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# Analysis of the Heating Needs in Natatoriums

**Clément Charles Perisse**

Sustainable Energy
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Supervisor: Laurent Georges, EPT
Co-supervisor: Ole Smedegård, SIAT

Norwegian University of Science and Technology
Department of Energy and Process Engineering

NTNU

Norwegian University
of Science and Technology

Department of Energy
and Process Engineering

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# MASTER THESIS

for

Student Clement Perisse

Spring 2018

*"Analysis of the Heating Needs in Natatoriums"*
*Analyse av oppvarmingsbehov i innendørs svømmeanlegg*

**Background and objective**

Swimming halls represent a type of building with several challenges and distinctions. Some of these are high level of humidity and high indoor temperature, which gives a risk for condensation and problems related to moisture in the construction. Along with the obvious direct influence this indoor condition has on the heating demand, both the energy related to showering and to the process system in relation to the operation of the swimming pool itself (i.e. the water treatment system) are of a significant level. In addition, the air handling system (AHU) operating to maintain a satisfying air quality under the strong influence from emissions in relation to the water treatment and the impact of the swimmers.

The objective of the master thesis is to analyse the heating demands in natatoriums, a prerequisite for the proper design of the HVAC system. The student shall carry out this analysis using the building performance simulation tool IDA-ICE. After these heating needs will be characterized, possible ways to cover them will be investigated. This theoretical analysis will be combined with a practical case. At Jøa in Fosnes commune, a new swimming hall was opened in January 2017. The swimming hall is part of a multipurpose sport centre and serve as a therapeutic pool. Its innovative and advanced HVAC system is a "state-of-the-art" in Norway and include both an innovative air flows distribution system and an advanced renewable thermal energy supply system. A comparison of the IDA-ICE model(s) with this building shall be performed.

The work is connected to a PhD work at SIAT, Centre for Sport Facilities and Technology, entitled "*Optimizing energy and climate system in buildings with swimming facilities*".

**The following tasks are to be considered:**

1. Literature review
2. Modelling a generic swimming pool along with the air handling unit (AHU) in IDA ICE.
3. Analysis of the heating needs for the different main design parameters (sensitivity analysis).
4. Discuss the influence of these loads on the HVAC system design and operation.

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Within 14 days of receiving the written text on the master thesis, the candidate shall submit a research plan for his project to the department.

When the thesis is evaluated, emphasis is put on processing of the results, and that they are presented in tabular and/or graphic form in a clear manner, and that they are analysed carefully.

The thesis should be formulated as a research report with summary both in English and Norwegian, conclusion, literature references, table of contents etc. During the preparation of the text, the candidate should make an effort to produce a well-structured and easily readable report. In order to ease the evaluation of the thesis, it is important that the cross-references are correct. In the making of the report, strong emphasis should be placed on both a thorough discussion of the results and an orderly presentation.

The candidate is requested to initiate and keep close contact with his/her academic supervisor(s) throughout the working period. The candidate must follow the rules and regulations of NTNU as well as passive directions given by the Department of Energy and Process Engineering.

Risk assessment of the candidate's work shall be carried out according to the department's procedures. The risk assessment must be documented and included as part of the final report. Events related to the candidate's work adversely affecting the health, safety or security, must be documented and included as part of the final report. If the documentation on risk assessment represents a large number of pages, the full version is to be submitted electronically to the supervisor and an excerpt is included in the report.

Pursuant to "Regulations concerning the supplementary provisions to the technology study program/Master of Science" at NTNU §20, the Department reserves the permission to utilize all the results and data for teaching and research purposes as well as in future publications.

The final report is to be submitted digitally in DAIM. An executive summary of the thesis including title, student's name, supervisor's name, year, department name, and NTNU's logo and name, shall be submitted to the department as a separate pdf file. Based on an agreement with the supervisor, the final report and other material and documents may be given to the supervisor in digital format.

☐ Work to be done in lab (Water power lab, Fluids engineering lab, Thermal engineering lab)
☐ Field work

Department of Energy and Process Engineering, 09. February 2018

Laurent Georges, Associate Professor
Academic Supervisor

Research Advisor:
Ole Øiene Smedegård, PhD student (SIAT)

# Abstract

In the quest of improving energy performance in buildings, swimming facilities stand out due to their excessive consumption. When it comes to scientific research, few publications have addressed this type of facility compared to residential or commercial buildings. The specificity of natatoriums is due to complex processes that challenge indoor environment, building structure maintenance, and energy use in a great extent. Energy saving potential becomes all the more important as these processes are highly consuming and appear not to be optimally tackled. Large discrepancies in energy consumption exist among swimming facilities, and W. Kampel estimated the final annual energy consumption could be lowered by 28% in total in Norway.

This study aims at analyzing energy use and power demand in natatoriums. Two models were built utilising the building performance simulation tool, IDA-ICE, along with the Ice Rinks & pool extension. The consumption per square meter of water surface within the model was slightly above the Norwegian statistical average of 4000 kWh/m_ws^2. The analysis brought a deeper understanding about both the software code related to the pool and the heating needs of swimming facility. Energy need due to evaporation appeared to be equivalent to the space-heating needs and even showed to have higher power peak values.

Sensitivity analysis were run to investigate influences of key parameters. Four parameters stood out: the pool temperature set-point, the pool area, the n_50 infiltration coefficient, and the pool activity factor. Further analysis focused on the effect these parameters have on the system's behavior. Studies were conducted to explore whether a parameter triggers higher power demand from the air-handling unit or increased loss through the structure, or higher evaporation rate leading to higher energy demand for heating of the pool water.

Finally, a comparison with the practical case of a swimming pool at Jøa was established thanks to data retrieved from sensors and energy meters set up there. The use of an integrated heat pump in the air-handling unit at Jøa can be misleading and then only domestic hot water consumption and thermal energy released to the pool are comparable. Both seem to be much higher in the IDA-ICE models. Due to short time framing and high complexity, a model gathering all specific features from Jøa still need to be built and further investigations are required.

# Preface

This Master's thesis of 30 ECTS credits is submitted to the Norwegian University of Science and Technology (NTNU) in Trondheim. It represents the conclusion of the Master of Science grade in energy use in buildings. I would like to thank my supervisor Laurent Georges and my co-supervisor Ole Øiene Smedegård for introducing me to this interesting topic, and for their guidance.

I would especially like to thank Florent Dulac whose continuous support and friendship along the way made this thesis a pleasant journey.

Hope you enjoy the reading.

Clément Perisse

Trondheim, 06.07.2018

# Table of Contents

# 1 Framework

This chapter will introduce the design characteristics of swimming facilities and the specific challenges they face. An overview of the work that has been done regarding swimming facilities is presented through relevant literature.

## 1.1 Swimming Pools Design

Swimming facilities are facing complicated indoor processes that trigger harm to the structure, deterioration of the indoor environment quality, and enhanced energy consumption.

### 1.1.1 General Challenges

Even in the diversified group of sport facilities, swimming pools have special overall design due to the special indoor air environment and special building needs, e.g. technical system for water purification and climate control. Several characteristics distinguish swimming facilities from other building (Kampel, 2015):

- Temperature and humidity level in the pool hall
- Strong evaporation of the pool water
- Ample need of warm tap water for pool and showers
- Water treatment system
- Wide use of heat recovery systems
- High energy use

Regarding the European policy about building construction, sport facilities tend naturally towards high insulted envelope and recent facilities are made according to the passive house standards. However, for swimming facilities neither a standard nor a best practice code exists for the energy use (Kampel, 2015). Indeed, as regard of the singular processes within a swimming facility, lowering U-values of the envelope is likely not to be sufficient to increase energy savings. It appears that climate has a negligible impact compared to the energy demanding processes happening inside the building.

A major problem concerning the structure is condensation. This issue occurs on cold surfaces of the inner side of the building envelope where temperature is below the dew point. Due to relatively high humidity, this dew point temperature is high and it becomes a challenge to avoid condensation even during the hottest day of the year. To avoid cold surfaces, walls, roof, windows must have a low enough U-value and be air-tight. A general solution to counteract possible condensation is to place convective heater below windows in order to create a warm, dry curtain of air. Finally, the roof must also be protected from rising warm and humid air from the pool. Condensation may be fatal to the building envelope by causing moistures among others.

The indoor climate is very special. Due to pool activities during the day, evaporation rate fluctuates widely throughout the day and is the main cause of indoor climate variation. Evaporation gives rise to two main issues in energy consumption in swimming facilities:

- Evaporation raises the load for water heating. First, this is due to the new need in fresh water that has to be heated up to the desired value. Second, evaporation process takes out heat from the pool water which, as a matter of fact, cools down the water and lower its temperature. This process is similar to perspiration which lowers skin temperature
- Evaporation affects the indoor air quality by bringing large amount of humidity in the air. Thus, the necessary increased ventilation controls and limits this humidity level. However, this induces larger energy consumption for heating the makeup air to the required temperature.

A heat recovery device has to be implemented to keep humidity at a constant level. Also, the role of the heat recovery unit is substantial to dilute polluted air caused by chlorine by-products originating from the pool water while avoiding losing an enormous quantity of energy due to high inside air temperature.

Furthermore, water consumption is considerable in swimming facilities. First, Norwegian regulation recommend exchanging 30 liters for every visitor if the water is below 34°C (*Retningslinjer for vannbehandling i offentlige bassengbad*, 2000). Second, large flows of water are used through the showers. Since this water came from the grid at a very low temperature (roughly 5°C) and is heated until very high temperature (roughly 70°C) to kill pathogens, grey water heat recovery becomes essential to reach high energy efficiency. Also, the water must be treated. Chlorine is widely used compared to seawater adding to avoid corrosion issues.

## 1.1.2 Thermal Analysis of a Pool

The interesting study by Kuyumcu and Yumrutaş (2016) gives a proper overview of interactions between the pool and the indoor environment. This study goes into specific concerning all kinds of energy transports that can happen within a pool

Heat losses from the indoor swimming pool occur in five different ways:

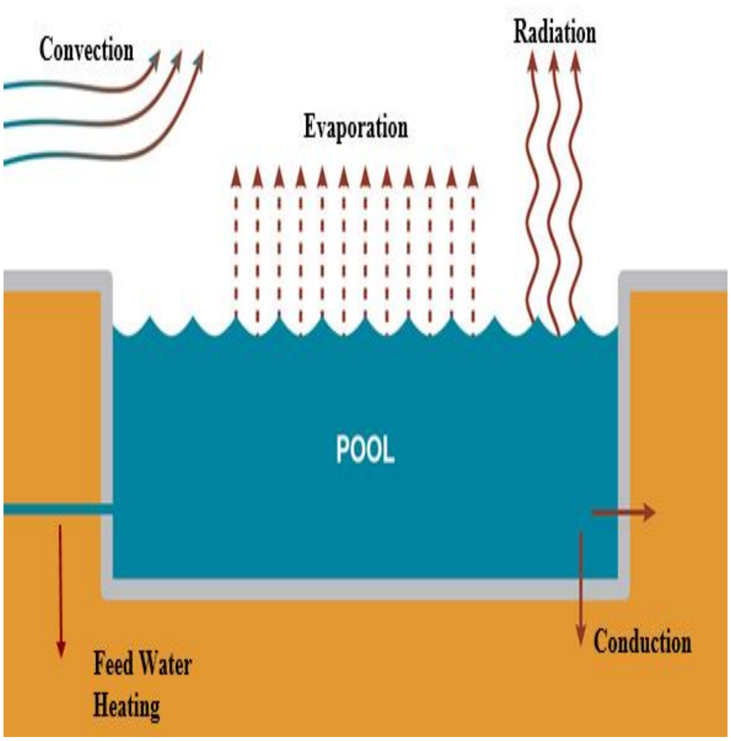



Figure 1-1 Schematic view of the pool system's energy cost.

- Convection heat loss
- Conduction heat loss from bottom surface and side wall to the ground
- Latent heat loss due to evaporation from the surface of the water
- Radiation heat loss that occurs between the surface of the pool and the ceiling
- Energy requirements for feed water heating.

Kuyumcu and Yumrutaş (2016) set up a MATLAB model describing these complex heat transfers. Clearly here, the system at stake is the swimming pool. Kuyumcu and Yumrutaş (2016) made the following assumptions:

- The swimming pool area is A=1250 m² which is the size of an Olympic one.
- The pool water is supposed constant, 26°C.
- Oddly enough, authors state in their study that the indoor air is supposed to vary throughout the year between 10 and 15°C. I suspect a mistake in the sentence and the indoor air temperature would vary between 20 and 25°C.
- ASHRAE evaporation formula (see equation 1-1) gives the evaporation load.
- Air speed above the water is u=0.15 m/s.
- Conduction heat transfer coefficient of ceiling component k=0.035 W/mK. $h_i$ and $h_o$ are 10 and 20 W/m²K respectively and stands for indoor and outdoor convection heat transfer coefficients. Then, for an optimal (according to the study results) thickness of 3cm, this gives a heat transfer coefficient of $U = \frac{1}{\frac{1}{h_i}+\frac{e}{k}+\frac{1}{h_o}} = 1.01$ W/m²K. This value is high, compared to usual insulation values. However, the study focuses only on the swimming pool system and this thickness is optimal only for cutting down the radiation losses.
- Temperature of fresh water is 10°C and mass flow of fresh water is the sum of evaporated water plus 0.2 kg/s for Olympic sized swimming pool.

Figure 1-2 shows the yearly average energy fraction of a swimming pool heat loss. One may be careful about these values considering the assumptions made. First, the insulation thickness is rather low, and in a case closer to the reality heat radiation from the pool surface to the ceiling will be lower. Indeed, with higher insulation the inner surface of the roof shows higher temperature, closer to the indoor air temperature. Since ceiling temperature will be higher and then closer to that of the water, the emissivity will be lower and the heat radiation as well. Also, the supply of fresh water is low compared to our study case in IDA-ICE (see later). The model includes in average a 0.2 kg/s mass rate supplied to the pool and in this analysis the amount is the same (without the evaporation) but for a swimming pool 12.5 times larger. However, when it comes to evaporation and temperature of fresh water, assumptions are realistic or at least comparable to the standard model established in IDA-ICE. Although the incoming temperature of fresh water is 10°C, it remains close to our study case (8°C). This analysis takes the ASHRAE formula to calculate the evaporation rate.

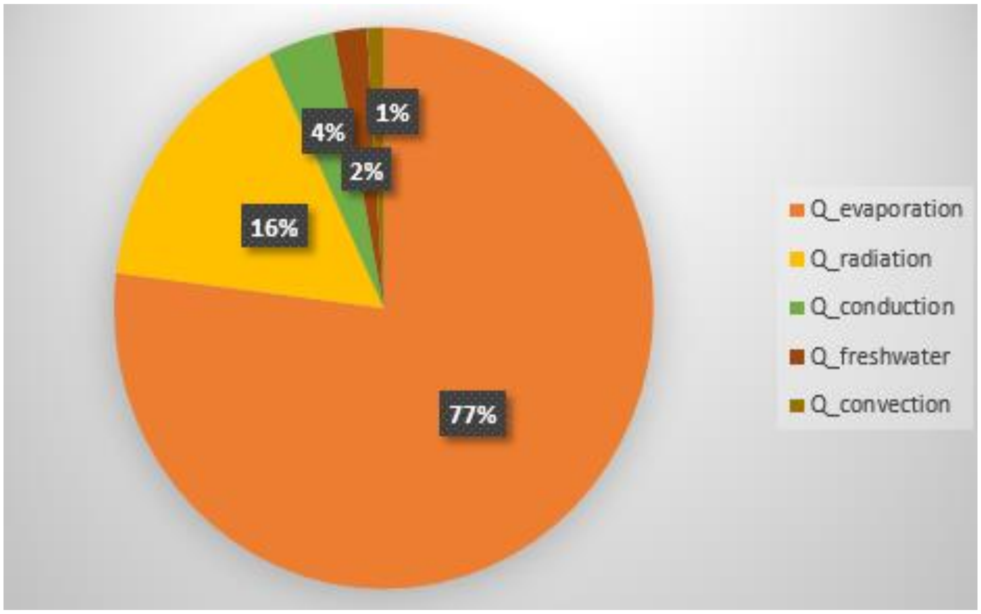



*Figure 1-2 Yearly average energy fraction of a swimming pool heat loss. (Kuyumcu & Yumrutaş, 2016)*

## 1.1.3 Evaporative Load

The most distinctive thermodynamic mechanism occurring in natatoriums is evaporation. Evaporation occurs at the air-water interface when the vapor-pressure of the air is less than the saturation pressure of the water. Indeed, within a tiny layer above the water, the air is saturated in vapor due to molecular mixing: some molecules at the water surface separate and are released into this tiny layer. What happens is an osmosis phenomenon. As long as the content in water vapor in the air is less than the one in the tiny layer, vapor is transferred to the indoor air. Intensity of this phenomenon is directly correlated to the difference between the partial pressures for water vapor right above the water surface, and for room air.

Many papers try to provide reliable method to calculate evaporation load. Both ASHRAE (USA) and VDI (Germany) have distributed guidelines for calculation/estimation of evaporation rate which probably are the most used calculation guidelines today. The similarity of these methods lies in the driving force, the pressure difference between the partial pressure in the surrounding air and the vapor pressure at the water surface. The way the equations approach the usage, with tabulated values of the activity level, is also similar and both have a structure that approaches hand calculation and easy implementing to BPS-programs.

Parsons, American Society of Heating, and Air-Conditioning (1995) estimated an evaporation rate ($w_p$ in kg/s). The value in equation 1-1 is valid with normal activity levels, and with a limited area of wetted deck.

$$w_p = \frac{A(p_w - p_a)(0{,}089 + 0{,}0782\,V)}{Y} \tag{1-1}$$

$A = area\ of\ pool\ surface, m^2$

$p_w = saturation\ vapor\ pressure\ taken\ at\ surface\ water\ temperature, kPa$

$p_a = saturation\ pressure\ at\ room\ air\ dew\ point, kPa$

$V = air\ velocity\ over\ water\ surface, m/s$

$Y = latent\ heat\ of\ vaporization\ at\ surface\ water\ temperature, kJ/kg$

By assuming a heat of vaporization Y equal to 2330 kJ/kg, a V value of 0.10 m/s, ASHRAE formula 1-1 turns into:

$$w_p = 4{,}16.10^{-5}\,A(p_w - p_a)F_a \tag{1-2}$$

$F_a = activity\ factor$

ASHRAE (1999) gives the following values of activity factor depending on the type of pool.

*Table 1-1* Typical activity factor according to the type of pool. (ASHRAE, 1999)

| Type of Pool | Typical Activity Factor ($F_a$) |
|---|---|
| Residential Pool | 0,5 |
| Condominium | 0,65 |
| Therapy | 0,65 |
| Hotel | 0,8 |
| Public, Schools | 1,0 |
| Whirlpools, spas | 1,0 |
| Wave pools, water slides | 1,5 (minimum) |

## 1.1.4 Common Design

### *1.1.4.1 Indoor Air Conditions*

The choice of boundary conditions is an essential element of an optimized system. ASHRAE (1999), chapter 4, recommends maintaining the natatorium air temperature between 2 and 4°C above the pool water temperature but not above the comfort threshold of 30°C. The reason of this threshold is to find a balance between the sizing of the dehumidifier and the energy cost of rising and maintaining the indoor air temperature. By setting the indoor air temperature 2 degrees above the pool water temperature, a balance is found to reduce evaporation, and therefore the size of the dehumidifier, and the energy cost of associated heating. Also, the swimmers will not feel cold when they leave the water. When setting this temperature, one dealt with:

- Heating loads of the pool water
- Evaporation rate
- Ventilation need
- Comfort of the users (both outside and within the water)

It has been shown that a 2°C difference between air and pool water temperature strikes an optimal balance for both energy consumption and comfort quality. For higher indoor air temperature, users feel discomfort entering the water due to temperature difference. Also, pool environment can be too warm. Evaporation rate gets higher which induces higher heating load for water heating and ventilation needs to maintain acceptable humidity level. Lower air temperature would induce higher sensible heat losses from the pool water to the ambient air, also higher evaporation rate (given a constant relative humidity).

About the humidity, ASHRAE (1999) suggests a relative humidity kept between 50% and 60%. Lower percentages would be counterproductive since it would significantly affect operation costs. Indeed, lower indoor relative humidity would give rise to increased evaporation from the pool surface, also it may create discomfort from the swimmers who exit the pool due to evaporating cooling from their bodies. Higher relative humidity jeopardizes building structural elements, furnishing, and support systems such as lighting (Aire, 2015). Condensation can destroy key building features and impair insulation values if it goes through the building structure. Constant presence of moisture on the glass and steel structure can lead to corrosion, shortening the lifespan of the facility and ultimately becoming dangerous.

Table 1-2 sums up the temperature and humidity conditions advised by ASHRAE (1999).

*Table 1-2* Typical Natatoriums Design Conditions. (ASHRAE, 1999)

| Type of Pool | Air Temperature (°C) | Water Temperature (°C) | Relative Humidity (%) |
|---|---|---|---|
| Recreational | 24 to 29 | 24 to 29 | 50 to 60 |
| Therapeutic | 27 to 29 | 29 to 35 | 50 to 60 |
| Competition | 26 to 29 | 24 to 28 | 50 to 60 |
| Diving | 27 to 29 | 27 to 32 | 50 to 60 |
| Whirlpool/Spa | 27 to 29 | 36 to 40 | 50 to 60 |

A vapor barrier is a material or film that prevents moisture penetration. The vapor barrier is usually located on the outside of the building's insulation. Vapor barrier should be place on the side where there is the highest moisture content but because of the high humidity in pool room, the vapor barrier is required to be on the inside for at least structure in North-America (Aire, 2015). The structure is then protected from humidity that could hamper its performance or prompt mold. Equivalently, adjacent and interior rooms need to be protected. Moisture content in these rooms is far lower due to lower air temperatures and relative humidity. Moisture is expected to migrate from the pool hall to the rest of the building. This phenomenon causes moisture damages if there is no appropriate vapor barrier implemented.

Baxter (2012) puts forward that, among all the disinfectant by-products (DBPs) present in a pool atmosphere, trichloramine vapor is the main compound accounting for air quality problems that cause adverse physiological responses in humans. Trichloramine is a potent respiratory irritant. It is toxic and can exist as a vapor in the air. Although not universally agreed, a pH-dependent substitution reaction would be the cause of its formation. These reactions requires acidic conditions that are not present in the pool water. Nevertheless, the omnipresence of organic nitrogen compounds, such as urea, due to the swimmers would trigger trichloramine formation. The core issue of this extremely volatile compound lies on the fact that its density amounts to several times that of dry air (Baxter, 2012). Consequently, it accumulates in low-lying places, which makes it difficult to remove. Furthermore, high-speed ventilation air above the pool surface causes higher evaporation rate (ASHRAE, 1999) which potentially causes a drastic rise in energy consumption. According to Baxter (2012), most modern pool HVAC systems are designed to limit air velocity across the pool surface, and this limited air movement is not sufficient to dislodge and lift to the exhaust grilles the dense trichloramine bubble. HVAC designers have to strike a balance between an increased rate of evaporation along with a swimmer-chilling problem and a satisfactory concentration of trichloramine.

Baxter (2012) presents a “source capture and exhaust” strategy shown Figure 1-3. An extra-exhaust air is set up close to the waterline to capture and remove toxic DBP due to high density vapor. The system needs to be dimensioned so that air velocity across pool surface does not exceed a certain speed e.g. 1 m/s. Cavestri and Seeger-Clevenger (2009) tested this exhaust strategy successfully in a model pool using test vapor with known properties similar to those of trichloramine vapor. They found out a top-level recirculation design does not efficiently clear the test vapor. On the opposite, a deck-level exhaust to the outside achieves complete elimination of the test vapor while having solely 1 to 2 fpm (0.005 to 1 m/s).

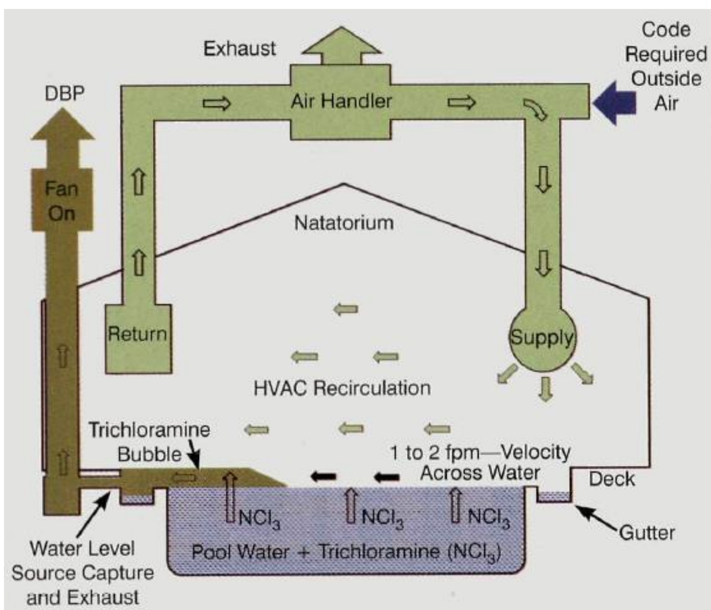



*Figure 1-3* Source capture and exhaust strategy to remove toxic DBP. (Baxter, 2012)

### *1.1.4.2 Ventilation Requirements*

Most local codes are based on ASHRAE Standard 62.1, the industry accepted ventilation code for indoor air quality. ASHRAE 62.1, table 6.1 provides the following levels of outdoor air to the breathing zones listed below. According to IECU (1994), the fresh air requirements in the swimming pool area, during operation hours, amounts to 10 $m^3/h$ per square meter of pool water surface. This is usually equivalent to 5 ach (air change per hour). To provide sufficient air to flush the walls and windows, prevent stratification and deliver air down to the breathing zone, ASHRAE Applications Handbook recommends the air change rates listed below (ASHRAE, 1999):

- 4 to 6 ach for pools with no spectator area
- 6 to 8 ach for pools with spectator areas
- 4 to 6 ach for therapeutic pools

*Table 1-3:* Minimum outdoor air volume rates to breathing zones requirement for swimming hall. (Ashrae, 2013)

| | IP units | SI units |
|---|---|---|
| Pool & wet deck | Area ($ft^2$)*0,48 ($cfm/ft^2$) | Area ($m^2$)*2,4 ($L/s.m^2$) |
| Remaining floor (the room without the pool, the wet deck, and the bleacher) | Remaining Area ($ft^2$)*0,06 ($cfm/ft^2$) | Remaining Area ($m^2$)*0,30 ($L/s.m^2$) |
| Spectator & Bleacher | Spectator area ($ft^2$)*0,06 ($cfm/ft^2$)+number of spectators*7,5(cfm) | Spectator area ($m^2$)*0,30 ($L/s.m^2$)+number of spectators*3,8 (L/s) |

A recent adjustment in natatoriums design has been to modify the way fresh air is supply to the room. From an energy performance point of view, grilles should no longer be aimed at the water surface so that air velocity remains below 30 feets per minute (equivalent to 1,524 m/s). Besides, the ASHRAE evaporation load formula assumes that air velocity is below this limit (Shah, 2014). According to Aire (2015), given a 3500 sq.ft, or 325.16 $m^2$, at 82°F water temperature and 84°F air temperature, or respectively 27.78°C and 28.89°C, should the airflow above the pool surface be increased from 30 fpm to 125 fpm, the evaporative load would grow by 40%.

As for the exhaust grilles, higher locations optimize the recovery of the higher temperature and humidity containing air, since hot humid air rises.

Spectator loads have potentially a great impact on the indoor environment through breathing, body heat, and perspiration. Ashrae (2013) requires additional volume of fresh air supplied to the zone when spectators are present. Because spectator occupancy is a very flexible variable, most building owners and HVAC engineers choose to install a dedicated outdoor air system (DOAS) which allows for independent control of temperature, supply air and exhaust air, and duct placement. According to Aire (2015), this independence could reduce energy cost compared to a combined system for both pool, wet deck, and spectator area.

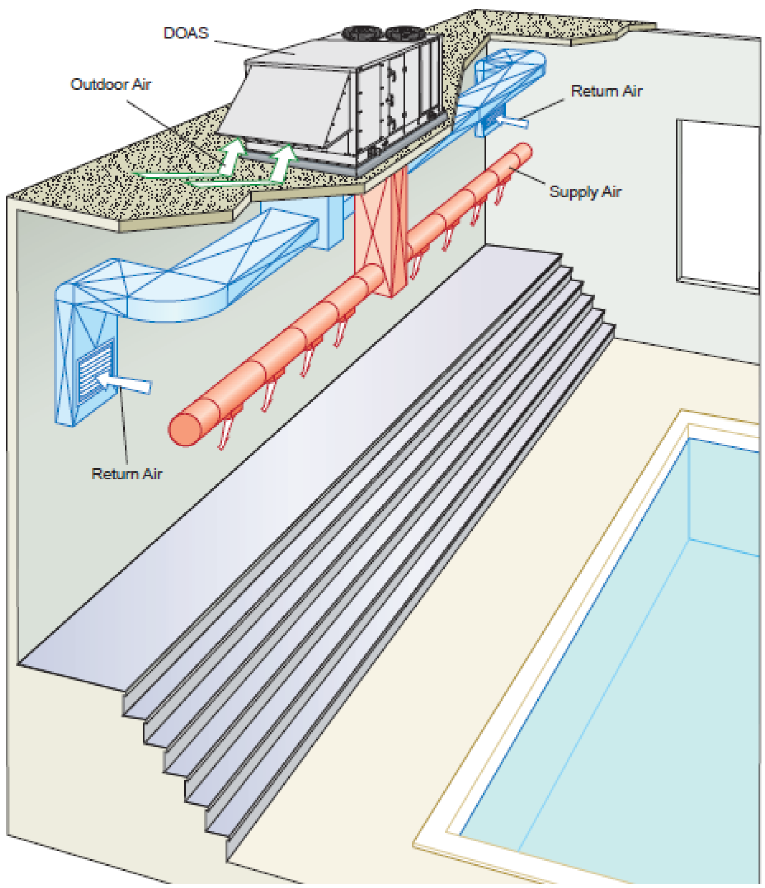



*Figure 1-4* Schematic view of a DOA system for spectator's area. (Aire, 2015)

### *1.1.4.3 Dehumidifier Systems & Condensation Control*

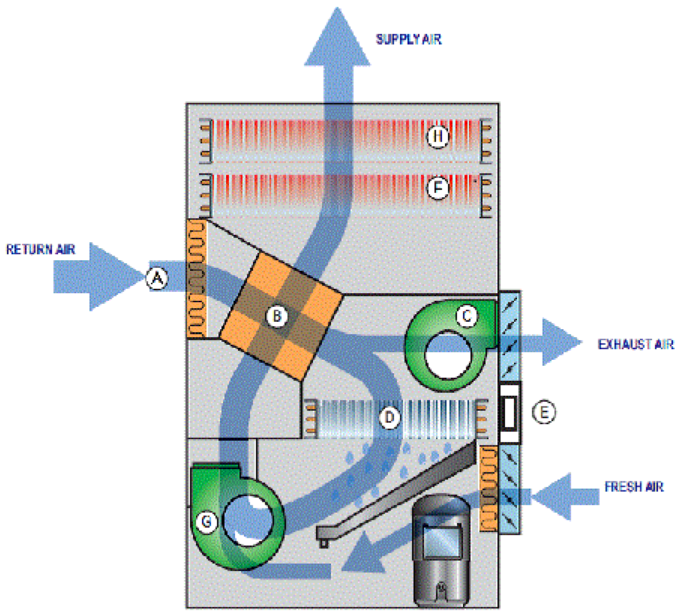
Dehumidifiers are simply air handlers that remove moisture from the air. They are dimensioned according to the calculated evaporation load due to the pool and other water equipment (Jacuzzi, speed slides, wave pools…).

The major part of dehumidifiers used in HVAC system works according to the same principle: thermal condensation. Since the saturation vapor pressure of water decreases with decreasing temperature, by cooling down the air below the dew point temperature, condensation happens and separates moisture from the air. Therefore, dehumidifiers can be seen as a special chiller unit. Figure 1-5 shows one example. The main steps are listed below:

1. The air goes through the air heat exchanger and gives away sensible heat
2. The air circulates through the evaporator, its temperature goes below its dew point, and moisture is removed by condensation. This allows to recover both sensible and especially latent heat contained in the air
3. This low-moisture content air mixes with fresh outdoor air
4. This mix takes heat by going through the air heat exchanger
5. Eventually the air is thrown back to the building after being re-heated by the condenser of the chiller unit

Thus, a dehumidifier works according to the same thermodynamic principles as heat pump units. Besides, the compressor appears at the bottom right-hand corner of Figure 1-5.

The heat sink in Figure 1-5 is the recirculated air thrown back to the inside but there can be many others. A water coil, generally a tube-in-tube heat exchanger, may be added as an additional heat sink. For example, this heat sink could be the pool water, domestic hot water or hydronic heat water. In case of a large dehumidifier that satisfies respective set-points of the heat sinks, a remote condenser dissipates the heat to the surrounding environment (Aire, 2015).




| | |
|---|---|
| A | Inside |
| B | Heat recovery |
| C | Exhaust fan |
| D | Cooling coil (Evaporator) |
| E | Outside |
| F | Re-heating coil (Condenser) |
| G | Supply Fan |
| H | Additional heating coil |

*Figure 1-5* A common dehumidifier system.

The main function of such a HVAC component is to avoid any condensation risk. Condensation appears in certain case even though the dehumidifier is properly dimensioned and operated. In fact, it is important that the HVAC engineer and the architect keep on communicating so that the there is a common agreement on construction materials, vapor barrier locations, quantity of openings, and also that a satisfactory air distribution is achieved. All external surfaces must be adequately washed with fresh air to prevent condensation, especially in winter month.

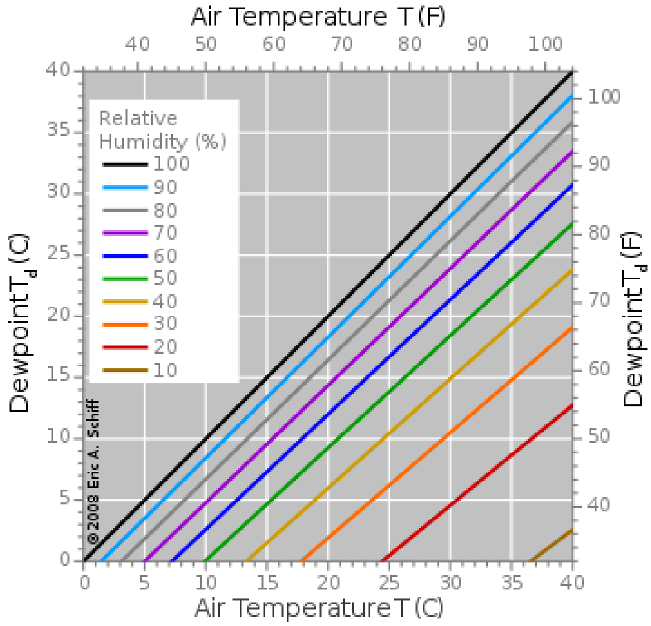



*Figure 1-6* Dew point temperature according to the air temperature (*https://en.wikipedia.org/wiki/Dew_point*)

Walls, doors, and external walls temperature must be kept above the dew point to avoid condensation. In Figure 1-6, for an indoor air conditions of 30°C (or 86°F), and 50% relative humidity, then the dew point temperature is roughly 18°C. Should the window or a part of the window be below this threshold, condensation will happen.

### *1.1.4.4 Air Handling Unit*

Most AHU suppliers specialized in swimming facilities include the dehumidifier directly inside the AHU. Figure 1-7 illustrates a typical AHU in Norway. This design is implemented in Jøa swimming facility and is the typical layout used by the two main suppliers in Norway.

Given a decent mixing in the hall and then a low air age, relative humidity and temperature at the exhaust grille give a trustworthy idea of actual humidity and temperature inside the hall. Depending on these values, control valve 3 (see Figure 1-7) regulates the incoming fresh volume. When the building is not under operation for instance, Control valve 3 is almost closed while control valve 1 is fully open. It means almost no fresh air enters while almost the whole exhaust air is being recirculated. And if humidity remains low enough, control valve 2 opens directly and air is recirculated even before the heat exchanger and the dehumidifier is switched off. The dehumidifier is shown inside the dotted rectangle in Figure 1-7. Basic principles of dehumidifier are explain section 1.1.4.3. In this case moisture is withdrawn from the exhaust air before being partly recirculated through control valve 1. Thus, when mixing with the fresh air, it dries the air supplied and allow a control of the hall relative humidity by adjusting:

- Control valve 1 opening
- The compressor volume rate and then the overall dehumidifier efficiency.
- The supply of fresh air with control valve 3
- The amount of exhaust air directly recirculated with control valve 2

Temperature control comes after the humidity control. Depending on the dehumidifier's heat energy release, supply air is re-heated by a secondary system. This secondary system corresponds to the heating coil linked to the pump Figure 1-7. The energy may come from a heat pump or a more conventional heating system. In the case of Jøa, this secondary system releases heat thanks to a $CO_2$ heat pump.

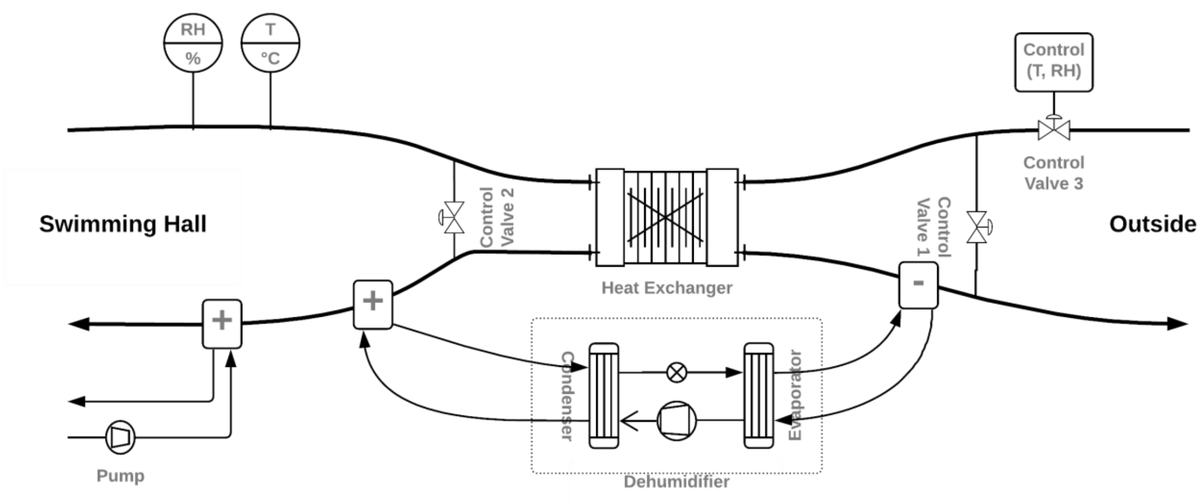



*Figure 1-7* Schematic representation of a standard AHU in Norway.

