Investigating different opportunities to increase energy efficiency in buildings by retrofitting Heat pump Coupled heat recovery systems

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Investigating Different Opportunities to Increase Energy Efficiency in Buildings by Retrofitting Heat Pump Coupled Heat Recovery Systems

Doctoral Thesis

By

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Abstract

Energy use in buildings has been a hot topic in Europe since the implementation of the European Parliament directive on the energy performance of buildings. To reach the energy efficiency goals set for the buildings it is vital that energy efficiency measures are implemented into the existing building stock.

Two important sources of heat loss in a building are the exhaust air of the ventilation system and the drain water discharged from the building. It would be desirable to reduce the heat loss from these sources; however, the temperature level of the discharged heat is often so low that it is difficult to use a passive heat recovery system. In these cases, an active system that uses a heat pump may be an interesting concept.

The investigations presented in the present thesis were carried out through a combination of experimental and numerical work. Investigations of the drain water heat recovery system were mainly experimental and the analysis of the ventilation heat recovery systems was mainly performed through simulations. Two different test facilities were built: one test facility for investigation of drain water heat recovery efficiency and one test facility for a run-around ventilation heat recovery system.

Investigations of the drain water heat recovery system revealed that the system's ability to recover heat depends on the flow rate of the drain water; therefore it was important to investigate how the system performs under transient conditions similar to a real drain waster heat recovery system. The results show that the heat recovery ratio was 69 %, for a low flow scenario 45 % for an average flow scenario and 34 % for a high flow scenario. The investigation also indicates that sizing of the heat pump is important. If the heat pump used in the test facility is doubled in size, the heat recovery ratio increases from 68 % to 81 %.

To investigate the possibilities for ventilation heat recovery systems, a run-around coil and a thermal energy wheel heat recovery system were investigated. Simulations show that the efficiency for a run-around coil heat recovery system can be increased by 18 percentage points by retrofitting a heat pump in a Stockholm case. For the thermal energy wheel heat recovery system, simulations indicate that the annual recovery rate can be improved by 24 percentage points by implementing a heat pump system.

Keywords: Drain water heat recovery, Ventilation heat recovery, Energy efficiency, Heat recovery, Heat pump heat recovery system.

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On a personal level I would like to give some love to my dear friends Jimpa, Niiiinis, Pralle, Sami P. and Tiiiiiitis you are probably the reason I'm still somewhat sane. To my sister Ami who is almost a real Dr (only PhD in medicine), you are still 2 years younger and a girl ;-). Love to my dad who is always there for me when I need help. To Sssssss, you are amazing in every way ♥. I also would like to acknowledge my 264 Facebook friends; I don't know how I would cope with all the boring work-related tasks without you. And Google my friend, you are the best!

Finally, the first line of an old The Hollies song reads "*The road is long. With many a winding turn. That leads us to who knows where..*". This is an excellent start to a song that describes my years as a PhD student in a good way. The song describes the unstoppable force that can be generated by a single person to tackle a task that seems unbearable to others. So Google the song and let it play while reading the rest of the thesis. This thesis ain't heavy, he's my brother...

Jörgen Wallin

Stockholm, June 2014

Publications

The thesis is based on the following publications:

Journal papers

- Wallin, J., Claesson, J., *Investigating the Efficiency of a Vertical Inline Drain Water Heat Recovery Heat Exchanger in a System Boosted with a Heat Pump,* Energy and Building Journal 80, pp. 7–16, September 2014, DOI:10.1016/j.enbuild.2014.05.003.
- Wallin, J., Claesson, J., *Analyzing the Efficiency of a Heat Pump Assisted Drain Water Heat Recovery System that uses a Vertical Inline Heat Exchanger,* Accepted for publication in Sustainable Energy Technologies and Assessment Journal, 2014.
- Wallin, J., Madani, H., Claesson, J., *Run-around coil ventilation heat recovery system: A comparative study between different system configurations*, Applied Energy 90, pp. 258–265, February 2012.
- Wallin, J. Claesson, J., *Improving heat recovery using retrofitted heat pump in air handling unit with energy wheel*, Applied Thermal Engineering 61, pp. 823-829, November 2013.

Other publications by the author not included in the thesis

- Wallin J., Madani, H., Claesson, J., *Ventilation heat recovery with run around coil: System analysis and a study on efficiency improvement – Part I*, Proceeding of ASHRAE Region-At-Large conference, Kuwait, 2009.
- Madani, H,. Wallin, J., Claesson, J., *Ventilation heat recovery with run around coil: System analysis and a study on efficiency improvement – Part II*, Proceeding of ASHRAE Region-At-Large conference, Kuwait, 2009.
- Madani, H,. Wallin, J., Claesson, J., *Retrofitting a variable capacity heat pump to a ventilation heat recovery system: modeling and performance analysis*, Proceedings of ICAE 2010, Singapore, 2010.
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- Wallin, J. Claesson, J., *P22 - Modell för identifiering av lämplig effektivisering av energitekniska system med värmepumpar i befintligt byggnadsbestånd – När/Var/Hur?,* Reviewed technical research report, EFFSYSII, 2010

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- Ngo, P., *Evaluation and Analysis of Chillers at SAAB AB, Järfälla (Utvärdering och analys av kylmaskiner på SAAB AB, Järfälla)*, Master thesis, Royal Institute of Technology (KTH), School of Industrial Technology and Management, 2013.
- Bjurling, F., Ngo, P., *Energy Recovery from Sewage Water (Energiåtervinning ur avloppsvatten)*, Bachelor thesis, Royal Institute of Technology (KTH), School of Industrial Technology and Management, 2011.

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1 Introduction

Energy use in buildings has been a hot topic in Europe since the implementation of the European Parliament directive on the energy performance of buildings (European Parliament, Council, 2002) in late 2002. The directive concludes that 40 % of the total final energy use can be attributed to the residential and tertiary sector, a sector which mainly consists of buildings. In 2007, the European Council implemented the 2020 goals for energy and climate change. The 2020 goals include the 20-20-20 targets (European Commission, 2010) where one of the keys is a 20 % increase in energy efficiency in the European Union. The legislation surrounding the energy efficiency came into force in 2012 (European Parliament, Council, 2012), to ensure that the member states will meet the 20 % increase in energy efficiency.

To reach the energy efficiency goal set for the buildings in the European Union, it is vital that energy efficiency measures are implemented into the existing building stock. This is, of course, because compared to the number of existing buildings, new buildings are quite few.

Breakdown of the energy use in buildings reveals that heating, ventilation and air-conditioning (HVAC) systems in buildings account for half of their total energy use (Perez-Lombard et al., 2011). Together with the tap water heating, sometimes referred to as domestic hot water (DHW), they represent a large portion of the energy use in the average building.

As an example of breakdown of energy end use, data describing the energy use in the United States residential building sector (U.S. Energy Information Administration (EIA), 2013) is presented in Figure 1.

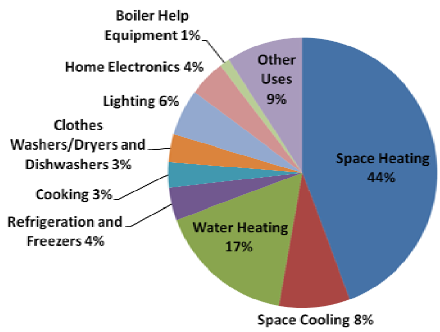

Figure 1 - End use in U.S residential building stock, data from EIA 2013.

The energy supplied for space- and water heating in the building will be lost through

- Transmission – Losses through the building envelope
- Ventilation – Losses through the ventilation system
- Infiltration – Losses through air leaks in the building envelope
- Drain water – Losses in the drain water system

The ratio between these will be different depending on the ambient conditions, design of the building envelope, type of ventilation system and type of activities in the building.

In Figure 2, heat losses from a building over one typical year are illustrated (Elmberg et al., 1996), visualizing how the different losses relate to each other in a general case. It should be noted that when the building envelope is improved (as in the future buildings), the losses from ventilation and drain water will become the most important heat losses in buildings.

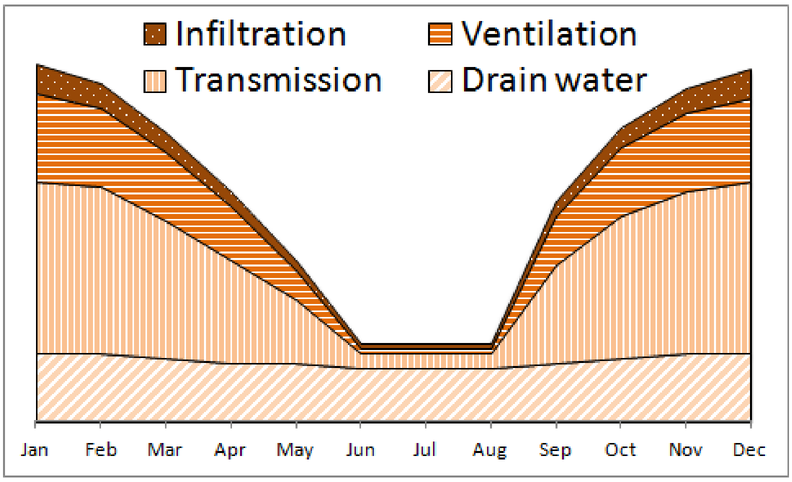

Figure 2 – Illustration of the heat loss from a building over the year. Adapted from "Hus i Sverige – perspektiv på energianvändningen" (Elmberg et al.1996)

A breakdown of the heat losses for a more specific case is presented in Figure 3; in this case, losses from a Swedish multi-family dwelling are presented according to the Swedish Energy Agency (Ekelin et al., 2006).

In light of the information in the presented figures, it is interesting to address the energy use of the ventilation and drain water systems in order to increase the energy performance of buildings since these two commonly account for about half of the heating demand.

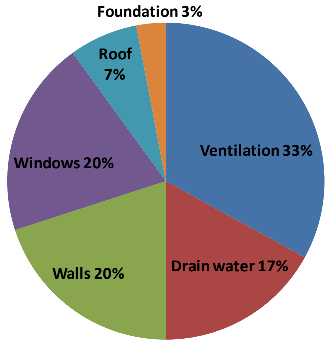

Figure 3 – An example of breakdown heat loss in a Swedish multifamily dwelling built before 1940 with a heat demand of 185 kWh/m^2 annually. Source: The Swedish Energy Agency (Ekelin et al., 2006)

Heat is lost in the exhaust air of the ventilation system and in the drain water discharged from the building. This fact introduces a possibility to increase the energy efficiency of these systems by recovering a part of the discharged heat. For old systems where heat recovery is already installed, there might be a possibility of increasing the efficiency of those systems.

However, the temperature level of the heat discharged from the building is often so low that it is difficult to utilize this energy in a useful way with a passive system. In these cases, a viable concept could be an active heat recovery system using a heat pump to increase the temperature level of the recovered heat.

Several different types of heat recovery systems are available for ventilation and drain water. The system design varies from case to case; there are, however, a few standard designs and system setups that are commonly used.

Drain Water Heat Recovery Systems:

Since a large part of the heat loss in many types of buildings can be attributed to losses in the drain water, heat recovery from the drain water could be one measures to reach the energy efficiency goal set within the European Union. Heat from the drain water in buildings can be extracted with different methods, for example (illustrated in Figure 4):

- Vertical falling film heat exchanger
- Horizontal tilted shell and tube heat exchanger
- Storage tank heat exchanger

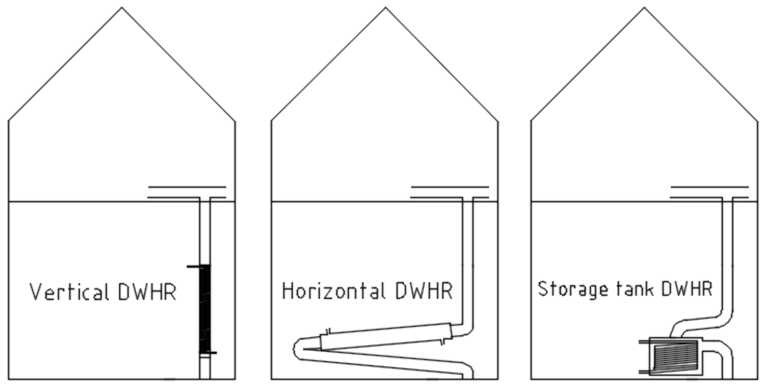

Figure 4 – Example of different ways
to extract heat from drain water

In the most traditional system used for retrofitting, the recovered heat is used to increase the temperature of the incoming cold water to the building. In this way, the hot water (DHW) is directly preheated by the heat recovery system. The traditional system setup only allows for the recovered heat to be used for DHW. For a more flexible heat recovery system configuration, a heat pump can be added to the design. The heat-pump-assisted design makes it possible to increase the temperature of the recovered heat and therefore the heat can be used in other installations.

In the literature, some studies can be found on the topic; Cooperman et al. (2011) described how heat can be recovered from the drain pipe by pre-heating the incoming tap water. They presented three different ways to connect the system. Eslami-Nejad and Bernier (2009) investigated the impact of greywater heat recovery by simulating demand in TrnSys. In their model, they also heat-exchanged the drain water with the incoming tap water. They concluded that the heat recovery ratio[1] varied between 10.4-21.5 % depending on the situation. Zaloum et al. (2007) investigated 5 different standing falling film heat recovery heat exchangers also with the outside coil pre-heating the incoming tap water. Their conclusion was that 9-27 % of the gas to the water heater could be saved. The amount is dependent on the heat exchanger design and the water draw profile of the building. For situations when no showers were taken, the savings of gas to the water heater were between 3.3-5.4 %. To increase

[1] The heat recovery ratio is defined as the power performance ratio of the system in the present thesis.

the heat recovery ratio, a heat pump can be fitted to the heat exchanger, Baek et al. (2005) concluded that for a hotel with a sauna 90 % of the heating load can be covered by using a heat-pump-assisted recovery system; TrnSys was used to do the analysis.

The conducted literature survey provides information on a number of investigations into heat recovery from drain water. Most of these studies are on systems that use incoming water mains coupled directly to the heat exchanger. In a master's thesis, Anders Nykvist (2012) presented a survey on the future potential for drain water heat recovery systems. The thesis presents different types of system that can be installed to recover heat from drain water in multi-family buildings and their potential. Results from the Nykvist (2012) investigation are that a passive system recovers between 10-25 % of the heat in the drain water, whereas for active systems using a heat pump the indicated heat recovery ratio is 50-70 %.

Heat Recovery Ventilation Systems:

There are several common types of heat recovery systems used for ventilation:

- Rotating heat exchanger system
- Cross-flow heat exchanger system
- Run-around coil heat recovery system
- Heat pipe heat exchanger system
- Exhaust air heat pump
- Using return air recirculation

All of the systems have advantages and disadvantages. Rotating heat exchangers have the advantage of being efficient but with the disadvantage of leakage between the exhaust and supply air. The performance and operation of heat recovery systems have been investigated by a number of researchers. For example, Roulet et al. (2001) investigated the real heat recovery of 13 air handling units. Fehrm et al. (2002) investigated different system setups of exhaust air heat pump recovery systems, making economic and technical comparisons. Juodis (2006, 2003) investigated how the heat recovery of the ventilation system operates if the effects of internal gains are considered. Lazzarin et al. (1998) investigated technical and economic analysis of recovery systems for ventilation. Riffat et al. (1998) studied the efficiency of different types of heat pipe heat recovery systems used in naturally ventilated buildings. With regard to analyzes related to run-around coil heat recovery systems, Emerson (1984) , Forsyth et al. (1988a, 1988b) and Zeng et al. (1992) studied the design and efficiency.

The investigations mentioned above shed light on many of the interesting aspects of heat recovery in ventilation. However, higher energy prices and the legislation within the European Union to reduce energy use in the building sector provide new incentive and opportunities to improve existing heat recovery systems. Very few publications in this field deal with the implications that surround this type of efficiency measure.

1.1 Motivation

As mentioned previously, the European Union has adopted legislation to reduce the energy use in the built environment. To reach the goals set, the energy efficiency of the existing building stock needs to be improved. Analyzing the characteristics of the energy demand of the building stock, it can be concluded that one of the main target areas for energy conservation is the heating demand for ventilation and hot water (DHW).

Existing buildings can be divided into two categories: buildings with upcoming major renovation and buildings with no upcoming major renovation.

Many of the existing buildings need no major renovations in the near future and therefore measures to reduce the heating demand of these buildings has to be easy to implement without the need for any major renovation.

Buildings where major renovation is being undertaken have a different set of possibilities available to them compared to buildings without any renovation need.

One way to meet the criteria identified for designing systems that are easy to install and that limit the need of refurbishment is to use heat pumping technologies to increase the efficiency of existing heat recovery systems for ventilation and to recover heat from the drain water. The heat pump is critical to obtaining the flexibility necessary in a system that is easy to install but still delivers high efficiency.

Very little attention has been given this topic in the research community and therefore it is important to investigate the role heat pumps can play when addressing the energy performance of existing buildings.

The main motivation of the present thesis is to investigate interesting heat pumping systems that can be used to decrease the heating demand in buildings without any major need for renovation.

1.2 Overall Aim of the Study

The present thesis investigates different methods to reduce the heat losses from ventilation systems that already have a heat recovery system installed and drain water systems in existing buildings. The focus has been on analyzing different system options where a simple plug-and-play type of heat-pump-assisted heat recovery system can be retrofitted without the need for major refurbishment of the building or system.

By investigating different options to reduce the heat losses, this thesis aims to provide a decision-making foundation for building owners who are considering different alternatives to increase the energy performance of their buildings.

1.3 Structure of Thesis

The present thesis is based on a collection of scientific papers published and submitted to reviewed journals. Chapter 1 introduces the reader to the subject and puts forward the motivation and aim of the thesis and briefly introduces findings from the literature survey. The literature survey is elaborated on in Chapter 4 and 5. Chapter 2 present general information regarding heat recovery ventilation and drain water heat recovery systems. Chapter 3 gives an overview of the methodology of the work performed. In Chapter 4, the investigation of the characteristics and performance of the proposed drain water heat recovery system is presented. Chapter 5 presents the investigations of two different heat recovery ventilation systems. In Chapter 6, the results of the investigations are discussed and in Chapter 7 the conclusions are presented. Chapter 8 puts forward some ideas for future work. After Chapter 8, the Nomenclature and References relevant to the work are presented. Appended at the end of the thesis are the four scientific papers on which the thesis is based.

2 Heat Recovery in Ventilation and Drain Water Systems

Heat recovery systems are essential in order to achieve high energy efficiency in buildings. The main areas where heat recovery is possible are ventilation and drain water. Heat recovery from ventilation is far more common than heat recovery from drain water, at least in installations in the residential sector. Installations of drain water heat recovery systems for greywater in hotels, sports facilities and laundries are the most common. Experiences from earlier installations in Sweden have not been promising, mainly due to large demand for maintenance (Bergrén, 2009).

2.1 Describing Drain Water Heat Recovery Systems

Concerning drain water heat recovery systems in buildings, it is important to distinguish which type of drain water the heat is recovered from. Basically, the drain water can be categorized as greywater or blackwater.

Greywater is usually defined as the water coming from sources in the building other than toilets; this water contains a low fraction of solid materials.

Blackwater is then usually defined as drain water from all of the sources in the building, including toilets. This water may contain a significant fraction of solid material.

Heat can be recovered more easily and efficiently from greywater than from blackwater since the greywater can be used in efficient heat exchangers with a limited need for filtering. Blackwater can only be used in heat exchangers that are adapted to fluids containing a solid fraction.

The system configurations available for a specific installation will depend on the following set of parameters:

- Drain water composition (grey- or blackwater)
- Design stage (if it is before the building has been built or if it is a system retrofitted to an existing building)
- Scope of the installation (only one building or a cluster of buildings)
- Size of the building (the flow rate and pattern of the drain water flow)
- Type of activities in the building (resulting flow rate and temperature of drain water)

2.1.1 Drain Water Characteristics

Knowledge about the drain water characteristics is important when designing a heat recovery system. The temperature, flow profile and composition are the most important parameters to know for the specific case.

Composition of Drain Water

According to the company that is responsible for collection and treatment of the sewage water in Stockholm (Stockholm Vatten), the content of drain water is hard to describe since people dispose of a wide variety of chemicals, liquids and solids into the drain. The only thing they can say with certainty is that there are detergents, chlorine, caustic soda and human excrement and that the pH value is in the range of 7-8 (Mattsson, 2013).

Few scientific investigations have been found which present the composition of drain water in residential buildings. Almeida et al. (1999) conducted an investigation and developed a model to predict the composition of the drain water in a residential building, Table 2-1 presents results from this model.

Table 2-1 – Quality of drain water from residential buildings (% of total volume or mass) (Almeida et al., 1999)

Appliance	Volume	CODt	NH3 ±N	NO3 ±N	PO4 ±P	TSS
WC	30.8	43.9	97.1	3.8	79.8	77.4
Kitchen sink	13.0	23.2	0.3	38	9.4	10.1
Wash basin	12.6	1.7	0.1	10.7	1.3	2.1
Bath	15.7	2.5	0.6	15.3	1.1	1.3
Shower	11.7	6.4	0.7	24.6	4.1	5.1
Washing	16.2	22.3	1.2	7.6	4.3	4.0
Total per person/day	0.1 m3	111.88 g	2.38 g	0.37 g	6.68 g	56.10 g

The conclusions from the Almeida et al. study are that the largest source of the drain water volume comes from the toilets and that the kitchen drain and toilets account for the majority of the solids in the drain water (TSS).

Temperature of Drain Water

Average temperature of drain water is an important parameter to consider when designing a heat recovery system. Investigations indicate that the average temperature of blackwater is somewhat lower than greywater. One study on blackwater from a multi-family dwelling shows that the average temperature is 27 °C (Bergrén, 2009), while another study presents average temperatures between 23 to 26 °C of the blackwater from multifamily and student residences (Seybold and Brunk, 2013). In two other studies, the average temperature of greywater in both a multifamily dwelling and a recreation center is around 30 °C (Grette and Hallstedt, 2004; Jonsson, 2005).

Drain Water Flow Profile

Flow in drain water system is fairly complicated since the flow is intermittent and can be considered as a three-phase flow (water, air and solids) (Wise, 2002). In standing drain pipes, the flow is annular and in the horizontal branches, the flow is free surface flow (Wise, 2002). The flow is gravity-driven which leads to different velocities throughout the drain water sys-

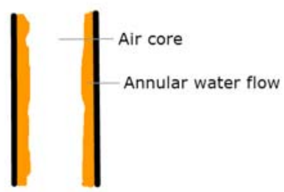

Figure 5 - Annular flow in standing drain water pipe

tem.

It is common practice to have vertical drain water stacks open to the ambient atmosphere at the top to allow for ventilation of foul air; by doing so, air will be ejected by the falling water down into the sewer system. This means that the annular flow in standing drain water pipes will consist of a drain water film on the inside of the drain pipe wall and a core of air traveling in the same direction. Figure 5 present how the water film clings to the wall of the pipe creating the annular flow pattern. This behavior is important to a vertical heat recovery heat exchanger as it enhances the heat transfer properties compared to flow in a full pipe.

2.1.2 Drain Water Heat Exchangers

As mentioned earlier, depending on the characteristics of drain water, different heat exchangers are available for the system designer.

Vertical Heat Exchangers (Gravity Film Heat Exchangers):

Two different types of heat exchangers are common in systems where the heat exchanger is installed vertically to recover heat from drain water.. The heat exchanger is usually either a “double pipe” type or the type where a tube is wound on the outside of a center tube. Examples of the two types of heat exchangers are presented in Figure 6.

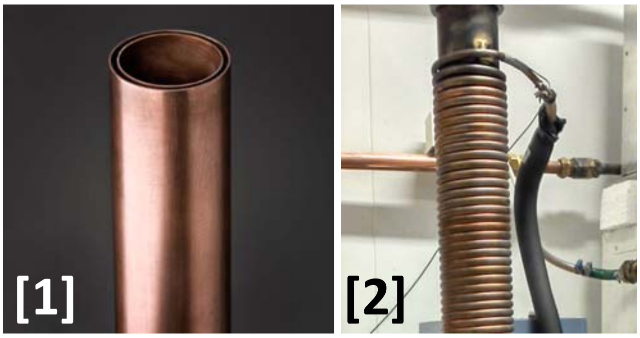

Figure 6 – Two different vertical heat exchangers used. [1] Double pipe (used with permission from 1 World Solar Ltd.), [2] Coiled outer tube

The vertical type of heat exchanger introduces the possibility of easy installation compared to the horizontal type where a portion of the drain system needs to be redesigned. For the vertical type, a portion of the standing drain pipe is to be cut and replaced by the heat exchanger. The heat exchanger is installed using flexible couplings between the existing drain pipe and the heat exchanger.

The effectiveness of 5 different vertical heat exchangers has been studied and reported by Zaloum et al., (2006). That study concluded that the effectiveness ranged between 0.40-0.56 for a steady state case with a flow rate that is equal or close to a shower with a low flow shower head. The tested heat exchangers have different length and mass.

Fouling is an important topic for heat exchangers and especially the ones subjected to complicated fluids like drain water. Vertical heat exchangers have the advantage over horizontal heat exchangers of not being affected by sedimentation of solid material. However, bio-fouling is still a risk in these installations; no scientific studies have been found presenting long-term analysis of performance degradation of vertical heat recover heat exchangers.

