Cooling system for solar housing in the Middle East

A design study based on a concept

Master of Science Thesis in the Master's Programme Structural engineering and Building Performance Design

MATTHIEU MAERTEN

AUDE TAN

Department of Energy and Environment

Division of Building Services Engineering

CHALMERS UNIVERSITY OF TECHNOLOGY

Göteborg, Sweden 2011

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Cover:
Drawing of the Solar House by Marcus Rydbo, Norconsult, 2006.

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Abstract

In the 21th century, energy use and resources have become a major issue. This is even tougher in the Middle East, one of the first producers of oil in the world, and also a high consumer. Air conditioning is a huge part of the consumption. On top that, no renewable energy is used at the moment in the region.

The solar house concept studied in the following thesis is a direct answer to this concern. It includes an energy-efficient building using a low-consumption air conditioning system. This system comprises a chiller and a water tank to create and store cooling for the house on a daily basis. The chiller is directly linked to photovoltaic cells.

The aim of this work is to investigate how much electricity can be spared for a family house cooling system with such a design. Simulations are made using a modelling tool to determine the cooling loads and the cooling system behaviour. Especially, the main interest is to get the area of solar panels needed, to balance the energy used to run the cooling system for one year. This study is made for three buildings for two climates.

The results show that the highest heat gain source is the solar radiations. Care should be taken to reduce the windows areas. Besides, the choice of chiller and the strategy used can highly influence size of the water tank, although the solar panels area is reasonable. Both might be further decreased, if the strategy used is more precisely defined throughout the year.

Key words:

Cooling system, solar house, storage tank, fan coil, chiller, energy efficient, vapour-compression

Système de climatisation pour un projet de « maison solaire » au Moyen-Orient
Une étude basée sur un concept

Master of Science Thesis in the Master's Programme Structural engineering and Building Performance Design

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Résumé

Au 21eme siècle, l'utilisation des ressources du sol, la production et la consommation d'énergie sont devenus des problèmes majeurs. Cela est d'autant plus vrai au Moyen-Orient, première région productrice de pétrole. Etant donné l'abondance et le faible coût de cette ressource, cette région en est aussi une importante consommatrice. Les systèmes de climatisation, dans ces pays aux températures élevées, participent pour beaucoup dans cette consommation. De plus, aucune énergie renouvelable n'est utilisée à ce jour dans ces pays.

La maison solaire « Solar house », étudiée dans cette thèse de master, est une réponse directe à ce problème. Elle est composée d'un bâtiment à faible demande énergétique, utilisant un système de climatisation basse-consommation. Ce système comprend une machine frigorifique, pour créer du froid, et un réservoir à eau, pour le stocker et le réutiliser pendant la nuit, et cela pour des cycles journaliers. La machine frigorifique est alimentée par des panneaux photovoltaïques.

Le but de cette thèse de master est d'analyser et de calculer la quantité d'électricité consommée pour la climatisation par une famille avec ce concept. Les simulations sont réalisées avec un outil de modélisation. Elles permettent de déterminer la demande de froid et d'obtenir ainsi les conditions intérieures idéales, mais aussi de calculer la surface de panneaux solaires nécessaire pour compenser l'électricité utilisée à faire fonctionner la machine frigorifique. Les simulations sont toujours réalisées sur une année. L'analyse est faite sur trois maisons et pour deux climats différents.

Les résultats montrent que la source qui apporte le plus de chaleur au bâtiment vient des radiations solaires. La surface vitrée de la maison doit donc être considérée avec attention. Le choix de la machine frigorifique, la taille du réservoir à eau et la surface de panneaux solaires dépendent les uns des autres. En fonction de la stratégie utilisée pour produire la quantité de froid nécessaire, ces différents paramètres sont optimisés.

Mots clés : Système de climatisation, maison solaire, réservoir à eau, machine frigorifique, bâtiment basse-consommation, ventilo-convecteur, réfrigérateur à compression de vapeur.

Contents

Preface

The study, carried out from January to June 2011, is part of a bigger housing project in Saudi Arabia, directed by the company Norconsult. This thesis was enabled thanks to the support from the companies Norconsult and Agera, as well as the Department of Civil and Environmental Engineering, Chalmers University of Technology, Sweden.

Also, we would like to thank our supervisors Marcus Rydbo from Norconsult and Mattias Larsson from Agera, for their support during this master thesis. We highly appreciate working in Norconsult's office, as part of the company. For the technical support, Kent Barry from Agera has been really helpful all along the project with his advice, especially when it came to HVAC design. We are also very grateful to Angela Sasic Kalagasidis for her general guidance and specific advice on how to model the building, as well as her help to get the different climate data. Finally, we are thankful to our examiner Jan Gustén for his participation to the elaboration of this report and his connection to Chalmers.

Göteborg May 2011

Matthieu Maerten
Aude Tan

Notations

Roman upper case letters

A	area [m^2]
A_g	windows area [m^2]
B	width [m]
C	thermal capacity [-]
COP	coefficient of performance [-]
F	view factor between the surfaces [-]
F_{sky_surf}	view factor between the sky and the given surface [-]
G	indoor moisture production [kg/h]
I_{lw}	irradiation of the surface due to long-wave radiations [W/m^2]
I_{sol}	irradiation of the surface due to sun radiations [W/m^2]
L	length [m]
Q	heat flow rate [W]
RH	Relative Humidity [%]
T	temperature [K]
$\bar{T}_{12}$	mean temperature of two surfaces [K]
T_0	annual average outdoor temperature [K]
T_{in}	indoor temperature [K]
U	thermal transmittance [W/m^2.K]
$\dot{V}$	volume flow [l/s]
V	volume [l]
W	compressor work [W]

Roman lower case letters

c	specific heat capacity [J/kg,K]
c_p	specific heat capacity [J/ m3.K]
d	thickness[m]
d_p	periodic penetration depth [m]
dh	height of the shadow [m]
d_{hm}	length of the overhang [m]
g	density of moisture flow rate [kg/s]
h	enthalpy [J/kg]
hp	height of the wall [m]
n	air exchange rate [h^{-1}]
q	heat flow rate [W/m^2]
r	reflected solar radiation [-]
t	temperature [°C]
t^r	sky temperature [°C]
t_a	air temperature [°C]
t_{eq}	equivalent temperature [°C]
t_{ground}	ground temperature [°C]
t_{out}	outdoor temperature [°C]
t_p	time period [s]

u	wind speed [m/s]
w	moisture content in the material [kg/m^3]
x	moisture content in the air [kg/m^3]
x_s	moisture at saturation in the air [kg/m^3]

Greek lower case letters

α_{sol}	absorptivity of the solar radiations through a material [-]
α_c	convective coefficient α_c [W/m^2.K]
α_e	total heat transfer coefficient at the surface [W/m^2.K]
α_r	heat transfer coefficient due to long wave radiation [W/m^2.K]
δ_v	vapour permeability of the material [mm^2/s]
ε	emissivity of the surface [-]
γ	height of the sun [rad]
λ	thermal conductivity [W/m,K]
η_t	temperature efficiency [%]
η_x	moisture efficiency [%]
ρ	density of the material[kg/m^3]
Φ	azimuth of the sun [rad]
τ	transmittance for solar radiation [-]
θ	incidence angle of the sun [rad]

1 Introduction

In this part will be described first the background leading to this project, then the scope of this thesis.

1.1 Background

Saudi Arabia is one of the biggest oil producers in the world. This resource is abundant and cheap there. Artificially low power price has increased the demand on electric utilities (averaging 5 to 7 percent annual growth). On the graph below, there is a real increase in electricity consumption (twice as big for the last ten year), and it is just a start. With the ever increasing life standard in this developing country, and the fast demographic growth (2% per year), the electricity demand will be much higher in the next few years. Saudi Arabia's Water and Electricity Ministry estimates that the country will require at least 35 Gigawatts (GW) of additional power generating capacity by 2023-2025 – more than double the capacity in 2005. Also, building new power plants cannot be done fast enough to cope the demand of the country. Moreover, these only run on non-renewable sources, like oil and gas.

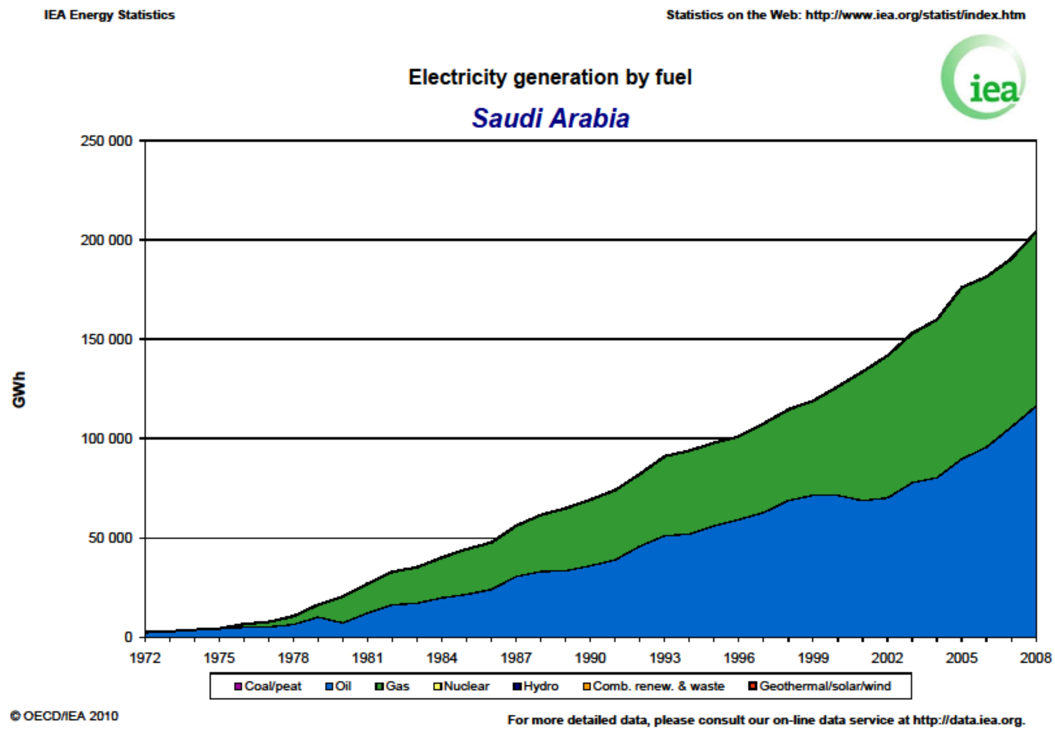


Figure 1 Electricity [GWh] generated in Saudi Arabia (IEA, 2011)

Saudi Arabia is, with Japan, one of the country in which the highest amount of electricity is produced from oil resources (116 TWh in 2008). As to avoid producing even more energy in the next few years and building power plants all over the country, something has to be done. This is even more needed as energy transportation and electricity supply to each and every building create huge losses due to the big size of the country.

Table 1 Countries with the highest electricity production from oil during 2008 (IEA i.e., 2011)

Oil	TWh
Japan	139
Saudi Arabia	116
United States	58
Mexico	49
Indonesia	43
Iraq	36
Kuwait	36
Islamic Rep. of Iran	36
India	34
Pakistan	32
Rest of the world	532
World	1 111

2008 data

Table 2 Electricity production and consumption of electricity in Saudi Arabia during 2008(IEA, 2011)

Electricity	Unit: GWh
Production from:	
- oil	116238
- gas	87962
- other sources	0
Total Production	204200
Energy Industry Own Use	16668
Losses	17472
Final Consumption	170060
Industry	21144
Residential	96687
Commercial and Public Services	48529
Agriculture / Forestry	3461
Other Non-Specified	239

Moreover, ventilation and air conditioning systems represent 65% of the total electricity consumption in buildings (Syed Hasnain, 1999). Especially in summer, the demand to cool a building can be really high (peak load). The electricity production in Saudi Arabia can hardly follow this increase in consumption and, often in summer, shutdowns of the network occur. More power plants should be created to face the demand for these specific days.

The electricity production company in Saudi Arabia, SCECO has created a load demand reduction programme to limit the amount of electricity used for air conditioning of office buildings between 13h00 and 17h00 (the peak load period) during the summer. (SM Hasnain, 2000).

Each building should create its own electricity to minimise its transportation and all the effects from the increase in electricity demand. Therefore, photovoltaic panels to create electricity are a very suitable solution. Even more, using photovoltaic panels to cool a building will match together the electricity production and consumption.

For now, renewable energy production does not exist in Saudi Arabia, even if there is a high potential with solar energy, due to the typical weather condition in this area.

There are two ways to cool a building with solar energy. One is to use solar thermal collectors to heat up circulating water which will be utilized for cooling purpose (absorption or desiccant cooling). The other is to use electricity to run a cooling machine (vapour compression).

1.2 Scope

In this part will be defined all the structure of the master thesis, among which, the scope, outlines, expectations and limitations.

In this master thesis, the focus is made on a project developed by 2 companies (Norconsult and Agera) to create residential houses with a good indoor thermal comfort in Saudi Arabia. In this project, a system has been developed and is patent pending, using photovoltaic panels (PV) to cool a building. As these PV panels can only work during the day, a storage system has been created to be able to cool the building even at night. This also leads to reducing the electricity demand during the peak load.

As seen in Table 2, there is a real need of this kind of system, as most of the electricity produced in Saudi Arabia is used by residential buildings (96687 GWh on 170060 GWh available from the production plant). This energy source will also be environmental-friendlier than oil or gas. The amount of carbon dioxide rejected to the atmosphere to produce electricity will be much lower.

It is to be mentioned that part of this project has already been considered before. Norconsult have already established the house design. Together with Agera, they have also defined the cooling system, transforming solar energy to cool the air. Some drawings, explanation texts and cost calculations have been made, and can be used further as ground for this thesis. The project has also been presented in Saudi Arabia. Yet, no deep analysis has already been done.

Therefore, this thesis is the opportunity to investigate further, in 20 weeks, the whole system comprising the energy-efficient design of the house and the cooling system as described earlier. The aim is to design the cooling system as a viable solution. This work will be organised as following.

To start, the concept is expounded, as to understand its principles, advantages and possible improvements. Then, each part of it will be designed in detail.

First, the input data will be determined: the indoor climate, the outdoor climate and the building characteristics. Two different climates are considered, both in Saudi Arabia: Riyadh, with hot and dry weather, and cool nights; and Jeddah, hot and humid weather with warmer nights. As mentioned before, the building has been clearly designed by Norconsult's architects. For this reason, its shape or layout will not be modified. The current drawings will be used as grounds for future calculations. Yet, as explained later, alternative buildings will also be studied: they have exactly the same layout, and only the materials will be changed to compare the results for different thermal characteristics. For this part, it could have been possible to investigate other optimisation means, such as the use of passive systems or of natural ventilation, but these possibilities will only be discussed in the last part.

Second, the need of cooling will be determined. It is one of the most important parameters for this project. Indeed, reducing the need of cooling will reduce the energy use of the house, and thus decrease the costs of the cooling machine and storage tank. All parameters used will be clearly exposed. However, some parameters may not be mentioned: only the ones with the highest impact on the results and the cooling system design are taken into consideration.

Third, the design of the cooling system will be made in detail for the air part, describing the strategy used.

Fourth, the whole concept for cooling system as imagined by the companies will be analysed. The different components will be designed depending on the cooling load. The interactions between the solar energy, which can only be available during the day to create electricity, and the need of cooling at night, are optimised. The tank to store the cold water, the chiller and its electricity need are dimensioned. It also induces the solar panel characteristics. In this part will be calculated the amount of solar panels needed to decrease the electricity consumption peak load. Different strategies could be used depending on the choice and schedule for using grid or solar electricity. Two different strategies will be compared.

Finally, the results will be analysed and discussed. This part will be an extension to further research on the subject. Also, this project is of main importance, as it could be extended to other kinds of buildings. Norconsult is also willing to apply it to a United Nation project in Soudan if possible. The system can be used in different countries where the need of cooling is important and the outside temperature high.

All the calculations made in this thesis are done using the MATHWORKS simulation tool SIMULINK. This means that only the assumptions and formulas implemented in the tool are used.

This report will follow the work structure. Chapter 2 is a description of the concept as defined by Norconsult. In chapter 3, all the initial conditions are defined: the indoor climate, the outdoor climate in Saudi Arabia, the building characteristics. In chapter 4, the need of cooling is calculated. Chapter 5 defines exactly the different components of the air cooling system and their efficiency. The detailed design of the water cooling system is done in chapter 6. In the last part of this report, chapter 7, the different upgrading and difficulties of the cooling system are discussed, as well as some other suitable solutions for cooling purposes in Saudi Arabia.

2 The solar housing concept

In this chapter, the innovative concept of the SolarHouse will be explained, with more focus on how the cooling system works.

2.1 The background

In the Middle East, the hot outdoor climate induces a need for cooling inside the buildings. Especially, the solar radiations are significant throughout the year.

Nowadays, air conditioning is very common in this region, and sometimes overused, leading to high peak loads for the electricity grid. Many parameters can help reducing this peak load. First, reducing the internal loads will lower the need for cooling. Especially, protecting the inside from solar radiations helps keeping low temperatures indoors. Meanwhile, one can notice that in summer, the air conditioning load is highly related to the solar radiation.

Thus, the idea is to use the energy from the solar radiations to run the air conditioning system and reduce the peak demand on the electrical grid.

Also, when using solar panels, it is most preferable to immediately use the solar energy than to keep it for later. Therefore, as to avoid using much electricity from the grid at night to run the chillers, the idea is to produce and store during the day enough cold to keep a good indoor climate for the whole night.

2.2 Solar photovoltaic, HVAC and energy storage system

The system developed by Norconsult consists in four main points. First, the building should be well insulated, with low thermal transferring windows, roof and walls. Second, the building will be equipped with solar photovoltaic panels (PV) integrated to the climate shell of the roof. The electricity produced will be first used for the air conditioning system, and then for the house appliances if possible. The first one is the main energy consumer and should be designed efficiently. Third, the compressor is oversized so that during the day, enough cooling capacity is produced to cool the building for a whole day and night. The coolant flows through the fan coil unit when needed, and the surplus is stored in a water tank for later, when the cells are not producing any electricity. This is much cheaper than using a battery to run the chiller at night. Also, the house will be quieter when the compressor is off at night. Last, a managing device regulates the need for cooling at any time, depending on the indoor temperature.

The advantage of using PV panels is that excess energy produced is exported to the grid and may be credited to the owner. Reversely, in case there is not enough sun radiation during the hottest days, energy can be provided to the compressor from the grid.

A patent is pending (No. PCT/SE2006/001301) for the combination of this storage system with the improved building characteristics (mentioned below).

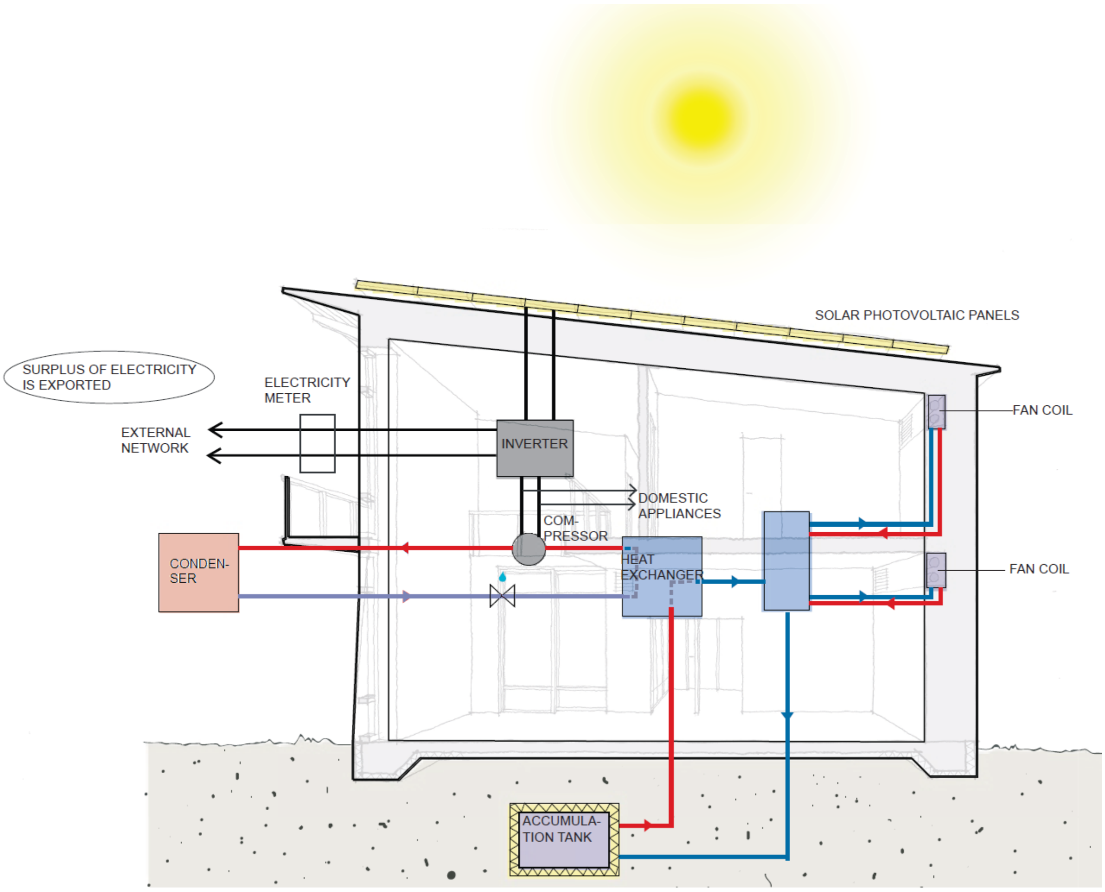


Figure 2 Sketch of the Solar House concept as defined by Norconsult

2.3 The cooling system

The studied cooling system is a vapour-compressor type, as explained in the following part. The compressor in the cool generating system is powered by electricity from the solar panels built in the roof. An inverter converts the electrical power from the solar cells into suitable electrical power for the compressor (from DC to AC).The excess can be transferred to an external or internal supply network (batteries). To reduce energy losses during the transportation, the converter is placed as close as possible to the solar panels. The evaporator enables energy transfer from a coolant to the refrigerant. The coolant circulates within the coolant feeding system, which comprises a tank and a cooling device (fan coils). Different arrangements can be made for the coolant feeding device. In this report, only the first arrangement will be studied, as called in the patent pending.

2.3.1 The vapour-compressor system

In the following part, the numbers refer to the elements noted on Figure 4.

The vapour-compressor cooling system comprises a compressor (2), a condenser (5), a check valve (4) and an evaporator (3). A refrigerant flows during the day through all these units in a closed loop.

The compressor is used to increase the pressure of the vapour refrigerant flowing from the evaporator. The refrigerant at high pressure and high temperature then gets to the condenser where it condenses, rejecting heat. The condenser is located outside the house so that the exhaust heat does not affect the in-house environment. The expansion device diminishes the pressure and the temperature of the refrigerant, until it gets to the evaporator. In there, the refrigerant is evaporated, absorbing heat to change phase. In the evaporator, the coolant flows like in a heat exchanger, releasing heat for the refrigerant to absorb it. The refrigerant and the coolant flow in opposite directions, and at the output, the refrigerant is hot (and in vapour form) and the coolant cold.

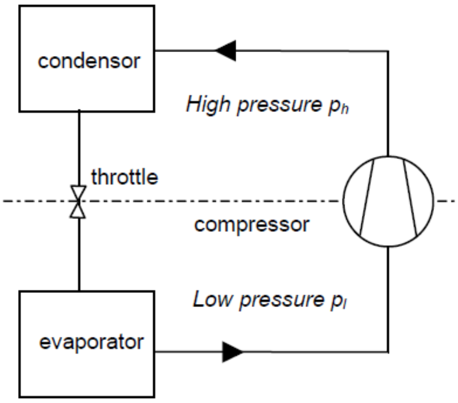


Figure 3 Compression cooler (Eicker, 2003)

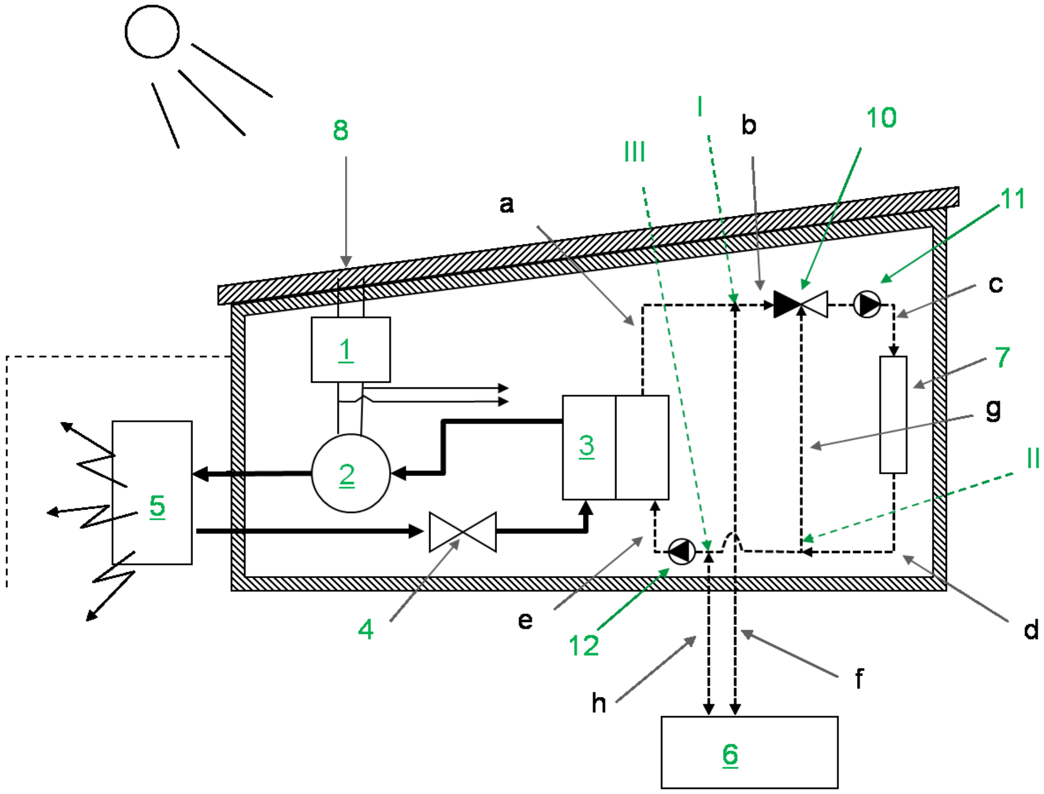
The compressor is the component that uses most energy in this system, so it must be powered with solar energy from the PV panels.

2.3.2 First arrangement of the coolant feeding device

In the following part, the numbers refer to the elements noted on Figure 4.

In the first arrangement, there are a valve (10) and 2 pumps (11 and 12). The control switches at the points I to III can be shunt-like devices, in the form of diverters and relief valves. The fluid control system is then controlled using an algorithm.

During the day, the system works as following, as can be seen on the left part of Figure 5. The cold coolant flows from the energy transfer unit (3) to the cooling device (7) (a-b-c). If there is a lower need of cooling than the capacity of the cooling system, the shunt at the connection point I will deviate part of the coolant to the tank (f), while keeping enough coolant for the cooling coil to meet the indoor climate demands (c). The temperature of the water flowing to the fan coil (7) can be adjusted by mixing the cold coolant (b) with the warm liquid (g) coming out of the fan coil (7), thanks to the valve 10. The warm water coming out from the fan coil unit (d) principally flows back to the energy transfer unit (e), to get cooled down by thermal exchange with the refrigerant in the condenser.



1	inverter	10	shunt	a	first conduit
2	a compressor	11	first pump	b	second conduit
3	energy transfer unit	12	second pump	c	third conduit
4	a check valve			d	fourth conduit
5	a condenser	I	first connection point	e	fifth conduit
6	tank	II	second connection point	f	sixth conduit
7	cooling device	III	third connection point	g	seventh conduit
8	solar cell system			h	eight conduit

Figure 4 First arrangement of the cooling system

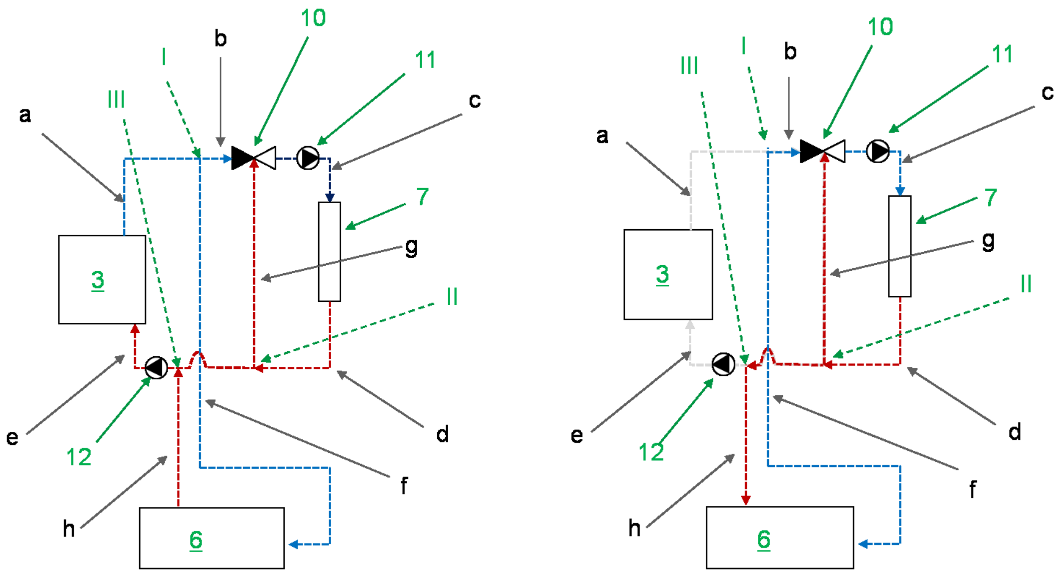


Figure 5 Day and night utilisation of the first arrangement of the cooling system. See legend on Figure 4

During the night, there is no energy from the solar panels. So, the cool generating device is shut down (2, 3, 4, 5). Then, there is no need to use pipes a and e and to take the coolant through the condenser (refer to the right sketch on Figure 5). At connection points I and III, the shunts shut down the access to pipes a and e. Instead, the cold water is directly taken from the tank to the cooling device through pipes f, b and c. During the night, the cooling demand should be less than during the day, and the coolant flowing out of the cooling device (d) should not be so much warmer than the temperature of the coolant flowing in. It can thus be sent back to the tank (h). In case of high desired cooling effect, the coolant should flow faster through the cooling device. The tank may have a device separating warmer water from the cooling device, from the cold water accumulated earlier. Pipe g might be used to control the temperature of the coolant in pipe c.

2.3.3 Benefits of the system

One benefit of the invention is that the electricity used comes from a harmless source for the environment. The solar cells also produce free electricity for the owner of the house, which is a long-term savings means. Yet the area of the solar cells has to be designed correctly as to correspond to the power consumption of the cool generating device.

Also, as setting the use of the cool generating device only for daytime, this induces the system to run almost continuously when turned on. Even if the compressor changes its efficiency with the change of inclination of the sun, it works during long cycles as the coolant is almost fed continuously to the tank during the day. The system then runs at optimum efficiency when in use, and is switched off at night time. This increases the life expectancy of the compressor and thus of the whole cooling system.

From a financial point of view, it is cheaper to have a tank for the coolant than a battery to store energy. Also, the tank is an easy and silent way to provide a cool environment during night time in the house. Of course, the tank and the pipes are insulated to avoid accidental heating of the fluid during transportation through the different devices.

Another benefit is the fact that water can be used as coolant, which makes it a cheap and harmless solution. Other possible liquid fluid can be ethylene glycol, especially if the system should create higher cooling loads.

Nevertheless, some electricity needs to be used at night to drive the cooling system. Some ideas are to use batteries or store electricity from the solar panels or from the wind, or either use potential energy to create the flow in the system.

3 Indoor and outdoor conditions, building design

In the following part will be analysed the given conditions for further use in the calculations. First, the choice for the indoor climate will be discussed. Then, outdoor climate data will be defined for the studied cities. Last, the building used for the studies will be precisely defined.

3.1 Indoor climate

To define the indoor condition in a residential building in Saudi Arabia, the *ASHRAE standard 55P* will be used. This is the American national standard created by the American society of heating, refrigerating and air-conditioning engineers. Saudi Arabia does not have any such standard, so it is common to use the most international one.

The six factors that define the indoor climate conditions and its feeling by people are: metabolic rate, clothing insulation, air temperature, radiant temperature, air speed, and humidity.

The operative temperature is the average of the air temperature and the radiant temperature.

3.1.1 Metabolic rate

It is defined depending on the activities of the people living in the building. It measures the energy produced by organisms.

Sleeping: 40 W/m^2 (0,7 Met)
Seated: 60 W/m^2 (1 Met)
Standing: 70W/m^2 (1,2 Met)
Walking, cooking: 100W/m^2 (1,7 Met)

When the metabolic rate increases above 1,0 Met, the sweating depending of the activity will have a bigger and bigger influence on the thermal sensation.

3.1.2 Clothing insulation

It has to be defined as to know the ideal indoor temperature. Saudi Arabia is a warm climate, so people do not wear so many clothes. So, for an ensemble of a thin trousers and short sleeve shirt, the I_{cl}=0,45clo.

When sleeping, it is hard to know the clothing insulation due to the use of sheets, which provide thermal insulation.

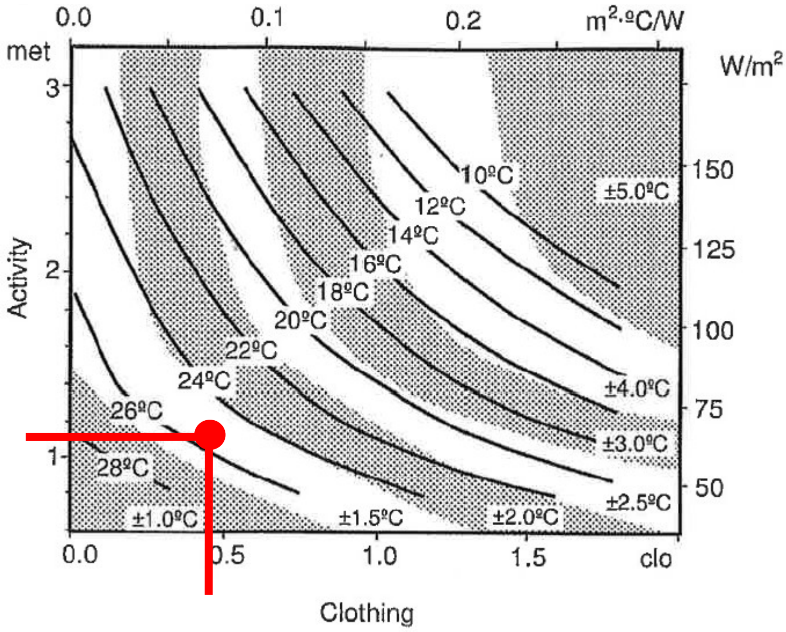


Figure 6 Optimal operative temperature depending of the activity and the clothing (ASHRAE, standard 55P, 2003)

According to this graph the optimal operative temperature would be 25,5°C in summer.

3.1.3 Factors for thermal sensation

To have an idea of the temperature felt by people living in the building, two factors will be used.

PMV (Predicted mean value) is a scale to quantify people thermal sensation. It is from -3 to +3. (+3 hot, +2 warm, +1 slightly warm, 0 neutral, -1 slightly cool, -2 cool, -3 cold)

PPD (Predicted percentage dissatisfied) defines the percentage of people who do not have a good thermal sensation in a specified indoor climate. It is always connected to the PMV like on the graph below. For a PMV of -0,5, 10% of the persons are dissatisfied.

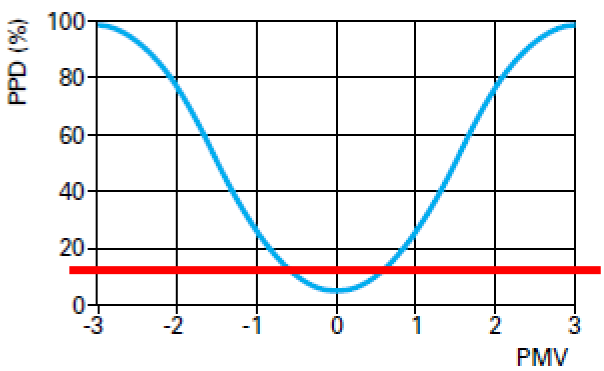


Figure 7 Correlation between PPD and PMV

Three classes of comfort are defined. In this design study, class B will be used. It means that the PPD should be below 10 % and the PMV between -0,5 and 0,5.

3.1.4 Humidity

For a Met-value between 1 and 1,3 and clothes insulation around 0,5, the graph below shows the thermal comfort zone in which the PMV is between +0,5 and -0,5. For an operative temperature of 26 °C, all the humidity ratios are accepted. It will not create any discomfort for people living in the building. Yet, it should not be higher than 0,012kg/m^3. That means that the relative humidity should not be higher than 57%.

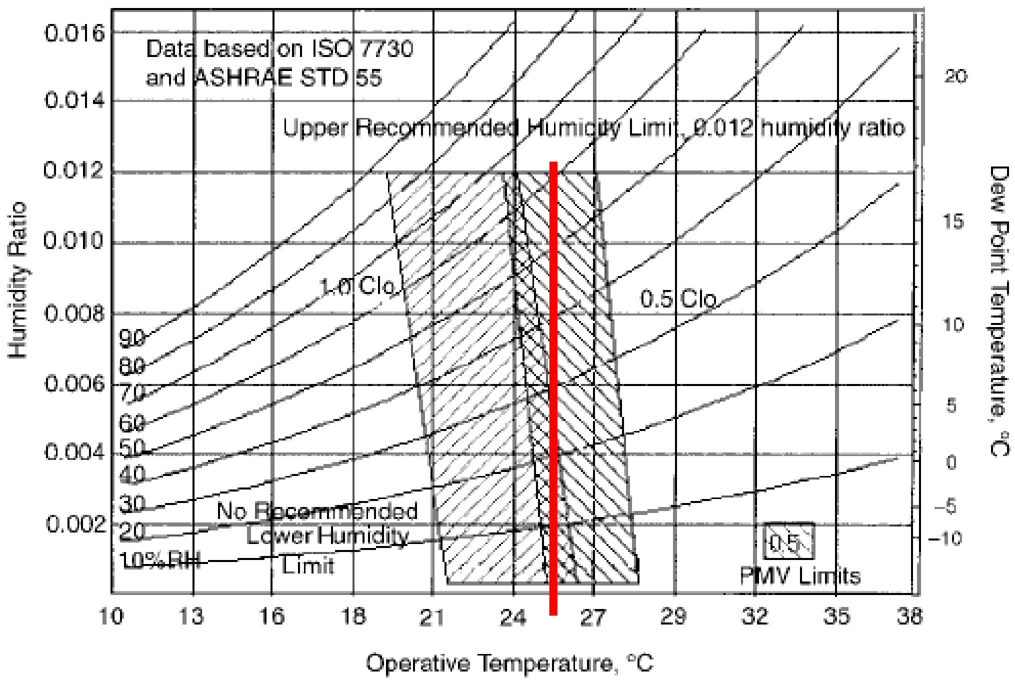


Figure 8 Acceptable range of operative temperature and humidity of the indoor climate (ASHRAE, standard 55P, 2003)

If it is assumed that the relative humidity of the air in the house is 40%, to obtain a good thermal comfort with a PMV of 0,5, the operative temperature will be 24°C-27°C in summer (0,4 clo).

The mean radiant temperature takes into account the radiation received by people. It will increase the temperature felt by them. It could come from the solar radiations or walls. If a mean radiant temperature of 26°C-29°C is considered – which is reasonable because the windows in the house are protected by overhangs and the walls inside the house are at the ambient temperature – it is better to include the ambient temperature between 22°C and 25°C.

