plane of the building there is

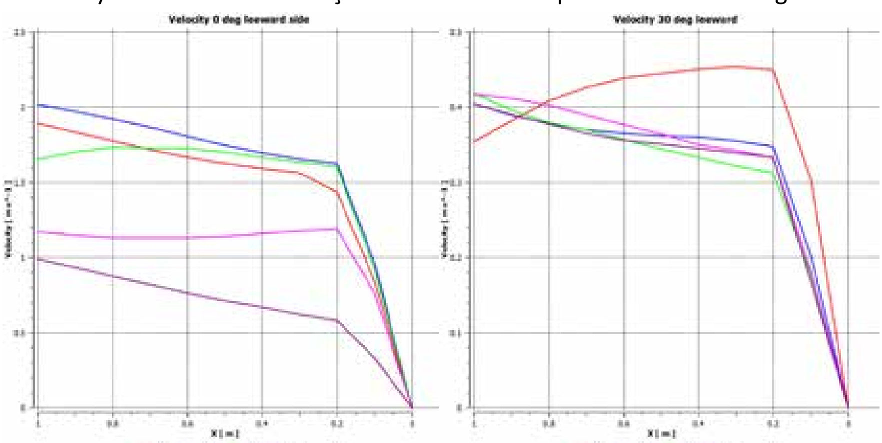


Figure 41: Wind speed results of different heights at the leeward side of the building. The x-value represents the distance perpendicular to the facade. Left wind at 0 degrees, right at 30 degrees.

no possible wind movement. And due to the size of the mesh at the façade, the first 10-15 cm will not be used for the data due to this boundary condition. As can been seen in figure 41, there is a dynamic pressure at a distance of 0m from the façade. This is not possible as result of the velocity at that point being 0 m/s. When looking at the formula for the dynamic pressure used by Ansys, formula [18], they also take into account the static pressure. This is only used at the points where the velocity is 0, at the facades, because the other values for the dynamic pressure can be calculated with the standard Bernoulli equation for dynamic pressure (formula [19]) (ANSYS Academic Research, 2014).

$$P_{dyn} = P_{stat} + \frac{1}{2} * \rho * v^2 \quad [23]$$

$$P = \frac{1}{2} * \rho * v^2 \quad [24]$$

Leeward side

The dynamic pressures in the middle of the building as a result of the different angles are different. At an angle of 30 and 60 degrees the dynamic pressures are about the same and will increase when further away from the building. There is an exception at 25m at the angle of 30 degrees. This is a result of the diagonal vortex created by the wind blowing over the roof. The highest wind speeds and dynamic pressures at the leeward side of the building are generated at an angle of 0 degrees. The highest dynamic pressures are measured in the upper half of the building. Directly at the façade the dynamic pressure will start at about 1.4 Pa and then rise towards 2.3 Pa at 1m distance from the façade. At the angles other than 0 degree, the dynamic pressures are very low due to the low wind speeds in the boundary layer created by the angle of the building. This can also be seen when the wind is parallel with the building (90 degree). The boundary layer and vortexes created by the flat frontal surface of the building result in very low wind speeds and dynamic pressures. Even at a distance of 1 meter from the façade the wind speeds are relatively low, 0.4 m/s.

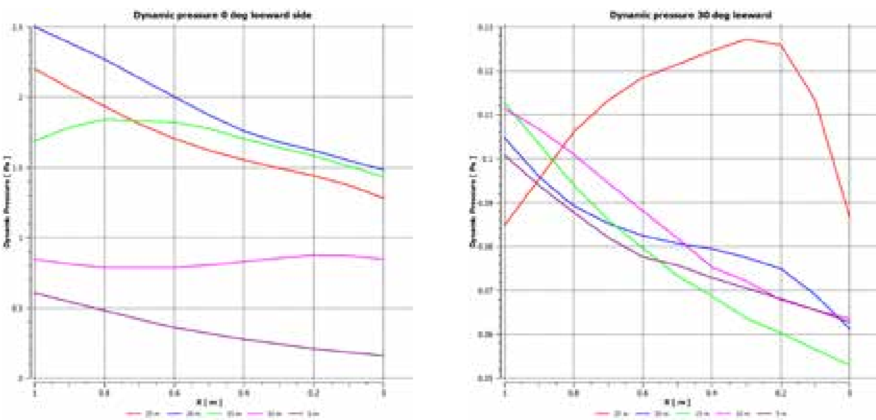


Figure 42: Dynamic pressures at different heights at the leeward side of the building. The x-value represents the distance perpendicular to the facade. Left at 0 degrees, right at 30 degrees.

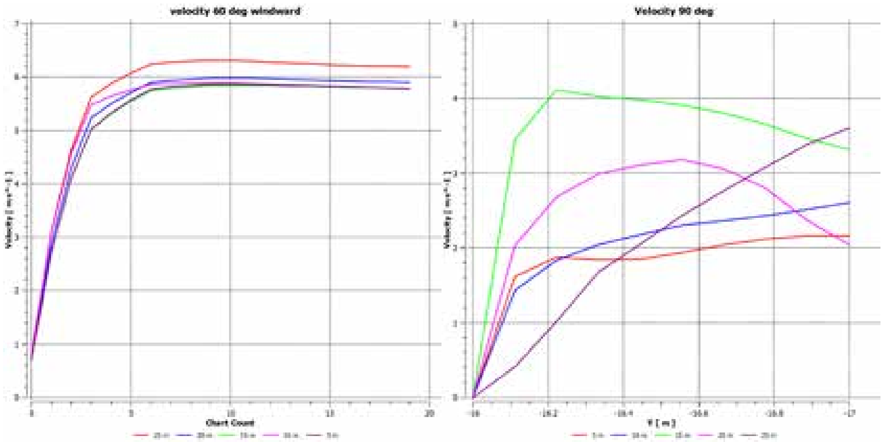


Figure 43: Velocities at different heights at the leeward side of the building. The x-value represents the distance perpendicular to the facade. Left at 60 degrees, right at 90 degrees.

Windward side

The wind speeds at the windward side are the opposite of the wind speeds at the leeward side of the building. As expected the highest wind speeds at an angle of 0 degrees are calculated at the top of the building as a result of the wind being forced over the building at the top (Bottema, 1993). These wind speeds can reach values around 3.4 m/s at a 6 m/s main wind speed. These are the top wind speeds at a height of 25 m. At the other heights, the wind speeds are relatively low due to the effect of the frontal vortex, as this creates low wind speeds at the facades of the building. Wind speeds in those areas are between 1 and 2 m/s. The highest

[Data]		Leeward side				Windward side			
		0 deg	30 deg	60 deg	90 deg	0 deg	30 deg	60 deg	90 deg
X [m]	Z [m]	0 Velocity Velocity [m/s]				Velocity [m/s]			
0.00	25.00	1.45	0.38	0.26	0.09	3.25	3.00	1.84	0.09
0.10	25.00	1.50	0.43	0.26	0.09	3.27	3.08	1.88	0.09
0.20	25.00	1.54	0.46	0.27	0.10	3.29	3.18	2.06	0.10
0.30	25.00	1.57	0.46	0.28	0.12	3.34	3.25	2.18	0.12
0.40	25.00	1.60	0.45	0.30	0.15	3.38	3.31	2.27	0.15
0.50	25.00	1.63	0.45	0.32	0.18	3.42	3.37	2.39	0.18
0.60	25.00	1.67	0.44	0.34	0.21	3.46	3.45	2.52	0.21
0.70	25.00	1.73	0.43	0.35	0.23	3.46	3.54	2.64	0.23
0.80	25.00	1.79	0.42	0.37	0.24	3.44	3.63	2.77	0.24
0.90	25.00	1.85	0.40	0.39	0.23	3.40	3.71	2.89	0.23
1.00	25.00	1.90	0.37	0.41	0.20	3.34	3.77	3.01	0.20

[Data]		0 deg	30 deg	60 deg	90 deg	0 deg	30 deg	60 deg	90 deg
X [m]	Z [m]	0 Velocity Velocity [m/s]				Velocity [m/s]			
0.00	20.00	1.56	0.32	0.26	0.17	1.97	2.53	1.66	0.17
0.10	20.00	1.60	0.34	0.27	0.17	1.98	2.55	1.72	0.17
0.20	20.00	1.63	0.35	0.29	0.17	1.99	2.59	1.86	0.17
0.30	20.00	1.66	0.36	0.31	0.16	2.00	2.65	1.98	0.16
0.40	20.00	1.70	0.36	0.33	0.17	2.02	2.71	2.08	0.17
0.50	20.00	1.75	0.36	0.34	0.20	2.05	2.80	2.19	0.20
0.60	20.00	1.81	0.37	0.36	0.22	2.09	2.88	2.30	0.22
0.70	20.00	1.88	0.37	0.37	0.24	2.13	2.97	2.42	0.24
0.80	20.00	1.93	0.38	0.38	0.26	2.19	3.05	2.56	0.26
0.90	20.00	1.98	0.40	0.40	0.28	2.22	3.10	2.69	0.28
1.00	20.00	2.03	0.42	0.41	0.28	2.23	3.14	2.82	0.28

Table 11: Wind speed dataset from Ansys.

wind speeds at the windward façade can be found if the wind is at an angle of 30 degrees. At this angle the wind speeds at the entire façade are around the 2.5 m/s with again top speeds at 25m of 3.7 m/s. If the wind speeds will be converted to dynamic pressures, it will vary between 0 and 7.28 Pa at 0 degrees and 2.33 and 8.64 Pa at 30 degrees.

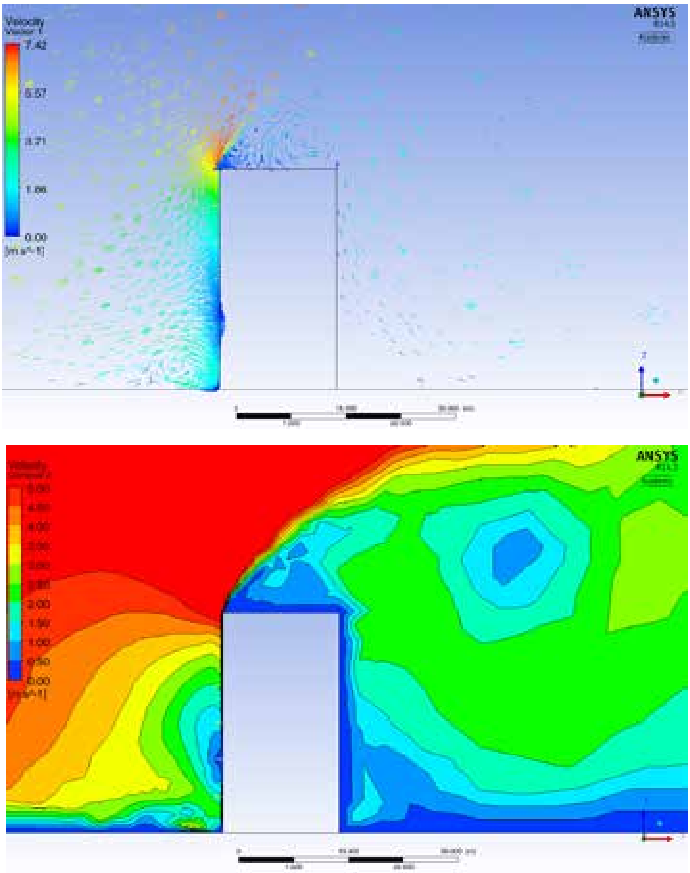


Figure 44: Velocity vector (top) and the dynamic pressure (bottom) of the wind at 0 degree at the windward facade.

4.5 Comparison

If the calculated data around the building are compared to the expected values regarding the correction factors as stated by Stathopoulos and Baskaran (chapter 8), there are no large differences. The values at the façade though, are a lot lower than the predicted values. This can be a result of the surface roughness. The measured values at the facade can be used as a guideline as the surface roughness and so the boundary layer may vary for each building. The values measured at the facades can be directly used for the pressure and velocity created at the inlet of the ventilation system and with the correction factor, stated in chapter 2.7, for the outlet.

4.6 Result Analysis

When looking at the data calculated by the Ansys models, it can be noticed that the wind speeds at the leeward side of the building do not create high enough dynamic pressures to overcome the pressure created by the heat exchanger. In an optimal inlet and outlet cowl dynamic pressures can be doubled and will result in values lower than the 4 Pa pressure loss of the heat exchanger. As a result of this it can be stated that complete natural ventilation by openings at the façade at the leeward side of the building cannot be created. In this case a mechanical fan will be needed in order to create the needed airflow for ventilation. If the wind direction during the colder months, sept – april, will be taken into account (figure 45), one side of the façade will need permanent mechanical ventilation due to the wind direction.

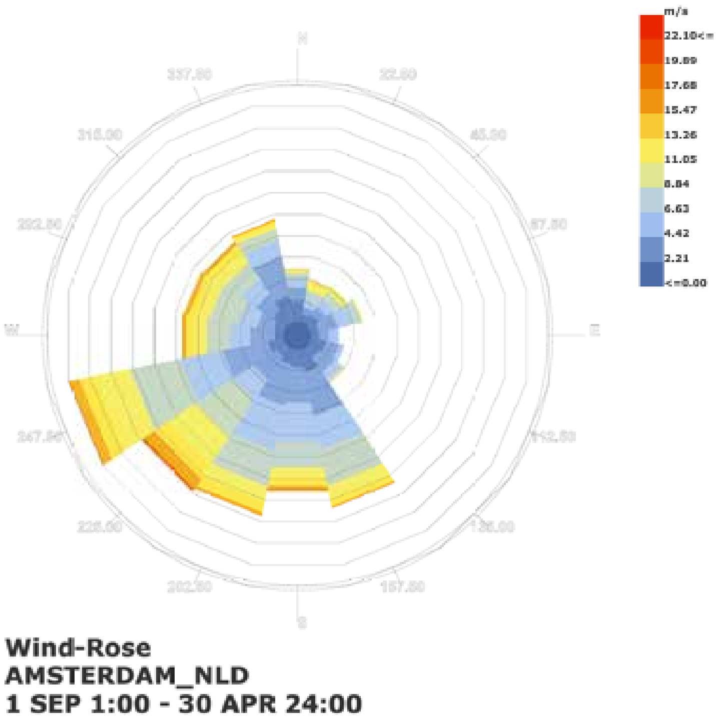


Figure 45: Wind rose for Amsterdam in the period of 1 september to 30 april.

The windward side of the building (facing west) will be able to make use of the wind speeds created at the facades. For optimal of the high wind speeds at angles of 30 and 60 degrees, the façade should be facing west as most winds during the period of September to April will come from SSW (195°) and SW (225°) direction.

If the wind speed data gathered by the Ansys simulation and the cowl correction factor are combined, the boundary conditions for the windward side outlets of the ventilation system are created. For the inlet a simple curved cured cowl can be used as this will have a correction factor of 0.9 according to de Gids and den Ouden (de Gids & den Ouden, 1986). The outlet of the ventilation system will be a venturi cowl (figure 47) with a correction factor of -0.8. This will generate enough pressure difference to overcome the pressure drop of the heat exchanger. The cowl distance to the façade will be about 0.2 m in order to be outside the boundary layer created by the surface roughness of the building. With these correction factors the minimum wind speed at the façade needs to be 1.86 m/s (formula [13]). But even at the windward façade there will be locations where this value will hardly be reached. These locations are mainly located at the lower levels of the facades.

4.7 In/Outlet Simulations

The inlet and outlet of the ventilation system play an important role. The inlet allows the fresh outdoor air to be guided into the system while the outlet extracts the air from the system. As shown in chapter 2.7 there are several types of systems. For the inlet only 1 system would meet the demands of redirecting the outside air to the system. This wind cowl will be rotated into the wind all time by using a wind vane on the back of the curve. The size of the inlet is of great influence to the total air flow as can been seen in figure 45. In figure 45 the opening which will be turned into, has a diameter of 100mm while figure x has a diameter of 150mm. They both will redirect the wind into the system but the larger opening will do this more efficient. The average wind velocity in the 150 mm inlet pipe is 2.2 m/s compared to an average velocity of 1.3 for the 100 mm diameter.

For the outlet there are several possibilities as can been seen in chapter 2.7. The most effective variant according to the de Gids and den Ouden (de Gids & den

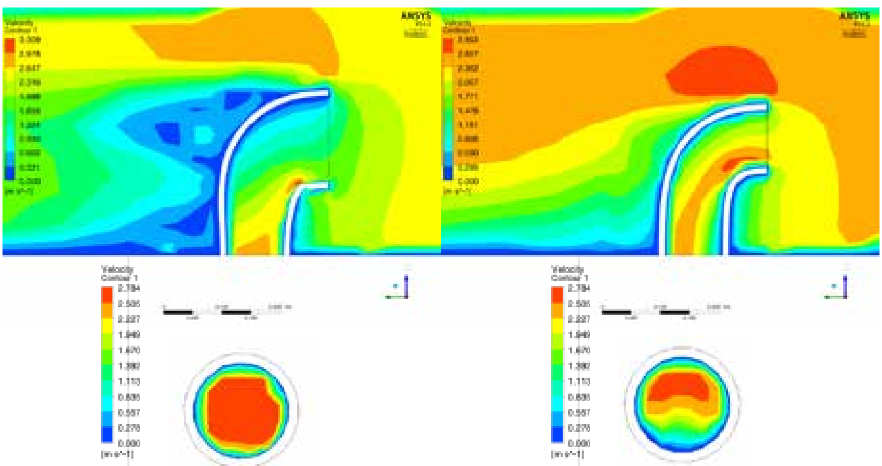


Figure 46: Velocity contour of the 150 mm inlet (left) and the 100 mm inlet (right) with their corresponding cross-section at the base.

Ouden, 1986) would be the venture shaped cowl. Several simulations have been conducted but the values generated by their research could not be reproduced. One of the possibilities is that the 3d venture system has a precise configuration. A difference of a couple of millimetres could result in a large pressure difference. Another possible solution for the outlet is to place a sort of hill on the façade. This hill will increase the wind velocity and create a negative pressure at the outlet, extracting air from the system. Several hill configurations were tested. The main changed parameters were the height and the distance between the outlet and the start of the slope of the hill. However, one of the downsides of the hill-outlet is the lack of wind protection. If the wind would be perpendicular to the façade it is possible that the wind would neutralized the negative pressure created by the parallel winds. This would completely disturb the system. Also the air velocities are only 0.34 m/s with a positive pressure of 0.4 Pa. The standard hill would double the air velocity at the outlet 0.6 m/s and generate a pressure of -0.3 Pa.

In order to simulate the desired system the hill will be used instead of the venturi shaped outlet. This system will still make use of the negative pressure but will not be used in the final design.

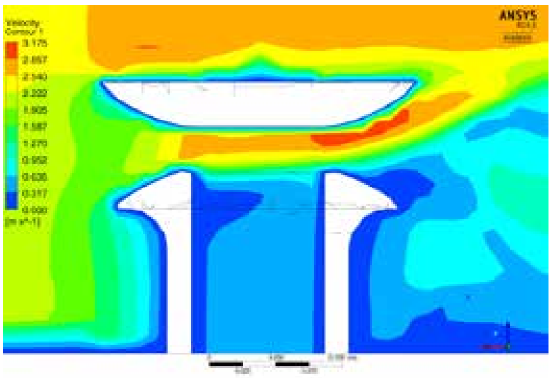


Figure 47: Venturi outlet simulation. Velocity contour.

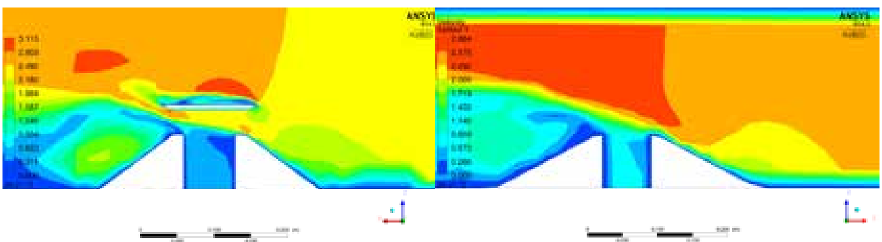


Figure 48: Velocity contour of the hill variant with cap (left) and without cap (right)

5 Final Design

By combining the simulations and the sketch design a final design is created. This design will be explained in chapter 5.1. In chapter 5.2 the final design will be simulated in the CFD program. This chapter will discuss the minimal air velocities as well as the pressure loss over the Fiwihex heat exchanger. A mock-up at scale 1:1 was also placed in a simple wind tunnel and air velocity measurements were conducted. This data will be discussed in chapter 5.3. The last chapter will explain the possibilities of the new ventilation system. Not only discussing the architectural boundary conditions but also the influence of the system with different floor layouts.

5.1 Final Design

Location

In combination with the data collected during the research with the Ansys models and the in/outlet vents the first sketch design was remodelled. It will still be a thin 'box' system which can be mounted next to the window on the inside of the façade. Due the manual control possibilities there will be two systems mounted. Each one creating and preheating an air flow of 50 m3/h. In this design the inlet grid will be at a height of 1m and the exhaust vent will be located just below the ceiling.

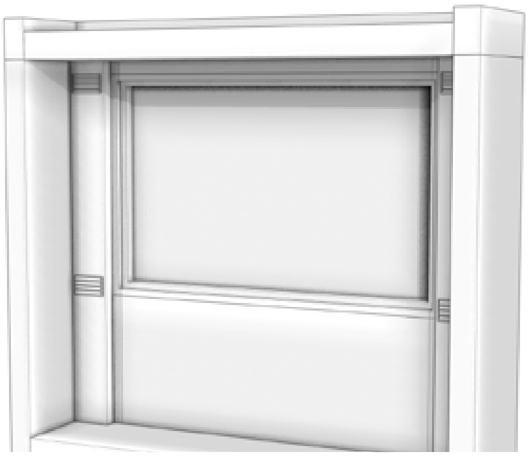


Figure 49: Interior view of the system mounted on both sides of the window.

Figure 50: Outside of the facade with the 2 inlets (top) and outlets (bottom).

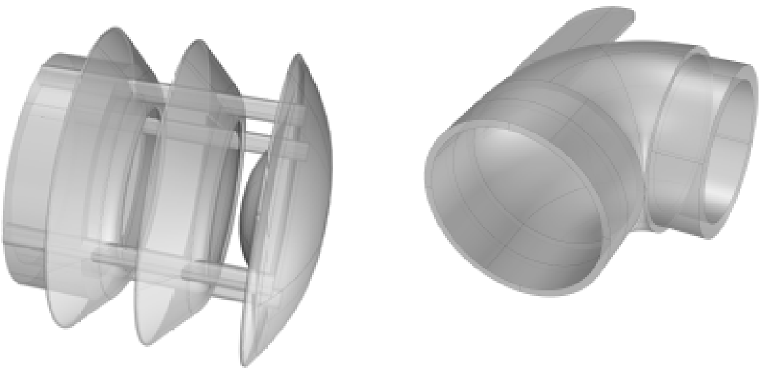


Figure 51: Details of the facade vents. Outlet (top) and the Inlet (bottom).

Façade vents

There will be two different façade systems used in order to let air in and out of the system. The inlet of the system exists of a rotating wind cowl. This system will always have the opening directed into the wind direction by using the metal fin mounted at the back. So at every possible angle the inlet will be turned into the wind. The outlet is a stationary system and is used to create a negative pressure at the outlet of the ventilation system. This element acts as a small 3D venturi element. Due to the shape it will create a negative pressure at any given wind direction at the façade.

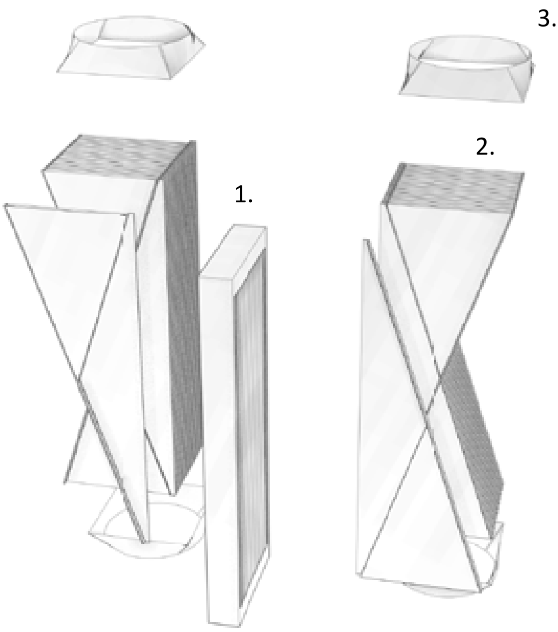


Figure 52: Exploded view of the heat exchanger.
1: Fiwihex unit, 2: flow guidance, 3: pipe entrance

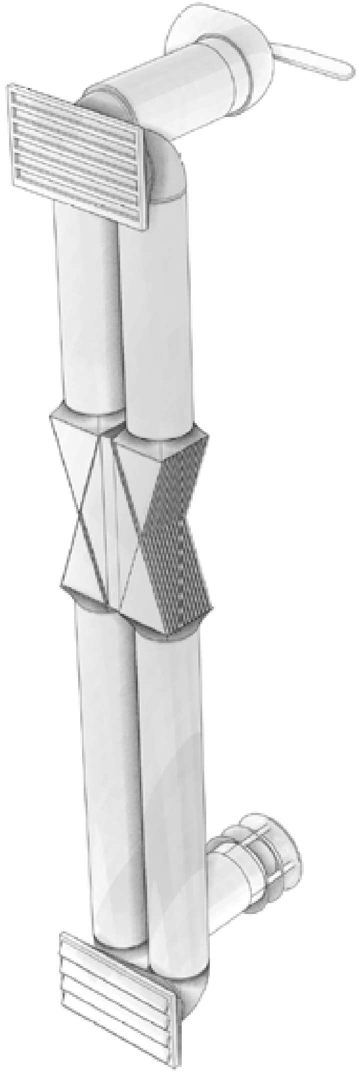


Figure 53: Elements of the ventilation system.

Heat exchanger

The heat exchanger used is the Fiwihex system. This system of copper wires acts like a flat plate heat exchanger with an extended surface made of the copper wires. The fresh air flow and the exhaust air flow will be guided towards this heat exchanger by using circular pipes. The heat exchanger will be placed perpendicular to the façade and due to this the inlet and exhaust pipes will be able to be placed next to each other. As a result of the orientation of the heat exchanger the box which holds all the systems will only be 15 cm thick and 30 cm wide.