### *1.1.4.5 Negative Air pressure Strategy Inside Pool Hall*

Natatoriums should be maintained at a negative air pressure relative to the outdoors and the adjacent rooms (ASHRAE, 2007). Due to the high temperature of the air in pool halls, the maximum value for humidity content is high. This means each kilo of air is capable of transporting higher amount of water vapor, and 50% of relative humidity at 30°C corresponds to 90% relative humidity at 20°C. Therefore, it makes sense that the pool hall indoor air should be restrained to this area and not propagate. This is even more important in order to prevent the dispersal of chloramines and other noxious contents used for cleaning of the pool water.

It exists two options when it comes to maintaining favorable negative air pressure:

- Static pressure control
- Active pressure control

In static methods, the automated system controls dampers to deliver the right amount of air according to the differential pressure across HVAC system components (e.g. dehumidifier's evaporator coil or exhaust fan).

According to ASHRAE (2007), active methods are the most suitable solutions and may be necessary when active humidity control strategy uses outdoor air. In active methods, a Variable Frequency Drive (VFD) or an Electronically Commutated Motor (ECM) controls the speed of the fan as well as the volume flow of exhaust air. This amount of exhaust air matches the real-time load and needs of the natatorium's negative pressure requirement. This method is more accurate and compensate any unexpected event like a supplementary exhaust source if someone opens a window.

Figure 1-8 illustrates the building pressure strategy according to Aire (2015). Locker rooms stands also for dressing room and food preparation spaces. They need to have a negative relative pressure with the other adjacent rooms. In case of chemical storage area, these must show a negative pressure even relative to the pool. In this case, the ventilation strategy ensures protection of inner structures from the warm and humid air of the pool hall. It prevents both the moisture and other by-products to propagate through the other parts of the building.

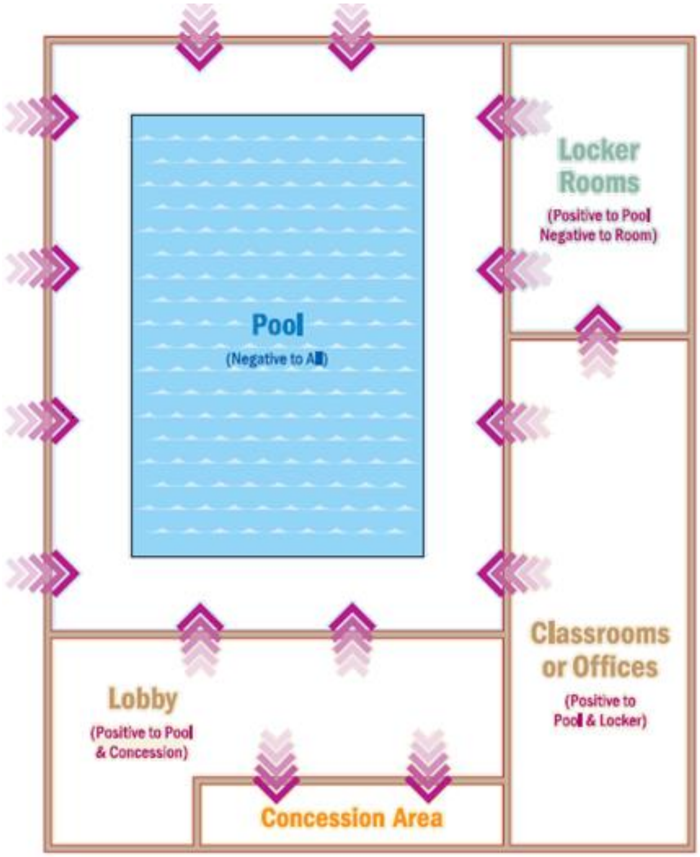



*Figure 1-8* Airflows through different part of a swimming facility showing an adequate pressure strategy (Aire, 2015).

## 1.2 Benchmarking

Benchmarking is the first step towards energy performance. Energy benchmarking provides a comparative evaluation of the energy performance of a building compared to other existing buildings. It allows comparison of buildings of similar functions and characteristics. Facility managers use benchmarking to situate the building's relative performance in comparison to its peers and then to identify potential for improvement. Energy Usage Intensity (EUI) in kWh/m²/year is the most commonly used indicator for benchmarking. (Priyadarsini, 2014)

One may analyze energy consumption figures very carefully. Energy consumption figures, given in various research papers, may be measured at different level of the building. Several performance metrics exist including source and site Energy Use Intensity (EUI).

### 1.2.1 Source and site EUI

The core of the matter comes from the types of energy used (electricity, natural gas, steam) and how they are distributed to the buildings. Buildings owners need to compare the same kind of energy use so that their building is properly benchmarked. The 1-100 ENERGY STAR benchmark advises using source energy. (EnergyStar, 2016a)

- **Site EUI –** this is the energy you pay for. It is the amount of heat and electricity consumed by a building as reflected in your utility bills. In other words, it is the delivered energy to the building taking into considerations all internal gains (occupancy, equipment…), consumer behaviors, and daylight.
- **Source Energy –** There are two forms the site energy can be supplied, either primary or secondary energy (EnergyStar, 2016a). The former is the raw fuel burned onsite to create heat and electricity, and the latter represents the heat or electricity created elsewhere from a raw fuel and purchased from the grid or received from a district heating network. To assess relative efficiencies of buildings with fluctuating proportions of primary and secondary form, it is necessary to convert these two types of energy into equivalent units of raw fuel consumed to generate that one unit of energy consumed on-site. The overall result represents the source energy and accounts for total energy use. This metric is preferable because reliable.

The calculations for source energy consumption use source-site ratio that convert primary and secondary energy to a total equivalent source energy. For primary energy, the ratio takes into account losses during storage, transport, and delivery of fuel to the building. While for secondary energy, it takes into account losses during production, transmission, and delivery to the site. To reach equitable comparisons between buildings, it appears relevant to use national source-site ratios. It then ensures that no specific buildings will be credited or penalized according to its utility provider.

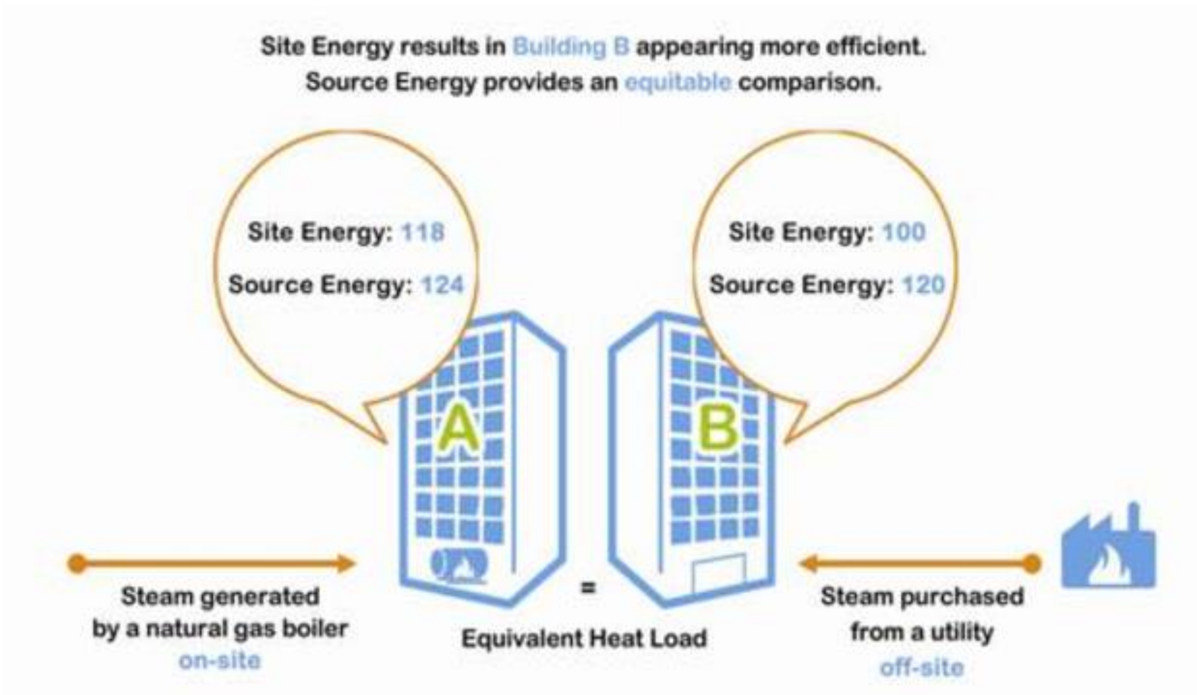



*Figure 1-9* Scheme explaining the site-to-source EUI conversion of primary and secondary energy. (EnergyStar, 2016a)

## 1.2.2 Typical USA and UK benchmark

The most commonly used benchmark in USA comes from the data used in the ENERGY STAR target finder and Portfolio Manager. ENERGY Star provide a target finder tool, which gives a national energy performance rating. It consists in an external benchmark that helps energy managers to assess how efficiently their buildings use energy, relative to similar buildings nationwide. The system rates each building on a 1-100 scale (EnergyStar, 2016b). A 50 indicates average energy performance while rating of 75 or better indicates top performance. This rating system was originally released for office buildings in 1999 but has expanded ever since to include the following space types.

- Bank/Financial Institutions
- Courthouses
- Data Centers
- Dormitories
- Hospitals
- Hotels
- Houses of Worship
- K-12 Schools
- Medical Offices
- Offices
- Retail Stores
- Supermarket
- Warehouses
- Wastewater Treatment Plants

As for the UK, Trust (2006) provides a typical benchmark for leisure facilities comprising a pool. Figures in Table 1-4 are for the whole building and not only the pool area. Also, typical values represent the median values and not the averages.

*Table 1-4* Median values of Energy Use Intensity (EUI) for swimming facilities in the UK. (Trust, 2006)

| Type | Good practice – fossil fuel (kWh/m²/yr) | Good practice – electricity (kWh/m²/yr) | Typical – fossil fuel (kWh/m²/yr) | Typical – electricity (kWh/m²/yr) |
|---|---|---|---|---|
| Centre with 25m Swimming pool | 573 | 152 | 1336 | 237 |
| Centre with leisure pool | 573 | 164 | 1321 | 258 |
| Combined center (with more than one type of facility) | 264 | 96 | 598 | 152 |

Gathering energy information of a representative sample of the building stock is very expensive and technically complex (Priyadarsini, 2014). To substitute from it, virtual data set using simulation for a range of energy parameters could constitute a reliable and time-saving alternative for the real building data collection required (Nikolaou, Skias, Kolokotsa, & Stavrakakis, 2009). Data sets can be created depending on which type of application of building and by running simulations for a range of energy parameters including building size, construction, operational characteristics, and climatic conditions.

## 1.2.3 Energy use intensity and allocation data

### *1.2.3.1 Hellenic swimming facilities*

In the framework of the European Commission SAVE program, Trianti-Stourna et al. (1998) paper "Energy conservation strategies for sports centers: Part B. Swimming pools", is a summary of energy audits performed in Hellenic swimming facilities to improve indoor conditions and optimize energy use. The focus was on retrofitting of existing building though proposed design and management principles could also be followed in new projects.

Hellenic pools are the second most popular type of sport facility (Trianti-Stourna et al., 1998) and their use are diverse: official races, diving, water polo and recreation. Generally, they include dressing rooms, shower, training rooms, lavatories, managerial offices, storage spaces, mechanical rooms, etc…

The paper gathered data from IECU (1994) of swimming facilities around Europe and Table 1-5 shows both their density and size in average. Belgium is the country with the highest density. The country has one 25m long swimming pool for every 23000 inhabitants. The data include all types of indoor sports facilities, used at all levels of competition, including professional, public, private and school facilities.

*Table 1-5* Swimming Pools population in Europe and average effective area of the installations. (Trianti-Stourna et al., 1998)

| Member state | Total number of swimming facilities | Effective area ($m^2$) per number of facilities |
|---|---|---|
| Denmark | 189 | 270 |
| France | 750 | 300 |
| Germany | 3168 | 316 |
| Greece | 29 | 300 |
| Ireland | 89 | 800 |
| Italy | 1489 | 300 |
| The Nederland | 300 | 400 |
| Portugal | 116 | 300 |
| Belgium | 1468 | 100 |
| Spain | 1025 | 233 |

| Member state | Total number of swimming facilities | Effective area (m²) per number of facilities |
|---|---|---|
| United Kingdom | 2900 | 261 |

According to Trianti-Stourna et al. (1998), energy consumptions is heavily dependent on location, type and use. Overall, the specific consumption for continental climate ranges from 600 to 6000 kWh/m². When the specific consumption is given in square meter of pool area:

- Mediterranean type climates have an average total energy consumption of about 4300 kWh/m² pool area
- Continental European zone consumption can be as high as 5200 kWh/m² pool area

Typically, energy cost represents the second biggest payment after labor cost and accounts for 30% of the total operating charge (Trust, 2006). Also, Trianti-Stourna et al. (1998) studied energy allocation and found out the two main sources of energy consumption were ventilation of the pool hall and heating of the pool water. Both account for nearly 80% of the total consumption. Figure 1-10 illustrates this energy allocation.

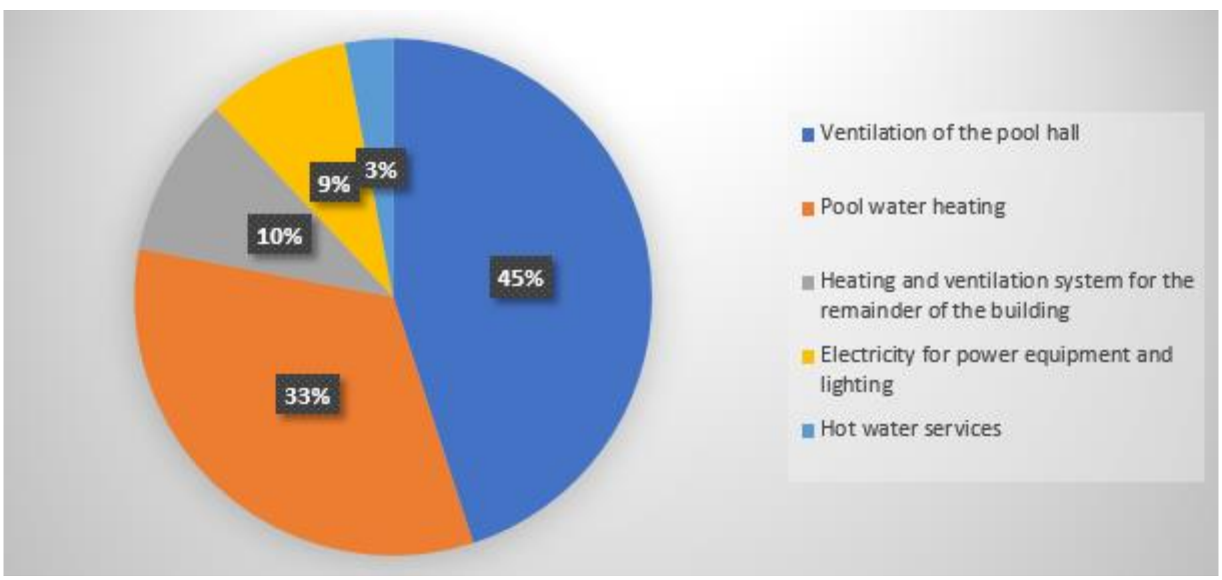



*Figure 1-10* Typical values from (Trianti-Stourna et al., 1998) showing energy allocation for European swimming facilities.

### *1.2.3.2 Aquatic Centers in Victoria*

Priyadarsini (2014) investigated energy performance of aquatic centers in Victoria. The findings are stated below.

The average proportion of gas and electricity is around 75% and 25% respectively. Space heating and pool-water heating constitute 20-30 per cent each of the gas consumption. In addition, transmission losses also constitute a significant proportion of the gas consumption. Majority of the electricity consumption is for pumps and fans. Six centers were included in the final analysis thanks to their complete and accurate data.

The total Energy Use Intensities (EUI) per square meter of floor area varies from 632 kWh/m² to 2247kWh/m² among the six facilities. Normalizing with the pool hall area, the average was found to be 3850 kWh/m². By converting energy usage per visit, the paper states EUI ranged from 8 to 17 kWh/visit.

### *1.2.3.3 Kampel's work*

Kampel's thesis (2015) gathered swimming facilities consumption statistics in European countries and compared it to the Norwegian ones. This comparison paved the way for identification of potential energy savings. Here is a non-exhaustive summary of his findings.

Concerning the high-use of energy, it is very difficult to draw a typical final energy consumption for it is highly dependent on the types (leisure center, sport, smaller pool to teach children), the variation in age, the technology and the characteristics of the swimming facilities. For instance, the specific energy need per $m^2$ could be very low because the pool stands for a small part of the total area. Nonetheless, Kampel (2015) summed up a large amount of information which is represented in the Figure 1-11. The blue columns are related to left axis which is the final annual energy consumption (FAEC) per $m^2$ of usable area, and the red ones to the right axis which is the FAEC but per $m^2$ of water surface. This broad amount of data must be used very carefully. Indeed, most studies do not include a sufficient amount of data to represent the majority of swimming facilities (Kampel, 2015). Also, it is not always clear whether the authors made a weather data correction before presenting the results. In the end, one must be very cautious with Figure 1-11.

Abrahamsen, Bergh, and Fedoryshyn (2013), called here statistics Norway, published a report indicating the final annual energy consumption (FAEC) of Norwegian facilities. However, the reliability is unsure since no additional information was given concerning type or location.

Another source from the “byggforskningsinstitutt” gathered data from Norwegian swimming facilities: the “Bade- og svømmeanlegg” book written by Bøhlerengen et al. (2004). Contrary to statistics Norway, this book seems more reliable and provide us with data from 27 swimming facilities.

Based on 17 facilities among the 475 ones in Sweden, Energimyndighet (2011) released similar results to statistics Norway: a 403.3 kWh/m² total area or 1302.7 kWh/m² water surface consumption per year. There is no further information about the selected swimming facilities except they are not pool schools or multipurpose facilities.

The Danish Institut (2016) stated that average energy consumption among Danish facilities varied from 2425 kW/m² pool area in 2006 to 2510 kWh/m² pool area in 2016. The source only specified data came from the biggest facilities.

According to Kampel (2015), in a report called “Use of Energy in Swimming Pools”, the British Amateur Swimming Association states the consumption of swimming pools. It differentiates between “typical” and “good” practice. There is, unfortunately, no information about the sample behind the numbers.

Values under the legend *Trianti-Stourna* are related to the study made in 1.2.3.1 about Hellenic facilities.

Finally, values under Finland legend are related to Saari, Sekki, and Saari (2008) paper. The particularity of these values is that they come from simulated data. Saari et al. (2008) ran simulations based on operating data from a swimming facility in Kirkkonummi. Surprisingly, authors did not say which software they used. The high consumption per square meter area is likely to be due to the seven saunas and the Turkish-type wet steam bath. They consume a lot of energy but do not add in water surface.

Data Figure 1-11 are very erratic and the graph does not allow for any specific conclusion. Further study must be conducted to investigate reasons of these relatively big value differences. However, it gives deeper knowledge about swimming facility consumptions. Values per square meter of available area lays way above the 145 kWh/m² TEK17 requirement for sport facilities (TEK17, 2017). As for the values in square meter per water surface, they are clearly dependent on the building layout: the more important the pool is for the building, the bigger the pool size is and the lower the value (in blue) is. This energy performance indicator (EPI in kWh/m² water surface) is better for benchmarking according to Kampel (2015) conclusions if no reliable data about number of visits is available and if they offer the same kind of services. The large differences indicate the huge potential of energy savings.

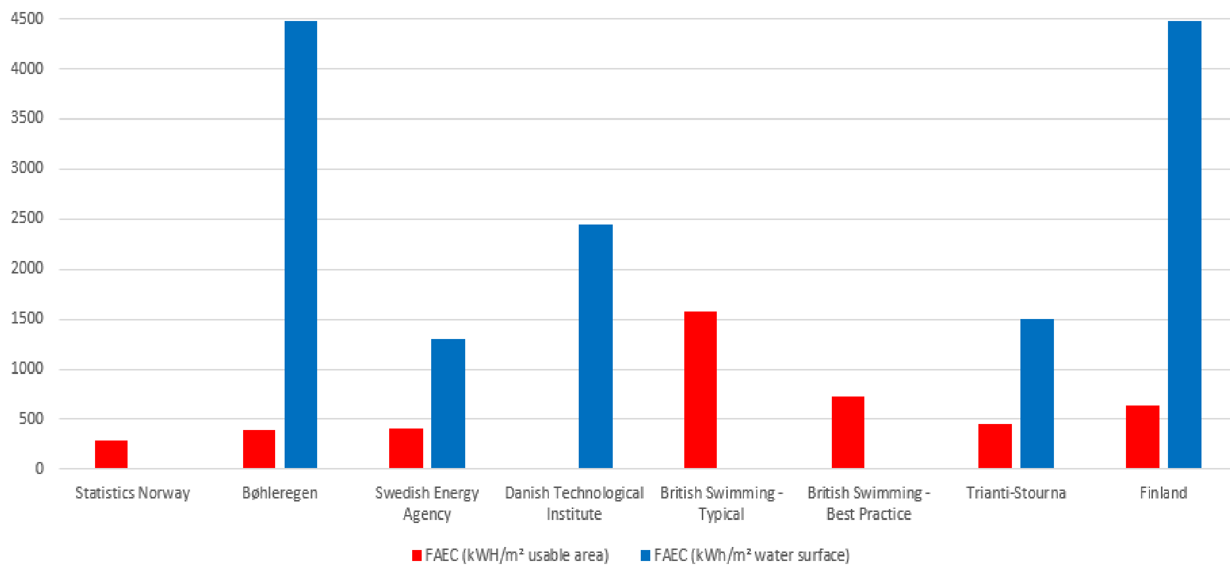



*Figure 1-11* Energy use in swimming facilities according to various publications. FAEC stands for Final Annual Energy Consumption. Values scaled to m² of usable area are in red while m² water surface ones are in blue. (Kampel, 2015)

In a second time, allocation of the energy use is crucial to define where savings can be done. To aim at reducing overall energy consumption, it is crucial to investigate the share of energy use in different subsystems. Same as for the energy use, W. Kampel made a summary from four articles that shows the distribution of energy use in swimming facilities. Unfortunately, it is impossible to generalize the share that goes into different subsystem since only little information has been published. However, Figure 1-12 is still interesting for it gives a general idea of energy allocation within swimming facilities. Rotating equipment refer to technical equipment, fans and pumps. Around 50% of the energy appears to go into the ventilation. Ventilation is covering space heating needs and air exchange as well as energy losses due to evaporation. Water heating is the second source of energy consumption, around 25%, and rotating equipment seems to be the third. To tackle energy consumption, one must therefore focus on these three aspects.

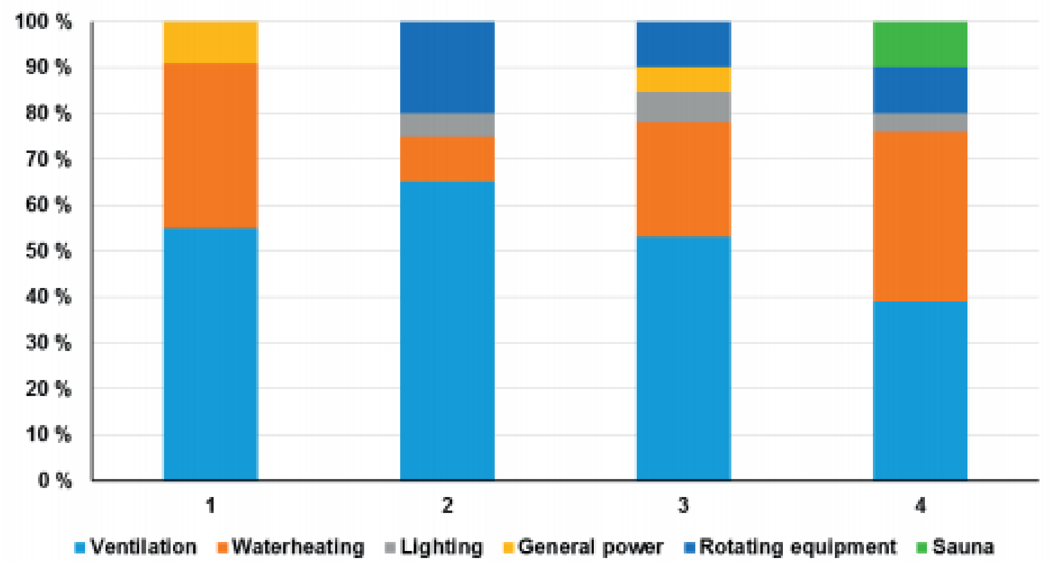



*Figure 1-12* Energy distribution to different subsystems from four cases in a publication by Trianti-Stourna et al. (1998).

# 2 Method

In this chapter, both the building and the pool hall behavior will be investigated using detailed dynamic simulation. Models will then be built and their design will fit standards so that the models imitates as much as possible a typical consumption. The idea is to run energy simulations throughout an entire year. Sensitivity analysis will be conducted afterwards. To ensure reliability of the results, it is very important to keep indoor conditions constant so that results between simulations can be comparable. Thus, two different models were built. The first, called “early stage model” gathers all features described further except for the ventilation strategy, the space-heating system, and the AHU which is a standard one. This model gives a reliable idea of general energy need within the building. The second model, called “detailed model” has a more complicated ventilation strategy and AHU. This model keeps indoor environment constant and is used for sensitivity analysis.

The chosen software to perform our study is IDA Indoor Climate and Energy (ICE). IDA-ICE is a dynamic simulation tool developed by the Swedish company EQUA Simulations AB and uses the principles of equation based modelling and Neutral Model Format (NMF). It is a model based multi-zone simulation. The underlying equations behind each object are transparent. The user can inspect how every component works by looking into the NMF code. Also, it is possible to log any variable into the calculations which turns out to be useful for detailed investigations.

IDA-ICE has been validated by numerous standard as e.g. ASHRAE 140, 2004, CEN Standard EN 15255 and 15265, CEN standard EN 13791.

Basically, there are three levels of complexity for the user to operate the program (Ole Ø. Smedegård, 2017):

1. **Wizard** – Fast and simple simulation of a single room to estimate heating and cooling loads.

2. **Standard** – Study of energy consumption and indoor climate of individual zones in a multi-zone building model using the available concepts and objects (windows, heating devices etc…)
3. **Advanced** – As the standard mode but includes the possibility to edit mathematical models.

## 2.1 Pool Extension in IDA-ICE

This extension allows the user to have open ice and water surfaces in a zone. According to EQUA (2013), the models account for both mass and heat transfer between the surface and the zone. Needed cooling and heating of the water in order to reach and maintain desired set-point temperatures are modeled.

The IDA-ICE pool model contains four separate internally connected component models. For solving complex phenomena, IDA-ICE uses equations and factors provided in the ASHRAE handbook. This pool model is connected both to the supply and exhaust water from the plant and the zone.

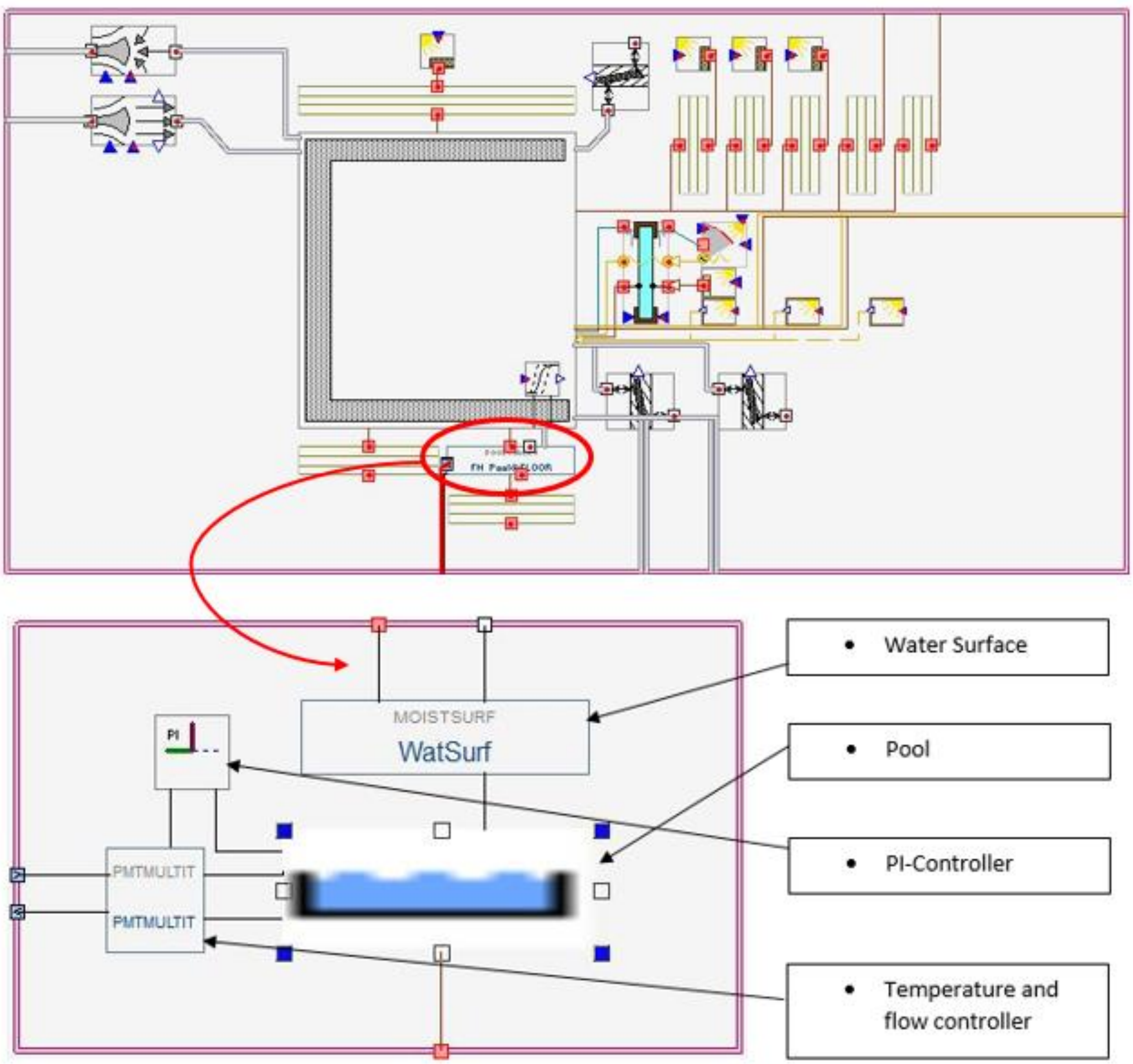



*Figure 2-1* Zone (above) and Pool (below) schematic views from IDA-ICE model.

1. **Water Surface** - The model is connected between pool water and the actual zone. Incorporated equations give sensible and latent heat transfer as well as the moist mass flow rate escaping the pool. Calculations are made according to ASHRAE formula. Basically, everything is based from the value of moist volume rate transfer leaving the pool. IDA-ICE's NMF code calls this variable HUMF. The value of water transfer by evaporation is given by equation 2-1 which corresponds exactly to equation 1-2 based on certain assumptions. In particular, IDA-ICE's

model deemed air velocity above the pool water to be around 0.10 m/s. From this moist transfer, the code deduces enthalpy transfers and energy transfers between this thin layer above the pool, the pool, and the zone.

$$HUMF = 4.10^{-5}\ AreaPool.(p_w - p_a).F_a \qquad \textit{2-1}$$

2. **Pool** - It represents the actual pool water and is linked to the water surface model and the controller where fresh heated water enters. The model calculates the heat flux due to conduction through the walls and the bottom, as well as the water temperature and humidity at the water surface.
3. **PI-Controller -** It monitors the pool temperature by comparing the actual pool temperature with its set-point. Depending on this comparison, it delivers a control signal between 0 and 1 to the temperature controller.
4. **Temperature and flow controller -** First, the circulated water is constant. However, according the signal from the PI-controller, an amount of fresh water comprised between a minimum and a maximum value is added to the supplied water. The supplied water is actually a mix between this fresh water and a fraction of exhaust water which is re-circulated. The fresh water is much warmer and comes from the boiler exhaust water of the plant. As a consequence, pool water temperature remains constant to the set-point throughout the year. Figure 2-2 shows pool temperature for an energy simulation when the set-point temperature of the water is 28°. The average value is 28.025°C with a standard deviation of 0.11°C.

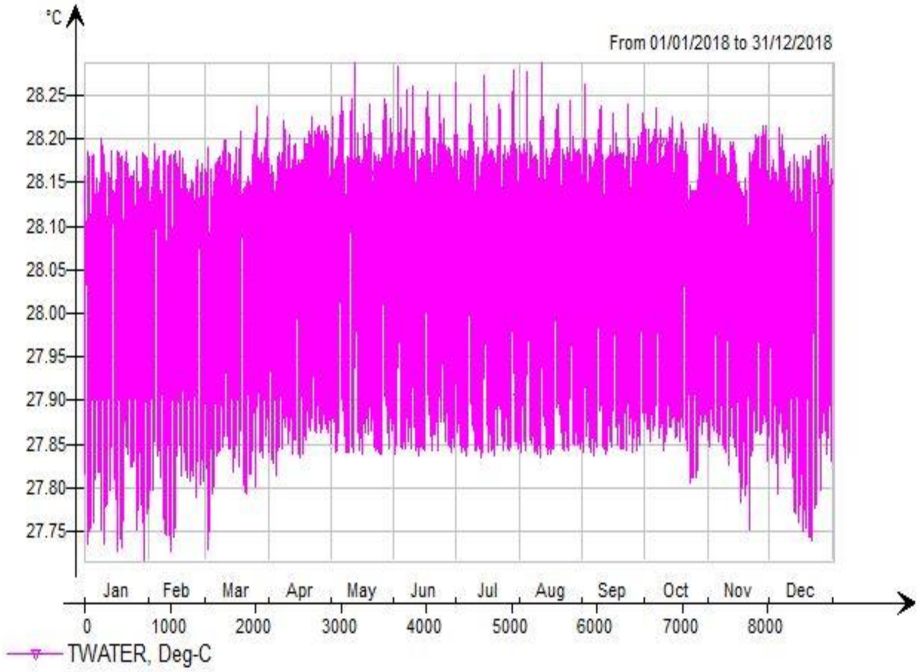



*Figure 2-2* Pool water temperature throughout the year as part of an IDA-ICE simulation.

Figure 2-3 and equations below illustrate how the model in IDA-ICE works. All values on the figure are mass flow rates in kg/s. These values are discussed in 2.2.5.

$$M_{In} = M_{out} = Constant \tag{2-2}$$

$$M_{supply} = M_{max} * ControlValuePI + M_{min} * (1 - ControlValuePI) \tag{2-3}$$

$$T_{In} = \frac{T_{supply} * M_{supply} + T_{poolwater} * M_{ret}}{M_{In}} \tag{2-4}$$


*Figure 2-3* Schematic illustration on how IDA-ICE model of the pool works.

## 2.2 Model's Characteristic

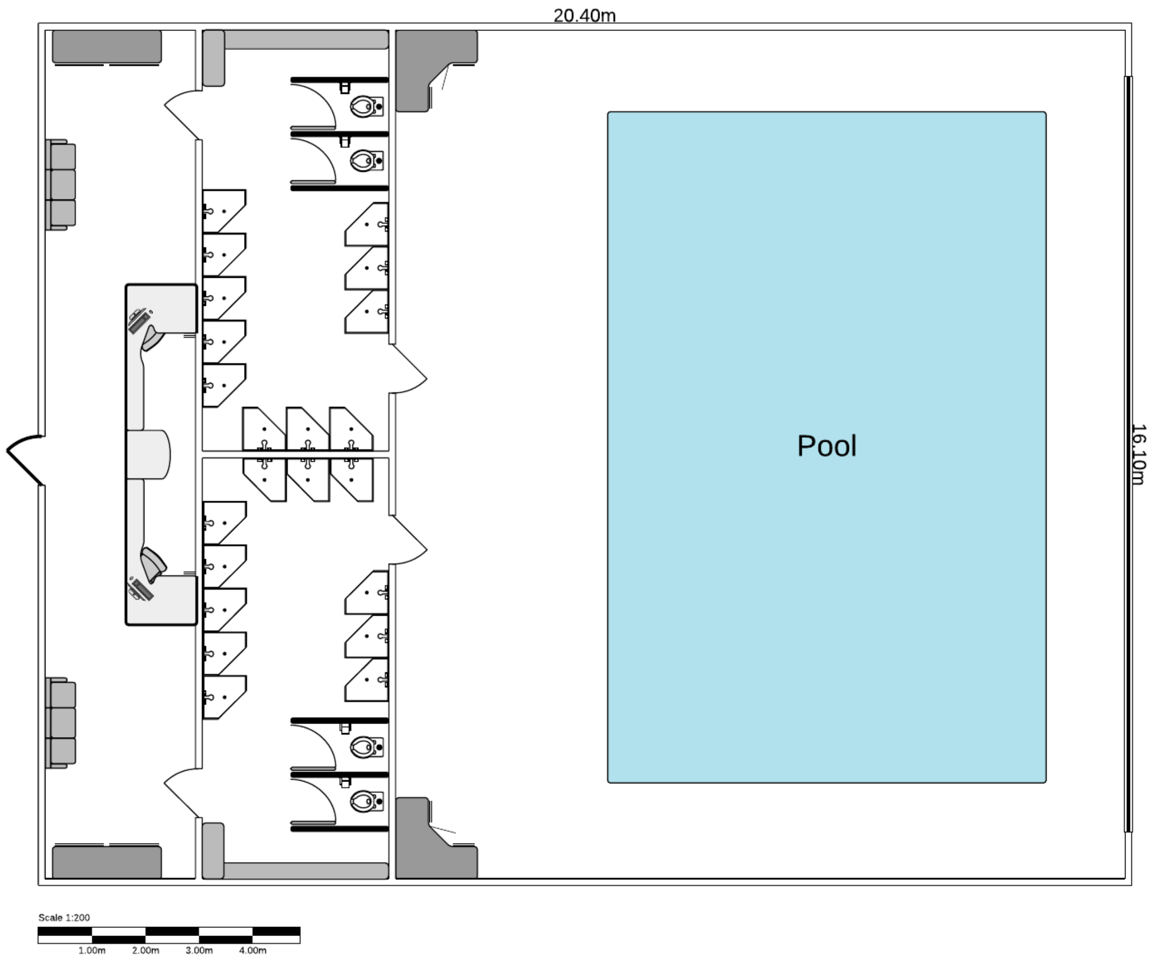



*Figure 2-4* Schematic Plan of the model.