Horizontal Heat Exchangers:

Heat recovery systems for drain water in buildings are not common compared to other types of heat recovery system. A common place where heat recovery systems for drain water are installed in Sweden is in facilities that generate large quantities of greywater, for example, in sporting facilities. In these cases, heat exchangers installed in a horizontal orientation are often used. Other places where horizontal heat exchangers can be used are in shower installations.

The horizontal shell and tube type is often used in greywater heat recovery systems in recreation centers and similar buildings. In those buildings, the heat exchanger is often horizontal since the flow rate is high and in order to obtain a high heat recovery ratio a large surface area is required. Different designs of shell and tube heat exchangers are available.

In residential buildings where separation of greywater and blackwater is not performed, a horizontal heat exchanger might require a higher degree of maintenance than a vertical heat exchanger.

Horizontal plate heat exchangers are not so common in drain water heat recovery systems, but one exception where plate heat exchangers are used is for shower installations where the compactness makes it possible to install and recover from a single shower. A study performed by Wong et al. (2010) concluded that 4-15 % energy savings can be achieved with a single-pass horizontal heat exchanger. If horizontal plate heat exchangers are uncommon, the vertical plate heat exchangers are almost unheard of. Plate heat exchangers are not suitable for fluids with large solid fractions and therefore not suitable for blackwater heat recovery systems.

Many different heat exchangers especially developed for installation in showers can be found today. The design can be either for integration or as a module to put on the floor of the shower. Many different products are available on the market. One example of a drain water heat recovery heat exchanger is presented in Figure 7.

Figure 7 – Shower heat recovery heat exchanger (used with permission from 1 World Solar Ltd.)

Storage Heat Exchangers:

Depending on whether the drain water includes water from toilets or not, different types of storage heat exchangers are available. Greywater solutions are most common at present, but solutions for blackwater are available.

In one example of a system, the heat exchanger is a tank with an integrated coil (Nykvist, 2012). Since drain water will be standing in the tank, fouling is an issue and some type of automated cleaning is often integrated in the tank. If the discharge from toilets is included in the drain water, the demand for cleaning increases.

2.1.3 Drain Water Heat Recovery in Buildings

In buildings, the heat from the drain water is normally recovered to preheat the tap warm water or to space heating. Recovery to space heating requires the use of heat pumping technology to be able to discharge the recovered heat into the space heating system. Recovery to preheat tap warm water can be made by direct exchange between incoming water mains and the drain water. Oak Ridge National Laboratory suggests three different installation methods where the incoming mains water is preheated (Oak Ridge National Laboratory, 2005). The installation methods are presented in Figure 8, Figure 9 and Figure 10.

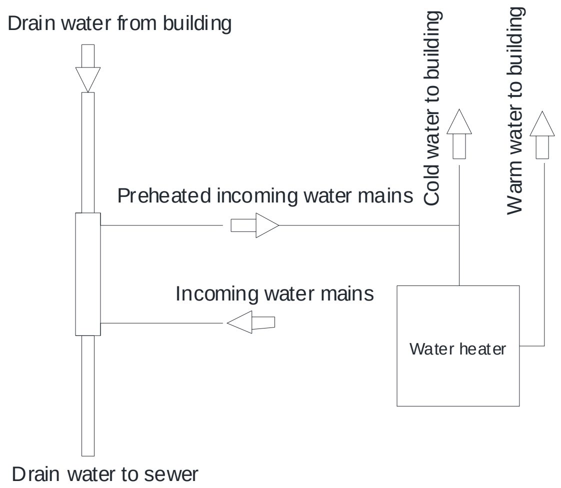

Figure 8 – Balanced flow installation of heat recovery heat exchanger

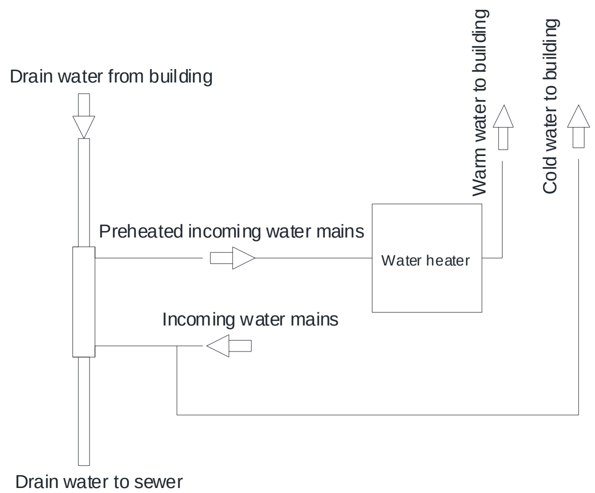

Figure 9 – Example 1 of Unbalanced flow installation of heat recovery heat exchanger

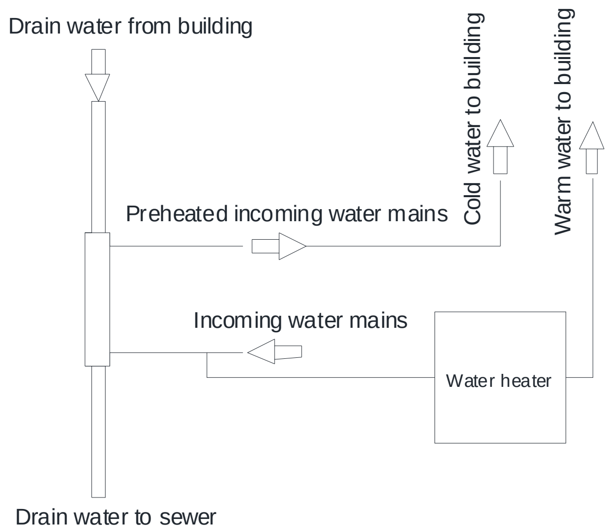

Figure 10 - Example 2 of Unbalanced flow installation of heat recovery heat exchanger

Figure 8 describes an installation method with balanced flow; this means that the flow rate in the drain is the same as the flow rate in the coil. The other two installation methods are unbalanced flow installations, meaning that the flow rate in the drain is higher than the flow rate in the coil.

Balanced flow leads to the highest heat recovery ratio (Oak Ridge National Laboratory, 2005) due to the fact that when the unbalanced installation method is chosen, the drain water flow rate in the drain water pipe is higher than the flow rate in the outside shell. This is because the drain water consists of both cold and warm water while the flow in the outside shell only consists of the cold or warm water depending on the installation method. The balanced flow installation raises the temperature of the cold water into the building which for some cases might not be desired.

Installation of a system with a heat pump can be made in many different ways; one example of installation is presented in Figure 11 where the heat pump is connected in series with the water and space heater. Both heaters need to have auxiliary heaters to ensure the specified temperature level when recovered heat does not cover the entire demand.

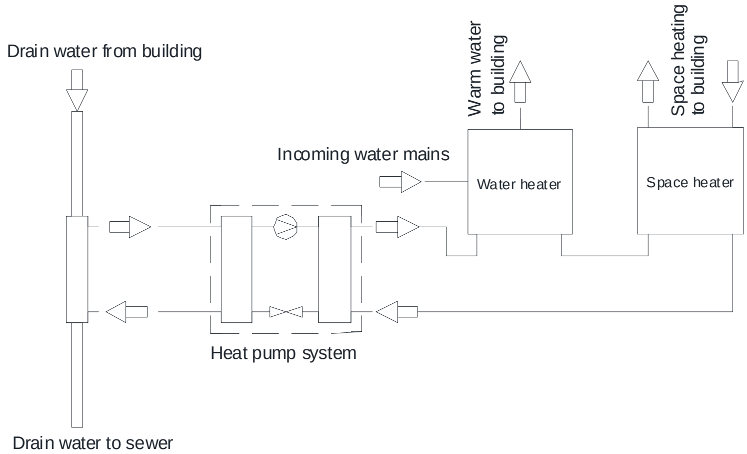

Figure 11 – Example of heat exchanger installation with heat pump

Most of the investigations related to drain water heat recovery found in the literature survey focus on greywater, which is a problem for most of the buildings that are already built. In these buildings, separation of grey- and blackwater usually means that the drain water pipes need to be rebuilt, leading to a large investment cost. There are, however, applications for separation of grey- and black water ("International Wastewater Systems," 2014); these systems are predominantly used for larger applications. Separation is commonly performed by implementation of a filtering process to remove the solid fraction of the drain water. Figure 12 present an example of a drain water filter.

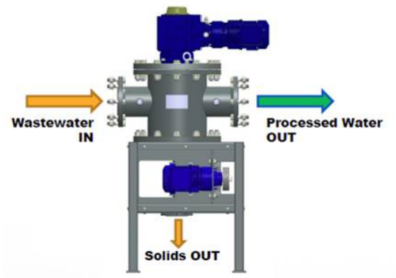

Figure 12 – Example of drain water filter (used with permission from International Wastewater Systems)

The separated greywater is pumped to a heat exchanger and the solid fraction is transported to the sewer pipe. With this method, heat can be extracted from blackwater with a more efficient heat exchanger than in-line heat exchangers.

2.2 Describing Ventilation Heat Recovery Systems

As concluded in the Introduction of this thesis, one of the largest sources of heat loss from a building is the ventilation heat loss. The amount of heating needed for air handling is dependent on the ambient climate and the demanded ventilation flow rate. Installation of a heat recovery system can reduce the heat demand by a great deal. The type of recovery system appropriate in a specific case will be dependent on the type of ventilation system installed, on whether exhaust air is allowed to mix with supply air and on the placement of the fan units.

The efficiency of the heat recovery system will be affected by the type, size and properties of the heat exchanger and its surface.

In the "Introduction" of the present thesis, six common types of heat recovery systems were presented.

- Rotating heat exchanger system
- Cross-flow heat exchanger system
- Run-around coil heat recovery system
- Heat pipe heat exchanger system
- Exhaust air heat pump
- Using return air recirculation

These systems can be divided into three categories: systems with regenerative heat exchanger, systems with recuperative heat exchanger and systems without heat exchanger.

Regenerative Heat Recovery Systems

A regenerative heat recovery system has a heat transferring mass that alternates between the exhaust- and supply-air side. This is usually done by having a wheel-shaped mass (heat exchanger) that is rotating between the hot and the cool side of the air handling unit. This means that the exhaust and supply sides need to be adjacent to each other. Two different types of regenerative heat exchangers are available: exchangers to recover sensible heat (thermal energy wheel) or exchangers to recover both sensible and latent heat

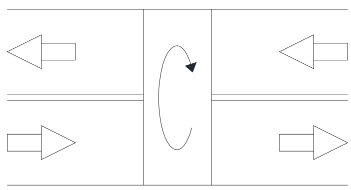

Figure 13 – Rotating regenerative heat exchange

(enthalpy energy wheel). In order to recover both the sensible and latent heat, the surface is usually coated with a hygroscopic material.

Since the mass alternates between the airstreams, leakage between the airstreams cannot be avoided. The leakage can be in the range of 5-10% (Abel et al., 2008). Figure 13 presents a schematic regenerative heat recovery system.

Typical temperature efficiency for this type of system is 0.7-0.8 (Abel et al., 2008). Regenerative heat exchangers do not have the same risk of frost growth as recuperative types, due to the fact that the heat transferring surface alternates between the warm and cold side of the air stream.

Recuperative Heat Recovery Systems

Recuperative heat recovery systems can be either direct or indirect. In a direct recuperative system, the two air streams are separated by a heat transferring wall. The heat is transferred from the hot air stream to the cold air stream through the walls of the heat exchanger. The walls are usually relatively thin to promote heat transfer properties. This means that the two air streams need to be placed side by side. Figure 14 presents an illustration of an air-handling unit with direct recuperative heat exchanger. There is a risk of leakage between the airstreams through the joints in the heat exchanger. These systems typically have a temperature efficiency between 0.6-0.8 (Abel et al., 2008), which may result in the temperature of the exhaust air being too low. If humidity is generated within the building, there is an imminent risk of frost growth on the heat exchanger surface at low ambient conditions with this type of heat exchanger.

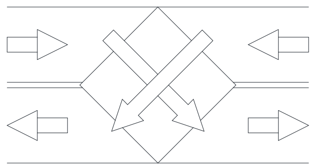

Figure 14 – Direct recuperative heat exchange

Indirect recuperative systems have a secondary heat transferring fluid between the two air streams. Hydronic heat exchangers are installed in each of the airstreams and a fluid is pumped between these heat exchangers. The fluid is typically a water and glycol solution with corrosion inhibitors added, to prevent the fluid from freezing and

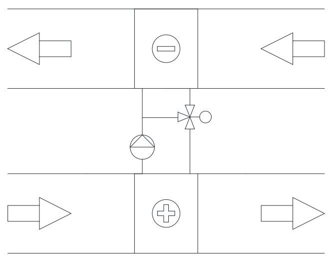

Figure 15 - Indirect recuperative heat exchange

to protect the system from corrosion. Figure 15 presents an illustration of this type of system. Indirect systems have two obvious advantages, the first being that the two airstreams are separated so that no leakage can occur between supply and exhaust airstreams. The second advantage is that the supply and exhaust airstreams can be spaced apart from each other. This type is used in buildings where contamination of the supply air is not allowed or where there are high demands on the quality of the supply air, for example, in hospitals or laboratory buildings. The drawback is that these systems typically have a lower efficiency than other system types, temperature efficiency ranging from 0.5-0.6 (Abel et al., 2008). Maintenance demand is also higher than for other system types.

Of the previously identified common heat recovery systems, the cross flow heat exchanger system and the heat pipe heat exchanger system can be categorized as direct recuperative systems. Exhaust air heat pump and run-around coil heat recovery systems can be categorized as indirect recuperative systems.

System without Heat Exchanger (Direct Exchange)

In this context, the system without heat exchanger refers to a system that uses return air recirculation to reduce the heating demand of the building ventilation. Using return air recirculation is the easiest way to recover heat from the exhaust air. The system recovers heat by mixing the supply air with a portion of filtered exhaust air. A system schematic is presented in Figure 16. This system type was common in buildings with large ventilation demand in the Northern parts of Europe in the ´70s, but these days this system type has been replaced with other types of heat recovery systems. In other parts of the world where the weather is warmer and more humid, like parts of the USA, the use of return air recirculation is still a common practice (Nilsson, 2003). This is because the warm and humid climate introduces the demand for cooling and drying of the supply air. In these climatic conditions, recirculation is considered to be necessary for cost reasons.

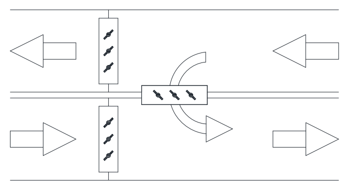

Figure 16 – Return air recirculation system

The main reason for abandoning these types of system and using 100 % fresh air is the fear of supplying the building with undesirable substances that might be found in exhaust air.

2.2.1 Ventilation Heat Recovery Efficiency

For heat recovery systems the efficiency can be defined in several ways. Normally, two different efficiency measures are used: temperature efficiency (η_T) and annual heat recovery efficiency (η_Q). The temperature efficiency describes the efficiency of a heat recovery unit (Equation 2.1); it does not, however, describe how much heat is recovered.

$$\eta_T = \frac{t_{recovery} - t_{ambient}}{t_{return} - t_{ambient}} \qquad \text{Equation 2.1}$$

How much energy is recovered is instead described by the annual heat recovery efficiency, Equation 2.2.

$$\eta_Q = 1 - \frac{Q_{annual\ demand\ with\ heat\ recovery}}{Q_{annual\ demand\ without\ heat\ recovery}} \qquad \text{Equation 2.2}$$

The difference between the two depends on the ambient temperature of the location, the exhaust temperature and the supply temperature.

For a specific case (Stockholm Arlanda, Energy plus weather data file) with the return temperature set to 21 °C, the difference between the temperature efficiency and the annual heat recovery efficiency at five different supply temperatures is presented in Figure 18.

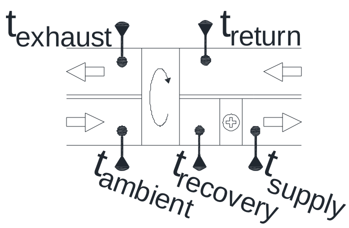

Figure 17 – Nomenclature of tempera-

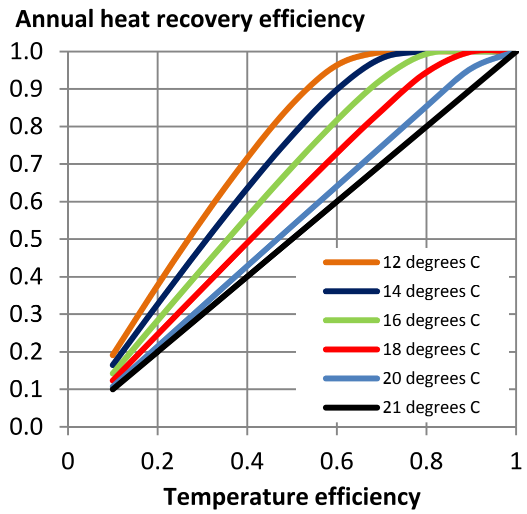

Figure 18 – Example of difference between temperature efficiency and annual heat recovery ratio at different supply temperatures

It can be seen in Figure 18 that when the supply temperature is the same as the return temperature, temperature efficiency and annual heat recovery efficiency are the same. If the supply temperature is lower than the return temperature (the most common case in Sweden), the annual heat recovery efficiency is higher than the temperature efficiency.

2.2.2 Control of Ventilation Heat Recovery Systems

At a certain ambient temperature, the heat recovery system is able to cover 100 % of the ventilation heating demand. This breakpoint temperature ($t_{amb,critical}$) is dependent on the temperature efficiency of the heat exchanger. When the ambient temperature is above this critical level ($t_{amb,critical}$), the temperature efficiency of the heat recovery system needs to be decreased in order to avoid high supply temperatures. In Figure 19 is an example of how the temperature efficiency and the temperature after the heat recovery heat exchanger ($t_{recovery}$) varies depending on the ambient temperature. The chart is valid for a case where the maximum temperature efficiency is 70 %, the supply temperature is 18 °C and the return temperature is 21 °C. Ambient temperatures from Energy plus weather data for Stockholm Arlanda are used for calculations.

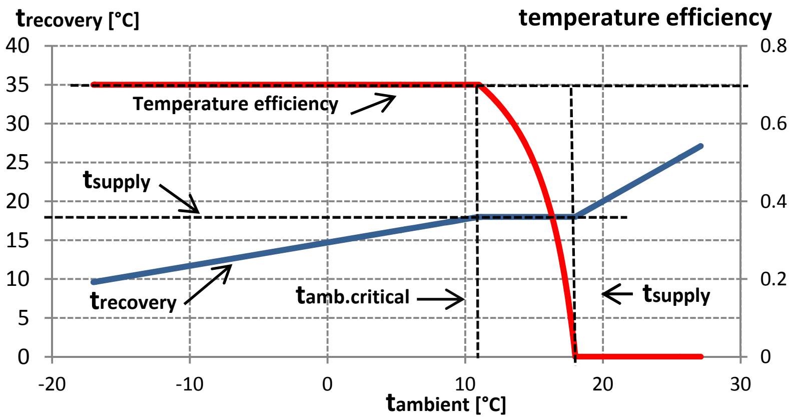

Figure 19 – Temperature efficiency and temperatures after heat re-
covery heat exchanger on the supply side at different ambient tem-

The temperature efficiency is controlled by different methods depending on the system type. In rotating regenerative systems, the control can be carried out by changing the rotational speed of the heat exchanger. For direct recuperative systems, the use of a bypass duct with a regulating damper is common practice. Indirect recuperative systems are controlled by changing the flow rate of the secondary fluid, either with the use of a shunt valve or by controlling the speed of the pump. Systems with return air recirculation are simply controlled by changing the amount of air re-circulated.

3 Methodology

The investigations presented in this thesis were carried out through a combination of experimental and numerical work. In the analysis of the drain water heat recovery system, the work was mainly experimental, but a simple simulation model was created in Excel to evaluate the influence of heat pump and storage tank sizing. The analysis of the ventilation heat recovery systems was mainly performed with the use of the simulation tool TrnSys. However, some experimental work was performed to support the simulations.

3.1 Drain Water Heat Recovery Analysis

To accommodate the analysis, a test facility has been built in a lab setting; the layout of the test facility can be seen in Figure 20. The test facility makes it possible to test the performance of a drain water heat recovery system with a vertical heat exchanger installed in an environment that mimics the situation of a drain water system.

A few different steps were needed to provide the necessary information; firstly, the characteristics of the heat exchanger needed to be established as the baseline for the analysis. With the knowledge of the heat exchanger, it will be possible to compare the results from this investigation with future investigations. This information also makes it possible to produce a semi-empirical model of the drain water heat recovery system.

Secondly, since the flow rate of the drain water in a real building will vary depending on the user behavior, it was important to investigate how this intermittent flow pattern influences the performance of the heat recovery system.

Thirdly, investigation into the size of the heat pump and storage tank is needed in order to describe the design criteria for this type of system.

3.1.1 Design and Operation of the Test Facility

The test facility was built to enable tests to be run continuously and automatically for long periods of time. To meet this demand, the facility was built with four separate liquid loops. In Figure 20, these loops are displayed with different colors. The test facility has more features than a real installation would have; see Figure 11 for a schematic of a typical installation in a building. The blue loop is a continuous circulating loop cooling the storage tank and the green loop is used for the cooling of the drain water heat exchanger. These two loops would also be present in a real installation. The orange loop is used to heat the water in the drain water storage barrel in the test rig. In a real system, this loop would be used to provide heat to any heating load; examples could be a radiator system or a domestic hot water system. The maroon-colored loop is used to pump drain water from the storage barrel to the top of the drain pipe. This loop would not be present in a real installation.

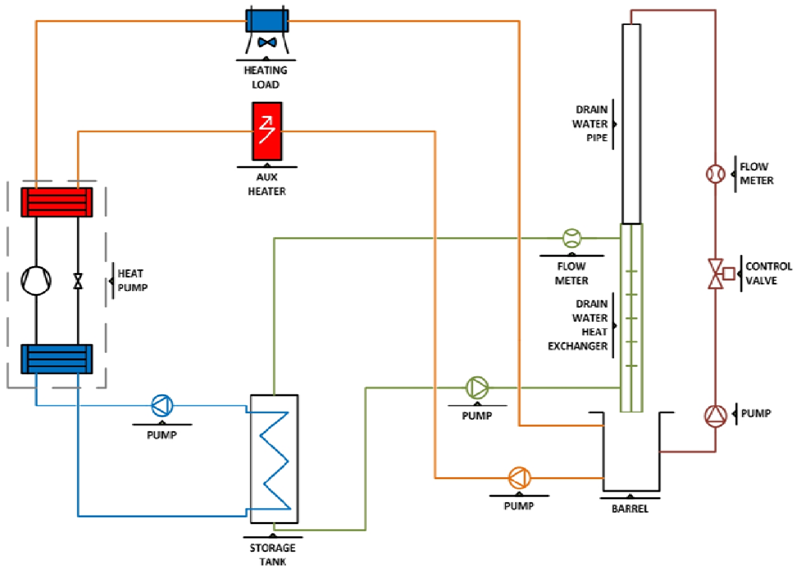

Figure 20 – Schematics of the test facility used

The heat exchanger is made up of a 1100 mm vertical copper pipe with a inside diameter of 70 mm, which gives the heat exchanger an inside heat transfer area of 0.20 m². On the outside of the copper pipe, a second copper pipe is wound to create the heat exchanger. Figure 21 displays the heat exchanger. The outside coil consists of a 25 meter long copper pipe giving the heat exchanger a 0.57 m² heat transfer area of the coil.

In order to reduce the contact resistance between the inner and outer pipes, a waterproof jacket is fitted on the outside of the heat exchanger and the inside of the jacket is filled with water. In this design, the space between the outside and inside pipes is water-filled instead of air-filled, which is the usual case for this type of heat exchanger. This leads to lower contact heat resistance and a more efficient heat exchanger.

Figure 21 – The drain water heat exchanger used

A 200-liter barrel is used to collect the drain water after it has passed through the drainpipe. The barrel is also used as a reservoir for the heat pump condenser cooling. Since the whole purpose of the system is to recover heat from the drain water and transport it back to the building heating system, this barrel can be seen as a part of the building heating system. The temperature in the barrel is held somewhat constant with the help of the hot (condenser) side of the heat pump. The aim is to keep the temperature in the barrel around 27-30 °C as this is what is reported as an average drain water temperature for multi-family houses in Sweden (Bergrén, 2009; Jonsson, 2005).

A storage tank is used to store cold water from the system heat pump; the cold (evaporator) side of the heat pump is connected to a coil in a storage tank, cooling the water in the tank down to the set point of 8 °C. From the storage tank, water is then pumped to the outside coil of the drain water heat exchanger.

The drain water flow rate is controlled by a motor valve, which is computer-operated by a user-designed Excel-based schedule.

The heat pump consists of a small R134a compressor, a thermostatic expansion valve, two brazed plate heat exchangers acting as evaporator and condenser, and a high/ low pressure switch to protect the heat pump is also fitted to the heat pump. The compressor unit run cycle is controlled by the high/low pressure switch. The cooling capacity of the heat pump is somewhere around 800 W under the running conditions in the investigations.