3.1.5 Draft-air speed

To avoid too high turbulence in the air and to create draft (unwanted local cooling by air movement), the mean air velocity should be restricted. A turbulence intensity of

20% is a good design. So with a local temperature of 23,5 °C, the mean velocity can reach 0,23 m/s as a maximum to avoid discomfort.

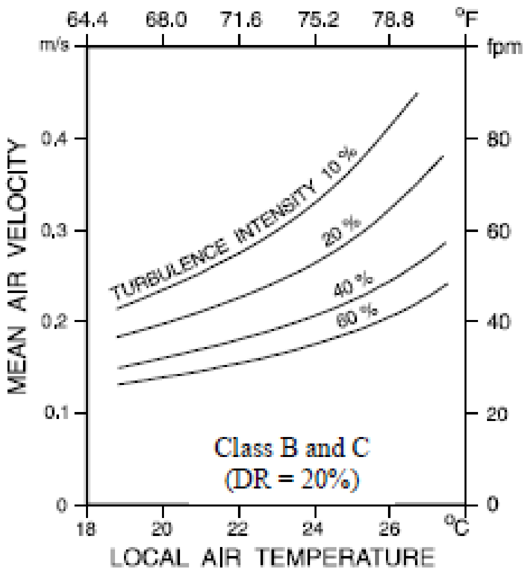


Figure 9 Allowable mean air speed as function of air temperature and turbulence intensity (ASHRAE, standard 55P, 2003)

3.2 Outdoor climate

The outdoor climate has a major impact on the energy use in a building. There are several factors related to the energy performance, including the outdoor temperature, the solar radiations, the long-wave radiations and the wind conditions.

The indoor comfort is also affected by the outdoor humidity and the indoor surface temperatures.

For this thesis, two cities are studied in Saudi Arabia: Riyadh and Jeddah.

3.2.1 Outdoor temperature

All data for the outdoor temperature is taken directly from the software METEONORM. The values are given on hourly basis and are directly used as input in SIMULINK.

Riyadh (24° 38′ 0″ N, 46° 43′ 0″ E), the capital city of Saudi Arabia is situated in the middle of the country.

Jeddah (21° 32′ 36″ N, 39° 10′ 22″ E), is on the Red Sea coast in the west side of the country.

Figure 10 Map of Saudi Arabia (al-Islami)

In Riyadh, the climate is arid and dry with hot days and cool nights. The temperature difference between the summer and winter is important. The hottest day is the 7th of August with a temperature of 46,6°C. The moisture varies all year long too, but will never be very high: between 60% relative humidity in winter and 10% in summer.

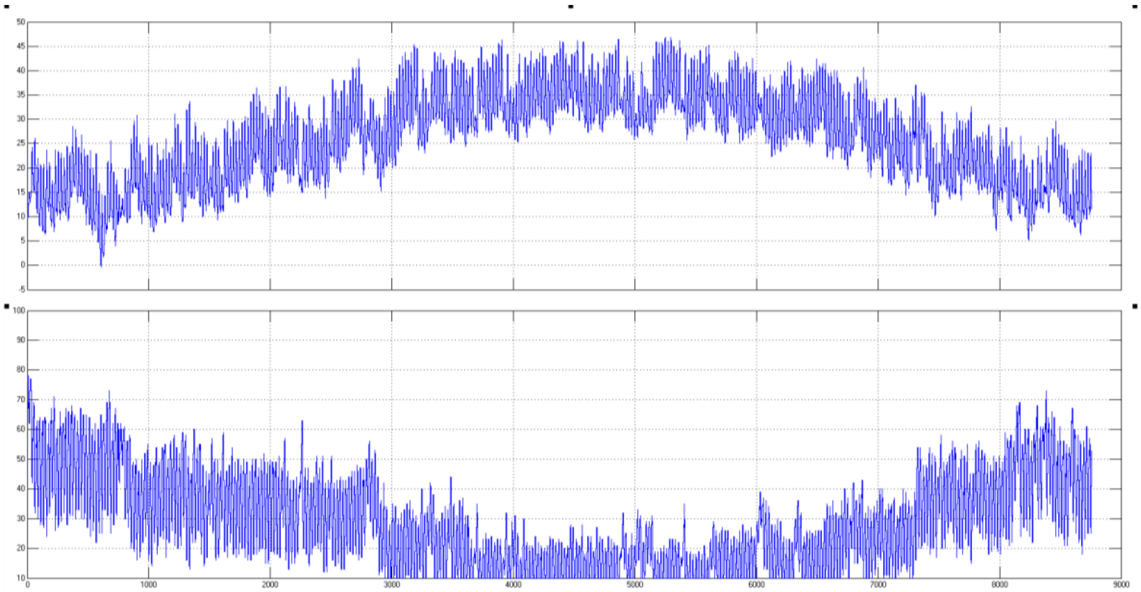


Figure 11 Temperature (-5 to 50°C) and relative humidity (10 to 100%) in Riyadh for one year (8760h)

In Jeddah, the climate is defined as hot and humid throughout the year. There is nearly no moisture difference between summer and winter (50-75% relative humidity). But sometimes it can reach 100% RH. The hottest day is the 13th of July with 42,2°C.The peak temperature is lower than in Riyadh, but the temperature is high all year long. The mean temperature on a year is 28,1 °C.

The climate in Jeddah is close to the climate on the Arabian Gulf coast, in the west of Saudi Arabia in cities like Dhahran. The cooling characteristics in Jeddah and this other part of Saudi Arabia are then comparable.

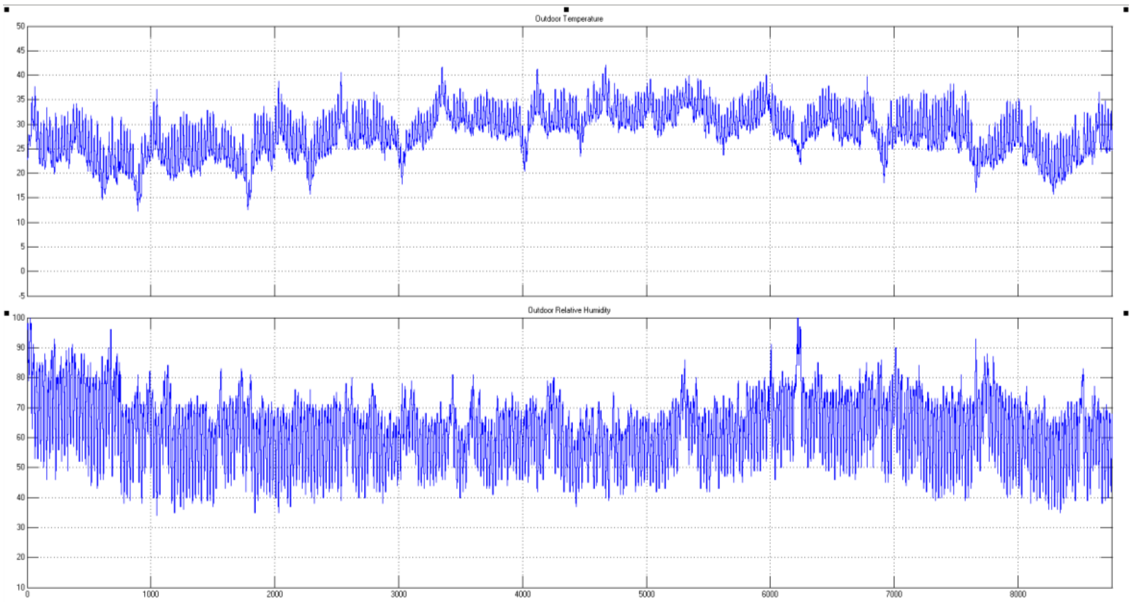


Figure 12 Temperature (-5 to 50°C) and relative humidity (10 to 100%) in Jeddah for one year (8760h)

3.2.2 Solar radiations

In Saudi Arabia, the solar radiation may have a huge impact on the energy use of a building. Indeed the solar constant, which is the intensity of the solar radiation that reaches the upper limit of the atmosphere, has a mean value of 1,39 kW/m². With some dispersion and reflection through the atmosphere, its intensity is decreased when reaching the earth surface, but still has a high value for the cities in study. Heating to the building is provided by both direct and diffuse solar radiations. Actually, the solar radiation is directed in 3 different ways when reaching a surface:

- One part is reflected back. It is noted r.
- One part is absorbed by the material, creating heat. Convection, conduction and radiation will then transport this heat away, some amount going back to the outside of the building, and some to the inside. The absorptivity of the solar radiations through a material is noted α_{sol}.
- One part is transmitted through the material (mostly for windows) to the inside of the building. The transmittance for solar radiation is noted τ. It is dependent on the angle of incidence of the incoming solar radiation. Yet, as to simplify the model, this angle will not be taken into account in the calculations. Doing this for a building in Saudi Arabia is taking the worst case, when the solar transmittance is the highest.

It is to be noted that:

$$r + \alpha_{sol} + \tau = 100\% \qquad (1)$$

The solar radiation on each surface is calculated from:

- The irradiance of the beam. It is the direct radiation received by a surface perpendicular to the sun radiations.
- The diffuse radiation received by an horizontal surface
- The orientation of the wall (North, East, South or West)
- The inclination of the surface (90° for the walls,0° for flat roof)
- The longitude
- The latitude
- The reflectivity of the ground (0,2 for sand)
- The local standard time meridian (45° for Saudi Arabia). It is the real position of the sun on earth compared to the time zone (+3h GMT)

These data give the direct solar radiation, the diffuse solar radiation using the different angles of sun in the sky. The sum of all this will give the total solar radiation.

To calculate the solar radiation depending on the facade, a MATLAB program is used. It has been created by Toke Rammer Nielsen in 2000 and using the method explained in the report "All-weather Model for sky luminance distribution-preliminary configuration and validation" written by Richard Perez, Robert Seals and Joseph Michalsky from the state University of New-York in USA in 1993.

To be more accurate, the shadow created by the overhangs on the façade has also been calculated. The overhangs will protect the walls from direct sun radiations. Thus, it will reduce the peak solar load. To calculate this shadow, different angles of the sun in the space are used:

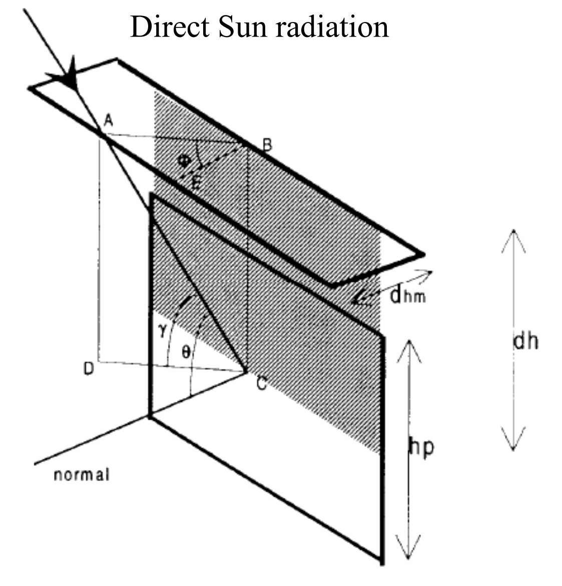


Figure 13 Shadow on a wall from an overhang and different inclinations of the sun (Centre Scientifique et Technique du Bâtiment, 2008)

With:

θ [rad] the incidence angle. It is the angle between the direct sun radiation and the normal to the surface.

γ [rad], the height of the sun. It is the angle between the direct sun radiation and the horizontal projection.

Φ [rad], the azimuth. It is the angle between the horizontal projection of the direct sun radiation and the normal of the wall.

d_{hm} [-], the length of the overhang.

dh [-], the height of the shadow.

hp [-], the height of the wall.

The height of the shadow is calculated by:

$$\mathrm{dh} = \mathrm{Maximum}\ (0, \mathrm{d_{hm}} \times \frac{\tan\gamma}{\cos\Phi}) \qquad (2)$$

In Appendices D.a. and F.a., the total solar radiation for different surfaces are shown for Jeddah and Riyadh.

The radiation on the north façade is really small (from 200 to 300 W/m^2); on the south façade, the radiations are high in winter (900W/m^2) but low in summer (250W/m^2). There are two main reason of this. First the sun is higher in the sky in summer so it has a big γ angle, the shadow on the wall is important. The highest radiation received by the wall is the normal radiation, when γ and ϕ are equal to 0. The radiation level is reduced by the angle made with the normal to the surface. These angles are higher in summer. The west and east façade received the highest radiation level (600 to 900 W/m^2). These façades receive more sun (the height of the sun is lower than on the south) in summer and during a longer time.

As to reduce the solar heat gain through the window, it is crucial to decrease its solar factor. Different types of coatings can be used for this purpose by decreasing the direct transmittance.

Furthermore, a low-emissivity coating on the inner side of the outer glazing will reduce act as a better thermal insulation by reflecting long-wave radiations to the outside and letting short-wave radiations pass to the inside.

3.2.3 Long-wave radiations

Long wave radiations from the sky to the outer parts of the building change its surface temperature. To determine the surface temperature, the “sky temperature” or temperature of the atmosphere is consider, which takes into account the temperature of radiation temperature of the sky. As defined by (Hagentoft, 2001), the sky temperature is calculated as following:

$$t^r = 1{,}2 \times t_a - 14 \text{ [°C] for a horizontal surface, clear sky} \qquad (3)$$

$$t^r = 1{,}1 \times t_a - 5 \text{ [°C] for a vertical surface, clear sky} \qquad (4)$$

$$t^r = t_a \text{ [°C] for a cloudy sky} \qquad (5)$$

Where t_a [°C] is the air temperature.

Both Jeddah and Riyadh will be taken as clear sky regions. In reality, Jeddah should be considered as cloudier, especially due to its high relative humidity, leading to a foggy sky. For this report, it was interesting to determine the impact of the long-wave radiations on the air temperature, therefore the skies were assumed to be clear. (WeatherReports.com).

As the long-wave radiations are taken from the software METEONORM, the long-wave radiation heat flux to a specific surface q_{lw} can be derived using this formula:

$$q_{lw} = F_{sky\_surf} \times \varepsilon \times (I_{lw} - \sigma \times T_a^4) \text{ [W/m}^2\text{]} \quad (6)$$

Where:

F_{sky_surf} [-] is the view factor between the sky and the given surface

ε [-] is the emissivity of the surface considered

I_{lw} [W/m^2] is the irradiation of the surface due to long-wave radiations, as taken from METEONORM. The cloudiness of the city is considered as already taken into account in this data.

$\sigma = 5{,}67 \times 10^{-8}$W/m^2.K^4 is Stefan-Boltzman constant

3.2.4 Wind conditions

The wind conditions influence the heat transfer by convection of the building envelope. The convective coefficient α_c [W/m^2.K] can be calculated from the known wind speed, as taken from METEONORM. (Hagentoft, 2001)

$$\alpha_c = 5 + 4{,}5 \times u - 0{,}14 \times u^2 \text{on a windward side with } u \leq 10 \text{ m/s} \quad (7)$$

$$\alpha_c = 5 + 1{,}5 \times u \text{ on a leeward side with } u \leq 8 \text{ m/s} \quad (8)$$

$$\alpha_c = 6 + 4 \times u \text{ on a parallel side with } u \leq 5 \text{ m/s} \quad (9)$$

Where u [m/s] is the wind speed.

In Jeddah the wind speed and wind direction are very constant through the year. Its values range between 0,5 and 1 m/s and the direction is always from the North. It is to be noted that these values are very low, so the wind has almost no influence on the cooling by convection of the building.

Table 3 Wind conditions in Jeddah

Façade:	**North**	**East**	**South**	**West**	**Roof**
Wind blowing from: North	Windward	Parallel	Leeward	Parallel	Parallel
Min [W/m^2.K]	*7,215*	*8*	*5,75*	*8*	*8*
Max [W/m^2.K]	*9,36*	*10*	*6,5*	*10*	*10*

Table 4 Wind conditions in Riyadh

Façade:	**North (0°)**	**East (90°)**	**South (180°)**	**West (270°)**	**Roof**
Wind blowing from: North (0-45, 315-360°)	Windward	Parallel	Leeward	Parallel	Parallel
Min [W/m²K]	*6,3374*	*7,2*	*5,45*	*7,2*	*7,2*
Max [W/m²K]	*7,6496*	*8,4*	*5,9*	*8,4*	*8,4*
Wind blowing from: East (45-135)	Parallel	Windward	Parallel	Leeward	Parallel
Wind blowing from: South (135-225°)	Leeward	Parallel	Windward	Parallel	Parallel

3.2.5 Equivalent surface temperature

As described in the previous paragraphs, the heat transfer between the building and the surroundings is increased by several factors. Especially, the sun irradiation of the envelope, the exposure to long wave radiations and the convective heat transfer at the external surface can change the heat transfer magnitude of the building envelope. To take into account these processes, it is possible to determine an equivalent temperature for the walls and roof of a building. As on the inside, there is no or little exposure to sun or long-wave radiations, the heat transfer on the inside can be neglected compared to the heat transfer between the outer side of the envelope and the exterior. Therefore, the surface temperature will be approximated to the equivalent outdoor temperature. Also, in the present case, latent heat is disregarded. Hagentoft then gives the equivalent temperature as following:

$$t_{eq} = t_a + \frac{1}{\alpha_e} \times (q_{sol} + q_{lw}) \text{ [°C]} \qquad (10)$$

Where:

$q_{sol} = I_{sol} \cdot \alpha_{sol}$ [W/m^2] is the heat due to solar radiation per surface area

α_e[W/m^2.K] is the total heat transfer coefficient at the surface

$$\alpha_e = \alpha_c + \alpha_r \text{[W/m}^2\text{.K]} \qquad (11)$$

In this equation α_r is the heat transfer coefficient due to long wave radiation. It is calculated as following for 2 surfaces

$$\alpha_r = \frac{4 \cdot \sigma \cdot \bar{T}_{12}{}^3}{\frac{1-\varepsilon_1}{\varepsilon_1} + \frac{1}{F_{12}} + \frac{1-\varepsilon_2}{\varepsilon_2} \cdot \frac{A_1}{A_2}} \text{[W/m}^2\text{.K]} \qquad (12)$$

Where:

$\bar{T}_{12}$ [K] is the mean temperature of the surfaces as calculated in Equation (13)

ε [-] is the emissivity of a surface

F [-] is the view factor between the surfaces

A [m^2] is the area of a surface

Here, one surface will be a wall or the roof, and the other one is the sky. Therefore the mean temperature will be:

$$\bar{T}_{12} = \bar{T}_{sky-surf} = \frac{t_{surf}+t^r}{2} + 273{,}15[\mathrm{K}] \quad (13)$$

Where t_{surf}[°C] is considered equal to the outdoor air temperature

Also, in this case, F [-] is the view factor between the sky and the surface. It is equal to 0,5 for the walls. A simplification is made for the roof: instead of using the view factor corresponding to the real pitch angle of the roof of 18°, the view factor will be considered equal to 1 as if the roof was horizontal.

Finally, when considering the sky as one surface, the heat transfer coefficient due to long wave radiations can be simplified, as the area of the sky is infinite

$$\alpha_r = 4 \times \sigma \times \varepsilon \times \bar{T}_{sky-surf}{}^3 [\mathrm{W/m^2.K}] \quad (14)$$

Where:

ε [-] is the emissivity of the wall or the roof. For a white painted surface, it is taken equal to 0,85. For a window, it is equal to 0,92.

3.2.6 Ground temperature

The heat transfer to the ground also differs from the heat transfer through another part of the envelope to the exterior. This is due to the value of the ground temperature as well as other factors.

In the present case, the ground is considered as a semi-infinite region filled with sand. The steady-state temperature in the ground below the centre of the slab is given hereafter as a function of the shape of the slab and the degree of thermal insulation, as well as the annual average outdoor temperature and the indoor temperature.

$$T_{ground} = T_0 + (T_{in} - T_0) \times u(\tfrac{d}{B}, \tfrac{L}{B}) \; [\mathrm{K}] \quad (15)$$

Where:

T_0 [K] is the annual average outdoor temperature

T_{in} [K] is the indoor temperature

d [m] is the equivalent insulation thickness of the soil, calculated as $d = \lambda \cdot R_{tot}$, with λ [W/m.K] the thermal conductivity of the sand and R_{tot} [m^2.K/W] the total thermal resistance of the floor structure, from the soil surface to the interior of the boundary temperature. It includes the interior surface resistance and the thermal resistance of the floor structure.

$u(\frac{d}{B}, \frac{L}{B})$ [-] is the center temperature of the ground. It depends on the shape of the rectangle slab (B [m] is the width, and L [m] the length). This temperature is taken from the plots in the handbook written by Hagentoft (2001), p.191.

3.2.7 Moisture risks

At the same time, moisture problems in the wall are checked. In Jeddah, the outdoor humidity can be really high. The software 1D HAM is used. It analyses the heat and moisture transport in multi layers walls. The analysed wall is composed of 3 layers: 20 cm of light weight concrete, 5 cm of polyurethane insulation, 20cm of light weight concrete.

The input data are: the equivalent temperature on the outdoor surface of the wall (t_{eq}), the indoor temperature, the indoor and outdoor moisture content in the air, and the solar radiations on the surface.

Below, the different characteristics of the materials are set. w_1 and w_2 are characteristics points on the sorption isotherms of the materials (see in Appendix C, the characteristics of the light weight concrete). w_2 is the upper limit of the hygroscopic region. δ_v, or D_v in 1D Ham, is the vapour permeability of the material.

Layer	Lengtl [m]	Cells	Material name (<=resize=>)	Dv [mm²/s]	Phi [%]	w1 [kg/m³]	w2 [kg/m³]	w(start) [kg/m³]		Lambda [W/(m·K)]	Capacity [MJ/(m³·K	Tstart [°C]
1	0.20000	15	Light-w conc. 400	4.600	80.00	26.00000	110.0000	40.00000		0.1200	0.5500	23.00
2	0.05000	10	polyurethane	0.390	80.00	0.25000	0.60000	0.30000		0.0280	0.0420	23.00
3	0.20000	15	Light-w conc. 400	4.600	80.00	26.00000	110.0000	40.00000		0.1200	0.5500	23.00

Figure 14 Characteristics of the materials used in 1D-HAM

The purpose is to divide the structure of the wall in different cells (until 100). The numerical model is based on finite difference technique.

The transfer of moisture to and from the cells is governed by the humidity by volume in the cells and the humidity at the boundary. The increase of the moisture content, w (kg/m³), of cell number i, with the width Δx_i due to the net flow rate of moisture Δg_i ((kg/m²·s)), is given by:

$$\Delta w_i = \Delta t \times \frac{\Delta g_i}{\Delta x_i} \tag{16}$$

And

$$\Delta g_i = \delta_v \times \frac{\Delta v_i}{\Delta x_i} \tag{17}$$

Here Δt is the time step considered. In this simulation, it is equal to one day.

For the simulation, there indoor relative humidity was set to 40% and the temperature to 23°C, with a moisture production of 1,2 kg/h. In the fan coil design part, calculations are made with a moisture production of only 0,45 kg/h. The results show that there is no moisture problem even in the worst case, when there is a lot of moisture production inside. The wall is vapour permeable: the light-weight concrete and the polyurethane layers do not act as vapour barriers.

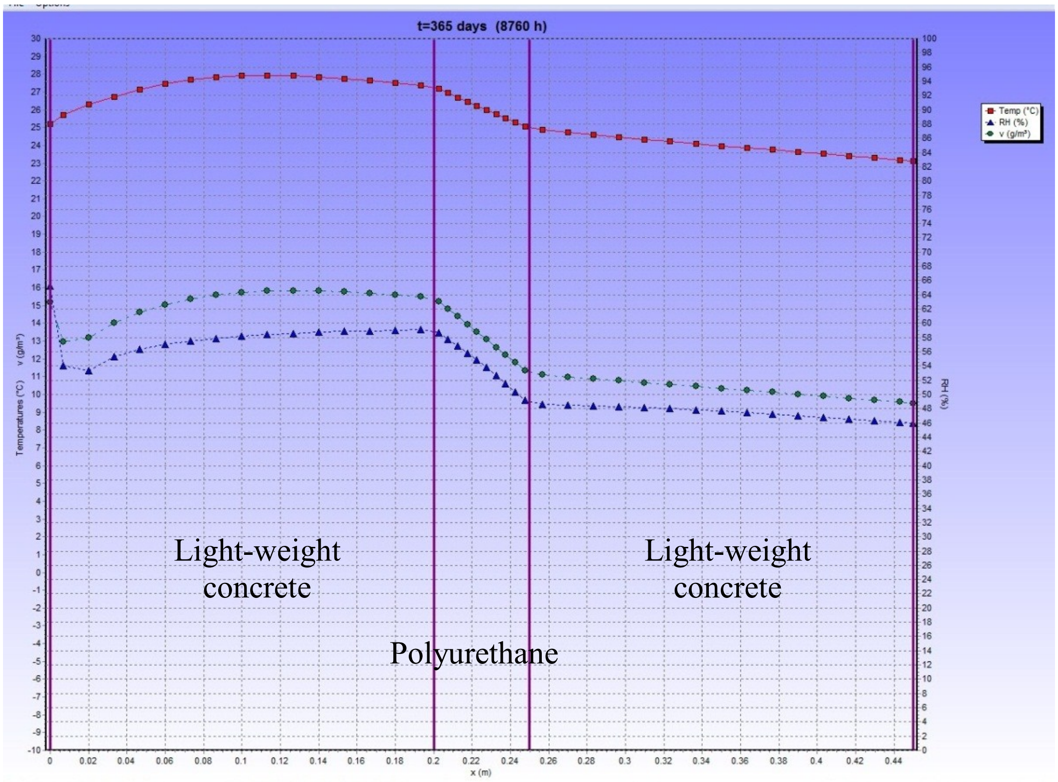


Figure 15 Moisture and temperature distribution through the wall (simulation using 1D-HAM). The outdoor surface of the wall is on the right and the indoor one on the left.

3.3 Building characteristics

In the following part will be described the three building studied. It came originally from the building designed by Norconsult for the patent. The size, orientation and architecture of the buildings are exactly the same.

3.3.1 Building 1 - Traditional building techniques

The main idea behind the choice of materials for the building, as designed by Norconsult, is that the building techniques used should be close to the traditional ones, thus making it easy to build and to maintain for local workers in the Middle East. Therefore, blocks of light-weight concrete were chosen for the walls, as the technique to erect them is well-known. Columns and light-weight concrete blocks will be used for the façade. Light-weight concrete material was preferred to common concrete for its thermal properties.

Table 5 Thermal characteristics of light-weight concrete (YtongThermopierre)

	Minimum	Maximum
Density	365 kg/m^3	425 kg/m^3
Thermal conductivity	0,09 W/m,K	0,14 W/m,K
Thermal capacity	1000 J/kg,K	1300 J/kg,K
Vapour diffusion resistance coefficient	5	36

For the roof structure, the use of concrete and foamglas was preferred. Indeed, foamglas is both a mechanically resistant material and a good insulation material.

Table 6 Thermal characteristics of foamglas (Foamglas T4)

	Minimum	Maximum
Density	100 kg/m^3	165 kg/m^3
Thermal conductivity	0,038 W/m,K	0,050W/m,K
Specific heat capacity	840 J/kg,K	1000 J/kg,K
Vapour diffusion resistance coefficient	36	38

3.3.2 Building 2 – Insulated walls

Regarding the current situation in Saudi Arabia, especially concerning oil use, it is assumed that a law will be soon voted to compel to use a certain amount of thermal insulation material in buildings. As light-weight concrete is not considered as an insulation material, other materials had to be chosen.

In this sub-part will be discussed the choice of materials for the walls, roof and windows, as to reduce further on the need for cooling.

To stick with the architectural idea of the Solar House, the only change in the design was to add insulation as a “sandwich wall” with 2 layers of light-weight concrete blocks and insulation in-between.

3.3.2.1 Polyurethane

Life Cycle Cost

According to Ahmad (December 2002), polyurethane is the best solution regarding Life Cycle Costs benefits. Indeed, for Riyadh city, the study shows that the optimum insulation thickness for polyurethane is 5,0 cm regarding both thermal resistance and total cost (average insulation cost and estimated cost for energy use to get the desired indoor climate). 5 cm thickness polyurethane costs about 7,8 US$ per square meter (2002) and thus has a payback period of about 2,1 years. Another study shows that using 5 cm polyurethane can reduce the building annual energy use by up to 45% for residential buildings in Riyadh, compared to non-insulated buildings; the impact on the peak cooling load is up to 37% reduction in the same conditions (Al-Houmoud, 2003).

Thermal characteristics

Polyurethane insulation is made of small closed gas cells having a lower thermal conductivity than air. Thus, it has a very good thermal conductivity ranging from 0,021 W/(m,K) to 0,028 W/(m,K). It is usually produced as foam within 2 aluminium foils about 50 µm thick. Those ensure the air tightness of the insulation as well as keeping the good thermal properties of the insulation through the age. (ISOVER, 2008).

Table 7 Thermal characteristics of polyurethane (Kingspan Insulation Kooltherm K8)

	Minimum	Maximum
Density	30 kg/m^3	50 kg/m^3
Thermal conductivity	0,021 W/m,K	0,028 W/m,K
Specific heat capacity	1400 J/kg,K	
Vapour diffusion resistance coefficient	36	38

3.3.2.2 Extruded Polystyrene

Extruded polystyrene is only used for insulating the ground. This is valid for all three buildings studied.

Table 8 Thermal characteristics of extruded polystyrene (DOW Styrofoam IB)

	Minimum	Maximum
Density	25 kg/m^3	40 kg/m^3
Thermal conductivity	0,029 W/m,K	0,035W/m,K
Specific heat capacity	1300 J/kg,K	
Vapour diffusion resistance coefficient	100	

As a result, "sandwich" type of wall will be used, including 200 mm of light-weight concrete, 50 mm of polyurethane insulation, 200 mm of light-weight concrete. Indeed, according to the studies of Al-saadi (May 2006), using polyurethane with light-weight concrete blocks (Siporex) is the best association regarding thermal issues.

3.3.3 Building 3 – Optimization of the windows

When studying the buildings and their needs of cooling, it was noticed that the sun constituted the main heat gain source (refer to chapter 3.2.2 and 4.4). Also, to decrease the need of cooling, the best way was to improve the design of the windows for the house.

Optimizing the size, location and characteristics of the windows influence very much the indoor climate. Preferably windows should be avoided on the western façade, as it is the most difficult ones to protect from direct sunlight during summer afternoons, inducing high solar heat gains. On the southern side, it is also very useful to protect the windows with roof eaves and solar protected glazing. Yet, there should be enough windows to get natural light to the building, and avoid using additional electrical appliances. According to ISOVER (2008), the total windows area should be around 20% of the heated area to reach a good thermal comfort, while keeping good lighting.

As defined on the architectural plans, the studied building has a roof eave of about 0,5m. There will be no windows on the southern façade, 44 m² on the northern façade, 14 m² on the eastern façade, and 30 m² on the western façade.

For the choice of glazing, the main characteristics considered are: the U-value and the shading coefficient. The most energy-efficient choice at the moment is triple-glazing windows with a U-value of 0,7 W/m².K and a solar shading coefficient of 17%.

It should be noted that this U-value is for the window only. When taking the frame into account, the U-value is increased. In the following study, it will then be considered equal to 1,0 W/m².K

3.3.4 Characteristics of the buildings to be compared

In the Appendix A, the drawings of the house are shown. It is a building with two floors with no windows on the south façade. Building 1 presented in the table below, is the one already designed. All its characteristics are shown. The two other buildings keep the same shape but some parameters are changed.

Table 9 Characteristics of the 3 types of buildings

		Building 1			Building 2			Building 3		
		Solar House (basic design)			Insulated building			Solar House (improved design)		
		Area [m²]	Composition	U[W/m^{2}K]	Area [m²]	Composition	U [W/m^{2}K]	Area [m²]	Composition	U [W/m^{2}K]
N	Walls	63,12	50 cm LWC	0,231	63,12	20 cm LWC 5 cm PUR 20 cm LWC	0,170	62,16	50 cm LWC	0,231
	Windows	48,88	SHGC 0,32	0,70	48,88	SHGC 0,32	0,70	49,84	SHGC 0,32	1,0
E	Walls	44,4	50 cm LWC	0,231	44,4	20 cm LWC 5 cm PUR 20 cm LWC	0,170	48,24	50 cm LWC	0,231
	Windows	18,17	SHGC 0,15	0,70	18,17	SHGC 0,15	0,70	14,33	SHGC 0,15	1,0
S	Walls	88	80 cm LWC	0,146	88	20 cm LWC 20 cm PUR 30 cm LWC	0,072	88	80 cm LWC	0,146
W	Walls	24,77	50 cm LWC	0,231	24,77	20 cm LWC 5 cm PUR 20 cm LWC	0,170	56,82	50 cm LWC	0,231
	Windows	37,33	SHGC 0,15	0,70	37,33	SHGC 0,15	0,70	5,28	SHGC 0,15	1,0
Roof		222	20 cm C 20 cm FG	0,179	222	20 cm C 10 cm PUR	0,197	222	20 cm C 50 cm FG	0,074
Ground		214,5	20 cm C 50 cm XPS	0,061	214,5	15 cm C 10 cm XPS	0,289	214,5	15 cm C 10 cm XPS	0,289
Middle slab		150	20 cm C		150	20 cm C		150	20 cm C	
Equivalent U-value[W/m^{2}K]		Without ground: 0,2858 Including ground: 0,2225			Without ground: 0,2639 Including ground: 0,2708			Without ground: 0,2512 Including ground: 0,262		
Heat capacity		53870 Wh/K			54025Wh/K			54248Wh/K		

C	Concrete	XPS	Extruded Polystyrene	FG	Foamglas
LWC	Light-weight concrete	PUR	Polyurethane		

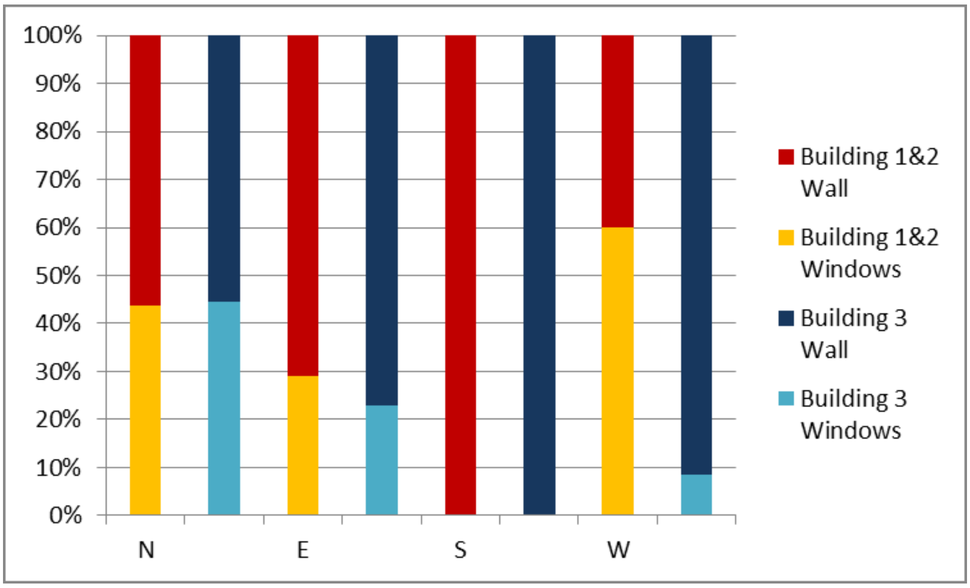


Figure 16 Walls and windows distribution for the different facades of the three buildings.

4 Need for cooling

For the following chapters, the focus is put on building 2 for Jeddah's climate. The results for the other buildings are put in the annex part.

In this chapter, the need for cooling to the building will be calculated.

4.1 Principles

The need for cooling is defined as the surplus heat provided to the building, and that is to be got rid of to reach good indoor thermal conditions.

In the following part, several sources of heat gain to the building will be considered: gains through the envelope, solar gains and leakages. Solar heat gains to the building can be transmitted in two ways: through the windows and by heating of the wall surfaces. The heating of the wall surfaces gives the equivalent temperature.

4.1.1 Heat gains through the envelope

The heat gains through the envelope are calculated as following:

$$Q_{transmission} = \mathrm{UA} \times (t_{out} - t_{in}) \text{ [W]} \tag{18}$$

There are different values for $Q_{transmission}$, depending on the orientation of the façade, the roof or the slab.

UA has a different value in each case (see the data about the characteristics of the house).

t_{out} [°C] is taken as t_{eq} for the walls and the roof (depending on the orientation of the façade too). For the slab, it will be equal to t_{ground}.

For the whole envelope, the transmission losses are summed up, giving:

$$Q_{trans} = \Sigma Q_{transmission} \text{ [W]} \tag{19}$$

4.1.2 Solar gains

The heat gains by solar radiations through the windows Q_{solar} can be calculated with:

$$Q_{solar\ heatgain} = g \times A_g \times I_{sol} \text{ [W]} \tag{20}$$

Where:

g [-] is the solar factor of the window pane, or solar heat gain coefficient: it is the part of the radiation due to direct transmittance and re-radiated radiation from the absorbed part, as seen on

A_g [m^2] is the windows area

I_{sol} [W/m^2] is the solar radiation on the surface considered

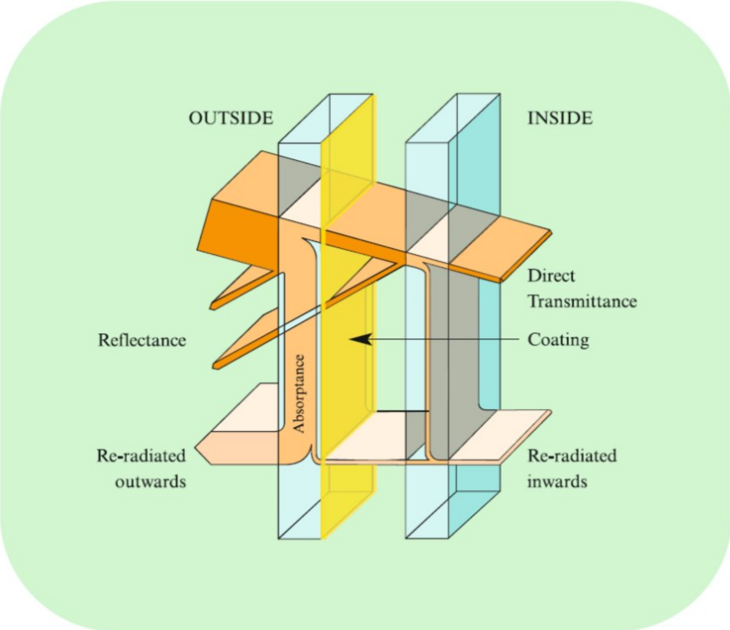


Figure 17 Insulating Glass Unit incorporating coated solar control glass (Pilkington, 2010)

Sun radiations provide internal heat gain to the building during the day. For the whole building,

$$Q_{sol} = \Sigma Q_{solar} \text{ [W]} \qquad (21)$$

To reduce the solar heat gains, the windows should be chosen with a low solar heat gain coefficient, reducing the solar transmission to the inside. Another solution is to reduce the windows area, especially the ones facing east and west.

4.1.3 Air leakages

Air tightness is another factor that can increase the thermal transmissions between a building and its environment. Also, as described later, the building studied uses mechanical supply-and-exhaust ventilation with heat recovery. In that case, air leakages are of importance for the energy use, as in case of leakages the air short-circuits the heat recovery, thus requiring more energy use. Therefore, having an air tight building is a requirement in the design.

Usually, the air tightness of a building is measured by a pressurisation test (the Blower Door test). There are regulations specifying the maximum pressure difference that should be obtained during this test. For the present building, the limit value is taken as 0,05 l/s,m^2 of heated area.