### 2.2.1 Location and Climate

The chosen location is Oslo, Norway. Oslo is a reference location and IDA-ICE simulation tool's database contains an ASHRAE IWEC 2 weather file for this location. This file corresponds to a station in Oslo Gardermoen: Latitude 60.2°N and longitude 11.083°E. The weather file includes hourly values for air dry-bulb temperature, relative humidity, wind speed, cloudiness, and both diffuse and direct normal radiation.

To determine the wind's influence on our building, IDA asked for two additional coefficients to complete the weather file: $a_0$ $and$ $a_{exp}$. These two coefficients are part of equation 2-5 where U (m/s) is the wind velocity at a height H (m). $U_{weatherfile}$ is the wind speed at a specific height $H_{ref}$, usually 10 meters. The chosen wind profile is urban and IDA-ICE gives the default values 0.67 and 0.25 for $a_0$ and $a_{exp}$ respectively.

$$U = U_{weatherfile} * a_0 * \frac{H^{a_{exp}}}{H_{ref}} \tag{2-5}$$

However, the model does not take into wind driven forces without any further information concerning pressure coefficient. Pressure coefficients are specific to each external surface and are depending on building shape and surroundings. There is no need to calculate them since IDA-ICE proposes an auto-fill by barely picking either “exposed”, “semi-exposed”, or “sheltered”. The “semi-exposed” auto-fill is chosen. The wind has an even greater influence on the model behavior when it comes to natural ventilation strategies. Nevertheless, the model shall not comprise natural ventilation and the wind shall have a rather low impact.

As for the orientation, the building is oriented toward north (0° of orientation in IDA-ICE see Figure 2-5).

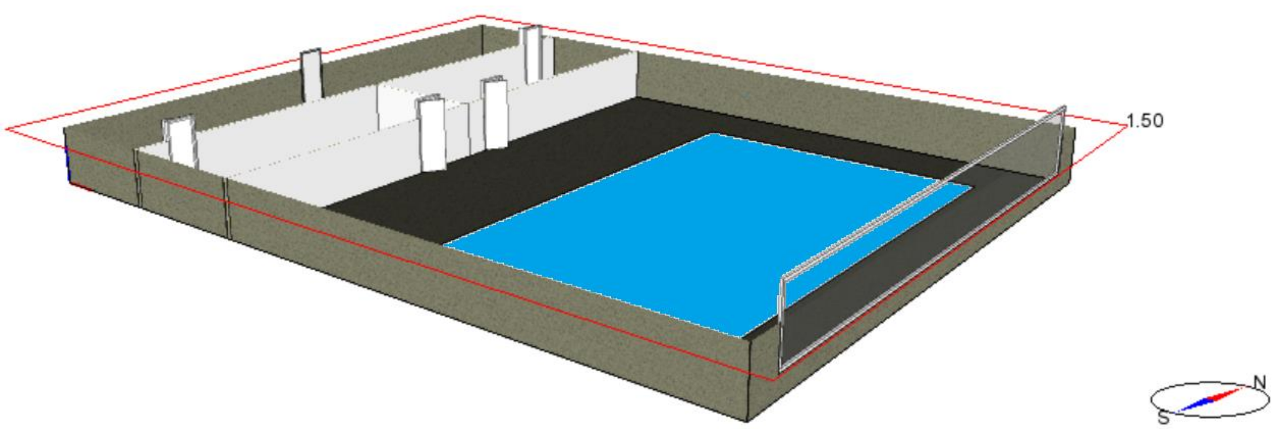



*Figure 2-5* 3D view of the model in IDA-ICE. The building is section cut from Z=1,5m. The swimming pool is represented in blue.

## 2.2.2 Building Geometry

To be realistic, the model should not include solely a swimming hall but the showers and other neutral area as well. The model includes a standard teaching pool which is usually 100 m ².

Standard areas are given point 2.2.5 in a Norwegian Swimming Federation document (NSF, 2006).

*Table 2-1* Typical Area of different type of pool. (NSF, 2006)

| Type of swimming Pool | Typical Area (m x m) |
|---|---|
| Competition Pool | 50 x 25 |
| Diving Pool | 25 x 20 |
| Teaching/therapy Pool | 12,5 x 8 |

The document from NSF (2006) gives typical values for showers' dimensions. Martin Øen thesis, about energy need and efficiency in pool facilities, uses these dimensions in a SIMIEN model (Øen, 2010). It also investigates a 100 m² pool. Therefore, and as an opportunity to compare both model afterwards, the building geometry will incorporate the same showers' geometry. Finally, the model has an entrance zone. Figure 2-6 illustrates the building geometry. Building's layout is straightforward and overall size is pretty small and compact. The pool fraction of floor area represents a substantial part of the total area.

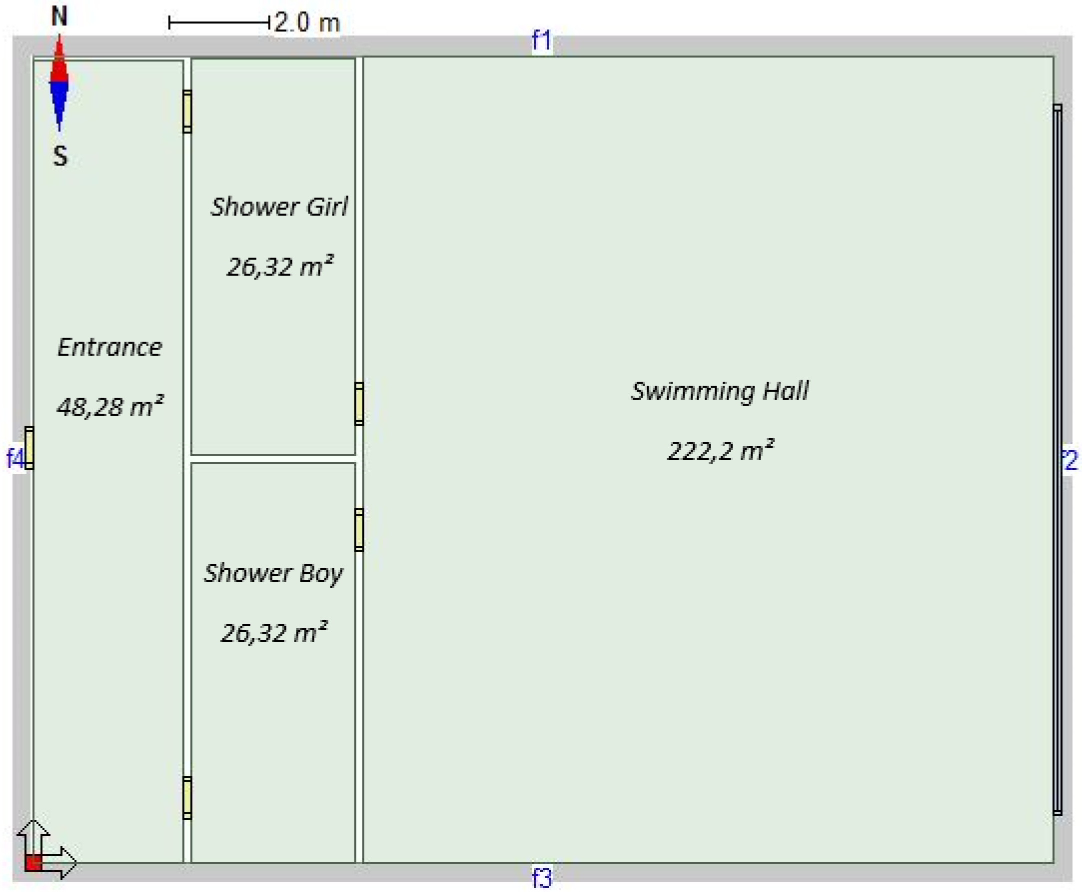



*Figure 2-6* Floor plan of the model in IDA-ICE.

## 2.2.3 Structure

The model should represent a standard structure. The idea is to choose minimum allowed insulation values according to Norwegian standards and then measure their influence by improving the thermal performance of the structure in further study.

IDA-ICE has a layer-based approach. For instance, the user can choose how many layers the walls will be formed of and which material is attributed to each layer. Three parameters define the material: thermal conductivity (W/mK), density (kg/m³), and specific heat (J/kgK). Obviously, there are several different layers combinations to achieve a certain U-value and then a certain envelope performance. However, when one changes the set of material composing the structure, it modifies building's thermal mass and then its thermal inertia. Higher inertia means the building is less influenced by sudden change in boundary conditions e.g. external environment. The building becomes less responsive to change.

In our case, only one type of insulation material is chosen: light insulation. It is a material by default in IDA-ICE's database. It has a thermal conductivity of 0.036 W/mK, a density of 20 kg/m³, and a specific heat of 750 J/kgK. Table 2-2 describes insulation characteristics of the model in comparison with TEK17 and NS3700 passive house standard minimum requirements (NS3700, 2013).

The only window in the IDA-ICE model is situated at the east façade (see Figure 2-5). It has an area of 25.38 m². As for external doors, only the entrance door is built and represents 1.6 m². TEK17 requires the overall windows and door area to be below 25% of the total floor area. In this case, the total heated area is 328.4 m² and then windows and door areas together stand for $\frac{1.6m^2+25.38m^2}{328.4m^2} = 8.21\%$ so the requirement is plainly fulfilled. The type of window for the base model is a Pilkington Artic Blue (6ab-15Ar-S(3)6). This simplified window model has been chosen for its U-value close to TEK17 requirement. It has a solar heat gain coefficient of g=0.33 which describe how much solar energy is being transmitted to the building zone.

The minimum requirement for infiltration is 1.5 air change per hour (ach) in TEK17 (2017). However, an article from Holøs and Relander (2009) conducted studies about airtightness in building using blower door method. It shows that a $n_{50}$ coefficient of 1 could be easily reachable even with common material and craftsmen without any specific training in airtight building. Therefore, a starting value of one air change per hour due to infiltration was set.

*Table 2-2* Model's insulation values and minimum requirements from TEK17 and passive house standard NS3130

| | Model | TEK17 minimum requirements | Passive house minimum requirements | Units |
|---|---|---|---|---|
| Walls | 0.1586 | <0.22 | <0.10 - 0.12 | W/m²K |
| Roof | 0.1258 | <0.18 | <0.08-0.09 | W/m²K |
| Floor | 0.1457 | <0.18 | <0.08 | W/m²K |
| Windows | 1.1 | <1.2 | <0.8 | W/m²K |
| Doors | 1.085 | <1.2 | <0.8 | W/m²K |

| | Model | TEK17 minimum requirements | Passive house minimum requirements | Units |
|---|---|---|---|---|
| Normalized Thermal Bridge | 0.1 | <0.05 – 0.07* | <0.03 | W/m²K |
| Infiltration (n50 coefficient) | 1** | <1.5 | <0.6 | $h^{-1}$ (air change per hour) |

*these are requirements for residential buildings. Table B.3 in NS3031 (2014) stances a 0.1 W/m²K normalized thermal bridge for concrete structure.

**(Holøs & Relander, 2009). See Above.

## 2.2.4 Internal Gains

### *2.2.4.1 Occupancy*

There are standards in Norway describing typical occupancy. However, swimming facilities do not especially fit into one category. Sport facilities would be the closest one, but the model will rely on a paper from the Norwegian technical pool association formed of biggest pool installations companies ("Norsk bassengbad teknisk forening" NBTF). Chapter 3 "kapasitetsberegning" stances that every person should have 4.5 m² of space in deep pool. Then, the occupancy for the 100 m² pool amounts to 22.22 persons. This number is the base of the reasoning for building occupancy.

Except during public holydays and on week-ends, the swimming pool is open from 8 a.m. to 4 p.m. Every two hours 4 phases described in APPENDIX take place. A group of 22.22 people occupies the pool hall and uses the pool before leaving the building while another group comes in. Detailed occupancies in each room can be also found in APPENDIX. Below is a detailed description of what happens:

1. **Phase 1** – Let's suppose a group, which will be called group 1, has already been using the pool from 10 a.m. The building occupancy in the building is 0 except in the pool hall where it amounts to 22.22.
2. **Phase 2 -** At 11h45 a.m., group 1 stops using the pool and enters in the two showers (boys and girls) while another group, which has the same size as group 1

and which will be called group 2, enters in the building and stand in the entrance room. The occupancy in the pool hall goes down to 0 while occupancy in the entrance becomes 22.22, and the occupancy in each shower becomes 11.11. This phase lasts 15 minutes.

3. **Phase 3** – At 12 noon, group 1 has finished showering and moves to the entrance. In the meantime, group 2 has entered in the showers. The different occupancies remain the same as in the previous phase, the groups have solely switched place.
4. **Phase 4** – At 12h15 noon, group 1 has left the building and group 2 starts using the pool hall. Occupancy has become the same as in phase 1.

To get through phase 1 to phase 4 usually lasts 30 minutes. However, when the pool opens and closes, phase 2 and 3 take place with only one group. To simplify the model and only use one schedule for both entrance and showers, one assumption is made. At the opening, one group comes in the entrance and go to the showers in solely 15 minutes while being supposedly in the showers and in the entrance simultaneously. This does not change neither the overall time and load of occupancy and simplifies the schedules (see Appendix 7.1). Generally, people occupying showers and entrance have an activity level of 1 met and the ones occupying the pool hall, 2 met.

### *2.2.4.2 Light and Equipment*

NS3031 (2014) gives normalized yearly electricity consumption due to lighting. This consumption depends on the type of building. The model takes the value in table A.10 corresponding to sport center: 21 kWh/m² per year. Since operation hour of the building occurs between 8 a.m. and 4 p.m. during week days, it amounts to 2080 operation hours during which light is lit. Thus, it ends up to $\frac{21\, kWh/m^2 year}{2080\, h} = 10.1\, W/m^2$. Every zone receives the corresponding amount according to their area.

No special equipment was added to the model and the internal gain related to equipment was set to 0 in IDA-ICE.

## 2.2.5 Pool

The pool heavily influences indoor environment through heat and mass transfer at the pool surface. Indeed, the evaporation process removes heat from direct surroundings and most of this latent heat comes from the water and released as vapor in the indoor atmosphere.

Pool dimension, temperature of the water, and air temperature play a crucial role. The bigger the swimming pool area, the more evaporation happens. As for temperatures, it is in reality the vapor pressure difference with the saturated pressure that directly determines evaporation. However, temperature and evaporation are connected since the saturation pressure increases with the temperature (see 1.1.3). The higher the temperature, the more evaporation will happen, but the lower relative humidity is for a fixed amount of water vapor in the air. In the model, the pool is 100 m², pool temperature set-point is 28°C, and air temperature set-point is 2°C above water temperature: 30°C.

Apart from pool dimensions and temperature, evaporation hinges on the activity factor. Indeed, with agitation of the water surface, liquid molecules break free from the surface tension more easily and evaporation rate increases. Besides, the activity factor appears in the ASHRAE formula for evaporation (see equation 1-2) which is the formula taken into the NMF code of IDA-ICE. Table 1-1 gives typical values of this factor. In this case, the activity factor is set to 1 during occupation hours. It is very important to notice that outside occupancy hours, this factor is set to 0.5.

Another parameter related to the pool model is graywater. What swimming pools usually do is to flush a certain amount of water during the night to clean the filters. The value is set to ensure there is enough fresh water supply every day. There are two different parameters when it comes to water supply in swimming pool. Similarly to air in ventilation systems, some water is extracted from the pool and is being circulated through filters, cleaning systems, and re-heated before being reinserted to the pool, while some water is thrown away. Some fresh water from the city grid needs to be supplied to compensate for the pool greywater. Regulations give typical mass flow rates for water circulation and fresh water supply. At least 2 m³/h per user of water should circulates (*Retningslinjer for vannbehandling i offentlige bassengbad*, 2000). It ends up to a value of the IDA-ICE variable $M_{CIRC}$ of 12.34 kg/s. Oddly enough, if a mathematical model is not created in the advanced level of the simulation tab, this parameter cannot be modified. Whereas in the advanced model, there is no information about the amount of circulated water and it seems the parameter is calculated automatically. Besides, at least 30 liters of fresh water per day per user should be supplied when the temperature is below 34°C (*Retningslinjer for vannbehandling i offentlige bassengbad*, 2000). Given pool occupancies, it ends up to 2.67 m³ per day and then 0.7417 kg/s of greywater mass flow rate between 3 a.m. and 4 a.m.

## 2.2.6 Domestic Hot Water (DHW)

Domestic hot water represents a considerable use of energy in swimming facilities. Indeed, all users take showers before and after entering the pool. Also, they surely use toilets and water sinks. In the end, the total need in fresh city water that must be heated up is high, and fresh water need to cover pool water losses and renewal is not considered in DHW.

Similarly to assumptions made in the thesis from Øen (2010), showers consumption is 8 L/min and each user takes a 6 minutes long shower. Thus, a standard shower accounts for 48 liters. The value asked by IDA-ICE should be in liter per occupant per day. Gathering the two showers, the occupancy amounts to 22.22 persons. However, four groups of 22.22 persons will be using the showers daily. Instead of 48 liters per person per day, 192 liters per person per day should be the DHW consumption. Yet, 192 liters per person per day represents the actual shower consumption. When the focus is on energy consumption, less than 192 liters of fresh water should be heated up to an exhaust temperature of about 50-60°C to get the needed value but at shower water temperature. Assuming showers provide 40°C water and knowing that city water is 5°C in IDA-ICE, the DHW demand amounts to 134.4 liters per person per day. Equation 2-6 explains how the value is found. By mixing water at different temperature, the total enthalpy is conserved. Dividing by the specific water enthalpy gives equation 2-6.

Thus, 134.4 liters per person per day is the value set in the IDA-ICE model because it represents the actual energy consumption for DHW.

$$DHW_{demand} * 55°C + (192 - DHW_{demand}) * 5°C = 192 * 40°C \tag{2-6}$$

$$DHW_{demand} = 134.4\ L/person.day \tag{2-7}$$

## 2.2.7 Indoor Air Environment Set-points

### *2.2.7.1 Pool water temperature*

Table 1-2 in section 1.1.4.1 gives typical water temperatures according to the type of pool. The model will follow this ASHRAE standard and a 28°C pool water temperature is chosen. This temperature is typical to e.g. recreational and competition pools.

### *2.2.7.2 Air temperature and Humidity*

The choice of which kind of indoor environment the building is to reach remains crucial to energy consumption. Standards describe, for example, temperature set-points depending on the building's type. However, swimming pools category does not exist in these standards. This type of facilities can be included in sport building or hospital category depending on which aspect the focus is put. For instance, when it comes to water consumption, swimming facilities fits more to the hospital category. Table 2-3 sums up the set-points temperature according to NS3031 (2014). The model will consider these temperatures for the entrance and the showers. A 19°C set-point temperature seemed too cold for showers that will rather follow the hospital requirement of 21°C which more realistic. To simplify the model, the entrance temperature set-point will also be 21°C.

When it comes to the swimming hall, temperature set-point refers to section 1.1.4.1 where it has been discussed. Since the pool water temperature is 28°C, the set-point for air temperature in the hall becomes 30°C, 2°C above the water temperature.

*Table 2-3* Extract from table A.9 (NS3031, 2014) giving indoor set-point temperatures depending on the building's type. Included is the two categories a swimming pool can be filled in.

| Building Category | Set-point Temperature for Heating (°C) | | Set-point for Cooling (°C) |
|---|---|---|---|
| | During Operation | Outside Operation | |
| Hospital/nursing | 21 | 19 | 24 |
| Sport Centers | 19 | 17 | 24 |

Standards do not give precise range for humidity. Not considering the swimming hall, the usual range of 40%-60% is considered as the set-point. In the early stage model, there is no control on humidity though.

In real operation in swimming hall, indoor air humidity set-point usually depends on the outside air temperature. In the early stage simulation, no humidity control will be installed but for the detailed model, a constant humidity control will be considered. As seen in section 1.1.4.1, the advised range is 50 to 60% relative humidity. The set-point then become 50%.

## 2.2.8 Ventilation

Similarly to all building types, ventilation achieves a desired air quality. In addition to keeping the air clean from contaminants, ventilation strategies implemented in swimming facilities keep humidity below a certain level. This indoor humidity set-point can vary depending on the outside temperature. Internal load and moist transfer from the pool are much higher during operation hours. What is usually done in swimming halls is to implement a CAV ventilation with two thresholds: one airflow rate during operation hour and a lower airflow rate outside occupancy. For instance, it is the case for Jøa swimming pool operation (see Appendix 7.2). For ventilation rate for the detailed model, see Appendix 7.3.

### *2.2.8.1 Ventilation rates*

The most common way to set technical requirements related to ventilation is to fulfill the pre-accepted quantitative performance found in the guideline in the Norwegian building code (TEK17, 2017). Air change rate grants an acceptable level of indoor air quality. Relative humidity, concentration of $CO_2$ or other pollutants are kept between suitable ranges thanks to the ventilation system. Satisfactory fresh air supply is assessed thanks to three different conditions, A, B, C. The minimum requirement for ventilation design and operation is the largest value between (A+B) and C. Values given by TEK17 (2017) are shown in Table 2-4. The list of special equipment, process, or activities, for requirement C, is not exhaustive.

A. Ventilation should remove pollutants from persons (through his/her breathing, perspiration…).

B. Ventilation should remove pollution from materials, products and installations (material load).
C. Ventilation should remove pollution from any special activities and processes.

*Table 2-4* Minimum requirements for fresh air change in the Norwegian building code (TEK17, 2017).

| Column Head | Minimum Fresh air supply requirement | Comments |
|---|---|---|
| Requirement A | 26 m³/h per person | Persons with light activities (1 met). |
| Requirement B | 2.5 m³/h per m² of floor surface | In use |
| | 0.7 m³/h per m² of floor surface | Not in use |
| Requirement C | 54 m³/h per shower | bathroom |
| | 36 m³/h per toilet/urinal | Toilet |
| | 2.5 m³/h per m² of floor area | basement |

No special activities take place in the entrance, ventilation rate is then set according to A+B requirements. The floor area is 48.28 m². With the occupancy of 22.22 people, the minimum requirement becomes 2.75 L/s.m². In the sport facility category, NS3031 (2014) sets a minimum requirement of 2.22 L/s.m².

Since showers are deemed as special processes that need special ventilation requirement, the model follows the minimum requirement C of 54 m³/h per shower given by TEK17 for the two bathrooms. For each room, there are eleven showers. Eleven showers seem to be completely fair with respect to the area. Indeed, standard showers dimensions are 0.9m x 0.9m (Bøhlerengen et al., 2004) or approximately 1 m² size. In total, the minimum requirement ends up to 6.29 L/s.m² which is the value used in the model.

In the case of the swimming hall, it becomes more complex to find average ventilation rate. There is no precise requirement for swimming pool in TEK17. By setting ventilation rate according to requirements in Table 2-4, the model would not reflect the reality. Even the minimum requirement from NS3031 (2014) of 2.22 L/s.m² for sport facility would not be sufficient enough. In the natatorium section chapter 5, Ashrae (2015) recommends a supply air delivery rate of 4 ach to 6 ach for recreational pools and 6 ach to 8 ach for competition pools with spectators. Our base model fits more in the recreational type so a middle ground is found by setting 5 ach in the model. This is equivalent to 3.612 L/s.m². In the case the pool is not occupied, the minimum value of 2.22 L/s.m² from NS3031 (2014) is set.

### *2.2.8.2 Negative air pressure strategy*

As mentioned in 1.1.4.5, it is, in swimming pools, common practice to implement negative air pressure to reduce the likelihood of hot and moist air going beyond the structure and condenses with subsequent moisture damage. A 5 Pa pressure difference between the swimming hall and the other rooms (swimming hall having a lower pressure) would be a minimal requirement to implement this strategy as said section 5.2.6 (Øen, 2010). Given the complexity of the system, there is no simple calculation method to find the necessary difference in airflow needed to maintain this pressure difference. In his thesis, supervisor of Martin Øen (2010) claimed it is common practice to implement a 300 m³/h difference in airflow rate and then set the system more precisely during operation.

Consequently, supply airflow rate in the entrance is raised by +300 m³/h. Final airflow rates become 3.95 L/s.m² and 2.22 L/s.m² for supply and exhaust respectively. As for the swimming hall, supply and return airflow rates during operation become 3.612 L/s.m² and 3.988 L/s.m² respectively. Outside operation, supply and return airflow rates are 2.22 L/s.m² and 2.596 L/s.m² respectively. Detailed graphs are shown in Appendix 7.3 and results to this strategy are discussed in 3.2.

## 2.2.9 Air Handling Unit

First, in the early stage simulation, the AHU used is the standard one given by default in IDA-ICE with some modifications explained in the following section 2.2.9.1.

Second, in a more detailed model, two AHUs will be set up. The first is dedicated to the pool hall and the second for the other rooms (showers and entrance). Their features are explained below in both 2.2.9.2 and 2.2.9.3.

### *2.2.9.1 Standard AHU for an early stage simulation*

The AHU has constant air volume system (CAV). Volume rates are the ones specified in section 2.2.8.1. However, volume rates for the hall are kept constant as if the pool was continuously occupied. Then, the hall receives constantly 3.612 L/s.m² of fresh air.

It is worthy to note that fans in IDA-ICE have an ideal pressure control and a constant efficiency according to the specific fan power. Here, SFP equals 1.5 kW/(m³/s). They also give an extra 1°C to the air during operation. Fans are always on. Table B.6 in NS3031 (2014) advises a 0.70 efficiency for rotary wheel and 0.60 for cross-flow. Since a rotary wheel is assumed installed in the AHU, efficiency is set to 0.7. One can notice in Figure 2-7 one hydronic heating coil and one hydronic cooling coil installed right after the rotary wheel. Heating coil has a 100% efficiency while the cooling coil has a 0% efficiency. It means the latter is set off. Indeed, temperature set-point is chosen 16°C constant which means that the heat recovery, the two hydronic coils are used whether the inlet temperature for each is not 16°C. It is not worth to cool down the air when outside air temperature is over 16°C. There is no humidity control whatsoever.

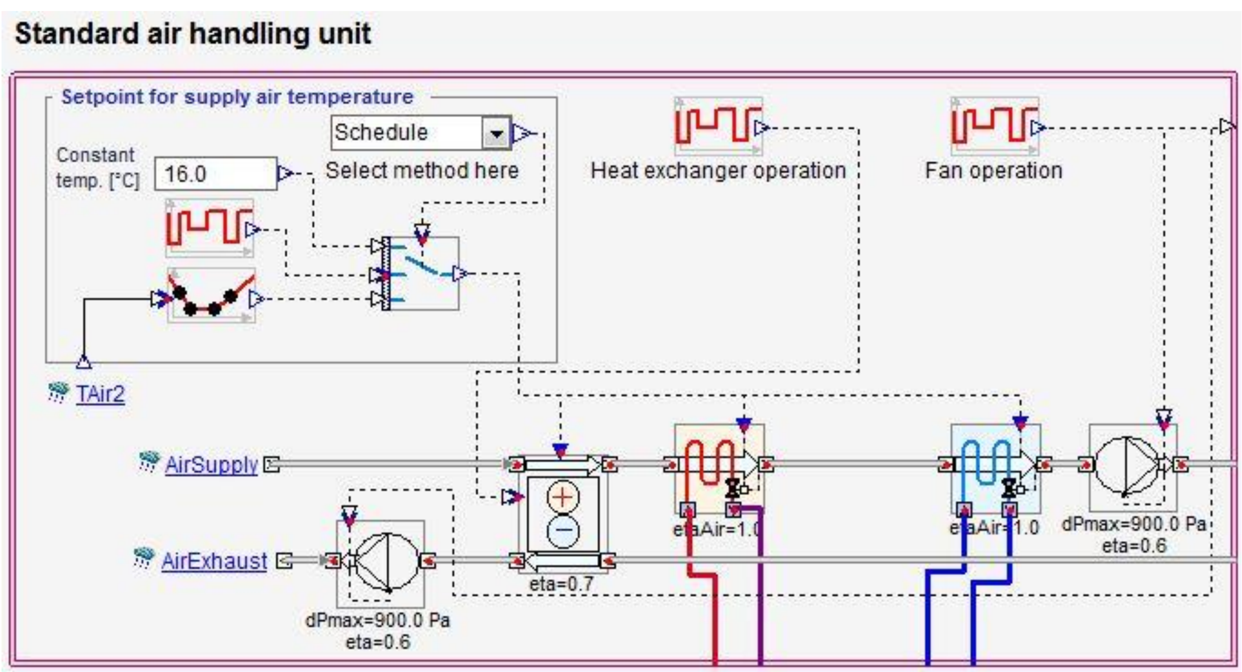



*Figure 2-7* Screenshot of the standard Air Handling Unit in IDA-ICE.

### *2.2.9.2 Actual AHU for the Entrance and the Showers*

For the detailed model, the AHU becomes much more complicated. It includes now two kinds of control: humidity and temperature. The idea is to keep indoor air conditions constant. Except for the new components for temperature and humidity control, all other components work the same way as described in the previous section 2.2.9.1.

Since IDA-ICE assumes a complete and perfect mixing inside the room, it means temperature and humidity are uniform inside every zone. Then, by controlling the exhaust air temperature and humidity, temperature and humidity inside the zone are controlled. Also, this AHU is specific for the entrance and showers ventilation only Figure 2-8 illustrates this AHU.

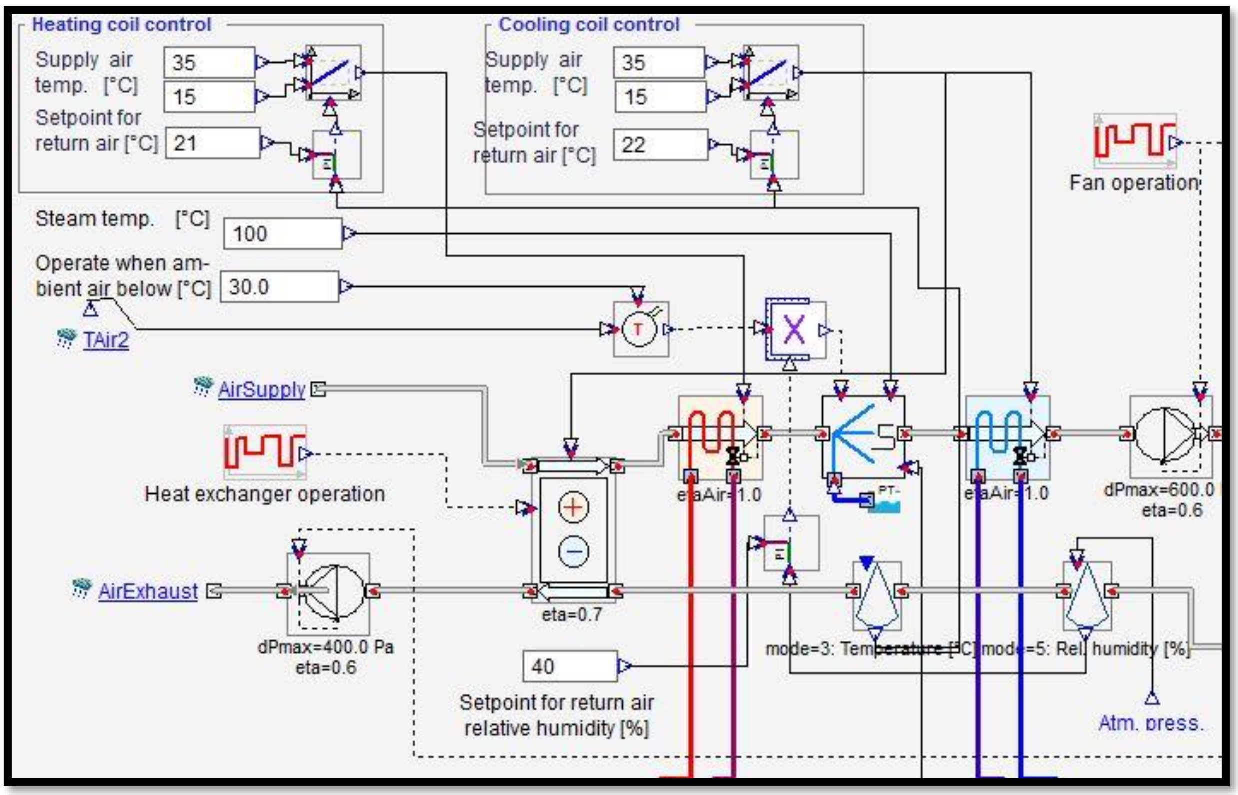



*Figure 2-8* Schematic view extracted from IDA-ICE of the AHU in charge of showers and entrance.

#### *2.2.9.2.1 Temperature Control*

A sensor installed in the exhaust duct gives in real time the exhaust temperature corresponding to the indoor temperature. This value is the bottom line of the control. On the top appears both the control for the heating coil and the control for the cooling coil. Both are relatively similar. Figure 2-9 is a general illustration of the temperature control implemented to the pool hall (or the other rooms).

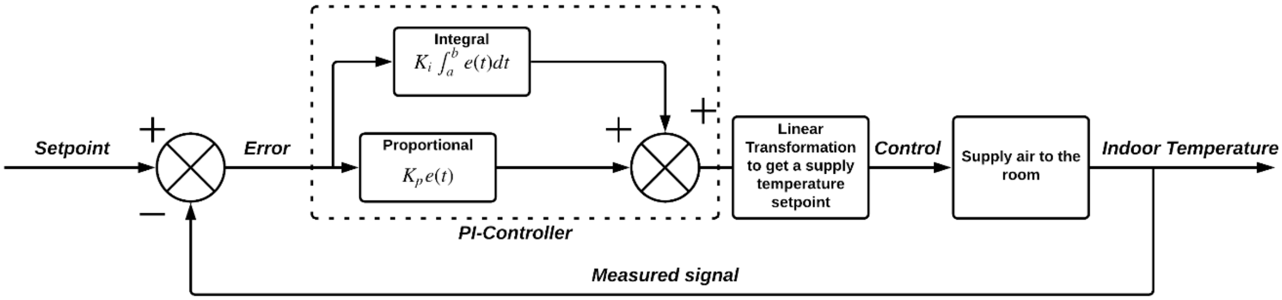



*Figure 2-9* Overview of temperature control of the indoor temperature by the AHUs.

For the heating coil control, the PI-controller delivers a value comprised between 0 and 1 depending on the difference between the set-point temperature of 21°C and the exhaust temperature. In general, a PI-controller measures an error ε (t), which is a difference between a signal and its set-point. It then applies a correction based on proportional and integral terms. If the exhaust temperature is more than 21°C, then the PI output is 0. If not, the output value is positive. After a certain threshold of a temperature difference (26°C minus the exhaust temperature), the output is 1 even though the exhaust temperature keeps decreasing. Furthermore, the output from the PI-controller is transformed into a temperature comprised between 15°C and 35°C (see Figure 2-8). This transformation is a linear relation between [0 1] and [15 35]: 0 corresponds to 15 and 1 corresponds to 35. Finally, the [0 35] output stands for the temperature set-point after the heating coil.

For the cooling coil, the control is relatively similar except that the PI-controller is a bit modified. Now, what gives the PI output between 0 and 1 is the difference between 22°C and the exhaust temperature (see Figure 2-8). In other words, 22°C is now the new threshold and is set so that cooling does not occur before 22°C. It is also very important to check whether both coils are used simultaneously which would entails useless energy consumptions. Table 2-5 displays temperature set-points for supply air depending on the measure of exhaust ar. These values have been validated by simulations conducted with the parametric run function (see section 3.3.1). The summary of these simulations is in Appendix 7.4. It shows that PI parameters are set so that the response to any difference with the set-point, even tiny ones, gives a clear-cut output: either 0 or 1.

The choice of the maximum and minimum temperature set-points is crucial to get a well-working AHU. Indeed, since the air volume rate is constant, if the maximum air supply temperature (here 35°C) is too low, then the AHU will not be powerful enough to counteract thermal losses. The same reasoning is relevant for a low enough minim temperature set-point (here 15°).

*Table 2-5* Output characteristics of AHU for temperature control.

| Exhaust temperature | PI-controller Output for heating | Heating Coil Set-point (°C) | PI-controller Output for Cooling | Cooling Coil Set-point (°C) | Heating Coil | Cooling Coil |
|---|---|---|---|---|---|---|
| <20.999 | 1 | 35 | 1 | 35 | On | Off |
| [20.999 21] | [0 1] | [0 35] | 1 | 35 | On/Off | Off |
| [21 21.999] | 0 | 15 | 1 | 35 | Off | Off |
| [21.999 22] | 0 | 15 | [0 1] | [15 35] | Off | On/Off |
| >22 | 0 | 15 | 1 | 15 | Off | On |

### *2.2.9.2.2 Humidity Control*

Since creating and implementing a dehumidifier in IDA-ICE seemed rather complex, the control is managed by a humidifier and the idea is to maintain a high enough level of humidity inside and keep it constant.