Regulation of the air flow

The airflow can be regulated by a slider which narrows the cross-section of inflow pipe. The created smaller section will generate more drag resistance, resulting in a lower dynamic pressure and lower air velocities. This system should also be used if the main wind speed outside is too high and the occupants experience the resulting indoor air flow as a draft e.g. higher than 0.2 m/s. The airflow can also be automatically be regulated by using a self-adjusting inlet vent. This type of vent uses of a membrane which will move into the airflow as a result of the increased air flow. The membrane will decrease the cross-section and reducing the air velocity. The velocity of the air will only increase a bit due to this system rather than keep increasing together with the outdoor air velocity (VLA, 2013).

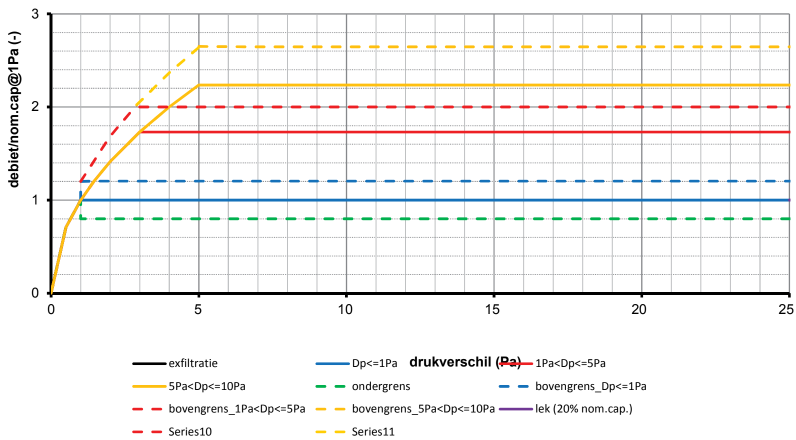


Figure 54: Effect of the selft regulating inlet vents. The air flow will stagnate after a set pressure, in stead of increasing with together with the pressure in the system. (VLA, 2013)

Backup system

The wind will not always be sufficient to create a large enough airflow to ventilate the office. If this occurs, a mechanical backup system is needed. This system consists of 2 small fans integrated in the supply and exhaust of the system. These will generate the airflow if the natural principles does not work sufficient.

5.2 Design Simulation

The final design was divided into two parts; the indoor air flow and the pressure generated by the façade elements.

Façade elements

The façade elements will be tested on the generation of pressure, positive and negative, with a wind velocity at the façade of 2.4 m/s. This corresponds with a distance of about 15 cm at angles of 0, 30 and 60 degrees. As a result of the simulation the inlet element directed almost all the air into the system. There was however a small vortex in the bend of the system but also an increased speed at back of the system. These two values counter act each other, resulting in an average wind velocity in the system of 1.3 m/s.

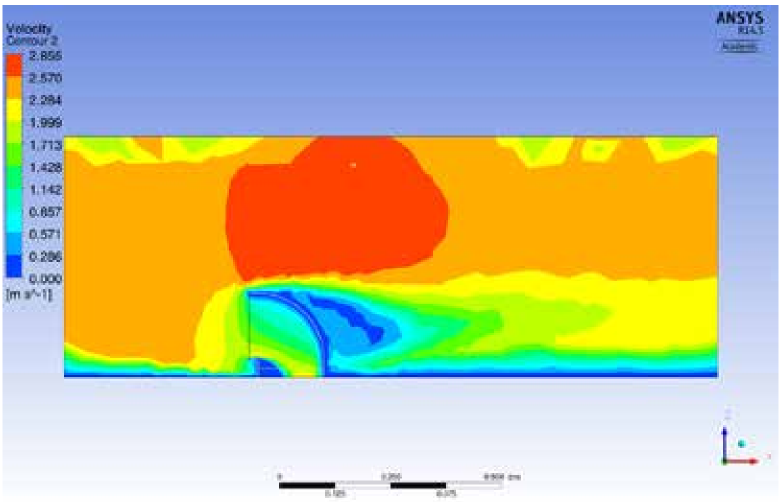


Figure 55: Velocity simulation results at location of the inlet cowl. v_{wind}= 2.0 m/s

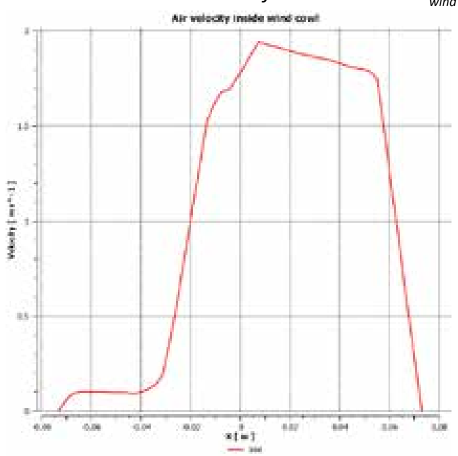


Figure 56: Wind velocities at cross section of the inlet.

The outlet on the façade was simulated as a small hill on the façade. This small hill will push the airflow upwards and over the opening. As a result of this airflow, a negative pressure will be created at the opening and thus extracting the air from the room. The negative pressure by this system will be 0.4 Pa and will help to overcome the pressure drop of the heat exchanger.

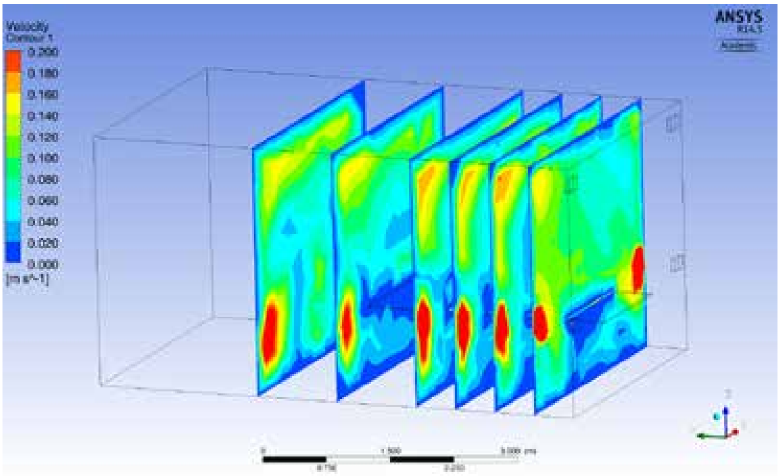


Figure 57: Air velocities throughout the room at an inlet velocity of 0.4 m/s.

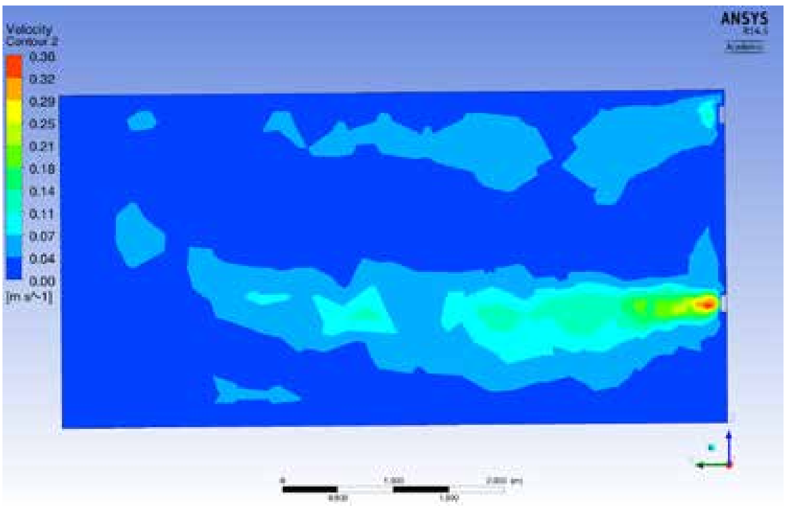


Figure 58: Air velocities at the inlet with inlet velocity of 0.36 m/s.

Interior

For the interior simulation the airflow inside the room was visualized. In order to meet the set boundary condition of a maximum wind velocity of 0.2 m/s. The wind velocity coming from the inlet is set to 0.38 m/s due to the opening area of 0.0375 m^2 resulting in a total air flow of about 50 m^3/h. The heat load of the office and the corresponding airflow due to the temperature difference will be created by the table in the room. Looking at the resulting data, velocities higher than 0.2 m/s will only be located in the first couple of meters of the inlet air flow. At other locations in the office the velocities will be relatively low with speeds lower than 0.04 m/s.

Heat exchanger

As the heat exchanger would create a pressure difference in the system due to its air resistance it had to be modelled correctly. As no specific data were available the input was first tested with the given parameters which state that there will be a pressure drop of 4 Pa at 100 m^3/h. For this simulation a simple tube was created with a 1 cm thick porous zone halfway (figure 59, 60). In order to create a unidirectional zone like in the actual heat exchanger the values which were not parallel to the air flow were set to high numbers. As a result of this the air was only able to flow in the x-direction through the porous zone. The next step was to use a sort of trial-and-error method in order to get a pressure drop of 4 Pa over the 1cm thick porous zone. This data were then used as the input for the heat exchanger in the final design simulation.

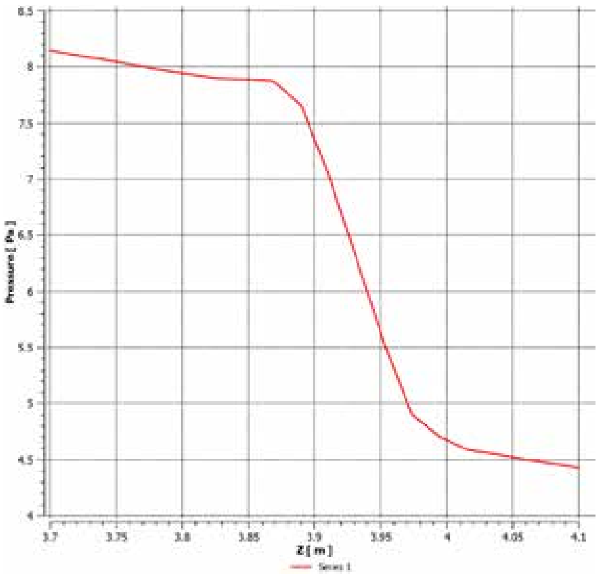


Figure 59: Simulated pressure drop of 4 Pa over the heat exchanger. Q = 100 m^3/h

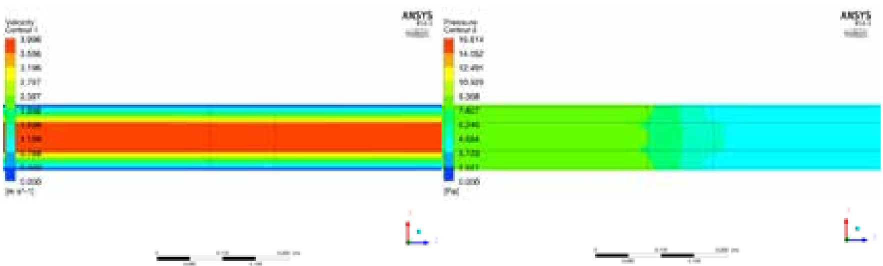


Figure 60: Velocity (left) and pressure (right) in the simulated heat exchanger. Q = 100 m^3/h

Total design simulation

These elements combined resulted in a simulation of the total design. This includes a box for the office , a porous zone for the heat exchanger and a tunnel for creating the wind. Inside the tunnel both façade elements, inlet and outlet, are modeled. The wind velocity in the tunnel was set to the average wind velocity at the façade of 2.4 m/s. This resulted in an air velocity of 1.5 m/s in the system. In combination with the diameter of 0.11m of the tube will result in a total air flow of 50 m^3/h. The pressure drop over the heat exchanger was a lot lower than expected. According to figure 65 which was created in the middle of the tube through the heat exchanger, the pressure drop was about 0.2 Pa.

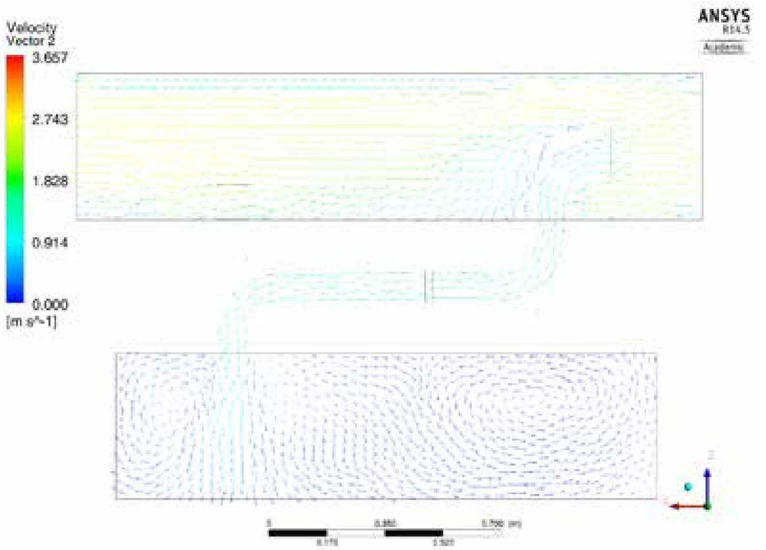


Figure 61: Velocity vectors at a section through the inlet of the ventilation system.

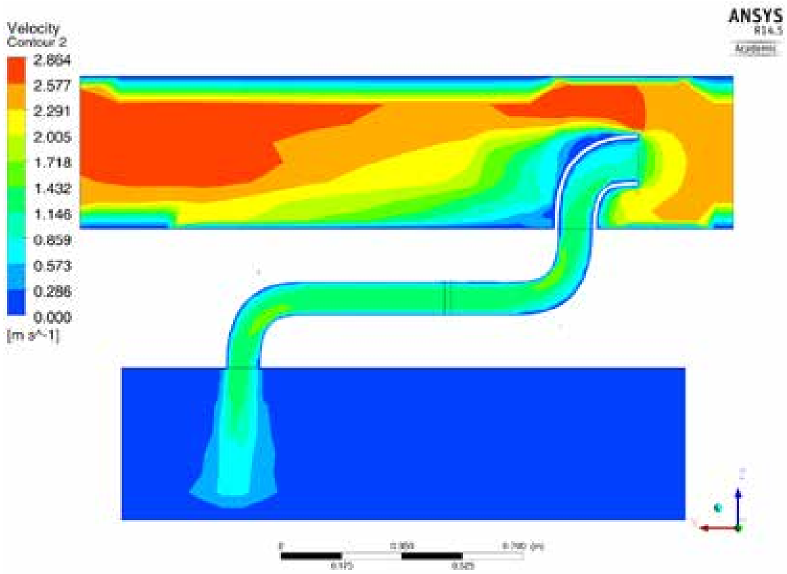


Figure 62: Velocity contour at a section through the inlet of the ventilation system.

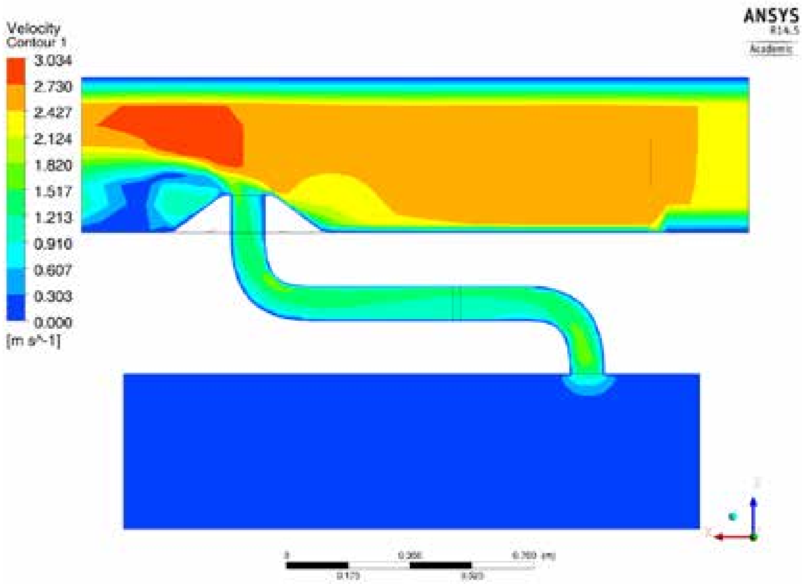


Figure 63: Velocity contour at a section through the outlet of the ventilation system.

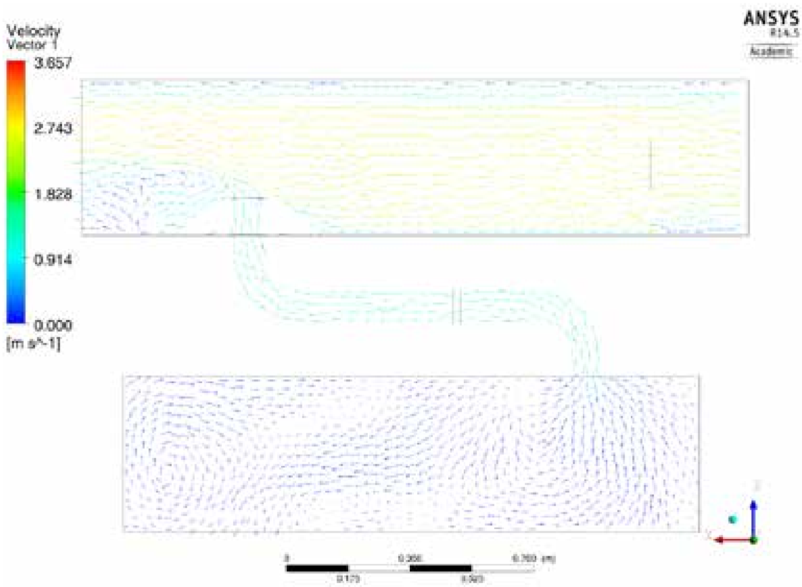


Figure 64: Velocity vectors at a section through the outlet of the ventilation system.

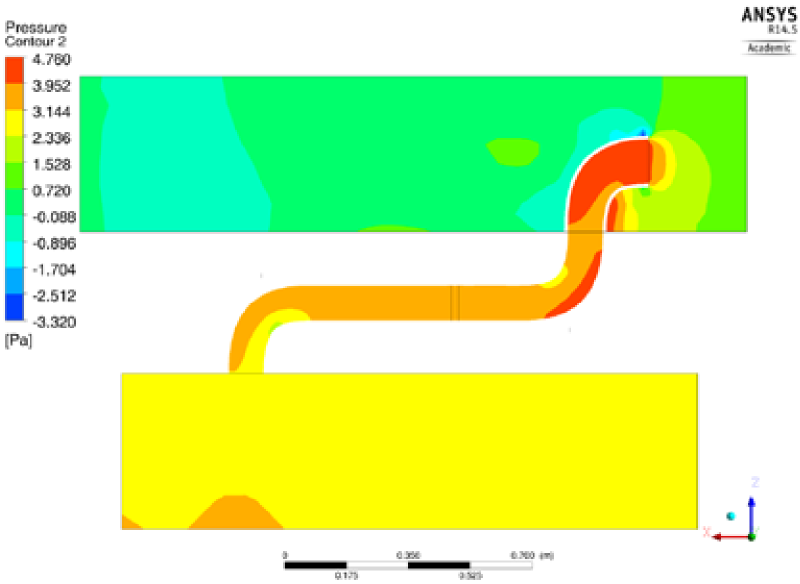


Figure 65: Pressure contours at a section through the inlet of the ventilation system.

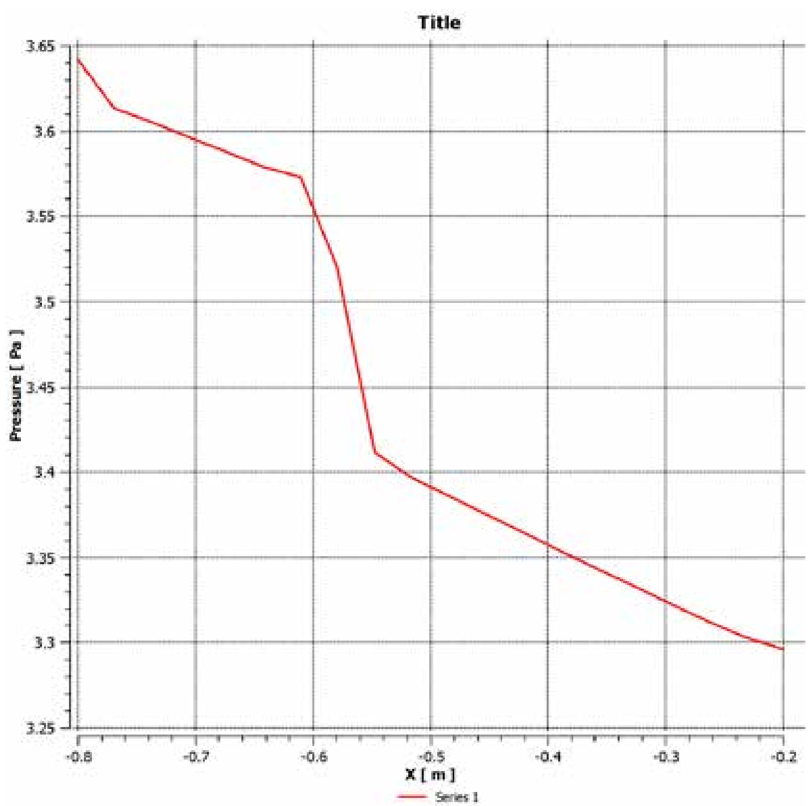


Figure 66: Pressure drop over the heat exchanger in the final design. Q= 50 m³/h

5.3 Test Setup 1

The test setup will be used to compare the simulated data with the model. For the test a wind tunnel simulation was created at a scale of 1:1. The Fiwihex air to air heat exchanger was given by Vision4Energy in order to complete the model. The total model exists of a wind tunnel in which the inlet and outlet will be situated. Underneath the wind tunnel the heat exchanger will be placed together with an 'office'. This office will be a the bend in order to limit the complexity of the system.

Figure 67: Heat exchanger mock-up. Scale 1:1.

Adjustments

For the simulation the model was slightly adjusted. In order to include the 'office' the inlet and the outlet were placed on the same side of the heat exchanger. This made it easier for the office to be integrated as the interior inlet and outlet were located on the same side of the heat exchanger. These changes will not interfere with the total air flow throughout the system. Only the direction of the airflows inside the heat exchanger will be changed. In the design the airflow inside the heat exchanger is parallel, in the model it is a cross flow. This will only influence the efficiency of the heat exchanger but this will not be tested in the given setup.

Figure 68: Wind tunnel setup with the fan (left) and the diffusor (right) visible.

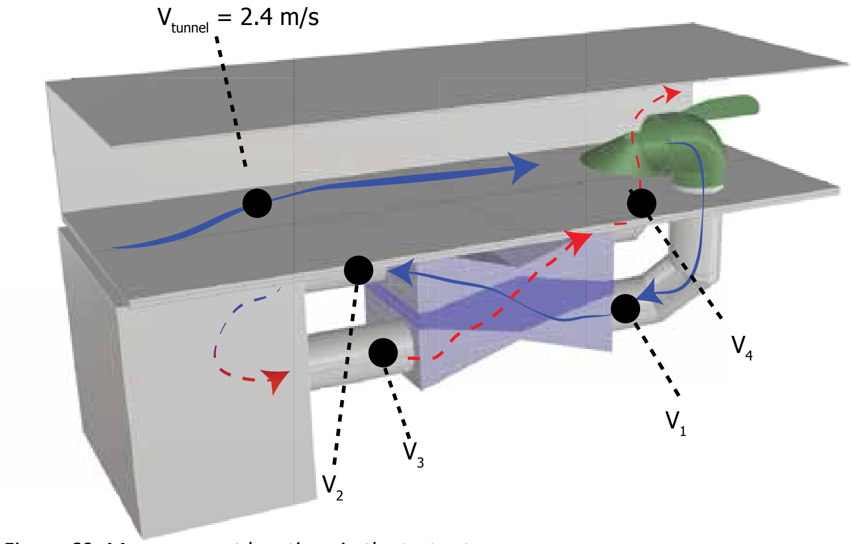


Figure 69: Measurement locations in the test setup.

A standard table fan will be used for creating the air flow extracting air from the tunnel. The air will be sucked into the tunnel through a diffuser created by 5cm long tubes. This diffuser makes redirect non-parallel air flows, so a uniform airflow will be created.

Equipment

For the measurements of the wind velocity the VelociCalc air velocity meter, Model 9515 was used (figure 70). This device is able to calculate the air velocity with a precision of 0.025 m/s or 5% of the show value by using a hot wire air velocity sensor. (TSI Incorporated, 2013).

Figure 70: VelociCalc air velocity meter model 9515 (left) and the hot wire air velocity sensor (right)

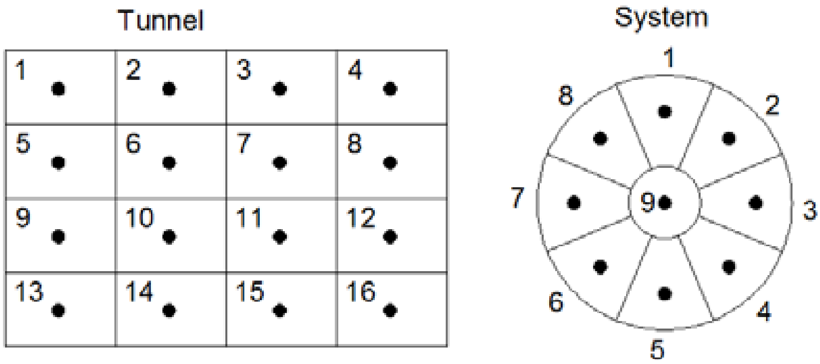


Figure 71: Measurement locations in the cross sections of the setup.

Measurement

In the setup there are a couple of measurement points. The first one is located in the wind tunnel itself (V_{tunnel}). At this point the wind speed in the tunnel will be measured and used in order to create a constant air flow of 2.4 m/s. The next measurement points are located just before and after the first pass through the heat exchanger. These values will be used to check for any air leakage in the system. The last two points will be measured just before and after the second pass through the heat exchanger. Again this will be used to check for any air leakages. The data collected from the four points inside the system will also be used to have a more average result for the air velocity inside the system.

