Norwegian University of
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High efficiency heat recovery from ventilation air

Effektiv varmegjenvinning fra ventilasjonsluft

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MASTER THESIS

for

Student Ranja Therese Nørgård-Hansen

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High efficiency heat recovery from ventilation air
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Background and objective

Standards for energy use in buildings are becoming more restrictive. To cope with future demands for energy use in buildings, heat recovery from ventilation air has become almost compulsory and it has to be very efficient. Some field measurements have shown that there are certain deviations between the design efficiency and the measured efficiencies in practice. The objective of this work is to measure efficiencies in several selected buildings and to study what could be done to improve current solutions or to develop feasible options.

The master thesis is a continuation of the work started during the candidate's specialization project.

The project is connected to a SINTEF Building and Infrastructure research project funded by Husbanken. The study will be limited to residential buildings and kindergartens.

The following tasks are to be considered for the specialization project:

1. Update and deepen the literature study related to heat recovery from ventilation air in general and on effectiveness specially (both thermal and moisture recovery). Also measuring techniques are to be analysed.
2. Do a few more of the preliminary measurement done in the specialization project and evaluate the test methods
3. Discuss out of the literature how to calculate effectiveness so that it is more representative. Consider the condensation challenge.
4. Plan how to measure the effectiveness in field
5. Perform preliminary field tests and evaluate the results. Discuss possible improvements in measurement technique
6. Perform tests in several buildings.
7. Evaluate results against TEK requirements, suppliers' data sheet and planner's design data.
8. Suggest improvements

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

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

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

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

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

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

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

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

Department of Energy and Process Engineering, 15. January 2017

Hans Martin Mathisen

Hans Martin Mathisen
Academic Supervisor

Research Advisor: Maria Justo Alonso

Abstract

The objective of this thesis has been to conduct both theoretical and practical work to investigate how to the calculations of effectiveness can be more representative.

A literature study regarding field test of heat recovery in ventilation air has been conducted. The main focus has been on the effectiveness concerning both thermal and moisture recovery. Measuring techniques for mass flow have been studied and presented.

Preliminary field tests have been conducted in Øya Kindergarten in Trondheim. An anemometer was used to measure both temperature and velocity. The results from measurements are presented and discussed. The effectiveness has also been calculated with the measured mass flow and temperature values with a result of >1. The results have been compared to heat balance ratio for validation and the uncertainty of the effectiveness of have been estimated. The calculated effectiveness that is based on measurements does no correlate with heat transfer theory or design data. The heat balance ratio confirms that the result is invalid.

The supply mass flow was estimated with the exhaust mass flow and the ratio between fan speed in the air handling unit (AHU). This results in an effectiveness of 81,5%, which corresponds to both theory and design data. The effectiveness of 81,5% for the heat exchanger is within the requirements in NS 3701 (2012), but slightly lower than the nominal efficiency given by the supplier. This result is based on assumptions and should not be credited as valid.

From the discussion of literature and results, velocity measurements can be a reliable method if it is measured in laminar flow. By establishing knowledge of the flow pattern in the ducts, the measurements can be conducted in the most suitable sites for determining mass flow. It comes forth that the velocities should be measured in the inlet ducts since the airflow is assumed to be turbulent in the outlet ducts, based on the results.

For further work, it is proposed to conduct measurements at several sites and use more than one unit for measuring and include latent efficiency.

Sammendrag

Formålet med denne oppgaven har vært å gjennomføre både teoretisk og praktisk arbeid for å undersøke hvordan effektivitetsberegninger kan være mer representativ.

En litteraturstudie om felttesting av varmegjenvinning i ventilasjonsluft er utført. Hovedfokuset har vært på effektivitet med hensyn til både termisk og fuktig gjenvinning. Målingsteknikker for massestrøm er studert og presentert.

Felttester er gjennomført i Øya barnehage i Trondheim. Et anemometer ble brukt til å måle både temperatur og hastighet. Resultatene fra målingene er presentert og diskutert. Effektiviteten er også beregnet med målte massestrøm- og temperaturverdier med et resultat på >1. Resultatene er sammenlignet med varmebalanseforholdet for validering og usikkerheten til effektiviteten er estimert. Den beregnede effektiviteten som er basert på målinger samsvarer ikke med varmeoverføringsteori eller designdata. Varmebalansforholdet bekrefter at resultatet er ugyldig.

Massestrømmen til tilførselsluft ble estimert basert på massestrømmen til avkastluft og forholdet mellom viftehastighetene til viftene i ventilasjonsaggregatet. Dette resulterer i en effektivitet på 81,5%, som samsvarer med både teori og design. Effektiviteten på 81,5% for varmeveksleren er innenfor kravene i NS 3701 (2012), men litt lavere enn den nominelle effektiviteten gitt av leverandøren. Dette resultatet er basert på forutsetninger og bør ikke krediteres som gyldig.

Fra diskusjon av litteratur og resultater kan hastighetsmålinger være en pålitelig metode hvis den måles i laminær strømning. Ved å tilegne seg kunnskap om strømningsmønsteret i kanalene, kan målingene utføres på de mest egnede stedene for å bestemme massestrømmen. Det fremgår at hastighetene skal måles i innløpskanalene, da luftstrømmen antas å være turbulent i utløpskanalene, basert på resultatene.

For videre arbeid foreslås det å gjennomføre målinger på flere steder og bruke mer enn en enhet til måling og inkludere latent effektivitet.

Preface

This thesis is part of the Master of Science for the Department of Energy and Process Engineering at the Norwegian University of Science and Technology (NTNU) with specialization in Energy and Indoor environment. The thesis is connected to the SINTEF Byggforsk and Infrastructure research project funded by Husbanken.

The thesis is a study of heat recovery in ventilation units installed in the field. The objective of this thesis is to compare results from field work with theory within the field and other specifications. The thesis addresses engineers, students and other professionals with knowledge regarding this discipline.

I would like to thank my supervisors, Maria Justo Alonso, and professor Hans Martin Mathisen for advises and inputs. I would also like to thank Peng Liu for answering all of my questions. At last, I would like to thank the employees at Øya Kindergarten, Synne Kathinka Bertelsen and Trondheim municipality for helping me with the measurements.

I also want to thank my friend Sigurd Eivindson Løkse for his guidance, my fellow students Silje Marie Smitt for always sharing a bottle of wine after a long day and Else Høeg Sundfør for her valuable input.

Dedicated to the memory of by brother Lars-Erik Andreassen. This one is for you.

Ranja Therese Nørgård-Hansen Trondheim, 10th of June 2017

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Abbreviations

AHU	Air handling unit
CAV	Constant air volume
COP	Coefficient of performance
Exhaust air	Exhaust air from the building, warm side of heat exchanger
HVAC	Heating, ventilation and air-conditioning
IAQ	Indoor air quality
NTNU	Norges teknisk-vitenskapelige universitet (Norwegian University of Science and Technology)
Outdoor air	Air from the surrounding environment
SFP	Specific fan power [kW/ (m^3/s)]. Power used to move 1 m^3 air per second through the ventilation system.
Supply air	Supply air to the building, cold side of heat exchanger
SINTEF	Stiftelsen for industriell og teknisk forskning (The foundation for scientific and industrial research)
VAV	Variable air volume

Nomenclature

A	Cross –sectional area [m^2]
A_s	Area of surface
C_c	Flow-stream capacity rate of cold-side fluid
C_h	Flow-stream capacity rate of hot-side fluid
C^*	Ratio of the minimum to maximum heat capacity rate of the air streams
C_r^*	Matrix heat (or moisture) capacity ratio on the supply or exhaust side
c_p	Specific heat capacity
ε	Effectiveness [-]
ε_{cf}	Effectiveness for a counterflow direct type
f	General function
h	Heat transfer coefficient
k	Thermal conductivity [W/mK]
L	Characteristic length
$\dot{m}$	Mass flow rate of dry air
n	Number of measurements

NTU	Number of transfer units on the supply or exhaust side
NTU_o	Overall number of transfer units
q	Heat Flow [W]
$\dot{Q}$	The heat flux between two media [W]
q_v	Gross amount of circulated air through the fan [m^3/s]
q''_x	Heat flux per unit area [W/m^2]
Re	Reynolds number [-]
s	Standard deviation
T	Bulk temperature
U	Heat transfer coefficient [W/(m^2K)]
U_R	Resulting uncertainty
U_s	Instrument accuracy
U_T	Uncertainty in mean value
V	Velocity [m/s]
ν	Kinematic viscosity [m^2/s]
x	Individual measure value
$\bar{x}$	Average value for a set of measurements
ΔT_{LM}	Logarithmic mean temperature difference [°C]
$\sum P$	Sum of electrical power used by fans in the ventilation system [kW]
Σx	Sum of measurements

Subscripts

ave	average
c	Cold side
e	Exhaust side
h	Hot side
i	Inlet flow
min	Minimum
max	Maximum
o	Outlet
s	Supply side
t	total

1 Introduction

People spend on an average over 90% of their time indoors. This leads to the high demand of fresh air to provide good indoor air quality (IAQ) to avoid sick building syndrome. As the buildings are becoming more airtight due to regulations, natural ventilation in buildings will not provide enough fresh air to keep the IAQ good enough. Thus mechanical ventilation systems have to be applied in buildings [Novakovic et al., 2007].

About 40% of the total energy consumption in developed countries is due to buildings. The amount used for heating, ventilation, and air-conditioning (HVAC) represents half of the consumption. Ventilation losses may represent 50% or higher of the total energy loss in high-level thermal insulated buildings [Nizovtsev, 2016].

TEK10 states that the efficiency in heat recovery from ventilation systems should be equal or higher than 80% in non-residential buildings to not exceed the demands for energy efficiency [Direktoratet for byggkvalitet, 2016] [NS 3701,2012].

The performance of a heat exchanger is determined by laboratory tests. Only the heat exchanger itself is installed in a casing with only the necessary duct connecting elements. In some cases, the fans and pumps are included. None of the additional components in the HVAC system is included. Thus the performance is determined with close to steady-state conditions, which will not appear in installed units in the field [NS-EN 308, 1997].

1.1 Background

The tests for determining the efficiency of a heat exchanger are conducted in close to steady-state conditions in a laboratory. There will not be steady-state conditions in units installed in the field. Changes in temperatures due to influence from the surroundings and variations in the airflow will affect the efficiency of the heat exchanger.

Rotary heat exchangers can transfer both sensible and latent heat, dependent of the heat transfer media [Mull 2004]. These heat exchangers are compact, mostly self-cleaning and frost-resistant. By adjusting the revolutions per minute, the efficiency can be regulated. It is possible to achieve efficiency from 70% to 80% for heat transfer with the recommended airflow. A typical diameter for air-ventilating regenerators is 0.25m to 3m, and a rotational speed of 10 rpm can be achieved. [Shah and Sekulic, 2003]

Therefore it is of interest to investigate whether real heat recovery in installed air handling units is within the requirements from regulations and correlate with the efficiency designed by the supplier of the ventilation system.

1.2 Objective

The objective of this thesis consists of several tasks. A literature study related to heat recovery from ventilation air and effectiveness shall be conducted. It is of particular concern to investigate both sensible and latent heat recovery, as well as to analyze different measuring techniques for mass flow. Based on the literature review a discussion regarding how to make calculations more representative shall be conducted, where the condensation challenge is considered. Planning of conduction of measurement shall be performed. Test in the field shall be conducted, and the result should be discussed and evaluated. The measuring technique shall also be discussed, and possible improvements in measuring technique should also be presented. It is desired to conduct measurements in several buildings to compare results from different heat recovery systems. The obtained result shall be compared against requirements from TEK 10 and design data from the supplier.