The core of the humidity control strategy comes from the humidifier code. Following equations sums up how it works. The moisture $W_{out}$ $(kg_{water}/kg_{air})$ at the outlet is the sum of the incoming humidity content $W_{In}$ and the added moisture brought by the steam $W_{steam}$. This $W_{steam}$ quantity depends on the maximum possible outlet humidity, $W_{max}$, set by a chosen constant parameter, and a 0-1 control variable. Included in Equation 2-10 are two specific NMF functions. SATPRES gives the saturation pressure for a certain air temperature, it means the maximum water vapor pressure possible at a certain temperature. HUMRAT gives the maximum moisture content in $kg_{water}/kg_{air}$ in the air depending on the air pressure and saturation pressure. By multiplying by the relative humidity, it gives the humidity ratio. In this case, it is the maximum humidity ratio that is obtained because it has been multiplied by the maximum relative humidity.

$$W_{out} = W_{In} + W_{steam}\ (kg_{water}/kg_{air}) \tag{2-8}$$

$$W_{steam} = Control * (W_{max} - W_{min}) \tag{2-9}$$

$$W_{max} = RH_{max} * HUMRAT(P_{air}, SATPRES(Tair_{outlet})) \tag{2-10}$$

The control acts on the control variable. The control variable will be again given by a PI-controller. Depending on the difference between a certain level of relative humidity, here 40%, and the exhaust air relative humidity, the PI-controller gives an output comprised between 0 and 1. This output will become the control variable within the humidifier if and only if the outside temperature does not go beyond a certain level, here 30°C. It is usual to implement such a control when injecting moisture in supply air. More specifically, for a relative humidity below 40% and an outside temperature below 30°C, the steam humidifier injects moisture at a maximum rate.

### *2.2.9.3 Actual AHU for the Pool Hall*

The AHU allocated to the pool hall is almost the same as the previous one. Temperature and humidity controls are the same but for other set-points. The temperature is to be comprised between 30°C and 32°C and the maximum set-point temperature for supply air is then set higher. The maximum set-point temperature for supply air is now 40°C to ensure enough power to the mechanical inflow to counteract all thermal losses. Contrary to the previous AHU, there is no air cooling, then the cooling coil efficiency is set to 0.

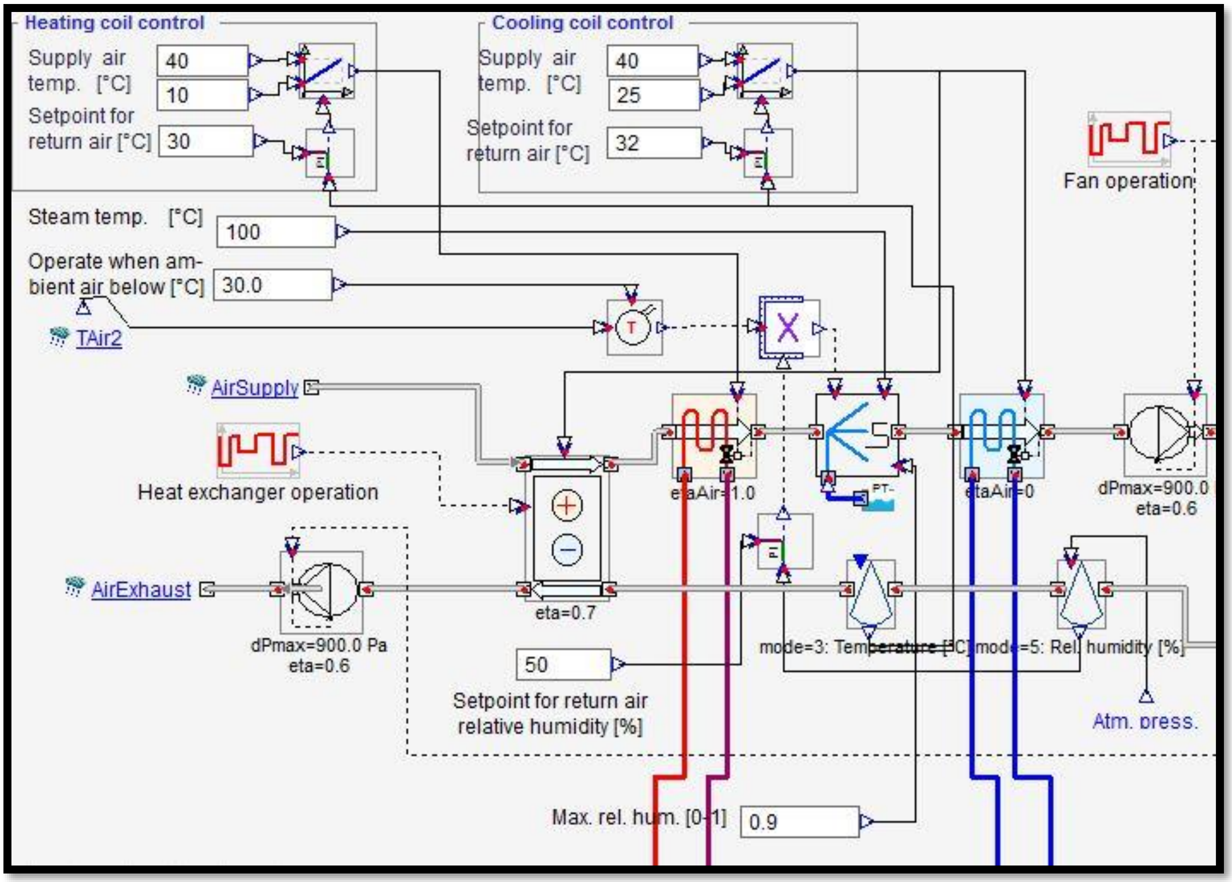



*Figure 2-10* Schematic view extracted from IDA-ICE of the AHU in charge of swimming hall.

### 2.2.10 Space heating system

For the early stage simulation, the standard AHU does not provide enough heat to the zones to maintain indoor temperature to the set-points. Indeed, the supply air is constant equal to 16°C. Then, the installed space heating system, made of ideal heaters and coolers, supplies energy to the rooms. Ideal heaters are a fictional heat distribution system that provide (or remove) the exact amount of energy to keep the temperature between the set-points without any dynamics. However, they possess a maximum power limit. After a few test simulation, values for the maximum power granted to ideal heaters is set to 500 W/m², which seems more than sufficient. Their efficiencies are arbitrarily set. In this case, their emission efficiencies are set to 1 which means the whole amount of energy provided by the plant is released to the room, there is no losses. One ideal heater and one ideal cooler are placed in every room except in the swimming hall. In the swimming hall, only one ideal heater is set up.

For the detailed model, the AHU unit monitor temperatures inside the building and is supposedly powerful enough. Since the airflow rate is constant, only the value of the maximum supplied air temperature, that the AHU can handle, determine the maximum power. Based on few initial simulations, it appears that the AHU struggles to maintain indoor temperature in the entrance. So an ideal heater is set up in the entrance to ensure a constant temperature. This ideal heater power is set to 100 $W/m^2$ so that it does not have too large influence. Maintaining constant indoor conditions is the bottom-line of this model.

# 3 Simulations & Results

## 3.1 Early Stage Simulation

To be clear with this model's characteristics, below is a summary of all differences (read the whole section 2 for more details).

- All characteristics are the same as the detailed model, that is to say the ones described section 3 except for the following points.
- The AHU unit is standard or by-default one from IDA-ICE (see section 2.2.9.1).
- Ventilation rate in the hall is constant equal to 3.612 L/s.m².
- There is no negative pressure strategy. All specific supply and exhaust airflow rates are equals within each room.
- Ideal heaters and coolers are installed in all the rooms (except an ideal cooler in the hall) and their set-points are summed up Table 3-1. Below the minimum temperature, ideal heaters operate at full power and above the maximum temperature ideal coolers operate at full power.

*Table 3-1* Temperature set-points for ideal heaters and coolers in the early stage model.

| | Minimum Temperature | Maximum Temperature |
|---|---|---|
| Entrance | 21 | 24 |
| Showers | 21 | 24 |
| Hall | 28 | 30 |

### 3.1.1 Analysis

The delivered energy corresponds to the site EUI discussed section 1.2.1. The delivered energy for one year to the building is 1421 kWh/m² or 4593 kWh/$m_{ws}^2$ (ws stands for water surface). In average for the year, DHW represents 15.3% of the total consumption, and fuel heating for 76.3%. Fuel heating is the energy delivered by the plant. The standard plant in IDA-ICE is a simple fuel boiler with a 0.9 efficiency, meaning 90% of the chemical energy incorporated into the fuel ends up in the water by raising us the temperature. Behind fuel heating are zone heating, AHU heating and heating of the pool water (for heating of the pool water, see explanations below). Below is the allocation of delivered energy to the various building processes. DHW consumption stays almost perfectly constant and fluctuates due to holydays or number of days in a month. Similarly, HVAC Aux, which stands for fans and pumps consumption, and the lighting fluctuates slightly throughout the year. What triggers the variations is the fuel heating consumption.

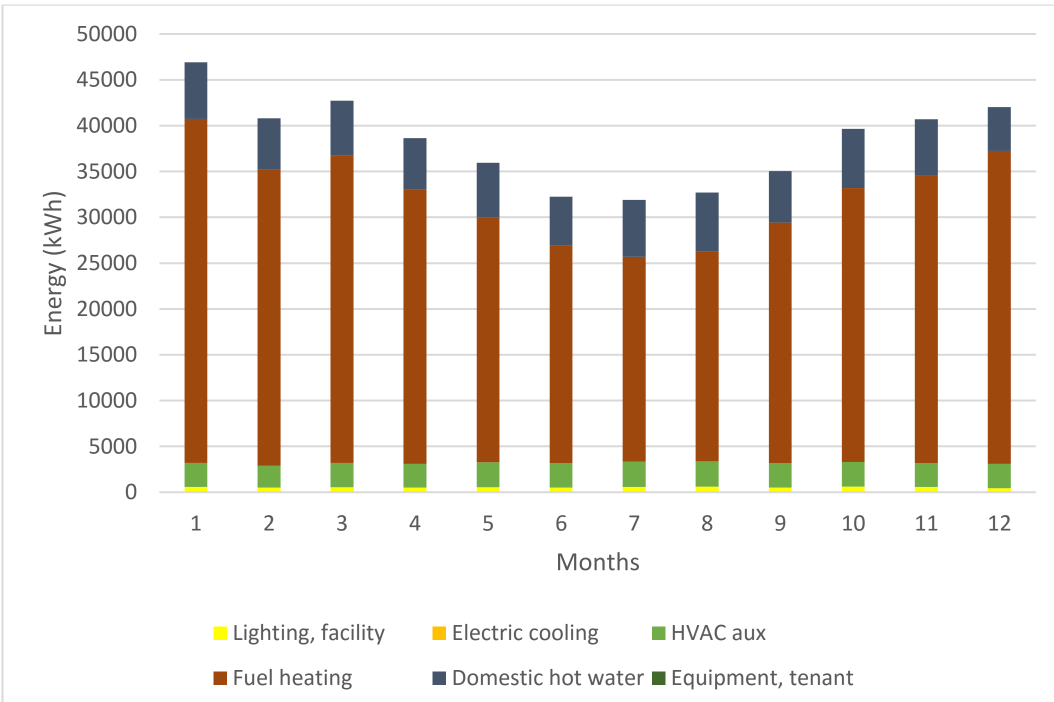



*Figure 3-1* Allocation of the delivered energy throughout the year in the IDA-ICE simulation for the early stage model.

In order to investigate the fuel consumption, IDA-ICE offers a result tab where energy used in different systems is described. Figure 3-2 sums up the data. All data do not stand for the previous fuel heating consumption, but further analysis can be conducted with it. What is included in fuel heating is zone heating and AHU heating. The first month, the consumption for fuel heating is 37535 kWh. Still during this first month, energy consumption for zone heating and AHU heating is 32431 kWh and 1351 kWh respectively. Taking into account the 0.9 efficiency of the boiler, 90% of the previous 37535 kWh gives exactly (with 0.5 kWh error) the sum of zone heating and AHU heating: 33782 kWh. AHU heating clearly represents energy released by the heating coil to the supplied mechanical air. However, zone heating could be wrongly interpreted as energy released by ideal heaters to the air in the rooms. However, it is not true at least with the ice rinks & swimming pools extension of IDA-ICE. It comprises what IDA-ICE calls "water-based heating" as well. It basically is heating of pool water. A study about water-based heating has been conducted later in this chapter.

Figure 3-2 illustrates how important heat recovery is. The part of heat recovery is worth 44% of zone heating on an average monthly basis. It even reaches 75% of zone heating during January.

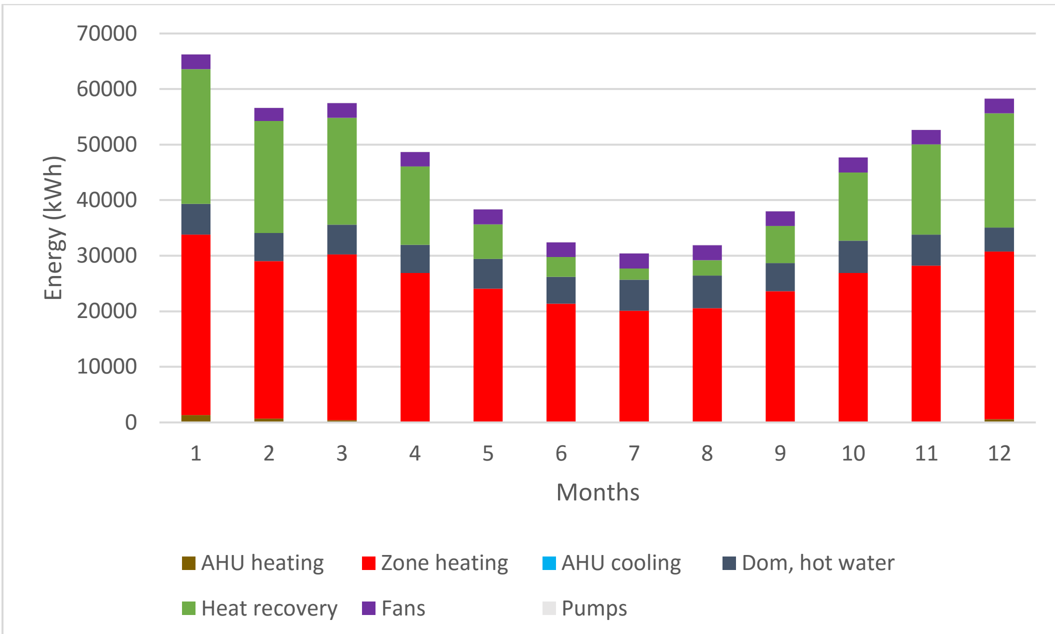



*Figure 3-2* Energy provided to the building as a whole according to different systems. Data come from IDA-ICE.

A new sort of energy consumption, called water-based heating, appears in the results for total heating in IDA-ICE. To study it, $M_{supply}$, $T_{supply}$, and $T_{exhaust}$ variables were logged to the results (see Figure 2-3). By investigating these values, Equation 3-1 is found out. Values are not perfectly equal, but their average difference is only -3.8 W with a standard deviation of 1.5 W. These numbers are tiny compared to the range of values. Average water-based heating throughout the year is 16046 W.

$$Water\ based\ Heating\ (Watt) = M_{supply} * C_{p_{water}} * (T_{supply} - T_{exhaust}) \quad 3\text{-}1$$

$$C_{p_{water}} = 4186\ J/kg.K \quad 3\text{-}2$$

Figure 3-3 shows monthly power average for the four following processes: heating coil, water-based heating, ideal heaters, and DHW. First, it is important to notice that these monthly averages do not accurately represent energy consumptions because IDA-ICE's simulation time steps are not perfectly constant. For instance, DHW seems to represent a much bigger part than before but when it comes to energy, it is previous figures that are relevant. Results for power pinpoint the significant part of water-based heating compared to other consumptions and especially ideal heaters consumption. Water-based heating is in average 81% of the power demand for ideal heaters. However, its power curve is clearly more erratic than for ideal heaters.

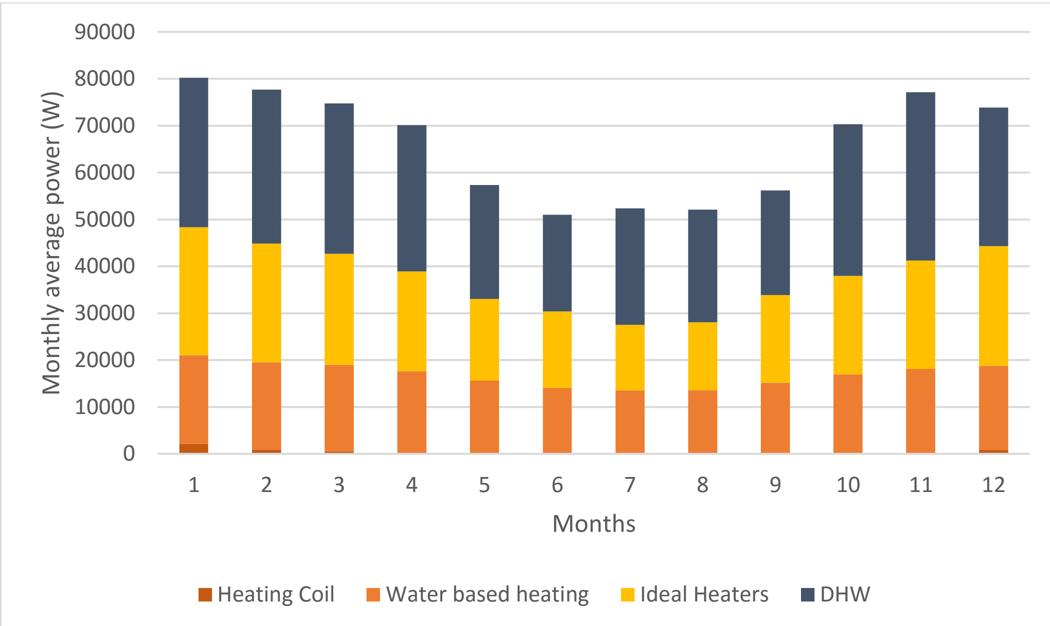



*Figure 3-3* Monthly average of power demand for heating coil, water-based heating, ideal heaters, and DHW. Data are retrieved from IDA-ICE.

What Figure 3-3 does not show is the volatility of the power values. Figure 3-4 illustrates this volatility during the first week of simulation. This first week is deemed to be relatively characteristic of the entire year. One can notice that 1$^{st}$ of January is part of the holydays so the first day of simulation is not a regular weekday. Data during other weeks do not have exactly the same values but they do have the same shape. First, DHW power curve appears to be singular. Peaks values appear regularly during the day and are considerably higher than the rest. They almost reach $2.10^5$ W. This is due to the high demand of shower water during short turnovers. Basically, forty-four people take 6 min showers within a 30 minutes span. It logically ends up in a significant power demand in the case no water tank is set up. When it comes to AHU heating, it is only influenced by the outside dry-air temperature and how much heat is recovered from the rotary wheel. Since AHU heating equals zero from the fifth day, it necessarily means the rotary wheel provides enough heat to the supplied air to reach the 16°C.

Water-based heating has a regular pattern throughout the week days and shows a flat curve during week-ends. Every day, two peaks arise. Remembering that values cannot be deemed as 100% representative of the whole year, the first power peak is about 35 kW and the second is roughly 30 kW and happens during the day (and during DHW power demand). By going into specifics, it appears the 3-to-4 a.m. flush down of graywater triggers the night peak of 35 kW. It is understandable since pool water is thrown out at a 0.7417 kg/s mass flow rate (see section 2.2.5). During the day, it is the evaporation rate that triggers heating of pool water. Both energy and mass leave the water through the pool surface. Thus, this phenomenon entails power demand for water-based heating. Further study about evaporation is conducted below.

Ideal heaters appear less erratic. It seems that two levels of power demand take place, roughly 30 kW during night, and roughly 25 kW during day. On first thought, one would put forward day and night temperature difference would influence ideal heaters. However, ideal heaters' power demand stays constant throughout week-ends (days 6 and 7 Figure 3-4). It means outside dry-air temperature barely has a minimal impact on sensible heat consumption variations inside the building. It seems that occupancy and lighting internal loads are the only ones playing a role in these day and night ideal heaters' power variations. Rapid power calculations for light and occupants give a 10.1 W/m² from light and roughly 20 W/m² from occupants, assuming 100 w/person. These numbers are in average per square meter for the whole building. It ends up to roughly 10 kW which is comparable in size with ideal heaters' power demand variations.

Also, it is very important to highlight that water-based heating brings no ideal heaters' consumption reduction whatsoever. For instance, the peaks that appears during week days do not induce a reduction in power demand for ideal heaters, it is quite the opposite. In fact, water-based heating is directly due to nigh flush of water and to evaporation. Evaporation releases latent heat to the indoor air. The heat is stocked in the vapor, which can be release by condensation. Contrary to sensible heat, latent heat does not trigger temperature variations. Therefore, evaporation makes the pool release latent heat but ideal heaters (or space heating in general) must deliver the sensible heat needed to counteract energy losses and drops of temperature.

Therefore, the evaporation process wastes heat through water-based heating and even induces high humidity environment. This is the reason why this phenomenon should be curb as much as possible.

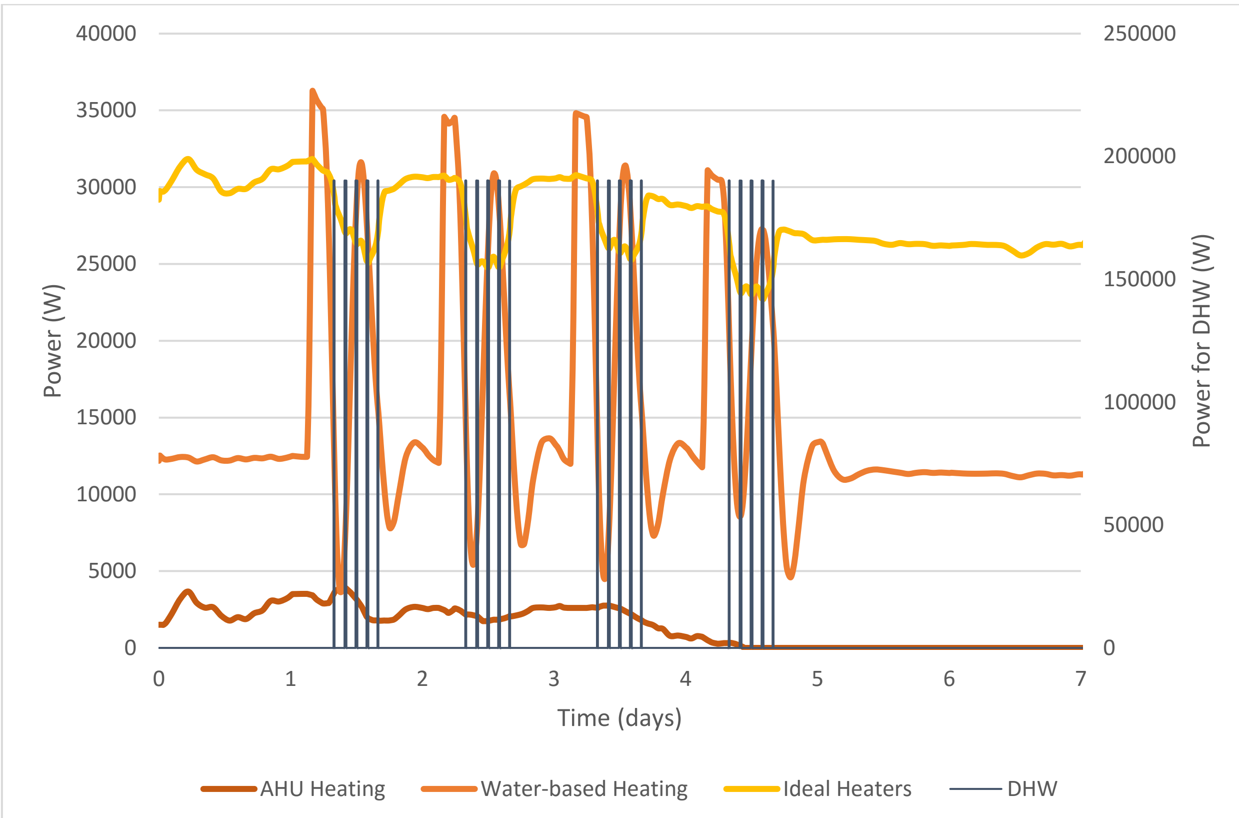



*Figure 3-4* Power demand for processes linked to the simple fuel boiler in IDA-ICE. Second vertical axis on the right-hand side is related to DHW power demand.

How heat energy and mass transfer appear due to evaporation? How evaporation occurs in IDA-ICE? When looking into the NMF code, it is possible to spotlight how IDA-ICE cope with this phenomenon. Four variables stand out: $Q_{surf}$(W), $Q_{moist}$(W), $Q_{zone}$(W), and $HUMF$(kg/s). HUMF is the variable describing mass transfer of pool water to the pool surface and then to the indoor air. The value is calculated at every time step according to ASHRAE formula for evaporation (see section 2.1). Figure 3-5 gives an illustration on how variables interact with the environment. $Q_{zone}$ is the sensible heat transfer going from the zone to the pool surface. It relies directly on temperature difference between indoor air and surface air. Then, temperature of the pool also acts on its value. $Q_{moist}$ stands for the latent heat due to evaporation. It is the power needed to evaporate the mass flow rate of water and raise its temperature to the vapor temperature of the indoor air. The value of $Q_{moist}$ is negative which means the latent heat goes from the water surface to the indoor environment.

$$Q_{surf} = Q_{moist} + Q_{zone} \tag{3-3}$$

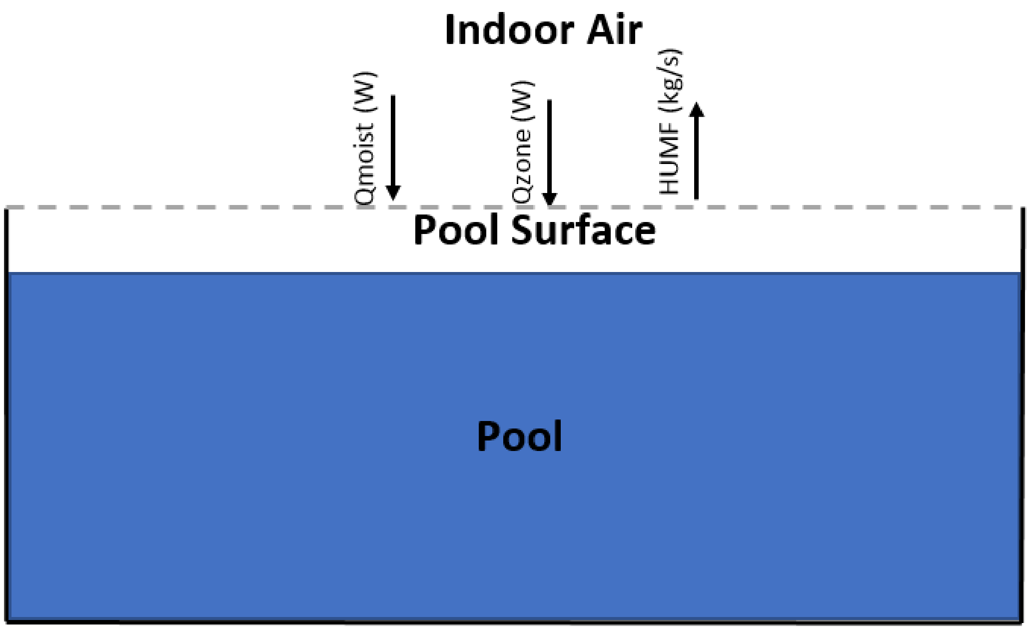



*Figure 3-5* Illustration of IDA-ICE variable on the pool. N.B. the arrow for $Q_{moist}$ is in the right direction. Nevertheless, its value is negative.

Figure 3-6 illustrates power exchanges between the pool surface and the indoor environment. Both $Q_{surf}$ and $Q_{moist}$ are negative which means the energy is actually leaving the surface to the air. They both have a special pattern during weekdays (the first day of the week is 1$^{st}$ of January and it is part of the holydays). Due to occupancy of the pool and an activity factor that goes up to 1, evaporation rises and so does $Q_{moist}$. There are some "waves" during the days due to turnovers. During turnovers, activity factor remains the same, but doors are opened (see Appendix 7.5) and, since there is no negative pressure strategy, the warm and humid air leaves the room. Thus, by supplying the pool hall with air having a clearly lower absolute humidity, partial vapor pressure difference between pool surface and indoor air is enlarged and evaporation is enhanced. Also, there are noticeable smaller peaks, of both $Q_{moist}$ and $Q_{zone}$, happening regularly around 3 to 4 a.m. This phenomenon is due to pool water being flushed out at that time which entails a drop in the pool water temperature. As shown Figure 3-7, this drop amounts to 0.3°C. A pool water temperature decrease means a larger temperature difference between indoor air (normally 30°C) and the very pool water temperature (normally 28°C). A larger temperature difference induces a larger sensible heat transfer $Q_{zone}$. A decrease in pool water temperature entails also a decrease in partial vapor pressure at the pool surface giving rise to a slower evaporation and then a decrease in absolute value of $Q_{moist}$.

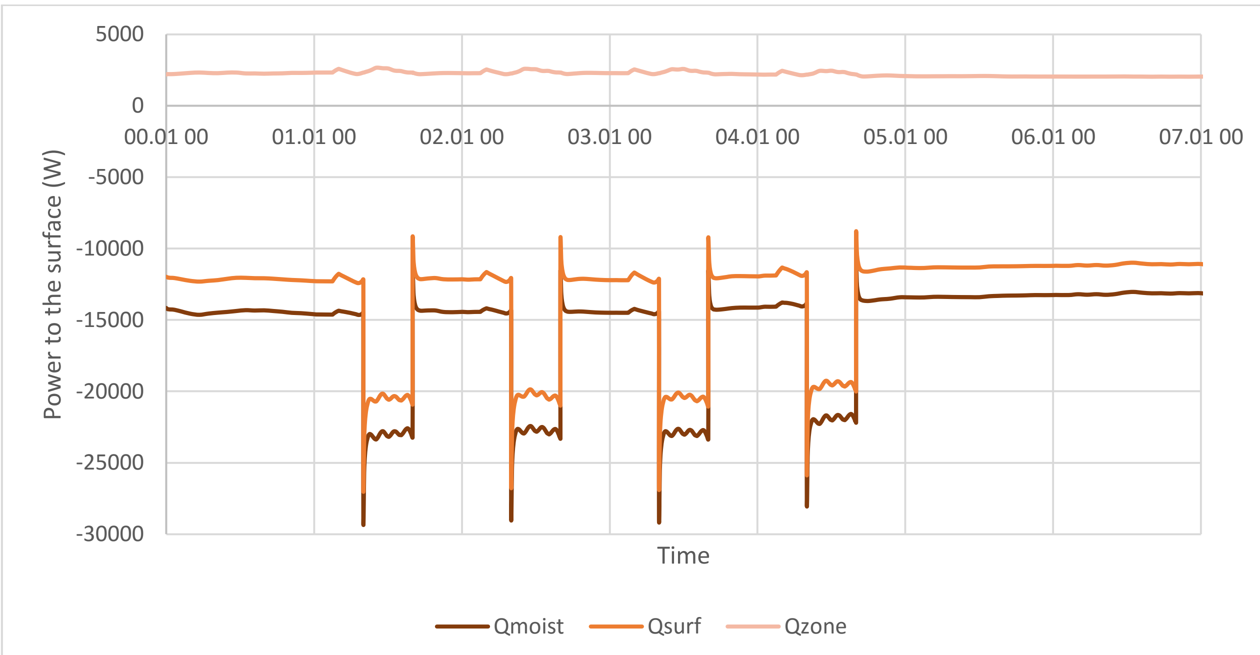



*Figure 3-6* Power exchanges between the pool surface and the indoor environment

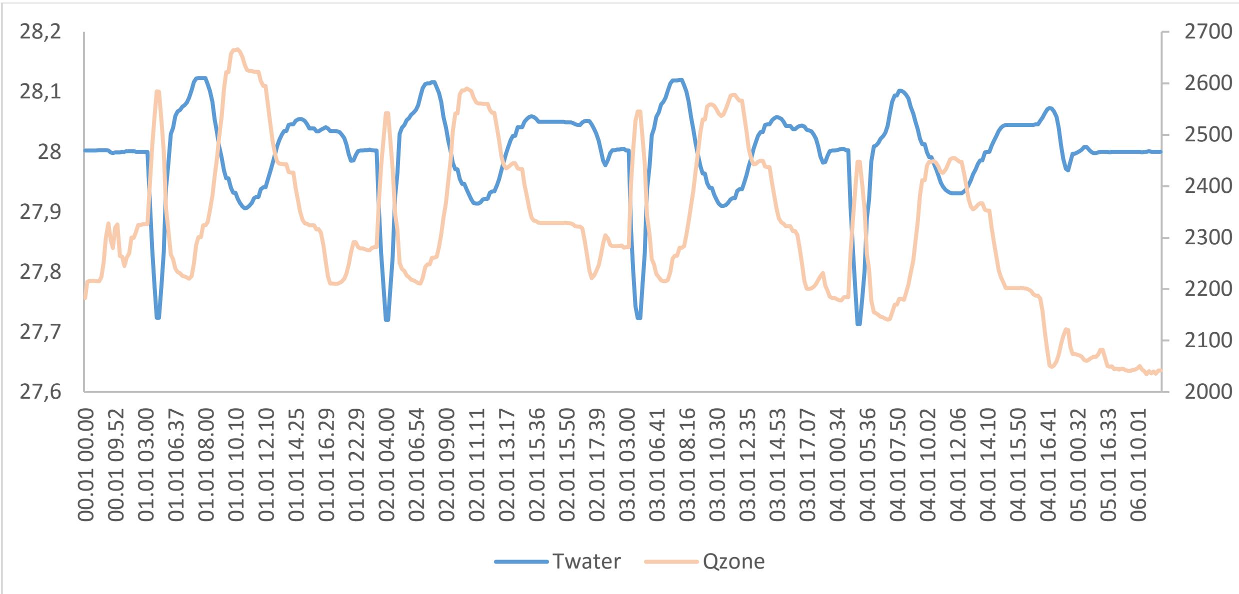



*Figure 3-7* Temperature of the pool water throughout the week and proof of its correlation with Qzone.

Then, $Q_{surf}$ represent the sum of a loss of the zone's sensible heat, through $Q_{zone}$, and an undesired gain for the zone in latent heat, through $Q_{moist}$. Its value must be lowered as much as possible. Figure 3-8 shows the correlation of its value and the evaporation rate.

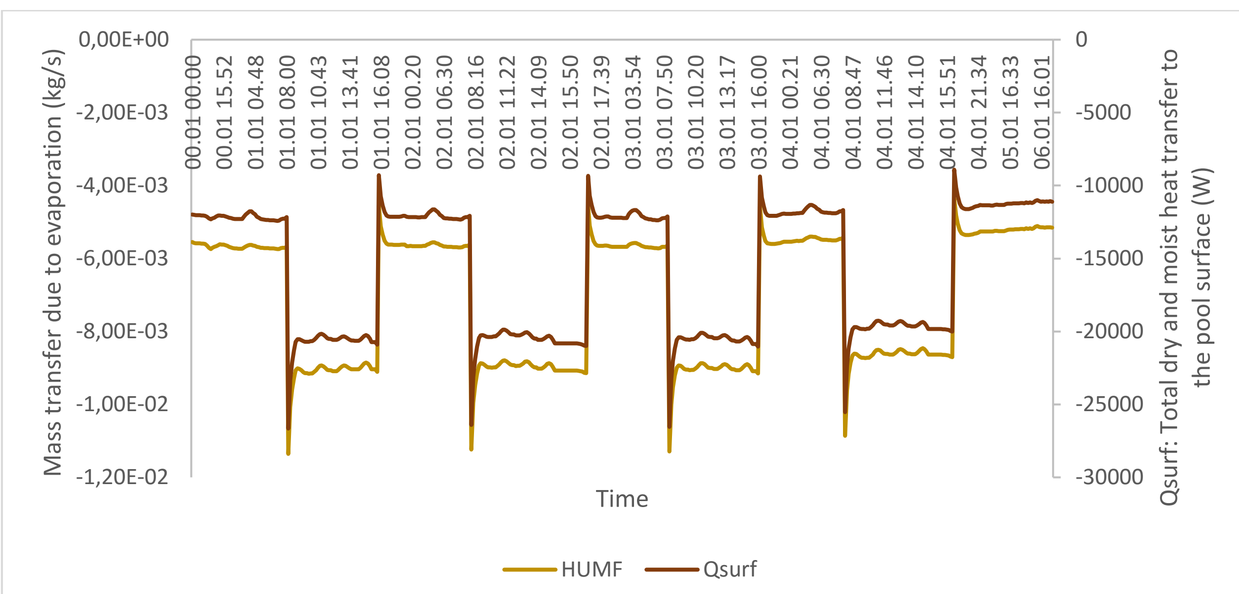



*Figure 3-8* First week data from IDA-ICE of evaporation rate and the dry and moist heat transfer to the pool water surface.

The indoor environment is investigated next. As shown Figure 3-9, all zones reach their set-point perfectly. Indeed, every standard deviation are less than 0.5°C for all zones. It means temperatures stay almost at the set-point level throughout the year. The reason behind this perfection is the very high maximum power of ideal heaters that are therefore able to maintain the desired temperature conditions.

