



Hochschule Reutlingen
Reutlingen University



Achieving better thermal comfort and energy savings by low-temperature heating

An experimental study of radiator boosters

Bachelor Thesis

Written by:

Nicklas Ganter

Born on August 1st, 1993, in Tübingen, Germany

Matriculation number: 723197

Supervisor at KTH Royal Institute of Technology:

Prof. Sture Holmberg

Supervisor at Reutlingen University:

Prof. Stephan Pitsch

Handover date:

29.07.2016

Declaration of academic honesty

I hereby confirm that the presented thesis is solely my own work and that if any text passages or diagrams from books, papers, the Web or other sources have been copied or in any other way used, all references – including those found in electronic media – have been acknowledged and fully cited.

Location, Date

Signature

Abstract (English)

Climate change proceeds and is more and more visible in our lives. For this reason, every opportunity to achieve energy savings must be considered. One area with a great potential is the residential sector. One retrofittable option are radiator boosters. They are expected to combine energy savings and improvements of thermal comfort in a keen and easy way.

This experimental study aims to examine the impacts of radiator boosters on both thermal comfort and possible energy savings. A single-family house in the northern suburban part of Stockholm was chosen in order to conduct measurements during the heating season. Within six weeks, radiator boosters were switched on an off alternately for one week, respectively.

It was shown that the radiators boosters' benefit was dependent on the buildings layout. Greater air circulation was considered as the major driver for energy savings and improvements of thermal comfort. Their benefit was greater with lower outdoor temperatures. Energy savings and improvements of thermal comfort were within the uncertainty of measurement devices admittedly. However, this study implies potential if implementing the radiator boosters in a more appropriate way.

Keywords: Low-temperature heating, thermal comfort, building energy performance

Kurzzusammenfassung (German)

Der voranschreitende Klimawandel wird immer sichtbarer und realer. Aus diesem Grund sollte jede Möglichkeit für Energieeinsparungen in Betrachtung gezogen werden. Die Beheizung von Gebäuden stellt dabei eine nicht unerhebliche Möglichkeit für diesen Zweck dar. Eine nachrüstbare Option sind „Radiator Booster“. Von ihnen erhofft man sich, den thermischen Komfort zu erhöhen und gleichzeitig den Energiekonsum zu reduzieren.

Diese Bachelorarbeit beschäftigt sich mit den Auswirkungen dieser „Radiator Booster“ auf den thermischen Komfort und den Energiekonsum. Hierfür wurden Messungen in einem Einfamilienhaus im Norden Stockholms innerhalb der Heizperiode durchgeführt. Innerhalb dieser sechs Wochen wurden die „Radiator Boosters“ abwechselnd für je eine Woche an-, beziehungsweise ausgeschaltet.

Es zeigte sich, dass deren Effekt vom Grundriss des Gebäudes abhängig war. Stärkere Luftzirkulation wurde als Hauptursache für Energieeinsparungen und einen verbesserten thermischen Komfort gesehen. Der Effekt vergrößerte sich mit sinkenden Außentemperaturen. Verbesserungen in der Energiebilanz und im thermischen Komfort waren zwar innerhalb der Unsicherheit der Messgeräte. Diese Studie zeigt jedoch trotzdem ein gewisses Potential im Falle einer passenderen Implementierung auf.

Schlagwörter: Niedrigtemperatur, thermischer Komfort, Energieleistung

Table of contents

Declaration of academic honesty	I
Abstract (English)	II
Kurzzusammenfassung (German)	III
Table of contents	IV
1 Introduction	1
1.1 Motivation.....	2
1.2 Structure of this thesis	3
1.3 Theoretical introduction and foundations	4
1.3.1 Indoor environmental conditions.....	4
1.3.2 The principles of heat transfer	6
1.3.3 The principles of a heat pump	8
1.3.4 The energy balance of a building	10
1.4 State of the art.....	13
1.4.1 Embedded surface heating.....	14
1.4.2 Baseboard radiators	14
1.4.3 Ventilation radiators	15
1.4.4 The radiator booster.....	16
1.4.5 Comparison.....	19
1.5 Objectives	20
2 Methodology	21
2.1 The building and its properties	21
2.2 The heat pump's control system	23
2.3 Procedure and schedule	24
2.4 Calculations	25
2.5 Measurement devices.....	29
2.6 Calibration of measurement devices.....	31
2.7 Expected results	34
3 Results	36
3.1 Limiting aspects and other occurrences.....	36
3.2 Measurement results	37
3.2.1 Presentation of polished results	38
3.2.2 Polishing of measured data.....	43
3.2.3 Results of polished data.....	44
3.3 Explanations for measurement results	46
3.4 Estimation of additional possible energy and cost savings.....	48
3.5 Uncertainty analysis.....	51
3.6 Reassessing of the hypothesis and expected results	52
4 Résumé	54
4.1 Overall result	54
4.2 Lookout.....	55

Nomenclature.....	IX
List of figures	XII
List of tables	XIII
Appendix A	XIV
Appendix B.....	XV

1 Introduction

Climate change will probably be the most challenging issue in the future for human kind. Significant increases in emission of greenhouse gases are leading to higher temperatures all around the world. The most dangerous gases are carbon dioxide (CO_2), methane (CH_4) and dinitrogen monoxide (N_2O). With CO_2 being the main driver, the mean temperature on earth has been increasing by 0.8°C since 1880. [2] In other words, climate change probably began with the industrialization. This fact can be seen in Figure 1.1. In 2012, the total energy use in the world amounted to a total of 100 400 TWh, increasing from year to year. This depicts a rise of 40 % since 1990. It is mainly due to emerging and developing countries [3].

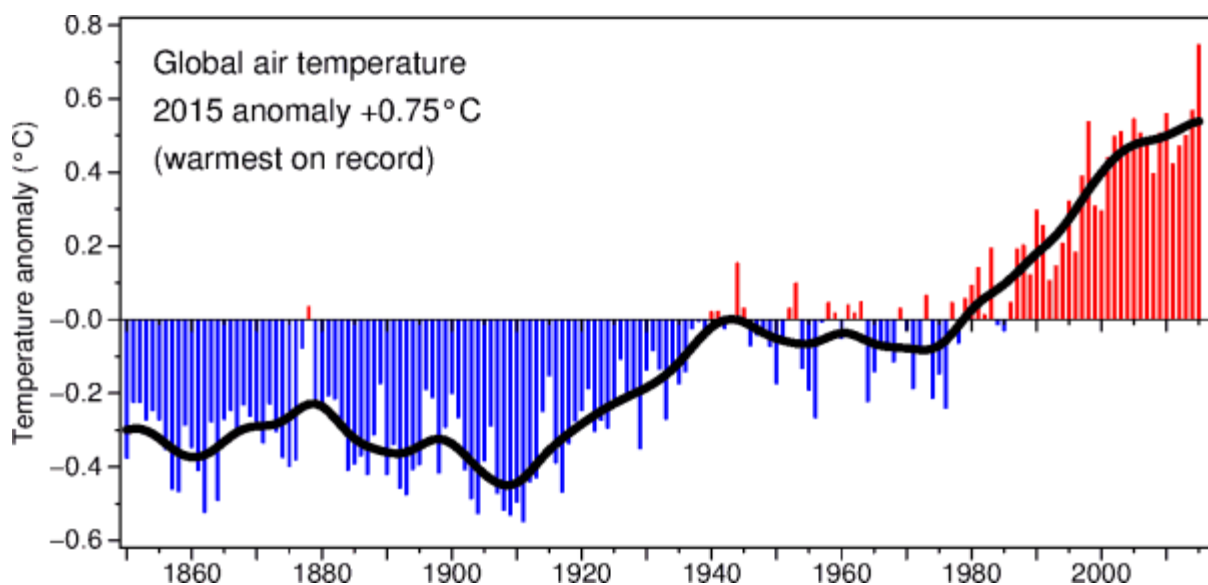


Figure 1.1: Development of global mean temperature [4]

As many as 13 of the hottest 14 years since beginning of weather records happened in this century. Increased temperatures are responsible for the melting of glaciers and the enhancement of the sea level, what consumes more and more land. Weather extremes ruin the crop and limit nourishment production. Changing and destroyed habitats lead to extinction of animals and plants. Climate change is not only a danger for nature, it also affects people directly since healthcare and infrastructure will be strained to a higher extent. In addition, more political tensions about limited natural resources can be expected.

Due to this, the European Union agreed on a strategy to limit the temperature rise to 2°C . The so called strategy “Europe 2020” sets the following targets (figures compared to 1990):

- Reduction of greenhouse gases by 20 %,
- Reduction of energy use with the help of improved energy efficiency by 20 %,
- Increasing the share of renewable energy use by 20 %.

Consequences of global warming appear slowly and progressively in long periods of times. Since these targets only constitute the first step and are not sufficient to limit global warming, new targets were set with more ambitious goals until 2030 and 2050. [2]

1.1 Motivation

Being aware of different stages of development as well as of variances in access to different forms of energy within the European countries, the EU has set different targets for different countries. Sweden, for instance, is already advanced in implementing the use of renewable energy. With a share of 52 % in 2013, Sweden has surpassed the targets of the EU by far. In addition, it has also surpassed its own goals. Planned share in the use of renewable energy was 50 % until 2020. Moreover, Sweden uses renewable energy in the transport sector for 15 % of the total use (planned until 2020: 10 %). Figure 1.2 illustrates the distribution of final energy usage in Sweden (2013).

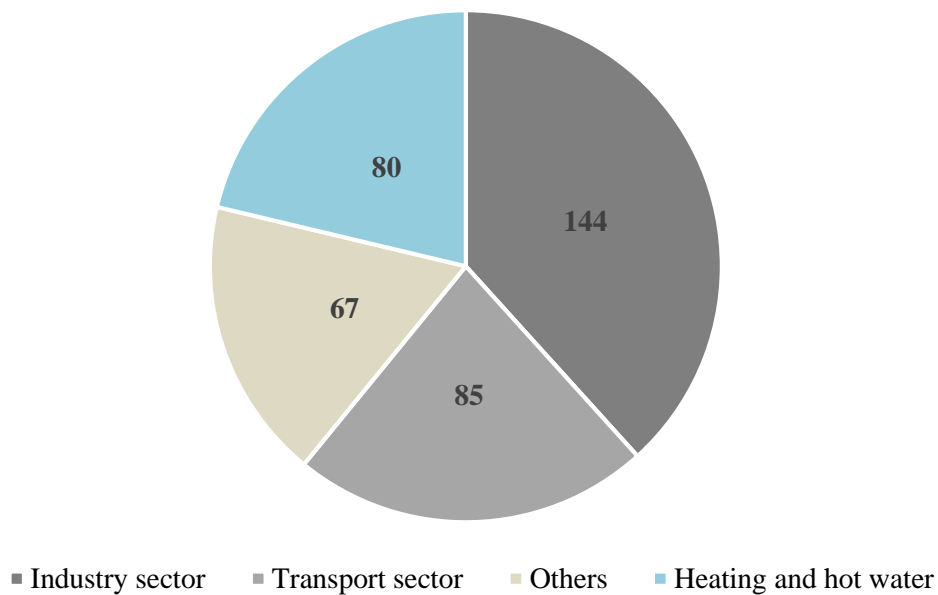


Figure 1.2: Own depiction of relative final energy usage in Sweden (2013), referring to Energimyndigheten [3]

The total energy use in Sweden has been staggering between 500 – 600 TWh for years and amounted to 565 TWh in 2013. Subtracting losses in production and transportation, the final energy use is 375 TWh. Of this amount, 39,2 % is connected to the residential and service sector, consuming 147 TWh. This depicts the greatest field of energy use. It comprises not only households, but also public administration, commercial, agriculture, forestry, fishing and construction. However, households and non-residential buildings are responsible for approximately 90 % of the energy use in this sector. With 80 TWh of the energy being used for heating and water, this sector constitutes a major driver for energy use and implies a high potential for savings and improvements. In total, 21.3 % from the total final energy use in Sweden is used for heating and hot water. [3]

Heating demand of modern buildings has been decreasing lately [3]. This is due to several reasons: Besides new techniques for reducing ventilation losses, better insulation is applied for new and retrofitted buildings. In order to further reduce energy use, low valued energy should be used. Low valued energy is mainly available in residual and ambient heat. In addition, it is provided by renewable sources. Since these sources are not able to provide high temperatures, they are mostly used for low-temperature heating. [5] Different supply and return temperature levels are defined in Table 1.1:

Table 1.1: Temperature ranges for different heating systems [5]

System	Supply / Return temperature, °C
Very low temperature heating	35 / 25
Low temperature heating	45 / 25-35
Medium temperature heating	55 / 35-40
High temperature heating	90 / 70

Low temperature heating systems (LTH) have multiple benefits. Besides leading to less heat loss in pipes, it stands out with a better efficiency of boilers and a better thermal comfort and indoor air quality (IAQ). This is reached through lower temperatures and larger surfaces. An example for LTH using large surfaces is floor heating. Here the complete floor serves as a radiator. LTH provides an adequate thermal comfort at significantly lower temperatures, compared to medium- or high-temperature heat sources. The lower supply and return temperature allow the usage of more renewable energy and low valued energy sources, due to the already mentioned reasons [6]. In addition, the energy production by using e.g. heat pumps or sun panels, is more efficient [7].

1.2 Structure of this thesis

The present thesis is divided into four main chapters. The introduction (chapter 1) outlines the necessity for doing research for energy savings in the residential area. Technical basics and principles, are then introduced. They comprise thermal indoor conditions, the principles of heat transfer and a heat pump, the radiator booster itself and the energy balance of a building. A brief review introduces the reader to the previously done research that aimed to decrease energy use by remaining the same thermal comfort.

Subsequently, the focus will lie on the methodology of the experimental study (chapter 2). Therefore, the building's properties and the heat pump's control system are focused on. The procedure of this experimental study is followed by the used calculations and the measurement devices. After having approached all necessary issues affecting this study, expected results are conducted.

Before the obtained results are presented (chapter 3), the limitations of this study need to be addressed. Specific occurrences made it necessary to refurbish the data. The conducted measures and calculations are presented and final results presented and evaluated. The next subchapter then suggests possible explanations. Based on these results, an estimation was made about how much annual energy and costs can be saved with the deployment of radiator boosters based on the degree-day method.

Finally, the overall results are stated, obtained data is compared to the expected results and a lookout is given (chapter 4).

1.3 Theoretical introduction and foundations

A heat supply chain can be divided into three main parts. The heat is provided by a heat source. The type is dependent on the purpose, the energy storage and the location. One-family houses mostly rely on boilers, being fuelled by natural gas, oil and wood, while for multi-family houses and commercial buildings district heating is a common mean. The distributing system then provides the heat to the consumer. Usually it is distributed by hydronic systems or by air. The former one is the dominating distribution system in colder areas. However, distribution systems based on air are common in, for instance, the United States or in commercial buildings, such as shopping malls. Finally, space heating supplies the heat in the individual rooms. Radiators, convectors and floor heating are common means for hydronic distribution systems. [8] Figure 1.3 illustrates the heat supply chain and its components being present in this experimental study. The heat was provided by a heat pump, distributed by a hydronic distribution system and provided in the individual rooms by one- and two-panel radiators.

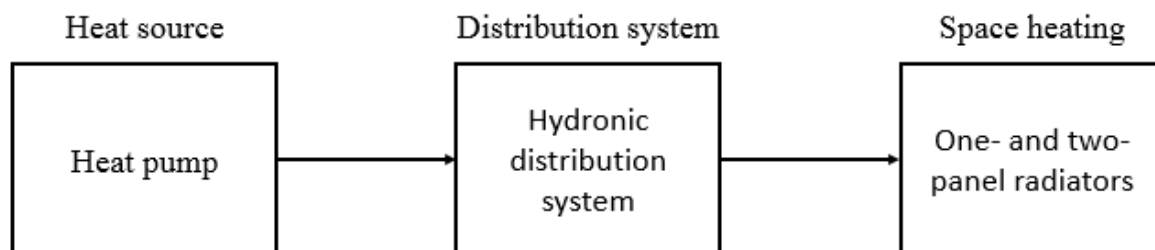


Figure 1.3: Heat supply chain referred to this experimental study (own depiction)

1.3.1 Indoor environmental conditions

There are two major associations for heating, cooling and air-conditioning. The American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc. (ASHRAE) is the most important one for the American area, while the Federation of European Heating and Air Conditioning Associations (REHVA) is the most important one for Europe. Their standards partly overlap, thus they are used as a complementary.

Since residential buildings are occupied by human beings, thermal indoor conditions should not be neglected when changing heating conditions. Both IAQ and thermal comfort are influenced by the selected heating system. They try to cover comfort and health requirements.[9] Indoor environmental conditions are divided into thermal environment, air quality, acoustics and lightning. Although the different dimensions can affect each other, they are treated separately by default. [1]

IAQ is mainly influenced by the amount of polluting substances in the air. For example, pollutants can cause undesired odours, allergies or irritations. Individuals perceive and react to pollutants and thermal conditions differently. Thus, IAQ is mainly measured by the percentage of persons who consider a specific environmental parameter as unacceptable. The fewer dissatisfied people, the higher the IAQ. Residential buildings are not the only locations that need to consider IAQ. A

hospital, for instance, is dependent on a high IAQ, since some pollutants can cause diseases or inflammation after surgeries. Consequently, IAQ comprises the following aspects:

- Perception of the air by humans,
- Effects of pollutants on health and, for example in an office, productivity
- Effects on products or processes (i.e. hospitals, cleanrooms). [9]

Thermal environment describes the aspects that affect heat exchange between the human body and the environment. Here, the thermal comfort is taken into consideration. It is a complex issue and can be measured in several different ways. Factors, influencing the thermal comfort, are the following:

- Surrounding air temperature,
- Humidity of surrounding air,
- Relative velocity of surrounding air,
- Temperatures of surrounding surface that, hence, can exchange radiation,
- Metabolic rate of the individual
- Clothing insulation (clo) of the individual.

One approach for determining acceptable thermal conditions in occupied spaces is the operative temperature. [1] More complex models take into account the humidity, air speed, metabolic rate and the clothing insulation [10]. However, none of these aspects were measured in this study. In addition, for air velocities lower than 0.2 m/s a simplification can be made. Therefore, only mean surface and ambient air temperatures $\bar{\theta}_{Surface}$ and θ_{air} are taken into consideration.

Depending on clothing insulation, humidity and humidity ratio, ASHRAE provides an acceptable range for operative temperature. Since measurements have been taken place in colder periods of a year, a clo value of 1.0 for clothing insulation is assumed. A standard provides figures for this item. As an example, typical winter clothes, such as trousers, a long-sleeve sweater and a T-shirt, lead to a clo value of 1.01, what depicts a value in the higher range. This value increases with thicker clothing. Based on this value, a range between 21.5 and 26.5 °C is considered as appropriate for the operative temperature θ_{op} in this study. [10]

The vertical temperature gradient shall not exceed certain limits. Lower gradients imply a higher thermal comfort. According to REHVA, the vertical temperature difference should stay below 3°C per meter, in the best case below 2°C.

Finally, floor surface temperature was considered. Compared to other surface temperatures, it has a major importance since occupants have direct contact with the floor. Recommended floor temperatures depend on if people wear shoes or are bare feet. Figure 1.4 illustrates the percentage of dissatisfied people wearing normal indoor shoes dependent on the floor temperature. It was the baseline for assessing the impact of radiator boosters on the floor temperature. [1]

In conclusion it can be said that the thermal comfort focuses on temperatures and temperature gradients and thus on thermal perception of humans, whereas the IAQ takes into account the cleanliness and pollution of the air, representing health aspects. Since the object of measurements is an ordinary residential building, health aspects play a minor role in this experimental study and indoor air quality is not considered. In contrast, thermal comfort will be a part of the investigation.