1.3 Limitations

Access to equipment and the system has been an obstacle. The premises for borrowing some of the necessary tools from the lab at NTNU were limited, therefore equipment had to obtained elsewhere. They had to be sent from Northern Norway, which took a week. The measuring device was suddenly borrowed by a fellow student without an agreement and was suddenly delivered back without notice. This caused the measurement to be further postponed.

Due to a series of disturbing incidents, reduced opening hours in the kindergarten, lack of staff with authorization to give access to the technical room, borrowed equipment, the number of measurements conducted are not as many as desired. Also had a hard time finding literature related to this issue. Even though the work with measurements started in March, the testing was not completed until May. A comparison of methods and equipment for measuring temperature will not be carried out in this thesis, since measuring mass flow is pointed out as the greatest challenge in the project thesis, but it is of interest for making the calculations more representative.

2 Theoretical Background of Heat Recovery

The technical regulations from 2010 (TEK10) states with §13-3 that all buildings shall have a ventilation system that adjusts to the pollution- and moisture load in the rooms [Direktoratet for byggkvalitet. 2016]. The indoor air quality (IAQ) should be satisfactory regarding odors and pollutions. A satisfying IAQ is achieved by replacing the polluted air with fresh air. In Norway, with its cold climate, it will cause heat loss. Thus the building regulations require heat recovery from ventilation air with a minimum efficiency [Novakovic et al., 2007].

For non-residential buildings, the requirement for minimum temperature efficiency is 70%, while for non-residential buildings of passive house standard it is 80% [NS 3701,2012]. For heat recovery in ventilation systems, plate heat exchangers and heat wheels are the most common in Norway.

Part of the presented theory is taken from the project thesis completed during the fall of 2016.

2.1 Heat transfer theory

To understand how a heat exchanger works, it is important to comprehend the basic heat transfer theory. Heat transfer is a process where thermal energy is in transit due to temperature differences. There are three different heat transfer modes, convection, conduction, and radiation.

2.1.1 Convection

Heat transfer between a fixed solid and a moving fluid or gas at different temperatures is referred to as convection. Convection comprises two mechanisms classified by the nature of the flow; natural and forced convection. In natural convection, the density difference induces the flow causing small air movements. The heat transfer rate increases with greater air speed, thus forced convection is desired for large heat transfer processes. In forced convection, the flow is caused by external means such as a fan or a pump. The Reynolds number determines whether the airflow is laminar or turbulent, and is given by:

$$Re_L = \frac{VL}{\nu} \qquad (2.1.1)$$

2.1.2 Conduction

Heat transfer across a medium is referred to as conduction. The process takes place at the molecular level. Energy is transferred in substance from the more energetic particles to the less energetic. Energy transfer is caused by interactions between the particles. The transfer is in the direction of decreasing temperature. For heat conduction in a one-dimensional case, the rate equation is expressed by:

$$q_x'' = -k\frac{dT}{dx} \qquad (2.1.2)$$

The heat is transferred in the direction of the decreasing temperature, thus the minus sign in equation 2.2. During steady-state conditions, the heat flux is given by

$$q_x'' = -k\frac{T_2 - T_1}{L} \qquad (2.1.3)$$

2.1.3 Radiation

Thermal radiation does not require the presence of a material medium, in contrast to convection and conduction. Radiation transfer occurs most efficiently in a vacuum since energy is transferred from surface to surface, in other words, the amount of heat transfer due to radiation in air-to-air heat exchangers is of limited extent.

2.2 Thermodynamic theory

In this chapter, the thermal design theory relevant for this thesis is presented. Efficiency theory and theory for thermal performance specially for recuperators and regenerators are given.

2.2.1 General efficiency calculations

More efficient use of energy is the most effective way of reducing the energy demand. Thus it is of interest to measure the thermal efficiency, effectiveness, and energy efficiency of a heat exchanger.

Thermal Efficiency

The relation between the supply flow and exhaust flow influences thermal efficiency in a heat exchanger. Thermal efficiency is defined as the gained temperature divided by the maximum temperature lift possibly. For heat/ energy wheel the sensible effectiveness is expressed as

$$\varepsilon_s = \frac{actual\ heat\ transfer}{maximum\ possible\ heat\ transfer} = \frac{\dot{m}_s(T_i - T_{O)}|s}{\dot{m}_{min}(T_{s,i} - T_{e,i})} \qquad (2.2.1)$$

Latent effectiveness is given as

$$\varepsilon_l = \frac{actual\ moisture\ transfer}{maximum\ possible\ moisture\ transfer} = \frac{\dot{m}_s(W_i - W_O)|s}{\dot{m}_{min}(W_{s,i} - W_{e,i})} \qquad (2.2.1)$$

Effectiveness

Effectiveness describes how much energy has been recovered. The effectiveness NTU-method can be used to determine the heat transfer effectiveness of the heat exchanger if the inlet temperatures are known.

Heat transfer effectiveness is defined as Effectiveness= Actual heat transfer/ Maximum heat transfer: $\varepsilon = \dot{Q} / \dot{Q}_{max}$.

Actual heat transfer can be found by calculating either energy lost by the hot fluid or energy gained by the cold fluid:

$$\dot{Q} = (\dot{m}c_p)_h(T_{h,in} - T_{h,out}) = (\dot{m}c_p)_c(T_{c,out} - T_{c,in}) \qquad (2.2.3)$$

The expression for maximum possible heat transfer is given by:

$$\dot{Q}_{max} = (\dot{m}c_p)_{min}(T_{h,in} - T_{c,in}) \qquad (2.2.4)$$

The actual heat transfer rate is expressed as:

$$\dot{Q} = \varepsilon(\dot{m}c_p)_{min}(T_{h,in} - T_{c,in}) \qquad (2.2.5)$$

Thus the average efficiency is

$$\varepsilon_{ave} = \frac{\dot{m}_s(T_{s,o} - T_{s,i}) + \dot{m}_e(T_{e,i} - T_{e,o})}{2 \cdot \dot{m}_{min}(T_{e,i} - T_{s,i})} \qquad (2.2.6)$$

Global efficiency

Global efficiency of a heat recovery system equals the effectiveness if there is no exfiltration or neither external nor extract-to-supply recirculation. If not, the global efficiency is lower than the nominal efficiency of the heat recovery unit.

Energy Efficiency

How much energy that is recovered by the heat exchanger concerning the total heating demand is referred to as the energy efficiency. The total energy transfer effectiveness for energy wheels is defined by

$$\varepsilon_t = \frac{actual\ energy\ transfer}{maximum\ possible\ energy\ transfer} = \frac{\dot{m}_s(h_i - h_o)|s}{\dot{m}_{min}(h_{s,i} - h_{e,i})} \qquad (2.2.7)$$

2.2.2 Recuperators

In a heat exchanger flow arrangement there are several independent and dependent variables causing design problems for a heat exchanger. Therefore dimensionless groups are formulated based on six independent and one or more dependent variable.

$$\underbrace{T_{h,o}, T_{c,o}\ or\ q}_{\substack{dependent\\ variables}} = f\ (\underbrace{\underbrace{T_{h,i},\ T_{c,i},\ C_h,\ C_c}_{\substack{operating\ condition\\ variables}}\ \underbrace{U, A, flow\ arrangement}_{\substack{parameters\ under\\ desginers\ controll}}\ )}_{independent\ variables\ and\ parameters} \qquad (2.2.8)$$

The dimensionless groups simplify the analysis, and the results can be presented in a more straightforwardly way. One of these methods is the $\varepsilon - NTU$ method as shown in Eq. (2.2.9)

$$\varepsilon = f\left(NTU, \frac{C_{min}}{C_{max}}, flow\ arrangement\right) \qquad (2.2.9)$$

$$NTU = \frac{UA_s}{C_{min}} \qquad (2.2.10)$$

For a counter flow heat exchanger with plain surfaces, the effectiveness can be expressed by

$$\varepsilon = \frac{1 - exp\left[-NTU(1 - \frac{C_{min}}{C_{max}})\right]}{1 - \frac{C_{min}}{C_{max}} exp\left[-NTU(1 - \frac{C_{min}}{C_{max}})\right]} \qquad (2.2.11)$$

Eq. (2.2.11) representing the generalized solution for the effectiveness where C_{min}=C_c, C^*=C_c/ C_h and NTU=UA/ C_{min} = UA/C_c for a single-pass counter-flow exchanger. [Shah and Sekulic, 2003]

2.2.3 Regenerators

The ε-NTU$_o$ and $\Lambda - \Pi$ methods are used for the regenerator thermal performance analysis, respectively for rotary and fixed-matrix regenerator. The aim of the method is to decide the efficiency for given conditions[1]. For a counter-flow regenerator, the following solutions for determining the regenerator effectiveness apply. The assumptions made are that there is no heat loss to the environment, constant heat transmission coefficient through the heat exchanger, constant fluid flow and simplified models for the flow arrangement.

The ε-NTU$_o$ method presented by Shah and Sekulic. (2003), was developed by Coppage and London. Their model is expressed by

$$\varepsilon = f(NTU_o, C^*, C_r^*, (hA)^*) \qquad (2.2.12)$$

Where NTU$_o$ is the modified number of transfer units. U$_o$ is termed as a modified overall heat transfer coefficient since there is not direct heat transfer between the fluids in a regenerator. Then

$$\frac{1}{U_oA} = \frac{1}{(hA)_h} + \frac{1}{(hA)_c} \text{ and } NTU_o = \frac{U_oA}{C_{min}} \qquad (2.2.13)$$

Since $C^* > 0.8$ in most regenerators, (hA)* can be eliminated from the Eq. (2.2.12). The influence of (hA)* on the regenerator effectiveness for the range $0.25 \leq (hA)^* < 4$ is of e negligible size. This was shown by Lambertson, and presented by Shah and Sekulic (2003).

Kays and London present a simple empirical formulation which shows the influence of C_r^* on ε, for $\varepsilon \leq 90\%$ as

$$\varepsilon = \varepsilon_{cf}\left[1 - \frac{1}{9(C_r^*)^{1.93}}\right] \qquad (2.2.14)$$

[1] Other conditions than the designed conditions

Where

$$\varepsilon_{cf} = \frac{1-\exp\left[-NTU_o(1-C^*)\right]}{1-C^*\exp\left[-NTU_o(1-C^*)\right]} \xrightarrow[C^*=1]{} \frac{NTU_o}{1+NTU_o} \qquad (2.2.15)$$

The results from Equation 2.3.7 are within 1% the tabular results of Lambertson for $C^* = 1$ for the ranges: 2 < NTU_o < 14 for $C_r^* \leq 1.5$, $NTU_o \leq 20$ for $C_r^* \leq 2$, and a complete range of NTU_o for $C_r^* \geq 5$. For decreasing values of C^*, the approximation causes an increasing error with lower values of C_r^* [Shah and Sekulic, 2003].

2.3 Heat Exchangers

The main objective of heat exchangers is to transfer heat from a hot fluid to a cold fluid, without mixing them together. They are used in HVAC technology as well as in different applications such as power engineering chemical industries, petroleum refineries, and food industries.

Two different forms, namely sensible and latent heat define thermal energy. With sensible heat transfer, heat is transferred due to a difference in temperature. In latent heat transfer, heat is absorbed or released caused by a phase change.

