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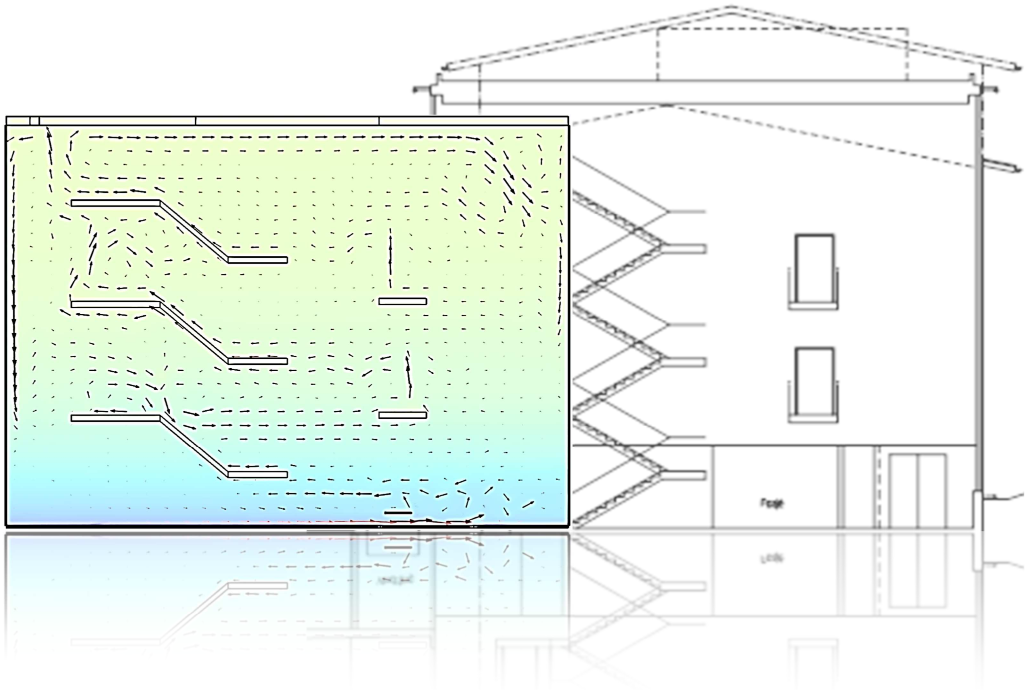

Radiant floor cooling systems
A measurement and simulation study

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

JOSEF JOHNSSON
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Department of Civil and Environmental Engineering
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CHALMERS UNIVERSITY OF TECHNOLOGY
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ABSTRACT

Radiant floor cooling systems have been around for quite some time now, but are not extensively used or much talked about in the industry. The background for this thesis lies in a question formulated by a HVAC – consultancy firm, wanting to know if the radiant floor cooling system they had installed worked in an adequate way regarding given cooling effects. Other issues raised were the precision of current calculation methods as well as important design factors to retrieve a high cooling effect. The question was broadened to consider the effect of the system on the thermal climate, where mainly temperature profiles and surface temperatures were studied.

The thesis is divided into one simulation- and one measurement study. The measurement study is, of course, study specific, and an atrium in the Centre of Gothenburg is studied. The measurements are mainly conducted to study the thermal climate in the atrium, as well as retrieve in data to the cooling effect calculations. Through simulations the in data was altered yielding different scenarios (i.e. with and without floor cooling), investigating the impact on the thermal climate.

To retrieve as high cooling effect as possible, the thickness and thermal conductivity of the screed are important factors. Other important factors, more connected to the balancing of the system, are water temperatures and flows. When the cooling effect was calculated, a finding was that the current method to approximate the cooling effect was not accurate enough. As for the results from the measurement study; they show a clear effect from the radiant floor cooling system on the temperature levels on the first floors. Higher up in the atrium the effect is not as evident.

Some study specific and general recommendations were found in this thesis; among them were the importance of thoroughly thinking through the activities and room conditions. A discussion concerning the viability of the system concludes the thesis, and the answer to the question is; it depends, partly on the activities in the room.

Key words: Radiant floor cooling, COMSOL Multiphysics, Radiant floor heating

Golvkylsystem

Examensarbete inom Building Performance Design

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SAMMANFATTNING

Tekniken att använda kalla golv för att kyla har funnits relativt länge, men används ännu inte i stor utsträckning och är inte heller speciellt omtalat i VVS – branschen. Bakgrunden till denna uppsats var en fråga formulerad av ett VVS – konsultbolag, vilka ville ha undersökt om denna typ av kylsystem fungerade på ett tillfredsställande sätt och gav givna kyleffekter. Ytterligare frågor som belyses i rapporten tillförlitligheten av de nuvarande beräkningsmetoderna, samt viktiga faktorer som påverkar kyleffekten av systemet. Frågan utvidgades för att även ta hänsyn till vilken påverkan golvkylan har på det termiska klimatet, där främst temperatur profilen och yttemperaturen undersöktes.

Rapporten är uppdelad i en simulering- och en mätningsstudie. Mätningarna är studiespecifika, där en ljusgård i centrala Göteborg studerades. Mätningarna utfördes främst för att undersöka det termiska klimatet, men även för att hämta indata till kyleffektberäkningarna. I simuleringarna modifierades erhållen indata för att efterlikna olika scenario (t.ex. med och utan golvkyla) och därmed se påverkan på det termiska klimatet och luftvolymen.

För att maximera systemets kyleffekt bör tjockleken och den termiska ledningsförmågan på avjämningsmassa samt golvmaterial beaktas. Andra faktorer som påverkar kyleffekten och är kopplade till injustering av systemet är t.ex. tilloppstemperaturer och flöden. Den nuvarande beräkningsmetoden för att approximera kyleffekten visade sig vara mindre exakt medan den akademiska metoden som används i denna studie vara mer noggranna. Resultaten från mätningarna visade att golvkylan hade en tydlig effekt på temperaturen på den första våningen, däremot var effekten inte lika uppenbar högre upp i ljusgården.

Vissa objektsspecifika samt allmänna rekommendationer finns i rapporten, där generella rekommendationer är t.ex. att beakta rumsaktiviteter och val av golvmaterial. Diskussionen tar upp frågan om det verkligen finns en vinst i att installera golvkyla. Svaret är att det beror på, delvis på vilken verksamhet som bedrivs i rummet.

Nyckelord: Golvkyla, COMSOL Multiphysics, Golvvärme

Contents

Preface

In this master thesis a radiant floor cooling system is studied through a measurement and simulation study. The project was partly carried out at the Division of Building Technology and Building Physics, Chalmers University of Technology, Sweden and partly at a HVAC consultancy firm. Furthermore field measurements were conducted at a building in the Centre of Gothenburg. The thesis is divided into a measurement and simulation study on a floor cooling system type A.

The measurement study could not have been done without the help of the HVAC consultancy firm. Special thanks to Jonas Sköld and Jasko Bećirović; thank you for all your patience and help! Furthermore, we send out a thank you to Sandra Olsson, always willing to help. The property technician at the study object, Ulf Löfdahl, also deserves gratitude. To our supervisor Angela Sacic Kalagasidis; Thank you for continuously steering us in the right direction, as well as always encouraging us.

We would like to conclude with saying that even though at times this thesis has taken the life out of us, it has been a great learning experience. We are glad that we have emphasized a subject that will be more common in the future.

Göteborg, 2012

Josef Johnsson

Linnéa Westerlund

Terms and definitions

The terms and definitions are partly taken from standard EN-15377-1:2008 where the following terms and definitions apply:

Embedded surface heating and cooling system

A system consists of circuits of pipes embedded in floor construction, distributors and control equipment

Circuit

Section of system connected to a distributor which can be independently switched and controlled

Active area

Area of the surface covered by a heating/cooling system

Peripheral area

Surface area which may be heated or cooled to a higher or lower temperature. It is generally an area of 1 m maximum in width along exterior walls. It is not an occupied area

Occupied area

Area within the heated or cooled surface occupied for long periods. Normally the zone between the floor and a height of 1.8 m and 1.0 m from external walls/windows and 0.5 m from internal walls

Surface heating and cooling components

Components are:

- insulating layer (for thermal and/or impact noise insulation)

- the protection layer (to protect the insulating layer)

- the pipes or plane sections

- the load and thermal distribution layer, where pipes are embedded

- covering

Heat flow density (q)

Heat flow between the space and surface divided by the heated/cooled surface area

Thermal conductance (γ_{up})

Upward thermal conductance of the floor construction. Taking pipe distance etc. into account calculated by HEAT 2

Total heat exchange coefficient (α_{tot})

The total heat exchange coefficient for a surface, combining radiation and convection

Supply temperature

Temperature of the water that enters the cooling pipes embedded in the floor structure

Symbols and units

Table Symbols

Symbol	Unit	Quantity
A	m^2	Surface area
$A_{diffuse}$	m^2	Area exposed to diffuse radiation
A_{direct}	m^2	Area exposed to direct radiation
$I_{sol}^{diffuse}$	W/m^2	Solar intensity diffuse radiation
I_{sol}^{direct}	W/m^2	Solar intensity direct radiation
K_c	W/K	Convective thermal conductance
K_{cd}	W/K	Conductive thermal conductance
Q	W	Heat flow rate
q	W/m^2	Heat flow rate density
R_w	m^3/s	Water flow rate
T_a	K	Air temperature
T_{in}	K	Indoor temperature
T_{op}	K	Operative temperature
T_{out}	K	Outside air temperature
$T_{r,mean}$	K	Mean radiant temperature
T_s	K	Outside surface temperature
T_{wR}	°C	Return water temperature
T_{wS}	°C	Supply water temperature
α_c	W/m^2K	Convective surface heat transfer coefficient
α_{cd}	W/m^2K	Conductive heat transfer coefficient
α_r	W/m^2K	Radiative heat transfer coefficient
α_{sol}	-	Absorptivity, solar radiation
ε	-	Emissivity of a surface
$\tau_{diffuse}$	-	Transmittance, diffuse radiation
τ_{direct}	-	Transmittance, direct radiation
λ	W/mK	Thermal conductivity
ΔT	K	Temperature difference
γ_{up}	W/mK	Thermal conductance upward

1 Introduction

This story begins with a consultancy firm hungry for knowledge and two students eager to learn, ending up with this master thesis, focusing on radiant floor cooling systems. The cooling technique has actually been around for quite some time, but in-depth knowledge concerning the system is lacking. In fact, in general, floor cooling systems are not solely designed as floor cooling systems, but are instead incorporated in radiant floor heating systems. Thus focus seldom lies entirely on the cooling system when designing. The consultancy firm established one of the main aims of this master thesis which was:

- Does the radiant floor cooling system work in an adequate way and what are the delivered cooling effects of the system?

The aim was, eventually, developed into further questions, one concerning the existing ways of calculating the delivered cooling effect. The company's current method of approximating delivered cooling effect is very simplified, only following a rule of thumb. Furthermore there are various other existing ways of calculating the delivered cooling effect. This leads us to the second issue:

- Is their current method accurate enough? Is it comparable to other existing methods and reality?

As mentioned, in-depth knowledge concerning design issues is lacking, this brings us to the following important issue that needs to be addressed:

- Which factors will influence the delivered cooling effect? Are there any possibilities to improve an existing system?

The above mentioned aims concern the systems delivered cooling effect. Another, equally interesting factor, since it essentially affects the wellbeing of humans, is how we perceive the indoor climate. The floor cooling system will have an influence on the thermal climate, exactly in what way is difficult to say beforehand, but will be studied in the thesis. The final aim of the thesis is thus:

- How is the thermal climate affected by the floor cooling system?

To relate the study to reality, a reference object was selected. All the measurements and simulations in this thesis are related to this object. A suitable object is, as it turns out, great in volume and height, has great glass window area and is most appropriate an entrance hall.

1.1 Aim

The aim of this thesis can be divided into two main parts. The first part deals with technical questions, primary considering effect issues like:

- What are the delivered cooling effects?
- Are the methods the consultancy firms use today accurate enough?
- Which are the main parameters that will influence the delivered cooling effect?
- Are there any possible improvements in an existing system?

The second category addresses more soft parameters, with the primary purpose to investigate how a floor cooling system will affect the indoor climate. Factors accounted for are:

- Floor surface temperatures
- Temperature profiles
- Humidity

Another essential question should be answered by the end of this thesis:

- Is installing a floor cooling system viable?

1.2 Limitations

Radiant floor systems can be divided into air-based, water-based and electric systems. Focus in this thesis will lies on a water-based system. Furthermore, this paper focuses mainly on a specific case study in a commercial building with a glazed entrance hall; hence the conclusions might not be applicable to other cases.

1.3 Methodology

Issues concerning the cooling effect are studied through measurements on a suitable object and a literature review. The literature includes various standards and scientific articles, as well as guidebooks. HEAT 2 was utilized to evaluate and adjust the heat transfer coefficient, which is required in the cooling effect calculations. Furthermore, the results from the measurements are utilized in cooling effect calculations to evaluate the effect issues further. The cooling effect is calculated by means of three different methods.

Thermal climate limits are investigated through the literature review. Through the measurement study some thermal climate factors are evaluated, subsequently simulations in COMSOL Multiphysics are performed to evaluate the impact of the floor cooling system on the behaviour of the air volume (CFD-analysis) in different cases.

A number of existing objects, with a floor cooling system installed, were provided by the consultancy firm involved in this project. To be able to decide on a suitable object, an inventory was made.

2 About radiant floor systems

The story of the radiant floor heating system goes way back in time. However, the radiant floor cooling system, although have been around for quite some time, is not nearly as well- known and used as the floor heating system. The following chapters will give a short history of the radiant floor system, as well as radiant floor cooling benefits.

2.1 History of the radiant floor system

The radiant floor system originates from China and Korea, who invented the radiant floor heating system already thousands of years ago. Their radiant heating system consisted of a fireplace and smoke ducts, and hot gas from the fire was led through the ducts. By means of conduction, beds or in some cases even entire floor, were heated up (see Figure 1). The heat transfer continued through the room by means of convection and radiation. [1]

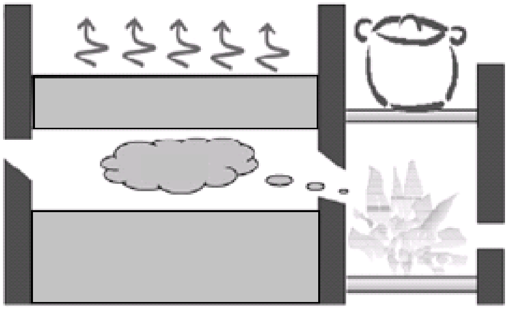


Figure 1 Chinese radiant bed heating system "Kang" [2]

Although the Asians were well before, the ancient Romans often get recognition for being the first to invent the floor heating systems, but they were as late as 500 B.C. and their system was very similar to the Asian system, also consisting of a fire place and smoke ducts (see Figure 2 for illustration). [1]

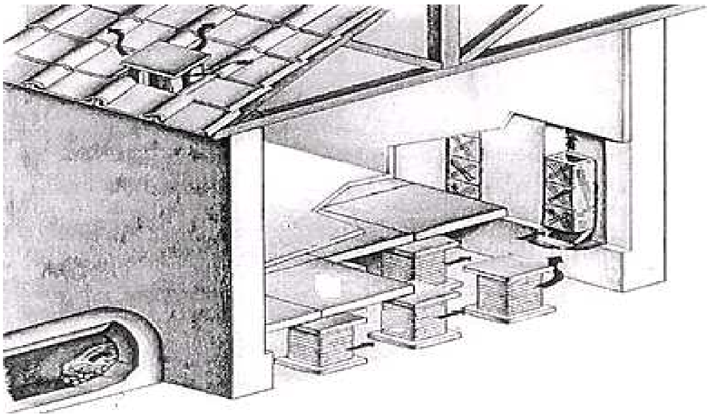


Figure 2 Roman radiant floor heating system "Hypocaust" [2]

In more recent history (19th century) the hydronic floor heating system we use today developed in Europe, with a water-based system and a boiler. It was not until 1930, that a man named Faber, started to use the water based system not only for heating, but also for cooling purposes. At that time they had issues with condensation, which was due to that the floor surface temperature was lower than the dew point temperature. Condensation issues could not be managed properly in the beginning, but was successively handled and in the later part of the 20th century the knowledge of controlling this kind of system deepened. [3]

Currently, low temperature radiant heating systems, which are fluid-based, are widely used [3]. High temperature cooling systems are used as well, but not to the same extent, and most often not solely as a cooling system, but is instead incorporated in a floor heating system.

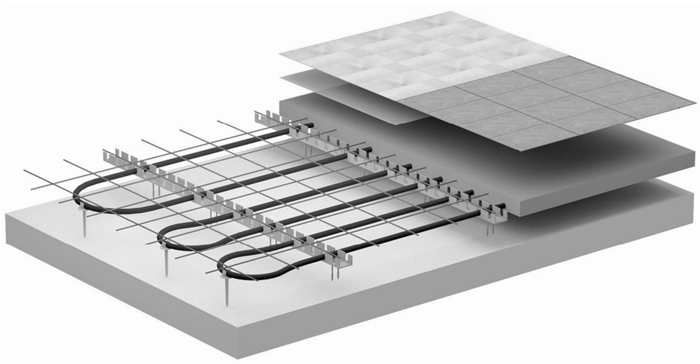


Figure 3 Radiant floor systems as we know them today [4]

2.2 Benefits of a radiant floor cooling system

Why a radiant floor cooling system? Cold feet and a low convective heat transfer must decrease the viability of the system? These statements are true, and the radiant floor cooling system is not the obvious choice in every single situation. With this said, there are some clear benefits with a radiant floor cooling system, as well. In fact, in spaces with large height, great volume and large window area, e.g. many modern entrance halls, this can be the best choice. A number of radiant floor cooling benefits are further described in this chapter, starting with the positive effect the radiant floor system has on the operative temperature.

As is known, the air temperature, which is the temperature of the surrounding air, is not on its own an adequate way of describing the indoor thermal climate. Instead the operative temperature, which is combination of the air temperature and the surrounding mean radiant temperature, is a better way to describe the thermal climate (see equation 1).

$$T_{op} = \frac{\alpha_r * T_{r,mean} + \alpha_c * T_a}{\alpha_r + \alpha_c} \qquad (1)$$

where:

T_{op}	- Operative temperature	(K)
α_r	- Radiant surface heat transfer coefficient	(W/m^2K)
$T_{r,mean}$	- Mean radiant temperature	(K)
α_c	- Convective surface heat transfer coefficient	(W/m^2K)
T_a	- Air temperature	(K)

The radiant cooling system can in theory achieve the same operative temperature at a higher air temperature than a convective all air system. This is due to that the overall mean radiant surface temperature will be lower. [5]

Some other benefits, with a radiant floor cooling system, worth mentioning are:

- In a floor cooling system the installations are concealed, this is often beneficial, and sometimes even required by architects and property owners
- As water has a higher heat capacity than air, less water than air is needed for the same heat capacity, hence installations take less space
- Why not a radiant cooling system in the ceiling, as it gives a higher convective heat transfer? One answer to this is that the floor has a higher view angle than the roof, especially in spaces with great height
- The efficiency of a heat pump may increase with the possibility to use it for both heating and cooling purposes, this is valid when utilizing a borehole
- The piping system can be used for both heating and cooling purposes, the only extra installation needed is an extra heat exchanger connected to the piping system and the buildings main cooling system

3 Types of high temperature cooling systems

There are essentially two main design types of hydronic floor heating- and cooling systems, these are thermo active building systems and ordinary floor systems. The second group includes the systems in Table 1; these are all insulated from the main building structure. The thermo active systems are not insulated from the main building structure; instead the system makes use of the thermal mass (shown in Table 2). The systems are introduced briefly in this chapter, for more detailed information see standard SS-EN 15377.

3.1 Ordinary systems

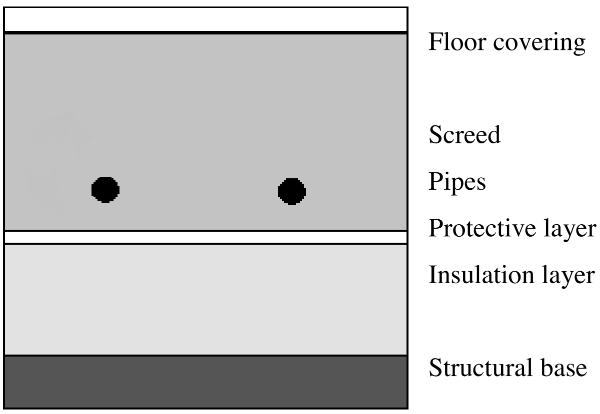
Type A is the most common type of radiant floor system in Europe. The pipes are, as seen in Table 1, embedded in the screed or concrete. Above this layer the floor covering is placed, and below a protective layer is followed by thermal insulation and a structural base. [6]

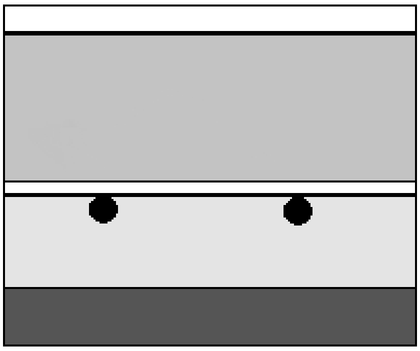
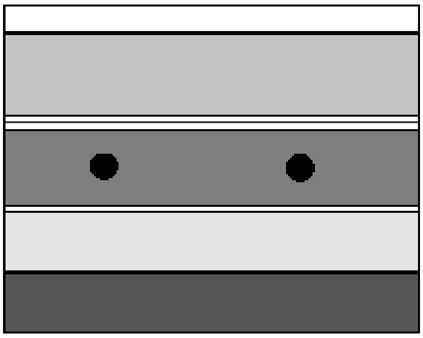
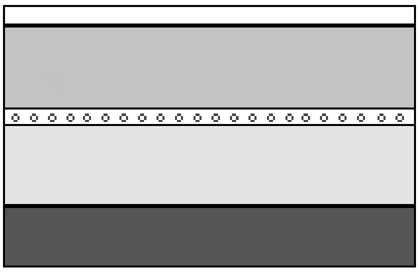
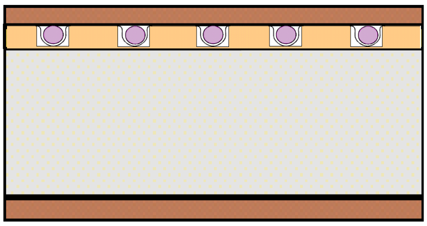
In a type B system the pipes are not embedded in the screed but are, for instance, embedded in the insulating layer. This type of system will have to rely on heat conductive plates to improve the distribution of the heat flow. The plates need to be linked properly with the pipes. [6]

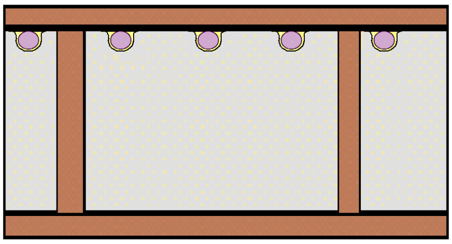
Type C is a variation of type A, but with an extra levelling layer where the pipes are embedded and placed below the screed, see Table 1 for design. [6]

The last two systems that are insulated from the main structure are plane section systems and systems where pipes are embedded in a wooden construction. These will not be described thoroughly here, since they are out of the scope of this thesis. The design can be seen in Table 1.

Table 1 Ordinary floor cooling designs

Type	
A	 *Pipes embedded in screed or concrete*

B	 *Pipes embedded outside screed*
C	 *Pipes embedded in screed/levelling layer*
D	 *Plate section system*
G	 *Tube in subfloor*

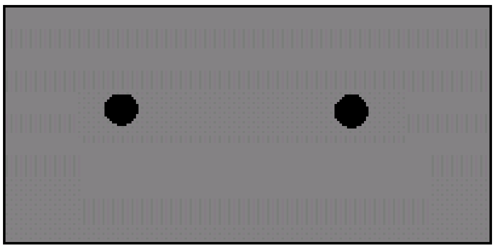
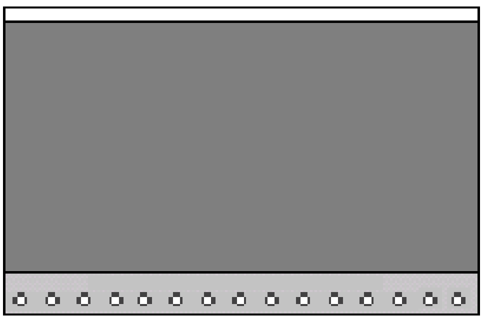
	 *Tube under subfloor*

3.2 Thermo active slabs

The thermo active slabs (TABS), where the pipes are in thermal contact with the load bearing structure, will response slow to internal gains and consequently, will not be able to counteract against fast changes in the indoor thermal climate. On the upside the system will be more energy efficient.

A distinction can be made between two types of TABS. In a type E system the pipes are placed in the concrete slab, and in a type F system capillary pipes are placed at the concrete surface. In Table 2 TABS designs can be seen. [6]

Table 2 TABS designs

Type	
E	 *Pipes embedded in a massive concrete layer*
F	 *Capillary pipes embedded in a layer at the inner surface*

The radiant floor system studied in this master thesis is system type A, which will be described further.

4 Heat transfer principles

Generally, there are three difference heat transfer mechanisms, which are heat conduction, convection and radiation; all of them involved in the heat transfer in the study object, see Figure 4. Equations are presented in this chapter, for description of the symbols; see Symbols and units.

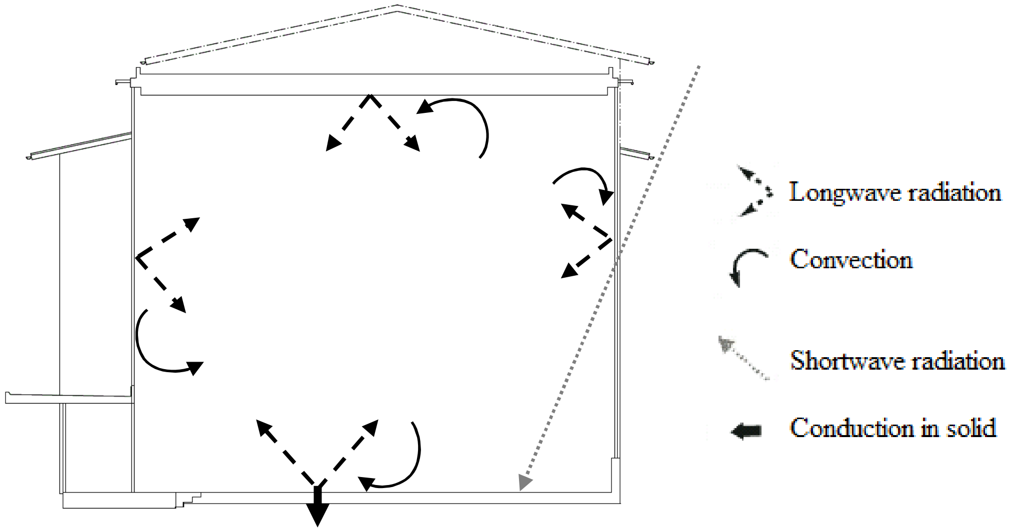


Figure 4 The main heat transfer mechanisms inside the atrium

The main heat transfer mechanism in homogenous material, i.e. the floor construction, is conduction (see equation 2). Factors that will influence the heat exchange in the radiant cooling system are for example: purely engineering aspects, such as spacing and thickness of the pipes, as well as fluid velocity and construction design. Another influencing factor is material properties. [7] The conductive heat transfer coefficient is dependent on the thermal conductivity, the surface area and the thickness of the material (see equation 3).

$$Q = \alpha_{cd} * A * \Delta T \qquad \text{(W)} \qquad (2)$$

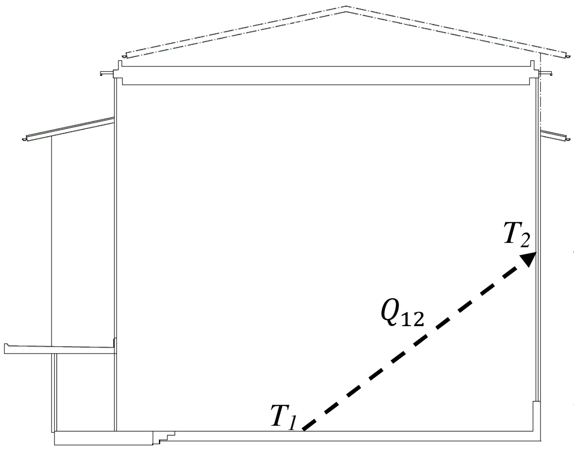
$$\alpha_{cd} = \frac{\lambda * A}{d} \qquad \text{(W/m}^2\text{K)} \qquad (3)$$

At the floor surface, radiation and convection are the main heat transfer mechanisms. The heat is exchanged with the surroundings by different means in the two cases; convection transfers the energy from the surface to the air directly, while radiation transfers energy by electromagnetic waves (long wave radiation) to other surfaces, which in turn heats the air. Solar radiation is another type of radiation which is a short wave radiation.

In equation 4 the basic convective heat transfer relationship is shown. The convective driving force is either the density difference, dependent on temperature difference (natural convection), or external effects like ventilation (forced convection). [8]

$$Q = \alpha_c * A * \Delta T \qquad \text{(W)} \qquad (4)$$

Long wave radiation is the main heat transfer mechanism between surfaces. Important factors influencing the radiant heat exchange in the atrium are e.g.; emissivity (ε) of the radiating surface, dependent on colour and material properties, and view factor (F_{12}), which depend on the geometry of the room and surrounding surfaces.



$$Q_{12} = \alpha_r * A_1 * \Delta T \qquad \text{(W)} \quad (5)$$

where:

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

$$\bar{T}_{1,2} = \frac{T_1+T_2}{2} \qquad \text{(K)}$$

Figure 5 Radiant heat transfer from one surface to the another in the room

The heat transfer rate can be described by equation 5. The equation is valid for the condition when T_1 is close to T_2, which is generally the case in building physic applications. The magnitude of the view factor (F_{12}) depends on the distance between the person and the radiating surface, as well as the overall surface area. The floor usually has the highest view factor [5], which is why a radiant floor often is preferable (especially in spaces with great volume). The equation of the view factor can be seen in Appendix A.

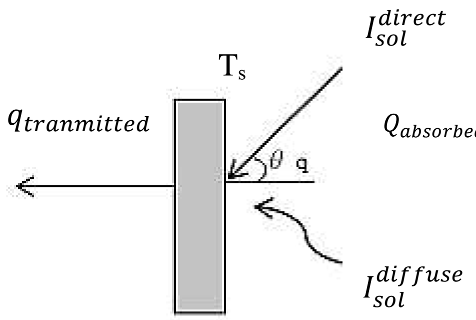
The incoming radiation from another surface or the sun can be absorbed, reflected or transmitted. Table 3 shows some emissivity values of different surfaces, which are utilized in the simulations. Larger emissivity of the surfaces partaking in the radiation, results in a greater heat transfer, which also can be seen in equation 5.

Table 3 Emissivity of different material

Material	**Emissivity**
Marble	0.55
Paint, white	0.85
Glass	0.92

The magnitude of the incoming solar radiation will be affected by a number of different factors, including altitude of the sun and azimuth (see Appendix A). Furthermore, the magnitude of the radiation is dependent on the weather and direction of the building (see chapter 6 for direction of the study object). The incoming radiation can be divided into direct and diffuse sun radiation, where direct sun radiation is unidirectional and thus dependent on all the factors mentioned above, while diffuse radiation is omnidirectional, and therefore not dependent on any angles or directions. On a clear day the maximum total sun intensity (I_{sol}) is around 1kW/m^2, the diffuse corresponding to around one third of the direct sun intensity. The simulations and selected parts of the measurements are studied during such a day.