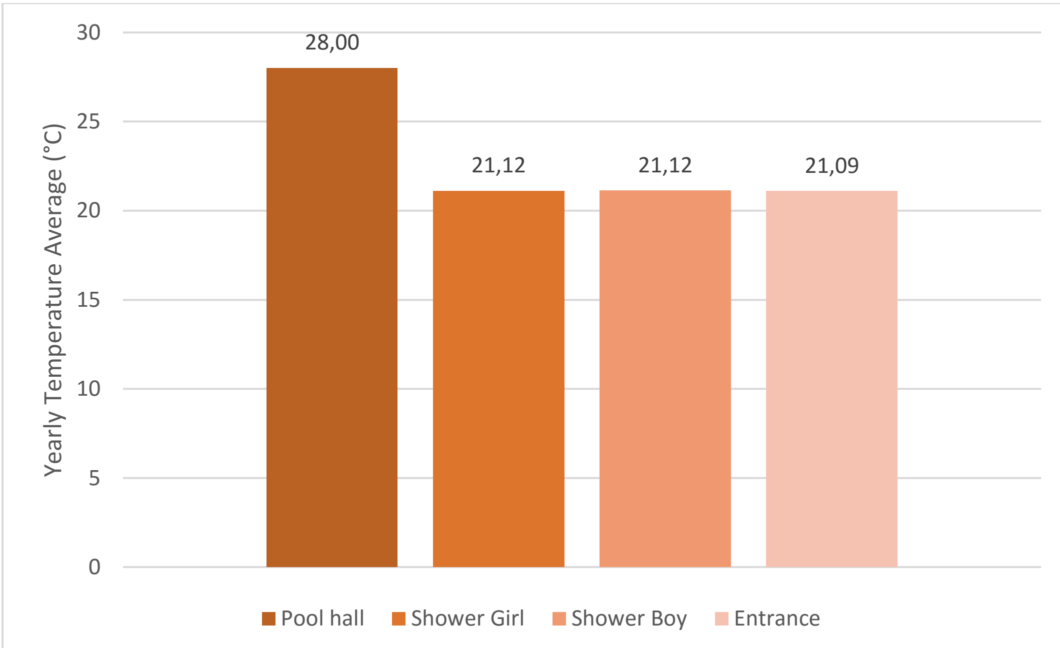



*Figure 3-9* Yearly average temperatures for all zones. All zones match their set-point perfectly.

When it comes to humidity, no control exists for this early stage model. Values are way more volatile as shown Figure 3-10. Most likely, the pool occupancy during the day from 8 a.m. to 4 p.m. triggers all the variations. The high vapor content in the pool hall spread to the other zones during turnovers when doors are opened. Further studies are conducted below. In a general manner, the rise of humidity in the middle of the simulation is due to higher absolute humidity values of the outside air during summer. What is surprising in Figure 3-10 is the low level of humidity for all zones. Without humidity control, one might have supposed humidity to be and stay too high within the building. It is the case in real life since HVAC designers install dehumidifier systems.

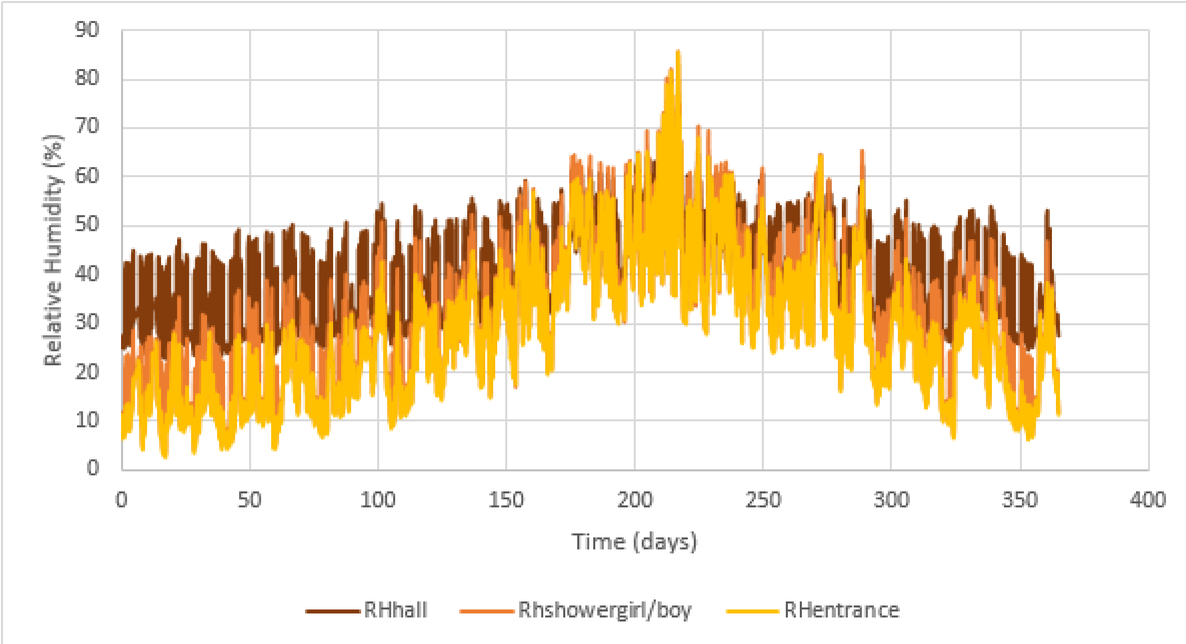



*Figure 3-10* Relative humidity for all zones. The two showers are gathered in one curve given they are very nearly equal.

Figure 3-11 goes into specific by drawing RHs for one week. Peaks' rise of around 15% take place every day in the hall. Peaks' amplitude for the showers is lower, 10%, and the entrance peaks' amplitude even lower, few degrees only. This figure shows clearly the influence of the swimming pool on the hall first and then on the rest during turnovers. The zoom aims at highlighting the fact that entrance and showers RHs are correlated but inversely correlated to the pool. It is completely normal since when doors are opened, RH in the hall goes down while the others rise with the incoming hot humid air.

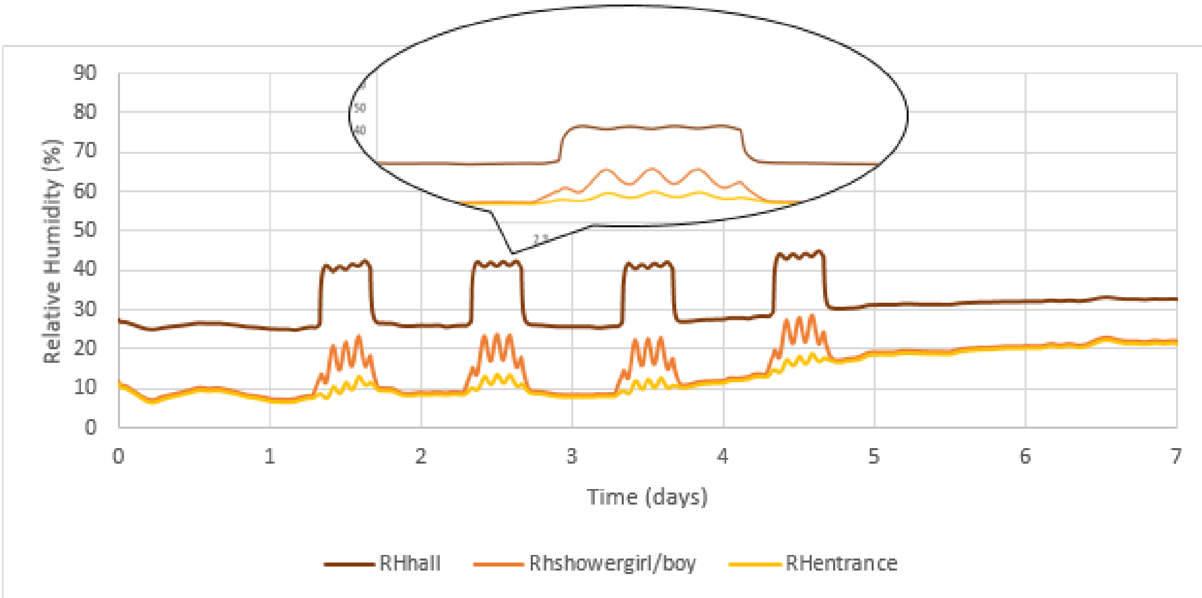



*Figure 3-11* Relative humidity for all zones. The zoom serves to show that peaks are inversely correlated between pool hall and the rest of the building.

### 3.1.2 Discussion

First, general energy results can be surprising at first sight. With an overall delivered energy of 1421 kWh/m², a performance in building engineer would suggest building wall around the building. However, many model features imply that this value is sound. The building is very small in size. A real swimming facility would comprise many other types of rooms. There are no precise statistics about what a usual area is but, for the same size of pool, a building having an area twice as larger would not be surprising. The consumption would then reach about 700 kWh/m², which is more usual for swimming facilities. In addition, when consumption is brought per square meter of pool area, it becomes 4593 kWh/$m_{ws}^2$. As deeply discussed section 1.2.3, this number is completely sound and acceptable. Furthermore, the part of energy used for space heating and heating of the water in the model appears to be fully usual in percentage of the overall consumption. In the model, delivered energy for both space heating and water-based heating represent 76.3% which is statically very correct as discussed in section 1.2.3. Even though consumption might be pretty high, the model reacts well and meets real facilities' statistics.

Reasons for a relatively too-high consumption are various. The consumption of DHW is tremendous compared to the building size and the pool occupancy seems pretty high as well. An important remark is that the building structure is not optimal either. Insulation levels barely follow TEK17 minimum requirements. Also, heat recovery is not optimal. First, as discussed in section 3.1.1, the share of sensible heat recovery is huge while the rotary wheel has an efficiency of only 0.7. Any improvement regarding this efficiency could bring substantial energy savings. Second, the rotary wheel heats up the air until 16°C only, and as seen previously, a major part of the time the heating coil is not even used because the wheel is sufficient. It means the rotary wheel's potential is not fully fulfilled and the wheel could recover even more energy. Third, only sensible heat is recovered. As illustrated in Figure 1-7, a dehumidifier installed in the AHU brings latent heat recovery and latent heat recovery has especially big potential in swimming pool due to evaporation.

Another remark is that processes inside the building, like the demand for high temperatures and the direct and indirect energy demand by the pool, are so significant that the outside air temperature has less impact on energy consumption. In fact, AHU heating may seem very low in Figure 3-3 but it is actually the other processes that are very high. DHW power demand is tremendous and must be significantly lowered by operation of a water tank. Water-based heating raises the power demand by two since it almost represents the same amount of power in average than ideal heaters and the peaks are even higher. Through water-based heating, the huge influence of the swimming is perceived.

Figure 3-11 shows how needed is a negative pressure strategy. It spotlights the hall's hot and humid air propagating inside the other rooms when opening the doors. This phenomenon can damage the building's structure.

Results show an unexpected low level of humidity especially during winter. The reason of this low level of humidity is likely to be a high ventilation rate and a swimming pool model that does not reflect perfectly real evaporation rate. As stated before, ventilation rate for the hall is equivalent to 5 ach, even not occupied. The supply of fresh air stays then high day and night. As for the model for evaporation and as discussed section 2.1, IDA-ICE supposes constant air velocity above the pool water of 0.10 m/s. Considering the occupancy that the model has and the fact that people may cause water splash among other water surface perturbations, even more evaporation might occur in real conditions compared to the model. From this analysis, it has been decided to use this low humidity to maintain constant indoor conditions for the detailed model. This strategy has been explained in section 2.2.9.2.2.

## 3.2 Detailed Model Simulation

The aim of this model is to simulate a close-to-the-reality system while keeping indoor environment conditions constant. By maintaining temperature and humidity constant, it gives a valuable understanding about how the system behaves. In fact, a specific model could be less energy demanding while maintaining colder indoor conditions. So, the indoor conditions are crucial and must be specified.

Our humidity control strategy is purely artificial, no swimming facility set up a humidifier in the supply air duct. The idea is to keep the humidity constant, but it also entails unnecessary or unrealistic energy consumption.

### 3.2.1 Analysis

Delivered energy is now higher than in the early stage model. It amounts to 1477.5 kWh/m² and 4774.6 kWh/$m_{ws}^2$. However, the consumption characteristics have been surely modified. Fuel heating now represents only 57% of the total delivered energy in average over the year. A new type of consumption has appeared due to HVAC Auxiliary. As shown Figure 3-12, HVAC Auxiliary stands for 26.5% of delivered energy in average over the year. HVAC Aux stands for fans, pumps, and humidifiers consumption. In this particular case, humidifiers consumption represents 78% of the delivered energy attached to HVAC Aux, while fans consumption represent almost all the rest. As a matter of fact, even though the setup of humidifiers in a swimming pool is unrealistic, a large part of the energy consumption goes into it because of the humidity control. Also, one can notice that the absolute fuel consumption values are lowered than in the early stage model. This is due to two phenomena. It is important to remember that space heating happens now through ventilation whereas through ideal heaters in the early stage model. However, heating coils efficiency is set to 1 and ideal heaters are, by definition, ideal so their efficiencies are 1 as well. Thus, space heating transmission loss does not cause the decrease in fuel heating because it remains perfect. The first and main reason is how space heating performs to reach the indoor temperature set-points. Further study below in this section will show that temperatures set-points are not meet as perfectly as with ideal heaters. Then, it induces a decrease in fuel consumption because the average temperature is lower. The second reason is less important but still plays a role. Humidifiers release latent heat to the air but also some sensible heat. Temperature of the air entering the humidifier is slightly lower than the one going out. For the AHU in charge of showers and entrance, the humidifier brings in average an extra 0.28°C. For the other AHU in charge of the pool hall, this value amounts to 0.42°C. It means, some of the space heating consumption is displaced to the HVAC Aux category.

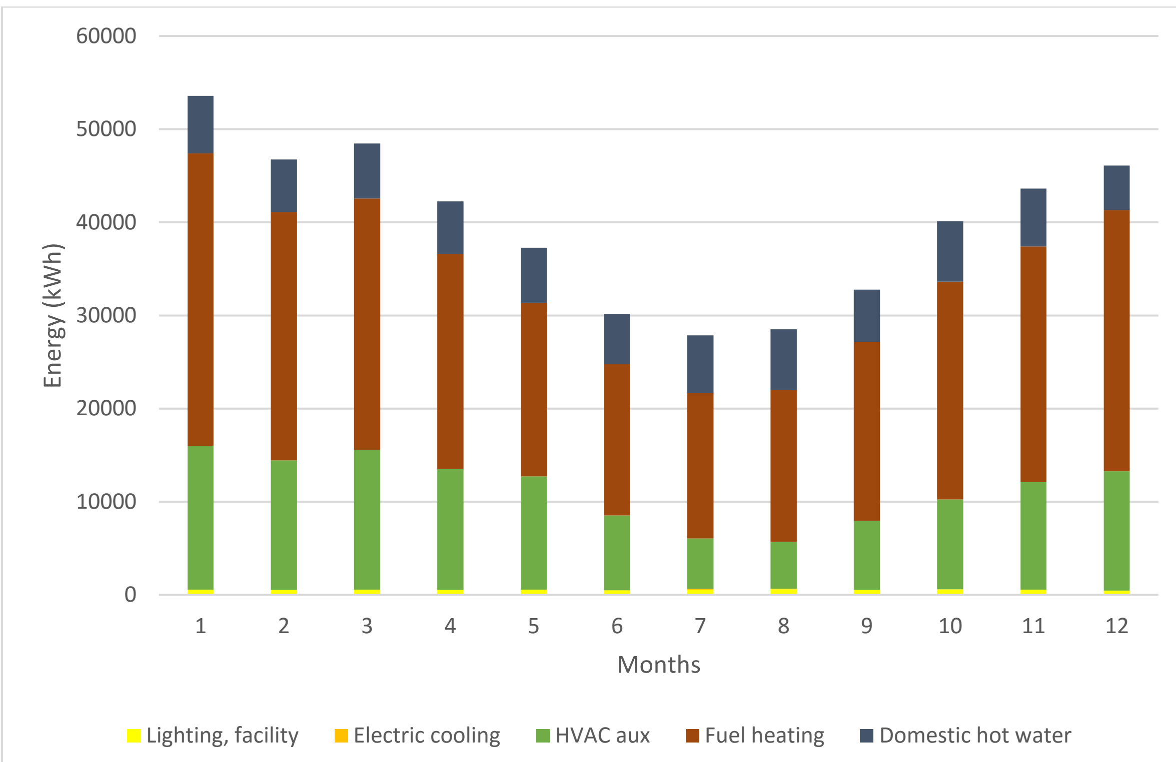



*Figure 3-12* Delivered energy over the twelve month of the simulated year. Data are retrieved from IDA-ICE.

In this model, all space heating is ensured by the two AHUs. Figure 3-13 illustrates behaviors of the AHUs. Consumption for the AHU dedicated to the hall is logically higher. Pool area stands for 75-80% of the total floor area and indoor condition set-points are clearly more demanding than for the rest of the building. The yearly energy consumption for this AUH is 168026 kWh (without the heat recovery), while the other one uses 84591 kWh. The part dedicated to heating is lower than the energy recovered by the rotary wheel. A greater share of energy, compared to the total, is used to heating in the AHU dedicated to the hall. This is due to higher demand in temperature and then higher demand for the heating coil. Adding the two yearly energy consumption dedicated to heating of the supply air gives 122870 kWh. This value is retrieved to 90% of the yearly fuel heating consumption of 270980 kWh to find the energy use for water heating of the pool: 121012 kWh/year. One may see the noticeable impact that humidifiers have on energy consumption. Actually, for both AHUs, humidification stands for 25% of energy consumption within AHUs. Finally, cooling for the AHU in charge of showers and entrance is almost non-existent.

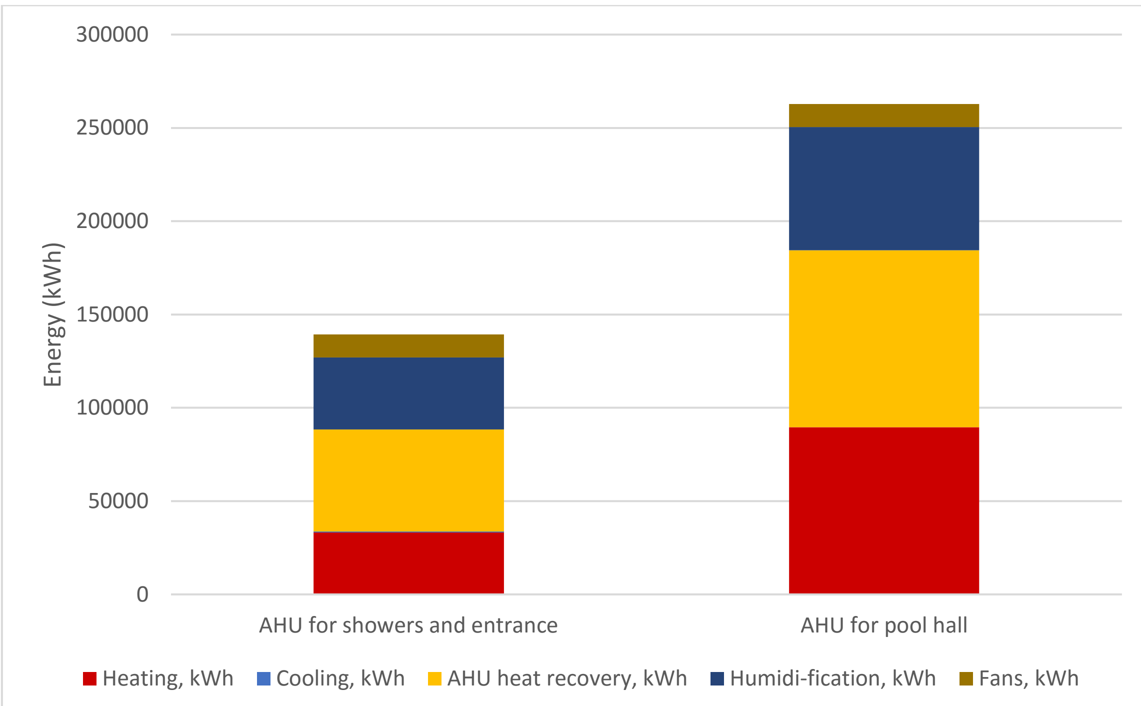



*Figure 3-13* Yearly energy consumption of the AHUs. Data are extracted from IDA-ICE.

Figure 3-14 illustrates power consumption for heating coils, DHW, the ideal heater in the entrance, and heating of the pool water throughout the first week of the simulated year. Even though no charts is drawn for this purpose, water-based heating has exactly the same pattern than with the early stage model. In fact, when drawing the two curves on the same chart, they almost perfectly look alike. There is a small difference though. When the average power demands for water-based heating are calculated for the two data set from the two models, a 7% difference appears. These data are to be taken very carefully since the time step is not constant and can vary depending on the simulation. Therefore the average power demand can be a bit misleading. However, this difference seems normal because in the second case relative humidity is kept constant and higher than the early stage simulation where it fluctuates but within lower range of values. Indeed, higher humidity in the hall curb the evaporation phenomenon which entails lower power demand for water-based heating. The main source of power which ensures space-heating is now coming from the AHUs. AHUs heating has a characteristic pattern. It has a sudden rise during opening hours. In fact, the ventilation strategy is CAV with two different values depending on operation hours (see section 2.2.8.1). During operation hours, the volume flow rate is higher and so is the heating power released to the hall. According to data, the temperature set-points remain most of the time constant and are constant at least during the week studied Figure 3-14. Since the temperature set-points (which depend on return air temperature) are constant and so do airflows, fluctuations come from another process. It is the heat exchanger which causes the fluctuations. Thus, fluctuations are indirectly linked to the indoor air temperature. When the indoor temperature gets warmer, heat exchangers recover more heat and less power is demanded for heating coils. Fluctuations of indoor air temperature come from the turnover periods when all indoor doors are open.

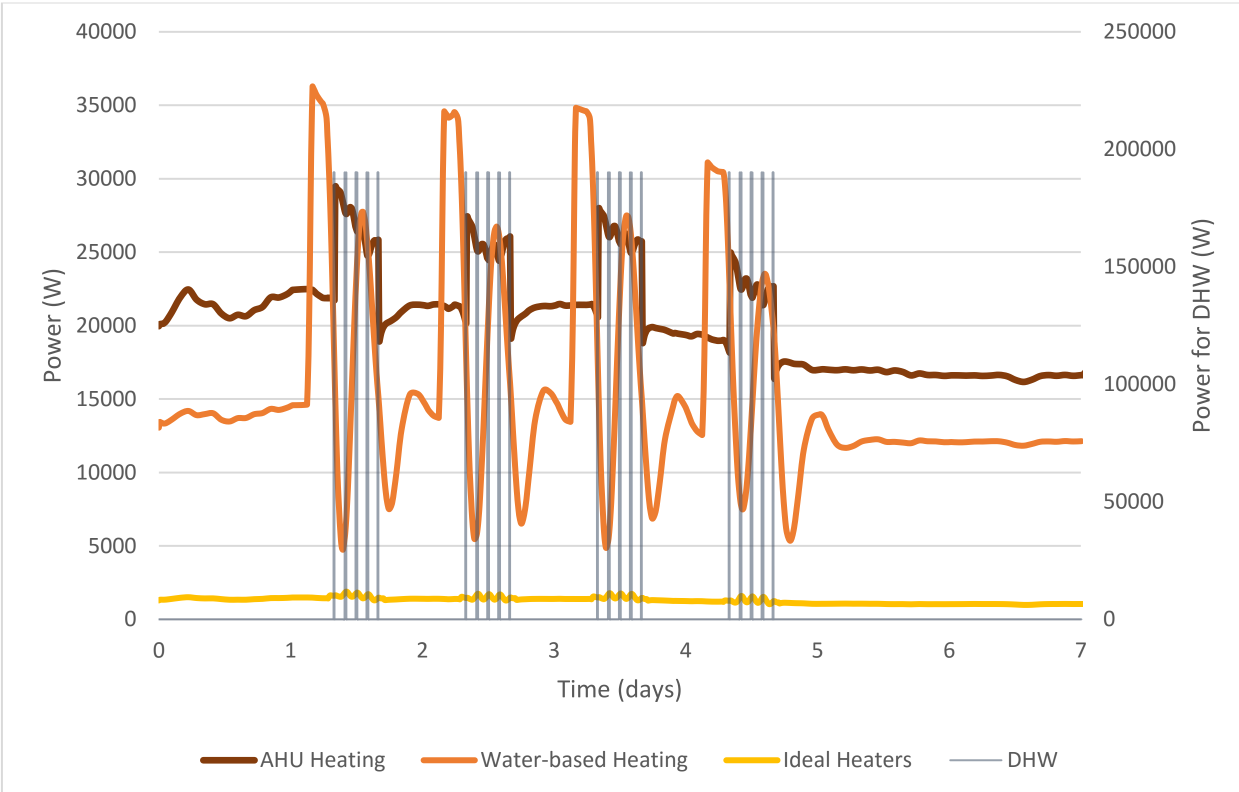



*Figure 3-14* Overall power consumption throughout the first week according to data retrieved from IDA-ICE.

Figure 3-15 shows monthly averages power demand for heating coils, heating of pool water, DHW, and ideal heater in the entrance. What is important to notice is the non-negligible share of water-based heating that even becomes greater than AHU heating during summer. Although yearly energy consumption from AHU heating is very slightly above water-based heating according to IDA-ICE results, power demand for heating of the pool water appears greater in average during the year than AHU heating since AHU represents 80% of water-based heating. As discussed before, these power demand averages cannot be perceived as energy consumption since time-step is not constant but it gives fruitful information.

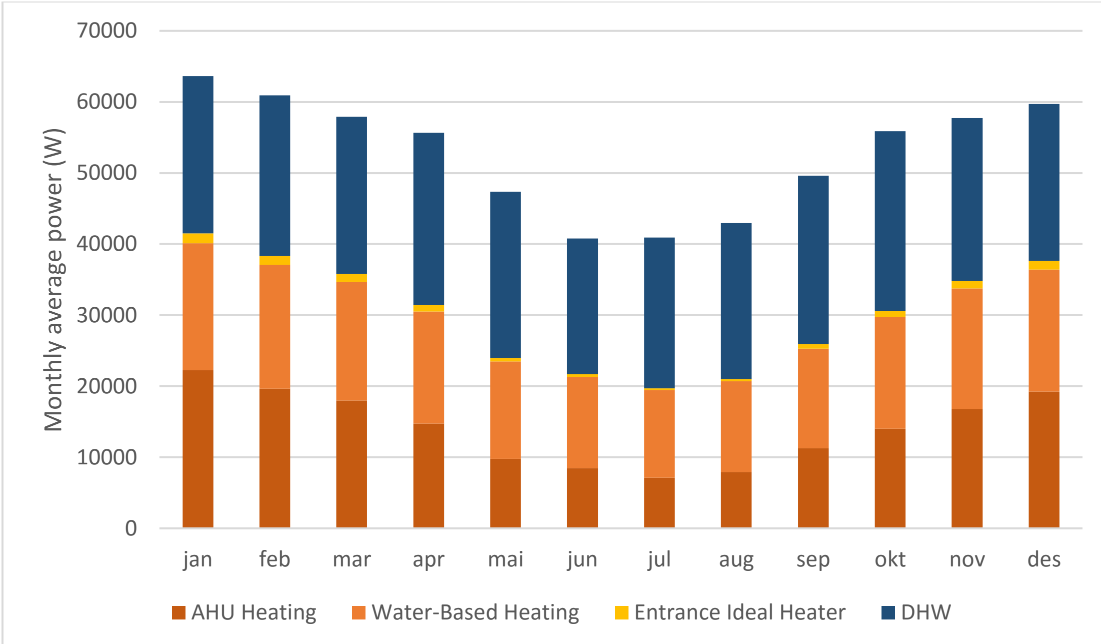



*Figure 3-15* Monthly average power demand for processes related to heating. Data are retrieved from IDA-ICE.

The indoor is investigated next. Figure 3-16 sums up the data about temperature retrieved from IDA-ICE. Standard deviations for each set of data appear on the graph to highlight whether values throughout the year are volatile. Except the pool hall, temperatures are pretty constant close the desired value. Though, it appears that temperature within the pool hall is more volatile below the desired value of 30°C.

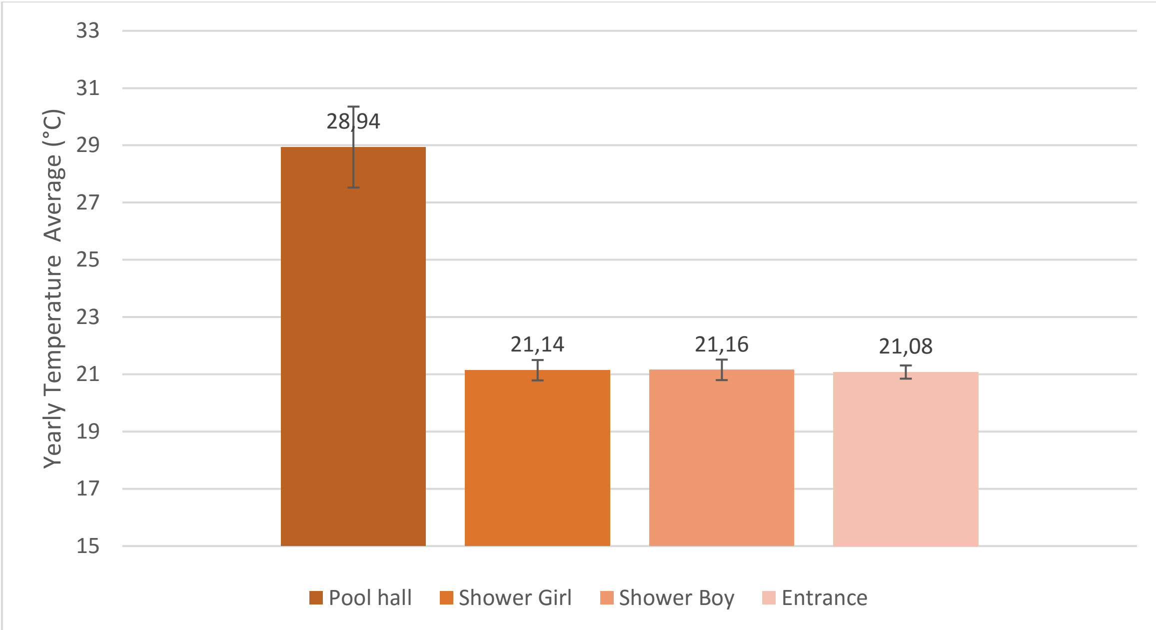



*Figure 3-16* Temperature averages in the zones with their own standard deviations. Data are retrieved from IDA.

When it comes to humidity, the pool hall meets very well the set-point of 50%. However, in the rest of the building, there is a control to keep RH at 40% and values are more volatile. Averages are near 40% but standard deviations are higher than in the pool hall. Figure 3-17 illustrates this phenomenon. Since the control lies on increasing the humidity and not decreasing it, by having a lower set-point, here 40% instead of 50%, entails a less-effective control. However, indoor conditions can be deemed constant.

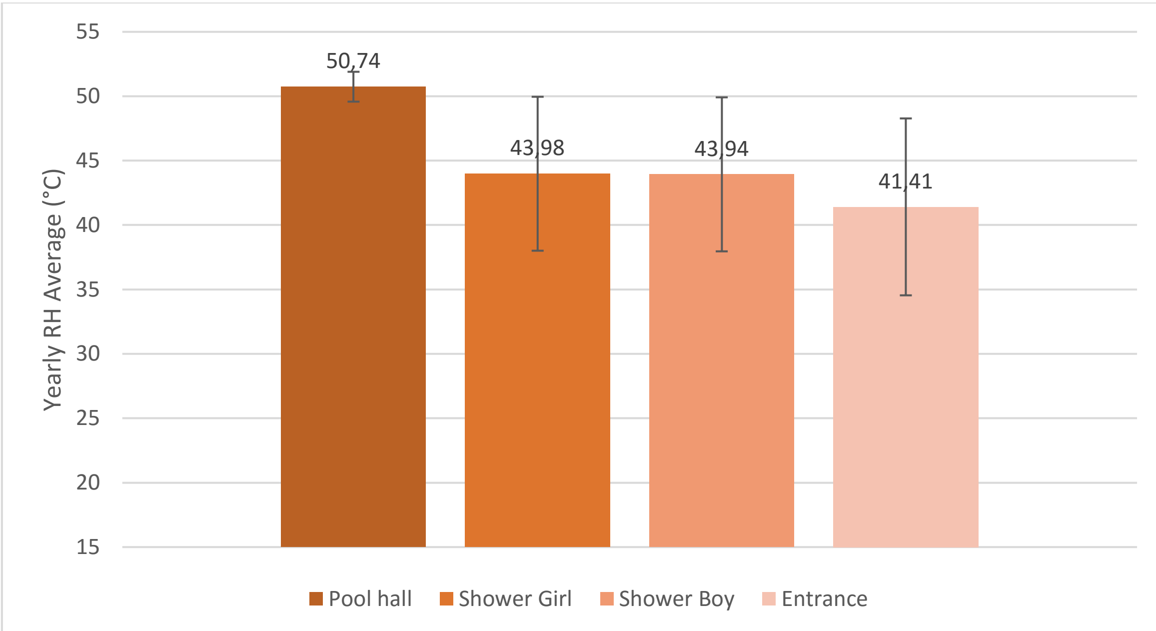



*Figure 3-17* Average relative humidity throughout the simulated year and respective standard deviations.

To study the efficiency of the negative pressure strategy implemented within the HVAC system, RH in all zones is drawn as in Figure 3-11. The influence of warm and humid air within the pool hall appears to be less accurate. Compared to the rise of 10-15% without the negative pressure strategy, RH in showers rises by a few percentage. Given, that temperature in the rooms haven't changed between the two simulations, the strategy implemented works. During operation hours, the internal air flow from the entrance to the hall increases (by roughly 15 L/s for the first week). Then, more fresh outside air penetrates through the structure at the entrance. Since at this period of the year the outside air is drier, RH in the entrance decreases a bit during operation hours.

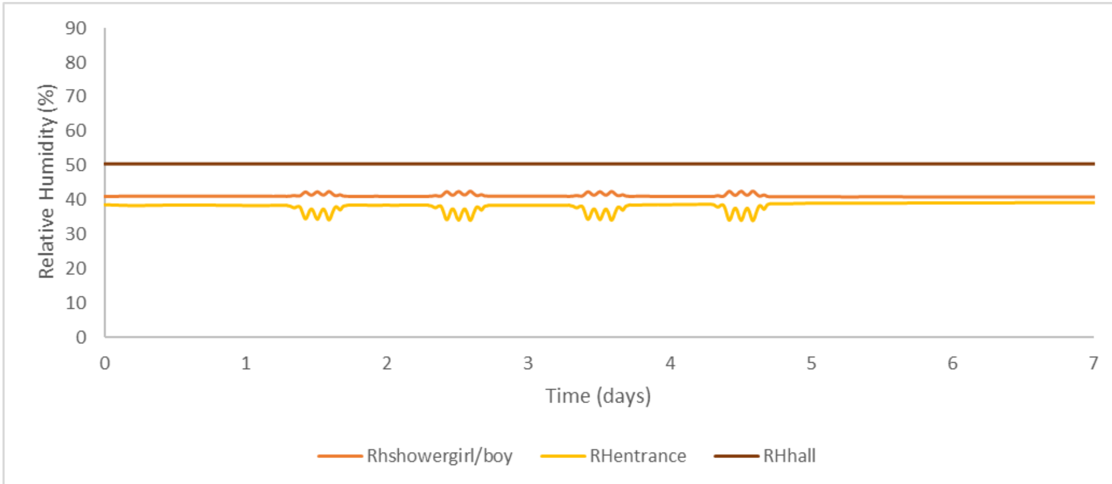



*Figure 3-18* RH in the different parts of the building for the first week of the simulated year.

One typical issue from this model that affects the relevance of the results compared to reality is the humidifiers consumption. Another approach is possible to avoid this matter. Instead of studying energy consumption in a general scale, one could focus on gain and losses within the building. By investigating how the building behaves, through e.g. envelope losses or mechanical air heat gain, the unrealistic humidifiers consumption is put apart. Figure 3-19 illustrates the two different systems.

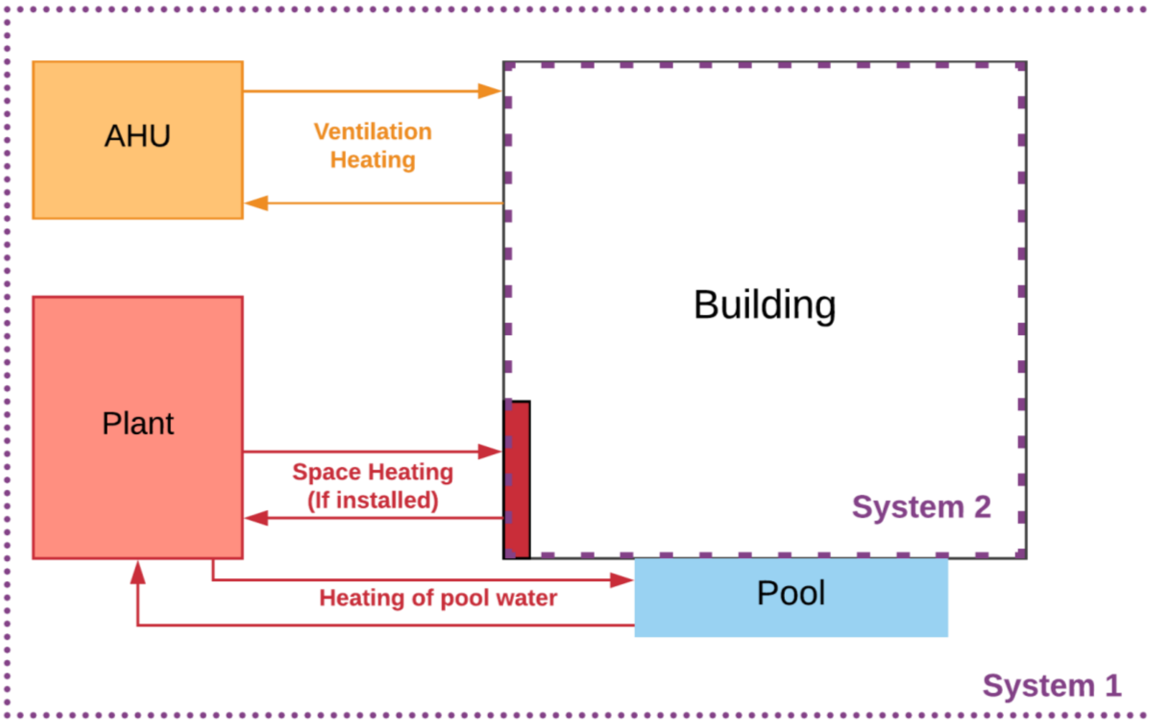



*Figure 3-19* Illustration of the two different scopes of energy study: system 1 and 2.