Therefore the heat loss corresponding to these leakages is equal to:

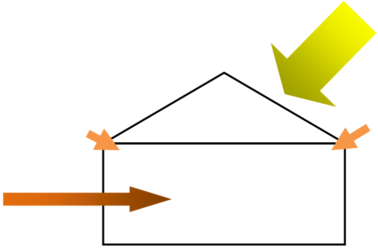
$$Q_{leak} = 0.05 \cdot 10^{-3} \times A_{heated} \times (\rho \cdot c_p)_{air} \times (t_{out} - t_{in}) \text{ [W]} \qquad (22)$$

With:

A_{heated}[m^2] the heated area of the building

$(\rho \cdot c_p)_{air} = 1200$ J/m^3.K the volumetric heat capacity of the air.

4.1.4 Need of cooling calculation



With the previous equations, one can derive the need of cooling for the building:

$$C\frac{dt_{indoor}}{dt} = Q_{transmission} + Q_{solar\ heatgain} + Q_{leakage}\ [W] \qquad (23)$$

One can notice that the thermal capacity of the building is taken into account in this equation. It reflects how the envelope can buffer heat coming through the building.

The thermal capacity C is derived from:

$$C = \Sigma(\rho \times A \times c \times d_p)\ [J/K] \qquad (24)$$

With d_p, the periodic penetration depth, equal to

$$d_p = \sqrt{\frac{\lambda \times t_p}{c_p \times \pi}}\ [m] \qquad (25)$$

t_p, the time period of the variation, is 24h in this design, reflecting the daily temperature variations.

The periodic penetration depth represents the thickness of the envelope on the indoor side for which the conditions will be influenced by the indoor temperature on 24 hours. It depends on the materials:

- for light weight concrete, $d_p = 8{,}4$ cm

- for concrete, $d_p = 13{,}2$ cm

The walls inside the building and the middle slab are also included in the calculations for the thermal capacity.

All of them act as buffer for the variations of the indoor temperature.

4.1.5 Comparison of the heat gains

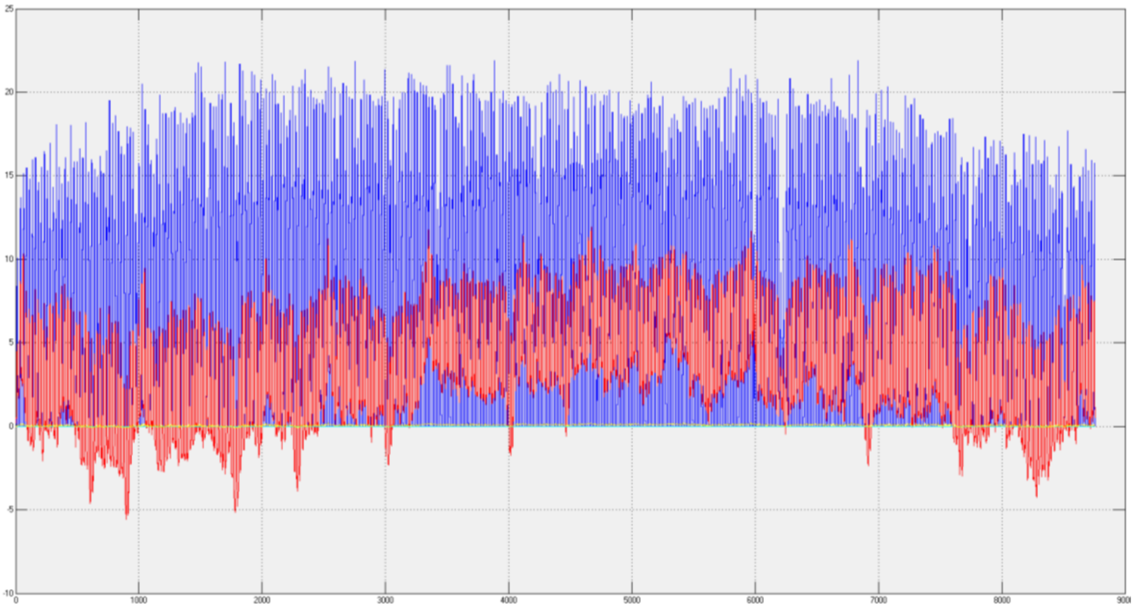


Figure 18 Heat gains to the building (-10 to 25W/m^2), for one year (8760h). In blue: the solar gain, in red: the transmissions through the building envelope, in green: the leakages, in yellow: the hygienic ventilation (Jeddah Building 1) (simulation using the modelling tool Simulink)

Due to their low value compared to the other gain sources, heat gains provided by people activity and leakages will be disregarded from now on, leading to:

$$C\frac{dt_{indoor}}{dt} = Q_{transmission} + Q_{solar\ heatgain} \qquad (26)$$

4.2 Definition of the ventilation system

After having determined the heat gains, it is possible to derive the cooling load. Also, it is important to know the cooling and ventilation system to get right load.

The cooling provided to the building will be done using a fan coil. Therefore the air is taken from the rooms and directly cooled down inside the rooms. Yet, for every building there is a minimum amount of fresh air to be provided: the hygienic flow.
For this study, the hygienic flow is considered as a certain amount of fresh air that does not provide cooling to the building. Therefore, it is considered to be an input of air at the indoor temperature. Nevertheless, there is a need of energy to bring this fresh air – at the outdoor temperature – to the indoor temperature.

4.3 Cooling load

As regards to the previous definition, the heat gain is the amount of heat provided to the building through transmission through the walls and solar gains.

$$\text{Heat gains} = Q_{transmission} + Q_{solar\ heatgain}\ [W] \qquad (27)$$

The cooling demand would then be the same amount of cooling needed to compensate this heat gain.

$$\text{Cooling demand} = -\text{Heat gains} = -Q_{\text{transmission}} - Q_{\text{solar heatgain}} \quad [\text{W}] \tag{28}$$

The cooling load is the cooling demand to cool down the building, in addition to the cooling demand to cool down the hygienic air.

$$Q_{\text{cooling}} = -Q_{\text{transmission}} - Q_{\text{solar heatgain}} - \dot{V}_{\text{hygienic}}(\rho c_p)_{\text{air}}(t_{\text{out}} - t_{\text{in}}) \ [\text{W}] \tag{29}$$

With $\dot{V}_{\text{hygienic}}$ the hygienic flow, equal to 93 l/s.

4.4 Comparison of the cooling demands

In this part, the need of cooling for the different situations presented above will be compared.

Figure 19 Transmission heat gains through the walls for the different buildings (-15 to 15W/m^2) in Jeddah for one year (8760h) [W/m² floor area]. (simulation using the modelling tool Simulink)

Buildings 1 and 3 are both made of light-weight concrete in the same dimensions and thus have the same transmission heat gains for the walls. Building 2 has insulation material to buffer the heat gains. Therefore, building 2 should have less heat gains than building 1 and 3. Yet, as there is a huge reduction in the windows area for building 3, it also improves the equivalent U-value of the walls and windows. Here, the gains are diminished by half (from about 10 W/m^2 for buildings 1 and 2 to 8 W/m^2 for building 3).

Figure 20 Solar heat gains through the windows for the different buildings (0 to 25W/m^2) in Jeddah for one year (8760h) [W/m² floor area]. (Simulation using the modelling tool Simulink)

Buildings 1 and 2 have the same windows characteristics. Building 3 has more than 50 % less windows. Also, the solar heat gains are reduced by half for building 3 (from 20 W/m^2 for buildings 1 and 2 to 10 W/m^2 for building 3).

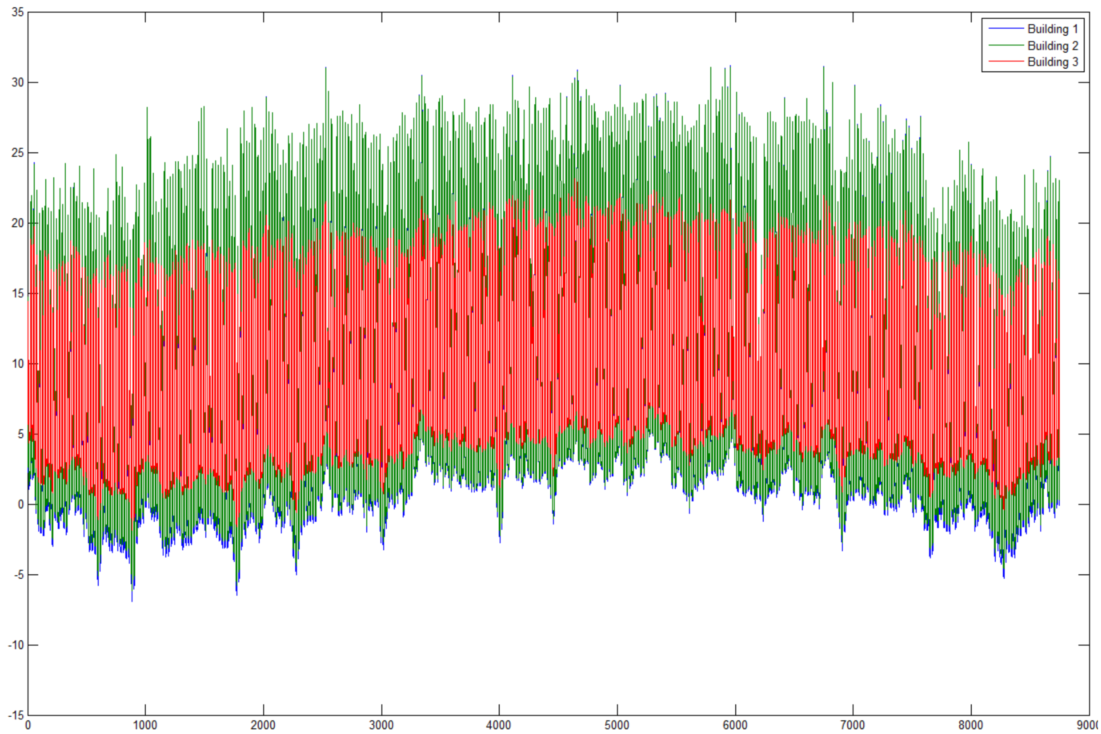


Figure 21 Total heat gains to the building (-15 to 35W/m^2) in Jeddah. In blue: building 1, in green: building 2, in red: building 3. for one year (8760h) [W/m² floor area]. (Simulation using the modelling tool Simulink)

Finally, from the graph above, the total heat gain of building depends a lot on the solar radiations (reduction from 30W/m^2 with the building 2 to 20W/m^2 with the building 3). The insulation has an influence on the diminution of the heat gains to the building, but a reduction of the windows area, a low solar heat gain coefficient g, or a better layout of them (avoiding west and east facades) is really the decisive factor in this kind of climate.

4.5 Climate without active cooling

Without fan coil in the building to keep the indoor temperature between 22°C and 25°C, the latter varies with the external conditions. Equation (26) is used to calculate the indoor temperature. It is shown for both Riyadh and Jeddah and for the three different buildings, on the two figures below.

Again, the comparisons made before are also valid here. Building 3, with the reduction of the windows area, has a much lower indoor temperature in summer (32,5°C in Riyadh, 34°C in Jeddah). The insulation layer, in building 2, reduces the summer temperature of 3°C compared to building 1, in both climates (49°C to 46°C for example in Jeddah).

Of course, the indoor temperature cannot vary so much. It is interesting to see that it is very far from the comfort temperature of 25°C. That is why fan coils are necessary. In the next part all the cooling system is calculated.

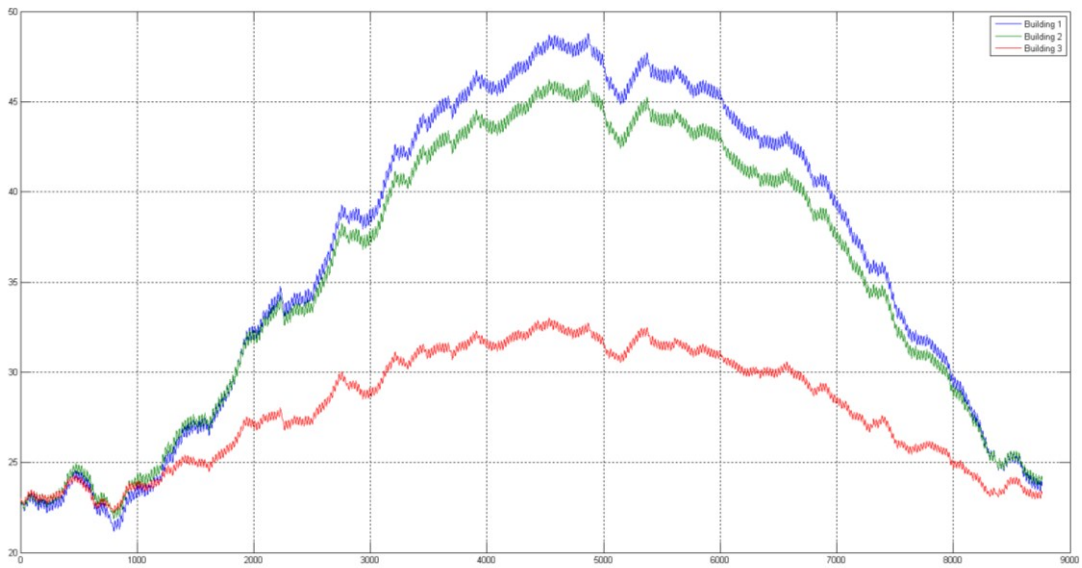


Figure 22 Temperatures in Riyadh without active cooling (20 to 50°C) for one year (8760h) [°C] (simulation using the modelling tool Simulink)

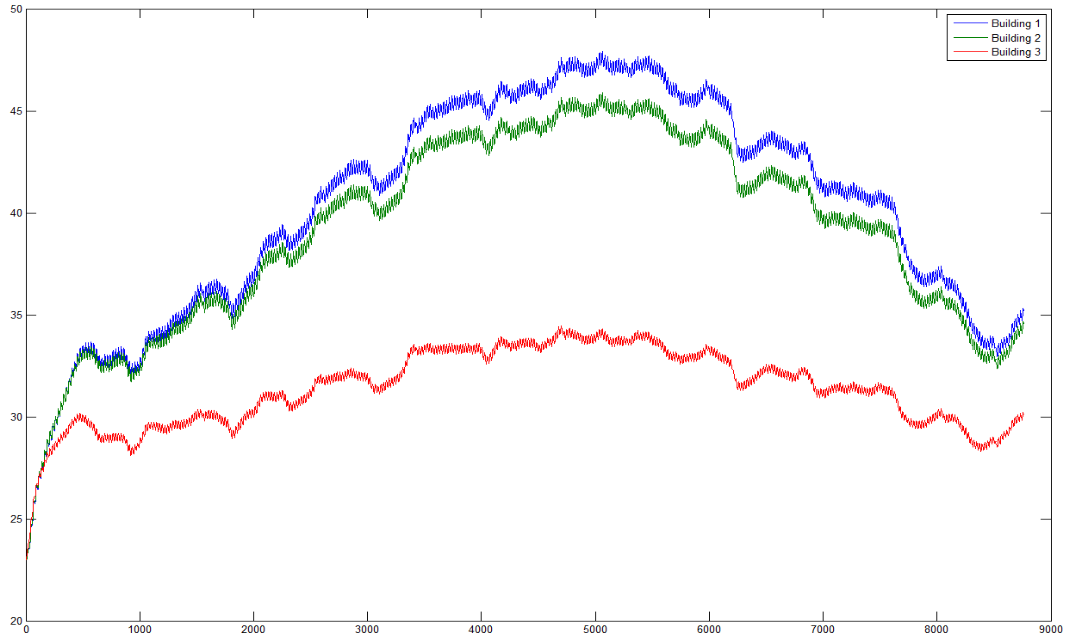


Figure 23 Temperatures in Jeddah without active cooling (20 to 50°C) for one year (8760h) [°C] (simulation using the modelling tool Simulink)

4.6 Thermal capacity

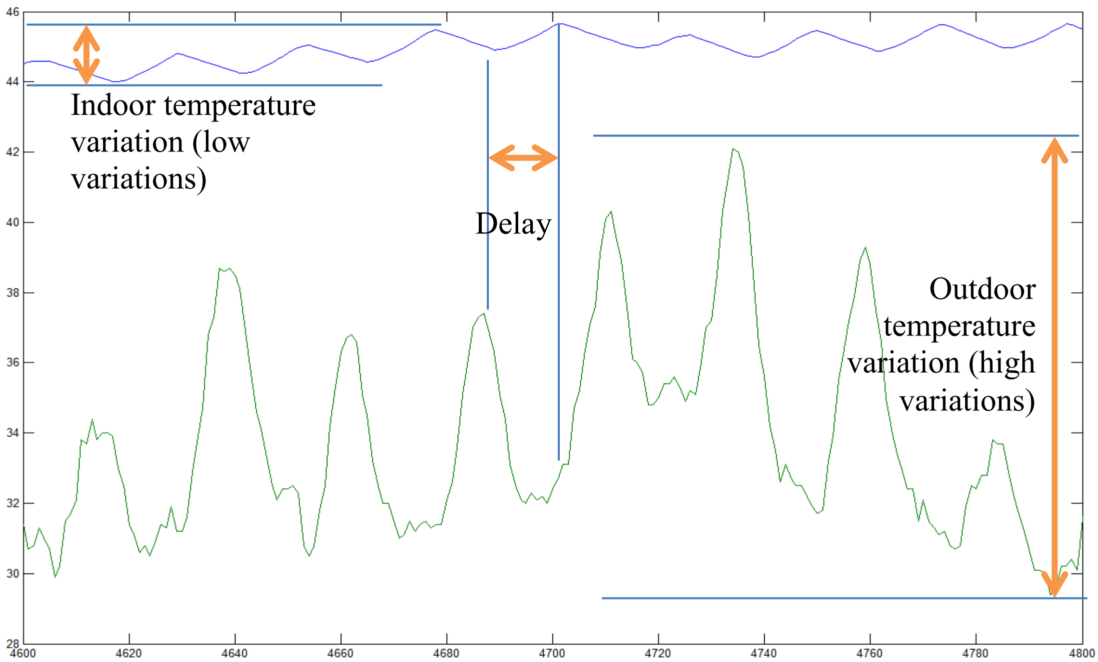


Figure 24 Thermal capacity of the building. Comparison between the outdoor temperature (in green) and the indoor temperature (in blue) (28 to 46°C). (simulation using the modelling tool Simulink)

The influence of the thermal capacity can be seen in the delay between the indoor and outdoor temperature variations. The following figure shows the indoor and outdoor temperature for Building 2 in Jeddah. Here, there is about 12 hours of delay. Also, the thermal mass helps the building buffer the climate variations. While the outdoor climate varies in a range of 14°C, the indoor temperature only varies in a range of 2°C.

5 Air cooling system

5.1 Cooling strategy

The scheme below shows the strategy used to cool the house.

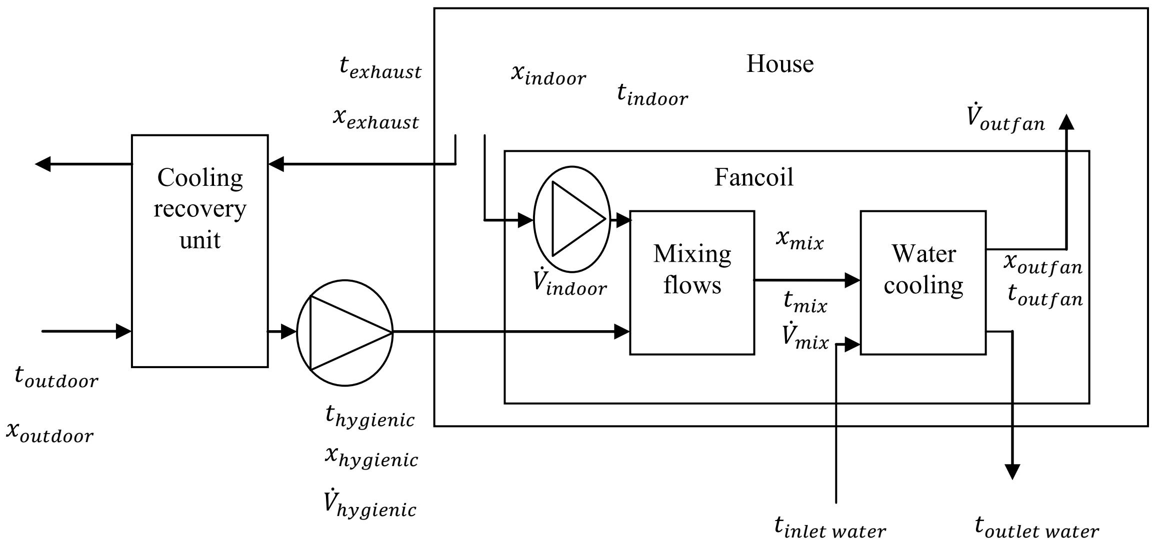


Figure 25 Strategy for the air cooling system

With:

t [°C] the temperature

x [g/m^3] the moisture content in the air

$\dot{V}$ [m^3/s] the flow through the ducts

The relative humidity (RH) [%] is equal to $\frac{x}{x_s}$ with:

x_s [g/m^3], the moisture content at saturation, represents the maximum amount of moisture that can be present in the air. It depends on the temperature. And:

$$x_s = \frac{288{,}68 \times (1{,}098 + \frac{t}{100})^{8{,}02}}{461{,}4 \times (t + 273{,}15)} \text{ [g/m}^3\text{]} \quad (30)$$

This formula is basically applied when the temperature t is between 0°C and 30°C. But this formula still works for a temperature of 45°C with a margin of ±0,02 g/m^3.

The cooling is done in 2 parts:

- The cooling recovery unit
- The fan coil

There is mixing of the hygienic air with the indoor air before cooling by water. The moisture is considered in this part too.

In the rest of the study, no more care will be taken for the cooling demand, as defined in paragraph 4.4. Also, when writing about the need of cooling, it will always refer to the cooling load.

5.2 Cooling recovery

The weather in Saudi Arabia is not only hot, it can be wet too. Therefore, the cooling strategy will be to use a comprehensive moisture and heat recovery unit. The indoor condition will be around 23°C and 40% relative humidity. The outdoor condition can rise to 42-46°C and 60-80% relative humidity. In the current market, the efficiency of these cooling energy recovery unit with hygroscopic wheel made in aluminium is 80 % for temperature and 70% for moisture. They recover both sensible and latent heat.

The efficiencies correspond to:

Temperature:

$$\eta_t = \frac{t_{outdoor} - t_{hygienic}}{t_{outdoor} - t_{exhaust}} \text{ [-]} \qquad (31)$$

Moisture:

$$\eta_x = \frac{x_{outdoor} - x_{hygienic}}{x_{outdoor} - x_{exhaust}} \text{ [-]} \qquad (32)$$

This will reduce the energy consumption to dehumidification and cooling of the air. So it will reduce the investment cost for cooling, more specifically with the reduction of fan coils or heat pump and pipes.

In Figure 26, the cooling capacity of the cooling energy recovery all along the year with different climates is shown. This cooling capacity can be 5 kW, especially in Jeddah, where the moisture content in the outdoor inlet air is high. The outlet temperature and moisture content after the cooling recovery is shown too. This air is used only as hygienic flow, which is needed for hygienic reasons inside the building: it is the air being renewed. As it is a house of 300 m^2 of floors with few people in it, and few activities, this flow is very low. It will correspond to 0,35 ACH, i.e. 0,35 times the volume of the house, which is $300m^2 \times 3{,}2m$. This flow will be 336 m^3/h or 93 l/s.

The cooling capacity of the cooling energy recovery is:

$$Q_{cooling\ recovery} = \dot{V}_{hygenic} \times \rho \times c_p \times (h_{hygienic} - h_{outdoor}) \text{ [W]} \qquad (33)$$

Where

$$h = c_p \times t + c_w \times x \times t + x \times r_o \text{ [kJ/kg]} \qquad (34)$$

c_p, the specific thermal capacity of the dry air, is 1kJ/(kg,K)

c_w, the specific thermal capacity of the water vapour, is 1,85 kJ/(kg.K)

r_o, the latent heat of vaporization of water at 0°C, is 2500 kJ/kg

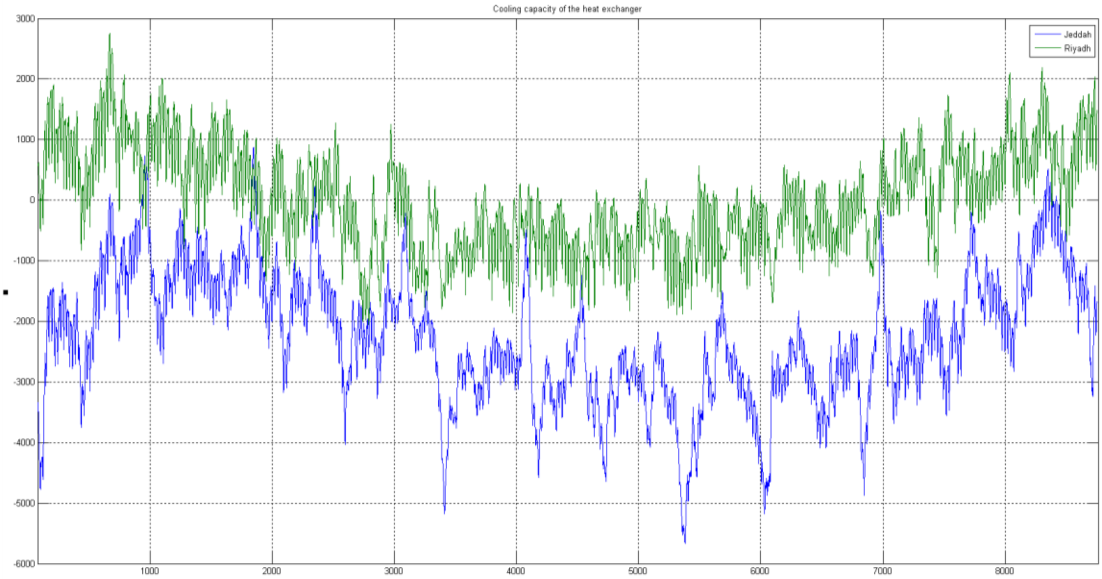


Figure 26 Cooling recovery unit capacity (-6000 to 3000W), for one year (8760h). In blue: Jeddah, in green: Riyadh. (simulation using the modelling tool Simulink)

On the graph above, the cooling capacity of the recovery unit is shown. The cooling capacity is higher in Jeddah (until -5000W) than in Riyadh (until-2000W) because there is a higher amount of moisture to be transferred between the exhaust air and the outdoor one, so the difference of enthalpy is higher. All the values are negative because the difference of enthalpy in the equation is negative.

The hygienic flow temperature varies between 22°C and 28°C depending on the season. The temperature of the hygienic flow is shown in the Appendix D b. This is perfect to be used to cool the house. The moisture transferring part is a key part. Without this desiccant wheel, the moisture would be a big issue in the hygienic flow, because some condensation would occur with a hygienic flow temperature below 26°C, if the moisture content is the same as the one outside. The air would be saturated. On the graph in Appendix D b, it is possible to see that there will be no moisture problem anymore in the air after the cooling recovery. The relative humidity is small enough, between 40% and 60%.

More in Riyadh, and a little bit in Jeddah, the recovery wheel will work as a heat recovery unit. The nights can be quite cold in Riyadh with this arid climate. The main purpose of the recovery unit is to reduce the difference between the hygienic flow conditions and the indoor conditions. That is why, on the graph for Riyadh, the cooling capacity is above 0W in winter. It means that the unit is not cooled but heated. The heat recovery unit will supply enough amount of energy to keep the temperature inside the house above 22°C.

It should be noted that, at the time when the report is written, no cooling recovery unit (for temperature and moisture) exists for such a small air flow. The calculations are still made considering the same recovery efficiencies, but these are not based on data from a real unit.

5.3 Mixing of air

The hygienic air coming from the outlet of the cooling exchanger is not redistributed directly to the rooms. It is mixed with the indoor air at the inlet of the fan coil to be cooled to the desired temperature through it. The indoor air flow through the fan coil is much bigger compared to the hygienic flow through it. This indoor air flow will be a fixed value all along the year and, as to meet the required cooling needed inside the house, it will be either equal to:

- 0 m^3/s or 0,8 m^3/s in Jeddah
- 0 m^3/s or 1 m^3/s in Riyadh

By fixing this air flow, only the outlet temperature of the fan coil will vary.

There will be a step change of the inlet flow in Riyadh and Jeddah. The step change is mostly due to the difference between day and night conditions. Sometimes the hygienic flow is enough to cool the building ($\dot{V}_{indoor}$= 0 m^3/s and $\dot{V}_{hygienic}$=0,093 m^3/s). Sometimes a much bigger flow is needed to create cooling ($\dot{V}_{indoor}$= 1 or 0,8 m^3/s and $\dot{V}_{hygienic}$=0,093 m^3/s). The sun radiations influence a lot the need of cooling inside the house. In Jeddah, the flow is always at the maximum during summer.

To calculate the temperature and moisture of the mixed air, this common equation is used:

$$t_{mix} = \frac{t_{indoor} \times \dot{V}_{indoor} + t_{hygienic} \times \dot{V}_{hygienic}}{\dot{V}_{indoor} + \dot{V}_{hygienic}} \text{ [°C]} \tag{35}$$

$$x_{mix} = \frac{x_{indoor} \times \dot{V}_{indoor} + x_{hygienic} \times \dot{V}_{hygienic}}{\dot{V}_{indoor} + \dot{V}_{hygienic}} \text{ [g/m}^3\text{]} \tag{36}$$

$$\dot{V}_{mix} = \dot{V}_{indoor} + \dot{V}_{hygenic} \text{ [l/s]} \tag{37}$$

5.4 The fan coil

In the fan coil, the air will be cooled at the contact with the water system. The inlet and outlet temperature of the water is 8/14°C, which makes a mean temperature of 11°C. On the Mollier chart below, the fan coil cooling process is shown. It takes into account the latent and the sensible heat: a wet fan coil is used. The cooling capacity of the fan coil is written as:

$$Q_{fancoil} = \dot{V}_{mix} \times \rho \times c_p \times (h_{outfan} - h_{mix}) \text{ [W]} \tag{38}$$

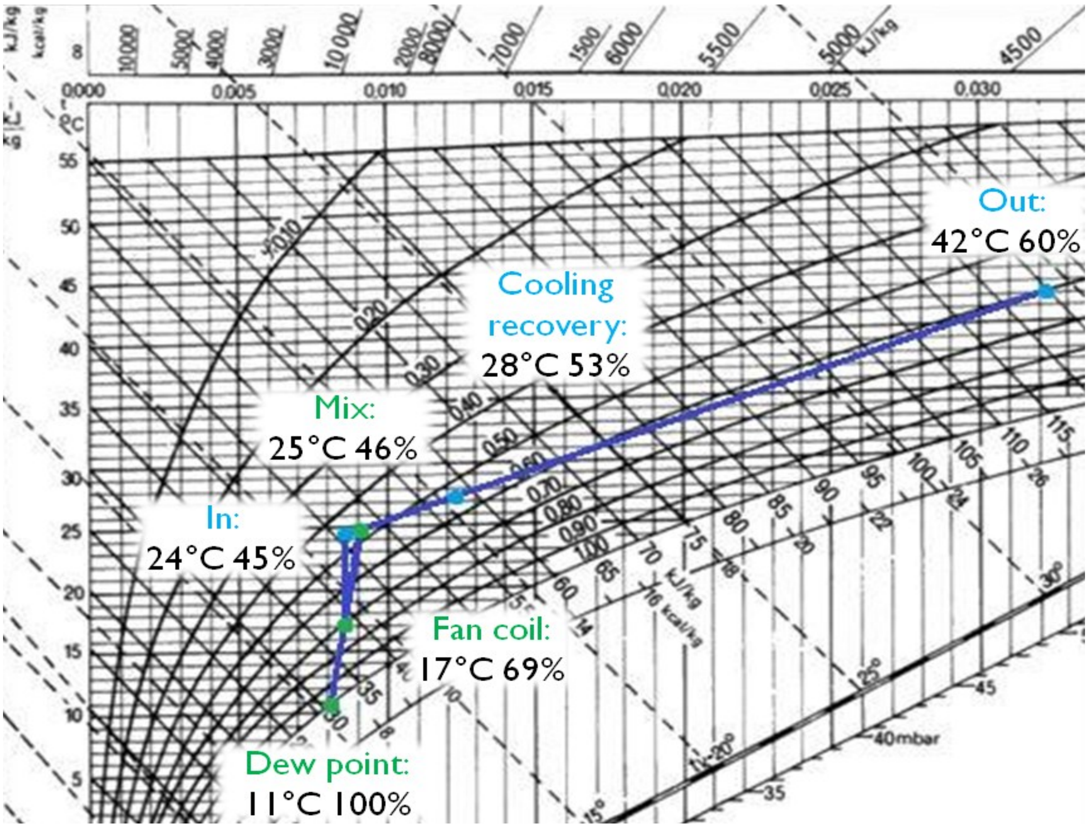


Figure 27 Mollier chart for a hot outdoor climate

There will be dehumidification during the process of cooling. That is why the cooling capacity equation with enthalpy difference is used. There is both sensible and latent cooling. Of course, there is a need to take care of the water obtained from the dehumidification in the fan coil, but this is always integrated in these fan coils, especially for this kind of climate.

To get the required temperature conditions inside the building (between 22°C and 25°C), the fan coil needs to be parameter depending on the indoor temperature; the efficiency of the fan coil will vary.

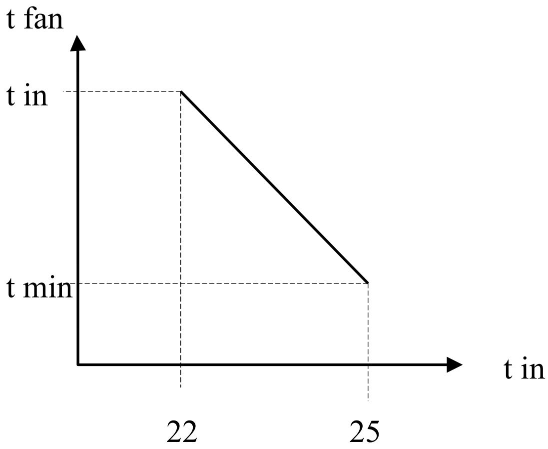


Figure 28 Correlation between the outlet fan coil temperature and the indoor temperature

To get a correct outlet temperature from the fan coil, it is adjusted linearly depending on the indoor temperature, as seen on the figure above. The maximum outlet temperature is equal to the indoor temperature, and the minimum outlet temperature depends on the need of cooling.

Therefore, the outlet temperature is equal to:

$$t_{outfan} = \frac{t_{min} - t_{in}}{25 - 22} \times (t_{in} - 25) + t_{min} \text{ [°C]} \qquad (39)$$

With: t_{min} set to 14°C in summer, and to 17°C in winter, to avoid moisture problems.

On the figures in Appendix D b, the moisture and the temperature at the outlet of the fan coil are shown together with the outdoor conditions and after the cooling recovery unit. The temperature at the outlet of the fan coil varies between 16°C and 21°C, while the relative humidity varies between 45% and 65%.

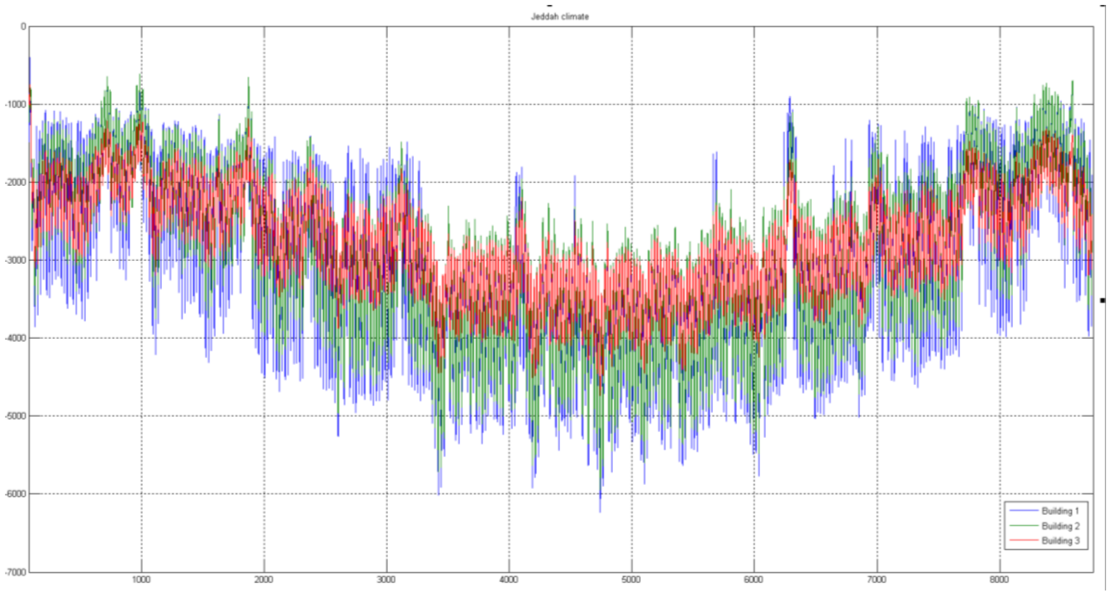


Figure 29 Fan coil cooling load (-7000 to 0W), in Jeddah, for one year (8760h). Building 1 in blue, building 2 in green, building 3 in red. (simulation using the modelling tool Simulink)

On the figure above, the cooling capacity of the fan coil is shown. Depending on the building, the need of cooling is different; hence the fan coil cooling load is different. For building 3, with fewer windows, it reaches -4500 W in summer, whereas for building 2, with the insulation layer, it reaches -5500W and -6000W for building 1. All the values are negative because it is a need for cooling: the difference of enthalpy in the equation is negative.

5.5 Indoor Moisture

There are many different moisture sources in a house.

- Human produce moisture depending on the activity: 50g/h when sleeping, 90g/h when standing/walking.
- Cooking: 2kg/day
- Shower: 0.2kg/day
- Laundry. 0.5kg/day

Assuming 6 people inside the house, the total moisture production inside is:
G = 0,45 kg/h
To deal with moisture indoors, a transient equation with moisture source is used.

$$G - \dot{V}_{outfan} \times (x_{indoor} - x_{fancoil}) - n \times V_{house} \times x_{indoor} = \frac{dx_{indoor}}{dt} \times V_{house} \text{ [kg/h]} \quad (40)$$

The rotating wheel in the recovery unit has a big effect on the reduction of the moisture. This is the key component of the design.

5.6 Choice of fan coils

To simplify the calculations, a big fan coil has been studied, but in fact, it is better to have 10 of them distributed all over the house. For the case of the solar house, 10 fan coils with a cooling capacity of 1000W and an airflow of 396 m^3/h, i.e. 0,11 m^3/s would be a good system. It is possible to change the parameters of either the airflow through the fan coil, or the temperature of the water, or the fan coil outlet temperature to obtain the same cooling capacity.

To be sure to avoid draft problems, the size of the ducts is important. With a local temperature of 23,5°C, the mean velocity can reach 0,23 m/s as a maximum to avoid discomfort, as it is written in chapter 3. With an airflow of 0,11 m^3/s through the fan coil, a supply pipe with a radius of 40 cm, i.e. 0,5 m^2 section, creates an airflow of 0,22 m/s, below the discomfort value for draft air.

5.7 Resulting indoor climate

The most important here is to reach the indoor conditions to have a good thermal comfort. No boundaries have been set on the indoor temperature and relative humidity. They depend on $\dot{V}_{indoor}$ and t_{fan}. Thanks to the equations written before, these two graphs are obtained. They are the results of the simulation using the modelling tool Simulink. The initial condition (at the beginning of the year) has been set arbitrarily. Besides, the indoor relative humidity needs some time to be stabilized.

On the graph below, the moisture inside the room is not critical. The indoor relative humidity varies around 40%.