In this thesis the focus will lie on absorbed and transmitted energy, as these will influence the thermal climate in the study object. The heat absorbed, due to solar radiation, is shown in equation 6. Another interesting factor is the increased outside surface temperature of the glass wall, due to short wave radiation (see equation 7). Increased surface temperature may, in fact, affect the thermal climate quite considerably.



$$Q_{absorbed} = \alpha_{sol} * I_{sol}^{direct} * \cos(\theta) * A_{direct} + \alpha_{sol} * I_{sol}^{diffuse} * A_{diffuse} \quad \text{(W)} \quad (6)$$

$$T_s = \frac{K_c * T_{out} + Q_{absorbed} + T_{in} * K_{cd}}{K_c + K_{cd}} \quad \text{(K)} \quad (7)$$

$$Q_{tranmitted} = \tau_{direct} * I_{sol}^{direct} * cos(\theta) * A_{direct} + \tau_{diffuse} * I_{sol}^{diffuse} * A_{diffuse} \quad \text{(W)} \quad (8)$$

Figure 6 Absorbed and transmitted radiation, as well as surface temperature

The energy that is not absorbed, nor reflected, will be transmitted into the building. The total transmitted sun radiation is show in equation 8. The transmitted part will vary depending on the properties of the building envelope, and will be high at the glass facades, where the transmittance factor (τ) will be relatively high, while at the remaining facades the transmittance is zero.

Transmitted solar radiation can be rather problematic in modern building with large total window area, where little thought has been taken to sun screening. The impact on the indoor thermal climate is quite large, with increased surface and room temperatures; this is illustrated further in the measurement and simulation chapters.

5 Cooling capacity of radiant floor cooling systems

There are some important factors that need to be correctly designed to give the highest possible cooling effect; these will be discussed in the following chapter. Furthermore, quantities of the total heat exchange will be accounted for, as well as methods to calculate the cooling capacity of a radiant floor system.

5.1 Important construction design details

The cooling capacity will be limited by a number of local thermal climate reasons, for example: lowest acceptable surface temperature, vertical air temperature difference, radiant asymmetry, as well as dew point issues. Furthermore, the floor construction, with its floor covering and slab thickness, and the design of the hydronic system will influence the cooling capacity. In the system design, factors like centre to centre distance, type of material and water velocity, will have an impact on the cooling capacity. Some of these parameters should be designed differently depending on if the systems main task is to cool or heat the space. For instance, when designing a floor heating system, a normal centre to centre distance is around 150 mm or more [5], but for improved cooling capacity the distance may need to be smaller, around 100-150 mm [9]. Moreover, the temperature difference between the supply and the return water temperature should be designed differently depending on the main purpose of the system. A common temperature difference between supply and return water in heating mode is 10°C, while for cooling purposes this is preferably lower at 3-5°C. The lower temperature difference will result in an increased water flow rate and pressure drop over the circuit, which needs to be corrected by shorter circuit or more powerful circulation pump [5].

In a recent study the main design parameters that influence the heat transfer were concluded to be the cover material and thickness, less important factors were the pipe material and diameter, as well as the pipe spacing which only had a marginal value [10]. Furthermore, Jin et al., conclude in their study that the velocity of the water plays a minor role on the performance even if the flow is laminar [11]. With this said, a lowering of velocity to save energy may be good to consider in certain cases.

If the floor cooling system is combined with a convective air system, the floor system takes care of sensible load and the air system will take care of the latent load. This will result in a higher cooling capacity since the dew point will be lowered by removing moisture from the air (latent cooling) [5].

5.2 Total cooling effect, quantities

The heat exchange from the surface to the atrium is, as mentioned before, carried out by means of radiation and convection. The total heat exchange coefficient can be calculated by a number of different methods. The method described in ASHRAE handbook HVAC systems and equipment, gives a maximum possible total heat exchange of 7 W/m^{2}K, for a floor cooling system. The radiant heat exchange coefficient accounts for 5.5 W/m^{2}K, giving a very low convective heat exchange coefficient [12]. As a comparison; for a floor heating system, the maximum possible total heat exchange coefficient is calculated to be larger at 11 W/m^{2}K, while for a lower temperature difference it is 9 W/m^{2}K (see Figure 7). The higher heat exchange coefficient is due to a larger convective heat transfer (greater buoyancy effect).

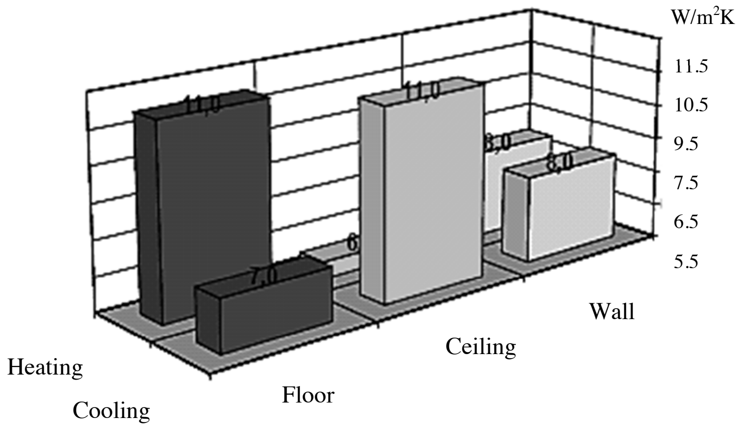


Figure 7 Heat exchange coefficient for radiant heating and cooling systems [13]

The absolute maximum delivered cooling effect is around 100 W/m^2, though this is only be valid if the floor is exposed to direct sunlight. In other cases the maximum cooling effect (which has been determined by experimental studies conducted by REHVA [7]) is presented in Figure 8.

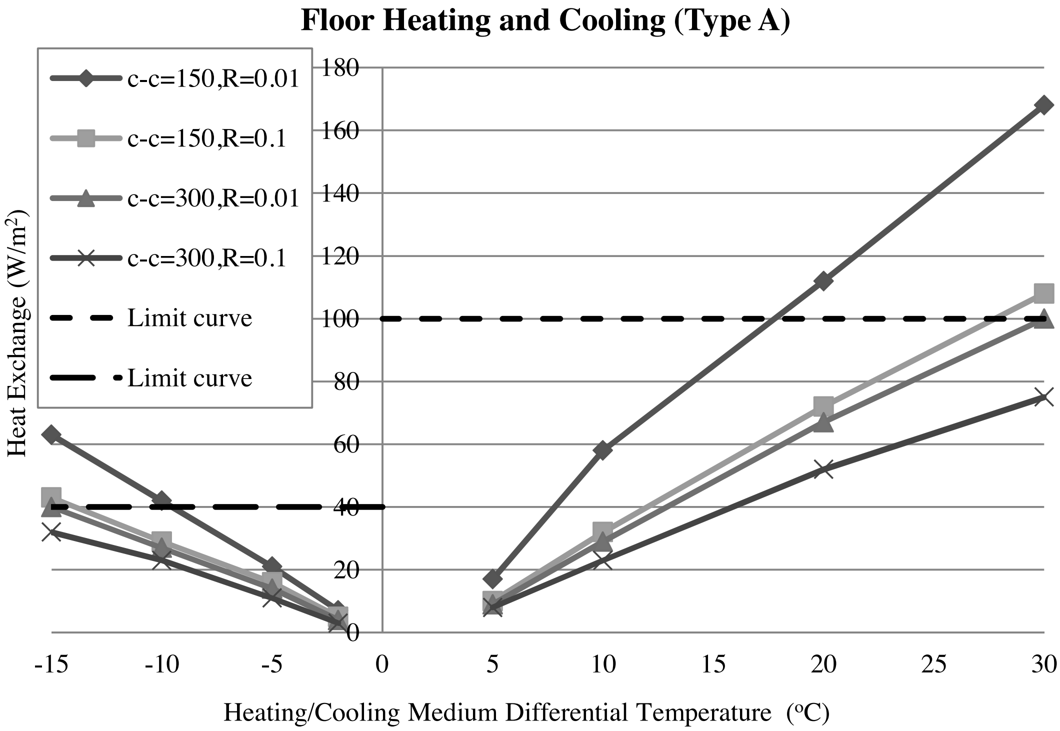


Figure 8 Heat exchange expressed as a function of the temperature difference, floor covering (R) and pipe spacing (c-c) (Limit curves shows the restrictions with regards to comfort criteria, i.e. maximum and minimum (here 20^{o}C) allowed floor surface temperatures. Room temperatures assumed: summer 26^{o}C, winter 20^{o}C) [7]

In Figure 8, the maximum delivered heating effect of a radiant floor heating system is shown as well. The maximum is 100 W/m^2, compared to a cooling effect of 40W/m^2 as a maximum, for a floor cooling system.

5.3 Methods determining the cooling capacity

The cooling capacity delivered by the floor cooling system is, in this thesis, determined by a number of methods and subsequently compared. Method waterside is used as a reference value, as this is the total heat flow from the pipes.

5.3.1 Method Waterside

This method is used to calculate a total heat flow from the circuit. To be able to make use of the calculations, information on flows and temperatures in the system is needed.

$$Q = \rho * c * R_w * (T_{wR} - T_{wS}) \tag{9}$$

where:

Q	- Heat flow rate	(W)
ρ	- Density	(kg/m^3)
R_w	- Water flow rate	(m^3/s)
T_{wR}	- Return temperature water	(°C)
T_{wS}	- Supply temperature water	(°C)

The calculated heat flow density changes with different water temperatures. As this is the maximum heat flow rate it sets an upper limit, hence the delivered cooling effect to the room must be lower than the total heat flow from the pipes.

5.3.2 Method Academic

The academic method is based on a lecture held by Henrik Karlsson from SP. It is used in this thesis for further calculations, as it includes the water flow rate and insulation properties. The method is based on input data from simulation tools and basic building physic equations [2], [8]. In the calculations, there are two variables which are rather challenging to determine, the first one being the real thermal conductivity of the floor construction. This is because the heat flows from or to the pipe is transferred from the whole perimeter of the pipe to the surface, which is seen in Figure 9 a). The second variable is the water temperature in the pipe circuit, which changes along the pipe, and in so doing altering the heat flow.

In this study the simulation tool HEAT 2 was used to calculate the thermal conductance of the floor, γ_{up}. The thermal conductance depends on a number of variables e.g. floor materials, distance between pipes and vertical position of the pipes. To find γ_{up} a model of the floor section was generated, which can be seen in Figure 9 b). A temperature of 1°C was prescribed to the upper and lower boundaries and a temperature of 0°C was set in the pipes. In Figure 9 a) the simulation results can be seen. The results show a heat flow density for a temperature difference of one degree. This value is γ_{up}, which is used to calculate the delivered heat flow density for different water temperatures. For study specific calculations see Appendix I.

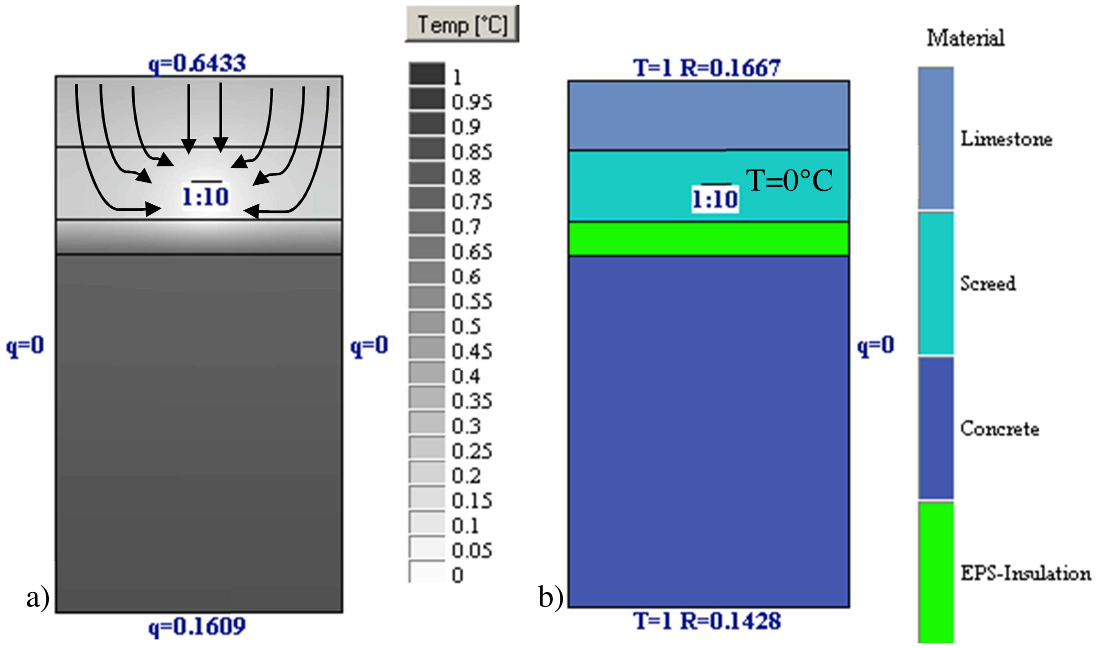


Figure 9 Model used in HEAT 2 to calculate the thermal conductance, γup

The water temperature calculations are based on the equation for a channel with transverse heat flow (see Figure 10), although γ is used as the heat conductivity.

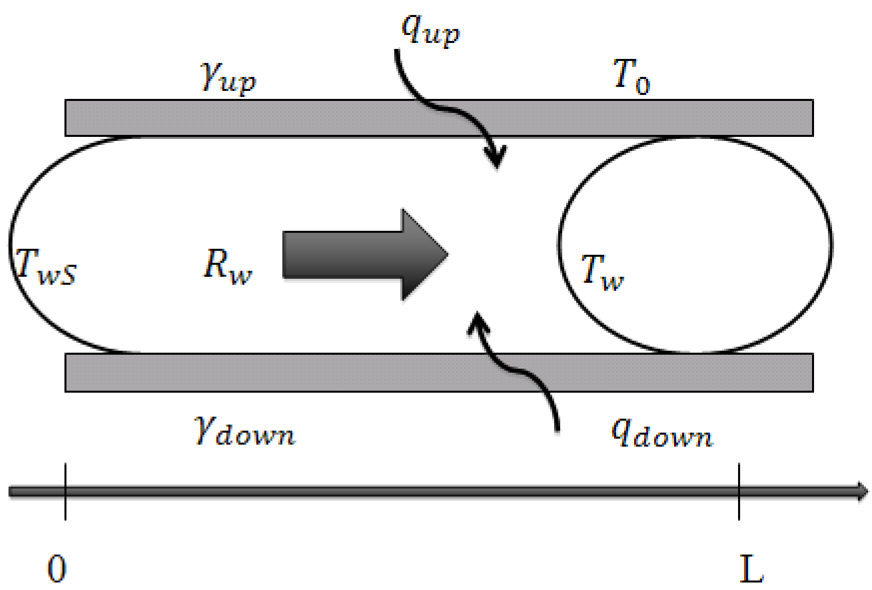


Figure 10 Modified transverse heat flow model

Equation 9 is used to calculate the heat flow; however the temperature is replaced by T_w from equation 10.

$$T_w(x) = T_0 + (T_{wS} - T_0) * e^{\frac{-x}{l_c}} \quad (10)$$

$$l_c = \frac{\rho * C_{pa} * R_w}{\gamma_{up} + \gamma_{down}} \quad (11)$$

where:

T_w	- Water temperature	(°C)
T_0	- Surrounding weighted temperature	(°C)
T_{wS}	- Supply temperature water	(°C)
l_c	- Characteristic length	(m)
ρ	- Density	(kg/m^3)
C_{pa}	- Heat capacity	(J/kgK)
R_w	- Water flow rate	(m^3/s)
γ_{up}	- Upward thermal conductance	
γ_{down}	- Downward thermal conductance	

The characteristic length is interesting as it has an impact on the heat transfer. The aim when designing the system is to keep the length of the pipe below the characteristic length of the system. This is due to that the temperature drop along the pipe will be the greatest in the beginning of the pipe. By keeping the length of the pipe below the characteristic length the entire circuit will be active in transferring heat. To use this method the design of the floor construction must be known, as well as the flow and supply temperature.

5.3.3 Method EN

The method is from a European standard used to calculate the effect of a floor heating system, nevertheless as the physics are the same in a floor cooling system this method can be used here as well. When the cooling capacity is calculated with standard EN1264, a simplified calculation model based on FEM (Finite Element Method) is used for system type A. This Single Power Function (see following equation) incorporates all main heat transfer parameters. [14]

$$q = B * a_B * a_{cc}^{m_{cc}} * a_D^{m_D} * a_U^{m_U} * \Delta T_H \qquad (W/m^2) \qquad (10)$$

where:

$B = B_0 = 6.7\ W/m^2$ for a conductivity in the pipe of $\lambda_R = \lambda_{R,0} = 0.35$ W/mK, and the pipe thickness is $S_R = S_{R,0} = (d_{outer} - d_{inner}) / 2 = 0.002$ m. For other cases variable B needs to be calculated (see SS-EN 1264 for equation)

a_B - surface covering factor, dependent on λ_E and $R_{\lambda,B}$ (-)
$\lambda_E = 1.2$ W/mK for cement with reduced humidity and for levelling layer. With another value the accuracy should be controlled.

a_{cc} - Pipe spacing factor, dependent on $R_{\lambda,B}$ (-)

a_D - Outer pipe diameter factor, dependent on cc and $R_{\lambda,B}$ (-)

a_U - Screed covering factor, dependent on cc and $R_{\lambda,B}$ (-)

For further information and tables on how to determine the factors, see Appendix B

$m_{cc} = 1 - \frac{cc}{0.075}$ valid for $0.050\ m \leq cc \leq 0.375\ m$ (-)

cc - Piping distance (m)

$m_D = 250(D - 0.02)$ valid for $0.01\ m \leq d \leq 0.03\ m$ (-)

D - Outer pipe diameter (m)

$m_U = 100 * (0.045 - S_u)$ valid for $Su \geq 0.015\ m$ (-)

S_u - Thickness of the screed layer above pipe (m)

$$\Delta T_H = \left| \frac{T_{wS} - T_{wR}}{\ln\left(\frac{T_{wS} - T_i}{T_{wR} - T_i}\right)} \right| \qquad (11)$$

where:

ΔT_H - Cooling medium differential temperature (K)

T_{wS} - Fluid supply temperature (^oC)

T_{wR} - Fluid return temperature (^oC)

T_i - Nominal room temperature (^oC)

Accordingly, the following factors will influence the magnitude of the cooling capacity: the pipe spacing, diameter and thickness, as well as conductivity of piping material and the thermal conductivity of the covering and screed. Some issues with the method are; it does not consider the water flow in the pipes, as well as it is presumed that the construction is well insulated.

5.3.4 Method Rule of thumb

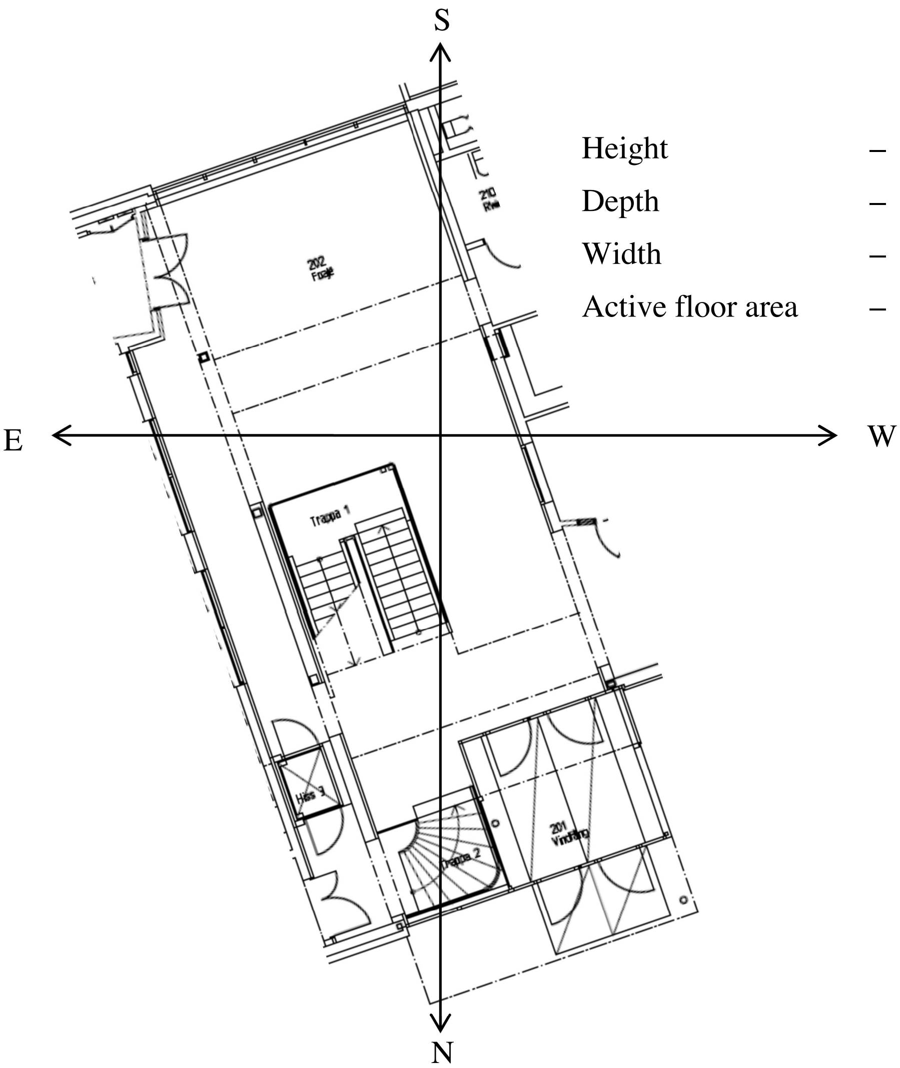
The method Rule of thumb is used by the HVAC-engineers to approximate the cooling capacity of the radiant floor cooling system. The method is basic, only following a rule of thumb, saying that the possible cooling effect is one fourth of the maximum heating effect.

The cooling capacity of the system in the study object was calculated according to all the methods described in chapter 5. The methods yielded results that differed, which is rather interesting. For further information, see chapter 8.

6 The study object

A large volume atrium located in Gothenburg Centre was chosen as a suitable reference object. The atrium is intended as an entrance hall and in this case, a lower demand than in an office space, is set on the thermal climate. The temperature is allowed to vary over the day, as there is no absolute upper temperature limit in the atrium.

The short sides of the entrance hall are located in a southeast – northwest direction and are made out of glass (see Figure 11), enabling the sun to shine directly into the atrium from around 9 a.m. until 1 p.m. Later in the afternoon, around 3 p.m., the sun reaches the other side of the atrium, affecting the indoor thermal climate until the sun is set. The long sidewalls in the atrium are mainly made out of gypsum board covered by sound insulation. For information concerning the orientation of the atrium in the main building, see Appendix G.



Height	–	13 m
Depth	–	19 m
Width	–	8.5 m
Active floor area	–	127 m^2

Figure 11 Direction of the atrium

A radiant floor system, which mainly is designed for heating purposes during winter, is installed in the entrance hall. The system is used for cooling purposes as well, as extra piping and an extra heat exchanger connects the radiant floor system to the buildings cooling system (see Figure 13). Accordingly the system runs cold water through the pipes during summer, decreasing the over temperature hours, making the climate more endurable. The summer design supply temperature should be sustained at 19°C if the outdoor temperature is above 17°C, according to the Technical specifications of the control and monitoring system. The flooring configuration can be seen in Figure 12. The radiant system is a type A system, which is described in chapter 3.

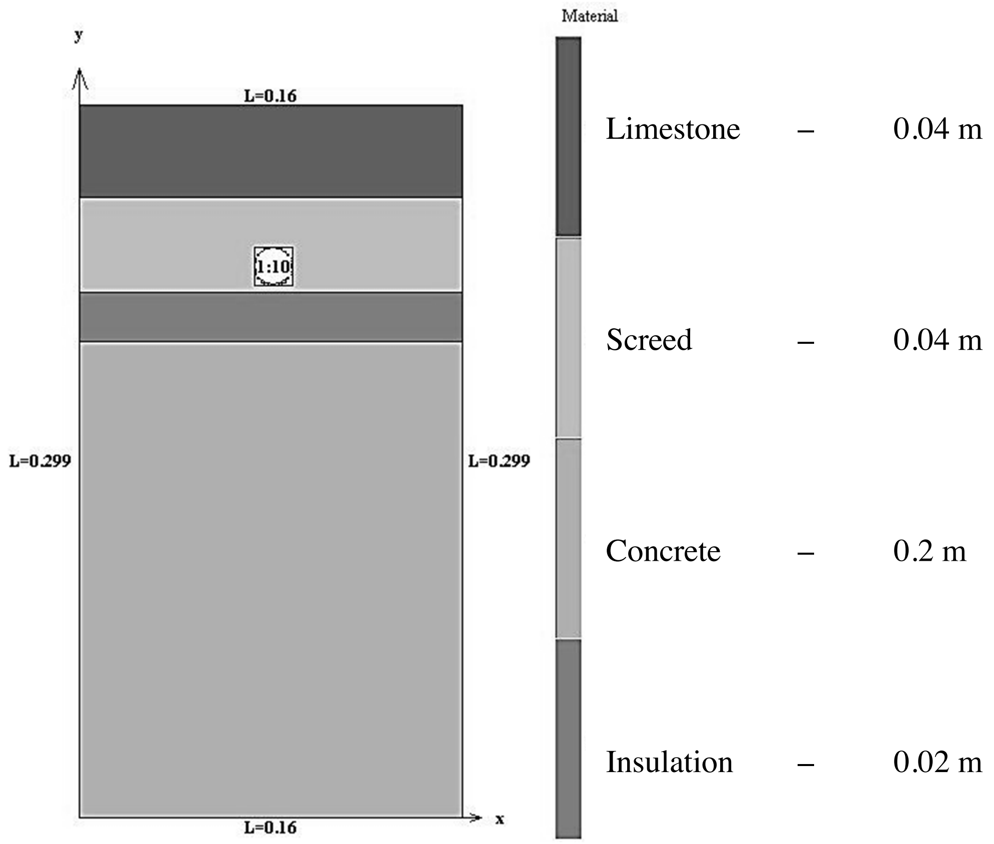


Figure 12 Floor construction in the study object

Figure 13 shows the functional scheme of the pipe installation. The interesting part is highlighted by a rectangle and zoomed in, showing the configuration of the floor cooling systems piping, the valves and the extra heat exchanger.

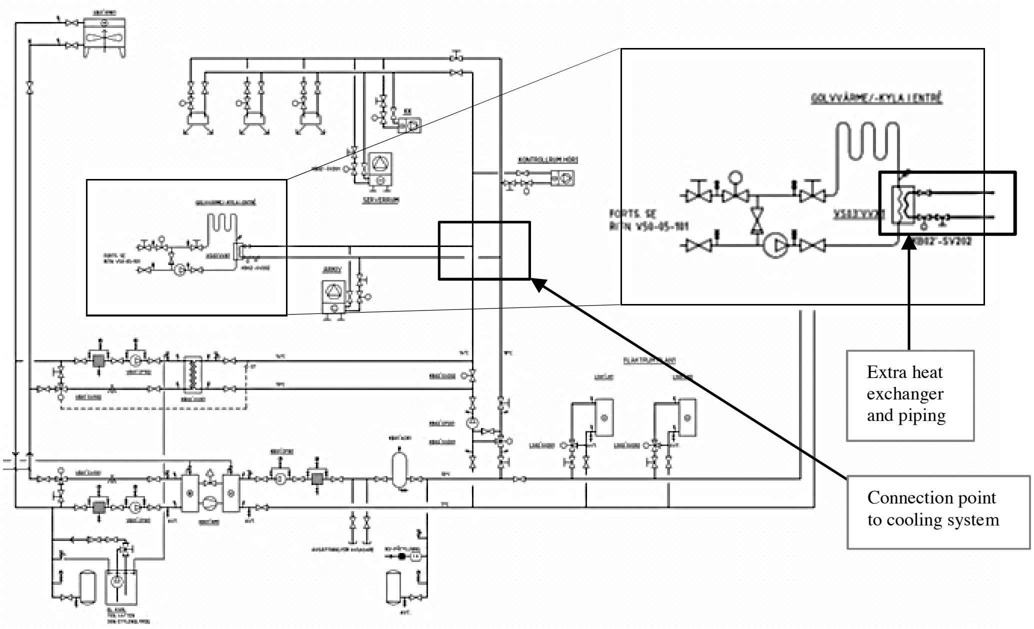


Figure 13 Functional scheme – cooling system piping

One of the main benefits with having radiant floor heating- and cooling system is the possibility to use the same piping, only adding an extra heat exchanger and piping to connect to the buildings main cooling system, see Figure 13.

7 Measurements - Study object

The purpose of these series of measurements was to get a basic understanding of the thermal climate in the entrance hall, as well as to obtain input data to the simulations. The relative humidity inside the atrium was studied to evaluate the dew point temperature and subsequently investigate the possibility to decrease the supply water temperature. The estimated error of the dew point temperature was calculated (see Appendix D). Another factor studied was the temperature profile, which was done by placing measuring devices at different heights in the atrium.

The study object chosen for the measurement series is, as mentioned before, an atrium, which is great in height and has a rather large floor surface area. Some difficulties were encountered, regarding where to place the measuring devices; this was corrected by conducting preparatory measurements, before the long-time measurements. Different areas of the floor were measured, and thereafter analysed, to find suitable placements of the measuring devices. Furthermore the measurements were meant to give an overview of the thermal climate in the building. The preparatory measurements will not be discussed thoroughly in this report, but a short description can be seen in Appendix E for those interested.

Long-time measurements were conducted during the period from 2011-07-04 to 2011-08-01. For information about the measuring devices, see Appendix C. For accuracy control of the measuring devices, see Appendix F. In Appendix G pictures that show the building and the placement of the measuring devices can be seen.

7.1 Placement of measuring devices

The long-time measurement program was designed based on the information obtained during the preparatory measurements (see Table 4 and Figure 14 for placement of the devices for the long-time measurements).

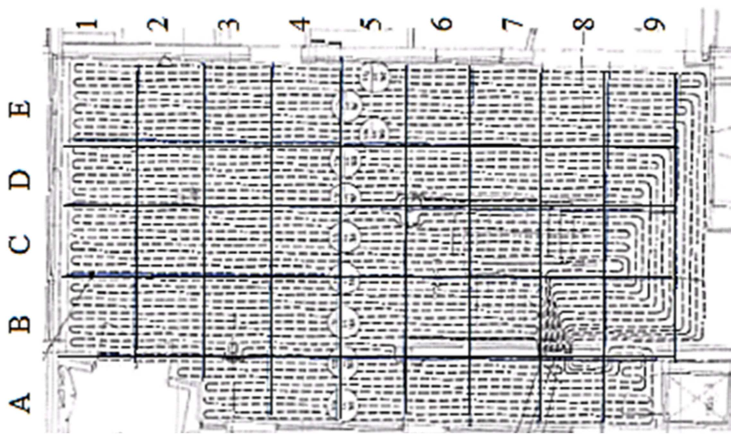


Figure 14 Division of floor surface area in squares

Floor surface temperatures were measured at three different locations beneath the stairs, see Table 4. Two thermometers were placed in a shaded area (T F and T J) and one (T E) was positioned to be affected by direct sunlight during approximately one hour per day.

Table 4 Measuring devices and their placements

Measuring device ID	Placement	Additional information
RH 2	Supply air	RH, Air temperature
RH 3	1.6 m above floor	RH, Air temperature
RH 4	Outside in shade	RH, Air temperature
RH 5	Exhaust air 12.9 m above floor	RH, Air temperature
T 7	9 m above floor	Air temperature
T 8	0.1 m above floor	Air temperature
T 9	5.3 m above floor	Air temperature
T E	Beneath stairs in square C5	Surface temperature, partly in sun at noon
T F	Beneath stairs in square C6	Surface temperature, in shade
T G	Supply water	Temperature
T I	Return water	Temperature
T J	Beneath stairs in square C8	Surface temperature

The temperature profile was measured, to get a basic understanding of the thermal climate inside the atrium. The measuring devices were placed at five different heights, see Figure 15. Two of them measured the relative humidity as well. Return and supply water temperatures were measured as well, see Appendix G for photo.