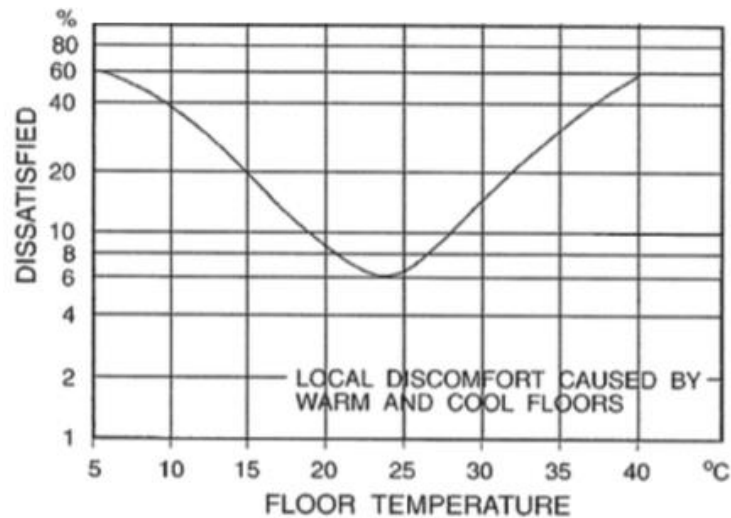


Figure 1.4: Percentage of dissatisfied people dependent on floor temperature [1]

1.3.2 The principles of heat transfer

Heat transfer is the transit of thermal energy. It always occurs whenever there is a temperature difference. It can occur in three different types:

- Conduction is the heat transfer in a stationary medium. It can happen within both a fluid or a solid.
- Convection is described as the heat transfer from a surface to a moving fluid. Among others, it depends on the surface geometry, the nature of the fluid motion and the fluid thermodynamic properties.
- In contrast to the first two types of heat transfer, radiation does not require a material medium. Energy is emitted by matter of electromagnetic waves, that also can go through vacuum. The sun depicts the most important and popular example for radiation. [11]

All three types of heat transfer happen during heating. When the hot water flows through the radiator, it gives its heat to the metal panel by convection. The heat is conducted through the metal panel by conduction, which then transfers the heat to the surrounding air by radiation and convection. The amount of transferred heat is mainly determined by the thermal conductivity k , which is a material value. Good insulators have a low value of thermal conductivity k .

Convection and radiation take place between ordinary radiators and the surrounding air. Since the objective of this experimental study is to examine the impacts of the deployment of radiator boosters, the convective part of heat transfer needs to be focused on. Convection can be separated into two different parts: Free, or natural convection and forced convection. While the former happens due to buoyancy forces, that are induced by temperature differences within the fluid that cause density variations, the latter is applied when external means are responsible for the flow of the fluid.

Formula (1.1) describes the heat transfer rate for both cases:

$$\dot{Q} = h * A * (\theta_s - \theta_f) \quad (1.1)$$

\dot{Q}	Heat transfer rate, W
h	Convective heat transfer coefficient, $\frac{W}{m^2 * K}$
A	Area of the surface, m^2
$(\theta_s - \theta_f)$	Temperature difference between surface and fluid, K

The direction of the heat flux is dependent on the temperatures: With $\theta_s > \theta_f$, heat is transferred from the surface to the fluid. With $\theta_s < \theta_f$, heat is transferred from the fluid to the surface. Typical values for the convective heat transfer coefficient for gases are $2-25 \frac{W}{m^2 * K}$ and $25-250 \frac{W}{m^2 * K}$ for natural and forced convection respectively. Consequently, forced convection implies a higher heat exchange when switching from natural to forced convection.

Figure 1.5 illustrates the development of the boundary layer on a heated vertical plate. [11] Here, θ_s equals T_s and θ_f equals T_f . Due to heat exchange, air close to the radiator has an increased temperature and a decreased density compared to air that is further away from the radiator. As a result, buoyancy forces make the heated air rise vertically along the radiator. Air's particle velocity is assumed to be zero when they make contact with the surface. Far away from the radiator the air's velocity is also zero. Consequently, the air's velocity is zero for $y = 0$ as well as for $y \rightarrow \infty$ and it is maximal in the middle area of the boundary layer. [11] The heat transfer coefficient is not a fixed parameter. It is dependent on the type of the radiator's surface, the air velocity, in case of forced convection, and the temperature difference between surrounding air and the surface for natural convection [1].

In case of combined types of heat transfer, an overall heat transfer coefficient U is applied. For convection and conduction, it is dependent on the thermal conductivity k and the convective heat transfer coefficient h (formula (1.2)). U increases if both of them increase as well. [11]

$$U = \frac{1}{\frac{L}{k} + \frac{1}{h}} \quad (1.2)$$

U	Overall heat transfer coefficient, $\frac{W}{m^2 * K}$
k	Thermal conductivity, $\frac{W}{m * K}$

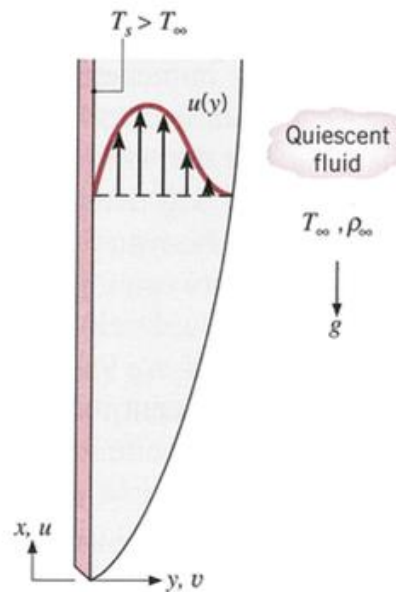


Figure 1.5: Boundary layer development on a heated vertical plate [11]

1.3.3 The principles of a heat pump

There are many possibilities to heat a building. One can choose from traditional heating systems, such as oil and gas heating systems, to more recent and advanced systems. One example is biomass. Although the need for changing the energy source was made clear in the introduction, still 81 % of the total energy use on the world is supplied by fossil fuels. However, the share of oil products used in the residential and service sector in Sweden has been steadily decreasing by 70 %, compared to 1990. Opposing to the development of oil use is the development of heat pumps. In 2013, the amount of one- or two-dwelling buildings having installed a heat pump in Sweden amounted to nearly 1 million [3].

Since the object of measurement has installed a heat pump as well, it is important to have a closer look on the functional principle of the classic heat pump and its benefits. A heat pump is a system, that moves energy from a low-temperature source to a higher temperature sink. In order to accomplish this, it is necessary to perform mechanical work. For this purpose, a compressor is integrated in the system. Heat pumps can be divided into three different types (heat source - sink):

- Water-water
- Air-water
- Air-Air. [12]

In this case, a water-water heat pump is deployed. Ground-water is used as a heat source and transferred to a heat carrier medium, the brine of the heat pump. The brine then transfers its heat to the water in the heating system. By flowing through the radiators, they are heated up and exchange

heat with the surrounding air by convection and radiation. The functional principle of a heat pump is illustrated by Figure 1.6:

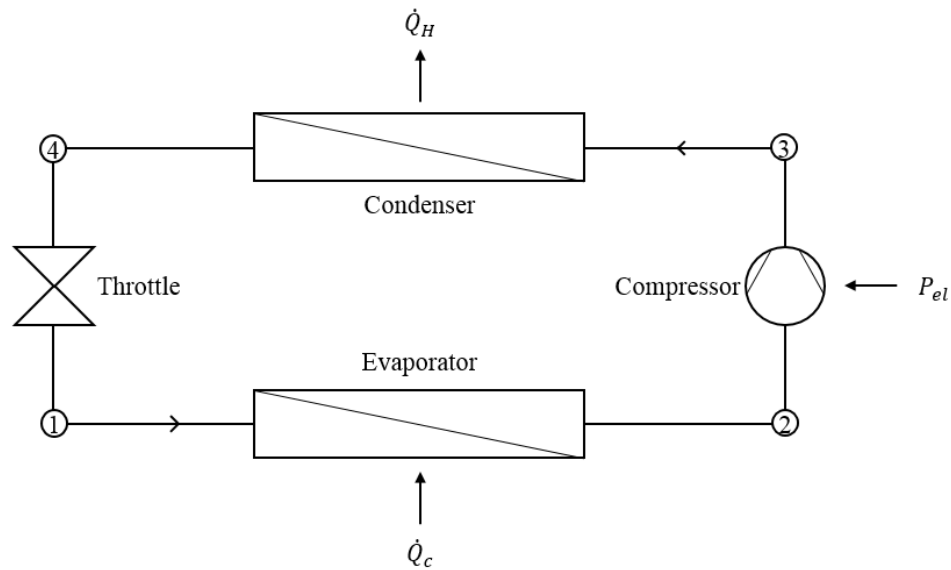


Figure 1.6: Own depiction of heat pump cycle

The explanation of the cycle starts after the throttle (1). In this stage, the refrigerant is liquid and relaxed. Within the evaporator the refrigerant is heated up by the heat source (\dot{Q}_c). Consequently, the temperature rises and the refrigerant begins to boil. When leaving the evaporator, the phase change is completed, the refrigerant is hot and it is fully gaseous (2). The compressor then performs mechanical work (P_{el}) in order to increase the pressure. Thus the temperature rises again.

In stage (3) the refrigerant is gaseous, overheated and has a high pressure. The condenser is connected to the energy sink. In this case, water in the heating cycle is the heat sink. Since heat is exchanged with the colder medium (\dot{Q}_H), the refrigerant turns liquid and cools down. However, it is still exposed to high pressure (4). The throttle is the counterpart to the compressor and releases pressure. As a result, the temperature decreases as well and the refrigerant is back to the condition in stage (1). The refrigerant is back to the condition in stage (1). The cycle will start anew. [13]

In order to determine the efficiency of a heat pump, the coefficient of performance calculates the relation between the provided heating/cooling power and the required input. Figure 1.7 shows the energy flow of a heat pump. As can be seen, the amount of released energy in form of the heat flow

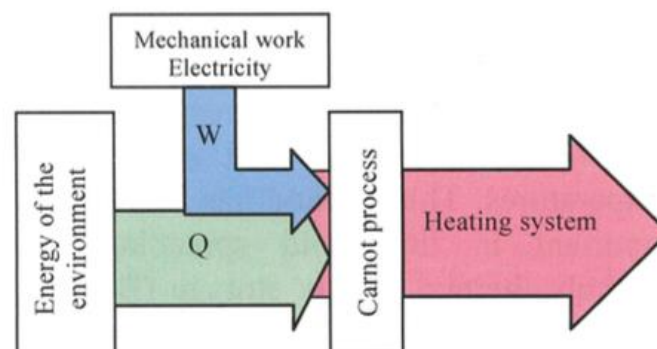


Figure 1.7: Energy flow chart of a basic heat pump based on the "Carnot-process" [1]

\dot{Q}_H is greater than power P_{el} put into the system by the heat pump's compressor. A good relation of those values is about 4:1 and can be retrieved in the COP. Here, the COP for heating is considered (formula (1.3)) [13]:

$$COP = \frac{Output}{Input} = \frac{\dot{Q}_H}{P_{el}} \quad (1.3)$$

COP Coefficient of performance of a heat pump

\dot{Q}_H Heat transfer rate, provided to the heating system, W

P_{el} Power, provided by the compressor, W

The COP considers losses in the process. In contrast, the Carnot-process is a theoretical, perfect process, without any losses. Its formula (1.4) shows, that it increases with a lower temperature difference between heat sink and source. This leads to the Carnot efficiency η_c (formula (1.5)), that sets up a relationship between the theoretical COP_c and the actual COP. Common values for the Carnot-efficiency η_c lie between 0.4 and 0.6, considering all losses. [13]

$$COP_c = \frac{\theta_c}{\theta_c - \theta_e} \quad (1.4)$$

$$\eta_c = \frac{COP}{COP_c} \quad (1.5)$$

θ_c Temperature in condenser, K

θ_e Temperature in evaporator, K

η_c Carnot efficiency

A higher evaporator temperature θ_e implies a higher temperature in the heat source. As an example, solar radiation heats up the soil. Consequently, a heat pump's efficiency is always dependent on environmental circumstances. As a results, a heat pump works less efficient in January than in March and is dependent on the seasons. The COP can not only be improved by increasing the temperature in the evaporator. Another possibility to improve the COP and hence decrease energy use, is to decrease the condenser temperature θ_c . This implies a lower hydronic supply temperature of the heat pump to the radiators. This can be achieved through LTH.

1.3.4 The energy balance of a building

For assessing the obtained results, it is important to understand the energy balance of a building. In the winter, a building emits heat to the environment. This heat loss must be compensated by the heat source. The heat loss differs for every building. But there are more aspects than only the heat loss when it comes to the energy balance. Three different parts determine the energy balance. They can be seen in Figure 1.8 (A, B and C).

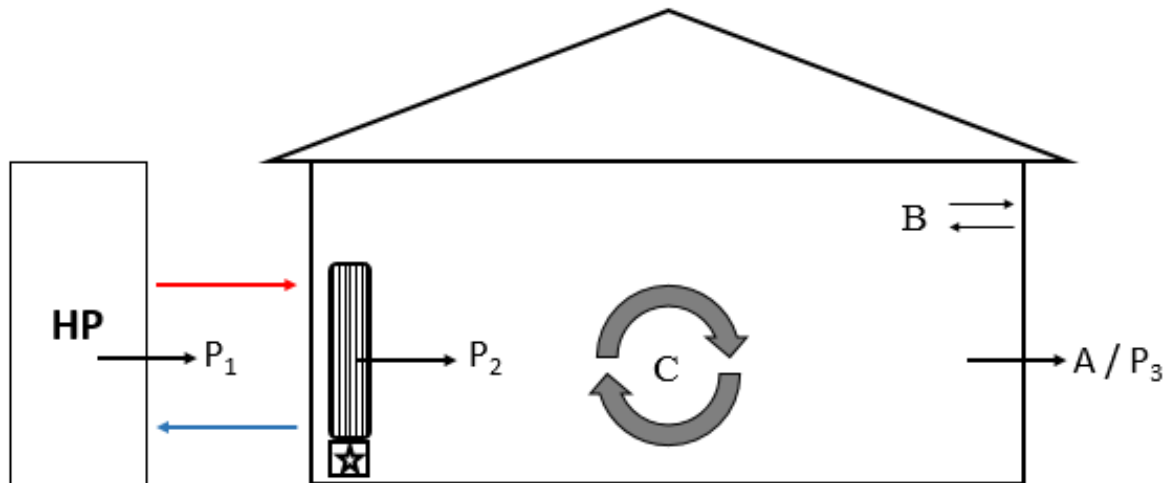


Figure 1.8: Own depiction of the energy balance of a building

A: The transport of heat through the envelope of the building: This happens through transmission and infiltration of air through the wall, roof and windows. Every element has its own share in the heat loss. Transmission is determined by the elements' different conductivity and their surface areas. The conductivity of an element is expressed by means of the "U-value". It can be reduced by better insulation of walls and windows. The heat loss through the windows is usually significantly higher than through the walls. Transmission and infiltration heat losses can be calculated by using the following formulas (1.6) and (1.7) [8], respectively:

$$\dot{Q}_{\text{transmission}} = \sum U_i * A_i * (\theta_{\text{indoor}} - \theta_{\text{outdoor}}) \quad (1.6)$$

$$\dot{Q}_{\text{infiltration}} = \dot{V}_i * \rho * c_p * (\theta_{\text{indoor}} - \theta_{\text{outdoor}}) \quad (1.7)$$

θ_{indoor} Inside air temperature, K

θ_{outdoor} Outside air temperature, K

\dot{V}_i Volumetric flow through an element, $\frac{m^3}{s}$

ρ Density, $\frac{kg}{m^3}$

c_p Specific heat capacity of the air, $\frac{J}{kg * K}$

B: The storage of heat in the envelope of the building: The envelope of a building stores energy. The amount depends on the material and its specific heat capacity. Solar radiation heats it up. As soon as there is a temperature difference between envelope and indoor air, heat exchange will occur. If the indoor air temperature decreases, the envelope will emit heat to the room. If the indoor air temperature increases, the envelope will store heat from the room. Depending on the properties of the material of the envelope, this can have a delaying effect on the heat balance of the building. The so-called time constant of a building characterizes this effect. Indoor air climate is not immediately changed when outdoor temperatures decrease and vice versa.

C: The internal generation of heat: Occupants, lightning and other electrical equipment emit heat to the room. In addition, solar radiation is absorbed by walls and the floor, which emit the heat to the room. As a result, the internal generation of heat is highly dependent on the occupants' behaviour, the daytime and the season. Thus, it cannot be considered constant and will not play a role in calculations. [8]

The energy balance is important for understanding the consequences of the deployment of radiator boosters. Using the energy balance, the results of this experimental study can theoretically be deduced. The building is equipped with a ground source heat pump. It was manufactured by the company "IVT Värmepumpar", based in Sweden. The model is "Premium Line HQ". The functional principle of a heat pump is explained in chapter 1.3.3.

The heat pump (abbreviated with HP in Figure 1.8) provides hot water to the radiator. The amount of provided power is labelled as P_1 . It is dependent on the hydronic mass flow \dot{m}_H , the specific heat capacity of water c_p and the temperature difference of supply and return temperature $\theta_{supply} - \theta_{return}$, indicated through the red and blue arrows, respectively. Finally, following formula (1.8) for P_1 is obtained [11]:

$$P_1 = \dot{m}_H * c_p * (\theta_{supply} - \theta_{return}) \quad (1.8)$$

P Power, W

\dot{m}_H Hydronic mass flow, $\frac{kg}{s}$

θ_{supply} Temperature of supply water, °C

θ_{return} Temperature of return water, °C

The provided hot water heats up the radiator. Due to the heat transfer, the temperature decreases from the water inlet to the outlet. P_2 describes the heat that is then exchanged between radiator and the surrounding air (formula (1.9)) [14]:

$$P_2 = U * A * \Delta\bar{\theta}_m \quad (1.9)$$

$\Delta\bar{\theta}_m$ Logarithmic mean temperature difference between radiator and surrounding air, °C

The logarithmic mean temperature difference is calculated by using the formula (2.4) in chapter 2.4. In order to remain a constant indoor air temperature, the heat pump has to compensate the heat loss. This leads to following formula for the heat losses, described with P_3 (formulas (1.10), (1.11) and (1.12) [8]:

$$P_3 = \dot{Q}_{transmission} + \dot{Q}_{infiltration} \quad (1.10)$$

$$P_3 = \sum U_i * A_i * (\theta_{inside} - \theta_{outside}) + \dot{V}_i * \rho * c_p * (\theta_{inside} - \theta_{outside}) \quad (1.11)$$

It is obvious that a lower outside air temperature $\theta_{outside}$ requires a higher compensation, since the heat loss P_3 increases. Assuming the outdoor air temperature to be constant and neglecting losses in pipes and other process steps, following assumption for a constant indoor air temperature θ_{inside} can be made:

$$P_1 = P_2 = P_3 = constant \quad (1.12)$$

As discussed in chapter 1.3.2, forced convection increases the convective heat transfer coefficient h and thus the U -value. Since the radiator's surface area A is constant, the logarithmic mean temperature difference $\Delta\bar{\theta}_m$ should decrease in order to keep P_2 constant. A lower logarithmic mean temperature difference would also imply a decrease in the difference between the hydronic supply and return temperature $\theta_{supply} - \theta_{return}$. As a consequence, in order to keep P_1 constant, the hydronic mass flow is estimated to increase since the specific heat capacity of water obviously cannot be changed manually. Table 1.2 lists all expected results.

Table 1.2: Theoretically expected alteration of energy performance influencing parameters

Value	Expected alteration
h , and therefore U	↑
$\Delta\bar{\theta}_m$	↓
$\theta_{supply} - \theta_{return}$	↓
\dot{m}_H	↑

1.4 State of the art

As seen in the previous chapters, the heating system has to cover the heat losses, that mainly happen through transmission and air infiltration, in order to remain a steady temperature in a building or room. In terms of energy, the input must equal the output. One target of the radiator booster is to reduce the energy use. Therefore, the heating system must work more efficiently since the heat loss cannot be influenced. For this purpose, one of the three parts of the heat supply chain (Figure 1.13, p. 17) must be attacked. So either the heat source, the distribution system or space heating can be improved. In this overview, space heating shall be focused on. The heat exchange between radiator and surrounding air follows formula (1.13):

$$\dot{Q} = U * A * \Delta\bar{\theta}_m \quad (1.13)$$

With a constant outdoor air temperature, the heat loss does not change. Consequently, the heat pump has to provide a constant amount of heat and \dot{Q} stays constant. Increasing the logarithmic mean temperature difference $\Delta\bar{\theta}_m$ along the radiator implies a higher supply temperature. This is not desired due to an increasing compressor energy use and is opposing LTH. As a result, either the radiator surface area A or the overall heat transfer coefficient U can be increased in order to achieve previously mentioned target. In this way, the supply temperature could be reduced as well.

The following presented studies dealt with increasing the radiators efficiency in order to provide the same thermal comfort and reduce the energy use. For baseboard radiators and floor heating the radiators surface area A was increased. Ventilation radiators and radiator booster attacked the convective heat transfer coefficient by applying forced convection.

1.4.1 Embedded surface heating

For this type of heating, the heating elements are embedded in a surface of the building. The “radiators” are embedded in either the floor, the wall or the ceiling. One type which is increasingly used in central and northern Europe, especially in newly built buildings, is floor heating. 30 – 50 % of new residential buildings in Germany, Austria and Denmark are equipped with floor heating. In Korea it has a long tradition, hence approximately even 90 % of residential buildings are equipped with floor heating. For this reason, it shall serve as an example for this category.

