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EFFICIENCY OF HEAT RECOVERY UNITS IN VENTILATION

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DESCRIPTION

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Efficiency of heat recovery units in ventilation

Abstract

The main aim of my bachelor thesis was to calculate the annual efficiency and the temperature ratios of the heat recovery unit and compare them with the manufacturer's data and requirements of European standards. Another aim was to estimate the influence of using the heat recovery unit on heat energy consumption of the air handling unit. Furthermore, the aim was to compare real costs of heat energy for the air handling unit with the heat recovery unit with costs of heat energy which would be if there wasn't the heat recovery unit in the air handling unit.

The air handling unit with the heat recovery unit which is located in D-building of Mikkeli University of Applied Sciences was chosen for research. The type of the heat recovery unit is a heat recovery unit with a rotating wheel. The research is based on data which was obtained from the measuring devices of the handling unit. The calculations of the annual efficiency and the temperature efficiency of the heat recovery unit were performed according to EN308 and other reliable sources.

The main result of the research is that the heat recovery has high annual heat recovery energy efficiency for supply air (77,3%) and high temperature efficiency for cold months (the maximum value is 83,4% on the 3th of January, 2011 at 9:00). Comparing with manufacturer's data wasn't successful because the manufacturer's data was obtained in different conditions compared with conditions of the research. There isn't any information about the annual efficiency and the temperature efficiency of the heat recovery unit in European standards. So the comparing with standards was impossible. Another result is that using the heat recovery unit really lead to save the heat energy for the air handling unit (by 119 MWh per calculated year) and significantly reduced costs of heat energy (6559 EUR were saved). At the same time the annual heat energy consumption of the coil was 35,1 MWh per year and the annual heat energy costs were 1930 EUR per year.

Subject headings, (keywords)

Heat recovery unit, rotating wheel, plate heat exchanger, run-around exchanger, efficiency of heat recovery, temperature ratio, annual heat recovery energy efficiency

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CONTENTS

NOMENCLATURE

t_{outd} outdoor air temperature (℃)

t_{SHR} supply air temperature after a heat recovery unit (℃)

t_{ex} exhaust air temperature before a heat recovery unit (℃)

t_{EHR} exhaust air temperature after a heat recovery unit (℃)

t'_{EHR} measured exhaust air temperature after the heat recovery unit (℃)

t_s supply air temperature (℃)

t_e exhaust air temperature after fan (℃)

η_s temperature ratio of a heat recovery unit for supply air (%)

$\eta_{s,m}$ energy efficiency of a heat recovery unit for supply air per month (%)

$\eta_{s,a}$ energy efficiency of a heat recovery unit for supply air per year (%)

$\sum_1^n \eta_s$ the sum of the values of temperature efficiency of a heat recovery unit for supply air per hour (%)

η_e temperature ratio of a heat recovery unit for exhaust air (%)

$\eta_{e,m}$ temperature efficiency of a heat recovery unit for exhaust air per month (%)

$\eta_{e,a}$ temperature efficiency of a heat recovery unit for exhaust air per year (%)

$\sum_1^n \eta_e$ the sum of the values of temperature efficiency of a heat recovery unit for exhaust air per hour (%)

n the amount of hours when the heat recovery unit is working for heating (h)

m the amount of hours when the air handling unit is set on (h)

$\sum m$ sum of the amount of hours per year when the air handling unit is set on

R volume flow ratio

$q_{v,s}$ supply air flow (m^3/s)

$q_{v,s}^a$ mean supply air flow per year (m^3/s)

$\sum q_{v,s}$ the sum of supply air flows for the whole month which has values more than 1 m^3/s

$q_{m,s}$ mass supply air flow (kg/s)

$q_{v,e}$ exhaust air flow (m^3/s)

$\sum q_{v,e}$ the sum of exhaust air flows for the whole month when supply air flows are more than 1 m^3/s

Q_{HR} heat energy of a heat recovery unit (kWh)

$\sum Q_{HR}$ heat energy of a heat recovery unit per month (MWh)

$\sum Q_{HR,a}$ heat energy of a heat recovery unit per year (MWh)

c_p specific capacity (kJ/ (kg·℃)) (for air c_p=1,0 kJ/(kg·℃))

ρ density (kg/m^3) (for air ρ =1,2 kg/ m^3)

ρ_{meas} measured density (kg/m^3)

Q_{total} heat energy consumption of air handling unit without heat recovery unit (kWh)

$\sum Q_{total}$ heat energy consumption of air handling unit without heat recovery unit per month (MWh)

$\sum Q_{total,a}$ heat energy consumption of air handling unit without heat recovery unit per year (MWh)

τ one hour (h)

η_a annual heat recovery energy efficiency for supply air (%)

η_Q heat recovery efficiency for supply air per hour (%)

$\eta_{Q,m}$ heat recovery efficiency for supply air per month (%)

Q_{coil} heat power of a coil (kWh)

$\sum Q_{coil}$ heat energy of a coil per month (MWh)

$\sum Q_{coil,a}$ heat energy of a coil per year (MWh)

W_e need of electricity for air handling unit (MWh)

W_{ei}, W_{ei+1} meter registration of electricity consumption (kWh)

Q_{ci}, Q_{ci+1} meter registration of heat consumption (MWh)

q_{mep} external leakage mass flow at positive pressure (kg/s)

q_{men} external leakage mass flow at negative pressure (kg/s)

q_{mn} nominal air mass flow indicated by the manufacturer (kg/s)

q_{mil} internal leakage mass flow (kg/s)

q_{mco} carry-over mass flow (kg/s)

Δp pressure drop (Pa)

Δp_{meas} measured pressure drop (Pa)

x_{11} moisture content of the exhaust air before the heat recovery unit (g/kg)

x_{22} moisture content of the supply air after the heat recovery unit (g/kg)

x_{12} moisture content of the exhaust air after the heat recovery unit (g/kg)

x_{21} moisture content of the supply air before the heat recovery unit (g/kg)

η_x the humidity ratio

Pi heat effect (kW)

$\sum W_{electricity}$ electricity consumption of air handling unit per month (kWh)

$\sum W_{electricity,a}$ annual electricity consumption of air handling unit (kWh)

SFP specific fan power ($kW/(m^3/s)$)

1 INTRODUCTION

Energy conservation is an actual topic in our world. All countries try to save energy. There are many ways and technologies to reduce the energy consumption. In buildings HVAC systems are one of the main fields where energy conservation measures are necessary due to high energy costs. It is known that the improvement of HVAC systems leads to decrease building energy costs by 30-60%. In ventilation using heat recovery units is the one method of many others energy saving measures.

Nowadays heat recovery units are common in ventilation, especially in countries with cold climate. These devices make it possible to reduce energy consumption of the air handling unit. When we use heat recovery unit we use energy less to heating coils and use natural processes without using electricity or other types of energy which we can get only when we use energy resources. Heat recovery units can also be installed directly in the room and are a part of decentralized ventilation system. These systems are used for providing necessary air exchange complicated due to using multiple glass panes. If heat recovery units are used in systems like these, heat losses of room/building reduce by the value of heat losses through ventilation. It means that heat demand of the heating system of the building decreases as well and a smaller size of a heat source is needed for heating needs due to reduction of energy and power consumption.

Therefore, the advantage of using heat recovery units is energy saving, and as a result, savings on costs for the operation of the ventilation system. Disadvantage is a necessary additional initial investments to install a heat recovery unit.

The topic of this bachelor thesis is efficiency of heat recovery units. The main aims are to find out heat recovery units have the efficiency which manufacturers promise to consumers or not, what the efficiency of them is in practice and it corresponds to requrements of European standards or not. Furthermore, the aim is to calculate how much heat energy consumption of the air handling unit is reduced due to using the heat recovery unit.

At first, there is a theoretical background about heat recovery processes and methods of efficiency calculations in this thesis. Then there is a description of research, i.e. what exactly were calculated and what methods were used for this aim.

After this, there is an example of efficiency calculations of the heat recovery unit with a rotating wheel located in D-building of Mikkeli University of Applied Sciences. This heat recovery unit is a component of the air handling unit. The calculations are made on the basis of data which was obtained from the building automation system. Measuring equipment is installed in the air-handling unit of D-building. The data is from a period of more than one year. Data collecting has begun since the end of 2010.

Finally, there are results of all research in the form of summary tables and charts. All obtained data were analyzed comparing them with manufacturer's data and European standards and a conclusion about efficiency of using heat recovery in practice was drawn.

2 AIMS

The main aim of my bachelor thesis is to find out the annual energy efficiency and the temperature ratios of the heat recovery unit with a rotating wheel which is the component of the air-handling unit located in D-building. D-building is one of the buildings of Mikkeli University of Applied Sciences.

Another aim is to calculate the electricity and heat energy consumption of the air handling unit with the heat recovery unit and without it. The benefit of using the heat recovery should be calculated as well.

Furthermore, the results of calculations of the temperature ratios of the heat recovery unit will be compared with manufacturer's data about this equipment. The aim is to check how obtained data corresponds to manufacturer's data.

These results of calculations of the temperature ratios and annual efficiency of the heat recovery unit will also be compared with Finnish National Building Codes and other European standards.

3 METHODS

To achieve aims collecting the initial data is done. The initial data is the data obtained from the measuring devices which are installed in the air handling unit chosen for research. They are outdoor air temperature, supply air temperature after the heat recovery unit, exhaust air temperature before the heat recovery unit, exhaust air temperature after the heat recovery unit, supply air flow, exhaust air flow, electricity (for fans), heat power (for coil). All initial data is per hour. Collection of the initial data had been started since September, 2010. It is still going on. All initial data are in the form of reports for every month. The reports have .xml extention. Therefore, they can be opened in Microsoft Excel.

After the collecting data the calculations of temperature ratio of the heat recovery unit with different ways, electricity per hour for fans, heat power for coil, heat power for the heat recovery unit, heat power for the air handling unit without the heat recovery unit are made.

Also, temperature efficiency of the heat recovery unit per month, electricity per month for fans, energy per month for coil, energy per month for heat recovery unit, energy per month for the air handling unit without the heat recovery unit are calculated. Annual values of these parameters are obtained as well. All calculations are made in Microsoft Excel according to EN 308.

Then obtained values are analyzed and compared with manufacturer's data. The main results are tabled. The charts of electricity and energy consumption of the air handling unit for whole year (2/2011 – 1/2012) are drawn. Furthermore, the charts of electricity and energy costs for 2/2011 – 1/2012 are drawn.

Also, the comparing method is used to check compliance of results of calculations with Finnish National Building Codes and other European standards. Based on this, the conclusion about performance quality of the air-handling unit was done.

4 HEAT RECOVERY UNITS

In this chapter the definition of heat recovery is given. Furthermore, the different types of heat recovery units which are used in ventilation systems are considered. The main advantages and disadvantages of them are shown.

4.1 Heat recovery

Heat recovery units are used in air-handling units in order to save energy. The principle of operation is to heat the supply air with the exhaust air heat in cold season and to cool the supply air with the energy of the exhaust air in a warm season(if there is an air conditioning system in selected room/building).

There are two types of heat recovery mechanisms. Both of them have their advantages and disadvantages and values of efficiency. But using either of them leads to decrease energy consumption of air-handling unit, so saves money.

The first one is a recuperative heat exchange. It means that heat transfer happens through a surface. This mechanism is used in two kinds of heat recovery units: a plate heat exchanger and a heat recovery unit with intermediate heat-transfer agent. /1./

The second one is a regenerative heat exchange. The principle of this process is that one heat-transfer agent delivers heat to a surface and then another heat-transfer agent goes to this surface and take this heat. It means that heat-transfer agents streams this surface by turns. A rotating wheel is a kind of heat recovery units which uses this mechanism. /1./

4.2 A plate heat exchanger

Plate heat exchangers are the most common heat recovery units. Supply and exhaust air flows cross each other in a plate heat exchanger. Air flows aren’t mixed because they are separated by plates. Due to it only sensible heat is transferred and humidity ratio of air flows doesn’t change. Various materials can be used in producing flat plates, e.g. plastic, aluminum, etc. /2, p.1./

There are many advantages of using a plate heat exchanger as a heat recovery unit. First of all, it has high efficiency. Besides, installation and operation a system with such device cause low costs. These units have low pressure drops and using them is effective action for noise-damping purposes. A cross flow plate heat exchanger hasn't moving parts, so it doesn't require mechanical maintenance. As a result, this equipment is very reliable. If there are dust or contaminating substances in air, it is necessary to provide suitable filters upstream of the heat exchanger. So it's easy to clean these devices./3./

The example of a cross-flow plate heat exchanger is shown in Figure 1.

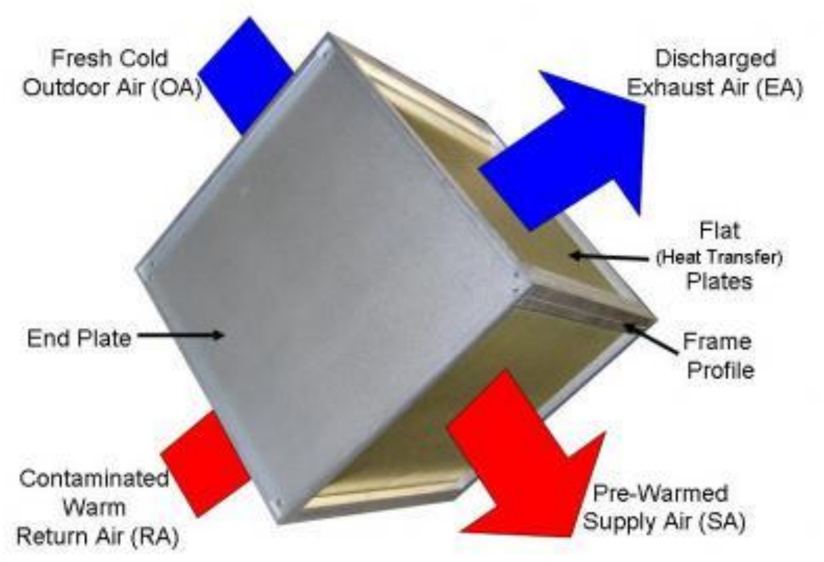

FIGURE 1. The example of plate heat exchanger /2, p.1/

There is one main disadvantage of this equipment. If a temperature of exhaust air which gives its heat to supply air becomes lower than a dew point temperature, water vapour from this air will condensate on a surface of flat plate. It can lead to icing and formation of hoar frost on internal equipment surfaces (if a temperature of plates surface is 0℃ or has a negative value) and to condensate accumulation (if a temperature of plates surface has a positive value). A dew point temperature of the exhaust air depends on its relative humidity and temperature. The more the moisture content of air the higher a dew point temperature. Pressure drop in a heat recovery unit increases due to freezing, so air flows through this device decreases and efficiency of the unit decreases as well./2, p.2-4./

Therefore, if we use a plate heat exchanger we should take into account condensation. A heat recovery unit should be oriented so that water due to condensation can easily flow downgrade out of the unit. Also the condensate shouldn't leaks into the supply air flow. To ensure it, the pressure of the supply air flow should be higher than the pressure of the exhaust air flow.