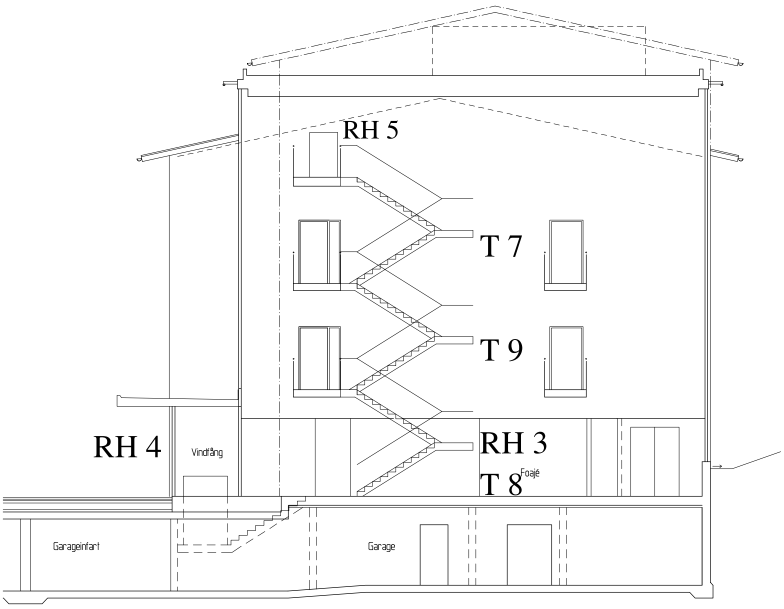


Figure 15 Positioning of the thermometers measuring the vertical temperature profile

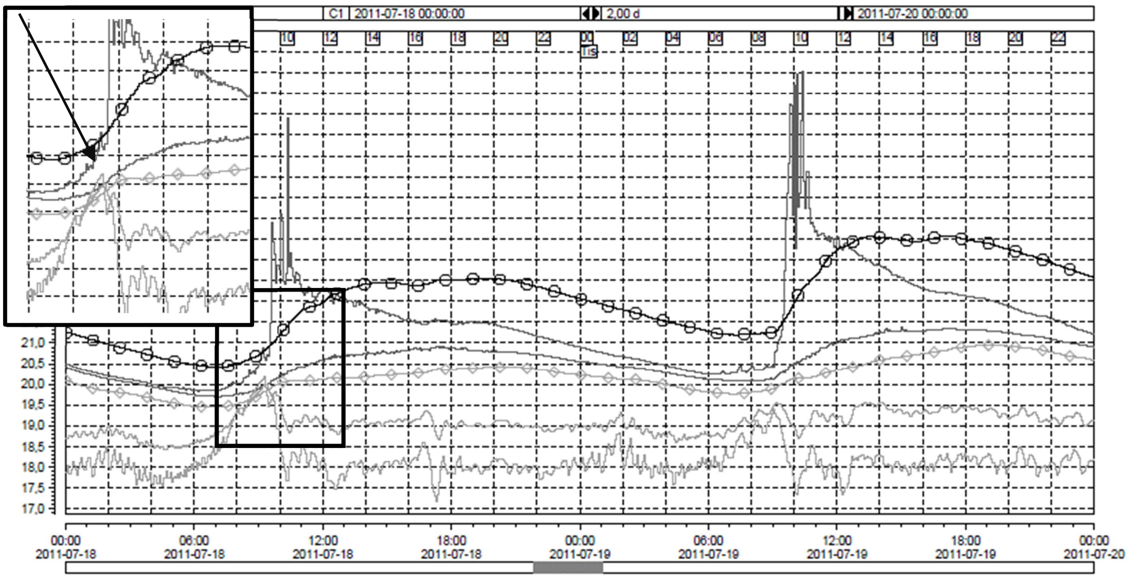
7.2 Results

In the following chapters, results from the measurements in the study object will be presented. A specific period, from 110718 to 110720, was selected for further analysis. This time period was partly chosen for the reason that the floor cooling was active during this time, and partly because the weather was clear and sunny, influencing the solar radiation, resulting in interesting data. The same period applies for all figures if not else specified. Some interesting figures from other periods are presented in Appendix G.

7.2.1 Floor surface temperatures

This chapter deals with the floor surface temperatures in different areas. For further information concerning the placement of the measuring devices, see previous chapter.

The data from the measuring devices varied greatly, which is shown in Figure 16. This can be explained by the fact that one of these deliberately was placed in a partly sunlit area, to see the influence of the sun radiation on the temperature (cell C5). The two remaining devices, which measured the floor surface temperature in shaded areas in cell C6 respectively cell C8, gave relatively equal temperatures.



	Färg	Enhet	Med	Min	Max	Linje-bredd	Benämning	Punkttyp
011		°C	22,07	20,43	23,57	—	RH 3 - Air temperature, 1.6 m above floor	O
004		°C	21,41	19,84	27,55	—	T E - Floor surface temperature in cell C5	
005		°C	20,59	19,70	21,37	—	T F - Floor surface temperature in cell C6	
008		°C	20,21	19,45	20,95	—	T J - Floor surface temperature in cell C8	◇
007		°C	19,08	18,42	20,01	—	T I - Water temperature, return	
006		°C	18,17	17,18	20,18	—	T G - Water temperature, supply	

Figure 16 Floor surface temperatures, supply and return water temperatures and air temperature

The temperature at height 1.6 m above floor can be seen as a reference temperature, when regarding the thermal climate on the entrance floor. Furthermore, the supply and return temperature is shown in the figure. The rectangle shows a period of interest where the supply water temperature rises and then, when the temperature is lowered again, an immediate response is shown in the floor surface temperature in cell C8; however in cell C6 and cell C5, the response is not as distinctive.

7.2.2 **Air temperatures**

The air temperature results will be accounted for in this chapter. Positioning of the measuring devices can be seen in Table 4.

Figure 17 and Figure 18, which show the temperature profile at specific times during 110718 and 110719, indicate that the air temperature varies greatly with height. In Figure 17 this is most prominent during the evening and least during the night hours. In Figure 18 the temperature profile changes gradually and peaks at 4 p.m..

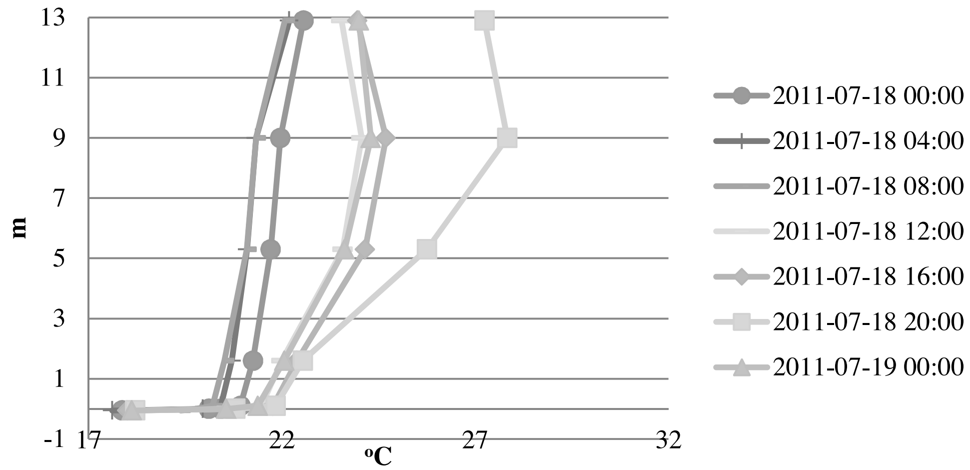


Figure 17 Temperature profile in the atrium 110718

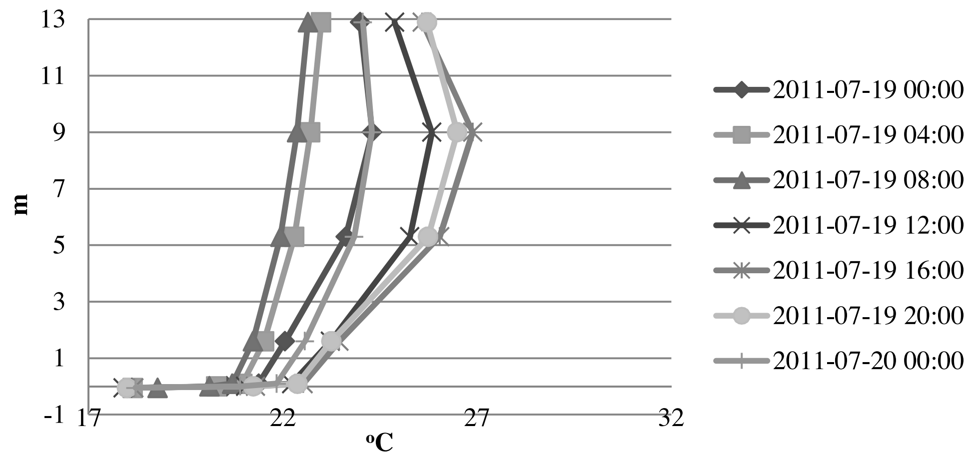
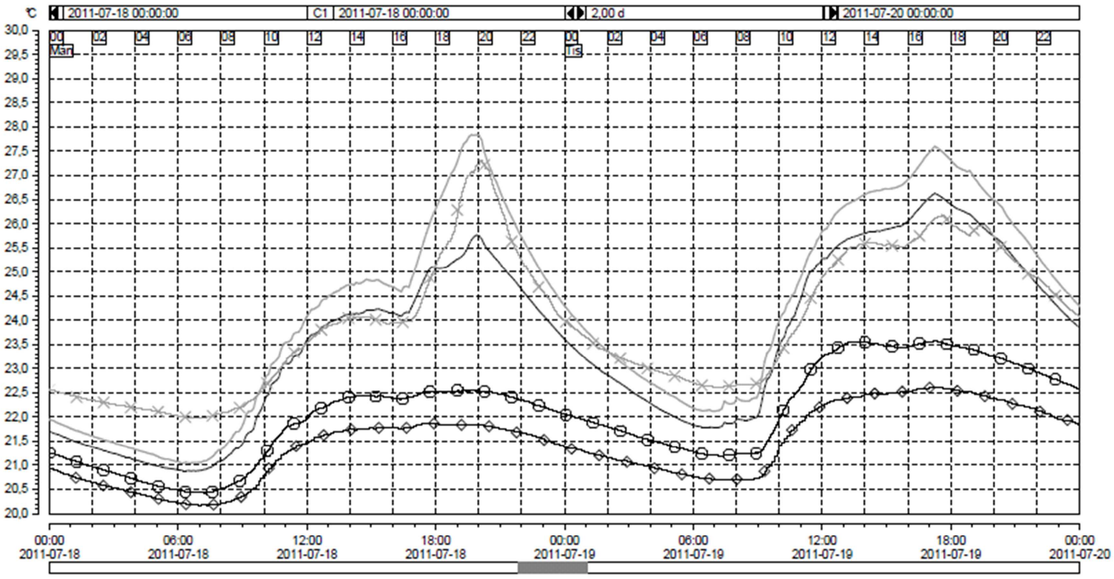


Figure 18 Temperature profile in the atrium 110719

The lines are approximately linear up to a height of 5.3 m, below that height the temperature increase is rather rapid, thereafter the air will be exposed to stratification. The device placed at the exhaust air shows a temperature decline, which is unrealistic. The fact that the temperature declines in this point may mean that the exhaust device does not only exhaust air from the entrance hall, but likely from elsewhere as well. This is also noticeable in Figure 19, which show the temperature at different heights during the period from 110718 to 110720. The exhaust temperature is lagging behind and thus results in smaller temperature fluctuations between night- and day-time, than for temperatures at height 5.3 m and 9 m.



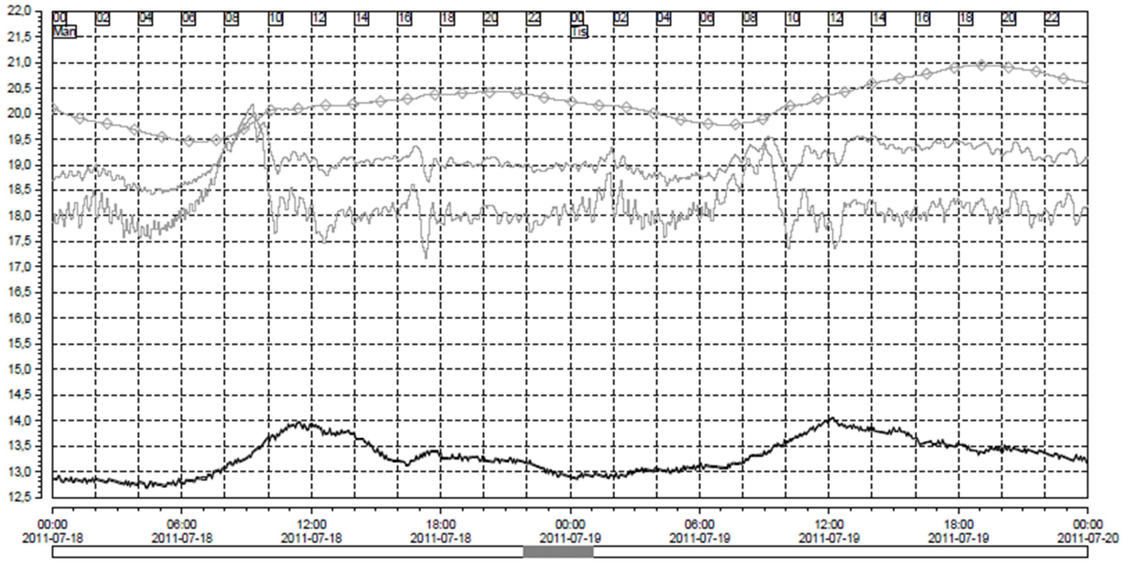
	Färg	Enhet	Med	Min	Max	Linje-bredd	Benämning	Punkttyp	
001		°C	24,23	21,05	27,84		T 7 - Air temperature, 9 m above floor		
015		°C	23,96	21,98	27,31		RH 5 - Exhaust air temperature, 12.9 m above floor	×	
003		°C	23,60	20,88	26,62		T 9 - Air temperature, 5.3 m above floor		
011		°C	22,07	20,43	23,57		RH 3 - Air temperature, 1.6 m above floor	o	
002		°C	21,44	20,17	22,61		T 8 - Air temperature, 0.1 m above floor	◇	

Figure 19 Temperature gradient from 110718 to 110720

At 5.3 m above the floor the maximum temperature is around 26.5°C, and at greater height the temperature reaches approximately 28°C. At the entrance floor the maximum temperature is around 23.5°C. During the entire measurement period, the highest temperature reaches 31.5°C at heights above 5.3 m. At the entrance floor this value is 26°C (for detailed information see Appendix H).

7.2.3 Water temperatures and condensation risk

Throughout the period from 110718 to 110720 (see Figure 20), the supply water temperature is relatively constant at 18°C. The temperature difference between supply and return water temperature is much lower than expected, around 1.5°C, which is not deviant, but valid for the entire time period. The small difference may indicate a low heat flow or a high water flow.



	Färg	Enhet	Med	Min	Max	Linje-bredd	Benämning	Punkttyp	
008		°C	20,21	19,45	20,95		T J - Floor surface temperature in cell C8	◇	
007		°C	19,08	18,42	20,01		T I - Water temperature, return		
006		°C	18,17	17,18	20,18		T G - Water temperature, supply		
026		°C	13,30	12,70	14,06		Tdew.alt.calculation		
025		°C	13,28	12,67	14,04		Tdew		

Figure 20 Water temperatures, dew point temperature and floor surface temperature

Figure 20 shows that the dew point is considerably lower than the supply water temperature. Furthermore, an even greater difference is seen between the dew point and the surface temperature. As the dew point temperature is much lower than the surface temperature the risk of condensation is very small. For more information concerning water temperatures and condensation risks see Appendix F.

7.3 Analysing the thermal climate in the study object

The thermal climate can be divided into smaller parts, see Figure 21. As seen, the PMV-PPD index and the operative temperature are included in the general thermal climate and the local thermal climate constitute of e.g. vertical temperature gradient and floor surface temperature [5].

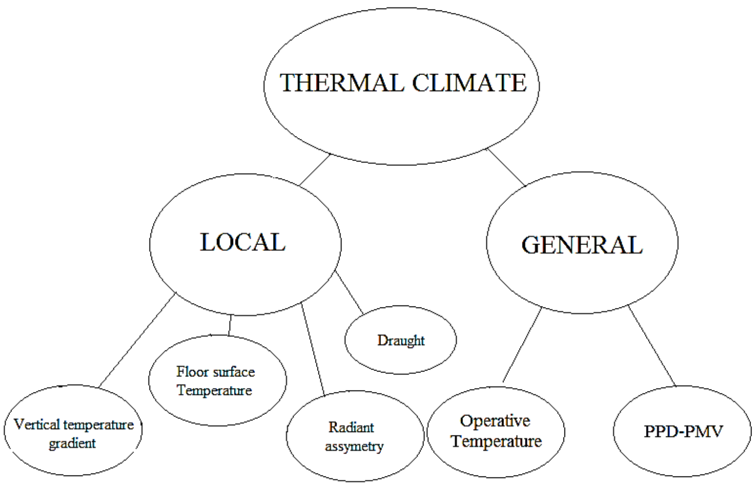


Figure 21 The thermal climate divided in local and general factors

Some other factors affecting the thermal climate are personal thermal factors, like metabolic rate and clothing insulation. These will not be discussed thoroughly here, but it can be mentioned that the degree of activity and the insulation of clothing plays a major role in the perception of the thermal climate since the human body will emit and hold latent and sensible heat differently depending on these factors [15], [7]

Local thermal factors are emphasized and discussed in the following chapter.

7.3.1 Floor surface temperatures

A very important factor when regarding the local thermal comfort and radiant floor systems is the floor surface temperature. In the standards a distinction between inactive occupancy and higher level of activity is made. For the first the lowest acceptable floor surface temperature is 20°C. For the latter, which is the case in the study object, a lowest temperature of 18°C is sufficient [6]. Further lowering of the surface temperature might be tolerable at higher activity level [7]; however the dew point temperature needs to be taken into account. The previous mentioned temperature levels are valid in both the perimeter and the occupied zone (see terms and definitions). If the radiant floor system is used for heating purposes as well, a maximum floor surface temperature of 29°C is set in the occupied zone, while in the perimeter zone the allowed maximum is 35°C [6].

The lowest acceptable floor surface temperature is far from reached in the measurements, both during the shorter time period and entire measurement period; see Figure 20 and Appendix F.

7.3.2 Vertical temperature gradient

According to standards the vertical temperature gradient (ΔT) must be less than 3°C/m [6], since a higher difference between the ankles and the head may cause discomfort. In an experimental study conducted by [16] the vertical temperature difference in a radiant floor cooling system was looked at. It was concluded that the temperature difference was very small (around 0.5°C) and occupants will not perceive any discomfort.

In the study object the vertical air temperature difference is well in the acceptable limit, see Figure 19. Although during the evening the gradient increase and get close to the limit, giving a much higher gradient than in the experimental study mentioned above.

7.3.3 Radiant temperature asymmetry

Radiant temperature asymmetry is another factor that affects the local thermal comfort and may cause discomfort. Most sensitive are people to warm ceilings, as well as cool window areas or walls. In large spaces with great height, a larger radiant asymmetry may be acceptable [17].

The radiant temperature asymmetry in the atrium is not thoroughly investigated, but with a large and warm glass wall and a cold floor surface, the radiant asymmetry is most likely rather large. Although, as the atrium is large, the effect on the thermal climate is reduced, based on the theory in the previous paragraph.

7.3.4 **Draught**

A positive effect of the radiant floor cooling system, considering the local thermal climate, is that draught issues is less likely to occur for this type of system than for convective systems, such as all air systems. Air velocities below 0.15 m/s at temperatures less than 24°C, are acceptable. Above this temperature higher velocities can be accepted, according to Olesen and during warm conditions it may even be favourable with a high air velocity. [18]

Draught issues are not studied in the measurements, as this was not considered to be a problem based on the theory in the previous paragraph.

7.3.5 **Relative humidity**

The relative humidity mainly affects the air quality, and in standard EN ISO 3370 the humidity range is set to 30-70% more for that reason, than for thermal climate reasons [18]. With this said, at greater relative humidity, the thermal climate may be compromised as well, since the ability of evaporating sweat from the skin can be limited. The influence on the thermal climate is otherwise relatively small. In a study conducted by Fanger el. al. different levels of air temperature and humidity in the ranges 18–28°C and 30-70% were analysed by a group of people. The results confirm that with increasing temperature and humidity the acceptability of the climate decrease and the perception of the air quality declines. [19]

During the measurement period in the study object, the relative humidity limit is never reached, see Appendix F for figures.

7.4 Analysis of the measurements

Discussed in this first paragraph is the overall experience of the thermal climate in the atrium, which is entirely subjective. The thermal climate in the entrance floor feels, though it at times is quite warm, relatively good during most of the day. At higher levels in the atrium the temperature is less endurable. The temperature reaches over 33°C at times, which is not a comfortable indoor temperature. This issue is probably due to the stratification phenomenon described in earlier.

In this paragraph an active and non-active system will be discussed and compared. The temperature profile and floor surface temperatures will be accounted for. In Figure 22 the temperature profile for the preparatory measurements is shown. The cooling system was inactive during this period. Figure 23 accounts for an active system.

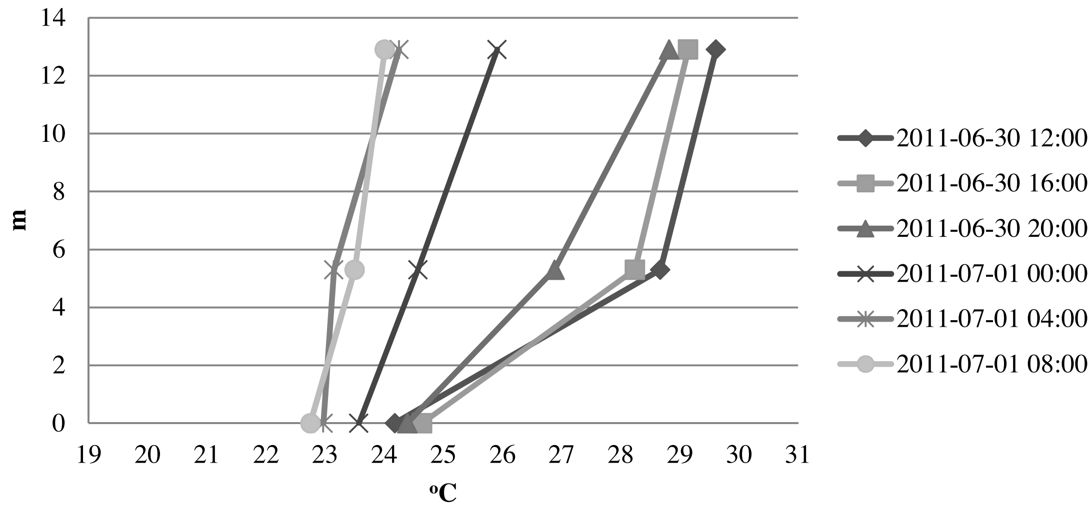


Figure 22 Temperature profile 110630-110701, inactive floor cooling system

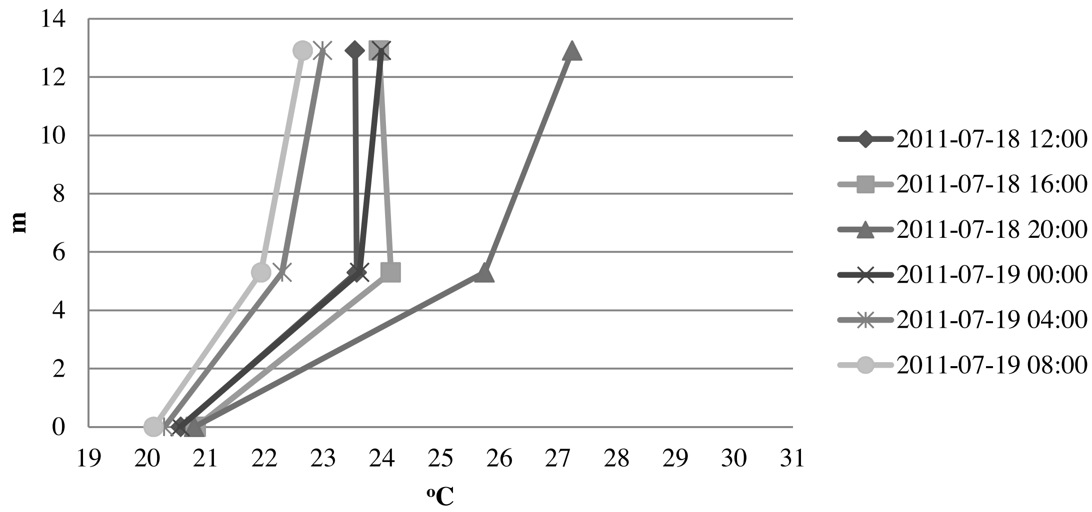


Figure 23 Temperature profile 110718-110719, active floor cooling system

The temperature profiles are preferably coupled with the outside temperature and sun intensity, before conclusions are drawn. In Table 4 the sun intensity and temperature during the days at hand are shown.

Table 5 Mean outside temperature and mean sun intensity

110630-110701	**110718-110719**	**110630-110701**	**110718-110719**
Temperature (ºC)	Temperature (ºC)	Sun intensity (Wh/m^2)	Sun intensity (Wh/m^2)
20	18	3100	4200

From the profiles it is evident that the floor surface temperature and the temperature at the entrance floor are lowered significantly by the floor cooling system. The temperatures at greater heights seem to be substantially lowered as well; this statement needs further research before drawing any conclusions though.

A number of conclusions can be drawn from the measurements.

1. The floor surface temperature is affected by the water temperature and this indicates that the floor cooling has some effect on the thermal climate.
2. The magnitude of the cooling effect can be discussed though, and the answer might be not enough to justify the system; however one should bear in mind that the system is used for floor heating purposes during the winter as well. This will increase the viability of the system since the extra cost of turning it into a cooling system, is not that great. The effect issue will be further investigated in the following chapter.
3. The positive effect on the thermal climate decreases with height, and at greater heights the effect is not as evident. Due to that people stay in this environment when they climb the stairwell, this might be a factor to consider. Perhaps a ventilation window or greater supply and exhaust air flow may be considered.
4. The improvement potential of the system seems relatively large, since the dew temperature is much lower than the water temperatures, as well as the surface temperatures. This leads to a possibility to decrease the supply water temperature and hence lower the floor surface temperature, subsequently lowering the air temperature. It is important to bear in mind that the moisture supply was very low during the measurement period, due to that the building was closed. If the moisture supply increases, the dew point will most probably increase as well. This requires further investigation, but can be done through estimates of the number of people in the atrium and their moisture contribution. Furthermore, the calculation of uncertainty of the dew point temperature needs to be considered, as a maximum deviation of 1.2°C is calculated.
5. And to conclude, the system can probably be improved if the control system has a delay before changing between heating and cooling mode. This is due to that the thermal mass will be used more efficiently. Currently the fast change between the modes results in that the full potential of the system is not reached. Some points of improvement will be studied further in the following chapter.

8 Delivered cooling effect - Study object

The cooling effect delivered by the floor cooling system was studied through a number of different calculation methods. The aim of the calculations was to find possibilities to improve the system, as well as to compare the calculation methods. The different methods are described in chapter 5.

8.1 Calculated cooling effect

The rule of thumb method approximates the possible cooling effect as one fourth of the deigned heating effect. In this case the heat flow density was 105 W/m^2, giving a cooling effect of about 25 W/m^2.

The total heat flow from the pipes was calculated to be 20 W/m^2 with method waterside. The calculated heat flow density changes with changed water temperatures, and this value is valid for a hot summer day. As this is the maximum heat flow rate it sets an upper limit for the real cooling capacity of the system, hence the delivered cooling effect must be lower than 20 W/m^2.

The academic method incorporates Heat 2 to find $\gamma_{up.}$ In Figure 24 the simulated results can be seen, where γ_{up} is 0.7143 W/mK. γ_{up} was subsequently used to calculate the delivered heat flow density. For calculations see Appendix I (Academic Method).

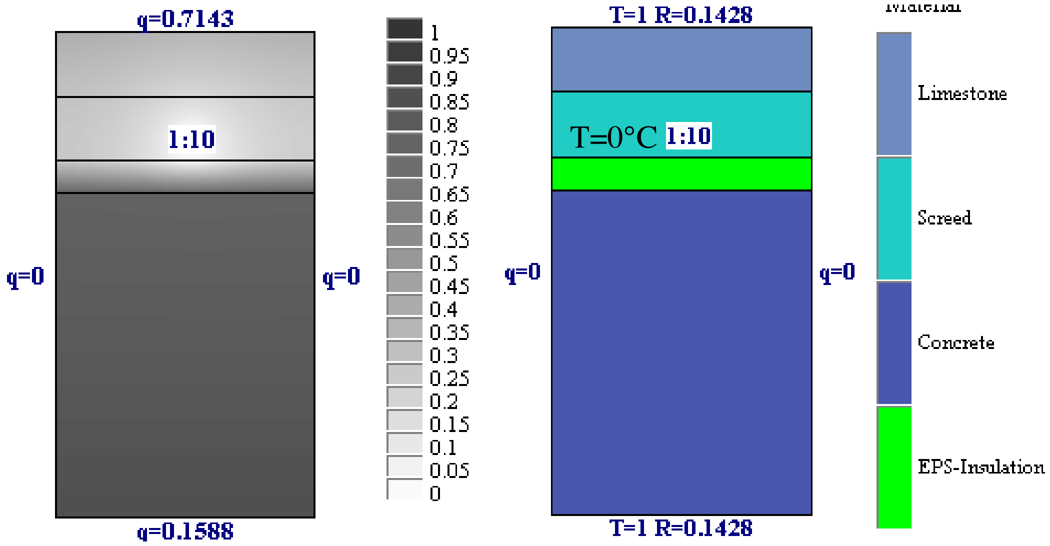


Figure 24 Model used in HEAT 2 to calculate the thermal conductance, γup

The academic method was found to be the most adequate method, and was consequently used to calculate the heat flow density. The system delivered a cooling effect of 16.5 W/m^2, for calculations see appendix I.

Calculations according to the EN1264 method resulted in a heat flow density of 19.6 W/m^2, which is rather high. The reason for this is that the method does not take the floor structure beneath the heat-conducting layer into account, instead the floor construction is assumed to be well insulated. This is not the case in the study object, as the loss downwards is more than the 10 % presumed in EN1264-2. Moreover, the method does not account for the water flow rate, instead it focuses more on the pipe material, which in the literature review is said to be less important. The calculations can be seen in Appendix I.

8.2 Evaluation of the methods

The above mentioned calculation methods have been compared in order to see how well they correlate. The calculated total heat flow from the pipes (method waterside) is showed as well, as an upper limit value. The calculation method in EN1264 has been used as a comparison to the academic method based on Henriks lecture. The delivered heat flow density calculated by these methods can be seen in the following table.