By using the complete floor, a huge radiator surface area A can be used to keep the temperatures very low. This especially supports renewable energy sources and is energy saving. For all types it is important to insulate the building’s basement. [1]

When designing a floor heating system, aspects such as piping layout, pipe distance and the water flow rate must be taken into consideration. For a zone with a higher surface temperature either the pipes can be run in a shorter distance to each other or the warmer beginning of the pipe is run here firstly. Since pipes are laid under the floor surface, floor heating systems allow a more efficient use of space. However, floor coverings with a high thermal resistance should be avoided. This would require a higher supply temperature and lead to higher losses through the ground. This is instead a major benefit of floor heating.

Moreover, a sufficient operative temperature can be achieved with a lower indoor air temperature, as surface temperatures are relatively high. This leads to less ventilation losses. Ventilation losses are caused by temperature differences between indoor and outdoor air temperature. Having installed no radiators, floor heating provides heat without producing noise or draft. Moreover, a uniform temperature distribution within the room is achieved since there is no single radiator but one that covers the complete room. [15]

1.4.2 Baseboard radiators

Baseboard radiators, also referred to as skirting heaters, are a hydronic heating system. Just as ordinary radiators, they have a supply and return water pipe. But in contrast, its height lies in between 120 and 180 mm. With a length of 8 – 15 m they occupy quite a big area of the walls. Figure 1.9 illustrates its dimensions. Another version of baseboard radiators adds the supply of ventilation air directly from outdoor.

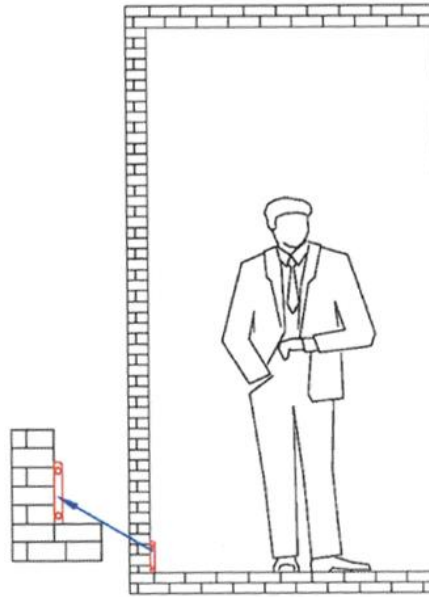


Figure 1.9: Relation between baseboard radiator and occupant in a room [19]

One of the benefits of baseboard radiators is the fact, that they are installed in the base of a room. As a consequence, they are exposed to the coldest air in the room. The higher temperature gradient leads to a higher heat transfer to the room. Its ability for heat transfer is stated to be higher by 50%, compared to conventional panel radiators. One study showed that supply temperature could be lowered by 5 °C, compared to a conventional radiator, in order to cover the same heat loss. [16]

1.4.3 Ventilation radiators

In contrast to floor heating and baseboard radiators, the ventilation radiator increases the convective heat coefficient instead of the surface area. This is attained by a difference between indoor and outdoor air temperature. Due to the resulting density and pressure difference, cold air enters the room through a vent in the wall. It is then led through a filter to the radiator, where it is heated up. Buoyancy forces lead to increased convection. Besides the higher mean air velocity, the high temperature difference between radiator and the entering cold air increases the heat output. There is also a variation that mixes the cold air from outside with warm air from inside before flowing through the radiator. In addition, this system combines ventilation air supply with heating in one unit. [17]

According to another study, ventilation radiators are able to create a more stable thermal climate. The radiator surface temperature was able to be lowered by 7.8 °C, what leads to energy and environmental savings. In contrast to the radiator boosters, this system does not lead to further electricity use. [18] Figure 1.10 illustrates the set-up:



Figure 1.10: Built-up of a ventilation radiator [21]

1.4.4 The radiator booster

A radiator booster mainly consists of direct current motor driven fans. Its purpose is to increase the convective heat coefficient h by accelerating the air flow around the radiator. Within this chapter the reader shall gain an insight about the functional principle of the radiator booster.

The radiator booster can be ordered in several designs. The fans are mounted on a bar in a distance of 20 cm to each other, being covered by white synthetic material (Figure 1.11). The amount of mounted fans depends on the necessary length of the bar and consequently on the radiator size. Sets are available between 30 to 305 cm. Included in this package are the bar with the cover, mounting, the fans and electricity supply (Figure 1.12). The manufacturer indicates a noise level of the direct current motor driven fans of 17,6 dB(A) when being operated with a voltage of 7.5 V. Accordingly, they would then consume electricity of approximately 0.4 W per fan. The voltage can be controlled in steps and so the speed of the fans. In this study the voltage was constantly kept at 7.5 V.



Figure 1.11: Five fans mounted on the cover [19]

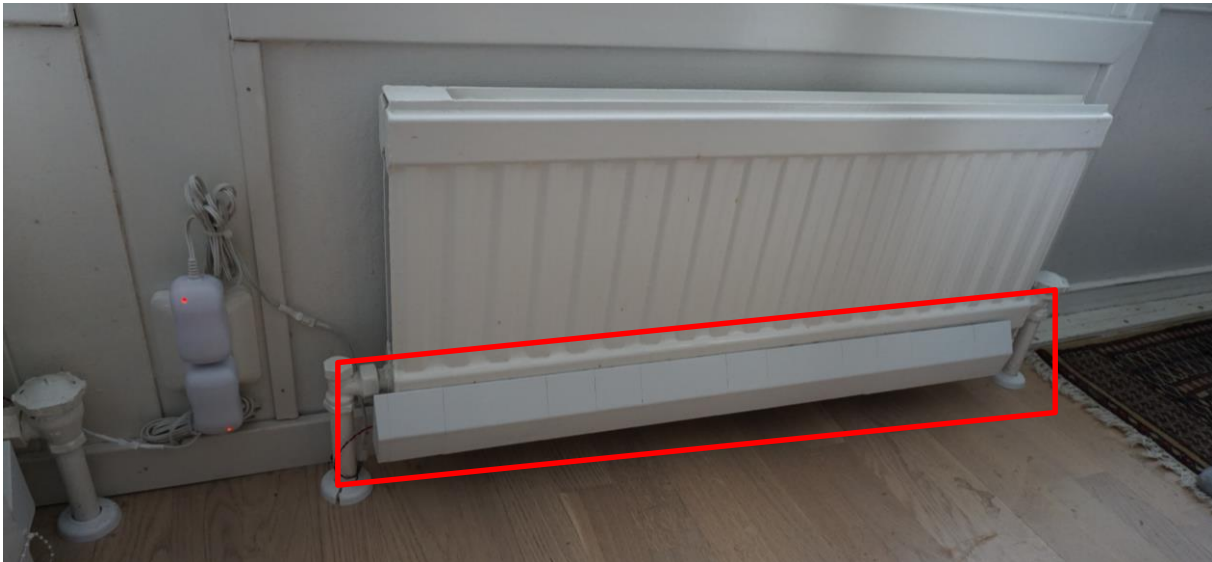


Figure 1.12: Mounted radiator booster (in the red frame) in operation mode in object of measurement

The fans draw cold air from below the radiator and force it to pass the radiators' hot panels. As a result, the air is heated up. The convection changes from natural to forced. Both principles were explained in chapter 1.3.2. As a result, the heat exchange from the radiator to the air increases. Warm air rises to the ceiling. Thereby it pushes the "old" air away and down to the floor. Since the air flow is accelerated, the air movement and thus the warm air cover a larger volume. [19] Air flows with and without deployment of radiator booster are shown in Figure 1.13:

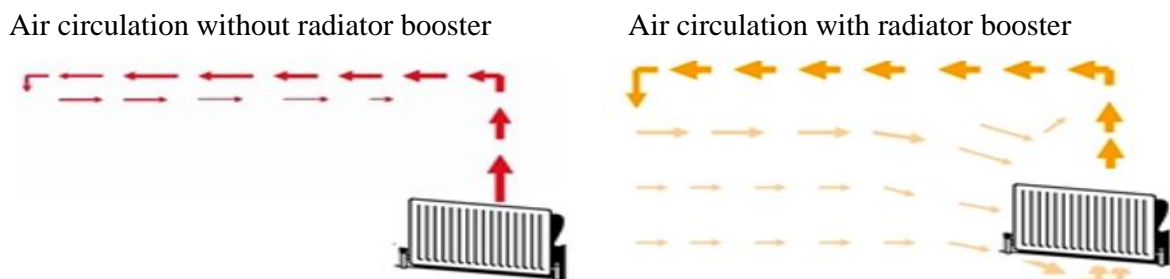


Figure 1.13: Air circulation with and without deployment of radiator booster [19]

A measurement, that addresses the temperature increase in dependency on the time within a room was conducted by the manufacturer of provided radiator booster, by the company "A-Energi AB" [19]. Therefore, two similar offices were chosen, but only one was equipped with a set of radiator boosters. Both offices were cooled down to approximately 12.5 °C before radiators and radiator boosters were switched on. Subsequently, the temperature was measured within the rooms. Measurements took place for 24 hours (Figure 1.14).

The office, being equipped with a radiator booster, not only heats up significantly faster. It also reaches a significant higher final room temperature of approximately 26 °C. In comparison, the office without a radiator booster reaches only about 18.5 °C after 24 hours. The considerably greater temperature rise manifests itself in the fact, that room temperatures were increased by approximately 6.0 and 1.7 °C after 1.2 h with and without radiator boosters, respectively.

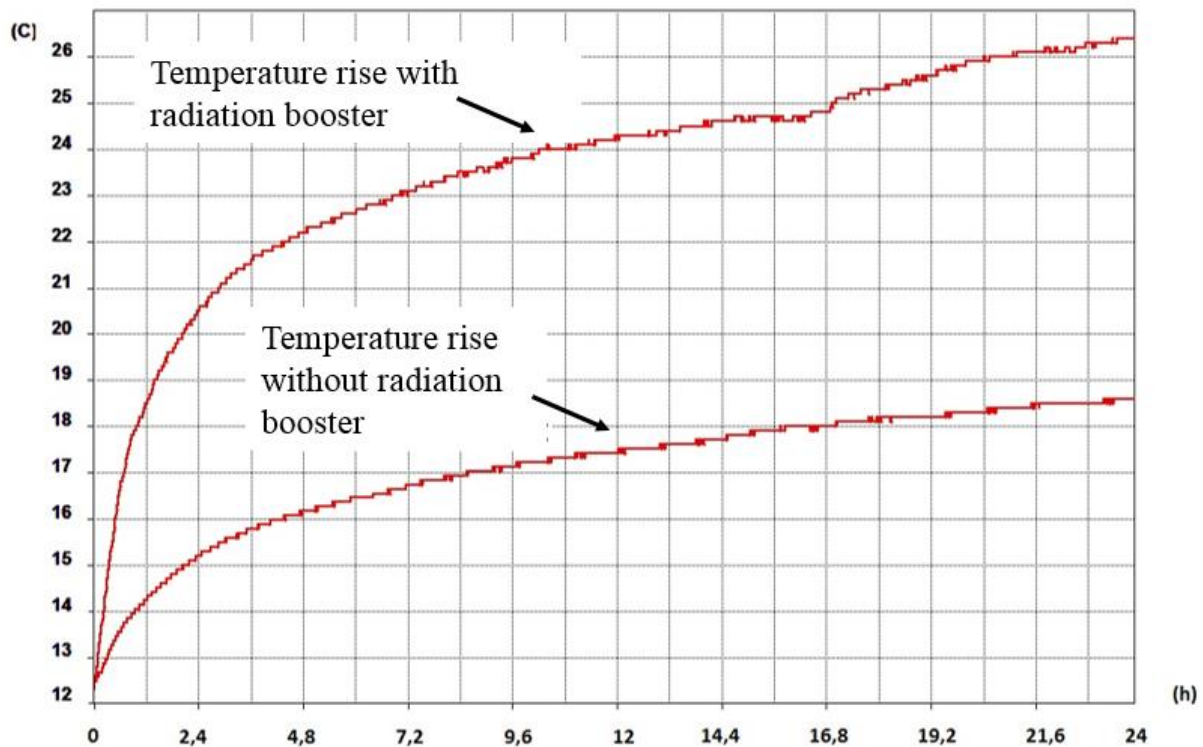


Figure 1.14: Room temperature development of two offices, only one being equipped with a radiator booster [19]

The radiator boosters have not been studied extensively so far. Only one study could be found. In an office in Lund, Sweden, built in 1960, experimental studies have been conducted by Johansson. In a pre-set interval, the “add-on-fan blowers” were switched on and off in time intervals of one hour. During this time, both mass flow and supply water temperature was not changed but kept constant. When applying the add-on-fan blowers, the return temperature was lowered by 5 °C. It dropped from 39 to 34 °C. This higher temperature drop led to an increased heat output of more than 60 %. [20]

Comparing this study and the present one, some differences can be distinguished. Inputs, such as supply temperature and mass flow, were fixed during the measurements. Outputs, such as the return temperature, were open and unlimited. Consequently, the possible potential was not limited and probably has led to an overheated office. In contrast, the present experimental study has a fixed output (target indoor air temperature, that limits that radiators heat output) and measures the required input in form of supply and return water temperature and compressor energy use, what might be a limitation for the possible potential of the radiator booster.

Following additional differences can be found: There is no statement about the amount of fans mounted on the radiators and their distance to each other. Indoor air temperature was not measured, thus impacts on thermal comfort are not known.

The study indicates an increase in the heat output of 60 %. While the supply temperature was kept constant, the return temperature dropped. The increased gradient causes the rise in the heat output. However, energy use of the heat source is not known. It is assumed that the heat source needs to work more in order to keep the supply temperature constant. This fact is not considered in this study.

Fans were driven with a voltage of 12 V. In contrast, in this experimental study the radiator boosters were driven with 7.5 V. Consequently, the fans did not provide the same power. In other words,

radiator boosters have a lower impact on the heat output when being operated with a lower voltage. Finally, there is only one radiator in the room and it is not known, whether the room was occupied within the period of measurement.

1.4.5 Comparison

Hesaraki et. al compared the energetic performance of conventional radiators, ventilation radiators and floor heating in a climate chamber [21]. It was run with the mean outdoor air temperature in Copenhagen of 5 °C. The results showed a necessary supply water temperature of 30°C for floor heating, 33 °C for ventilation radiators and 45 °C for the conventional radiators in order to cover the simulated heat losses. These results led to a reduced energy use by 17 and 22 % for ventilation radiators and floor heating, respectively.

This study confirms the statements about low-temperature heating in the introduction. Moreover, it underlines the efficiency of floor heating and ventilation radiators compared to conventional radiators. But also baseboard radiators have positive impacts, as previously presented studies show. However, one disadvantage, especially compared to floor heating, is the blockage of usable space. Table 1.3 summarizes the mentioned products:

Table 1.3: Overview of different kind of innovations being elaborated in order to achieve a more efficient space heating

Product	Increased value
Embedded surface heating	Surface area A
Baseboard radiators	
Ventilation radiators	Convective heat coefficient h
Radiator boosters	

The first three different types of LTH systems have in common that they are not able to be retrofitted easily but come along with a high effort. In contrast, radiator boosters are installed in a short period of time. Radiator boosters are relatively newly developed. Consequently, there is not a lot of previously conducted studies about them. Due to this reason, it is necessary to examine their impact on a heating system and their possible contribution to LTH.

1.5 Objectives

The following hypothesis shall be reassessed in this thesis:

“Deployment of existing radiators with radiator boosters, connected with a heat pump, theoretically will increase the heat output of radiators. Since heating demand cannot be influenced, supply temperature can be lowered by implication. This will lead to energy savings. Due to the deployment, warm air will be distributed more equally within the room and thus thermal comfort will be improved.”

For this purpose, an experimental study in a single-family house in the northern suburban area of Stockholm was conducted in March and April 2016. The building from the 1960s is equipped with radiator boosters on the ground floor on a total area of 88 m². Radiator boosters were switched on and off alternating with a duration of one week in order to balance the increase of outdoor temperature.

Benefits in terms of thermal comfort are mainly expected to be mirrored in an improved operative temperature. In addition, floor surface temperature and vertical temperature gradient are taken into account when evaluating the benefits of radiator boosters.

Main performance indicator for energy performance are the coefficient of performance (COP) of the heat pump and the net energy savings. Additional energy use, caused by the fans, are expected to be compensated by the decreased energy use of the compressor, that is induced by the radiator boosters. This circumstance would lead to positive net energy savings.

Finally, improved energy performance and thermal comfort are the baseline for further calculation of possible energy and cost savings with radiator boosters.

2 Methodology

Within this chapter, the procedure, the set-up of the measurements as well as the circumstances will be explained in detail. After having read this part of the thesis, the reader should have a clear idea about the proceeding. Important information about the building (chapter 2.1), the control system of the heat pump (chapter 2.2) and the procedure (chapter 2.3), that all influence the measurements results, will be outlined. After presenting the used equations and calculations (chapter 2.4), measurement devices will be introduced (chapter 2.5) and then their calibration presented (chapter 2.6). The final chapter will deal with the results that were expected before the measurements were conducted (chapter 2.7).

2.1 The building and its properties

The object of measurement is located in the northern suburban area of Stockholm. It is a 2-storey single family house. It was constructed in the 1960s with brick foundations without basement (Figure 2.1b). On the ground-floor the living room, dining room as well as the kitchen are combined and heated by ordinary hydronic radiators. In addition, there are two bedrooms that are equipped with hydronic radiators as well. The rest of the ground-floor, consisting of entrance area, bathroom and laundry room, is provided with heat by floor-heating. Consequently, this area was not considered in this experimental study. The ground plan of the ground floor can be seen in Figure 2.1a. Grey areas are equipped with floor heating. The white area is equipped with hydronic radiators. This area amounts to 88 m².

There is no ventilation system installed in the building. Thus, only natural ventilation occurs. The kitchen hood is only switched on when cooking. Other devices in the area of interest that emit heat are TV, fireplace, oven, stove and lighting. According to occupants the fireplace was not used during the measurements. Since this arrangement does not change between the periods, there is no further need to consider their contribution. Their contribution to internal heat gains is assumed to be balanced over the measurement periods.

The first floor is equipped with electrical radiators, being only used for guests and storage occasionally. Thus, it was not included into the measurements. In total there were nine hydronic radiators installed, with two of them being single-panel radiators and seven of them being double-panel radiators. The build-up of both arrangements can be seen in Figure 2.2a and Figure 2.2b. Two of the nine radiators were long enough to be equipped with two but one radiator booster bars. As a result, eleven bars were installed. In total, 48 fans are deployed.

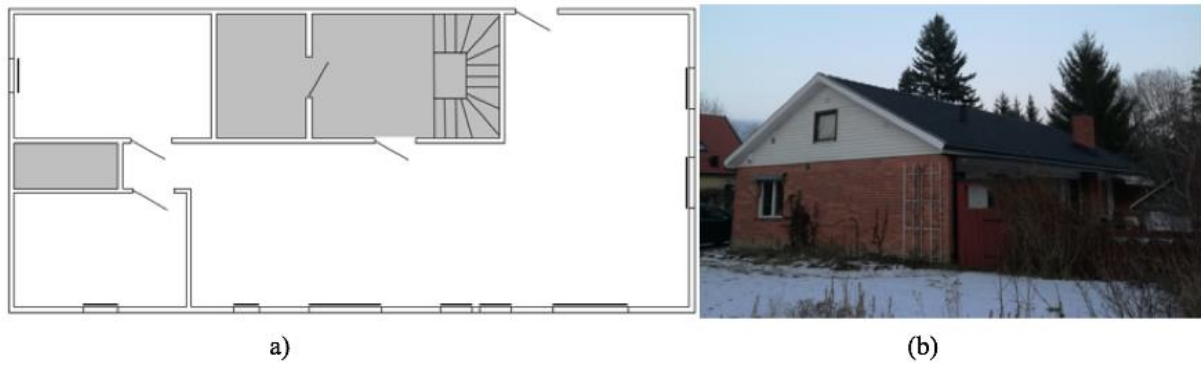


Figure 2.1: a) Ground plan of ground floor of object of measurement and (b) the selected single-family house

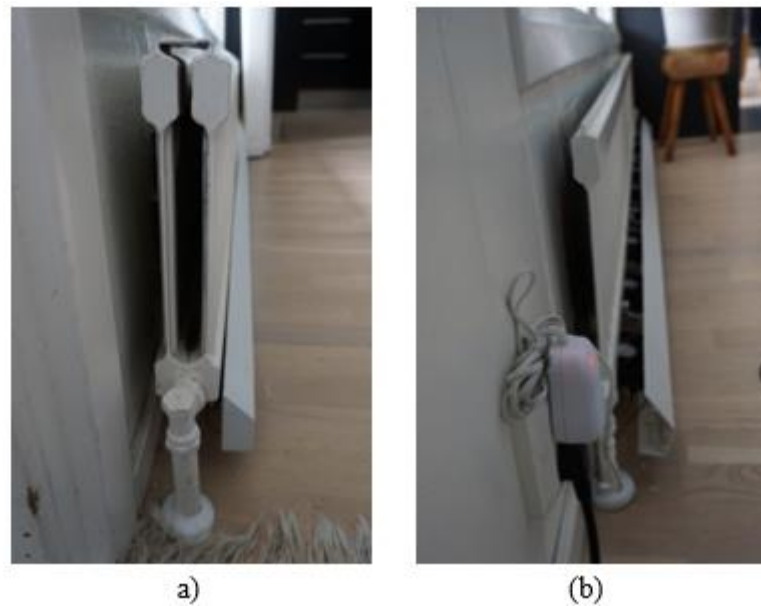


Figure 2.2: a) Double-panel radiator with radiator booster being installed and b) single-panel radiator with radiator booster being installed and connected to time switch

The radiators differed in height and length. Table 2.1 lists all radiators and its sizes. Being connected in line leads to an uneven distribution of hydronic supply water from the heat pump. Thus, the radiators and the results were considered as one system and single contributions of a radiator were not taken into consideration. The target of this approach is to depict a realistic scenario, as buildings mostly differ in radiator size, type of radiator and radiators per area that needs to be heated. It is assumed that the customer is more interested in the final result, namely the total energy savings as well as associated cost savings. In addition, it is expected to be unlikely to equip single rooms with a radiator booster instead of equipping a complete building.