Losses in heat exchangers are due to exchange of heat across a finite temperature difference, fluid friction, material and manufacturing of the heat exchanger and heat exchange with the environment. The surface of the heat exchanger is insulated to reduce thermal losses.

Installation of a heat exchanger in a ventilation system demands the supply and exhaust air to be forced. An airtight building is required; otherwise, the obtained recovery will be compromised [Sintef, 2007].

The ratio of the enthalpy flow delivered to the supply air over the enthalpy flow in the exhaust air is the energy efficiency of the heat recovery system.

The efficiency of heat recovery systems is dependent on two main factors, the contact surface area between the heat exchanger and the fluid, and the temperature difference between the two fluid streams.

A higher contact area leads to higher efficiency of heat recovery. Fins, plates, and coils are used to increase the surface area, causing an increase in pressure drop across the heat exchanger. An additional fan or pump power can be used to overcome the pressure drop. It is important to clean the surfaces regularly to maintain the heat exchangers performance. Accumulation of particles, fouling, on the surfaces will increase the roughness and increase the pressure drop. Fouling also causes reduced heat transfer and performance.

Higher temperature differences between the two fluids cause more efficient heat recovery. Thus cold climates are more beneficial for heat recovery systems than mild climates. [Krarti, 2011]

There are two general categories of heat exchangers, regenerative and recuperative which are further presented and described.

2.3.1 Recuperative heat exchangers

In recuperative heat exchangers sensible heat is transferred from the hot exhaust air, through the separating plates, to the cold supply air. Cross-plate and run-around coils are examples of recuperative heat exchangers. Since they only can transfer sensible heat, the efficiency is lower than of regenerative heat exchangers [Shah and Sekulic, 2003].

Plate and tube heat exchangers

In a plate exchanger, several plates form a wall between the supply air and the exhaust air. The heat transfers through the walls by conduction. The tube exchanger works according to the same principle, where the supply air usually moves through the pipes, while the exhaust flows around it. Ice can form if the surface temperature falls below zero [Novakovic et al., 2007].

Run- around coil

The heat exchanger, an indirect exchanger, consists of a cooling and heating coil. The cooling coil is placed in the exhaust duct, whereas the heating coil is put in the supply duct. The heat exchanges through heat carriers like gas, vapor or liquid media. The heat exchanger is utilized when the ducts for exhaust and supply can not be side-by-side [Novakovic et al., 2007].

Heat pipe

Heat pipe heat exchangers are based on the same principle as the run-around heat exchanger. A refrigerant evaporates when absorbing heat from the exhaust air, moving towards the supply air side due to density. When the evaporated refrigerant transfers heat to the supply air, it condenses and flows back to the exhaust air side. Bypassing of the supply air can be used to regulate the temperature efficiency and frost protection [Novakovic et al., 2007].

2.3.2 Regenerative heat exchangers

In regenerative heat exchangers, both sensible heat and latent heat can transfer between the supply air and exhaust air. Heat from the hot fluid is intermittently stored in a thermal storage medium and then transferred to the cold fluid. Regenerative heat exchangers can achieve an efficiency of 80-90 % for heat recovery. Therefore in ventilation systems, regenerative heat exchangers, such as rotary heat exchangers, are commonly used [Shah and Sekulic, 2003].

Heat Wheels

Rotating heat exchangers that transfer sensible heat are referred to as heat wheels. Heat is transferred between two counter-flow air streams with different temperature. They are common in gas turbine plants and electrical power generating stations. Thermal energy is recovered from the exhaust gases and used to preheat inlet combustion air, increasing the thermal efficiency of the overall plant. Heat wheels are found in thermal power plants as well as in HVAC systems [Simonson, 2007].

Energy Wheels

Both heat and moisture can be transferred between two air streams in energy wheels. Energy wheels use minimal of external energy input for heat and moisture transfer. Thus the market share for air-to-air/ heat exchangers for energy wheels has increased significantly since indoor air quality conditions and the humidity level inside affects the thermal climate. The operation of energy wheels is similar to the operation of heat wheels, but the energy wheel is coated with a desiccant for storage of moisture. Their main application is in HVAC systems [Simonson, 2007].

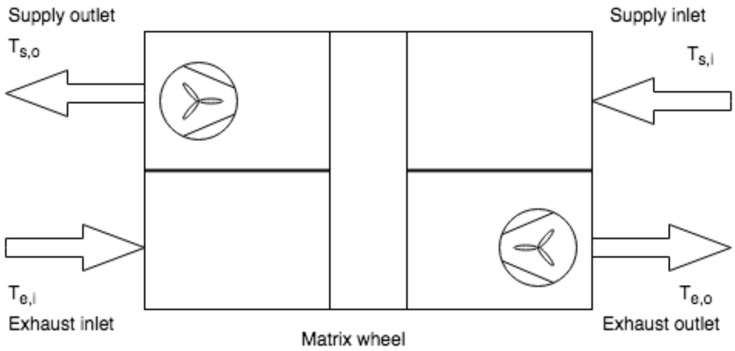

Figure 2.1 Simple illustration of a heat/ energy wheel (Nørgård-Hansen, 2016)

The wheel matrixes in heat and energy wheels are constructed with different materials, although aluminum is the most common due to its high thermal conductivity and thermal capacitance [Shah and Sekulic., 2003]. The matrix is the core of the wheel and allows transfer of heat and moisture between the two airstreams, as shown in Figure 2.1.

Each material used in matrixes has a specific range of applications, performance, and limitations. Besides aluminum, other materials that are used are ceramics, stainless steel, plastics, and paper. There is also a wide range of desiccants that is utilized [Shah and Sekulic, 2003].

Fixed matrix heat exchanger

The fixed matrix heat exchanger consists of two separate compartments and a damper housing. Each of the compartments contains material that stores the heat (matrix). Each matrix has a large surface. The surfaces alternately dissipate heat to supply air or absorb heat from the exhaust air. A damper regulates the airflow, so the absorbed heat from the exhaust air can be used to heat the fresh supply air. A timer controls the sequence. The transportation of odors and pollution is present, though lower in a rotary wheel. Changing the setting for the timer can regulate the efficiency [Novakovic et al., 2007].

2.3.3 Comparison of heat exchangers

The advantages and disadvantages of the different heat exchangers are different. The previously mentioned heat recovery methods are compared in Table 2.1.

	Type	Sensible efficiency	Latent efficiency	Advantages	Disadvantages
Energy wheel	Regenerative	80 - 85 %	80-85 %	High efficiency	Side-by-side necessary. Displacement of fans could cause leakages.
Fixed matrix	Regenerative	70 – 80 %	-	High efficiency	Side-by-side necessary. Minor frost formation
Plate	Recuperative	60 – 90 %	-	Simple construction. No leakages	Freezing could occur. Side-by-side
Run-around	Recuperative	65 – 70 %	-	Side-by-side ducts not necessary	Freezing could occur
Heat pipes	Recuperative	50 – 60 %	-	Simple construction. No leakages	Side-by-side.

Table 2.1 Comparison of heat exchangers [Novakovic et al., 2007]

2.4 Condensation

During cold weather operation conditions, condensation may occur in air-to-air heat/ energy exchangers. The same applies for frosting. The warm air may be cooled to dew point temperature as it passes through the exchanger.

In energy wheels, the desiccant coating may be degraded by enormous condensation. A condensation drain should be implemented if condensation occurs during system design conditions [Simonson, 2007].

Freezing of the heat exchangers increases the hydraulic resistance, drop of the airflow due to partial or complete blockage and an overall reduction of efficiency. Measures to avoid freezing can be applied, although they cause additional energy costs. The heat exchanger efficiency is also reduced.

Condensation may occur when the temperature of the exhaust air drops below the dew point. Since a rotary heat exchanger exposes the same surface for both the exhaust and supply air, any moisture from condensation will most likely evaporate into the supply air [Simonson, 2007].

In installed ventilation system in the field, several moisture sources provide moisture to the exhaust air. Moisture caused by everyday activities such, as cooking and hygiene are individual from user to user [Smith and Svendsen, 2016].

2.5 Carryover Leakages

The supply and exhaust airflows are balanced or varied. If a VAV-system is used, the building is put under a slight pressure difference between surrounding conditions. The efficiency will be affected by leakages of warm indoor air to the surroundings [Roulet et al., 2001]. When a CAV-system is used, air at varied temperatures is provided at constant air volume. Then it is important to calibrate the unit to avoid overpressure in the building, leading to air being pushed into the structure causing a heat loss [Novakovic et al., 2007].

A small fraction of flow leakages from hot to cold streams and vice versa is referred to as carryover leakages. Seals are applied to the rotating wheel to prevent leakages from happening. Carryover leakages in the heat exchanger may become significant with increased rotational speed. This causes a reduction in the regenerator performance. For heat exchangers operating at high effectiveness the influence of pressure drop and carryover leakages is important. The mass flow rate of the carryover leakage is a function of total void volume, rotor speed and fluid density [Shah and Sekulic, 2003].

2.6 Fault Analysis

To quantify a physical quantity a set of five fundamental conditions have to be determined, such as what is measured, how will the measurement be conducted, which equipment is suitable, how accurate must the measurements be and which processing routines shall be followed. Errors in measurements can be divided into three main categories, severe errors, systematic errors, and random errors.

The main cause of a severe error is the lack of attention or sloppiness. These types of errors must be avoided. Systematic errors are divided into instrument errors and personnel errors. Systematic errors may be due to fault in the instrument or incorrect usage by the user. Systematic errors can be avoided. Random errors cannot be eliminated, only reduced. These may be unsystematic affections or imperfect solutions on analog or digital instruments [Novakovic et al., 2007].

2.6.1 Unassembled Measurements

In unassembled measurements, the results emerge directly in/on the instrument, such as temperature on a thermometer. With repeated measurement, it may occur that the reading values are different [Novakovic et al., 2007]. To estimate the real value, the average value is calculated for several measurements:

$$\bar{x} = \frac{\sum x}{n} = \text{mean value} \qquad (2.6.1)$$

From the number of measurements, n, the individual measurements, x, and the mean value, $\bar{x}$, the uncertainty in the individual measurement may be expressed by the standard deviation, s

$$s = \sqrt{\frac{\sum(x-\bar{x})^2}{n-1}} = \text{standard deviation} \qquad (2.6.2)$$

The uncertainty in the mean value is then given by

$$U_T = \pm\frac{s}{\sqrt{n}} = \text{random error} \qquad (2.6.3)$$

The uncertainty of the instrument is termed as U_S. Thus the resultant uncertainty can be estimated by the expression

$$U_R = \pm\sqrt{U_T^2 + U_S^2} \qquad (2.6.4)$$

2.6.2 Assembled Measurements

The uncertainty in the result depends on several individual factors.

$$N = f(u_1, u_2, \dots, u_n) \qquad (2.6.5)$$

Where $u_1, u_2, \dots, u_n$ are the directly measures sizes, while N is the resulting value from the assembled measurement. Every u is measured with uncertainty (Δu), so

$$u_1 \pm \Delta u_1, u_2 \pm \Delta u_2, \dots, u_n \pm \Delta u_n$$

Thus the directly measured sizes give a resulting uncertainty in the assembled measurement:

$$N \pm \Delta N = f(u_1 \pm \Delta u_1, \dots, u_n \pm \Delta u_n) \quad (2.6.6)$$

The resulting uncertainty, ΔN, can be determined from the differential:

$$(N \pm \Delta N) - N = \pm \Delta N \qquad (2.6.7)$$

From Taylor

$$f(u_1 \pm \Delta u_1, \dots, u_n \pm \Delta u_n) - f(u_1, \dots, u_n) = \frac{\partial f}{\partial u_1}\Delta u_1 + \dots + \frac{\partial f}{\partial u_n}\Delta u_n \ (2.6.8)$$

If it can be presupposed that the uncertainty in the individual measurements is independent of each other, then ($\Delta N = U_R$):

$$U_R = \pm \sqrt{\left(\frac{\partial f}{\partial u_1} \times \Delta u_1\right)^2 + \dots + \left(\frac{\partial f}{\partial u_n} \times \Delta u_n\right)^2} \ (2.6.9)$$

Determining the supply airflow

If the supply air mass flow is unknown, it can be found by assuming a quasi-steady operating condition. The supply air mass flow is given by

$$\dot{m}_s = \dot{m}_e \left(\frac{T_{e,o} - T_{e,i}}{T_{s,i} - T_{s,o}}\right) \qquad (2.6.10)$$

3 Literature study

Analysis of heat exchangers is difficult to perform. The geometries and complex physical phenomena complicate the analysis. Thus they consist of assumptions for simplification and dependent equations. Experimental studies mostly consist of preparation of the experimental set-up [Patil et al., 2016].