Table 6 Delivered heat flow density with different methods

Case 1	Original system	**Case 5**	Decreasing supply temp -
Case 2	Increasing water flow+	**Case 6**	Decreasing supply temp --
Case 3	Increasing water flow ++	**Case 7**	Decreasing supply temp ---
Case 4	Increasing water flow +++	**Case 8**	Compare between EN1264 and Academic

		CASE 1			**CASE 2**			**CASE 3**			**CASE 4**		
		Waterside	EN1264	Academic	Waterside	EN1264	Academic	Waterside	EN1264	Academic	Waterside	EN1264	Academic
T_air	°C	23.5	23.5	23.5	23.5	23.5	23.5	23.5	23.5	23.5	23.5	23.5	23.5
T_supply	°C	17.9	17.9	17.9	17.9	17.9	17.9	17.9	17.9	17.9	17.9	17.9	17.9
T_return	°C	19.6	19.6	-	19.6	19.6		19.6	19.6		19.6	19.6	
Water flow	l/hr	130	-	130	150	-	150	170	-	170	200	-	200
Results													
Heat flow density	W/m²	20.3	19.6	16.5	23.4	19.6	17	26.5	19.6	17.3	31.2	19.6	17.8
T_return	°C	-	-	19.6	-	-	19.4	-	-	19.3	-	-	19.1

		CASE 5			**CASE 6**			**CASE 7**			**CASE 8**		
		Waterside	EN1264	Academic	Waterside	EN1264	Academic	Waterside	EN1264	Academic	Waterside	EN1264	Academic
T_air	°C	23.5	23.5	23.5	23.5	23.5	23.5	23.5	23.5	23.5	23.5	23.5	23.5
T_supply	°C	17	17	17	16	16	16	15	15	15	17.9	17.9	17.9
T_return	°C	19.6	19.6	-	19.6	19.6		19.6	19.6		19.6	19.6	-
Water flow	l/hr	130	-	130	130	-	130	130	-	130	150	-	140
										η (Insulation efficiency)			0.9
Results													
Heat flow density	W/m²	31	21.2	19	43	22.9	23.3	54.9	24.6	26.9	21.9	19.6	19.3
T_return	°C	-	-	18.9	-	-	18.4	-	-	17.8	-	-	19.6

In Table 6 selected cases are compared. In Case 1 the measured temperature and water flow is used as input data, thus representing the existing system. Case 2-4 are improvement by altering the water flow and in Case 5-7 the supply temperature is changed. In Case 8 the Academic method is modified by altering the insulation efficiency and water flow in order to give the same result as in method EN1264. Case 8 supports that the EN1264 method assumes a 10% loss of the heat flow density downwards, based on that the insulation efficiency was changed to almost 0.9, corresponding to a 10% loss.

8.2.1 **Improving the floor cooling system**

Based on table Table 6 the following two figures were generated. In Figure 25 the heat flow density, dependent on the water flow rate, is shown. The values are based on the Academic method, as it took the water flow rate into account.

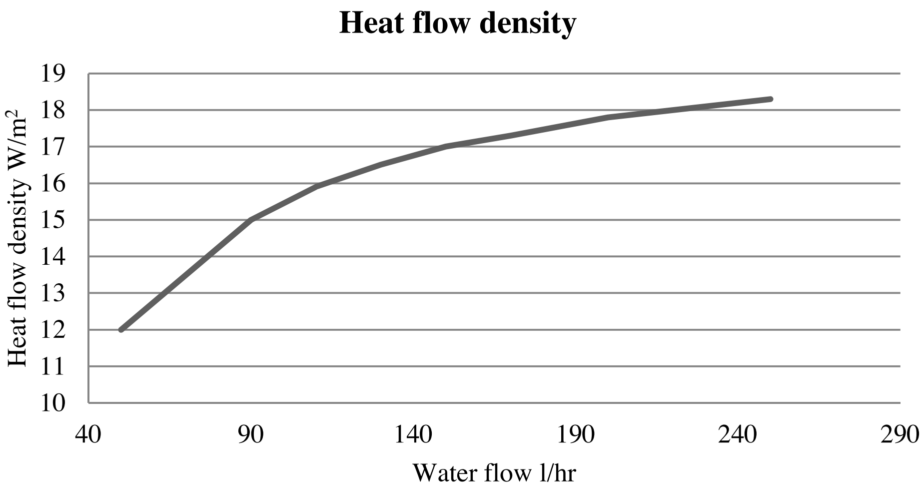


Figure 25 Heat flow density depending on water flow rate

In Figure 26 the effect of varying the supply water temperature is shown.

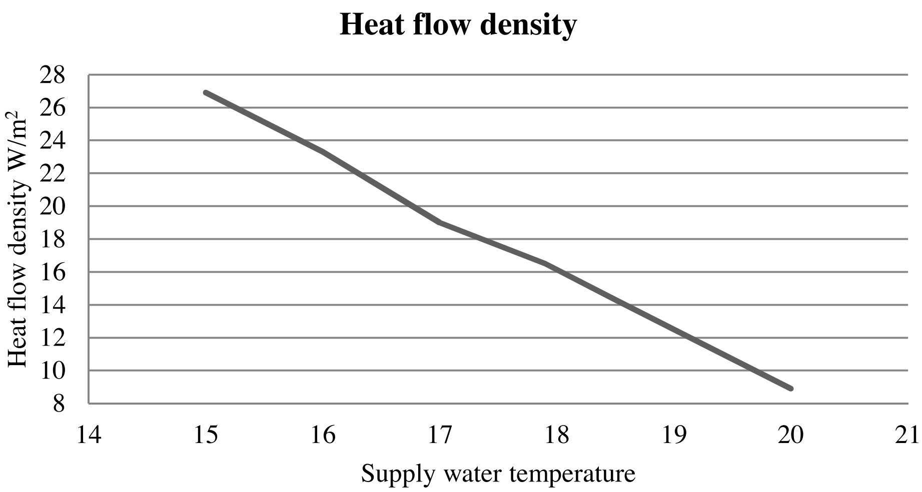


Figure 26 Heat flow density depending on supply water temperature

By comparing Figure 25 and Figure 26 it can be concluded that a change in water flow rate does not affect the heat flow density much, instead a lowering of the supply water temperature is more beneficial, acquiring a larger increase in the heat flow density.

9 Simulations - Indoor Thermal Climate

One of the main objectives of the master thesis was to investigate, both through measurements and simulations, how the thermal climate in the atrium was affected by the floor cooling. COMSOL Multiphysics, which will be described briefly in the following chapter, was used to perform the simulations.

The thermal climate is affected by a number of different heat transfer mechanisms, which all need to be combined, ending up with a rather complex and challenging matter to understand and simulate. Therefore the physics and models needed to be verified in COMSOL, see Appendix H. The final model, with a number of different scenarios is presented in chapter 9.

9.1 Simulating in COMSOL Multiphysics

COMSOL Multiphysics 4.2 is a simulation software, in which it is possible to work with a graphical user interface, where a model can be created and analysed. By using different modules a wide spectrum of physics can be studied. In this study the heat transfer module and CFD (Computational Fluid Dynamics) module has been used, since it corresponds well with the aim of studying the thermal climate and air movement. In order to decrease the computational time the simulations were performed on a cluster system. One node in the system was used, which encompassed 8 cores and 48 GB of memory, thereby decreasing the computational time.

9.1.1 Model

The model in the main simulations encompasses obstacles to simulate a realistic environment, with stairs and footbridges, see Figure 27. Some variables have been altered in the model, e.g. the surface temperatures, as well as ventilation flow and supply air temperature are changed to simulate different scenarios. The simulations start with the same initial temperature and run over a time of 36000 seconds.

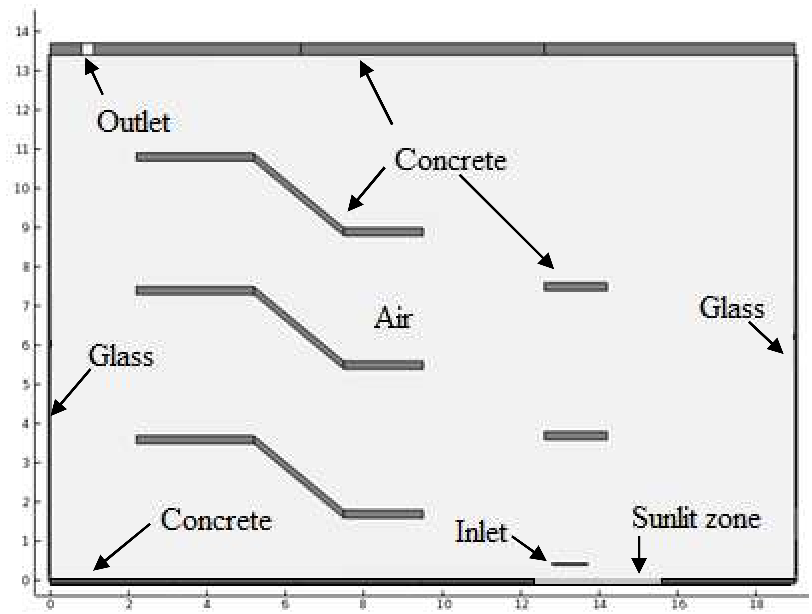


Figure 27 2D room simulated, including obstacles

In Figure 28 the model builder view from COMSOL can be seen. This figure is presented to give the reader an overview on how the software works, as well as to see the software possibilities. In Figure 28 a) the materials included in the simulations can be seen, these have been altered to better fit the real situation in the study object. As an example: the windows are simulated as massive glass, with the thermal conductivity changed to represent a U-value of 1.2 W/m^{2}K. In Figure 28 b) the Non-isothermal flow node is seen, which was used to simulate the heat transfer in the air volume and in the solids. In the same node it was possible to change the inlet and outlet properties to regulate the ventilation. Figure 28 c) illustrate a possible solver configuration, which variables to be solved are set, as well as in which order they should be solved. The basic solver configuration was utilized.

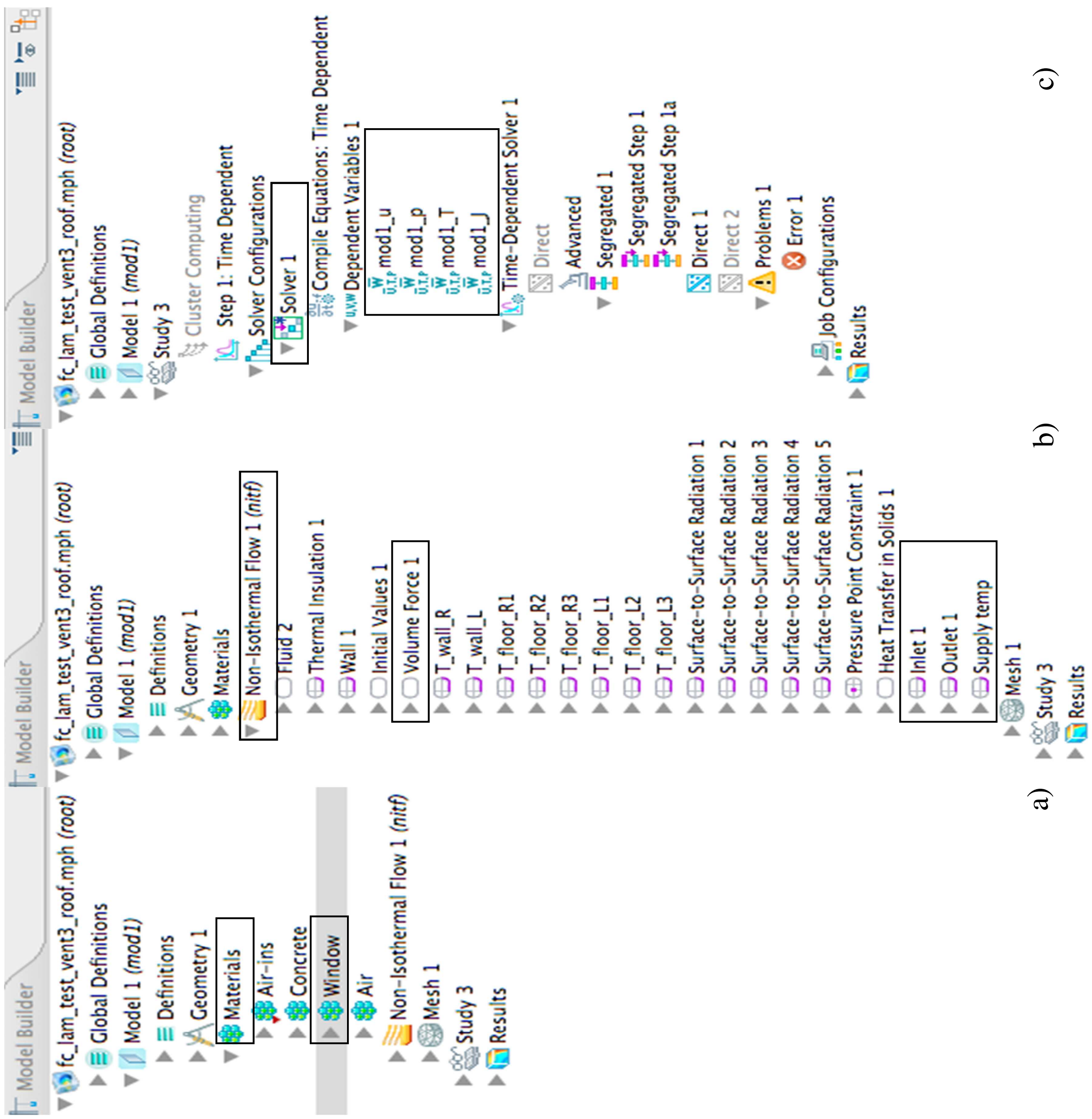


Figure 28 Model Builder view from COMSOL showing parts of the configuration for the simulations

9.2 Results

Five scenarios, and one baseline case, is looked at and discussed in this thesis. The simulations studies are the following:

0. A baseline case, no cooling
1. A study with solely floor cooling
2. A study with solely ventilation
3. A combined study with both floor cooling and ventilation
4. A study with solely ceiling cooling
5. A study with floor cooling and ventilation (changed placement of supply air device to ceiling)

In the simulations, one floor surface zone is constant at 32°C to simulate a heated sunlit floor area.

9.2.1 Study 0 - Baseline case

Study 0 is intended to be viewed as a baseline scenario, as the ventilation system and floor cooling system is inactive. Dependent variables are seen in the table below.

Table 7 Dependent variables in the simulations

Variable	Value	Comment
Initial temp	27°C	Standard initial temperature
Ventilation	Off	-
Cooling	No	32°C in one “sunlit” zone
Obstacles	Yes	

As seen in Figure 28 the temperature is uniform at around 27°C in the entire atrium. The velocity field shows rather low air speeds, which is expected as the air volume is not influenced by external effects, like ventilation, or much by temperature differences (giving a low natural convection). One exception is the right glass wall, where the warm air from the sunlit areas will ascend. The surface temperature at the left glass wall is set as cold; hence the air will be colder close to the wall and consequently descend.

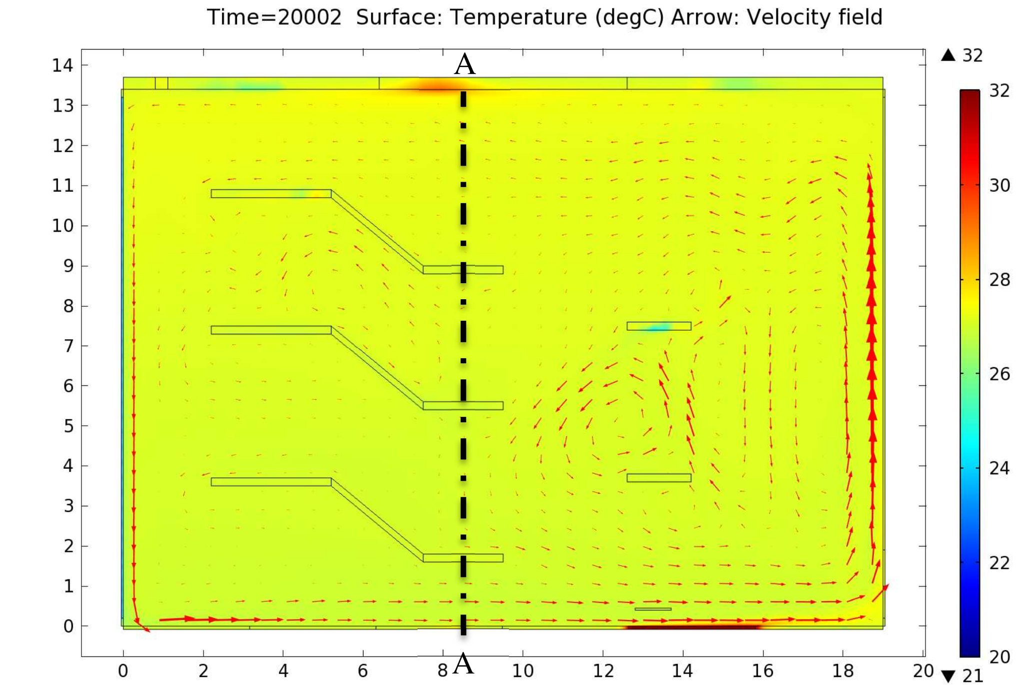


Figure 28 Temperature and velocity field baseline case

In Figure 29 the temperature profile in cutline A-A (figure above) is shown at different times. As mentioned before the temperature is rather uniform, giving a virtually unexciting temperature gradient.

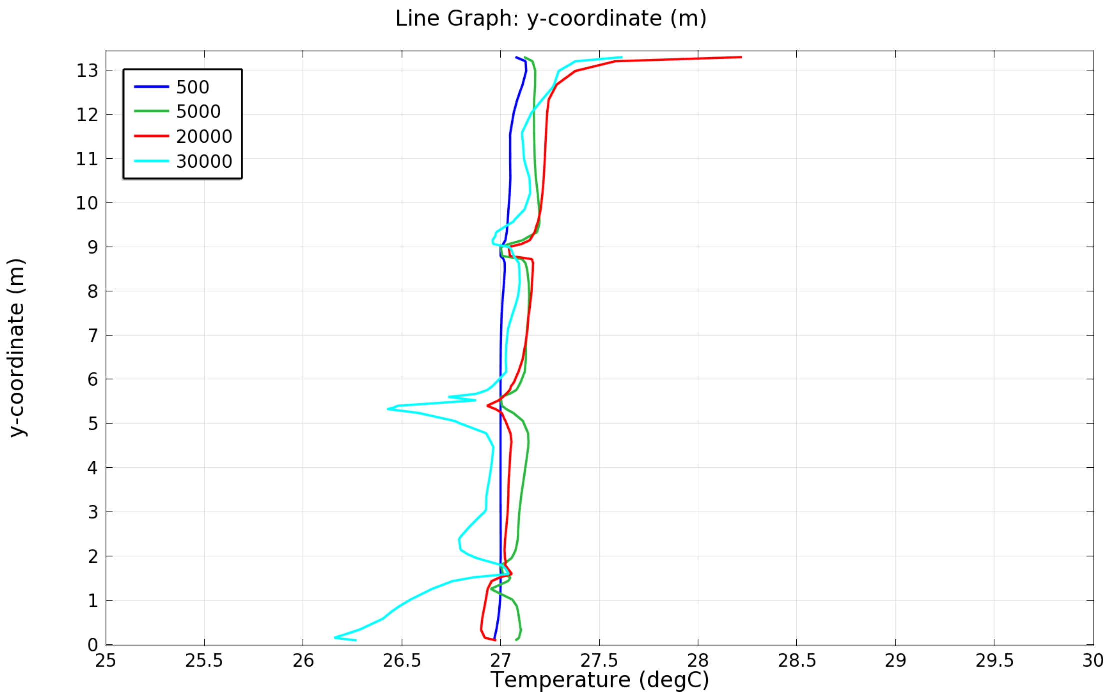


Figure 29 Temperature profile baseline case

In the following studies the temperature profiles are taken in cutline A-A as well. The arrows representing the velocity field are of relative length and they cannot be compared between studies.

9.2.2 Study 1 - Effect of the floor cooling system

In this study the effect on the thermal climate by the floor cooling system was studied. Selected variables important to the simulation are seen in Table 8.

Table 8 Dependent variables in the simulations

Variable	Value	Comment
Initial temp	27 °C	Standard initial temperature
Ventilation	Off	No ventilation
Cooling	Yes	Floor surface temperature 20°C, 32 °C in one "sunlit" zone
Obstacles	Yes	

The results from the simulation are seen in Figure 31 and Figure 30. In the first figure the stratification phenomenon, described in the measurement chapter, is clearly shown. The air movement is mainly limited to areas where the surface temperature is dissimilar to the surrounding air temperature.

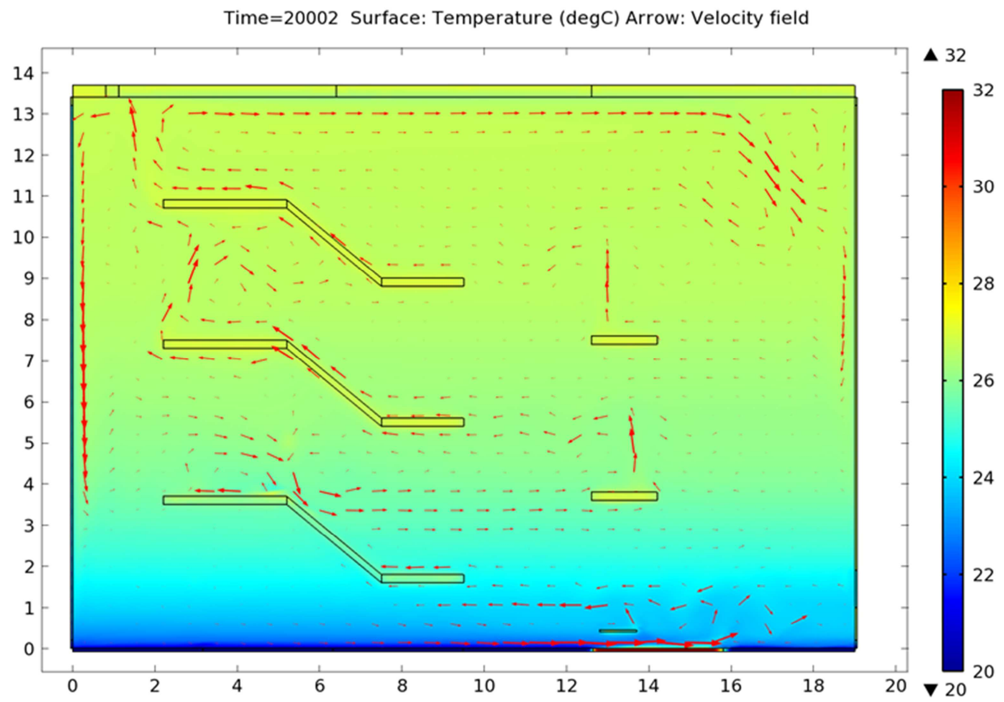


Figure 31 Temperature and velocity field with solely floor cooling

In Figure 30, which shows the temperature profile, the impact of the floor cooling system on the thermal climate is illustrated. The temperature at the first floor is substantially lowered by the floor cooling system, while higher up in the atrium the temperature is not affected as much. The dents in the curves are explained by that the cut line is taken through the stairs, which has a higher thermal mass and thereby a slower response to temperature changes.

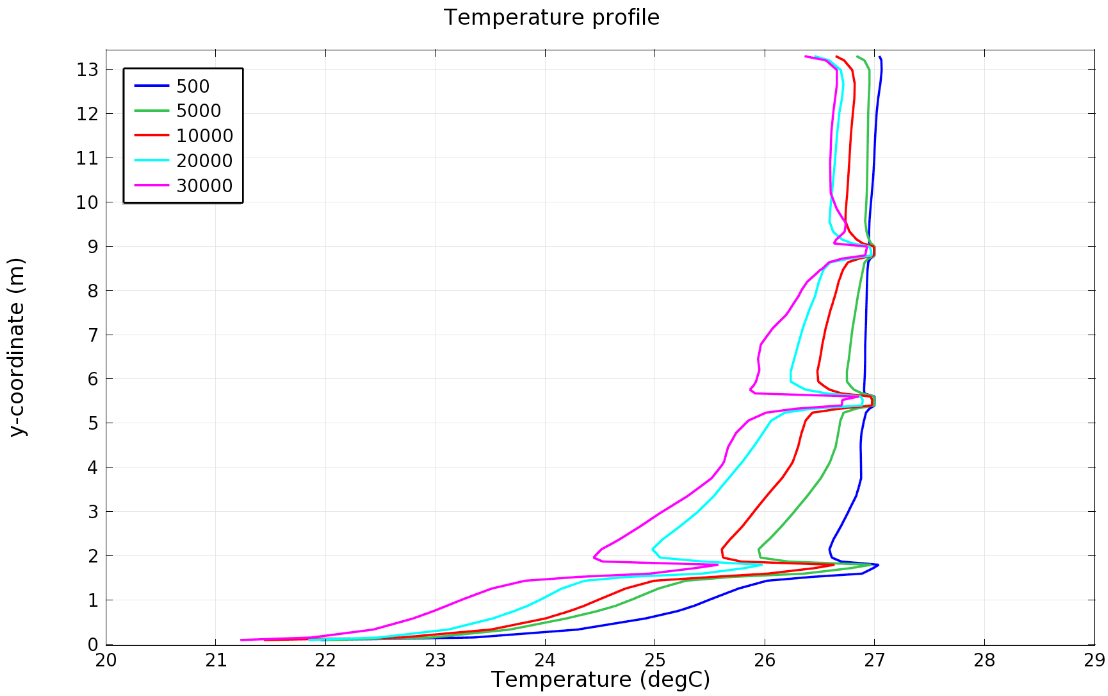


Figure 30 Temperature profile with solely floor cooling

9.2.3 Study 2 - Effect of the ventilation system

In Study 2 the impact the ventilation system has on the thermal climate was studied. A constant air supply temperature and flow, which corresponds to the real measured values, is set.

Table 9 Dependent variables in the simulations

Variable	**Value**	**Comment**
Initial temp	27 °C	Standard initial temperature
Ventilation	On	Supply air temperature 22°C
Cooling	No	Floor surface temperature 32°C in one "sunlit" zone
Obstacles	Yes	

In Figure 31 the stratification phenomenon is not seen as clearly as in **Fel! Hittar inte referenskälla.**, instead the temperature difference is more linear, which is due to a well-mixed air volume.

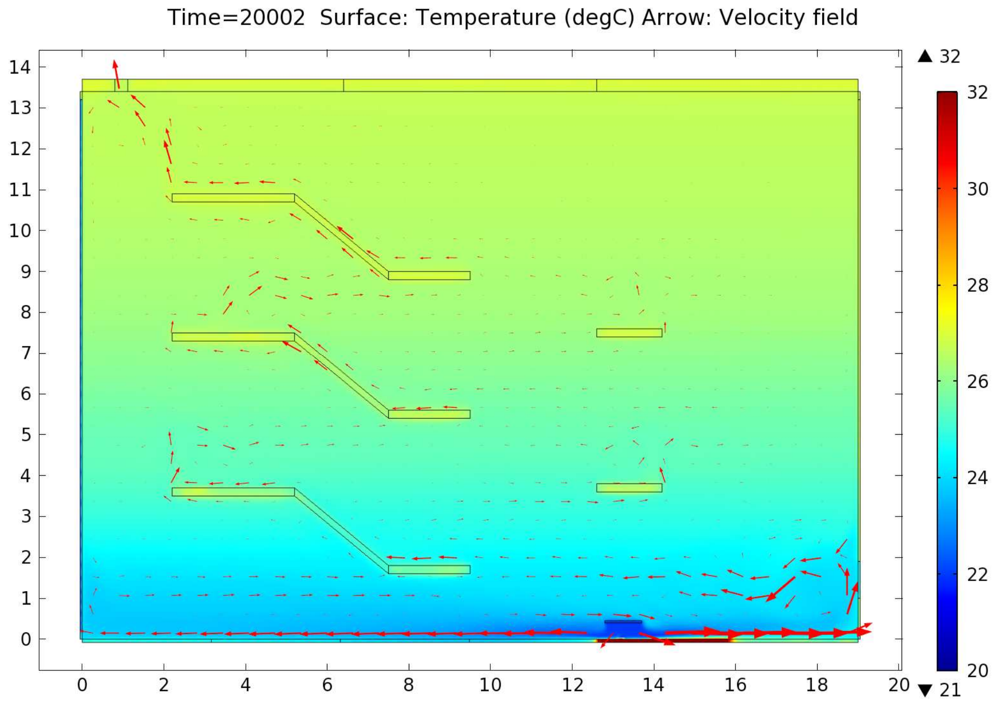


Figure 31 Temperature field and velocity field with solely ventilation

Figure 32 shows two interesting aspects, one is that temperature gradient is even, and the other one is that the ventilation affects the temperature in the entire space. This is, again, an indication that the ventilation mixes the air and thereby evens out the temperature in the room.

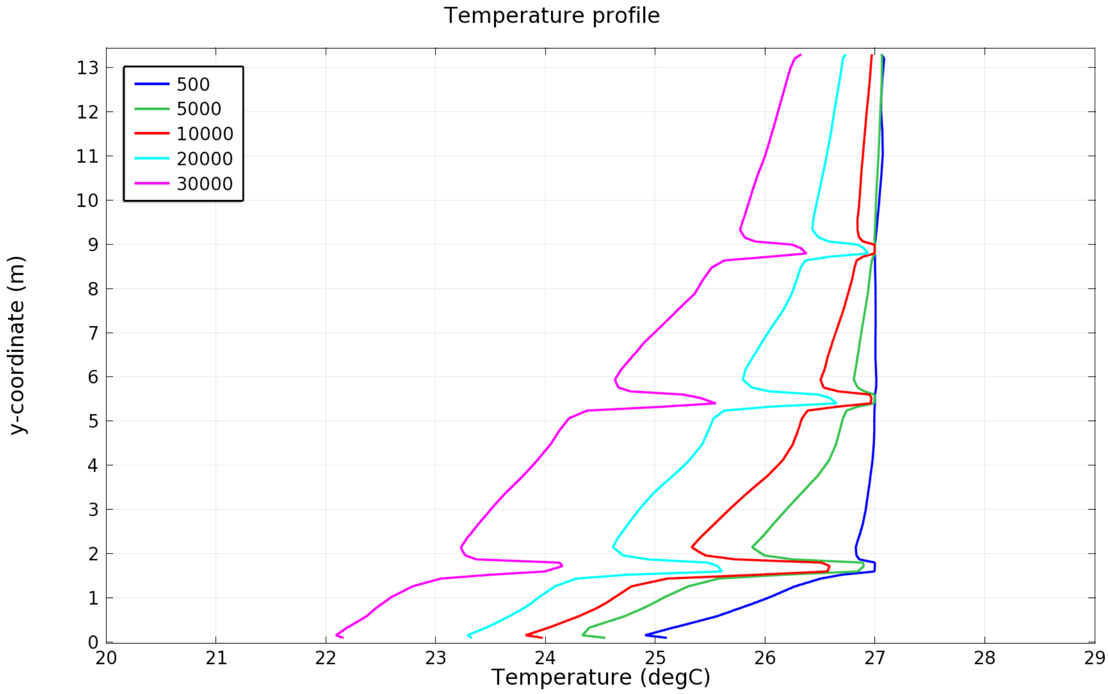


Figure 32 Temperature profile with solely ventilation

9.2.4 **Study 3 - Effect of the combined system**

In Study 3 the ventilation, as well as the floor cooling is active during the simulation, to see the combined effect of the systems.

Table 10 Dependent variables in the simulations

Variable	**Value**	**Comment**
Initial temp	27°C	Standard initial temperature
Ventilation	On	Supply air temperature 22°C
Cooling	Yes	Floor surface temperature 20°C, 32°C in one "sunlit" zone
Obstacles	Yes	

The temperature field in Figure 33 clearly shows a stratification phenomenon at the first floor, which is due to the floor cooling system. The effect is less apparent at greater height due to a well-mixed air as a result of the ventilation system.

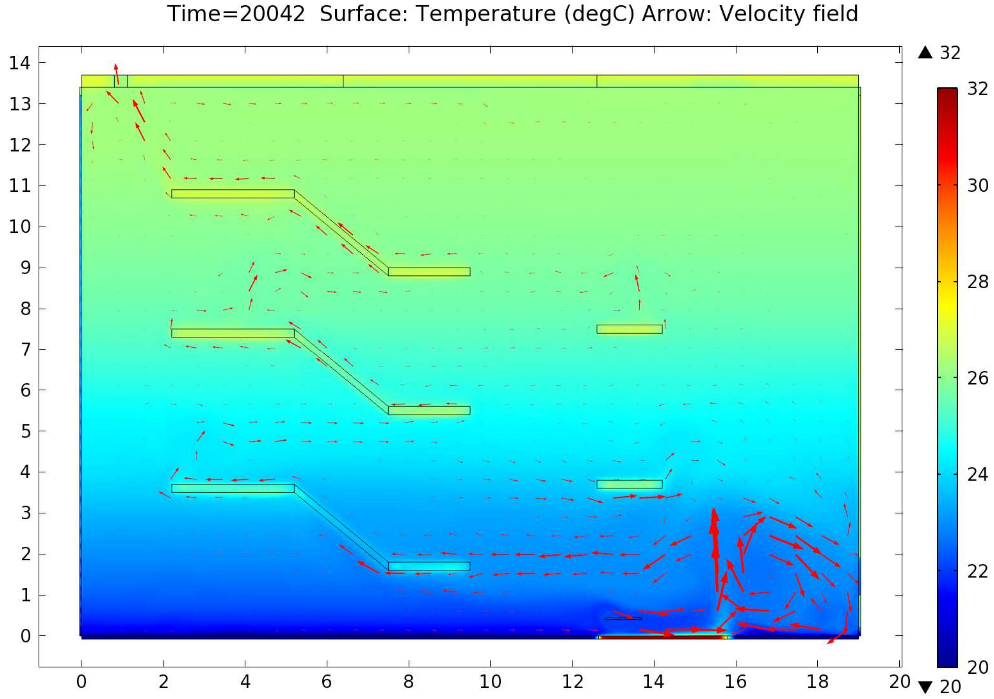


Figure 33 Temperature field and velocity field with floor cooling and ventilation

In Figure 34 the temperature profile is seen. In Study 3 the temperature is lower than in the first two studies, both at the upper and the lower levels. This is expected as the combined system has a higher cooling effect. What is interesting is that the combined system contributes to a positive effect on the thermal climate at higher levels, although the effect is only about 1.5°C. Most effective is, not surprisingly, the combined system at the first and second floor, where the climate is substantially improved.