Table 2.1: List of all types of radiators

#	Location	Height [mm]	Length [mm]	Area [m ²]
1	Dining- and Livingroom, Kitchen	580	1040	0.60
2		580	1270	0.74
3		284	1905	0.54
4		284	800	0.23
5		284	800	0.23
6		284	1820	0.52
7		580	635	0.37
8	2 x Bedroom	580	880	0.51
9		580	880	0.51

2.2 The heat pump's control system

The functional principle was already explained in chapter 1.3.3. Here, it will only be focused on the heat pumps control system, as it cannot be assumed that all heat pumps work in the same manner.

A heat pump is controlled by means of a heating curve: An example for such a heating curve can be seen in Figure 2.3b. A linear graph indicates the heat pumps set supply temperature in dependency on the outdoor temperature. The resulting point on the graph for a specific outdoor temperature is referred to as “set point”. The lower the outdoor temperature, the higher the heat losses and hence the higher necessary the supply temperature. There are several heating curves programmed in the heat pump. Depending on his preferences, the customer can choose one heating curve. The heating curve remained the same during the measurements.

A thermostat that measures the indoor air temperature is optional and can be retrofitted. If doing so, the indoor air temperature can influence the supply temperature as well. At the thermostat, the desired indoor air temperature can be set. It will be referred to as set temperature in the following. In addition, the customer has to determine an “influence factor”. This factor can be set from 0 to 10 and indicates to what extent the actual indoor air temperature shall be taken into account when calculating the supply temperature. This influence factor is set high if occupants want the building to be heated up faster.

Following example shall illustrate the functional principle for the used heat pump, that is equipped with an indoor thermostat. In order to compute the necessary supply temperature, the outdoor temperature is measured and a set point is obtained. For instance, an outdoor temperature of $-10\text{ }^{\circ}\text{C}$ results in a supply temperature of $40\text{ }^{\circ}\text{C}$. Now the difference between actual indoor air temperature and set temperature is calculated. Assuming a desired indoor temperature of $20\text{ }^{\circ}\text{C}$ and an actual indoor temperature of $18\text{ }^{\circ}\text{C}$, the temperature difference is $2\text{ }^{\circ}\text{C}$. It is then multiplied with the influence factor and added to the previously obtained set point. In this experimental study it was set to 7. Consequently, $14\text{ }^{\circ}\text{C}$ are added to the set point and the new supply temperature is $54\text{ }^{\circ}\text{C}$.

A high influence especially helps with heating up the building in a fast way. Moreover, it can be used to balance and compensate an inadequate heating curve in order to still achieve the desired

indoor climate. However, as the new supply temperature is significantly higher than the old one, the heat pump has to perform more work and therefore uses more energy. The heat pumps control system needs to be considered when assessing the measurement results. It is not known which influence factor supports the radiator boosters the most. However, since they heat up a room faster (see chapter 1.4.4) a high influence factor seems to be redundant. Figure 2.3a illustrates the heat pumps control system. [22]

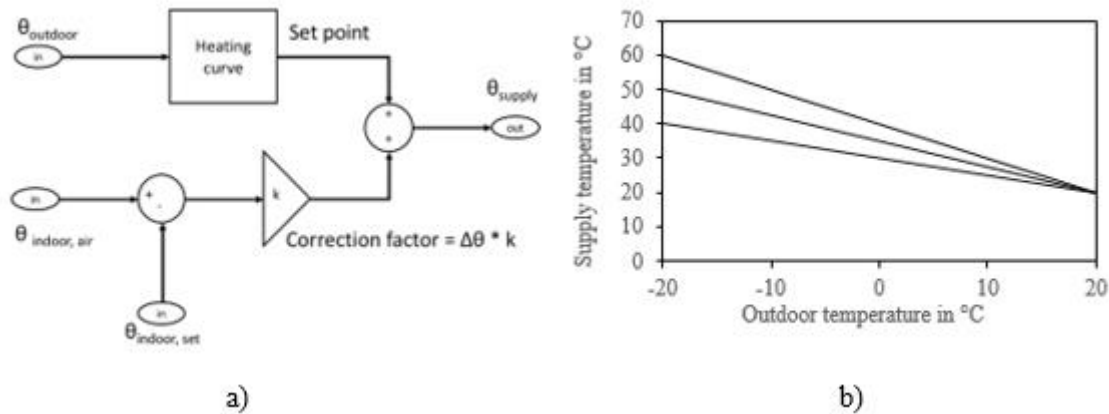


Figure 2.3: a) Own depiction of control cycle of heat pump and b) own depiction of an example for three different heating curves, based on provided manual by manufacturer “IVT Värmepumpar”

2.3 Procedure and schedule

In total, the measurements have been taking place for six consecutive weeks. Radiator boosters have been switched on and off for three weeks, respectively. In order to obtain comparable results, the radiator boosters were switched on and off alternately for one week, since the outside temperature was increasing constantly. In the best case, the outdoor air temperatures are similar in periods one and two, three and four and five and six. In the following, they are referred to as period couples. This allows a comparison of respective periods. Figure 2.4 illustrates the schedule.

The measurements have been taken place from March 9th until April 19th. Changing the set-up only once and after three weeks would not have been comparable, due to different mean temperatures. Another criterion to obtain comparable results was to keep the set temperature of the thermostat constant for the measurement periods. Otherwise, the heat pump would obviously have provided a different amount of heat and it would not be possible to determine, whether this was due to the radiator booster or a different set temperature. A measurement period of one week was considered to be suitable to cover the changing behaviours of occupants from weekdays to weekends. Moreover, slight daily variations can be absorbed and compensated.

Data is captured every ten minutes and saved in a cloud service. In total 1008 data points per period were collected. The radiator boosters can be switched on and off automatically by means of remote controlling via internet. As a consequence, there is no need to do it manually and switching on/off can happen on time. It was always switched on and off on Tuesdays, 12 pm.

Monday	Tuesday	Wednesday	Thursday	Friday	Saturday	Sunday	
7	8	9	10	11	12	13	MARCH
14	15	16	17	18	19	20	
21	22	23	24	25	26	27	
28	29	30	31				
				1	2	3	APRIL
4	5	6	7	8	9	10	
11	12	13	14	15	16	17	
18	19	20	21	22	23	24	

Holiday	
Vacation	
On	
Off	

Period 1	9.03. - 15.03.
Period 2	16.03. - 22.03.
Period 3	23.03. - 29.03.
Period 4	30.03. - 05.04.
Period 5	06.04. - 12.04.
Period 6	13.04. - 19.04.

Figure 2.4: Schedule of measurements

2.4 Calculations

This chapter deals with the calculations and formula being used in this study. Figure 2.5 shows the locations of the measurement devices on the ground floor. Wall 1 and wall 2 temperatures $\theta_{wall,1}$ and $\theta_{wall,2}$ as well as the thermostat were mounted on the wall. The thermostat measured the indoor air temperature θ_{indoor} . Their locations were influenced by the buildings arrangement of furniture. In addition, the devices are not supposed to influence occupants in the daily life. As standard the heat pump was equipped with a thermo sensor that measures the outdoor temperature $\theta_{outdoor}$. Additional data that was measured includes the hydronic mass flow \dot{m}_H , flowing from the heat pump to the radiators, and the brine pump's and compressor's energy use E_{brine} and E_{comp} . If a mean value was needed in the calculation, the arithmetic medium was chosen in all cases.

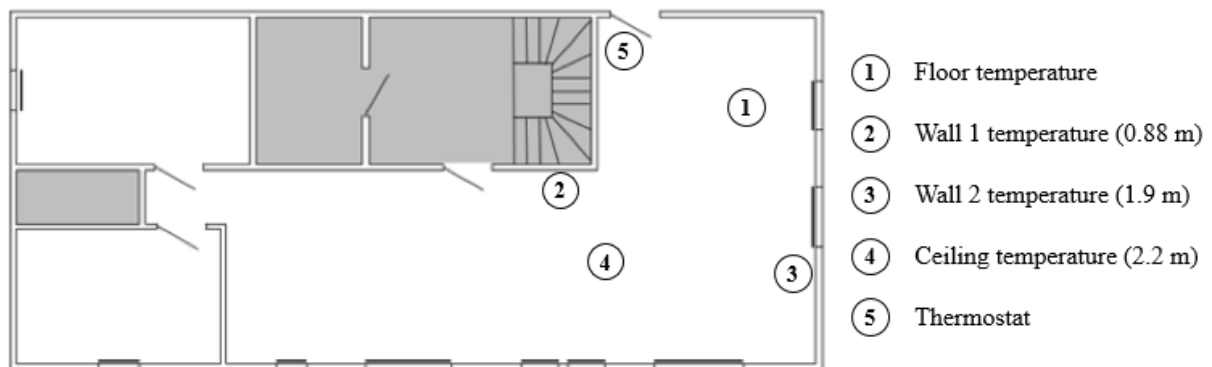


Figure 2.5: Locations of measurement devices in the object of measurement

In order to calculate the vertical temperature gradient within the building, floor and ceiling surface temperatures θ_{floor} and $\theta_{ceiling}$ were measured. The latter was measured at a height of 2.2 m. In addition, wall surface temperatures $\theta_{wall,1}$ and $\theta_{wall,2}$ were measured at a height of 0.88 m and 1.9 m, respectively. Formula (2.1) was used to calculate the vertical temperature gradient per meter:

$$\Delta\theta_{vert} = \frac{\theta_{ceiling} - \theta_{floor}}{2.2 \text{ m}} \quad (2.1)$$

$\Delta\theta_{vert}$ Vertical temperature gradient between floor and ceiling, °C

$\theta_{ceiling}$ Ceiling surface temperature, °C

θ_{floor} Floor surface temperature, °C

For operative temperature θ_{op} , mean surface as well as indoor air temperatures $\bar{\theta}_{surface}$ and θ_{indoor} were used. The former was calculated by means of formula (2.2). Formula (2.3) was used to compute operative temperature θ_{op} [10]:

$$\bar{\theta}_{surface} = \frac{\theta_{floor} + \theta_{wall,1} + \theta_{wall,2} + \theta_{ceiling}}{4} \quad (2.2)$$

$$\theta_{op} = \frac{\bar{\theta}_{surface} + \theta_{indoor}}{2} \quad (2.3)$$

$\bar{\theta}_{surface}$ Mean surface temperature, °C

θ_{wall} Wall surface temperature, °C

θ_{op} Operative temperature, °C

Mean logarithmic temperature difference $\Delta\bar{\theta}_m$ was calculated with the help of formula (2.4) [14]. Here, $\theta_{ambiente}$ is the thermostat's set temperature, namely 24.5 °C.

$$\Delta\bar{\theta}_m = \frac{\theta_{supply} - \theta_{return}}{\ln\left(\frac{\theta_{supply} - \theta_{ambiente}}{\theta_{return} - \theta_{ambiente}}\right)} \quad (2.4)$$

Hydronic supply and return temperatures θ_{supply} and θ_{return} as well as the hydronic mass flow \dot{m}_H were measured directly at the heat pump. As a result, the total heat transferred from all radiators in the room can be calculated. Using formula (2.5), the mean heat transfer rate \dot{Q}_H within a period of 10 minutes was obtained[11]:

$$\dot{Q}_H = \dot{m}_H * c_p * (\theta_{supply} - \theta_{return}) \quad (2.5)$$

c_p Specific heat capacity of water, $\frac{J}{kg \cdot K}$

The total heat output Q_{total} within a period of one week was calculated by the mean value of \dot{Q}_H multiplied with the duration T_w of 168 h (formula (2.6)):

$$Q_{total} = \bar{Q}_H * T_w \quad (2.6)$$

Q_{total} Total heat output in a period, Wh

T_w Duration of a period, h

The outdoor air temperature $\theta_{outdoor}$ was measured to relate impacts of the radiator booster to outdoor air temperature and to check, whether periods are comparable to each other. The compressors energy use E_{comp} within a period was read directly from the heat pump. In order to calculate the mean COP for a period, formula (2.7) was used [13]:

$$\overline{COP} = \frac{Output}{Input} = \frac{Q_{total}}{E_{comp}} \quad (2.7)$$

E_{comp} Compressor energy use, Wh

For calculating the total operational energy usage TOEU within a period, the energy use of the radiator boosters $E_{boosters}$ must be known. According to manufacturer's information, previously conducted measurements indicated a mean energy use of 0.432 W for a single fan. Here, the radiator booster's total energy use amounts to

$$E_{boosters} = 0.432 \text{ W} * 48 \text{ pcs} * 168 \frac{\text{h}}{\text{period}} = 3.19 \frac{\text{kWh}}{\text{period}}$$

$E_{boosters}$ Energy use of all radiator boosters together, Wh

Electricity use for the brine pump E_{brine} was read directly from the heat pump. However, compared to the compressor, its energy use was pretty small (0.01 – 0.06 % of compressor energy use), thus it was not included in this calculation. Formulas (2.8) and (2.9) are used for calculating the total operational energy usage, while $TOEU_{on}$ depicts the periods with the radiator boosters being switched on and $TOEU_{off}$ depicts the periods with the radiator booster being switched off. The circulation pump's energy use could not be measured. Thus, it is not included in this formula.

$$TOEU_{on} = E_{comp} + E_{boosters} \quad (2.8)$$

$$TOEU_{off} = E_{comp} \quad (2.9)$$

$TOEU$ Total operational energy use, Wh

An important aspect that needs to be elaborated is whether energy savings $E_{savings}$ exceeded the electricity use of the radiator boosters $E_{boosters}$. Energy savings are noticeable in a lower compressor energy use, since this is the only effort for heating. Formula (2.10) was used to calculate energy savings. Energy can be saved if $E_{savings} > 0$.

$$E_{savings} = TOEU_{off} - TOEU_{on} \quad (2.10)$$

$E_{savings}$ Energy savings between two periods, Wh

If measurement results indicate an increased operative temperature, the degree day (DD) method will be used in order to calculate additional possible energy savings. Degree days use the assumptions, that internal heat generation and solar energy will compensate the heat loss of a

building as soon as the mean daily outdoor temperature exceeds a certain temperature. This base temperature is mostly determined to 18 °C. For instance, if the indoor set temperature is 20 °C, the final 2 °C are attained by internal heat gains. In addition, fuel use is believed to be proportional to the difference between mean daily outdoor temperature and the base temperature. Energy use is proportional to the degree days and additional possible energy savings can easily be calculated. Formula (2.11) is used to calculate the degree days in dependency of a base temperature θ_{base} : [12]

$$DD = \frac{(\theta_{base} - \theta_i) * N}{24} \quad (2.11)$$

DD	Degree days, °C * d
θ_{base}	Base temperature, °C
θ_i	Actual outdoor temperature, °C
N	Number of hours the actual outdoor temperature was valid

The actual degree days for the year 2015 are obtained from “BizEE”¹, since this data are more accurate and reliable than calculating manually. The radiator boosters were expected to increase the thermal comfort, which was mainly measured by the operative temperature. In order to estimate possible energy savings, degree days will be once obtained for a base temperature of 18°C. Afterwards, the degree days will be obtained for a base temperature that is lower by the difference that the measurement results indicate. For instance, if the results indicated an operative temperature that was higher by 1 °C, the new base temperature would be 17 °C. Since degree days are assumed to be proportional to fuel use, the new energy use when deploying radiator boosters will be then calculated by using the formula (2.12):

$$E_{new} = \frac{\text{Degree days (17 °C)}}{\text{Degree days (18 °C)}} * E_{old} \quad (2.12)$$

E_{new}	New energy consumption when lowering the operative temperature θ_{op} , Wh
E_{old}	Old energy consumption, Wh

For calculating the possible cost savings, the average price for electricity (C) from the year 2015 will be used² (formula (2.13)).

$$\text{Energy savings} = (E_{old} - E_{new}) * C \quad (2.13)$$

$$\text{Cost savings} = C * E_{old} * \left(1 - \frac{\text{Degree days (17 °C)}}{\text{Degree days (18 °C)}}\right) \quad (2.14)$$

C	Price for electricity, €
-----	--------------------------

¹Webpage: www.degreedays.net (last checked on 07.06.2016)

² Average price for electricity in Sweden in 2015 is obtained from: <http://strom-report.de/strompreise-europa/> (last check: 10.06.2016)

All in all, Table 2.2 lists up all data that is measured or calculated:

Table 2.2: List of measured and calculated data

Measured data		Calculated data:	
Indoor air temperature θ_{indoor}	[°C]	Vertical temperature gradient $\Delta\theta_{vert}$	[°C]
Outdoor air temperature $\theta_{outdoor}$	[°C]	Operative temperature θ_{op}	[°C]
Floor, ceiling and wall surface temperatures $\theta_{floor}, \theta_{ceiling}, \theta_{wall,1}, \theta_{wall,2}$	[°C]	Temperature difference of supply and return temperatures $\Delta\theta_{hydraulic}$	[°C]
Heat pumps supply and return temperatures θ_{supply} and θ_{return}	[°C]	Hydronic mass flow through condenser \dot{m}_H	[kg / s]
Hydronic mass flow through condenser \dot{m}_H	[pulse] ³	Heat flux for time interval \dot{Q}_H	[W]
Electricity use of brine pump E_{brine}	[Wh]	Total heat output Q_{total}	[Wh]
Electricity use of compressor E_{comp}	[Wh]	Coefficient of performance (COP)	[-]
		Total operational energy usage TOEU	[Wh]
		Net energy savings $E_{savings}$	[Wh]
		Degree days DD	[°C*d]
		Energy savings based on DD	[Wh]
		Cost savings	[€]

2.5 Measurement devices

With their accuracy, measurement devices play a major role in determining the validity of these results. In this experimental study, several devices were used and will be introduced in this chapter. Table 2.3 lists all devices.

The accuracies for the flow meter as well as the energy meter were found in their manuals. With 0.4 % and 2 %, respectively, they depict the most reliable devices. In contrast, accuracies for the thermo sensor from the company “Proove AB” were not stated on public information portals. After contacting the supplier, it indicated an uncertainty of ± 2 °C for temperatures above 20 °C.

³ According to manufacturer’s information, 1 pulse equals 10 litre.

Table 2.3: Overview of measurement devices applied in this experimental study

Value	Manufacturer	Article number / type	Accuracy (given by manufacturer)	New accuracy
Indoor air temperature	Unknown	NTC Sensor 15 k Ω at 25 °C	-	$\pm 5 \%$ -
Outdoor air temperature				
Floor surface temperature	Proove AB ⁴	TSS 320	Over 20 °C: ± 2 °C	± 0.12 °C
Wall 1 surface temperature, 0.88 m			Under 20 °C: ± 1 °C	
Wall 2 surface temperature, 1.9 m	Proove AB	TSS 320	Over 20 °C: ± 2 °C	± 0.12 °C
Ceiling surface temperature			Under 20 °C: ± 1 °C	
Supply temperature	Kamstrup ⁵	65-00-0A0-219	± 0.05 °C	-
Return temperature				
Hydronic mass flow	ABB ⁶	WaterMaster DS/WM-EN Rev. W	$\pm 0.4 \%$	-
Compressor electricity use	Kamstrup ⁷	382 B	$\pm 2 \%$	-
Brine pump electricity use				

⁴ See Appendix B for manufacturer statement via eMail

⁵ Internet link:

<http://heatcombustion.co.uk/downloads/may%20datasheets/Kamstrup%20Sensors%20and%20Pockets.pdf> (last check: 18.07.2016)

⁶ Internet link: <https://library.e.abb.com/> (last check: 18.07.2016)

⁷Internet link: http://descargas.futurasmus-knxgroup.org/DOC/GB/Lingg&Janke/9443/Kamstrup_382_205810-559-GB.pdf (last check: 18.07.2016)

The accuracies for the flow meter as well as the energy meter were found in their manuals. With 0.4 % and 2 %, respectively, they depict the most reliable devices. In contrast, accuracies for the thermo sensor from the company “Proove AB” were not stated on public information portals. After contacting the supplier, it indicated an uncertainty of ± 2 °C for temperatures above 20 °C.