The amount of measurement points in the cross-section of the location was determined by looking at the standard for measurements in air ducts (CEN, 2007). This standard states that for a pipe diameter smaller than 0.35 m only 1

location	#	Velocity (m/s)	average (m/s)
Tunnel	1	0.16	0.77
	2	1.52	
	3	1.13	
	4	0.65	
	5	0.30	
	6	1.69	
	7	0.92	
	8	0.58	
	9	0.30	
	10	1.53	
	11	1.74	
	12	0.42	
	13	0.52	
	14	0.28	
	15	0.13	
	16	0.37	

Table 12: Velocity data inside the tunnel.

location 1

#	Velocity (m/s)	average (m/s)
1	0.44	0.78
2	0.74	
3	0.71	
4	1.02	
5	1.16	
6	0.75	
7	0.88	
8	0.56	
9	0.74	

location 2

#		Velocity (m/s)	average (m/s)
1		0.21	0.22
2	2	0.16	0.16
3		0.11	0.11
4	4	0.35	0.35
5		0.30	0.30
6	6	0.25	0.25
7		0.25	0.25
8	8	0.11	0.11
9		0.21	0.21

location 3

#	Velocity (m/s)	average (m/s)
1	0.28	0.20
2	0.27	
3	0.32	
4	0.16	
5	0.08	
6	0.21	
7	0.05	
8	0.20	
9	0.19	

location 4

#		Velocity (m/s)	average (m/s)	
1		0.26	0.24	0.24
2	2	0.27	0.27	
3		0.14	0.14	
4	4	0.20	0.20	
5		0.23	0.23	
6	6	0.24	0.24	
7		0.26	0.26	
8	8	0.23	0.23	
9		0.29	0.29	

Table 13: Velocity data inside the system. V1 (top left), V2 (top right), V3 (bottom left) and V4 (bottom right).

measurement point would be sufficient but more would be preferred due to the large fluctuations in air speed in the cross section. As a result of this 9 points will be used for the measurements. 8 out of these 9 will be located 1 cm from the wall and at angles of 45 degrees. The 9th point will be located in the centre of the cross section (figure 71). For the rectangular cross section of the wind tunnel 16 data points are placed in a grid of 4x4. The standard (CEN, 2007) states that a minimum of 4 data points on at least 2 side divisions should be used.

Results

During the test it was clear that the fan used in the system was not able to create a strong enough air flow by extracting the air from the tunnel as the measured wind velocity was only 0.1 m/s. This in combination with the error margin of the measurement device will give an accuracy of 20% above or below the measured value. As a result of this, the fan was use to blow air into the system. The vortex created by the fan was weakened by the use of the diffuser. Due to this setup there was not an evenly spread air velocity throughout the system. This can be seen in the data from the measurements inside the tunnel. The values vary between the 0.13 m/s at the sides to 1.74 m/s in the center (table 12) with an average of 0.7 m/s.
It should be mentioned that due to the vortex there still was a fluctuation of the air speed. For the gathered data the velocities were calculated during a period of 2 minutes and the mean value was used for the data.

The averages measured at the 3 measurement locations (location 2,3 and 4) in the system, average at respectively 0.22, 0.20 and 0.24 m/s. The average of the first location (0.78 m/s) differs from these values but it matches the wind velocity in the wind tunnel. This is probably a result of the way there was measured. For this first measurement the heat exchanger was disconnected from the inlet. As the heat exchanger would create a static pressure in this inlet not all the air would flow into the system. This was not the case during the measurements so the air could flow freely through the inlet with the same velocity as the wind inside the tunnel creating this high value.

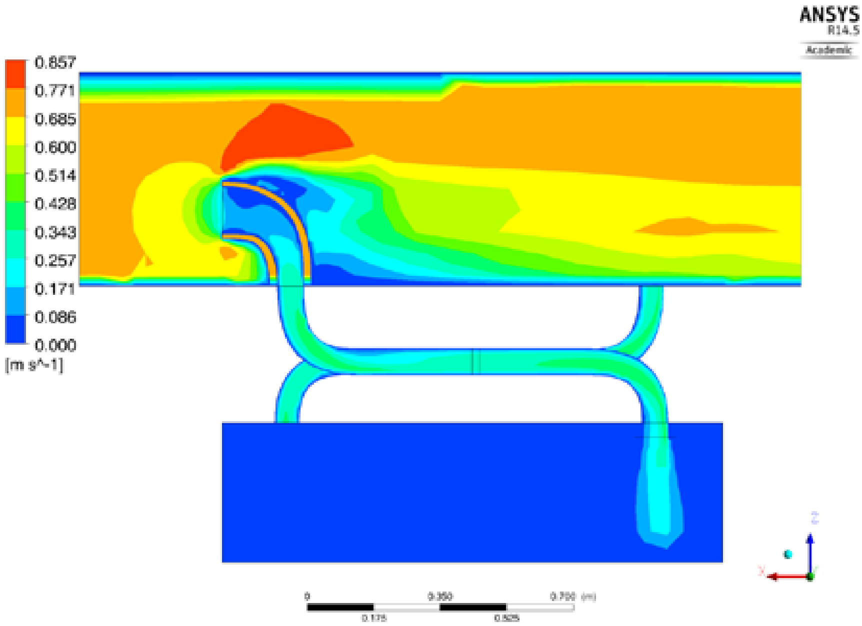


Figure 72: Simulation of the setup with a main air velocity of 0.72 m/s.

Simulation vs test data

Due to the lower wind velocity in the tunnel a new simulation was done in order to compare the measured results with the simulated data. In the simulation, the air velocities in the ducts of the system were averaged at 0.28 m/s with a peak in the center of 0.31 m/s. This corresponds with the data measured during the test. There is a 20% difference between the simulated and measured data. One possible explanation could be the quality of the setup. Due to the use of simple PVC pipe elements and wooden boxes, there could be more air resistance in the system itself. Another explanation could be the fluctuations of the wind velocity due to the fan. A more constant wind velocity would have given more accurate data.

5.4 Test Setup 2

Due to the low air velocities in the tunnel during the first test another test was done. This test was executed at the research laboratory of DPA Cauberg-Huygen in Zwolle. During this test not only the wind velocity in the system was measured, also the pressure difference created by the heat exchanger.

Due to the small capacity of the fan used during the first test, another fan was used for generating higher wind velocities. Also the fan was extracting air from the tunnel. This would remove the turbulent vortex created by the fan in the first test.

Equipment

For this test the same equipment for calculating the air velocity was used. For the air velocity in the tunnel the Testo 417 vane anemometer was used (figure 73 right). This anemometer has an accuracy of 0.1m/s or 1.5% of the measured value. For the pressure a micro manometer was used. The Betz 2500 micro manometer (figure 73 left) uses two pitot tubes which were installed just before and after the heat exchanger (figure 74). It also has an accuracy of 0.1 Pa.

Figure 73: Betz 2500 micro manometer (left) and the Testo 417 vane anemometer (right)

Figure 74: Pitot tube inserted into the pipes.

Measurement

The same locations as in the first test were used. In order to get a better understanding of the pressure difference of the heat exchanger six different air velocities were tested. The pitot tubes were inserted through a hole in the pipes at location 1 and 2 (figure 75). Due to the higher accuracy of the vane anemometer this system was used for calculating the air velocity in the tunnel. Due to its size only 6 measurements points were used in the cross section of the tunnel.
For the calculation of the air velocity in the tubes of the system only 5 measurement points were used this time. This is still enough according to the current standards (CEN, 2007). For this air velocity test the air velocity of in the tunnel was set at 2.4 m/s in order to generate data which could be compared to the computer simulations.

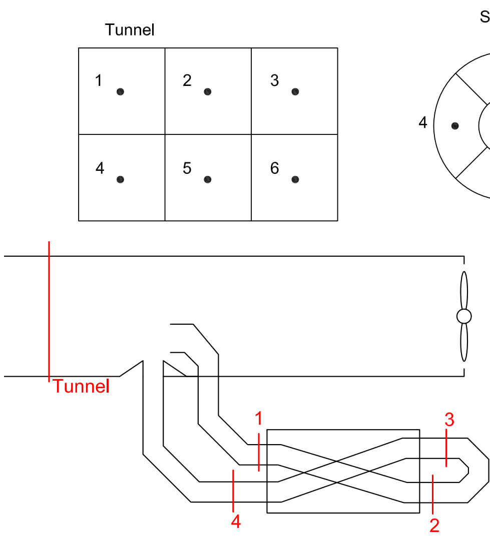


Figure 75: Measurement locations

Figure 76: Self-made test setup at the laboratory of DPA Cauberg-Huygen, Zwolle.

Results

The new fan created a more consistent air velocity throughout the whole cross section of the tunnel. The calculated values hardly differ so more accurate data was collected. The pressure difference is almost linear with the air velocity in the tunnel (table 14). The pressure difference over the heat exchanger measured at 2.4 m/s however differs with the value created by the simulation program. According the simulation it was a pressure difference of 0.2 Pa but a value of 1.5 Pa was measured during the test. The measured value would be more logical as the pressure difference at a double velocity would be 4 Pa. At the tunnel velocity of 2.4 m/s the air velocities in the tubes was also measured. The average velocities of the cross sections were respectively 0.74, 0.81, 0.89 and 0.95 m/s (table 15).

Simulation vs test 2

In the simulation, the air velocities in the ducts of the system were averaged at 1.07 m/s (figure 77). During the simulation the average wind velocity in the ducts was 0.85 m/s resulting in a difference between simulation and test of around 20 % compared with the simulation. The reason for this 20 % are possible the same as with the first test; The quality of the setup. All the bends in the system would generate a higher pressure resistance blocking the air. Also the difference in pressure resistance of the heat exchanger can be of great influence as this is a difference of 87%.

location	#	Velocity (m/s)	average (m/s)	dP HE (Pa)
Tunnel	1	6.15	6.29	10.80
	2	6.52		
	3	6.60		
	4	5.75		
	5	6.32		
	6	6.38		

location	#	Velocity (m/s)	average (m/s)	dP HE (Pa)
Tunnel	1	3.75	3.79	4.80
	2	3.81		
	3	3.81		
	4	3.78		
	5	3.75		
	6	3.82		

location	#	Velocity (m/s)	average (m/s)	dP HE (Pa)
Tunnel	1	5.55	5.59	8.60
	2	5.52		
	3	5.68		
	4	5.44		
	5	5.53		
	6	5.80		

location	#	Velocity (m/s)	average (m/s)	dP HE (Pa)
Tunnel	1	2.23	2.34	1.50
	2	2.35		
	3	2.48		
	4	2.10		
	5	2.44		
	6	2.42		

location	#	Velocity (m/s)	average (m/s)	dP HE (Pa)
Tunnel	1	4.65	4.78	6.00
	2	5.01		
	3	4.81		
	4	4.80		
	5	4.74		
	6	4.68		

location	#	Velocity (m/s)	average (m/s)	dP HE (Pa)
Tunnel	1	1.00	1.01	0.70
	2	1.05		
	3	1.00		
	4	0.99		
	5	1.05		
	6	0.95		

Table 14: Velocity and pressure difference inside the tunnel.

location	#	Velocity (m/s)	average (m/s)
Tube 1	1	0.90	0.94
	2	0.94	
	3	0.99	
	4	0.91	
	5	0.95	

location	#	Velocity (m/s)	average (m/s)
Tube 2	1	1.19	1.00
	2	1.00	
	3	1.10	
	4	0.79	
	5	0.90	

location	#	Velocity (m/s)	average (m/s)
Tube 3	**1**	**1.04**	**1.10**
	2	**0.83**	
	3	**1.29**	
	4	**1.18**	
	5	**1.16**	

location	#	Velocity (m/s)	average (m/s)
Tube 4	**1**	**1.17**	**1.15**
	2	**1.11**	
	3	**1.11**	
	4	**1.21**	
	5	**1.13**	

Total average (m/s)
1.05

Table 15: Velocity data inside the system. V1 (top left), V2 (top right), V3 (bottom left) and V4 (bottom right).

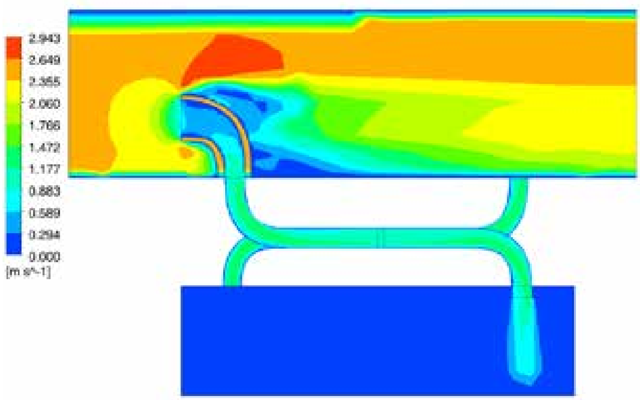


Figure 77: Simulation of the setup with a main air velocity of 2.4 m/s.

5.5 Design Possibilities

The proposed natural ventilation system has multiple design possibilities. There are for example several possibilities for the façade, interior and even the layout. For the integration of the system only several boundary conditions should be met in order to use the system efficiently.

Façade

For the design of the façade there are a couple of boundary conditions which should be met. The first is the minimum distance of the inlet and outlet to the façade. They both have to be at least 10cm distance from the façade. This is a result of the boundary layer created by the material of the façade cladding and the possible vortexes created by the window frames. Another boundary condition for the façade is the distance between the in- and outlet of the system. They have to be at least 1m apart from each other. Due to the 'dirty air' coming out of the exhaust, it should not directly be blown into the inlet as 'fresh air'. If these boundary conditions for the façade are met, there are several possibilities for the integration of the ventilation system. One could for example create a more horizontal alignment in the façade by using 2 different types of cladding (figure 77 ,left). The same principle can be used to create a more vertically oriented façade (figure 77, right). For these designs the size of the window can also be easily adjusted. You could for example create a wide but narrow window but also create a large floor to floor window (figure 78, right). Another possibility could

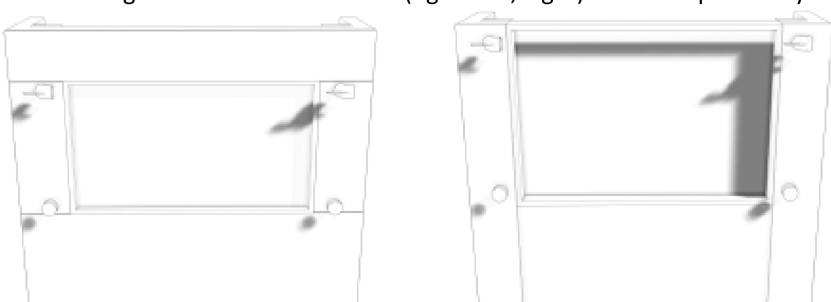


Figure 78: Possible facade design options. Horizontal (left) and vertical (right) variants of the standard office facade with possibility desk location at the facade.

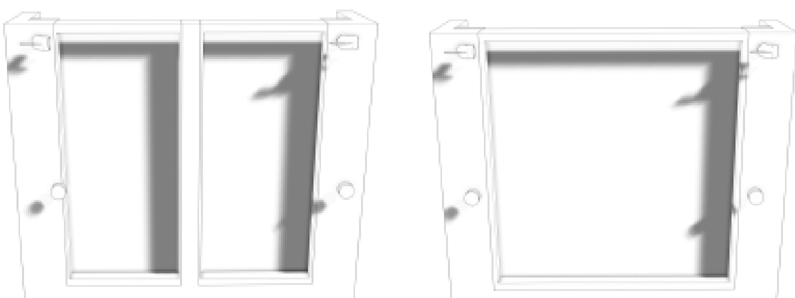


Figure 79: Possible facade design options. A double window (left) or a large floor to floor window (right)

be the use of two windows next to each other (figure 78, left). This also gives the possibility to locate the ventilation systems in the centre of the façade element. The last possibility is the use of curved cladding. As the boundary condition of the system is that it should be at 10 cm distance from the façade you could create more movement in the façade by using double curved cladding panels. This could even help to increase the efficiency of the ventilation system due to the redirected air (figure 79).

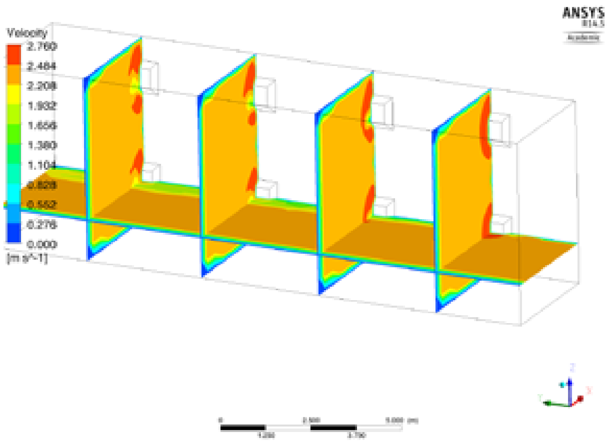


Figure 80: Air velocity vector contour if a simple double curved elements are located on the facade

Floor layout

The ventilation system also gives some possibilities regarding the layout of the floor. Due to the distance the air could go to the back of the room, the most optimal office depth would be 6m. The width of the office can be adjusted to the type of building. During this research the office width was kept at 3.6m due to the 70's building typology. The ventilation system would also work for offices wider than that.

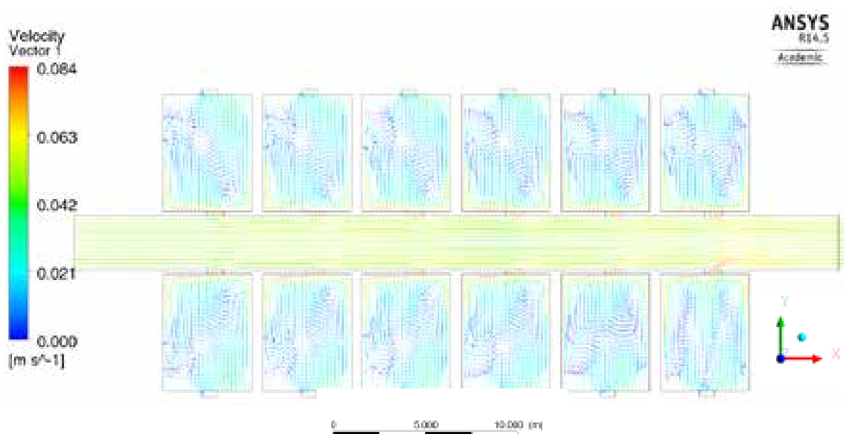


Figure 81: Air velocity vector in the hallway if there is an overpressure in the hallway.

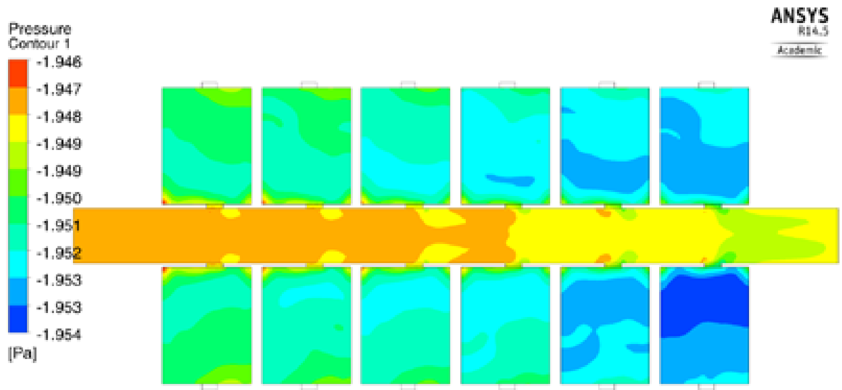


Figure 82: Air velocity contour in the hallway if there is an overpressure in the hallway.

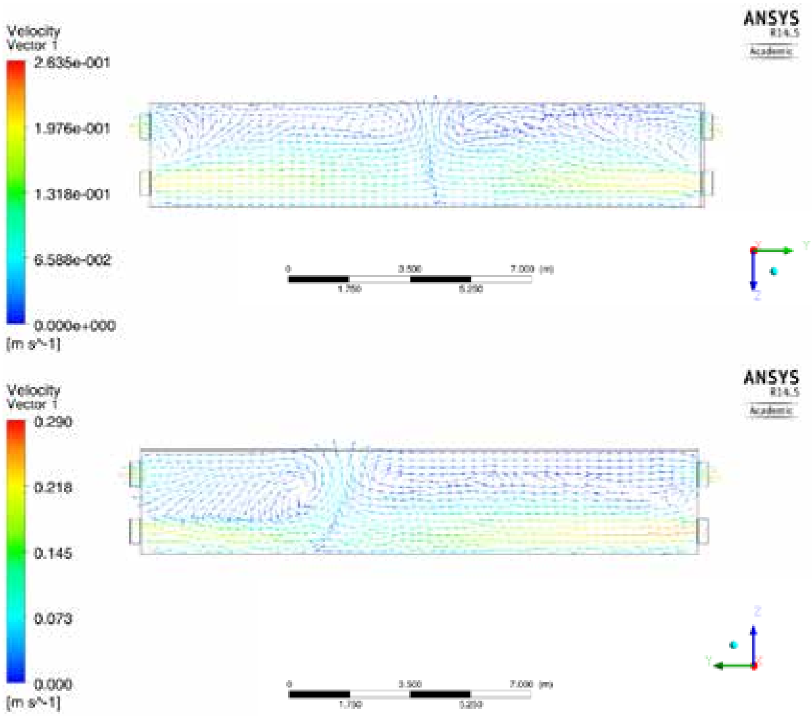


Figure 83: Air velocity vectors in a cross section of a open plan office. Top: equal air velocities on both facades. Bottom: different air velocities on both opposing facades.

The closed air loop might be disrupted if some employees keep their door towards the hallway open. As the hallway also needs ventilation the resulting airflow will disturb both air flows. As a result of this one of the boundary conditions for this typology will be that a small overpressure in the corridor, e.g. 2 Pa, will be needed. This can be the result of a mechanical or even natural system. The air inside the office will hardly go towards the hallway and thus creates a closed system inside the office (figure 80).

It is also possible to use the system in an open plan office. If the system is placed on both sides of the façade and the mechanical backup system on the leeward side creates about the same air velocity as the natural principles on the windward sides there are possibilities for an open plan office. Even small differences in air velocities between the facades will not result in a disfunctional system as they both keep their own effective area inside the office (figure 81). One of the boundary conditions for the open plan office is that there can be no freestanding rooms, e.g. board rooms or toilets, in the middle of the floor. Not only will these rooms need ducts under the ceiling but they will also interfere with the airflow from the ventilation systems, so all the enclosed rooms need to be located at the facade.

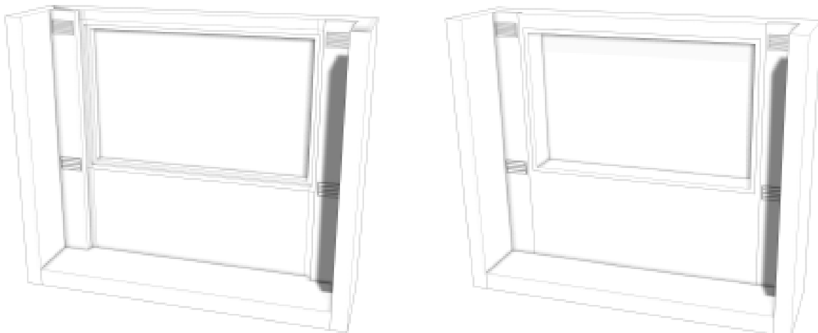


Figure 84: Internal design possibilities. Added to an existing facade system (left) or integrated in a new facade element (right)

Interior

There are also some possibilities regarding the interior. One of the boundary conditions is that there should be a distance of at least 1.5m between the inlet and outlet vent in the office. The fresh air will not only spread through the office better but the ventilation system can also take advantage of the pressure created by temperature difference between the floor and the ceiling, about 0.5 Pa. The location of the system on the façade can be linked to the visual aspects regarding the façade. Both systems can be located at the walls but also in the centre of the façade element (figure 77,left). The ventilation system can also be very well integrated into the wall panel. However, this will result in a thicker façade element (figure 78,right).

Technical possibilities

Summer situation

During the summer the heat exchanger will be used to cool down the outside air. The air outside the office might be warmer (28 deg) than inside the office (22 deg). A possible solution to prevent overheating will be to integrate a bypass system. This system will let air directly from the outside to the office without passing through the heat exchanger. Another advantage of the bypass is the possibility to use night ventilation. During the night the bypass system allows the colder outside air to flush the office and cool down the mass of the structure resulting in a cooler office during the day.

Toilets

Due to the large airflow needed in the toilets a natural ventilation system would not be sufficient. It is a possibility to make use of the same system but fully mechanical. This allows a standardized façade panel for the entire façade.

5.6 Energy Savings

One of the targets of this research was to reduce the energy consumption of the building. As stated in chapter 1.5, a mechanical ventilation systems uses about 55% total energy consumption of a building. Out of these 55% about 20% is needed for ventilation. If the mechanical fans are replaced by natural principles for half the façade it will result in an energy saving of 5% on the total energy consumption. Another advantage of the system is the high efficiency of the heat exchanger. Due to this high efficiency even more energy could be saved on heating and cooling energy.

For the comparison of the decentralized ventilation system with the newly designed system the total energy costs of the building is needed. The Dutch Green Building council stated in a report that the average energy consumption per square meter is 0.53 GJ/m2 or 155 kWh/m2 annually (Balvers & van Uffelen, 2012) with an average energy cost per kWh of € 0.23.

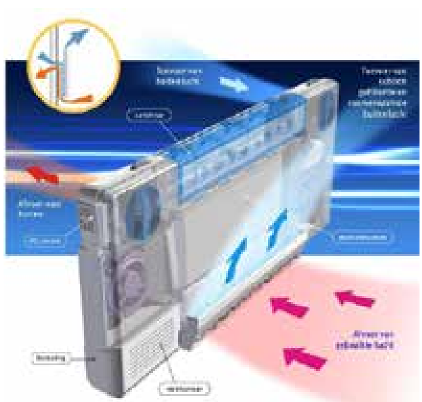


Figure 85: Climarad ventilation system. (Climarad BV, 2014 a)

In order to take note of the advantages of the hybrid ventilation system it will be compared with several decentralized mechanical ventilation systems. The total energy usage will be halved as a result of the systems on the leeward side of the building which would need mechanical backup.

The first system is the Climarad system (see chapter 3.3)(figure 84). This system uses about 36W in order to power the fans and to increase the efficiency of the heat exchanger. Compared with the other systems (table 16) this system has a relativly high energy usage per square meter. If this system would be replaced by the new natural ventilation system it would give a possible energy saving of € 1.66 per square meter. This system can be compared with the new design as they are both only used for ventilation. They don't have any additional heating or cooling systems.