Sometimes to avoid freezing and condensation the efficiency of a heat recovery unit should be decreased. The high efficiency means that the heat exchanger transfers more heat to the supply air flow from the exhaust air flow, therefore, the temperature of exhaust air flow becomes low and it causes condensation and freezing. If efficiency is lower the temperature of exhaust air is higher and condensation and freezing don't occur.

There are many other ways to prevent a heat recovery unit from freezing. But all these methods don't help to solve this problem completely. If the device is totally frozen, it is recommended to stop its work or reduce the supply air flow while the warm exhaust air flow will provide defrosting of the device. Furthermore, there is a method which is called "Full Bypass". It is that the cold supply air goes around the exchanger through a bypass when its temperature is less than a certain temperature (temperature of a "frost limit" for surface of a heat recovery unit). This process leads to heating the device by warm exhaust air. Another method is called "Face-and-Bypass". When an automation system indicates that freezing has begun, a part of supply air flow goes through the heat recovery unit and another part goes into bypass line. It means that supply air flow decreases to certain amount due to which the temperature of exhaust air flow doesn't achieve a dew point temperature, therefore, freezing doesn't occur. "Cold Corner Bypass" is a method of defrosting as well. It is that the air channels are blocked mechanically in a part of the device, so the cold air flow reduces in the cold corner of the unit. Also, "Traversing Frost Control" is known as another way of defrosting. It is that the portions of the supply air channels are blocked temporarily, so it is given time to defrosting these ones. Last known method is called "Pre-Heat". It is used in very cold conditions. The supply air is preheated before it goes through the heat recovery unit./2, p.4-5./

A heat recovery unit can consist of some plate heat exchangers. There are some configurations of plate heat exchangers which are in Figure 2.

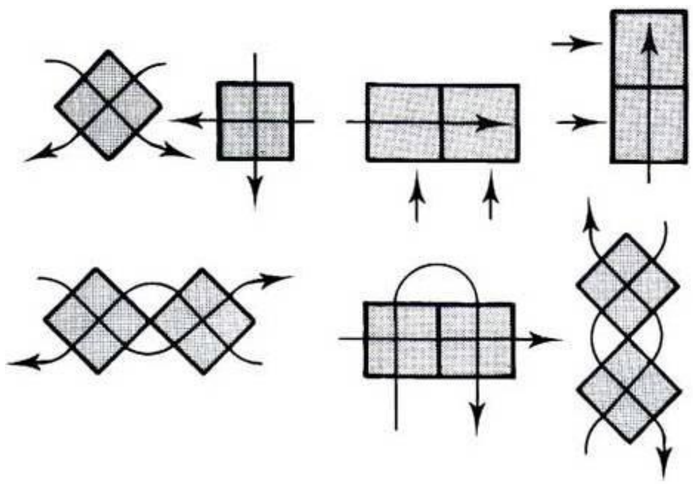

FIGURE 2. The example of plate heat exchangers' configurations /4/

4.3 A heat recovery unit with intermediate heat-transfer agent (Run-around coil heat exchangers)

A heat recovery unit with intermediate heat-transfer agent consists of two detached coils. The supply air flow goes through one of them, the exhaust air flow goes through another one. These coils are connected with pipes. There is waterglycol or water as a heat-transfer agent inside these pipes. The principle of operation is that heat from the exhaust air flow is transfered to glycol through the surface of the one coil and this heat is transfers from glycol to the supply air flow through the surface of the another coil. /5, p.48./ Figure 3 shows a scheme of a heat recovery unit with intermediate heat-transfer agent.

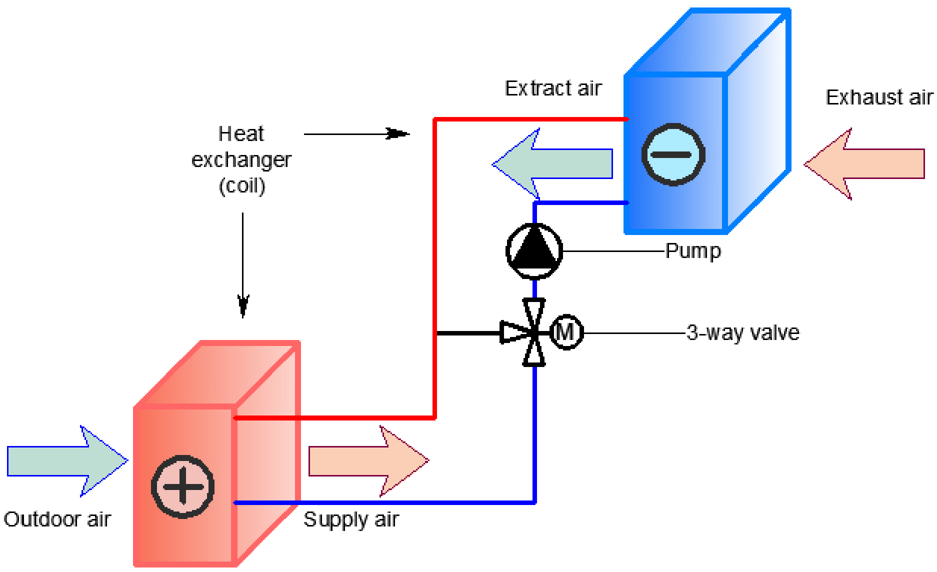

FIGURE 3. Run-around coil heat exchanger – functional scheme

A three-way valve is used in this heat recovery unit. It is needed for protection the exhaust coil from freezing. This valve provides a value of a heat-transfer agent temperature 5℃ or above. Also using of this valve ensures that the temperature of supply air flow has a certain value. /5, p.48./

Because the supply and exhaust air flows aren’t in contact with each other, the coils can be separate from each other. This fact leads to the reason that the coils are suitable for renovation. Another advantage is that water glycol or water loop doesn’t transfer humidity between the supply and exhaust airflows. Therefore, seal leakages between both flows are impossible. /5, p.48./

The main disadvantage of this type of a heat recovery unit is that the circulation pump consumes a lot of energy. This fact reduces the efficiency of heat recovery and sometimes it is unreasonable to use heat recovery in the system. Another important thing that it is necessary to maintain this unit to significant extent due to a large number of valves and fittings and a pump. Also there isn’t humidity exchange between the supply and the exhaust air flows. It shouldn’t be acceptable for buildings where humidity is one of a determining factor/6, p.92./

The efficiency of a heat recovery unit with intermediate heat-transfer agent is changed if the flow of waterglycol or water in the loop is reduced or shunted/7, p.336/.

4.4 A rotating wheel

A heat recovery unit with a rotating wheel (a thermal wheel) is a device which consists of a wheel through which the supply and exhaust air flows go in turns. It means that at first the exhaust air flow transfers the heat to an upper part of the wheel then it rotates and this heated part moves downward, the supply air flows through this one and becomes warmer. An electromotor provides rotation of the wheel.

Rotating wheels transfer both moisture and heat between two air flows**.** It is important in systems which should provide thermal comfort and indoor air quality conditions in buildings where humidity is a key variable. Rotating wheels are over half of all new air-to-air heat exchangers installed in buildings./8./ They are the most effective of heat recovery units. Depending on application type these devices have efficiency from 50 to 85 percent/9/.

Figure 4 shows a scheme of a heat recovery unit with a rotating wheel.

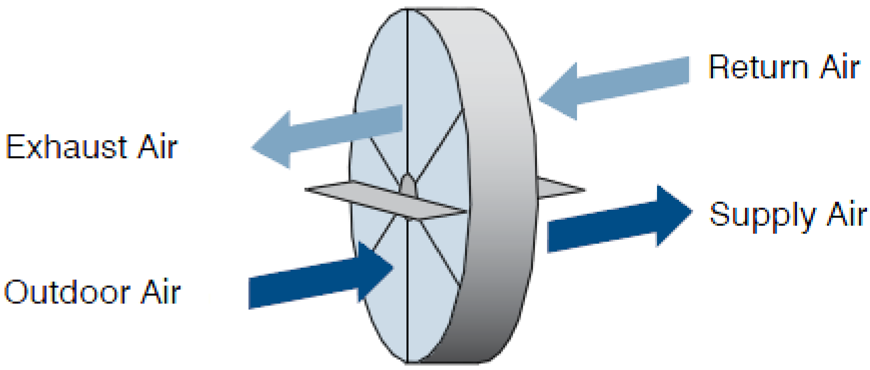

FIGURE 4. Heat recovery unit with a rotating wheel/10, p.9/

There are two types of rotating wheels. The first type is a sensible heat wheels.They transfer only sensible heat. It means that only temperature of supply and exhaust air is changed. Humidity is not transferred in this type of rotating wheels. This equipment is often used in office buildings and other types of buildings where humidity isn't a critical factor. The second type of rotating wheels is a total energy wheel, also known as enthalpy wheels. A desiccant(or absorbent) is used for humidity transfer. It can be a material such as molecular sieve or silica gel. The air flow with a higher moisture content transfers a portion of the humidity to a flow of lower humidity. The ideal amount of humidity should be taken into account in the air properties of designing calculations. The ventilation systems with enthalpy wheels should be used in schools, hospitals and other environments where it is important to maintain comfortable humidity. /9, p.1-2./

The efficiency of a rotating wheel can be changed by adjustment of the speed of the rotor /7, p.336/.

The main disadvantages of this type of heat recovery units are that, firstly, the using them is possible if the ducts of supply and exhaust air are closed to each other, secondly, the using these devices leads to electricity extra consumption which needs for the electromotor, thirdly, polluted air partially can be carried to supply air flow. Pollution can be reduced by using special constructive measures (e.g. purge zone), but they can't ensure complete treatment of air./10, p.92./ Therefore, a rotating wheel isn't allowed to use in ventilation systems of cleanrooms. Also, if the exhaust air from a bathroom exhaust system contains odors or it is toxic using a wheel isn't acceptable. To add a purge section into the wheel is a good solution from polluted air. A purge uses a small quantity of supply air which is bypassed through the purge section before the main supply air flow goes through the wheel. The wheel with the purge section uses more energy for the fan, but this decision is effective for eliminating unhealthy exhaust air./9, p.2./

Condensation and frosting are problems that can occur when a rotating wheel is used. The reason and solutions of these problems are the same as for a plate heat exchangers. However, total rotating wheels are much less susceptible to freezing and condensation than other sensible heat exchangers due to change of moisture and temperature of flows. It means when exhaust air flow transfers its heat it also gives moisture to the supply air flow, so it is dried. Therefore, the temperature of exhaust air is less likely to achieve a dew point temperature./8./

5 CALCULATION OF HEAT RECOVERY EFFICIENCY

The efficiency of heat recovery depends on factors like the type of heat exchanger, the size of the heat exchanging surfaces and the heat transferring properties of these surfaces. The efficiency can be evaluated by using results of calculations of temperature ratio for supply and exhaust air .

The temperature ratio of a heat recovery unit for supply air can be calculated using the equation 1 /11, p.7/.

$$\eta_s = \frac{t_{SHR} - t_{outd}}{t_{ex} - t_{outd}} \cdot 100 \qquad (1)$$

The temperature ratio of a heat recovery unit for supply air is about 50-60% for run-around coil heat exchangers. It is about 70-80% for plate heat exchangers and about 70% for rotating wheels./7, p.335./

The temperature ratio of a heat recovery unit for exhaust air can be calculated using the equation 2

$$\eta_e = \frac{t_{ex} - t_{EHR}}{t_{ex} - t_{outd}} \cdot 100 \qquad (2)$$

In practice η_e can be calculated with the equation 3

$$\eta_e = R \cdot \eta_s \qquad (3)$$

Where R is equal to the equation 4

$$R = \frac{q_{v,s}}{q_{v,e}} \qquad (4)$$

Also it can be obtained with equation 5:

$$R = \frac{\eta_e}{\eta_s} \qquad (5)$$

Annual heat recovery energy efficiency can be calculated with the equation 6

$$\eta_a = \frac{\sum Q'_{HR}}{\sum Q'_{total}} \cdot 100 \qquad (6)$$

where $\sum Q'_{HR}$ - the sum of energy of a heat recovery unit in each month of year, where the heat energy of a heat recovery unit per month is the sum of heat energy of a heat recovery unit per hour;

$\sum Q'_{total}$ - energy of air handling unit without heat recovery unit for whole year which is the sum of energy of air handling unit without heat recovery unit in each month of year, where energy of air handling unit without heat recovery unit per month is the sum of heat energy of air handling unit without heat recovery unit per hour;

Q_{HR} per hour can be found with the equation 7

$$Q_{HR} = c_p \cdot \rho \cdot q_{v,s} \cdot (t_{SHR} - t_{outd}) \cdot \tau \qquad (7)$$

Q_{total} per hour can be found with the equation 8

$$Q_{total} = c_p \cdot \rho \cdot q_{v,s} \cdot (t_s - t_{outd}) \cdot \tau \qquad (8)$$

6 LEGISLATION

There are many requirements to ventilation system with heat recovery units in Finnish Building Code D2. A heat recovery unit shouldn’t be used in systems if the exhaust air is polluted and it prevents the operation of the heat recovery unit or the temperature of exhaust air is lower than +15°C during the heating season. A significant value exhaust air shouldn’t be transferred to supply air in heat recovery unit. Therefore, construction and pressures of the device should be such as to provide this requirement. /12, p.27./

A pressure difference between supply and exhaust air flows or a direction of a leakage air flow in a heat recovery unit depends on the class of exhaust air flow which goes through the unit. Classes of exhaust air are presented in Table 1.