IDA-ICE offers two result data sets that comprehend system 2 behavior: *heat balance* and *energy balance*. The former includes both sensible and latent heat transfers whereas the latter which is only sensible heat. While the sensible heat estimation from *energy balance* makes sense, it seems the *heat balance* results do not fit with the pool&Ice-Rink extension. Results from both these latent and sensible heat transfers seem not to consider interactions with the swimming-pool, which are paramount. It is not fully surprising since programmers from IDA-ICE probably designed it without knowing a pool extension would be available later and it has not been updated ever since.

First, results from *energy balance* are investigated. Investigations are only made for the pool hall since it represents most of the building area and comprises all relevant processes. Figure 3-20 shows sensible energy use for each month of the simulated year. It comprehends the following elements (AB, 2013):

- Envelope & Thermal Bridges - Heat lost by conduction through external walls, roof, floor, and thermal bridges.

- Internal Walls and Masses - Heat gained through internal walls, ceiling, floor, and internal masses.
- External Window and Solar – Net heat gain through external windows as well as via transmission through pane and frame.
- Mechanical Supply Air – Heat supplied by mechanical ventilation.
- Infiltration & Openings – Heat supplied via air from leaks and openings.
- Occupants – Heat from people in the zone, excluding heat from perspiration.
- Lighting – Heat from artificial lighting.

Values are no longer comparable to the previous analysis since the system is different. For example, AHU could use a huge amount of energy to heat up the external air to the indoor temperature but since the indoor air and the air supplied would have the same temperature in this case: mechanical supply air would be equal to zero in *energy balance*. Sums of losses and gains are not equal Figure 3-20 because latent sources and thermal masses are neglected. The most favorable situation would be to get small losses and then small gains from the AHU through mechanical supplied air. It appears that the two main losses are heat conduction through the envelope including thermal bridges and heat loss via air through infiltration or openings. The latter actually accounts for 72% of the total losses in average through the year. Except lighting and heat brought by occupancy, the main source of energy to counteract the losses is the mechanical supplied air and then AHU heating. It amounts to 66000 kWh per year (different from the 122870 kWh for AHU heating).

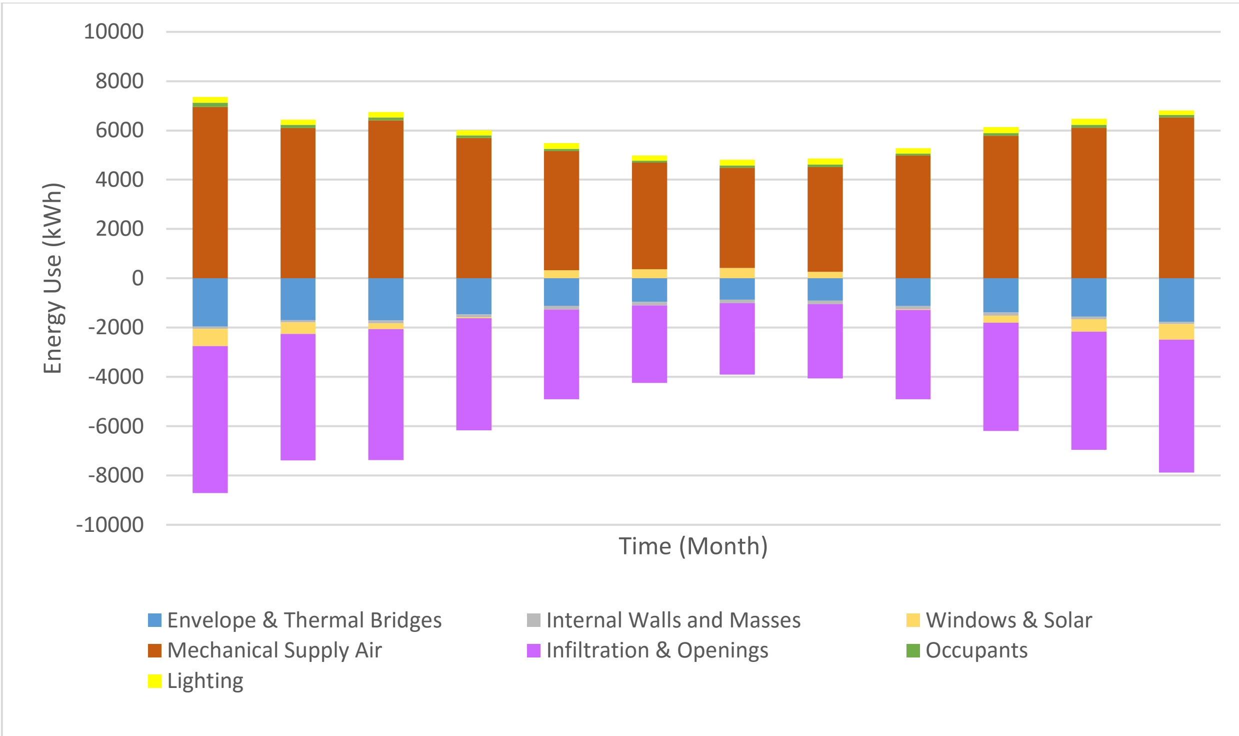



*Figure 3-20* Sensible heat balance within the hall for each month of the simulated year.

To describe more precisely these interactions, Figure 3-21 shows the four main interactions within the hall during the first week. Data given for the whole year can be found Appendix 7.6 along with the power gain curves due to occupancy and lighting. The power gain provided by the mechanical ventilation rises during occupancy hours and has the same shape as AHU heating. Indeed, during occupancy hours, volume air flow rate rises and since the indoor temperature is lower than the set-point value of 30°C (at least during this first week), supply air is delivered at 40°C and the gain coherently increases. During other period of the year and especially summer, it is possible that no increase occurs during occupancy hours. In fact, the volume air increases and more power should be delivered to the room but since the indoor temperature approaches 30°C, the supply temperature set-point decreases and so does the power gain brought by mechanical air to the room. As for the envelope losses including the thermal bridges, it simply responds to inside and outside temperature difference, and appears to be higher during operation due to higher indoor temperature. The same reasoning applies to infiltration and openings. Moreover, it is more sensible to turnovers (when swimmers leave or enter the pool) since it takes loss due to colder air in the showers entering the hall into account. Therefore, it fluctuates more.

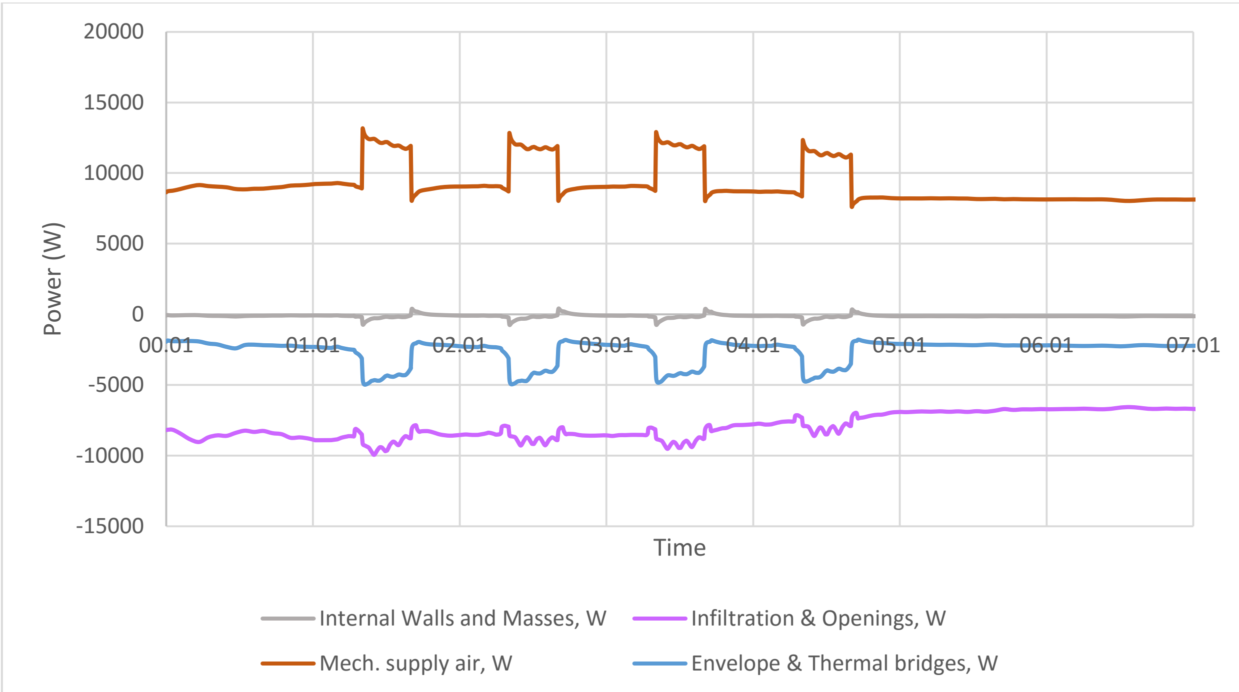



*Figure 3-21* Power influence of the four main interactions within the pool hall during the first week of the simulated year.

However, it is important to notice that this assessment is incomplete since the latent heat behavior of the pool as long as the energy consumption for heating of the pool water are discarded in the previous results. A comprehensive view of latent and sensible heat transfer within the pool hall is given in *heat balance* results. The set of data represent power curves of ten parameters. In this particular case, only seven are not constant equal to zero. Occupancy and lighting seem like the previous results in *energy balance* for sensible heat except that latent heat from occupants are considered which gives rise to higher power values for occupancy. Windows' influence are split in two parameters. The first gives the solar contribution going through the window while the other represents the conducted heat and retransmitted absorbed solar radiation. All previous parameters seem sound and reasonable. However, the three last parameters behave oddly and seem not to comply with reality. Figure 3-22 shows these three parameters during the first simulated week (AB, 2013):

- Heat from Air Flows - gathers power gains or losses from mechanical ventilation but also from infiltration or openings.
- Heat from Thermal Bridges – Simply heat losses due to thermal bridges.

- Heat from Walls and Floor – It accounts for heat conduction through the structure but also heat from internal masses. The heat conduction can be the net transmission or any internal heat sources such as floor heating. The pool influence should be included here.

It seems the values do not consider the influence from the pool. In fact, a large amount of latent heat is released during the day from the pool to the zone, $Q_{moist}$ can even reach 20 kW of latent heat during operation hours. Unfortunately, "Heat from Walls and Floors" decreases during operation hours. The explanation lies on the fact that without considering the pool influence, losses through the structure are enhanced during operation hours due to a higher indoor temperature. It is not shown here but the indoor temperature rise by 3°C during operation hours. This rise is lower after a few weeks when temperature is more stable.

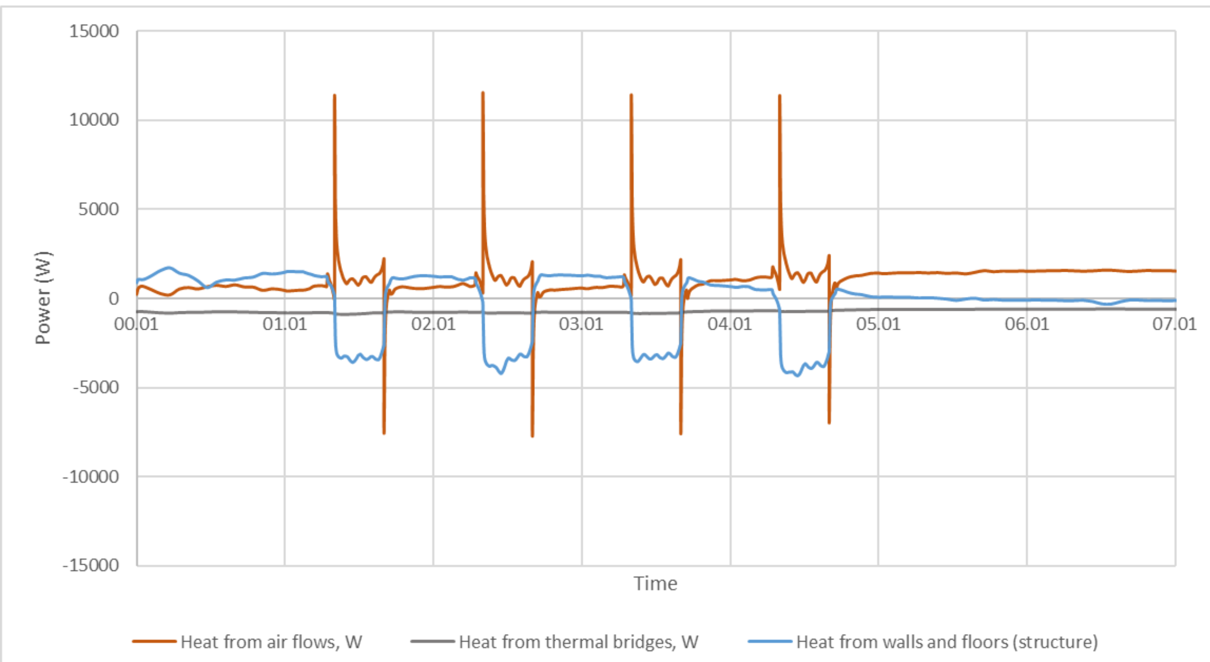



*Figure 3-22* Power curves of three parameters from *Heat Balance* during the first simulated week. Both latent and sensible heat are considered.

What is also surprising is when monthly averages of the power curves are calculated *Figure 3-23*. Losses through walls and floors are more important during summer. Since sensible heat decreases during summer, a reasonable explanation would be that a singular loss in latent heat occurs in the summer. In the case of the pool, it is the opposite: less latent heat is released during summer. It seems that if one add $Q_{zone}$ (sensible heat from the zone to the water due to temperature difference) to Envelope & Thermal bridges values from *Energy Balance* then a curve remotely similar to Walls and Floors is obtained. However there is still no clear answered and the problem remains unsolved.

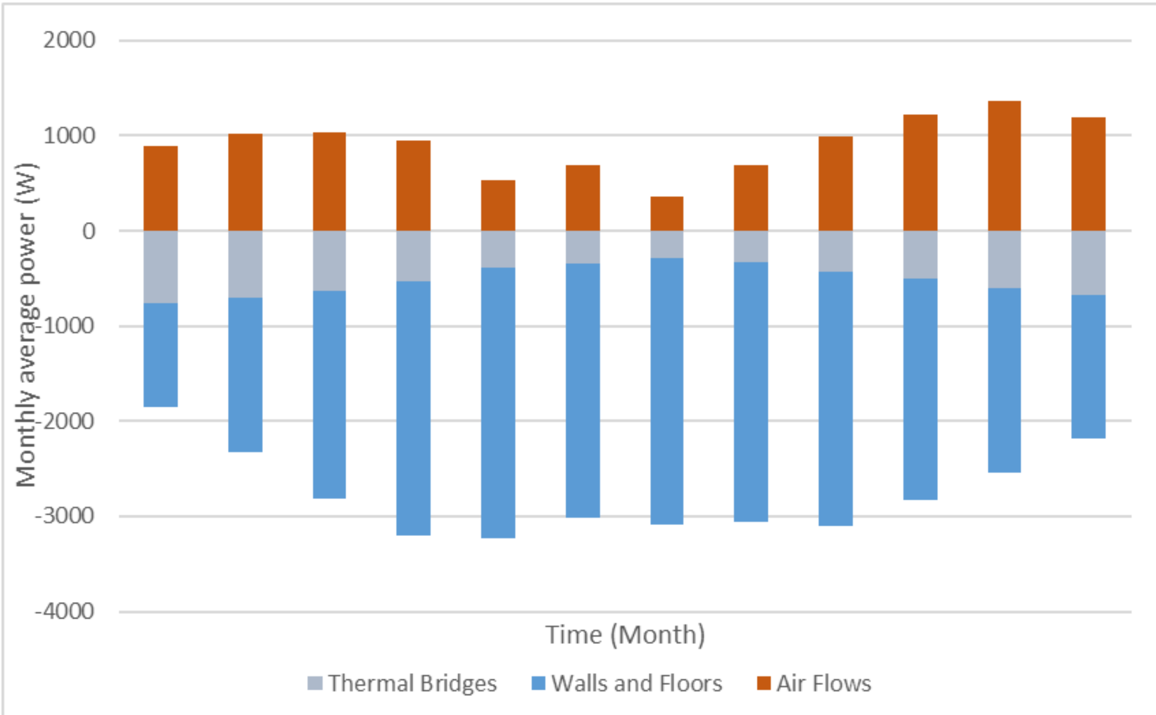



*Figure 3-23* Monthly average power of three processes within the pool hall according to IDA-ICE results.

## 3.2.2 Discussion

The detailed model performs well. Although the overall delivered energy throughout the year has slightly increased by 4% compared to the early stage model, 20.7% (78%*26.5% see analysis section 3.2.1) is now dedicated to humidifiers to maintain constant RH inside the building. It means that without the humidity control, which is unusual compared to actual pool systems, the yearly consumption would be lower and even closer to the current pool consumption statistics. By implementing a dehumidifier that would retrieve latent heat escaping as water vapor through ventilation ducts, the overall energy consumption of our model will be positioned perfectly within typical energy consumption for pool facilities in Norway, between 3000 kWh/$m_{ws}^2$ and 4000 kWh/$m_{ws}^2$. When the focus is on consumptions dedicated to heating, there are now two different sources: AHU heating and Heating of the pool water. The former represents a 122870 kWh yearly consumption and the latter 121012 kWh. Energy dedicated to pool heating and then directly caused by evaporation or obligatory water renewal corresponds to a significant share of energy use in this kind of facility. Apart from these two sources, energy is consumed into the ideal heater placed in the entrance to ensure acceptable temperature conditions. Though, its influence is limited and represents 4% of the total AHU heating power demand in average over the year.

When it comes to temperature control, the AHU system seems logically less robust than the ideal heaters in the early stage model. It gave rise to more temperature fluctuations, especially in the pool hall. Other rooms' temperature stay pretty accurately to the set-point temperature of 21°C. In the hall, the average is roughly 29°C with 1.41°C standard deviation. When the focus is put on humidity control, the opposite phenomenon appears. The pool hall shows more accuracy than the rest of the building. In fact, the rest of the building's relative humidity set-point is set to 40% which induces some difficulties in maintaining this level during the summer when the outside air humidity becomes higher. Average relative humidity levels are situated in a 40-45% range with a standard deviation around 6%. The bottom line is that indoor conditions are not perfectly constant but at least irregularities are known.

Energy used for humidifiers is unusual compared to real facilities and serves only the humidity control. To avoid taking into account this source of consumption, the scope of study can be reduced. Moving to the *system 2* (see Figure 3-19) allows an energy study that only focuses on the system behavior while clearing away all external consumption. It comprehends power effects from the mechanical supplied air, from the envelope, from the infiltration, and from the pool influence. However, for the pool influence, IDA-ICE does not provide any adapted results. This could be a downside of the pool & ice rink extension: the general software features have not been adapted to the extension. For instance, the result tabs do not provide the user with a comprehensive view of the pool influence and can be misleading. Also, the catalog of default HVAC components is not updated to the swimming pools and the ice-rinks. Further investigations are then needed. To highlight pool's influence on the zone, the sensible heat results from *energy balance* is used along with the following variables: $Q_{surf}$ (W), $Q_{moist}$ (W), $Q_{zone}$ (W), and HUMF (kg/s). The curves concerning heat transfers for the whole year can be found in Appendix 7.6 Figure 7-14 which gives a general view of their behaviors. An analysis of these three variables has already been made section 3.1.1. Figure 3-24 illustrates heat and mass transfer between the zone and the pool during the first week of the simulated year. Comparing the power values to the ones from *energy balance* related to sensible heat (see Figure 3-21), $Q_{surf}$ shows clearly higher values with peaks that can reach 25 kW in absolute value.

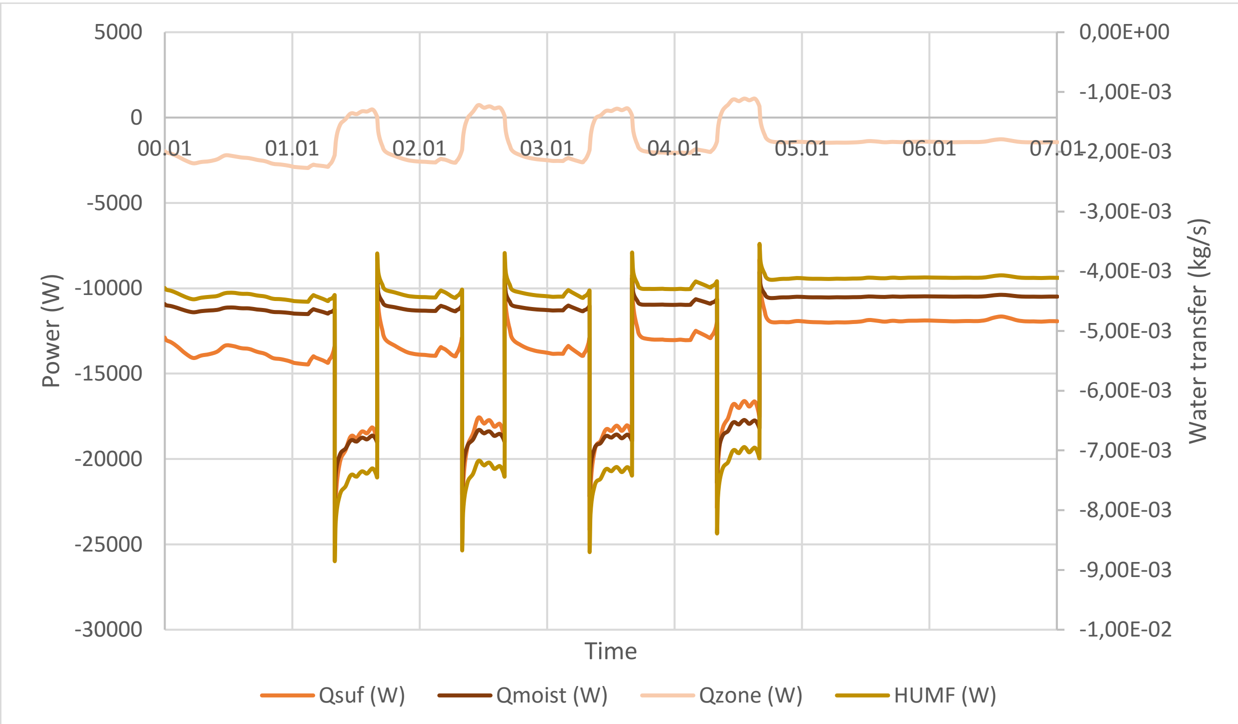



*Figure 3-24* Heat and mass transfer between the pool and the zone during the first week of the simulated year.

## 3.3 Sensitivity Analysis

The aim is to find parameters that weight the more on the energy consumption. It is very interesting to know whether e.g. the energy consumption is multiplied by two when the pool depth is multiplied by two. A sensitivity analysis of a function consists in varying its variables and inspect the change of the function output. If Equation *3-4* represents the function, the percentage of change of the output compared to its original value will be drawn in a graph against the percentage of change of the variables $p_{1,2,3,\ldots}$ compared to their original value.

$$Y = f(p_1, p_2, p_3, \ldots) \qquad 3\text{-}4$$

First, the study focuses on the total delivered energy in the early stage model depending on various parameters which will be described further. This model has a straightforward AHU but results shall give accurate knowledge on how the system behaves and reacts to a certain modification. Second, a sensitivity analysis will be conducted on the detailed model. The significant upside of this model is that indoor conditions are kept constant which makes the analysis even more relevant. However, the investigated system should not be the same. Instead of the total energy delivered, the focus will be set on the inner behavior of system 2 (see Figure 3-19).

### 3.3.1 Parametric Run Function in IDA-ICE

A parametric run can be attached to any IDA system. In this paper case, any parametric run will be attached to the ICE building.

It supports multiple types of simulation e.g. "energy simulation" or "heating simulation". The users must enter the list of modifications of the system. It can be variation of parameters or a change in the data structure. The user must as well enter the list of variables that should be reported in the summary.

The IDA extension "Parametric runs" allows then:

- Running a series of simulations with systematic variation of parameters and/or other changes in data structure (as inserting or replacing different objects).
- Running external optimization tool.
- Visualization of simulation.

Coupled with an optimization tool called GenOpt, IDA-ICE allows parameter optimization with more simplicity directly within the software. GenOpt directly acts on IDA-ICE parameters to find the optimized value of the desired variable. Even though this new tool seems promising, the investigations will only be based on sensitivity analysis performed by parametric run tool of IDA-ICE. It means, IDA-ICE will automatically run all simulations varying each parameter one by one. Instead of using the visualization part which is limited, an analysis tool will be used.

## 3.3.2 Parameters at stake

*Table 3-2* List of the gathered parameters that are studied during each sensitivity analysis.

| Parameter | Original Value | Values for Sensitivity Analysis | Comments |
|---|---|---|---|
| Orientation (°) | 0 | (90 180 270) | The 4 main directions. |
| Thermal Conductivity (W/mK) | 0.036 | (0.005 0.01 0.02) | Thermal conductivity of the only component of walls, roof, and floors. By changing this value, the overall insulation is changed proportionally. |
| Thermal Bridge (W/m²K) | 0.1 | (0.05 0.07) | Given per square meter of floor area. |
| Infiltration ($n_{50}$ coefficient $h^{-1}$) | 1 | (0.3 0.5 0.7) | NS3031 passive house standard requires $n_{50}$<0.6 $h^{-1}$. |
| Window (U-value in W/m²K) | 1.1 | (0.6 0.95 1.9) | Not only U-values are being changed but all parameters that come along with a window type.* |
| Pool Depth (m) | 1.5 | (0.75 2 3) | - |
| Pool Area (m²) | 100 | (25 50 75 112.5) | Due to the building size, it was not possible to have larger area values without changing the whole structure. |
| Activity Factor | 1 | (0.5 1.5 2) | These are the value during occupancy hours only. In each case, the AF comes down to 0.5 when the pool is not occupied. |
| Pool Temperature Set-point (°C) | 28 | (26 27 29 30) | Set-point for the indoor air temperature remains the same even though it should change along with the pool temperature. |
| Pool Hall Occupancy | 100% | (50% 150% 200%) | 100% corresponds to 22.22 people during occupancy and 0 when the swimming pool is closed. |

* The four types of windows are default types from IDA-ICE database and are, starting from the original one: Pilkington Arctic Blue (6ab-15Ar-S(3)6); Pilkington Optitherm S3 (4S(3)-15Ar-4-15Ar-S(3)4); Saint-Gobain T4-12 m, PLANITHERM ONE+ar; 3 pane glazing, clear, 4-12-4-12-4;

## 3.3.3 Early stage model

The sensitivity analysis concerning this model focuses on the total delivered energy. This variable gives a general understanding of the building's energy consumption. More in-depth results were also calculated to go into specific if need is. Results of all simulations bring the percentage of change in delivered energy according to the percentage of change in the parameter, as shown Figure 3-25. For instance, the figure shows that by dividing the pool area by two, it gives rise to a 12.5% decrease in delivered energy over the year.

One may for example notice that placing the window toward south would bring a 1% decrease to the total delivered energy over the year. Considering the huge amount of energy consumed over the year, 1% is not negligible. Also, it seems that pool depth does not bring any change whatsoever. Energy use remains similar whatever the depth, but a big influence that the depth could have is in the case depth compels the owner to raise water circulation or the amount of fresh water renewal. The glazing has no influence given the very small surface covered by the only window of the building.

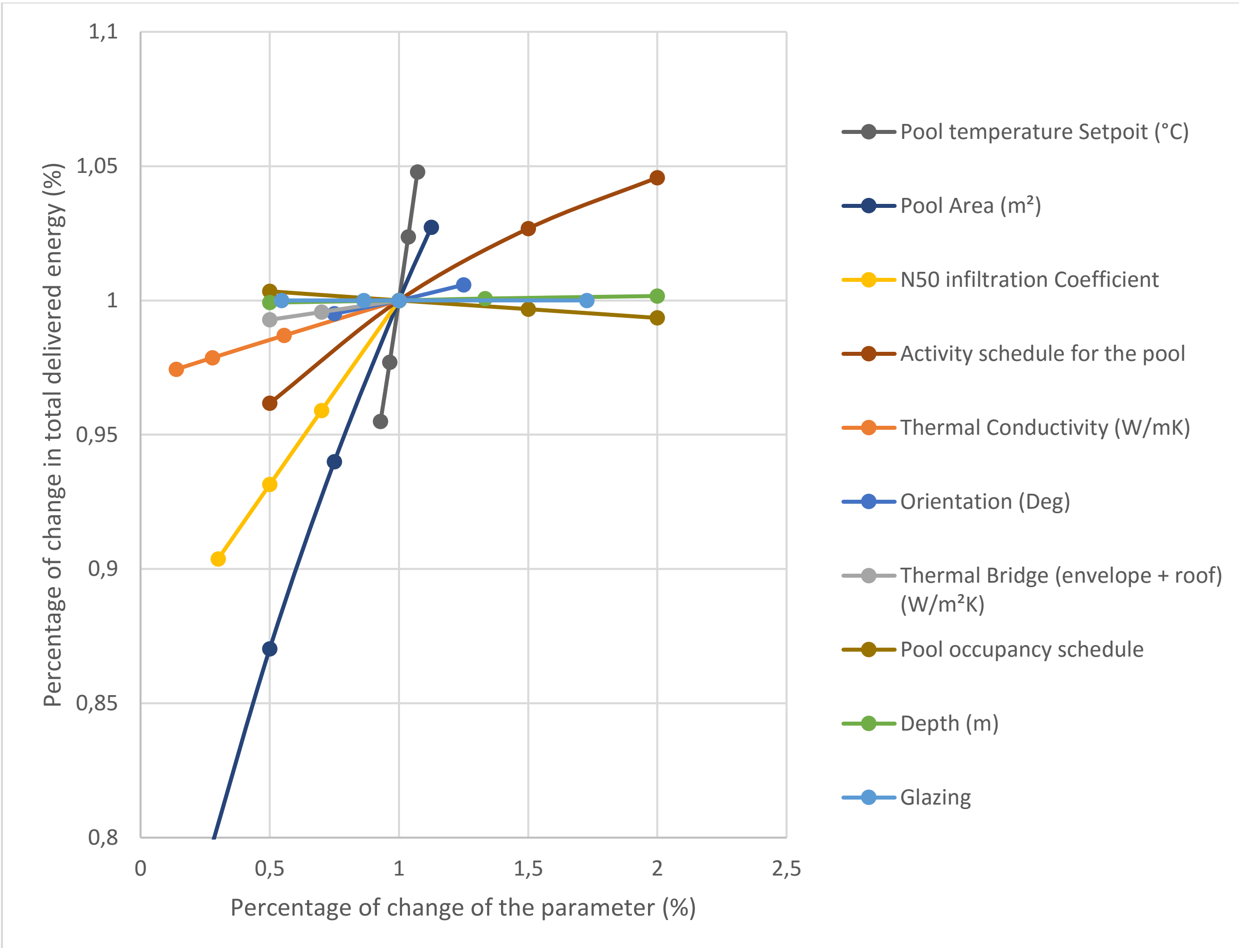



*Figure 3-25* Percentage of change in total delivered energy against the percentage of change of each parameter.

**Error! Reference source not found.** gathers the slopes of the curves drawn Figure 3-25, and then their respective influence on the delivered energy over the year. The most influencing parameter is the pool temperature with a 0.65 slope. The pool area and the coefficient of infiltration follows with slopes of 0.27 and 0.14 respectively. It appears that the envelope insulation does not have the greatest influence. It is due to the high energy consuming processes happening inside like evaporation or heating of the pool water. It is worth noticing that the activity factor has a larger influence than the envelope insulation even though the change is made only during occupancy hours. Whatever the simulation, the pool activity factor is 0.5 from 4 p.m. to 8 a.m., during week-ends, and during holydays.

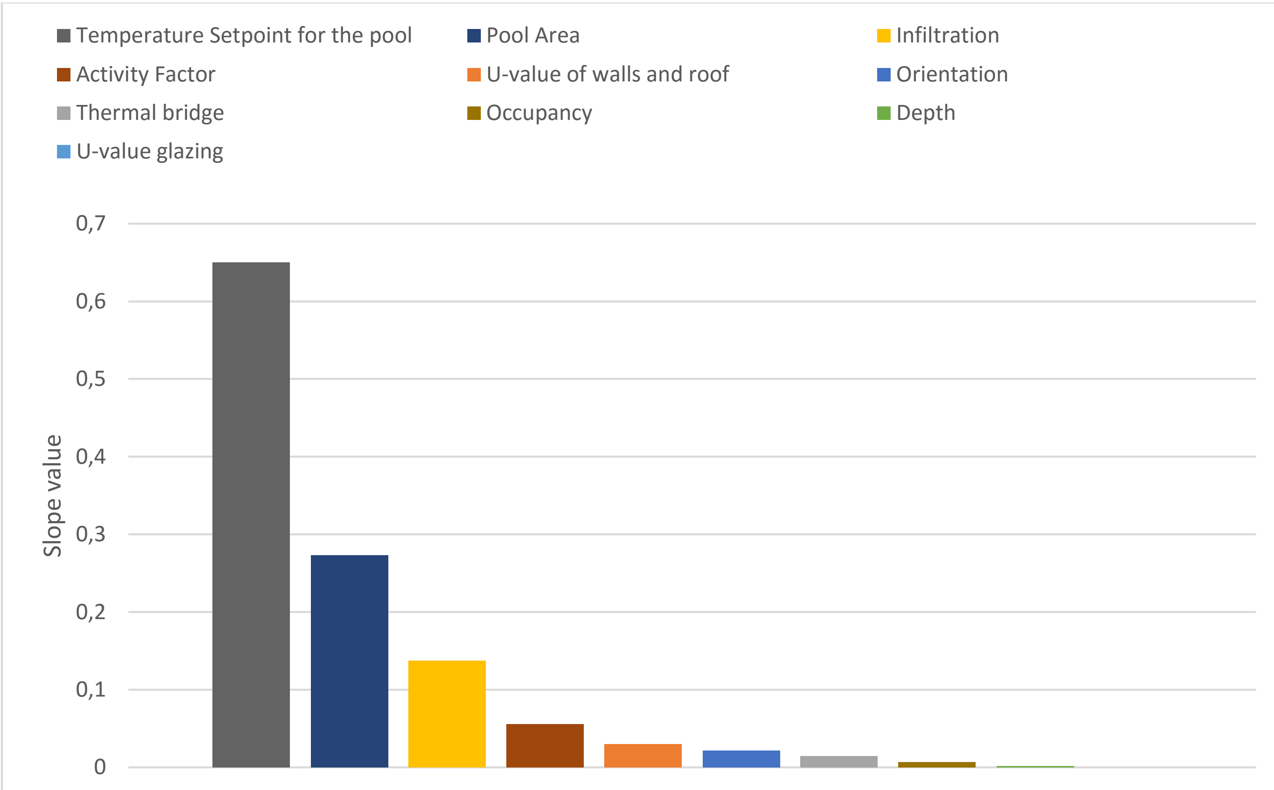



*Figure 3-26* Absolute slope value of the total delivered energy against one specific parameter. It represents the influence of each parameter on energy use.

The totality of all simulations gives different values for the total delivered energy solely due to a change in total fuel heating. Virtually all changes happen in the fuel heating consumption. Furthermore, the energy consumption delivered to the AHU heating coil remains quite the same throughout all simulations as well. Thus, parameters changing entails variations for almost only both space heating via ideal heaters and water-based heating.

What is interesting to see is what each parameter influences to change the overall consumption. Detailed studies have been conducted on water-based heating. Every percent of change in pool area brings 0.71% change in water-based power demand in average over the year. In the case of water temperature, every percent of change for the temperature set-point brings 3.38% of change in water-based power demand. However, when it comes to infiltration, the value is much lower. Every percent change for the coefficient of infiltration brings only 0.0041% of change for water-based power demand. Temperature set-point, pool are, and infiltration are the three most important parameters (see Figure 3-26). Nevertheless, they do not influence consumption the same way. When it comes to water-based heating (heating of the pool water), it appears that both pool area and pool temperature set-point have a lot of influence contrary to infiltration which has none. The full influence of infiltration lies on space heating via ideal heaters. In other terms, each parameter influences the total delivered energy consumption either via the consumption of ideal heaters for space heating, or via the heating of pool water, or both.

Indoor conditions have been studied as well. Temperatures in showers and in the entrance remain almost exactly the same thanks to the effectiveness and the huge power of ideal heaters and coolers. They stay between the desired values of 21°C and 24°C. Temperature in the hall is a bit more volatile. Figure 3-27 shows maximum and minimum temperature over the year according to each simulation. There is no precision on which column stands for which simulation but the goal is to have a general idea. Minimum temperatures are perfectly the same thanks to the ideal heater but since there is no ideal cooler installed, maximum temperatures are remotely different. Two peaks appear: one at 30°C when the window is oriented toward south and one at 28.58°C when the swimming pool is 26°C.