Another decentralized mechanical ventilation system is the Trox Schoolair-V (Trox, 2011). This system just like the designed system be placed next to the window and has about the same dimensions. The Schoolair-V system uses only 17W of energy to power the fans in the system. If these would be replaced by natural principles it would save € 8.45 for every office annually. This seems not a lot but due to the low efficiency of the heat exchanger, 55%, more can be saved on heating and cooling with the use of the designed ventilation system. Also the enthalpy heat exchanger system would still allow the indoor contaminations to settle in the system and be blown back into the room.

Figure 86: Trox Schoolair V. Average annual energy demand of 73 kWh for ventilation. (Trox, 2011)

Name system	Air flow (m3/h)	Energy use (W)	Energy use (kWh/year)	Energy use (kWh/m2)	cost (€/kWh)	cost (€/m2)	Possible energy savings (€/m2)	Efficiency (%)
Climarad	100	36	311	14.40	0.23	3.31	1.66	80
Trox Schoolair-V	200	17	147	3.40	0.23	0.78	0.39	55
Smartbox	100	40	352	16.30	0.23	3.75	1.87	92
4VE-Smart ventilation (Fiwihex)	50	12	104	9.60	0.23	2.21	1.10	85
inVENTer	29	3	26	4.14	0.23	0.95	0.48	78
Sonair A+	65	9.6	83	5.91	0.23	1.36	0.68	0 - 100

Table 16: Comparison of different decentral mechanical ventilation systems and the new natural ventilation system.

The third system is the smartbox system (chapter 3.2). This system is using a high efficiency enthalpy heat exchanger of 92% at a capacity of 100 m3/h. This 100 m3/h is equal to the use of 2 naturally ventilated systems in an office of 21.6 m2. In the system it also includes heating, cooling and humidifiers to create the perfect indoor comfort. One of the downsides of this system is the high energy use off all these combined system. This complete system is also one of the advantages as no additional systems would be needed to create a perfect indoor climate.

The next system is the 4VE-Smart ventilation system with the fiwihex heat exchanger (chapter 3.1). Compared with the new system the total energy savings are not that large as the new system is based on the ventilation system.

The fifth decentralized ventilation system which will be compared is the inVENTer (Teulings Consult, 1998). This system uses ceramic plates to store energy and then releases as the airflow is inverted each 15 minutes. This also allows it to act as an enthalpy heat exchanger. Due to the simple design it doesn't use a lot of energy, 3 Watt. However the total capacity of the system is very low, 29 m3/h. Also the system has a low efficiency of the heat exchanger, making the need for additional heating necessarily.

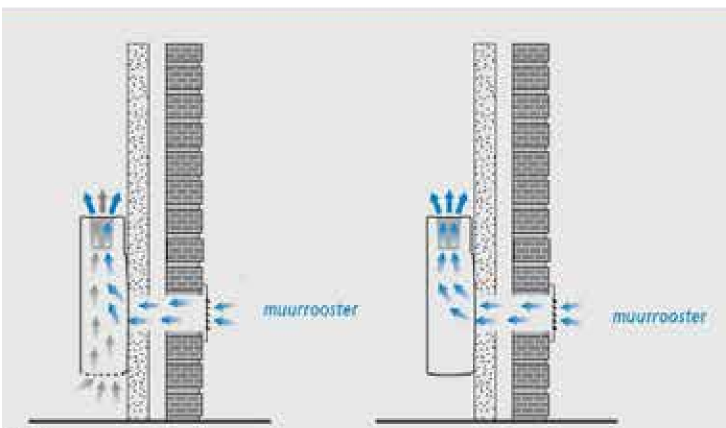


Figure 87: Princple of the Sonair A+ showing the possibilities for ventilation and heat recovery (Brink, 2014)

name system	savings (€/m2)	savings(€/office)(21.6 m2)	savings (€/building)(7344 m2)
Climarad	1.66	35.77	12161.66
Trox Schoolair-V	0.39	8.45	2871.50
Smartbox	1.87	40.48	13763.20
4VE-Smart ventilation (Fiwihex)	1.10	23.85	8107.78
inVENTer	0.48	10.28	3494.73
Sonair A+	0.68	14.67	4989.40

Table 17: Total energy savings if the mechanical ventilation would be replaced by the new natural ventilation system.

The last ventilation system which will be compared with the new natural system is the Sonair A+ (Brink, 2014). The heat recovery of this system is created by mixing the fresh indoor air with a part of the indoor air. Also the total energy savings if it would be replaced by the natural ventilation is relatively low. One of the advantages of this system is the complete manual control of the heat exchanger. This allows a lot of control over the inlet temperature. Due to this simple system the energy savings are relatively low on at a yearly base.

Compared with the six other decentralized mechanical ventilation systems, the newly designed system saves energy (table 17). It is however per system different what this total energy savings are, as they range from 4.73%, Climarad, to only 1.1 % of the Trox Schoolair-V. The total energy savings are not the only thing as the efficiency of the system also differs. The Trox Schoolair-V does for example not save that much on the total energy bill but it still needs a lot of additional heating due to the low efficiency of the heat exchanger. In comparison with these mechanical systems more energy can be saved as a result of the natural ventilation. Natural ventilation systems allow lower temperatures in the office (see chapter 2.4). This will reduce the amount of additional heating during the start-up.

6 Conclusion

During the analytical and the experimental studies of this thesis, all the research questions were answered. But during the research new questions appeared of which some were answered and some need some further research. The answered research questions will be described in chapter 6.1 and the topics for further research will be discussed in chapter 6.2.

6.1 Conclusion

Natural ventilation is a good alternative for mechanical ventilation. As the current used mechanical ventilation systems use 11% of the total energy usage of a building, a large progress could be made. During the literature studies, recent researches have shown that naturally ventilated buildings not only save energy but also create a more pleasant work environment. Next to the savings on the ventilation, a lot of energy can be saved on heating and cooling as naturally ventilated buildings accept respectively lower and higher temperatures.

For natural ventilation different principles like cross- or single-sided ventilation can be used. Also the driving forces, wind and buoyancy, can be used in different ways with each having their own advantages. A hybrid ventilation system can make use of one of these principles with a mechanical backup system if the created air flow wouldn't be sufficient. This will also allow the users to adjust the system more easily.

During the research it became clear that the main focus would be on describing and analysing the wind direction and velocity at the façade. There was some research done but this was mainly at street level. As the CFD models showed the average wind velocity next to the façade was around 2.4 m/s with a main wind velocity of 6 m/s (4 Bft). This would increase if the wind would blow from an angle (30, 60 and 90 deg) to the façade.

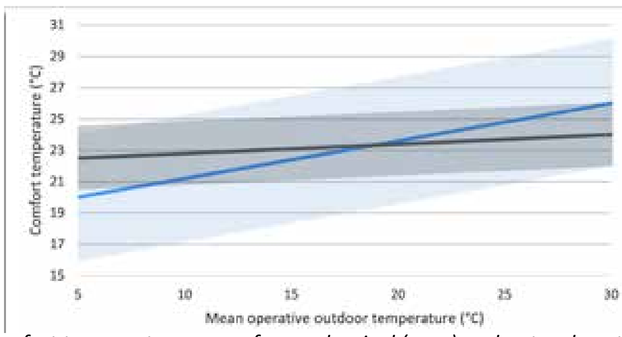


Figure 88: Comfort temperature range for mechanical (grey) and natural ventilation (blue). (Fountain, Brager, & de Dear, 1996)

The designed ventilation system is based on some of the current mechanical ventilation systems like the Troxx schoolair V, the Fiwihex and the Smartbox. By combining the advantages of these systems like the location, efficiency and shape, with the results from a research from de Gids and den Ouden the new system was created. The new natural ventilation system consists of a Fiwihex heat exchanger combined with a wind cowl and a 3D-venturi outlet at the façade. This system was not only successfully tested with the CFD program but also successfully tested in a 1:1 scale model.

Final calculations of this system show that an energy saving of 5% can be reached on the total energy use of the building. In comparison with other decentralized mechanical ventilation systems savings up to € 13763 are possible. During simulations it was also clear that the decentralized hybrid ventilation system can be used in enclosed offices as well as open-plan offices if the boundary conditions are met.

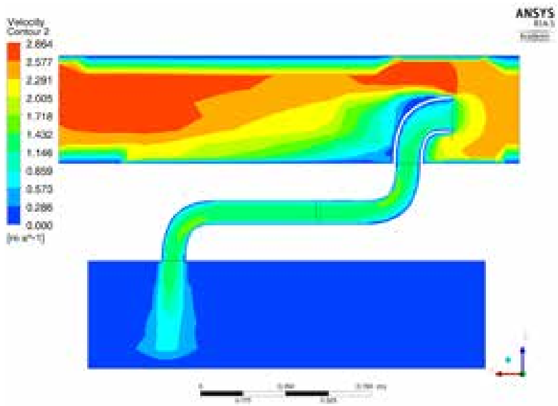


Figure 89: Velocity contour at a section through the inlet of the ventilation system.

6.2 Further Research

During the literature research as well as the analytical research, additional questions appeared. Some of these questions were answered during the research.

One of these topics was the wind velocity directly at the façade of a building. During the research these values were created by a CFD model but these might differ with wind tunnel simulations. Literature even shows the difference between the CFD programs itself. So some further research can be done towards the wind velocities at the façades of the building.

Another topic for further research is the influence of the environment on the wind velocity at a building. During the research of this thesis the building was set in an open environment but the wind velocities will vary a lot when the building will be placed in for example an urban environment.

Another topic is the design of the in- and outlet on the façade. During the research it became clear that different types of systems can be used to direct air into the system or extract air from it. Small changes in for example the diameter, could lead to completely different results.

The last recomendation is about the adaptivness during the summer. What are the possibilties of bypassing the heat exchanger for example and how can this be integrated into the design. This would lead to an even better intergrated system which works for the entire year.

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8 Appendices

A. *Wind around buildings Ansys*

A1. *Wind 0 °*

1. File Report

Table 1. File Information for Wind grid test

Case	Wind grid test
File Path	D:\Documents D;\Afstuderen\Calculations\3 01900.dat.gz
File Date	03 November 2014
File Time	03:19:19 PM
File Type	FLUENT
File Version	14.5.0

2. Mesh Report

Table 2. Mesh Information for Wind grid test

Domain	Nodes	Elements
model_space	613880	534536

3. Physics Report

Table 3. Domain Physics for Wind grid test

Domain - model_space	
Type	cell

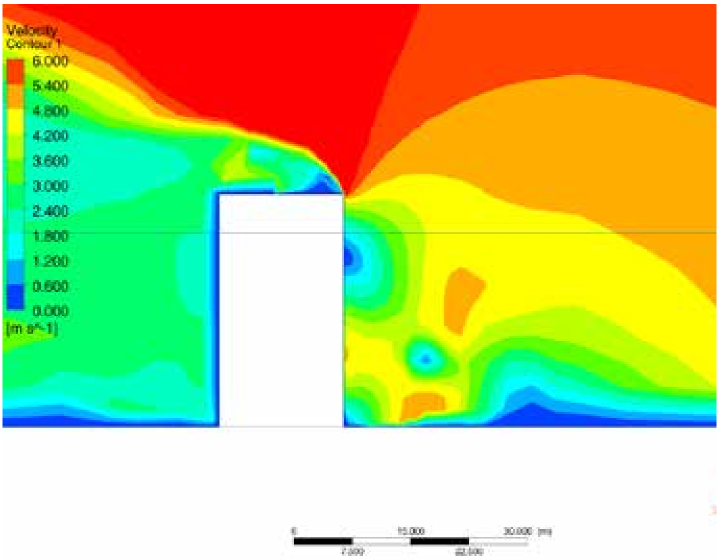
Table 4. Boundary Physics for Wind grid test

Domain	Boundaries	
model_space	**Boundary - inlet**	
	Type	VELOCITY-INLET
	Boundary - outle	
	Type	OUTFLOW
	Boundary - wall model_space	
	Type	WALL

4. Solver Input

```
Models
------
 Model                     Settings
 -------------------------------------
 Space                      3D
 Time                       Steady
 Viscous                    Laminar
 Heat Transfer              Disabled
 Solidification and Melting Disabled
 Species                    Disabled
 Coupled Dispersed Phase    Disabled
 NOx Pollutants             Disabled
 SOx Pollutants             Disabled
 Soot                       Disabled
 Mercury Pollutants         Disabled
Material Properties
-------------------
 Material: air (fluid)
   Property                     Units    Method    Value(s)
   ---------------------------------------------------------------
   Density                      kg/m3    constant  1.225
   Cp (Specific Heat)           j/kg-k   constant  1006.43
   Thermal Conductivity         w/m-k    constant  0.0242
   Viscosity                    kg/m-s   constant  1.7894e-05
   Molecular Weight             kg/kgmol constant  28.966
   Thermal Expansion Coefficient 1/k     constant  0
   Speed of Sound               m/s      none      #f
 Material: aluminum (solid)
   Property             Units    Method    Value(s)
   ---------------------------------------------------
   Density              kg/m3    constant  2719
   Cp (Specific Heat)   j/kg-k   constant  871
   Thermal Conductivity w/m-k    constant  202.4
Cell Zone Conditions
--------------------
 Zones
   name         id  type
   -------------------------
   *model_space  2  fluid

Boundary Conditions
-------------------
 Zones
   name              id  type
   ---------------------------------------
   outle             6   outflow
   inlet             5   velocity-inlet
   wall-*model_space 7   wall
 Setup Conditions
   outle
     Condition          Value
     --------------------------
     Flow rate weighting  1
   inlet
     Condition                              Value
     -------------------------------------------------
     Velocity Specification Method            1
     Reference Frame                          0
     Velocity Magnitude (m/s)                 0
     Supersonic/Initial Gauge Pressure (pascal) 0
     Coordinate System                        0
     X-Velocity (m/s)                         6
     Y-Velocity (m/s)                         0
     Z-Velocity (m/s)                         0
     X-Component of Flow Direction            1
     Y-Component of Flow Direction            0
     Z-Component of Flow Direction            0
     X-Component of Axis Direction            1
     Y-Component of Axis Direction            0
     Z-Component of Axis Direction            0
     X-Coordinate of Axis Origin (m)          0
     Y-Coordinate of Axis Origin (m)          0
     Z-Coordinate of Axis Origin (m)          0
     Angular velocity (rad/s)                 0
     is zone used in mixing-plane model?      no
   wall-*model_space
     Condition                                     Value
     -------------------------------------------------------
     Enable shell conduction?                        no
     Wall Motion                                     0
     Shear Boundary Condition                        0
     Define wall motion relative to adjacent cell zone?  yes
     Apply a rotational velocity to this wall?       no
     Velocity Magnitude (m/s)                        0
     X-Component of Wall Translation                 1
     Y-Component of Wall Translation                 0
     Z-Component of Wall Translation                 0
     Define wall velocity components?                no
     X-Component of Wall Translation (m/s)           0
     Y-Component of Wall Translation (m/s)           0
     Z-Component of Wall Translation (m/s)           0
     Rotation Speed (rad/s)                          0
     X-Position of Rotation-Axis Origin (m)          0
     Y-Position of Rotation-Axis Origin (m)          0
     Z-Position of Rotation-Axis Origin (m)          0
     X-Component of Rotation-Axis Direction          0
     Y-Component of Rotation-Axis Direction          0
     Z-Component of Rotation-Axis Direction          1
     X-component of shear stress (pascal)            0
     Y-component of shear stress (pascal)            0
     Z-component of shear stress (pascal)            0
     Fslip constant                                  0
     Eslip constant                                  0
     Specularity Coefficient                         0
Solver Settings
---------------
 Equations
   Equation  Solved
   ----------------
   Flow      yes
 Numerics
   Numeric                       Enabled
   ---------------------------------------
   Absolute Velocity Formulation  yes
 Relaxation
   Variable     Relaxation Factor
   ------------------------------
   Pressure     0.3
   Density      1
   Body Forces  1
   Momentum     0.7
 Linear Solver
              Solver    Termination  Residual Reduction
   Variable   Type      Criterion    Tolerance
   -----------------------------------------------------
   Pressure   V-Cycle   0.1
   X-Momentum Flexible  0.1          0.7
   Y-Momentum Flexible  0.1          0.7
   Z-Momentum Flexible  0.1          0.7
 Pressure-Velocity Coupling
   Parameter  Value
   ----------------
   Type       SIMPLE
 Discretization Scheme
   Variable  Scheme
   ----------------------------
   Pressure  Standard
   Momentum  Second Order Upwind
```

5. Output data



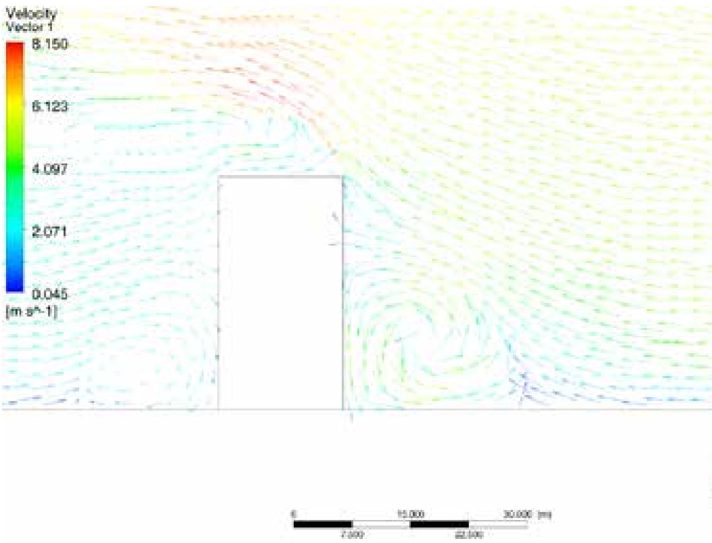
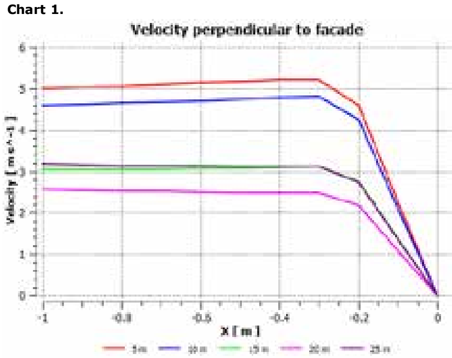
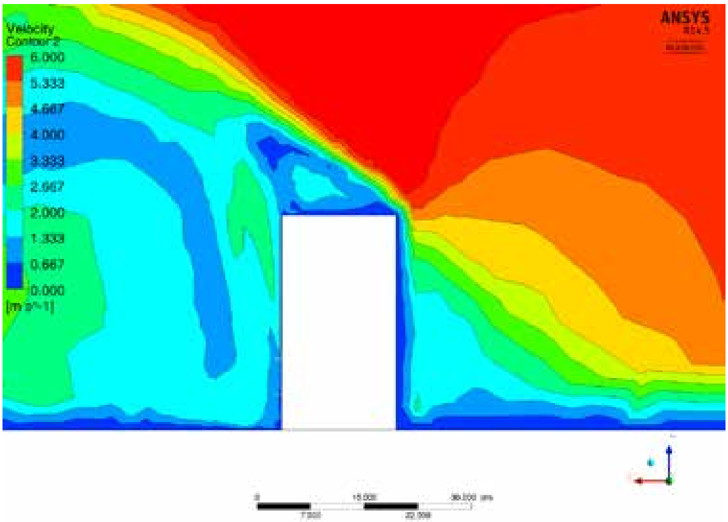
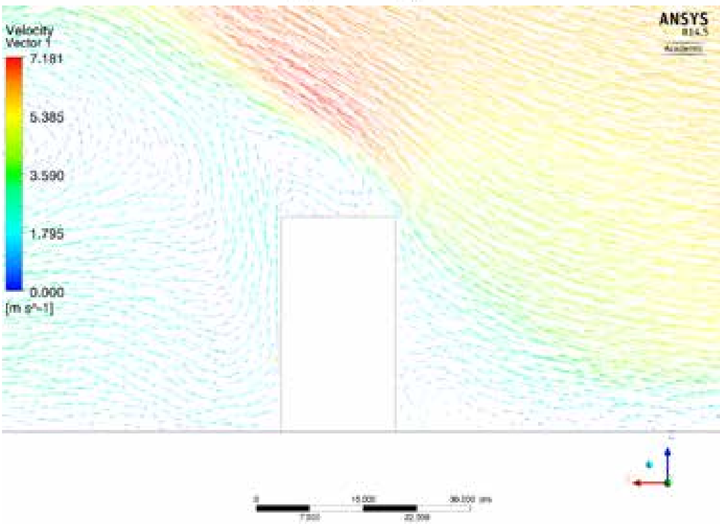


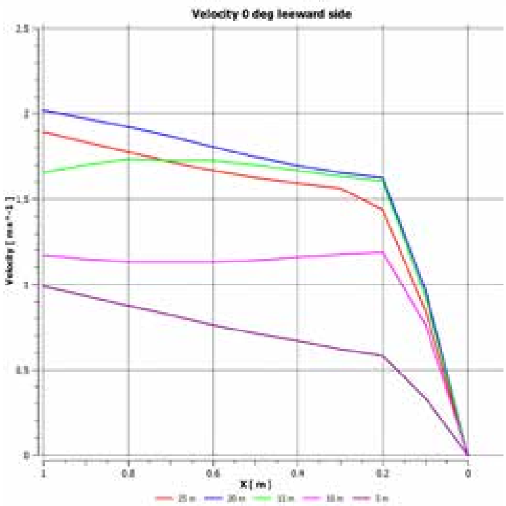
Chart 1.



Leeward 0 °


Velocity
Contour 2
6.000
5.333
4.667
4.000
3.333
2.667
2.000
1.333
0.667
0.000
[m s^-1]
ANSYS


Velocity
Vector 1
7.181
5.385
3.590
1.795
0.000
[m s^-1]
ANSYS


Velocity 0 deg leeward side
2.5
2
1.5
1
0.5
0
1
0.8
0.6
0.4
0.2
0
X [m]

A2. *Wind 30 °*

1. File Report

Table 1. File Information for Windward 30 deg rect mesh

Case	Windward 30 deg rect mesh
File Path	D:\Documents D;\Afstuderen\Calculations\3d files for wind around building\Wind 0 deg mesh_files\dp0\FFF-10\Fluent\FFF-10.2-1-00548.dat.gz
File Date	05 November 2014
File Time	11:16:21 AM
File Type	FLUENT
File Version	14.5.0

2. Mesh Report

Table 2. Mesh Information for Windward 30 deg rect mesh

Domain	Nodes	Elements
model_space	725078	646894

3. Physics Report

Table 3. Domain Physics for Windward 30 deg rect mesh

Domain - model_space	
Type	cell

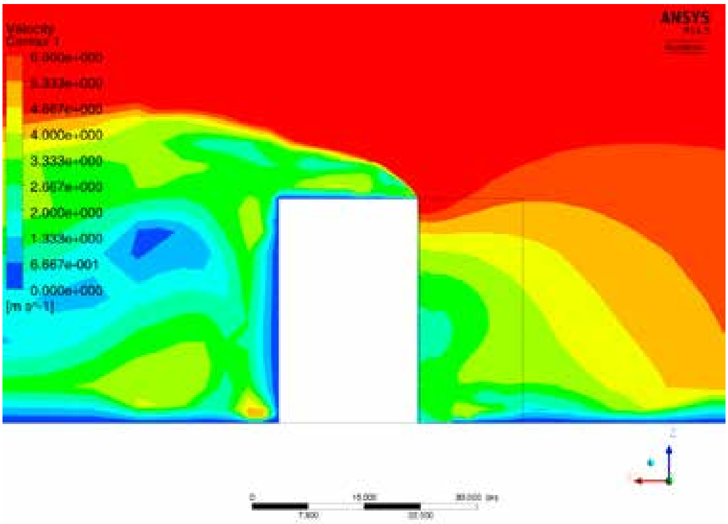
Table 4. Boundary Physics for Windward 30 deg rect mesh

Domain	Boundaries	
model_space	**Boundary - inlet**	
	Type	VELOCITY-INLET
	Boundary - outlet	
	Type	OUTFLOW
	Boundary - wall	
	Type	WALL