In addition to the previously mentioned components, there is a heater and a cooler installed in the heat pump condenser circuit. The heater is installed to enable the system to be tested at different temperatures in the building heating system and the cooler acts as the building heating load, lowering the temperature of the circuit to the desired drain water temperature.

The system is monitored by a computerized data collection system with the capability of presenting real time data to the operator via a graphical user interface plus collecting and storing data from 18 temperature sensors and two flow meters. The computer system also has the ability to calculate and display real time data of recovered heat and the efficiency of the heat recovery system.

3.1.2 Quality of Measurements

Evaluation of the system requires many points of measured data. Quality in the measurements is essential in order to be confident of the results. Therefore, the temperature sensors are calibrated using a calibrated temperature bath, and a calibration polynomial equation is developed to adjust the readings from the temperature sensors. The temperature sensors used to measure temperature difference over the heat exchanger are calibrated as pairs to increase the accuracy of the temperature measurements.

The flow meters come calibrated from the supplier but in order to attain a better accuracy they are calibrated in situ. The method used in the present research is calibration using gravimetric weighing of the amount of water that flows through the meter. This done by capturing the amount of water that flows through the flow meter over a period of time in a container with a known dry-weight. The container is weighed with a calibrated high accuracy scale to obtain the mass of the water. The measured time and the mass of the water give us the average flow rate over the calibration time. This value is then used to produce a calibration factor for the flow meters.

From the calibration results of temperature sensors and the flow meters, the propagation of uncertainties is analyzed to identify the parameters that have the greatest influence on the uncertainty using the Kline and McClintock (1953) approach. The procedure is described in more depth in Appendix A.

3.1.3 The Experimental Analysis

The experimental analysis was divided into two different investigations: firstly, the system performance and the heat exchanger characteristics are evaluated under steady state conditions to attain knowledge about the system and heat exchanger behavior. The analysis is done using a modified version of the classical Wilson Plot method, developed by Khartabil et al. (1998). This information is essential for the creation of the simple numerical model of the system. It is also important to know the characteristics of the heat exchanger in order to establish the baseline of the investigation.

Secondly, the performance of the heat recovery system under dynamic behavior is evaluated. The dynamic behavior analysis is performed to provide information on how this type of installation will perform in a more realistic environment. A 24-hour schedule is used to create the dynamic flow pattern of the drain water. The schedules were created from measured values of hot water (DHW) in a multi-family building presented in a report by the Swedish Energy Agency (Swedish Energy Agency, 2009a). The choice of using a hot water profile was forced by the absence of any source for measured drain water flow rate in a multi-family building.

With the information from the measurements and the characteristics of the heat exchanger, a simplistic numerical model is created and used to evaluate how the sizing of the heat pump and the storage tank influences the performance of the system.

3.2 Ventilation Heat Recovery Analysis

Two different ventilation heat recovery system types were identified as suitable candidates for the use of a retrofitted heat pump in order to increase the efficiency. The system types were those using a run-around coil loop and those using a thermal energy wheel[2] to recover heat from the ventilation exhaust air stream. The motivation for investigating a run-around coil system is the ease of installation combined with the fact that the heat recovery ratio is usually lower for these than for other types of heat recovery system. The reason for investigating a system using a thermal energy wheel was that the heat recovery ratio is usually the high-

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[2] In the present thesis, a thermal energy wheel is defined as a rotary air-to-air heat exchanger that recovers sensible heat.

est of all the different types, creating a worst-case scenario for the installation.

The investigations were mainly performed through simulations with the software TrnSys. However, some initial experimental work was performed on a run-around coil ventilation heat recovery test facility that was build. The purpose of the experimental work was to attain knowledge on what parameters that influence the performance of these system types. Measurements were also used to validate the run-around coil part of the simulation model.

3.2.1 Run-around Coil Investigations

The investigations on increasing energy efficiency in run-around coil heat recovery systems presented in the present thesis is based on a paper by Wallin et al. (2012) appended to this thesis. This paper is a comparative study summarizing the findings from three earlier conference papers (Madani et al., 2010, 2009; Wallin et al., 2009a). The study mainly focuses on ventilation systems in countries that have a long heating season and where the annual heat demand is far greater than the cooling demand, as in Sweden, Norway and Finland, for example. But the information may perhaps also be of interest in other areas where the climate conditions are different.

3.2.1.1 Methodology - Run-around Coil Investigations

In two previous conference papers by Wallin et al. (2009b) and Madani and Wallin et al. (2009), the performance of run-around coil heat recovery system is investigated. Wallin et al. (2009b) performed experimental investigations to determine factors that influence the heat recovery ratio, focusing on the brine side of the system. The aim was also to accumulate knowledge of the system parameters so that a reliable simulation model could be built in TrnSys (Kline et al., 2007). One of the key findings in the investigation was that the UA-value of the heat exchangers can be considered to be constant even though the ambient air temperature changes. This information was essential when building a model of the run-around coil system with a retrofitted heat pump in TrnSys to be used in the Madani and Wallin et al. investigation (2010, 2009).

TrnSys is a simulation software where transient systems can be analyzed. TrnSys ships with many validated modules that can be combined to build a theoretical model of a thermal system.

To build the model of the run-around coil system in TrnSys, the following five standard modules were combined:

- Climate module (TrnSys Type 15 with using Meteonorm climate file (Remund et al., 2009))
- Heat exchanger module (TrnSys Type 5)
- Pump module (TrnSys Type 3)
- Fan module (TrnSys Type 663-3)
- Control unit module (TrnSys Type 23)

The different modules that were used are described in more detail in the Madani, Wallin et al. paper (2009) and the mathematical reference for the models can be found in Appendix B.

Investigations were divided into a few different steps, the aim of the first step being to build a simulation model of the run-around coil heat recovery system and perform simulations to establish a baseline case. The output was then used for comparison with the results from the second and third step simulations.

The second step investigated how the performance of the run-around coil heat recovery system would be affected if a three-stage on/off controlled heat pump was retrofitted to the system. In order to be able to investigate this scenario a heat pump module was added to the TrnSys model of the run-around heat recovery system. The heat pump module used in the simulation model was TrnSys Type 668, a water-to-water heat pump using performance data supplied by the user. The user-supplied heat pump data are heating capacity and compressor power at different incoming load and source temperatures. The performance data was generated by making a three-stage heat pump model in EES (Engineering Equation Solver) (Kline, 2010), which was validated against experimental data (Madani et al., 2009). The EES model was subsequently used to create the performance data needed as input to the TrnSys heat pump model. The main reason for choosing a three-stage heat pump was to obtain a smooth, efficient operation and a good heat recovery ratio over the wide operating conditions and to better meet ventilation heating demand. The heat demand varies depending on the ambient outdoor temperature: low ambient outdoor temperature gives a higher heating demand and vice versa. In the early stages of the investigation it was found that a single-stage heat pump will have a sizing mismatch, resulting in a low heat recovery ratio. These findings triggered testing to find a heat pump sizing that provides a reasonable heat recovery ratio. Choosing a three-stage heat pump and sizing it for the Stockholm case provided a heat recovery ratio that was close to the theoretical maximum. Table 3-1 provides information about the three-stage heat pump performance.

Table 3-1 – Three-stage heat pump performance table used in simulation

Tw load [°C]	Tw source [°C]	Q_dot_1 [kW]	E_dot [kW]	COP [-]
-5	15	170.0	17.0	10.0
0	15	170.3	20.6	8.3
5	15	170.5	24.1	7.1
10	15	170.5	27.6	6.2
11	15	170.5	28.3	6.0
11	15	123.8	15.8	7.8
12	15	123.7	16.3	7.6
15	15	123.6	17.8	6.9

The model of the run-around coil system with the three-stage heat pump attached was used to perform annual simulations. The result from the simulations could then be compared to the baseline case to establish the change in system energy performance.

Madani and Wallin et al. (2009) found that the sizing of the heat pump is of major importance for the run-around coil heat recovery system to obtain a good efficiency. Therefore, it was interesting to investigate how the system efficiency would be affected if the on/off heat pump was changed to a variable capacity heat pump. In the third step, a speed controlled heat pump module was incorporated in the run-around coil heat recovery simulation model (Madani et al., 2010). In this case, the heat pump module is created and simulated in EES (Kline, 2010). EES is connected to the TrnSys model so that data can be exchanged between the two models at every time step. In EES, the heat pump is modeled as a semi-empirical model by combining mathematical models of compressor, condenser, evaporator and expansion valve. The component models are created using data from measurement and the created complete heat pump model is validated with measured data. The heat pump was sized to cover about 80 % of the annual ventilation heating demand, i.e. the same rate of coverage as for the three-stage heat pump. Table 3-2 shows performance parameters for the variable capacity heat pump.

Table 3-2 – Capacity controlled heat pump performance table used in simulation

Frequency [Hz]	Tw load [°C]	Tw source [°C]	Q_dot_1 [kW]	E_dot [kW]	COP [-]
30	10	15	67.4	7.6	8.9
40	10	15	85.3	10.2	8.4
50	10	15	102.1	13.1	7.8
60	10	15	117.8	16.3	7.2
70	10	15	132.8	19.7	6.7
80	10	15	147.0	23.3	6.3
90	10	15	160.7	27.0	6.0
100	10	15	173.7	30.8	5.6
110	10	15	186.3	34.8	5.4
120	10	15	198.5	38.9	5.1

With the EES heat pump model connected to TrnSys, annual simulations were performed to estimate the recovery rate of the run-around coil heat recovery system using the proposed setup. The output was then compared with the base case and the on/off heat pump system setup established in the earlier investigation. The investigation is described in more depth in the Madani and Wallin et al. paper (2010), which analyzes the modeling and investigation details for the run-around coil heat recovery system with the variable capacity heat pump retrofitted.

3.2.2 Energy Wheel Heat Recovery Investigations

The main objective of the investigation where a heat pump is retrofitted to an thermal energy wheel heat recovery systems is to analyze the possibility of increasing the heat recovery ratio in an air handling unit in which a thermal energy wheel installed. The heat pump extracts heat from the exhaust air stream and "lifts" the temperature to a level enabling utilization of the recovered heat to reheat the return water flow from the ventilation heating coil. This type of system setup should be technically easy to install in many existing ventilation systems.

3.2.2.1 Methodology – Energy Wheel Recovery Investigations

In order to investigate the possibilities of increasing the heat recovery ratio of a thermal energy wheel heat recovery system, a simulation model has been built in the simulation tool TrnSys (Kline et al., 2007). The model is based on the specification and measurements of an air handling unit installed in an office building in Stockholm, Sweden. The aim was to evaluate the seasonal performance of the air handling unit equipped with thermal energy wheel heat recovery and to investigate how the efficiency is influenced by different characteristics of a retrofitted heat pump set up according to Figure 22.

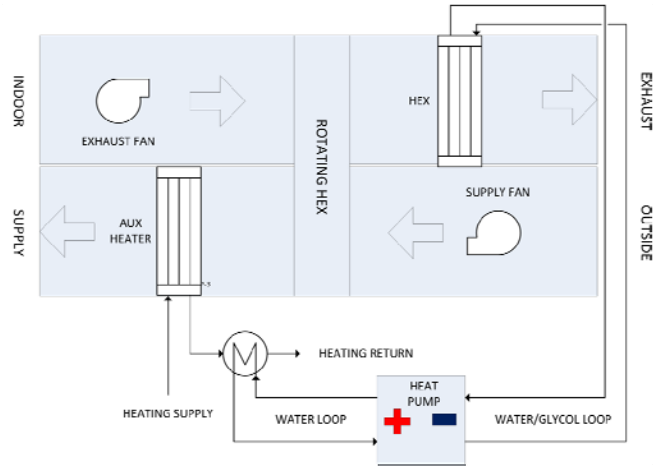

Figure 22 – Schematics of the proposed heat recovery system

Connecting the heat pump to the return on the water side avoids the need for an extra air side heat exchanger; it also enables the possibility of having a higher water side return temperature than the water side supply.

In order to attain relevant results from the simulations, it is important that the model have its roots in reality. Therefore, a simulation model of a real office building has been created in TrnSys and the output from the simulation model of the building has been validated against measurement data. Hourly measurement data from the building has been collected for more than one year. This philosophy allows simulations for any period of time, up to whole-year simulation.

Even though the model has a set value of 0 °C of the minimum temperature of the air after the retrofitted extra heat recovery heat exchanger, the lowest temperature during one time step in the simulation is -1.6 °C. The reason for this is that the control in the simulation model is slow and not working perfectly. This means that the temperature of the water/glycol liquid loop going from the heat pump evaporator side to the retrofitted heat recovery heat exchanger is below freezing at some of the time steps. For these time steps, frost could start to accumulate on the heat exchanger surface, which could perhaps introduce the need for defrosting, which is not considered in the investigation. The controller also shuts down the heat pump when the rotating heat exchanger covers the entire demand of the air handling unit.

The model is qualitatively described as a number of black box models (simulation modules) where the data is sent between the different boxes in every time step. Calculations are carried out with new data in the boxes at every time step. The calculation results for every time step are written to a data file, which can easily be imported in a number of different analyzing software.

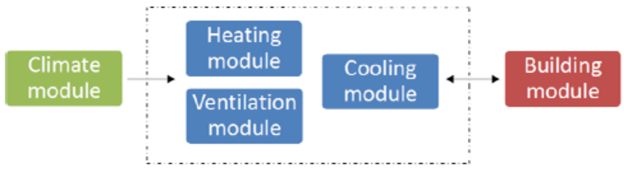

Figure 23 – Qualitative description of the simulation model

The quantitative model of the air handling unit with the heat recovery system is created by using standard validated models supplied with the TrnSys software and the TESS component library (Thornton et al., 2008). Since the real system, which the system is modeled upon, does not have the retrofitted exhaust heat pump recovery, the air handling unit module is validated in the state before the exhaust heat pump is connected to the air handling unit. The heat pump unit is validated separately as a standalone unit before being connected to the model.

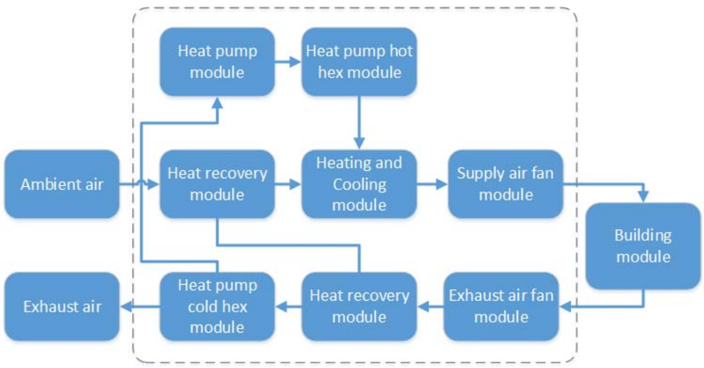

Figure 24 - Quantitative description of the simulation model

The different submodules are connected to each other according to Figure 24 to form the air handling unit model. The different submodels are considered as individual black box models.

In the creation of the air handling unit different TrnSys standard modules are combined to achieve the desired functionality. The submodule types, inputs, outputs and dimensioning data are described in Table 1.

Table 3-3 – Description of the different TrnSys modules used in the creation of the model

Sub module	**TrnSys Type**	**Description**
Exhaust air fan	112b	For exhaust fan a single speed fan module is used. The fan module has the building room temperature as input and outputs a constant air flow rate at a temperature that is the same as room temperature. No consideration is given to any heat gain from the fan. The air flow and temperature from the exhaust fan is connected to the thermal energy wheel module.
Energy wheel	667d	Input to the thermal energy wheel is air temperature and air flow from ambient and the exhaust fan. Design data is the heat exchanger sensible effectiveness. Output from the thermal energy wheel is air temperature and flow rate; the output is connected to the heat pump exhaust heat recovery heat exchanger and the auxiliary heating module.
Heat pump cold exhaust heat exchanger	5e	This heat exchanger is a cross-flow heat exchanger that has exhaust air temperature and flow on one side and water temperature and flow rate from the heat pump cold side on the other. Design data is fluid properties and the overall heat transfer coefficient of the heat exchanger. Outputs are connected to the heat pump cold side and the ambient.

Heat pump hot exhaust heat exchanger	5b	This heat exchanger is a counter-flow heat exchanger that has the return water temperature and flow rate as input on the cold side and heat pump hot side temperature and flow rate on the other side. Design data is fluid properties and the overall heat transfer coefficient of the heat exchanger. Outputs are connected to the heat pump hot side and the ventilation heating system.
Heating	753d	The heating coil has air flow rate and temperature from the thermal energy wheel as inputs on the air side. On the water side, inputs are heating system supply temperature and flow rate, the flow rate is controlled so that the output temperature on the air side is 20 °C. Design data is the liquid side specific heat.
Cooling	508c	Cooling is not used for this simulation.
Supply air fan	664-4	The supply air fan module has the air flow rate and temperature as input and outputs the same air flow rate and temperature. Heat gain from fan is not taken into consideration. The outputs are connected to the building module.
Heat pump	668	This is a water-to-water heat pump that has water temperature and flow rate from and to hot and cold exhaust heat exchangers as input and output. The heat pump interpolates the output performance from a design matrix. The design matrix is created from measurements of heat pump performance for a real heat pump.
Building model	56a	The building module has as input climatic conditions and the design parameters include the building characteristics (walls, roof, windows and so on). Relevant output is room temperature.

All the standard modules in TrnSys have been individually validated but when modules are combined the output needs to be validated. In this case, the model is validated against measured data from the real building that the model is based on. However, since the real building does not have the combined heat pump and thermal energy wheel heat recovery system, it has not been possible to validate the model with this system setup. Instead, the model is validated before the heat pump module is connected to the model. The heat pump model is then only validated as a standalone module by TrnSys and not individually validated in the present thesis, which may introduce some uncertainty.

In the real building, one air handling unit is chosen and measurements have been carried out in order to accumulate data for validation of the heat recovery model. The measurements performed were air flow of supply and exhaust air stream and air temperatures before and after the heat recovery heat exchanger on the exhaust side. This allows calculation of the recovered heat by the heat recovery system; the measured calculated heat is then compared with the simulated recovered heat. The result from this validation can be seen in Figure 25.

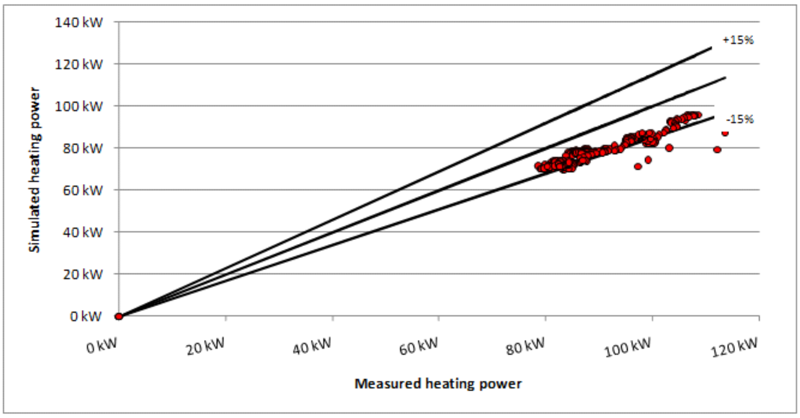

Figure 25 – Validation of recovered heat from the model

In Figure 25 it is obvious that the measured heat recovery data is consistently higher than the simulated data; this could perhaps be due to air leakage or a systematic measurement error. With regard to the air leakage, two possible scenarios are possible: leakage between the outside air and the exhaust air stream causing the measurement of the exhaust temperature to be lower than the real temperature after the heat exchanger. The second possibility is that air leaks over to the supply air stream before the heat exchanger and since the trace gas used to measure the flow rate is introduced into the exhaust duct long before the air handling unit, if there is a leak before the heat exchanger the real flow over the heat exchanger will be lower than the measured flow rate. This causes the temperature difference over the heat exchanger to be larger than it would be if all of the air were to pass over the heat exchanger, but it does not affect the concentration of the trace gas and therefore may introduce a measurement error of the flow rate. Table 3-4 displays an example of the results from the measurements of the flow rates of the air handling unit.

Table 3-4 – Example of one measurement point on the real air handling unit

T ambient [°C]	**T extract [°C]**	**T exhaust [°C]**	**Heat recovery ratio [%]**	**Exhaust flow rate [m3/s]**	**Supply flow rate [m3/s]**
-0.1	21.4	9.2	56.7	6	5.7

In the simulation model created in TrnSys, the standard water-to-water heat pump component 663 is used. This component represents a single-stage heat pump that uses table data supplied by the user. The data supplied is heat capacity and compressor power at different load and source temperatures. The component takes incoming source and load temperatures and returns the corresponding heating and compressor power.

Type 668 calculates the outgoing load and source temperatures from the returned heating and compressor power data. The user-supplied data used for the heat pump comes from manufacturer-specified data for a smaller heat pump; this data is then adapted with a factor so that the cooling capacity roughly matches the possible extra heat recovery. Table 3-5 shows the user-supplied performance table of the heat pump.

Table 3-5 – Heat pump performance table used in the simulation

Condenser power [kW]	Compressor power [kW]	Tw load [°C]	Tw source [°C]	COP [-]
50.4	16.2	32.2	-13.9	3.1
69.2	18.1	32.2	4.4	3.8
87.7	20.1	32.2	12.8	4.4
106.2	22.2	32.2	21.1	4.8
124.8	24.1	32.2	29.4	5.2
49.3	17.6	37.8	-13.9	2.8
67.5	19.7	37.8	4.4	3.4
85.8	21.8	37.8	12.8	3.9
104.0	23.9	37.8	21.1	4.4
122.3	26.1	37.8	29.4	4.7
48.4	18.8	43.3	-13.9	2.6
66.5	21.0	43.3	4.4	3.2
84.2	23.3	43.3	12.8	3.6
102.1	25.6	43.3	21.1	4.0
108.2	28.1	43.3	29.4	3.9
47.4	20.1	58.9	-13.9	2.4
64.8	22.6	58.9	4.4	2.9
82.3	25.0	58.9	12.8	3.3
94.1	27.5	58.9	21.1	3.4
94.1	29.9	58.9	29.4	3.1

4 Investigating Vertical Inline Drain Water Heat Recovery in Buildings

Many different types of drain water heat recovery heat exchangers are available on the market and manufacturers present performance data for the heat exchanger. However, the heat exchanger performance is not equal to the performance of the heat recovery system, which is the important parameter to consider. Few studies investigating the efficiency of an installed heat recovery system are available in the literature.

4.1 Introduction

The heat recovery ratio in a vertical inline drain water heat recovery system depends not only on the design of the system and its components but also on the flow profile and characteristics of the drain water (Bartkowiak et al., 2010). It is, therefore, of interest to investigate how the flow profile of the drain water influences the heat recovery ratio of a drain water heat recovery system. But in order to establish under which conditions the results from the investigations are valid, it is important to map the characteristics of the heat exchanger.

Hence, the first step in the investigations is to evaluate the heat exchanger under steady state conditions to establish the heat exchanger characteristics. When the heat exchanger parameters are determined, the next step is to evaluate how the system performs under dynamic conditions. In this step, the choice of system design is made. In the present thesis, the system design chosen is one with a heat pump; see Figure 20 for reference.

Several studies investigate the performance of drain water heat exchangers and systems. Collins et al., (2013) looked into the effectiveness of falling film drain water heat recovery systems at equal flow conditions. The systems investigated were of the traditional type where incoming mains water is heat-exchanged directly with the drain water. Equal flow rate conditions will most likely not occur in a real installation due to the time lag between the flow of the incoming water mains and the drain water. Eslami-Nejad and Bernier, (2009) tested how different draw water flow profiles influenced the electrical demand of the hot water heater with drain water heat recovery installed. The different draw profiles were investigated using a simulation model. The heat recovery system simulated in the Eslami-Nejad and Bernier paper used direct heat exchange between the incoming mains and the drain water. In the literature, most of the systems that have been investigated are of this type. As mentioned earlier, when it comes to investigations about vertical inline heat recovery systems, Zaloum et al. (2007) analyzed five different heat exchangers in a system where the incoming mains were directly preheated by the drain water heat recovery system. The testing was performed using four different daily draw profiles and two different system configurations. Thirty-three days were devoted to benchmarking the heat recovery system. The main reason for the extended testing was the fluctuation of the cold water temperature. The temperature of the cold incoming water was between 19.4 °C and 9.5 °C. The draw profiles were related to the number of occupants in an apartment. In the Zaloum et al. (2006) investigation, a 2, 3, 4 and 4+ occupants schedule was used. The conclusion from that study was that the gas to the water heater can be reduced by 9 to 27 % with such a system and that the amount is dependent on the draw profile of the building. The number of showers has a large influence on the performance; decrease in gas demand for the water heater without showers was on average only 4.1 %.

Domestic shower heat recovery systems have been studied by several authors, for example, McNabola et al. (2013) and L.T. Wong et al. (2010). A shower heat recovery system is a form of drain water heat recovery systems that heat-exchanges outgoing drain water with incoming water to the shower. The system type is not similar to the characteristics of the proposed system in the present investigation.

Few papers present findings from drain water heat recovery system using a heat pump. One study (Baek et al., 2005) investigated a drain water heat recovery system with a heat pump installed in a hotel with a sauna. The analysis was made using simulations in TrnSys and the conclusion was that the heat recovery system could cover 90 % of the instant hot water load. In the Baek et al. investigation, the drain water was accumulated in a storage tank and not directly heat-exchanged. Ni et al. (2012) investigated a system to recover heat from greywater in a single family residential building. The investigation was made using a theoretical model describing a system with heat pump and a collection tank for the greywater. Conclusions drawn from the investigation were that the energy demand for space heating, cooling and hot water can be reduced by 33.9 % for a building in New York. The authors also investigated the outcome in 14 other cities in different climate zones, the results of the savings varying between 17 and 57.9 %.