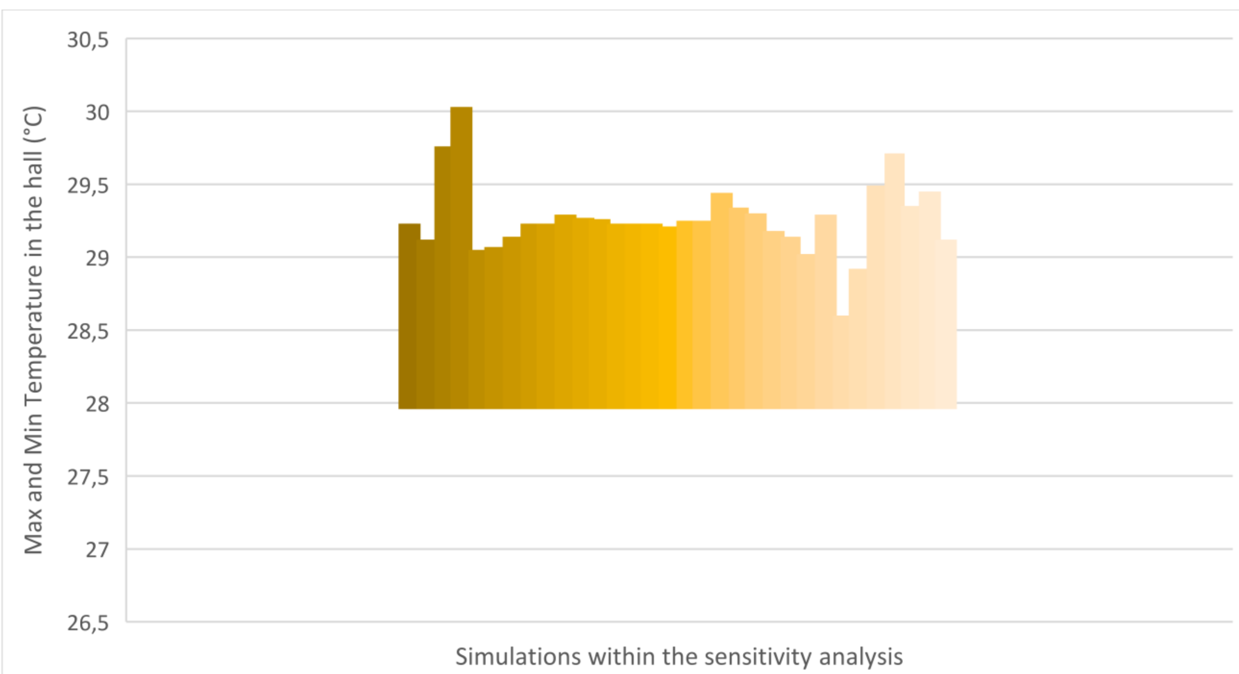



*Figure 3-27* Graphic illustration of both maximum and minimum temperatures for each simulation within the sensitivity analysis

Similarly to the figure above, Figure 3-28 shows maximum and minimum RH within the hall for each simulation. The range is very large since no control on humidity is implemented in this model. However the range of values stay pretty much the same throughout all simulations except when pool area changes. It triggers then a sizeable modification on the range of values for RH. As for ranges for the other zones, they are much more stable and not illustrated.

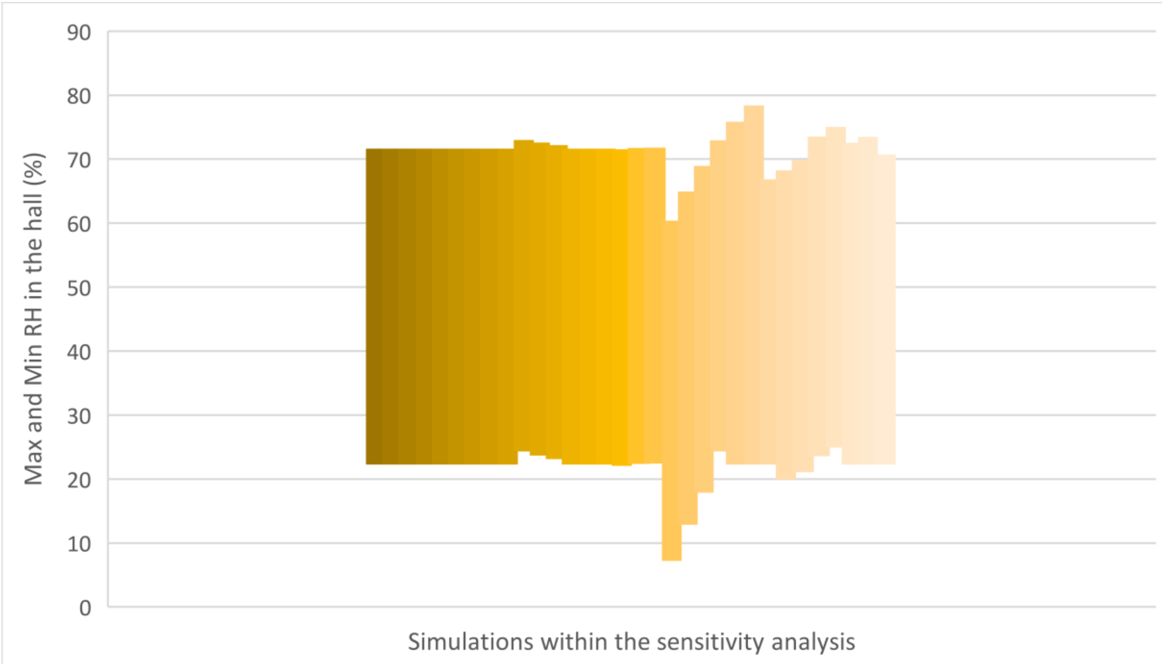



*Figure 3-28* Graphic illustration of both maximum and minimum RH for each simulation within the sensitivity analysis

## 3.3.4 Detailed Model

Two sensitivity analysis have been conducted. The first one is the same sensitivity analysis as before with the same inputs and outputs but with the detailed model. Due to the unusual consumption of humidifiers to keep indoor environment constant, a second parametric run has been run with different output but with the same set of parameters (inputs).

### *3.3.4.1 System 1*

In a first place, the focus is on the overall building and its consumption. Like in section 3.3.3, the analysis first lies on the variation of the total delivered energy throughout the year. Parameters are the ones in Table 3-2.

Figure 3-29 illustrates the results. Two parameters clearly weight more than all the others: Pool temperature set-point and $n_{50}$ infiltration value. Parameters, that used to have influence in the previous model, have lost their impact on energy consumption. For instance both activity schedule of the pool and the pool area do not influence energy consumption anymore. Every percent of change in these parameters brings barely around 0.01 percent of change in the final consumption. Figure 3-30 exposes more clearly each parameter's influence. The steeper the slope is, the greater the influence is. It confirms the fact that only two parameters remain influent.

When looking in detail results from this sensitivity analysis, it appears that these results are not relevant. Influences of certain parameters are diminished by humidifiers operation. For example in the case of the pool area, this parameter is no longer influent whereas in the previous model. What happens in this new model is that the pool area does proportionally influence the total consumption except for humidifiers consumption where it is inversely proportional. Indeed, when pool area decreases, evaporation decreases, moisture transfer to the indoor air decreases, and humidifiers operate even more to counteract it and maintain constant humidity. Since humidifiers operation serve to maintain indoor air conditions but is not usual in such facilities, results are erroneous and inaccurate.

An artificial way to correct it would be to inversely count humidifiers' influence. Indeed, a decrease in humidifiers' consumption means that more humidity has been released due to evaporation. Since moisture transfer is, in the opposite, a loss in sensible energy and then a rise in energy consumption, it makes sense to count inversely count variations in humidifiers' consumption. Figure 3-31 sums up the result after having made the correction. The influence of pool temperature set-point and pool area are greater but otherwise the graph obtained is similar to the early stage model. This proves even more the influence ranking of each parameter and the size of their relative influence between each other. Four parameters seem to stand out. The four most influent parameters are, from the first to the fourth:

1. Pool Temperature Set-point
2. Pool Area
3. Infiltration ($n_{50}$ value)
4. Activity Factor

Sizes of their influence are clear-cut, except for infiltration and activity factor which are relatively similar. What comes afterwards are structure features like insulation U-values or thermal bridges.

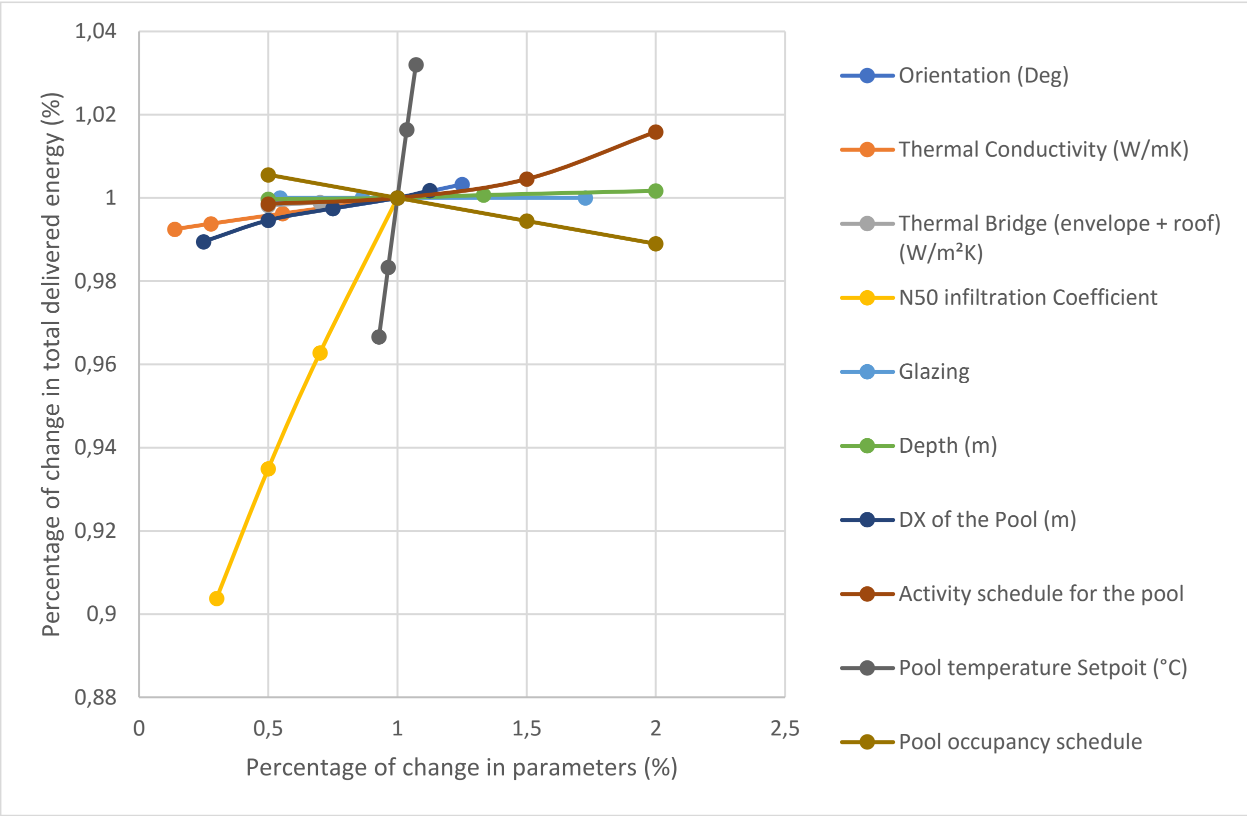



*Figure 3-29* Variation of total delivered energy against variations of each parameter for the detailed model.

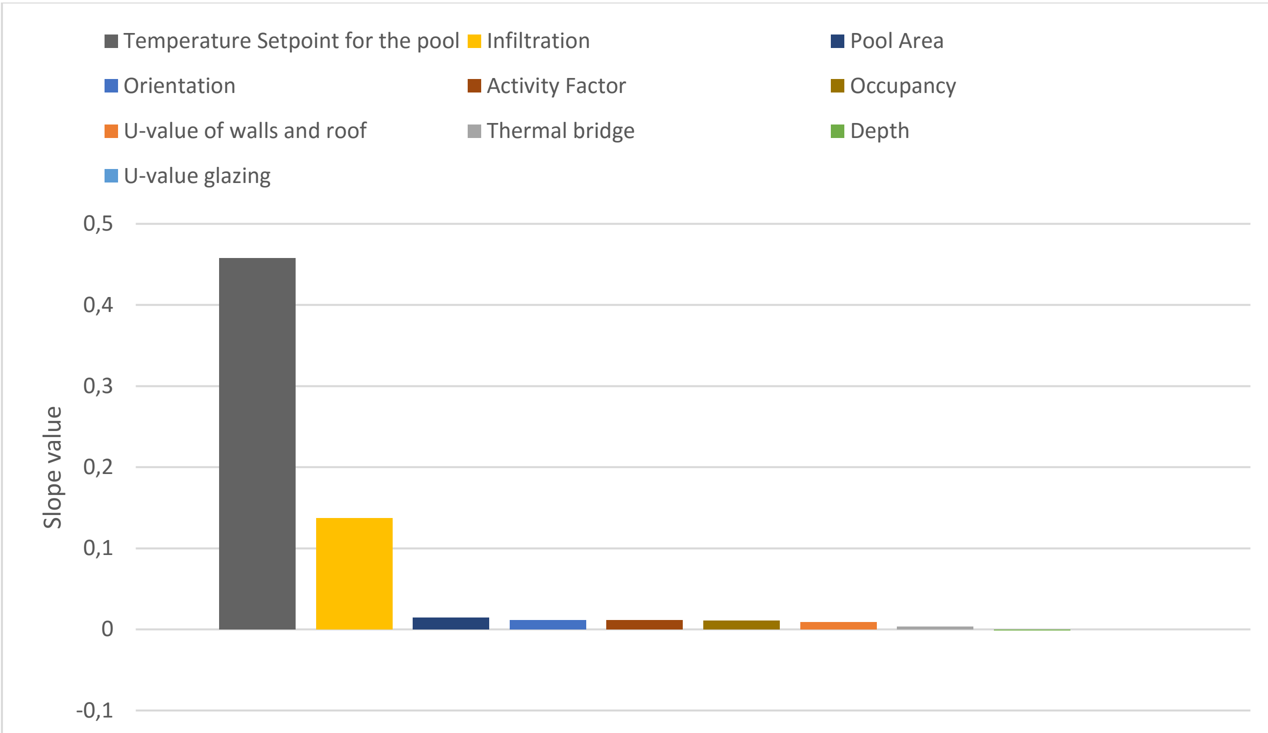



*Figure 3-30* Absolute value of slopes in *Figure 3-29* for each parameter.

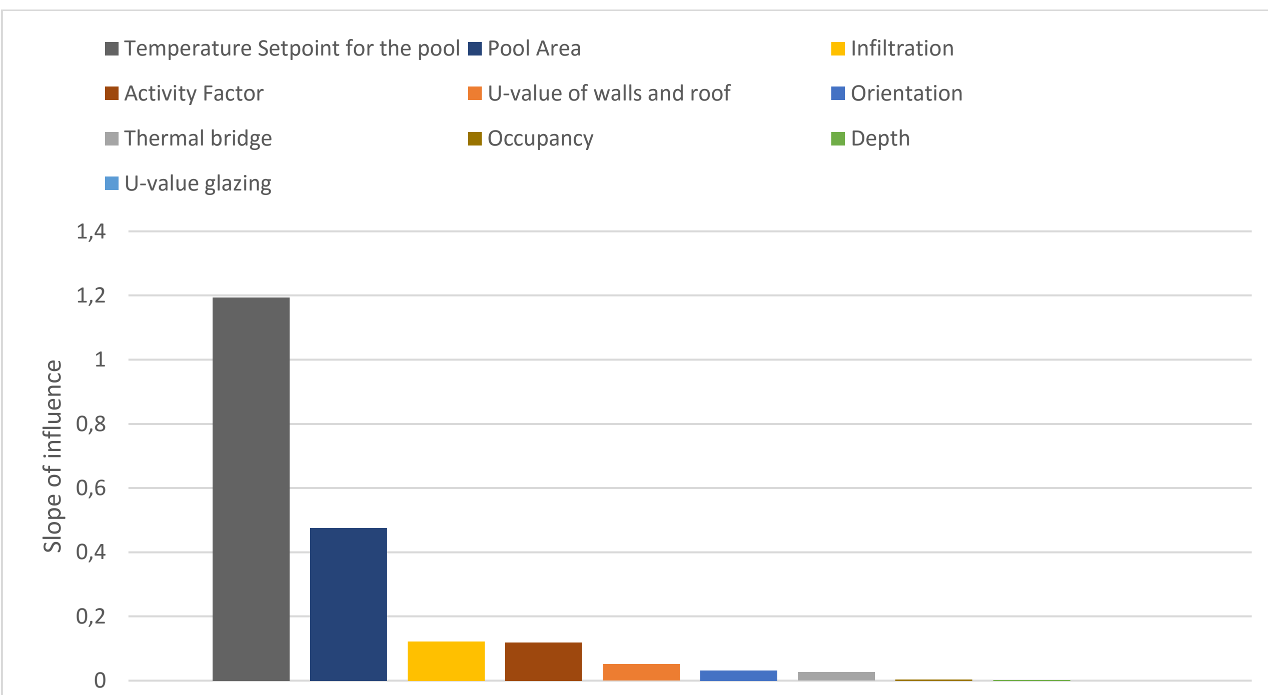



*Figure 3-31* Slopes from sensitivity analysis for each parameter considering an inversely proportional correction concerning humidifiers' consumption.

### *3.3.4.2 System 2*

Similarly to the analysis made section 3.2.1, the inner behavior of the pool hall is studied. Results for sensible heat from *energy balance* in IDA-ICE are used along with $Q_{surf}$ values to make a general assessment of the zone. Contrary to the previous analysis, there are three different outputs. These three outputs are:

- Gains: Positive sensible heat released to the zone (pool hall). It is formed of the heat brought by the mechanical supplied air, the sensible heat from occupants, and the heat from lighting. Heat due to occupancy and lighting stays virtually constant, and variations come from the mechanical supply air input.
- Losses: Sensible energy given away from the hall. It is constituted of energy loss due to infiltration, to internal walls conduction, to transfer through windows, and to thermal transfer through the envelope accounting thermal bridges. In the case of windows loss, it is pretty special since the solar contribution is considered and the "loss" through the window can become a "gain" during summer. In such a case, it still belongs to the loss category and is counted negatively. Except that, variations are mostly caused by variation in envelope energy losses and infiltration.
- $Q_{surf}$: The sensible and moist heat transfer from the pool to the zone. The value is composed of a negative sensible heat loss (mainly but can be gain) due to temperature difference between the indoor air and the pool temperature (precisely the surface temperature). A positive moist heat transfer composes it as well, it is the latent heat transfer due to vapor released to the room. The moist heat transfer is much larger in absolute value than the sensible heat transfer, around ten times larger.

By changing parameters during the sensitivity analysis, variations from these three outputs were recorded. Considering one output, lines are drawn and they represent the percentage of variation of this output against the percentage of variation of each parameter. Recovering the slopes of these lines gives each parameter's influence on the very output chosen.

Figure 3-32 sums up all slopes values, and all values can be found Appendix 7.7. This figure is very important. The same four parameters as in previous sensitivity analysis appear to stand out. It seems that the pool temperature set-point influence is much more important than the rest. Three other parameters have a relatively high influence as well: pool area, infiltration, and activity factor. It is important to notice that the activity factor is only changed during operation hours between 8 a.m. and 4 p.m.

The first column is related to pool temperature set-point, it means this parameter has a very large influence on gains, losses, and $Q_{surf}$. When the temperature set-point is increased, gains through the mechanical ventilation significantly decreases while $Q_{surf}$ significantly increases. Precisely, every percent of increase in pool temperature set-point compared to 28°C brings 4.81% rise in $Q_{surf}$, 0.44% rise in losses, and -1.44% decrease in gains. Gains decreases since higher pool temperature either diminish heat transfer from the air to the water or increase a positive input of heat from the water to the air. This gains decrease is beneficial and larger than the losses increase but it also entails significant increase in water-based heating (see Figure 3-33). Pool area appears to have no consequences on sensible gains and losses, but does have a big impact on energy consumption for heating of pool water. Contrary to pool area, $n_{50}$ infiltration coefficient has influence mainly on gains and losses which is very logical since higher infiltration gives higher losses which lead to higher gains to maintain the indoor environment. Thus, even though a parameter has a large proportional influence on gains it is not positive since it means that the need in heat brought by mechanical ventilation is higher. Activity factor plays a decisive role in energy consumption. Similarly to pool area, it influences solely $Q_{surf}$.

Purpose of Figure 3-33 is to enlarge the system (*system 2*) and include the pool. By including the pool, the output replacing $Q_{surf}$ naturally becomes “water-based heating”. The reason of this new chart is that many parameters relate to the pool which is external of *system 2*. However, results are highly correlated.

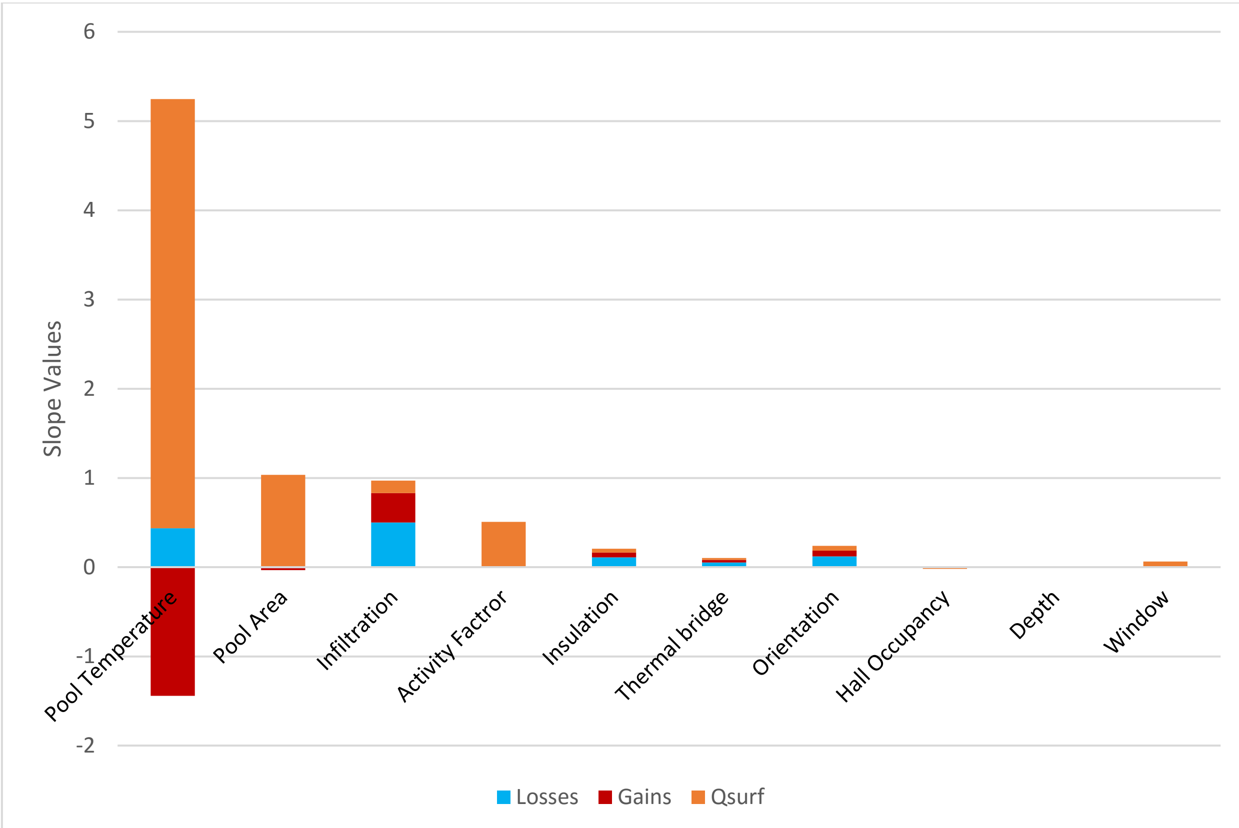



*Figure 3-32* Values of sensitivity analysis slopes for losses, gains, and $Q_{surf}$ according to every single parameter.

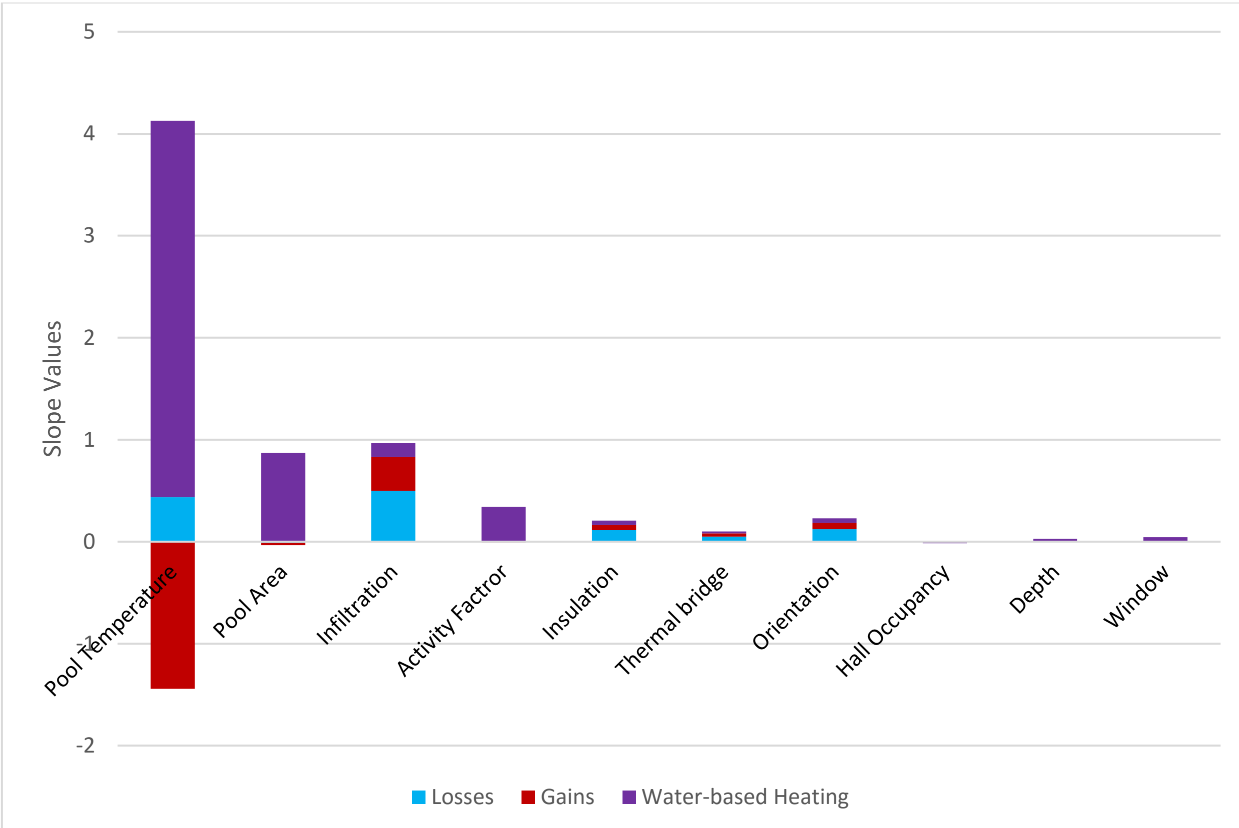



*Figure 3-33* Values of sensitivity analysis slopes for losses, gains, and water-based heating according to every single parameter.

Since the core of these sensitivity analysis relies on the fact that indoor environment is kept constant, it is very important to investigate both temperature and relative humidity inside the zone. Figure 3-34 shows yearly average temperatures and their respective standard deviations for each simulation in the sensitivity analysis. Similarly to the original simulation, temperatures seem pretty volatile with a standard deviation of roughly 1.4°C in average. Except for the case with low infiltration where the average temperature leaps to 29.8°C, indoor temperature feature can be considered constant throughout all simulations. As for relative humidity, Figure 3-35 shows the yearly average relative humidity along with their respective standard deviations. Parameters related to the pool have an impact on the average relative humidity, especially the activity factor and then the way the pool is used.

A change in indoor conditions would give untrustworthy results. For example, it is important to take into account the average temperature obtained. If the value is higher than the base case, it means the total amount of energy consumed would become lower for equal indoor conditions. The same reasoning is true for relative humidity only in the case a dehumidifier is installed. Indeed, higher average relative humidity means that the zone is provided with higher amount of latent heat. However, if this latent heat is to have an impact somehow on energy consumption, a dehumidifier is needed to recover this potential. Otherwise this surplus of latent heat becomes an inevitable loss.

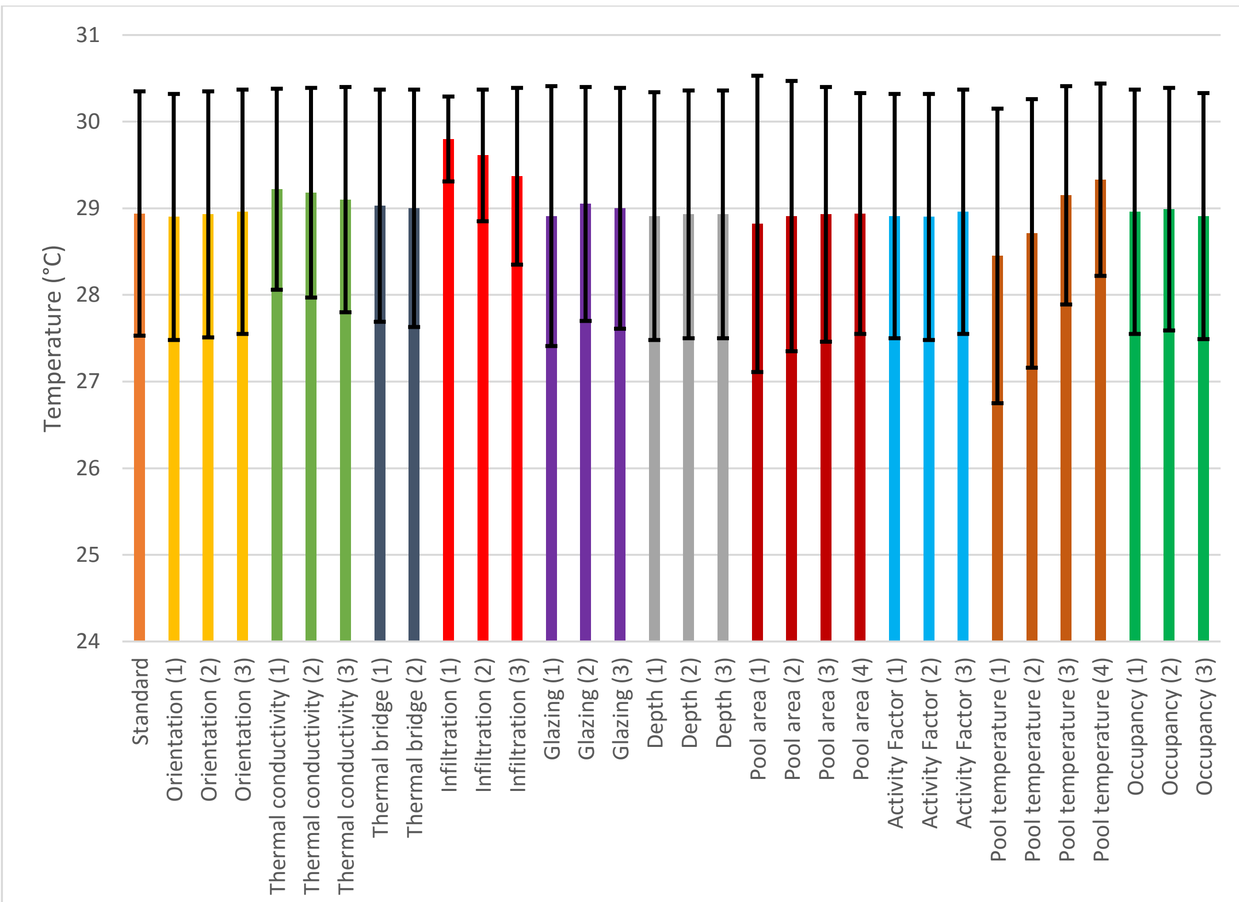



*Figure 3-34* Average Temperature in the hall and standard deviation in each simulation as part of the Sensitivity Analysis.

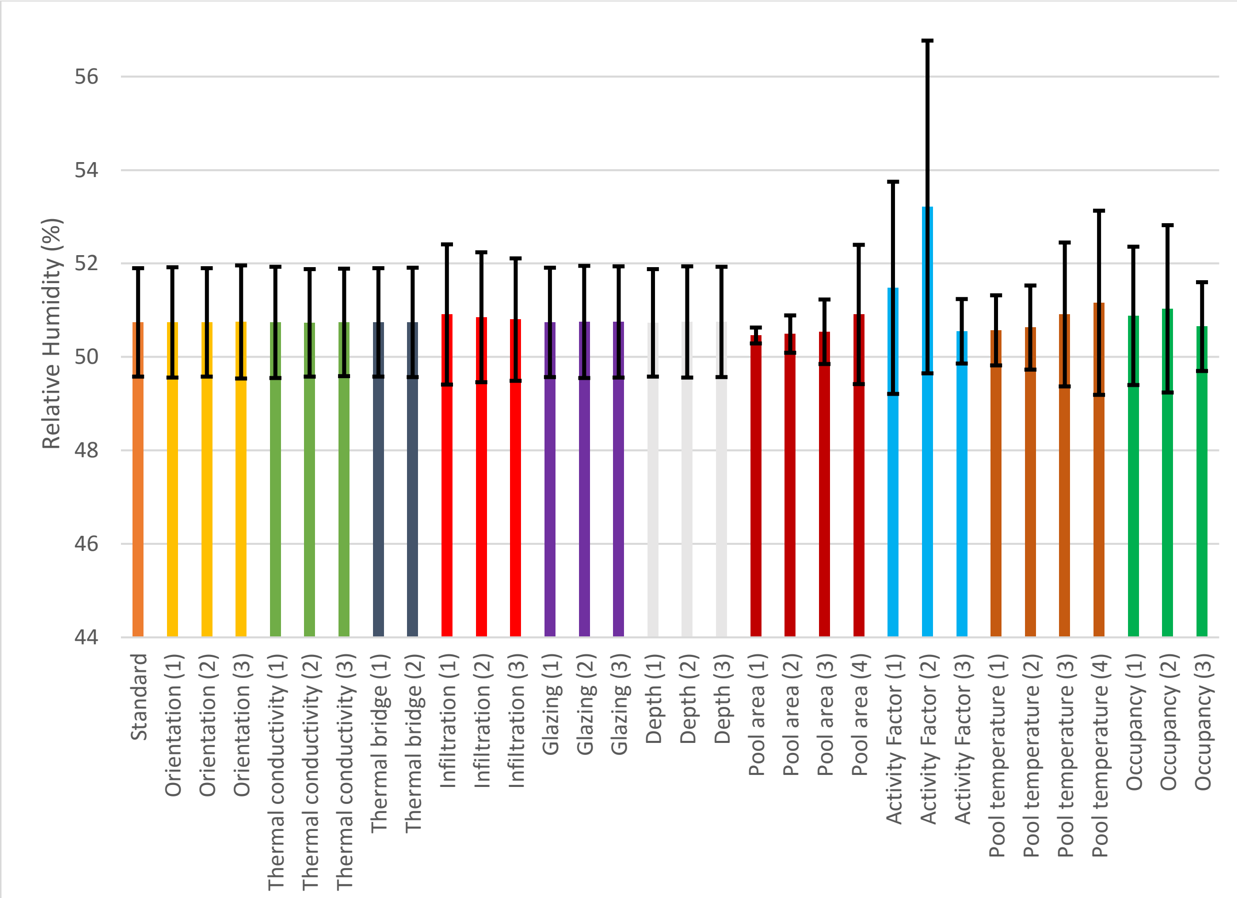



*Figure 3-35* Average relative humidity in the hall and standard deviation in each simulation as part of the Sensitivity Analysis.

# 4 Comparison with Jøa's data

The building of interest is the newly inaugurated and commissioned multi-purpose sports facility of Jøa. This building was opened autumn 2016. The building is located at Jøa, an island of the municipality of Fosnes, at 64.6°N, 11.2°E, and 65 meters above the mean sea level. The sport facility, owned by the municipality, contains several functions beside the swimming pool facility, such as:

- A sport hall (handball, indoor football, basketball…)
- Shooting range
- Library
- Café
- Fitness room
- Central unit for heat supply to the nearby school building
- Undergoing work achieving an outdoor ice-rink

This study will mainly focus on energy consumption related to the swimming hall. The actual hall is 266 m² (13.7 m per 19.43 m) including a 100 m² swimming pool (8 m x 12.5 m). This pool is purposely the same size as in IDA-ICEs model studied before.

Table 4-1 sums up statistics over a 3 months period (from 26.02 to 17.05) retrieved from Ole Smedegård as part of a PhD project (Ole Øiene Smedegård, 2018). These data account for a total number of visitors of 707. Then the specific total energy use related to the swimming hall amounts to 44.8 kWh/visitor. The specific total energy for the whole building in the case of the early stage model amounts to 23 kWh/visitor. To get this number, the 5 weeks of holydays and the week-ends were taken into account to get the total number of visitor knowing that every working day, 88 (4 x 22) people use the pool. The number is lower than the statistics. Energy consumption may be higher but the model considers a substantially higher number of occupant everyday than the poor occupancy of Jøa. A total of 707 throughout the period represents around 10 persons per day of occupancy which seems very low. The reason might lie on the fact that population density in Jøa's area is low.

The period during which statistical data are measured represents 21.9% of one year. The thermal energy released in the AHU by the heating coil is hardly comparable between the value in Table 4-1 and the value in the IDA-ICE model. In fact, the value in the table below is obtained during operation with an AHU where an integrated heat pump is installed. This heat pump recovers the latent heat content in the exhaust air by condensation after the recovery wheel and releases heat to the supply air via the evaporator. Thus, heating coil's consumption is significantly reduced and cannot be compared to an AHU without an integrated heat pump working as a dehumidifier. What can be compared are both the thermal energy pool circuit and the energy use due to DHW. Considering 21.9% of the total energy use for DHW throughout the year in the early stage model gives a 15435 kWh DHW consumption. This value is much higher than 2556 kWh (see Table 4-1). Again, number of visitors in Jøa is very low and is around eight times lower than the one in the early stage model. By multiplying by eight the DHW consumption of 2556 kWh, the two figures are relatively comparable. When it comes to thermal energy pool circuit, an estimation of the total energy use for the year is made with the power demand curve of water-based heating. Thus, 21.9% of the estimated consumption for the early stage model amounts to 26826 kWh which is roughly three times higher than the 9578 kWh retrieved from the energy meters at Jøa. Nonetheless, there are few possible explanations. First, the integrated heat pump within the AHU pre-heat the fresh water before it goes to the pool circuit. Therefore, a part of the consumption is already covered in the case of Jøa. Second, the pool in Jøa is covered every night which prevent any evaporation during night. This is equivalent to set the activity factor to zero outside the occupancy hours, and as shown before, this has a non-negligible impact on water-based heating of the pool. Third, the very low occupancy of the pool in Jøa influence the evaporation load as well. When no one uses the pool the mass and moist transfer is lower.