3.1 Field tests of ventilation units in residential buildings

Merzkirch et al. (2015) conducted field tests of centralized and decentralized ventilation units in residential buildings. The paper addresses the shortcomings of installed ventilation systems.

The parameters that were considered were main airflows, internal and external recirculation, and sensitivity to differential pressure, SFP and heat recovery efficiency. The measurements were conducted between the years 2013 and 2014. The outside temperatures were between 0 and 4 °C and the wind speed low. By deriving real performance data from parameters investigated, it contributes to an increased accuracy of input parameters for energy calculations.

A tracer gas test was used to measure the volume flow in the ducts. The airflow in the exhaust duct was not measured and therefore assumed to be equal to the extract flow and reduced by the internal leakage ratio, which was calculated. For measuring the air temperatures, thermal wires with an accuracy of $\pm$3% within a temperature range of -25-400 °C. The thermal wires were placed as close to the heat exchanger as possible.

For the units with a low heat recovery efficiency, Merzkirch et al. point towards the unbalance in volume flow and the resulting excess of extract airflow as the reason.

For the decentralized systems, the mean heat recovery efficiency had a standard deviation of 17%, while the centralized systems with mean heat recovery efficiency of 65% had a deviation of 23%.

The paper concludes that only a well-balanced and well-installed system can lead to a good overall system. Also, every single factor should be taken into consideration.

3.2 Real heat recovery with air handling units

Roulet et al. (2001) conducted measurements with tracer gas tests to detect various malfunctions in thirty units located in Switzerland and Germany. The main focus in the report is air leakage in the ventilation system itself or leakage due to a pressure difference between the building and outside conditions causing leaking in the building envelope. The efficiency of ventilation and heat recovery decreases due to leakage. Calculation of the mass flow is done based on the tracer gas test. Measurements of the temperature were conducted. Thus the effectiveness could be calculated.

They found that in one-third of the audited units more than 50% of the supply air is lost due to leakages. The results presented shows that in the three best cases the real, global heat recovery efficiency was between 60 and 70% for units with an 80% nominal efficiency. For the three worst cases, Roulet et al. (2001) discovered that the real, global heat recovery efficiency was less than 10%.

3.3 Uncertainty analysis in the testing of air-to-air heat/ energy exchangers installed in buildings

Johnson et al. (1998) conducted tests on typical air-to-air heat exchangers to obtain data from the field test. They performed an analysis of the uncertainties with particular attention to the non-uniform temperature distribution in the ducts. The uncertainty analysis is used to determine the most representative method to calculate the effectiveness. The introduction ends with this quote:

As a consequence of the complexities of operating conditions noted above for installed air-to-air heat/energy recovery systems and the absence of an in-situ field testing standard, there are few published field test data and analyses of installed systems to guide designers and to verify savings for owners of air-to-air recovery systems (Johnson et al., 1998, p. 1640).

Huang and Niu (2016) also mention the relatively low number of publications within the field.

In the introduction, Johnson et al. (1998) it is mentioned that measuring airflow rates in ducting systems is difficult due to large uncertainties. Pressure loss coefficients have large uncertainties for ducting components. Non-uniform temperature distributions may affect the efficiency. The effectiveness can be reduced by misdistributions of flow. The paper also addresses the issue that standard for laboratory test conditions does not apply for a field test. Measurements of the relative humidity indicated small changes from the inlet to the outlet. Therefore no humidity data are presented in the paper, only the sensible effectiveness. Tracer gas tests for measuring the cross-leakage between supply and exhaust airstreams were not conducted, justified by that a tracer gas test is challenging to do in the field.

The paper concludes that the errors in the calculations of effectiveness are reduced by accurate calculation of mean temperature and the humidity by using data from local temperature and humidity. It is recommended to use extra sensors if possible when duct properties are non-uniform, to reduce the uncertainty in effectiveness, for more accurate bulk properties.

3.4 Influence of condensation on the efficiency of regenerative heat exchanger for ventilation

The condensation and the following latent heat release affect the overall thermal performance of heat exchangers in both partially and fully wet operating conditions. To be able to develop more energy saving systems it is necessary to compare the performance of heat exchangers during dry and wet operating conditions. [Patil et al., 2016]

Nizovtsev et al. (2017) developed a physical and mathematical model for calculating the air-to-air heat exchanger with periodically changing direction of airflow. Determination of sensible and latent efficiency and their dependence of relative humidity as well are presented in the paper. The paper concludes that the temperature efficiency at low humidity indoor air is high and independent of indoor air humidity. With a moderate or high humidity indoor air, the sensible efficiency is decreased, while the latent efficiency increases.

Al-Ghamdi (2006) did an analysis of air-to-air rotary energy wheels for his Doctor of Philosophy thesis. He developed a condensation model for the energy recovery ventilator. It describes the heat and mass transfer in a rotary wheel designed with a non-desiccant porous matrix. The paper concluded with the extent of condensation affects the effectiveness.

3.5 Measuring techniques for field test

To determine the efficiency, several measuring techniques can be used. The different methods are listed and described below.

Pitot

The turbulence in the duct has to be reduced when a Pitot tube is used. The Pitot tube measures the total pressure and the static pressure to determine the velocity pressure. The air velocity is derived from the velocity pressure. The tip of the Pitot tube has to point directly into the air stream. The accuracy of a Pitot tube depends on the geometry of the duct and its effect on the flow patterns, so the flow over the tube is laminar [Klopfenstein Jr, 1998].

Tracer gas method

Within tracer gas, there are several methods for measuring. On a general basis, tracer gas is described as one of the most feasible ways to measure airflows in ducts. Tracer gas is injected at a carefully chosen location since it is crucial to have a good mixing of tracer gas and the air. Concentrations of the tracer gas are measured at a different location to determine all required airflow rates for the equations from conservation of airflow. The sampling site has to be carefully chosen as well to make sure that the sample represents the total concentration. Leakages and shortcuts in the system can be detected by this method. To get the tracer gas to fully blend with the air, a minimum duct length is required or mechanical mixing placed inside the ducts. Therefore it is difficult to measure the airflow rates in buildings with mechanical ventilation since it requires suitable sites for placement of injection and sampling [McWilliams, 2002].

Anemometer

Anemometer has been widely used for measuring air velocities. One of the disadvantages is that the probe disturbs the flow field. This effect will be magnified when several points are measured. With a very small probe, the disturbance will be reduced. When measuring airflow with an anemometer the flow profile in the duct should be known [McWilliams, 2002].

3.6 Methods used by producers

In theoretical analysis of steady-state heat exchangers, it is necessary to make some assumptions. The most significant assumptions are steady-state conditions i.e. constant flow rates and fluid temperatures, negligible heat losses to or from surroundings, no thermal energy sources or sinks in the exchanger, uniform temperature in the fluid over every cross section of the exchanger and at last uniformly distributed thermal resistance in the exchanger [Shah and Sekulic, 2002].

For quantifying the performance of energy exchangers, the most important parameter is the effectiveness. It determines the economic viability or feasibility of the heat exchanger. The effectiveness is determined by steady-state test conditions since the inlet operating conditions change slowly in their application. Steady-state or transient methods can be used to measure the effectiveness, but then the uncertainty has to be taken into consideration. Thus follow that the uncertainty in the gained effectiveness is low. For heat/ energy wheels the effectiveness can range from 50% to 85% for commercial wheels [Simonson, 2007]

The Nordtest method (NT VVS 130) describes laboratory and calculation procedures for testing and rating a set of characteristics listed in Table 3.1 for balanced ventilation systems. The test is valid for all climates with heating and/ or cooling season. It is necessary to mention that only the heat recovery unit is tested and is accurate for all air-to-air heat exchangers. An equation for calculations of the different parameters is included in the Nordtest method.

The gained results from the testing are compared with tables given in the Nordtest method and should be within a certain range with the assigned tables.

Characteristics tested and rated for balanced ventilation systems
- Recirculation due to casing and internal air leakages, and external local short-circuiting for non-ducted units
- Fan performance (SPF) and net air exchange capacity
- Net heat and moisture recovery efficiency under various specified operating conditions
- Annual net heat recovery efficiency and COP for a given building type and local climate, for use in standard methods for calculating building energy use

Table 3.1 Characteristics tested and rated using the Nordtest method

For the sampling, the specimen should preferably be picked randomly from the production line by a neutral body for testing. In the test report the declared features and options given by the manufacturer be documented. The test rig consists of two chambers, an outdoors-climate, and one indoors-climate chamber.

During the testing, the several parameters are measured such as the dry bulb air temperature, humidity ratio, flow rate, barometric pressure, static pressure, electrical power, and voltage and tracer gas.

The laboratory procedure consists of four steps that are given in a certain order due to dependencies between them. Both step 2 and 3 depend on the result from step 1, whereas step 2 is sensible before step 3. The procedure is listed below.

1. Tracer gas measurements
2. Aerodynamic (fan) performance measurements (optional)
3. Thermal performance measurements (net heat & moisture recovery)
4. Annual thermal efficiency calculation (optional)

The main object for tracer gas tests it to determine the fraction of recirculated air in each of the outlets. The pressure distribution of the unit is measured in the aerodynamic performance measurements. The purpose of the thermal performance measurements is to determine the conduction heat loss through the casing as well as measuring the influence of low outdoor temperature on the thermal and aerodynamic performance.

The Nordtest method document refers to appropriate test standards such as NS-EN 308 (1997) for the test procedure for establishing the performance of air to air and flue gas heat recovery devices instead of duplicating them. According to NS-EN 308, the uncertainty of measurement shall not exceed $\pm 0{,}2$ K for dry bulb temperature and $\pm 0{,}3$ K for wet bulb temperature. The air should be mixed to avoid uneven temperature, and the maximum allowed deviation in a measuring plane is equal to $0{,}55(t_{s,o} - t_{s,i})$, where t is the temperature. The uncertainty shall not be greater than $\pm 3\%$ for measurement on airflow rates. Tests for cold climates should run for at least 6 hours. For the result of the measurement to valid, the heat effect ratio P_1/P_2 shall be within $\pm 5\%$ of 1. The heat ratio is valid if:

$$0{,}95 \leq P_1/\ P_2 \leq 1{,}05 \qquad (3.6.1)$$

Where $P_i = c_p \cdot \dot{m}_i \cdot \Delta T$. It is proposed to calculate the heat balance after every test to validate the result [NS-EN 308, 1997].