With respect to studies of inline drain heat recovery systems using heat pump in multi-family houses, the literature survey returned few relevant investigations. One investigation (Nykvist, 2012) presents a survey on different drain water heat recovery systems suitable for installation to recover heat from drain water in existing buildings. The investigation presents measurements conducted on installed passive systems indicating that 10-15 % of the DHW heat demand can be recovered. In this investigation, simulations of the performance of such a system are also conducted and they indicate that a passive heat recovery system should be able to recover 20-25 % of the DHW demand. Nykvist (2012) also performed simulations for active drain water heat recovery systems with a heat pump installed. The conclusion from simulations on active heat recovery systems is that they can recover much more energy than passive systems: between 50-70 %.

The motivation for including a heat pump in the recovery system is that it might increase the heat recovery ratio since the recovered heat can be used more freely. Therefore, it will be possible to obtain a better match between demand and availability of the heat in the drain.

4.2 Aim of the Study

The aim of the study of the drain water heat recovery system is to investigate experimentally how a vertical inline heat exchanger with a heat pump performs under steady state and transient scenarios. The results from the experimental analysis are then used to theoretically describe the heat exchanger in order to investigate how different sizes of heat pump and storage tank influence the system performance.

For the analysis under transient conditions, two 24-hour flow scenarios have been created. The two different scenarios are created from measured incoming water mains to a 110-apartment building. The measured incoming flow profile is scaled down to represent two- and five-apartments. A third scenario was also used, emulating the drain water flow rate of a three-minute shower.

The main aim of this study is to analyze how the drain water flow in a multi-family house scenario influences the energy heat recovery ratio of an inline heat recovery system with a heat pump.

4.3 Analysis

The analysis is mainly conducted through experimental work, but a simplified simulation model is constructed from the experimental work to analyze how the size of the heat pump and storage tank influence the system performance.

4.3.1 Experimental Analysis

The system performance and the heat exchanger characteristics are evaluated to attain knowledge about the system and heat exchanger behavior. This information will be valuable for subsequently determining how the system should be set up to achieve high heat recovery ratios for the system operating in a real life setting.

4.3.1.1 Evaluating the Individual Heat Transfer Resistances

To establish the base for the experimental analysis, the characteristics of the heat exchanger are investigated. Because of the chosen heat exchanger, the heat transfer resistance between the outer coil and the inner pipe is significant. For coiled tube heat exchangers, the Dean number is used to characterize the flow regime (Srinivasan et al., 1968). Since the geometry of the coil affects the flow profile, the standard straight tube procedure of using the Reynolds number is not sufficient. In the present investigation the Dean number of the heat exchanger is >3 000 for the cases when the Reynolds number of the coil is higher than the critical Reynolds number. The critical Reynolds number defines when the flow regime changes from laminar flow to the transitional flow region (Srinivasan et al., 1968). The Dean number and the critical Reynolds number are defined, according to the work of Srinivasan et. al (1968), by Equation 4.1 and Equation 4.2 respectively.

$$De = Re \cdot \left(\frac{D_i}{D_c}\right)^{0.5} \qquad \text{Equation 4.1}$$

$$Re_{cri} = 2100 \cdot \left[1 + 12 \cdot \left(\frac{D_i}{D_c}\right)^{0.5}\right] \qquad \text{Equation 4.2}$$

Few correlations have been found in the literature that are valid for Dean Numbers higher than 2 000 and no correlations have been found that have been specifically developed for the heat exchanger design used in the present study. Therefore, a modified Wilson Plot method is used to investigate the characteristics of the heat exchanger and to develop a correlation that is valid for the heat exchanger used in this thesis.

The analysis method used in the present thesis to investigate the heat transfer characteristics of the heat exchanger is a modified Wilson Plot method, as proposed by Khartabil et al. (1998), and recommended by Shah (1990). The proposed method is suggested for use in cases where all three thermal resistances are unknown. The overall thermal resistance of the heat exchanger is calculated according to Equation 4.3.

$$\frac{1}{U \cdot A} = \frac{1}{h_{hot} \cdot A_{hot}} + R_{tot} + \frac{1}{h_{cold} \cdot A_{cold}} \quad \text{Equation 4.3}$$

Khartabil et al. (1998) combined the method proposed by Young and Wall (1957) and the method proposed by Briggs and Young (1969) to construct a methodology to iteratively obtain the five unknowns needed to determine the three thermal resistances. The equation Khartabil et al. presented in their modified version of the Wilson Plot method is described by Equation 4.4.

$$\left(\frac{1}{U \cdot A} - R_{Tot}\right) \cdot \left[Re^{d} \cdot Pr^{0.4} \cdot \frac{A}{D}\right]_s = \frac{1}{C_t} \cdot \frac{\left[Re^{d} \cdot Pr^{0.4} \cdot \frac{A}{D}\right]_s}{\left[Re^{a} \cdot Pr^{0.4} \cdot \frac{A}{D}\right]_t} + \frac{1}{C_s} \quad \text{Equation 4.4}$$

Equation 4.4 describes the correlations for both sides of the heat exchanger. It is assumed when using this equation that the flow rate is in single phase and the Nusselt number is a function of both the Reynolds number and the Prandtl number. Equation 4.4 has the form of a straight line, y=mx+b. Briggs and Young (1969) showed that successive linear regressions can be used to determine the unknowns. The Briggs and Young method requires that the shell side Reynolds number exponent is known, which is not the case in the present study. Khartabil et al. (1998) modified this method and introduced a third linear regression to acquire all the parameters, thus eliminating the need to know the behavior of one of the sides of the heat exchanger.

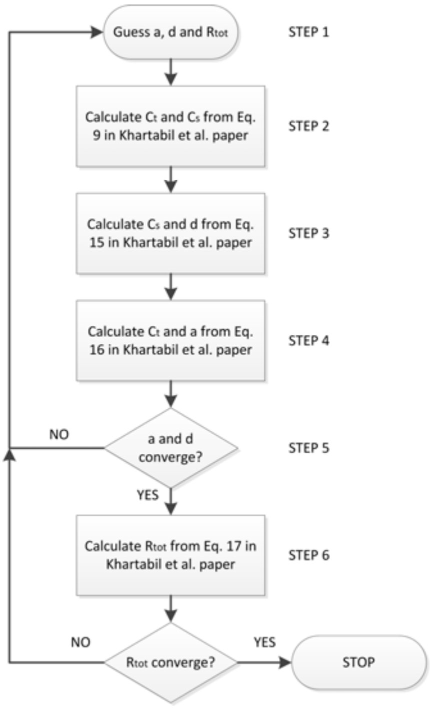

Figure 26 - Flowchart for the Khartabil et al. scheme

Khartabil et al. (1998) developed a scheme for the modified Wilson Plot method that describes the steps taken to obtain the unknown parameters a, d, R_{tot}, C_s, C_t in Equation 4.4 and to develop correlations for the three heat transfer resistances. Figure 26 shows the scheme. Three sets of measurement data are used in the method. By following the steps presented in the flow chart (Figure 26), the characteristics of the heat exchanger are determined.

With all the five unknowns determined, correlations for the heat transfer resistances on both sides can be developed for the heat exchanger used in the present study.

The correlation will be in the form described by Equation 4.5.

$$\frac{1}{h_{s,t} \cdot A_{s,t}} = \frac{1}{C_{s,t} \cdot \left[Re^{a,d} \cdot Pr^{0.4} \cdot \frac{A_{s,t}}{D_{s,t}}\right]} \qquad \text{Equation 4.5}$$

4.3.1.2 Evaluating the Steady State Heat Recovery Ratio

The steady state heat recovery ratio for the drain water heat recovery system is evaluated by measuring the recovered heat and comparing it with the available heat in the drain pipe. In this case, the available heat is defined as the amount of heat that would be recovered if the temperature of the drain water is lowered to the temperature of the incoming water mains, since legislation in Sweden stipulates that the temperature of the drain water leaving the building cannot be lower than this supplied temperature. Since the temperature of the water mains changes depending on the ambient temperature, a fixed temperature of 8 °C has been chosen. Equation 4.11 shows how the heat recovery ratio is calculated.

$$\eta_{8°C} = \frac{\dot{q}_{measured}}{\dot{m}_{drain} \cdot c_p \cdot (t_{drain\ in} - 8)} \qquad \text{Equation 4.6}$$

Since the flow rate in the drain for most cases is higher than the flow rate of the coil and the temperature difference used for calculating the maximum available heat in the drain is not the difference between the incoming temperatures to the heat exchanger, the heat recovery ratio will not be the same as the heat exchanger effectiveness. The heat recovery ratio is, therefore, a measure of the system efficiency and the heat exchanger effectiveness is a measure of the heat exchanger efficiency. The heat exchanger effectiveness is calculated according to Equation 4.7.

$$\varepsilon_{measured} = \frac{\dot{q}_{measured}}{C_{min} \cdot (t_{drain\ in} - t_{coil\ in})} \qquad \text{Equation 4.7}$$

For reference cases of the heat exchanger effectiveness, theoretical calculations are made using Equation 4.8 through Equation 4.10, using the methodology described by Incropera and DeWitt (1996).

$$\varepsilon_{counter\ flow} = \frac{1 - exp(-NTU \cdot (1 - C_r)}{1 - C_r \cdot exp(-NTU \cdot (1 - C_r)} \qquad \text{Equation 4.8}$$

$$\varepsilon_{cross\ flow} = 1 - exp\ [(\frac{1}{C_r}) \cdot (NTU)^{0.22} \cdot \{exp[-C_r \cdot (NTU)^{0.78}] - 1\}] \qquad \text{Equation 4.9}$$

$$\varepsilon_{all\ hex\ Cr=0} = 1 - exp(-NTU) \qquad \text{Equation 4.10}$$

The method used involves keeping the flow rate of the cooling water in the coil constant and varying the flow rate of the drain water. This is done for several different fixed flow rates in the cooling coil. The heat recovery ratio is calculated as the heat recovered divided by the maximum heat recovery. The maximum heat recovered is calculated from the flow rate of the drain water and the temperature difference between the two inlets to the heat exchanger.

4.3.1.3 *Evaluating the Performance of the Heat Recovery System under Transient Conditions*

To create transient conditions for the system, the flow rate of the drain water changes according to developed schedules. The literature survey was unsuccessful in identifying any relevant information on drain water flow rates in multi-family houses; therefore in this investigation, the schedules are created from a draw profile for incoming tap water mains to a multi-family house consisting of 110 apartments (Swedish Energy Agency, 2009a). The drain water flow rates created by the schedules for the two 24-hour scenarios are presented in Figure 27 .

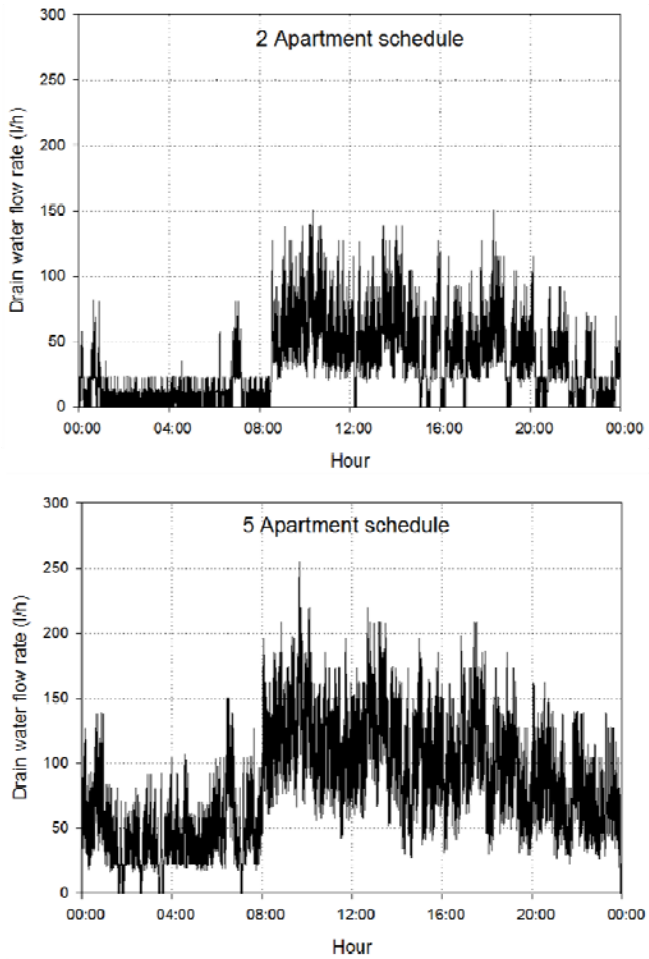

Figure 27 – Measured drain water flow rates for the two 24-hour schedules

The schedules were created from measured values presented in a report by the Swedish Energy Agency (2009a). Since the draw profile represented 110 apartments, this profile was adapted to the two desired scenarios (two- and five-apartments) by dividing the measured data by the number of apartments (110) and then multiplying the result by the desired number of apartments in the present investigation. This creates a schedule that gives a lower maximum flow rate due to the coincidence factor of the drain water if compared with measurements representing a single apartment. The aim of using this approach is to obtain an investigation result that is more related to larger multi-family buildings. The flow profiles are not entirely similar since the desired flow rate is transformed to an output voltage signal to the control valve. The transformation voltage is calculated using a polynomial equation developed from test measurements, but the flow rate at one specific voltage output may be different depending on the output signal on the previous time step (the schedule time step during experiments is 30 seconds).

Measurements of the drain water- and coil flow rate and the inlet and outlet temperatures are conducted during the 24-hour analysis period with a sample time of 30 seconds.

4.3.2 Theoretical Analysis

A simulation model was constructed using the characteristics of the heat exchanger evaluated according to Equation 4.5 and Table 4-1. The procedure to determine the coefficients in Table 4-1 is shown in Chapter 4.3.1.1.

Table 4-1 – Heat exchanger properties used for correlation in simulation model

Ct (coil side) [-]	Cs (drain side) [-]	a (coil side) [-]	d (drain side) [-]	Rw [K/W]
0.00000033	0.848	1.842	0.658	0.001

The simulation model uses data from the measurements and calculates a new tank temperature at every time step. Since the measured incoming temperature is used at each time step, both incoming temperatures are known for every time step, enabling the use of ε-NTU method. The ε-NTU method is used to calculate the recovered heat based on the incoming heat temperatures and the previously identified characteristics of the heat exchanger. The recovered heat is then used to calculate both the exit temperatures from the heat exchanger. These temperatures are then used to extract the heat pump cooling power from a heat pump performance table, again based on previous measurements. When the heat pump cooling power is determined, the added or extracted heat from the tank can be calculated in the next time step, giving the new tank temperature. Figure 28 presents a flow chart with the simulation steps.

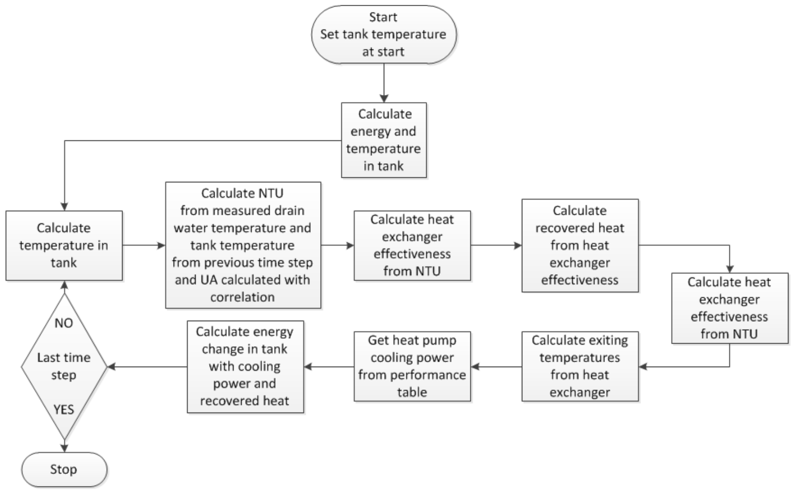

Figure 28 – Simulation process flowchart

The equations used in the simulation model are presented in Equation 4.11 through Equation 4.19.

Energy in tank:

$$Q_{tank\ time\ step\ 1} = m_{tank} \cdot c_p \cdot T_{tank} \qquad \text{Equation 4.11}$$

$$Q_{tank\ time\ step\ 2\ldots i} = Q_{tank\ previous\ time\ step} + Q_{energy\ change} \qquad \text{Equation 4.12}$$

Tank temperature:

$$T_{tank\ time\ step} = T_{tank\ previous\ time\ step} + \frac{\left(Q_{tank\ previous\ time\ step} - Q_{tank\ time\ step}\right)}{m_{tank} \cdot Cp} \qquad \text{Equation 4.13}$$

Heat transfer calculations:

$$NTU = \frac{UA}{C_{min}} \qquad \text{Equation 4.14}$$

$$C_{min} = \dot{m} \cdot c_p \qquad \text{Equation 4.15}$$

The effectiveness is approximated at $\frac{C_{min}}{C_{max}} = 0$ because of the nature of the drain water flow profile. In Figure 36, it can be seen that this is a good approximation for low Reynolds numbers on the drain water side. For high Reynolds numbers, correlations for counterflow in a concentric tube provide a better fit. In the present thesis, we operate with both high and low Reynolds numbers.

$$\varepsilon = 1 - \exp(-\text{NTU}) \qquad \text{Equation 4.16}$$

$$q_{recovered} = \varepsilon \cdot C_{min \cdot (T_{drain\ in} - T_{cooling\ in})} \qquad \text{Equation 4.17}$$

The outlet temperatures from the heat exchanger are calculated as:

$$T_{drain\ out} = T_{drain\ in} - \frac{q_{recovered}}{\dot{m}_{drain} \cdot c_p} \qquad \text{Equation 4.18}$$

$$T_{cooling\ out} = T_{cooling\ in} - \frac{q_{recovered}}{\dot{m}_{cooling} \cdot c_p} \qquad \text{Equation 4.19}$$

The heat pump performance table is constructed from the measurements and the heat pump cooling capacity is shown in Figure 29.

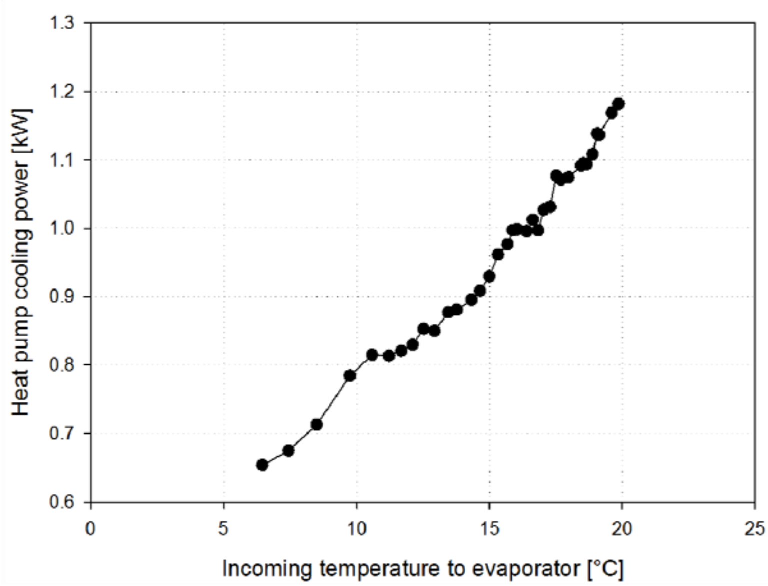

Figure 29 – Heat pump performance chart

To evaluate how different sizes of the heat pump and storage tank influence the performance of the system, simulations are performed with different sizes of heat pump and tank. The simulation model uses a scaling factor to increase or decrease the size of the tank or heat pump. Simulations are compared to the original measured tank temperature (temperature exiting the tank on coil side) to validate the model. In Figure 30, it can be seen that the output from the model is within ±10 % compared with the measured data.

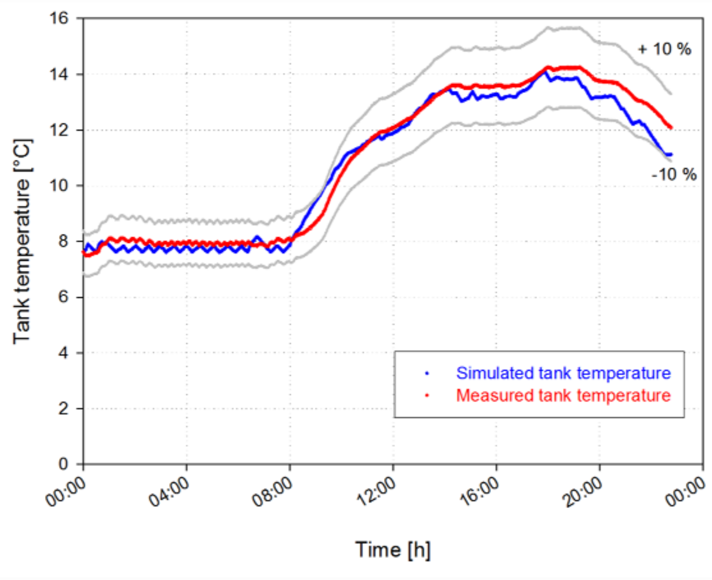

Figure 30 – Tank temperatures for validation of simulation model

4.4 Results from DWHR Investigations

Measurements have been undertaken to evaluate the performance and characteristics of the heat exchanger and the drain water heat recovery system. The characteristics of the heat exchanger are important to investigate and present so that the results can be compared with others' work and to obtain information for the development of the simulation model. Results from the three scenario analyses contribute to the information on how the drain water flow profile influences the performance of the heat recovery system as a whole. As mentioned before, the scenario analyses performed are two 24-hour scenarios with drain water flow profiles developed from measurements of incoming tap water to an apartment building with 110 apartments and a study of a drain water flow rate equivalent to what could be expected for one three-minute shower.

The measurements used to analyze the heat exchanger and heat recovery system are heat recovery ratio and the recovered heat, both for every 30 seconds and as an average over 24 hours depending on the analysis. Two different reference levels are used: a heat recovery ratio compared with what is theoretically possible and a heat recovery ratio compared with cooling the drain water to 8 °C. In theory, it would be possible to cool the drain water to the same temperature as the incoming cooling water using an infinitely large heat exchanger. Any real heat exchanger would perform less efficiently. Equation 4.20 through Equation 4.24 describe the evaluation procedure.

$$\dot{q}_{recovered} = \dot{m}_{drainwater} \cdot C_p \cdot \left(t_{drainwater\_in} - t_{drainwater\_out}\right) \qquad \text{Equation 4.20}$$

$$\dot{q}_{max} = \dot{m}_{drainwater} \cdot C_p \cdot \left(t_{drainwater\_in} - t_{coolingwater\_in}\right) \qquad \text{Equation 4.21}$$

$$\dot{q}_{\max\ \_8°C} = \dot{m}_{drainwater} \cdot C_p \cdot \left(t_{drainwater\_in} - 8\right) \qquad \text{Equation 4.22}$$

$$\eta_{max} = \frac{\dot{q}_{recovered}}{\dot{q}_{max}} \qquad \text{Equation 4.23}$$

$$\eta_{\max\ \_8°C} = \frac{\dot{q}_{recovered}}{\dot{q}_{\max\ \_8°C}} \qquad \text{Equation 4.24}$$

The motivation for the use of the heat recovery ratio at 8 °C is that many municipalities, which are the governing organizations of the sewage systems in Sweden, stipulate that the outgoing drain water temperature may not be lower than the incoming water mains. According to a report by the Swedish Energy Agency (Swedish Energy Agency, 2009b) the yearly average temperature of the incoming water mains is 8.4 °C.

4.4.1 *Results Quality of Measurements*

Before starting the measurements, their quality needs to be ensured. A great effort was made to be in control of the expected uncertainty in the system. The temperature sensors and the flow meters were all calibrated and calibration equations were developed to enhance the quality of the measurements.

4.4.1.1 *Uncertainty of Temperature Measurements*

The uncertainty of the temperature sensors at different temperatures and the overall uncertainty were analyzed using the methodology described in

Chapter 3.1.2 and more in depth in Appendix A. The result from the analysis is presented in Table 4-2.

Table 4-2 – Uncertainty of temperature sensors

Average Measured Temperature [°C]	Average Reference Temperature [°C]	Average Dev. Of Mean	Standard Dev.	No. Measurements	Standard Dev. Of Mean [°C]	Uncertainty 95 % Conf. Level [°C]
40.11	40.31	5.4E-15	0.0225	756	0.00082	0.0016
35.13	35.28	1.7E-14	0.0186	396	0.00093	0.0018
30.11	30.23	9.7E-15	0.0158	360	0.00083	0.0016
25.13	25.18	3.6E-14	0.0160	828	0.00056	0.0011
20.09	20.13	1.0E-16	0.0163	576	0.00068	0.0013
15.10	15.08	-8.3E-15	0.0170	558	0.00072	0.0014
10.14	10.03	3.7E-15	0.0191	432	0.00092	0.0018
5.15	4.98	-2.5E-15	0.0212	1008	0.00067	0.0013
			Overall Uncertainty 95 % Conf. Level [°C]			0.0015

The data from the analysis is used to develop the calibration correlation. The calibration equation is in linear form, to obtain sufficiently low uncertainty of the temperature differential measurements used in the experiments. Equation 4.25 shows the data for the temperature sensors.