According to the ASHRAE, the critical value of thermal comfort value, with a PPD of 10%, is 12g/m^3 of moisture in the indoor air. Here, it is around 8g/m^3, so way below.

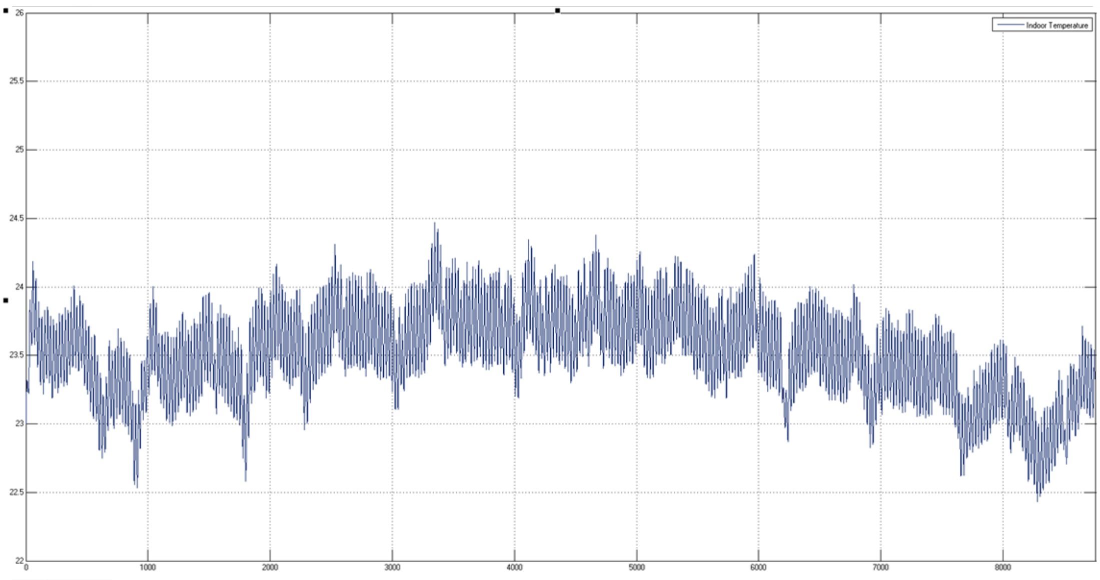


Figure 30 Indoor temperature (22 to 26°C), in Jeddah, for one year (8760h) (simulation using the modelling tool Simulink)

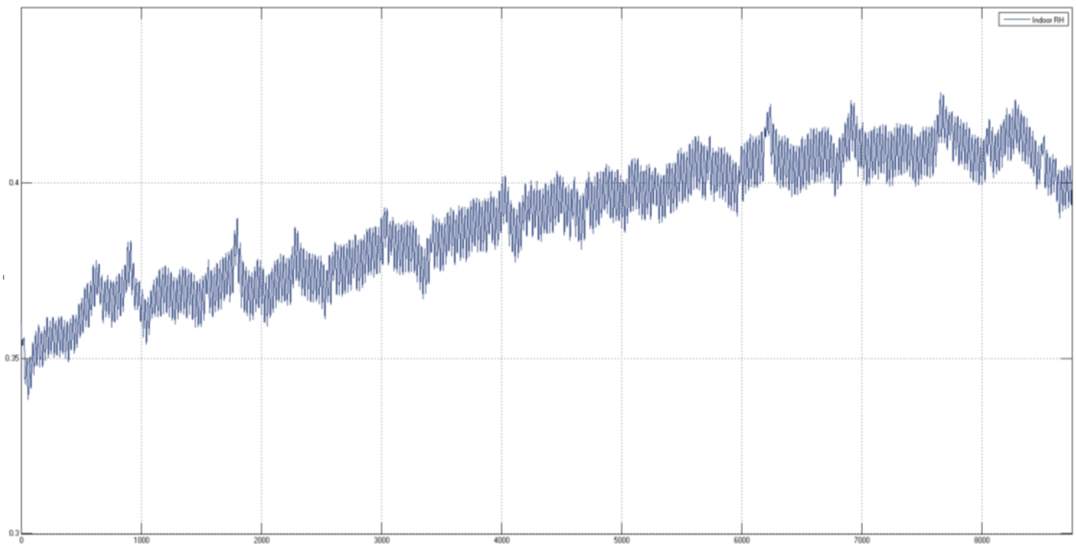


Figure 31 Indoor relative humidity (30 to 45%), in Jeddah, for one year (8760h) (simulation using the modelling tool Simulink)

6 Water system design

In this chapter, the first part consists of obtaining the amount of cooling delivered by the chiller from the photovoltaic panels or the grid. Then the second part is the design of the connection between all the components: the chiller, the fan coil and the tank as for the chiller to be able to deliver the necessary cooling capacity to the fan coil, for day and night.

Proceeding this way, the chiller cooling capacity will match the fan coil cooling demand. Optimising it leads to deriving the area of the solar panels, the chiller size, and the tank size.

6.1 Solar panels and their behaviours in this climate

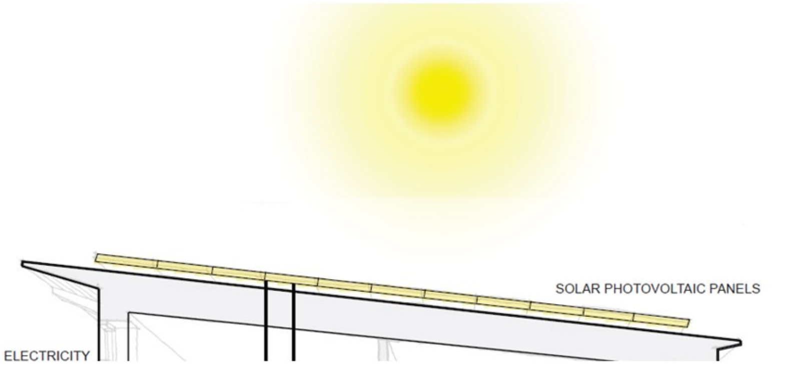


Figure 32 Part of the cooling system to be analysed: solar panels

6.1.1 Angle of the panels

To optimize the electricity delivered by the solar panels, the right angle should be determined.

The same MATLAB program as the one to calculate the irradiation on the façade and the roof is used. But this time, a south direction of the roof is set and its horizontal angle varies. The optimal angle is the one that can create the highest amount of electricity both all year long and during the peak period.

The peak period is defined as the time of the year that requires the most important cooling need. It is possible to get it from the fan coil design chapter. For Jeddah, this period goes from the 15th of May to the 15th of September (hour n°3217 to 6192). For Riyadh, it runs from the 6th of May to the 15th of September (hour n°3001 to 6192).

Table 10 Irradiations on the solar panels for different periods, depending on the roof angle for Jeddah.

Angle roof (°) Jeddah	Irradiation panels for one year (Wh/m^2)	Irradiation panels for the cooling peak period (Wh/m^2)
0	$2{,}1714 \times 10^6$	$8{,}3457 \times 10^5$
15	$2{,}3034 \times 10^6$	$8{,}1252 \times 10^5$
18	$2{,}3160 \times 10^6$	$8{,}0326 \times 10^5$
21	$2{,}3238 \times 10^6$	$7{,}9250 \times 10^5$
24	$2{,}3268 \times 10^6$	$7{,}8002 \times 10^5$
30	$2{,}3184 \times 10^6$	$7{,}5071 \times 10^5$
45	$2{,}2174 \times 10^6$	$6{,}5557 \times 10^5$

For Jeddah, the optimal angle to obtain the highest amount of irradiation all year long is 24°, but the angle to have the highest amount of irradiation during the peak period is 0°. An angle of 18° has been chosen because the variation of irradiations with the optimal angle, in both cases, is still small.

Table 11 Irradiations on the solar panels for different periods, depending on the roof angle for Riyadh

Angle roof (°) Riyadh	Irradiation panels for one year (Wh/m^2)	Irradiation panels for the cooling peak period (Wh/m^2)
0	$1{,}9610 \times 10^6$	$8{,}3211 \times 10^5$
15	$2{,}0979 \times 10^6$	$8{,}1738 \times 10^5$
20	$2{,}1207 \times 10^6$	$8{,}0366 \times 10^5$
25	$2{,}1318 \times 10^6$	$7{,}8574 \times 10^5$
27	$2{,}1331 \times 10^6$	$7{,}7763 \times 10^5$
30	$2{,}1318 \times 10^6$	$7{,}6446 \times 10^5$
45	$2{,}0637 \times 10^6$	$6{,}7961 \times 10^5$

For Riyadh, the optimal angle to obtain the highest amount of irradiation all year long is 27°, but the angle to have the highest amount of irradiation during the peak period is 0°. Therefore, an angle of 20° has been chosen.

On the graph below, it is possible to see the path of the sun at the equinoxes and the solstices. The angle of the sun with a horizontal surface, the sunset and sunrise are visible. Because the two cities are close to the cancer tropics (23° 26' 16" latitude north), the sun is nearly vertical the 21th of June. To remind it, the latitude in Riyadh is 24,43° and 21,41° in Jeddah. Therefore, the best roof angle for the peak load season is 0°. The sun has an angle with a vertical surface corresponding to the latitude of the cities at the equinoxes. This explains why the most efficient angle of the solar panels depends on the seasons.

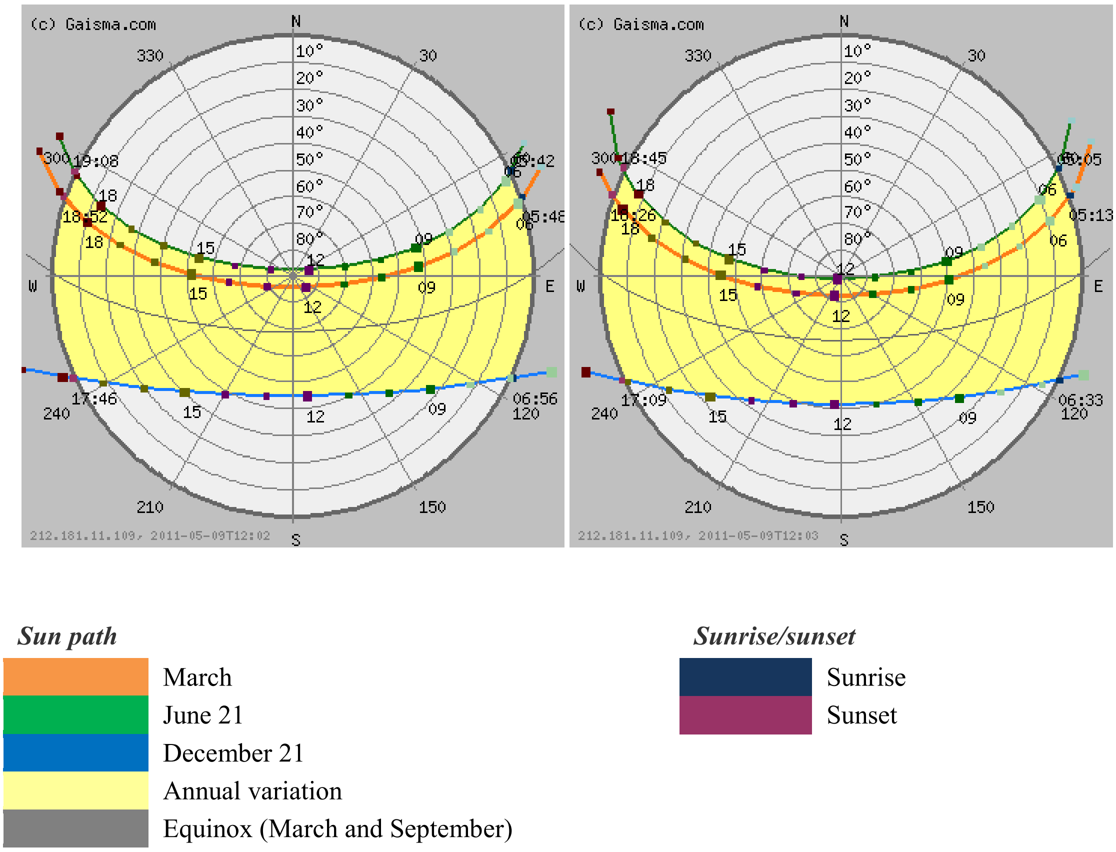


Figure 33 Sun path in the sky for Jeddah (on the left) and Riyadh (on the right) (Tukiainen, 2005-2011)

6.1.2 The reduction factors of the supply electricity

All powers of the solar panels are measured under the standard test conditions (STC): an irradiation of 1000W/m^2 and a temperature of 25°C. Of course, this does not represent the real outdoor conditions. The power delivered by the solar panels varies depending on the outdoor conditions. Below, the efficiency of different panels is shown under the STC conditions.

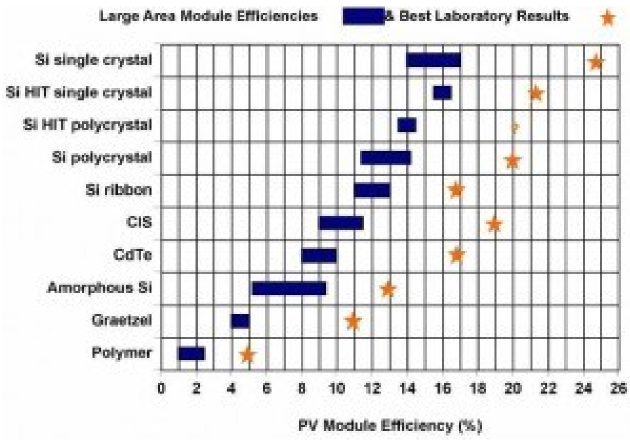


Figure 34 Comparison of different PV module efficiencies (Solar Navigator, 2008)

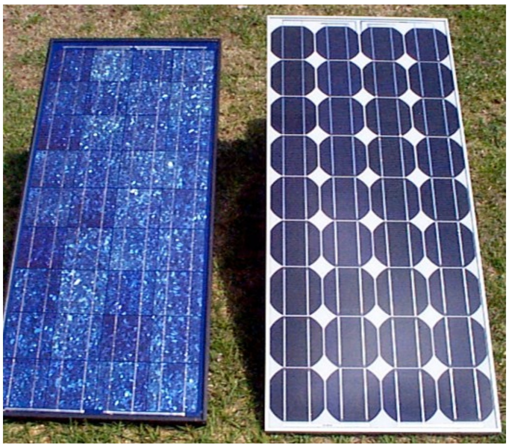


Figure 35 Mono and poly crystalline solar panels (Uni-solar)

6.1.2.1 Mono crystalline panels

The most efficient solar panel is the mono-crystalline one with an efficiency between 14% to 18%. They are quite more expensive than the poly-crystalline ones, but they have a longer life-span (25 to 50 years), better performance and efficiency.

All the panels have a reduced electricity production when the temperature of the panel increases. The thermal reduction coefficient for a mono-crystalline panel is 0,5% per degree above 25°C. This value is much bigger for a poly-crystalline panel. So if the equivalent temperature of the surface of the panels is equal to 60°C, the reduction coefficient would be (60-25)×0,005 = -17,5%. On Figure 36, it can be seen that this temperature can nearly reach 100°C in Jeddah, with high variations during the day. This temperature is slightly the same in Riyadh and reaches 100°C too. This is the most important reduction factor of the electricity produced by the PV panels.

To calculate the equivalent temperature on the solar panels surface, the same equation (41) as before is used, considering the convection, the conduction and the radiation. But this time, the absorptivity of the mono-crystalline panels is 0,92 and the emissivity is 0,84 (Muller, 2010)

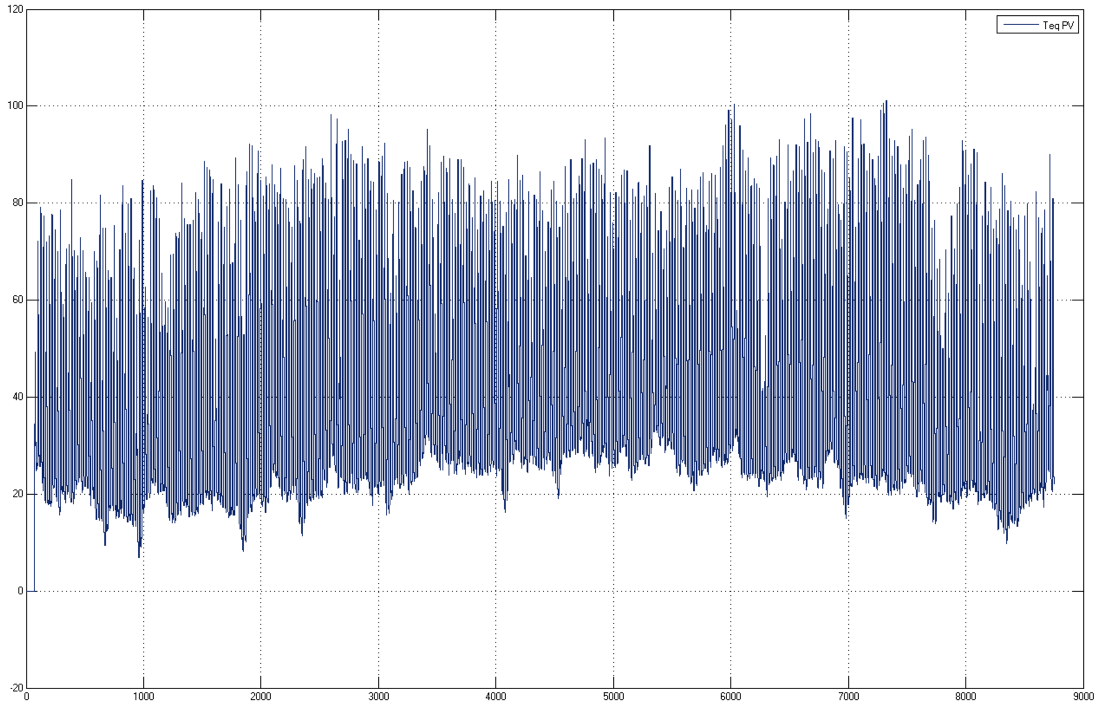


Figure 36 Equivalent temperature on the surface of the photovoltaic panel (-20 to 120°C), for one year (8760h) in Jeddah (simulation using the modelling tool Simulink)

6.1.2.2 Reduction factors

Assumptions have to be made when it comes to reduction factors. All variables that can influence the efficiency of the solar panels are:

- Weather data, equivalent temperature of the solar panels
- Soiling (dust on the panels): a small area of the panels are shaded or soiled and their efficiency can be much reduced. Especially for crystalline panels, the electricity output can nearly be reduced to 0 if two cells are really shaded or soiled.
- Shading (from a tree, another building...)
- The tilt angle of the panels: seen above, the electricity produced by the panels depends on its inclination with the sun
- Array mismatch: the voltage obtained from all the panels is usually an average of the voltage at maximum power of each panel. When the difference between these two values increases, the power delivered is reduced. This value is considered as 2% losses
- Inverter efficiency: the inverter has a large influence on the reduction of the electricity produced. It goes from 6% to 10% reduction losses. If a 3 phases system is used, and it is the case in this design (see chiller electrical data in appendix E), there is an added 2.5% to 3% reduction to be considered
- Distribution losses: quite small, can be considered as 1%
- Wiring losses: one should choose a right wire size that limits the losses ($I^2 \times R$) to less than 3% (Allan Gregg)

6.1.2.3 Triple junction amorphous silicon cells solar panels

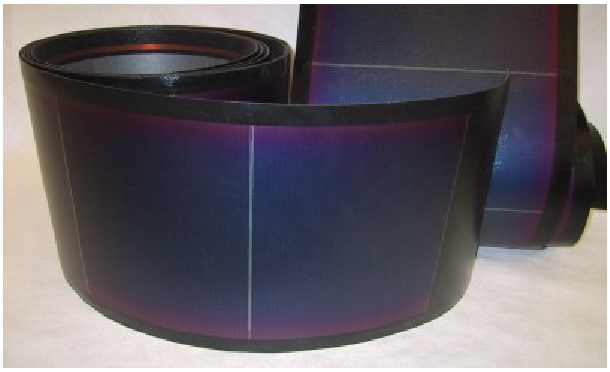


Figure 37 Triple junction amorphous solar panels (Uni-solar)

A triple junction amorphous silicon cells solar panel can be a good alternative. Its advantages are:

- It has a better efficiency than crystalline panels at low irradiation of the panels (beginning and end of the day or cloudy weather)
- It has a better efficiency when the equivalent temperature of the solar panel is high. The thermal reduction coefficient for an amorphous silicon cells panel is 0,21% per degree above 25°C. So when the panel's temperature is equal to 60°C, the reduction coefficient is equal to (60-25)×0,0021 = -7,35%.
- The panels are really flexible and lighter and can resist better to damages than the crystalline ones: they still produce most of their rated power. On the contrary, when the glass of a crystalline panel is broken, it becomes nearly inefficient and the changing of the glass can be really expensive.
- They cost much less to produce because less material is needed to build one panel as they are very thin.
- In case of soiling or shading, the power produced by the amorphous silicon cells is reduced of 9% only, because the cells are bigger (fewer cells on one panel) and each cell has a by-pass diode.

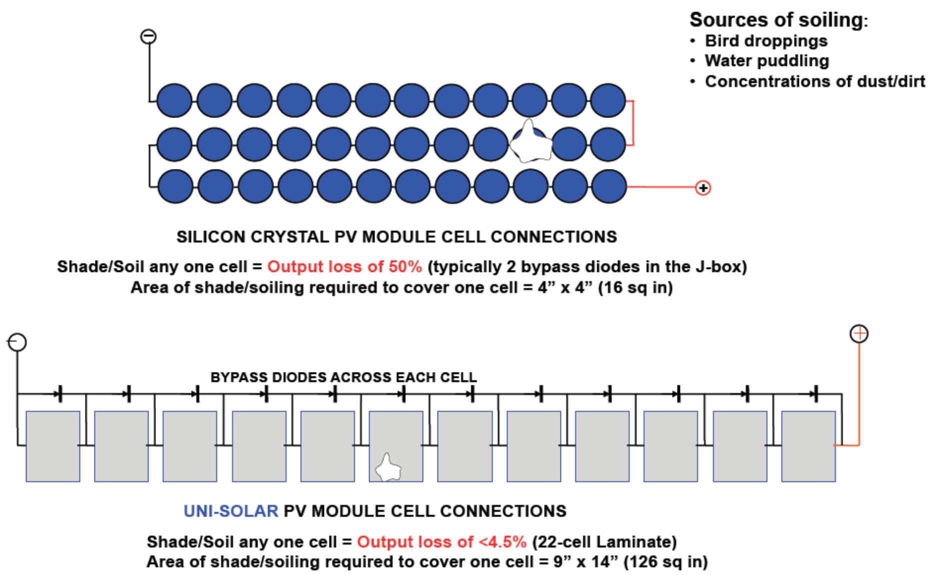


Figure 38 Effect of shading or soiling on the efficiency of the PV panels (Uni-solar)

However, compared to crystalline panels, their efficiency is smaller (between 5% and 9%), so a bigger surface of solar panels is needed. At very high equivalent temperature of the panels, both kinds of panels have the same efficiencies. For poly-crystalline panels, which are less efficient than mono-crystalline ones, it is the case. (Meike, 1998)

To conclude, both kind of panels have pros and cons. The amorphous silicon cells solar panels work better in the Saudi Arabia climate conditions, but are less efficient compared to the mono-crystalline panels.

In this study, the mono-crystalline panels are used to produce electricity. But the people who will live in this building should be aware of the maintenance, like cleaning the panels every two weeks or checking damages.

6.2 Chiller capacity and choice of chiller package unit

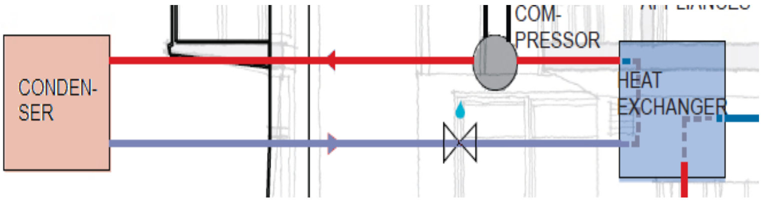


Figure 39 Part of the cooling system to be analysed: chiller

To know the capacity of the chiller, this one has to produce the cooling required for one day and the next night. Connected to solar panels, the chiller will only work during day time.

In this report, the main goal is not to focus on the chiller properties but more on the strategy between the different parts of this cooling system. Therefore, to know the work to be delivered to the chiller, the outdoor temperature should be taken into account. The higher the outdoor temperature, the less efficient the condenser, and the smaller the chiller's COP (coefficient of performance).

To calculate this COP, for a given refrigerant, the isentropic coefficient of performance is

$$COP_i = \frac{Q_{chiller}}{W} = \eta_{c,i} \times \frac{T_2}{T_1 - T_2} \text{ [-]} \qquad (42)$$

With: $\eta_{c,i}$ [-] the isentropic Carnot efficiency

T_2 [K] the temperature of the refrigerant on the evaporator side

T_1 [K] the temperature of the refrigerant on the condenser side

W [W] compressor power input

$Q_{chiller}$ [W] the cooling delivered by the chiller

This COP_i is not the real value of the COP of the chiller. This corresponds to an ideal system; the chiller process is not really isentropic.

$$COP_{em} = \eta_i \times \eta_{mt} \times COP_i$$
$$= \eta_i \times \eta_{mt} \times \eta_{c,i} \times \frac{T_2}{T_1 - T_2} = \eta_{c,em} \times \frac{T_2}{T_1 - T_2} \text{ [-]} \quad (43)$$

With COP_{em}, the electric motor coefficient of performance

η_i [-] the isentropic efficiency of the compression process

η_{mt}[-] the combined efficiency of the electric driving motor and transmission

$\eta_{c,em}$ [-] the electric Carnot efficiency. It is usually between 0,4 and 0,6.

This equation gives a more accurate value. A deeper analysis can be carried out on this part, for another project, considering super heating, sub-cooling, pressure losses, different efficiencies and temperatures on the condenser and evaporator side, the use of a suction gas heat exchanger, the refrigerant type… (Lindholm, November 2009)

A good tool to carry this study is to use the software EES, especially with the cool pack package created by the technical university of Denmark. A downloading version can be found on their website: http://www.et.web.mek.dtu.dk/coolpack/uk/index.html

To be as close as possible to the real working conditions, some characteristics of real chillers are used. The data used come from the chiller 30 RA 005-015 from Carrier. It is an air-cooled liquid chiller and has a nominal capacity of 5-15kW. In the appendix E, "Chiller characteristics", the characteristics of the chiller are presented. (Carrier)

From the cooling capacities given in the technical data, a correlation between the COP of the chiller and the outdoor temperature is made. This is presented in the graph below.

The unit power input is used to calculate the COP of the chiller because it includes the power required to the compressor, to the fan on the condenser side, to the control circuit and to the pump.

The outlet water temperature on the evaporator side of the chiller is 7°C. The condenser entering air temperature depends on the outdoor temperature. Two characteristics of the chiller are drawn:

- One with a cooling capacity of 14,7 kW at 35°C outdoor, the 30 RA 015
- One with a cooling capacity of 11,2 kW at 35°C outdoor, the 30 RA 013

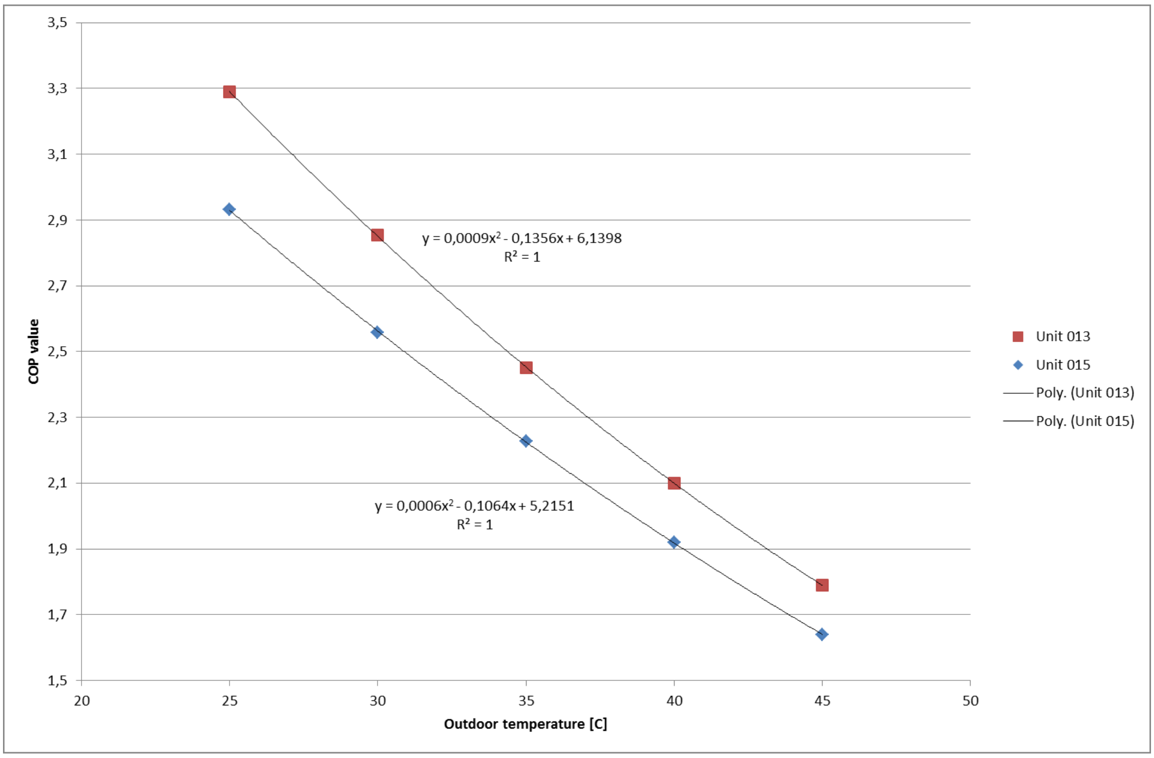


Figure 40 COP values for 2 units of constant chiller, depending on the outdoor temperature

6.3 The cooling strategy

The strategy consists of using the meteorological data for the upcoming day and night to predict the amount of cooling that the chiller should produce.

In the next part of the report, the figures showing the four seasons drawings always present the same periods of the year

- Summer case: hour 4705 – 4780: 16th July (midnight) to 19th July (4am)
- Spring case: hour 2400 – 2475: 10th April (11pm) to 14th April (am)
- Winter case: hour 1225 – 1300: 21th February (midnight) to 24th February (4am)
- Fall case: hour 6480 – 6555: 27th September (11pm) to 1st October (3am)

6.3.1 Estimation of the tanks losses to the ground

An estimation of the losses through the tank envelope for one day has to be predicted. The worth case is considered. This gives a margin in case the weather conditions are worse than predicted (hotter and wetter than the current climate).

$$Q_{prediction\ losses\ through\ the\ tank} = UA_{tank} \times (t_{out} - t_{tank})\ [\mathrm{W}] \qquad (44)$$

The worst case happens when the tank is full of chilled water, so $t_{tank} = t_{outlet\ chiller}$. This case would happens only the hottest day of the year as the size of the tank is designed to contain enough water to cool the house for this special day.

A cylindrical shape of the tank is considered, because it reduces the contact surface between the tank and the outside environment.

The tank is made of plastic sheet and a 5cm insulation polyurethane insulation (the characteristic values are the same as the one used for the polyurethane wall insulation). The tank is buried in the ground ($t_{out} = t_{ground}$).

6.3.2 Evaluation of the cooling needed for one day

The energy needed for one day corresponds to the cooling needed for the fan coil during the day and during the night (energy stored in the tank) and the losses through the tank. This energy is produced by the chiller.

$$E_{predicted\ cooling\ chiller\ one\ day} = \int Q_{fancoil\ day}\,\delta t + \int Q_{fancoil\ night}\,\delta t + \int Q_{prediction\ losses\ through\ the\ tank}\,\delta t \quad [W\times 24\ hours] \qquad (45)$$

From Figure 41 below, it is possible to see the cooling needed to be produced by the chiller for a whole day. It is calculated at every sunrise. To be accurate, a cycle of the cooling system lasts from a sunrise to the following day's sunrise. It is set on 24h on a day/night cycle beginning at sunrise. All values are negative, because it corresponds to cooling. Three entire days are shown for different seasons of the year, as explained before.

Each hour, the predicted cooling for one day is compared to the real cooling produced by the chiller after every hour during the day. This cooling load is summed every hour to know how much has been produced so far. On Figure 41 below, it corresponds to the curve which is reset every morning and that decreases during the day. When the real cooling produced by the chiller is lower than the prediction, the chiller stops working and the system is switched to night mode (when the blue curve gets below the green one).

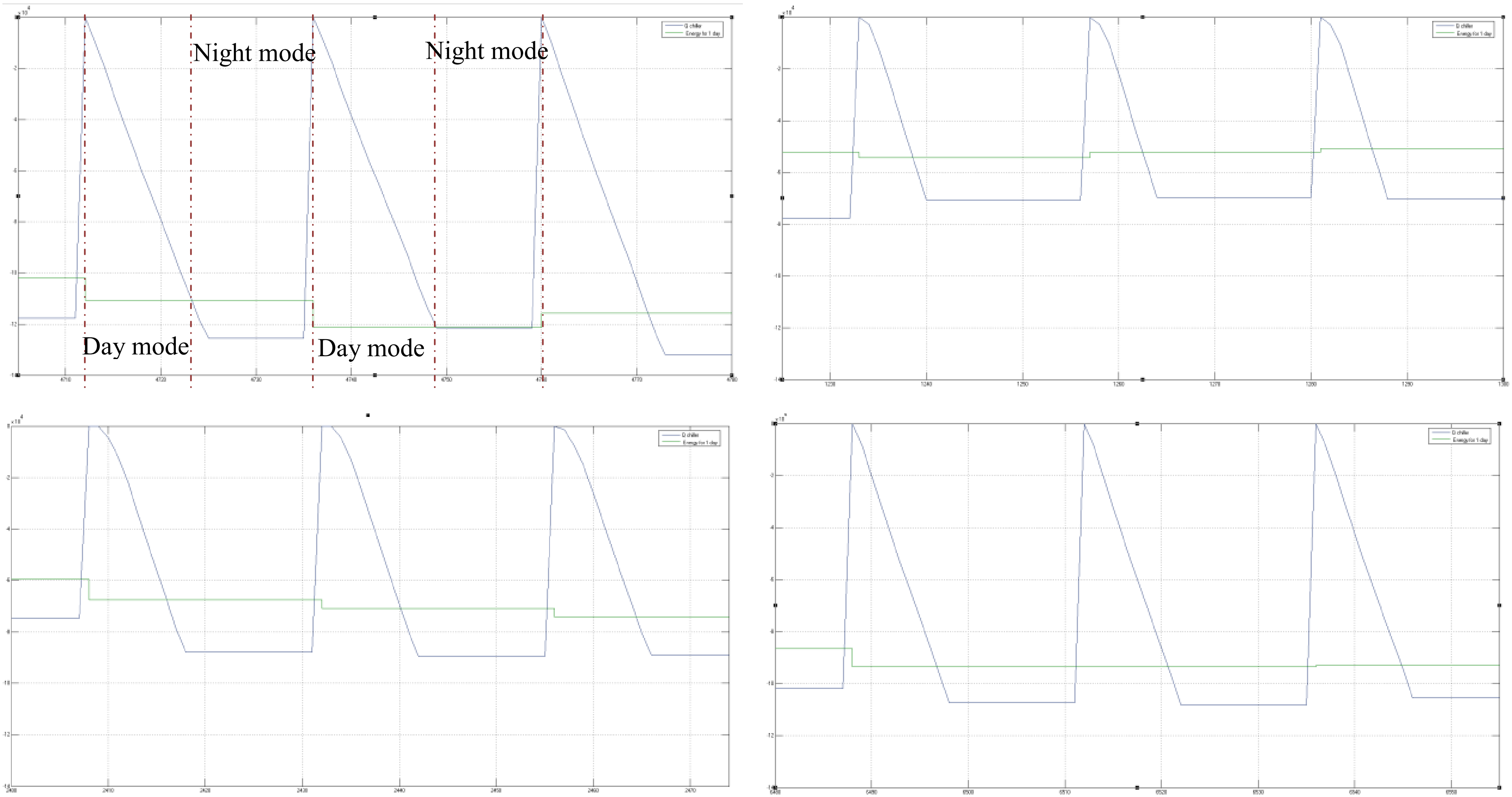


Figure 41 In green: Cooling need for a whole day (24h); in blue: Cooling provided by the chiller. (-14 to 0·10^4 W), Top left: summer time, top right: winter, bottom left: spring, bottom right: fall. In Jeddah. (simulation using the modelling tool Simulink)

6.4 Off-grid/On-grid

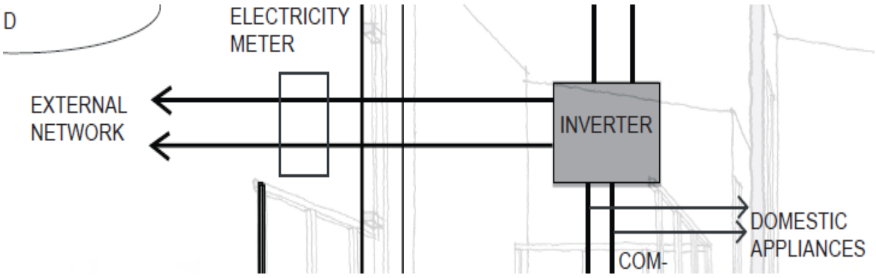


Figure 42 Part of the cooling system to be analysed: grid connection

Now, the characteristic of the chiller is set and the cooling strategy too. To know the cooling provided by the chiller and to compare it with the cooling needed for a whole day, the delivered electricity to the compressor must be known.

This electricity comes from the photovoltaic panels or the grid. Two strategies for providing electricity to the compressor are analysed.

6.4.1 Strategy 1: variable work chiller

A variable work chiller is a chiller with a maximum work load, and that can work at a lower rate of this maximum load. This rate is a certain percentage of the maximum load, and will be called "step" in the following parts. In this study, the chosen variable work chiller is considered having a power range of 10 possible steps from 10% to 100%. The strategy when using a variable work chiller is to adjust the amount of electricity provided to the chiller depending on the solar radiations electricity production and the need of cooling. The electricity provided to the chiller corresponds to a certain step.

There is a need to optimise the electricity use and production throughout the year. It is obvious that in winter, only a small amount of the electricity provided by the solar panels will be used for the air conditioning. On the other hand, in summer, there is a need to take electricity from the grid. Therefore, using the solar energy production and the cooling load previsions, it is possible to determine an optimal electricity use strategy.

In the calculations, the previsions are made as following:

First, a certain amount of solar panels is taken as the maximum area to be used. The requirement for this area is to provide enough energy to the chiller to achieve the cooling load for every day of the year.

Then, the amount of surplus energy provided to the chiller is derived. Depending on that amount, a series of 10 steps are defined for the work provided to the chiller. For example, if the cooling load is barely achieved, then the work will be almost always equal to 100% of the chiller work capacity. On the other hand, when the solar energy production is barely used, the chiller may work only at 10% of its capacity for the whole day. This has a huge advantage, as the chiller gets the better COP value with the lower percentage of the work load. Nevertheless, the change in the COP value is not taken into account in the calculations, as these are based on real chiller data.

This amount is then the electricity use provided to the chiller. With that one, it is possible to reduce the PV panel area, allowing the use of grid electricity during some of the days. The only requirement is to have as much electricity provided by the solar panels to the grid as the one imported from the grid on 1 year.

Yet, it should be noted that at the time when the report is written, no variable chiller exists for the low cooling capacity that is used in this project. There are only variable chillers designed for commercial buildings size, but not for residential ones. For the calculations, the data used come from real one-step work chillers, and it is assumed that they can work at lower rates of the maximum load.