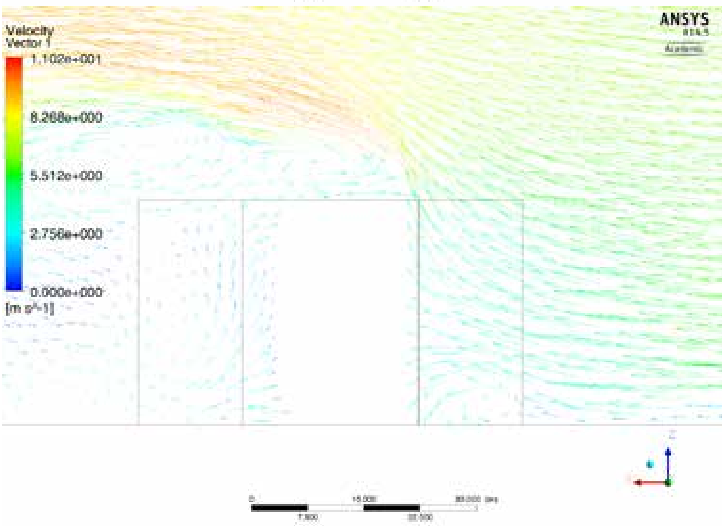
4. Solver Input

```
Fluent
Version: 3d, dp, pbns, lam (3d, double precision, pressure-based, laminar)
Release: 14.5.0
Title:
Models
------
  Model                        Settings
  -------------------------------------
  Space                        3D
  Time                         Steady
  Viscous                      Laminar
  Heat Transfer                Disabled
  Solidification and Melting   Disabled
  Species                      Disabled
  Coupled Dispersed Phase      Disabled
  NOx Pollutants               Disabled
  SOx Pollutants               Disabled
  Soot                         Disabled
  Mercury Pollutants           Disabled
Material Properties
-------------------
  Material: air (fluid)
    Property                      Units     Method     Value(s)
    -----------------------------------------------------------
    Density                       kg/m3     constant   1.225
    Cp (Specific Heat)            j/kg-k    constant   1006.43
    Thermal Conductivity          w/m-k     constant   0.0242
    Viscosity                     kg/m-s    constant   1.7894e-05
    Molecular Weight              kg/kgmol  constant   28.966
    Thermal Expansion Coefficient 1/k       constant   0
    Speed of Sound                m/s       none       #f
  Material: aluminum (solid)
    Property             Units   Method    Value(s)
    -----------------------------------------------
    Density              kg/m3   constant  2719
    Cp (Specific Heat)   j/kg-k  constant  871
    Thermal Conductivity w/m-k   constant  202.4
Cell Zone Conditions
--------------------
  Zones
    name          id   type
    -------------------------
    *model_space  2    fluid
Boundary Conditions
-------------------
  Zones
    name     id   type
    ---------------------------
    outlet   6    outflow
    wall     5    wall
    inlet    7    velocity-inlet

  Setup Conditions
    outlet
      Condition            Value
      --------------------------
      Flow rate weighting  1
    wall
      Condition                                             Value
      -------------------------------------------------------------
      Enable shell conduction?                              no
      Wall Motion                                           0
      Shear Boundary Condition                              0
      Define wall motion relative to adjacent cell zone?    yes
      Apply a rotational velocity to this wall?             no
      Velocity Magnitude (m/s)                              0
      X-Component of Wall Translation                       1
      Y-Component of Wall Translation                       0
      Z-Component of Wall Translation                       0
      Define wall velocity components?                      no
      X-Component of Wall Translation (m/s)                 0
      Y-Component of Wall Translation (m/s)                 0
      Z-Component of Wall Translation (m/s)                 0
      Rotation Speed (rad/s)                                0
      X-Position of Rotation-Axis Origin (m)                0
      Y-Position of Rotation-Axis Origin (m)                0
      Z-Position of Rotation-Axis Origin (m)                0
      X-Component of Rotation-Axis Direction                0
      Y-Component of Rotation-Axis Direction                0
      Z-Component of Rotation-Axis Direction                1
      X-component of shear stress (pascal)                  0
      Y-component of shear stress (pascal)                  0
      Z-component of shear stress (pascal)                  0
      Fslip constant                                        0
      Eslip constant                                        0
      Specularity Coefficient                               0
    inlet
      Condition                                    Value
      -----------------------------------------------------
      Velocity Specification Method                1
      Reference Frame                              0
      Velocity Magnitude (m/s)                     0
      Supersonic/Initial Gauge Pressure (pascal)   0
      Coordinate System                            0
      X-Velocity (m/s)                             6
      Y-Velocity (m/s)                             0
      Z-Velocity (m/s)                             0
      X-Component of Flow Direction                1
      Y-Component of Flow Direction                0
      Z-Component of Flow Direction                0
      X-Component of Axis Direction                1
      Y-Component of Axis Direction                0
      Z-Component of Axis Direction                0
      X-Coordinate of Axis Origin (m)              0
      Y-Coordinate of Axis Origin (m)              0
      Z-Coordinate of Axis Origin (m)              0

      Angular velocity (rad/s)                     0
      is zone used in mixing-plane model?          no
Solver Settings
---------------
  Equations
    Equation   Solved
    -----------------
    Flow       yes
  Numerics
    Numeric                          Enabled
    ---------------------------------------
    Absolute Velocity Formulation    yes
  Relaxation
    Variable      Relaxation Factor
    -------------------------------
    Pressure      0.3
    Density       1
    Body Forces   1
    Momentum      0.7
  Linear Solver
                 Solver    Termination   Residual Reduction
    Variable     Type      Criterion     Tolerance
    --------------------------------------------------------
    Pressure     V-Cycle   0.1
    X-Momentum   Flexible  0.1           0.7
    Y-Momentum   Flexible  0.1           0.7
    Z-Momentum   Flexible  0.1           0.7
  Pressure-Velocity Coupling
    Parameter   Value
    -----------------
    Type        SIMPLE
  Discretization Scheme
    Variable   Scheme
    -----------------------------
    Pressure   Standard
    Momentum   Second Order Upwind
```

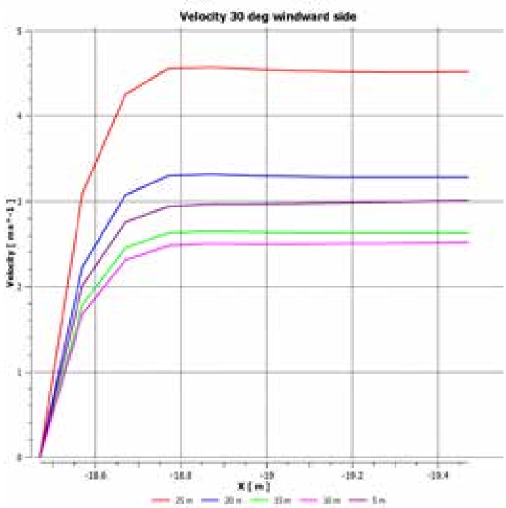
Windward 30 °



ANSYS
Velocity
Contour 1
6.000e+000
5.333e+000
4.667e+000
4.000e+000
3.333e+000
2.667e+000
2.000e+000
1.333e+000
6.667e-001
0.000e+000
[m s^-1]

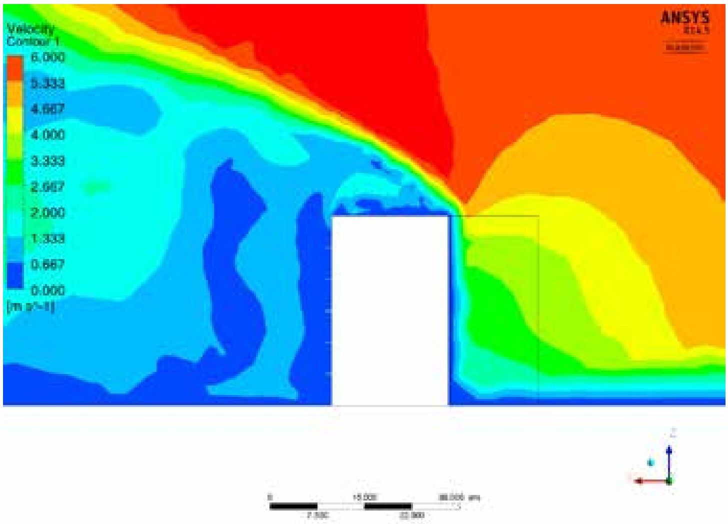


ANSYS
Velocity
Vector 1
1.102e+001
8.268e+000
5.512e+000
2.756e+000
0.000e+000
[m s^-1]

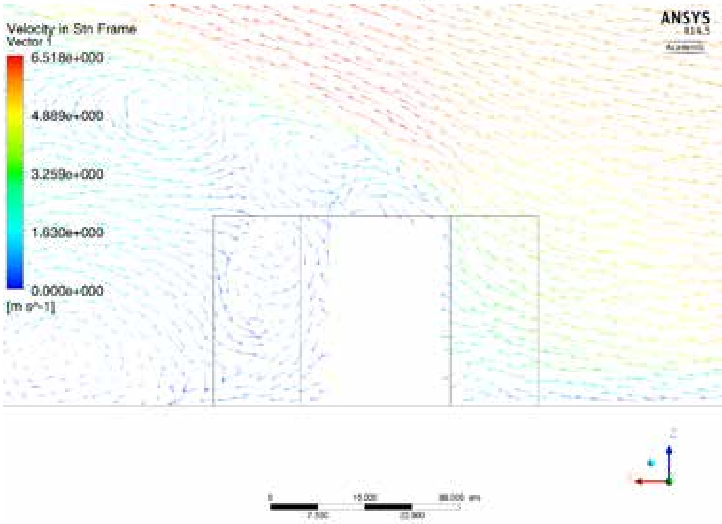


Velocity 30 deg windward side
Velocity [m s^-1]
X [m]

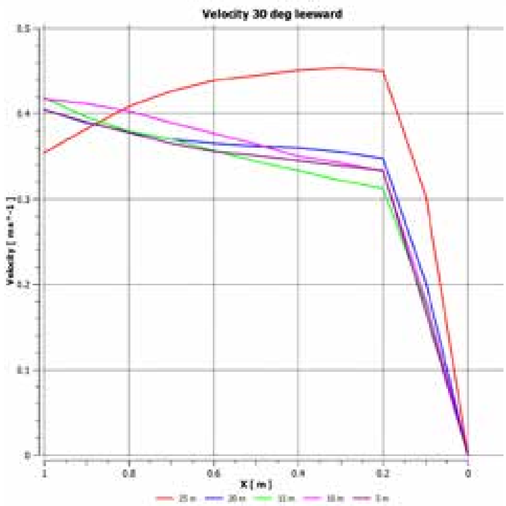
Leeward 30 °



Velocity
Contour 1
6.000
5.333
4.667
4.000
3.333
2.667
2.000
1.333
0.667
0.000
[m s^-1]
ANSYS



Velocity in Stn Frame
Vector 1
6.518e+000
4.889e+000
3.259e+000
1.630e+000
0.000e+000
[m s^-1]
ANSYS



Velocity 30 deg leeward
X [m]

A3. *Wind 60 °*

1. File Report

Table 1. File Information for Windward 60 deg rect mesh

Case	Windward 60 deg rect mesh
File Path	D:\Documents D;\Afstuderen\Calculations\3d files for wind around building\Wind 0 deg mesh_files\dp0\FFF-11\Fluent\FFF-11.1-4-00250.dat.gz
File Date	05 November 2014
File Time	12:25:50 PM
File Type	FLUENT
File Version	14.5.0

2. Mesh Report

Table 2. Mesh Information for Windward 60 deg rect mesh

Domain	Nodes	Elements
model_space	635474	555980

3. Physics Report

Table 3. Domain Physics for Windward 60 deg rect mesh

Domain - model_space	
Type	cell

Table 4. Boundary Physics for Windward 60 deg rect mesh

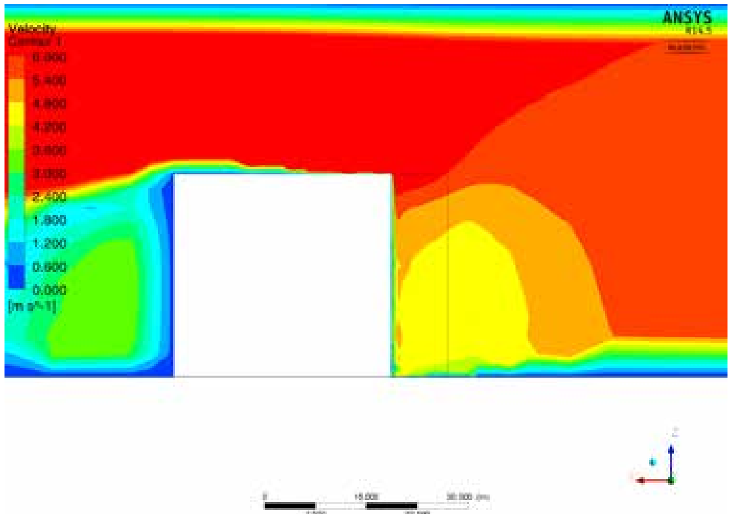
Domain	Boundaries	
model_space	**Boundary - inlet**	
	Type	VELOCITY-INLET
	Boundary - outlet	
	Type	PRESSURE-OUTLET
	Boundary - outlet_top	
	Type	WALL
	Boundary - wall model_space	
	Type	WALL

4. Solver Input

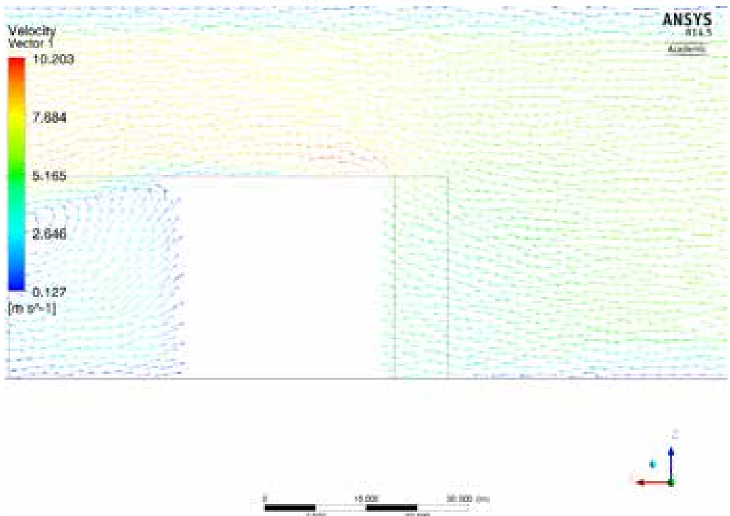
```
Models
------
  Model                     Settings
  -------------------------------------
  Space                     3D
  Time                      Steady
  Viscous                   Laminar
  Heat Transfer             Disabled
  Solidification and Melting   Disabled
  Species                   Disabled
  Coupled Dispersed Phase      Disabled
  NOx Pollutants            Disabled
  SOx Pollutants            Disabled
  Soot                      Disabled
  Mercury Pollutants        Disabled
Material Properties
-------------------
  Material: air (fluid)
    Property                  Units     Method     Value(s)
    ---------------------------------------------------------------
    Density                   kg/m3     constant   1.225
    Cp (Specific Heat)        j/kg-k    constant   1006.43
    Thermal Conductivity      w/m-k     constant   0.0242
    Viscosity                 kg/m-s    constant   1.7894e-05
    Molecular Weight          kg/kgmol  constant   28.966
    Thermal Expansion Coefficient   1/k     constant   0
    Speed of Sound            m/s       none       #f
  Material: aluminum (solid)
    Property               Units    Method    Value(s)
    ---------------------------------------------------
    Density                kg/m3    constant  2719
    Cp (Specific Heat)     j/kg-k   constant  871
    Thermal Conductivity   w/m-k    constant  202.4
Cell Zone Conditions
--------------------
  Zones
    name          id   type
    -------------------------
    *model_space   2   fluid
Boundary Conditions
-------------------
  Zones
    name               id   type
    -----------------------------------------
    outlet_top          7   wall
    inlet               5   velocity-inlet
    outlet              6   pressure-outlet
    wall-*model_space   8   wall
  Setup Conditions
    outlet_top
      Condition                                    Value
    ---------------------------------------
    Absolute Velocity Formulation   yes
  Relaxation
    Variable      Relaxation Factor
    ------------------------------
    Pressure      0.3
    Density       1
    Body Forces   1
    Momentum      0.7
  Linear Solver
                Solver     Termination   Residual Reduction
    Variable    Type       Criterion     Tolerance
    ---------------------------------------------------------
    Pressure     V-Cycle    0.1
    X-Momentum   Flexible   0.1          0.7
    Y-Momentum   Flexible   0.1          0.7
    Z-Momentum   Flexible   0.1          0.7
  Pressure-Velocity Coupling
    Parameter   Value
    ----------------
    Type        SIMPLE
  Discretization Scheme
    Variable   Scheme
    -----------------------------
    Pressure   Standard
    Momentum   Second Order Upwind
```

```
      ---------------------------------------------------------
      Enable shell conduction?                          no
      Wall Motion                               0
      Shear Boundary Condition                          0
      Define wall motion relative to adjacent cell zone?   yes
      Apply a rotational velocity to this wall?         no
      Velocity Magnitude (m/s)                          0
      X-Component of Wall Translation                   1
      Y-Component of Wall Translation                   0
      Z-Component of Wall Translation                   0
      Define wall velocity components?                  no
      X-Component of Wall Translation (m/s)             0
      Y-Component of Wall Translation (m/s)             0
      Z-Component of Wall Translation (m/s)             0
      Rotation Speed (rad/s)                            0
      X-Position of Rotation-Axis Origin (m)            0
      Y-Position of Rotation-Axis Origin (m)            0
      Z-Position of Rotation-Axis Origin (m)            0
      X-Component of Rotation-Axis Direction            1
      Y-Component of Rotation-Axis Direction            0
      Z-Component of Rotation-Axis Direction            0
      X-component of shear stress (pascal)              0
      Y-component of shear stress (pascal)              0
      Z-component of shear stress (pascal)              0
      Fslip constant                            0
      Eslip constant                            0
      Specularity Coefficient                       0
    inlet
      Condition                              Value
      ------------------------------------------------
      Velocity Specification Method            1
      Reference Frame                        0
      Velocity Magnitude (m/s)                 0
      Supersonic/Initial Gauge Pressure (pascal)   0
      Coordinate System                      0
      X-Velocity (m/s)                       6
      Y-Velocity (m/s)                       0
      Z-Velocity (m/s)                       0
      X-Component of Flow Direction            1
      Y-Component of Flow Direction            0
      Z-Component of Flow Direction            0
      X-Component of Axis Direction            1
      Y-Component of Axis Direction            0
      Z-Component of Axis Direction            0
      X-Coordinate of Axis Origin (m)          0
      Y-Coordinate of Axis Origin (m)          0
      Z-Coordinate of Axis Origin (m)          0
      Angular velocity (rad/s)               0
      is zone used in mixing-plane model?        no
    outlet
      Condition                                  Value
      --------------------------------------------------------
      Gauge Pressure (pascal)                        0
      Backflow Direction Specification Method           1

      Coordinate System                            0
      X-Component of Flow Direction                  1
      Y-Component of Flow Direction                  0
      Z-Component of Flow Direction                  0
      X-Component of Axis Direction                  1
      Y-Component of Axis Direction                  0
      Z-Component of Axis Direction                  0
      X-Coordinate of Axis Origin (m)                0
      Y-Coordinate of Axis Origin (m)                0
      Z-Coordinate of Axis Origin (m)                0
      is zone used in mixing-plane model?              no
      Radial Equilibrium Pressure Distribution         no
      Specify Average Pressure Specification            no
      Specify targeted mass flow rate                no
      Targeted mass flow (kg/s)                    1
      Upper Limit of Absolute Pressure Value (pascal)   5000000
      Lower Limit of Absolute Pressure Value (pascal)   1
    wall-*model_space
      Condition                                    Value
      ---------------------------------------------------------
      Enable shell conduction?                          no
      Wall Motion                               0
      Shear Boundary Condition                          0
      Define wall motion relative to adjacent cell zone?   yes
      Apply a rotational velocity to this wall?         no
      Velocity Magnitude (m/s)                          0
      X-Component of Wall Translation                   1
      Y-Component of Wall Translation                   0
      Z-Component of Wall Translation                   0
      Define wall velocity components?                  no
      X-Component of Wall Translation (m/s)             0
      Y-Component of Wall Translation (m/s)             0
      Z-Component of Wall Translation (m/s)             0
      Rotation Speed (rad/s)                            0
      X-Position of Rotation-Axis Origin (m)            0
      Y-Position of Rotation-Axis Origin (m)            0
      Z-Position of Rotation-Axis Origin (m)            0
      X-Component of Rotation-Axis Direction            0
      Y-Component of Rotation-Axis Direction            0
      Z-Component of Rotation-Axis Direction            1
      X-component of shear stress (pascal)              0
      Y-component of shear stress (pascal)              0
      Z-component of shear stress (pascal)              0
      Fslip constant                            0
      Eslip constant                            0
      Specularity Coefficient                       0
Solver Settings
---------------
  Equations
    Equation   Solved
    -----------------
    Flow       yes
  Numerics
    Numeric                  Enabled
```

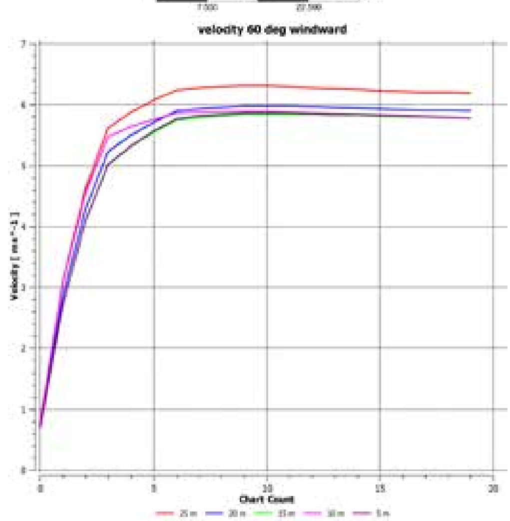
Windward 60 °



Velocity
Contour 1
6.000
5.400
4.800
4.200
3.600
3.000
2.400
1.800
1.200
0.600
0.000
[m s^-1]
ANSYS

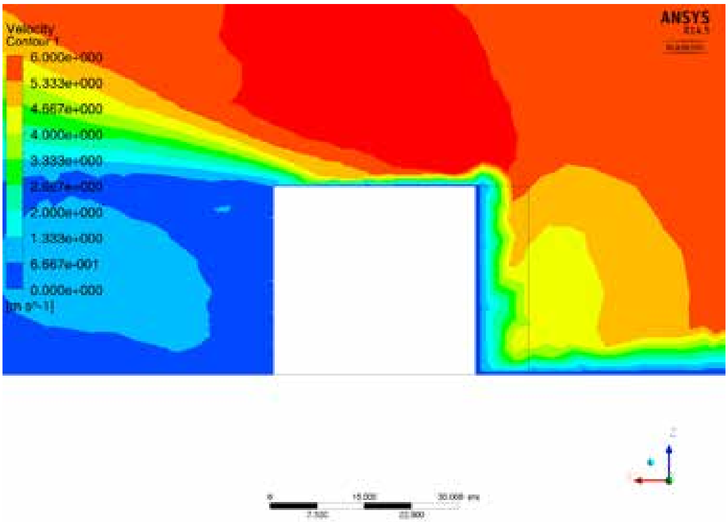


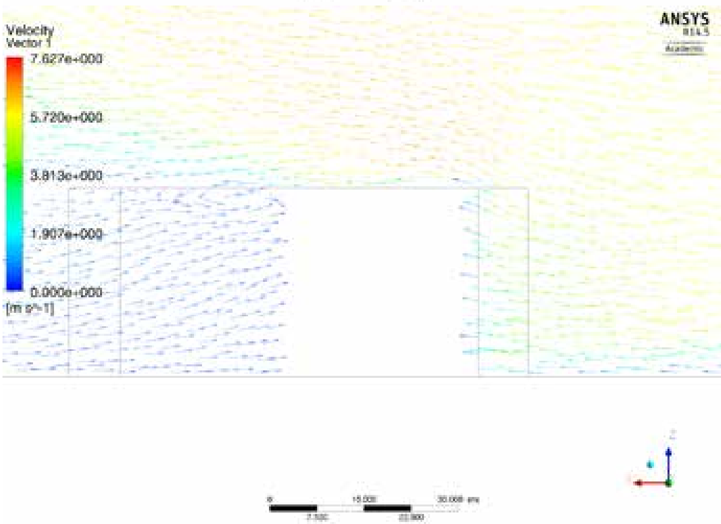
Velocity
Vector 1
10.203
7.684
5.165
2.646
0.127
[m s^-1]
ANSYS

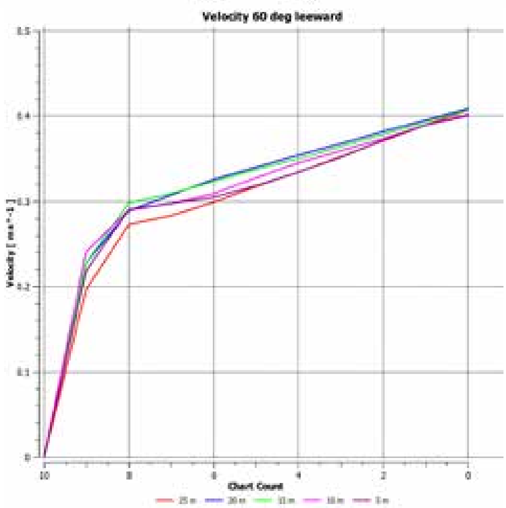


velocity 60 deg windward
Velocity [m s^-1]
Chart Count

Leeward 60 °


Velocity
Contour 1
6.000e+000
5.333e+000
4.667e+000
4.000e+000
3.333e+000
2.667e+000
2.000e+000
1.333e+000
6.667e-001
0.000e+000
[m s^-1]
ANSYS


Velocity
Vector 1
7.627e+000
5.720e+000
3.813e+000
1.907e+000
0.000e+000
[m s^-1]
ANSYS


Velocity 60 deg leeward
Chart Count

A4. *Wind 90 °*

1. File Report

Table 1. File Information for Wind 90 deg rect mesh

Case	Wind 90 deg rect mesh
File Path	D:\Documents D;\Afstuderen\Calculations\3d files for wind around building\Wind 0 deg mesh_files\dp0\FFF-12\Fluent\FFF-12-1-00500.dat.gz
File Date	05 November 2014
File Time	01:16:20 PM
File Type	FLUENT
File Version	14.5.0

2. Mesh Report

Table 2. Mesh Information for Wind 90 deg rect mesh

Domain	Nodes	Elements
model_space	715859	624883

3. Physics Report

Table 3. Domain Physics for Wind 90 deg rect mesh

Domain - model_space	
Type	cell

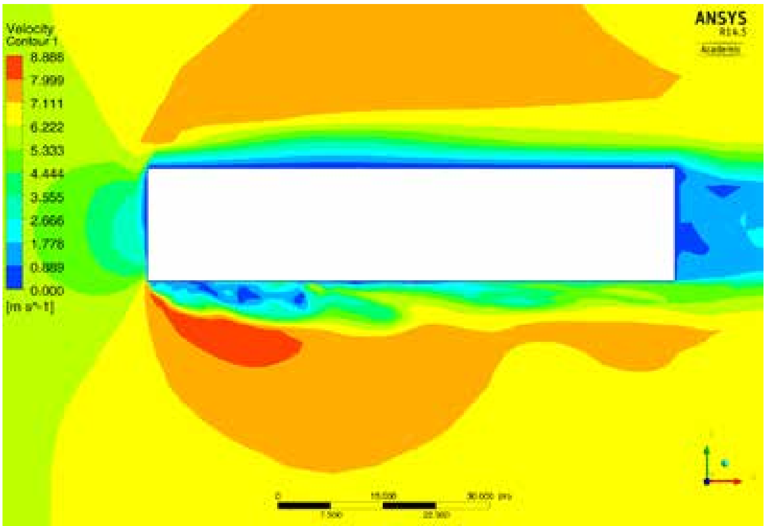
Table 4. Boundary Physics for Wind 90 deg rect mesh