Calibration equation for temperature sensors:

$$t_{calib} = t_{measured} \cdot 1.01018 - 0.19856 \qquad \text{Equation 4.25}$$

Using the temperature calibration equation, the uncertainty of the temperature differential measurements is reduced to ±0.00077 °C at 95 % confidence level. This uncertainty does not include the uncertainty of the calibration method, i.e. the uncertainty of the calibration temperature instrument.

4.4.1.2 Uncertainty of Flow Measurements

The flow meters were calibrated by the manufacturer on delivery, but to ensure reliable flow measurements, the flow meters were calibrated in situ using the gravimetric weighing method described in Chapter 3.1.2 and Appendix A.

Table 4-3 and Table 4-4 describe the outcome of the uncertainty analysis for the two flow meters.

Table 4-3 - Uncertainty of Flow meter 1 (Drain side flow meter)

Flow meter 1 - Drain side (warm)						
Weight [kg]	Logg time [s]	Ref. Ave. Flow rate [kg/s]	Ref. Ave. Flow rate [l/h]	Measured Flow rate [l/h]	Deviation [l/h]	Deviation [%]
4.32	40.00	0.108	388.70	389.06	-0.41	-0.11
3.76	40.00	0.094	338.81	338.73	-0.76	-0.22
2.93	40.00	0.073	263.34	263.82	-0.05	-0.02
3.14	60.00	0.052	188.28	188.44	-0.22	-0.11
1.83	60.00	0.031	109.92	109.92	-0.21	-0.19
2.34	120.00	0.020	70.24	69.90	-0.48	-0.69
1.84	120.00	0.015	55.13	55.96	0.72	1.30
1.04	180.00	0.006	20.73	21.04	0.27	1.30
0.66	180.00	0.004	13.17	13.53	0.34	2.55

The uncertainty at 95 % confidence level for Flow meter 1 is according to the measurements, ±0.40 l/h.

Implementing Equation 4.26 to correct the flow measurement according to the calibration, the uncertainty of the flow measurement is reduced to ±0.29 l/h.

Calibration equation for flow meter 1:

$$\dot{\mathrm{V}}_{\mathrm{calib}} = \dot{\mathrm{V}}_{\mathrm{measured}} \cdot 1.002 - 0.264 \qquad \text{Equation 4.26}$$

Table 4-4 - Uncertainty of Flow meter 2 (Coil side flow meter)

Flow meter 2 - Coil side (cool)						
Weight [kg]	Logg time [s]	Ref. Ave. Flow rate [kg/s]	Ref. Ave. Flow rate [l/h]	Measured Flow rate [l/h]	Deviation [l/h]	Deviation [%]
3.36	90.00	0.037	134.68	133.06	-1.62	-1.20
3.26	90.00	0.036	130.83	129.26	-1.57	-1.20
3.40	90.00	0.038	136.11	134.64	-1.46	-1.07
3.28	60.00	0.055	197.20	195.33	-1.87	-0.95
3.29	60.00	0.055	197.74	195.33	-2.41	-1.22
3.27	60.00	0.055	196.82	195.33	-1.49	-0.76
4.09	60.00	0.068	246.13	242.36	-3.77	-1.53
3.91	60.00	0.065	235.04	234.50	-0.54	-0.23
3.82	60.00	0.064	229.81	225.47	-4.34	-1.89

The uncertainty at 95 % confidence level for flow meter 2, according to the measurements, is ±2.15 l/h.

By implementing the calibration Equation 4.27 for flow meter 2, the uncertainty of the flow measurement is reduced to ±0.91 l/h.

Calibration equation for flow meter 2:

$$\dot{V}_{calib} = \dot{V}_{measured} \cdot 1.012 - 0.106 \qquad \text{Equation 4.27}$$

4.4.1.3 Propagation of Uncertainty in Calculations

To evaluate which of the parameters influences the uncertainty the most in the evaluation of the heat recovered during experiments, the propagation of uncertainty in recovered heat, Equation 4.28, is analyzed.

$$\dot{q} = \dot{m} \cdot C_p \cdot \Delta t \qquad \text{Equation 4.28}$$

Table 4-5 shows the propagation of uncertainty and how the different parameters influence the total uncertainty.

Table 4-5 - Uncertainty of calculated recovered heat

Recovered heat [kW]	Uncertainty 95 % Conf. Level [kW]	Uncertainty 95 % Conf. Level [%]	Flowmeter part of the total Uncertainty [%]	Density part of the total Uncertainty [%]	Specific Heat part of the total Uncertainty [%]	dT part of the total Uncertainty [%]
9.04	0.022	0.24	95.3	4.4	0.3	0.1
7.09	0.019	0.27	96.4	3.3	0.2	0.1
4.90	0.017	0.35	97.8	2.1	0.1	0.1
3.06	0.015	0.49	98.8	1.1	0.1	0.0
1.53	0.013	0.83	99.6	0.4	0.0	0.0
0.81	0.011	1.31	99.8	0.1	0.0	0.0
0.52	0.008	1.63	99.9	0.1	0.0	0.0
0.15	0.006	4.37	100.0	0.0	0.0	0.0
0.06	0.004	6.85	100.0	0.0	0.0	0.0

It is evident that the uncertainty of the flow meters is the critical component in the analysis. As mentioned before, the flow meters are calibrated in situ so that the installation effect is included in the uncertainty.

4.4.2 Results from the Heat Exchanger Characteristics Investigation

The analysis of the three heat transfer resistances revealed that the resistance between the outer coil and the inner pipe is significant and of the same magnitude as the resistance between the water and the wall on the inside of the inner pipe. The heat resistance between the water and the wall of the coil is small compared with the other two. The analysis resulted in the development of a correlation for the heat transfer resistance of the coil side, the drain side and the resistance between the inner and outer pipes (the form of the correlation is described by Equation 4.5). The constants and exponents used are presented in Table 4-1.

Using the correlation and the data from Table 4-1 to calculate the total heat transfer resistance and comparing it with the measured total heat transfer resistance is presented in Figure 31.

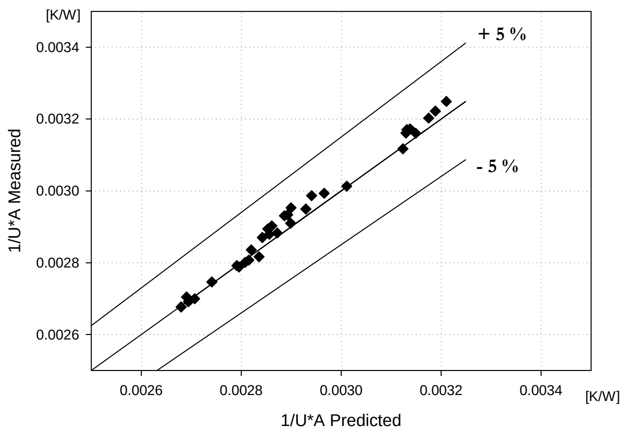

Figure 31 - Measured heat transfer resistance compared to resistance predicted by the correlation

How different flow rates influence the individual heat transfer resistances for the coil and drain side is presented in Figure 32 and Figure 33.

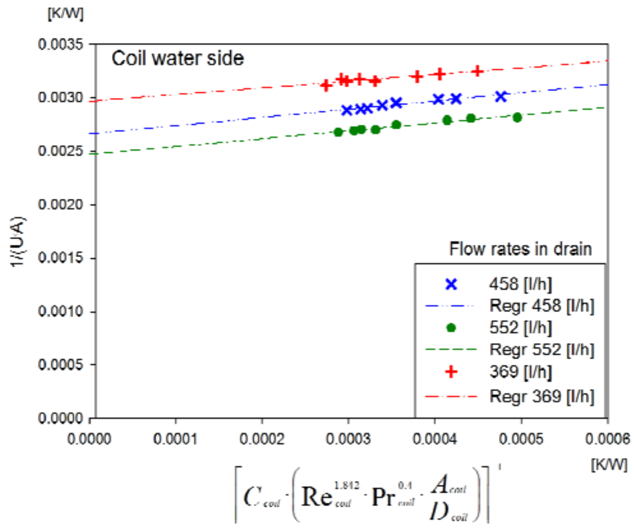

Figure 32 - Overall heat transfer resistance for three different drain water flow rates at different flow rates on the coil side

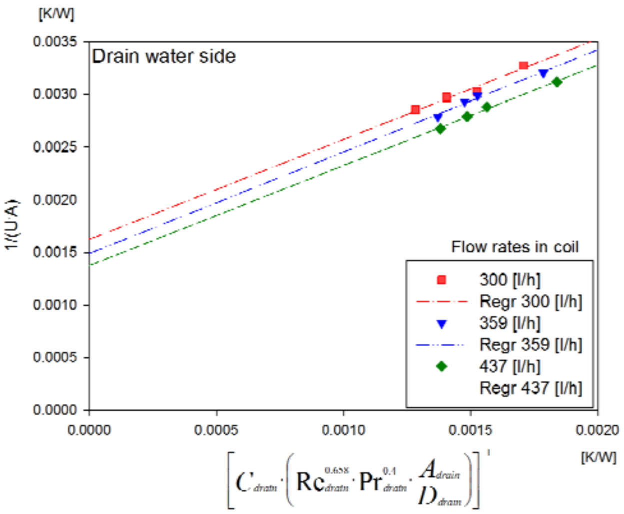

Figure 33 - Overall heat transfer resistance for three different coil flow rates at different flow rates on the drain side

By combining the information in Figure 32 and Figure 33, it is clear that the heat resistance on the drain side and the contact resistance are dominant over the coil heat resistance. From the figures, the contact resistances (R_w) will be approximately 0.001 K/W (0.0025-0.0015).

Due to the design of the heat exchanger, it has a high contact heat resistance between the inner and outer pipes. Since the outer pipe is wound around the outside of the inner pipe and only fixated at the top and bottom, the contact between the two pipes is not good. In the present study, the space between the outer and inner pipes is water-filled to reduce the contact resistance; which is still high compared with a traditional tube and shell heat exchanger. This means that there is significant room for improvement in heat exchanger design and performance.

4.4.3 Results from the Steady State Heat Recovery Performance Investigation

The ability of heat recovery systems to recover heat at different flow rates is investigated. As expected, the measurements indicate that the heat transfer properties are severely reduced when the flow regime turns laminar, usually considered at a Reynolds number of approximately 1700 for water in falling film (Karapantsios et al., 1989). This seems consistent with the results from the measurements in the present study; see Figure 34.

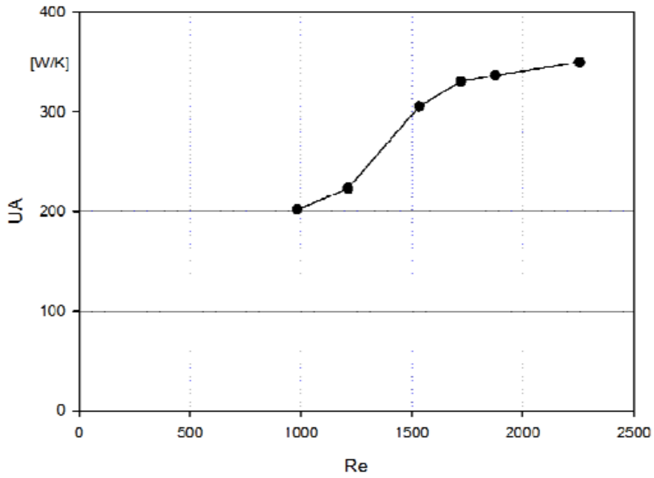

Figure 34 - Overall heat transfer coefficient at different flow rates of the drain water

How the heat recovery system performs at different flow rates is presented in Figure 35.

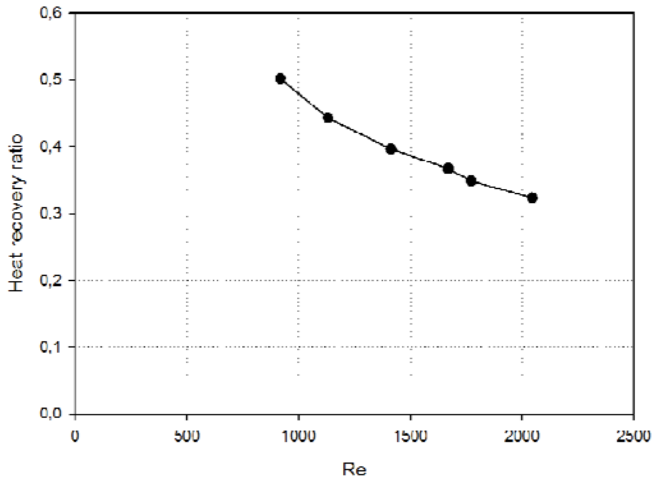

Figure 35 - Heat recovery ratio at different flow rates of the drain water

Under the conditions that the heat exchanger was tested, the heat recovery ratio ranges between 32 and 50 %, the ratio being dependent on the flow rate of the drain water. The heat recovery ratio decreases when the flow rate increases, even though the heat flux increases. Higher flow rates lead to higher exiting temperature of the drain water, which in turn leads to a larger temperature difference between the wall and the average drain water temperature. This larger temperature difference introduces a larger driving force for the heat transfer, thereby increasing the heat flux. The decrease in heat recovery ratio depends on how it is defined. As seen in Equation 4.24, the only parameter in the equation that affects the heat recovery ratio due to the increase in flow rate is the exiting temperature of the drain water. Incoming temperature to the drain water varies somewhat but the temperature to the coil side is close to 8 °C. Table 4-6 presents temperatures and flow rates for the different runs.

Table 4-6 - Temperatures and flow rates for the different runs

Run	T drain in [°C]	T drain out [°C]	T coil in [°C]	T coil out [°C]	q flow drain [l/h]	q flow coil [l/h]	Re_drain [-]
1.	24.6	19.3	8.2	16.8	544	336	2045
2	22.4	17.4	8.2	15.2	483	349	1772
3	23.3	17.7	7.9	15.2	457	351	1669
4	27.6	19.9	8.9	17.1	370	348	1413
5	22.0	15.9	8.2	13.9	316	347	1131
6	22.7	15.4	8.1	13.6	257	349	918

To evaluate how the experimental heat exchanger effectiveness compares with the theoretical effectiveness, several different heat exchanger configurations are used as comparison. In Figure 36, the outcome of the comparison is presented.

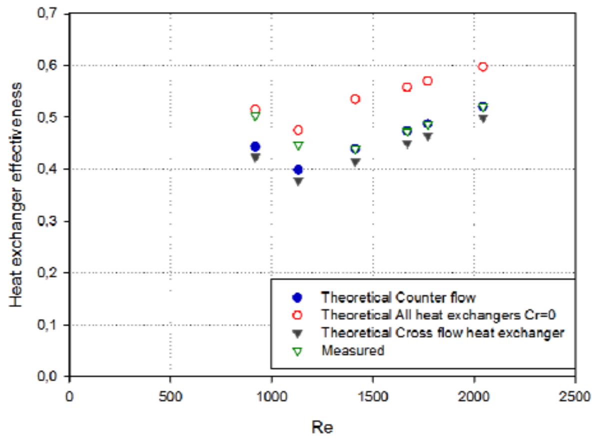

Figure 36 - Comparison between experimental and theoretical heat exchanger effectiveness for several heat exchanger configurations

The heat exchanger behaves similarly to a counter-flow heat exchanger when the flow rate on the drain side is higher than the flow rate on the coil side. When the flow rate on the coil side is higher than on the drain side (Re<~1200), the heat exchanger tends to behave more like a heat exchanger with a heat capacity ratio C_r=0. The heat capacity ratio is defined as $C_r=C_{min}/C_{max}$.

As concluded earlier, the heat exchanger effectiveness describes the performance of the heat exchanger. If it is assumed that there are only very small variations in liquid properties, and that the UA value of the heat exchanger due to changes in temperature is constant throughout the whole temperature range of the experiment, this would mean that the heat exchanger effectiveness should be constant if the flow rates are constant. This would hold even if the incoming temperatures to the heat exchanger change, if the changes in the fluid properties are neglected.

For the heat recovery ratio, the situation is different. If the temperature of the incoming water to the coil deviates from the lowest allowed temperature leaving the building (set to 8 °C in this thesis), the heat recovery ratio will change depending on the temperature difference between the incoming drain water and the incoming temperature to the coil. Figure 37 shows how the heat recovery ratio will change depending on the incoming temperatures to the heat exchanger.

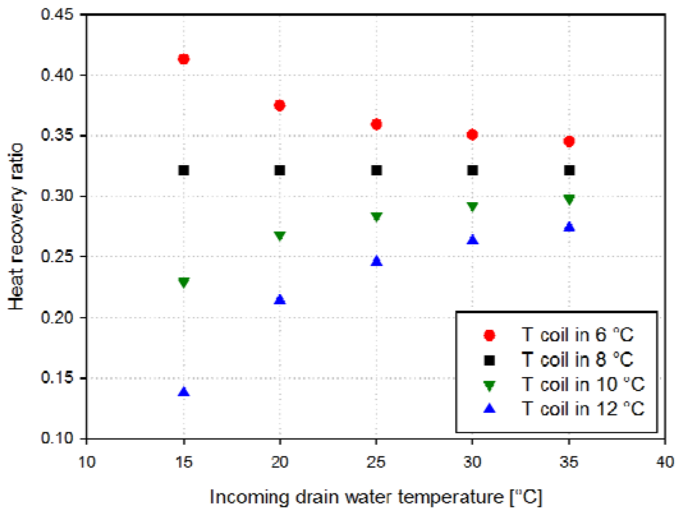

Figure 37 - Heat recovery ratio at different temperatures of the incoming fluids to the heat exchanger

It should be noted that in this particular case, presented in Figure 37, the drain water flow rate is higher than the coil flow rate. If the case had instead been the opposite, the heat recovery ratio would have been the same as the heat exchanger effectiveness if the incoming temperature to the coil had been 8 °C. This is because C_{min} will depend on the drain water flow rate.

The analysis indicates that it would indeed be interesting to investigate the amount of energy that is possible to recover when the flow rate is intermittent, similar to a real life situation

4.4.4 Results from the Investigation of the Performance of the Heat Recovery System under Transient Conditions

Investigation has been carried out with three different scenarios; as described in Chapter 4.3.1.3. Since the investigations of the steady state heat recovery performance indicated that the drain flow rate is an important factor for system efficiency, two different 24-hour drain flow rate scenarios are chosen for the investigation. To complement these investigations, a scenario similar to a three-minute shower is also investigated.

4.4.4.1 Results from the Two-Apartment Scenario

Figure 27 presents the flow rate of the drain water in the study of the heat recovery ratio in a 24-hour scenario for two apartments. The result of the average heat recovery ratio for the two-apartment scenario is presented in Table 4-7.

Table 4-7 - Recovered energy and available energy from the drain water for the two-apartment scenario

Scenario - two-apartment		
Recovered Energy [kWh]	Energy in drain 8 dgC [kWh]	Energy in drain max [kWh]
14.8	21.5	18.2
Heat recovery ratio	69%	81%

The difference between the two heat recovery ratios is due to the fact that the designed heat recovery system cannot keep the incoming cooling temperature to the heat exchanger constant at 8 °C. However, the system recovers 81 % of what is theoretically possible for this system design.

Figure 38 displays the variation in the heat recovery ratio over 24 hours. It is clear that the system can keep the cooling temperature constant at 8 °C until around 8:00 in the morning, after which the temperature increases and the two heat recovery ratios start to differ. The reason for this is that the system is undersized compared with the available energy and the heat exchanger capacity, which means that the system will recover less energy than the heat exchanger is capable of.

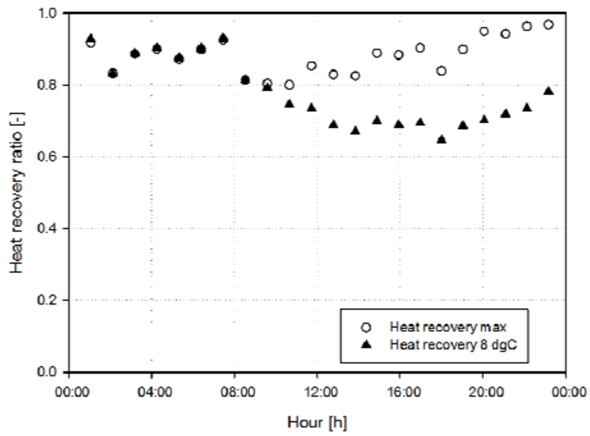

Figure 38 - Average Heat recovery ratios for the two-apartment scenario

In Figure 39, the theoretically available energy is presented together with the recovered energy.

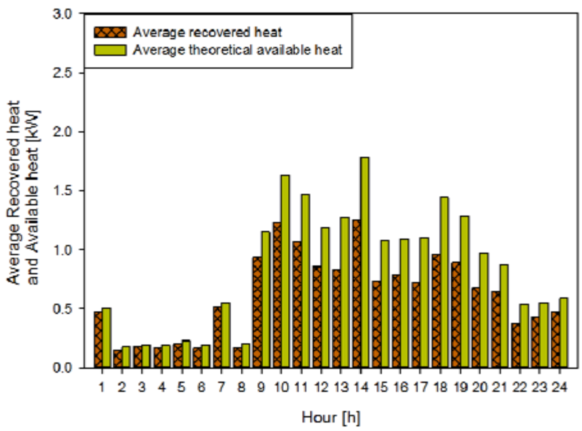

Figure 39 - Average recovered heat and theoretically available heat for the two-apartment scenario

It can be seen that the system can on average recover up to just over 1.2 kW even though the capacity of the heat pump is only about 800 W. This is, of course, because the system has a storage tank and for many hours of the day the average demand is lower than 800 W, so storage is possible.

4.4.4.2 Results from the Five Apartment Scenario

The five-apartment scenario analysis is conducted in the same way as the two-apartment analysis with the one difference of the drain water flow profile. The difference in the two drain water profiles can be seen in Figure 27. The result of the average heat recovery ratio for the five-apartment scenario is presented in Table 4-8.

Table 4-8 - Recovered energy and available energy in the drain water for the five-apartment scenario

	Scenario - five-apartment	
Recovered Energy [kWh]	Energy in drain 8 dgC [kWh]	Energy in drain max [kWh]
23.3	51.8	35.4
Heat recovery ratio	45%	66%

In this case, the difference between the two heat recovery ratios is larger than for the two-apartment case. This is because now there is even more available energy in the drain water; which enables energy to be stored in the storage tank smaller and therefore pushes up the average temperature of the incoming cooling water. In a perfectly designed system, the incoming temperature to the cooling coil should be kept constant in all operating points. This means that the system is undersized compared with the available heat in the drain water.

The heat recovery ratio over the 24 hours is presented in Figure 40. It is apparent that the system can keep the temperature of the incoming temperature to the cooling coil at 8 °C in the early hours of the morning. After about 3 hours, the system cannot keep the temperature constant because the heat pump is too small.

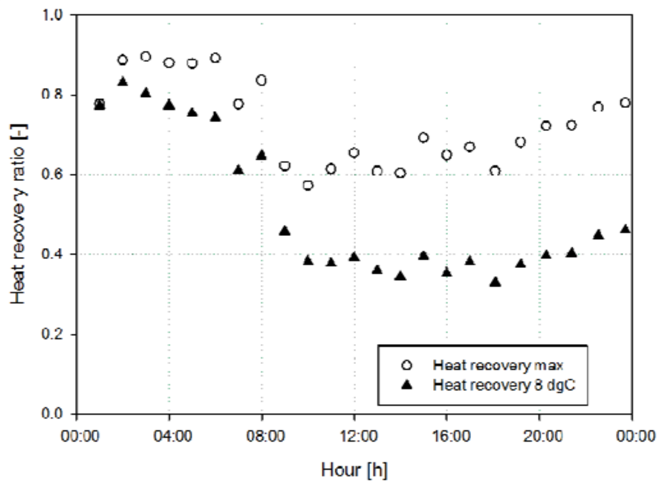

Figure 40 - Average Heat recovery ratios for the five-apartment scenario

Since the system cannot deliver the amount of cooling power that is needed in order to keep the temperature of the cooling water sufficiently low the heat recovery system has a reduced capacity. This can be seen in Figure 41.

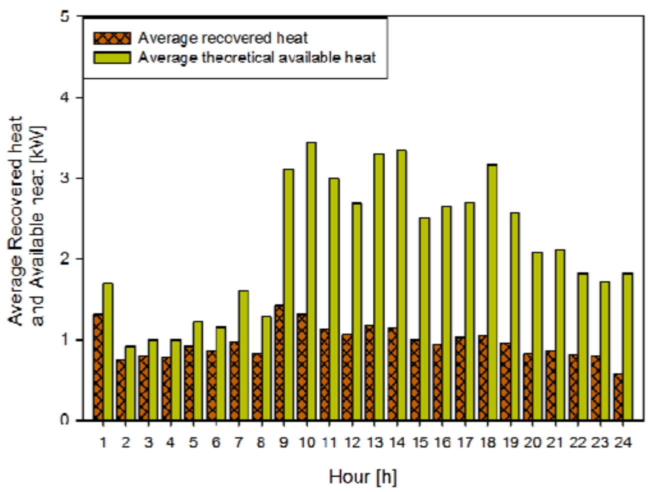

Figure 41 - Average recovered heat and theoretically available heat for the five-apartment scenario

In Figure 39, which represents the average recovered heating power for the two-apartment scenario, the maximum recovered heating power was just over 1.2 kW. In the five-apartment scenario, the average maximum recovered heating power is about 1.2 kW, even though the available heat is much higher in this scenario. This can be attributed to the fact that the incoming temperature to the cooling coil is higher. Figure 42 shows the behavior of the incoming temperature to the cooling coil.