4 Basis for field test

During the fall of 2016, Øya kindergarten, as well as several other sites connected to Trondheim municipality with a rotary wheel for heat recovery in the ventilation system, was inspected with supervisor Hans Martin Mathisen and co-supervisor Maria Justo Alonso to consider the challenges on site. Øya kindergarten is located within walking distance from NTNU, Gløshaugen and the AHU and surroundings were satisfying regarding space and easy access. It was therefore decided to conduct tests in Øya Kindergarten. From reviewing pictures taken during the inspection and the measurement equipment, the decision fell on drilling holes in the ducts and measure the temperature and velocity in them. The available equipment for measurement is a thermal anemometer that can measure temperature and velocity.

4.1 Locations of field measurements

Øya kindergarten was completed in 2015 and is a passive house standard building where the occupants consist of 136 children and staff during the daytime. The kindergarten is built with massive wood and have a total area of 1 730 m^2 [Trondheim kommune, 2017].

4.2 Occupants behavior

The opening hours for the kindergarten are between 06.45–16.30. Beyond this, there is usually no occupiers in the building. The children in the care of the kindergarten are from 0- 6 years old. [Trondheim kommune, 2017]

4.3 Specifications from Trondheim municipality

Øya kindergarten has two AHUs in its ventilation system which is referred to as 360.01 and 360.02. The AHU used for the field test in the work of this thesis is 360.02. Systemair provides the specification for the AHUs. The specifications given in the technical data for the 360.02 include fluid data, AHU data, and operating point specification. The AHU consists of a rotary energy wheel, filter at the inlet to the AHU for both exhaust and supply air, fans for both supply air and exhaust air placed in the outlet, a heating coil with heated water as working fluid.

Operating point

The nominal values are based on the values for operating point 2, given in specifications that can be found in Appendix C.

The field tests are conducted on AHU 360.02. The nominal airflow given for the operating point is 9000 m^3/h for both supply and exhaust air, with chosen density of 1,205 kg/m^3. The nominal temperature efficiency given is 83,5 % for the rotary wheel during the working mode. The relative air humidity for supply air before and after the heat exchanger is given respectively as 90 % and 31 %. For exhaust air, the relative humidity is before and after the heat exchanger is respectively as 30 % and 97 %.

The filters used in the AHUs are F7-filters, which were last changed in November 2016. The fan control system is a VAV system, delivering a constant airflow at variable temperatures. The AHU 360.01 is set to be a CAV-system, while the AHU 360.02 is set to be a VAV system, but a service technician will be able to adjust the regulation to fit the users needs for both AHUs. The airflow is regulated separately for both supply and exhaust in both AHUs. The efficiency of the fans with static and total pressure is respectively 72,8 % and 80,3 %. The electric motors for the fan are permanent magnets motors. The motor power is given as 3,90 kW, with efficiencies at the operating point as 91,8%. In the specifications, the air temperature before and after the heating coil is given, respectively at 13,5 °C and 22,0 °C.

4.4 Assumptions for measurements

Ducts

The ducts for supply outlet and exhaust inlet have a diameter of 0,5, while the duct for supply inlet has a diameter of 1m. The supply inlet air ducts for both of the aggregates are connected to each other, with only one air intake, as shown in figure 4.1. It is assumed that the air supply for AHU 360.01 does not affect the supply air for AHU 360.02.

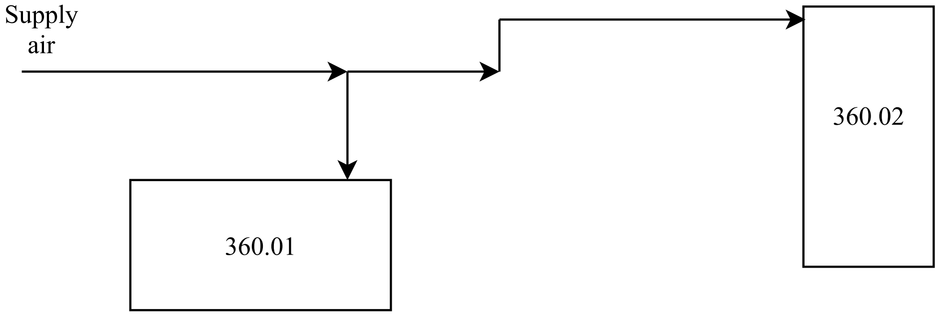

Figure 4.1 Flowchart of supply inlet air, Øya Kindergarten

Availability for measurements

For conducting measurements is necessary to have, access to the kindergartens technical room, enough time to finish measurements, external help from authorized personnel to control the AHU to wanted settings, available equipment.

4.5 What to measure

Since the ventilation system in the kindergarten is a VAV system, the airflows in the supply air, and exhaust air are assumed to be different. Measurements to determine the mass flow have to be performed. With the available equipment, the air velocity can be measured. Therefore the diameters of the ducts have to be measured, so the volume flow can be estimated before converting it to mass flow. Temperatures in the ducts before and after the heat exchanger will be measured.

4.6 Expectations and assumptions

It is expected that the measured temperatures in the ducts are close to uniform, and are close to the given temperatures from specifications regarding indoor temperature, 19°C for supply outlet and 22 °C for exhaust inlet, and outdoor temperature for supply inlet.

It is assumed that the wheel installed in the AHU is a heat wheel, which means that latent heat transfer is not occurring. Thus the humidity has not been measured. It is also assumed that the values gained are measured at the same time. The effectiveness is expected to be lower than the given 83,5%.

Since the flow pattern in the ducts is unknown, it is assumed laminar flow in the ducts.

5 Preliminary Measurement / Field test

In this chapter the, the procedure for measurements has been described. Fieldwork was performed at Øya kindergarten to analyze the effectiveness of the heat exchanger in the AHU. The fieldwork consisted of several experiments of measuring the temperature and velocity of the exhaust and supply air. The measurements were conducted in the ducts, except exhaust outlet.

5.1 Measuring Equipment

An anemometer of the type VelociCalc Plus 8360 measures air velocity and temperature. Measurement range and accuracy for the device is; velocity: 0,15 to 50 m/s with an accuracy of 3% or 0.02 m/s, whichever is greater. Temperature range: -10 to 60°C, with a resolution of 0,1 °C and precision of $\pm 0{,}3°C$.

5.2 Measuring points

The ducts are circular, therefore the measurement will be in a horizontal and a vertical direction. Two holes were drilled into each duct, horizontal and vertical. Through these, the probe was inserted into the duct. The size of the holes was 10mm, the same size at the thickest of the measuring device, to prevent leakage through the holes as far as possible.

The measuring points used for measurement are shown in Figure 5.1 and Figure 5.2 as dots, while the numbering indicates the distance from the inlet in cm. The division of sections is shown in Figure 5.1 and Figure 5.2. The division is done in the middle of two measuring points. It is assumed that the measurements in a point, both temperature, and velocity, represent the entire area. The measuring points are 5 cm apart from each other. These points were used for both measuring temperature and velocity of the air in the different ducts.

Figure 5.1 shows the measuring point for the ducts for air supply outlet and exhaust inlet air. Both ducts have a diameter of 0,5 m.

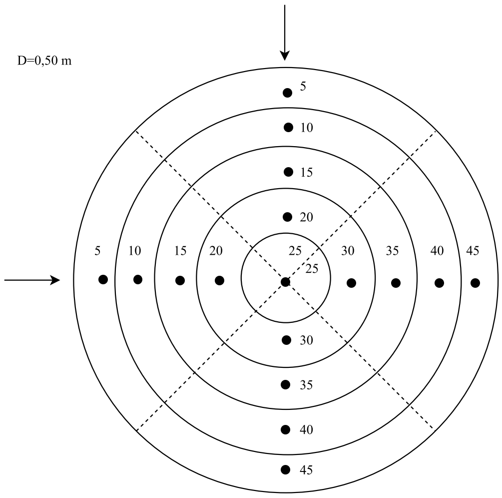

Figure 5.1 Cross-section of duct with diameter 0,5 m

Figure 5.2 shows the measuring point for the supply inlet. As can be seen in the figure, the section for measuring point 75 is significantly larger than the other parts. The large area connected is due to the length of the measuring equipment. Due to the placement of the duct is close to the wall and floor; it was not possible to enter the duct from opposite side. The area connected to each measuring point 75 is 15% of the total area of the cross-section of the duct.

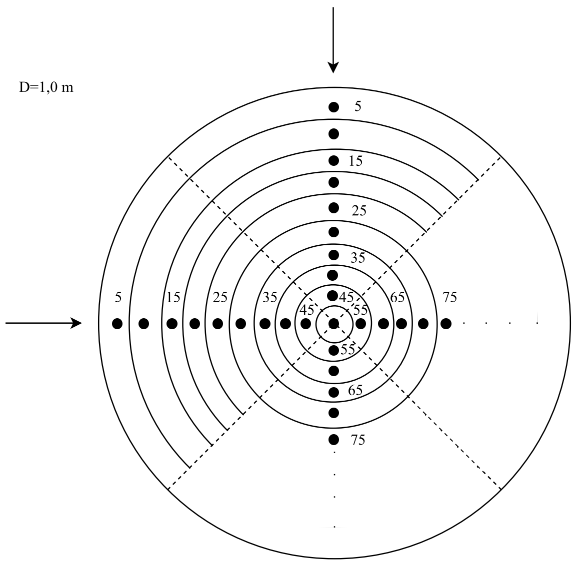

Figure 5.2 Cross-section of duct with diameter of 1 m

5.3 Values

Velocity: For each measuring point, ten measurements were done for the velocity, then the average of these ten was chosen to represent the measuring point, due to turbulent flow. The velocity is given in m/s.

Temperature: Since there were no changes in the temperature, the values are based on one measurement for each measuring point. Temperature is given in °C.

5.4 Settings for the system

Since the outdoor temperature is above 9°C the day of fieldwork, the matrix wheel is running at a lower speed. To cope with the high supply inlet temperature, the wheel was forced to run at 100%. The matrix was originally running at 70%. The heating coil turned off, so the change in temperature is only due to the heat wheel and some heat gain from the fan motor.

6 Results

This chapter presents the results of the measurements. The measurements of the velocity and temperature are conducted at separate occasions. The reason for this is because an employee in the municipality has to be available during the measurements. It was therefore not enough time to conduct the both measurements on the same day. It is assumed that the temperature does not affect the velocity and vice versa.

6.1 Velocity measurements

A total of 30 points were measured in the supply inlet duct, 15 in the vertical direction and 15 in the horizontal direction. For the supply outlet and the exhaust inlet, a total of 18 points were measured, 9 in each direction. As previously explained the exhaust outlet is not measured because it was out of reach. In each point, a set of ten measurements is conducted to represent the average velocity at that exact point. Then all the points have been summarized and averaged this is done with Eq. 6.1.1.

$$\frac{u_1 \cdot A_1 + u_2 \cdot A_2 + .. \ + u_n \cdot A_n}{A_{TOT}} \qquad (6.1.1)$$

Since the areas are known for each point, the volume flow in each duct can be estimated. By multiplying the velocity with the correlating area, the volume flow for each is found. Total volume flow for the duct is found by summarizing the volume flow for each area. Knowing the media, the mass flow can also be found by multiplying the volume flow with the density of air at the given temperature. The results of the averaged measurements of velocity, volume flow, and mass flow are displayed in Table 6.1.