6.4.2 Strategy 2: one-step work compressor

In this strategy, the chiller receives a constant work load, which is defined. It is switched on with the sunrise and works until it produces enough cooling for the whole day. This maximum work value is set for the chiller to be able to provide enough cooling for the hottest day of the year by working from sunrise to sunset. It means that, during winter, it will only run a few hours. The electricity received by the chiller does not follow the amount of solar radiations received by the photovoltaic panels, whereas for the variable work chiller, advantage is taken from the solar curve shape.

6.4.3 Results

6.4.3.1 Comparison between the two strategies

From the simulation results, specific values have been picked for this table:

Table 12 Comparison between two strategies: 1-step and variable work compressors

	1-step work	Variable work
Maximum work to the chiller [W]	4500	5000
Electricity provided to the chiller for 1 year [kWh]	13600	13710
Cooling capacity of the chiller for 1 year [kWh]	33500	43000
PV panels area [m^2]	71	72
Total energy produced by the solar panels for 1 year [kWh]	13650	13750
Total energy exported to the grid by the solar panels for 1 year [kWh]	3800	2740
Total energy imported from the grid for 1 year [kWh]	3740	2700

With a variable work strategy, the maximum work to the chiller is bigger because the compressor works by step and follows the amount of sun radiation received by the solar panels, hence the electricity delivered by the panels. Compared to the one-step work compressor, it provides less electricity to the chiller at the beginning and end of the day, but more in the middle of the day, when the electricity coming from the photovoltaic panels is high. That is why the maximum work to the chiller is higher in the middle of the day and the size of the chiller is also bigger for the variable work strategy.

For the one-step work strategy, the chiller 30RA 013 has been chosen and for the variable work strategy, the 30RA 015 unit is used for a bigger cooling capacity.

The requirement to have as much electricity exported by the solar panels to the grid as the one imported from the grid for 1 year is shown in the previous table (3800/3740kWh for the one-step work chiller and 2740/2700 kWh for the variable one). This means that, for one year, the electricity provided to the chiller is the same as the electricity produced by the solar panels (13600/13650 kWh). The electricity provided to the chiller is the same for both strategies in one year time. This was expected, as the cooling load to be provided to the room is the same, i.e. the cooling capacity of the chiller differs during the day, but the chiller produces the same amount of cooling for one whole day with both strategies.

The PV panels area is the same for this reason too, but as the variable work strategy follows the electricity produced by the panels, the electricity received by the chiller is better optimised to avoid importing and exporting electricity from the grid and use as much electricity provided by the PV panels as possible (2700 kWh for the variable work strategy against 3800kWh for the one-step work strategy).

6.4.3.2 Electricity used with the variable work strategy

In this part will be presented the results obtained when using a variable work chiller.

On Figure 43 below is shown in blue the electricity provided to the compressor to create enough cooling, using the strategy explained above. The electricity provided by the photovoltaic panels is in green. There is a graph for each season of the year. They are designed for Jeddah and represent 3 days with the beginning of the electricity production at sunrise.

The simulation shows that the electricity provided to the chiller follows the electricity production of the photovoltaic panels. It is easy to see the difference between the day mode of the chiller, when electricity is provided to it, and the night mode, when no electricity is provided.

The difference between the real electricity needed for the chiller and the electricity provided by the PV panels represents the grid part:

- When this difference is positive, electricity is imported from the grid.
- When this difference is negative, electricity is exported to the grid.

The area between the two curves always represents the amount of either imported or exported electricity from the grid. This total amount is shown on Figure 44 for a whole year.

In summer case, electricity is only imported from the grid all day long. The chiller is used at its maximum work capacity (5000W) during a long time in the middle of the day. All the electricity produced by the solar panels is used by the chiller. The chiller is switched off at sunset.

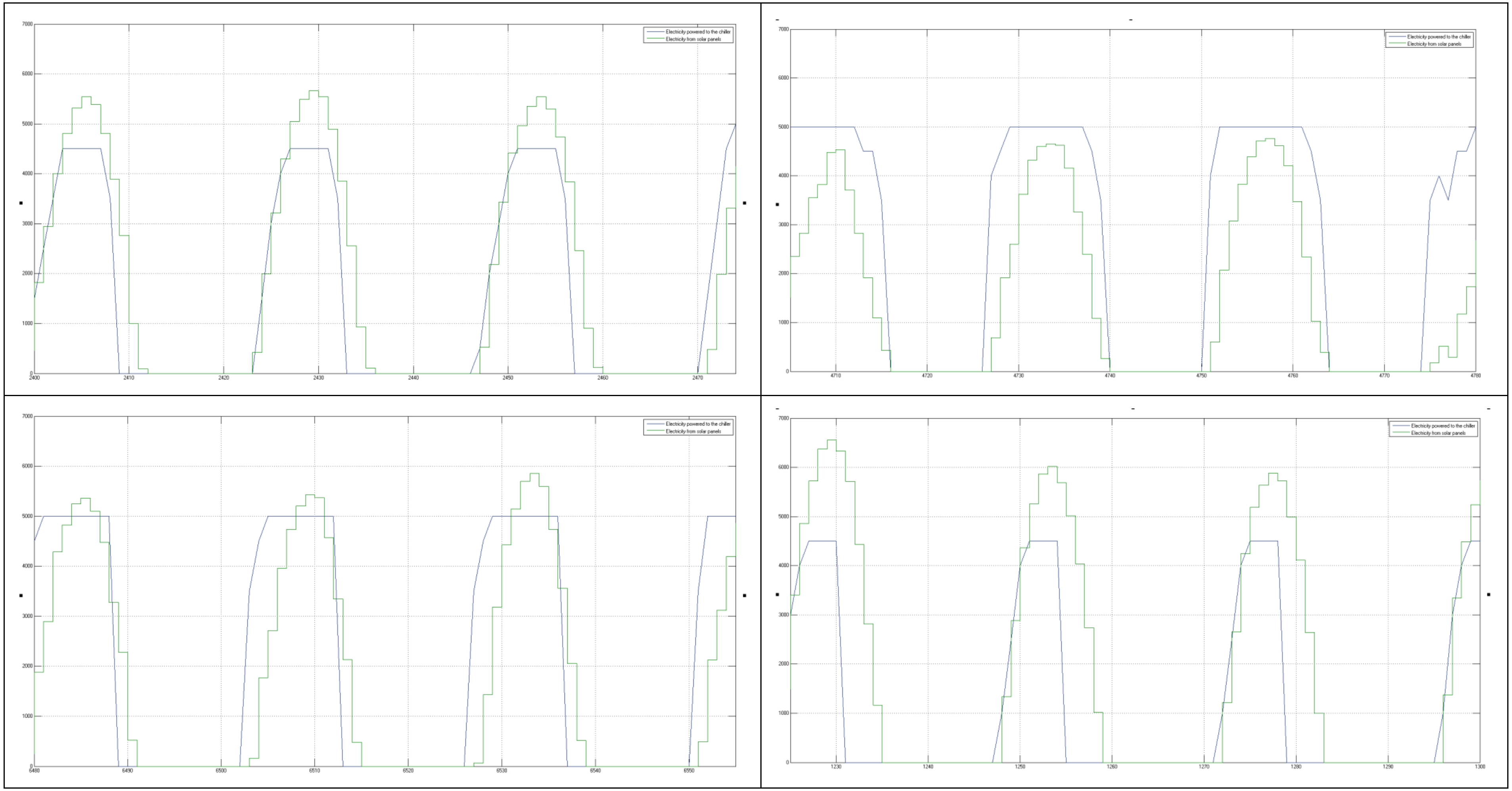


Figure 43 In green the electricity provided by the PV panels, in blue: the real electricity need for the chiller (0 to 7000 W) Top left: spring case, top right: summer case, bottom left: fall case, bottom right: winter case. (Simulation using the modelling tool Simulink)

In the fall case, some of the electricity is imported to the grid and some is exported. Even if the predicted cooling needed for the whole day is set, it is hard to provide electricity to the chiller all day long by following the electricity produced by the solar panels, and reach this predicted cooling need. That is why some electricity is imported from the grid in the morning. And, if the chiller provides enough cooling before the sun sets, it is switched off and the electricity from the solar panels is exported to the grid. This is a preferable strategy than not providing enough cooling for the upcoming night. The system is optimised to avoid as much as possible this imported and exported electricity on the same day. The chiller works at its maximum load, i.e. 5000W.

For “early spring” and winter, the electricity produced by the photovoltaic panels is enough to run the chiller for a whole day. There is only electricity exported to the grid and no electricity imported from it. As soon as the chiller produces enough work for the next day/night cycle, it is switched off. All the electricity from the PV panels goes to the grid. The maximum work load during the day is 4500W.

6.4.3.3 Imported and exported electricity to the grid with the variable work strategy

The figure below shows that much electricity is exported during winter time (with many peaks of about 5000W) since the cooling demand is low. On the contrary, nearly no electricity is exported during the summer time. The total amount of this surplus production for one year on an hourly basis is 2740kWh (see Table 12).

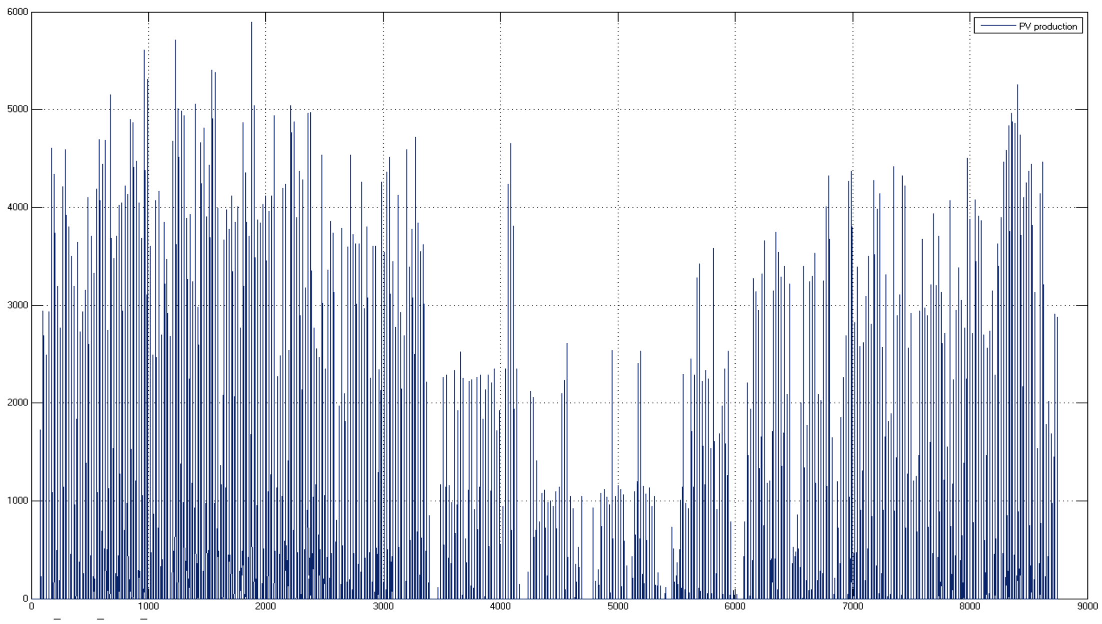


Figure 44 Electricity surplus production by the PV panels (0 to 6000 W) through the year (8760h) with the variable work strategy (simulation using the modelling tool Simulink)

On the figure above, one can notice that even in summer some electricity is produced in surplus. This happens sometimes when the chiller is turned off before sunset, usually about one hour before. There are two causes for that: either the cooling load is low, and there is no need to run the chiller for the whole day, or the cooling provided exceeds the cooling need, already one hour before sunset. This means that the strategy to choose which step to be used is not designed in an optimum way. Indeed, when the chiller has a large work capacity, it is harder to adjust precisely its electricity provision. The difference between two steps is so big that for some days, the amount of electricity is barely enough to produce the required cooling load, whereas some days, the production is largely enough, and requires the chiller to be turned off earlier. These two cases can be seen on Figure 45 for summer and fall cases.

The imported and exported electricity is shown in Figure 45 in more detail, for the same four periods of the year. When the curve is above zero, the electricity is exported to the grid and below zero, it is imported.

In summer case, the graph shows that a lot of electricity is imported at the beginning of the day and at its end (until 3500W). There is nearly no electricity need at noon, when the PV panels produce the highest amount of electricity.

In Saudi Arabia, the electricity demand peak load is in the afternoon, especially at the end of it. One improvement would be to try reducing the electricity imported from the grid at the end of the day, to avoid participating in the increase in the electricity consumption during the peak load. It would also be safer, as the Saudi Arabian electricity company has trouble providing so much electricity during the peak load and power cuts can occur.

In the “early spring” and winter case, the graph shows that the electricity is only exported all day long, especially when the chiller is switched off before sunset, when the exported electricity can be really high (until 5000W).

In the fall case, the explanation given on the graph with the electricity provided by the PV panels and the real electricity need for the chiller are observable here. Electricity is imported at the beginning of the day (3000W). As the PV panels produce the highest amount of electricity at noon the electricity imported from the grid is null. There is even some electricity exported. The last peak of imported electricity corresponds to the moment when the chiller switches off before sunset (2000W).

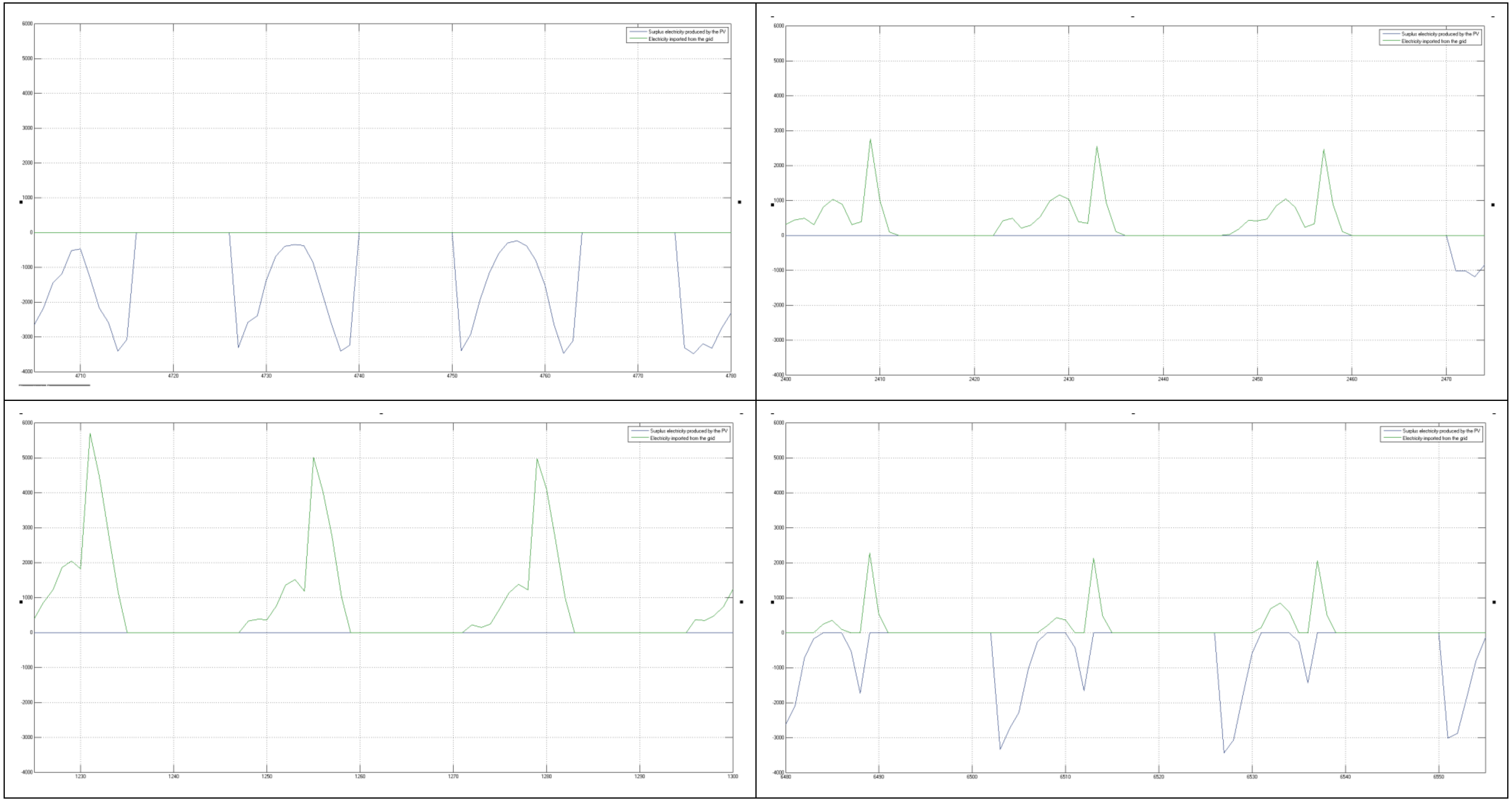


Figure 45 In green: electricity produced by the PV panels and exported to the grid (-4000 to 6000 W). In blue: electricity imported from the grid. Top left: summer case, top right: spring case, bottom left: winter case, bottom right: fall case. (Simulation using the modelling tool Simulink)

6.5 Detailed design of the distribution pipes

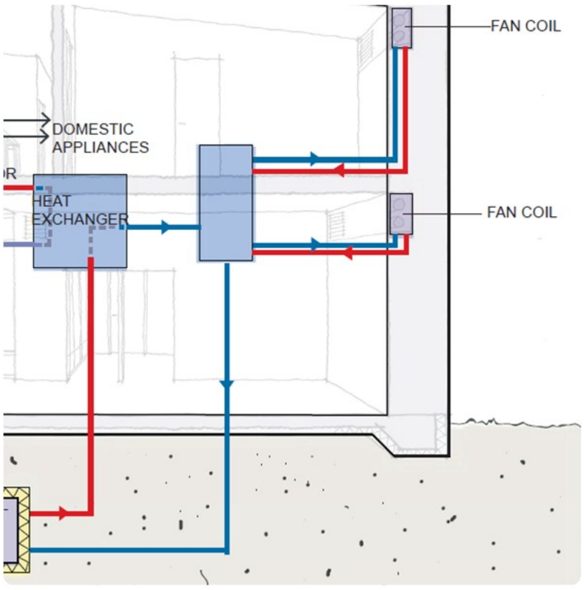


Figure 46 Part of the cooling system to be analysed: distribution pipes

Now that the fan coil cooling load and the cooling provided by the chiller are defined, the purpose is to calculate the flows and the temperatures in the different parts of the pipes connections and the water tank. Thanks to this, the chiller is able to provide the cooling load both to the fan coil – to cool the building directly – and to the tank – for cooling at night. Some values are set; some others have to be defined.

All the figures in the water tank design and the distribution pipes come from the simulation with the variable work strategy. A comparison with the one-step work strategy is made in the tables.

6.5.1 Day mode (chiller on)

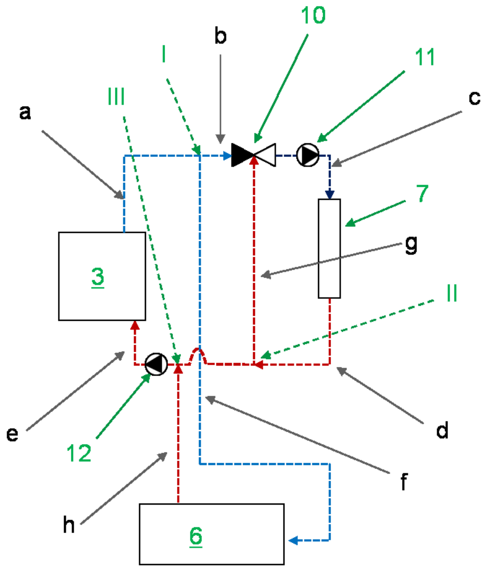


Figure 47 Water system in day mode

The numbers and letters designing the different part of the water system on the drawings in day and night mode refer to the table in Figure 48.

To design the water pipe system, the equation between different flows and temperatures at a mixing point is used. For example, at the mixing valve 10,

$$t_b \times C_b + t_g \times C_g = t_c \times C_c \text{ [W]} \qquad (46)$$

With the thermal capacity flow $C = \dot{V}\rho c_p$ [W/°C]

The specific thermal capacity and the density of the water are assumed to be constant. To be accurate, the density and the heat capacity of the water are not the same at different temperatures. For example between 6°C and 12°C, there is a variation of 12°C, changing the thermal capacity of 0,3%. This value is so small that it will not influence the results but one must be aware of it.

Table 13 Water characteristics for different temperatures

Water temperature	ρ (kg/m^3)	c_p (kJ/kg.K)	ρc_p (kJ/m^3.K)
6°C	999,91	4,204	4203,75
12°C	999,498	4,193	4190,90

So,

$$t_b \times \dot{V}_b + t_c \times \dot{V}_c = t_g \times \dot{V}_g \text{ [l.°C/s]} \qquad (47)$$

$$\dot{V}_b + \dot{V}_c = \dot{V}_g \text{ [l/s]} \qquad (48)$$

In the same way, at point III,

$$t_d \times \dot{V}_{d-g} + t_h \times \dot{V}_h = t_e \times \dot{V}_e \;\; [\text{l.°C/s}] \tag{49}$$

$$\dot{V}_{d-g} + \dot{V}_h = \dot{V}_e \; [\text{l/s}] \tag{50}$$

The water flows in a closed circuit: the same volume of water is used in all the pipes and the tank; no water is supplied. So the flow entering the tank is the same as the one leaving it: $\dot{V}_h = \dot{V}_f$

Moreover, $Q_{chiller}$ (the cooling provided by the chiller) and Q_{fancoil} (the cooling needed to the fan coil) are known from the previous parts of the cooling system design.

On the evaporator side of the chiller,

$$Q_{chiller} = \dot{V_a} \times \rho \times c_{p,eau} \times (t_e - t_a) \;\; [\text{W}] \tag{51}$$

On the water side of the fan coil,

$$Q_{fancoil} = \dot{V_c} \times \rho \times c_{p,eau} \times (t_c - t_d) \;\; [\text{W}] \tag{52}$$

Table 14 Known and unknown variables to solve the water system equations for the day mode

Known variables			Unknowns variables	
$t_a = t_b = t_f$	outlet chiller temperature	7°C	$\dot{V}_e = \dot{V}_a$	flow through the chiller
$t_d = t_g = t_{d-g}$	outlet fan coil temperature	14°C	$\dot{V}_f = \dot{V}_h$	flow through the tank
t_c	inlet fan coil temperature	8°C	$\dot{V}_g$	flow in the return connection
t_h	outlet tank temperature		$\dot{V}_b = \dot{V}_{d-g}$	inlet and outlet return connection flow
$Q_{chiller}$	cooling from chiller		$\dot{V}_c = \dot{V}_d$	flow through the fan coil
$Q_{fancoil}$	cooling to the fan coil		t_e	inlet chiller temperature

6.5.2 Night mode (chiller off)

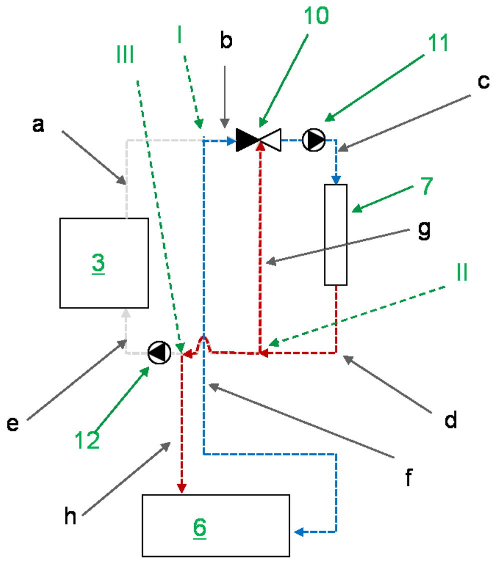


Figure 49 Water system in night mode (legend in Figure 4)

For the night mode, the equation, at the mixing valve 10, is still the same

$$t_b \times \dot{V}_b + t_c \times \dot{V}_c = t_g \times \dot{V}_g \qquad (53)$$

And at point II:

$$\dot{V}_{d-g} + \dot{V}_h = \dot{V}_e \text{ [l/s]} \qquad (54)$$

And on the water side of the fan coil,

$$Q_{fancoil} = V_c^{\cdot} \times \rho \times c_{p,eau} \times (t_c - t_d) \text{ [W]} \qquad (55)$$

Table 15 Known and unknown variables to solve the water system equations for the night mode

Known variables		Unknowns variables	
$\dot{V}_c = \dot{V}_d$	flow through the fan coil	$\dot{V}_f = \dot{V}_b = \dot{V}_h$	flow through the tank
$t_d = t_g = t_h$	outlet fan coil temperature	$\dot{V}_g$	flow in the return connection
$t_b = t_f$	outlet tank temperature	t_c	inlet fan coil temperature
$Q_{fancoil}$	cooling to the fan coil		

6.5.3 Flows through the system

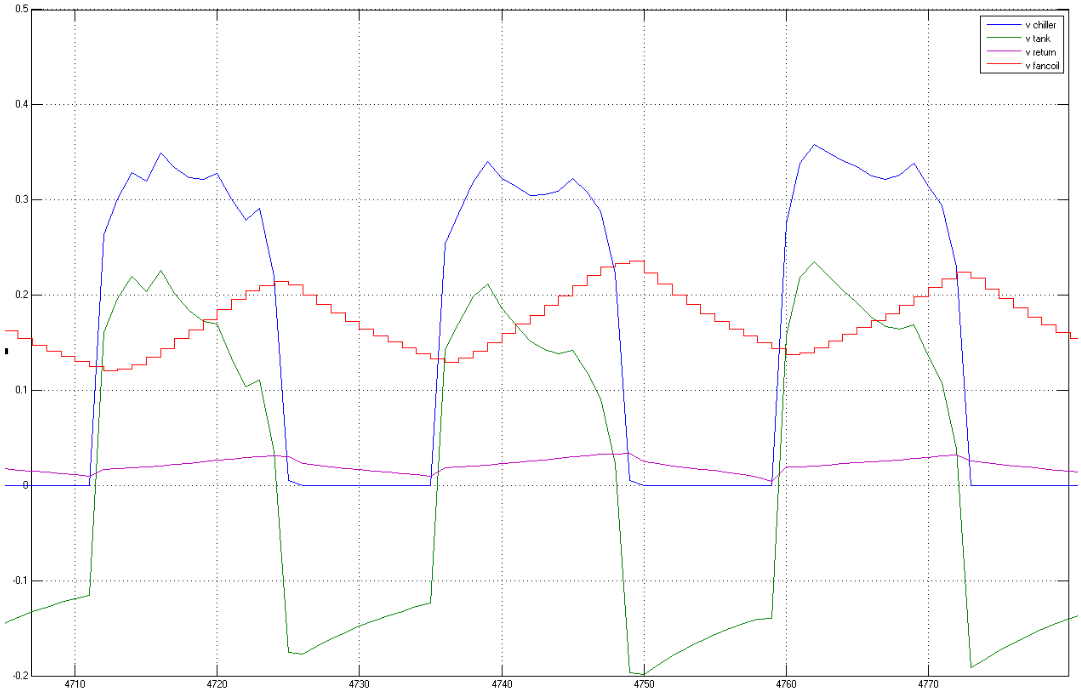


Figure 50 Flows through the system (-0,2 to 0,5 l/s) in summer time. In blue: chiller, in green: tank, in purple: return flow, in red: fan coil. (Simulation using the modelling tool Simulink)

In the figure above, the flows are shown for three days in summer time. The flow through the chiller is only positive during daytime, as the compressor is turned off at nights. The flow through the tank also follow this day/night mode: it is positive during the day and negative during the night, as the flow is reversed through the tank when the chiller is off: the cool water is taken from the tank and sent to the fan coil.

The fan coil flow has a peak value at the end of the afternoon. Indeed, as explained before, it is at this time of the day the cooling demand is the highest. It is the lowest in the beginning of the morning, at sun rise. The return flow also follows this shape, and is slightly higher during the peak cooling demand. Here, it is about 0,03 litres per second, at the highest. When increasing the return flow, it avoids using too much cold water at the inlet of the fan coil. As a result, it both decreases the water tank volume and the cooling capacity of the chiller.

Table 16 Flows through the system

	Fixed work	Variable work
Maximum flow through the fan coil[l/s]	0,236	0,236
Maximum flow through the chiller [l/s]	0,449	0,495
Maximum day flow through the tank [l/s]	0,428	0,440
Maximum night flow through the tank [l/s]	0,198	0,199
Maximum return flow [l/s]	0,034	0,034

6.6 The water tank

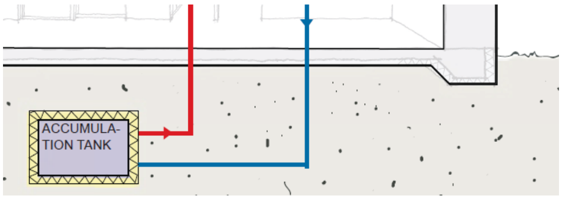


Figure 51 Part of the cooling system to be analysed: water tank

In the following part will be described the principles of the water tank design.

The water tank is a major component of the design. Thanks to it, part of the cooling capacity, produced by the chiller during the day, can be used all night long. With the equations from the previous part, it is possible to know the flow coming through the tank.

The main goal of this design is to avoid filling the tank every day totally when it is not necessary. Consequently, in the cooling prediction part, only the cooling needed by the fan coil for the upcoming day and night is considered. With this, the chiller produces only the exact amount of cooling every day and the tank only stores the amount of cooling necessary to finish a day/night cycle. Surplus electricity is redistributed to the grid.

The tank is split into 2 parts:

- The top one with a warm water temperature t_{top} and a volume V_{top}. The warm water has always a smaller density than the cold one. During the day mode, the top part of the tank sends water to the pipes system to be cooled by the chiller, so its volume decreases. During the night mode, the top part of the tank receives water from the outlet of the fan coil, so its volume increases.
- The bottom one with a cool water temperature t_{bottom} and a volume V_{bottom}. During the day mode, the bottom part of the tank receives water from the outlet of the chiller to store water for the night mode, so its volume decreases. During the night mode, the bottom part of the tank sends cool water the fan coil, so its volume decreases.

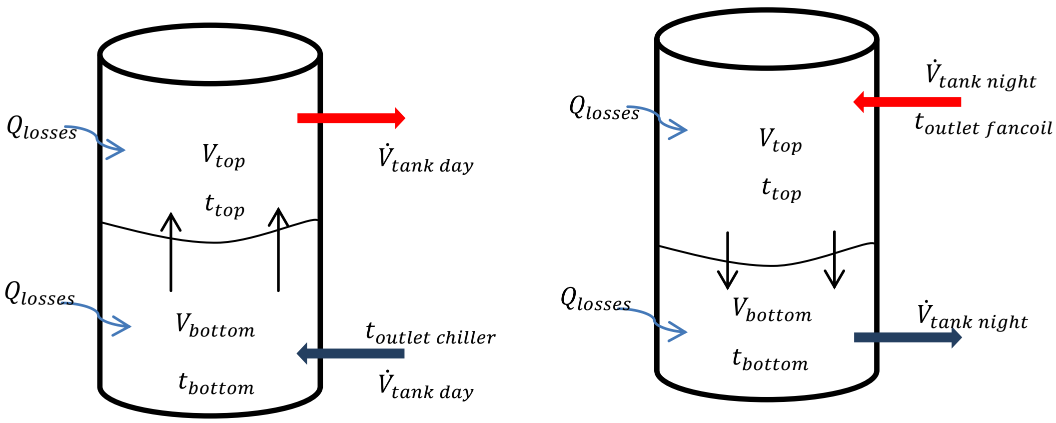


Figure 52 Explanation of the tank filling strategy

At the end of a day/night cycle, the volume V_{bottom} left is considered as a part of V_{top}. There is no volume V_{bottom} at sunrise in the tank, only a volume V_{top} to be cooled by the chiller.

The day with the maximum need of cooling for the building defines the maximum V_{bottom}, which is the size (volume) of the tank. Since the total volume of water in the tank does not change, V_{top} and V_{bottom} are supplementary. The sum of these two volumes always represents the total volume of the tank:

$$V_{top} = V_{tank} - V_{bottom} \text{ [litres]} \tag{56}$$

And (with a reset at every sunrise)

$$V_{bottom} = \int \dot{V}_{tank}\ \delta t \text{ [litres]} \tag{57}$$

As a result, $\dot{V}_{tank}$ is positive during the day (increase ofV_{bottom}) and negative during the night (decrease ofV_{bottom}).

After predicting the energy required for one day, the simulation is run for each hour to see the variation of temperature and volume in the two parts of the tank and the variation of temperature and flow at different points of the piping system.

For the top part, the temperature equation is:

$$C\frac{dt_{top}}{dt} = UA_{top} \times (t_{out} - t_{top}) + \rho c_{p,water}\dot{V}_{tank\,night} \times (t_{outletfancoil} - t_{top}) \text{ [W]} \tag{58}$$

For the bottom part,

$$\mathrm{C}\frac{\mathrm{dt_{bottom}}}{\mathrm{dt}} = UA_{bottom} \times (t_{out} - t_{bottom}) - \rho c_{p,water}\dot{V}_{tank\,day} \times (t_{outletchiller} - t_{top}) \text{ [W]} \tag{59}$$

Of course, A_{bottom} and A_{top} vary and correspond to the area of the envelope of the tank in contact with the top and bottom volume. To calculate C, the equation is the same as before but both heat capacities of the tank envelope and the water has to be considered.

6.6.1 Filling of the tank

The simulation is launched for both variable and fixed work strategies. On the table below, the tank volumes corresponding to the maximum of V_{bottom} are shown.

Table 17 Comparison of the tank volumes

	Fixed work	Variable work
Maximum tank volume [litres]	8825	9670

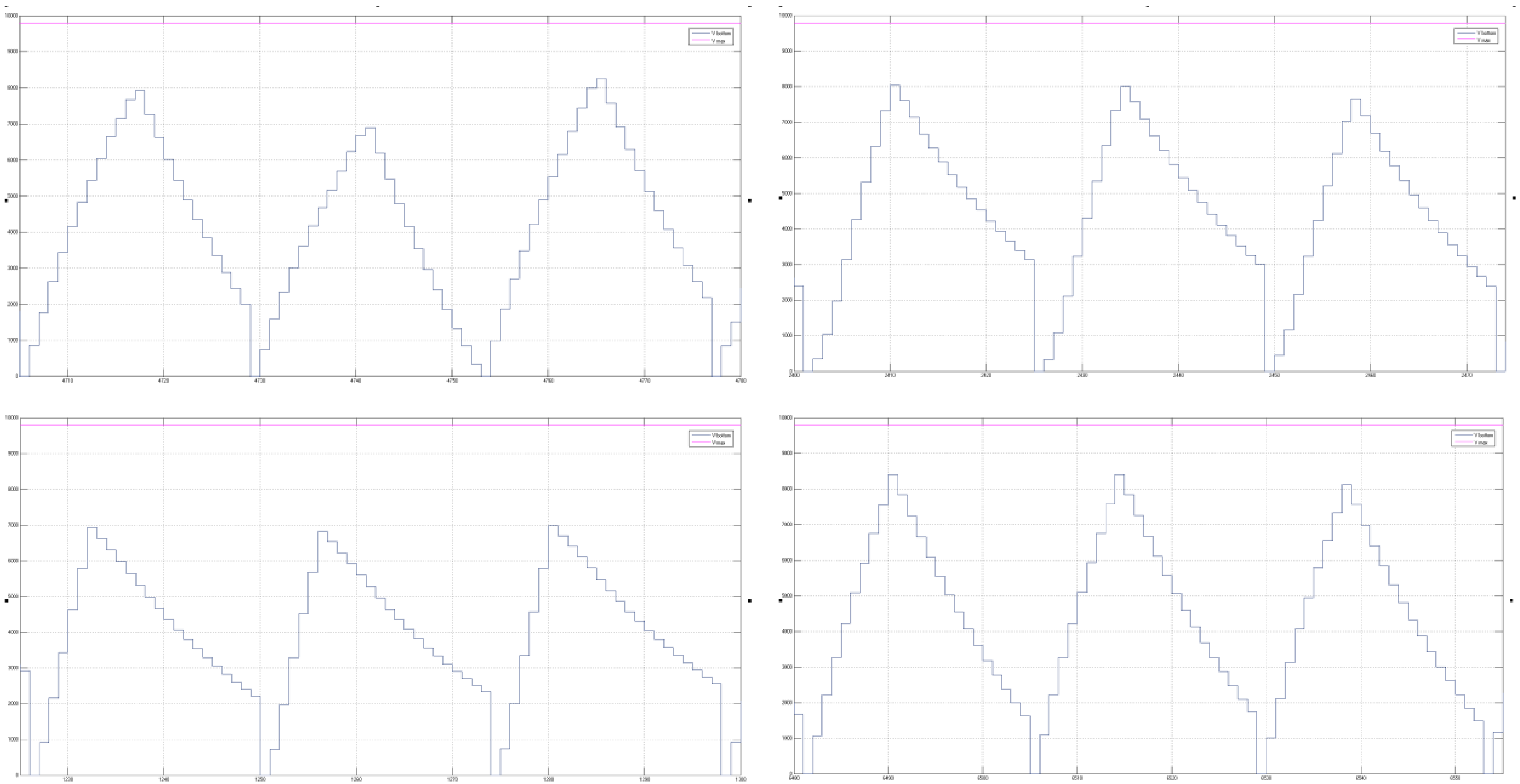


Figure 53 Filling of the bottom part of the tank (in blue); maximum volume of the tank (in pink). (0 to 10000 litres) Top left: summer time, top right: spring, bottom left: winter, bottom right: fall. (Simulation using the modelling tool Simulink)

On Figure 53, the bottom volume of the tank, with the variable work strategy, is shown. This volume increases during the day mode with cool water supplied by the chiller to the tank, and decreases during the night mode with cool water from the tank supplied to the fan coil. Depending on the cooling needed, i.e. on the time of the year, the tank is more or less filled with cool water: 8300 litres for summer case, 8000 litres for spring case, 7000 litres for winter case and 8400 litres for fall case.

For the calculations, the predicted heat losses appear to be higher than the real losses. Therefore, a certain volume of water always remains in the bottom part of the tank at the end. As the weather can change between the forecast and the reality, this is the safety margin. For example, for the 2nd day shown on the summer case graph, the V_{bottom} left at the end of a day/night cycle is 300 litres on the 7000 litres present at the beginning of the night mode. It is possible to see too, that the volume of cooled water is considered to be 0 at the beginning of every new cycle (at sunrise).

6.6.2 Temperatures in the tank

Because of the high heat capacity of the water, the temperature variations in the two volumes of the tank, due to heat losses to the ground, are small (less than 1°C for the bottom one with cooled water at 7°C). Because the top volume receives water from the outlet of the fan coil during the night mode, it tends to reach the same temperature, which is set to 14°C (refer to the distribution pipes design chapter). This is the same reason for the bottom volume and the 7°C water temperature received from the outlet of the chiller during day mode.

The mean temperature of the tank varies a lot while the volumes of V_{bottom} and V_{top}vary. It decreases with the increase of V_{bottom} filled with cooled water and the decrease of V_{top} during the day mode. It increases with the increase of V_{top} filled with warmer water and the decrease of V_{bottom} during the night mode.