TABLE 1. Exhaust air classes/12, p.15/

Exhaust air class	Description and restrictions on use	Example of premises
1	Extract air that contains impurities in low concentrations. The main sources of impurities are human metabolism and emissions from structures. This air is suitable for use as recirculation and/or transferred air.	Office premises and related small storage areas, customer service areas, teaching areas, certain assembly areas and commercial areas with no odour load.
2	Extract air that contains some impurities. This air shall not be used as recirculation air for other rooms but it can be conducted as transferred air to e.g. toilets and wash rooms	Dwelling rooms, dining rooms, café kitchens, stores, office building storage rooms, dressing rooms and restaurants where smoking is forbidden.
3	Extract air from areas where damp, processes, chemicals and odours substantially impair the quality of extract air. This air shall not be used as recirculation and/or transferred air	Toilets and wash rooms, saunas, apartment kitchens, distribution and teaching kitchens, areas for copying drawings.
4	Extract air that contains odors or impurities detrimental to health in significantly higher concentrations than those acceptable for indoor air. This air shall not be used as recirculation and/or transferred air.	Fume cupboards in professional use, grills and local kitchen exhausts, garages and traffic tunnels, rooms for handling paints and solvents, rooms for unwashed laundry, rooms for foodstuff waste, chemical laboratories, smoking rooms as well as hotel and restaurant premises where smoking is permitted

If there is the exhaust air flow of class 1, there aren't special requarements to a pressure difference between supply and exhaust air flows or a direction of a leakage air flow. For class 2 exhaust air flow a pressure difference should be such as the direction of a leakage air flows is mainly from supply to exhaust air side. If the exhaust air of class 3 transfer its heat a pressure difference should be such as the direction of a leakage air flows is from supply to exhaust air side. /12, p.20./

It isn't allowed to mix supply and exhaust air flows for 4 class of exhaust air. Therefore, a heat recovery unit with intermediate heat-transfer agent should be used for this class of exhaust air. Regenerative exchangers(e.g. a rotating wheel) can be used if the contents of class 3 exhaust air flow isn't more than 5 %. But it is allowed to use regenerative exchangers if the contents of class 3 exhaust air flow is more than 5 % in one family dwelings. /12, p.20./

According to EN 13779 plate heat exchangers can be installed in air-handling units through which exhaust air goes from toilets and other rooms with exhaust air category 3 /5,p.44/.

According to D5 the temperature of the exhaust air after the heat recovery unit shouldn’t exceed the desired setpoint. Limiting this air temperature makes it possible to prevent the heat recovery unit from freezing. If there aren’t the data of this temperature of the heat recovery unit from a manufacturer, the minimum values of temperature of the exit air after the heat recovery unit for protection from freezing it which should be used for calculation are in the Table 2.

TABLE 2. The minimum values of the exhaust air temperature after the heat recovery unit for different types of heat recovery units/13, p.19/

Type of building	A plate heat exchanger	A rotary heat exchanger
Residential	+5°C	0°C
Others	0°C	-5°C

If there aren’t any data about the temperature ratio of the heat recovery unit, the values which can be used for calculations are shown in Table 3.

TABLE 3. Values of temperature ratio for supply air for different types of heat recovery units which is used for calculation of the annual efficiency of heat recovery/13, p.20/

Heat exchanger type	Temperature ratio η_t
Heat exchanger with fluid circulation	0.45
Cross-flow heat exchanger	0.55
Counter-flow heat exchanger	0.70
Regenerative heat exchanger	0.75

According to D3 the annual heat recovery energy efficiency of the heat recovery unit should be 45 %. This value is used for compensation calculations for reference when the building has parameters which is given in D3. For example, certain values of U-value for different types of building envelope. For real case the annual heat recovery energy efficiency can be different from the value of 45 %. If the U-values of the buiding are poorer than they are in D3, the annual heat recovery energy efficiency should be more than 45 % and vice versa./ 11,p. 14./

According to EN 308 there are three categories of heat recovery units. The first category (I) are recuperators, e.g a plate heat exchanger. The second category (II) are heat exchangers with intermediary heat transfer medium, besides the category IIa is without phase-change and the category IIb is with phase-change. A run-around coil exchanger is the category IIa. The third category (III) are regenerators, moreover the category IIIa is non hygroscopic and the category IIIb is hygroscopic. Heat recovery units with rotating wheels have the category III./14, p.3/

There are some tests which should be performed before using a heat recovery unit installed in an air handling unit of the system. The first test is an external leakage test. It shows an amount of air which leaks to or from environment when air flows go through a heat recovery unit. At first all ducts are blanked off and sealed. Then the supply and the exhaust air sides of the heat recovery unit are connected to a fan. The scheme of the connection is shown in Figure 5. /14,p.7/

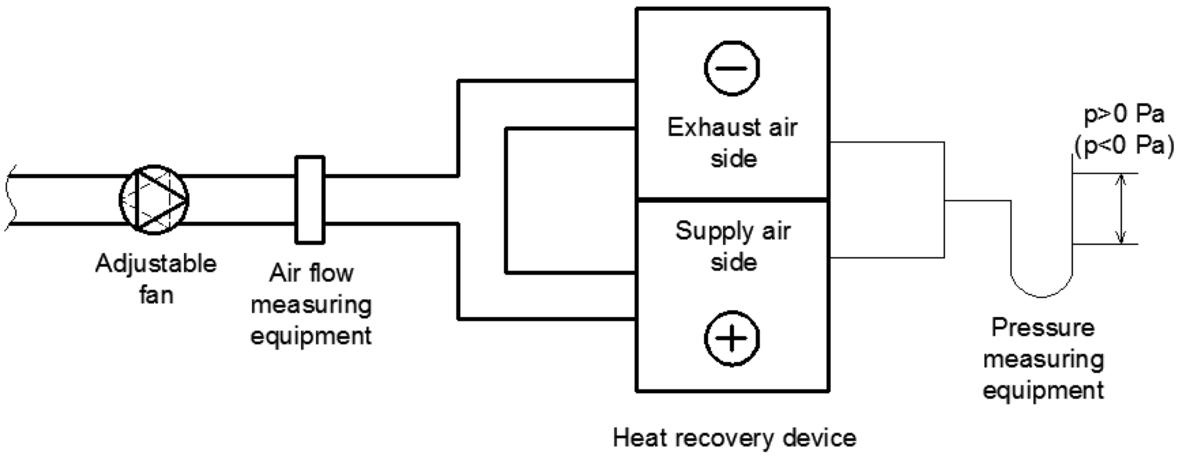

FIGURE 5. Test setup for the external leakage test/14, p.10/

The test is performed at positive and negative pressure of 400 Pa. However, if the static pressure of the system is equal or less than 250 Pa, the test pressure can be 250 Pa instead of 400 Pa. The mass air flow is measured at these values of pressure by air flow measuring equipment and is compared with nominal mass air flow of the heat recovery unit indicated by the manufacturer. All measured values are recorded in the test report. External leakage is written down as a percentage of the nominal air flow, which is calculated with equation 9/14, p.4/.

$$\Delta = \frac{q_{me}}{q_{mn}} \cdot 100\% \qquad (9)$$

During the test density of the air should be between 1,16 kg/m^3 and 1,24 kg/m^3. The accuracy of air flows measurements should be ±5% and the accuracy of the static pressure measurements should be ±3%./14, p. 7/

The second test is the internal exhaust air leakage test which is for I and IIa categories of heat recovery units. It shows a quantity of exhaust air which leaks to the supply air flow when air flows go through a heat recovery unit. First of all, all ducts are blanked off and sealed. Then the supply air side of the heat recovery unit is connected to an exhaust fan, the exhaust air side is connected to a supply fan. . The scheme of the connection is shown in Figure 6. /14,p.7/

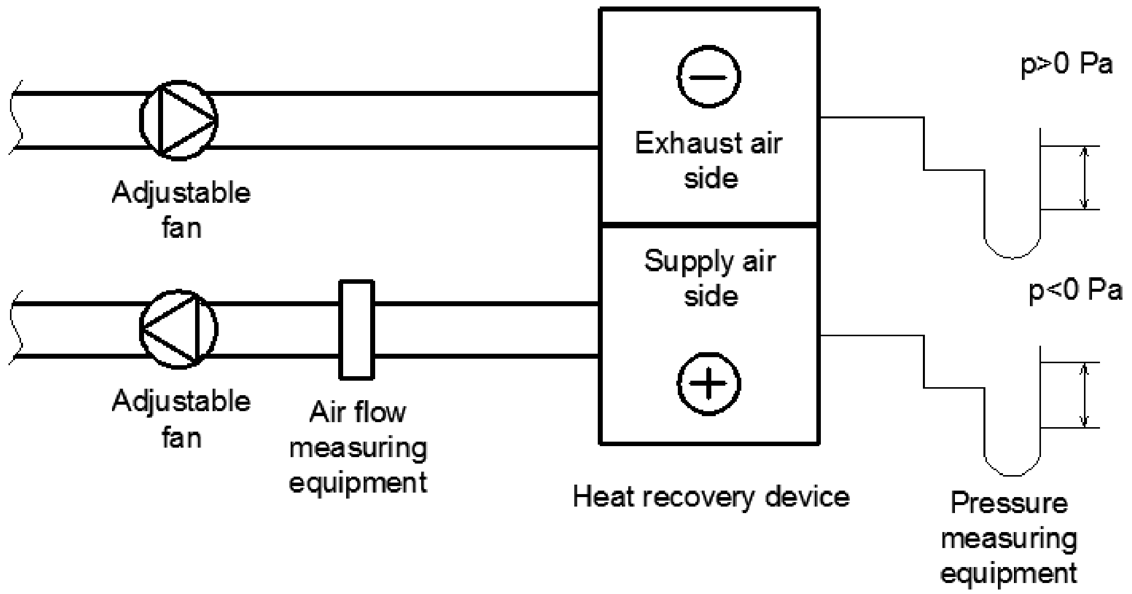

FIGURE 6. Test setup for the internal leakage test/14, p.10/

The testing pressure is 0 Pa for the supply side and 250 Pa for the exhaust side. Using the value of pressure 0 Pa leads to the internal exhaust air leakage only without any casing leakage. If the designed static pressure of the system with the heat recovery unit of category 1 is equal or less 250 Pa, the testing pressure for the exhaust side should be 100 Pa. The internal leakage is determined as a percentage of the nominal air flow with the equation 10:

$$\Delta = \frac{q_{mil}}{q_{mn}} \cdot 100 \qquad (10)$$

The result of this calculation is recorded to the test report. The inaccuracy of air flows measurements shouldn't exceed ±6%. The inaccuracy of measurements of static pressure difference between the supply and the exhaust side shouldn't exceed ±3%. During the test density of the air should be between 1,16 kg/m^3 and 1,24 kg/m^3. Also, internal exhaust air leakage test can be performed with the tracer gas technique.

The internal exhaust air leakage can be in the heat recovery units of category III, e.g. a heat recovery unit with a rotating wheel. The leakage flow depends on the effectiveness of insulation. So, overpressure on the supply side is used in these units and the manufacturer usually gives the information about leakage of supply air into the exhaust air side. In spite of overpressure requirements a small quantity of internal leakage can be obtained by the rotation of the rotor. This phenomenon is called carry-over and another type of the test is needed to define the mass exhaust air flow which leaks to the supply side of the heat recovery unit of category III./14, p.5/

Therefore, this type of the test is called a carry-over test. The test is performed with injecting inert tracer gas into the exhaust inlet section. The scheme of the heat recovery unit for the test is shown in Figure 7. Air samples are taken from sections 11, 22 and 21. The sample from section 21 is needed to check the purity of the supply air out. Air samples at sections 11 to 22 have different tracer gas concentrations a_{22} and a_{11}. Using these values the carry-over mass flow can be calculated with equation 11./14, p.7/

$$\frac{q_{mco}}{q_{m,s}} \cdot 100 = \frac{a_{22}}{a_{11}} \cdot 100 \quad (11)$$

The error of a_{11} measurements shouldn't exceed ±10%. The acceptable errors of a_{22} measurements are shown in Table 4.

TABLE 4. The acceptable errors of a_{22}

For carry-over values, %	Measuring inaccuracy for a_{22}, %	Carry-over error, %
>3	10	< ±15
0,3 to 3	20	< ±25
<0,3	50	<±50

The static pressure difference of the test should be from 0Pa to 20 Pa. During the test density of the air should be between 1,16 kg/m^3 and 1,24 kg/m^3. The supply and the exhaust mass flows should be the same and equal to the nominal air flow of the heat recovery unit.