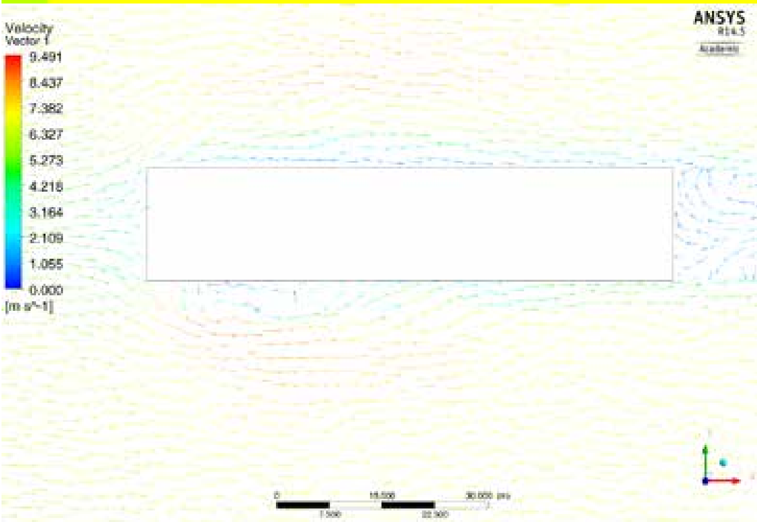
Domain	Boundaries	
model_space	**Boundary - inlet**	
	Type	VELOCITY-INLET
	Boundary - outlet	
	Type	PRESSURE-OUTLET
	Boundary - wall model_space	
	Type	WALL

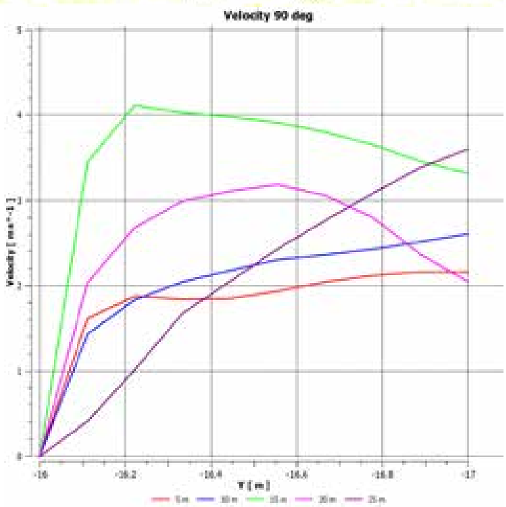
4. Solver Input

```
Models
------
 Model                        Settings
 -------------------------------------
 Space                         3D
 Time                         Steady
 Viscous                       Laminar
 Heat Transfer                 Disabled
 Solidification and Melting   Disabled
 Species                       Disabled
 Coupled Dispersed Phase      Disabled
 NOx Pollutants                Disabled
 SOx Pollutants                Disabled
 Soot                         Disabled
 Mercury Pollutants            Disabled
Material Properties
-------------------
 Material: air (fluid)
   Property                    Units     Method     Value(s)
   ----------------------------------------------------------------
   Density                     kg/m3     constant   1.225
   Cp (Specific Heat)          j/kg-k    constant   1006.43
   Thermal Conductivity        w/m-k     constant   0.0242
   Viscosity                   kg/m-s    constant   1.7894e-05
   Molecular Weight            kg/kgmol  constant   28.966
   Thermal Expansion Coefficient  1/k    constant   0
   Speed of Sound              m/s       none       #f
 Material: aluminum (solid)
   Property              Units    Method    Value(s)
   ---------------------------------------------------
   Density               kg/m3    constant   2719
   Cp (Specific Heat)    j/kg-k   constant   871
   Thermal Conductivity  w/m-k    constant   202.4
Cell Zone Conditions
--------------------
 Zones
   name          id   type
   -------------------------
   *model_space   2    fluid
Boundary Conditions
-------------------
 Zones
   name                id   type
   ----------------------------------------
   inlet               5    velocity-inlet
   outlet              6    pressure-outlet
   wall-*model_space   7    wall
 Setup Conditions
   inlet
     Condition                                  Value
     --------------------------------------------------
     Velocity Specification Method               1
     Reference Frame                             0
     Velocity Magnitude (m/s)                    0
     Supersonic/Initial Gauge Pressure (pascal)  0
     Coordinate System                           0
     X-Velocity (m/s)                            6
     Y-Velocity (m/s)                            0
     Z-Velocity (m/s)                            0
     X-Component of Flow Direction               1
     Y-Component of Flow Direction               0
     Z-Component of Flow Direction               0
     X-Component of Axis Direction               1
     Y-Component of Axis Direction               0
     Z-Component of Axis Direction               0
     X-Coordinate of Axis Origin (m)             0
     Y-Coordinate of Axis Origin (m)             0
     Z-Coordinate of Axis Origin (m)             0
     Angular velocity (rad/s)                    0
     is zone used in mixing-plane model?         no
   outlet
     Condition                                      Value
     --------------------------------------------------------
     Gauge Pressure (pascal)                        0
     Backflow Direction Specification Method        1
     Coordinate System                              0
     X-Component of Flow Direction                  1
     Y-Component of Flow Direction                  0
     Z-Component of Flow Direction                  0
     X-Component of Axis Direction                  1
     Y-Component of Axis Direction                  0
     Z-Component of Axis Direction                  0
     X-Coordinate of Axis Origin (m)                0
     Y-Coordinate of Axis Origin (m)                0
     Z-Coordinate of Axis Origin (m)                0
     is zone used in mixing-plane model?            no
     Radial Equilibrium Pressure Distribution       no
     Specify Average Pressure Specification         no
     Specify targeted mass flow rate                no
     Targeted mass flow (kg/s)                      1
     Upper Limit of Absolute Pressure Value (pascal)  5000000
     Lower Limit of Absolute Pressure Value (pascal)  1
   wall-*model_space
     Condition                                         Value
     ---------------------------------------------------------
     Enable shell conduction?                          no
     Wall Motion                                       0
     Shear Boundary Condition                          0
     Define wall motion relative to adjacent cell zone?  yes
     Apply a rotational velocity to this wall?         no
     Velocity Magnitude (m/s)                          0
     X-Component of Wall Translation                   1
     Y-Component of Wall Translation                   0
     Z-Component of Wall Translation                   0
     Define wall velocity components?                  no
     X-Component of Wall Translation (m/s)             0
     Y-Component of Wall Translation (m/s)             0
     Z-Component of Wall Translation (m/s)             0
     Rotation Speed (rad/s)                            0
     X-Position of Rotation-Axis Origin (m)            0
     Y-Position of Rotation-Axis Origin (m)            0
     Z-Position of Rotation-Axis Origin (m)            0
     X-Component of Rotation-Axis Direction            0
     Y-Component of Rotation-Axis Direction            0
     Z-Component of Rotation-Axis Direction            1
     X-component of shear stress (pascal)              0
     Y-component of shear stress (pascal)              0
     Z-component of shear stress (pascal)              0
     Fslip constant                                    0
     Eslip constant                                    0
     Specularity Coefficient                           0
Solver Settings
---------------
 Equations
   Equation   Solved
   -----------------
   Flow       yes
 Numerics
   Numeric                         Enabled
   ---------------------------------------
   Absolute Velocity Formulation   yes
 Relaxation
   Variable      Relaxation Factor
   -------------------------------
   Pressure      0.3
   Density       1
   Body Forces   1
   Momentum      0.7
 Linear Solver
              Solver     Termination   Residual Reduction
   Variable   Type       Criterion     Tolerance
   --------------------------------------------------------
   Pressure     V-Cycle    0.1
   X-Momentum   Flexible   0.1         0.7
   Y-Momentum   Flexible   0.1         0.7
   Z-Momentum   Flexible   0.1         0.7
 Pressure-Velocity Coupling
   Parameter   Value
   -----------------
   Type        SIMPLE
 Discretization Scheme
   Variable   Scheme
   -----------------------------
   Pressure   Standard
   Momentum   Second Order Upwind
```

Windward/leeward 90 °







B. *In/Outlet simulation Ansys*

Ansys nozzle setup

1. File Report

Table 1. File Information for FFF 14

Case	FFF 14
File Path	D:\Documents D;\Afstuderen\Calculations\Outlet nozzles\outlet nozzles_files\dp0\FFF-14\Fluent\FFF-14-1-01092.dat.gz
File Date	12 November 2014
File Time	10:46:49 AM
File Type	FLUENT
File Version	14.5.0

2. Mesh Report

Table 2. Mesh Information for FFF 14

Domain	Nodes	Elements
model_space	5412	4259

3. Physics Report

Table 3. Domain Physics for FFF 14

Domain - model_space	
Type	cell

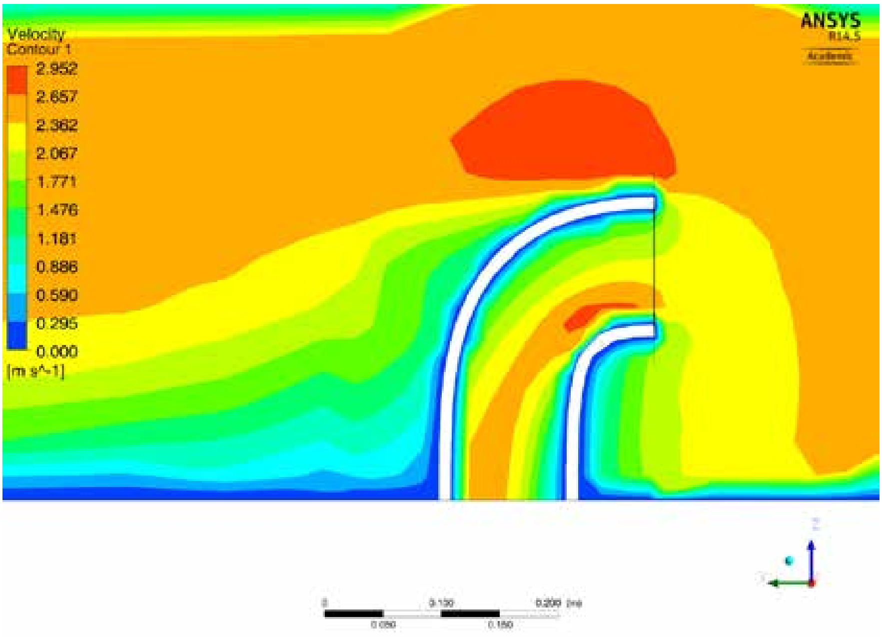
Table 4. Boundary Physics for FFF 14

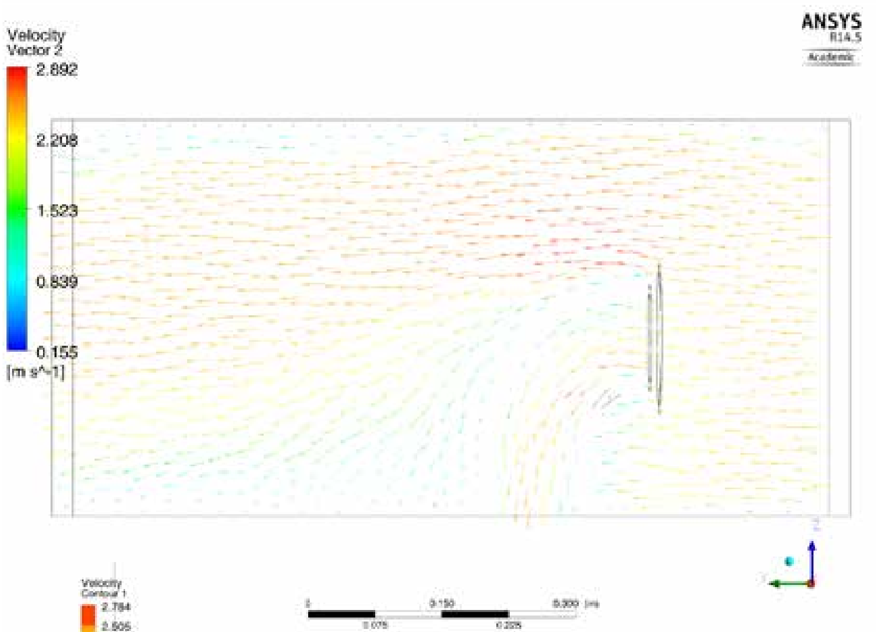
Domain	Boundaries	
model_space	**Boundary - inlet**	
	Type	VELOCITY-INLET
	Boundary - inlet_pipe	
	Type	PRESSURE-INLET
	Boundary - mirror	
	Type	SYMMETRY
	Boundary - outlet	
	Type	PRESSURE-OUTLET
	Boundary - wall model_space	
	Type	WALL

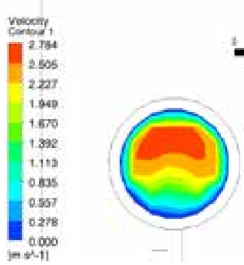
4. Solver Input

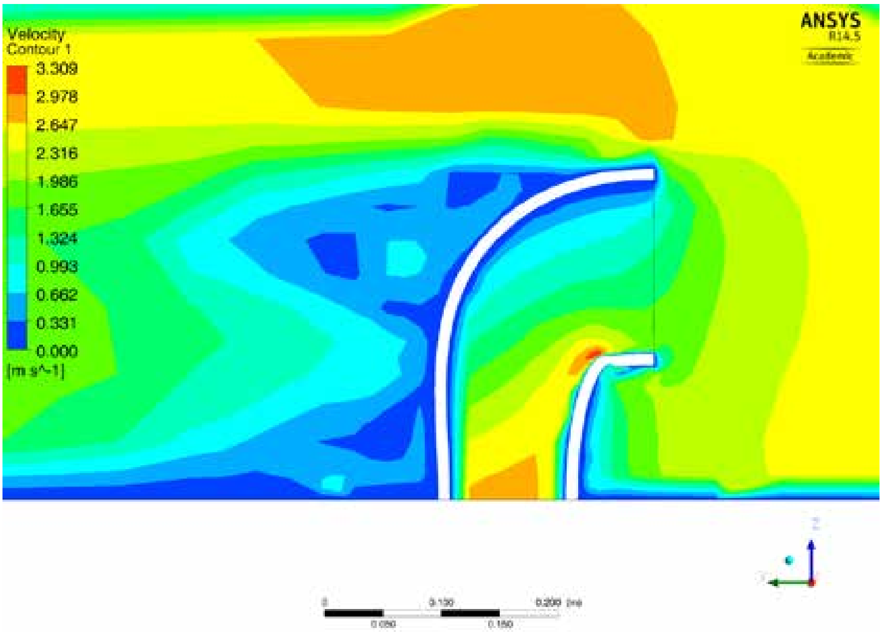
```
Fluent
Version: 3d, dp, pbns, ske (3d, double precision, pressure-based, standard k-epsilon)
Release: 14.5.0
Title:
Models
------
 Model                    Settings
 ---------------------------------------------------------------
 Space                    3D
 Time                     Steady
 Viscous                  Standard k-epsilon turbulence model
 Wall Treatment           Standard Wall Functions
 Heat Transfer            Disabled
 Solidification and Melting   Disabled
 Species                  Disabled
 Coupled Dispersed Phase      Disabled
 NOx Pollutants           Disabled
 SOx Pollutants           Disabled
 Soot                     Disabled
 Mercury Pollutants       Disabled
Material Properties
-------------------
 Material: air (fluid)
   Property                    Units      Method     Value(s)
   ---------------------------------------------------------------
   Density                     kg/m3      constant   1.225
   Cp (Specific Heat)          j/kg-k     constant   1006.43
   Thermal Conductivity        w/m-k      constant   0.0242
   Viscosity                   kg/m-s     constant   1.7894e-05
   Molecular Weight            kg/kgmol   constant   28.966
   Thermal Expansion Coefficient  1/k     constant   0
   Speed of Sound              m/s        none       #f
 Material: aluminum (solid)
   Property               Units    Method     Value(s)
   ---------------------------------------------------
   Density                kg/m3    constant   2719
   Cp (Specific Heat)     j/kg-k   constant   871
   Thermal Conductivity   w/m-k    constant   202.4
Cell Zone Conditions
--------------------
 Zones
   name          id   type
   ------------------------
   *model_space   2    fluid
 Setup Conditions
   *model_space
     Condition                                            Value
     --------------------------------------------------------------
     Material Name                                        air
     Specify source terms?                                no
     Source Terms                                         ()

     Specify fixed values?                                no
     Local Coordinate System for Fixed Velocities         no
     Fixed Values                                         ()
     Frame Motion?                                        no
     Relative To Cell Zone                                -1
     Reference Frame Rotation Speed (rad/s)               0
     Reference Frame X-Velocity Of Zone (m/s)             0
     Reference Frame Y-Velocity Of Zone (m/s)             0
     Reference Frame Z-Velocity Of Zone (m/s)             0
     Reference Frame X-Origin of Rotation-Axis (m)        0
     Reference Frame Y-Origin of Rotation-Axis (m)        0
     Reference Frame Z-Origin of Rotation-Axis (m)        0
     Reference Frame X-Component of Rotation-Axis         0
     Reference Frame Y-Component of Rotation-Axis         0
     Reference Frame Z-Component of Rotation-Axis         1
     Reference Frame User Defined Zone Motion Function    none
     Mesh Motion?                                         no
     Relative To Cell Zone                                -1
     Moving Mesh Rotation Speed (rad/s)                   0
     Moving Mesh X-Velocity Of Zone (m/s)                 0
     Moving Mesh Y-Velocity Of Zone (m/s)                 0
     Moving Mesh Z-Velocity Of Zone (m/s)                 0
     Moving Mesh X-Origin of Rotation-Axis (m)            0
     Moving Mesh Y-Origin of Rotation-Axis (m)            0
     Moving Mesh Z-Origin of Rotation-Axis (m)            0
     Moving Mesh X-Component of Rotation-Axis             0
     Moving Mesh Y-Component of Rotation-Axis             0
     Moving Mesh Z-Component of Rotation-Axis             1
     Moving Mesh User Defined Zone Motion Function        none
     Deactivated Thread                                   no
     Laminar zone?                                        no
     Set Turbulent Viscosity to zero within laminar zone?   yes
     Embedded Subgrid-Scale Model                         0
     Momentum Spatial Discretization                      0
     Cwale                                                0.325
     Cs                                                   0.1
     Porous zone?                                         no
     Conical porous zone?                                 no
     X-Component of Direction-1 Vector                    1
     Y-Component of Direction-1 Vector                    0
     Z-Component of Direction-1 Vector                    0
     X-Component of Direction-2 Vector                    0
     Y-Component of Direction-2 Vector                    1
     Z-Component of Direction-2 Vector                    0
     X-Component of Cone Axis Vector                      1
     Y-Component of Cone Axis Vector                      0
     Z-Component of Cone Axis Vector                      0
     X-Coordinate of Point on Cone Axis (m)               1
     Y-Coordinate of Point on Cone Axis (m)               0
     Z-Coordinate of Point on Cone Axis (m)               0
     Half Angle of Cone Relative to its Axis (deg)        0
     Relative Velocity Resistance Formulation?            yes
     Direction-1 Viscous Resistance (1/m2)                0
     Direction-2 Viscous Resistance (1/m2)                0
     Direction-3 Viscous Resistance (1/m2)                0
     Choose alternative formulation for inertial resistance?   no
     Direction-1 Inertial Resistance (1/m)                0
     Direction-2 Inertial Resistance (1/m)                0
     Direction-3 Inertial Resistance (1/m)                0
     C0 Coefficient for Power-Law                         0
     C1 Coefficient for Power-Law                         0
     Porosity                                             1
     Interfacial Area Density (1/m)                       1
     Heat Transfer Coefficient (w/m2-k)                   1
Boundary Conditions
-------------------
 Zones
   name               id   type
   ---------------------------------------
   inlet               5    velocity-inlet
   outlet_pipe         6    pressure-outlet
   outlet              7    pressure-outlet
   wall-*model_space   8    wall
 Setup Conditions
   inlet
     Condition                                      Value
     ------------------------------------------------
     Velocity Specification Method                  1
     Reference Frame                                0
     Velocity Magnitude (m/s)                       0
     Supersonic/Initial Gauge Pressure (pascal)     0
     Coordinate System                              0
     X-Velocity (m/s)                               0
     Y-Velocity (m/s)                               2.4
     Z-Velocity (m/s)                               0
     X-Component of Flow Direction                  1
     Y-Component of Flow Direction                  0
     Z-Component of Flow Direction                  0
     X-Component of Axis Direction                  1
     Y-Component of Axis Direction                  0
     Z-Component of Axis Direction                  0
     X-Coordinate of Axis Origin (m)                0
     Y-Coordinate of Axis Origin (m)                0
     Z-Coordinate of Axis Origin (m)                0
     Angular velocity (rad/s)                       0
     Turbulent Specification Method                 2
     Turbulent Kinetic Energy (m2/s2)               1
     Turbulent Dissipation Rate (m2/s3)             1
     Turbulent Intensity (%)                        5
     Turbulent Length Scale (m)                     1
     Hydraulic Diameter (m)                         1
     Turbulent Viscosity Ratio                      10
     is zone used in mixing-plane model?            no
   outlet_pipe
     Condition                                      Value
     -------------------------------------------------------
     Gauge Pressure (pascal)                        0
     Backflow Direction Specification Method        1

     Coordinate System                              0
     X-Component of Flow Direction                  1
     Y-Component of Flow Direction                  0
     Z-Component of Flow Direction                  0
     X-Component of Axis Direction                  1
     Y-Component of Axis Direction                  0
     Z-Component of Axis Direction                  0
     X-Coordinate of Axis Origin (m)                0
     Y-Coordinate of Axis Origin (m)                0
     Z-Coordinate of Axis Origin (m)                0
     Turbulent Specification Method                 2
     Backflow Turbulent Kinetic Energy (m2/s2)      1
     Backflow Turbulent Dissipation Rate (m2/s3)    1
     Backflow Turbulent Intensity (%)               5
     Backflow Turbulent Length Scale (m)            1
     Backflow Hydraulic Diameter (m)                1
     Backflow Turbulent Viscosity Ratio             10
     is zone used in mixing-plane model?            no
     Radial Equilibrium Pressure Distribution       no
     Specify Average Pressure Specification         no
     Specify targeted mass flow rate                no
     Targeted mass flow (kg/s)                      1
     Upper Limit of Absolute Pressure Value (pascal)   5000000
     Lower Limit of Absolute Pressure Value (pascal)   1
   outlet
     Condition                                      Value
     -------------------------------------------------------
     Gauge Pressure (pascal)                        0
     Backflow Direction Specification Method        1
     Coordinate System                              0
     X-Component of Flow Direction                  1
     Y-Component of Flow Direction                  0
     Z-Component of Flow Direction                  0
     X-Component of Axis Direction                  1
     Y-Component of Axis Direction                  0
     Z-Component of Axis Direction                  0
     X-Coordinate of Axis Origin (m)                0
     Y-Coordinate of Axis Origin (m)                0
     Z-Coordinate of Axis Origin (m)                0
     Turbulent Specification Method                 2
     Backflow Turbulent Kinetic Energy (m2/s2)      1
     Backflow Turbulent Dissipation Rate (m2/s3)    1
     Backflow Turbulent Intensity (%)               5
     Backflow Turbulent Length Scale (m)            1
     Backflow Hydraulic Diameter (m)                1
     Backflow Turbulent Viscosity Ratio             10
     is zone used in mixing-plane model?            no
     Radial Equilibrium Pressure Distribution       no
     Specify Average Pressure Specification         no
     Specify targeted mass flow rate                no
     Targeted mass flow (kg/s)                      1
     Upper Limit of Absolute Pressure Value (pascal)   5000000
     Lower Limit of Absolute Pressure Value (pascal)   1
   wall-*model_space
     Condition                                          Value
     -----------------------------------------------------------
     Enable shell conduction?                           no
     Wall Motion                                        0
     Shear Boundary Condition                           0
     Define wall motion relative to adjacent cell zone?   yes
     Apply a rotational velocity to this wall?          no
     Velocity Magnitude (m/s)                           0
     X-Component of Wall Translation                    1
     Y-Component of Wall Translation                    0
     Z-Component of Wall Translation                    0
     Define wall velocity components?                   no
     X-Component of Wall Translation (m/s)              0
     Y-Component of Wall Translation (m/s)              0
     Z-Component of Wall Translation (m/s)              0
     Wall Roughness Height (m)                          0
     Wall Roughness Constant                            0.5
     Rotation Speed (rad/s)                             0
     X-Position of Rotation-Axis Origin (m)             0
     Y-Position of Rotation-Axis Origin (m)             0
     Z-Position of Rotation-Axis Origin (m)             0
     X-Component of Rotation-Axis Direction             0
     Y-Component of Rotation-Axis Direction             0
     Z-Component of Rotation-Axis Direction             1
     X-component of shear stress (pascal)               0
     Y-component of shear stress (pascal)               0
     Z-component of shear stress (pascal)               0
     Fslip constant                                     0
     Eslip constant                                     0
     Specularity Coefficient                            0
Solver Settings
---------------
 Equations
   Equation    Solved
   -------------------
   Flow        yes
   Turbulence  yes
 Numerics
   Numeric                        Enabled
   ---------------------------------------
   Absolute Velocity Formulation  yes
 Relaxation
   Variable                     Relaxation Factor
   ---------------------------------------------
   Pressure                     0.3
   Density                      1
   Body Forces                  1
   Momentum                     0.7
   Turbulent Kinetic Energy     0.8
   Turbulent Dissipation Rate   0.8
   Turbulent Viscosity          1
 Linear Solver
                                Solver     Termination   Residual Reduction
   Variable                     Type       Criterion     Tolerance
   -----------------------------------------------------------------------
   Pressure                     V-Cycle    0.1
   X-Momentum                   Flexible   0.1           0.7
   Y-Momentum                   Flexible   0.1           0.7
   Z-Momentum                   Flexible   0.1           0.7
   Turbulent Kinetic Energy     Flexible   0.1           0.7
   Turbulent Dissipation Rate   Flexible   0.1           0.7
 Pressure-Velocity Coupling
   Parameter   Value
   -----------------
   Type        SIMPLE
 Discretization Scheme
   Variable                     Scheme
   ---------------------------------------------
   Pressure                     Standard
   Momentum                     Second Order Upwind
   Turbulent Kinetic Energy     First Order Upwind
   Turbulent Dissipation Rate   First Order Upwind
 Solution Limits
   Quantity                         Limit
   ---------------------------------------
   Minimum Absolute Pressure        1
   Maximum Absolute Pressure        5e+10
   Minimum Temperature              1
   Maximum Temperature              5000
   Minimum Turb. Kinetic Energy     1e-14
   Minimum Turb. Dissipation Rate   1e-20
   Maximum Turb. Viscosity Ratio    100000
```