Indoor air temperature was measured by means of the retrofitted thermostat. Outdoor air temperature was measured by a thermo sensor deployed outside. It is part of the heat pump. The manufacturer stated that a so-called NTC sensor with a resistance of 15 k Ω at 25 °C is used in both cases. However, no further information about a specific model or accuracy could be ascertained.

Thus an internet research was conducted in order to find out common accuracies of similar products. Four examples for such thermos sensor and its accuracies can be found in the appendix (Appendix A). Most sensors indicated an accuracy of 5 %. Thus, this value was applied in the uncertainty analysis.

2.6 Calibration of measurement devices

The four surface temperatures indicate a relatively high measurement uncertainty of ± 2 °C. Thus, possible deviation of the result due to the uncertainty could be bigger than actual difference within period couples. For example, if floor temperature increased by 0.5 °C from period one to two, this could be covered by the uncertainty of ± 2 °C. As a consequence, a calibration was conducted.

Measurement deviations can be divided into two parts. The systematic deviation has a constant value and algebraic sign. They always occur and depict an offset from the actual value. In contrast, random deviations differ in value and algebraic sign and are randomly distributed. The systematic deviation can also have the value 0. The goal of an uncertainty analysis is to define the deviation and to then to deduce counteraction to reduce the deviation. [23]

Since the heating season came to an end and measurement devices were not available before start, the calibration could not be conducted before the experimental study. Hence it is done subsequently. Unfortunately, the measurement device for measuring wall surface temperature $\theta_{wall,2}$ was damaged after the measurement and was then unable to be calibrated. However, residual measurement devices θ_{floor} , $\theta_{ceiling}$ and $\theta_{wall,1}$ were calibrated.

Therefore, a calibration machine was used. A reference thermo sensor and the measurement devices that were calibrated were put into a bowl filled with water, whose temperature could be set manually. The reference device's accuracy is $\sigma_B = 0.015$ °C. When water temperature was changed, reference and each of the three measurement devices' temperature were read. Figure 2.6, Figure 2.7 and Figure 2.8 show the obtained data. Mean value \bar{x} is the difference between device and reference temperature. Although the reference device is very accurate, its uncertainty σ_B also needs to be considered in the new, total standard deviation σ_{new} that was used for further uncertainty analysis. Standard deviation σ_A and new standard deviation σ_{new} were calculated by means of formulas (2.15) and (2.16):

$$\sigma_A = \sqrt{\frac{\sum(x - \bar{x})^2}{n - 1}} \quad (2.15)$$

$$\sigma_{new} = \sigma_A + \sigma_B \quad (2.16)$$

σ_A	Calculated standard deviation of the measurement device, °C
x	Measured value, °C
\bar{x}	Mean value of the difference between measurement and reference device, °C
n	Sample size
σ_{new}	New, total standard deviation, including the reference device's uncertainty σ_B , °C
σ_B	Standard deviation of the reference device, °C

Since systematic errors are constant and always occur, they were considered as correction factor. Thus the mean value \bar{x} was subtracted from all the measured values that have been measured with this specific device. The calculated standard deviation σ_A depicted the uncertainty of measurement devices. Together with the reference device's uncertainty σ_B it formed the new uncertainty σ_{new} of the specific measurement device. All calculations were conducted with the help of "Microsoft Excel". In total, three correction factors and new uncertainties were obtained. Table 2.4 lists the obtained results.

Standard deviation σ_A turned out to be the same for all three devices. Due to this reason and the fact, that it is the similar type, the same standard deviation was also used in the uncertainty analysis for the fourth measurement device. However, the systematic error differs and thus no offset was applied for the residual device. Figure 2.6, Figure 2.7 and Figure 2.8 show the obtained values.

Table 2.4: Results of calibration

Measurement device	Mean value of difference \bar{x}	Standard deviation σ_A	Total new standard deviation σ_{new}
Unit	°C	°C	°C
Wall 1	0.52	± 0.12	± 0.14
Ceiling	0.89	± 0.12	± 0.14
Floor	0.50	± 0.12	± 0.14

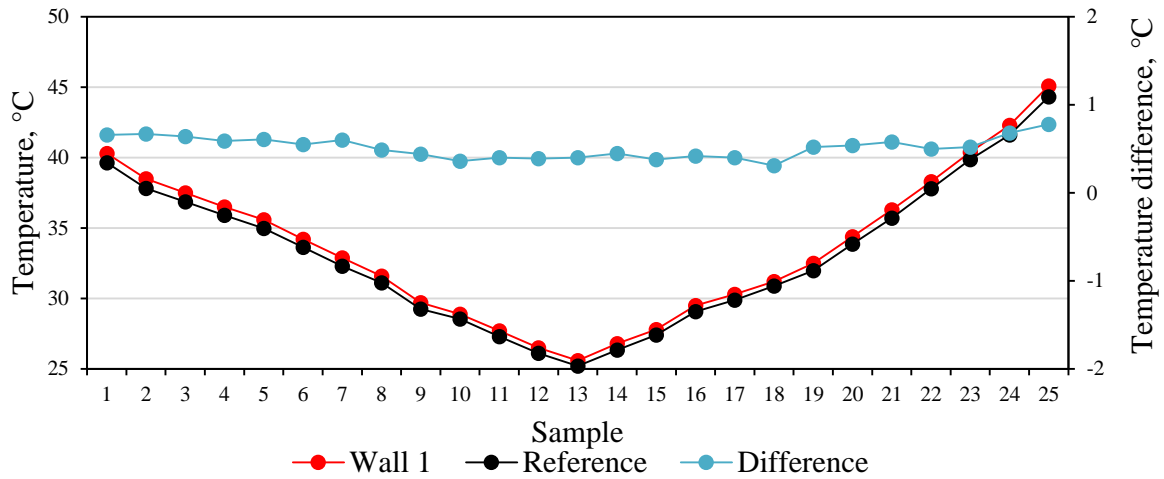


Figure 2.6: Measurement sample of "wall 1" measurement device

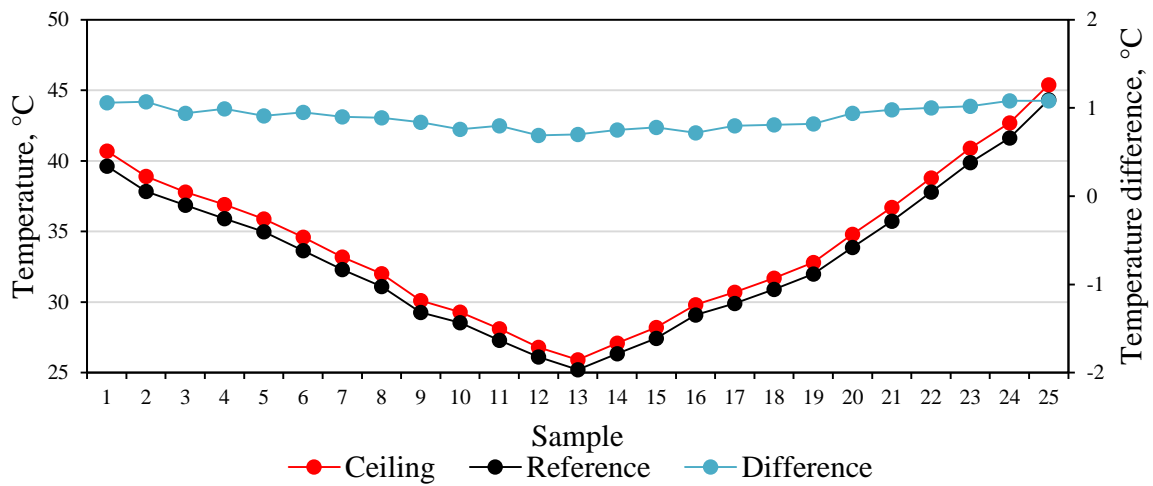


Figure 2.7: Measurement sample of "ceiling" measurement device

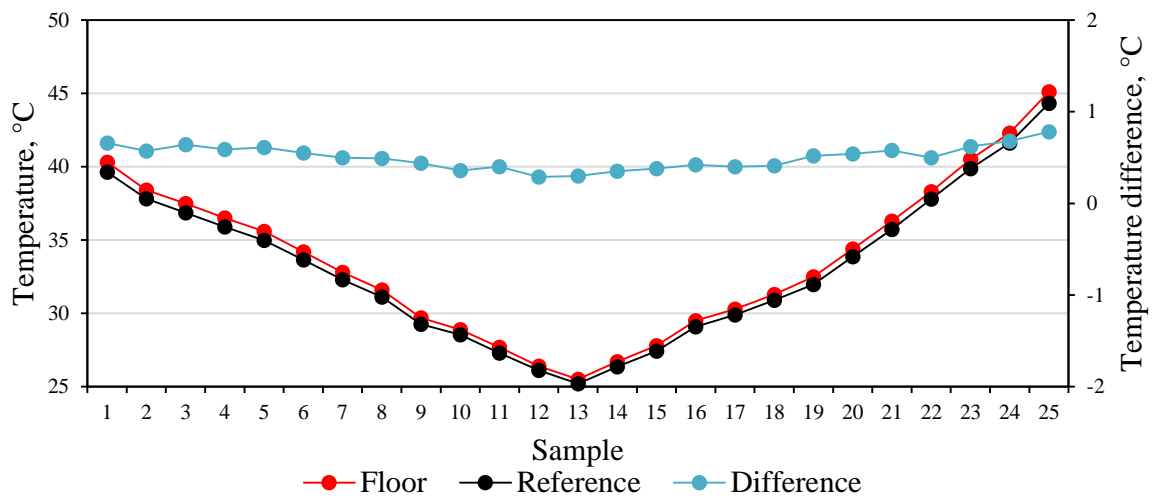


Figure 2.8: Measurement sample of "floor" measurement device

2.7 Expected results

After having addressed all important aspects of the methodology, the expected results are discussed (Table 2.5). Foremost it was expected that the radiator booster had impacts on thermal comfort and energy use. All values were referred to the obtained values when not using the radiator booster.

In terms of thermal comfort, the main performance indicator was the operative temperature θ_{op} . It was evaluated by the mean operative temperature within a period and the amount of hours it has been in a pre-defined temperature interval. It was expected to be higher with the radiator booster being switched on. The floor temperature was evaluated by the percentage of dissatisfied, according to ASHRAE standards. This percentage is expected to decrease, what entails an improvement of performance. Due to greater air movement, the floor temperature was expected to be higher. This also implies a decreased vertical temperature gradient $\Delta\theta_{vert}$ per meter.

As discussed in chapter 1.3.2 and 1.3.4, the heat exchange between radiator and air was expected to be higher. However, the heat output is fixed by the heating demand and the thermostat. As a consequence, the heat pump was expected to work less. This was expected to be mirrored in the heat pumps supply and return temperatures θ_{supply} and θ_{return} . The difference between them was estimated to drop, what mainly should be mirrored in the supply temperature. The electricity use of the compressor E_{comp} was expected to decrease with the decreasing workload of the heat pump. With the electricity use being decreased and the heating demand remaining constant, the COP value was expected to increase.

Table 2.5: Summary of expected results of retrofitting with radiator booster

Measured value	Expected development
Energy performance	
Difference in supply and return temperatures θ_{supply} , θ_{return} and supply temperature itself	↓
Mean logarithmic temperature difference $\Delta\bar{\theta}_m$	↓
Compressor energy use E_{comp}	↓
Coefficient of performance COP	↑
Total operational energy usage TOEU	↓
Net energy savings $E_{savings}$	Positive
Thermal comfort	
Operative temperature θ_{op}	↑
Vertical temperature gradient $\Delta\theta_{vert}$	↓
Floor temperature	↑

(↑: improved performance, ↓: diminished performance)

The final performance indicators of this study are the net energy savings. The compressor energy savings were expected to exceed the radiator boosters' electricity use E_{fans} . Then the total energy use is reduced and therefore the customer should be able to save energy and costs.

3 Results

The following chapter aims to present the obtained results of the experimental study. First, limitations of this experimental study are presented (chapter 3.1). Due to these, raw measurement results required an overwork, what is together presented in chapter 3.2. Final conclusions and explanations are based on refurbished data (chapter 3.3). Based on the degree-day method, additional possible energy savings are calculated (chapter 3.4), before an uncertainty analysis follows (chapter 3.5). Finally, the hypothesis is reassessed chapter 3.6.

3.1 Limiting aspects and other occurrences

There are several aspects that may limit the potential of the radiator boosters and the certainty of measured results. Some of them can be related to all periods, while others only happened in a single period. The universal aspects are listed in the following.

The heat pump is connected to a thermostat and works as soon as the measured indoor air temperature falls below the set temperature. It stops working as soon as the measured indoor air temperature is higher than the set temperature. As a consequence, the heat pump works intermittent. Thus it always needs some time to operate in the optimal range and the process has to be started first. A continuous process is expected to be both more efficient and accurate. This is expected to affect the COP in general. However, since the radiator boosters heat up the room faster, this effect could be even stronger. Thus, the compressor's efficiency could even deteriorate.

In contrast to the study conducted from Johansson [20], the output in this experimental study was "fixed" by the thermostat. Instead, the input was measured by means of compressor energy use, supply and return temperatures and hydronic mass flow. This circumstance is expected to limit the possible potential of the radiator boosters.

The effect of the influence factor is unknown. With a value of 7 (maximal 10) this factor has a great impact on the supply temperature. No studies have been conducted in order to examine the perfect factor or whether it improves or diminishes the heat pump's efficiency and accuracy. However, since "on" and "off" periods shall be compared to each other, the influence factor was kept constant during the complete six measurement periods.

Since measurements started in March, the outdoor temperature has been increasing from period to period. Although the measurements have been taking place within six periods, their slightly different mean outdoor temperature has to be taken into consideration. Compared to the average outdoor temperature in the heating season 2015, later periods indicate a higher one.⁸

A look at the layout of the ground floor (Figure 2.1a) shows that the living room is equipped with many windows. Nearly one complete sidewall is built of glass. Thus, the sun has an impact on indoor air temperature and can heat up the building to a great extent. The quantitative influence was not calculated but could lead to variation of heating demand within the periods.

Radiators on the ground floor are connected in line. As a consequence, the hydronic temperature decreases from radiator to radiator. This also implies that the possible benefit of the radiator booster

⁸ Daily mean temperatures of the heating season 2015 were obtained from www.smhi.se (last check: 08.06.2016)

decreases from radiator to radiator. This is a reason to evaluate all radiators as a system as well. In addition, it cannot certainly be said which radiator has been activated and which not. It is also not known, if a radiator actual contributed to the heat output, because remaining water flowing through it was not hot enough. For those being switched off or not contributing, a radiator booster obviously would be redundant.

Radiator boosters have always been switched on during the whole period. However, active heating was not always demanded, especially in the afternoon. In this period of time radiator boosters could also have been switched off in order to reduce their energy use and save further energy.

The following aspects occurred during single periods and are responsible for some deviations in measurement data. It is unknown why measurement devices did not work for a certain duration.

- The occupants living in the object of measurement have been on vacation from Monday, March 28th, until Sunday, April 4th. This is mainly affecting period 4, since less energy was needed, especially for domestic hot water. Consequently, a decreased heat output as well as compressor energy use is expected. “Vacation mode” of the heat pump, that leads to a lower indoor air temperature, was not activated.
- Compressor energy use was not recorded in period 5 for a period of time of approximately four days.
- Mass flow in compared periods are supposed to be as good as equal, with only slight deviations. However, period 5 indicated a significantly lower mass flow than period 6.
- Due to unknown reasons, the measurements of most of the measured variables in period 6 have stopped 1.5 days earlier. As a result, obtained data covers only 134.5 h instead 168 h.

3.2 Measurement results

After having introduced the reader to the technical background and the methodology of the measurement and its set-up, it is time to have a look at the obtained results. Raw results are presented first. Then measures for cleaning up deviating values are introduced. This was necessary in order to balance deviations and impacts of occupants’ behaviour. New, more reasonable data will be the baseline for final interpretation and conclusions. Table 3.1 shows the mean outdoor air temperatures in the single periods:

Table 3.1: Outdoor air temperature in the different periods

Period	Fan status	Mean Outdoor Air Temperature in °C
1	On	4.3
2	Off	4.6
3	On	6.9
4	Off	7.3
5	On	8.2
6	Off	8.3

With a maximal difference of only 0.4 °C between period 3 and 4 the couples 1 and 2, 3 and 4, 5 and 6 are able to be compared to each other. However, these little temperature differences must be taken into consideration when evaluating the data. Outdoor air temperature has been the lowest in periods 1 and 2. Period 1 and 2 took place in the middle of March. They are more suitable to represent the heating season, since temperatures in December, January and February are even colder and the boosters' impact is assumed to be greater.

Mean outdoor temperature in the heating season 2015 (January 1st – April 30 and October 1st – December 31st) has been 4.4 °C⁹ and thus nearly the same as in the first two periods. In addition, already mentioned aspects in chapter 3.1 lead to the assumption that the first two periods are the most reliable ones and reveal the highest potential of radiator boosters. Therefore, most presented charts and diagrams will obtain its data from period 1 and 2. Shorter amounts of data can be presented more clearly. However, measurement results from the other periods are not neglected and they are presented in the following chapters.

3.2.1 Presentation of polished results

Lower heating demand in period 4 and lacking measured data in period 5 and 6 did not influence the relations and temperature levels. Values describing the thermal comfort are still evaluable, but values in terms of energy performance are not comparable to other periods.

Part of the thermal indoor conditions are surface temperatures on the floor, the wall and the ceiling. In addition, indoor air temperature was measured and, based on the measured values, operative temperature as well as temperature gradient between floor and ceiling calculated.

With a mean outdoor air temperature of 4.3 and 4.6 °C in period 1 and 2, respectively, the environmental conditions for both periods have been quite the same. As can be seen in Figure 3.1a and Figure 3.1b, naturally outdoor air temperature and sunlight during the day also affect indoor air temperature. A new day starts with every given figure on the x-axis. Thus, highest temperature always can be seen in between two of them. Mean indoor air temperatures are 24.6 and 24.4 °C for period 1 and 2 respectively. Indoor air temperature differences from compared period couples are 0.2, 0.3 and 0.2 °C. Therewith the deployment of radiator boosters always has led to a slightly higher indoor air temperature. Target indoor air temperature of 24.5 °C has even slightly been exceeded in period 1.

Figure 3.2a illustrates the development of floor temperature in period 1 and 2. As can be seen, radiator boosters have mostly led to a much higher floor temperature. In the first two periods the difference was 1 °C, and then declined to 0.7 and 0 °C in the other two couples. Applying the ASHRAE standards, introduced in chapter 1.3.1, approximately 6.5 and 7.0 % dissatisfied people are obtained.