The fourth test is a ratio test. The temperature and humidity ratios are determined during the test. The period of measurements is at least 30 minutes. The temperature ratio is calculated with equation 1 using temperatures which are measured. The measured temperatures should be adjusted with ±0,5K. The scheme of air handling unit for the test is shown in Figure 7. /14,p.7/

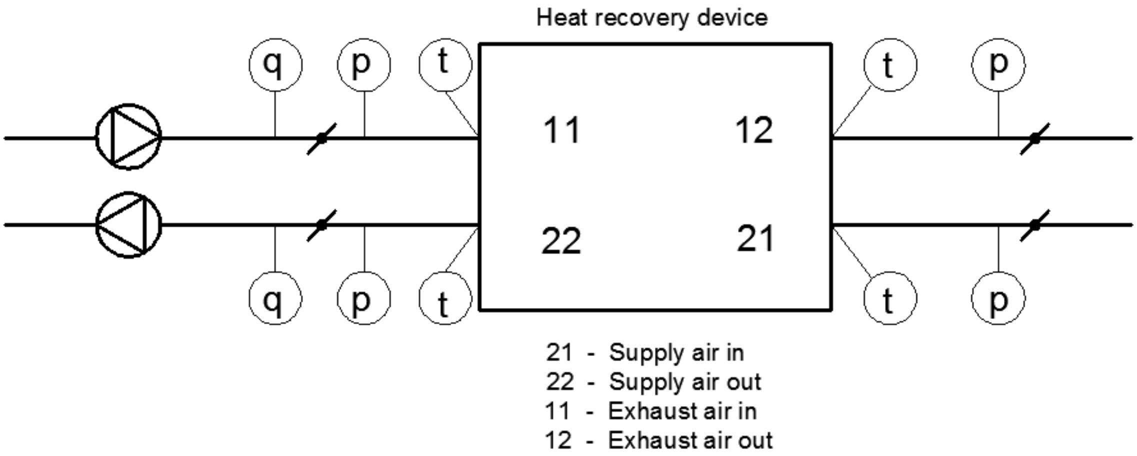

FIGURE 7. Test setup for ratio tests and pressure drop tests/14, p.11/

The dry and wet or dew point temperatures are measured to determine moisture content of the air. The humidity ratio is calculated with the equation 12:

$$\eta_x = \frac{x_{22}-x_{21}}{x_{11}-x_{21}} \qquad (12)$$

The inaccuracy of dry bulb temperature shouldn't exceed ±0,2K. The inaccuracy of wet bulb temperature shouldn't exceed ±0,3K. During measuring the wet bulb temperature velocity of the air flow should be between 3,5 to10m/s. The static pressure difference of the test should be from 0Pa to 20 Pa between sections 22 and 11. The ambient temperature should be between 17°C and 27°C, but for warm climates it can be from 25°C to 35°C. If the external and internal leakage exceed 3% of the nominal air flow the test shouldn't be performed because air leakages have an influence on temperature and humidity ratios./14, p.7/

The air conditions which should be during the test are shown in Table 5.

TABLE 5. The air conditions for the ratio test/14, p. 6/

Application mode	Category/Temperature, ℃	Category/Temperature, ℃
Recovery device category	I; II; IIIa	IIIb
Exhaust inlet air: Exhaust air temp. before HRU (t_{ex})	25	25
Wet bulb temp. before HRU	<14	18
Supply inlet air: Supply air temp. before HRU (t_{outd})	5	5
Wet bulb temp. before HRU		3

The fifth test is a pressure drop test. The scheme for test is in Figure 7. The supply and the exhaust air flows, the pressure drops on the exhaust-air side and on the supply-air side are measured during the test. The measurements should be carried out at a constant temperature. The pressure drops should be corrected for standard air with equation 13/14, p.8/.

$$\Delta p = \Delta p_{meas} \cdot \frac{\rho_{meas}}{\rho} \qquad (13)$$

The supply and exhaust air flows should be measured with errors less than ±3%. Errors of static pressure measurements should be less than ±3%/14, p.8/

Heat balance should be calculated for all tests. The heat effect ratio should be equal to 1. It is between the two flows. The error can be ±5%. Heat effect can be calculated with equation 14./14, p.8/.

$$P_i = c_p \cdot q_{mi} \cdot \Delta t_i \qquad (14)$$

It is calculated for the supply and exhaust air flows. The heat effect ratio is the heat effect of supply air divided by the heat effect of exhaust air.

7 STUDY CASE

This chapter is the practical part of this bachelor thesis. There are the description of the building and the ventilation system, the scheme of the air handling unit. Furthermore, the calculations of efficiency of the heat recovery unit are shown. There is an example of the calculations and the summary tables of obtained results.

7.1 Description of the building and its ventilation system

D-building is one of the buildings owned by Mikkeli University of Applied Sciences. It is an educational building, i.e. the classrooms take up the biggest part of the building. Therefore, D-building is a public building which has 3 storeys. There are technical rooms above the 3rd storey. Two storeys of the building was built in the 1970s. The third floor was designed in June, 2009 and built at the end of 2009. At the same time ventilation system was renovated and new air-handling units were installed.

There are the ventilation system serviced D and X-building and 6 air handling units(TK41; TK42; TK43; TK44; TK45; TK46) in this building. The ventilation system is a mechanical supply-and-exhaust ventilation system. All air handling units are in the technical rooms which are above the 3rd storey. TK41 and TK42 service the second floor. TK43 services the third floor of D-building, a basement of D-building and part of X-building. TK44 services the first and the third floors. TK45 services the first floor. TK46 services the second floor.

7.2 Characteristics of air handling unit chosen for research

TK43 was chosen for research. The heat recovery unit with a rotating wheel is in this air handling unit. Operation hours of the air-handling unit are from 7-20 5 days a week. The ventilation system doesn’t work at night and at weekends. But small fans work at that time to provide air exchange rate 0,2 1/h according to D2. The supply air flow is more than the exhaust air flow because the part of the exit air is exhausted through toilets. Furthermore, the exhaust ventilation in toilets is working all the time. Figure 8 shows a scheme of the air handling unit.

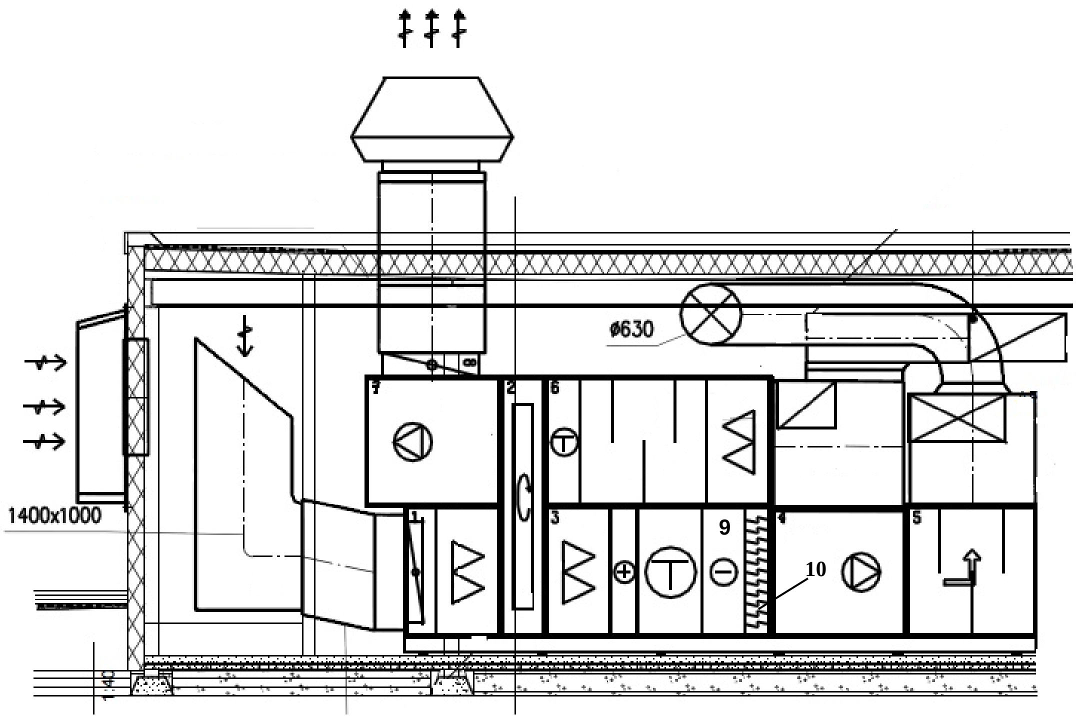

FIGURE 8. Scheme of TK43: 1 – air damper;2 – HR with a rotating wheel;3 – a filter; 4 a fan;5 – a silencer;6 – a maintenance block; 7 – a fan;8 – a damper;9 – a place for installation of a cooling coil;10 – a place for installation of a drop separator;

A place for a cooling coil and a drop separator is provided but these devices aren't installed. The owners of the building are thinking about the installation of the equipment in the future.

Designed supply air flow through TK43 is 4,00m^3/s. Designed exhaust air flow through the air handling unit is 3,60m^3/s. Designing velocity of supply air flow is 2,2m/s. Designing velocity of exhaust air flow is 2,0m/s

The manufacturer of the air handling units is KOJA, Finland. The model of the air handling unit are KOJA Future. The model of the heat recovery unit is FRTR-1512-R-2-1-AL-1-2-E-N.

There are three devices for measuring air temperature installed in the air-handling unit. The first one measures supply air temperature after the heat recovery unit (t_{SHR}). The second one measures exhaust air temperature before the heat recovery unit (t_{ex}). The third one measures exhaust air temperature after the heat recovery unit (t_{EHR}). Furthermore, two flow meters are installed for measuring supply and exhaust air flows. There are devices for measuring electricity consumption (for fans) and heat energy consumption (for coil) as well. There is a device which measures outdoor air temperature in the building automation system. It provides values of outdoor air temperature for all air handling units of the building. The data from measuring devices is sent every hour and logged in reports which are made for every month. The example of the report is shown in Appendix 1. Scheme of the air-handling unit for calculation is shown in Figure 9.

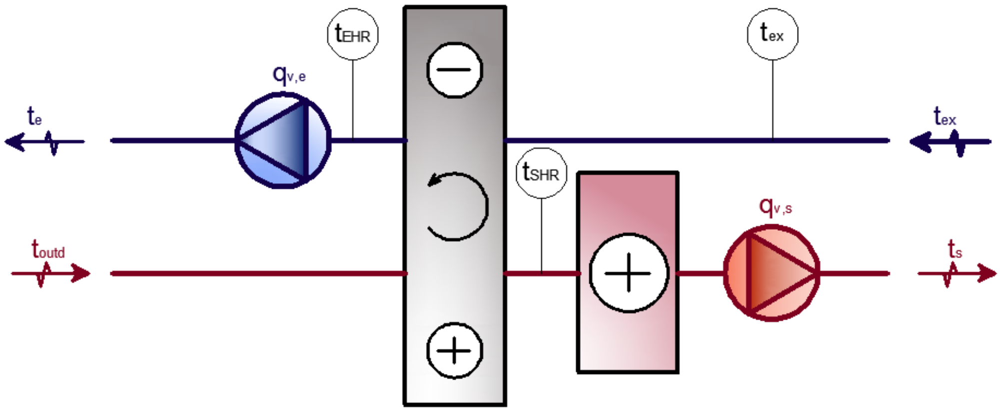

FIGURE 9. Scheme of the air handling unit for calculation

7.3 Ignored data and adopted values

During calculating many data were ignored for different reasons. First of all, the hours when the air flow rate was less than 1 m^3/s were ignored because it means that the air-handling unit and all its components are set off.

Due to the absence of data of supply air temperature (t_s) this temperature was adopted as +17°C. This value was used for the calculation of the heat energy of the air handling unit(AHU) without the heat recovery unit per hour (Q_{total}).

Furthermore, the sensor which is measuring the exhaust air temperature after the heat recovery unit is located after the bend which is before the heat recovery unit. So the measured temperature hasn't the right value. So, the correction of these data values should be done. The correction is -1,5°C, i.e. the equation 15 is

$$t_{EHR}=t'_{EHR}+(-1,5) \qquad (15)$$

Sometimes the temperature ratio of the heat recovery unit for supply air was obtained more than the temperature ratio of the heat recovery unit for exhaust air during calculating. It is impossible because the supply air flow is more than the exhaust air flow. So these results are incorrect and should be ignored. These results are obtained because of the small errors of measuring devices. Correct values of temperature ratio are obtained because errors of measuring devices haven't influence on calculation. However, sometimes these errors of measuring devices have influence on calculation because the values of data can have deviation in different side: one of the temperature value has deviation from the true value to negative side and another has deviation to positive side. Therefore incorrect values of calculation are obtained.

For example, some values of temperatures on the 10th of December, 2011 in Table 6.

TABLE 6. Data from measuring devices

Date	Hour	t_{outd}	t_{SHR}	t_{ex}	t_{EHR}
10.12.2011	08:00	-0,31	16,93	21,31	9,97
	09:00	-0,05	13,86	20,92	6,04
	10:00	0,07	13,93	20,90	6,22

The temperature ratio for supply air for these hours are equal according to equation 1 and 2:

for 8:00

$$\eta_s = \frac{16{,}93 - (-0{,}31)}{21{,}31 - (-0{,}31)} \cdot 100 = 79{,}7\%$$

$$\eta_e = \frac{21{,}31 - 9{,}97}{21{,}31 - (-0{,}31)} \cdot 100 = 52{,}5\%$$

The results for other hours are in Table 7.

TABLE 7. Calculation of temperature ratio

Date	Hour	η_s	η_e
10.12.2011	08:00	79,7	52,5
	09:00	66,3	78,1
	10:00	66,5	77,7

Also there aren't data of exhaust air temperature after the heat recovery unit for some months. They are September, October, November, December of 2010 and January of 2011. So it's impossible to calculate the values of the temperature ratio for exhaust air with the equation 2. The temperature ratios for exhaust air of these months are calculated only with one way with equation 3.

There are a small amount of operation hours of the heat recovery unit during summer time (June, July, August of 2011) and the calculated values of temperature ratios for supply and exhaust air are less than 40%. Therefore, it is adopted that the heat recovery unit was set off in these months. So, these values were ignored during calculations of annual heat recovery energy efficiency of the heat recovery unit.

7.4 Example of calculation of efficiency

According to obtained data the calculation of efficiency of the heat recovery unit was done. The data acquition has been begun since September, 2010.

For example, the calculation of efficiency of the heat recovery unit for the 2nd of January, 2012 at 12:01 was done like this:

Initial data/Appendix 1/:

- Outdoor temperature t_{outd}=-4,57°C;
- Supply air temperature after HRU(heat recovery unit) t_{SHR}=12,59°C;
- Exhaust air temperature before HRU t_{ex}=21,09°C;
- Exhaust air temperature after HRU t_{EHR}=2,62-1,5=1,12°C;
- Supply air flow $q_{v,s}$=3,47 m^3/s;
- Exhaust air flow $q_{v,e}$=2,84 m^3/s;
- Meter registration of electricity consumption W_{ei}= 69190kWh, W_{ei+1} = 69198kWh
- Meter registration of heat consumption Q_{ci}=82,22MWh, Q_{ci+1}=82,25MWh

1) The temperature ratio of the heat recovery unit for supply air (according to equation 1):

$$\eta_s = \frac{12{,}59 - (-4{,}57)}{21{,}09 - (-4{,}57)} \cdot 100 = 66{,}9\%$$

The maximum value of the temperature ratio of the heat recovery unit which was obtained during calculations is 83,4 % with data of the 3th of January, 2011 at 9:00.