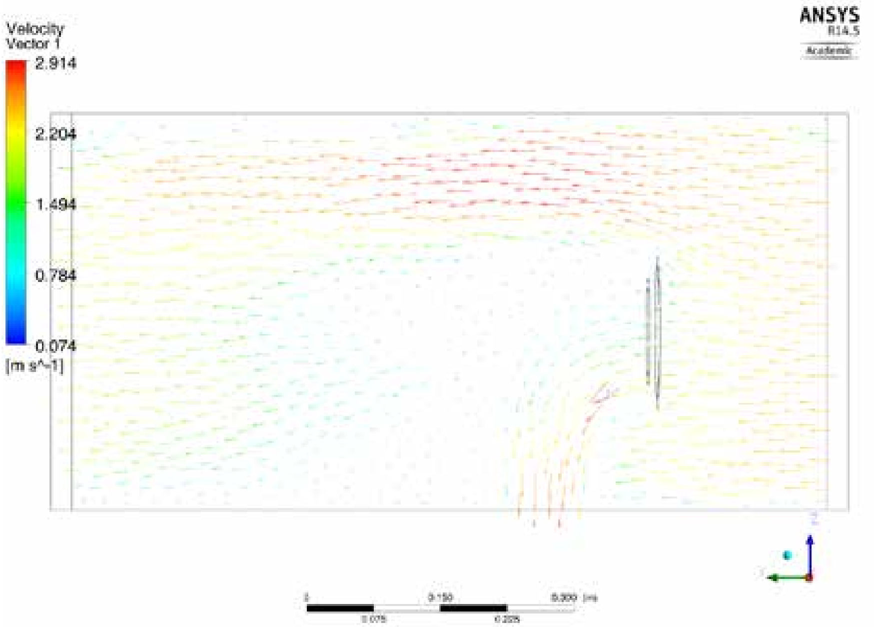
Inlet

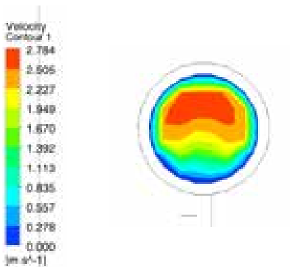

Velocity
Contour 1
2.952
2.657
2.362
2.067
1.771
1.476
1.181
0.886
0.590
0.295
0.000
[m s^-1]
ANSYS
R14.5
Academic


Velocity
Vector 2
2.892
2.208
1.523
0.839
0.155
[m s^-1]
ANSYS
R14.5
Academic

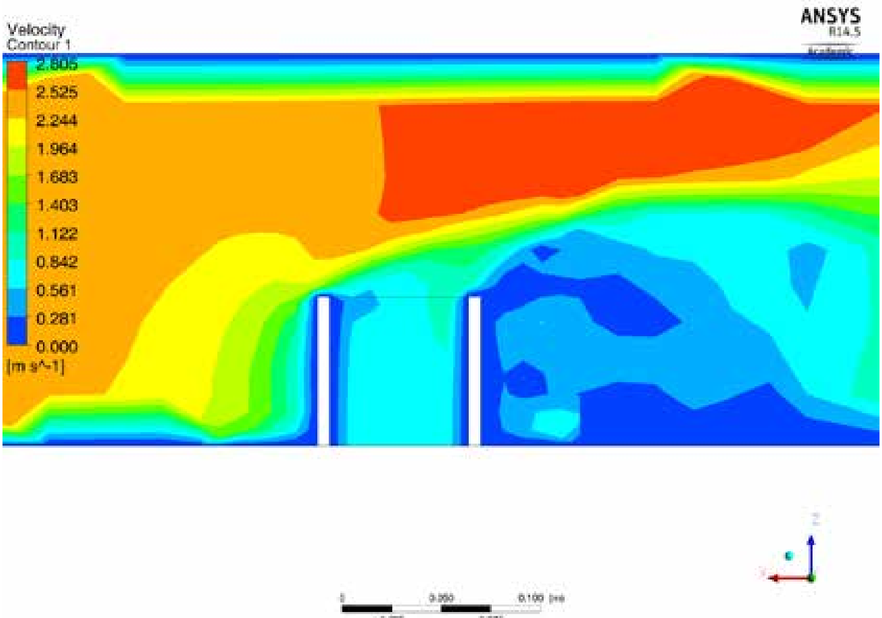

Velocity
Contour 1
2.784
2.505
2.227
1.949
1.670
1.392
1.113
0.835
0.557
0.278
0.000
[m s^-1]

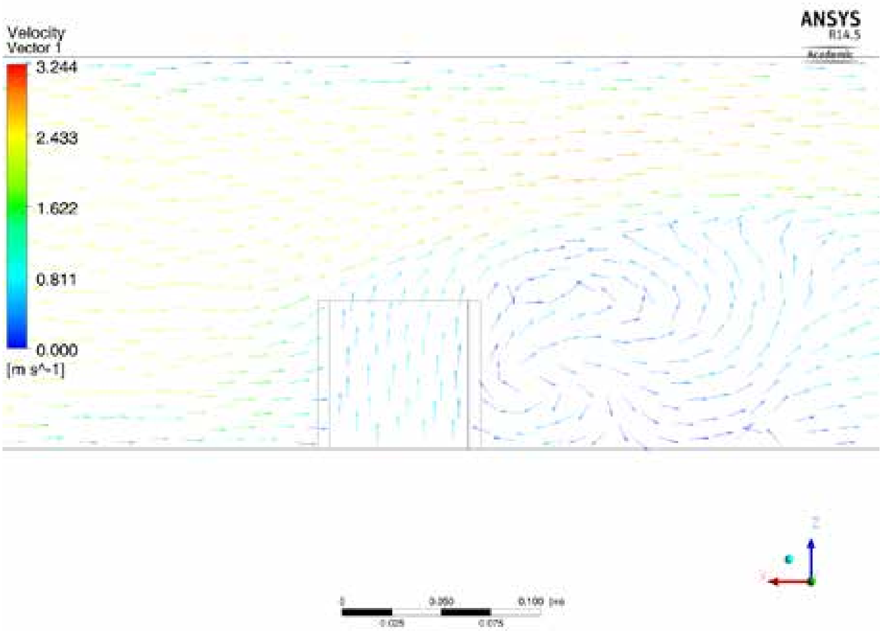

Velocity
Contour 1
3.309
2.978
2.647
2.316
1.986
1.655
1.324
0.993
0.662
0.331
0.000
[m s^-1]
ANSYS
R14.5
Academic
0
0.100
0.200 (m)
0.050
0.150

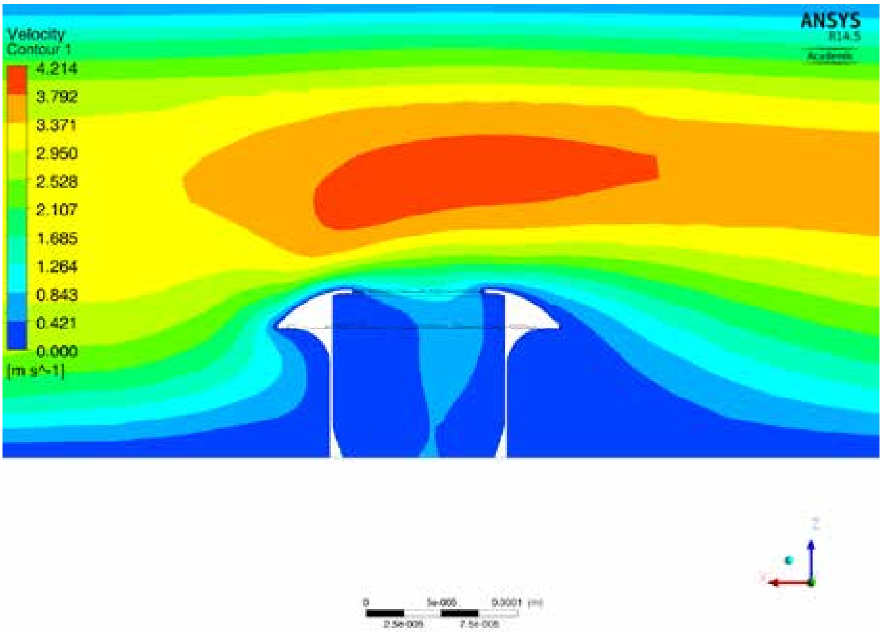

Velocity
Vector 1
2.914
2.204
1.494
0.784
0.074
[m s^-1]
ANSYS
R14.5
Academic
0
0.150
0.300 (m)
0.075
0.225

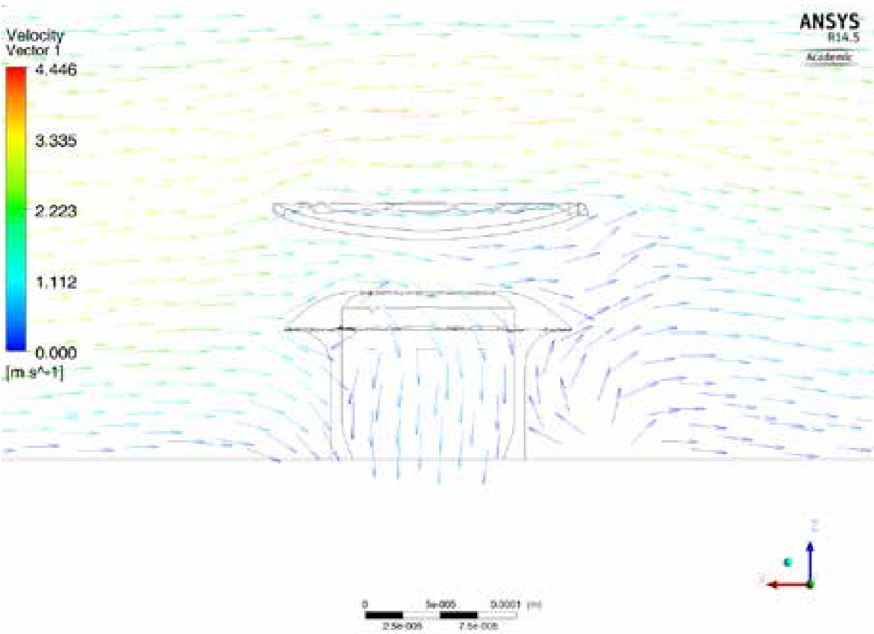

Velocity
Contour 1
2.784
2.505
2.227
1.949
1.670
1.392
1.113
0.835
0.557
0.278
0.000
[m s^-1]

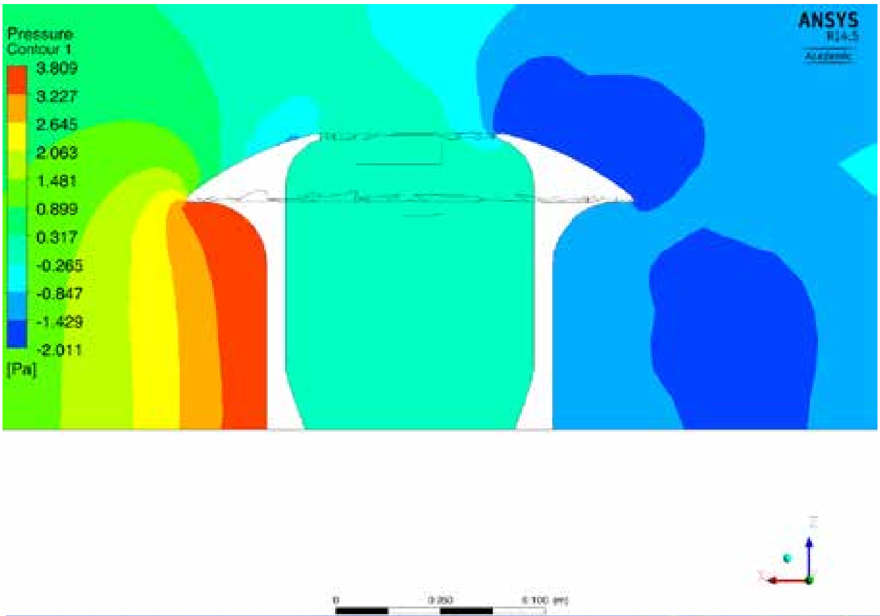
Outlet - Open pipe

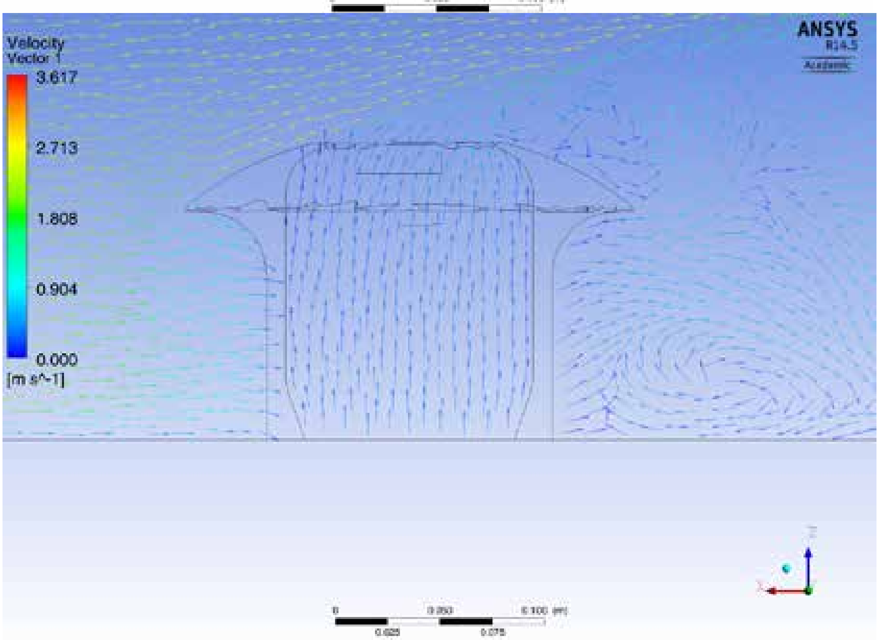




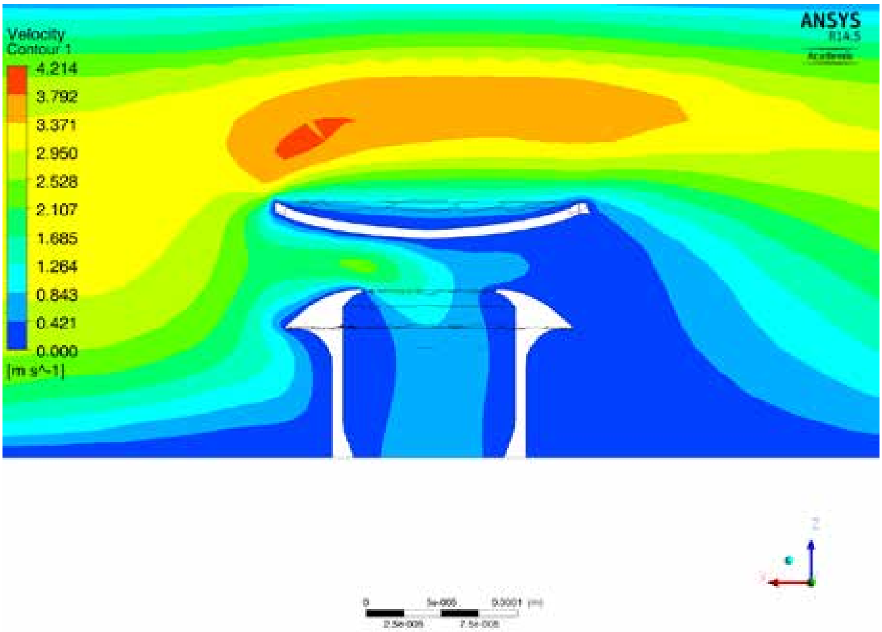

Velocity
Contour 1
4.214
3.792
3.371
2.950
2.528
2.107
1.685
1.264
0.843
0.421
0.000
[m s^-1]
ANSYS
R14.5
Academic

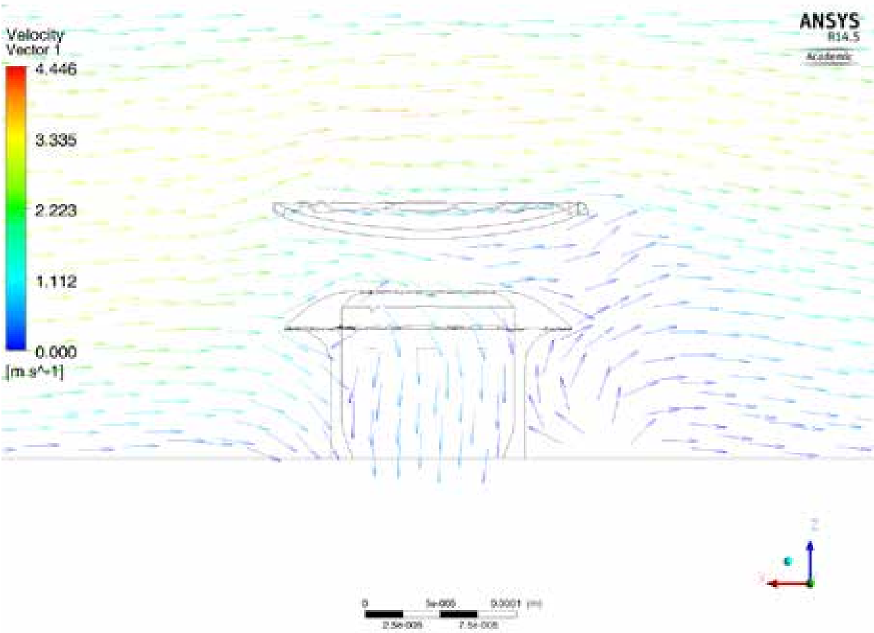

Velocity
Vector 1
4.446
3.335
2.223
1.112
0.000
[m s^-1]
ANSYS
R14.5
Academic

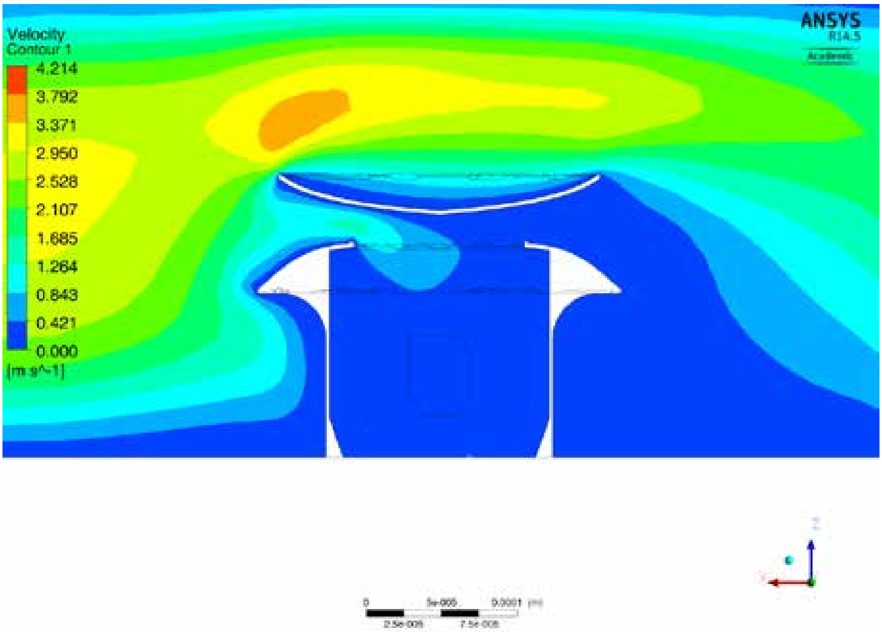

Pressure
Contour 1
3.809
3.227
2.645
2.063
1.481
0.899
0.317
-0.265
-0.847
-1.429
-2.011
[Pa]
ANSYS
R14.5
Academic

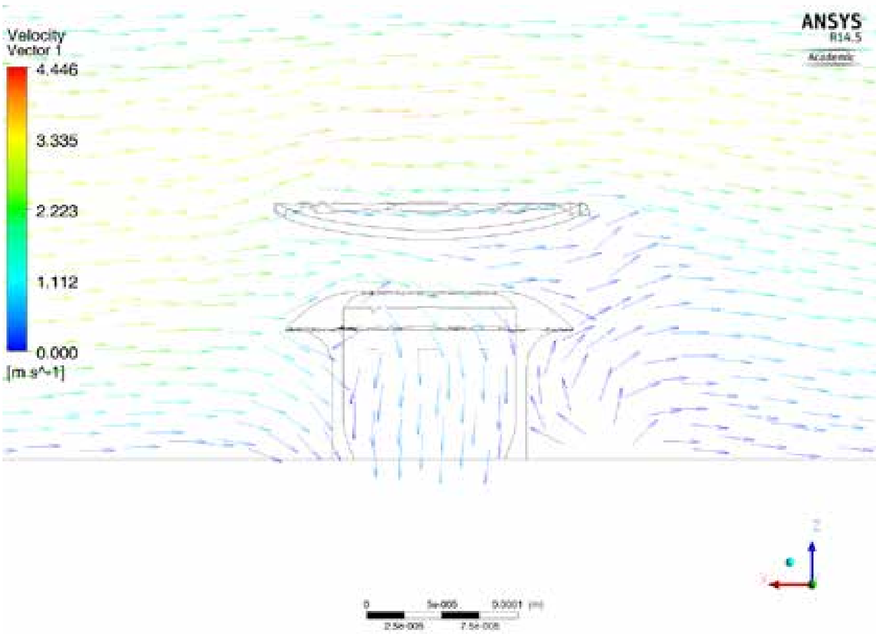

Velocity
Vector 1
3.617
2.713
1.808
0.904
0.000
[m s^-1]
ANSYS
R14.5
Academic

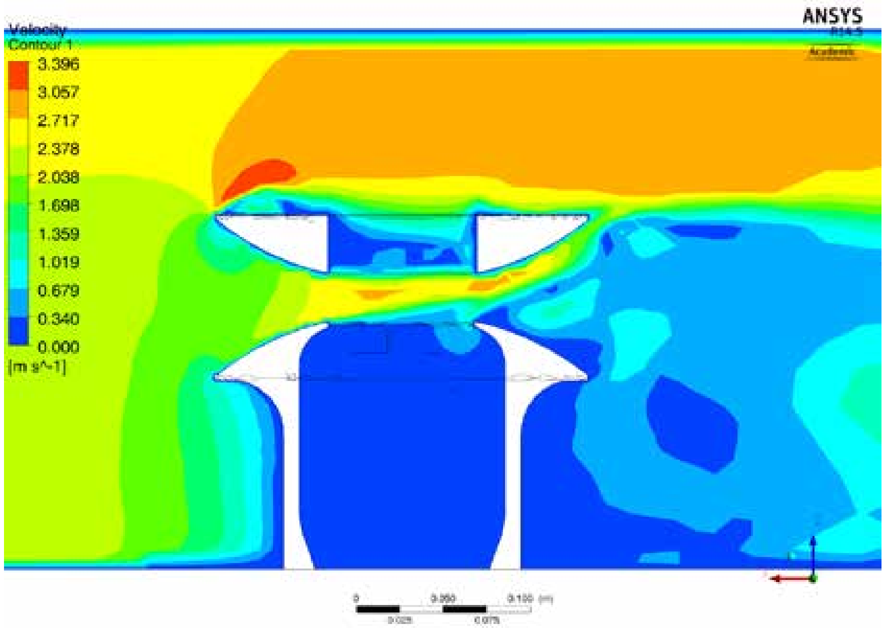
Outlet - Venturi

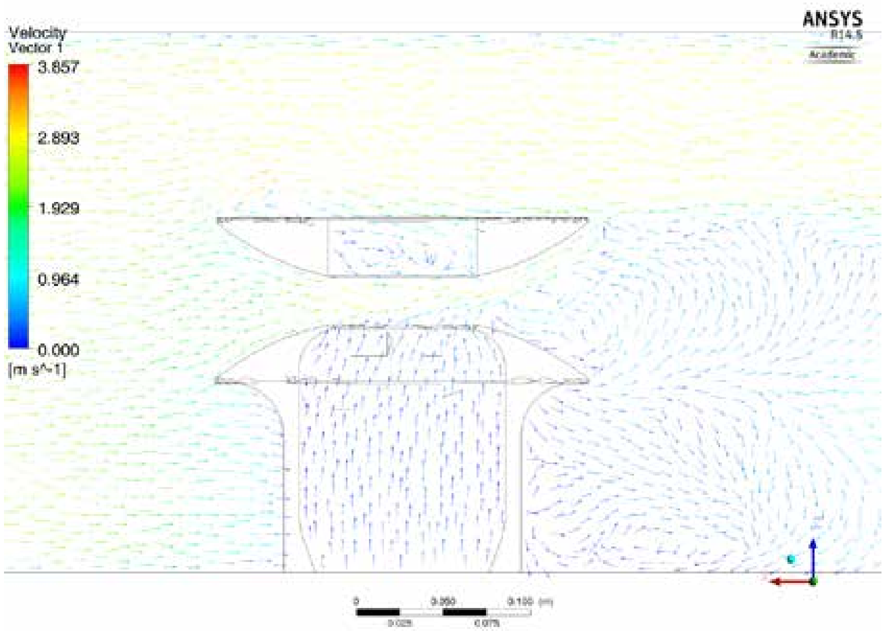


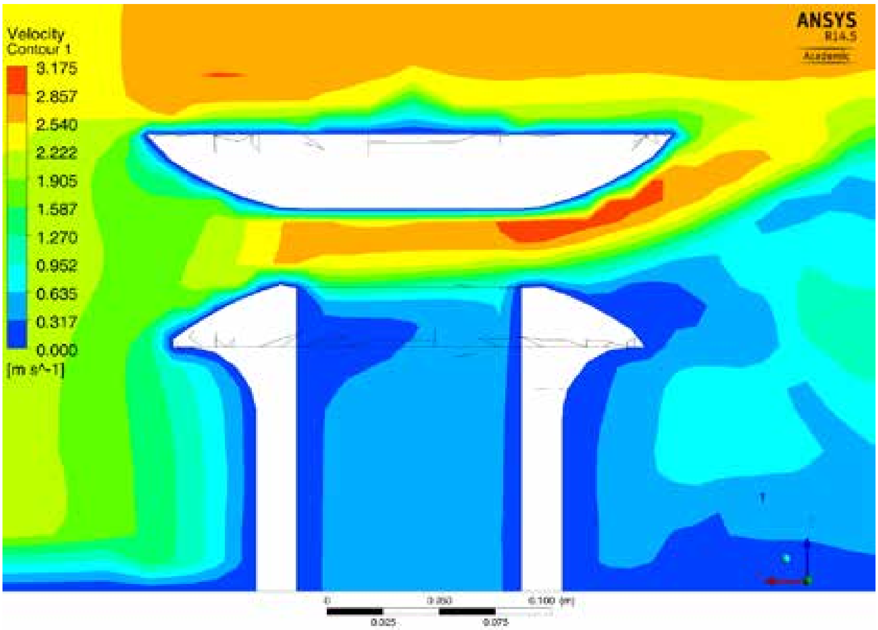


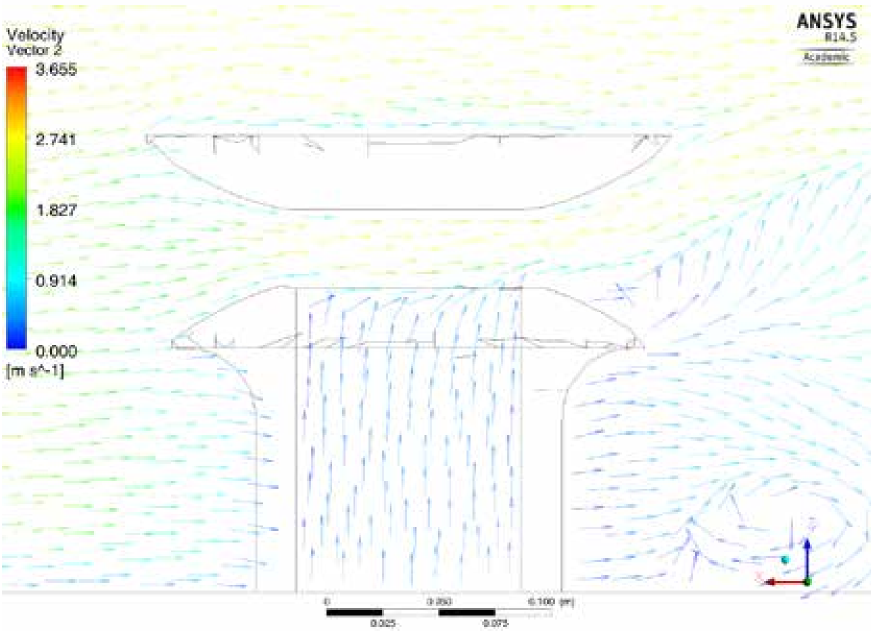

Velocity
Contour 1
4.214
3.792
3.371
2.950
2.528
2.107
1.685
1.264
0.843
0.421
0.000
[m s^-1]
ANSYS
R14.5
Academic