⁹ www.smhi.se (last check: 10.06.2016)

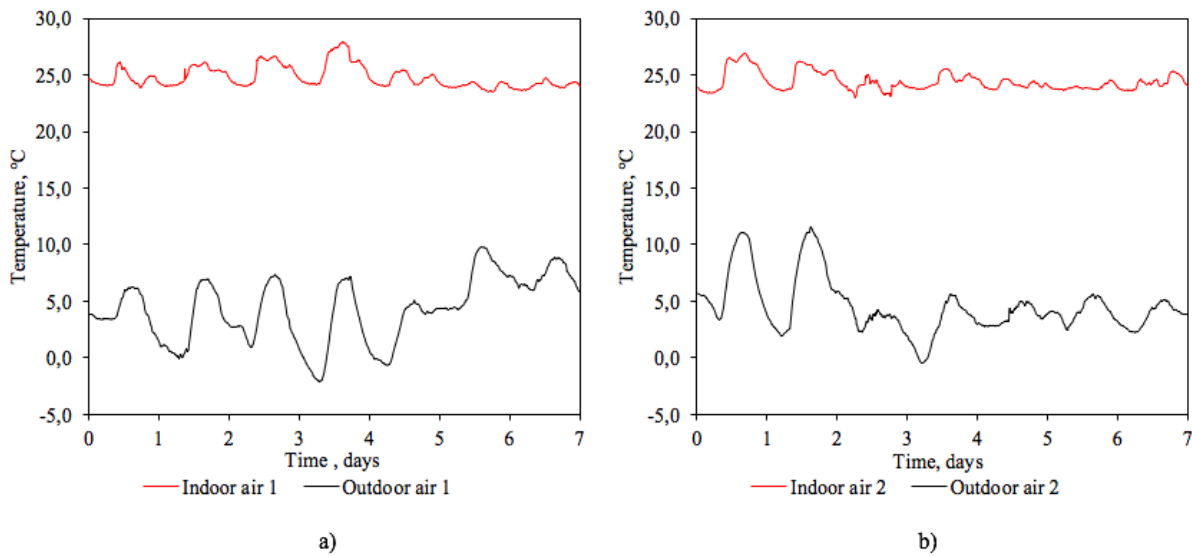


Figure 3.1: a) Development of indoor air and outdoor air temperature for period 1 and for period 2 in b)

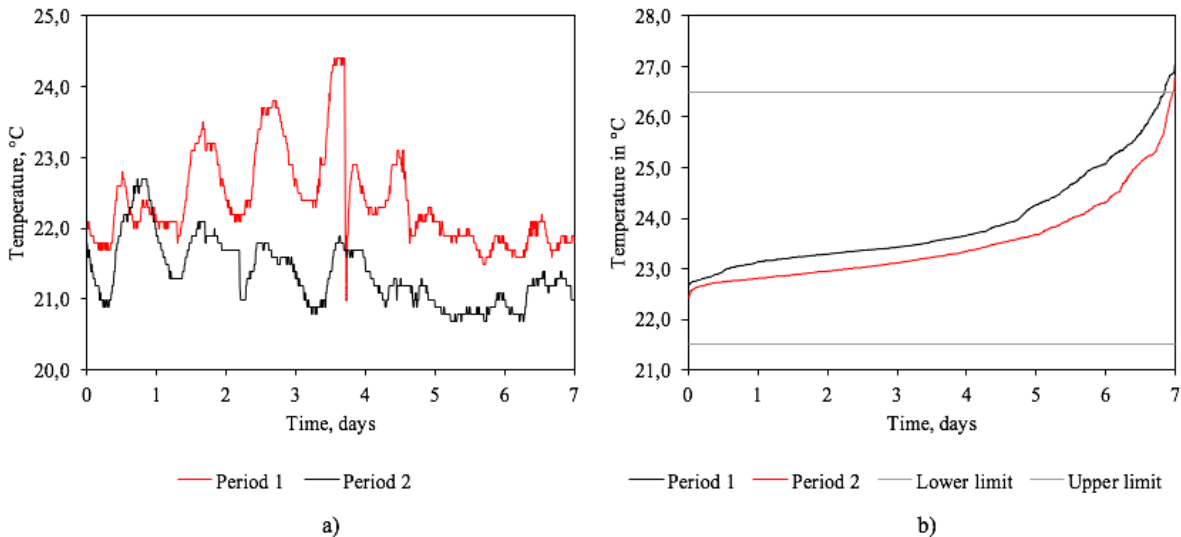


Figure 3.2: a) Development of floor temperature and b) operative temperature within temperature range, both for period 1 and 2

As seen in chapter 2.4, mean surface and indoor air temperature are used for calculating the operative temperature. Its development can be seen in Figure 3.2b. The upper and lower limits are 21.5 and 26.5 °C, respectively, and determine the acceptable range. In both cases, the lower temperature limit was never exceeded. When having the radiator booster switched on, the operative temperature has been within the desired interval for approximately 164 hours. This covers about 98 % of the total period of time. In period two, operative temperature exceeded the desired temperature interval for only 50 minutes.

Mean operative temperatures have always been increased with boosters being switched on. The difference varies between 0.4 and 0.1 °C within the period couples. Obviously, the differences decrease with higher outdoor temperatures.

Finally, vertical temperature difference between the floor and the ceiling was calculated. In all cases the difference was at least lower when the booster were switched off. In the first two couples the decrease amounted to 38 %. However, between period 5 and 6 no significant difference was measured. Table 3.2 summarizes the mean values in terms of thermal indoor conditions for all periods.

Table 3.2: Mean values for thermal indoor condition per period

Period	Mean indoor air temperature	Mean outdoor air temperature	Mean floor temperature	Mean operative temperature	Mean vertical temperature gradient
Unit	°C	°C	°C	°C	°C / m
1	24.6	4.3	22.4	23.9	0.5
2	24.4	4.6	21.4	23.5	0.8
3	24.4	6.9	21.9	23.5	0.4
4	24.1	7.3	21.2	23.2	0.7
5	24.2	8.2	20.9	23.2	0.8
6	24.0	8.3	20.9	23.1	0.8

The complementing part of the objective of this experimental study was to examine the impacts of radiator boosters on the energy performance of the heating system. Main performance indicators are the total heat output Q and the compressor energy use E_{comp} . Together they determine the heat pump's COP. The difference between the total operational energy usages TOEU of period couples are the net energy savings.

Return and supply water temperatures mainly influence the heat pumps energy use (chapter 1.3.3). Figure 3.3 and Figure 3.4 show the development of supply, return and outside temperatures in period 1 and 2. Self-evidently, there is a higher demand for heating in the night, when temperatures drop, sunshine is missing and indoor activities are reduced. Maximal supply temperature was increased by 2.5 °C in period 2, although minimal outdoor temperatures in period 1 and 2 were -2.1 and -0.4 °C, respectively. In other words, the supply temperature was lower although outdoor temperature was diminished as well. This can be seen on the fourth day in Figure 3.3 for period 1 and Figure 3.4 for period 2.

When boosters have been switched on, mean supply temperatures are 33.1, 31.6 and 31.0 °C for period 1, 2 and 3, respectively. With the boosters being switched off, mean supply temperatures are 35.9, 31.9 and 31.6 °C. In average, supply temperature was decreased by 1.2 °C and by 2.8 °C in period couple 1 in particular. However, it is important to keep in mind that the building was not occupied in nearly the complete period 4 and thus there was no need for domestic hot water. That

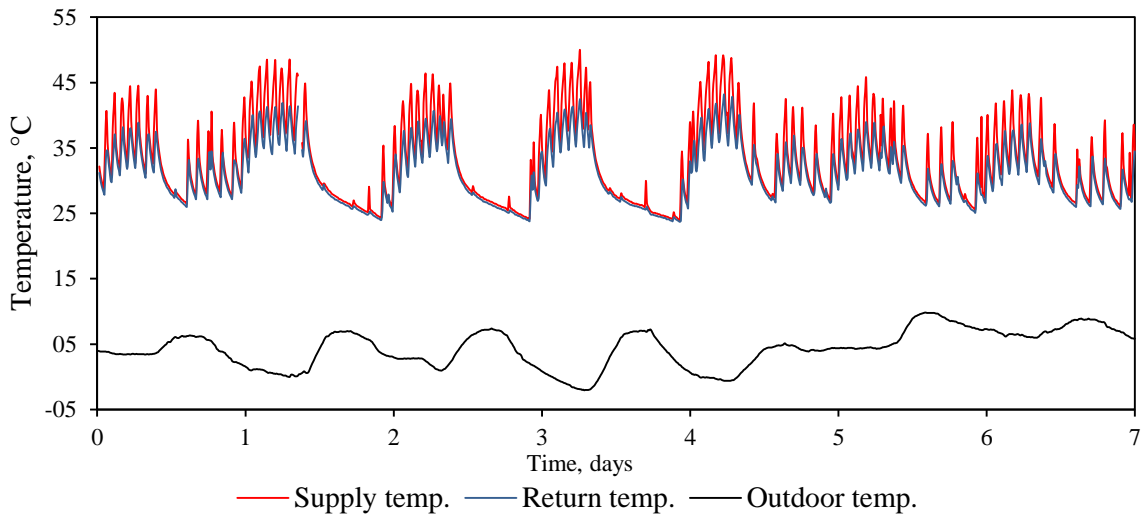


Figure 3.3: Development of supply, return and outdoor temperature in period 1

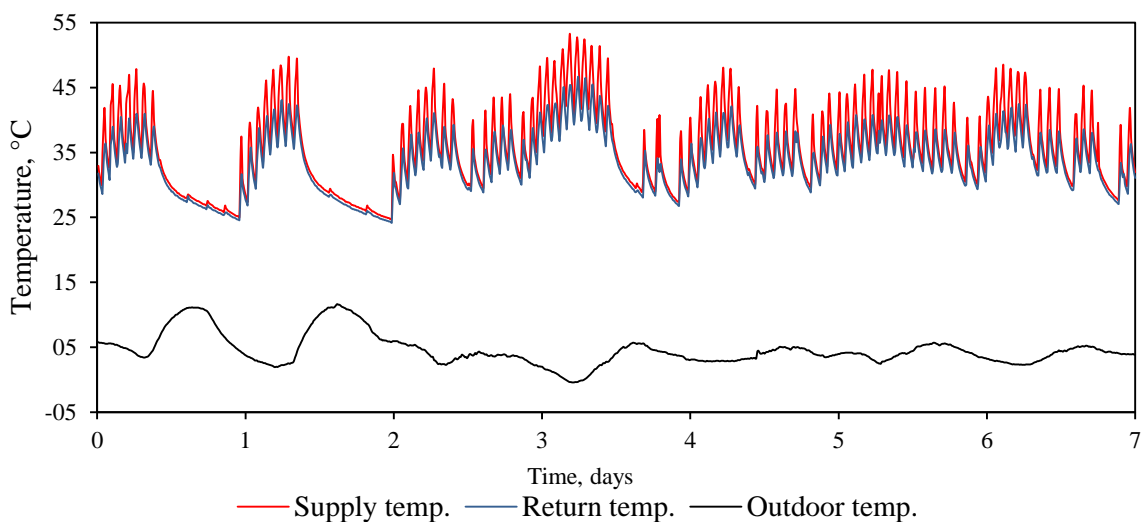


Figure 3.4: Development of supply, return and outdoor temperature in period 2

is why mean supply temperature difference is assumed to be higher in reality. According to the expected results, the differences between supply and return slightly decreased when switching on the boosters from 2.4 to 2.1 °C in the first period couple.

The mean logarithmic temperature difference indicates the mean difference between radiator surface and surrounding air. Due to heat transfer the radiator surface temperature decreases over the past “distance” of the water within the radiators. Measurement results have shown a decrease of up to 2.7 °C when activating the boosters. Increased indoor air temperature (+0.2 °C in period 1) and decreased supply temperature cause this decrease. Further declinations are 0.3 and 0.6 °C for couples 2 and 3, respectively. Again, diminished heating demand in period 4 and therefore reduced temperature levels have to be kept in mind. Actual difference is expected to be higher.

During the measurements it turned out that the hydronic mass flow is dependent on outdoor temperature. If this is constant, mass flow will not change when radiator boosters are switched on. This can also be seen in the measurements results. As a result, the circulation pump’s energy use can be assumed constant, if the hydronic mass flow did not change within period couples. As a consequence, they do not need to be considered in calculating the total operational energy usage

TOEU. Only exception is period five. Here it differed from its related period six. However, as already mentioned this is assumed to be a measurement error and will be compensated.

Table 3.3 and Table 3.4 show the results in terms of energy performance. Comparing the couples of the total heat output indicates a change of +10.5, -3.7 and -1.4 % when switching off the boosters. Obviously no pattern can be seen. Negative change from period 3 to 4 can be allocated to the low heating demand in period 4. Actual total heat output is expected to be higher than the current value, and thus an increase is assumed.

Table 3.3: Mean values for energy performance per period (1)

Period	Mean supply temp.	Mean return temp.	Log. mean temp. difference	Mean hydronic mass flow	Total heat output	Alteration
Unit	°C	°C	°C	kg/s	Wh	%
1	33.1	31.1	8.0	0.261	383 792	+10,5
2	35.9	33.5	10.7	0.262	428 780	
3	31.6	29.8	6.7	0.257	323 573	-3.7
4	31.9	30.1	7.0	0.257	312 117	
5	31.0	29.4	6.1	0.217	283 049	-1.4
6	31.6	29.9	6.7	0.253	279 209	

Table 3.4: Mean values for energy performance per period (2)

Period	Total heat output	E _{comp}	TOEU	COP	Net energy savings
Unit	Wh	Wh	Wh	-	Wh
1	383 792	124 658	127 851	3.08	2 370
2	428 780	130 221	130 221	3.29	
3	323 573	101 298	104 491	3.19	-11 312
4	312 117	93 179	93 179	3.35	
5	283 049	41 080	44 273	6.89	-31 695
6	279 209	75 689	75 968	3.68	

Due to already mentioned problems, no pattern can be seen in the compressor energy use. Comparing period one and two, a decrease can be assumed when switching on the fans. Since total

heat output and compressor energy use determine the COP, total operational energy usage and the net savings, no pattern can be discerned here as well. After the refurbishment of the data, the energy performance will be focused on again.

3.2.2 Polishing of measured data

Already mentioned limitations and influences in chapter 3.1 are tried to be balanced in this chapter. With the help of this cleaned up data, explanations and final conclusions are made. Following aspects have influenced the obtained values or are tackled when cleaning up the data (Table 3.5):

Table 3.5: Occurrences in single periods and countermeasures

Period	Affected value	Measure	Old value	New value
4	Heat flux and total heat output	Excluding from calculations for total heat output, compressor energy use, TOEU, net energy savings		
5	Mass flow	Use mass flow from P6	0.217 kg/s	0.253 kg/s
	Compressor energy use	Extrapolation	41 080 Wh	89 618 Wh
6	Total heat output	Use 168 h instead of 134.5 h	279 209 Wh	348 751 Wh
	Compressor energy use	Use 168 h instead of 134.5 h	75 968 Wh	94 889 Wh

Period 4 displays significant deviations from the other periods. In contrast to period 5 and 6, this is not due to lacking measurement data. Data, such as indoor air temperature, surface temperatures and its resulting calculated quantities are still valid. Mainly affected are the total heat output and the compressor energy use. For this reason, they are not included for further calculations of mean values of all periods.

In order to balance the diminished hydronic mass flow in period 5, the one from period 6 is used instead. New mean heat flux is calculated in the following way (formula (3.1)):

$$\dot{Q}_{new} = \frac{\dot{m}_6}{\dot{m}_5} * \dot{Q}_{old} \quad (3.1)$$

Recording of compressor energy use in period 5 did not take place for about four days. Since it influences the TOEU, the COP and the net energy savings it is important to obtain a value that is representative for this period. This was done by means of extrapolation (Figure 3.5). Since the heat

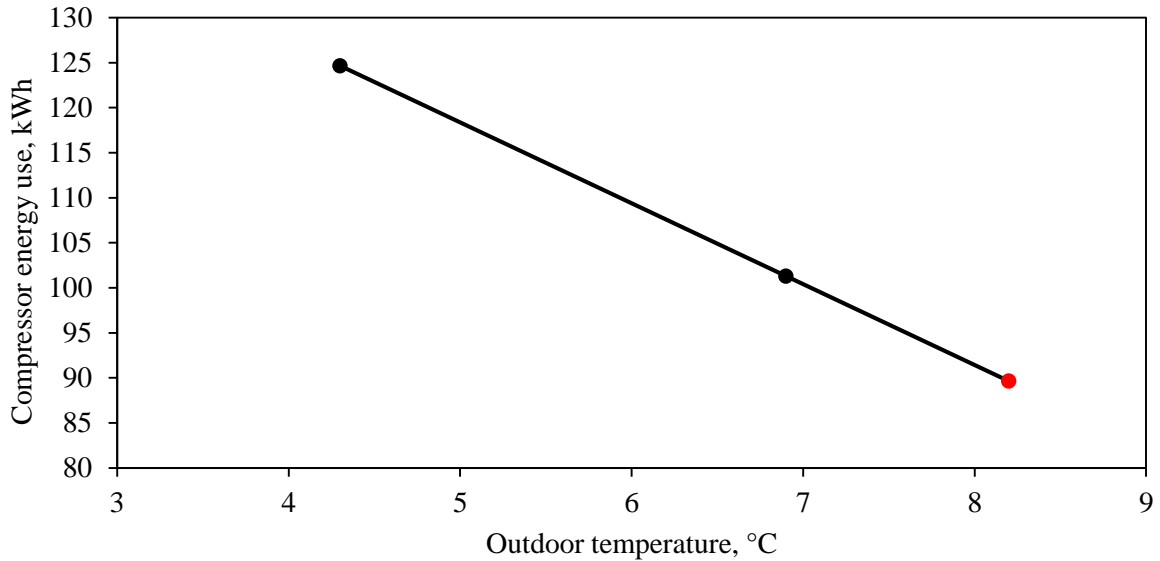


Figure 3.5: Estimation of compressor energy use in period 5 by means of extrapolation

pump follows a heating curve, which works accordingly to the outdoor temperature, compressor energy use should be proportional to the outdoor air temperature as well.

Not the complete week was recorded by the measurements in period 6. A simple extrapolation for the total heat output and the compressor energy use was done (formulas (3.2) and (3.3)):

$$Q_{H,new,6} = \frac{168 \text{ h}}{134.5 \text{ h}} * Q_{H,old,6} \quad (3.2)$$

$$E_{comp,new,6} = \frac{168 \text{ h}}{134.5 \text{ h}} * E_{comp,old,6} \quad (3.3)$$

The new values obviously have an impact on the results and are presented in the next chapter.

3.2.3 Results of polished data

After the clean-up new conclusions can be drawn. Additional to already presented results regarding thermal comfort, new findings based on the refurbished data are presented in the following. As values from Period 4 are not comparable to other periods, mean values for alterations in total heat output, compressor energy use, TOEU as well as net savings are stated by neglecting period couple 2. All stated values refer to the mean values of the fan status “on” and “off”:

In two of three period couples heat output was lowered. It was decreased by approximately 11 % and 45 kWh. In average, total heat output was decreased by 31.8 kWh (8.2 % savings).

Compressor energy use was increased in the “off” periods. This is a logical result of an increased heat output. In average, 5.4 kWh was saved through the deployment of radiator boosters (5 %). In the first two periods the difference amounted to 5.6 kWh, what depicts a decrease of about 4 %.

These figures lead to an average COP of 3.32 and 3.44 for “on” and “off”, respectively. Here, period couple 2 can be considered, since the relation between total heat output and compressor energy use

is not expected to be affected. The first two COPs are 3.08 and 3.29. All COPs are increasing with higher outdoor air temperature. A lower COP is surprising and contradictory to the assumptions that were made in chapter 2.7. However, it was focused on in chapter 3.3 again.

When the fans are switched on, total operational energy usage amounts to 110.3 kWh, in average. For the periods without the fans, measurement results indicate a value of 112.6 kWh. This implies a difference of 2.2 kWh and 2 %. A negative development can again be seen in period couple 2. However, this can be explained by the significantly lower compressor energy use in period 4. A change of 2.3 kWh was observed between period one and two (2.2 %).

Finally, net energy savings determine the economic feasibility of radiator boosters. With the given data, additional 2.2 kWh need to be raised when switching on the boosters. Approximately the same difference can be seen within the first two periods. In Figure 3.6 they can be seen as gap between both bars.

Table 3.6 sums up all new figures.

Table 3.6: Revised and cleaned up mean values for energy performance per period

Period	Total heat output	E_{comp}	TOEU	COP	Net energy savings
Unit	Wh	Wh	Wh	-	Wh
1	383 792	124 658	127 851	3.08	2 370
2	428 780	130 221	130 221	3.29	
3	323 573	101 298	104 491	3.19	-11 312
4	312 117	93 179	93 179	3.35	
5	330 203	89 618	92 811	3.68	2 078
6	348 751	94 889	94 889	3.68	
Average values, comparing periods with and without radiator boosters:					
On	356 997	107 138	110 331	3.32	2 224
Off	388 766	112 555	112 555	3.44	
Alteration [%]	-8.2	-4.8	-1.8	3.5	

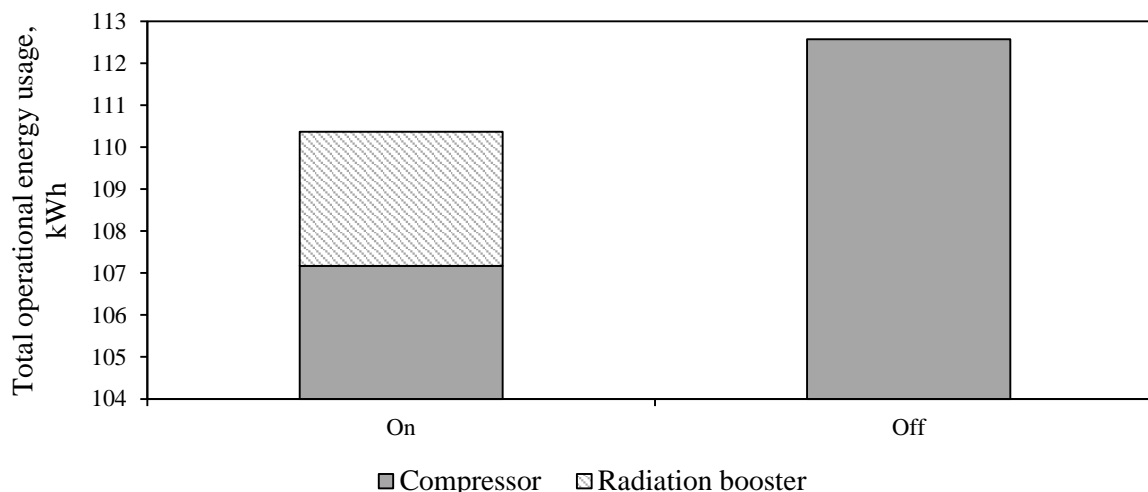


Figure 3.6: Mean total operational energy usage for periods with and without radiator boosters

For all values it is important to keep in mind, that period 4 does not reveal the full potential of the radiator booster. While aspects regarding thermal comfort can still be considered as reliable, those regarding energy performance, such as supply temperature, total heat output and compressor energy use are significantly influenced by the lack of domestic hot water production. As a result, those figures are expected to be higher than the present and improvements, caused by the radiator boosters, are assumed to be even higher.