	Supply inlet	Supply outlet	Exhaust Inlet	Assumed exhaust outlet
Mean velocity [m/s]	5,24	6,21	4,84	4,84
Volume flow [m^3/s]	4,119	1,221	0,951	0,951
Mass flow [kg/s]	5,145	1,471	1,136	1,136

Table 6.1 Mean velocity, volume flow in ducts

In the supply inlet, the greatest different between the measurements is 1,07 m/s which may indicate that the flow is somewhat steady. On the other hand, the volume flow at the supply inlet is over four times the amount of the volume flow at the exhaust inlet. This would result in a huge overpressure in the building, which is not wanted and not likely since the building has passive house standard. The velocities measured at the supply outlet shows quite large deviations. The measurements are within the range of 2,25-7,95 m/s. This is a good indicator that the flow is more turbulent and vortices may have occurred during the measurements, which is quite logical since the fan is ahead of the outlet.

Law of continuity states that the volume flow in should be equal to the volume flow out. It is therefore not quite clear why the volume flow at the supply outlet is almost four times lower than the volume flow at the supply inlet. One explanation can be that there are one or several leakages ahead of the measuring point. Since the measurements are not conducted simultaneously, there could arise large variations between the measurements. Another explanation can simply be that this measuring method is not optimal for this purpose. It is hard to pinpoint the exact reason, and it can be a combination of all of them.

In consultation with supervisor Hans Martin Mathisen, it was decided to neglect the velocity measurements of the supply inlet and assume the volume flow to be the same as the outlet flow. The basis of this choice was due to the overpressure that would have existed in the building if these measurements, in fact were, correct.

It should also be noticed that the exhaust inlet volume flow is lower than the supply outlet volume flow. This would create a slight overpressure in the building, which is expected since the fans in the AHU have different rpm. See Appendix C. Due to the rpm ratio of the two fans in the heat exchanger the overpressure should be around 0,01% higher than atmospheric pressure. 0,01 % would create a small overpressure, but big enough to drive the exhaust air into the ventilation heat exchanger. However, the difference is around 22%, which cannot be explained at the moment, but it will yield some trouble in the further calculation. Further, it is therefore needed to theoretically estimate a mass flow for either the supply side or the exhaust side. It is chosen to estimate the mass flow of the supply side based on the measurements of the inlet exhaust side. The mass flow at the exhaust side inlet is chosen because all measurements had pretty similar values and based on this it is decided to assume steady flow. The mass flow of the supply side is calculated with Eq. 6.1.2.

$$RPM_{Fan,Ex} \cdot \dot{m}_{Ex} = RPM_{Fan,sup} \cdot \dot{m}_{sup} \tag{6.1.2}$$

This yields a supply mass flow of 1,144 kg/s.

6.2 Temperature measurements

The temperature is measured in the same locations as the velocity but conducted at a different time. The average temperature is calculated by Eq. 6.2.1.

$$\frac{T_1 \cdot A_1 + T_2 \cdot A_2 + \,.. \, + T_n \cdot A_n}{A_{TOT}} \tag{6.2.1}$$

In addition to the measurements access to integrated temperature measurement equipment have been available. These can be found in Appendix D. Table 6.2 presents the averaged measured values of the temperatures in the duct, the measured values from the integrated measuring equipment as well as the deviation between the two.

	Supply inlet temperature, $T_{s,i}$	Supply outlet temperature, $T_{s,o}$	Exhaust inlet temperature, $T_{e,i}$	Exhaust outlet temperature, $T_{e,o}$
Temperature from measurements [°C]	9,4	19,8[2]	22,0	-
Temperature from integrated equipment [°C]	10,20	20,22[3]	22,12	13,69
Deviation	0,8	0,42	0,12	-

Table 6.2 Estimated temperature in ducts, temperature values from integrated equipment and deviation

The reading from the integrated equipment shows slightly higher temperatures than the values measured with the anemometer. The reason for this might be because the integrated sensors are placed closer to the heat exchanger and therefore be exposed to radiation.

The temperature at the supply outlet has to be corrected since the electric motor will cause a temperature increase due to heat loss. This correction is of significance to do before calculating the effectiveness. The electric motor has a total efficiency of 91,8%. The remaining 8,2% is assumed to be heat loss to the surroundings. Heat loss in Watt can be found with Eq. 6.2.2.

$$\dot{Q}_{loss} = \dot{Q} \cdot (1 - \varepsilon) = 3{,}9\, kW \cdot 0{,}082 = 0{,}3198\, kW = 0{,}32\, kJ/s \tag{6.2.2}$$

[2] Temperature $T_{s,o}$ have not been corrected regarding heat generation from el. motor.
[3] Temperature $T_{s,o}$ have not been corrected regarding heat generation from el. motor.

The increase in temperature can be calculated with equation 6.2.3

$$\Delta T = \frac{\dot{V}\cdot\Delta T}{\dot{V}} = \frac{0{,}266\ K\cdot m^3/s}{1{,}221\ m^3/s} = 0{,}2\ K \text{ (6.2.4)} \tag{6.2.3}$$

Where $\dot{V}\cdot\Delta T$ is given by equation 6.2.4 and $\dot{m}\cdot\Delta T$ is given by Eq. 6.2.5.

$$\dot{V}\cdot\Delta T = \frac{\dot{m}\cdot\Delta T}{\rho} = \frac{0{,}32\ K\cdot kg/s}{1{,}202\ kg/m^3} = 0{,}266\ K\cdot m^3/s \tag{6.2.4}$$

$$\dot{m}\cdot\Delta T = \frac{\dot{Q}_{loss}}{c_{p,air}} = \frac{0{,}32\ kJ/s}{1{,}0\ kJ/kg\cdot K} = 0{,}32\ K\cdot kg/s \tag{6.2.5}$$

This yields a temperature correction value of 0,2°C.

6.3 Calculation of effectiveness

With three known temperatures and the respective airflows, the effectiveness can be calculated. Since the airflow is not balanced, the efficiency can be calculated using Eq. (2.2.3) and Eq. (2.2.4)

$$\dot{Q} = 1{,}471\ {}^{kg}/_{s}\cdot 1\ {}^{kJ}/_{(kg\cdot K)}\cdot(19{,}6-9{,}4)K = 15{,}004 kJ/s \text{ (6.3.1)}$$

$$\dot{Q}_{max} = 1{,}136\ {}^{kg}/_{s}\cdot 1\ {}^{kJ}/_{(kg\cdot K)}\cdot(22-9{,}4)K = 14{,}314\ kJ/s \text{ (6.3.2)}$$

Then the effectiveness, ε is:

$$\varepsilon = \frac{\dot{Q}}{\dot{Q}_{max}} = \frac{15{,}004}{14{,}314} = 1{,}048 => 100\% \text{ (6.3.3)}$$

This is an invalid result.

From the effectiveness, the exhaust outlet temperature can be found. The belonging $T_{e,o}$ will then be 13,2℃. This is highly unlikely since this would mean that the supply air is gaining more heat than released heat from the exhaust air. The heat balance is calculated with Eq. (3.6.1). For this test gives a heat effect ratio of 1,5. Thus the results from measurement are not valid.

6.4 Effectiveness calculation with estimated supply air mass flow

Since the mass flow in supply outlet is assumed to be turbulent, the mass flow obtained with Eq. (6.1.2) will be used to calculate the effectiveness. The values for temperatures, $T_{s,i}$, $T_{s,o}$ and $T_{e,i}$, and the exhaust mass flow $\dot{m}_e$ from measurements, and estimated $\dot{m}_s$= 1,144 gives an estimated effectiveness of ε= 81,5%. This result is slightly lower than the nominal efficiency given in the specifications.

From the effectiveness estimated to be 81,5%, the exhaust outlet temperature can be found. The belonging $T_{e,o}$ will then be 11,7°C. This leads to a temperature difference of 10,2 K in supply air and 10,3 K in exhaust air. The deviation of 0,1K can be assumed to be heat loss to the surroundings. The heat balance ratio for this test by Eq. (3.6.1) is 1,00. According to NS-EN 308 (1997), this would validate the results from measurements.

The result from these calculation correlates with the designed conditions and are realistic with regards to heat transfer theory. Therefore this effectiveness will be used in comparison in Chapter 7.

6.5 Uncertainties

There are some uncertainties in the results from measurements due to equipment and performance. This implies some uncertainty in the result for effectiveness as a consequence. Thus determining the uncertainty of the effectiveness has to be carried out.

For calculation of the uncertainty of effectiveness based on obtained values from measurements, it is assumed that the uncertainties in the velocity or temperature measurements do not depend on each other. The uncertainties for temperature and mass flow measures are calculated based on equations in Chapter 2.6.1. The resultant uncertainty for temperature is $\pm$0,3°C. The resultant uncertainty for supply inlet mass flow is $\pm$0,115 kg/s, and for exhaust outlet mass flow it is $\pm$0,035 kg/s.

Then the relative error for the effectiveness can be calculated by using the equations in Chapter 2.6.2. As previous shown the effectiveness is given by the ratio between actual heat transfer and maximum heat transfer. Where the energy flow is determined by the expression

$$\dot{Q} = \int_{\Delta t} \dot{m} \cdot c_p \cdot \Delta\theta \cdot dt \;\; \left[{}^{J}/_{s}\right]$$

The actual heat transfer and the maximum heat transfer are found in chapter 6.3. The actual heat transfer, $\dot{Q}$= 15,004 kJ/s and the maximum heat transfer $\dot{Q}_{max}$=14,314 kJ/s.

It is assumed that $\varepsilon(c_p)=0$ and $\varepsilon(\Delta t)=0$. c_p is determined at 1005 J/(kg·K) and time of measurements is 1 second. From resultant uncertainty of velocity, the uncertainty of mass flow is $\epsilon_{sup}(\dot{m}) = \pm 0{,}115\ (kg/s)$ and $\epsilon_{exh}(\dot{m}) = \pm 0{,}035\ (kg/s)$. The resultant uncertainty of temperature is $\pm 0{,}3$°C.

Since it assumed that there is no correlation between the errors in measured mass flow and the measured temperature, the error in $\dot{Q}$ is determined by:

$$\epsilon(\dot{Q}) = \sqrt{\left(\frac{\delta\dot{Q}}{\delta\dot{m}} \cdot \epsilon_{sup}(\dot{m})\right)^2 + \left(\frac{\delta\dot{Q}}{\delta\Delta\theta} \cdot \epsilon(\Delta\theta)\right)^2}$$

$$\epsilon(\dot{Q}) = c_p \cdot \Delta t \sqrt{(\Delta\theta \cdot \epsilon_{sup}(\dot{m}))^2 + (\dot{m} \cdot \epsilon(\Delta\theta))^2}$$

$$\epsilon(\dot{Q}) = 1005 \cdot 1 \cdot \sqrt{(10{,}2 \cdot 0{,}115)^2 + (1{,}471 \cdot 0{,}3)^2} = 1{,}259\ \text{[kJ/s]}$$

The error in $\dot{Q}_{max}$ is then:

$$\epsilon(\dot{Q}_{max}) = \sqrt{\left(\frac{\delta\dot{Q}_{max}}{\delta\dot{m}} \cdot \epsilon_{exh}(\dot{m})\right)^2 + \left(\frac{\delta\dot{Q}_{max}}{\delta\Delta\theta} \cdot \epsilon(\Delta\theta)\right)^2}$$

$$\epsilon(\dot{Q}_{max}) = c_p \cdot \Delta t \sqrt{(\Delta\theta \cdot \epsilon_{exh}(\dot{m}))^2 + (\dot{m} \cdot \epsilon(\Delta\theta))^2}$$

$$\epsilon(\dot{Q}_{max}) = 1005 \cdot 1 \cdot \sqrt{(12{,}6 \cdot 0{,}035)^2 + (1{,}136 \cdot 0{,}3)^2} = 0{,}560\ \text{[kJ/s]}$$

The total error in the effectiveness is then finally expressed by:

$$\epsilon(\varepsilon) = \sqrt{\left(\frac{\delta\varepsilon}{\delta\dot{Q}} \cdot \epsilon(\dot{Q})\right)^2 + \left(\frac{\delta\varepsilon}{\delta\dot{Q}_{max}} \cdot \epsilon(\dot{Q}_{max})\right)^2}$$

$$\epsilon(\varepsilon) = \sqrt{\left(\frac{1}{\dot{Q}_{max}} \cdot \epsilon(\dot{Q})\right)^2 + \left(\frac{\dot{Q}}{\dot{Q}_{max}{}^2} \cdot \epsilon(\dot{Q}_{\mathrm{max}})\right)^2}$$

$$\epsilon(\varepsilon) = \sqrt{\left(\frac{1}{14{,}314} \cdot 1{,}259\right)^2 + \left(\frac{15{,}004}{14{,}314^2} \cdot 0{,}560\right)^2} = 0{,}097 = 9{,}7\%$$

The error of the effectiveness is $\pm$9,7 %. Since the effectiveness estimated is 100%, the effectiveness with corrections in 100-9,7 = 90,3%. This still gives an unlikely high effectiveness as shown in Chapter 6.3.