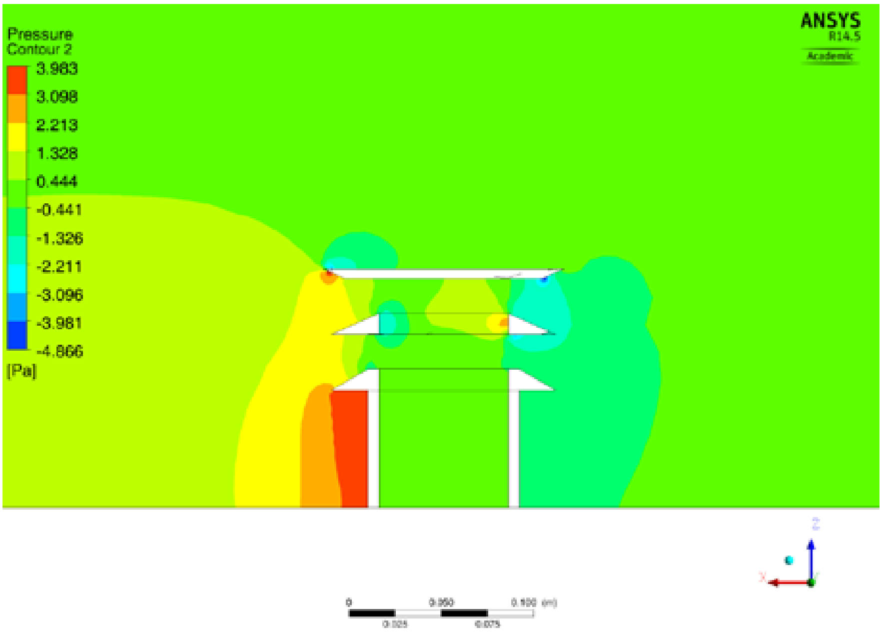

Velocity
Vector 1
4.446
3.335
2.223
1.112
0.000
[m s^-1]
ANSYS
R14.5
Academic

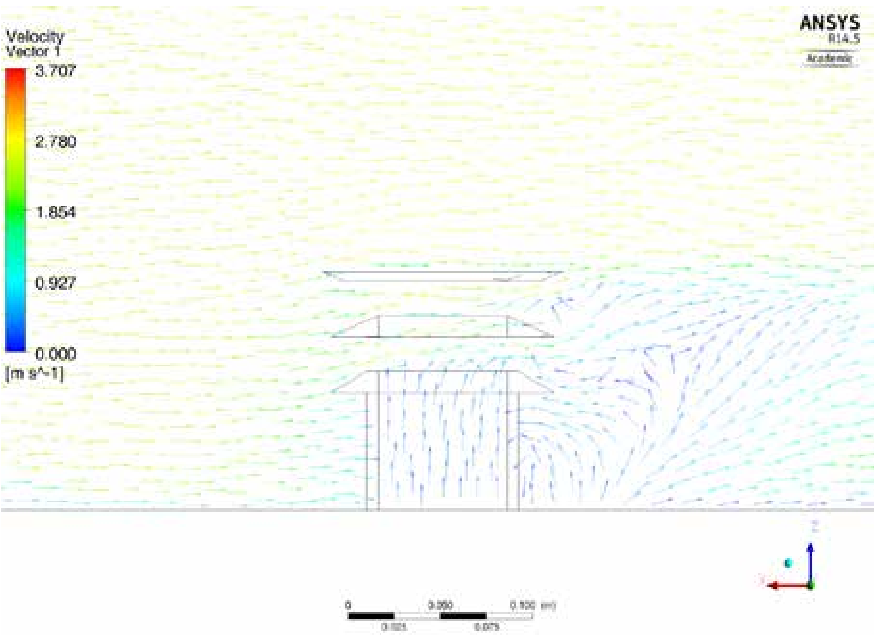

ANSYS
R14.5
Academic
Velocity
Contour 1
3.396
3.057
2.717
2.378
2.038
1.698
1.359
1.019
0.679
0.340
0.000
[m s^-1]

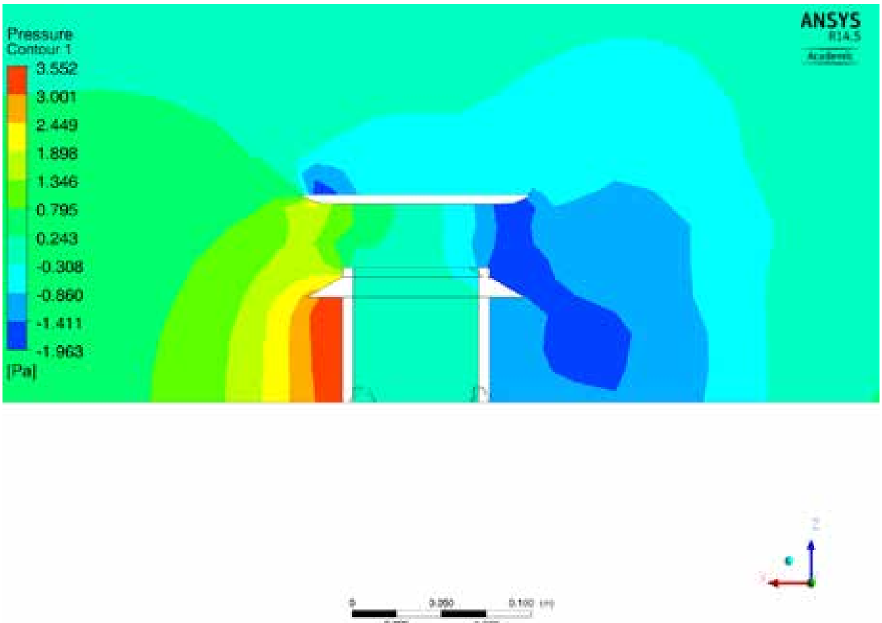

ANSYS
R14.5
Academic
Velocity
Vector 1
3.857
2.893
1.929
0.964
0.000
[m s^-1]

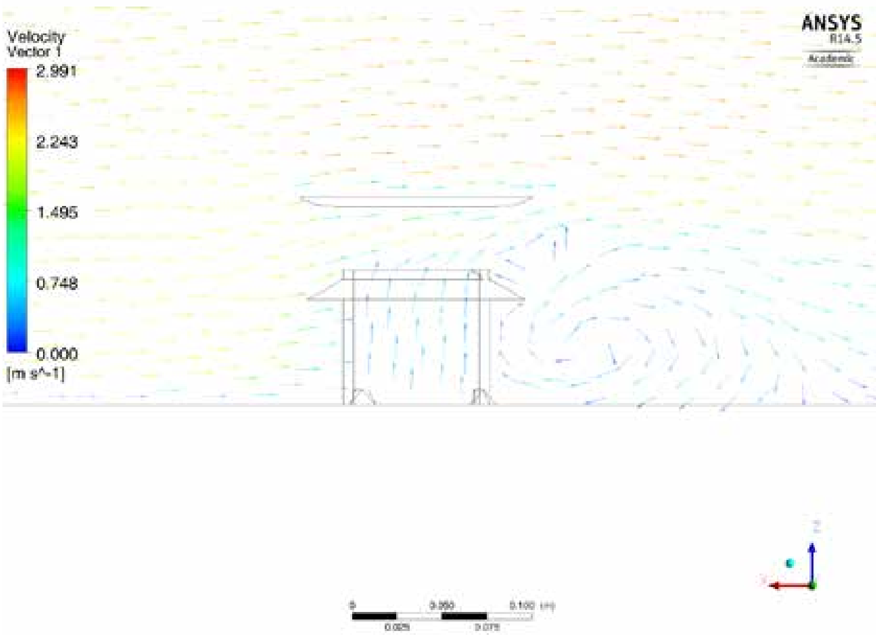

Velocity
Contour 1
3.175
2.857
2.540
2.222
1.905
1.587
1.270
0.952
0.635
0.317
0.000
[m s^-1]
ANSYS
R14.5
Academic

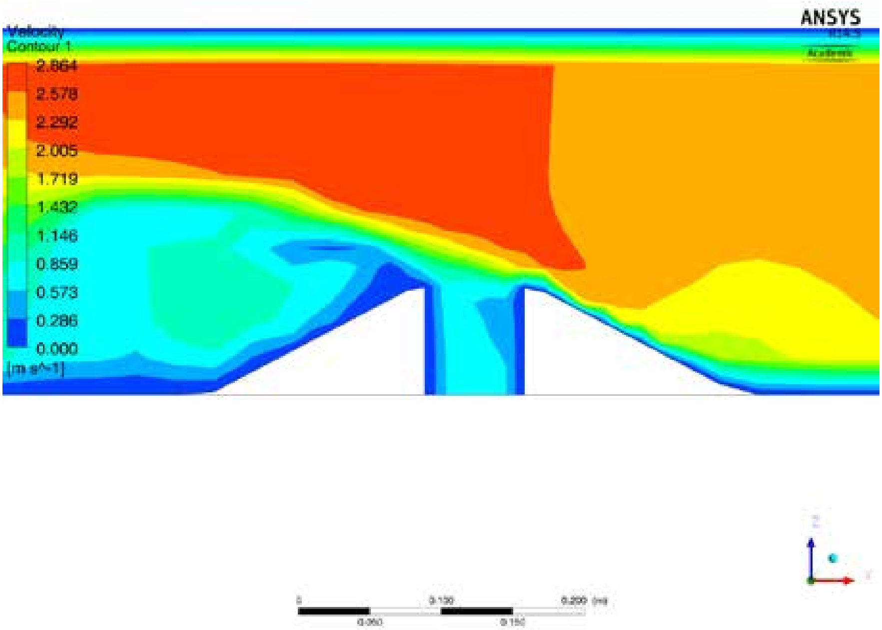

Velocity
Vector 2
3.655
2.741
1.827
0.914
0.000
[m s^-1]
ANSYS
R14.5
Academic

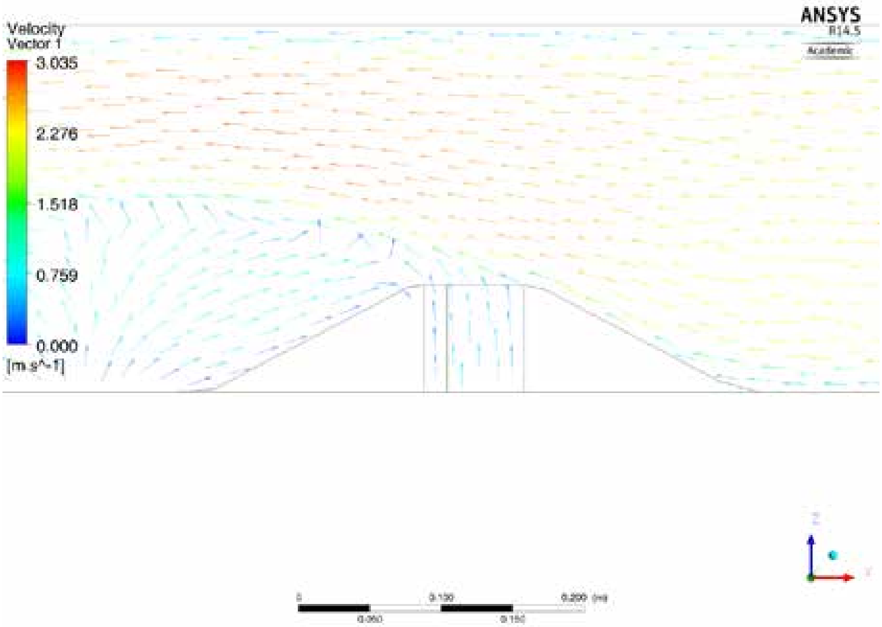

Pressure
Contour 2
3.983
3.098
2.213
1.328
0.444
-0.441
-1.326
-2.211
-3.096
-3.981
-4.866
[Pa]
ANSYS
R14.5
Academic

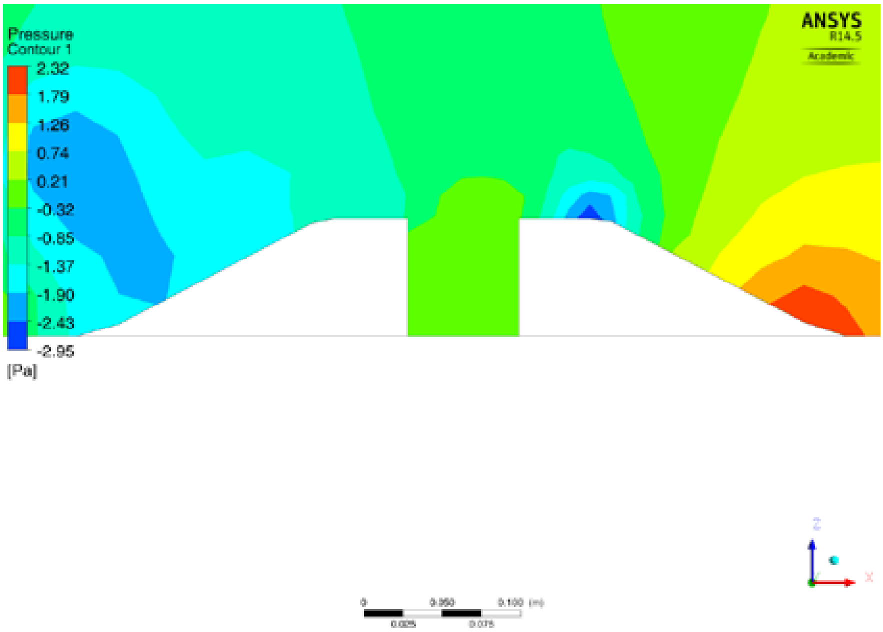

Velocity
Vector 1
3.707
2.780
1.854
0.927
0.000
[m s^-1]
ANSYS
R14.5
Academic


Pressure
Contour 1
3.552
3.001
2.449
1.898
1.346
0.795
0.243
-0.308
-0.860
-1.411
-1.963
[Pa]
ANSYS
R14.5
Academic

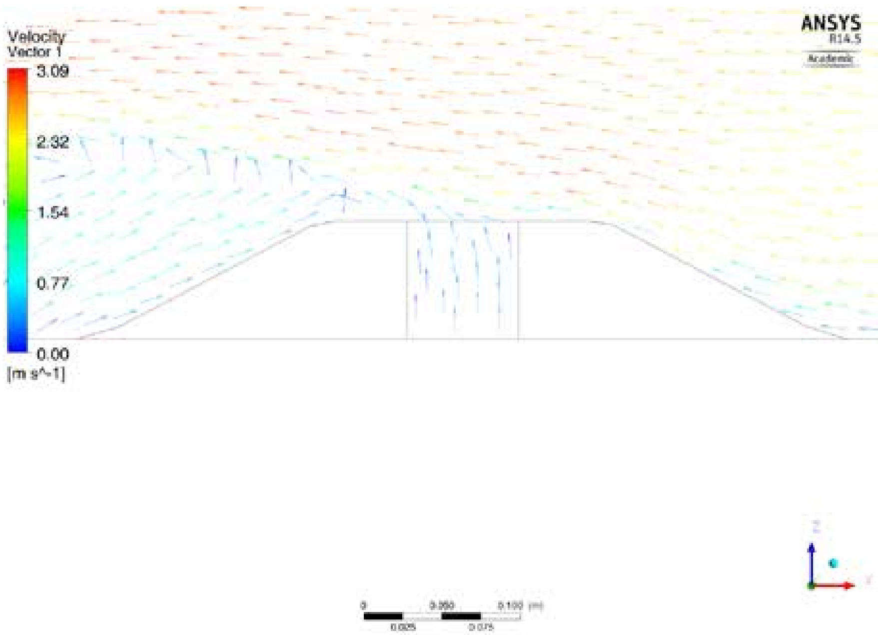

Velocity
Vector 1
2.991
2.243
1.495
0.748
0.000
[m s^-1]
ANSYS
R14.5
Academic


ANSYS
Velocity
Contour 1
2.864
2.578
2.292
2.005
1.719
1.432
1.146
0.859
0.573
0.286
0.000
[m s^-1]

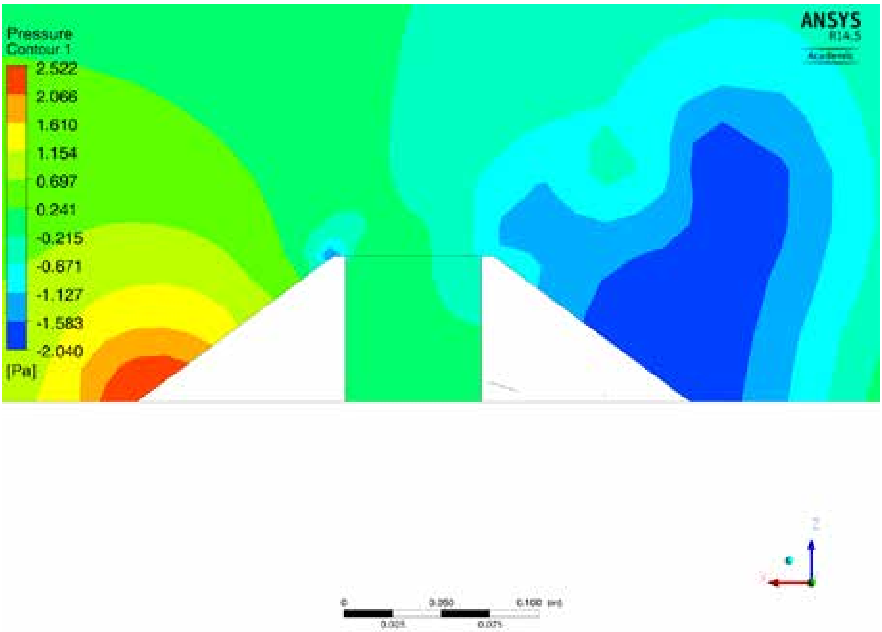

ANSYS
R14.5
Academic
Velocity
Vector 1
3.035
2.276
1.518
0.759
0.000
[m s^-1]

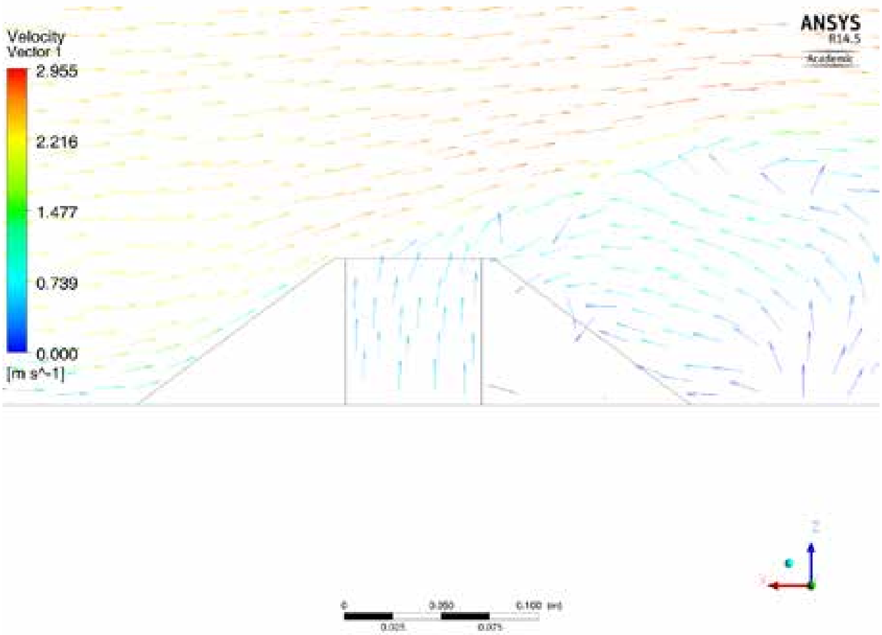


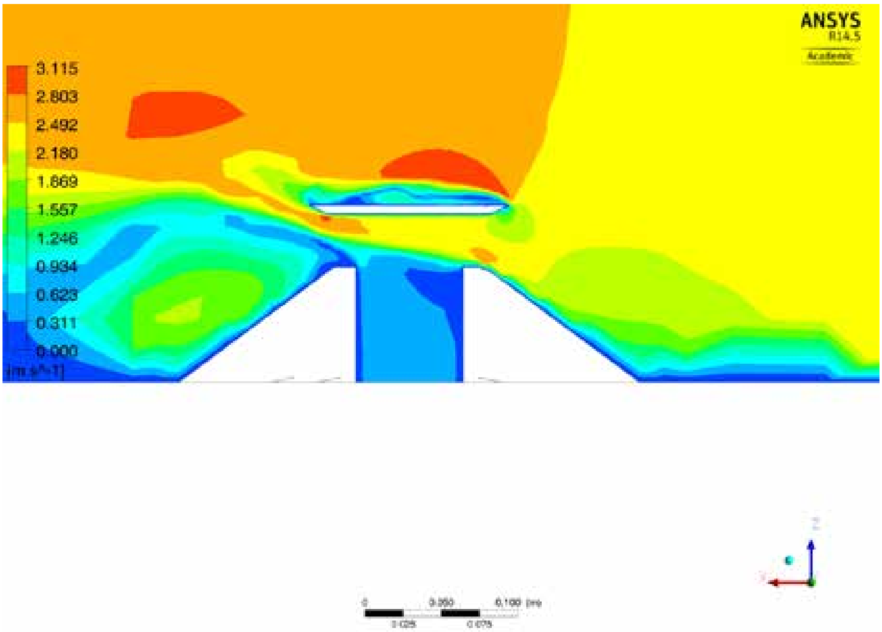
Pressure
Contour 1
2.32
1.79
1.26
0.74
0.21
-0.32
-0.85
-1.37
-1.90
-2.43
-2.95
[Pa]
ANSYS
R14.5
Academic

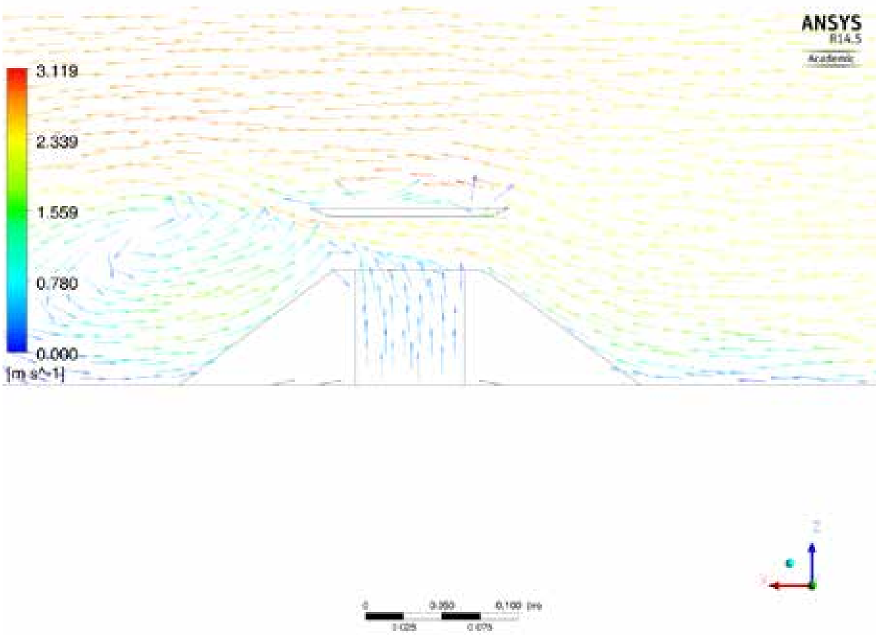


Velocity
Vector 1
3.09
2.32
1.54
0.77
0.00
[m s^-1]
ANSYS
R14.5
Academic


Pressure
Contour 1
2.522
2.066
1.610
1.154
0.697
0.241
-0.215
-0.671
-1.127
-1.583
-2.040
[Pa]
ANSYS
R14.5
Academic


Velocity
Vector 1
2.955
2.216
1.477
0.739
0.000
[m s^-1]
ANSYS
R14.5
Academic


ANSYS
R14.5
Academic
3.115
2.803
2.492
2.180
1.869
1.557
1.246
0.934
0.623
0.311
0.000
[m s^-1]


ANSYS
R14.5
Academic
3.119
2.339
1.559
0.780
0.000
[m s^-1]