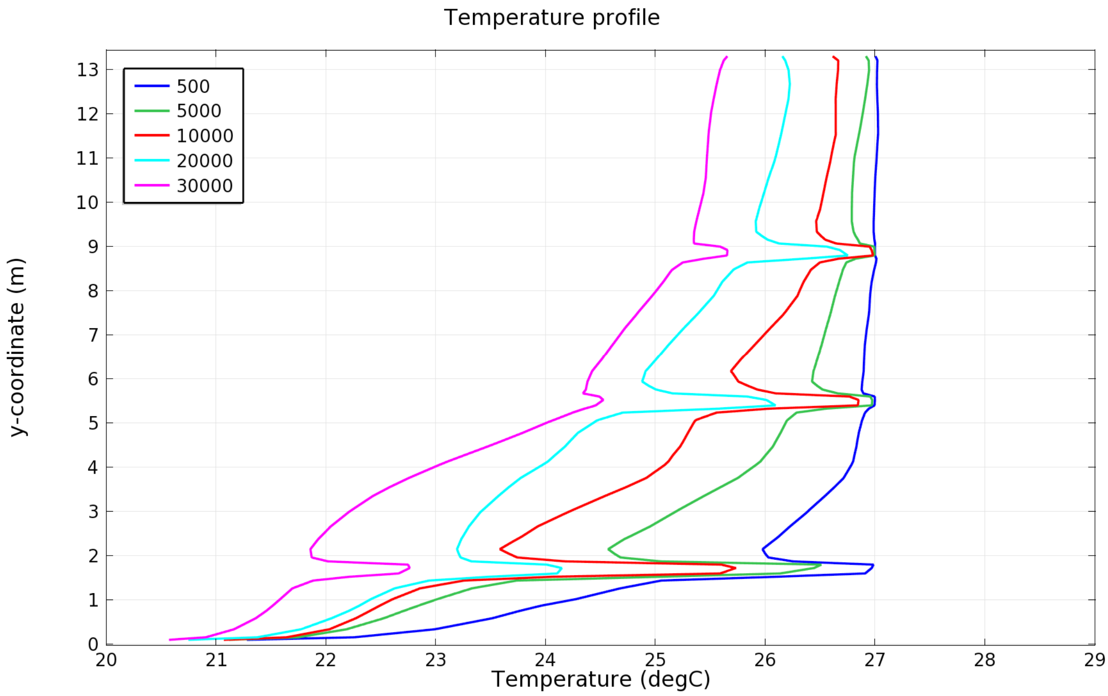


Figure 34 Temperature profile with floor cooling and ventilation

9.2.5 Study 4 - Effect of ceiling cooling

This study was conducted to see the effect of ceiling cooling, as opposed to floor cooling. The heat transfer coefficient becomes larger, increasing the cooling effect, thus making it interesting to compare with the radiant floor cooling system.

Table 11 Dependent variables in the simulations

Variable	Value	Comment
Initial temp	27°C	Standard initial temperature
Ventilation	Off	Supply air temperature 22°C
Cooling	Yes - Roof	Roof surface temp. 20°C, 32°C in one ”sunlit” floor zone
Obstacles	Yes	

The temperature field, see Figure 35, shows a rather even temperature. The velocity field shows rather high air speeds, this is expected due to the higher density of the cold air.

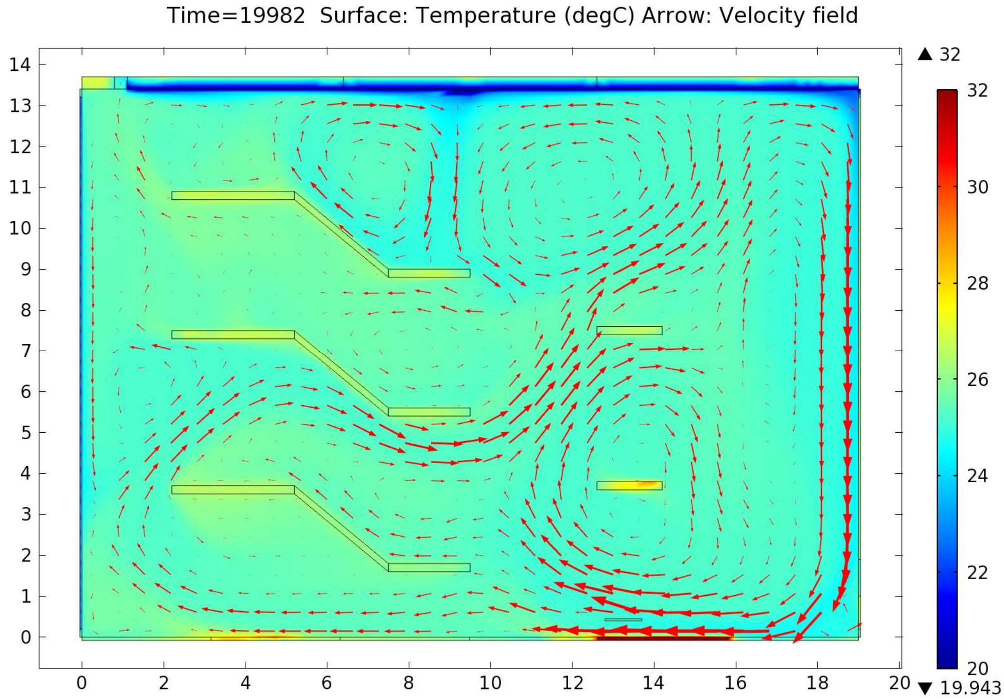


Figure 35 Temperature and velocity field with cooling in the ceiling

In Figure 36, which shows the temperature profile, an evenly distributed temperature is seen without the stratification phenomenon. Furthermore, the mean temperature in the room is lower compared to Study 1, which is expected as a result of a higher heat transfer coefficient.

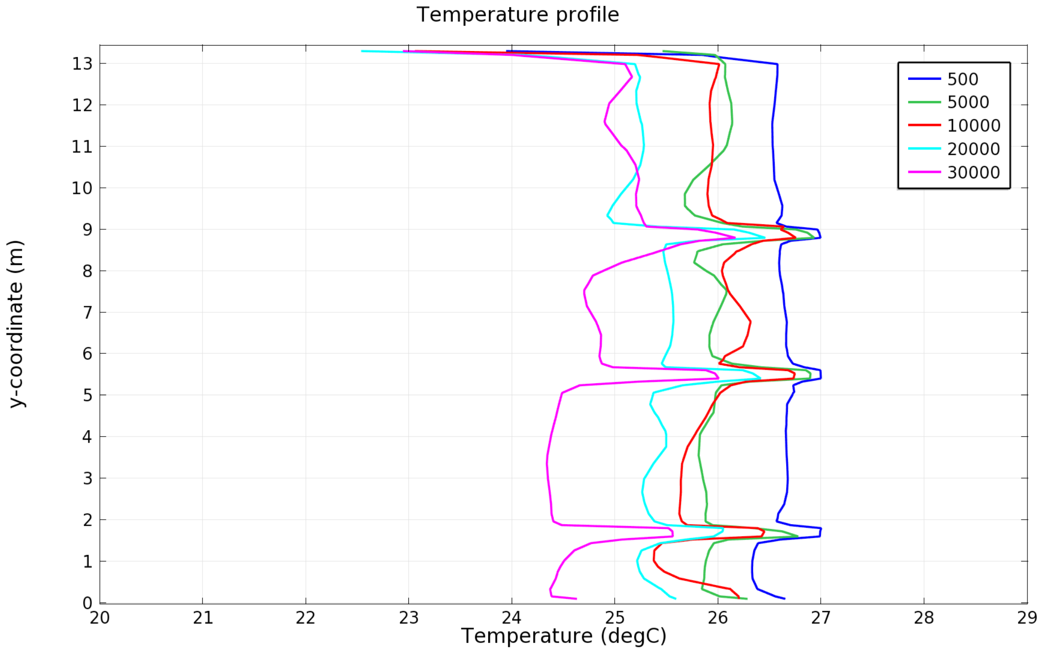


Figure 36 Temperature profile with cooling in the ceiling

The floor surface temperature, which is now rather high due to the thermal mass of the floor construction, will most likely decrease if the simulation continues for a longer period of time.

9.2.6 Study 5 - Effect of floor cooling with air supply in ceiling

This study was conducted as possible improvement to the current system, and the supply air device placement is changed from the floor level to the ceiling, resulting in a well-mixed air volume.

Table 12 Dependent variables in the simulations

Variable	Value	Comment
Initial temp	27°C	Standard initial temperature
Ventilation	On	Supply air temperature 22°C, placed in ceiling
Cooling	Yes	Floor surface temperature 20°C, 32°C in one “sunlit” zone
Obstacles	Yes	

Figure 37 shows that the air movement is increased, resulting in a well-mixed air volume and thereby a lower mean temperature in the room. The reason for this is mostly the new placement of the supply air device; however the sunlit floor zone with the higher surface temperature contributes to the mixing as well.

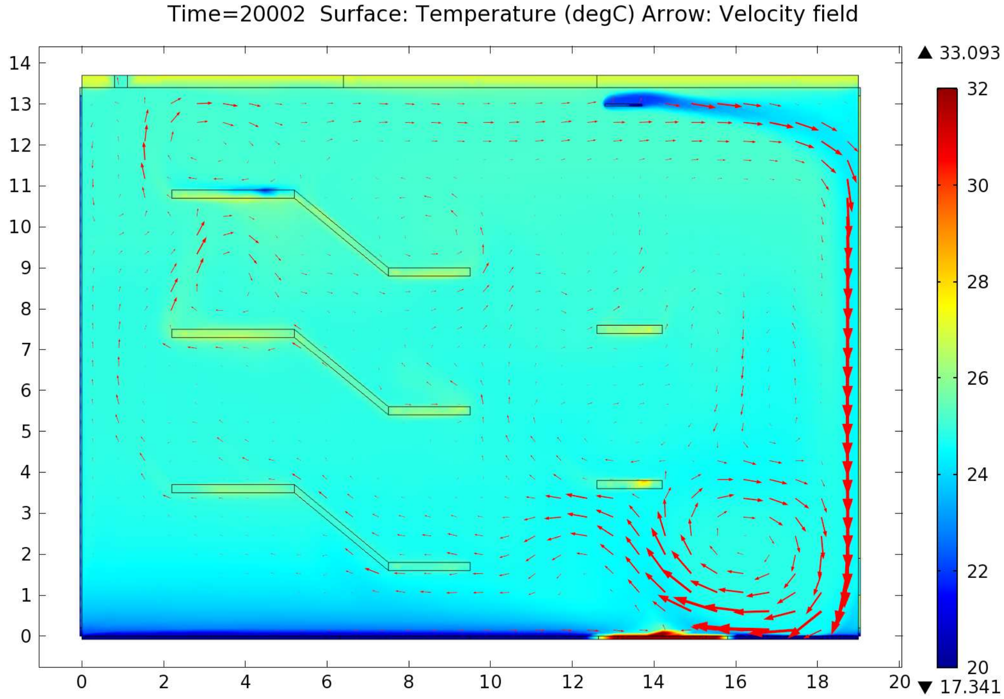


Figure 37 Temperature and velocity field with floor cooling and ventilation in ceiling

Figure 38, which shows the temperature profile, resembles Figure 36 somewhat. This is due to that the higher density of the colder air moves down and mixes with the rest of the air volume. A difference is the decreased temperature close to the floor, which is due to the low temperature prescribed at the floor surface.

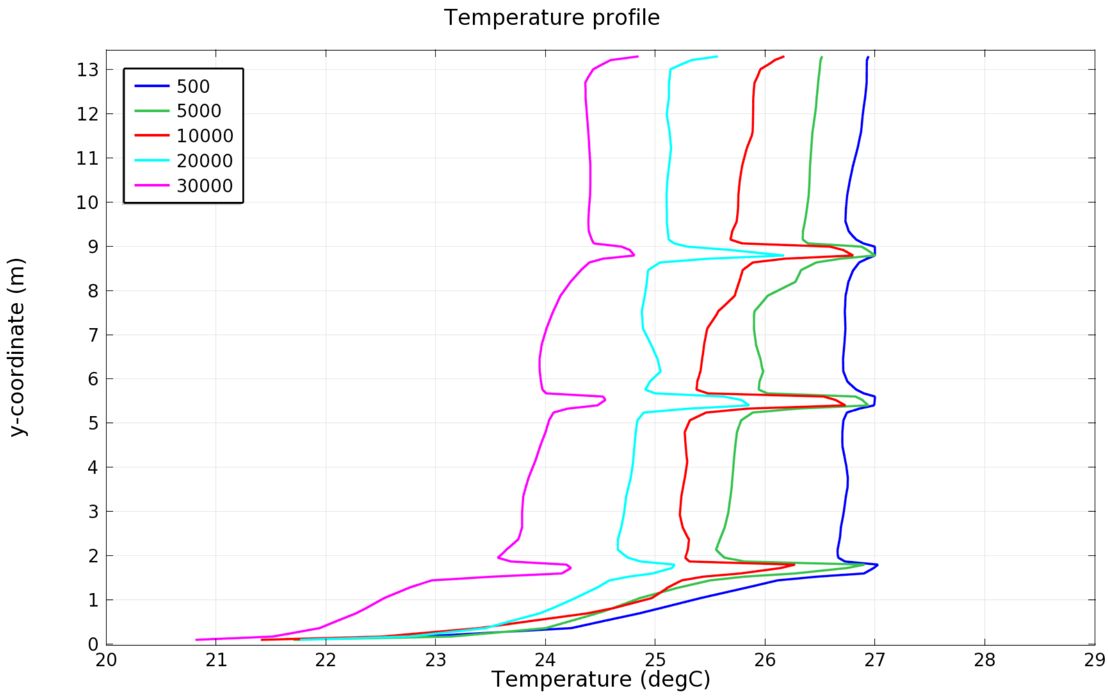


Figure 38 Temperature profile with floor cooling and ventilation in ceiling

9.3 Analysis of the simulations

The simulations have shown the behaviour of the air volume with different cooling systems. These were mainly conducted to see how the floor cooling system affected the thermal climate in a room. Study 2 (solely ventilation system) shows the importance of the ventilation system, which has a large impact on the temperature profile, especially at greater height. Through Study 1 (solely floor cooling) it can be concluded that the floor cooling affects the temperature at the first level but has lower impact on the higher levels. By combining Study 1 and 2, Study 3 (combined ventilation and floor cooling) is obtained, which corresponds the best to the real environment. Study 3 shows the lowest temperature at the first level; this is not surprising as the greatest cooling effect is installed in this study, giving the lowest mean temperature as well. Study 1 show that with solely the radiant floor cooling system a lower temperature close to the floor surface is obtained. To retrieve a well-mixed air volume the system requires help from the ventilation system. The opposite effect is seen in Study 4 (ceiling cooling), where the cold air moves downwards from the cold ceiling surface, resulting in a better mixed air-volume. In Study 5 (combined floor cooling and ventilation in ceiling) the effect of moving the supply air device from the first level to the ceiling was studied. Compared to remaining studies, the temperature is lowered in the entire atrium, except for in the first floor, where the lowest temperature is found in Study 3.

In Figure 39 the combined results from the simulations can be seen and compared. In Study 3 the best effect on the thermal climate at the first floor is seen, however the results in Study 5 indicate a better thermal climate at the upper floors.

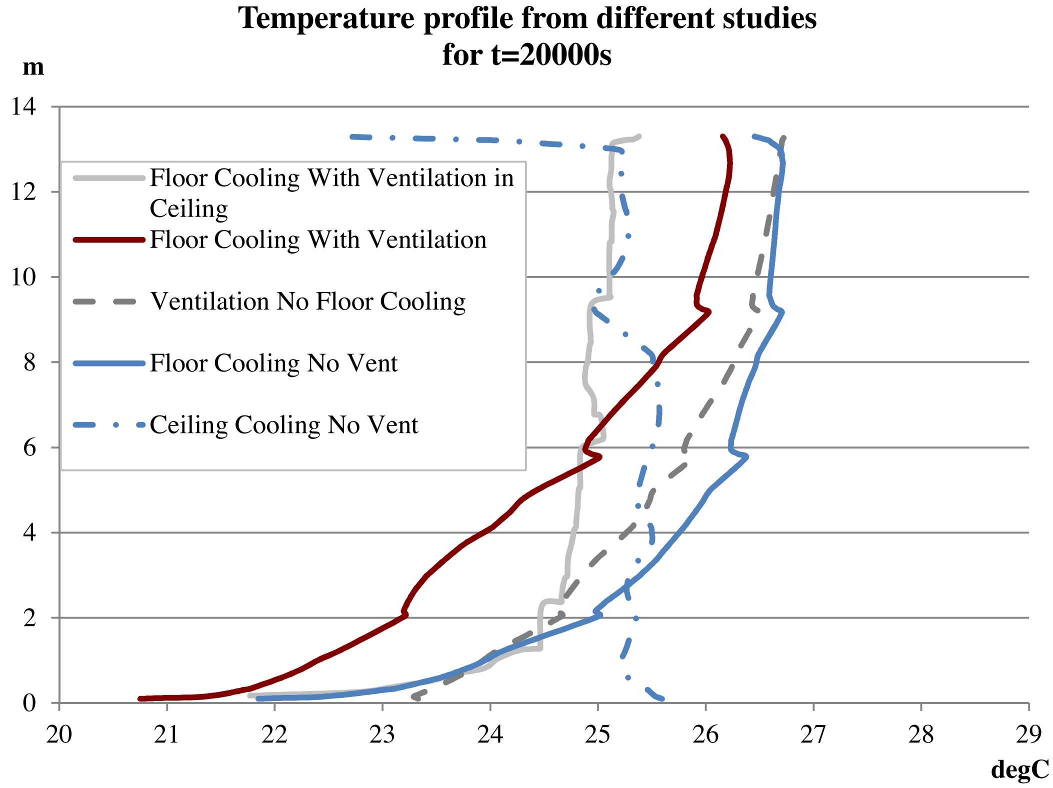


Figure 39 *Temperature profiles from the five studies, temperature values in solids cleaned*

In Figure 40 a comparison between the simulated and the measured values are seen. The simulated graph is taken from Study 3 (t=20000s) and the measured values are taken from the afternoon and evening. The graphs have the same appearance, which indicates a possibility to improve the model and re-use it for further simulations of air volume behaviour in similar spaces.

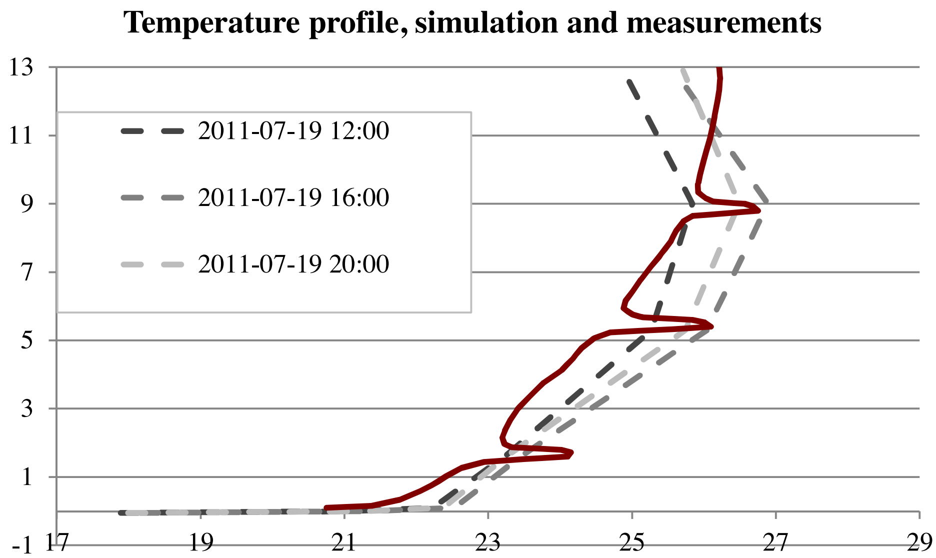


Figure 40 *Comparison between measured and simulated temperature profile*

10 Conclusions

The questions stated in the introduction will be answered in this chapter, starting with the first question:

- Does the radiant floor cooling system work in an adequate way and what are the delivered cooling effects of the system?

The existing floor cooling systems delivered cooling effect does not correspond to the cooling effect expected by method rule of thumb, as the method gave an approximate cooling effect of 25 W/m^2. The real value is somewhere in between 16.5 – 20 W/m^2, probably closer to the lower value. This is valid in this particular study object and is not applicable on other systems. The calculations have shown that it is possible to reach higher cooling effects by lowering the supply temperature.

Concerning the issue if the system works in an adequate way, the question must be related to how the thermal climate is perceived in the atrium. The measurement together with our personal opinion tells that the system has a major impact on the thermal climate at a low level in the atrium. The air temperature was about 22-23^{o}C in the occupied area at the first floor which is acceptable. At the higher levels in the atrium the temperature was not as comfortable, with temperatures at about 27^{o}C. Simulations from Study 3 (combined floor cooling and ventilation) support this as well. The simulations give that both the ventilation and floor cooling are important, though without the floor cooling the temperature at the first floor is a couple of degrees higher. Even if the radiant floor cooling system working in an acceptable way, there is room for system improvements. A very important aspect is the design and balancing of the system, which leads us to the second issue:

- Which factors will influence the delivered cooling effect? Are there any possibilities to improve an existing system?

The main factors that influence the delivered cooling effect are the water temperatures and water flows in the pipes. The design of the floor construction is also important, with focus on pipe distance and insulation amount below the pipes as well as screed and covering thickness and thermal conductivity. Generally, more insulation below the pipes is better and a pipe distance of below 160 mm is recommended to acquire greater cooling effects. An existing system is possible to improve by adjusting the water flow and lowering the supply temperature in the system. It is important to keep the dew point in mind, not letting the supply temperature fall below 1^{o}C above the dew point temperature. Another possible improvement can be to dehumidify the supply air and thereby lowering the dew point temperature. This would be a technical possible way to design a system but there can be other comfort related problems. Although it may not be possible to implement in the study object, the result from the Study 5 (floor cooling and ceiling ventilation), shows that this combination is worth studying further.

- Is their current method accurate enough? Is it comparable to other existing methods and reality?

Based on this analysis, the rule of thumb method is a simplified method that can be at the most used to obtain a key number of the highest possible cooling effect. The method does not give any information on how to design a radiant floor cooling system. Similarly, the EN method does not take into account water flows and how well insulated the floor is. The academic method on the other hand, accounts for the design of the floor construction by using a simple FDM tool. The water temperatures and water flows are calculated, as well. The academic method is the most comprehensive method, and can be used to adjust the system, thereby creating better prerequisites to reach as high designed cooling effect as possible.

- How is the thermal climate affected by the floor cooling system?

The measurement study shows a clear temperature difference at the first floor between when the floor cooling system is active and inactive. An issue though is the difficulty to differentiate the effect of the floor cooling system from external sources (e.g. outside temperature and solar radiation). Nevertheless as the measurements show a difference of 3°C, the effect of the radiant floor cooling cannot be disregarded. The simulations show that the effect does not solely depend on the floor cooling system; nevertheless without it the temperature would be higher.

With the above mentioned, it can be concluded that the floor cooling system affects the thermal climate positively, with a lowering of the air temperature, in the occupied zone where it counts the most. Although this is true, the air temperature at higher levels is not as affected by the floor cooling system. This is probably explained by a low installed cooling effect, which cannot counteract against the large solar heat gains and that the air volume is not well-mixed.

10.1 Recommendations

In this chapter, general and case specific recommendations regarding radiant floor cooling system design and use will be discussed.

10.1.1 General design recommendations

- **Activities in a room with floor cooling**

A general recommendation is to think about the purpose of the room. In some spaces, where people do not wear shoes and sit still, strict regulations on the floor surface temperature are needed. In other rooms where shoes are worn, lower demands can be set on the floor surface temperature. Consequently a low surface temperature (16-18°C) can be utilized, increasing the cooling effect.

- **Room proportions**

When using a surface to cool a space, there are some important geometry aspects to bear in mind. If the floor area is very small in comparison other surfaces, a radiant floor cooling system might not be the best option. For example: A small room with low height will have less floor area compared to the overall area of the walls and the ceiling. In this case the view angle of the floor will not be that different from the view angle of the ceiling. If a radiant cooling system is installed in the ceiling, the heat exchange coefficient will be greater as well, hence give a higher cooling effect. This is all good as long as the room height is not to great, as this will results in a low view angle, decreasing the positive impact on the thermal climate.

- **Decrease pipe distance to around 150 mm**

The temperature difference between supply and return water temperature is lower in a floor cooling system compared to a regular floor heating system; as a result a shorter pipe center to center distance should be considered (around 150mm).

- **Thermal conductance and thermal mass**

The thermal conductance should be regarded, when deciding materials for the floor construction. For example, floor surface materials preferably have a high thermal conductance, as well as a large thermal mass. Wooden floor constructions, with a low thermal mass, will response faster to internal gains, but will be less energy efficient due to a lower ability to store energy and transfer heat. Furthermore, a wooden construction will be more moisture sensitive, which is wise to take into account.

- **Careful detailing of floor construction**

If an ordinary system (see chapter 3.1) design is decided on, the floor construction should be well insulated from the remaining parts of the building; otherwise the system will partly act as a thermal active building system (TABS) and should be designed accordingly.

- **Humidity control**

The supply water temperature is preferably controlled by the dew point temperature in the room to avoid condensation on the floor. A supply temperature 2°C above than the dew point temperature is recommended. However, as different floor constructions have different thermal mass each system must be designed separately.

- **Delay in the control system**

It is important to have a delay programmed into the control system between the switching from heating to cooling mode and vice versa. The system is often regulated by the outside temperature and during a clear and cold summer night the temperature can get very low, switching the system into heating mode. This is an issue as after this clear night comes a clear day, which means high solar gains. Thus, the room is preferably cooled during the entire night.

- **Use a design method that calculates water temperatures and water flows**

It is recommended to not use a method that only calculates the delivered cooling effect, but provides information concerning the water temperatures and flows, as well. This additional information can be used to regulate the system in an adequate way.

10.1.2 Possible improvements in the study object

- **Delay in control system**

During the measurement study the system changed into heating system a number of times. This was due to that the system was designed to immediately shift into heating mode when the outside temperature dropped below 15°C. A system delay of at least 12 hours would be desirable.

- **Supply water temperatures**

According to the technical description of the control and monitoring equipment, the supply water temperature is designed to be 19°C. This is not the real case; instead the supply temperature varies with a lowest measured temperature about 17°C (average 19.5°C). From the dew point temperatures it was found that the mean dew point temperature was around 13.5°C with a maximum of 17.8°C. The lowest measured surface temperature was 19.4°C with a mean of about 21°C. Although the surface temperature should not be lower than the dew point temperature, there is a possibility to lower the design supply water temperature substantially. In fact, the measurements show that the supply water temperature can be decreased to about 16°C and, with the same humidity control as today, maintain a cooler climate, still keeping the condensation risk at an acceptable level.

- **General recommendations**

Supplementary measures against a low installed cooling effect can be e.g.; installing airing windows, increasing the supply air flow or lowering the supply air temperature. The foremost aspect to consider in spaces with great window area is solar shading; this is a design detail requiring attention prior to building construction.

11 Discussion

There is a clear application area for radiant floor cooling systems. What one must bear in mind is that the construction needs thorough thought and design. Another factor needing attention is the room application and prerequisites, which solely determines if it is viable or not to install a radiant floor cooling system. In the specific case of this thesis study object, it can be concluded that the floor cooling system is a good way of improving the thermal climate at the entrance floor. This is even though the system is most probably not balanced properly or design in a correct way (insulation thickness needs to be greater). So, is a radiant floor cooling system viable? Yes, we would say, but, as mentioned, the system design, the balancing and the room activities as well as dimensions are entirely decisive to if the system is viable or not.

In a large part of the world, there is a great need of cooling. With the dwindling oil reserves, and a lot of grand hotels in e.g. Dubai, the possibility to use a high temperature cooling technique with an associated bore hole, may be the future.

11.1 Retrospective learning

The thesis investigated the radiant floor cooling system with regards to different aspects and with different methods. As the authors of this thesis had a keen interest of understanding how the thermal climate was affected by the radiant floor cooling system, a measurement study was conducted. The study not only gave a broadened understanding of how a measurement study should be conducted, but an insight in the importance of having evaluated the placement of the measuring devices thoroughly, as well. A revelation is that one can never measure in too many places.

In the thermal climate simulations in COMSOL Multiphysics, some difficulties were encountered, partly due to that the complexity of them was underestimated. In-depth knowledge of the simulation tool and its CFD analysis was lacking, mostly due to the time restriction of the thesis. A basic physic network analysis model, including all heat transfer mechanisms is always a good start before initiating CFD simulations, as this can give a better understanding of the simulations.

11.2 Further research

The floor cooling system studied in this thesis is a type A system. A lot of the discussions about floor cooling system today are about TABS, which has not been studied in this thesis, instead the focus has been on a more simple system that is easier to control and gives the option to increase the use of already installed floor heating systems. A further study recommendation would be, as a result of the above, to look at the possibilities of TABS systems. During the thesis work it was found that there is an interest to use this technique in residential buildings. This is something that should be carefully thought about before doing, as there is seldom a need for an active cooling system in a residential building. Problems with over temperatures can often be solved by passive techniques such as solar shading instead.

An interesting prospect is the possibility to utilize the floor cooling system together with a geothermal heating system and solar collectors. With this the floor cooling system could be used to load the bore hole with energy during the summer which would increase the temperature in the bore hole and thereby increase the efficiency of the heat pump during the winter. The solar collectors are needed for the production of hot tap water during the summer. This would be a more complex system, which might not be suitable for common house owners.

12 References

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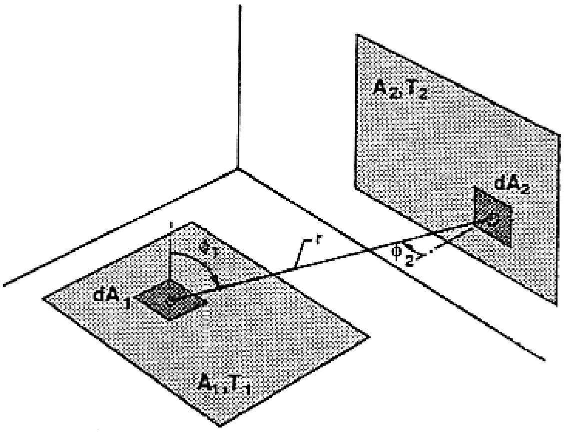
13 Appendices

Appendix A – Heat Transfer, View factor and Azimuth

The view factor is defined as follows:



$$F_{12} = \frac{1}{A_1} \int_{A_1} \int_{A_2} \frac{\cos\varphi_1 \cos\varphi_2}{\pi r^2} dA_1 dA_2 \quad \text{(A. 1)}$$

Figure A. 1 Geometric parameters for calculating the view factor [21]

The magnitude of the sun radiation will be affected by a number of different factors, including altitude of the sun and azimuth (see Figure A. 2). Furthermore, the amount of incoming radiation is dependent on the direction of the building.