The results obtained in this chapter imply that the method utilized in the field work of this thesis do not provide valid results and they should not be trusted.

7 Discussion

It is desired to make the calculation of effectiveness more representative for a unit installed in the field. The literature within the discipline and the results from the preliminary field test are to be discussed in this chapter. To come to a conclusion, it is important that the advantages and disadvantages of methods are discussed. The obstacles throughout conduction of field test and a discussion of possible solution and improvements are presented.

7.1 Evaluation of Test Results

Test results are somewhat discussed in the presentation of them in Chapter 6. Thus in this chapter, the discussion involving test results will not repeat all of the previous discussion of the test results.

7.1.1 Temperature

The gained values correlate with outdoor temperature and what was expected with regards to supply outlet and exhaust inlet temperatures. Since the anemometer had not been calibrated in advance of the measurement, it is unknown how accurate the results are. The correlation between gained temperatures and expected temperatures indicate that the inaccuracy of the anemometer is small if there are any. The values were steady during measurement, so there was not necessary to calculate the average value for each measuring point.

The somewhat close to uniform temperature distribution in the ducts was not expected ahead of testing. During the field work in the project thesis in advance of this master thesis, the temperature measured with thermocouples in a duct were non-uniform with great deviation. It is necessary to mention that the measuring equipment was proven to be faulty.

In the analysis by Johnson et al.(1998) the non-uniform temperature distribution is pointed at as an element of uncertainty in the calculation of effectiveness.

A comparison of methods and equipment for measuring temperature will not be carried out in this thesis, since measuring mass flow is pointed out as the greatest challenge, but it is of interest for making the calculations more representative.

7.1.2 Velocity

During the measurement of velocities, it is assumed that the velocities do not change for a point, while another is measured. There is also no knowledge of the flow pattern in the duct. The measurements show that the flow is turbulent. Therefore the sum of results from the individual measurements does not provide a comprehensive image of the mass flow.

The measurements of velocities for supply air and exhaust air should have been conducted simultaneously. The AHU chosen for testing operates with a VAV system, meaning that the fans regulate the volume flow.

Since the VelociCalc only has a range of 0,75 m, there is a significant area that was not measured at the supply inlet. The assumptions made for this duct is quite daring since the mass flow gives great manifestations in the effectiveness calculation. If the results for the other ducts are taken into account, where the velocity measured ranged from approximately 7 m/s to 2 m/s. Therefore it is reasonable to suspect that there is a large range in the velocity in the supply inlet duct as well. Thus the determination of neglecting the result for velocity for the supply inlet is justified.

Since the air supply for both AHUs installed in the building are connected, this may be the reason for the high mass flow in the supply inlet duct.

7.1.3 Calculation of the Effectiveness

The calculation of the effectiveness with the obtained values from tests gave a void result (>1). Therefore either the values for velocity or temperature or even both are incorrect.

As mentioned in Chapter 3, the effectiveness is dependent on airflow. The calculation of effectiveness is based on measurements of velocity that have uncertainties.

The effectiveness of 81,5% is close but less than the nominal efficiency. The calculation of the effectiveness is based the values for temperatures, $T_{s,i}$, $T_{s,o}$ and $T_{e,i}$, and the exhaust mass flow $\dot{m}_e$ from measurements, and estimated $\dot{m}_s$. The heat balance ratio also validates the results.

The volume flow calculations for supply and exhaust air shows that the air supply is greater than the exhaust air. This causes a slight overpressure between the inside of the building and the outdoors conditions. Therefore it may be assumed that there is a leakage of warm air to the surroundings, decreasing the efficiency of the heat recovery unit. This will accordingly to Roulet et al. (2001) reduce the efficiency.

Since the kindergarten has two AHUs with a connected supply duct, and none measurements were conducted on the second unit, it is unknown how the distributions of air between the two AHUs are.

7.1.4 Fault analysis

The total uncertainty for the effectiveness is 9,7%. An error of 9,7% in the effectiveness is a relatively large error. As previously mentioned, there was made some assumptions during measurements regarding the methodical conductions of measurements. Repeating the measurements does not reduce methodical errors. Thus it is important to minimize them.

7.1.5 Effectiveness in other operating points

The results are gained from an AHU forced to run at 100%. With outdoor temperatures of 9°C, the matrix wheel would usually rotate at 70% since the need for heat transfer is reduced. The effectiveness of the heat recovery for matrix speed at 70% can be found by using the ε-NTU_0 method described in chapter 2.3.2. As the effectiveness found based only on the measurements was invalid (>1), the effectiveness for other conditions than designed condition, matrix speed at 70%, would have been invalid as well. Thus it has not been calculated.

7.2 Evaluation of test methods

By measuring both temperature and velocity in every measuring point in all of the ducts with only one sensor took a long time (6+ hours). It was assumed that the gained values were measured at the same time. If there was a change in supply temperatures, this is not discovered. The difference in mass flow could also be different, but this is also not discovered.

As Johnson et al. (1998) mention in their paper, the uncertainty for airflow is large. As they also mention, there are not many publications regarding field tests of heat recovery systems. Thus the selected choice of method for measuring velocity was not the best.

7.2.1 Determination of mass flow

From chapter 6.5 it is reasonable to believe that the measurement for supply air is invalid. By calculating the mass flow of supply air based on the exhaust air and all temperatures, the calculated efficiency correspond with the data for the AHU.

When measuring volume flow with an anemometer, the flow pattern in the duct should be known. It was not it this case. The measuring technique can therefore not considered reliable. If the flow in the duct is somewhat laminar, and the flow pattern in known, the method could be reliable enough to estimate mass flow.

Tracer gas is a common, well-known method for estimation of the mass flow. The disadvantage is the requirement of long enough ducts for the trace gas to mix with the air. Some of the ducts at Øya Kindergarten might be long enough for tracer gas to properly mix with the air. It was therefore desired to conduct tests utilizing tracer gas, but the necessary equipment is not available at NTNU.

7.2.2 Temperature

The measurement of temperatures was conducted as stated in the Nordtest method for a circular duct, in a vertical and horizontal direction. The measurement was simple to do and the reading values were stabile.

The measurements of temperatures were done during a day with typical spring temperatures at 9 °C. It would have been interesting to investigate how efficient the heat recovery was during colder days. Since the occupants of the kindergarten are children from the age of 0-6 years, they are vulnerable to cold temperatures. Thus it would probably not be well received by children and employees if the heating coil was turned off during a cold day. In the specifications for the heating coil it is given that the air is heated from ca. 13 °C to 20°C. Thus it would be desired to performed measurements during outside the hours/ days the building is occupied.

7.2.3 Location

The measurements were conducted in the area where the available ducts had straight lengths. It also seemed most suitable due to surroundings and was as close as possible to the heat exchanger. The measurements should ideally be done as close to the heat exchanger as possible. The distance between the locations of measurement and the heat exchanger may be too far for the result to be valid due to heat loss and gain in the ducts.

Since the measurements were conducted with the anemometer, with its limitation in range, it would have been challenging to place the sensor closer to the heat exchanger and at the same time do measurements that covered the entire area in a satisfying order.

The condition in the exhaust outlet duct was not measured because of the inconvenient shape and placement of the duct. The duct is too wide for the entire cross-section to be within reaching without climbing on top of it. Due to respect of government property, climbing on the equipment was not an option.

7.3 Equipment

Since the VelociCalc does not have a logging memory, it is not possible to log over several days. Logging over several days would also demand several devices. This limits the application of a single unit.

The measurements were conducted in a vertical and horizontal direction. Thus the cross-section of the ducts was divided into sections, with a measuring point for temperature and velocity representing the entire area of the associated point. For the ducts with a diameter of 1 m, the problem was that the range of the measuring device is only 0,75 m. Therefore an assumption that the sector belonging to the measuring point furthest away was larger than the others, representing 15,65% of the total area. Thus there is great uncertainty regarding the velocity and temperature in these sectors.

Since it is unknown when the equipment was calibrated last time, it is necessary to be skeptical to the result. On the other hand, it is the temperature differential between inlet and outlet that is important for the calculations. The differential should be equal even if the accuracy is incorrect.

As the flow is turbulent, and the anemometer only measures in a single point, the device is not suitable for measuring values used for mass flow calculations. As the AHU delivers a constant air temperature at various airflow, the anemometer is suitable for measurements of temperatures. The calculation of error in effectiveness in Chapter 6.5 shows that the error is high. The heat ratio calculations for the tests gave 1,5 and 1,00 in ratio respectively for the result for measurement and the alternative presented in Chapter 6.4. The result is not in the equipment's favor since the uncertainties are connected to measurements temperature and velocity. Utilization of velocity measurements is not to be a proposed method for estimating mass flow in ducts if the flow pattern is unknown.

7.4 Difficulties in Conducting Measurements in the Field

During the work of this thesis, several measurements have been conducted in Øya kindergarten. The work with measurements started at the beginning of March. A set of holes was drilled in the ducts, but due to limited lifetime of batteries and availability of equipment, as well as limited access to the technical room, the preparations for measurement was more time consuming than anticipated.

During the first measurement of temperature and velocities, the outdoor temperature was low, but due to an early closing of the kindergarten that day, the measurement was aborted. Since the calculations require a full set of measurements, these were useless for the thesis.

A new attempt of completing the measurement was done a few days later. Due to heating of supply outlet air by the heating coil, these measurements were rejected. Only authorized personnel can control the system. As the operator in Øya kindergarten was assigned to other projects, the measurement was postponed further. The two first conducted measurements were of some value to this thesis; since the time required to finish all measurements were discovered and gave practice in using the equipment.

At last, the measurement of temperatures could be conducted with the assistance from the energy department in Trondheim municipality. This was organized due to a coincidental request from the energy department in Trondheim municipality. Unfortunately, there was not enough time to measure the velocities during this session before the kindergarten closed. Thus another session another day was necessary.

At the final day of measurement, only the velocities were measured. The settings for the AHU were the same as for the previous measurement. It was assumed that the conditions in the ducts were the same as the day of temperature measurement.