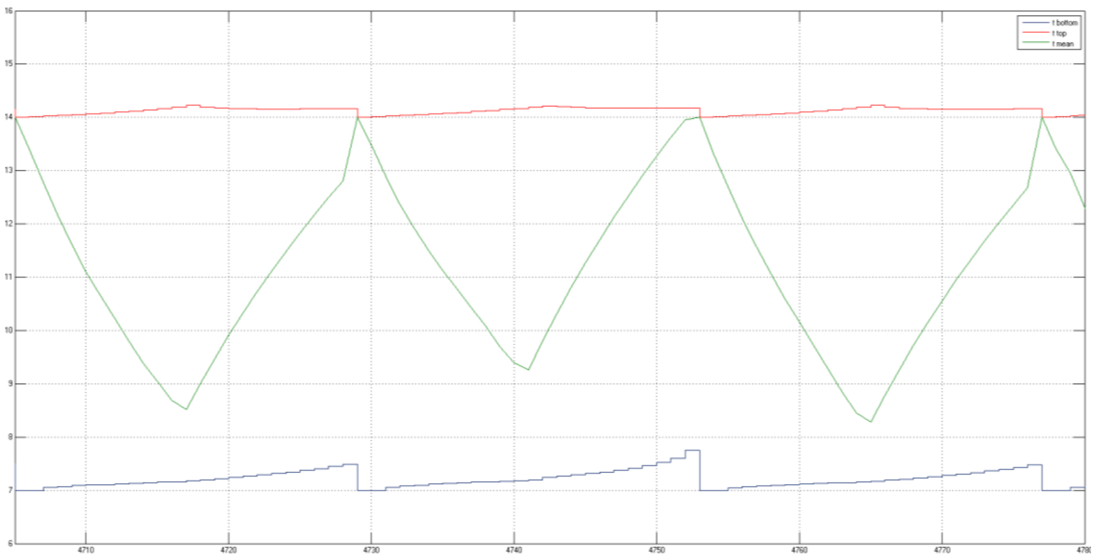


Figure 54 Temperature in the tank (6 to 16°C) In blue: bottom part; in red: top volume, in green: mean temperature of the tank. For the summer case. (Simulation using the modelling tool Simulink)

7 Discussion

The goal of this study was to analyse the cooling system explained at the beginning of this report. The concept has been created by the companies Norconsult and Agera. The aim of the present thesis work was to develop and design it in detail. This cooling system, using photovoltaic panels and storage tank in a climate with high outdoor temperatures and high solar radiations, is really suitable. The chiller has a high efficiency; the water has a high storage capacity. The strategy to minimise the electricity exchange from and to the grid, and to use as most electricity as possible from the solar panels is also very valuable.

The design of the cooling system and the connection between the different parts are ones of the possibilities which could be imagined. The results are positive and the photovoltaic panels' area is reasonable. Yet, some other cooling system possibilities could also be chosen. In this part, a short analysis discuss if they are better or worse than the ones designed in this report.

7.1 Same system, other alternatives

One major question when looking at the previous results is the possibility to improve the size of the tank. To solve that, some parameters can be changed, such as: increasing the inlet temperature of the fan coil and keeping the same temperature at the outlet of the chiller. Thanks to this, during the night mode, the return flow increases and the flow coming from the tank gets smaller. Whereas the temperature difference between the fan coil inlet and outlet is still kept to 6°C, the temperature at the outlet of the fan coil is higher. Thus, the temperature difference between the chiller inlet and outlet is increased: the chiller produces more cooling. Nevertheless, since the flow leaving the tank during the night mode is smaller, the maximum tank volume is also smaller. To put it in a nutshell, diminishing the tank volume can be done by adjusting the inlet and outlet temperature on different parts of the connection system between the chiller and the fan coil. Yet it should be noted that the fan coil would be less efficient.

Another assumption in this studied was to aim for having as much electricity provided by the solar panels to the grid as the one imported from the grid for 1 year. It means that, for a year, the building does not use electricity for cooling. Yet, during the peak load in summer, some electricity must still be provided by the grid. To avoid importing too much electricity from the grid at this time, it could be a good idea to increase the size of the solar panels. Of course, for a year, the system will export more electricity than it imports for cooling.

Also, in the previous design, the solar panels were assumed to be fixed on the roof. Using sun trackers, in vertical or horizontal directions, improve the solar radiations received by the solar panels, hence increasing the electricity produced by the photovoltaic panels. Otherwise, the panels can also be fixed, but not facing the south. By facing an angle between the south and the west side, they produce their highest amount of electricity, not at noon, but later in the afternoon, during the peak electricity consumption in Saudi Arabia. Subsequently, the cooling system reduces its electricity consumption from the grid during the peak load.

7.2 Modification of the system

Some changes in the building and the cooling design can be done and analysed to increase the efficiency of some parts of the system. All these cases can be studied separately and each of them can be designed for the Saudi Arabian climate to see if they are suitable.

7.2.1 Active cooling systems

In the previous study, the cooling system chosen was a vapour compression machine coupled with photovoltaic panels. As a solar cooling system, other systems also exist. On the figure below examples are shown of the different existing solar cooling systems. The most common ones are the vapour compression cooling, the absorption cooling and the dessicant cycle cooling. These last two systems are presented in the annex G, and both of them use solar thermal panels. These ones could be efficient with high outdoor temperatures and high solar radiations. As shown in this report, the equivalent temperature of the surface of the panels can reach 100°C, which can produce heat to really high temperatures.

The vapour compression system has a better COP than the absorption system, so it has a more efficient cooling purpose and is more suitable for use in small appliances like housing. For the advantages and drawbacks of the other systems, refer to annex H.

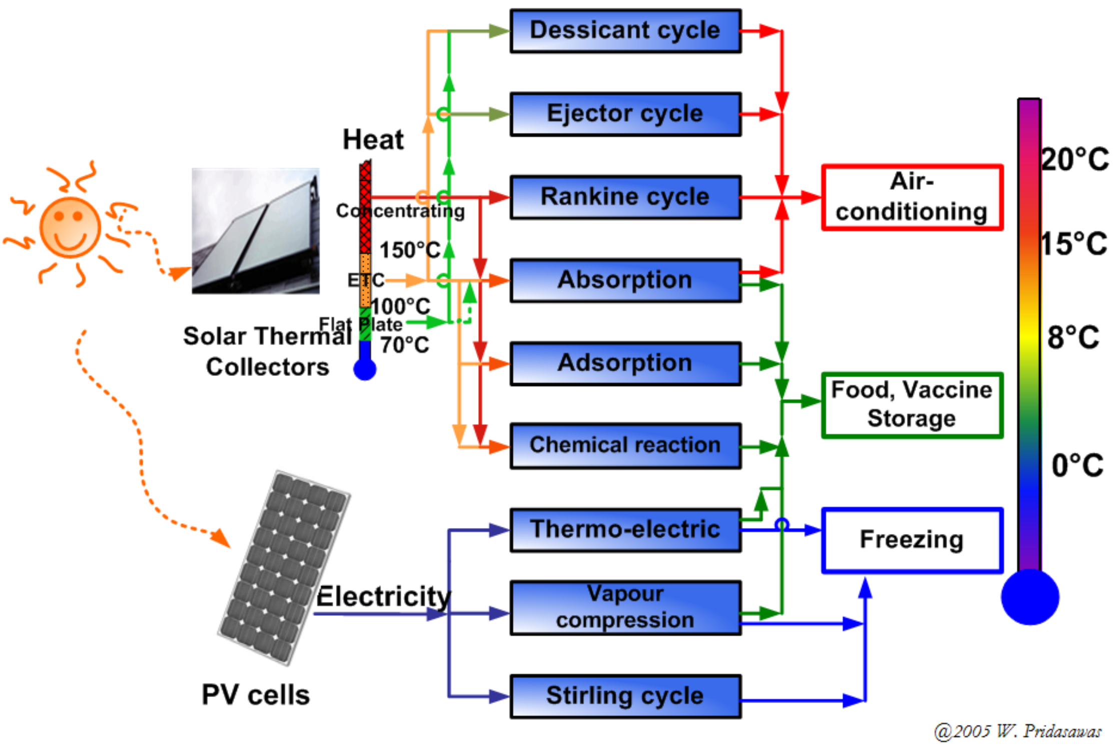


Figure 55 Difference between solar cooling systems (Pridasawas, October 2006)

To improve the efficiency of the chiller, a water to water system can also be used, which will not depend on the outdoor conditions but on the water temperature on the condenser side. This water can be cooled down in the ground, by drilling pipes under the building to use the mean outdoor annual temperature, which is lower than the outdoor temperature during the hot season.

Finally the efficiency of the system can also be increased by lowering the coolant temperature. Instead of water, glycol is used, since its freezing point is below the water one. As the vapour compressor system can produce a high cooling demand and reach freezing conditions, ice storage system will then be used, instead of liquid water storage. To know how it works, there are interesting reports on the subject with applications in Saudi Arabia. There are also comparisons with the cooled water storage solution. (Syed Mohmood Hasnain, 1999) (Hasnain, 1997) (International Energy Agency, 2002)

7.2.2 Passive cooling systems

Some passive cooling systems can be analysed too, to avoid using too much mechanical ventilation and cooling: optimisation of the orientation of the house, windows shading (inside and/or outside), landscape, natural ventilation, chimney effect, earth cooling... Some passive cooling solutions are already used on this building, like overhangs, triple glazed windows, insulation (for Building 2) or high thermal mass.

The design of the building could be rearranged in a shape with a longer west-east axis than the south-north one. With an overhang, and in the conditions of Saudi Arabia, where the sun is, most of the time, high in the sky, the south wall is really well protected from direct sun radiations. On the contrary, the west and east façades are not so well protected with overhangs because the sun is lower and the sun radiations are high. So, it is more suitable if these facades are small.

The best is to avoid windows on the west and east facades and install them on the north and south façades instead. Some cross ventilation can be done too by this way. An air speed of 0,5m/s with a 50% humidity air will create a temperature drop of 3°C. Non-opening windows on the walls submitted to sand storm or dusty wind has to be considered.

Vegetation around the building can provide fresh air and shading to the building. Of course, people have to be careful not to shade the photovoltaic panels at the same time. Trees and grass can decrease the surrounding temperature of 6 °C. Grass reflects only 20% of the solar radiation whereas concrete which reflects 40%. The vegetation effect is particularly efficient in hot and dry climates due to moisture release to the air.

Earth cooling can be created by underground pipes, providing fresh outdoor air to the building. The warm outside air is cooled down in the pipes thanks to the ground. In fact, the ground has a lower temperature than the outdoor air. The deeper the pipe is in the ground, the closer the temperature is to the annual average outdoor air temperature of. Yet, mould growing should be avoided in the pipe. It has to be well ventilated with use of a fan or require dehumidification in hot humid climate. Also, after a certain depth, the ground temperature starts to increase again when going deeper. Furthermore, one should not forget the high costs of digging in the ground, especially in the sand, as it is the case in the selected regions.

High thermal mass is a good solution in the arid climate of Saudi Arabia because the temperature at night can be very low even during summer, due to the lack of moisture in the atmosphere which usually traps the heat. So the walls facing the sun will be cooled down at the beginning of the day and it will take more time to heat up the wall all day long.

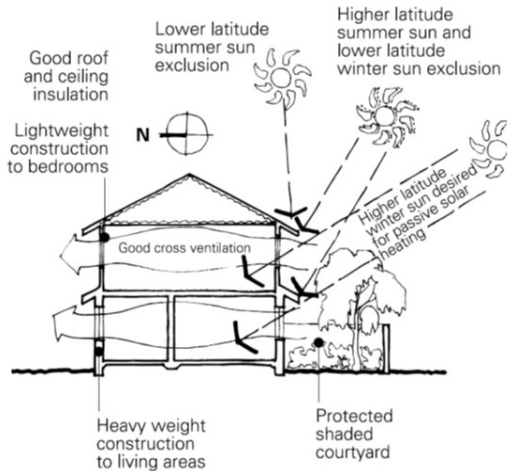


Figure 56 Effect of some passive cooling systems (Chris Reardon D. C.)

7.3 Conclusion

Even if, in all this study, most of the parameters that influence the cooling system have been analysed, included all of the major ones, there is still a difference between the study and the reality. This report tries to go as close to the reality as possible, but the best way is to test a model in real conditions. For example, it is hard to define how fast the efficiency of the solar panels decreases with dust, sand, or extremely high temperatures. Especially in a country where the main source of energy is oil, photovoltaic panel projects are still experimental. The most advance example with renewable energy use and low energy consumption systems is Masdar City. This completely new sustainable city is under construction, but it is a project realised in the United Arab Emirates, not in Saudi Arabia. (Masdar city, 2011) However, Saudi Arabia starts to understand the potential of solar energy as to face the high electricity demands due to important cooling load.

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Appendices

A. Architectural drawings

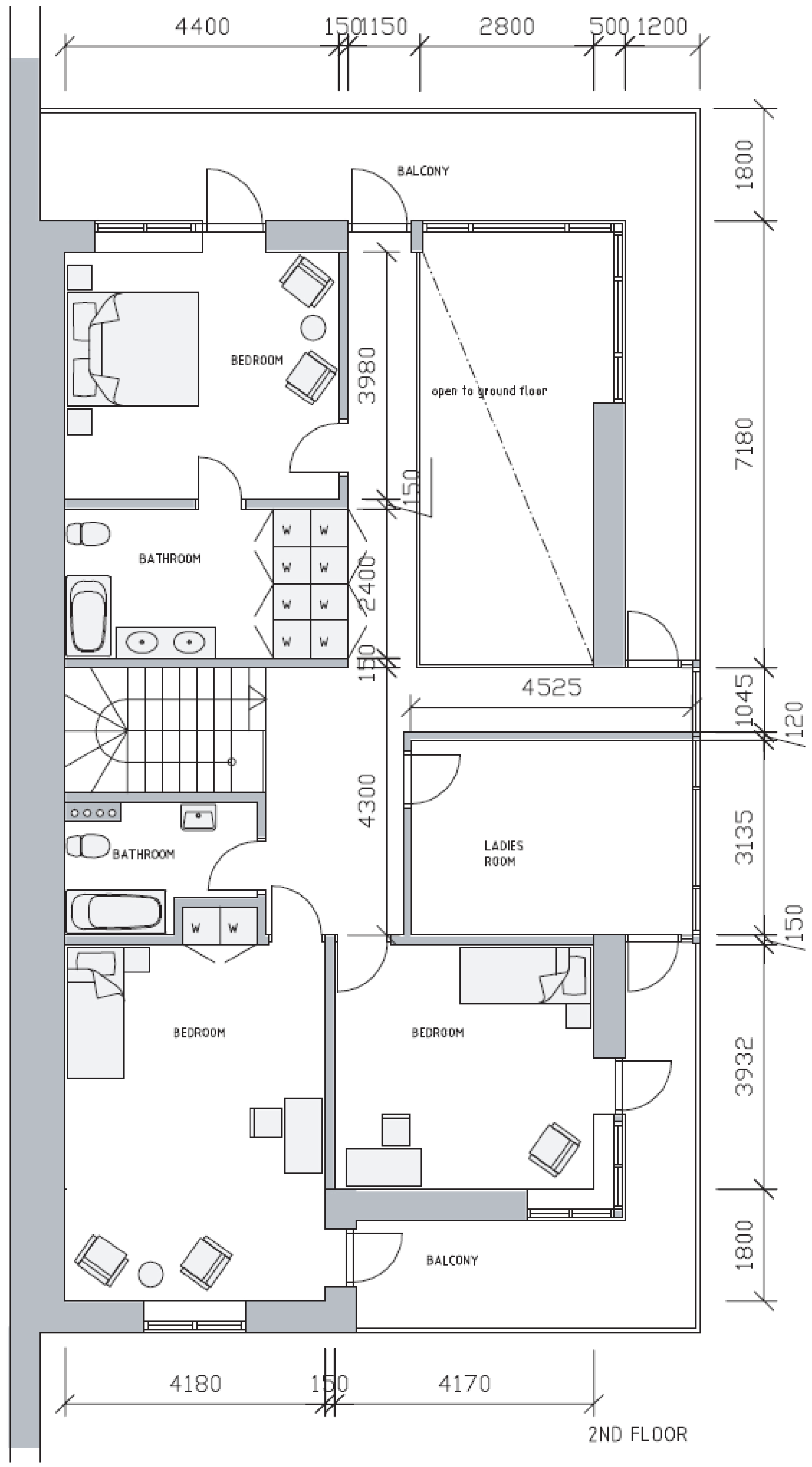


Figure 57 2nd floor of the house by Norconsult

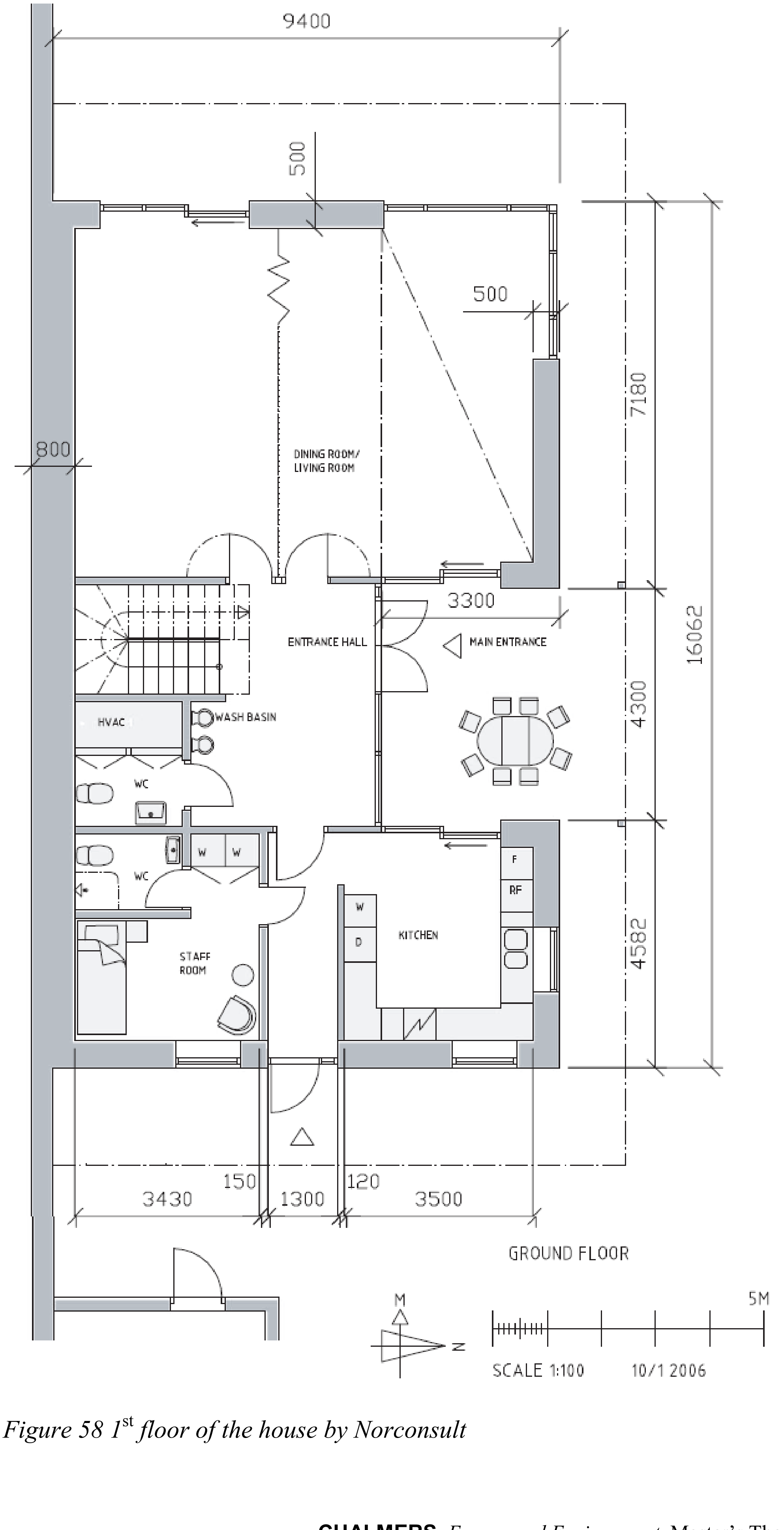


Figure 58 1st floor of the house by Norconsult

B. Simulink programmes

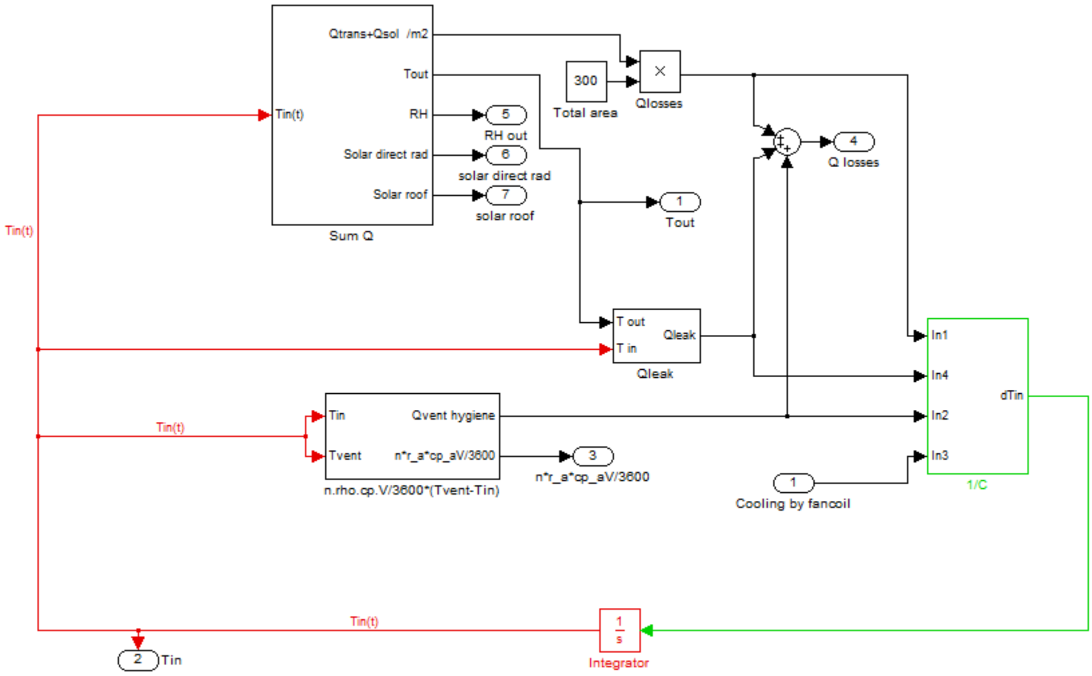


Figure 59 Programme to simulate the indoor temperature

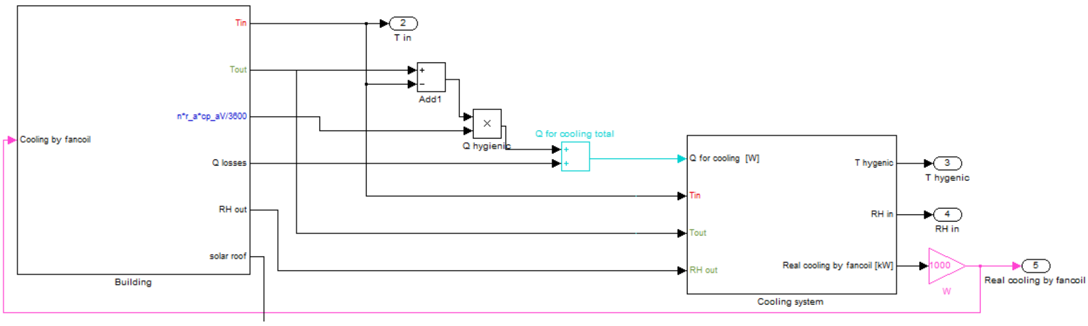


Figure 60 Programme to get the cooling load of the fan coil

C. Moisture properties of light-weight concrete

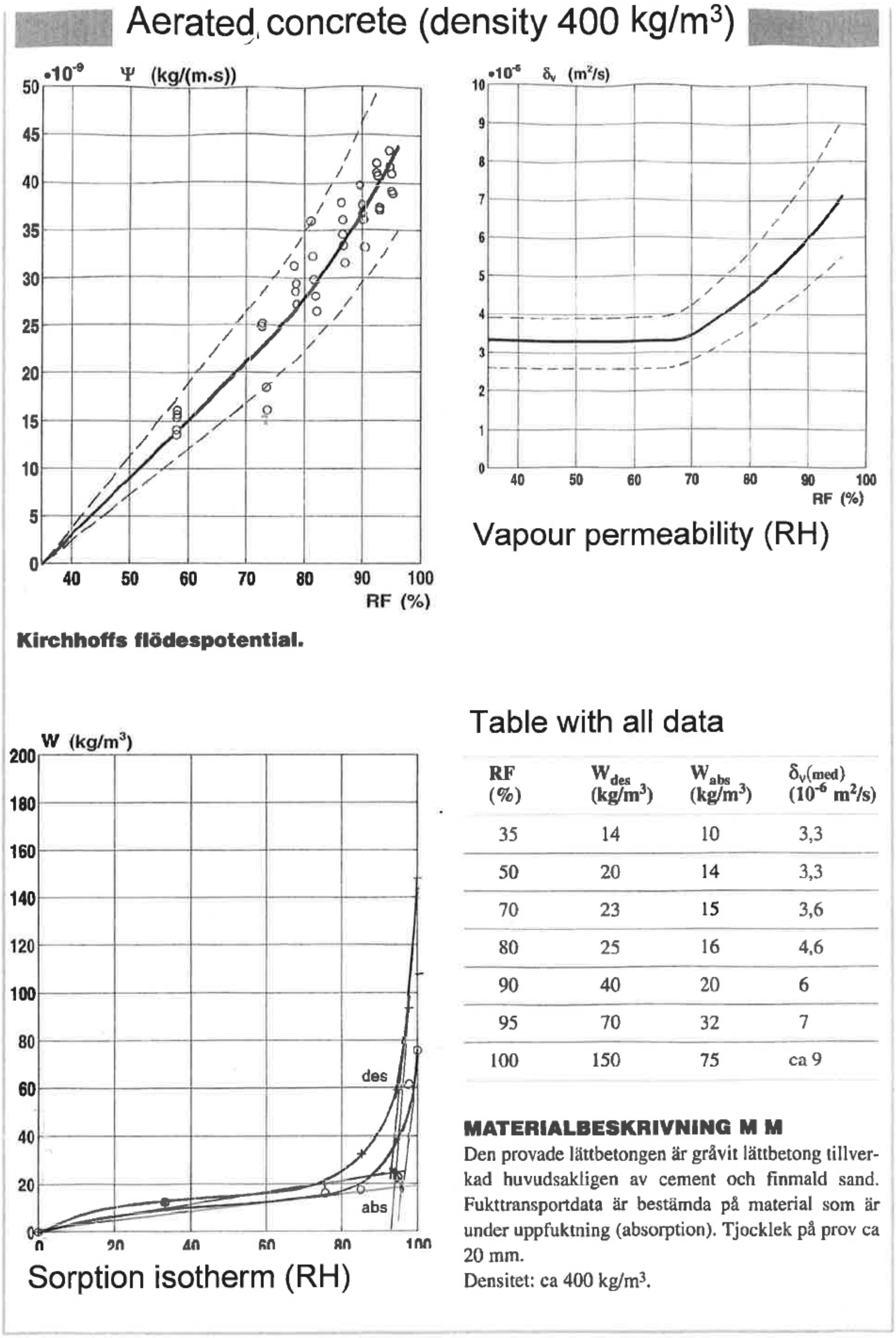


Table with all data

RF (%)	W_{des} (kg/m^3)	W_{abs} (kg/m^3)	δ_v(med) (10^{-6} m^2/s)
35	14	10	3,3
50	20	14	3,3
70	23	15	3,6
80	25	16	4,6
90	40	20	6
95	70	32	7
100	150	75	ca 9

MATERIALBESKRIVNING M M

Den provade lättbetongen är gråvit lättbetong tillverkad huvudsakligen av cement och finmald sand. Fukttransportdata är bestämda på material som är under uppfuktning (absorption). Tjocklek på prov ca 20 mm.

Densitet: ca 400 kg/m³.

Figure 61: Moisture properties of light-weight concrete

D. Jeddah further results

a. Solar radiations on facades

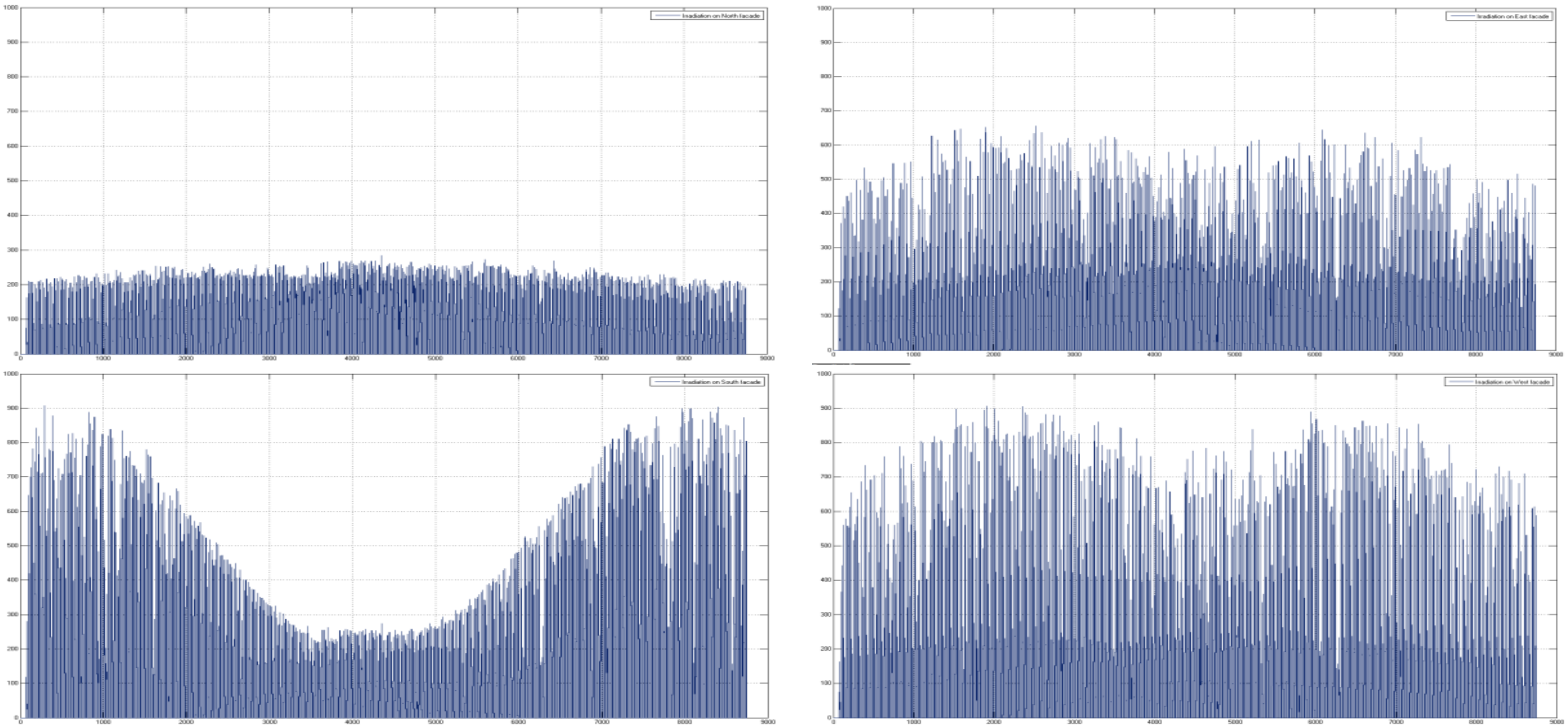


Figure 62 Solar radiations on different facades through the year 50 to 8760h), in Jeddah (from 0 to 1000 W/m²). Up left: North, up right: East, bottom left: South, bottom right: West. (Simulation using the modelling tool Simulink)

b. Temperature and Relative Humidity at different points of the air distribution system for Building 2

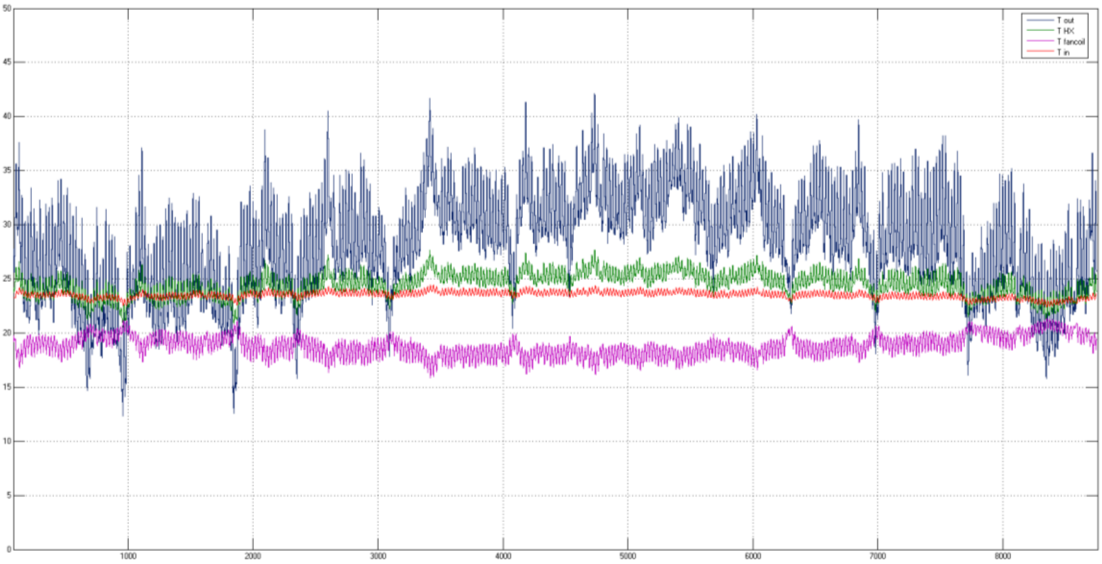


Figure 63 Comparison of the temperatures: outdoors (blue), after the cooling recovery unit (green), at the outlet of the fan coil (purple), and inside (red) for one year (0 to 8760h) (from 0 to 50°C). (Simulation using the modelling tool Simulink)

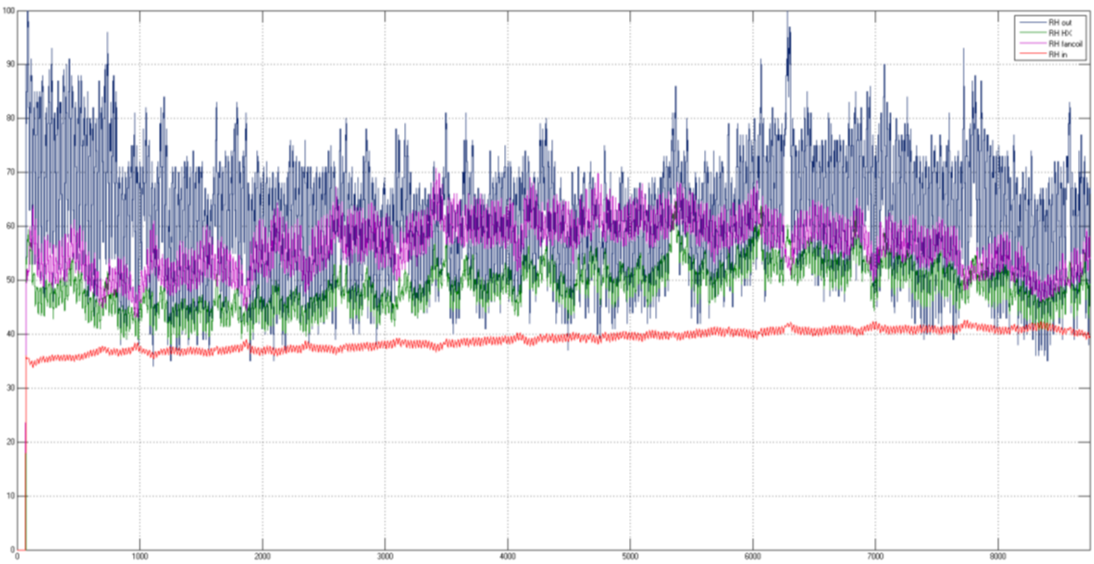


Figure 64 Comparison of the relative humidity 50 to 100%): outdoors (blue), after the cooling recovery unit (green), at the outlet of the fan coil (purple), and inside (red), for one year (0 to 8760h). (Simulation using the modelling tool Simulink)

c. Building 1

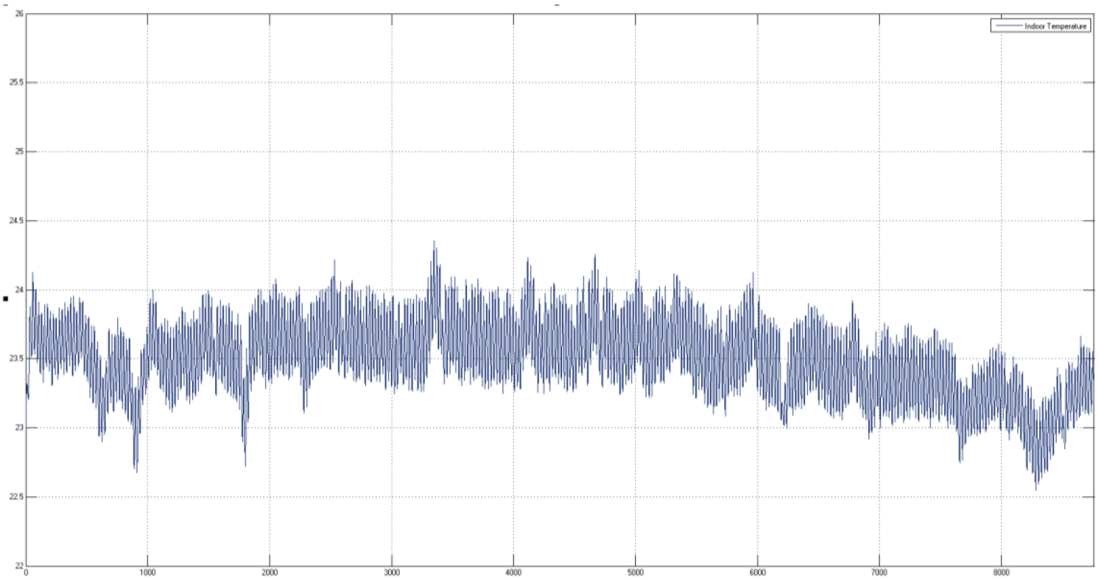


Figure 65 Indoor temperature (22 to 26°C), Building 1, Jeddah, for one year (0 to 8760h). (Simulation using the modelling tool Simulink)

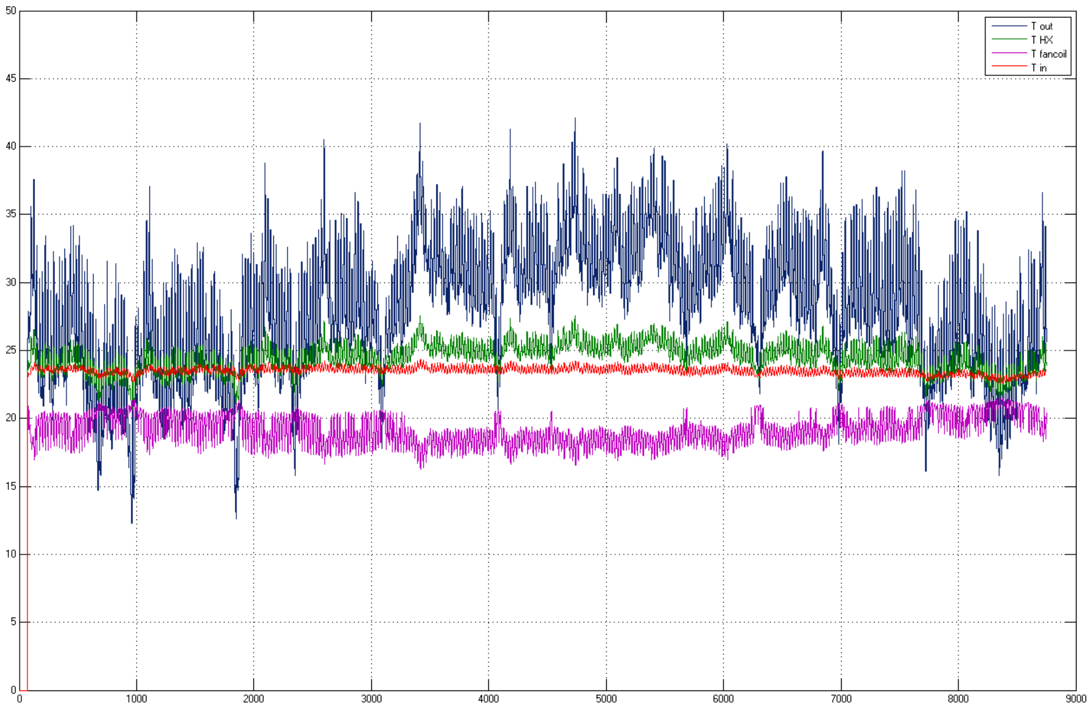


Figure 66 Temperatures (0 to 50°C) for the air cooling system for Building 1 in Jeddah , for one year (0 to 8760h). (Simulation using the modelling tool Simulink)

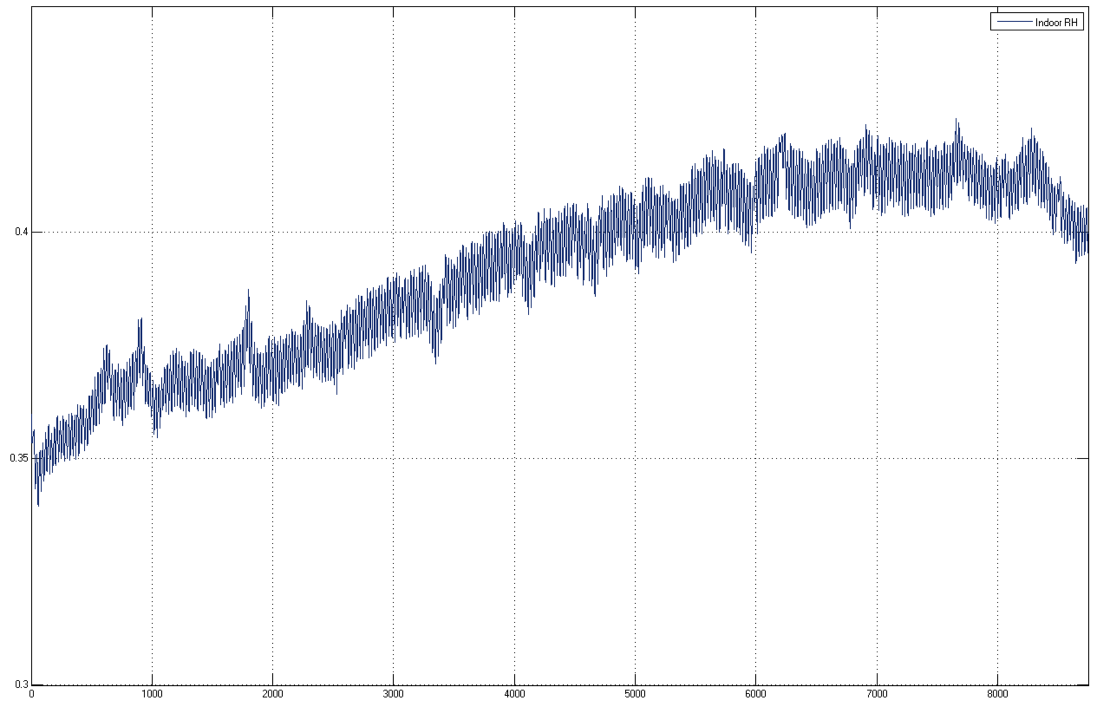


Figure 67 Indoor relative humidity (30 to 45%), Building 1, Jeddah, for one year (0 to 8760h). (Simulation using the modelling tool Simulink)

For the above simulation, the initial condition for the relative humidity was set to 35%, which is in fact not the appropriate initial condition. However, one can see that at the end of the year, the indoor relative humidity diminishes again, down to 40%. Therefore, 40% could be taken as initial condition and the results would be a curve within the range of 40 to 45%.