Due to a short time framing, further work need to be done on this matter. Building construction features as well as weather file need to be adjust to the ones at Jøa in order to get valuable values.

*Table 4-1* Statistics retrieved from Ole Øiene Smedegård (2018) concerning the period 26.02.18-17.05.18.

| | Energy Consumption (kWh) | Comments |
|---|---|---|
| Electricity supplied to the building | 74655 | Includes all electrical consumptions for the whole building. |
| Electricity supplied AHU swimming hall | 6100 | Electricity consumption related to the AHU, it includes the AHU heat pump compressor consumption |
| Electricity supplied pool circuit | 10376 | Pumps for circulating water but also cleaning and rinsing equipment. |
| Thermal energy AHU swimming hall | 3089 | Thermal input from the heating coil only. It does not include energy brought by the heat pump or the heat recovery system. |
| Thermal energy pool circuit | 9578 | Heating of pool water. |
| Total energy supplied swimming hall | 29143 | Sum of the four last consumptions |
| DHW | 2556 (3.6/visitor) | Thermal and electrical heating of hot water. |

# 5 Conclusion

The aim of this study was to design and investigate a model of a standard swimming pool facility. An extensive literature research was conducted to find characteristic features of swimming facilities when it comes both to the technical installation layout and the energy consumption. All challenges specific to swimming facilities were discussed, and especially the evaporation load and its drawbacks regarding energy consumption and power demand. Along with the help of Norwegian standards, the model was shaped to be as standard as possible. Thus, energy consumption throughout the year remains in the average. For instance, the building's structure of the standard model was not particularly insulated and did not follow the NS3031 passive house standard.

First, an early stage model was built. This model shows all characteristics of swimming facilities except its AHU and space-heating system. The implemented AHU is completely standard, the unit provides the building with a constant airflow rate at a constant temperature. To cover space-heating need, ideal heaters are installed. This model shows an energy consumption of 4593 kWh/$m_{ws}^2$ (per square meter of water surface). The part related to space-heating and heating of pool water represents 76.3%, and in general the model seems to be pretty trustworthy and comparable to the statistics. Through the results, the significant influence of the swimming pool has been highlighted. Due to moist and mass transfer from the pool water to the indoor air, the energy need in heating of pool water is roughly equivalent to the need in space-heating. Power demand for such a process is very erratic, and peaks reaches higher values than space-heating. In this case, peaks can reach 44.3 kW compared to those from space-heating whose maximum is 36 kW.

Second, a detailed model was built from the previous one. This model includes now an advanced AHU along with a ventilation strategy. All space-heating needs are covered by the mechanical ventilation. The core of this model is to keep indoor conditions constant, that is to say temperature and relative humidity. The total delivered energy amounts now to 4775 kWh/$m_{ws}^2$. The consumption is higher but nearly a quarter of it is due to the humidity control strategy and does not represent actual operation in real buildings. The idea is to reduce the scope and take the indoor air of the pool hall as the new system. This system behavior has been studied through sensible heat exchanges related among other to the structure, the mechanical supply air, and through latent heat exchanges with the pool. Again, the strong influence of the pool is highlighted. In average over the year, the pool releases 12.7 kW of latent heat to the zone which is large compared to the 7.4 kW released in average by the mechanical ventilation.

Third, three sensitivity analysis were run. The aim is to find which parameters have influence when it comes to energy consumption. The first sensitivity analysis deals with the early stage model. The second and the third are both run with the detailed model but with different systems at stake. From these three analysis, four parameters appear to stand out: the pool temperature set-point, the pool area, the $n_{50}$ infiltration coefficient, and the activity factor. They all affect different sections of energy consumption like space heating, ventilation heating, or heating of pool water. The building structure remains secondary compared to these four parameters.

Finally, a comparison was established between energy use in the IDA-ICEs model and the actual energy use at the swimming facility of Jøa. Due to an integrated heat pump installed within the AHU at Jøa, the thermal energy provided by the AHU cannot be compared to results from IDA-ICE. However, both DHW consumption and consumption for heating of the pool water are comparable and energy use at Jøa showed much lower values. Due to a short time framing, further study should be conducted. One need to build a relevant model with the same design features along with the right weather file to get reliable results. Energy consumption might remain higher than the data retrieved from the actual pool since it seems the pool at Jøa is not being used by many people. Besides, the $CO_2$ heat pump might be oversized since the power demand is not as high as expected.

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# 7 Appendix

## 7.1 Occupancy phases and detailed schedule

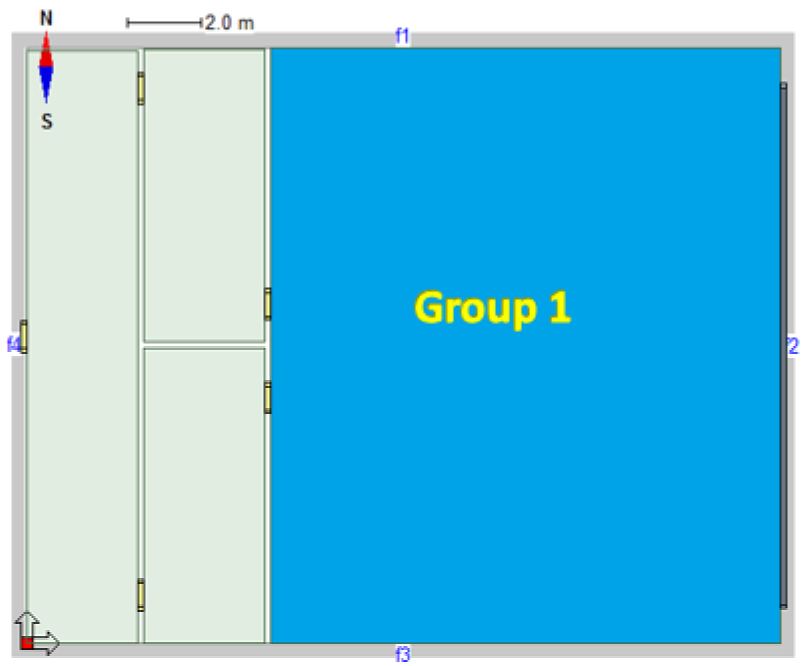


*Figure 7-1* Occupancy phase 1.

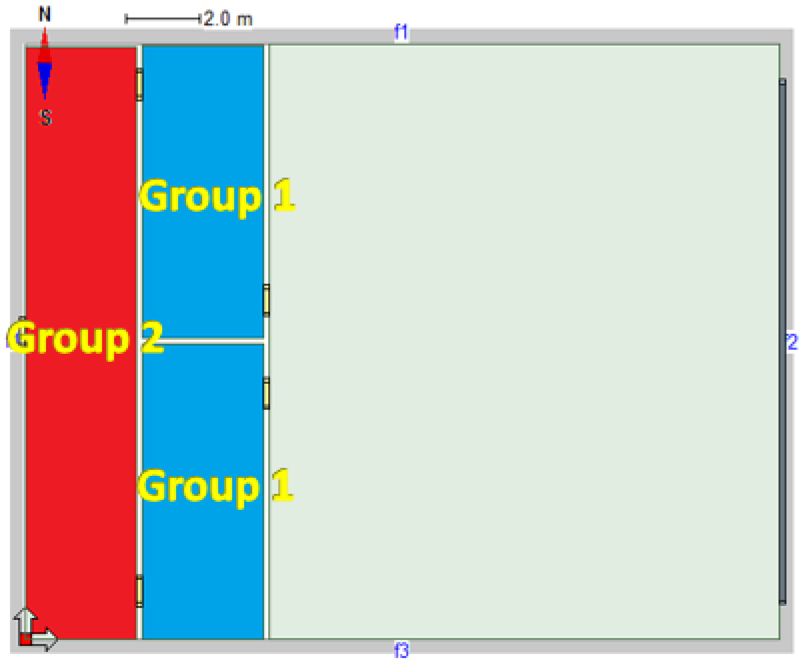


*Figure 7-2* Occupancy phase 2.

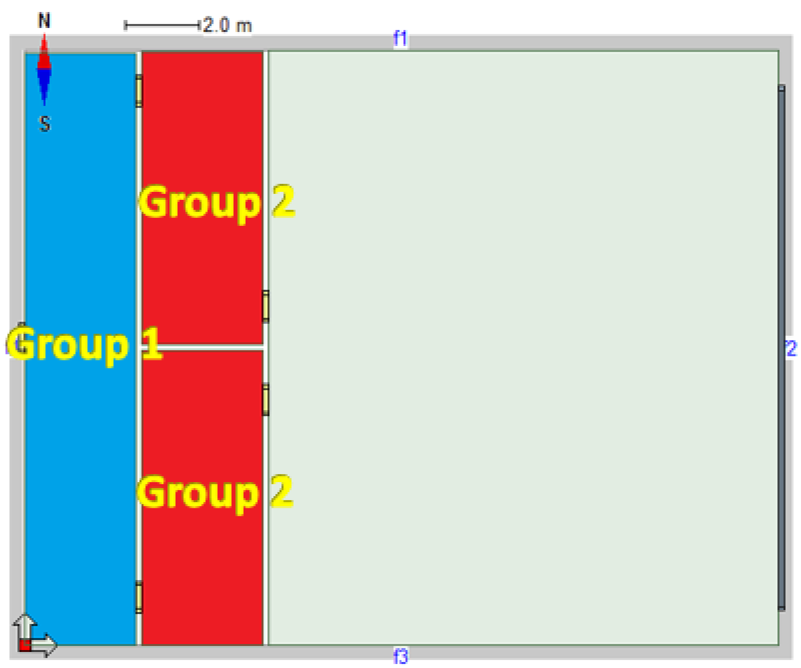


*Figure 7-3* Occupancy phase 3.

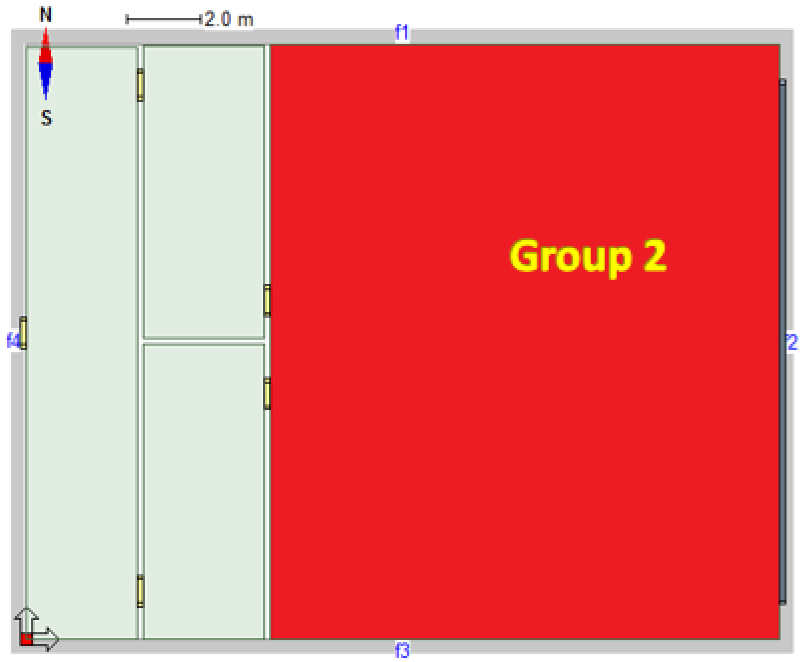


*Figure 7-4* Occupancy phase 4

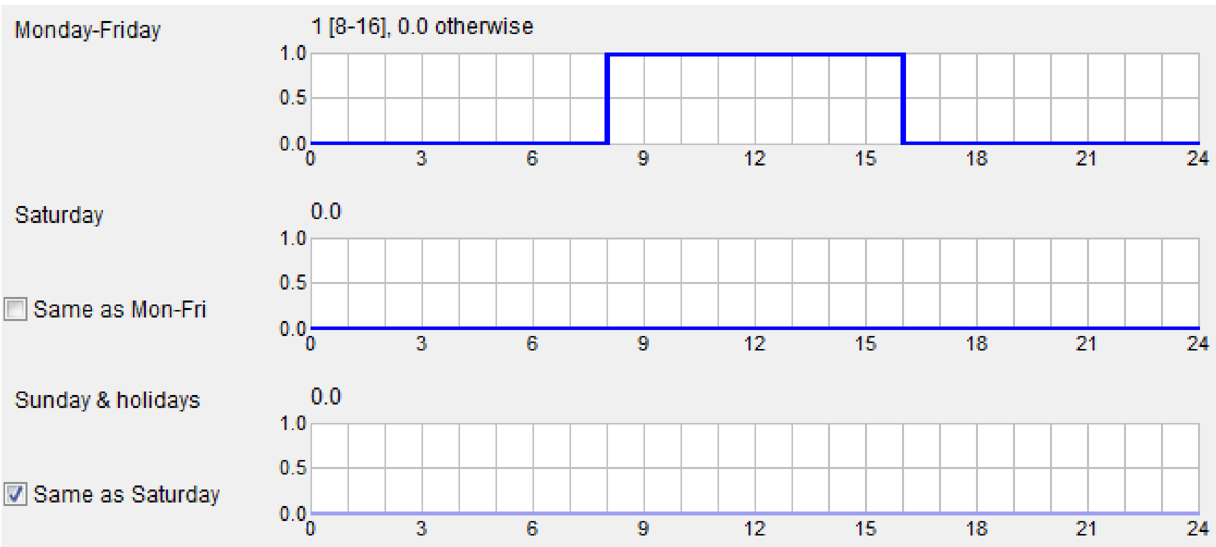


*Figure 7-5* Model's schedule for occupancy in the pool hall. The schedule is in percentage occupancy. A value of 1 means the occupancy is 100% of the reference occupancy (see 2.2.4.1).

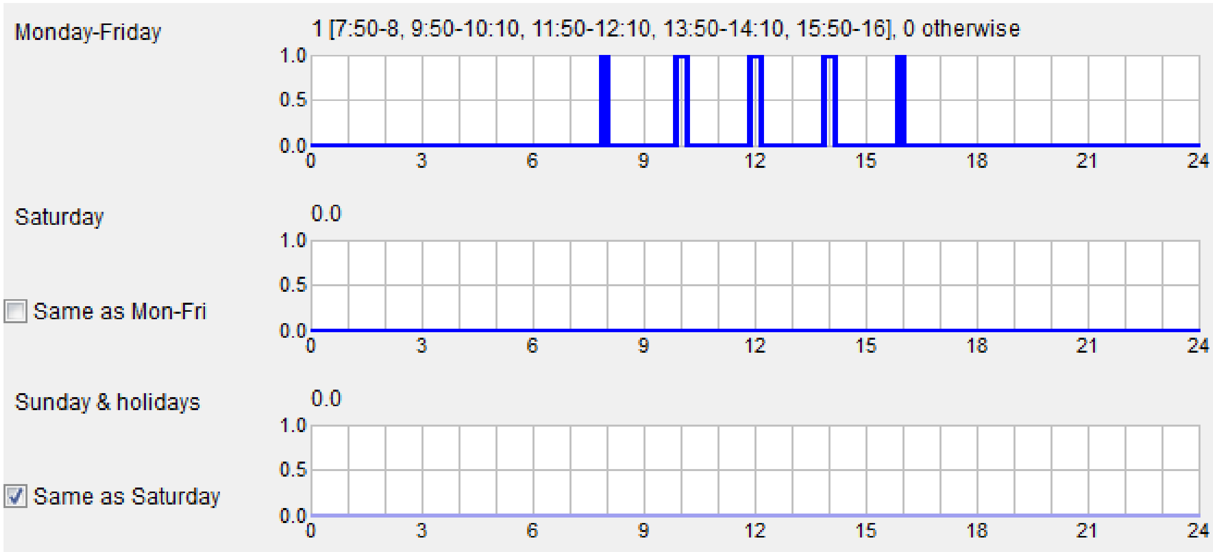


*Figure 7-6* Occupancy schedule for showers and entrance. The two peaks on the sides are fifteen minutes long and the three peaks in between are 30 minutes long (see explanations 2.2.4.1).

## 7.2 Airflow strategy at Jøa swimming facility

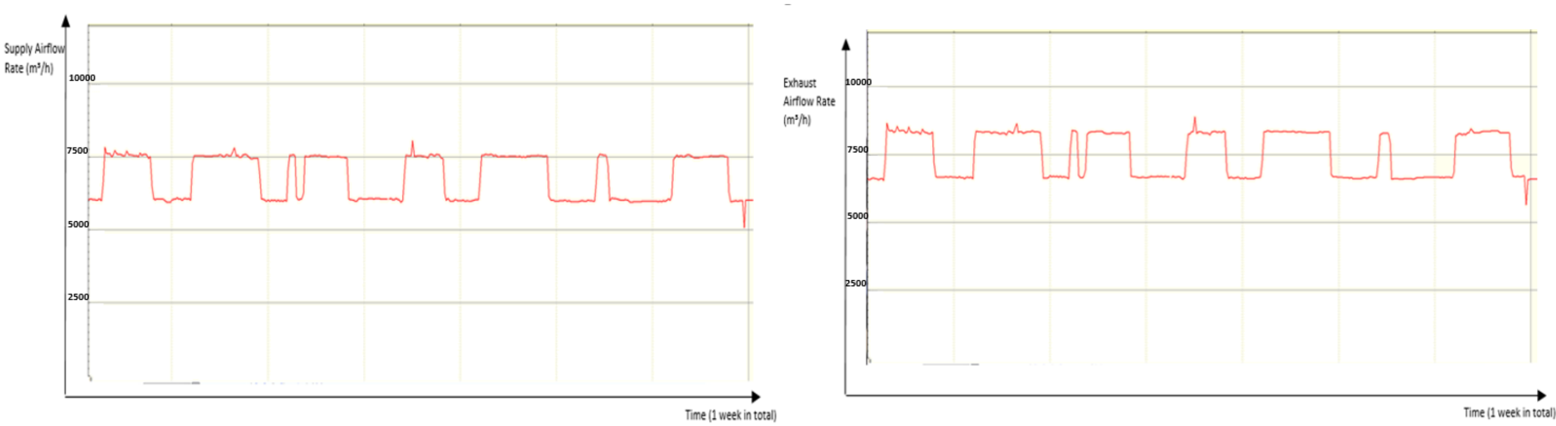


*Figure 7-7* Supply and exhaust Airflow rate for one week in Jøa swimming facility. Screenshot from the operation system.

## 7.3 Ventilation rates for the detailed model

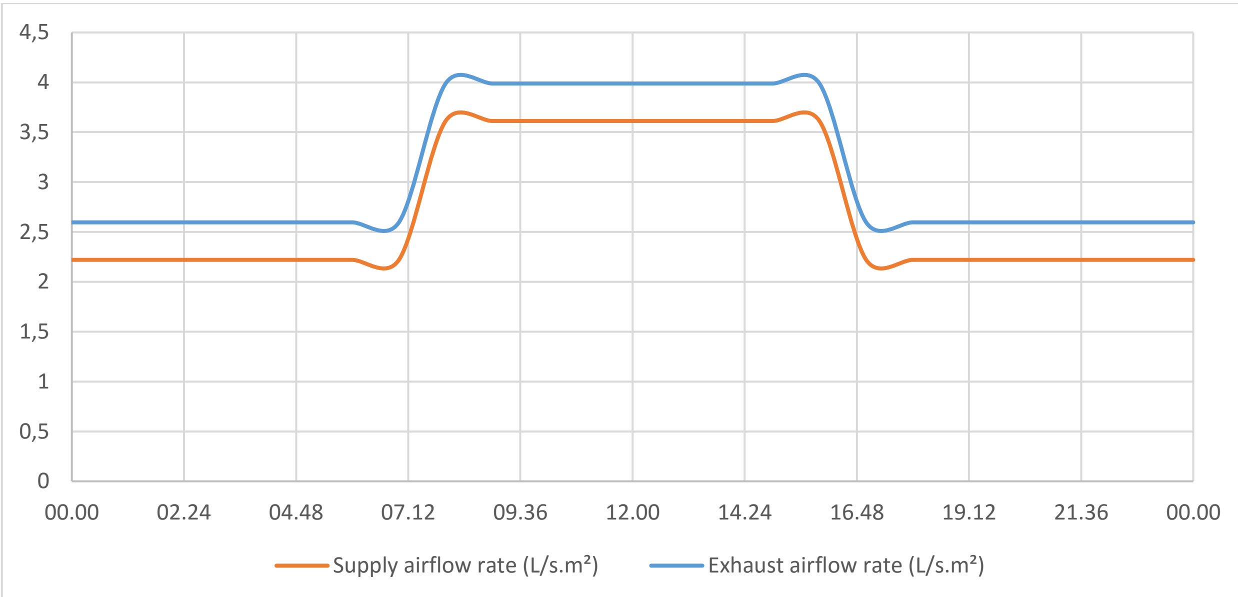


*Figure 7-8* Supply and return airflow rate in the swimming hall during a weekday when the swimming pool is open from 8 a.m. to 4 p.pm.

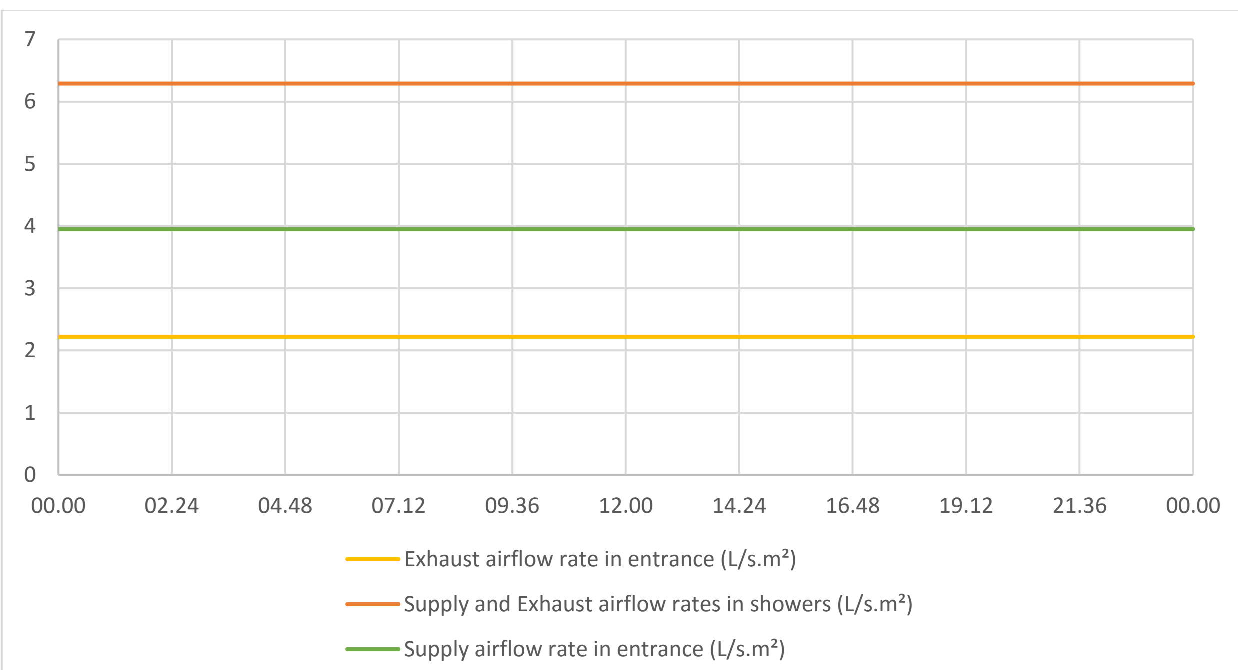


*Figure 7-9* Ventilation airflow rates in the entrance and in showers.

## 7.4 Investigation on the PI-controller behavior

*Table 7-1* PI characteristics depending on the measured signal (input). The set-point is constant, 21°C.

| PI input (°C) | PI output | PI input (°C) | PI output |
|---|---|---|---|
| 18 | 1 | 20.6 | 1 |
| 19 | 1 | 20.8 | 1 |
| 20 | 1 | 20.95 | 1 |
| 20.2 | 1 | 21 | 0 |
| 20.4 | 1 | 22 | 0 |
| 20.5 | 1 | 23 | 0 |

More precisely, illustrates how the PI behaves and shows that the dead-band is $10^{-4}$°C precise.

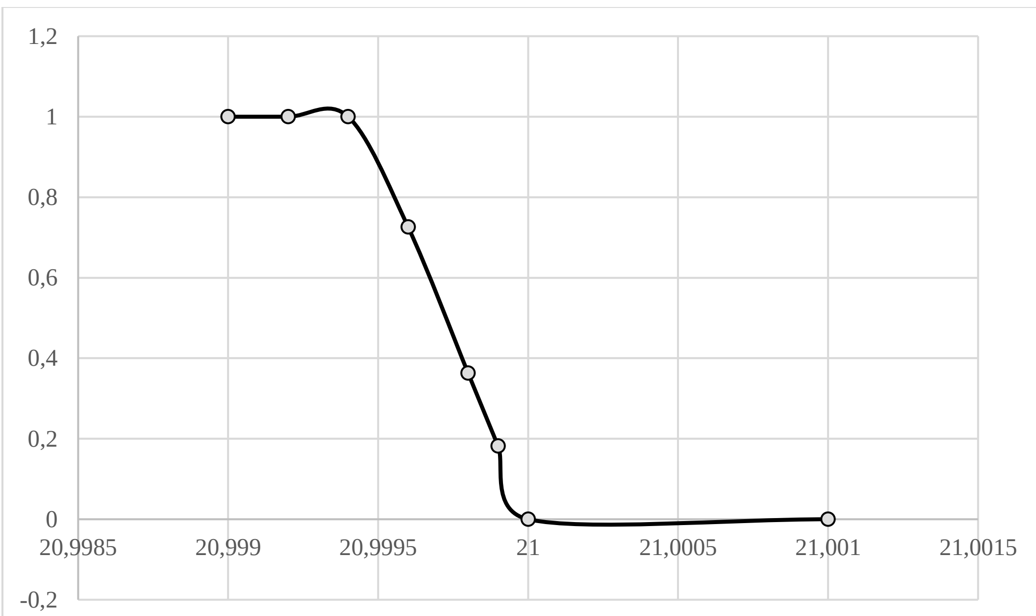


*Figure 7-10* PI Output in the y-axis against the input signal (°C) in the x-axis. The PI set-point is 26°C.

## 7.5 Inside door opening schedule

Doors are supposed to stay opened during turnovers. So every two hours, doors open for thirty minutes.

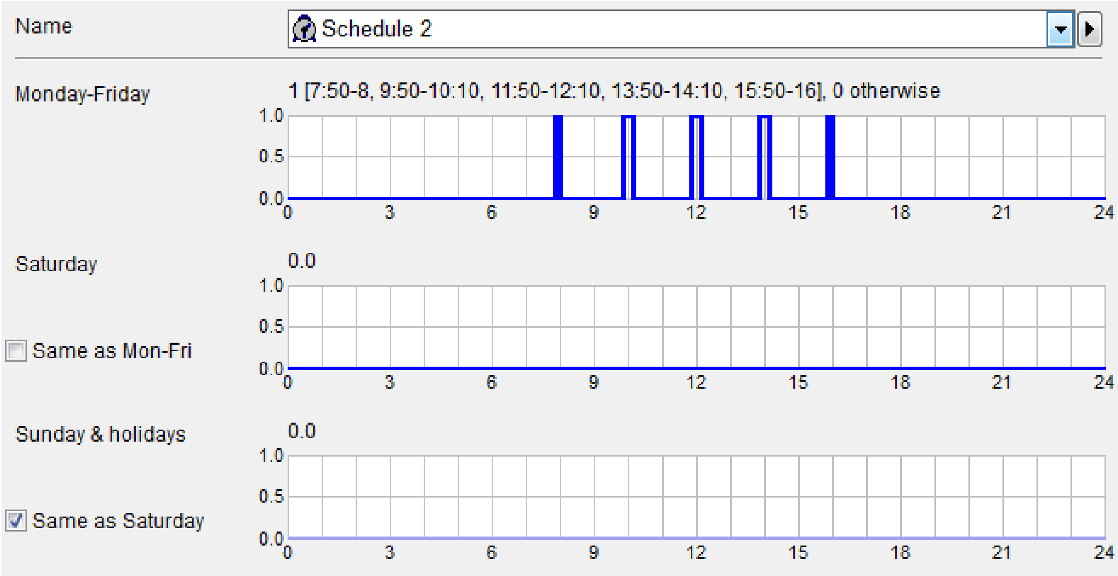


*Figure 7-11* Inside door opening schedule.

## 7.6 Additional results for the detailed model.

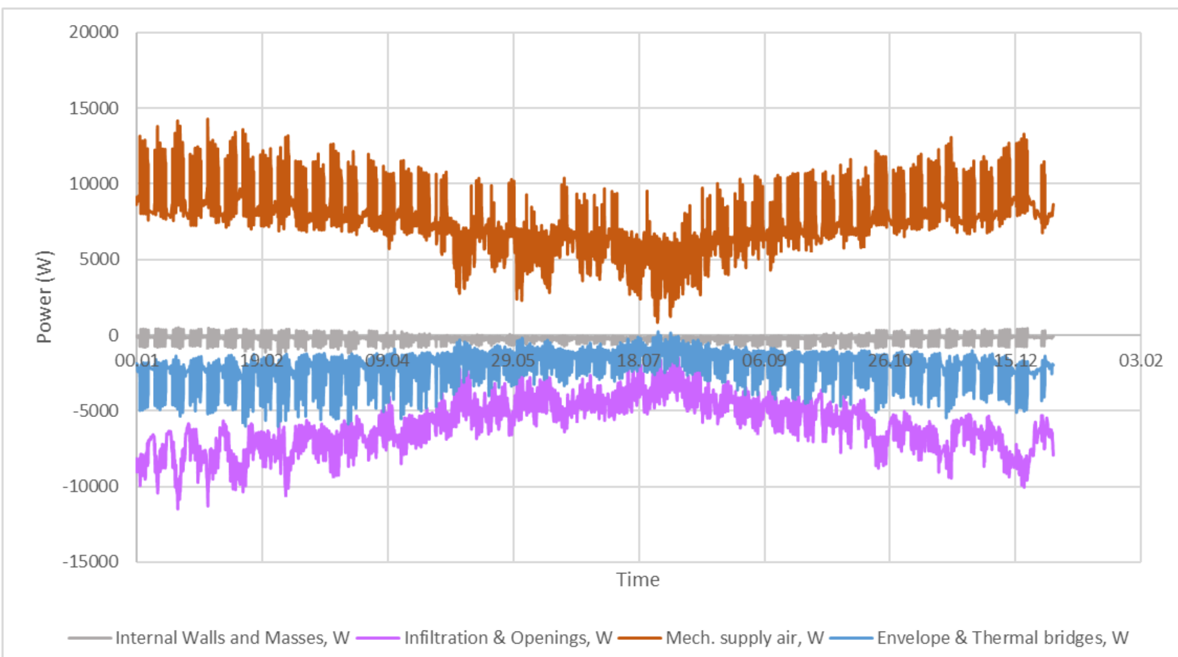


*Figure 7-12* Yearly power curves for the four main processes interacting within the pool hall excluding any processes related to the pool.

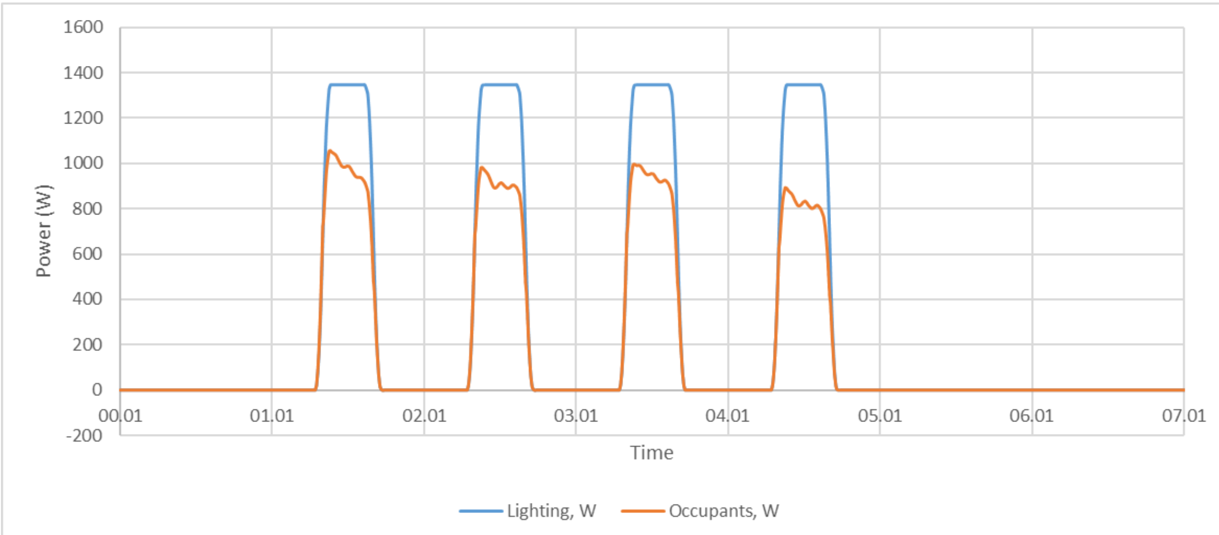


*Figure 7-13* Power influence of both occupancy and lighting during the first week of the simulated year.

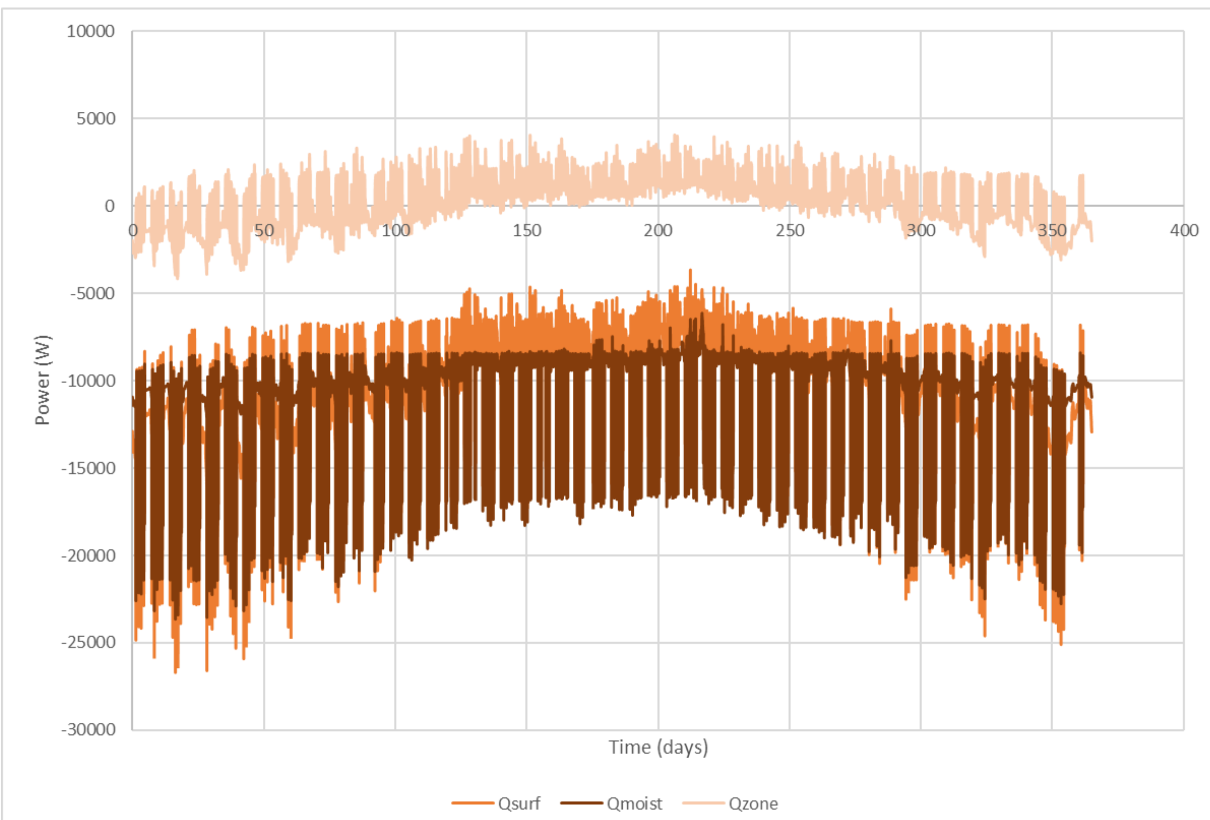


*Figure 7-14* Heat transfer between the pool and the zone for the whole simulated year.

## *7.7* Slopes' Values for the sensitivity analysis with *System 2*

*Table 7-2* Slope's values for the sensitivity analysis with *System 2*.

| | Slope Losses | Slope Gains | Slope $Q_{surf}$ | Slope Water-based heating |
|---|---|---|---|---|
| Orientation | 0.121 | 0.0638 | 0.0547 | 0.0439 |
| Insulation | 0.111 | 0.0517 | 0.043 | 0.0428 |
| Thermal Bridge | 0.0509 | 0.0293 | 0.023 | 0.0191 |
| Infiltration | 0.5 | 0.33 | 0.142 | 0.136 |
| Window | 0 | 0 | 0.0613 | 0.0448 |
| Depth | 0.000587 | -0.00109 | 0.00375 | 0.0284 |
| Pool Area | -0.0117 | -0.0218 | 1.03 | 0.871 |
| Activity Factor | -0.00538 | -0.00201 | 0.508 | 0.341 |
| Pool Temperature | 0.435 | -1.442 | 4.81 | 3.694 |
| Hall Occupancy | 0.000885 | 0.00626 | -0.0209 | -0.0146 |