3.3 Explanations for measurement results

This chapter aims to explain the results stated in the previous chapters. Following statements are based on refurbished data.

Periods with the fans switched off indicate a higher heat output and higher compressor energy use. The heat output equals the heating demand, as can be seen in chapter 1.3.4. Although it was supposed to be constant for the same outdoor temperature, it did increase when switching off the fans. One possible solution could be the greater distribution of warm air. Without the fans, it is assumed that there are “heat bulbs” in the area close to the radiators, as the warm air is not blown away into the room.

Just as in this case, radiators are usually placed below windows and on exterior walls. Driving force for heat transfer is a temperature gradient. With the radiator boosters not being deployed, this gradient between indoor and outdoor is now higher. Figure 3.7a and Figure 3.7b show the conditions without and with radiator boosters, respectively. A darker red indicates a higher density of heat. This leads to higher transmission and ventilation losses and thus to a higher heating demand. The area on the other side of the room does not influence the heat transfer on the exterior walls.

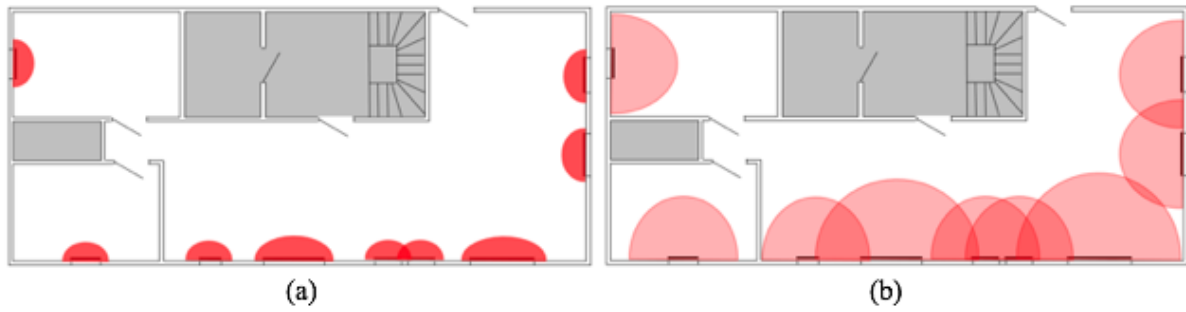


Figure 3.7: a) Distribution of warm air without and b) with deployment of radiator boosters

As a result, if these walls are heated up with the help of the radiator boosters, more energy is stored in the building's envelop and less energy is "lost" through transmission and ventilation. Due to the buildings layout, all of them are interior walls. They store the energy, do not lose it and give it back to the room as soon as indoor air temperature is smaller than the walls temperature. Finally, less energy is needed to heat the building.

A higher heating demand obviously also causes more work for the heat pump and thus a higher compressor energy use. This can be observed in most cases. If a significantly higher heating demand, due to a higher demand for hot water, in period 4 is assumed, it can be related to all periods. Another consequence of this is the improved thermal comfort. Surface temperatures and thus operative temperature did increase, while the vertical temperature gradient decreased.

A lower heating demand could perfectly explain the decreased supply temperature that could be observed in all period couples. However, another reason might be found in the fact, that rooms are heated up fast with the help of radiator boosters. As a consequence, target set temperature is reached faster and the heat pump switches off. One thing, that might even amplify this effect could be the control of the heat pump.

Figure 3.8 illustrates the difference between the actual indoor air temperature and the set temperature in period 1 and 2. The important part is between zero and the point of intersection with the grey line. At this point, there is no difference between actual and set indoor air temperature and the heat pump does not work anymore.

In period one, there was a negative temperature difference in the building for about 115 hours (4.8 days). In period two, the set temperature was not reached for about 151 hours (6.3 days). This implies a longer period of heating for about 31 %. The mean temperature difference within the first 4.8 and 6.3 days are -0.82 and -1.47 °C. This even amplifies the effect of an increased correction factor and leads to an increased supply temperature in period 2. As discussed in chapter 2.2, this increases the correction factor and thus the provided supply temperature. In theory, the provided supply temperature, given by the heating curve, should be increased by

$$\Delta\theta_{supply} = \frac{1.47^{\circ}\text{C} - 0.82^{\circ}\text{C}}{1.47^{\circ}\text{C}} * 7 = 3.1^{\circ}\text{C}.$$

$\Delta\theta_{supply}$ Difference of supply temperature between period 1 and 2, °C

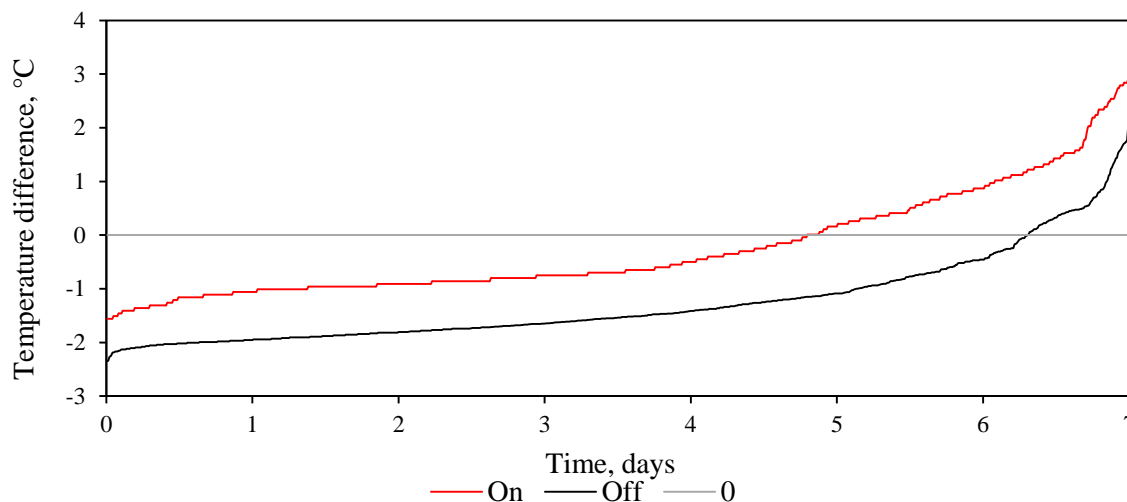


Figure 3.8: Difference between actual and set indoor air temperature in period 1 and 2

With a difference of 2.8 °C within the first period couple this value the reality lies quite close to the theoretical one.

As discussed, both the total heat output as well as the compressor energy use did decrease with the deployment of radiator boosters. However, in contrast to the expected results, the COP did not improve but deteriorate. The reason for this is that the total heat output and compressor energy use did not decrease in the same relation. The latter did not decrease as much as the total heat output. Since the fans heat up the room faster, the compressor's single working times are assumed to be shorter, as it has to stop earlier. It is assumed that this decreased the compressor's efficiency. Thus more energy is used, because the compressor needs to put into motion and before it is in a good operation mode it has to stop. However, there are no values measured that support this hypothesis.

An analogy can be found in kinematics. In order to move a body for a certain, resting on a surface, for a certain distance, static friction needs to be overcome at first. As soon as the body moves only sliding friction needs to be overcome to keep the body's velocity constant. Obviously, more energy is consumed with an increasing amount of intermediate stops since static friction needs to be overcome again and again.

3.4 Estimation of additional possible energy and cost savings

Net energy savings amounted to approximately 2.4 kWh in the first period couple. Since outdoor temperatures have been nearly the same as the mean outdoor temperature in the heating season 2015, this value will be used as baseline for calculation of additional possible energy and cost savings of one heating season. Final savings comprise already mentioned net energy savings and additional savings, achieved through a decrease of the operative temperature. In the same period couple, mean operative temperature was increased by 0.4 °C. By means of the degree-day method, additional possible energy savings are calculated, if lowering the mean operative temperature by 0.4 °C.

3.4 Estimation of additional possible energy and cost savings

First, net energy savings of one week have to be extrapolated to one heating season. One heating season comprises 212 days. Degree-days were calculated with the help of precise weather records¹⁰. Degree-days were already introduced in chapter 2.4. Fuel use is assumed to be proportional to the temperature difference. Thus, a linear decrease of energy use for heating can be expected for a decreased base temperature.

Second, additional possible energy savings are calculated. The base temperature can be changed in steps of 0.5 °C. For a base temperature of 18 °C, 92 064 hours are obtained. Decreasing the base temperature by 0.5 °C leads to 88 080 hours of active heating for one year. Assuming a linear graph, 88 877 hours can be related to a decrease in the base temperature of 0.4 °C. Accordingly, energy use and cost can theoretically further be reduced by 3.5 %. For these calculations formula (2.11) and (2.12) are used.

For further calculations the approximate annual energy use of a single family-house must be calculated. Figure 3.9 shows the space heat loss per square meter floor area of an average Norwegian single-family house up to 2000 [24]. This curve represents younger buildings. Since the object of measurement was built in the 1960s and insulation technology improved, it must be adapted.

Therefore, the energy use of period four was used. Here, the occupants have been on vacation and DHW usage was reduced to a minimum. Thus, two points can be assumed as fixed: The y-intercept, since this is the temperature without heat loss, and the point determined by the energy use in period four (31 W/m² at an outdoor air temperature of 7.3 °C).

This results in a new curve for the specific heat loss per square meter floor area for our building. Next, the outdoor air temperature profile of the heating season 2015 (Figure 3.10) is used to calculate the annual energy use. Daily mean temperatures for 212 days are given by SMHI¹¹ and used to calculate daily mean energy use. With possible savings of 3.5 % and an annual price for electricity of 0.19 €/kWh in 2015¹² energy and cost savings are obtained (Table 3.7).

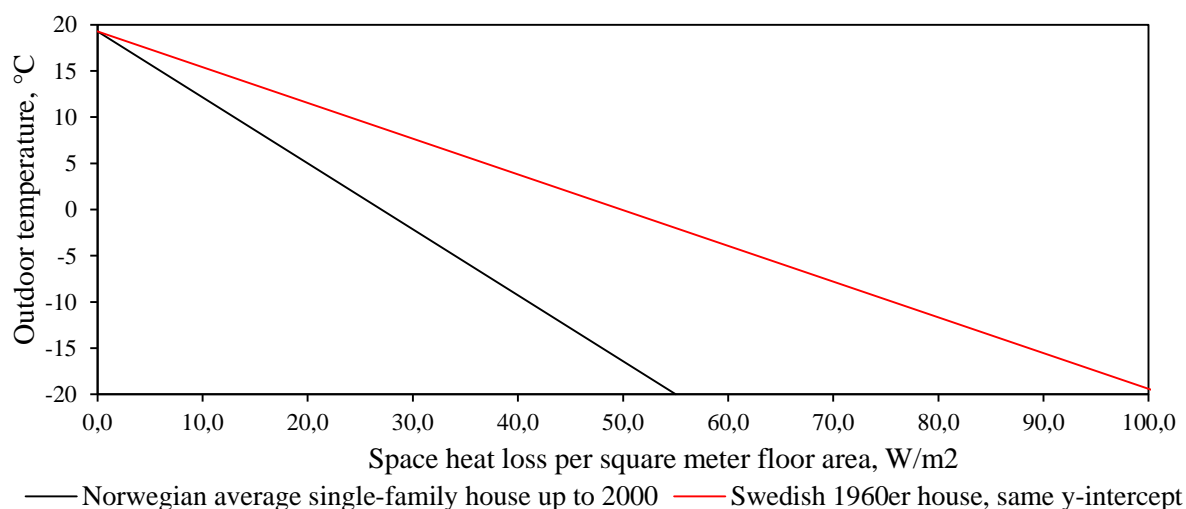


Figure 3.9: Calculated space heat loss per square meter floor area of a Swedish 1960er built single-family house based on a norwegian average single-family house up to 2000

¹⁰ www.degree-days.net calculates the degree-days based on data from <https://www.wunderground.com/> (last check: 10.06.2016)

¹¹ www.smhi.se (last check: 14.06.2016)

¹² <http://strom-report.de/strompreise-europa/> (last check: 15.06.2016)

3.4 Estimation of additional possible energy and cost savings

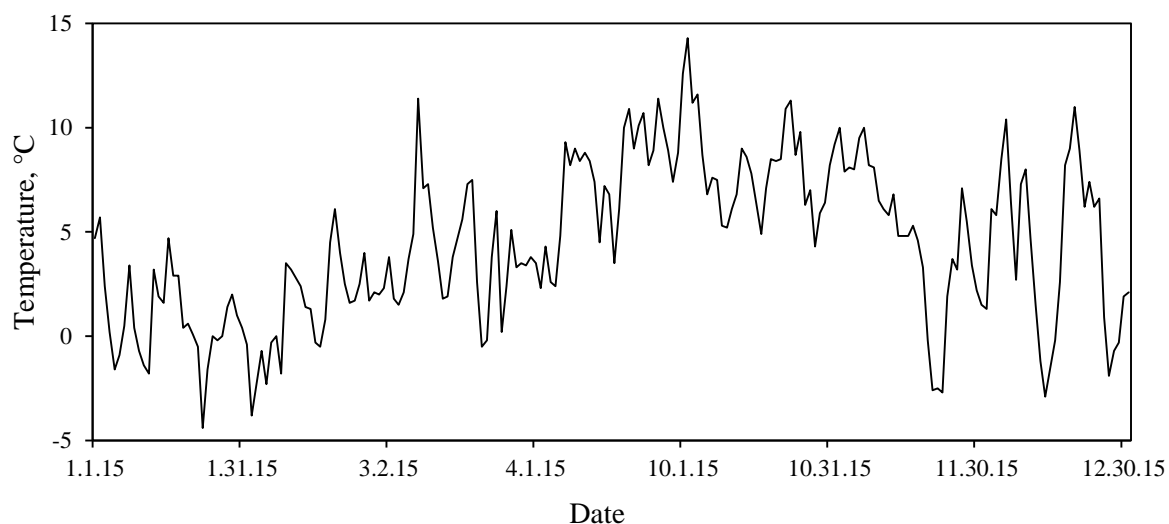


Figure 3.10: Development of outdoor temperature in the heating season 2015

Table 3.7: Estimated energy and cost savings based on degree-days method

Electricity price in Sweden in 2015	0.19 €/kWh	
Basic net energy savings per week	2.4 Wh	
Relative possible cost and energy savings	3.5 %	
Basic net energy and cost savings per heating season	72 kWh	14 €
Additional possible energy and cost savings per heating season, based on degree-day method	602 kWh	114 €
Resulting possible energy and cost savings	674 kWh	128 €

In conclusion, approximately 72 kWh and 14 € could already be saved per heating season by the deployment of radiator boosters. However, with further adjustments, significant more, additional energy and costs can be saved. Decreasing the operative temperatures by 0.4 °C could lead to additional energy and cost savings of approximately 602 kWh and 114 €, respectively. This can be achieved by choosing a lower heating curve what lowers the set temperature, for instance. Then still the same thermal comfort as without radiator boosters can be achieved. Finally, in total 128 € could be saved per year. This depicts about 3.5 % energy savings of the total energy use per year.

3.5 Uncertainty analysis

As a final step an uncertainty analysis needs to be performed. With the help of the calibration (chapter 2.6), measurement devices' uncertainties could be reduced. Thus, obtained data is expected to be more reliable and definite. For sums, as can be seen in formula (3.4), the absolute deviation was calculated by means of formula (3.5) [25]:

$$z = x + y \quad (3.4)$$

$$\Delta z = \sqrt{\Delta x^2 + \Delta y^2} \quad (3.5)$$

x, y, z Mean value of an item

$\Delta x, \Delta y, \Delta z$ Absolute deviation of an item

Products (formula (3.6)) are treated differently. For them, formula (3.7) was used [25]:

$$z = x * y \quad (3.6)$$

$$\Delta z = \Delta x * |y| + \Delta y * |x| \quad (3.7)$$

$|x|, |y|$ Absolute value of an item

For fractions (formula (3.8)), formula (3.9) was used [25]:

$$z = \frac{x}{y} \quad (3.8)$$

$$\Delta z = \frac{\Delta x * |y| + \Delta y * |x|}{y^2} \quad (3.9)$$

Such as in previous chapters, it is focused on period one and two. In all cases, period couple one is used. Table 3.8 summarizes the obtained absolute and relative deviations.

The column "Limit" indicates the maximal possible absolute deviation, so that it can be definitely concluded that the deployment of radiator boosters has led to an improvement in a specific item from period one to period two. In other words, the measurement device's uncertainty is smaller than the measured difference between period one and two. Hooks and crosses indicate that obtained data is valid and not valid, respectively.

As can be seen, this is the case for the floor temperature θ_{floor} , supply temperature θ_{supply} , vertical temperature gradient $\Delta\theta_{vert}$, total heat output Q_{total} , compressor energy use E_{comp} and the COP.

In contrast, the measured improvement in the operative temperature θ_{op} is smaller than the uncertainty of this value. As a result, it cannot be guaranteed that this value actually improved. This is also the case for the total operational energy usage TOEU and the energy savings $E_{savings}$.

3.6 Reassessing of the hypothesis and expected results

Table 3.8: Summary of absolute and relative deviations from performance indicators

Item	Mean value	Absolute deviation	Limit	Within boarder?
θ_{floor}	21.9 °C	0.14 °C	1.0 °C	✓
θ_{supply}	34.5 °C	0.04 °C	2.8 °C	✓
$\Delta\theta_{vert}$	0.6 °C	0.10 °C	0.6 °C	✓
θ_{op}	23.7 °C	0.61 °C	0.4 °C	✗
Q_{total}	406 286 Wh	12 056 Wh	44 988 Wh	✓
E_{comp}	127 440 Wh	2 549 Wh	5 563 Wh	✓
TOEU	129 036 Wh	2 549 Wh	2 370 Wh	✗
$E_{savings}$	2 224 Wh	3 605 Wh	2 224	✗
COP	3.19	0.16	0.21	✓

Since the probability, that values were always recorded on the outer limits of the uncertainty, is relatively low, these measurements at least give a hint that radiator booster lead to an increase/decrease in this item. As an example, the absolute deviation of TOEU is only slightly greater than the actual difference.

3.6 Reassessing of the hypothesis and expected results

In terms of thermal comfort, all assumptions were vindicated. Vertical temperature gradient, operative temperature as well as floor temperature did all improve. The biggest increase can be found in the floor temperature.

More aspects have been investigated in the energy performance. Here, not all values developed as expected. To begin with the correct assumptions, mean logarithmic temperature difference along the radiator, compressor energy use, TOEU and net energy savings did develop as expected. A decrease in the first three figures have led to positive net energy savings. This means less energy was used when fans have been switched on.

However, the difference between supply and return temperature did only slightly decrease in the first period couple. While it remained constant in the last one, period couple two even indicated a negative change. However, this can be related to the decrease heating demand. As a result, a general decrease can only be assumed. Another value that did not turn out to improve is the COP. Although it was expected to increase and energy was saved, it deteriorated with radiator boosters being switched on.

3.6 Reassessing of the hypothesis and expected results

However, the improvements in operative temperature, TOEU and energy savings are smaller than its uncertainty. This is mainly due to the accumulating uncertainty, as these values are calculated by means of the most different values. Consequently, these aspects need to be treated cautiously.

4 Résumé

In this experimental study the impacts of retrofitting ordinary hydronic radiators, equipped with a GSHP, with radiator boosters were examined. The study focused on the thermal comfort and the energy performance. It took place in a single-family house in the northern suburban area of Stockholm, Sweden, in the late heating season 2015 (March – April). In total, 48 fans mounted on nine radiators were installed in a total area of 88 m².