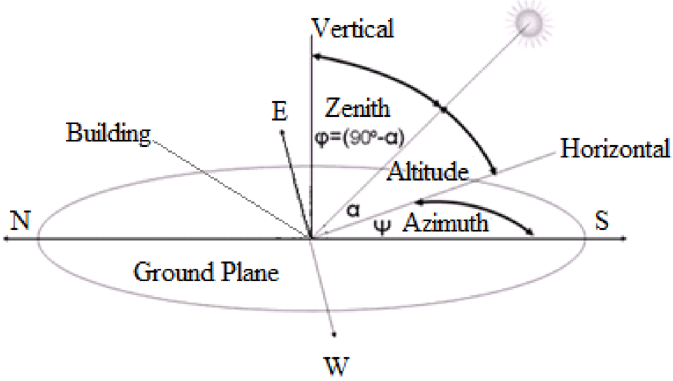


Figure A. 2 Azimuth and altitude for Northern latitudes

Appendix B – Tables cooling capacity

Table A. 1 Floor covering factor depending on thermal resistance $R_{\lambda,B}$ and thermal conductivity of the screed λ_E

$R_{\lambda,B}$ (m^2K/W)	**0**	**0.05**	**0.10**	**0.15**
λ_E (W/mK)	**a_B**			
2.0	1.196	0.833	0.640	0.519
1.5	1.122	0.797	0.618	0.505
1.2	1.058	0.764	0.598	0.491
1.0	1.000	0.734	0.579	0.478
0.8	0.924	0.692	0.553	0.460
0.6	0.821	0.632	0.514	0.433

COMMENT Floor covering factor a_B can be found from:

$$a_B = \frac{\frac{1}{\alpha} + \frac{S_{U,0}}{\lambda_{U,0}}}{\frac{1}{\alpha} + \frac{S_{U,0}}{\lambda_E} + R_{\lambda,B}}$$

where $\alpha = 10.8\ W/m^2K$, $\lambda_{U,0} = 1\ W/mK$, $S_{u,0} = 0.045\ m$

Table A. 2 Spacing factor a_T

$R_{\lambda,B}$ (m^2K/W)	**0**	**0.05**	**0.10**	**0.15**
a_T	1.23	1.188	1.156	1.134

Table A. 3 Coating factor a_U dependent on pipe spacing cc and the floor coverings thermal conductivity $R_{\lambda,B}$

$R_{\lambda,B}$	0	0.05	0.10	0.15
cc	a_U			
0.050	1.069	1.056	1.043	1.037
0.075	1.066	1.053	1.041	1.035
0.100	1.063	1.050	1.039	1.0335
0.150	1.057	1.046	1.035	1.0305
0.200	1.051	1.041	1.0315	1.0275
0.225	1.048	1.038	1.0295	1.026
0.300	1.0395	1.031	1.024	1.021
0.375	1.030	1.024	1.018	1.016

Table A. 4 Factor for outer diameter a_D dependent on pipe spacing cc and the floor coverings thermal conductivity $R_{\lambda,B}$

$R_{\lambda,B}$	0	0.05	0.10	0.15
cc	A_D			
0.050	1.013	1.013	1.012	1.011
0.075	1.021	1.019	1.016	1.014
0.100	1.029	1.025	1.022	1.018
0.150	1.040	1.034	1.029	1.024
0.200	1.046	1.040	1.035	1.030
0.225	1.049	1.043	1.038	1.033
0.300	1.053	1.049	1.044	1.039
0.375	1.056	1.051	1.046	1.042

Appendix C – Specification of measuring devices

To measure instantaneous surface temperatures a Testo 830-T2 infrared thermometer was used. For the long-time measurements three different models from Tinytag were used.

Table A. 5 Properties of handheld IR thermometer [22]

Technical data - Testo 830-T2	
Measuring range	-30 ... +400°C
Accurateness IR	±1.5°C or 1,5% of measured value (0 ... +400°C) ±2°C or 2% of measured value (-30 ... 0°C)
Resolution IR	0.5°C
Measuring optics IR	12:1

Table A. 6 Properties of thermometers [23]

Technical data Tinytag Transit 2	
Measuring range	-40 ... +70°C
Accurateness	±0.4 °C at 20°C
Resolution	0.01°C at 20°C
Technical data Tinytag Talk 2	
Measuring range	-40 ... +85°C
Accurateness	±0.5 °C at 20°C
Resolution	0.05°C at 20°C
Other	External probe to measure surface temperatures
Technical data Tinytag Ultra 2 Temp & RH	
Measuring range	-25 ... +85°C 0%... 85% RH
Accurateness	±0.45 °C at 25°C ±3% RH at 25°C
Resolution	0.01°C 0.3 % RH
Other	Combined RH and temperature

Appendix D – Dew point temperature calculations

The dew point temperature is calculated according to equation A.2.

$$T_{dew} = \frac{b*\alpha(T,RH)}{a-\alpha(T,RH)} \qquad (°C) \qquad (A.2)$$

where

$$\alpha(T,RH) = \frac{a*T}{b+T} + \ln(RH)$$
$$a = 17.2$$
$$b = 237.7$$

The formula is based on the August–Roche–Magnus approximation and it is valid for temperatures between 0-60°C, a relative humidity between 1-100% and a dew point temperature between 0-50°C

In the MathCad dew point temperature chapter (which follows) the uncertainty is calculated to be 0.7-1.2 °C for the dew point temperature, with regard to the specified accurateness of the thermometers and relative humidity sensors. The uncertainty has been calculated in two different ways. The lower value of 0.7 °C is based on equation A.3 where σT_{dew} is the uncertainty of the dew point temperature. The data for these calculations are taken from the specification of the measuring devices.

$$\sigma T_{dew}{}^2 = \sigma T^2 \left(\frac{\partial T_{dew}}{\partial T}\right)^2 + \sigma RH^2 \left(\frac{\partial T_{dew}}{\partial RH}\right)^2 \qquad (°C) \qquad (A.3)$$

after derivations the uncertainty becomes

$$\sigma T_{dew} = \sqrt{\sigma T^2 \left(\frac{a*b}{a*b-(b+T)*\ln(RH)}\right)^4 + \sigma RH^2 \left(\frac{a*b*(b+T)^2}{RH*(a*b-(b+T)*\ln(RH))^2}\right)^2} \qquad (°C)$$

The larger value of 1.2 °C is calculated by equation A.4 stating that the maximum error is the sum of the relative error of each variable.

$$\sigma T_{dew} = T_{dew} * \left(\frac{\sigma T}{T} + \frac{\sigma RH}{RH}\right) \qquad (°C) \qquad (A.4)$$

These two different ways of calculating gives that an uncertainty of about 1°C can be presumed. This should be considered in the chapters which include the dew point temperature.

Dew point temperature calculations - MathCad

Dew point temperature

The dew point temperature is calculated according to the following formula.
The uncertainty in the calculated dew point temperature is 0.4 C.
Found at: http://www.paroscientific.com/dewpoint.htm

$$T_d = \frac{b\,\gamma(T,RH)}{a - \gamma(T,RH)}$$

$$\gamma(T,RH) = \frac{a\,T}{b+T} + \ln(RH/100)$$

a = 17,27
b = 237,7

Calcultaion of the uncertainty of the dew point temperature

Input

$a := 17.27$

$b := 237.7$

$T := 25$ Measured temperature (0-60)

$RH := 0.85$ Measured RH (0-1)

$\sigma_T := 0.45$ Specified error temperature

$\sigma_{RH} := 0.03$ Specified error RH

Calculations

$$\sigma_{Td} := \sqrt{\sigma_T^{\,2}\cdot\left[\frac{a\cdot b}{a\cdot b-(b+T)\cdot\ln(RH)}\right]^4 + \sigma_{RH}^{\,2}\left[\frac{a\cdot b\cdot(b+T)^2}{RH\cdot[a\cdot b-(b+T)\ln(RH)]^2}\right]^2}$$

$\sigma_{Td} = 0.729$ Uncertainty of the calculated dew point temperature.

Dew point temperature

$$\alpha := \frac{a\cdot T}{b+T} + \ln(RH) = 1.481$$

$$T_d := \frac{b\cdot\alpha}{a-\alpha} = 22.296$$

Alternative calculation of uncertainty

$$\Delta T_d := T_d\cdot\left(\frac{\sigma_T}{T} + \frac{\sigma_{RH}}{RH}\right) = 1.188$$

Appendix E – Preparatory measurements

The preparatory measurements were made on a clear day in late June, with an outside air temperature of around 23°C.

Plan – positioning of the measuring devices

To get an overview of the temperature profile, as well as surface temperatures in different parts of the atrium, the room was divided into cells. The surface temperature was measured a number of times in the different cells (see Figure A. 3 for cell division and Figure A. 4 respectively Figure A. 5 for placement).

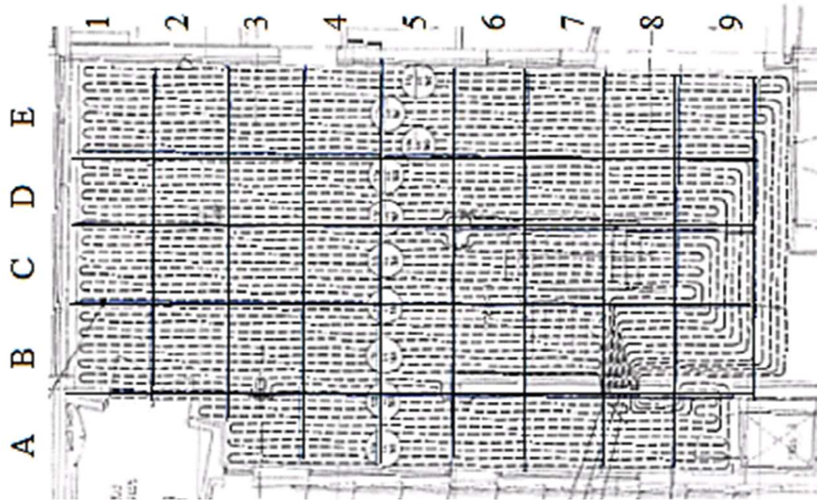


Figure A. 3 The entrance floor was divided into cells, giving an overview of the location of the measurements

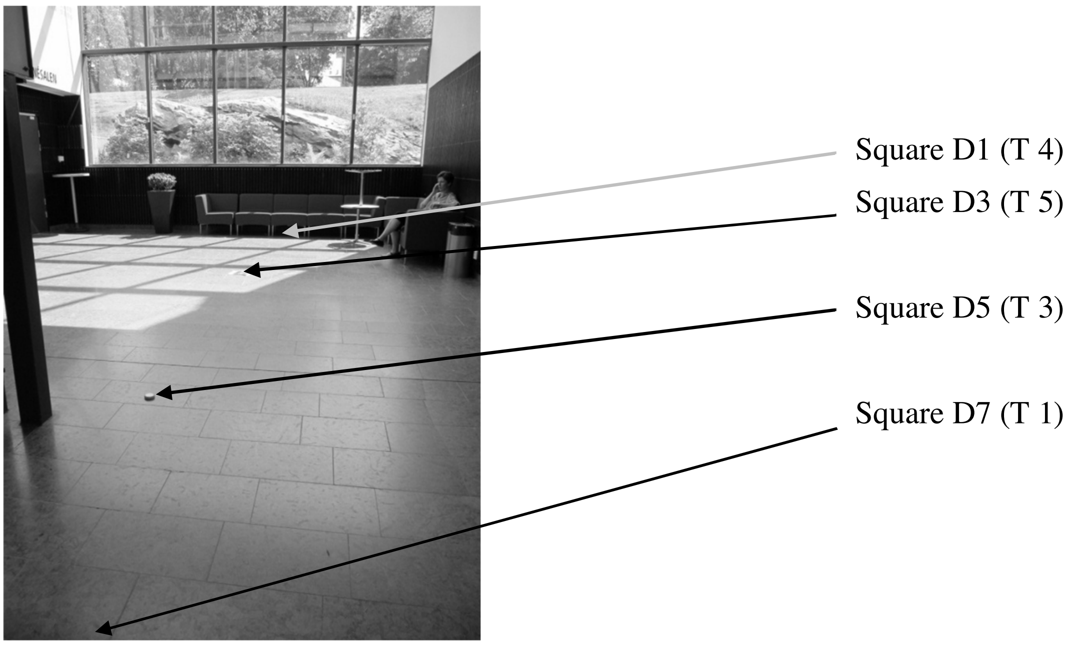


Figure A. 4 Placement of measuring devices on floor

Square D9 (T 2)

Figure A. 5 Placement of measuring devices on floor

Furthermore the air temperature and relative humidity was measured at different heights in the atrium, see Table A. 7 and Figure A. 6.

Table A. 7 Measuring devices and their placements

Measuring device ID	**Placement**	**Additional information**
RH 1	1.6 m above floor	RH
RH 3	Outside in shade	RH, Air temperature
RH 4	Exhaust air, 12.9 m above floor	RH, Air temperature
RH 5	Supply air	RH, Air temperature
T 1	Square D7	Surface temperature
T 2	Square D9	Surface temperature
T 3	Square D5	Surface temperature
T 4	Beneath sofas in square D1	Surface temperature
T 5	Square D3	Surface temperature, sunlit
T F	5.3 m above floor	Air temperature
T G	Beneath stairs in square B8	Surface temperature
T H	Supply water	Temperature
T I	Return water	Temperature
T J	Beneath stairs in square C8	Surface temperature

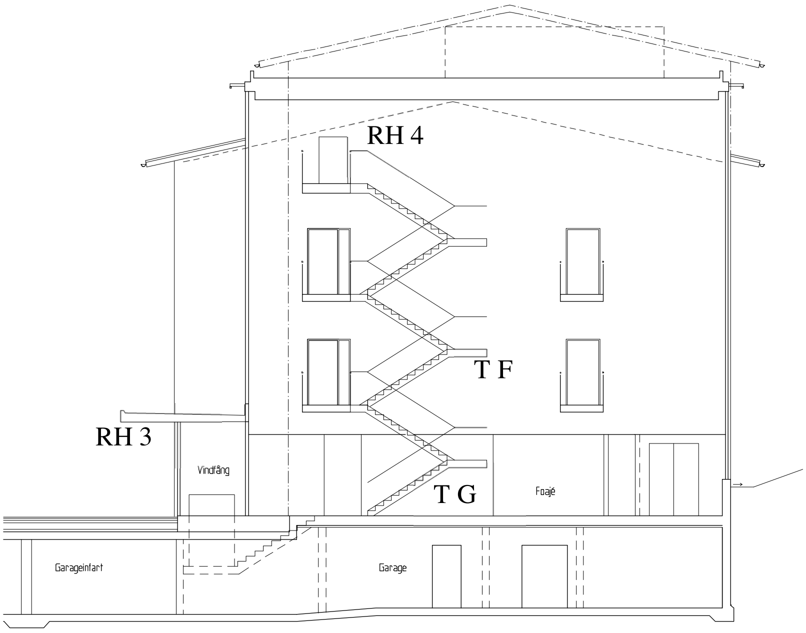
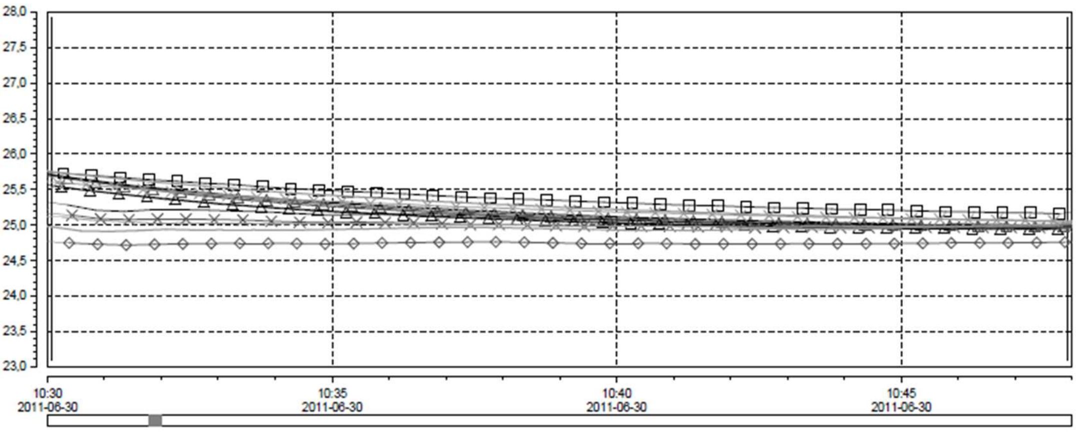


Figure A. 6 Positioning of the thermometers measuring the vertical temperature profile

Control of the correctness of the measuring devices

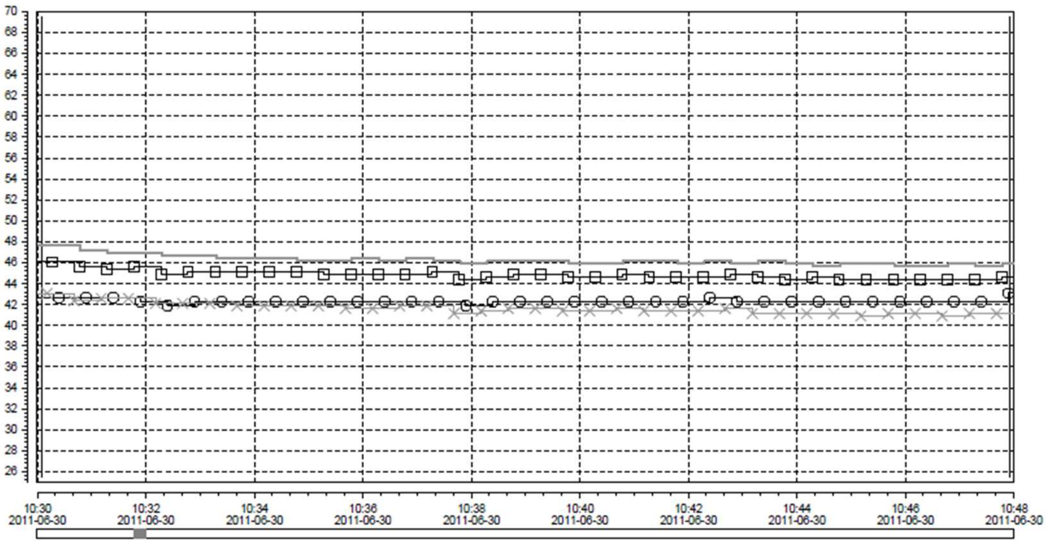
The measurement showed that when regarding the temperature, two of the thermometers differed from the mean value with around 0.2°C, see Figure A. 7. This is inside the specified fault of the devices.



	Färg	Enhet	Med	Min	Max	Linje-bredd	Benämning	Punkttyp	
012		°C	25,38	25,16	25,73		RH 3 - Outside air temperature	□	
016		°C	25,29	25,05	25,73		RH 5 - Supply air temperature		
014		°C	25,25	25,06	25,60		RH 4 - Exhaust air temperature, 12.9 m above floor	×	
003		°C	25,20	24,99	25,67		T 3 - Floor surface temperature in cell D5		
001		°C	25,18	24,98	25,65		T 1 - Floor surface temperature in cell D7	△	
002		°C	25,16	24,96	25,66		T 2 - Floor surface temperature in cell D9		
005		°C	25,11	24,93	25,55		T 5 - Floor surface temperature in cell D3		
004		°C	25,11	24,94	25,53		T 4 - Floor surface temperature in cell D1	△	
006		°C	25,10	25,00	25,28		T F - Air temperature, 5.3 m above floor		
007		°C	25,01	24,95	25,14		T G - Floor surface temperature in cell B8	×	
008		°C	25,00	24,97	25,11		T H - Water temperature, supply		
009		°C	24,94	24,91	25,00		T I - Water temperature, return		
010		°C	24,74	24,72	24,76		T J - Floor surface temperature in cell C8	◇	

Figure A. 7 Control of the thermometers

Concerning the relative humidity there was also a deviation from the mean value, see Figure A. 8.



	Färg	Enhet	Med	Min	Max	Linje-bredd	Benämning	Punkttyp	
017		%RH	46,21	45,66	47,68		RH 5 - Relative humidity of supply air		
013		%RH	44,83	44,40	46,06		RH 3 - Relative humidity of outside air	□	
011		%RH	42,32	41,88	43,08		RH 1 - Relative humidity of air, 1.6 m above floor	O	
015		%RH	41,62	40,90	43,07		RH 4 - Relative humidity of exhaust air, 12.9 m above floor	×	

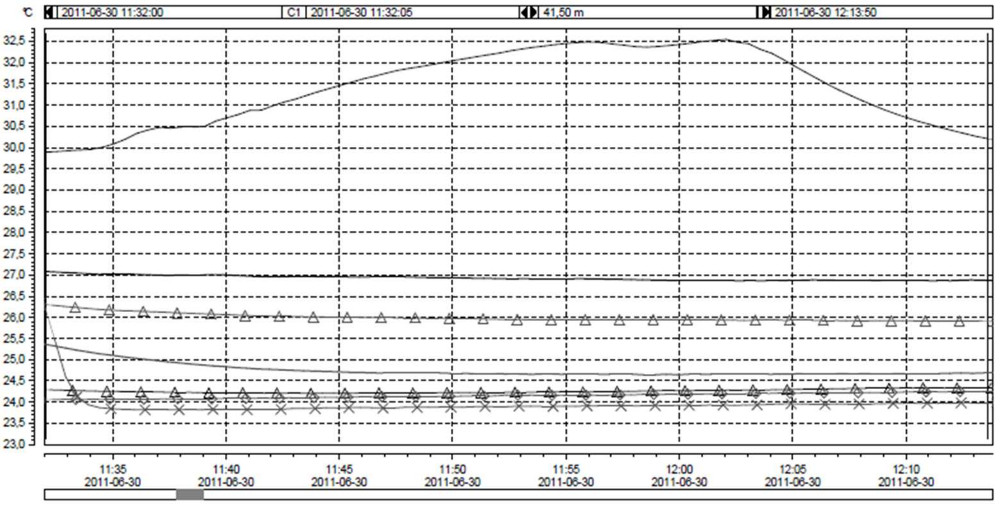
Figure A. 8 Control of the relative humidity sensors

Results

In the following chapters, results from the preparatory measurements on the study object will be presented. Floor surface temperatures and air temperatures will be accounted for, followed by a short discussion. The cooling system was turned off during this measuring period due to a defective three way valve, the air temperatures and the floor surface temperatures will hence be valid for a non-functioning system.

Surface temperatures

The surface temperature will be affected by a number of factors; however, one of the most important factors is the sun radiation which can cause a temperature rise up to 32°C over a short period of time. This phenomenon can be seen in Figure A. 9 where the device in square D3 is placed in a sunlit area and shows a maximum temperature of over 32°C, whereas remaining devices show a lower temperature. The lowest temperatures are found below the stairs and under the sofas in square D1, C8 and B8. The floor surface temperature in shaded areas is from around 24°C in some cells, up to 27°C in other. These can be divided in two categories, low and mid temperatures. The temperatures in squares D1, D9, C8 and B8 are included in the first category and the temperature varies from 24°C to 24.5°C. Included in the second category are the temperatures in square D5 and D7, these are between 26°C and 27°C. The temperature in the sunlit area is included in a category named high temperatures. These categories are also accounted for in the comment chapter, where placement of the devices for the long-time measurement period is discussed.



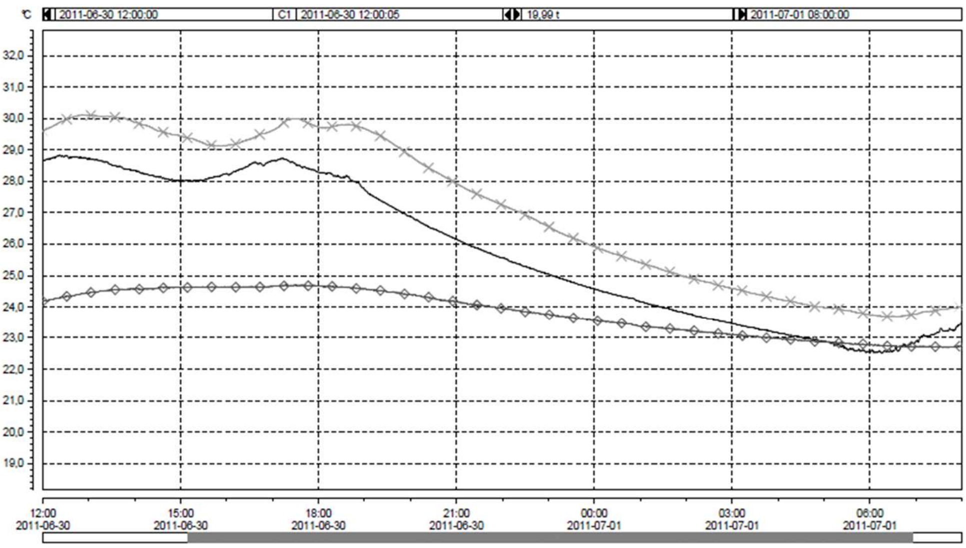
	Färg	Enhet	Med	Min	Max	Linje-bredd	Benämning	Punkttyp	
005		°C	31,41	29,89	32,54		T 5 - Floor surface temperature in cell D3		
003		°C	26,93	26,86	27,07		T 3 - Floor surface temperature in cell D5		
001		°C	26,00	25,91	26,29		T 1 - Floor surface temperature in cell D7	△	
002		°C	24,74	24,64	25,32		T 2 - Floor surface temperature in cell D9		
004		°C	24,25	24,20	24,33		T 4 - Floor surface temperature in cell D1	△	
010		°C	24,14	24,05	24,24		T J - Floor surface temperature in cell C8	◇	
007		°C	23,92	23,81	25,51		T G - Floor surface temperature in cell B8	×	

Figure A. 9 Surface temperatures

To ensure the correctness of the measuring devices, squares D1, D3, D5 and D7 were also measured with an IR-camera and an infrared thermometer.

Air temperature

The results from the measurements are valid for a non-active system, due to the default three-way valve described earlier. A maximum air temperature of 28.5°C at the second floor level was measured during the day and at the exhaust a temperature of 30°C was reached, see Figure A. 10.



	Färg	Enhet	Med	Min	Max	Linje-bredd	Benämning	Punkttyp	
014		°C	27,09	23,69	30,12		RH 4 - Exhaust air temperature, 12.9 m above floor	×	
006		°C	25,76	22,51	28,83		T F - Air temperature, 5.3 m above floor		
010		°C	23,83	22,69	24,69		T J - Floor surface temperature in cell C8	◇	

Figure A. 10 Temperature gradient and outside temperature for the preparatory measurements

Figure A. 11 shows the temperature profile from 110630 12:00 to 110701 08:00. The temperature outside was rather high, and so are the temperatures inside the atrium at greater height. Furthermore the floor surface temperatures are also quite high.

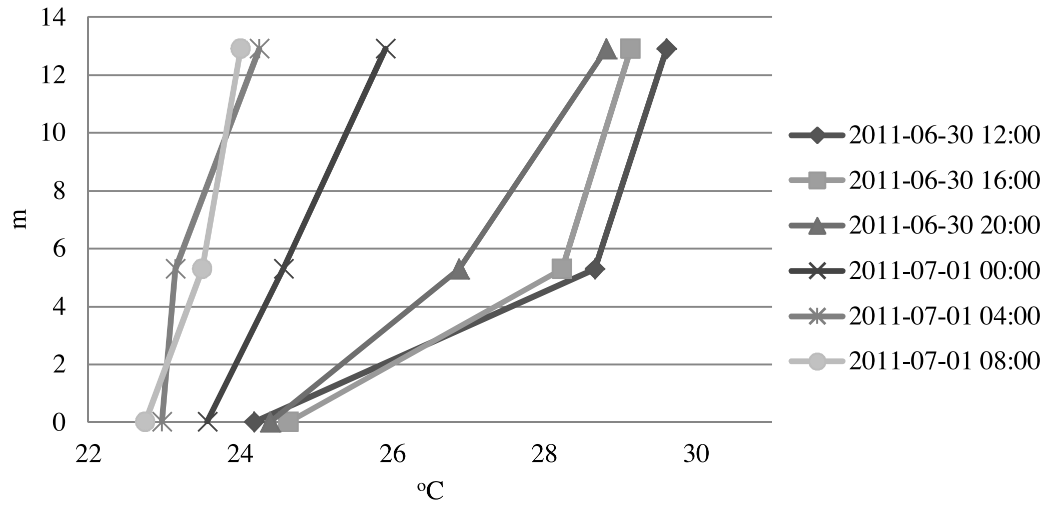


Figure A. 11 Temperature profile 110630-110701

Comments

One finding from the preparatory measurements was that the floor cooling system was out of function due to a defective three-way valve. The system fault resulted in high floor surface temperatures.

The main purpose of the preparatory measurements was to find adequate placements for the devices in the further measurements. The surface temperatures are, as mentioned before, divided in the three categories, low, mid and high temperatures. For the long-time measurements a measuring device was placed in each category. The placement is shown in Figure A. 12.

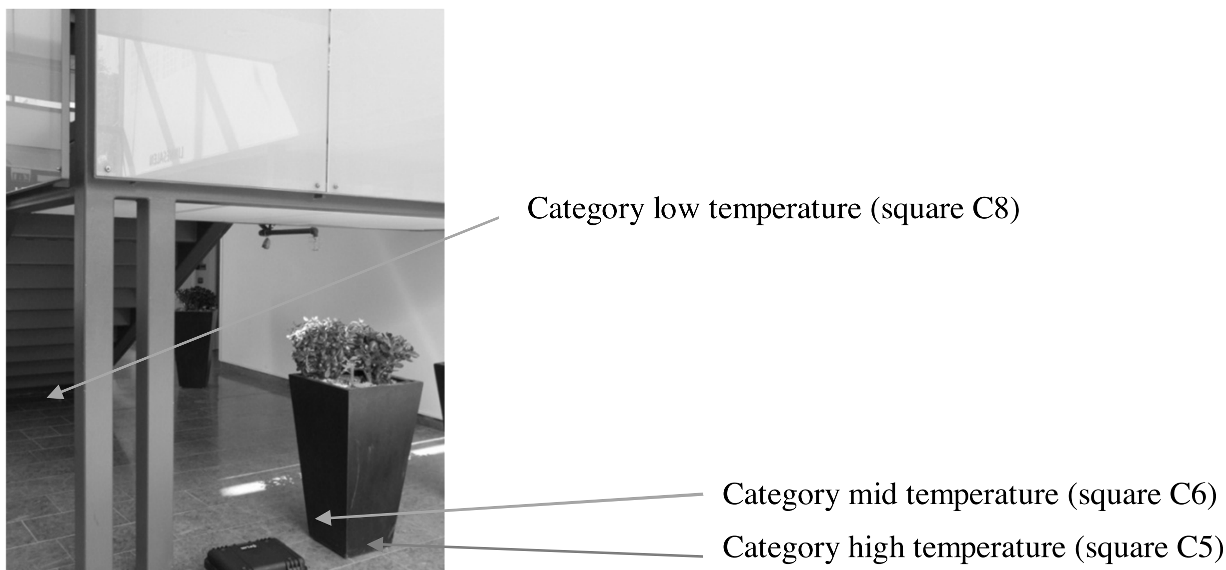
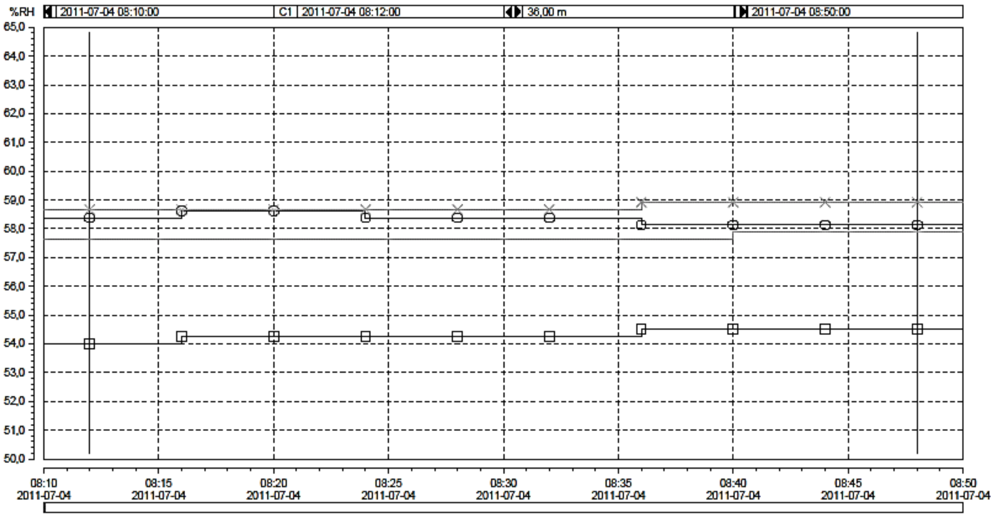


Figure A. 12 Placement of measuring devices

Appendix F – Long-time measurements

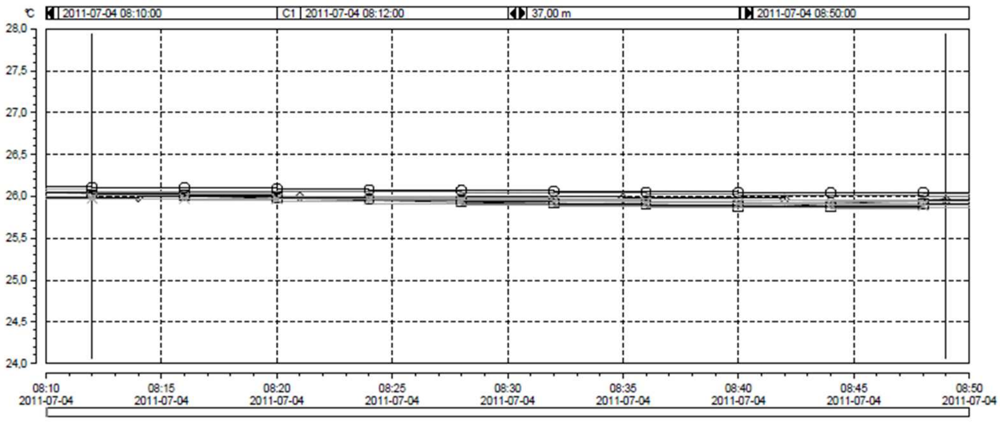
Accuracy control of the measuring devices

The maximum measured temperature difference of the devices was around 0.2°C, which is acceptable. The relative humidity differed more, with device RH 4 at 4 %, which is the same value as the control for the preparatory measurements. Therefore all values from device RH 4 will be corrected by adding 4 % to the existing data. The temperature accuracy can be seen in Figure A. 14, for control of the devices regarding the relative humidity, see Figure A. 13.