2) The temperature ratio of the heat recovery unit for exhaust air (according to equation 2):

$$\eta_e = \frac{21{,}09 - 1{,}12}{21{,}09 - (-4{,}57)} \cdot 100 = 77{,}8\%$$

3) According to equation 5: $R = \frac{77{,}8}{66{,}9}$ =1,16

4) The temperature ratio of the heat recovery unit for exhaust air (according to equation 3):

According to equation 4: $R = \frac{3{,}47}{2{,}84}$ =1,22

$\eta_e = 1{,}22 \cdot 66{,}9 = 81{,}6\%$

5) The heat energy of the heat recovery unit per hour according to equation 7:

$Q_{HR} = 1{,}0 \cdot 1{,}2 \cdot 3{,}47 \cdot \big(12{,}59 - (-4{,}57)\big)/1000 = 0{,}07MWh$

6) The heat energy consumption of the air handling unit (AHU) without the heat recovery unit per hour according to equation 8:

$Q_{total} = 1{,}0 \cdot 1{,}2 \cdot 3{,}47 \cdot \big(17 - (-4{,}57)\big)/1000 = 0{,}09MWh$

7) The heat energy for coil per hour is calculated with equation/16/:

$$Q_{coil} = Q_{ci+1} - Q_{ci} \qquad (16)$$

$Q_{coil} = 82{,}25 - 82{,}22 = 0{,}03MWh$

8) The electricity for fans per hour is calculated with equation/17/:

$$W_e = W_{ei+1} - W_{ei} \qquad (17)$$

$W_e = 69198 - 69190 = 8kWh = 0{,}008MWh$

9) The heat recovery efficiency per hour according to equation /5/:

$$\eta_Q = \frac{0{,}07}{0{,}09} \cdot 100 = 77{,}8\%$$

Example of a summary table with data per hour is in Appendix 2.

After the calculations of all days of the month were done like this, the calculations of efficiency of HRU per month had been done according to these data.

For example, for January, 2012:

1) The energy efficiency of a heat recovery unit for supply air per month is calculated with equation 18:

$$\eta_{s,m} = \frac{\sum_1^n \eta_s}{n} \qquad (18)$$

$$\eta_{s,m} = \frac{19630}{296} = 66{,}3\%$$

2) The temperature efficiency of a heat recovery unit for exhaust air per month is calculated with equation 19:

$$\eta_{e,m} = \frac{\sum_1^n \eta_e}{n} \qquad (19)$$

$$\eta_{e,m} = \frac{23159}{296} = 78{,}2\%$$

The temperature efficiency of a heat recovery unit for exhaust air per month which includes values of η_e calculated with equation /3/:

$$\eta'_{e,m} = \frac{23705}{296} = 80{,}1\%$$

The relative difference between two results is

$$\Delta_{\sum \eta_e} = \frac{80{,}1 - 78{,}2}{80{,}1} \cdot 100 = +2{,}4\%$$

3) The heat energy for coil per month is equal to the difference between the last and the first in this month values of meter registration of heat consumption:

$$\sum Q_{coil} = 91{,}37 - 82{,}12 = 9{,}3 MWh$$

4) The electricity for fans per hour is equal the difference between the last and the first in this month values of meter registration of electricity consumption:

$$\sum W_e = 71579 - 69153 = 2426 kWh = 2{,}4 MWh$$

5) The heat energy of the heat recovery unit per month is the sum of the values of the heat power of the heat recovery unit per hour:

$$\sum Q_{HR} = 23{,}6 MWh$$

6) The heat energy of the air handling unit (AHU) without the heat recovery unit per month is the sum of the values of the heat power of the air handling unit (AHU) without the heat recovery unit per hour:

$$\sum Q_{total} = 30{,}0 MWh$$

Also, it can be calculated with equation 20:

$$\sum Q_{total} = \sum Q_{coil} + \sum Q_{HR} \qquad (20)$$

$$\sum Q_{total} = 9{,}3 + 23{,}6 = 32{,}9 MWh$$

This value 32,9 MWh is more accurate than 30 MWh because in first case mainly errors of measuring devices have an influence on accuracy and they are smaller than the errors of calculation with adopted supply air temperature of 17 °C in any case.

$$\Delta_{\sum Q_{total}} = \frac{32{,}9 - 30{,}0}{32{,}9} \cdot 100 = 8{,}8\%$$

7) Amount of operation hours of the air handling unit is a quantity of hours when the supply air flow is more than 1 m^3/s. For January it is 302 hours. Amount of operation hours of the heat recovery unit is the amount of operation hours of the air handling unit minus hours when incorrect values of temperature ratios of the heat recovery were obtained ($\eta_s > \eta_e$). For January it is 302 hours minus 6 hours. It means that the amount of operation hours of the heat recovery unit is 296 hours for January. Total amount of hours is 744 hours in January. So, the amount of ignored hours is 744 hours minus 296 hours, i.e. 448 hours. It is 60,2% of the total amount of hours in January.

9) Mean supply and exhaust air flows per month are calculated with equations 21 and 22:

$$q_{v,s}^{m} = \frac{\sum q_{v,s}}{m} \qquad (21)$$

$$q_{v,e}^{m} = \frac{\sum q_{v,e}}{m} \qquad (22)$$

$$q_{v,s}^{m} = \frac{1050{,}7}{302} = 3{,}48\, m^3/s$$

$$q_{v,s}^{m} = \frac{880{,}5}{302} = 2{,}83\, m^3/s$$

8) The heat recovery efficiency for supply air per month can be calculated with two methods (equations 23 and 24):

$$\eta_{Q,m} = \frac{\sum_1^n \eta_Q}{n} \qquad (23)$$

$$\eta_{Q,m} = \frac{\sum Q_{HR}}{\sum Q_{total}} \cdot 100 \qquad (24)$$

$$\eta_{Q,m} = \frac{23471}{296} = 79{,}3\%$$

$$\eta_{Q,m} = \frac{23{,}6}{30{,}0} \cdot 100 = 78{,}8\%$$

The calculated values were tabulated. Table 8 shows a table of calculations per month for January, 2012.

TABLE 8. The calculations per month for January, 2012

Name of calculated value	Value	Units
$\eta_{s,m}$	66,3	%
$\eta_{e,m}$	78,2	%
$\eta_{e',m}$ (with air flows)	80,1	%
$\Delta_{\sum \eta_e}$	+2,4	%
ΣQ_{coil}	9,3	MWh
$\Sigma W_{electricity}$	2,4	MWh
ΣQ_{total}	30,0	MWh
ΣQ_{HR}	23,6	MWh
$\Sigma Q_{coil} + \Sigma Q_{HR}$	32,9	MWh
$\Delta_{\sum Q_{total}}$	8,8	%
$\eta_{Q,m(with\ energy)}$	78,8	%
$\eta_{Q,m(average)}$	79,3	%
Time which was ignored	448,0	h
Total time	744,0	h
Operation time of HRU	296,0	h
Operation time of AHU	302,0	h
Mean q_s per month	3,48	m3/s
Mean q_e per month	2,83	m3/s

Obtained values for other months are in the summary table which is in Appendix 3. After that the annual values of calculation were obtained as arithmetic average of monthly values or sum of them. To estimate the energy conservation effect monthly and annual costs of electricity and district heating (for coil) of the air handling unit were calculated. The prices of energy which were used were taken from the web site of Etelä-Savon Energia Oy company. It is the local energy company in Mikkeli. The price of electricity which was used for calculations of electricity costs is 100 EUR/MWh. The price of district heating which was used for calculations of district heating costs is 55,04 EUR/MWh. Costs of electricity and heat energy of the air handling unit per year were obtained. These data and other annual data of calculation are shown in Table 9.

TABLE 9. The calculations per year chosen for research

Annual values					
$\eta_{s,a}$	63,9	%	$\Delta_{\Sigma Qtotal,a}$	6,7	%
$\eta_{e,a}$	77,9	%	$\eta_{a(with\ Qtotal)}$	82,9	%
$\eta'_{e,a}$ (with air flows)	77,0	%	$\eta_{a(with\ \Sigma Q_{HR},a+\Sigma Qcoil,a)}$	77,3	%
$\Delta_{\eta e,a}$	1,1	%	**Time which was ignored**	6736	h
$\Sigma Q_{coil,a}$	35,1	MWh	**Total time**	8735	h
$\Sigma W_{electricity,a}$	30,9	MWh	**Operation time of HRU**	1999	h
$\Sigma Q_{total,a}$	143,8	MWh	**Operation time of AHU**	3776	h
$\Sigma Q_{HR,a}$	119,2	MWh	**Mean q_s per year**	3,43	m^3/s
$\Sigma Q_{HR,a}+\Sigma Q_{coil,a}$	154,2	MWh	**Mean q_e per year**	2,83	m^3/s
			R	1,21	
Annual Costs					
Costs of DH with HR	1 930 €				
Costs of operation (electricity)	3 089 €				
Costs of DH without HR	8 488 €				

7.5 Results

11 months of 2011 (from February to December) and 1 month of 2012 (January) were chosen for analyzing and obtaining values of annual heat recovery energy efficiency, annual energy consumption and annual costs of maintanance of the air handling unit.

According to obtained values of energy consumption of the air handling unit shown in Appendix 3 the diagram of the monthly energy consumption of TK43 was drawn. It is shown in Figure 10.

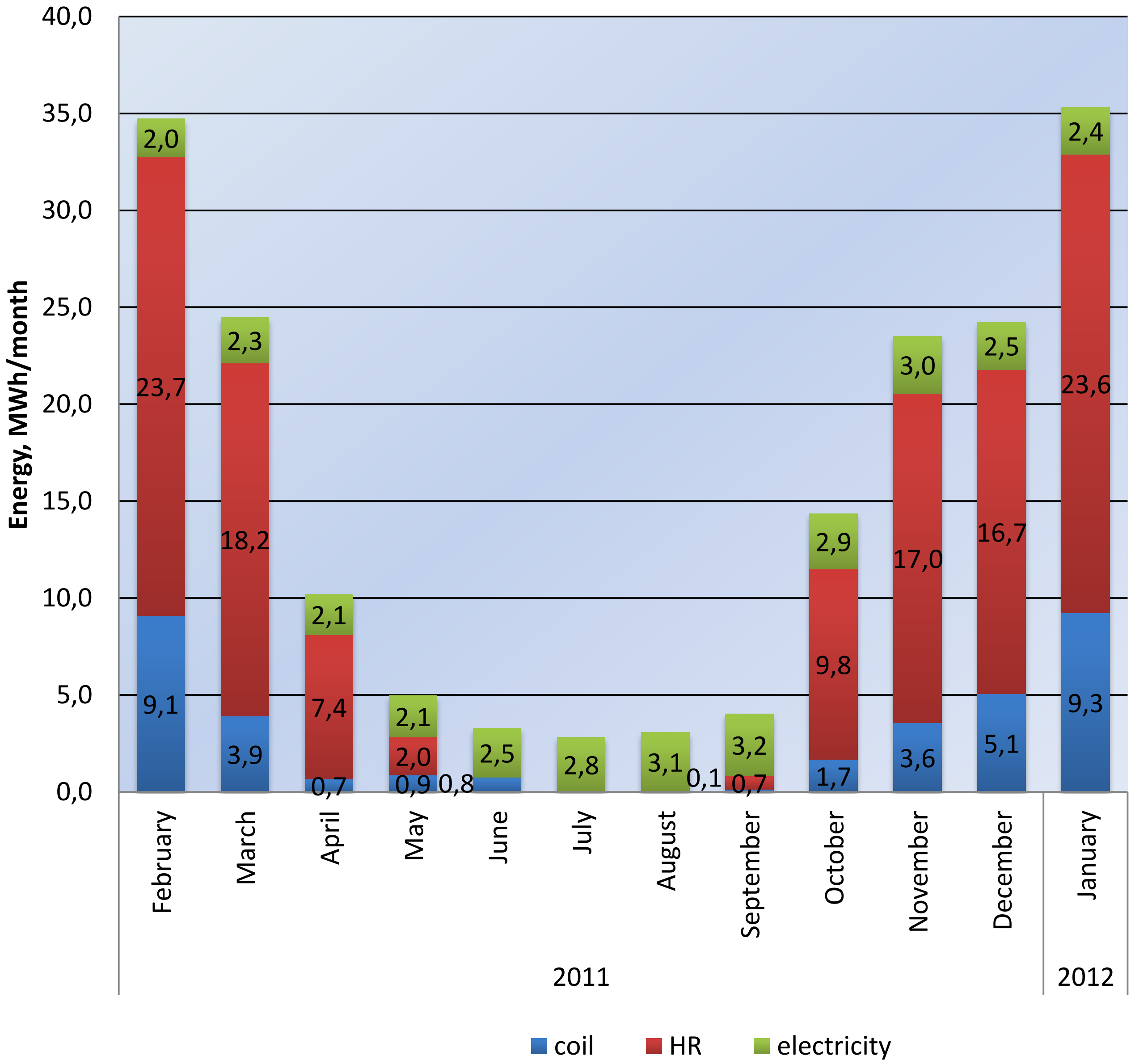

FIGURE 10. Monthly energy consumption of TK43, 2/2011 – 1/2012

It is seen that more heat energy for coil is required in winter time due to high difference between outdoor air temperature (t_{outd}) and supply air temperature (t_s). The maximum value of the energy of the coil (district heating load) is 9,3 MWh in January, 2012. The minimum value of it is 0,02MWh in August, 2012. The lowest value of the heat recovery efficiency ($\eta_{Q,m}$) is 78,8% in January, 2012 and the highest value is 93% in September, 2011. The electricity consumption fluctuates from 2 MWh to 3,2 MWh per month. More electricity for the air handling unit is required in summer months because the air flow rates are bigger in this period.