4.1 Overall result

Following conclusions are drawn from measurement results:

Most explanations and conclusions are based on the fact, that the radiator boosters seem to be greatly dependent on the building layout. In this case, radiators are mounted on exterior walls and warm air is pushed towards the middle of the building and towards interior walls, where no heat loss happens. This implies that actual figures could differ between buildings. If the warm air was pushed from interior to exterior walls, it is questioned if boosters had a benefit at all, as temperature gradients at exterior walls would increase. Consequently, all following conclusions are made under assumptions of similar layouts.

The impact of the radiator boosters on both thermal comfort as well as energy performance decreased with increasing outdoor temperature. This is obvious, since there is a higher need for heating in colder months and the heat pump is required to work longer.

Measurements show that the deployment of radiator boosters led to a decreased demand for heating. It is assumed they have “destroyed” the heat bulbs close to the radiators by moving the warm air towards the other side of the room. Thus, temperature gradients at exterior walls are smaller and heat loss reduced. The decreased heating demand (-10.5 % in period couple one) entails the lower compressor energy use (-3.8 %).

A lower COP, especially in colder periods, was observed during periods with the boosters being switched off. This fact gives a hint on a reduced compressor efficiency, what was already mentioned in chapter 3.3. A steadily working heat pump is assumed to show a different performance and could be more suitable when deploying radiator boosters. But in this case, the COP seems not to be appropriate as ultimate performance indicator, since actual compressor efficiency is not known. In contrast, net energy savings seem to depict the key performance indicator in this case.

When outdoor temperatures are cold enough, the boosters were able to save energy and costs. Net energy savings are then positive. In total, 3.5 % of the previous energy and costs expenses could be saved, if adapting the newly gained, increased thermal comfort to the previous level. The additional adaption of the heating curve, what was not done in this building, is considered the main driver for energy and cost savings. According to calculations, it could be responsible for about 89 % of the calculated total possible savings.

If not adapting, energy and cost savings reduce to a minimum (11 %). Then the main benefit is the increased thermal comfort. It was expressed by an increased operative temperature (+0.4 °C), increased floor temperature (+1.0 °C) and a decreased vertical temperature gradient (-0.3 °C/m) in period couple one.

Finally, it is important to keep in mind that improvements in the operative temperature, the total operational energy usage and the net energy savings are lower than the uncertainty of the specific, influencing parameters. This means these figure should only be conceived as a hint and not as a fact.

In conclusion, it can be said that radiator boosters do have a benefit in terms of thermal comfort and could have a benefit in terms of energy performance. The hypothesis cannot be confirmed entirely and with certainty. Measurement results and calculations indicate that there is still space for improvements. In addition, compared to other solutions, such as ventilation radiators, baseboard radiators and especially floor heating, radiator boosters are expected to provide the smallest contribution and benefits. However, they depict the easiest and only retrofittable one of all the four presented solutions.

Considering the whole picture, there are many adjusting screws in order to reduce energy use. Retrofitting with radiator boosters are just one of them. In terms of building energy use, other possibilities could be wall and façade renovation, window replacement, adding insulation on the attic/roof and ground-floor renovations. Together with the application of ventilation radiators, a study showed that all five retrofitting options together could decrease the energy use of 40 – 50 year old buildings by up to 50 % [26].

This outlines the need for attacking many components. Single contributions and small components do only have a small impact on the total operational energy usage. However, every contribution should be embraced and used for improving the existing heating system.

4.2 Lookout

The lower COP was explained with a lower compressor efficiency, due to shorter working times, as the boosters heat up a room faster. As an example, the analogy of moving a body on a surface was used. Due to this assumptions, the actual quantitative benefit of the radiator boosters is still hidden in the fog. Therefore, a further study would be helpful. In order to compensate the compressor efficiency problem and other limitations (chapter 3.1), following recommendations are presented:

The lower efficiency of the compressor could be avoided by using a heating curve, that provides a lower supply temperature at the same outdoor temperatures. In another study, the best fitting heating curve, that provides the same thermal comfort but with lower supply temperatures, should be find out and used in the periods with the radiator boosters being switched on. This would indicate how much the supply temperature can be decreased in fact, compared to the normal operation mode, and thus to which extend low-temperature heating is actually supported. Moreover, it could help dealing with the compressor efficiency problem, since working times are expected to be more equal.

Another important aspect of lowering the heating curve is to find out the actual energy savings. In this study, the energy and cost savings were calculated based on the degree-day method due to a higher operative temperature. Therefore, assumptions and extrapolation were used. Measuring the actual savings would reduce the uncertainty that comes along with calculations. This could be achieved by adapting the heating curve and by reducing the influence factor to 0. Since both a high influence factor and the radiator boosters lead to a faster heating up of the room, this could amplify the assumed inefficiency of the compressor when boosters are deployed. As a result of the adaption, greater possible savings than the obtained ones (3.5 %) and even lower supply temperatures are assumed.

Finally, the study should cover the complete heating season in order to enhance the accuracy. In this study, two of three period couples are object of disturbances. A higher amount of period couples therefor provides a picture about the development of the impact of the boosters in dependency on the outdoor temperature and more accurate data about the COP. In addition, deficient period couples are compensated.

There are also other possibilities to improve the radiator boosters and the system itself. Especially in the beginning and the end of a heating season (spring and autumn), heating can be required only in the night. Here, measurement results sometimes indicated an inactive heat pump for several hours without interruption. Radiator boosters still have run continuously. An adaption of the working times of boosters and heat pump would further reduce the energy use.

The idea of “heating on demand” aligns with this measure. As presented in chapter 1.4.4, radiation boosters are able to heat up a room much faster than an ordinary radiator. In order to benefit from this ability, the idea of “heating on demand” suggests to completely turn off the heating system when leaving the building. Turning on the heating system via remote control in sufficient time before coming home, e.g. a smartphone, should then lead to significant energy savings.

Table of references

- [1] J. Babiak, *Low temperature heating and high temperature cooling*. Brussels: Rehva, 2007.
- [2] *Klimaschutz: Wir gestalten unsere Welt, wir gestalten unsere Klima*. Luxembourg: Publications Office, 2014.
- [3] Swedish Energy Agency, “Energy in Sweden 2015,” Swedish Energy Agency, Eskilstuna, 2015. [Online] Available: <http://www.energimyndigheten.se/en/news/2015/you-can-now-read-energy-in-sweden---facts-and-figures-2015/>. Accessed on: Mar. 03 2016.
- [4] Climate Research Institute. [Online] Available: <http://www.cru.uea.ac.uk/cru/info/warming>. Accessed on: Jun. 11 2016.
- [5] H. Eijdens, A. Boerstra, and P. O. Veld, “The health, safety and comfort advantages of low-temperature heating systems: a literature review,” *Proceedings of the 6th international Conference on Healthy Buildings*, 2000.
- [6] M. Bojić, D. Cvetković, J. Skerlić, D. Nikolić, and H. Boyer, “Performances of low temperature radiant heating systems,” *Energy and Buildings*, no. 61, pp. 233–238, <http://dx.doi.org/10.1016/j.enbuild.2013.02.033>, 2013.
- [7] S. Frederiksen and S. Werner, *District Heating-Theory, Technique and Functionality (in Swedish)*. Lund: Studentlitteratur, 1993.
- [8] E. Abel, “The Energy Balance of a Building,” in *Achieving the desired indoor climate: Energy efficiency aspects of system design*, P.-E. Nilsson, Ed, Lund: Commtech Group, 2003, pp. 299–355.
- [9] L. E. Ekberg, P. O. Fanger, and P. Wargocki, “Physical Climate Factors,” in *Achieving the desired indoor climate: Energy efficiency aspects of system design*, P.-E. Nilsson, Ed, Lund: Commtech Group, 2003, pp. 91–210.
- [10] American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc, “Thermal Environmental Conditions for Human Occupancy,” American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc. (Atlanta) 2004, 2010. Accessed on: Mar. 09 2016.
- [11] F. P. Incropera and D. P. DeWitt, *Fundamentals of heat and mass transfer*, 5th ed. New York, NY: Wiley, 2002.
- [12] F. C. McQuiston, J. D. Parker, and J. D. Spitler, *Heating, ventilating, and air conditioning: Analysis and design*, 6th ed. Hoboken, N.J: Wiley, 2005.
- [13] A. Trüschel, “Generation of Heating and Cooling,” in *Achieving the desired indoor climate: Energy efficiency aspects of system design*, P.-E. Nilsson, Ed, Lund: Commtech Group, 2003, pp. 563–584.
- [14] Anders Trüschel, “Hydronic Heating and Cooling Systems,” in *Achieving the desired indoor climate: Energy efficiency aspects of system design*, P.-E. Nilsson, Ed, Lund: Commtech Group, 2003, pp. 459–537.
- [15] Olesen and Bjarne W, “Radiant Floor Heating In Theory and Practice,” *ASHRAE Journal*, pp. 19–24, https://www.taco-hvac.com/uploads/FileLibrary/Radiant_Floor_Heating_-_ASHRAE_Journal_2002-7.pdf, Jul. 2002.
- [16] A. Ploskić and N. Meiser, *Technical solutions for low-temperature heat emissions in buildings // Novel technologies for mode-locking of solid-state lasers*. Stockholm, Sweden: KTH Royal Institute of Technology, 2013.

- [17] J. A. Myhren and S. Holmberg, “Performance evaluation of ventilation radiators,” *Applied thermal engineering*, no. 51, pp. 315–324, <http://dx.doi.org/10.1016/j.applthermaleng.2012.08.030>, 2012.
- [18] J. A. Myhren and S. Holmberg, “Design considerations with ventilation-radiators: Comparisons to traditional two-panel radiation,” *Energy and Buildings*, no. 41, pp. 92–100, https://www.researchgate.net/publication/223626584_Design_considerations_with_ventilation-radiators_Comparisons_to_traditional_two-panel_radiators, 2009.
- [19] A-Energi AB, *ElementFläkten*. [Online] Available: http://www.elementflakten.se/pub_docs/files/Elementflakten.pdf. Accessed on: Jun. 06 2016.
- [20] P.-O. Johansson, *Buildings and district heating: Contributions to development and assessments of efficient technology*. Lund: Lund University, 2011.
- [21] A. Hesaraki, E. Bourdakis, A. Ploskić, and Holmberg Sture, “Experimental study of energy performance in low-temperature hydronic heating systems,” *Energy and Buildings*, no. 109, pp. 108–114, <http://dx.doi.org/10.1016/j.enbuild.2015.09.064>, 2015.
- [22] IVT Värmepumpar, *Troubleshooting of heat pump with Rego 600/634/637/638 control unit*.
- [23] K. Eden and H. Gebhard, *Dokumentation in der Mess- und Prüftechnik: Messen - Auswerten - Darstellen Protokolle - Berichte - Präsentationen*, 2nd ed. Wiesbaden: Springer Vieweg, 2014.
- [24] J. Rekstad, A. R. Kristoffersen, and M. Meir, “Control and energy metering in low temperature heating systems,” *Energy and Buildings*, no. 35, pp. 281–291, doi:10.1016/S0378-7788(02)00090-7, 2003.
- [25] L. Eldén and L. Wittmeyer-Koch, *Numeriska beräkningar: Analys och illustrationer med MATLAB®*, 4th ed. Lund: Studentlitteratur, 2001 ;.
- [26] Q. Wang, A. Ploskić, and S. Holmberg, “Retrofitting with low-temperature heating to achieve energy-demand savings and thermal comfort,” *Energy and Buildings*, no. 109, pp. 217–229, <http://www.sciencedirect.com/science/article/pii/S0378778815302875>, 2015.

Nomenclature

θ_s	Temperature of a surface
θ_f	Temperature of a fluid
θ_c	Condenser temperature
θ_e	Evaporator temperature
θ_{supply}	Supply temperature
θ_{return}	Return temperature
θ_{op}	Operative temperature
θ_{floor}	Floor surface temperature
$\theta_{wall,1}$	Wall surface temperature (height: 0.88m)
$\theta_{wall,2}$	Wall surface temperature (height: 1.9m)
$\theta_{ceiling}$	Ceiling surface temperature
$\bar{\theta}_{surface}$	Mean surface temperature
θ_{indoor}	Indoor air temperature
$\theta_{outdoor}$	Outdoor temperature
$\Delta\bar{\theta}_m$	Logarithmic mean temperature difference between radiator and surrounding air
$\Delta\theta_{vert}$	Vertical temperature gradient
θ_{base}	Base temperature
θ_i	Actual outdoor temperature
$\Delta\theta_{supply}$	Calculated difference of supply temperature between period 1 and 2
η_c	Carnot efficiency of a heat pump
h	Convective heat transfer coefficient
k	Conductive heat transfer coefficient
A	Surface area
U	Overall heat transfer coefficient

\dot{m}_H	Hydronic mass flow
\dot{V}	Volumetric flow
ρ	Density
c_p	Specific heat capacity of water
\dot{Q}_H	Heat flow
Q_{total}	Total heat output of radiators
E_{comp}	Energy use of compressor
$E_{booster}$	Energy use of radiator boosters
$E_{savings}$	Energy savings within a period couple
E_{new}	New energy consumption
E_{old}	Old energy consumption
DD	Degree days
C	Cost for electricity
T_w	Duration of a period
N	Number of hours the actual outdoor temperature θ_i was valid (for degree days)
σ_A	Standard deviation of a measurement device
σ_B	Standard deviation of the reference device
σ_{new}	Total standard deviation of a measurement device
x, y, z	Measurand
$\bar{x}, \bar{y}, \bar{z}$	Mean value of a measurand
$\Delta x, \Delta y, \Delta z$	Deviation of a measurand
$ x , y $	Absolut value of a measurand
n	Sample size
IAQ	Indoor air quality
COP	Coefficient of Performance
DHW	Domestic hot water
LTH	Low-temperature heating

TOEU

Total operational energy usage

List of figures

Figure 1.1: Development of global mean temperature [4]	1
Figure 1.2: Own depiction of relative final energy usage in Sweden (2013), referring to Energimyndigheten [3].....	2
Figure 1.3: Heat supply chain referred to this experimental study (own depiction)	4
Figure 1.4: Percentage of dissatisfied people dependent on floor temperature [1].....	6
Figure 1.5: Boundary layer development on a heated vertical plate [11].....	8
Figure 1.6: Own depiction of heat pump cycle	9
Figure 1.7: Energy flow chart of a basic heat pump based on the “Carnot-process” [1]	9
Figure 1.8: Own depiction of the energy balance of a building	11
Figure 1.9: Relation between baseboard radiator and occupant in a room [19].....	15
Figure 1.10: Built-up of a ventilation radiator [21].....	16
Figure 1.11: Five fans mounted on the cover [19]	16
Figure 1.12: Mounted radiator booster (in the red frame) in operation mode in object of measurement.....	17
Figure 1.13: Air circulation with and without deployment of radiator booster [19]	17
Figure 1.14: Room temperature development of two offices, only one being equipped with a radiator booster [19].....	18
Figure 2.1: a) Ground plan of ground floor of object of measurement and (b) the selected single-family house	22
Figure 2.2: a) Double-panel radiator with radiator booster being installed and b) single-panel radiator with radiator booster being installed and connected to time switch	22
Figure 2.3: a) Own depiction of control cycle of heat pump and b) own depiction of an example for three different heating curves, based on provided manual by manufacturer “IVT Värmepumpar” ²⁴	
Figure 2.4: Schedule of measurements.....	25
Figure 2.5: Locations of measurement devices in the object of measurement.....	25
Figure 2.6: Measurement sample of "wall 1" measurement device	33
Figure 2.7: Measurement sample of "ceiling" measurement device	33
Figure 2.8: Measurement sample of "floor" measurement device	33
Figure 3.1: a) Development of indoor air and outdoor air temperature for period 1 and for period 2 in b).....	39
Figure 3.2: a) Development of floor temperature and b) operative temperature within temperature range, both for period 1 and 2	39
Figure 3.3: Development of supply, return and outdoor temperature in period 1	41
Figure 3.4: Development of supply, return and outdoor temperature in period 2.....	41
Figure 3.5: Estimation of compressor energy use in period 5 by means of extrapolation	44
Figure 3.6: Mean total operational energy usage for periods with and without radiator boosters .	46
Figure 3.7: a) Distribution of warm air without and b) with deployment of radiator boosters	47
Figure 3.8: Difference between actual and set indoor air temperature in period 1 and 2.....	48
Figure 3.9: Calculated space heat loss per square meter floor area of a Swedish 1960er built single-family house based on a norwegian average single-family house up to 2000	49
Figure 3.10: Development of outdoor temperature in the heating season 2015.....	50

List of tables

Table 1.1: Temperature ranges for different heating systems [5].....	3
Table 1.2: Theoretically expected alteration of energy performance influencing parameters	13
Table 1.3: Overview of different kind of innovations being elaborated in order to achieve a more efficient space heating	19
Table 2.1: List of all types of radiators.....	23
Table 2.2: List of measured and calculated data	29
Table 2.3: Overview of measurement devices applied in this experimental study	30
Table 2.4: Results of calibration.....	32
Table 2.5: Summary of expected results of retrofitting with radiator booster	34
Table 3.1: Outdoor air temperature in the different periods.....	37
Table 3.2: Mean values for thermal indoor condition per period.....	40
Table 3.3: Mean values for energy performance per period (1).....	42
Table 3.4: Mean values for energy performance per period (2).....	42
Table 3.5: Occurrences in single periods and countermeasures.....	43
Table 3.6: Revised and cleaned up mean values for energy performance per period	45
Table 3.7: Estimated energy and cost savings based on degree-days method	50
Table 3.8: Summary of absolute and relative deviations from performance indicators.....	52

Appendix A

SMD210KJ375J	10,000	±5 %	3750	2,0 ±0,2	1,25 ±0,2	1,2 Max.	2 mW/°C	300 mW
SMD215KJ400J	15,000	±5 %	4000	2,0 ±0,2	1,25 ±0,2	1,2 Max.	2 mW/°C	300 mW
SMD220KJ400J	20,000	±5 %	4000	2,0 ±0,2	1,25 ±0,2	1,2 Max.	2 mW/°C	300 mW

Source: http://mcshaneinc.com/html/TS67_Specs.html?NewWin (last check:14.06.2016)

Specifications	Documents (1)	My Notes
Manufacturer:	TDK	<input checked="" type="checkbox"/>
Product Category:	NTC (Negative Temperature Coefficient) Thermistors	<input checked="" type="checkbox"/>
B Parameter:	3650 K	<input type="checkbox"/>
Resistance:	15000 Ohms	<input type="checkbox"/>
Power Rating:	-	<input type="checkbox"/>
Tolerance:	5 %	<input type="checkbox"/>

Source:

<http://eu.mouser.com/ProductDetail/TDK/NTCG203NH153JT1/?qs=YdQ7Kj7W0bxLGrB1X8hRwg%3d%3d> (last check: 14.06.2016)

QUICK REFERENCE DATA		
PARAMETER	VALUE	UNIT
Resistance value at 25 °C	3.3 to 470K	Ω
Tolerance on R ₂₅ -value	± 2; ± 3; ± 5	%
B _{25/85} -value	2880 to 4570	K
Tolerance on B _{25/85} -value	± 0.5 to ± 3	%

Source: <http://www.vishay.com/docs/29049/ntcle100.pdf> (last check: 14.06.2016)

Specification	
Item	Performance characteristics
SMD	SMD
Style	0402
Shape	0402
Lead Style	-
B Value	4201 to 4300 K
Resistance Tolerance	+/-5%
Resistance R25	15000 ohm
Packing	Paper Taping(2mm)

Source: <http://industrial.panasonic.com/ww/products/thermal-solutions/ntc-thermistor-chip-type/ntc-thermistor/ntc-thermistorchip-type/ERTJ0ER153J> (last check: 14.06.2016)

Appendix B

Von: **Tobias Andersson** tobias.andersson@proove.se
Betreff: Re: Uncertainty of measurement devices TSS 320
Datum: 30. Mai 2016 um 16:03
An: gantern Nicklas_Caspar.Ganter@Student.Reutlingen-University.DE

TA

Hi Nicklas

Nice to hear from you.

Regarding TSS320

The accuracy for this is as following

0-20 +C is +-1

Bellow 0 and above +20 it is +-2

Hope this is the information you are looking for.

Pls let me know if there are anything else I can help with.

Best regards
Tobias Andersson
Proove AB

30 maj 2016 kl. 21:25 skrev gantern <Nicklas_Caspar.Ganter@Student.Reutlingen-University.DE>:

Dear Mr. Andersson,

my name is Nicklas Ganter. I got your eMail-adress when I called to service of Proove. I am currently writing my bachelor thesis at Royal Institute of Technology in Stockholm in the department of Fluid and Climate Technology. Within my work I measured temperatures. Therefor I used the thermosensor TSS 320. In order to validate the results, I need to know the sensor's accuracy. Unfortunately I could not find any information about this on your webpage. I hope you can help me with this issue.

I am looking forward to your response. Kind regards,
Nicklas Ganter