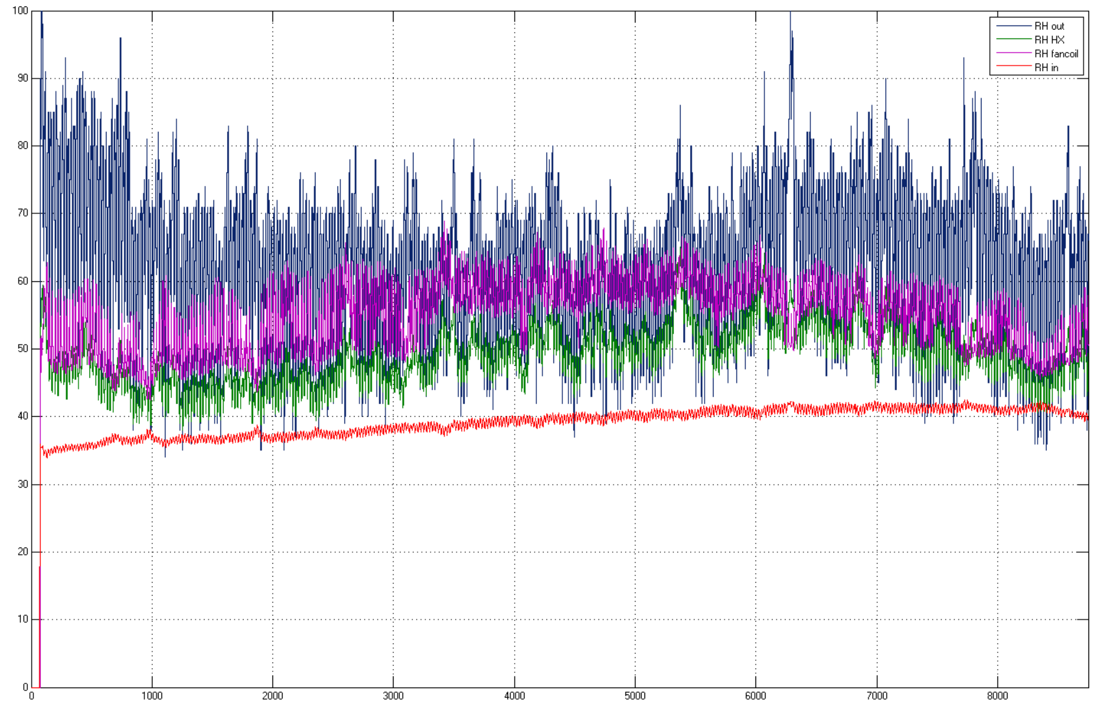


Figure 68 Relative humidity (0 to 100%) for the air cooling system for Building 1 in Jeddah, for one year (0 to 8760h). (Simulation using the modelling tool Simulink)

d. Flows through the system for building 2

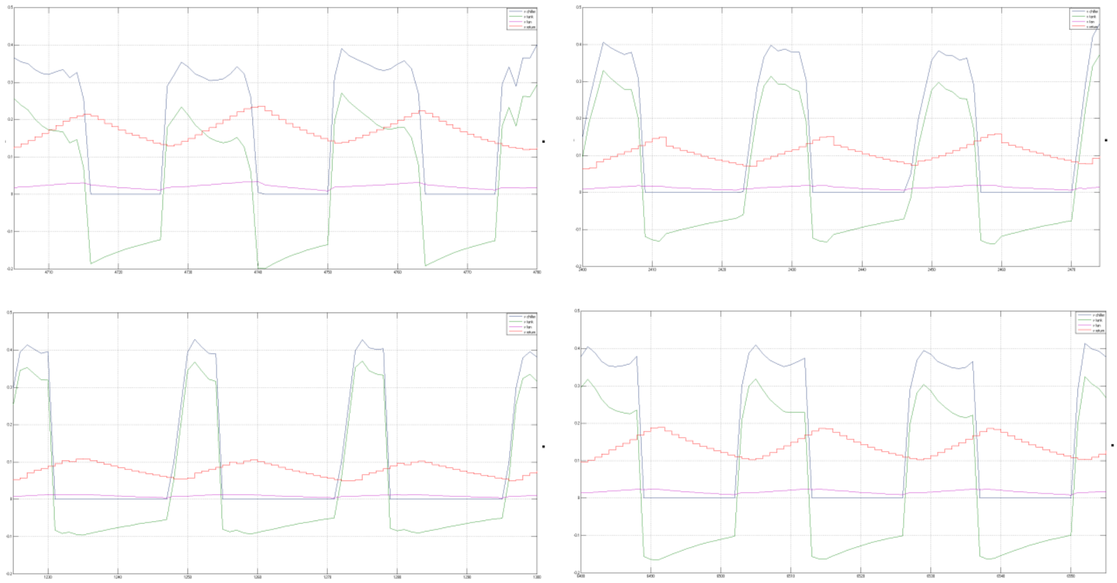


Figure 69 Flows (-0,2 to 0,5 l/s) through the system. Top left: summer, top right: spring, bottom left: winter, bottom right: fall. (Simulation using the modelling tool Simulink)

e. Building 3

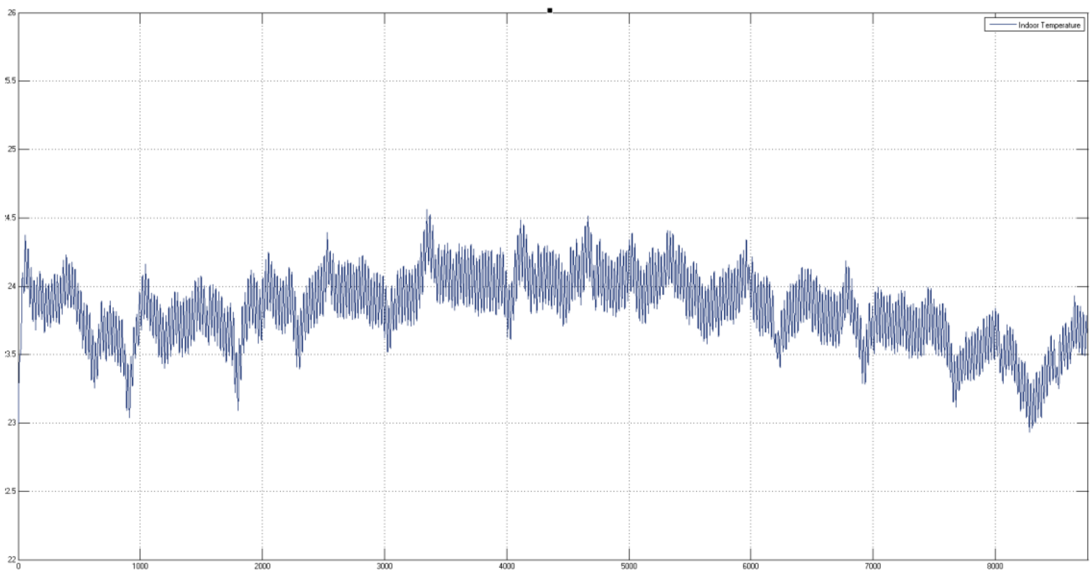


Figure 70 Indoor temperature (22 to 26°C), Building 3, Jeddah, for one year (0 to 8760h). (Simulation using the modelling tool Simulink)

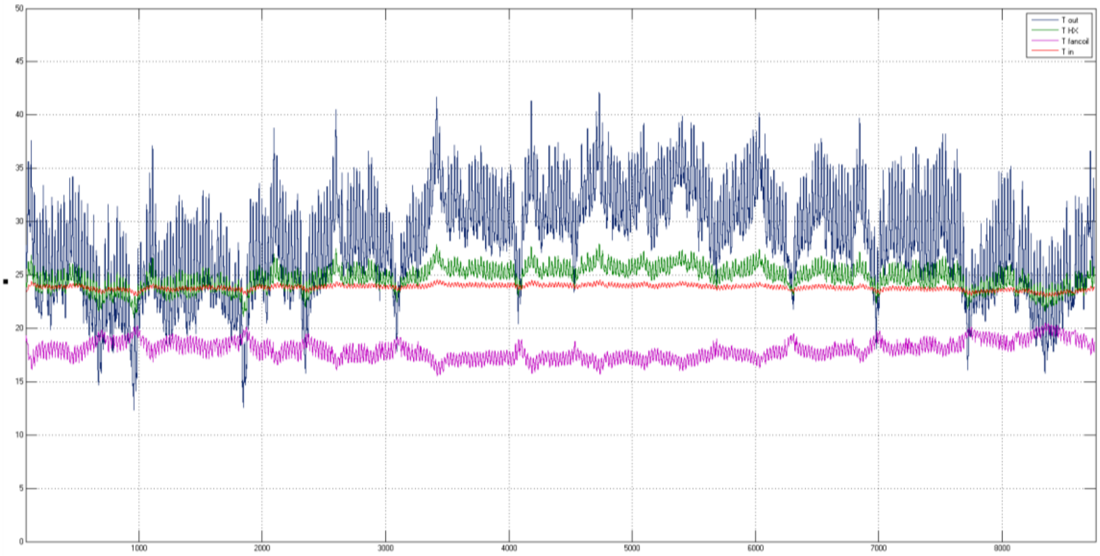


Figure 71 Temperatures (0 to 50°C) for the air cooling system for Building 3 in Jeddah, for one year (0 to 8760h). (Simulation using the modelling tool Simulink)

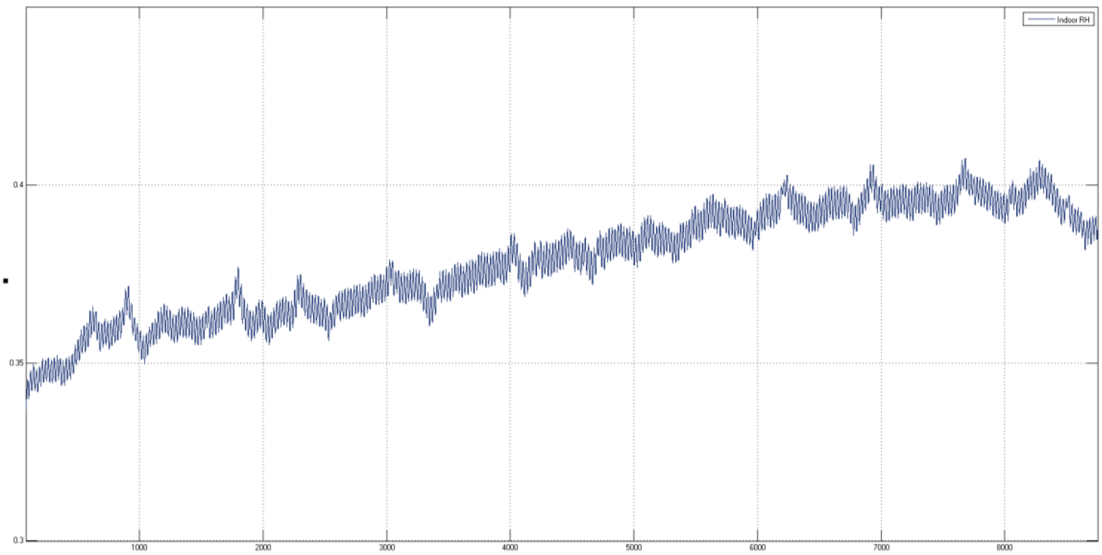


Figure 72 Indoor relative humidity (30 to 45%), Building 3, Jeddah, for one year (0 to 8760h). (Simulation using the modelling tool Simulink)

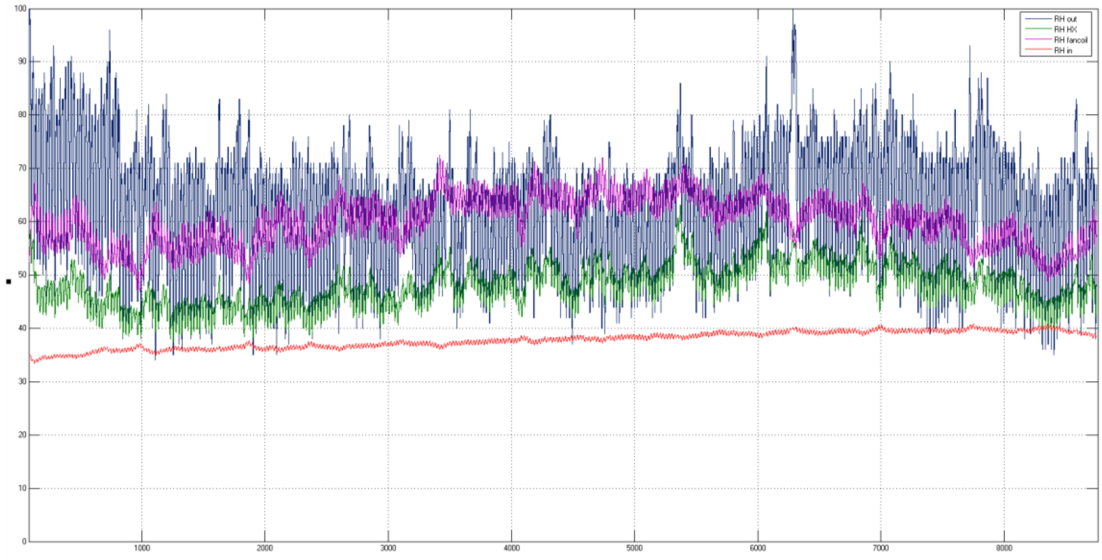


Figure 73 Relative humidity (0 to 100%) for the air cooling system for Building 3 in Jeddah, for one year (0 to 8760h). (Simulation using the modelling tool Simulink)

E. Chiller characteristics

Carrier is one of the world leaders in the HVAC field. They design HVAC building and industrial systems, home comfort, transport refrigeration. Their products are sold all over the world. (Carrier corporation, a UTC company, 2011)

Physical data

30RA		005	007†	009	011	013	015
Nominal cooling capacity*	kW	5.2	6.7/6.5	7.6	9.7	11.2	14.7
Operating weight	kg	71	73	85	108	118	135
Refrigerant type		R-410A	R-410A	R-410A	R-410A	R-410A	R-407C
Compressor		One scroll compressor					
Evaporator		One plate heat exchanger					
Net water volume	l	0.66	0.85	0.94	1.22	1.22	1.50
Max. water-side operating pressure	kPa	300	300	300	300	300	400****
Hydraulic circuit							
Pump		One three-speed pump (sizes 005-013) or single-speed pump (size 015)					
Available pressure**	kPa	44	35/36	49	53	54	150
Water inlet/outlet connections	in	1	1	1	1	1	1
Expansion tank volume	l	1	1	2	2	2	2
Fans		One or two propeller fans					
Number of fans/diameter	mm	1/370	1/370	1/370	2/370	2/370	2/370
No. of blades		4	4	4	4	4	4
Fan speed	r/s	14	18.2	17.2	17.2	17.2	17.2
Sound pressure level* **	dB(A)	36	40	41	42	44	50
Sound power level	dB(A)	64	68	69	70	72	78

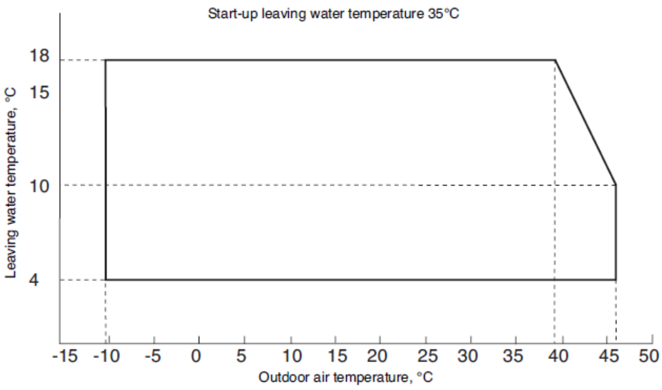
* Based on Eurovent conditions: evaporator entering/leaving water temperature 12°C/7°C, condenser entering air temperature of 35°C.
** At nominal flow and high pump speed
*** Sound pressure is measured at 10 m distance.
**** The safety valve is not installed on size 015; please ensure that it is installed on-site in the hydronic circuit.
† The first value is for single-phase units, the second value is for three-phase units.

Electrical data

30RA		005	007	007	009	011	011	013	015
Power supply	V-ph-Hz	230-1-50	230-1-50	400-3-50	400-3-50	230-1-50	400-3-50	400-3-50	400-3-50
Voltage range	V	198-264	198-264	342-462	342-462	198-264	342-462	342-462	342-462
Nominal power input*	kW	2.07	2.78	2.70	3.05	3.22	3.22	4.57	6.60
Maximum power input**	kW	2.90	3.80	3.60	4.30	4.30	4.40	6.30	8.00
Locked rotor current	A	58	82	35	40	97	48	64	75.5
Max. starting current with soft starter	A	-	-	-	-	-	-	-	45
Full load current	A	15	18	7.5	8	21.5	8.5	11.5	14.5
Water circulating pump (230-1-50)									
Current drawn	A	0.30	0.30	0.30	0.50	0.90	0.90	0.97	1.10
Fan motor (230-1-50)									
Current drawn	A	0.50	0.94	0.94	0.90	1.80	1.80	1.80	1.64
Compressor crankcase heater (230-1-50)									
Current drawn	A	0.11	0.11	0.11	0.11	0.11	0.11	0.11	-

* At standard Eurovent conditions
** Maximum unit power input at maximum operating conditions and worst power supply voltage

Operating limits



Cooling capacities

30RA — Condenser entering air temperature, °C

30RA	LWT	25						30						35						40						45					
	°C	CAP kW	COMP kW	UNIT kW	COOL l/s	COOL kPa	PRES kPa	CAP kW	COMP kW	UNIT kW	COOL l/s	COOL kPa	PRES kPa	CAP kW	COMP kW	UNIT kW	COOL l/s	COOL kPa	PRES kPa	CAP kW	COMP kW	UNIT kW	COOL l/s	COOL kPa	PRES kPa	CAP kW	COMP kW	UNIT kW	COOL l/s	COOL kPa	PRES kPa
005	5	5.02	1.52	1.64	0.24	14	45	4.98	1.71	1.83	0.24	14	45	4.81	1.9	2.02	0.23	13	47	4.48	2.09	2.22	0.21	11	50	4.01	2.29	2.42	0.19	9	53
007-7		6.89	2.07	2.29	0.33	11	33	6.54	2.31	2.53	0.31	10	37	6.2	2.56	2.78	0.3	9	38	5.88	2.81	3.03	0.28	7	42	5.57	3.07	3.29	0.27	7	44
007-9		6.72	1.99	2.19	0.32	10	35	6.37	2.23	2.44	0.3	9	38	6.03	2.48	2.69	0.29	8	40	5.71	2.74	2.94	0.27	7	44	5.41	3	3.21	0.26	6	45
009		8.01	2.25	2.45	0.38	26	46	7.71	2.53	2.73	0.37	25	47	7.27	2.83	3.03	0.35	23	49	6.72	3.14	3.34	0.32	19	52	6.03	3.46	3.66	0.29	16	54
011		9.73	2.24	2.58	0.46	20	52	9.59	2.54	2.89	0.46	19	53	9.24	2.86	3.2	0.44	18	55	8.68	3.18	3.52	0.41	16	58	7.91	3.51	3.86	0.38	14	60
013		12	3.43	3.76	0.57	28	50	11.4	3.8	4.13	0.54	26	53	10.8	4.21	4.53	0.51	22	57	10.1	4.64	4.97	0.48	18	62	9.39	5.1	5.43	0.45	16	65
015		15.7	5.28	5.5	0.74	25	137	15	5.79	6.01	0.7	23	146	14.2	6.34	6.56	0.66	20	156	13.3	6.93	7.15	0.62	18	165	12.3	7.56	7.78	0.57	16	176
005	6	5.16	1.54	1.66	0.25	15	44	5.14	1.73	1.85	0.24	15	44	4.98	1.92	2.04	0.24	14	45	4.67	2.12	2.24	0.22	12	49	4.21	2.31	2.44	0.2	10	52
007-7		7.11	2.09	2.31	0.34	12	31	6.76	2.32	2.54	0.32	10	35	6.42	2.56	2.78	0.31	10	37	6.11	2.81	3.03	0.29	8	40	5.8	3.06	3.28	0.28	7	42
007-9		6.94	2.01	2.21	0.33	11	33	6.59	2.25	2.45	0.31	10	37	6.26	2.49	2.69	0.3	9	38	5.94	2.74	2.94	0.28	7	42	5.64	2.99	3.2	0.27	7	44
009		8.12	2.26	2.46	0.39	28	45	7.85	2.54	2.74	0.37	25	47	7.45	2.84	3.04	0.35	24	48	6.93	3.15	3.35	0.33	21	51	6.29	3.48	3.68	0.3	17	54
011		9.79	2.24	2.58	0.47	20	52	9.72	2.55	2.89	0.46	19	53	9.44	2.87	3.21	0.45	19	54	8.96	3.2	3.54	0.43	17	56	8.26	3.54	3.88	0.39	14	59
013		12.2	3.45	3.78	0.58	29	49	11.7	3.82	4.15	0.55	27	52	11	4.22	4.55	0.52	24	56	10.3	4.65	4.98	0.49	19	61	9.58	5.11	5.44	0.45	17	64
015		16	5.31	5.53	0.75	26	134	15.3	5.82	6.03	0.71	23	143	14.4	6.36	6.58	0.68	21	152	13.5	6.95	7.17	0.63	19	163	12.5	7.57	7.8	0.59	16	173
005	7	5.3	1.56	1.68	0.25	15	44	5.3	1.75	1.87	0.25	15	44	5.15	1.94	2.07	0.24	15	44	4.85	2.14	2.26	0.23	13	47	4.42	2.33	2.46	0.21	11	50
007-7		7.33	2.11	2.33	0.35	13	29	6.98	2.34	2.56	0.33	11	33	6.65	2.57	2.79	0.32	10	35	6.33	2.81	3.03	0.3	9	38	6.03	3.05	3.27	0.29	8	40
007-9		7.16	2.03	2.24	0.34	12	31	6.81	2.26	2.46	0.32	11	33	6.48	2.49	2.7	0.31	10	37	6.17	2.74	2.94	0.29	8	40	5.87	2.98	3.19	0.28	7	42
009		8.22	2.27	2.46	0.39	28	45	7.99	2.55	2.75	0.38	26	46	7.63	2.85	3.05	0.36	24	48	7.15	3.16	3.36	0.34	22	50	6.54	3.49	3.69	0.31	18	53
011		9.85	2.23	2.57	0.47	20	52	9.85	2.55	2.89	0.47	20	52	9.65	2.88	3.22	0.46	19	53	9.23	3.22	3.56	0.44	18	55	8.61	3.56	3.91	0.41	16	58
013		12.5	3.46	3.8	0.59	32	46	11.9	3.84	4.17	0.56	28	50	11.2	4.24	4.57	0.53	25	54	10.5	4.67	5	0.5	20	60	9.77	5.13	5.46	0.46	17	63
015		16.3	5.33	5.56	0.77	27	130	15.5	5.84	6.06	0.73	24	139	14.7	6.38	6.6	0.69	22	149	13.8	6.97	7.19	0.64	19	160	12.8	7.59	7.81	0.6	17	171
005	8	5.44	1.58	1.7	0.26	17	42	5.45	1.77	1.89	0.26	17	42	5.32	1.97	2.09	0.25	15	44	5.04	2.16	2.28	0.24	14	45	4.62	2.36	2.48	0.22	12	49
007-7		7.55	2.13	2.35	0.36	14	27	7.21	2.35	2.57	0.34	12	31	6.88	2.57	2.79	0.33	11	33	6.56	2.8	3.02	0.31	10	37	6.26	3.04	3.26	0.3	9	38
007-9		7.38	2.05	2.26	0.35	13	29	7.04	2.27	2.48	0.34	12	31	6.71	2.5	2.7	0.32	10	35	6.4	2.73	2.94	0.3	10	37	6.1	2.97	3.17	0.29	8	40
009		8.32	2.27	2.47	0.4	29	44	8.13	2.56	2.76	0.39	28	45	7.81	2.86	3.06	0.37	25	47	7.36	3.18	3.38	0.35	23	49	6.79	3.51	3.71	0.32	19	52
011		9.92	2.22	2.57	0.47	20	52	9.99	2.55	2.9	0.47	21	51	9.85	2.89	3.23	0.47	20	52	9.51	3.24	3.58	0.45	19	54	8.96	3.59	3.93	0.43	17	56
013		12.7	3.48	3.82	0.6	33	44	12.1	3.85	4.19	0.58	29	49	11.4	4.25	4.58	0.54	26	53	10.7	4.69	5.01	0.51	21	58	9.95	5.15	5.47	0.47	17	63
015		16.6	5.36	5.59	0.78	28	126	15.8	5.86	6.09	0.74	25	136	15	6.41	6.62	0.7	23	146	14	6.99	7.21	0.66	20	157	13	7.61	7.83	0.61	17	168
005	9	5.58	1.6	1.73	0.27	18	40	5.61	1.79	1.92	0.27	18	40	5.49	1.99	2.11	0.26	17	42	5.23	2.18	2.3	0.25	15	44	4.82	2.38	2.5	0.23	13	47
007-7		7.77	2.15	2.38	0.37	15	24	7.43	2.36	2.58	0.35	13	29	7.1	2.58	2.8	0.34	12	31	6.79	2.8	3.02	0.32	10	35	6.5	3.03	3.25	0.31	10	37
007-9		7.6	2.07	2.28	0.36	14	27	7.26	2.28	2.49	0.35	13	29	6.93	2.5	2.71	0.33	11	33	6.62	2.73	2.93	0.32	10	35	6.33	2.96	3.16	0.3	9	38
009		8.43	2.28	2.47	0.4	29	44	8.27	2.57	2.76	0.39	29	44	7.99	2.87	3.07	0.38	26	46	7.58	3.19	3.39	0.36	24	48	7.05	3.53	3.72	0.34	22	50
011		9.98	2.22	2.56	0.47	21	51	10.1	2.56	2.9	0.48	21	51	10.1	2.9	3.24	0.48	21	51	9.78	3.25	3.6	0.46	20	52	9.31	3.62	3.96	0.44	19	54
013		12.9	3.5	3.83	0.62	35	43	12.3	3.87	4.2	0.59	31	47	11.7	4.27	4.6	0.55	27	52	10.9	4.7	5.03	0.52	22	57	10.1	5.16	5.49	0.48	18	62
015		16.9	5.39	5.62	0.79	29	122	16.1	5.89	6.11	0.76	26	132	15.3	6.43	6.65	0.71	23	143	14.3	7.01	7.23	0.67	21	154	13.3	7.63	7.85	0.62	18	166
005	10	5.72	1.62	1.75	0.27	18	40	5.76	1.82	1.94	0.27	20	38	5.66	2.01	2.13	0.27	18	40	5.42	2.2	2.32	0.26	17	42	5.02	2.4	2.52	0.24	14	45
007-7		7.99	2.17	2.4	0.38	16	22	7.65	2.37	2.6	0.36	14	27	7.33	2.58	2.81	0.35	13	29	7.02	2.8	3.02	0.33	11	33	6.73	3.02	3.24	0.32	10	35
007-9		7.82	2.09	2.3	0.37	15	24	7.48	2.3	2.51	0.36	14	27	7.16	2.51	2.72	0.34	12	31	6.85	2.73	2.93	0.33	11	33	6.56	2.95	3.15	0.31	10	37
009		8.53	2.28	2.48	0.41	30	43	8.41	2.58	2.77	0.4	29	44	8.17	2.88	3.08	0.39	28	45	7.8	3.2	3.4	0.37	25	47	7.3	3.54	3.74	0.35	23	49
011		10	2.21	2.56	0.48	21	51	10.3	2.56	2.9	0.49	22	50	10.3	2.91	3.25	0.49	22	50	10.1	3.27	3.62	0.48	21	51	9.66	3.64	3.99	0.46	19	53
013		13.2	3.51	3.85	0.63	36	41	12.6	3.89	4.22	0.6	32	46	11.9	4.29	4.62	0.56	28	50	11.1	4.72	5.05	0.53	24	56	10.3	5.18	5.5	0.49	19	61
015		17.2	5.41	5.65	0.81	30	118	16.4	5.91	6.14	0.77	27	128	15.5	6.45	6.67	0.73	24	139	14.6	7.03	7.25	0.68	21	151	13.5	7.65	7.87	0.63	19	163

Legend:

LWT	Leaving water temperature
CAP kW	Net cooling capacity = gross cooling capacity plus the capacity corresponding to the available pressure (flow x pressure/0.3).
Comp kW	Compressor power input
Unit kW	Unit power input (compressors, fans, control circuit and pumps) minus the capacity corresponding to the available pressure (flow x pressure/0.3).
Cool l/s	Evaporator water flow rate
Cool kPa	Evaporator and hydronic module pressure drop
Pres kPa	Available pressure at the unit outlet (unit with single-pump hydronic module)

▭ Capacity based on standard EUROVENT conditions

The published performances are in accordance with EUROVENT tolerances:
- 5% for heating and cooling capacities
+ 5% for power input
+15% for the pressure drop

Full load correction factors for Eurovent laboratory test:

Net cooling capacity	1.000
Energy efficiency ratio	1.000
Evaporator pressure drop	1.000

Application data:

Refrigerant: R-410A (sizes 005-013), R-407C (size 015)
Evaporator temperature rise: 5 K
Evaporator fluid: chilled water
Fouling factor: 0.000044 m^2 K/W

System water flow rate/volume

30RA		005	007	009	011	013	015
Nominal water flow rate	l/s	0.25	0.31	0.37	0.46	0.54	0.70
System water volume	l						
Minimum		17	22	27	32	41	53
Maximum		50	50	100	100	100	85

Sound power levels (dB)

30RA	125 Hz	250 Hz	500 Hz	1000 Hz	2000 Hz	4000 Hz	8000 Hz	dB(A)
005	69	64	64	59	52	45	39	64
007	72	67	66	63	58	53	47	68
009	71	71	67	63	60	54	48	69
011	71	70	68	66	61	56	50	70
013	72	71	71	66	62	57	52	72
015	72.5	71	68	68.5	62.5	58	52	78

F. Riyadh results

All the following graphs are shown for one year simulations: from 0 to 8760h.

a. General results

Figure 74 Transmission losses (from -15 to 15 W/m² floor area) in Riyadh (simulation using the modelling tool Simulink)

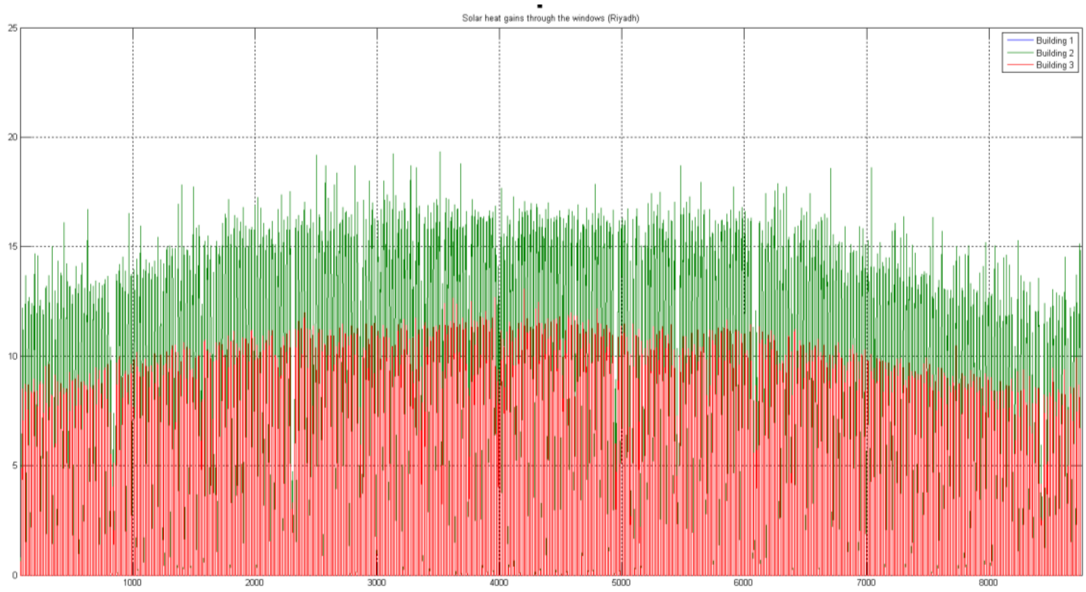


Figure 75 Solar heat gains (from 0 to 25 W/m² floor area) in Riyadh (simulation using the modelling tool Simulink)

b. Building 1

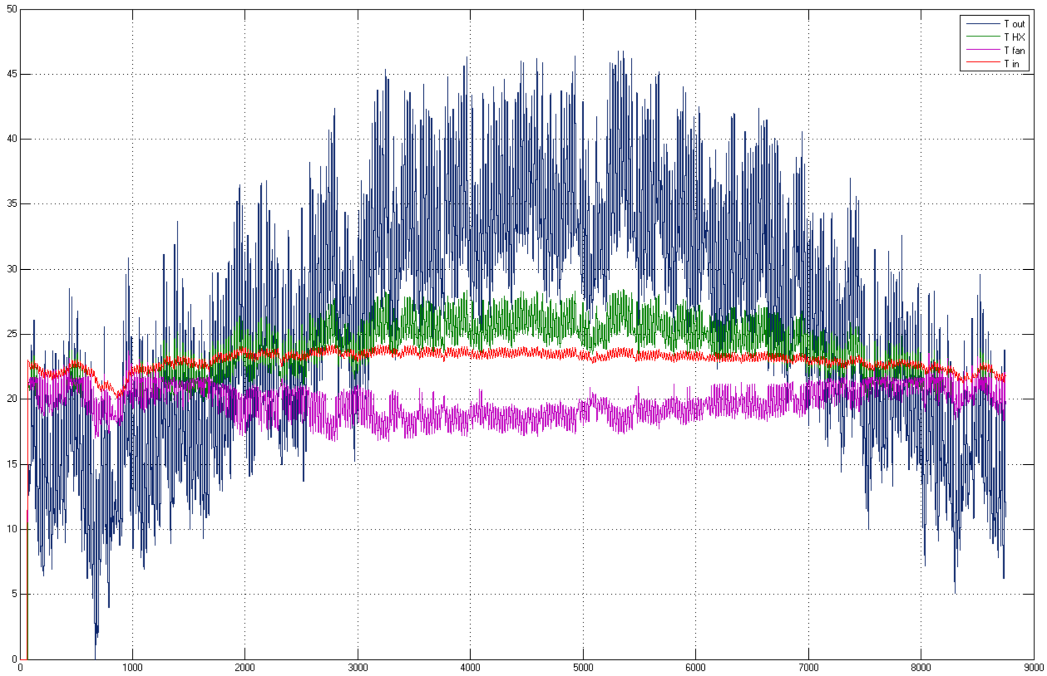


Figure 76 Temperatures (0 to 50°C) for the air cooling system for Building 1 in Riyadh (simulation using the modelling tool Simulink)

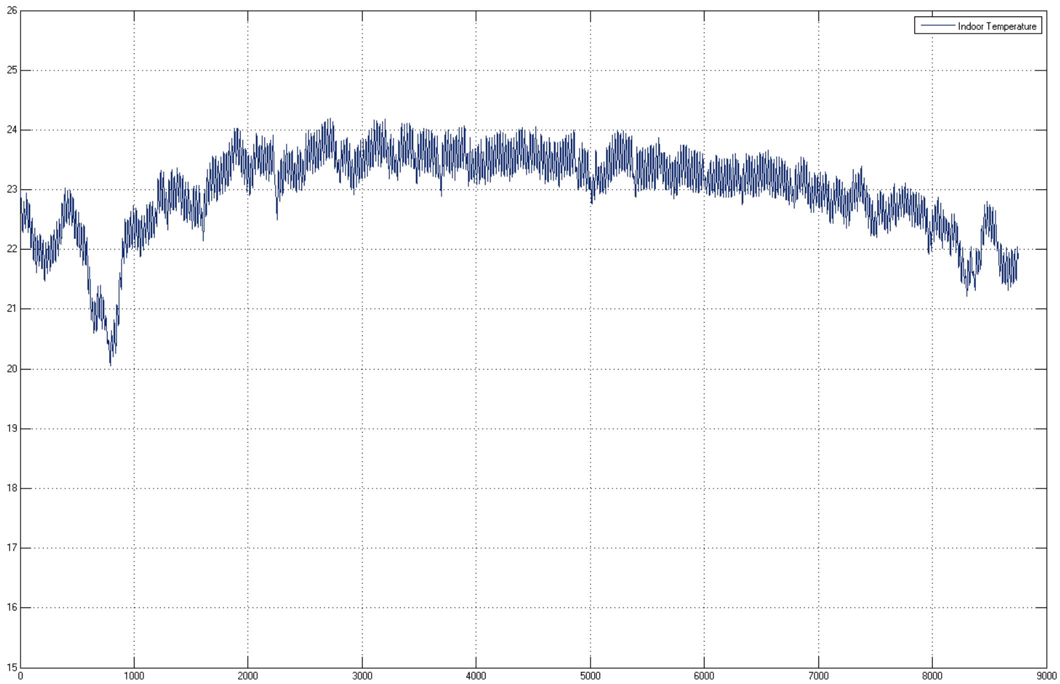


Figure 77 Indoor temperature (15 to 26°C), Building 1, Riyadh (simulation using the modelling tool Simulink)

In winter, one can see that temperature goes lower than the specified range defined in the indoor climate chapter. However, 20°C is still an acceptable temperature indoors. Therefore, there might be no need for a heating system in winter.