Using the heat recovery unit leads to decreasing the consumption of energy for the coil. The reduction is (23,6/(23,6+9,3))·100=71,9% at least (by example of January, 2012: the energy consumption of the coil – 9,3MWh; the energy consumption of the heat recovery unit – 23,6MWh). As a result the costs of district heating (for the coil) reduced as well. The diminution of the costs is ((156-48)/156)·100=69% at least according to Table 9 (by example of May,2011: the costs with the heat recovery unit - 48 EUR; the costs without heat recovery unit - 156 EUR).

Monthly mean supply and exhaust air flow rates were calculated and the diagram was drawn. It is shown in Figure 11.

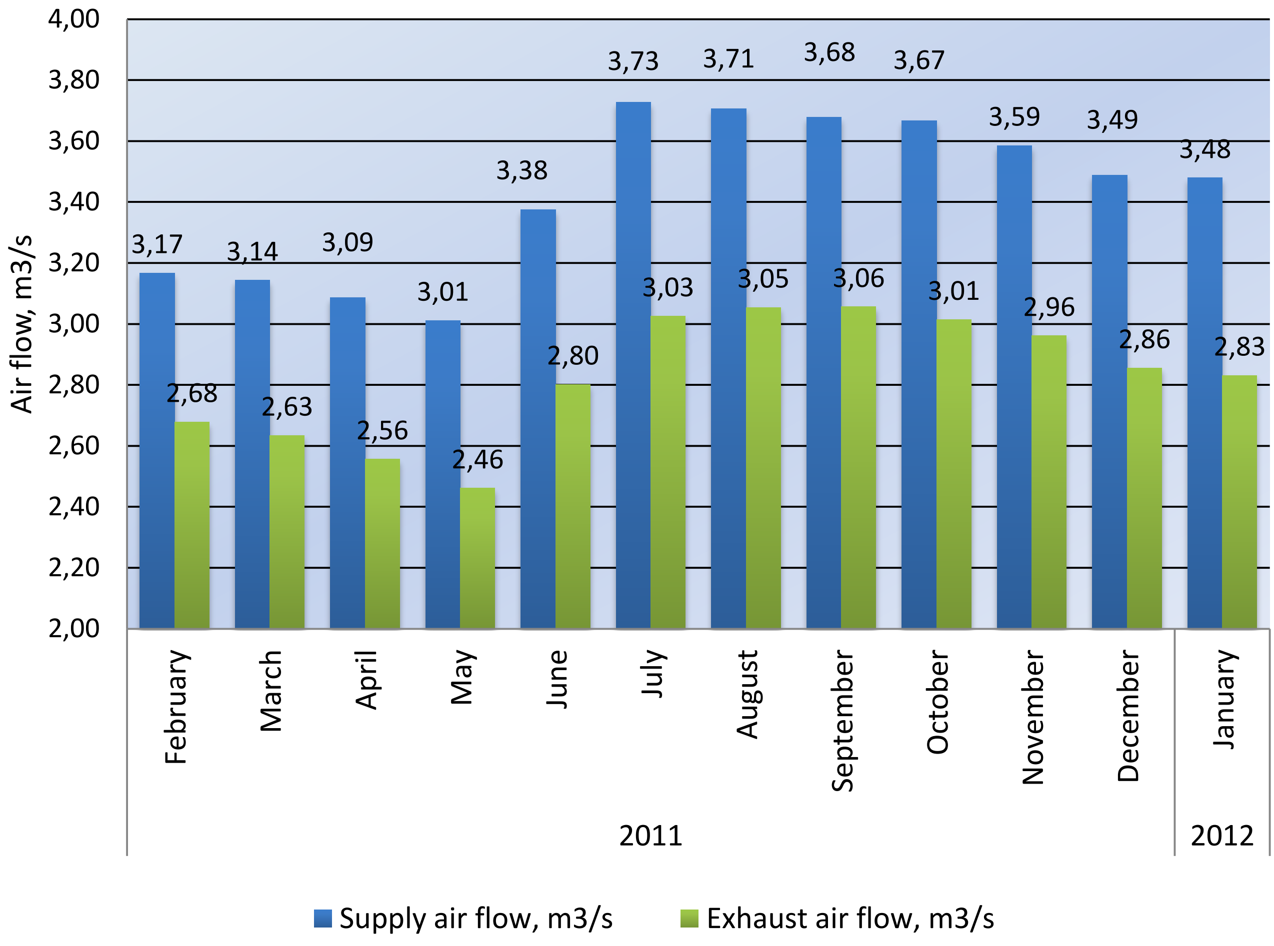

FIGURE 11. Mean air flow rates per month of TK43, 2/2011 – 1/2012

The values which are shown in the diagram are from Table 10.

TABLE 10. Mean air flow rates per month/per year of TK43, 2/2011 – 1/2012

Value		Per month												Per year	Units
		2011											2012		
		Feb.	March	April	May	June	July	Aug.	Sep.	Oct.	Nov.	Dec.	Jan.		
Mean Supply Air flow	q_s	3,17	3,14	3,09	3,01	3,38	3,73	3,71	3,68	3,67	3,59	3,49	3,48	3,43	m^3/s
Mean Exhaust Air flow	q_e	2,68	2,63	2,56	2,46	2,80	3,03	3,05	3,06	3,01	2,96	2,86	2,83	2,83	m^3/s
Volume Ratio (with eq. 4)	R	1,18	1,19	1,21	1,22	1,20	1,23	1,21	1,20	1,22	1,21	1,22	1,23	1,21	-

The maximum values of the mean supply and exhaust air flow rates are in September, 2011. The highest value of the mean supply air flow rate is 3,68 m^3/s. The highest value of the mean exhaust air flow rate is 3,06 m^3/s. It is the reason why the electricity consumption of this month is maximum. The minimum values of the mean supply and exhaust air flow rates are in May, 2011. The lowest value of the mean supply air flow rate is 3,01 m^3/s The lowest value of the mean exhaust air flow rate is 2,46 m^3/s.

Annual data of the energy consumption of the air handling unit was obtained by summing the values of energy consumption of the air handling unit and energy saved by the heat recovery unit. It is shown in Figure 12.

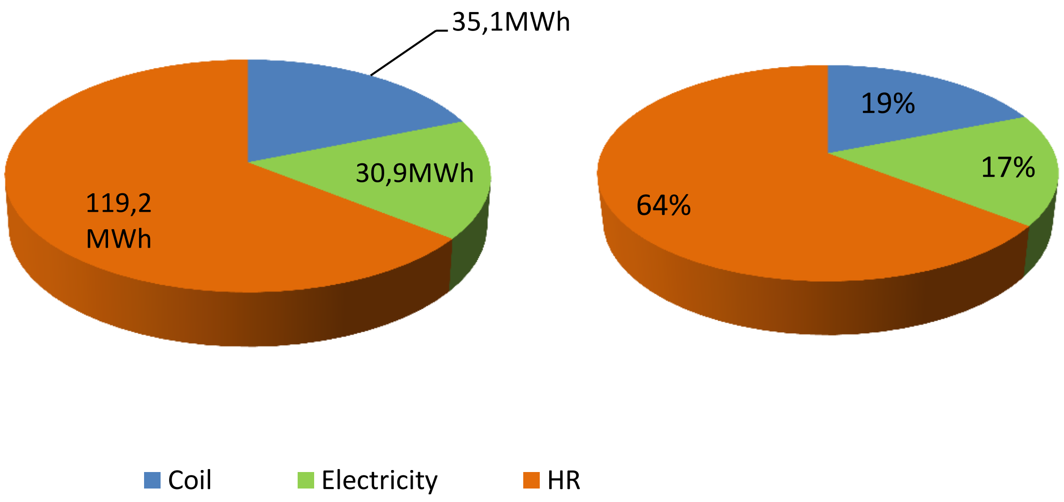

FIGURE 12. Annual energy consumption of TK43, 2/2011 – 1/2012

It is seen that the heat recovery unit saved 119,2 MWh of the heat energy of the coil. It is 64% of annual energy consumption of the air handling unit and ((119,2/(119,2+35,1))*100)=77,3% of annual heat energy consumption of the air handling unit.

Annual heat recovery energy efficiency for supply air was calculated in two ways with equation 6. The first way is to calculate with the sum of obtained ΣQ_{total} per each month of the calculated year when the supply air temperature was adopted +17°C. The second way is to calculate with the sum of ΣQ_{coil} and ΣQ_{HR} per each month of the calculated year.

The first way: $\eta_a = \frac{119,2}{143,8} \cdot 100 = 82,9\%$

The second way: $\eta_a = \frac{119,2}{154,2} \cdot 100 = 77,3\%$

Monthly costs of TK43 with the heat recovery unit are shown in Figure 13. Also in Figure 14 monthly costs of TK43 are shown in condition when there isn't the heat recovery unit in the air handling unit.

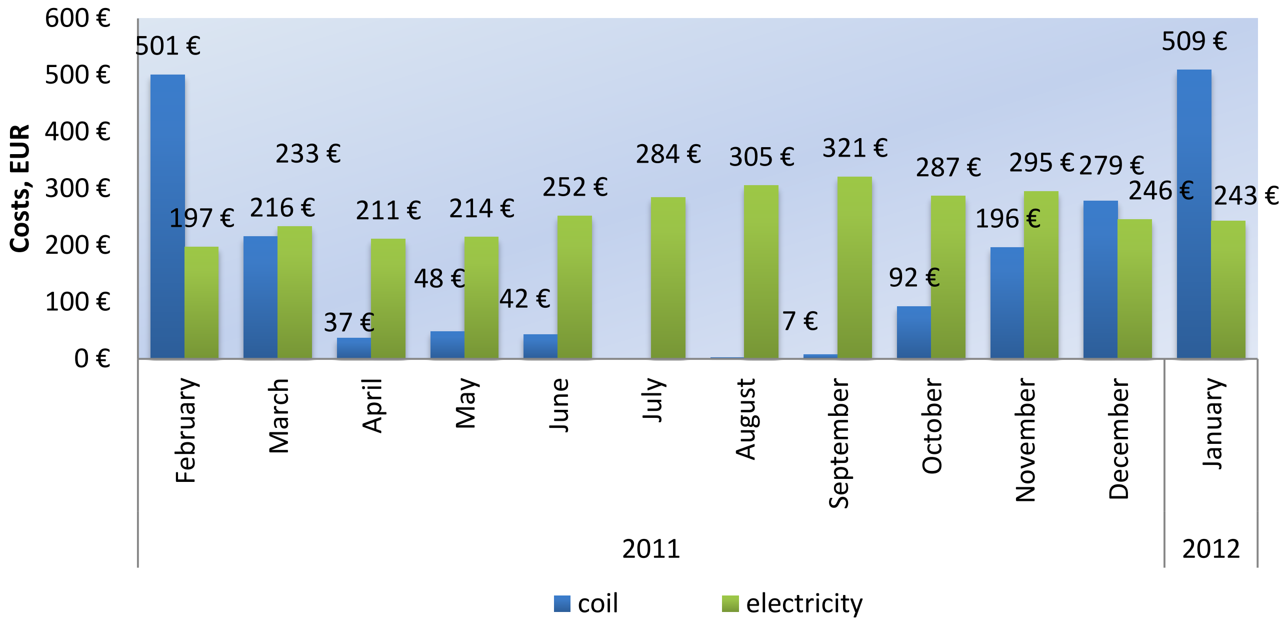

FIGURE 13. Monthly costs of TK43 with the heat recovery unit, 2/2011 – 1/2012

It is seen that electricity is more expensive than energy of district heating. The maximum costs of electricity is 321 EUR in September, 2011 due to maximum electricity consumption and the maximum costs of district heating is 509 EUR in January, 2012 due to maximum heat energy consumption of the coil.

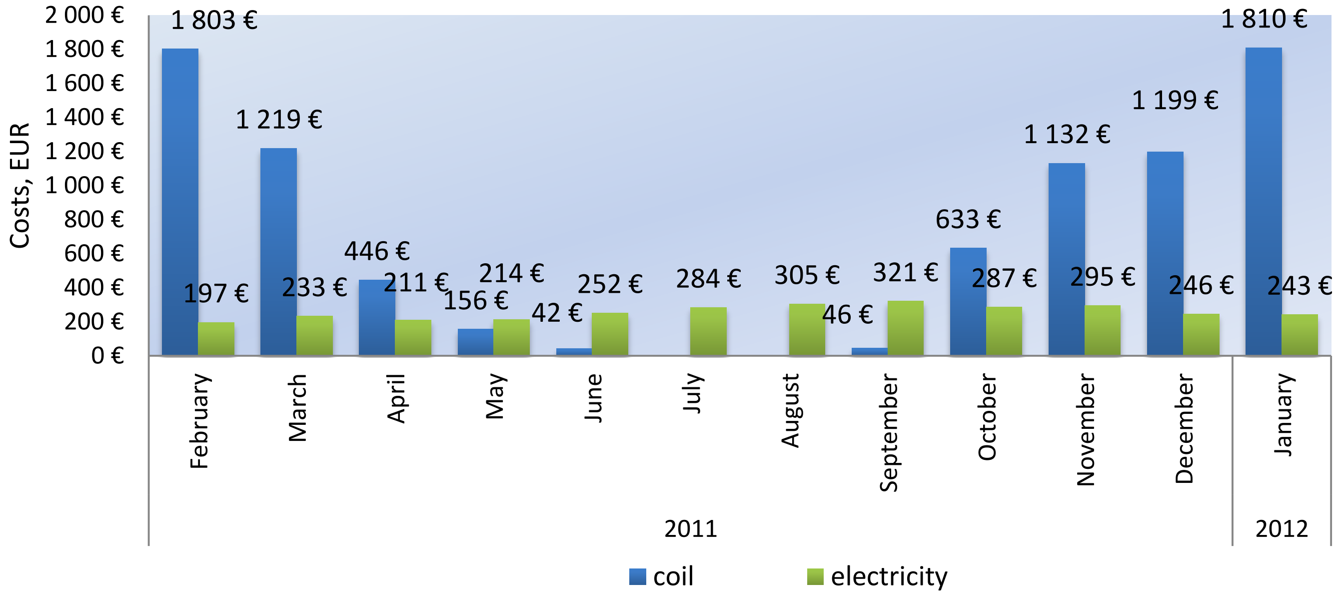

FIGURE 14. Monthly costs of TK43 without the heat recovery unit, 2/2011 – 1/2012

After Figure 14 was analyzed it became clear that the costs of the district heating increased sharply. For example, the costs of district heating is 509 EUR in January, 2012 for the real air handling unit with the heat recovery unit. The costs of district heating is 1810 EUR in January, 2012 for the theoretical air handling unit without the heat recovery unit. So, 1301 EUR was saved due to using the heat recovery unit. It is 3,6 times more than what was spent.