	Färg	Enhet	Med	Min	Max	Linje-bredd	Benämning	Punkttyp	
016		%RH	58,77	58,66	58,92		RH 5 - Relative humidity of exhaust air, 12.9 m above floor	×	
012		%RH	58,33	58,13	58,62		RH 3 - Relative humidity of air, 1.6 m above floor	○	
010		%RH	57,71	57,64	57,89		RH 2 - Relative humidity of supply air		
014		%RH	54,33	54,00	54,50		RH 4 - Outside air relative humidity	□	

Figure A. 13 Control of the relative humidity

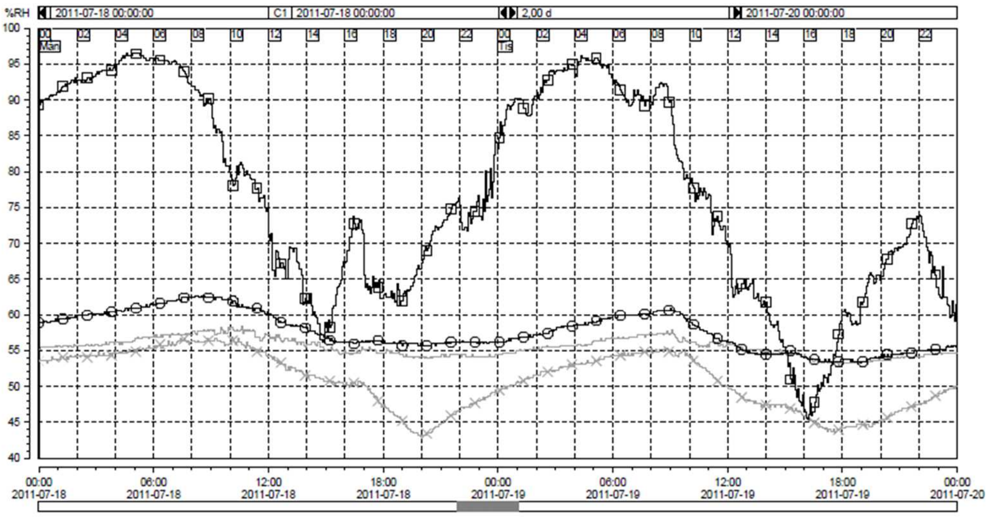


	Färg	Enhet	Med	Min	Max	Linje-bredd	Benämning	Punkttyp	
011		°C	26,07	26,04	26,11		RH 3 - Air temperature, 1.6 m above floor	○	
004		°C	26,04	26,00	26,06		T E - Floor surface temperature in cell C5		
002		°C	25,98	25,95	25,99		T 8 - Air temperature, 0.1 m above floor	◇	
008		°C	25,98	25,94	26,01		T J - Floor surface temperature in cell C8	◇	
009		°C	25,98	25,93	26,05		RH 2 - Supply air temperature		
001		°C	25,97	25,92	26,03		T 7 - Air temperature, 9 m above floor		
006		°C	25,97	25,92	26,06		T G - Water temperature, supply		
003		°C	25,94	25,88	26,02		T 9 - Air temperature, 5.3 m above floor		
015		°C	25,94	25,89	25,98		RH 5 - Exhaust air temperature, 12.9 m above floor	×	
013		°C	25,94	25,87	26,02		RH 4 - Outside air temperature	□	
005		°C	25,93	25,86	26,04		T F - Floor surface temperature in cell C6		
007		°C	25,90	25,84	25,96		T I - Water temperature, return		

Figure A. 14 Control of the temperature

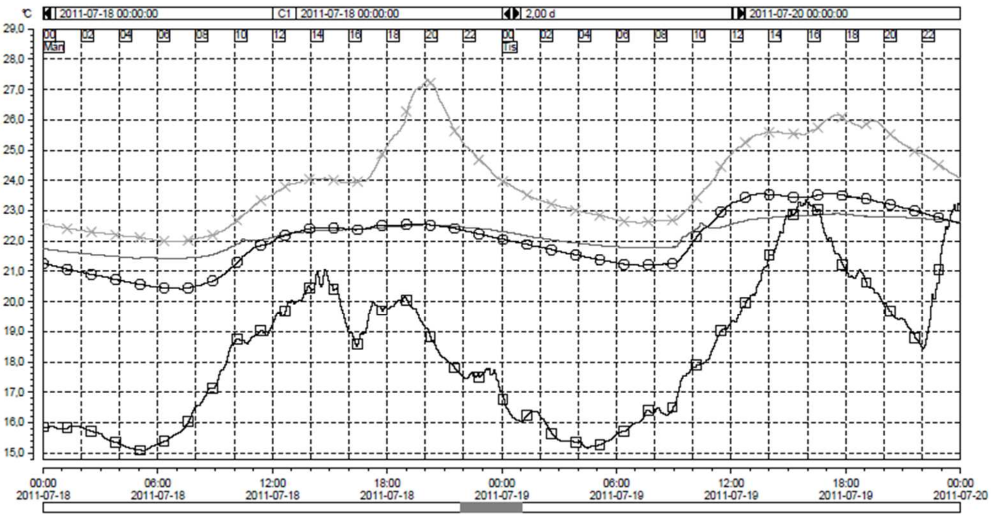
Relative humidity

Due to that the relative humidity of the air is closely linked to the temperature, the temperature will also be presented. The positioning of the different RH measuring devices is outside, at the supply air device and at the exhaust air device. Furthermore the RH device placed at height 1.6 m will be accounted for.



	Färg	Enhet	Med	Min	Max	Linje-bredd	Benämning	Punkttyp	
017		%RH	76,86	45,38	96,45		RHout	□	
012		%RH	57,65	53,20	62,69		RH 3 - Relative humidity of air, 1.6 m above floor	O	
010		%RH	55,45	52,91	58,39		RH 2 - Relative humidity of supply air		
016		%RH	50,81	42,94	56,87		RH 5 - Relative humidity of exhaust air, 12.9 m above floor	×	

Figure A. 15 Exhaust, supply, at height 1.6 m and outside relative humidity



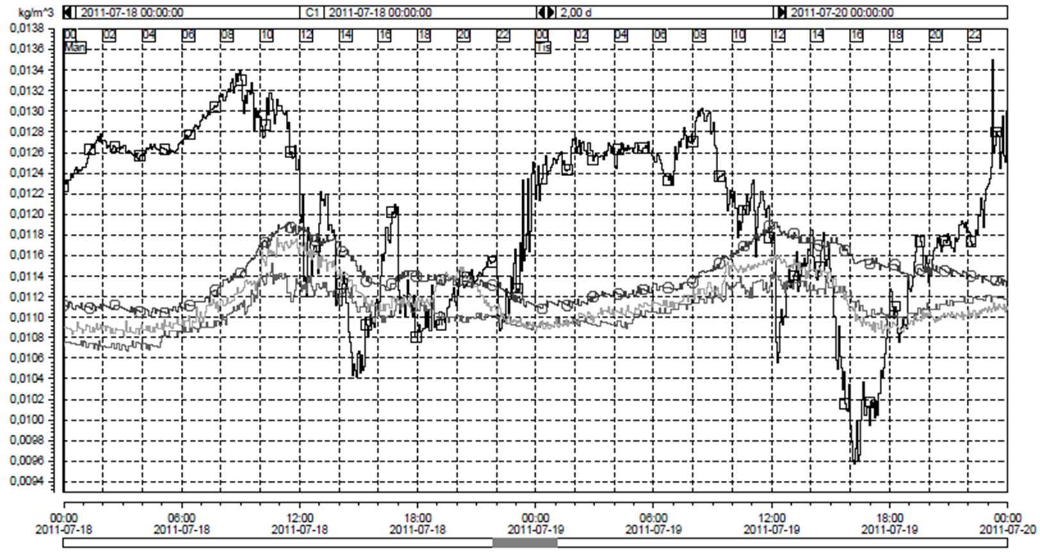
	Färg	Enhet	Med	Min	Max	Linje-bredd	Benämning	Punkttyp	
015		°C	23,96	21,98	27,31		RH 5 - Exhaust air temperature, 12.9 m above floor	×	
009		°C	22,21	21,42	22,90		RH 2 - Supply air temperature		
011		°C	22,07	20,43	23,57		RH 3 - Air temperature, 1.6 m above floor	O	
013		°C	18,26	15,08	23,37		RH 4 - Outside air temperature	□	

Figure A. 16 Exhaust, supply, at height 1.6 m and outside air temperature

Figure A. 15 and Figure A. 16 shows the close correlation between relative humidity and temperature, as T increase, the RH will decrease. The outside relative humidity varies the most, which is due to the great temperature variation between day and night. Furthermore the relative humidity outside is highest throughout most of the period, and thereafter the supply air RH and RH at 1.6 m is followed by the exhaust air, which, again, correlates very well with the temperatures at the same points. The RH of the supply air and the RH at 1.6 m above floor varies little in relation to the outside RH, however, the RH of the exhaust air, although much lower, correlates better with the outside variations.

Absolute humidity

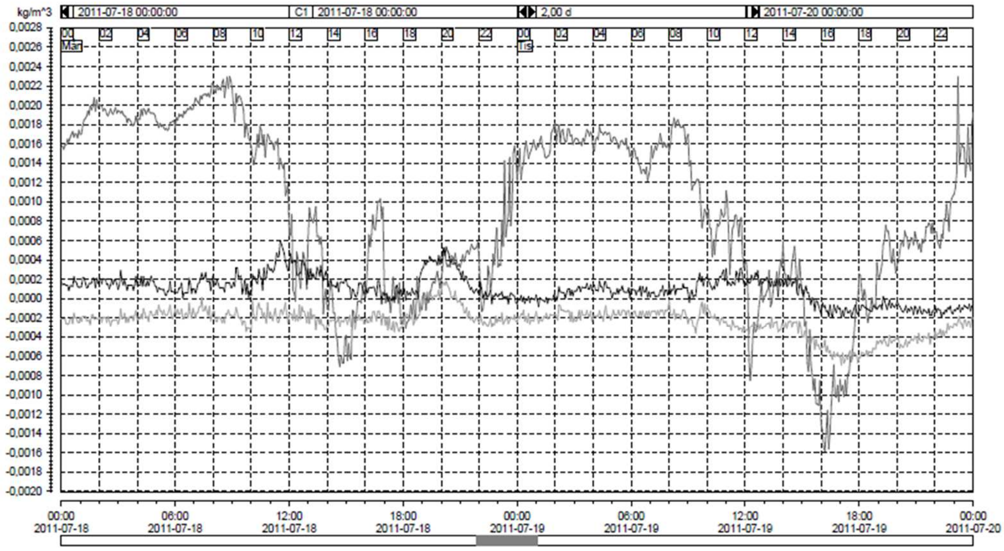
The absolute humidity in the outdoor and indoor air will be accounted for in this chapter. Furthermore the differences between absolute humidity in outside and supply air will be presented as well as differences of the absolute humidity in supply and exhaust air.



	Färg	Enhet	Med	Min	Max	Linje-bredd	Benämning	Punkttyp	
018		kg/m^3	0,0120	0,0096	0,0135		v.out	□	
023		kg/m^3	0,0114	0,0110	0,0119		v.at.1.6m	O	
021		kg/m^3	0,0112	0,0108	0,0118		v.exhaust		
019		kg/m^3	0,0111	0,0107	0,0115		v.supply		

Figure A. 17 Absolute humidity of the outside and the inside air

Figure A. 17 indicates that the greatest variation of the absolute humidity is in the outside air. Outside is also where the maximum absolute humidity is seen. Regarding the remaining absolute humidity at supply, exhaust and at height 1.6 m above floor, the results show a rather equal distribution, with a maximum difference of 1 g/m^3. The vapour content inside the atrium seems to reach a maximum at noon and a minimum during the evening and night hours. The absolute humidity of the outside air reaches a maximum a couple of hours earlier and the same applies to the minimum.



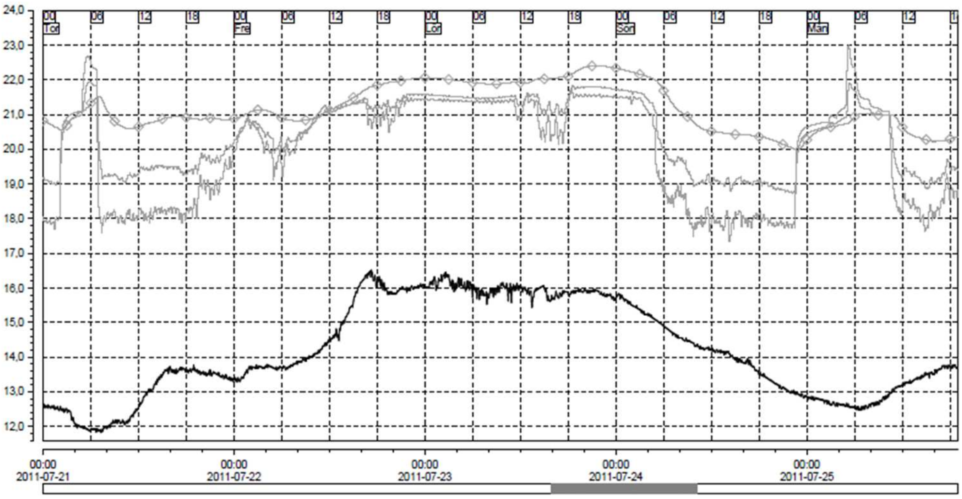
	Färg	Enhet	Med	Min	Max	Linje-bredd	Benämning	Punkttyp	
020		kg/m^3	0,0009	-0,0016	0,0023		v.DIFF.v.out.minus.v.supply		
022		kg/m^3	0,0001	-0,0002	0,0006		v.DIFF.v.supply.minus.v.exhaust		
024		kg/m^3	-0,0002	-0,0007	0,0002		v.DIFF.v.at.1.6m.minus.v.exhaust		

Figure A. 18 Difference in absolute humidity between outside, supply, at 1.6m above floor and exhaust air

The absolute humidity does not differ significantly between different heights in the atrium. Instead, the main difference can be seen between the outside air and supply air, which may be a consequence of the fact that the air is treated. For example the air may be dehumidified in the cooling process.

Water temperature and condensation risk

An interesting period regarding dew point temperatures is from 110721 to 110726, see Figure A. 19. The first sudden increase of the supply temperature is probably due to a low outside temperature, i.e. the winter mode is enabled, which also applies for the period in the end. From 110722 to 110724 however, the dew point temperature increase and so does the supply water temperature. This is probably caused by a dew point adjustment to prevent the risk of condensation.



	Färg	Enhet	Med	Min	Max	Linje-bredd	Benämning	Punkttyp	
008		°C	21,19	19,97	22,41		T J - Floor surface temperature in cell C8	◇	
007		°C	20,45	18,73	21,95		T I - Water temperature, return		
006		°C	19,98	17,35	22,99		T G - Water temperature, supply		
026		°C	14,22	11,84	16,52		Tdew.alt.calculation		
025		°C	14,20	11,82	16,51		Tdew		

Figure A. 19 Water temperatures, dew point temperature and floor surface temperature

The minimum supply water temperature during the entire measurement period is around 17^{o}C and the maximum extend up to 25^{o}C, see Figure A. 20. The maximum temperature is probably caused by a cold outdoor temperature (below 15^{o}C) switching the system into heating mode.

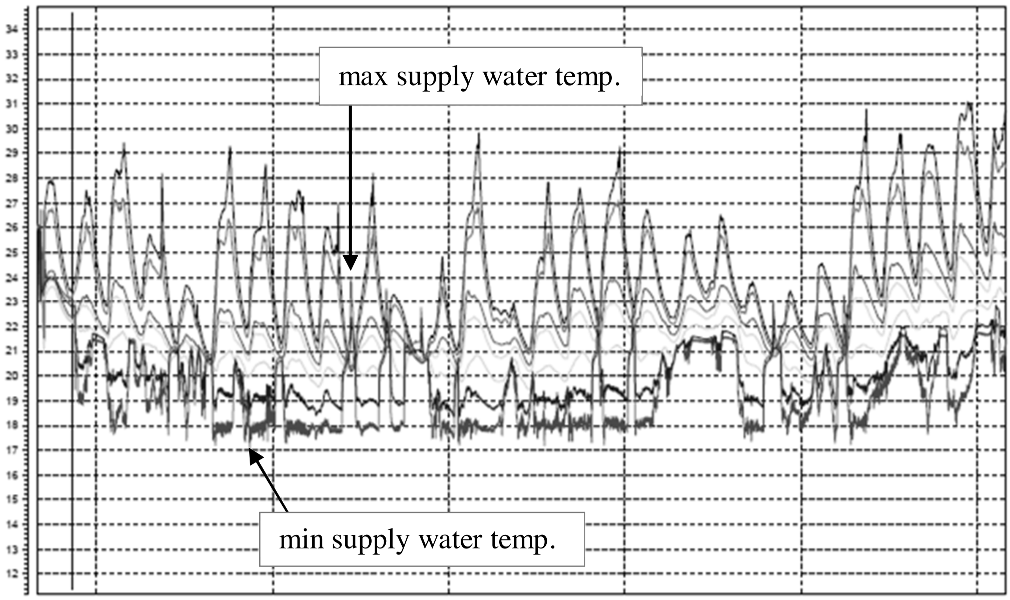


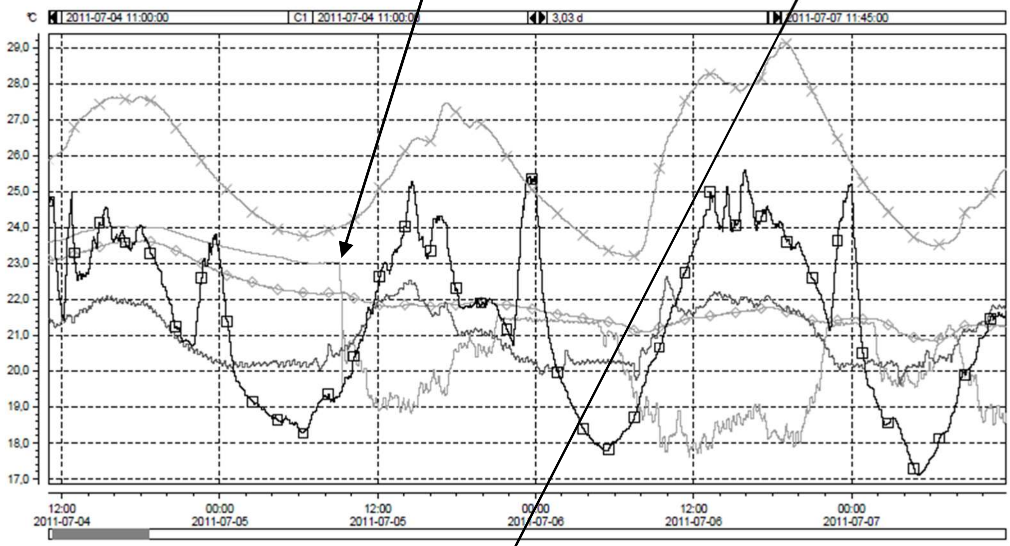
Figure A. 20 Temperature profile, floor surface temperatures and water temperatures from 110704 to 110801

Appendix G – Miscellaneous measurements

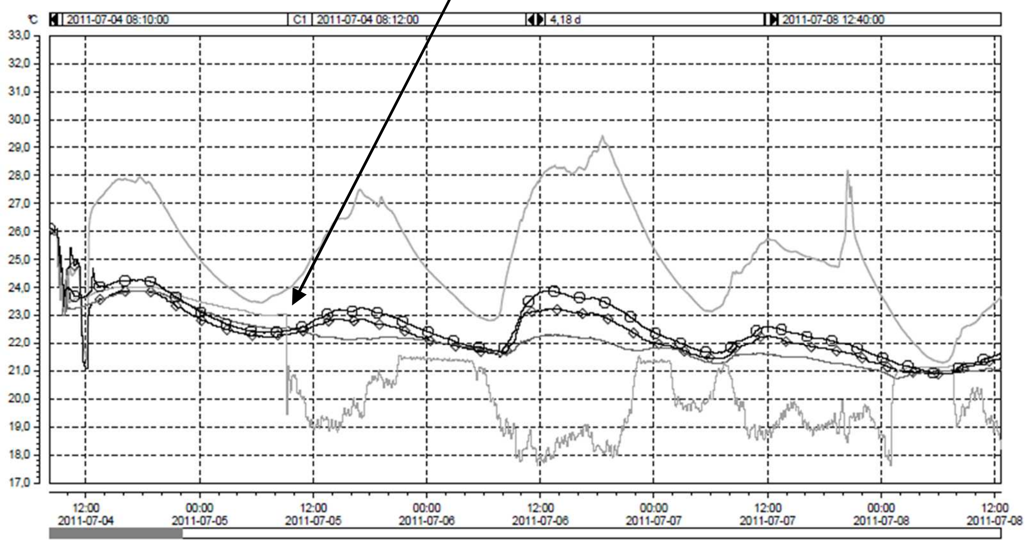
Floor cooling system, going from inactive to active

The following two figures show before and after the cooling system was turned on.

The floor cooling system starts to function. Changes in floor temperatures are evident, but not as evident in air temperature higher up in the atrium.



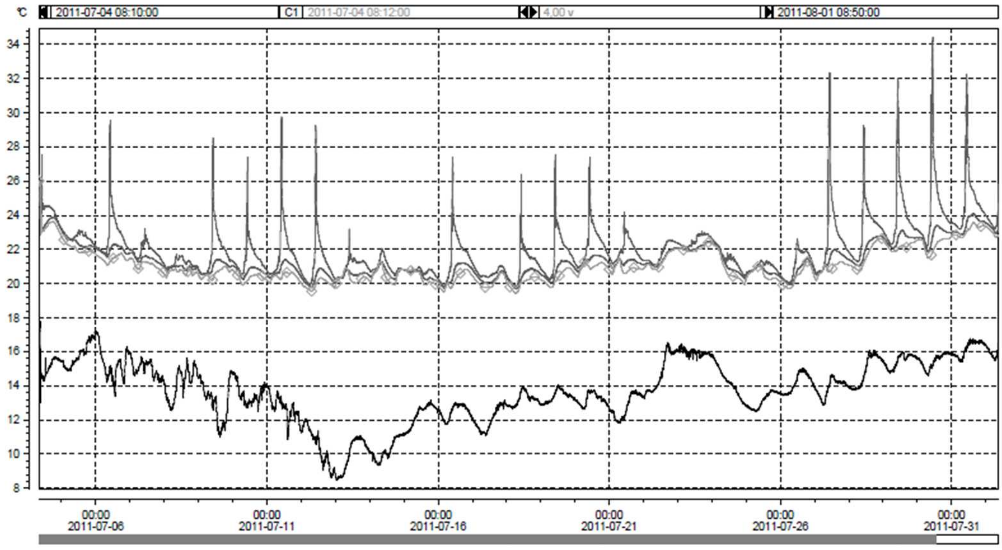
	Färg	Enhet	Med	Min	Max	Linje-bredd	Benämning	Punkttyp	
006		℃	21,00	17,62	24,03		T G - Water temperature, supply		
008		℃	21,96	20,85	23,63		T J - Floor surface temperature in cell C8	◇	
009		℃	21,00	19,60	22,64		RH 2 - Supply air temperature		
013		℃	21,49	17,12	25,59		RH 4 - Outside air temperature	□	
015		℃	25,73	23,19	29,15		RH 5 - Exhaust air temperature, 12.9 m above floor	×	



	Färg	Enhet	Med	Min	Max	Linje-bredd	Benämning	Punkttyp	
001		℃	25,13	21,29	29,42		T 7 - Air temperature, 9 m above floor		
011		℃	22,63	20,89	26,11		RH 3 - Air temperature, 1.6 m above floor	○	
002		℃	22,34	20,88	25,99		T 8 - Air temperature, 0.1 m above floor	◇	
005		℃	22,10	20,73	26,04		T F - Floor surface temperature in cell C6		
006		℃	20,77	17,62	26,06		T G - Water temperature, supply		

Figure A. 21 Before and just after start of the cooling system

Dew- and surface temperature during the entire measuring period



	Färg	Enhet	Med	Min	Max	Linje-bredd	Benämning	Punkttyp	
004		°C	22,08	19,84	34,42		T E - Floor surface temperature in cell C5		
005		°C	21,36	19,70	26,04		T F - Floor surface temperature in cell C6		
008		°C	21,05	19,45	26,01		T J - Floor surface temperature in cell C8	◇	
025		°C	13,68	8,47	17,79		Tdew		

Figure A. 22 Floor surface temperatures and dew point temperatures from 110704 to 110801

Orientation in building and photos of device placement

The following figure shows the orientation of the atrium in the building.

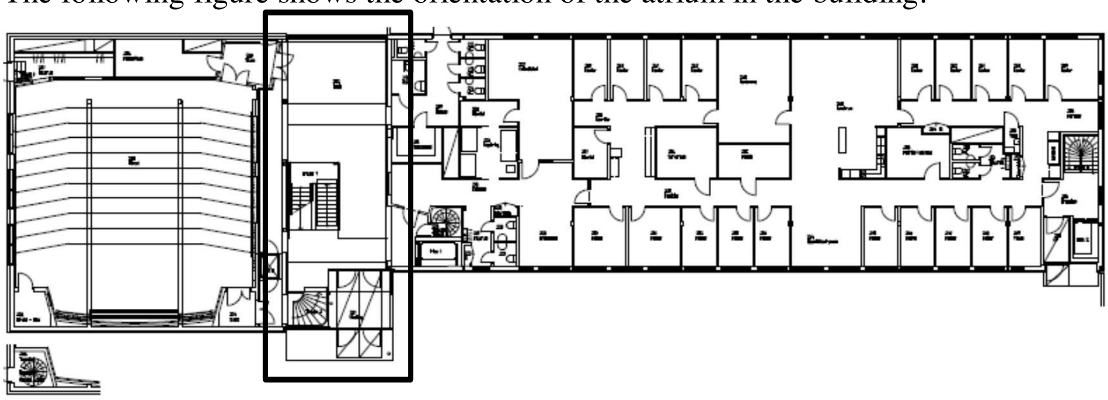


Figure A. 23 Orientation of the atrium

The following photos are presented to give an overview of the measuring devices placement.

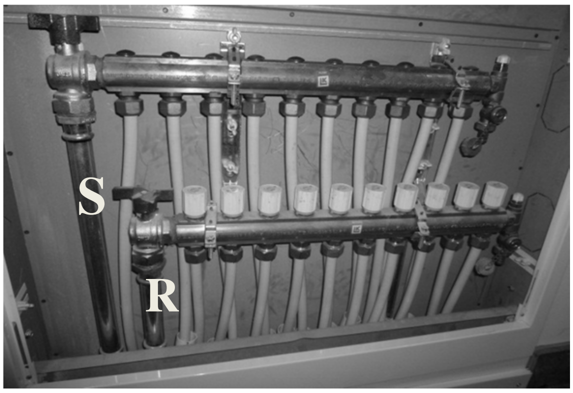


Figure A. 24 The supply (S) temperature and return (R) temperature was measured at the circuit manifold

Outside

RH 3 (preparatory measurements)

RH 4 (long-time measurements)

Figure A. 25 Placement of device outside

Exhaust air

RH 4 (preparatory measurements)

RH 5 (long-time measurements)

Figure A. 26 Placement of device at exhaust air

Figure A. 27 Placement of device at supply air

Figure A. 28 Placement of devices at 5.3 m and 9 m for temperature profile

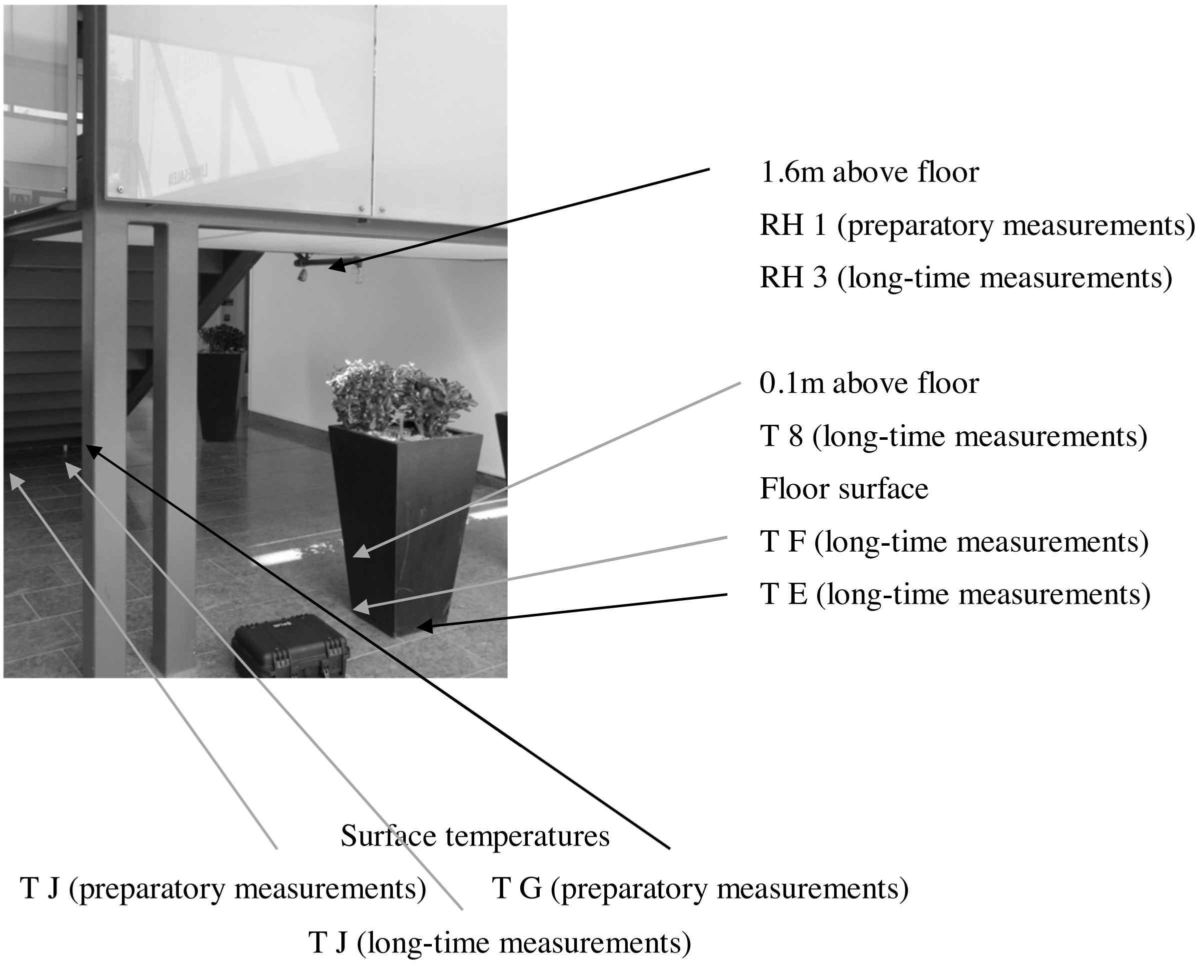


Figure A. 29 Placement of device on floor surface and 0.1 m above floor, as well as 1.6 m above floor

Appendix H – Verifying physics and model in COMSOL Multiphysics

The model and physics in the simulation study need to be verified. In the following chapters the equations and models for the different physics are accounted for as well, with the exception for conduction, which is assumed to be accurate.