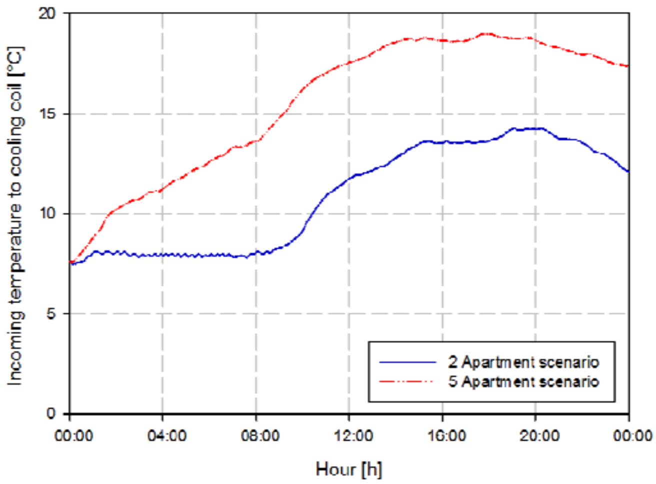

Figure 42 - Incoming temperature to the cooling coil of the drain water heat exchanger

For the five-apartment scenario, the temperature starts to deviate from the desired set point (8 °C) immediately, whilst for the two-apartment scenario the temperature is constant throughout the night and all the way until the morning peak sets in. The reason for the difference in temperature of the water to the coil is that the heat pump and storage tank combination are even more undersized in the five-apartment case than the two-apartment case. The effect of undersizing of the heat pump and storage tank is that the system is not able to cool the water in the same rate as it is heated by the drain water heat exchanger. It may also be seen that neither system is able to reduce incoming cooling coil temperature before the cycle starts over. This, again, shows the capacity mismatch between load and recovery system capacity.

4.4.4.3 Results from the Three Minute Shower Scenario

To evaluate the behavior of the system during a large peak for a short period of time, a scenario where the flow rate of the drain water is constant for three minutes is investigated. The flow rate is chosen to represent one person taking a three-minute shower. The flow rate is set to be approximately 7 l/min. Table 4-9 present the result from the analysis of the heat recovery ratio.

Table 4-9 - Recovered energy and available energy in the drain water for the Shower scenario

Scenario - Shower		
Recovered Energy [kWh]	Energy in drain 8 dgC [kWh]	Energy in drain max [kWh]
0.2	0.6	0.7
Heat recovery ratio	34%	33%

As expected, the heat recovery ratio is lower when the flow rate of the drain water is high but the recovery rate is still respectable. Figure 43 shows how the recovered energy compares with the available energy in the drain.

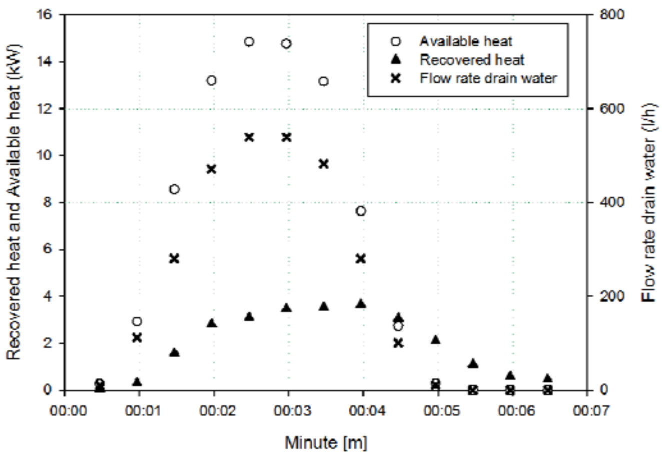

Figure 43 - Recovered heat and theoretically available energy for the Shower scenario

During the experiments, it was observed that it takes some time for the valve controlling the drain water flow rate to open and close, which can be seen in Figure 43, and there is also some latency between the two sides of the heat exchanger. The maximum heat recovered is 3.65 kW which is close to the maximum number that was achieved during the analysis for the two-apartment scenario.

4.4.4.4 Results from Simulation of Different Heat Pump and Tank Sizes

Simulations of heat recovery system performance with different sizes of heat pump and the tank were performed for the two-apartment drain water flow profile. Table 4-10 illustrates how the performance changes for different sizes of heat pump and Table 4-11 shows how the performance changes for different sizes of storage tank.

Table 4-10 - Simulated heat recovery at different heat pump sizes

Recovered heat at different heat pump size - two-apartment			
Fraction of original Heat pump size	HP Average Cooling [kW]	Recovered heat [kWh]	Heat recovery ratio [%]
0.4	0.37	11.3	52.8
0.6	0.49	12.7	59.3
0.8	0.58	13.8	64.3
1	0.66	14.7	68.4
1.2	0.73	15.6	72.7
1.4	0.78	16.4	76.4
1.6	0.80	17.0	79.0
1.8	0.82	17.3	80.7
2	0.82	17.4	81.1

The table shows that if the heat pump is twice as large as the one used during the measurements (fraction to original heat pump size is 2), the heat recovery ratio increases from 68 % to 81 %. In this case, the heat recovery ratio is defined by Equation 4.24. The increase of heat recovery ratio from heat pump size 1.8 to 2 times the size used in the investigation is very small compared to the situation when the heat pump size is increased between 1.4 and 1.6. This indicates that there is a maximum heat pump size that delivers the highest heat recovery ratio, and a larger heat pump will not increase the heat recovery ratio. In a well designed system, the heat pump needs to be able to deliver at least the same amount of cooling energy during one cycle as the amount of recovered heat from the drain water. This is the smallest capacity the heat pump should have.

Table 4-11 - Simulated heat recovery at different storage tank sizes

Recovered heat at different storage tank size - two-apartment			
Fraction of original Storage Tank size	HP Average Cooling [kW]	Recovered heat [kWh]	Heat recovery ratio [%]
0.4	0.68	14.0	65.2
0.6	0.67	14.2	66.1
0.8	0.66	14.5	67.4
1.0	0.66	14.7	68.4
1.2	0.66	15.0	70.1
1.4	0.65	15.2	71.0
1.6	0.64	15.3	71.1
1.8	0.64	15.6	72.7
2.0	0.63	15.6	72.9
2.2	0.62	15.7	73.2
2.4	0.62	15.8	73.8
2.6	0.61	15.8	73.6
2.8	0.60	15.8	73.6
3.0	0.60	15.9	74.2

Considering the size of the tank, there also seems to be a maximum size that yields the highest possible heat recovery ratio. However, the simulation model assumes that the tank is fully charged when the simulation starts. This means that a larger storage tank decreases the need of a larger heat pump.

How the temperature in the storage tank varies over the 24 hours for the different sizes of the heat pump and storage tank is also interesting to see. Figure 44 and Figure 45 present the storage tank temperatures for a few selected cases.

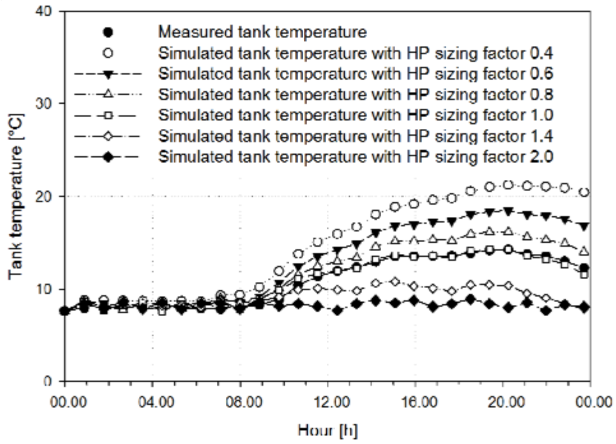

Figure 44 - Storage tank temperature for different heat pump sizes

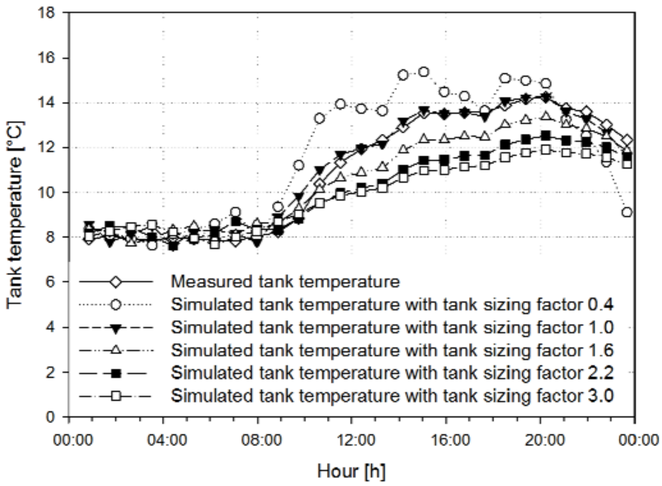

Figure 45 - Storage tank temperature for different tank sizes

For the heat pump sizes, it can be seen that the heat pump needs to be at least 1.4 times the size of the heat pump that is installed in the test rig to be correctly sized for the two-apartment drain water flow profile. This is because the temperature in the storage tank temperature needs to return to the starting level after hour 24. In addition, it may also be seen that the measured temperature matches the simulated temperature with a heat pump sizing factor of 1, which means that the model is able to capture the trends and heat recovery of the measurements.

For the storage tank sizes, it is evident that the temperature in the tank changes more rapidly when the tank is small, but there is not a very large influence on the end temperature if the tank increases. This indicates that the sizing of the heat pump is more important for the system performance and operation.

4.5 Discussion

The investigations of the heat exchanger performance indicate that the heat exchanger constructed and used in the present project is not very good. The reason for this might be related to the design of the heat exchanger and the way that it was produced. A system with a more efficient heat exchanger would be able to perform better. However, as shown, it is important to consider the sizing of the heat pump and storage tank as well as the system as a whole.

The schedules that have been used for the investigations on the transient behavior of the heat recovery system may not be completely accurate for multi-family houses with so few apartments. In a case with fewer apartments, the coincidence factor will not affect the drain water flow as much as for a building with many apartments. This leads to a relatively larger difference between average and maximum drain water flow rate. To obtain a good heat recovery ratio in a situation where there is a large difference between the average and maximum drain water flow rate, the system needs to be sized to meet the specific demand, which most likely means that the installation will be more costly. This is due to the fact that the cooling power demand compared with the available heat in the drain water will be larger than for a case with a more constant drain water flow rate. The fact that the difference between the present approach in this thesis with a scaled-down drain water flow rate and a flow rate from a building with few apartments will affect the heat recovery system performance was the reason for investigating how the present system performs under a shower scenario. The shower scenario gives an insight into how the system performs when the flow rate is high compared to what the heat exchanger is designed for.

The methodology chosen should be seen as a scaled-down investigation of a multi-family house with 110 apartments and therefore the results are more representative of buildings with many apartments where the coincidence factor has a clear effect. This becomes apparent when analyzing the heat recovery ratio of a three-minute shower where the maximum available heat in the drain is 14.8 kW compared with 3.6 kW for the two-apartment schedule.

The sizing of the heat pump was made for another, even less efficient, heat exchanger and therefore the test facility is slightly undersized for the two-apartment scenario. This is apparent in Figure 44, where it is obvious that the storage tank temperature is not returned to the starting point. Figure 42 shows that for the five-apartment case, the heat pump and storage tank combination is well undersized and the temperature entering the coil is high after 24 hours of operation. For a system to operate efficiently over time, the tank temperature needs to be returned to the original temperature after 24 hours. Preferably, the temperature should be kept constant over the entire 24 hours, but this is probably not economically feasible.

4.6 Conclusions

The investigation of the heat exchanger performance shows that the heat exchanger used in the analysis has large contact heat resistance between the inner and outer pipe, which is detrimental to its the ability to recover heat. Despite this fact, the analysis shows that with this heat exchanger design a fair amount of heat can be recovered in a steady state situation.

The correlations that have been developed for this heat exchanger can predict the heat resistances satisfactorily. However, it has not investigated whether these correlations can be used for other heat exchangers of the same type and this should not be assumed or be concluded.

An investigation has also been performed on how different drain water flow profiles affect the performance of an inline drain water heat recovery system that use heat pump to increase the temperature of the recovered heat. Two different 24-hour scenarios and one shower scenario have been investigated.

The investigation shows that a heat recovery system of this type is able to recover a large portion of the available heat in the drain water if the system has been sized to match the drain water profile. It can also be concluded that the drain water heat exchanger works well in combination with heat pump and storage tank.

It is also evident that sizing of the heat pump is important for the performance of the system; sizing of the storage tank is also important but not as critical.

Another conclusion from the study is that it is technically feasible to install drain water heat recovery systems of this type in multi-family houses.

This type of heat exchanger installation is relatively easy to implement and does not introduce an obstruction in the drain water pipe, which is an advantage over other type of systems where the drain water needs to be stored.

The investigation indicates that it is well worth expanding the knowledge concerning this type of heat recovery system. To do so, more studies are required to investigate the long term effects on the performance of the system and the nature of the drain water flow characteristics i.e. temperatures and flow rates over time.

5 Investigating Heat Recovery Ventilation in Buildings

One of the main sources of heat loss in a building in a cold climate is ventilation. Therefore, the energy performance of buildings in such climates is dependent on the amount of heat that is recovered from the exhaust air. In order to meet the new energy and climate goals, new ways are needed to increase the amount of heat that is being recovered from ventilation systems.

5.1 Introduction

The energy performance of any building located in a region where the heating demand is significant during the year depends on several factors. As mentioned in Chapter 1, the main factors are thermal losses through the building envelope, ventilation losses and losses due to air infiltration/exfiltration. In new buildings, the thermal losses through the building envelope and through air infiltration are greatly reduced compared with the buildings that were constructed more than 20 years ago, making ventilation the single biggest source of thermal heat loss (Liddament and Orme, 1998). For older buildings intended for commercial use, the thermal loss through the ventilation is normally the most dominant factor, the reason being that buildings intended for commercial use often have a high ventilation air change rate (ACH). The thermal loss from ventilation can account for more than 50 % of the total thermal loss from the building (Roulet et al., 2001). This often leads to poor energy performance in the building. To reduce the influence on the energy performance due to heating of ventilation air, a heat recovery system is often implemented.

With new demands on the efficiency of these heat recovery systems, new ways to improve the efficiency are needed. One possible way is to retrofit a system with a heat pump to support the existing heat recovery system. Two types of heat recovery systems have been identified as good candidates for such an installation: run-around coil- and thermal energy wheel heat recovery systems.

A run-around coil heat recovery system is often used in buildings where the extraction air contains volatile substances that can contaminate the supply air or in buildings where the supply and exhaust air fans are placed far apart. Hospitals are one example of a building type that matches this criterion; another one is buildings that contain a processing industry.

Several studies (Emerson, 1984; Forsyth and Besant, 1988a, 1988b; Wallin et al., 2012; Zeng et al., 1992) have investigated the important factors that influence the performance of ventilation heat recovery system with a run-around coil. Several factors have been identified in the previous studies; of special interest are the brine flow rate and the glycol concentration of the brine.

Energy wheel heat recovery systems are one of the most common types of heat recovery system for ventilation. Roulet et al. (2001) studied how the ventilation heat recovery systems performed through numerous field measurements. They showed that the global heat recovery ratio varies a great deal but the three best systems recover about 60-70 % of the energy in the exhaust air. This means that the losses from the ventilation system typically account for 30-40 % of the ventilation heating demand.

In recent years, the costs for energy have increased in Sweden and Europe. This introduces a new economic incentive to take energy conservation measures. The higher energy price provides new possibilities.

The cost of implementing measures to improve the energy performance of air handling units in existing buildings can be high relative to the savings; this means that the payback period is often longer than many organizations will allow.

However, there is little to be found in the literature on retrofitting heat pumps to existing heat recovery systems to increase the annual recovered heat. The present chapter presents the results from investigations of the potential of such an approach.

5.2 Aim

The aim of the study is to investigate the possibility of retrofitting a heat pump to an existing ventilation heat recovery system and to investigate how different parameters influence the energy performance of such an approach.

By theoretically investigating what can be achieved when a heat pump is retrofitted to an existing ventilation heat recovery system, conclusions can be reached that contribute to the knowledge about energy performance in buildings. Theoretical investigations allow several different system configurations to be analyzed on a limited budget.

The main aim is to develop new ways to increase the efficiency in existing ventilation heat recovery systems.

5.3 Analysis

The investigation was mainly conducted through simulations in TrnSys; however, some experimental work was also undertaken when analyzing the run-around coil heat recovery system.

5.3.1 Run-around Coil System Analysis

To investigate the performance of run-around coil systems and the possibility to increase the efficiency by retrofitting a heat pump to this type of system, a test facility and a simulation model of this facility was built. The main reason for this was to create a baseline for a run-around coil system and to identify the parameters that influence the performance, but also to study the impact of temperature on the overall heat transfer properties of the heat exchangers.

A run-around coil heat recovery system consists of at least two coiled heat exchangers. Figure 46 presents a schematic of such a system. The coils are connected via pipes to a loop in which a fluid flows. The fluid is usually a mixture of water and an anti-freeze fluid; in the present investigation, a mixture of water and ethylene glycol is used. Heat is "moved" from the hot side (extraction/exhaust air) to the cold side (supply air) via the secondary fluid. It is also possible to recover cooling energy during warm days.

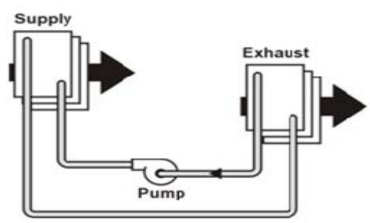

Figure 46 - Scheme of a run around coil heat recovery

The performance of a run-around coil system will be influenced by the brine flow rate, since higher flow rates, at the same air flow rate, introduce a smaller temperature difference between the warm and the cold side. Larger temperature differences will yield a higher system performance, as long as the flow rate of the brine is high enough to transfer the heat between the warm and the cold side of the system and as long as the flow in the heat exchanger is turbulent. If the heat exchanger can be freely chosen, the highest system performance is achieved when the ratio between the brine- and airside heat capacity rates is 1. This can be seen in Figure 47 where the performance of a theoretical run-around coil is presented.

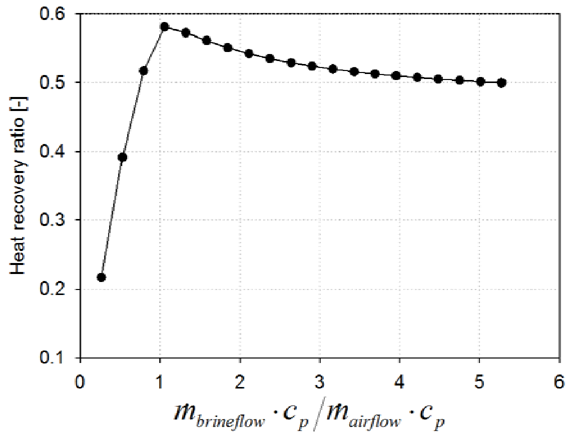

Figure 47 - Heat recovery ratio for a theoretical run around coil system at different ratios between the brine- and airside flow rates

For a real system, maximum performance may or may not occur when the ratio between the heat capacity rates is 1. This depends on the design of the heat exchangers and the air flow rate. In other words, if the air flow rate is lower than the heat exchanger is designed for, there is a risk that the flow in the heat exchanger could be laminar at the flow rate that yields a heat capacity rate ratio of 1. In 32 run-around coil systems analyzed by the author, ~20 % had an optimum system performance at a heat capacity rate ratio higher than 2 due to the fact that the brine flow becomes laminar in the heat exchanger.

These facts are interesting to consider when deciding on how to connect a heat pump to a run-around coil system. Two different ways to connect the heat pump may be considered, as shown in Figure 48.

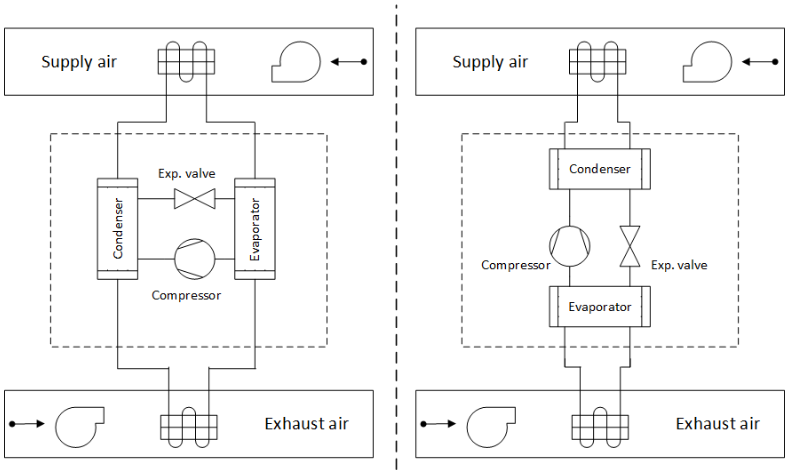

Figure 48 - Two different ways to connect a heat pump to a run around coil heat recovery system

The left system is an "inline" retrofit of a heat pump and the right system is an "in-between" option. At first glance, it might be hard to evaluate which system would have the higher system performance. The "inline" system has some obvious advantages, such as

- Working at all ambient temperatures regardless of heat pump size
- Low demand for control equipment
- Lower temperature difference between the condenser and evaporator

Drawbacks of the “inline” setup are that the performance will depend on the brine flow rate (heat capacity rate ratio). The “in-between” system setup will not be as dependent on the brine flow rate which is the greatest advantage. The “in-between” system has the disadvantage that the size of the heat pump influences at what ambient temperatures it will have a higher efficiency than the original run-around system. It also requires more control equipment compared with the “inline” option.

For systems where a heat capacity rate ratio of 1 is possible, without entering laminar flow in the heat exchangers, the “inline” option seems to be the system of choice. For systems where this is not possible, the “in-between” option could be the preferred one. The outcomes of the two systems then need to be carefully examined as the combination of relative heat pump size, size of heat exchanger of the air coils, and the heat capacity rate ratio all influence the performance of the system. Since the "in-between" option is the more complex of the two systems, this type seemed more interesting to analyze first.

In the simulation setup, a heat pump is retrofitted to a existing run-around coil system, as shown in Figure 49. Depending on the ambient temperature, the heat recovery system can be switched from run-around coil to the heat pump unit and vice versa. The different routes of the secondary fluid depend on the operation mode, and are denoted 1 and 2 in Figure 49.

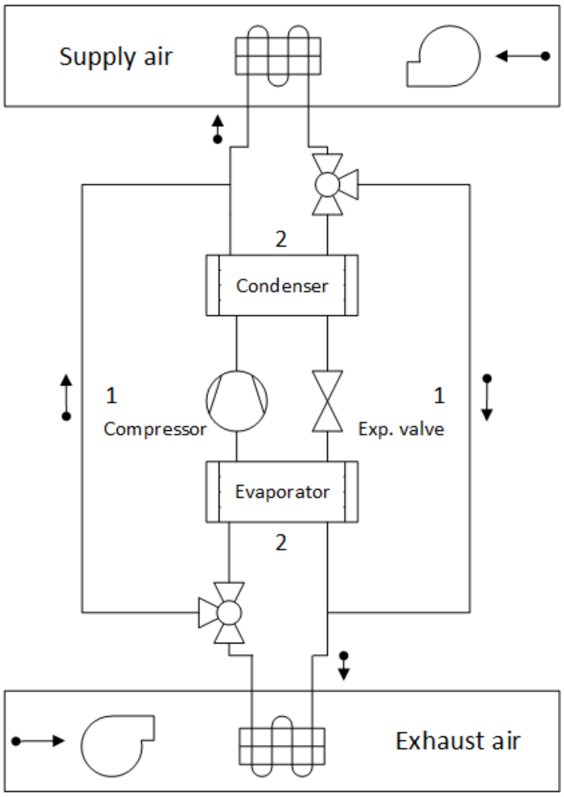

Figure 49 – Schematic of run around coil system with retro-fitted heat pump

Two different heat pump types have been evaluated, in the first case a three-stage on/off controlled heat pump unit was evaluated, secondly the system performance was evaluated using a variable capacity heat pump.

5.3.1.1 Results from the Investigation of Run-around Coil Heat Recovery System

The baseline simulation of the efficiency for the run-around coil heat recovery system indicated an annual heat recovery ratio of 47%. This model was then used to simulate the system performance with different retrofitted heat pump units. Figure 50 shows a comparison between the annual performances of the three different system setups in two European locations.