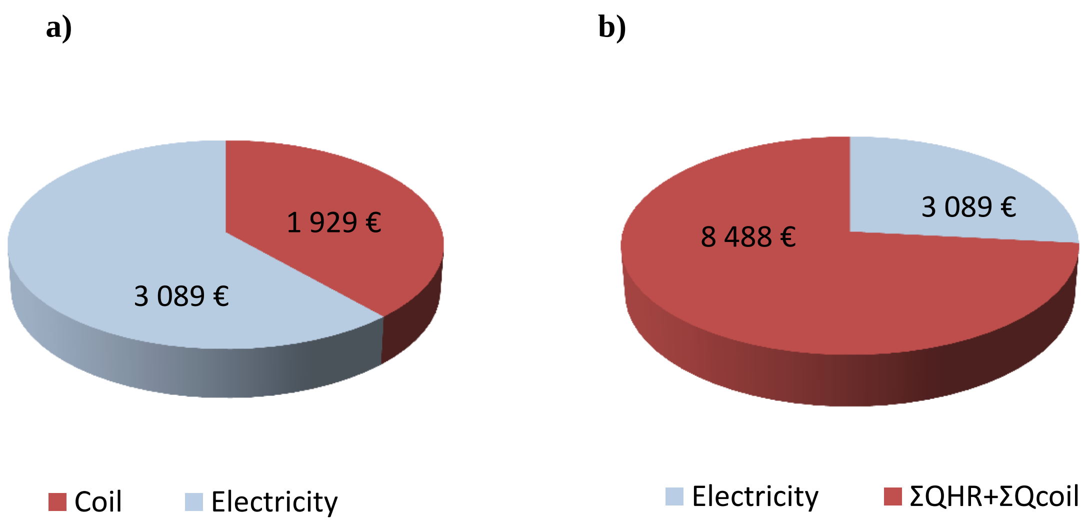

FIGURE 15. Annual costs of TK43, 2/2011 – 1/2012: a) with the heat recovery unit; b) without the heat recovery unit

Then, the annual costs of TK43 were analyzed and the diagrams were drawn. They are in Figure 15.

The annual costs of the real air handling unit with the heat recovery unit and the air handling unit without the heat recovery unit were compared. It is obvious that the costs decrease extremely using the heat recovery unit. The difference is 6559 EUR. So, the costs decreases by 77,2%.

All values of costs of TK 43 are shown in Table 11 as well.

TABLE 11. Costs of the operation of the air handling unit per month/per year of TK43, 2/2011 – 1/2012

Year	Month	Costs, EUR/month		
		Electricity	Coil+HRU	Without HRU (Only Coil)
2011	February	197	501	1803
	March	233	216	1219
	April	211	37	446
	May	214	48	156
	June	252	42	42
	July	284	0	0
	August	305	2	2
	September	321	7	46
	October	287	92	633
	November	295	196	1132
	December	246	279	1199
2012	January	243	509	1810
Sum		3089	1930	8488

Specific fan power (SFP) of the air handling unit was calculated with equation 25:

$$\mathrm{SFP} = \frac{\sum W_{\mathrm{electricity,a}}}{\sum \mathrm{m} \cdot q_{v,s}^{a}} \qquad (25)$$

$$\mathrm{SFP} = \frac{30890}{3776 \cdot 3{,}18} = 2{,}6\mathrm{kW}/(\mathrm{m}^3/\mathrm{s})$$

According to D3 SFP of mechanical supply and exhaust air system shouldn't be more than 2,0 kW/(m^3/s). The obtained value is bigger than that one. The reason is that the designing of the ventilation system was when Finnish National Building Code hadn't so strict requirements. The value of SFP which is 2,0 kW/(m^3/s) was approved in 2012 in new version of D3.

The data about the heat recovery unit which was given by the manufacturer is the temperature ratios for supply air exhaust air. The temperature ratio for supply air was obtained during the test according to EN308 with inlet air temperatures written down in Chapter 6, Table 5. It is 70 %. Then the temperature ratio for exhaust air was calculated with equation (3) with values of designed flow rates:
- the supply air flow rate is 4,00 m^3/s;
- the exhaust air flow rate is 3,60 m^3/s;
-the volume flow ratio calculated with the equation 4 is 1,11;
The value of the temperature ratio for exhaust air which was calculated by the manufacturer is 78 %.

The calculated values of the temperature ratios of the research can't be compared with that manufacturer's data because manufacturer's value of the temperature ratio for the supply air was obtained when the supply and exhaust air flows were equal to each other. It is a requirement of the test. In our case, these flows never are equal and the supply air flow rate is more than the exhaust one. Furthermore, inlet air temperatures don't corresponds to the required temperatures of the test. There aren't also any data of the research when the volume flow ratio is 1,11. It was always more than this value.

8.DISCUSSION

As a result of the research answers on main questions of the thesis were obtained. In practice the annual heat recovery energy efficiency of the heat recovery unit for supply air is high and it is equal to 77,3 %. It is impossible to compare this value with standards because there isn't any information about what value the annual heat recovery efficiency of the heat recovery unit exactly should have. It isn't any data about acceptable temperature ratios of the heat recovery unit in European standards as well.

Furthermore, an attempt of comparing the obtained data with manufacturer's data for the heat recovery device was made. However, it was impossible because the manufacturer give us only information about the temperature ratios of the heat recovery unit which was obtained during test procedure according to EN 308. This procedure is performed in certain conditions which are described in Chapter 6. The conditions of obtaining data of the research didn't correspond to the conditions of the test, i.e. the supply and exhaust air flows weren't equal to each other, the volume ratio of the obtained data was always more than the designed value of the volume ratio which was given by the manufacturer and inlet air temperatures aren't equal to the required ones. But the calculated value of annual heat recovery energy efficiency of the heat recovery unit was compared with the annual heat recovery efficiency of the heat recovery unit for the standard year of the second climate zone. Mikkeli is located in this climate zone. The value is 74,4 %/15/. The difference is only 2,9 %. So, these values are very close to each other and have the same order. It means that the operation of the heat recovery unit was effective during the researched year.

The annual heat energy consumption of the air handling unit with the heat recovery unit is 35,1 MWh. Using the heat recovery unit leads to reduction of the heat energy for coil by 119,2 MWh. So, 6559 EUR were saved in heating costs. There aren't any recommendations of heat energy saving for the air handling unit because the heat recovery unit saves the heat energy effectively. But there is a problem with the electricity consumption of the air handling unit because the specific fan power of the air handling unit (SFP) is more than it is required in D3. It is important because the price of electricity is high. Costs of electricity of the air handling unit is 61,6 % of the total annual costs of the operation. It is recommended to decrease pressure losses in the ventilation system, for example, by increasing diameters of the ducts. These measures will lead to reduction of the fan power. Therefore, the electricity consumption of the air handling unit will lower.

Finally, there are some recommendations for owners of the building about the location of the measuring devices of the air handling unit. During the research it was found out that the measured exhaust air temperature after the heat recovery unit hasn't correct value due to wrong location of the measuring sensor. So, it is recommended to change the location of this device to get correct values. If calculations of the efficiency of the heat recovery unit are planned in the future, it will be recommended to install a device for measuring a supply air temperature (after the coil) in the air handling unit. This device will provide data of the temperature which can be recorded and used for calculation of the real heat energy consumption of the air handling unit.

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A PART OF THE DATA REPORT FOR JANUARY 2012

Mikkeli University of Applied Sciences

D-building

TK 43

DATA	Hour (UTC +2)	t_{outd}	t_{SHR}	t_{ex}	t_{EHR}	$q_{v,s}$	$q_{v,e}$	Electricity (for fans)	Heat energy (for coil)
		°C	°C	°C	°C	m³/s	m³/s	kWh	MWh
02.01.2012	00:01:00	-6,64	23,43	22,33	18,67	0,65	0,78	69158	82,12
02.01.2012	01:01:00	-6,17	23,48	22,36	18,68	0,64	0,77	69159	82,12
02.01.2012	02:01:00	-5,7	23,51	22,35	18,7	0,66	0,76	69159	82,12
02.01.2012	03:01:00	-5,64	23,47	22,35	18,75	0,65	0,74	69159	82,12
02.01.2012	04:01:00	-5,61	23,49	22,39	18,79	0,64	0,74	69159	82,12
02.01.2012	05:01:00	-5,17	23,56	22,39	18,81	0,65	0,76	69160	82,12
02.01.2012	06:01:00	-4,76	23,47	22,37	18,83	0,63	0,76	69160	82,12
02.01.2012	07:01:00	-5,54	22,7	22,4	17,57	0,84	0,89	69161	82,12
02.01.2012	08:01:00	-5,32	12,67	21,17	2,8	3,48	2,84	69168	82,14
02.01.2012	09:01:00	-4,98	12,69	21,09	2,79	3,46	2,85	69175	82,17
02.01.2012	10:01:00	-4,98	12,65	21,11	2,67	3,48	2,84	69183	82,2
02.01.2012	11:01:00	-4,98	12,56	21,12	2,49	3,47	2,85	69190	82,22
02.01.2012	12:01:00	-4,57	12,59	21,09	2,62	3,47	2,84	69198	82,25
02.01.2012	13:01:00	-4,14	12,74	21,03	2,97	3,44	2,84	69205	82,27
02.01.2012	14:01:00	-3,87	12,8	21,05	3,18	3,46	2,84	69212	82,3
02.01.2012	15:01:00	-3,87	12,73	20,93	3,12	3,47	2,84	69220	82,32
02.01.2012	16:01:00	-3,87	12,74	20,95	3,11	3,45	2,83	69227	82,35
02.01.2012	17:01:00	-3,87	12,7	20,86	3,13	3,46	2,84	69235	82,37
02.01.2012	18:01:00	-3,87	12,65	20,74	3,13	3,48	2,86	69242	82,4
02.01.2012	19:01:00	-3,87	12,58	20,67	3,09	3,48	2,84	69250	82,42
02.01.2012	20:01:00	-3,87	12,53	20,65	2,99	3,48	2,83	69257	82,45
02.01.2012	21:01:00	-3,87	27,28	20,64	10,54	0,82	0,94	69257	82,45
02.01.2012	22:01:01	-3,87	24,97	21,04	13,95	0,61	0,8	69258	82,45
02.01.2012	23:01:00	-3,87	23,61	21,22	15,72	0,56	0,79	69258	82,45
03.01.2012	00:01:00	-3,87	23,05	21,34	16,7	0,63	0,78	69258	82,45
03.01.2012	01:01:00	-3,22	22,77	21,44	17,37	0,58	0,77	69258	82,45
03.01.2012	02:01:00	-3,11	22,66	21,47	17,74	0,57	0,75	69259	82,45

03.01.2012	03:01:00	-3,11	22,65	21,54	17,93	0,61	0,74	69259	82,45
03.01.2012	04:01:00	-3,11	22,64	21,62	17,97	0,6	0,74	69259	82,45
03.01.2012	05:01:00	-2,8	22,71	21,7	17,92	0,58	0,77	69259	82,45
03.01.2012	06:01:00	-2,64	22,63	21,69	17,85	0,62	0,73	69259	82,45
03.01.2012	07:01:00	-2,65	21,93	21,72	16,83	0,82	0,84	69260	82,45
03.01.2012	08:01:00	-2,65	13,23	21	4,25	3,46	2,8	69268	82,47
03.01.2012	09:01:00	-2,4	13,22	20,94	4,21	3,46	2,83	69275	82,49
03.01.2012	10:01:00	-2,28	13,27	20,98	4,34	3,48	2,81	69283	82,52
03.01.2012	11:01:01	-2,05	13,4	21,08	4,51	3,48	2,82	69290	82,54
03.01.2012	12:01:01	-1,68	13,5	21,1	4,71	3,45	2,82	69298	82,56
03.01.2012	13:01:00	-1,08	13,55	21,03	4,92	3,47	2,83	69305	82,58
03.01.2012	14:01:01	-0,8	13,74	21,06	5,29	3,45	2,82	69313	82,6
03.01.2012	15:01:00	-0,28	13,92	21,08	5,7	3,48	2,82	69320	82,62
03.01.2012	16:01:00	-0,28	13,97	21,04	5,88	3,48	2,83	69328	82,64
03.01.2012	17:01:00	-0,28	13,98	20,99	6,01	3,46	2,83	69336	82,66
03.01.2012	18:01:00	-0,28	13,87	20,76	6,04	3,47	2,84	69343	82,68
03.01.2012	19:01:00	-0,04	13,84	20,66	6,08	3,49	2,83	69351	82,7
03.01.2012	20:01:00	0,08	13,83	20,6	6,03	3,48	2,83	69358	82,72
03.01.2012	21:01:00	-0,69	24,82	20,66	12,31	0,86	0,92	69359	82,72
03.01.2012	22:01:00	-0,98	24,34	21,03	15,08	0,61	0,75	69359	82,72
03.01.2012	23:01:00	-0,26	23,44	21,22	16,4	0,6	0,76	69359	82,72
04.01.2012	00:01:00	-0,4	23,11	21,27	17,02	0,61	0,77	69360	82,72
04.01.2012	01:01:00	-0,53	23,03	21,37	17,6	0,6	0,72	69360	82,72
04.01.2012	02:01:00	-0,34	22,95	21,43	17,91	0,62	0,74	69360	82,72
04.01.2012	03:01:00	-0,03	22,9	21,48	18,14	0,62	0,75	69360	82,72
04.01.2012	04:01:00	0,03	22,89	21,52	18,33	0,62	0,73	69360	82,72
04.01.2012	05:01:00	0,03	22,89	21,58	18,47	0,6	0,73	69361	82,72
04.01.2012	06:01:00	0,03	22,66	21,57	18,55	0,56	0,73	69361	82,72
04.01.2012	07:01:00	0,03	22,16	21,59	17,67	0,77	0,89	69362	82,72
04.01.2012	08:01:00	0,03	14,24	20,96	6,63	3,45	2,84	69369	82,74
04.01.2012	09:01:01	0,03	14,1	20,8	6,36	3,46	2,86	69377	82,76