Convection

Model

To study the convection physics a simple model was generated (see Figure A. 30) modelling “Non Isothermal Flow” as physics. The simulation was performed over a time period of for two hours, with a time-dependent solver. The time stepping method BDF with free time stepping and a “normal” mesh size was chosen. In order to decrease the simulation time, a downscaled model was used. Different boundary conditions were set to the boundaries, having constant temperatures T_{hot}=50 °C, T_{cold}=25°C, $T_{intitial}$ = 20°C.

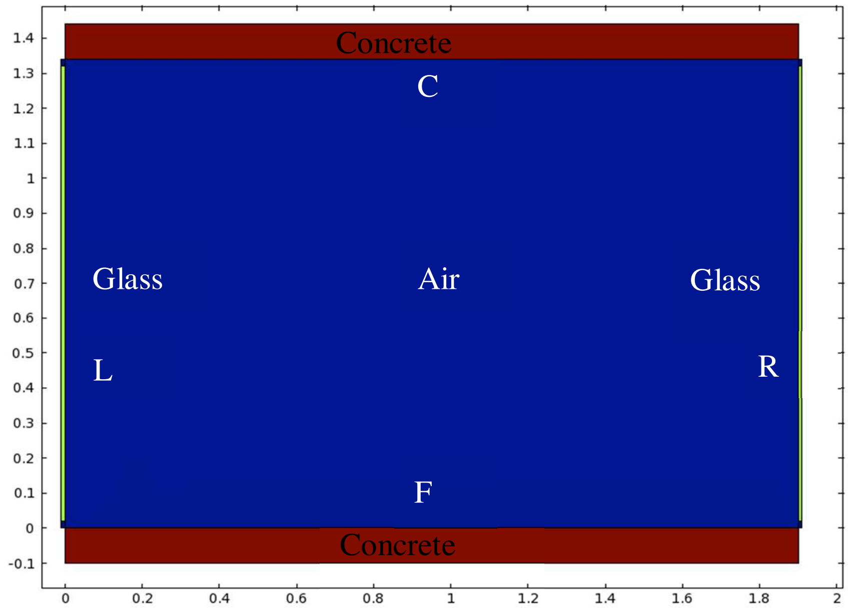


Figure A. 30 Materials and definition of variables

Equations

To simulate the effect of convection and the temperature profile, due to differences in air density, a volume force was added to the simulation, see equation A.5.

$$F = \rho * g \qquad \text{(A. 5)}$$

where:

F - Volume force (N/m^3)

ρ - Density (kg/m^3)

g - Acceleration of gravity (m/s^2)

Verifying the model

The set up and the expected flow pattern can be seen in Figure A. 31. The right wall is set to be hot and the left wall is set to be cold.

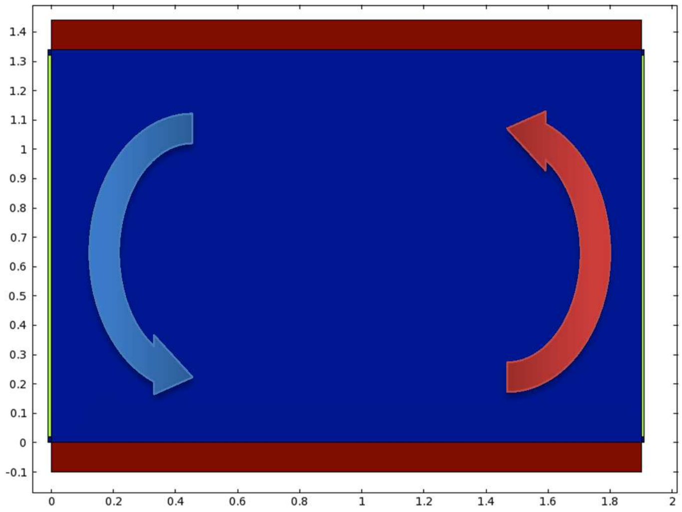


Figure A. 31 Model and expected flow pattern

The result from the simulations (after 3000 seconds) can be seen in Figure A. 32. The simulated flow pattern corresponds well to the expected one, therefore the model is assumed to be correct.

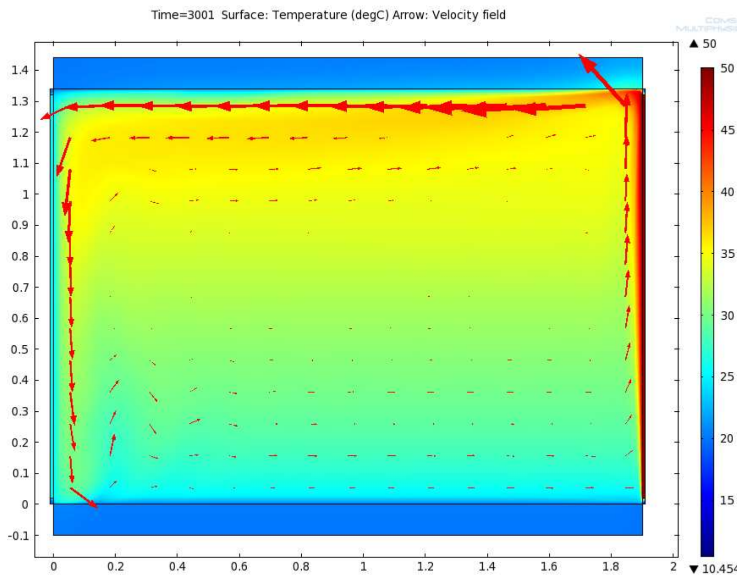


Figure A. 32 Results from simulation showing the temperature and the flow directions of the air

Radiation

Model

A similar model, as for the convection, was used for the radiation simulation, with one exception; the volume force. In its place the function “surface to surface radiation” was enabled in the heat transfer module. Utilizing this function the radiation affects the surfaces which are seen and the participating surfaces must be specified. The temperature was set at 40°C at the right wall, with an initial temperature of 20°C for all remaining parts. The simulation ran for 20 hours.

Verifying the model

The results from the 20 hour simulation can be seen in Figure A. 33.

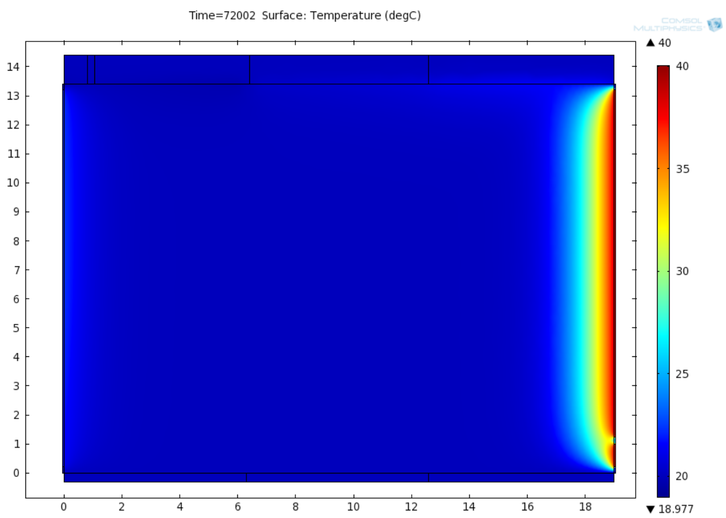


Figure A. 33 Radiation model after 20 hours

As seen in Figure A. 33, the temperature in the middle of the room maintains 20°C, while the left wall is affected by radiation from the right wall, resulting in a temperature increase to 22°C. Without radiation enabled there would not be a temperature increase along the left wall, while the air in the middle of the room was unaffected, hence the model seems to be accurate.

Combined

A model combined, with the three above mentioned physics, needed to be verified as well. Furthermore ventilation was added to the model.

Model

The model used in the radiation simulations was utilized for the combined simulations as well, with the addition of ventilation the volume force. The ventilation was specified by adding an outlet and inlet velocity, as well as inlet temperature of the air. The inlet was placed to the lower right and the outlet at the upper left, see Figure A. 34.

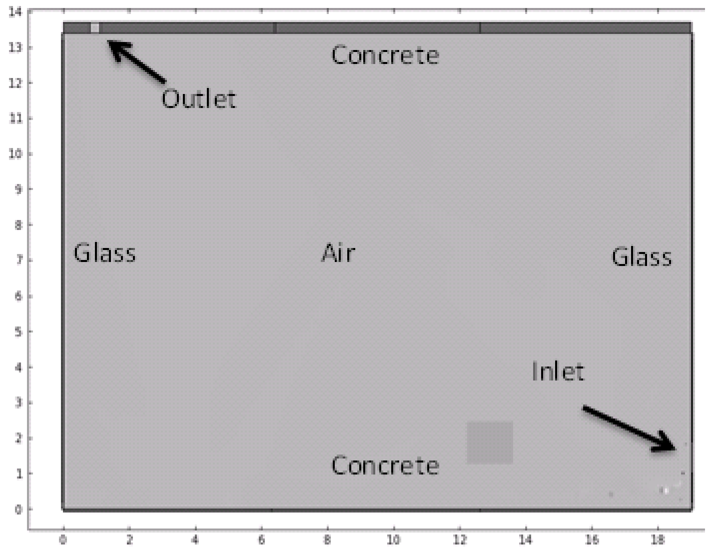


Figure A. 34 *Combined physics model*

Verifying the model

In this study different physics are at work, which may counteract each other, therefore there were some difficulties to interpret the results from the simulations. Simulations were conducted with and without the radiation turned on; a comparison can be seen in Figure A. 36.

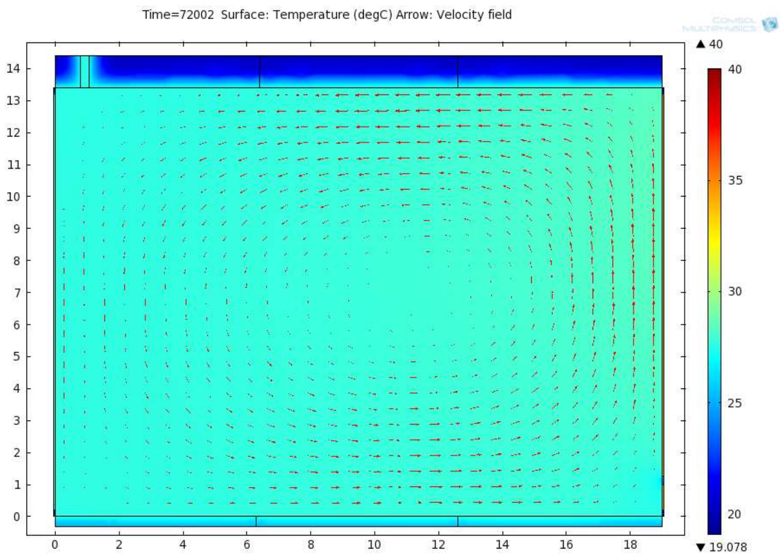
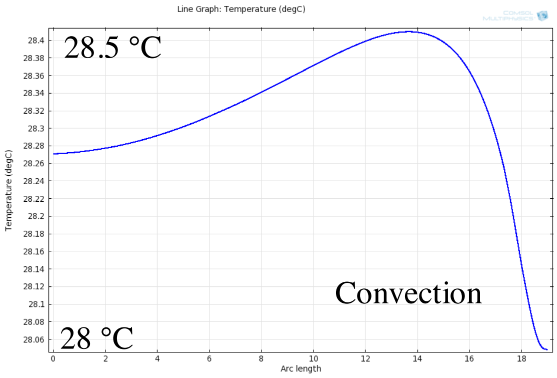


Figure A. 35 Simulated combined model after 20 hours

In Figure A. 35 the simulated results after 20 hours can be seen. As it is difficult to see differences in this figure, graphs of the boundary temperatures will be analysed instead.



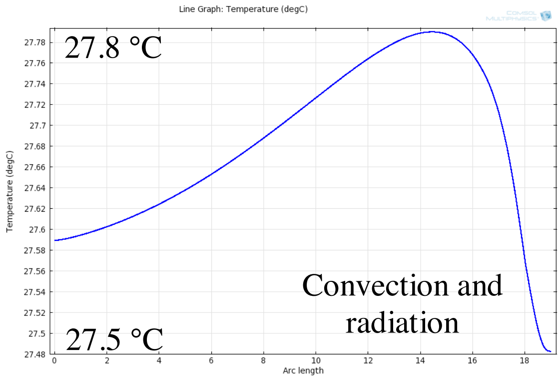


Figure A. 36 Mean temperature difference with and without radiation

The floor temperatures after 20 hours are shown in Figure A. 36. The interesting part in the graphs is the shape and the mean temperature difference. When radiation is added to the model the mean temperature decreases, which can be is seen in the right part of Figure A. 36. This can be explained by that the right wall surface temperature decreases, due to radiation form the remaining cold surfaces in the room. The lower surface temperature reduces the heat flow from the wall (due to lower temperature difference between the wall and indoor air temperature) and with that the overall mean temperature is decreased. This means that the combined model acts in an adequate way.

Appendix I - Heat flow density calculations

Cooling effect from on WATERSIDE of system

Basic input

$\rho_w := 1000 \frac{kg}{m^3}$ Density of water

$Cp_w := 4200 \frac{J}{kg \cdot K}$ Heat capacity of water

Specific input

$Area_a := 127m^2$ Floor area with pipes

$T_{f\_S} := 17.9\,°C$ Temperature fluid Supply Date 2011-07-11 kl 12:00

$T_{f\_R} := 19.6\,°C$ Temperature fluid Return

$$V_{flow\_tot\_designed} := (9 \cdot 115.4 + 72.4)\frac{l}{hr} = 1.111 \times 10^3 \cdot \frac{l}{hr}$$

$V_{flow\_tot} := 1300 \frac{l}{hr}$ Volume flow for floor cooling system in total
Measured value

Q.supply side

$$Q_{supply} := \rho_w \cdot Cp_w \cdot V_{flow\_tot} \cdot \left(T_{f\_R} - T_{f\_S}\right) = 2.578 \times 10^3\, W$$

$$q_{up} := \frac{Q_{supply}}{Area_a} = 20.302 \cdot \frac{W}{m^2}$$ Water side.

Cooling effect according to Academic method

Formulas can be found in example from session 8, lecture
held by Henrik Karlsson from SP 20110202

$\gamma_{up} := 0.7143 \frac{W}{m \cdot K}$ $\gamma_{down} := 0.1587 \frac{W}{m \cdot K}$ From HEAT 2 set fluid temp to 0 C and enviroment to 1

$T_i := 23.5\,°C$ $T_e := 18\,°C$ Date 2011-07-11 kl 12:00 Temperatures, i = inside, e = garage

$A_{floor} := 12.7m^2$ Looking at one circut

$T_{f.0} := 17.9\,°C$ Supply water temperature, $T_{.f}$(x=0). Date 2011-07-11 kl 12:00

$V_{dot.f} := 130 \frac{L}{hr}$ Water flow

$\eta := \frac{\gamma_{up}}{\gamma_{up} + \gamma_{down}} = 0.818$ Insulation efficientcy

$\rho_{water} := 1000 \frac{kg}{m^3}$ $c_{water} := 4200 \frac{J}{kg \cdot K}$ Properties of water

$\rho c_{water} := \rho_{water} \cdot c_{water} = 4.2 \times 10^6 \cdot \frac{J}{m^3 K}$

$l_c := \frac{\rho c_{water} \cdot V_{dot.f}}{\gamma_{up} + \gamma_{down}} = 173.73\,m$ characteristic length

$cc := 0.16m$ centre to centre distance

$L_{pipe} := \frac{A_{floor}}{cc} = 79.375\,m$ Length of pipe

$T_0 := \eta \cdot T_i + (1 - \eta) \cdot T_e = 22.5 \cdot °C$ Reference temperature. Weighted Mean surrounding temperature

$T_{f.L} := T_{f.0} \cdot e^{\frac{-L_{pipe}}{l_c}} + T_0 \cdot \left(1 - e^{\frac{-L_{pipe}}{l_c}}\right) = 19.587 \cdot °C$ Fluid temperature at end of pipe = Return water temperature

$T_{f.0} - T_{f.L} = -1.687\,K$

$T_{f\_mean} := \frac{T_{f.0} + T_{f.L}}{2} = 18.744 \cdot °C$ Mean water temp

$Q_{supply} := V_{dot.f} \cdot (T_{f.0} - T_{f.L}) \cdot \rho c_{water} = -255.877\,W$ Heat flow rate in pipes

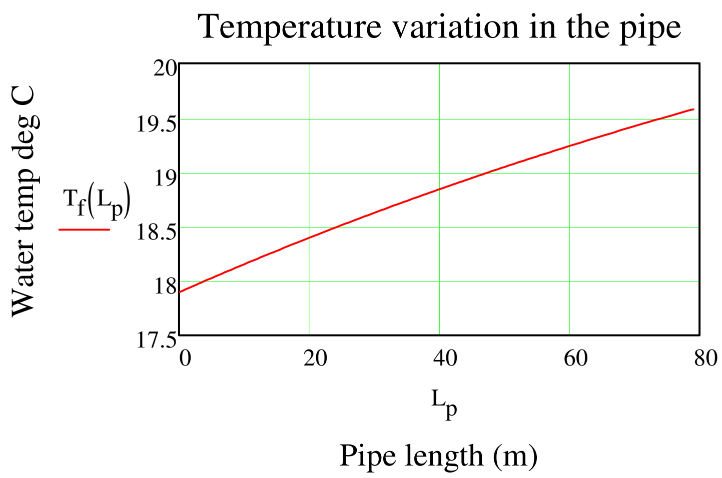
$Q_{up} := Q_{supply} \cdot \eta = -209.362\,W$

$\frac{Q_{supply}}{A_{floor}} = -20.148 \cdot \frac{W}{m^2}$ Heat flux density water side

$q_{floor} := \frac{Q_{up}}{A_{floor}} = -16.485 \cdot \frac{W}{m^2}$ Heat flux density

$$L_p := 0, 0.5m .. 1L_{pipe}$$

$$T_f(L_p) := \left[T_{f.0}\cdot e^{\frac{-L_p}{l_c}} + T_0\cdot\left(1 - e^{\frac{-L_p}{l_c}}\right)\right] - 273.15K$$



Cooling effect according to EN 1264-2:1998

Equations from EN1264-2:1998

$T_{f.0} := 17.9\,°C$ Supply water temperature, $T._f$(x=0). Date 2011-07-11 kl 12:00

$T_{f.L} := 19.6\,°C$ Return water temperature, $T._f$(x=L).

$T_{air} := 23.5\,°C$ Air temperature

$cc := 0.16$ Pipe distance cc

$D := 0.016$ Pipe diameter

$s_u := 0.07$ Thickness of cover above pipes

$$q_{floor} := B \cdot a_B \cdot a_T^{m_T} \cdot a_u^{m_u} \cdot a_D^{m_D} \cdot \Delta T_H = \blacksquare \cdot \frac{W}{m^2}$$ Heat flux density

Calculations

$$B := 6.7 \frac{W}{\left(m^2 K\right)}$$ Thermal conducatance and inmpact from pipe

$\alpha := 10.8$ $\lambda_{u.0} := 1$ $s_{u.0} := 0.045$

$\lambda_{screed} := 1.5$ $R_{\lambda.Limestone} := \frac{0.04}{1.33} = 0.03$

$$a_B := \frac{\frac{1}{\alpha} + \frac{s_{u.0}}{\lambda_{u.0}}}{\frac{1}{\alpha} + \frac{s_{u.0}}{\lambda_{screed}} + R_{\lambda.Limestone}} = 0.901$$

$a_T := 1.2048$ $a_u := 1.0504$ $a_D := 1.037$

$$m_T := 1 - \frac{cc}{0.075} = -1.133$$

$$m_u := 100 \cdot \left(0.045 - s_u\right) = -2.5$$

$$m_D := 250(D - 0.02) = -1$$

$$\Delta T_H := \frac{T_{f.0} - T_{f.L}}{\ln\left(\frac{T_{f.0} - T_{air}}{T_{f.L} - T_{air}}\right)} = -4.699\,K$$

$$q_{floor} := B \cdot a_B \cdot a_T^{m_T} \cdot a_u^{m_u} \cdot a_D^{m_D} \cdot \Delta T_H = -19.591 \cdot \frac{W}{m^2}$$ Heat flux density

Calculations valid for type A floors.

Comparisson of different calculations methods

	Case 1	Original system	**Case 5**	Decreasing supply temp -
	Case 2	Increase water flow+	**Case 6**	Decreasing supply temp --
	Case 3	Increase water flow ++	**Case 7**	Decreasing supply temp ---
	Case 4	Increase water flow +++	**Case 8**	Compare EN1264 and Henrik

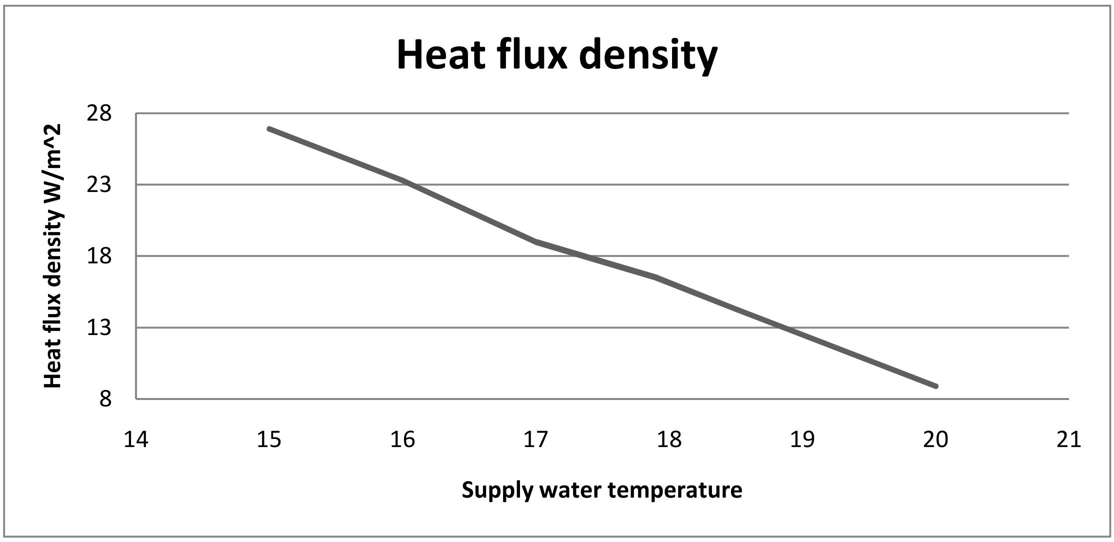
		CASE 1			**CASE 2**			**CASE 3**			**CASE 4**		
		Waterside	EN1264	Academic	Waterside	EN1264	Academic	Waterside	EN1264	Academic	Waterside	EN1264	Academic
T_air	°C	23,5	23,5	23,5	23,5	23,5	23,5	23,5	23,5	23,5	23,5	23,5	23,5
T_supply	°C	17,9	17,9	17,9	17,9	17,9	17,9	17,9	17,9	17,9	17,9	17,9	17,9
T_return	°C	19,6	19,6	-	19,6	19,6	-	19,6	19,6	-	19,6	19,6	-
Water flow	l/hr	130	-	130	150	-	150	170	-	170	200	-	200
Results													
Heat flux density	W/m2	20,3	19,6	16,5	23,4	19,6	17	26,5	19,6	17,3	31,2	19,6	17,8
T_return	°C	-	-	19,6	-	-	19,4	-	-	19,3	-	-	19,1

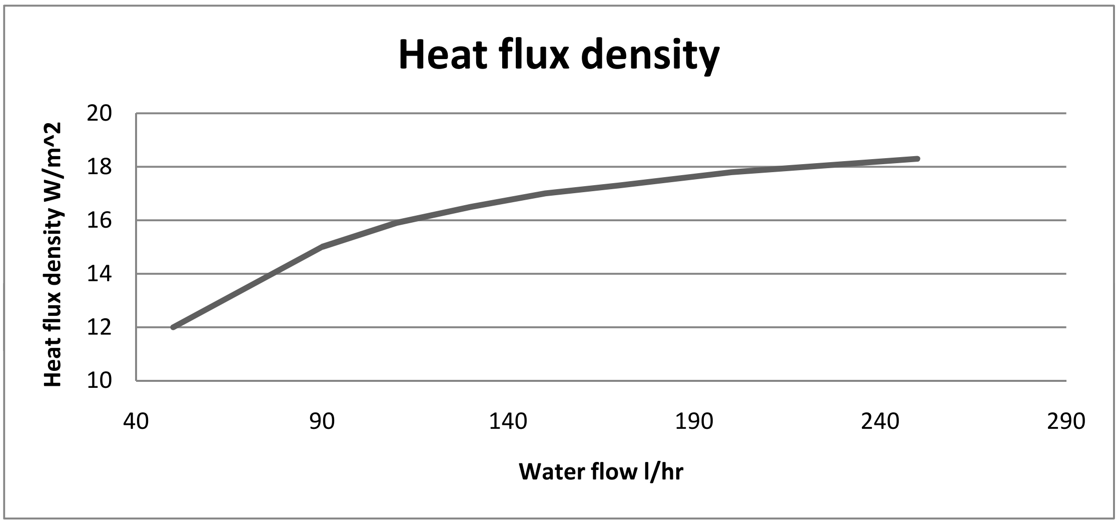
		CASE 5			**CASE 6**			**CASE 7**			**CASE 8**		
		Waterside	EN1264	Academic	Waterside	EN1264	Academic	Waterside	EN1264	Academic	Waterside	EN1264	Academic
T_air	°C	23,5	23,5	23,5	23,5	23,5	23,5	23,5	23,5	23,5	23,5	23,5	23,5
T_supply	°C	17	17	17	16	16	16	15	15	15	17,9	17,9	17,9
T_return	°C	19,6	19,6	-	19,6	19,6	-	19,6	19,6		19,6	19,6	-
Water flow	l/hr	130	-	130	130	-	130	130	-	130	150	-	140
											η (Insulation efficiency)		0,9
Results													
Heat flux density	W/m2	31	21,2	19	43	22,9	23,3	54,9	24,6	26,9	21,9	19,6	19,3
T_return	°C	-	-	18,9	-	-	18,4	-	-	17,8	-	-	19,6

Case 9	Increase water flow	**Case 13**	Increasing supply temp +
Case 10	Decarease water flow -	**Case 14**	Increasing supply temp ++
Case 11	Decarease water flow --	**Case 15**	Increasing supply temp +++
Case 12	Decarease water flow ---		

		CASE 9			**CASE 10**			**CASE 11**			**CASE 12**		
		Waterside	EN1264	Academic	Waterside	EN1264	Academic	Waterside	EN1264	Academic	Waterside	EN1264	Academic
T_air	°C	23,5	23,5	23,5	23,5	23,5	23,5	23,5	23,5	23,5	23,5	23,5	23,5
T_supply	°C	17,9	17,9	17,9	17,9	17,9	17,9	17,9	17,9	17,9	17,9	17,9	17,9
T_return	°C	19,6	19,6	-	19,6	19,6	-	19,6	19,6	-	19,6	19,6	-
Water flow	l/hr	250	-	250	110	-	110	90	-	90	50	-	50
Results													
Heat flux density	W/m2	39	19,6	18,3	17,2	21,7	15,9	14	21,7	15	14	21,7	12
T_return	°C	-	-	18,9	-	-	19,8	-	-	20,1	-	-	21,1

		CASE 13			**CASE 14**			**CASE 15**					
		Waterside	EN1264	Academic	Waterside	EN1264	Academic	Waterside	EN1264	Academic	Waterside	EN1264	Academic
T_air	°C	23,5	23,5	23,5	23,5	23,5	23,5			23,5			
T_supply	°C	18,5	18,5	18,5	19	19	19			20			
T_return	°C	19,6	19,6	-	19,6	19,6	-			-			
Water flow	l/hr	130	-	130	130	-	130			130			
Results													
Heat flux density	W/m2	13	18,5	14,3	7,2	17,5	12,5			8,9			
T_return	°C	-	-	20	-	-	20,3	-	-	20,9			


Heat flux density
Heat flux density W/m^2
28
23
18
13
8
14
15
16
17
18
19
20
21
Supply water temperature


Heat flux density
Heat flux density W/m^2
20
18
16
14
12
10
40
90
140
190
240
290
Water flow l/hr

HEAT2 report: FLOOR_HEATING_ELEMENT

Boundary conditions:

```
4 types
q=0 W/m
T=1°C, R=0.1428 m²·K/W
T=1°C, R=0.1428 m²·K/W
q=0 W/m
```

Calculation:

```
Steady-state simulation, 1459 iterations
Stable time step: 0.3523 s
Determined at cell (53,151) (dx,dy)=(0.0015,0.0015)
Number of cells: 21400 (Nx=107, Ny=200)
```

Internal flows:

```
Area,  lower,  right,  upper,  left,   total
       [W/m]   [W/m]   [W/m]   [W/m]   [W/m]
 1   -0.0963 -0.2339 -0.2984 -0.2444  -0.873
All areas: sum of positive flows=      0 [W/m]
All areas: sum of absolute flows=  0.873 [W/m]
All areas: net flow= -0.873 [W/m]
```

Boundary flows:

```
Sum pos flows:  0.873 W/m

Heat flows for each BC:
  BC           q [W/m]
  [2]          0.7143  (T=1 R=0.1428)
  [3]          0.1587  (T=1 R=0.1428)
          Sum:  0.873

Bound    q        q     length  BC
      [W/m²]    [W/m]     [m]
 1    0.9921   0.1587    0.16  [3] T=1 R=0.1428
 3    4.4642   0.7143    0.16  [2] T=1 R=0.1428
Sum flows:      0.873 W/m
```

Project info:

Input file: FLOOR_HEATING_ELEMENT.DAT (*.H2P), last saved on 2012-02-14 16:56:32

Materials:

	Lx [W/(m·K)]	Ly [W/(m·K)]	C [MJ/(m³·K)]
Limestone	1.33	1.33	1
Screed	1.5	1.5	1
Concrete	1.7	2.1	2.448
EPS-Insulation	0.033	0.033	1
MOD: J Pipe (given temp) - Constant	1	1	1