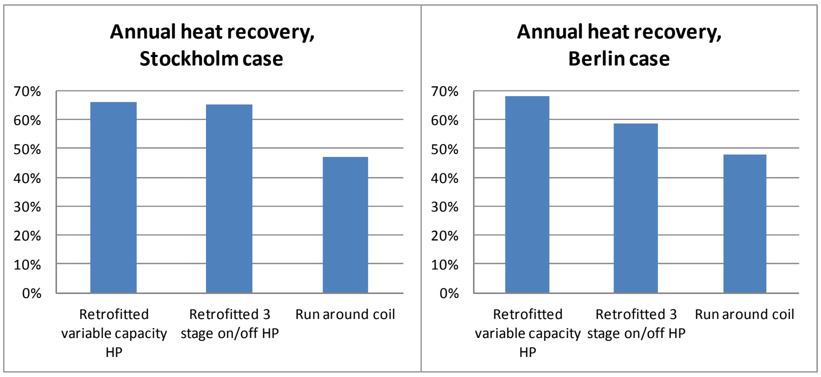

Figure 50 – Annual heat recovery for different system setups in Stockholm, Sweden and Berlin, Germany

Figure 50 shows that, for the Stockholm case, the heat recovery rate for the system with a three-stage heat pump and the system with the variable capacity heat pump is more or less the same. For the Berlin case, the performance of the retrofitted three-stage heat pump is not as efficient as the variable capacity heat pump. The reason is that the three-stage heat pump is designed for the Stockholm case. Since the heat demand is lower for the Berlin case, the three-stage heat pump is large compared to the demand. The retrofitted variable capacity heat pump system can adjust to the lower demand and the heat recovery ratio is about 2 percentage points higher than for the Stockholm case. For the three-stage heat pump recovery system, the recovery rate is around 6 percentage points lower, since the heat pump is oversized with respect to the demand. Hence, for the Berlin case the heat recovery system with the three-stage heat pump has a relatively longer period when the heat pump is not in operation compared with the Stockholm case. Thus, the sizing of the heat pump unit and system is crucial to achieving a high system performance; especially for the three-stage pump.

Another aspect of the system is to evaluate the total annual heating supplied to the air handling unit by the heat recovery system with heat pump. This may be called the degree of coverage. In this thesis, the degree of coverage for the heat pump cases is defined as the amount of energy supplied to the air going to the building, i.e. total recovery plus energy supplied to the compressor, divided by the total demand of the air handling unit. For the Stockholm case, the degree of coverage is 81 % with the variable capacity heat pump and 77 % with the three-stage heat pump retrofitted. Figure 51 shows the air handling unit total heating demand which is covered by the run-around coil heat recovery system and the heat pumps.

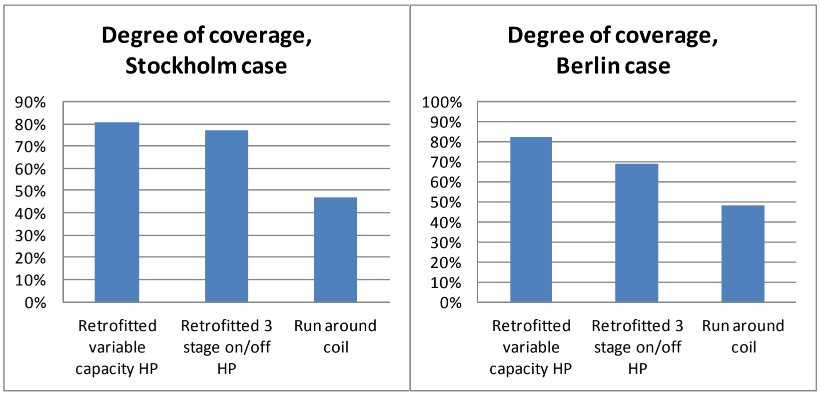

Figure 51 – Annual degree of coverage for different system setups in Stockholm, Sweden and Berlin, Germany

It is also interesting to break down the total heating energy demand of the ventilation air handling unit into the heat recovery system, the added compressor power and the auxiliary heater. In Figure 52, the distribution is illustrated.

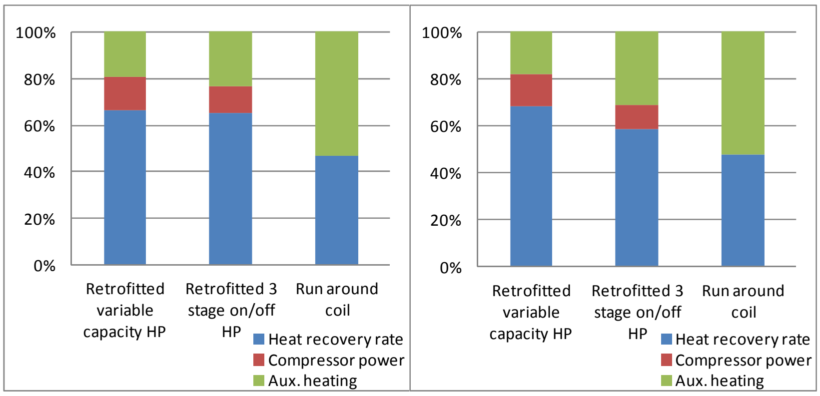

Figure 52 – Distribution of heating energy demand of air handling unit in Stockholm, Sweden and Berlin, Germany

One could also argue that the exhaust air temperature is a measure of how efficient the heat recovery system is, i.e. if the exhaust air temperature is the same as the ambient temperature, the heat recovery will be 100 %. In Figure 53, the exhaust air temperature is plotted together with the ambient temperature for the different system setups for January.

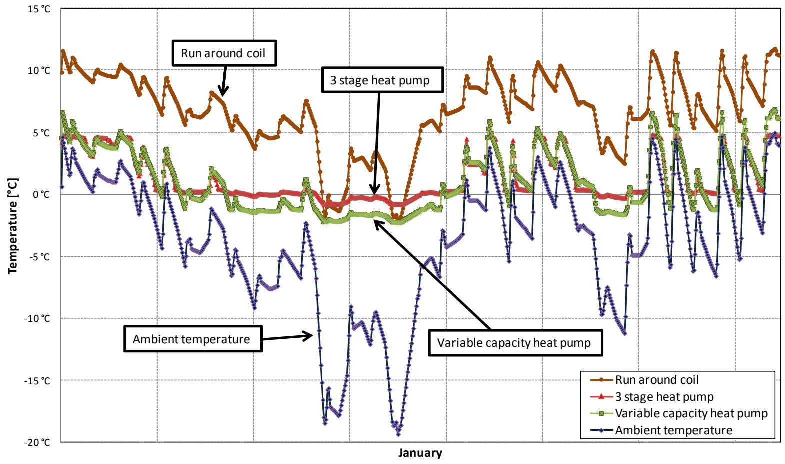

Figure 53 – Exhaust air- and ambient temperatures for different heat recovery systems in January

From Figure 53, the increase of the heat recovery is visible. It is also visible that there is an ambient temperature at which the run-around coil system alone is as efficient as the heat recovery system with heat pump retrofitted. For the three-stage heat pump system in Stockholm this is around -15 °C; for the variable capacity case, the temperature seems to be around -18 °C. However, the variable capacity heat pump reaches its maximum heat capacity at roughly -6 °C and at lower ambient temperatures, the supply air temperature after the heat recovery system with a heat pump retrofitted gradually approaches the supply temperature obtained from the run-around coil system.

5.3.2 Thermal Wheel System Analysis

To investigate the possibilities of the proposed system solution described in Figure 22, a simulation model has been built in the simulation tool TrnSys. The model is based on the specification and measurements of an air handling unit installed in an office building in Stockholm, Sweden. The objective is to evaluate the seasonal performance of the air handling unit equipped with thermal energy wheel heat recovery and to investigate how the efficiency is influenced by different characteristics of a retrofitted heat pump.

In the present investigation, a thermal energy wheel is defined as a rotating heat exchanger that transports sensible heat from the warm exhaust air to the cold supply air. Figure 54 shows the proposed system.

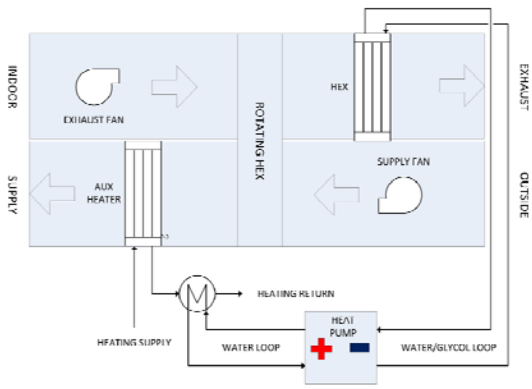

Figure 54 – Illustration of an air handling unit with thermal energy wheel heat recovery assisted with a heat pump recovery system

The advantage of this type of heat recovery system is the relative high heat recovery ratio compared with other passive[3] types of systems; the cost of the heat exchanger is also economically competitive. Examples of the drawbacks include the need to have the supply and the exhaust air-stream close to each other and the leakage between the two air streams. The heat recovery ratio is also fairly easy and inexpensive to control by adjusting the rotational speed of the heat exchanger.

Air extracted from a building is a near constant temperature heat source, which makes it a good heat source to be used together with a heat pump. Since the heat pump can "lift" the temperature of the source, heat pumping technology can make a low grade heat source more useful. Heat pump heat recovery systems can therefore be connected to the building heating system or even produce domestic hot water using a low temperature heat source. The efficiency of heat pumps depends on the temperature difference between the heat source and the heat sink. Since domestic hot water demands a higher temperature than space heating or heating of supply air, a heat pump will be more efficient if it is connected to a well designed low temperature heating system.

Generally, an exhaust air heat pump recovery system has the advantage of being flexible in the sense that the supply and exhaust air streams do not have to be close to each other. Another fact that makes it a flexible recovery system is that the recovered heat does not necessarily have to be used to heat the supply air: the ability of the heat pump to increase the temperature of the recovered heat enables heat to be recovered to a more diverse range of users. Drawbacks of this type of system are that they are much more complicated and require more maintenance than other heat recovery systems

5.3.2.1 Results from the Investigation of a Thermal Energy Wheel Heat Recovery System

Annual simulations of the heat recovery rate indicate that there is a potential to substantially increase the heat recovery ratio by retrofitting the air handling unit with a heat pump exhaust recovery. In the simulations, defrosting issues have not been considered. In Table 5-1, the results from the simulations are summarized.

[3] Heat recovery systems without heat pump

Table 5-1 - Results from simulation

Object	Result	Comment
Ventilation heat demand without heat recovery	895 MWh	Total annual energy demand of air handling unit. (heat recovery + auxiliary energy)
Energy wheel heat recovery	565 MWh	Annual total heat recovered by the energy wheel heat recovery
Energy wheel heat recovery	63%	
Heat pump exhaust heat recovery	210 MWh	Annual total heat recovered by the heat pump exhaust recovery (without compressor energy)
Heat pump exhaust heat recovery	24%	
Total heat recovery	775 MWh	The sum of annual heat recovery (energy wheel heat recovery + heat pump recovery)
Total heat recovery	87%	
Energy coverage with heat recovery + heat pump	872 MWh	Total annual energy supplied to the air handling unit (heat pump exhaust recovery + compressor energy + energy wheel heat recovery).
Energy coverage with heat recovery + heat pump	97%	
Auxiliary heating demand	23 MWh	Annual heating energy supplied to the air stream by the auxiliary heating coil.
Auxiliary heating demand	3%	

The simulation indicates that the recovery rate can be improved by 24 percentage points by implementing the proposed system. In this simulation, the heat pump annual efficiency (COP) is 3. With this COP, the energy coverage is 97 % of the total heat demand in the ventilation system (assuming no heat recovery at all). This indicates that if the system is sized and set up in this way, the maximum annual heat recovery ratio is close to 90 %.

The result from this simulation is that there is a maximum heat recovery ratio that is related to the heat pump efficiency (COP). This is because the heat demand of the air handling unit limits the amount of heat that the heat pump can deliver. Since the heat delivered by the heat pump is the sum of the heat recovered from the exhaust air and the electricity supplied to the compressor, a better COP allows a larger portion of recovered heat in the delivered heat to the air handling unit. This indicates that the heat pump sizing is an important factor to consider when building the proposed system.

The exhaust air temperatures will change depending on the outdoor temperature and the sizing of the heat pump. Figure 55 and Figure 56 show the variation of the exhaust air temperature after the thermal energy wheel and the heat pump exhaust heat recovery at a cold and warm period of a normal year in Stockholm.

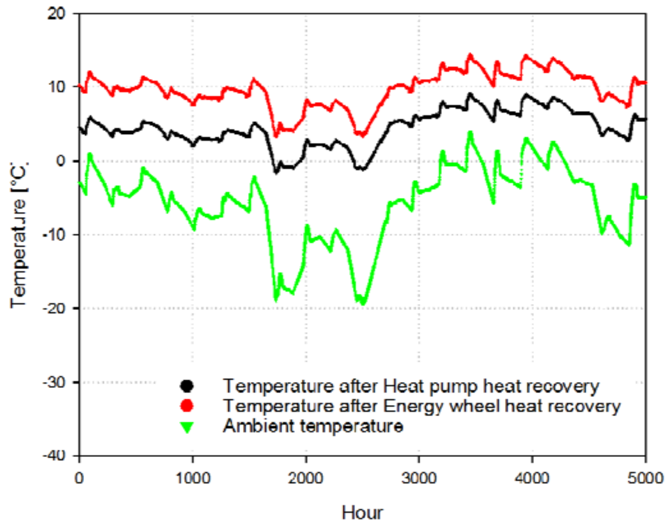

Figure 55 – Exhaust air temperatures after the two heat recovery heat exchangers in a cold period

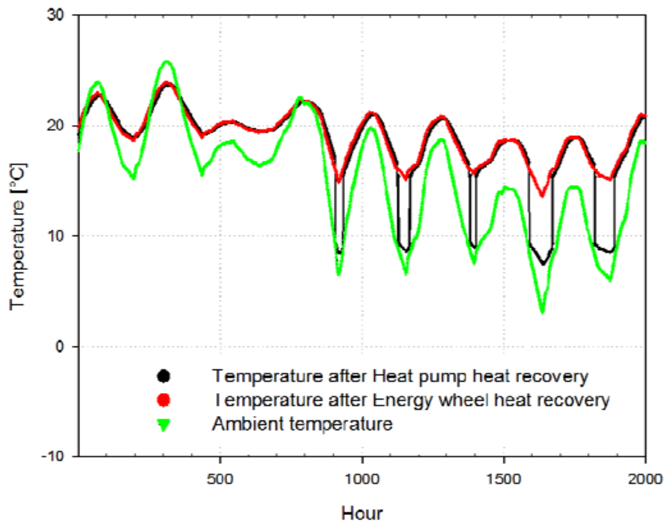

Figure 56 – Exhaust air temperatures after the two heat recovery heat exchangers in a warm period

The results from the simulation show that with the current sizing of the system the heat pump efficiency (COP) needs to be higher to recover more energy. The reason is that the energy coverage is close to 100 % already (97 %). In the cold period, the exhaust air temperature is around 4 °C higher than the ambient outdoor temperature, indicating that there is still more energy to recover from the exhaust air. This requires a larger heat pump which in turn means a higher investment cost, which could indicate that there is an economic optimum for heat pump size. As can be seen in Figure 55 and Figure 56, the heat pump recovery operates continuously when the temperature is roughly below 7 °C. At higher ambient temperatures, the heat pump is running in on/off operation. This indicates that a larger heat pump will have even more on/off cycles during one year since the cycling will start at a lower temperature.

5.4 Discussion

Investigations were undertaken regarding the possibilities of increasing the performance of ventilation heat recovery systems by retrofitting a heat pump system. Several concepts for how to increase the system performance have been identified and are presented in the present thesis.

5.4.1 Discussion Run-around Coil System

The results strongly suggest that there is a potential to retrofit a heat pump to a run-around coil system. There are several advantages in installing a heat pump to a system configuration such as this, the greatest perhaps being the ease of installation. Other advantages include a constant source temperature and a small temperature difference between source and load temperatures during most of the operating hours, which lead to an effective heat pump operation.

For the three-stage heat pump, sizing is crucial and it is somewhat complicated to achieve a high recovery rate. The variable capacity heat pump system is more flexible in sizing; however, these heat pumps are still relatively more expensive.

An interesting finding is that for the Stockholm case the variable capacity heat pump can cover 81 % of the demand for heating to the supply air temperature, and 77 % for the three-stage heat pump.

The energy recovery ratio is dependent on the supply air temperature; for a run-around coil heat recovery system; a higher supply temperature would mean a lower heat recovery ratio. The possibility to change the supply air temperature is limited since it will affect the indoor climate. In Sweden, the ventilation is not normally used to heat the building. When a heat pump is retrofitted to a system, a higher ventilation supply temperature still means a lower heat recovery ratio, but the increase in heat recovery ratio is dependent on the sizing of the heat pump.

The operational strategy for the system has been to have a brine-side supply temperature to the exhaust heat exchanger that is always over 0 °C for all operating points. This is to avoid the need for defrosting. It could perhaps be interesting to have a brine-side supply temperature to the exhaust heat exchanger that is lower than 0 °C; this would, however, require a larger heat pump and a more elaborate system for defrosting.

The possibilities would be that the heat recovery system with the retrofitted heat pump could cover 100 % of the heating demand of the ventilation air. In countries like Sweden where ambient outdoor temperatures are well below 0 °C for many hours of the year, it is impossible to cover 100 % of the ventilation air heating demand by the heat recovery system with a heat pump retrofitted without allowing brine temperatures below 0 °C. In such a case, there will be a need for defrosting the exhaust heat exchanger, thus somewhat offsetting the increased energy savings from the heat pump installation.

Another interesting option with a heat pump retrofitted to a run-around coil system is the possibility to recover cooling energy in the warm period of the year. This would perhaps be more interesting for buildings in zones that have a mixed climate, with warm summers and fairly cold winters. A system setup that offers the opportunity to recover both heat and cooling energy will be more technically complicated and require extra components, thus influencing the economy of the installation. For buildings in warm climates, a run-around coil system with heat pump could be an alternative to recover cooling energy instead of using it as a heat recovery system.

The payback for the two system configurations decreases when the size of the system increases because the price of the heat pump unit does not increase linearly with the size of the heat pump unit. Considering the lifetime (typically at least 15 years) of a heat pump system the results suggests that the life cycle profit for a retrofitting project would be considerable.

In this thesis, only one of the two possible ways to connect the heat pump to the system is analyzed. The other type, "inline" installation, will most probably be the better option if there is no restriction on the brine flow rate.

In those cases where a new run-around coil system is built, it is definitely recommended to consider a system setup incorporating a heat pump, most probably the "inline" system option

5.4.2 *Discussion Energy Wheel System*

Air handling units with thermal energy wheel heat recovery systems typically have a relatively high heat recovery ratio. New environmental goals within the European Union have introduced new requirements for the installed heat recovery systems in order to meet these goals. New system design ideas are needed that give the owners of the buildings the opportunity to increase the efficiency of their buildings with a reasonable payback on their investment.

The investigation into the idea of introducing a system with a heat pump and an extra air-side heat exchanger provides preliminary results about the technical possibilities of such a system. However, not all of the parameters have been taken into account in the analysis. For instance, it does not include the increase in air-side pressure drop that a retrofitted heat exchanger would introduce. A rough calculation of how the increase in pressure drop would affect the system that was simulated is presented in Table 5-2.

Table 5-2 – Difference in fan and motor power due to additional pressure drop by retrofitted heat recovery heat exchanger

Difference in fan and motor power due to extra heat recovery heat exchanger						
	Air flow rate [m^3/s]	drop [Pa]	Fan power [kW]	efficiency [-]	Motor power [kW]	SFP
Original setup	5.7	650	3.705	0.65	5.7	2.00
With HP HR	5.7	850	4.845	0.65	7.5	2.62
				Diff	1.8	
				Diff	0.3	
				Average extra heat recovery [kW]	24.0	

In the calculations, the extra pressure drop introduced by the retrofitted heat exchanger is assumed to be 200 Pa and the original pressure drop to be 650 Pa. The 200 Pa pressure drop was derived from manufacturers' specifications (Luvata AB, 2014) for a standard heat recovery heat exchanger designed for a flow rate of 5.7 m^3/s. The original pressure drop will, of course, depend on the system design, but the pressure drop of 650 Pa was chosen to give a specific fan power (SFP) of 2 $kW/m^3/s$. The Swedish buildings code (Boverket, 2011) stipulates that ventilation systems with heat recovery must have a SFP that is 2 $kW/m^3/s$ or better.

The specific fan power is calculated as the sum of the fan power for supply and exhaust fans divided by the larger of the supply or exhaust air flow rate. In the present investigation, the supply and exhaust air flow rate and fan powers are assumed to be the same.

The fan efficiency will not be the same for the original setup and the case with the retrofitted heat pump heat recovery: the efficiency may be better or worse depending on the operating point in the original case. However, the difference that a 200 Pa extra increase in pressure introduces into the efficiency will not be significant to the fan efficiency and it is, therefore, assumed to be constant in the calculations. It is assumed here that it is possible to achieve the new operating point by increasing the speed of the original fan. The motor power is calculated according to Equation 5.1.

$$P_{motor} = \frac{\Delta p \cdot q}{\eta_{total}} \qquad \text{Equation 5.1}$$

According to the calculations, the amount of extra heat recovered by the retrofitted heat pump recovery system is significantly higher than the extra fan power needed to overcome the added pressure drop caused by the retrofitted heat exchanger. The fan speed needs to be increased in order to keep the air flow rate at the same level as before the heat exchanger was retrofitted. It is, of course, not certain that it is possible to increase the fan speed to the level that is needed for all specific cases. There are two obvious ways to increase the fan speed: by changing the size of the belt drive pulleys or by changing the fan speed with a frequency inverter.

With regard to the heat recovery ratio, the flexibility of the heat pump system adds extra possibilities. If there is a possibility to dispose of heat to another heat sink with a larger heating demand than the air handling unit, for example a radiator system, it might be possible to size the system to have a lower exhaust temperature than the ambient temperature in certain operating points, thus increasing the efficiency of the system. In this way, the system can utilize the fact that for many hours of the year, heat recovery systems are able to deliver more than 100 % of the heating demand of the air handling unit. In these cases, defrosting will be a major issue to contend with because the surface temperature of the heat exchanger will be below freezing during many hours of operation. The effects of frost build-up on the heat exchanger surface and how to deal with this most efficiently are something that future investigations could target.

5.5 Conclusions

Two different existing ventilation heat recovery system types have been investigated regarding the technical possibility of increasing the efficiency by retrofitting a heat pump system. Heat pumps introduce interesting opportunities to increase the efficiency through the flexibility of the temperature level of the recovered heat.

5.5.1 Conclusions Run-around Coil System Analysis

Several factors that influence the performance of run-around coil heat recovery systems have been identified. Investigations have also looked into how heat pump units can be retrofitted to existing run-around coil system in order to improve system performance and consequently decrease the energy use of the system.

In conclusion, the annual modeling shows that by retrofitting a well-designed three-stage heat pump to the system described earlier in the thesis, the annual heat recovery ratio for the Stockholm case can be increased from 47 % to 65 %. For the retrofitted variable capacity heat pump, the numbers for the Stockholm case are an improvement in annual heat recovery from 47 % to 66 %.

5.5.2 Conclusions Energy Wheel System Analysis

The results from the investigation on the thermal energy wheel heat recovery system indicate that it is technically interesting to retrofit a heat pump exhaust heat recovery system to an air handling unit as described in Chapter 3.2.2. For the proposed system, the simulation indicates that the heat recovery can be increased by 24 percentage points, giving the recovery system energy coverage of 97 %.

The investigations also suggest that the efficiency of the heat pump is an important factor for the overall system efficiency and therefore it is important to consider this when sizing the system. In the present study, the heat pump has a seasonal efficiency (SCOP) of about 3; the SCOP is calculated in the simulations by dividing the annual heat supplied by the heat pump to the air-handling system with the annual electrical energy supplied to the compressor.

In the thermal energy wheel system investigation, defrosting issues have not been given any attention; if defrosting is needed, it will have a negative effect on the efficiency of the installation.

6 Discussion

New climate goals, increasing energy prices and cheaper electronics open the door for implementation of systems to increase the energy performance in existing buildings that in the past would not have been possible. The changing situation in the world also renders old standards obsolete. As an example, heat recovery systems in the past had an economic temperature heat recovery ratio somewhere between 50-70 %, depending on the system type. The industry seems slow to react to changing situations and it often applies a "business as usual" approach. In this context, it is important that the research community have the possibility to explore both old and new ideas when investigating how the energy performance of the existing building stock can be increased. In the present thesis, this type of approach has been applied to investigate how heat pump systems can be integrated into existing buildings to enhance the heat recovery ratio and thereby increase the energy performance of existing buildings.

It is clear that when investigating how different systems behave under certain conditions, many simplifications and assumptions are required. This is true regardless of whether the investigation is experimental or theoretical. It seems to be difficult for building owners and other interested parties to understand that these simplifications and assumptions need to be made in order to conduct the analysis with a reasonable effort. Perhaps this is the largest obstacle to overcome when researching new ways to reduce the energy demand in the built environment.

The present investigation introduces a few different ideas on how heat pump coupled heat recovery systems can be retrofitted to increase the energy performance of existing buildings. It is clear that this approach is interesting from a technical standpoint. However, economic analysis is needed to provide a complete picture. Installation effects of the systems in a real setting also need to be investigated before it is possible to conclude that the technology is mature. For the drain water system installation, the effect of operation, i.e. fouling of the heat exchanger surface, also needs to be investigated. The final piece of the puzzle that needs to be obtained is to prove that the proposed system types have a place in the portfolio of recognized energy efficiency measures in existing buildings.