APPENDIX 2 (1)

A PART OF THE SUMMARY TABLE OF CALCULATIONS PER HOUR FOR JANUARY

Mikkeli University of Applied Sciences
D-building
TK 43

Date	Hour (UTC +2)	Corrected t_{EHR}	η_s	η_e	$R_{HR}=\eta e/\eta s$	$R_{HR}=q_{v,s}/q_{v,e}$	$\eta_e=\eta_s \cdot R_{HR}$	Need of heat with HR Q_{HR}	Total need of heat Qtotal	Need of electricity Q_e	Energy for coil Q_{coil}	Heat recovery efficiency η_Q for heat
		°C	%	%	-	-	%	MWh	MWh	kWh	MWh	%
02.01.2012	00:01:00	17,17	0,0	0,0	0,00	0,00	0,0	0,000	0,000	0,0	0,00	0,0
02.01.2012	01:01:00	17,18	0,0	0,0	0,00	0,00	0,0	0,000	0,000	1,0	0,00	0,0
02.01.2012	02:01:00	17,2	0,0	0,0	0,00	0,00	0,0	0,000	0,000	0,0	0,00	0,0
02.01.2012	03:01:00	17,25	0,0	0,0	0,00	0,00	0,0	0,000	0,000	0,0	0,00	0,0
02.01.2012	04:01:00	17,29	0,0	0,0	0,00	0,00	0,0	0,000	0,000	0,0	0,00	0,0
02.01.2012	05:01:00	17,31	0,0	0,0	0,00	0,00	0,0	0,000	0,000	1,0	0,00	0,0
02.01.2012	06:01:00	17,33	0,0	0,0	0,00	0,00	0,0	0,000	0,000	0,0	0,00	0,0
02.01.2012	07:01:00	16,07	0,0	0,0	0,00	0,00	0,0	0,000	0,000	1,0	0,00	0,0
02.01.2012	08:01:00	1,3	67,9	75,0	1,10	1,23	83,2	0,075	0,093	7,0	0,02	80,6
02.01.2012	09:01:00	1,29	67,8	75,9	1,12	1,21	82,3	0,073	0,091	7,0	0,03	80,4
02.01.2012	10:01:00	1,17	67,6	76,4	1,13	1,23	82,8	0,074	0,092	8,0	0,03	80,2
02.01.2012	11:01:00	0,99	67,2	77,1	1,15	1,22	81,8	0,073	0,092	7,0	0,02	79,8
02.01.2012	12:01:00	1,12	66,9	77,8	1,16	1,22	81,7	0,071	0,090	8,0	0,03	79,6
02.01.2012	13:01:00	1,47	67,1	77,7	1,16	1,21	81,2	0,070	0,087	7,0	0,02	79,8
02.01.2012	14:01:00	1,68	66,9	77,7	1,16	1,22	81,5	0,069	0,087	7,0	0,03	79,9
02.01.2012	15:01:00	1,62	66,9	77,9	1,16	1,22	81,8	0,069	0,087	8,0	0,02	79,5
02.01.2012	16:01:00	1,61	66,9	77,9	1,16	1,22	81,6	0,069	0,086	7,0	0,03	79,6
02.01.2012	17:01:00	1,63	67,0	77,8	1,16	1,22	81,6	0,069	0,087	8,0	0,02	79,4
02.01.2012	18:01:00	1,63	67,1	77,7	1,16	1,22	81,7	0,069	0,087	7,0	0,03	79,2
02.01.2012	19:01:00	1,59	67,0	77,8	1,16	1,23	82,1	0,069	0,087	8,0	0,02	78,8
02.01.2012	20:01:00	1,49	66,9	78,1	1,17	1,23	82,2	0,068	0,087	7,0	0,03	78,6
02.01.2012	21:01:00	9,04	0,0	0,0	0,00	0,00	0,0	0,000	0,000	0,0	0,00	0,0
02.01.2012	22:01:01	12,45	0,0	0,0	0,00	0,00	0,0	0,000	0,000	1,0	0,00	0,0
02.01.2012	23:01:00	14,22	0,0	0,0	0,00	0,00	0,0	0,000	0,000	0,0	0,00	0,0
03.01.2012	00:01:00	15,2	0,0	0,0	0,00	0,00	0,0	0,000	0,000	0,0	0,00	0,0
03.01.2012	01:01:00	15,87	0,0	0,0	0,00	0,00	0,0	0,000	0,000	0,0	0,00	0,0
03.01.2012	02:01:00	16,24	0,0	0,0	0,00	0,00	0,0	0,000	0,000	1,0	0,00	0,0

APPENDIX 2 (1)

A PART OF THE SUMMARY TABLE OF CALCULATIONS PER HOUR FOR JANUARY 2012

03.01.2012	03:01:00	16,43	0,0	0,0	0,00	0,00	0,0	0,000	0,000	0,0	0,00	0,0
03.01.2012	04:01:00	16,47	0,0	0,0	0,00	0,00	0,0	0,000	0,000	0,0	0,00	0,0
03.01.2012	05:01:00	16,42	0,0	0,0	0,00	0,00	0,0	0,000	0,000	0,0	0,00	0,0
03.01.2012	06:01:00	16,35	0,0	0,0	0,00	0,00	0,0	0,000	0,000	0,0	0,00	0,0
03.01.2012	07:01:00	15,33	0,0	0,0	0,00	0,00	0,0	0,000	0,000	1,0	0,00	0,0
03.01.2012	08:01:00	2,75	67,1	77,2	1,15	1,24	83,0	0,066	0,082	8,0	0,02	80,8
03.01.2012	09:01:00	2,71	66,9	78,1	1,17	1,22	81,8	0,065	0,081	7,0	0,02	80,5
03.01.2012	10:01:00	2,84	66,9	78,0	1,17	1,24	82,8	0,065	0,081	8,0	0,03	80,7
03.01.2012	11:01:01	3,01	66,8	78,1	1,17	1,23	82,4	0,065	0,080	7,0	0,02	81,1
03.01.2012	12:01:01	3,21	66,6	78,5	1,18	1,22	81,5	0,063	0,077	8,0	0,02	81,3
03.01.2012	13:01:00	3,42	66,2	79,6	1,20	1,23	81,1	0,061	0,075	7,0	0,02	80,9
03.01.2012	14:01:01	3,79	66,5	79,0	1,19	1,22	81,4	0,060	0,074	8,0	0,02	81,7
03.01.2012	15:01:00	4,2	66,5	79,0	1,19	1,23	82,0	0,059	0,072	7,0	0,02	82,2
03.01.2012	16:01:00	4,38	66,8	78,1	1,17	1,23	82,2	0,060	0,072	8,0	0,02	82,5
03.01.2012	17:01:00	4,51	67,0	77,5	1,16	1,22	82,0	0,059	0,072	8,0	0,02	82,5
03.01.2012	18:01:00	4,54	67,3	77,1	1,15	1,22	82,2	0,059	0,072	7,0	0,02	81,9
03.01.2012	19:01:00	4,58	67,1	77,7	1,16	1,23	82,7	0,058	0,071	8,0	0,02	81,5
03.01.2012	20:01:00	4,53	67,0	78,3	1,17	1,23	82,4	0,057	0,071	7,0	0,02	81,3
03.01.2012	21:01:00	10,81	0,0	0,0	0,00	0,00	0,0	0,000	0,000	1,0	0,00	0,0
03.01.2012	22:01:00	13,58	0,0	0,0	0,00	0,00	0,0	0,000	0,000	0,0	0,00	0,0
03.01.2012	23:01:00	14,9	0,0	0,0	0,00	0,00	0,0	0,000	0,000	0,0	0,00	0,0
04.01.2012	00:01:00	15,52	0,0	0,0	0,00	0,00	0,0	0,000	0,000	1,0	0,00	0,0
04.01.2012	01:01:00	16,1	0,0	0,0	0,00	0,00	0,0	0,000	0,000	0,0	0,00	0,0
04.01.2012	02:01:00	16,41	0,0	0,0	0,00	0,00	0,0	0,000	0,000	0,0	0,00	0,0
04.01.2012	03:01:00	16,64	0,0	0,0	0,00	0,00	0,0	0,000	0,000	0,0	0,00	0,0
04.01.2012	04:01:00	16,83	0,0	0,0	0,00	0,00	0,0	0,000	0,000	0,0	0,00	0,0
04.01.2012	05:01:00	16,97	0,0	0,0	0,00	0,00	0,0	0,000	0,000	1,0	0,00	0,0
04.01.2012	06:01:00	17,05	0,0	0,0	0,00	0,00	0,0	0,000	0,000	0,0	0,00	0,0
04.01.2012	07:01:00	16,17	0,0	0,0	0,00	0,00	0,0	0,000	0,000	1,0	0,00	0,0
04.01.2012	08:01:00	5,13	67,9	75,6	1,11	1,21	82,5	0,059	0,070	7,0	0,02	83,7
04.01.2012	09:01:01	4,86	67,7	76,7	1,13	1,21	82,0	0,058	0,070	8,0	0,02	82,9

APPENDIX 3

THE SUMMARY TABLE OF CALCULATED VALUES FOR ALL MONTHS OF THE RESEARCH

Year	2010				2011	2011											2012	Units
Values	Sept. 9	Oct. 10	Nov. 11	Dec. 12	Jan. 1	Feb. 2	March 3	Apr. 4	May 5	June 6	July 7	Aug. 8	Sept. 9	Oct. 10	Nov. 11	Dec. 12	Jan. 1	
$\eta_{s,m}$	56,0	50,7	63,6	65,6	67,5	67,9	66,8	64,2	65,7	0,0	0,0	0,0	45,2	65,8	66,2	66,6	66,3	%
$\eta_{e,m}$	0,0	0,0	0,0	0,0	0,0	77,8	80,9	84,0	87,1	0,0	0,0	0,0	57,4	79,1	78,3	78,6	78,2	%
$\eta'_{e,m}$ (with air flows)	84,8	68,7	84,0	83,4	81,8	80,3	80,0	77,5	80,0	0,0	0,0	0,0	54,7	79,7	80,3	80,8	80,1	%
$\Delta_{\eta e,m}$	100,0	100,0	100,0	100,0	100,0	3,2	-1,1	-8,3	-8,8	0,0	0,0	0,0	-5,0	0,8	2,5	2,6	2,3	%
ΣQ_{coil}	2,8	10,4	13,5	16,7	12,1	9,1	3,9	0,7	0,9	0,8	0,000	0,020	0,1	1,7	3,6	5,1	9,3	MWh
$\Sigma W_{electricity}$	3,5	3,6	2,5	2,3	3,2	2,0	2,3	2,1	2,1	2,5	2,8	3,1	3,2	2,9	3,0	2,5	2,4	MWh
ΣQ_{total}	31,9	25,8	44,9	26,1	36,5	29,9	21,6	8,2	2,2	0,4	0,1	0,0	0,8	10,9	19,6	20,1	30,0	MWh
ΣQ_{HR}	27,0	19,4	34,4	19,8	29,5	23,7	18,2	7,4	2,0	0,0	0,0	0,0	0,7	9,8	17,0	16,7	23,6	MWh
ΣQ_{HR}+ΣQcoil	29,8	29,8	47,9	36,5	41,6	32,8	22,1	8,1	2,8	0,8	0,0	0,0	0,8	11,5	20,6	21,8	32,9	MWh
$\Delta_{\Sigma Qtotal}$	-7,3	13,2	6,4	28,4	12,1	8,6	2,6	-1,8	22,5	42,0	0,0	6,8	9,3	5,4	4,8	7,6	8,8	%
$\eta_{Q,m(with\ energy)}$	84,5	75,2	76,7	75,7	80,8	79,0	84,5	90,1	89,1	0	0	0	93,0	90,3	86,8	83,1	78,8	%
$\eta_{Q,m(average)}$	82,5	69,9	77,9	75,6	81,2	79,8	85,0	90,4	90,2	0	0	0	81,4	90,9	87,6	83,3	79,3	%
Time which was ignored	223	83	67	483	324	406	432	546	685	701	736	734	693	505	401	449	448	h
Total time	718	744	720	744	744	672	744	719	744	720	744	744	720	720	720	744	744	h
Operation time of HRU	495	661	653	261	420	266	312	173	59	19	8	10	27	215	319	295	296	h
Operation time of AHU	718	695	655	261	472	280	331	304	325	307	294	322	342	325	344	300	302	h
Mean q_s per month	6,93	2,85	2,74	2,98	3,22	3,17	3,14	3,09	3,01	3,38	3,73	3,71	3,68	3,67	3,59	3,49	3,48	m^3/s
Mean q_e per month	4,63	2,11	2,08	2,35	2,66	2,68	2,63	2,56	2,46	2,80	3,03	3,05	3,06	3,01	2,96	2,86	2,83	m^3/s
Costs of DH with HR	154	570	743	918	663	501	216	37	48	42	0	2	7	92	196	279	509	EUR
Costs of operation(electricity)	352	360	247	230	320	197	233	211	214	252	284	305	321	287	295	246	243	EUR
Costs of DH without HR	1 637	1 638	2 638	2 006	2 288	1 803	1 219	446	156	42	0	2	46	633	1 132	1 199	1 810	EUR