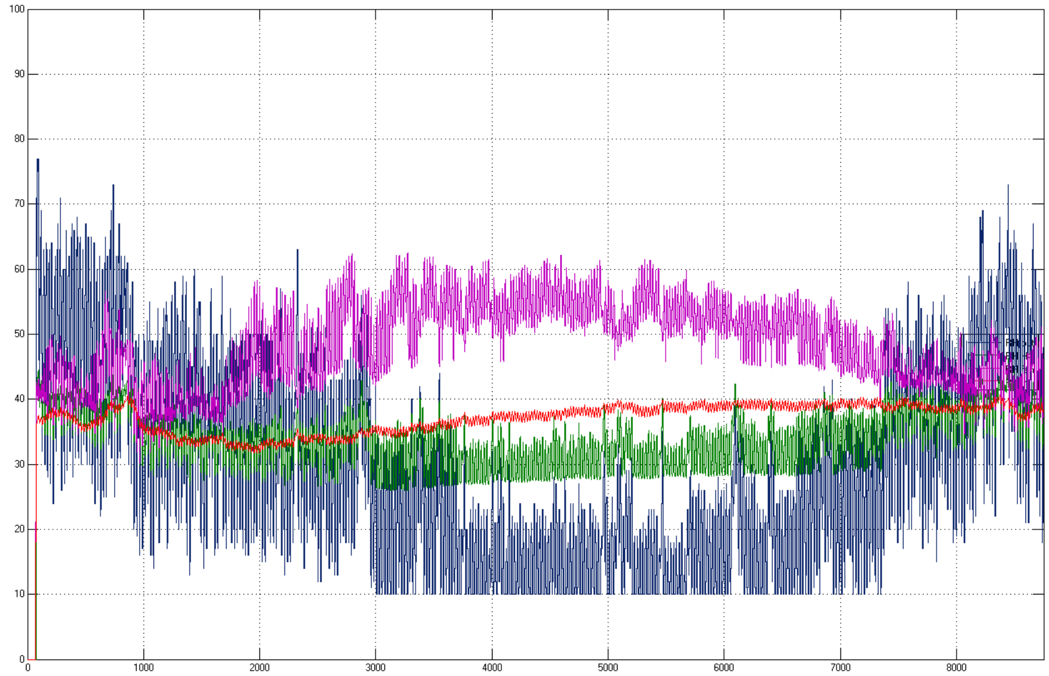


Figure 78 Relative humidity (0 to 100%) for the air cooling system for Building 1 in Riyadh (simulation using the modelling tool Simulink)

c. Building 2

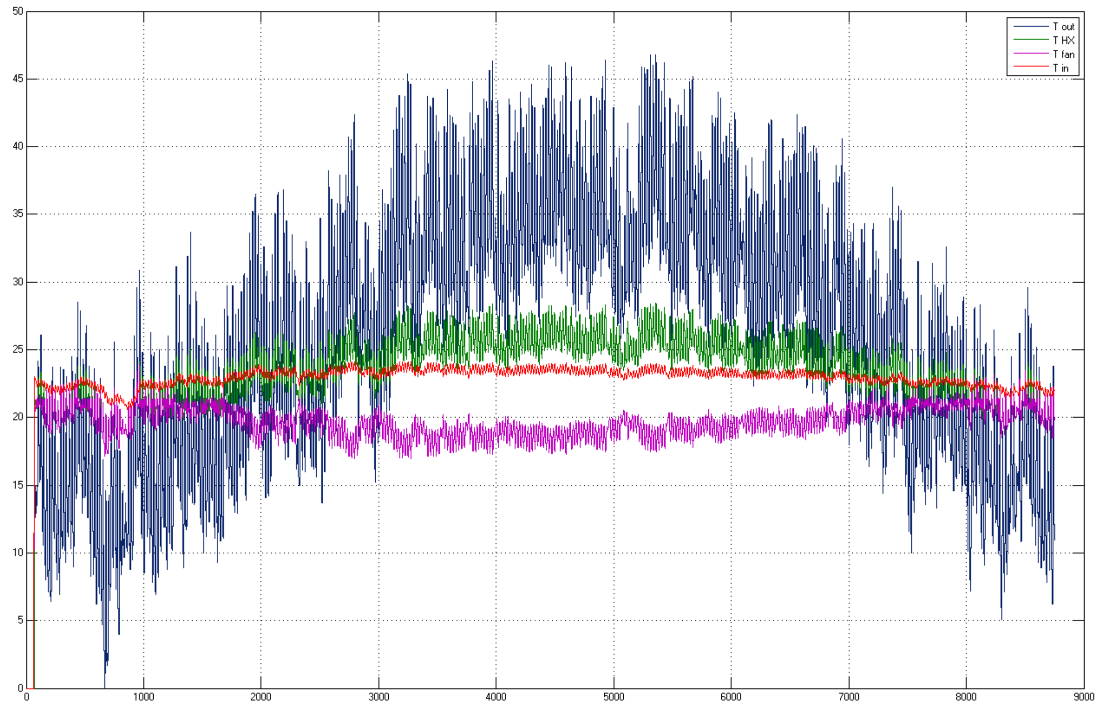


Figure 79 Temperatures (0 to 50°C) for the air cooling system for Building 2 in Riyadh (simulation using the modelling tool Simulink)

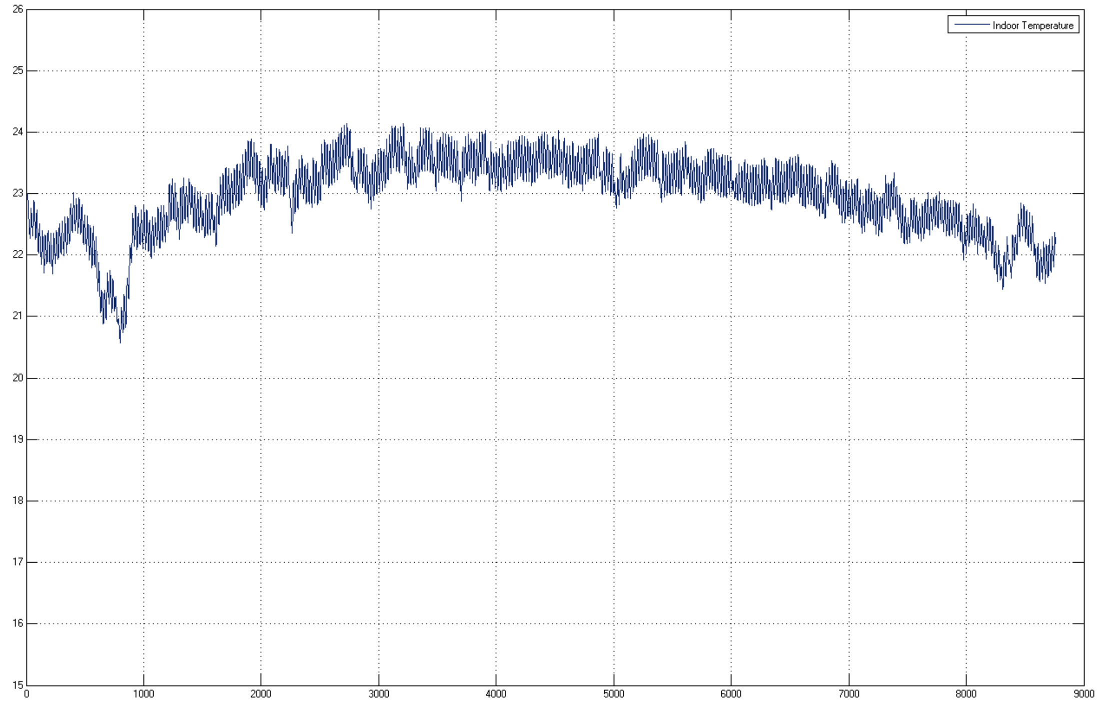


Figure 80 Indoor temperature (15 to 26°C), Building 2, Riyadh (simulation using the modelling tool Simulink)

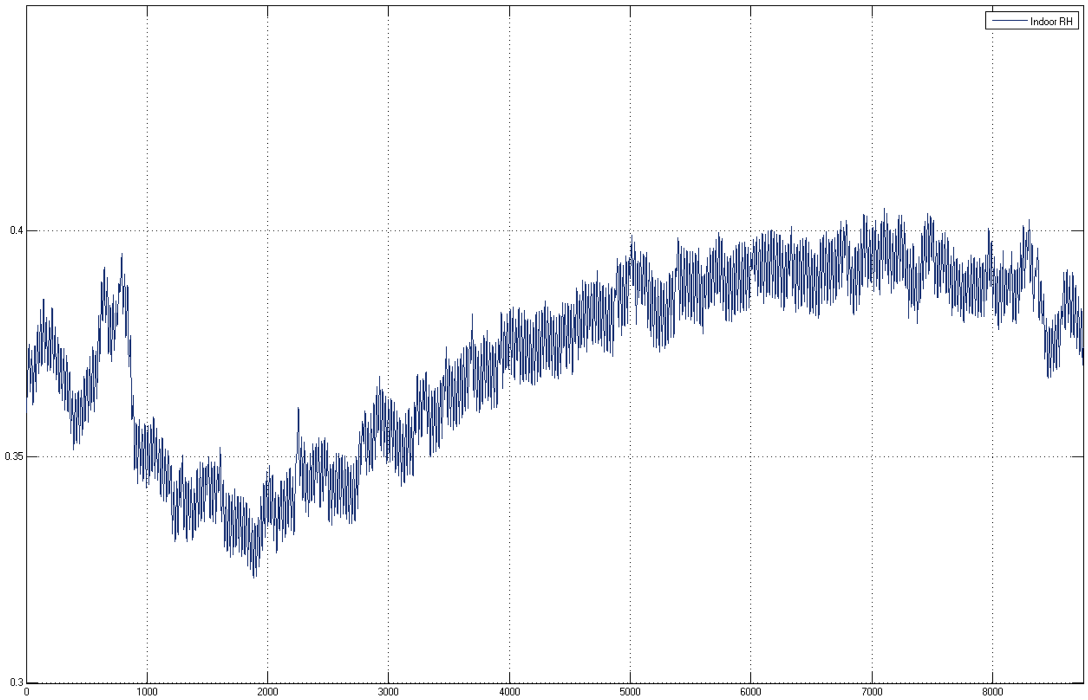


Figure 81 Indoor relative humidity (30 to 45%), Building 2, Riyadh (simulation using the modelling tool Simulink)

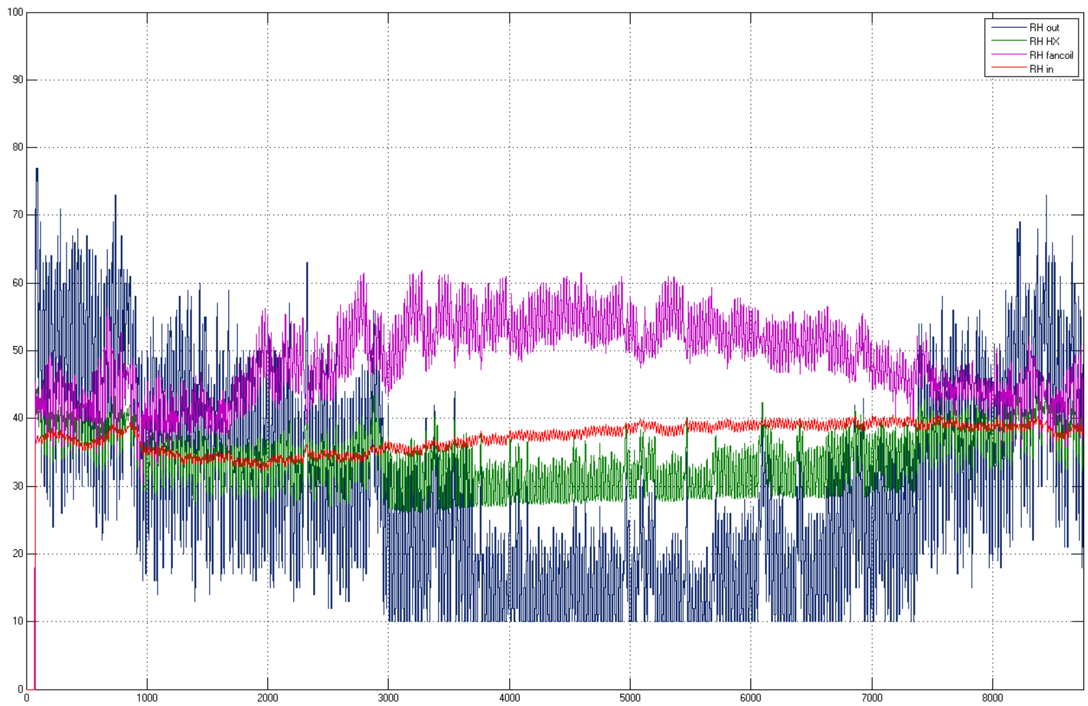


Figure 82 Relative humidity (0 to 100%) for the air cooling system for Building 2 in Riyadh (simulation using the modelling tool Simulink)

d. Building 3

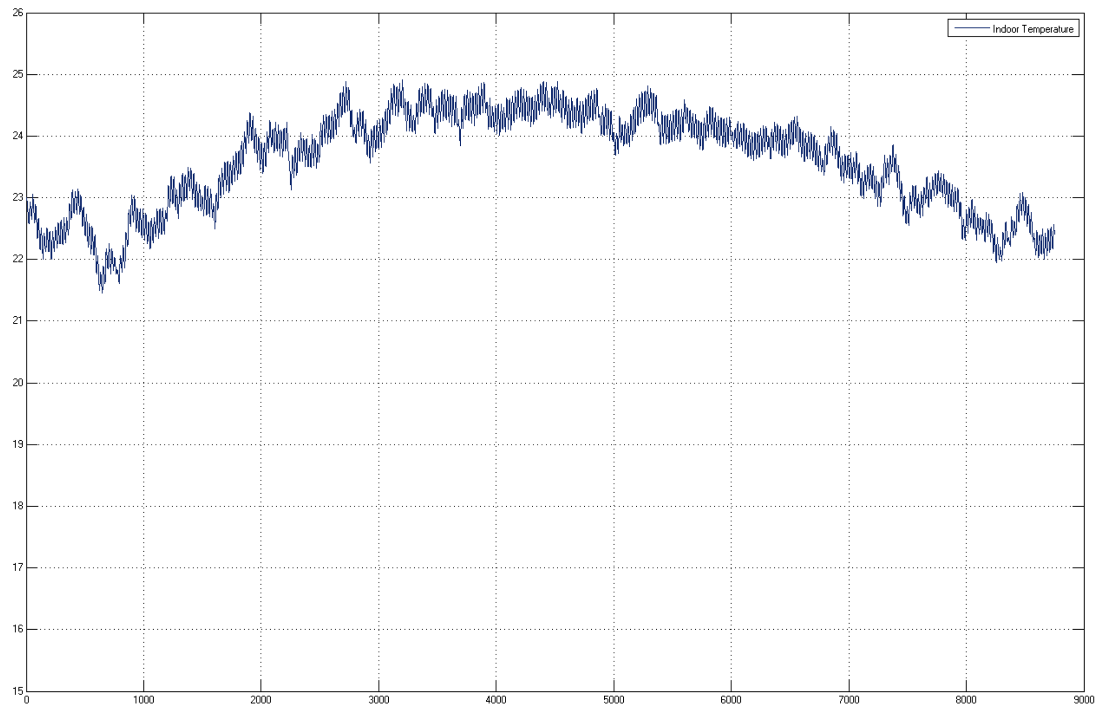


Figure 83 Indoor temperature (15 to 26°C), Building 3, Riyadh (simulation using the modelling tool Simulink)

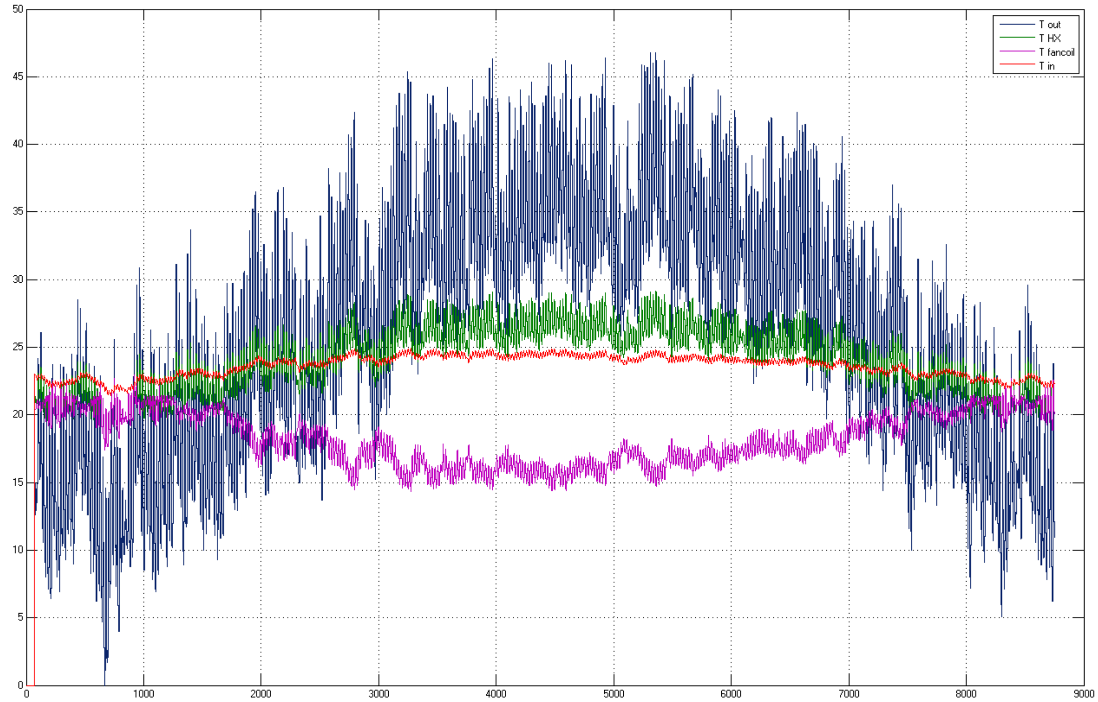


Figure 84 Temperatures (0 to 50°C) for the air cooling system for Building 3 in Riyadh (simulation using the modelling tool Simulink)

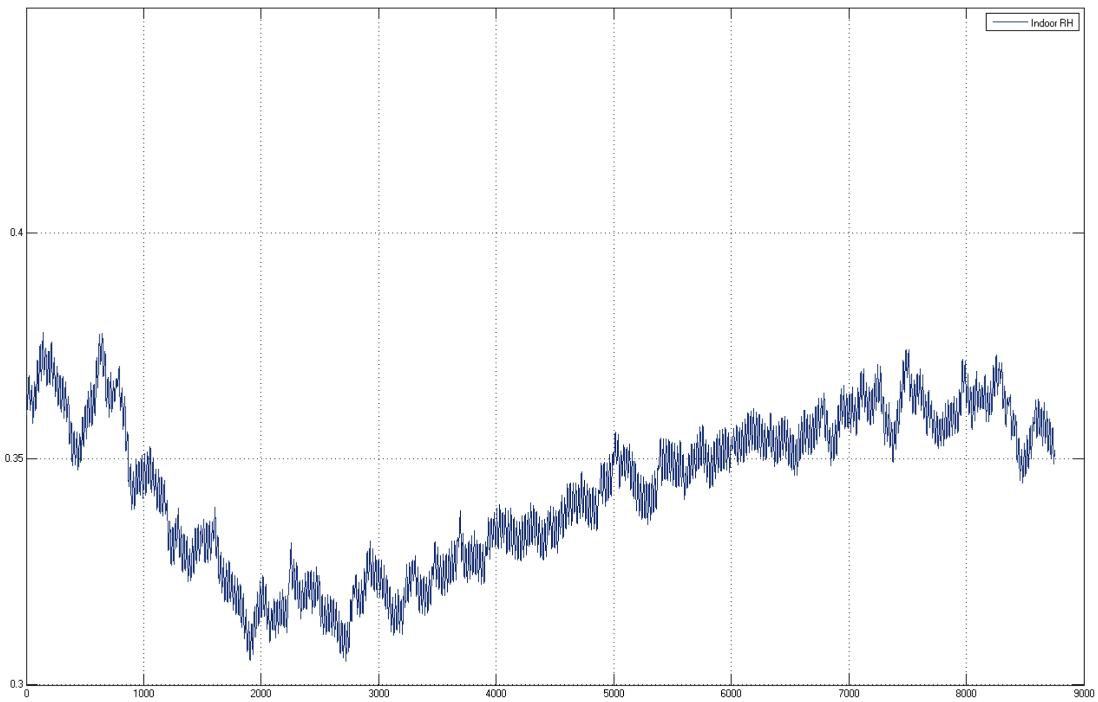


Figure 85 Indoor relative humidity (30 to 45%), Building 3, Riyadh (simulation using the modelling tool Simulink)

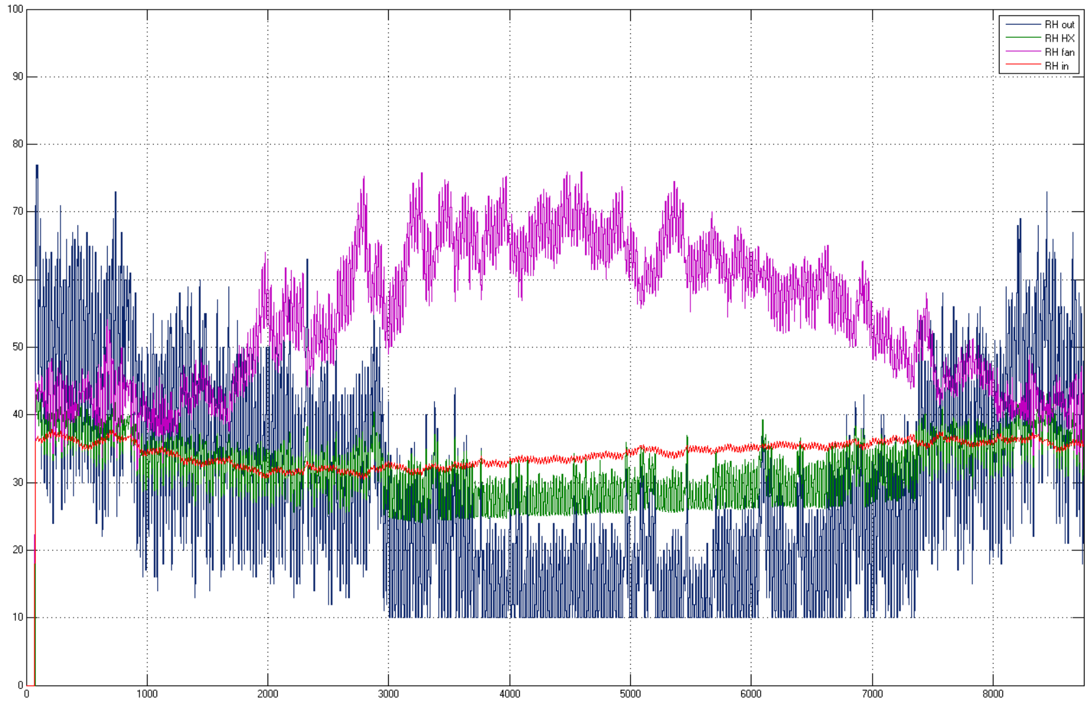


Figure 86 Relative humidity (0 to 100%) for the air cooling system for Building 3 in Riyadh (simulation using the modelling tool Simulink)

G. Absorption cooling

In this part will be described the absorption cooling, as another possible strategy of cooling. The absorption cooling machines should replace the chiller system of the design studied.

The absorption cooling machines are the most common chillers using thermal energy as input. They were also the first cooling machines invented, before the vapour-compression chillers. The reason why absorption cooling system could be interesting comes from the possibility to use heat from solar radiations to the generator of this system.

a. Working principle

In the absorption cooler, there are four components: the generator, the condenser, the evaporator, the absorber. The refrigerant (F) flows between these parts. The generator and the absorber act as the "compressor" in this thermally driven chiller. The principle is that, at low pressure, the refrigerant evaporates at a low temperature.

For a single-effect absorption chiller, there are 2 different pressure levels and 3 temperatures.

The steps of the absorption cycle are, as seen on Figure 87:

- In the evaporator EV, a low-pressure p_F is maintained. The liquid refrigerant evaporates by absorbing heat from the medium to be cooled (chilled water) at a low temperature θ_F. This results in the cooling effect of the medium by Φ_F.
- The refrigerant vapour flows to the absorber AB (step 1), where it is absorbed by a concentrated solution. This mixing produces heat, as well as there is latent heat from the condensation of the mixed solution (strong solution SR). At the same time, a cooling fluid from a cooling tower flows through the absorber in a tube bundle, removing this heat Φ_{AB}.
- The diluted solution is pumped by PS to the generator GE (step a-b).
- The generator is provided with heat (Φ_M) from the solar collector. This makes the solution boil: the refrigerant evaporates and the weak solution remains liquid. This is made possible by correctly selecting the absorbent and the refrigerant depending on their boiling temperatures and their ability to dilute one another.
- After the separation, the concentrated solution (weak solution SP) flows back to the absorber (step c-d). A heat exchanger EI is put between the weak and the strong solution. This enables the strong solution to be pre-heated, and thus reduces the need of energy input to the generator.
- The refrigerant vapour flows from the generator to the condenser CD (step 2). There, it condensates by rejecting heat Φ_{CD} to the cooling fluid flowing towards the cooling tower.
- The liquid refrigerant flows back to the evaporator through an expansion valve DT (step 3-4), lowering the pressure from p_C to p_F, to repeat the whole cycle.

(Duminil, 2008)

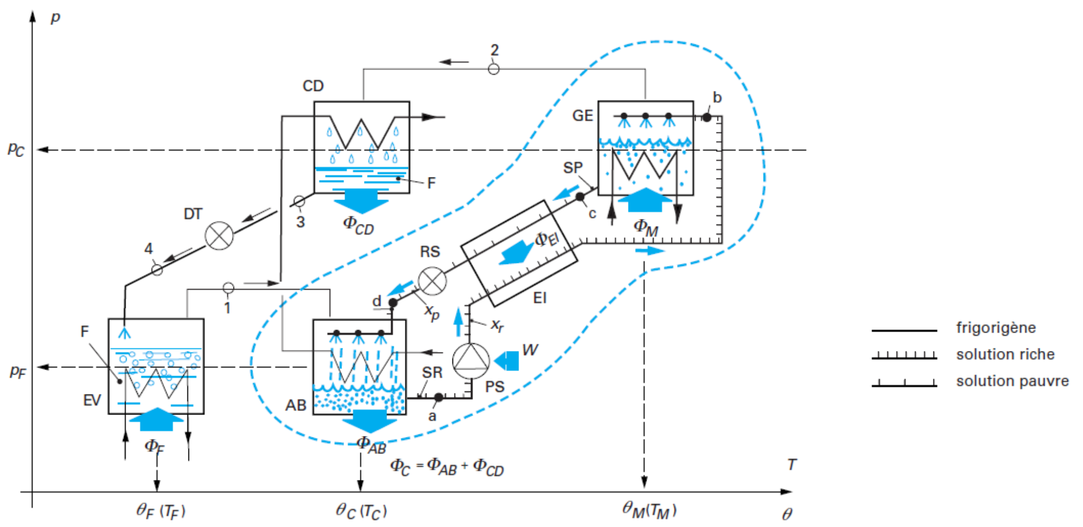


Figure 87 Absorption cooling machine (Duminil, 2008).

The dotted line comprehends the absorber AB and the generator GE, as to replace the compressor in the vapour-compressor chiller. PS is the pump for the strong solution to flow, EI the heat exchanger between the weak and the strong solution, RS the expanding valve of the weak solution.

When using absorption chillers for air conditioning in Europe, the temperature of the low temperature heat source should be higher than 5°C. Usually, the refrigerant/absorbent couple water/lithium-bromide (LiBr) is used, as it permits using lower driving temperatures (from 76°C to 99°C according to (Guyer, 2010)) than for ammonia systems (95°-120°C). At these temperatures, it is possible to use flat plate collectors to reduce a little bit the investment in the solar components, although concentrating collector operate at higher efficiencies. In both cases, solar components still constitute the main source of expense among the whole system.

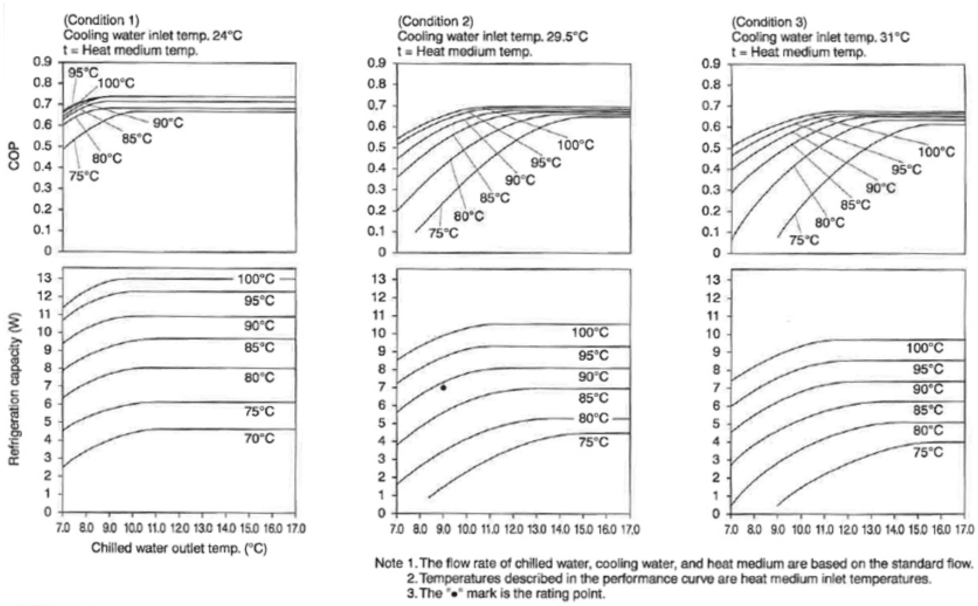


Figure 88 Capacity and COP of a single-effect hot-water-driven lithium bromide chiller. (IEA Heat Pump Programme, 2000)

b. Advantages

Compared to the vapour-compression system, the absorption cooling can use water as refrigerant, which has no Global Warming Potential (GWP) or Ozone Depletion Potential (ODP).

The whole system is also smaller than the electric-driven system with chiller and boiler. The absence of compressor reduces both noise and vibrations.

The energy needed for running the pumps corresponds to about 1% of the energy needed by the generator.

The absorption cooling represented 60% of the market of cooling machines in Europe in 2004, which means that it is quite easy to find a specific product in this area. In particular, much research has been done in this area and more chillers are now manufactured with a low capacity to fit better residential purposes.

Also, according to (Delbès & Vadrot, 2008), absorption coolers can be used on a wide power range from 10 to 100%, depending on the solar radiations and the outdoor conditions, without much change in the COP.

c. Disadvantages

Even if the absorption cooling uses solar energy and thus will need less energy from the electrical grid, the COP of the absorption chiller is smaller than the one of the conventional vapour-compression machines. The investment cost is also a major issue.

With the absorption cooling, there is a need of maintenance. One of the main problems that happen is crystallisation. More severe problems occur in air-cooled chillers than water-cooled units. To avoid it, there is a need to control the temperature of the heat supplied to the generator and the temperature of the cooling water that enters the absorber in case of water-cooled units. Otherwise, the cooling airflow rate should be controlled.

According to Delbès and Vadrot, the cost is almost the same for vapour-compression and absorption cooling systems. The latter is event reduced as there are few movable parts in the system and the average lifetime is about 30 years. Also, for most of the absorption cooling machines currently available, there is a need for a cooling tower, which requires more maintenance.

Finally, absorption coolers need at least one hour to reach their operating efficiency. A machine that is turned on and off regularly has a considerably lower average for the coefficient of performance than a machine used continuously. Guyer suggests having a cold storage unit to allow continuous operation of the absorption cooler. Thus, even if the absorption cooler is turned off at nights and needs time to start up, the cold storage unit can keep enough cold until the machine gets to its full efficiency.

d. Other possibilities

To increase the COP of the chiller, it is possible to use a double-effect or a triple-effect chiller, but this means bigger investments and also the driving temperature must be higher. To increase the temperature, there are two possibilities: investing in more effective thermal solar panels or using a secondary heat source, such as natural gas.

H. Desiccant cooling

a. Working principle

As opposed to the absorption cooling, the desiccant cooling is an open-cycle process: the refrigerant does not re-circulate.

This cooling method is based on the hygroscopic properties of certain materials called desiccants: they are used to attracting molecules of water and dehumidify the air before it enters in the indoor space. This works in two steps. First, they collect moisture, as the surface vapour pressure of the desiccants is smaller than the vapour pressure of the surrounding humid air. After some time, they get saturated and cannot take up more water, they need to be regenerated. By using a stream of hot air, they attract the moisture from the sorbent surface of the desiccant and drive it away.

There are many desiccant materials. Some are solid like silica gel, lithium chloride salt, activated alumina or zeolite. Others are liquid: lithium chloride, lithium bromide, calcium chloride solutions.

b. Solid desiccant cooling

In a desiccant cooling machine, the outside air (hot and humid) is first dehumidified in the sorption wheel where the solid desiccant is stored, it gets very dry (1-2). Then, the air goes through a heat recovery wheel where it is pre-cooled (2-3). After that, it is cooled by rehumidification (3-4). There might be a need of heating the air again to get it at the desired temperature (4-5), before supplying it indoors (6). The heating device might be connected to the thermal solar panel.

The return air is humidified first as close as possible to saturation (7-8). This enables a higher potential for cooling the supply air through the heat recovery wheel (2-3). Then it is pre-heated through the heat recovery (8-9). After that, a heater connected to the thermal solar panel brings the air to a high temperature (9-10). Therefore, the air is hot and dry when entering the sorption wheel (10). It can therefore take away the humidity from the desiccants (11).

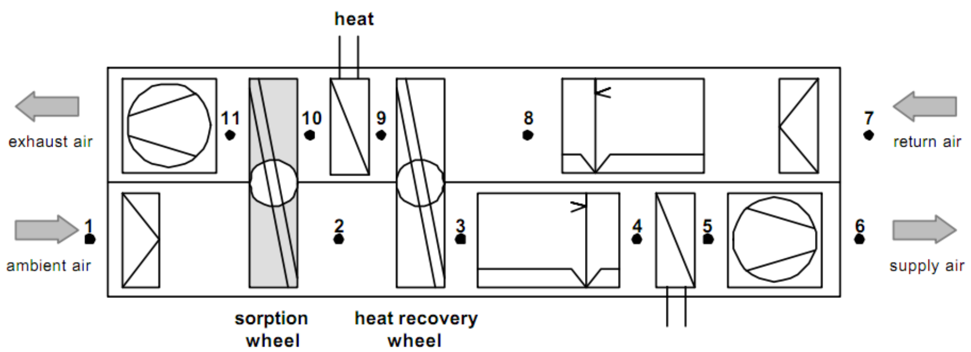


Figure 89 Schematic drawing of a solid desiccant cooling air-handling unit (Altener 2002)

c. Liquid desiccant cooling

The advantage of the liquid desiccant cooling is that the regeneration temperature of the desiccant can be lower than for the solid desiccant. Yet, there is a need of using a cooling coil.

d. Advantages

Compared to the vapour-compression and absorption system, desiccant cooling do not use any refrigerant, and so has no Global Warming Potential (GWP) or Ozone Depletion Potential (ODP).

In this system, the air temperature and relative humidity can be controlled independently. It makes it easier to meet the demands.

Moreover, the cost of the solar components are minimised thanks to the possibility of using flat plate collectors, which can heat up water up to 80C. Even the solid desiccants need a low temperature heat source to be regenerated, around 50 to 75C. At this temperature, desiccant cooling is more efficient than absorption cooling systems.

Also, all stages are operated at atmospheric pressure. This avoids using pressure resistant units and keeps the cost low.

Besides, it can be driven by solar energy, waste heat, or natural gas, thus keeping operation costs of the cooling system reduced (except for natural gas).

As well, it can be used in hybrid air-conditioning systems: the desiccant cooling handles the latent load by taking out moisture of the air; and it is used with an additional cooling machine (compression chiller), taking care of the sensible load. This one decreases the air temperature, reducing the amount of energy needed, and enhancing the efficiency by using a higher evaporator temperature.

e. Disadvantages

Several disadvantages have been listed for such systems.

The main one is that the COP of the desiccant cooling is only 0,7 and is highly related to climatic and operating conditions (indoor and outdoor climate).

Also, liquid desiccant systems use inorganic salts (LiCl) that have corrosive properties, inducing negative influence on the system.

Risk of crystallisation if too high concentration

One should not forget that the solar part still dominates the cost. Therefore, there is a need for better performance and cheaper price for the collector.

Finally, it should be mentioned that only few applications of liquid desiccant systems are made for solar-driven air conditioning at the moment.

I. Further results

Table 18 Results from further simulations

	1-step chiller	Variable work chiller	Variable work chiller	Variable work chiller
Inlet fan coil water temperature [°C]	8	8	12	12
Outlet fan coil water temperature [°C]	14	14	19	19
Outlet chiller water temperature [°C]	7	7	6	6
Fan air flow [m^3/s]	0,90	0,90	0,80	0,80
Fan coil maximum cooling load [W]	5500	5500	5250	5250
Outlet fan coil minimum air temperature [°C]	14	14	16	16
Indoor temperature [°C]	22,4-24,5	22,4-24,5	22,8-25,2	22,8-25,2
Compressor maximum work [W]	4500	5000	4750	6000
PV panels area [m^2]	71	72	50	62
Efficiency of PV panels [-]	13%	13%	18%	14,3%
Total installed power [kW]	9,2	9,4	9,0	8,9
Electricity provided to the chiller for 1 year [kWh]	13600	13710	13200	13000
Cooling capacity of the chiller for 1 year [kWh]	33500	43000	40000	35000
Total energy produced by the solar panels for 1 year [kWh]	13650	13750	13250	13050
Total energy exported to the grid by the solar panels [kWh]	3800	2740	2600	2400
Total energy imported from the grid for 1 year [kWh]	3740	2700	2550	2350
Tank volume [l]	8000	9600	5000	6000
Days without enough cooling	0	0	2	19