7 Conclusions

Many different systems in existing buildings have been designed according to old frameworks. This opens up the possibility to retrofit heat recovery systems to increase the energy performance of buildings.

When buildings were commissioned in the past, the focus on energy performance was not as prioritized as it is today. This means that there are opportunities to increase the energy performance by addressing the efficiency of the individual systems. The present thesis puts forward an approach to retrofitting heat pump coupled heat recovery systems to increase the energy efficiency of theses old systems.

The general conclusion from the investigation is that it is technically possible to increase the efficiency of existing systems for ventilation heat recovery and drain water systems. However, the scope of the investigation does not include all of the desired parts and additional information is needed to provide a complete picture. Until the long-term and installation effects have been investigated, the proposed approach cannot be considered to be mature.

Highlights from the investigation:

- Description and validation of a modified version of the Wilson Plot method using the Khartabil approach (Khartabil et al., 1998) to categorize a vertical coiled heat exchanger.
- Presentation of the promising potential of a vertical drain water heat recovery system coupled with a heat pump.
- Presentation of the influence of heat pump- and storage tank sizing for drain water heat recovery system.
- Showing that the heat exchanger overall UA value can be assumed to be independent of the temperature for a run-around coil heat recovery system and therefore can be considered to be constant when simulating such a system.
- Presenting new ways to increase the efficiency for two different common ventilation heat recovery systems by introducing a retrofitted heat pump coupled heat recovery system.

8 Suggestions for future work

The investigations presented in this thesis has been performed in fictive settings; the analysis has been either theoretical simulations or experimental analysis in a test facility built in a lab setting. This means that the effects that the installations would experience in a real life setting are not validated. This is perhaps one of the most interesting factors to consider when discussing future work in this field.

8.1 Future Work - Vertical Inline Drain Water Heat Recovery in Buildings

In order to obtain a more clear picture of how this type of system works in reality, several aspects of drain water heat recovery systems should be investigated: firstly, how the flow rate in a real building affects the performance. Secondly, how the heat exchanger performance decrease over time, meaning that the degradation due to fouling of the inside area of the heat exchanger should be investigated. Thirdly, typical drain water flow rate profiles that can be used in the design stage of a system need to be developed. No real investigations looking into drain water flow rate profiles in multi-family buildings can be found in the literature. The fourth suggestion for future work is to investigate different types of heat exchangers. This could lead to knowledge of which design is the most efficient for the proposed system design. The fifth and last suggestion is probably one of the most important factors that should be investigated: the financial aspects of the proposed system. In reality, no system will be interesting unless it is economically viable.

8.2 Future Work - Heat Recovery Ventilation in Buildings

There are a few interesting topics that may be studied further based on the findings on the heat recovery ventilation system presented in the study. As always, the economics of this type of installation need to be established.

For both of the proposed system types, investigation of real installations is needed in order to study any implications that might arise. It is evident that these types of installations are needed to investigate long-term effects on the performance and also to provide information for design and control.

Control strategies should be investigated for the run-around coil system with the retrofitted heat pump system because the present study provides information that the control is important to achieving high system performance.

Evaluation of the "inline" system setup and comparison with the "in-between" option are needed to provide recommendations on when either type should be used

For the thermal energy wheel system with retrofitted heat pump, more in-depth study is required into the extra fan power needed to overcome the pressure drop introduced by the additional heat exchanger in the exhaust flow stream. Also, the need of defrosting should be analyzed to provide information on how this would influence the system performance.

Since the efficiency of the heat pump is of great importance for the system performance of the thermal energy wheel system, it would be of interest to investigate how to properly size the heat pump to the system in order to achieve the highest possible system efficiency. Perhaps the use of a variable capacity heat pump could enhance this efficiency. It would also be interesting to investigate the appropriate size of heat exchanger and heat pump to evaluate how this influences the system performance. The increase in recovery rate is at such a level that it is interesting to conduct further analysis that can reveal the deeper technical challenges of this type of installation.

Nomenclature

<u>Roman</u>

A	*Heat transfer area (m^2)*
C	*Constant (-)*
C_{max}	*Largest value of Mass flow * Cp for heat exchanger ($J\ s^{-1}\ K^{-1}$)*
C_{min}	*Smallest value of Mass flow * Cp for heat exchanger ($J\ s^{-1}\ K^{-1}$)*
COD_t	*Chemical oxygen demand (%)*
COP	*Coefficient of performance (-)*
C_p	*Specific heat capacity ($J\ kg^{-1}\ K^{-1}$)*
D	*Diameter (m)*
De	*Dean number (-)*
E_dot	*Compressor power (kW)*
ESS	*Error sum of squares (-)*
h	*Convective heat transfer coefficient ($W\ m^{-2}\ K^{-1}$)*
m	*Degrees of freedom (-) or mass (kg)*
$\dot{m}$	*Mass flow rate ($kg\ s^{-1}$)*
n	*Number of observations (-)*
N	*Sample size (-)*
P	*Power (W)*

p	*Pressure (Pascal)*
Pr	*Prandtl number (-)*
$\dot{q}$	*Heat transferred per unit time (W)*
Re	*Reynolds number (-)*
R_{tot}	*Total heat transfer thermal resistance by contact resistance and conduction (K/W)*
s	*Standard deviation (-)*
s_y	*Propagation of uncorrelated uncertainty (-)*
S_{yt}	*Standard error of fitted equation (-)*
TSS	*Total suspended solids (%)*
t	*Temperature (°C)*
$\bar{t}$	*Average temperature (°C)*
$t_{\nu,95\%}$	*Student's t value at 95 % confidence level (-)*
$t'_{95\%}$	*Measurement uncertainty at 95 % confidence level (-)*
$t_{ambient}$	*Temperature of outside air (°C)*
$t_{recovery}$	*Temperature of air after heat recovery heat exchanger (°C)*
t_{return}	*Temperature of air extracted from building (°C)*
$t_{w\ source}$	*Incoming brine temperature to evaporator (°C)*
$t_{w\ load}$	*Incoming brine temperature to condenser (°C)*
U	*Overall heat transfer coefficient (W m^{-2} K^{-1})*
$\dot{V}$	*Volumetric flow rate (m^3 s^{-1})*
Q	*Heat demand (Wh)*

Q_dot_1	*Condenser power (kW)*
q	*Flow rate (m³/s)*

Greek

ρ	Density (kg/m³)
Δ	*Difference (-)*
ε	*Heat exchanger effectiveness (-)*
η	*Efficiency (-)*
η_T	*Temperature efficiency (-)*
η_Q	*Annual heat recovery efficiency (-)*

Subscript

annual demand with heat recovery	*Annual heat demand with heat recovery*
annual demand without heat recovery	*Annual heat demand without heat recovery*
c	*Coil or Calibrated*
calib	*Calibrated value*
cold	*Value contributed to the coil side*
cooling	*Value contributed to the coil side*
cooling in	*Inlet to the coil side of heat exchanger*
cooling out	*Outlet from coil side of heat exchanger*
coil	*Value contributed to the coil side*
cri	*Critical*

drain	*Value contributed to the drain side*
drain in	*Inlet to drain water heat exchanger*
drain out	*Outlet from drain water heat exchanger*
energy change	*Change of energy in storage tank compare to previous time step*
hot	*Value contributed to the drain side*
i	*Inner*
inner	*Value contributed to the drain pipe inside*
max	*Maximum value*
max_8°C	*Maximum value if outgoing temperature of water from drain is 8 °C*
measured	*Measured value*
recovered	*Value contributed to amount of recovery*
motor	*Value contributed to the motor*
s	*Drain side*
t	*Coil side*
tank	*Storage tank*
time step	*Time step of simulation*
tot	*Total*
total	*Total*

Superscript

a	*Constant*
d	*Constant*

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Appendix A - Quality of Measurements in Drain Water Analysis

A1 Calibration of temperature sensors

There are 18 temperature sensors installed in various positions in the test facility, most of the sensors being used for monitoring. Only four of the sensors are used for calculations, to measure temperature difference over both sides of the drain water heat exchanger. The temperature sensor used is Class 1 T-type thermocouples. Class 1 states according to EN 60584-2:1993 (INTERNATIONAL ELECTROTECHNICAL COMMISSION, 1991) that the tolerance value is ±0.5 °C for this type of thermocouple. However, it is possible to achieve better accuracy by calibrating the thermocouples, which has been done in the present study and described below.

The approach that was used in the present investigation, as mentioned earlier, was first to calibrate the temperature sensors together using a calibrated temperature bath. The uncertainty of the temperature bath is not considered in the analysis since we are more concerned with differential temperatures than absolute temperatures. However, one parameter that could have an impact is the uniformity of the temperature in the bath. The manufacturer states that the bath has a uniformity of 0.01 °C at 250 °C, leading to the assumption that the uniformity does not affect the outcome of the measurements.

The calibration procedure was as follows, that output from the sensors was recorded at eight different temperatures of the bath, ranging from 5 °C to 40 °C. This is the temperature range that the sensors will experience during operation of the test facility. During the testing, a minimum of 20 readings were made per sensor at each of the eight different temperatures, giving in total a minimum of 360 readings at each of the temperatures.

To evaluate the uncertainty and to develop the calibration polynomial equation, an approach described by Taylor (1997) was used. A polynomial calibration equation is developed to adjust the measured value, so that the values used in the calculations are less uncertain. The Taylor (1997) method provides a mean to evaluate the uncertainty in the measurement and also the uncertainty in the polynomial calibration result. The following equations were used in the evaluation;

Measured average temperature:

$$\bar{t} = \frac{1}{n} \cdot \sum_{i=1}^{n} t_i \qquad \text{Equation A.1}$$

Measured standard deviation:

$$s = \sqrt{\frac{1}{n-1} \cdot \sum_{i=1}^{n} (t_i - \bar{t})^2} \qquad \text{Equation A.2}$$

Measured standard deviation of the mean:

$$s_{\bar{t}} = \frac{s}{\sqrt{n}} \qquad \text{Equation A.3}$$

Temperature measurement uncertainty at 95% confidence level, without systematic errors:

$$t'_{95\%} = \pm t_{\nu,95\%} \cdot s_{\bar{t}} \qquad \text{Equation A.4}$$

To find the overall uncertainty for the measurement all the eight individual uncertainties (of the eight thermocouples) are combined using Equation A.5.

Overall uncertainty of temp (RMS):

$$t'_{overall \cdot 95\%} = \pm \sqrt{\left(\frac{1}{n}\left({t'_{95\%}}_1^{\,2} + {t'_{95\%}}_2^{\,2} + \cdots + {t'_{95\%}}_8^{\,2}\right)\right)} \qquad \text{Equation A.5}$$

To create the calibration polynomial equation, an assumption was made that the error is linear, normally distributed and that there are no systematic errors. The calibration polynomial function is assumed to be a simple linear function;

Calibration polynomial:

$$t_c = a_1 \cdot \bar{t} + a_0 \qquad \text{Equation A.6}$$

Linear least square regression is then used to find the constants a_0 and a_1 in Equation A.6. The method uses Equation A.7 through Equation A.10.

Error sum of squares:

$$ESS = \sum_{i=1}^{N} (\bar{t} - t_c)^2 \qquad \text{Equation A.7}$$

Finding the constants a_0 and a_1 is done by minimizing ESS, this is done with the MS Excel problem solver:

$$\frac{\partial ESS}{\partial a_{m=0,1}} = 0 \qquad \text{Equation A.8}$$

Standard error of the fitted equation:

$$S_{yt} = \sqrt{\frac{ESS}{N-m}} \qquad \text{Equation A.9}$$

Overall uncertainty of the measurement using calibration polynomial equation:

$$t'_{c\cdot 95\%} = \pm t_{\nu,95\%} \cdot \frac{S_{yt}}{\sqrt{N}} \qquad \text{Equation A.10}$$

A2 Calibration of flow meters

There are two flow meters in the test facility; both are of the magnetic induction type. The flow meters are delivered calibrated from the supplier but in order to increase the accuracy, calibration is also performed on the meters when installed in the test facility.

The calibration is done according to the classic gravimetric weighing method. In this method, a high precision scale is used together with a stopwatch to measure the amount of water that passes through the flow meters during a specific time interval. Nine different flow rates are measured and a calibration polynomial equation is developed for each of the meters. The process of creating the polynomial equation is done in the same way and with the same equations as for the temperature sensors described in section A1.

A3 Propagation of uncertainty in calculations

In the present study, the output heating power of the experiment is calculated from several data that are either measured or obtained from tables and other sources. All of the individual data come with their individual uncertainty that will influence the uncertainty in the end result of the experiment. Uncertainty for the individual data propagates through the calculations to the results of the calculations. To evaluate how the different uncertainties affect the end result of the calculations, the Kline and McClintock (Kline and McClintock, 1953) approach is used. The method provides a way to analyze which of the measurements influence the uncertainty the most and therefore it provides information on where to use more accurate (more expensive) sensors.

Analysis of the propagation of the relative uncertainty is given from the general form described by Equation A.11, valid for uncorrelated uncertainty.

$$s_y = \sqrt{\sum_{j=1}^{m} \left(\frac{\partial f(\cdots)}{\partial \bar{x}_j} \cdot s_{\bar{x}_j} \right)^2} \qquad \text{Equation A.11}$$

The general form will be developed to fit the different equations that will be used in the analysis of the experiment.

A4 Uncertainty of temperature measurements

The uncertainty of the temperature sensors at different temperatures and the overall uncertainty were analyzed using the methodology described in section A1. The uncertainty of the reference sensor is not included in the uncertainty. Hence, the uncertainties of the temperature sensors are relative to each other and not to absolute temperature. This is in this case believed to be a valid approach since the subsequent analysis only incorporates temperature differences. The result from the analysis is presented in Table A-1.

Table A-1 – Uncertainty in temperature measurements

Average Measured Temperature [°C]	Average Reference Temperature [°C]	Average Dev. Of Mean	Standard Dev.	No. Measurements	Standard Dev. Of Mean [°C]	Uncertainty 95 % Conf. Level [°C]
40.11	40.31	5.4E-15	0.0225	756	0.00082	0.0016
35.13	35.28	1.7E-14	0.0186	396	0.00093	0.0018
30.11	30.23	9.7E-15	0.0158	360	0.00083	0.0016
25.13	25.18	3.6E-14	0.0160	828	0.00056	0.0011
20.09	20.13	1.0E-16	0.0163	576	0.00068	0.0013
15.10	15.08	-8.3E-15	0.0170	558	0.00072	0.0014
10.14	10.03	3.7E-15	0.0191	432	0.00092	0.0018
5.15	4.98	-2.5E-15	0.0212	1008	0.00067	0.0013
				Overall Uncertainty 95 % Conf. Level [°C]		0.0015

The data from the analysis is used to develop the calibration correlation. The calibration equation is in linear form. Equation A.12 shows the data for the temperature sensors.

Calibration equation for temperature sensors:

$$t_{calib} = t_{measured} \cdot 1.01018 - 0.19856 \qquad \text{Equation A.12}$$

Using the temperature calibration equation, the uncertainty of the temperature used for differential measurements is reduced to ±0.00077 °C at 95 % confidence level.

A5 Uncertainty of flow measurements

The flow meters were calibrated on delivery, but to ensure reliable flow measurements they were also calibrated in situ using a gravimetric weighing method as described in section A2. Table A-2 and Table A-3 describes the outcome of the uncertainty analysis for the two flow meters.

Table A-2 – Uncertainty in drain-side flow meter

Flow meter 1 - Drain side (warm)

Weight [kg]	Logg time [s]	Ref. Ave. Flow rate [kg/s]	Ref. Ave. Flow rate [l/h]	Measured Flow rate [l/h]	Deviation [l/h]	Deviation [%]
4.319	40	0.1080	388.70	389.06	-0.41	-0.11
3.765	40	0.0941	338.81	338.73	-0.76	-0.22
2.926	40	0.0732	263.34	263.82	-0.05	-0.02
3.138	60	0.0523	188.28	188.44	-0.22	-0.11
1.832	60	0.0305	109.92	109.92	-0.21	-0.19
2.341	120	0.0195	70.24	69.90	-0.48	-0.69
1.838	120	0.0153	55.13	55.96	0.72	1.30
1.037	180	0.0058	20.73	21.04	0.27	1.30
0.658	180	0.0037	13.17	13.53	0.34	2.55

The uncertainty at 95 % confidence level for Flow meter 1, according to the measurements, is ±0.40 l/h.

Implementing the calibration equation (Equation A.13) the uncertainty is reduced to ±0.29 l/h.

Calibration equation for flow meter 1:

$$\dot{V}_{calib} = \dot{V}_{measured} \cdot 1.002 - 0.264 \qquad \text{Equation A.13}$$

Table A-3 – Uncertainty in coil-side flow meter

Flow meter 2 - Coil side (cool)						
Weight [kg]	Logg time [s]	Ref. Ave. Flow rate [kg/s]	Ref. Ave. Flow rate [l/h]	Measured Flow rate [l/h]	Deviation [l/h]	Deviation [%]
3.360	90	0.0373	134.68	133.06	-1.62	-1.20
3.264	90	0.0363	130.83	129.26	-1.57	-1.20
3.396	90	0.0377	136.11	134.64	-1.46	-1.07
3.280	60	0.0547	197.20	195.33	-1.87	-0.95
3.289	60	0.0548	197.74	195.33	-2.41	-1.22
3.274	60	0.0546	196.82	195.33	-1.49	-0.76
4.094	60	0.0682	246.13	242.36	-3.77	-1.53
3.909	60	0.0652	235.04	234.50	-0.54	-0.23
3.822	60	0.0637	229.81	225.47	-4.34	-1.89

The uncertainty at 95 % confidence level for Flow meter 2, according to the measurements, is ±2.15 l/h.

Implementing the calibration equation (Equation A.14), the uncertainty is reduced to ±0.91 l/h.

Calibration equation for flow meter 2:

$$\dot{V}_{calib} = \dot{V}_{measured} \cdot 1.012 - 0.106 \qquad \text{Equation A.14}$$

Appendix B - Mathematical Reference for TrnSys Simulations

All the simulations in TrnSys were made using validated standard components. The mathematical reference to all of the components is described in Volume 5 of the TrnSys handbook (Kline et al., 2007). In this appendix, the most important equations for the main components in the simulation models are described.

B 1 TrnSys component Type 5 – Heat Exchanger

The heat exchanger is modeled as different configurations depending on where in the model it is used. In the air stream, the heat exchanger is modeled as a cross-flow heat exchanger while the heat exchangers used in the brine loop are modeled in counter-flow conditions. The calculations are performed in the same way independent of the configuration. The hot and cold side inlet temperatures and flow rates are used to calculate the effectiveness at a fixed user-provided value of the overall heat transfer coefficient. Hence, the model uses an effectiveness minimum capacitance approach.

B 1.1 Nomenclature for component Type 5

ε=heat exchanger effectiveness

UA=overall heat transfer coefficient of heat exchanger

C_{min}=minimum capacity rate ($\dot{m}_{cold} * C_{p,cold}$ or $\dot{m}_{hot} * C_{p,hot}$ whichever is lowest)

C_{max}=maximum capacity rate ($\dot{m}_{cold} * C_{p,cold}$ or $\dot{m}_{hot} * C_{p,hot}$ whichever is highest)

B 1.2 Mathematical description for component Type 5

Equation B.1 describes the effectiveness for the counter-flow configuration.

$$\varepsilon = \frac{1 - \exp\left(-\frac{UA}{C_{min}}\left(1 - \frac{C_{min}}{C_{max}}\right)\right)}{1 - \left(\frac{C_{min}}{C_{max}}\right)\exp\left(-\frac{UA}{C_{min}}\left(1 - \frac{C_{min}}{C_{max}}\right)\right)} \qquad \text{Equation B.1}$$

Equation B.2Equation B. describes the effectiveness for the cross-flow configuration with both fluids unmixed.

$$\varepsilon = 1 - \exp\left[\left(\frac{C_{max}}{C_{min}}\right)\left(\frac{UA}{C_{min}}\right)^{0.22}\left\{\exp\left[-\frac{C_{min}}{C_{max}}\left(\frac{UA}{C_{min}}\right)^{0.78}\right] - 1\right\}\right] \qquad \text{Equation B.2}$$

B 2 TrnSys component Type 3 – Variable Speed Pump or Fan without Humidity Effects

The pump is modeled using the Type 3 component in TrnSys. The component can be used as a constant- or variable-flow-rate component. The flow rate is calculated from a user-defined maximum value and a linear control function between 0 and 1. Other user-defined data supplied to the component are the fluid thermodynamic properties and rated fan power.

B 2.1 Nomenclature for component Type 3

$\dot{m}_{out}$=mass flow rate out from pump

γ=the value of the linear control function

$\dot{m}_{max}$=user specified maximum mass flow rate of pump

T_{out}=temperature of the fluid out from the pump

T_{in}=temperature of the fluid in to the pump

P=power consumption of pump

F_{par}=user defined fraction of pump power converted to thermal energy

C_p=specific heat of the fluid

B 2.2 Mathematical description for component Type 3

The outlet flow rate is calculated using Equation B.3.

$$\dot{m}_{out} = \gamma * \dot{m}_{max} \qquad \text{Equation B.3}$$

The fluid outlet temperature is calculated as;

$$T_{out} = T_{in} + \frac{P * F_{par}}{\dot{m}_{out} * C_p} \qquad \text{Equation B.4}$$

B 3 TrnSys component 663-3 - Electric unit heater with variable speed fan and proportional control

Component 663 is used as a constant speed fan in the model with heating enabled on the supply fan. The component has external proportional control of the heating power. The heater is designed not to exceed a user supplied set point value; hence, if the output temperature exceeds the set point value due to the user-specified heating capacity and the external control value, the external control value will be overridden.

B 3.1 Nomenclature for component Type 663-3

$\dot{Q}_{heater}$=power added to air stream by the heater

$\dot{Q}_{air}$=power added to the air stream by the fan and motor

$\dot{m}_{air}$=mass flow rate of air

$h_{air,out}$=enthalpy of air exiting the heater

$h_{air,fan\ out}$=enthalpy of air after the fan and before the heater

$h_{air,fan\ in}$=enthalpy of air before the fan

$\dot{P}_{heater}$=maximum heater power

$\dot{P}_{fan}$=fan power

η_{heater}=heater efficiency

η_{motor}=fan motor efficiency

γ_{heater}=heater control signal (value between 0 and 1)

γ_{fan}=fan control signal (value between 0 and 1)

f_{air}=fraction fan motor power that contributes to enthalpy increase (user input)

B 3.2 Mathematical description for component Type 663-3

The power transferred to the air stream by the heater in the fan is calculated from the enthalpy change.

$$\dot{Q}_{heater} = \dot{m}_{air} * (h_{air,out} - h_{air,fan\ out}) \quad \text{Equation B.5}$$

The enthalpies are calculated as;

$$h_{air,out} = h_{air,fan\ out} + \frac{\dot{P}_{heater} * \eta_{heater} * \gamma_{heater}}{\dot{m}_{air}} \quad \text{Equation B.6}$$

$$h_{air,fan\ out} = h_{air,fan\ in} + \frac{\dot{Q}_{air}}{\dot{m}_{air}} \quad \text{Equation B.7}$$

And the energy transferred to the air stream from the fan and motor is calculated using Equation B.8.

$$\dot{Q}_{air} = \gamma_{fan} * \dot{P}_{fan} * (\eta_{motor} + (1 - \eta_{motor}) * f_{air} \quad \text{Equation B.8}$$

B 4 TrnSys component 668 – Water to water heat pump

Component 663 is a single-stage heat pump that uses table data supplied by the user. The data supplied is heat capacity and compressor power at different load and source temperatures. The component takes incoming source and load temperatures and returns the corresponding heating and compressor power. Type 668 calculates the outgoing load and source temperatures from the returned heating and compressor power data.

B 4.1 Nomenclature for component Type 668

$\dot{Q}_{absorbed}$=power absorbed from the source fluid stream

$\dot{Q}_{heating}$=heat pump heating power at current conditions

$\dot{P}_{compressor}$=compressor power at current conditions

$T_{source,in}$=temperature of liquid entering source side

$T_{source,out}$=temperature of liquid exiting source side

$T_{load,in}$=temperature of liquid entering load side

$T_{load,out}$=temperature of liquid exiting load side

$\dot{m}_{source}$=mass flow rate on source side

$\dot{m}_{load}$=mass flow rate on load side

$C_{p,source}$=specific heat of the liquid on the source side

$C_{p,load}$=specific heat of the liquid on the load side

B 4.2 Mathematical description for component Type 668

The amount of heating power absorbed from the source fluid is given by Equation B.11.

$$\dot{Q}_{absorbed} = (\dot{Q}_{heating} - \dot{P}_{compressor}) \qquad \text{Equation B.9}$$

The outgoing temperatures are calculated as;

$$T_{source,out} = T_{source,in} - \frac{\dot{Q}_{absorbed}}{\dot{m}_{source} * C_{p,source}} \qquad \text{Equation B.10}$$

$$T_{load,out} = T_{load,in} - \frac{\dot{Q}_{heating}}{\dot{m}_{load} * C_{p,load}} \qquad \text{Equation B.11}$$

Paper 1

Paper 2

Paper 3

Paper 4