In a retro perspective, it is clear that a direct communication channel with the energy department in Trondheim municipality should have been established during the start-up of the work with this thesis.

The measurements were postponed due to difficulties finding literature regarding field-test of heat recovery systems. This caused that several rookie mistakes were done such as measuring velocity of air without knowledge of the flow pattern, instead of pressure or use other known techniques for finding the mass flow. Relevant literature was discovered after the measurements were performed. The measurements should have been conducted during the semester's coldest period, which is February-March. Since the building is used during measurements, it would probably not have been acceptable to use the AHU without using the heating coil. Therefore the matrix wheel was forced to run at 100% during testing as an attempt to replace a low supply inlet temperature.

To be able to conduct measurement under circumstances that reduced the uncertainties as far as possible, it was in this case dependent on a person in Trondheim municipality controlling the system. Thus the availability of staff is crucial, and that schedules are corresponding. This caused neglecting of conducting measurements in several buildings since the measurements in Øya Kindergarten were completed at the end of May.

To be able to conduct proper measurements it is crucial to have full access to the system, the control system and be able to perform measurements in all ducts simultaneous. The flow pattern should be investigated ahead of measurements. Then the anemometer can be applied where the flows are most suited for measurements with regards to laminar flow. The measurements conducted close to the fan in the outlet duct is most likely in a turbulent flow. The inlet measurements appear more reliable. Thus the velocities should be measured in the inlet ducts. Based on results the airflow is assumed to be turbulent in the outlet ducts.

7.5 Discussion of literature study

Since there are few published papers on this matter, the literature study is only based on a few articles. Ideally, the number of reviewed papers should have been higher. Some articles were published nearly 20 years ago, but by comparing these with newer publications, they are still relevant.

Both Merzkirch et al. (2015) and Roulet et al. (2001) used tracer gas methods to calculate the mass flow in the ducts. While Johnson et al. (1998) describe the tracer gas test as hard to conduct in field-testing and neglected the measurement of humidity as test samples indicated similar humidity.

From these papers, it is clear that the mass flow has a significant effect on the efficiency of an AHU. The importance of correct measurement of the mass flow cannot be stated enough. It is clear that correct mass flow makes the calculation of the effectiveness more representative.

The utilization of Pitot tubes for measuring mass flow in a duct is not supported in any of the literature reviews so it can be ruled out in further projects.

All of the measurements in the mentioned papers are conducted over a long period.
The temperature difference is also an important factor in calculations. In Norway, the temperature variation between seasons is significantly large. Even typical temperatures for a season vary from year to year. Therefore measurement should be conducted over a longer period several times during a year. It is likely that the effectiveness based on measurement from a cold winter day will not be equal to an effectiveness based on measurement from a rainy day during spring.

If the estimation of mass flow is based on a less feasible method, conducting the same test several times and calculate the heat balance ratio after each test should validate it. Since the mass flow is challenging to measure, the uncertainties can be great. Thus one feasible method is better than several less feasible methods for measuring mass flow.

As it is desired to conduct tests during cold days to push the heat exchanger to its limits, the measurements should have been done during at least 6 hours according to standards. With only one device, this would have been difficult to perform. The heat balance ratio is useful way to determine the validation of result during measurements. Therefore it should be calculated between each test as proposed in the standard.

It is clear that the effectiveness depends on the condensation. At low or equal humidity it seems that the effect of condensation is small, close to negligible. Johnson et al. (1998) neglected measurement of humidity in their analysis. Thus the humidity of the air should be measured to determine if it is necessary to identify the sensible and the latent efficiency.

Latent heat transfer was not investigated in this thesis, but it should have taken into consideration. All of the authors of articles regarding condensation and its effect on the performance of a system conclude with that the humidity is affecting the efficiency. In Norway, the climate changes from season to season. There are cold temperatures with dry air, and there are warm temperatures with moist air. Thus the sensible and latent efficiencies for a heat exchanger could vary. Therefore it is suggested that the heat recover efficiency should be calculated for different seasons for the calculations to be more representative.

7.6 Evaluation of test results against TEK, suppliers data sheet and planners design data.

The minimum requirement for yearly mean temperature efficiency for heat exchanger is set to be ≥80% in non-residential buildings. The supplier's data sheet states that the efficiency is 79,9% when running with maximum volume flow. The planners design data state that the efficiency is 83,5% when running according to operating point.

The effectiveness discovered with estimated supply mass flow was 81,5%. This is within the requirements stated in NS 3701. The heat balance ratio is within requirements, although the values have not been obtained with measurements, but instead by estimation. It is also slightly lower than the given efficiency for designed operating point, which was expected before measurements.

Although it is necessary to mention that the result is gained from dubious measuring techniques and calculations with assumption that the mass flow is equal. The rotary heat wheel is running at 100% during the measurement.

7.7 Suggestion for improvements

As the most realistic effectiveness calculated was based on the estimation of mass flow with Eq. (6.1.2), the validation of this assumption should be conducted. Knowledge of the flow pattern in the ducts would be useful in determination of measuring site in the duct.

The limited amount of published literature has made the literature study more time-consuming than it should have been. To push the development of methods further and increase the motivation for research of real heat recovery of ventilation air, stricter regulations have to be set by the government within real heat recovery.

8 Conclusion

A literature study regarding field test of heat recovery in ventilation air has been conducted. Heat recovery in ventilation systems depends on several factors, such as air leakage in system, uniform temperature and velocity distribution in inlet of matrix wheel. Also, the type of heat exchanger utilized has a great influence on heat recovery in ventilation air.

With high or partly high relative humidity in the air, the latent heat efficiency will increase while the sensible efficiency will be reduced. With low relative humidity, the latent efficiency can be neglected. Thus the humidity should always be measured to determine the total efficiency.

The planning of measurements could have been better, but it is difficult to find literature regarding field measurements. Measurements were conducted at Øya Kindergarten, and due to the limited access and availability of the system, there were not enough time to conduct tests at other sites. Based on literature investigated and results from measurements, the greatest challenge within representative calculations is determining the mass flow through the heat exchanger.

Several techniques for measuring mass flow in a duct were investigated and analyzed. The method chosen was velocity measurements with an anemometer. The reason for choosing this method for measuring was solemnly based on the available equipment provided by NTNU. The anemometer was used to measure both temperature and velocity. The disadvantages of this method have been evaluated, and it comes forth that the flow pattern in the duct should be known when utilizing an anemometer for estimating mass flow. By establishing knowledge of the flow pattern in the ducts, the measurements can be conducted in the most suitable sites for determining mass flow. It comes forth that the velocities should be measured in the inlet ducts since the airflow is assumed to be turbulent in the outlet ducts, based on the results.

The results from measurements are considered unreliable for mass flow. The calculated effectiveness (>1) based on measurements do no correlate with heat transfer theory or design data. The heat balance ratio confirms that the result is invalid.

Assumptions were made to be able to estimate the effectiveness. This results in an effectiveness of 81,5%, which corresponds to both theory and design data. The effectiveness of 81,5% for the heat exchanger is within the requirements in NS 3701 (2012), but slightly lower than the nominal efficiency given by the supplier. Since the regulation by NS 3701 (2012) is fulfilled, this is acceptable.

To push the development of methods further and increase the motivation for research of real heat recovery of ventilation air, stricter regulations have to be set by the government within real heat recovery.

9 Further Work

Further work that is considered as important by the author is to further develop the test method, ensure access to systems that shall be used for tests and conduct more tests.

9.1 Test method

As tracer gas test is challenging to utilize in the field, the applied measuring technique in this thesis should be further developed. The method can easily be applied in several systems, which is its greatest advantage. It is proposed to obtain an instrument for velocity measurements that consists of several sensors, without disturbing the flow too much, so that the velocity can be measured at several points simultaneously in the duct. The equipment should also have the ability to log the values. There should be more than one device available so that measurements can be conducted in several ducts at the same time.

9.2 Access to HVAC systems

Unlimited access to the HVAC systems should be obtained, as well as authorization to regulate the different parameters in the control system.

9.3 Conduction of measurements

The measurement should be conducted in several buildings. There should be considered to have more than one to conduct measurements. With only one device the measurements took a lot of time. Standards for testing should be more applied in the conduction of tests. Including the latent efficiency should be individually evaluated for each site.

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Appendix A

SJA tittel: Gjennomføring av målinger i Øya barnehage, Trondheim		
Dato: 08.03.17	Sted: Trondheim	
Kryss av for utfylt sjekkliste:		

Deltakere:		
Ranja T. Nørgård-Hansen		
SJA-ansvarlig:Ranja T. Nørgård-Hansen		

Arbeidsbeskrivelse: (Hva og hvordan?) Målinger av temperatur og hastighet til luft i ventilasjonskanaler før og etter varmegjenvinneren på teknisk rom. Det må borres to hull i hver kanal, til horisontal og vertikal måling. Det er totalt fire kanaler, så det må borres åtte hull. Først må utvendig isolasjon (mykt materiale) fjernes for å komme til stålrøret. Dette gjøres med kniv for å skjære ut og for å få et pent snitt. For å borre hull brukes en elektrisk håndholdt drill med 10 mm bor. Målinger gjennomføres i ett hull om gangen. Etter målinger repareres hullene med VVS-teip fra Glava og isolasjonen legges på plass. Ved Glava-isolasjon teipes det også utenpå for å holde isolasjonen på plass.
Risiko forbundet med arbeidet: Kutt fra kniv eller borr, fragmenter i øyne. Rusk i luftkanalene.
Beskyttelse/sikring: (tiltaksplan, se neste side) Vernebriller og forsiktig håndtering av kniv og drill. Luftkanalene har enten overtrykk eller går gjennom et filter, liten risiko for rusk i ventilasjonsluften.
Konklusjon/kommentar: Ved forsiktighet er det liten risiko for kutt. Ved bruk av vernebriller er det liten risiko for fragmenter i øyne.

Anbefaling/ godkjenning:	**Dato/Signatur:**	**Anbefaling/ godkjenning:**	**Dato/Signatur:**
SJA-ansvarlig:		Områdeansvarlig:	
Ansvarlig for utføring:		Annen (stilling):	

HMS aspekt	**Ja**	**Nei**	**Ikke aktuelt**	**Kommentar / tiltak**	**Ansv.**
Dokumentasjon, erfaring, kompetanse					
Kjent arbeidsoperasjon?	X				
Kjennskap til erfaringer/ uønskede hendelser fra tilsvarende operasjoner?	X				
Nødvendig personell?		X			
Kommunikasjon og koordinering					
Mulig konflikt med andre operasjoner?		X			
Håndtering av en evt. hendelse (alarm, evakuering)?		x			
Behov for ekstra vakt?		X			
Arbeidsstedet					
Uvante arbeidsstillinger?		X			
Arbeid i tanker, kummer el.lignende?		X			
Arbeid i grøfter eller sjakter?		X			
Rent og ryddig?	X				

Verneutstyr ut over det personlige?		x			
Vær, vind, sikt, belysning, ventilasjon?	x				
Bruk av stillaser/lift/ seler/stropper?		x			
Arbeid i høyden?		x			
Ioniserende stråling?		x			
Rømningsveie r OK?	x				
Kjemiske farer					
Bruk av helseskadelige /giftige/ etsende kjemikalier?		x			
Bruk av brannfarlige eller eksplosjonsfarl ige kjemikalier?		x			
Må kjemikaliene godkjennes?		x			
Biologisk materiale?		x			
Støv/asbest?		x			
Mekaniske farer					
Stabilitet/ styrke/ spenning?		x			
Klem/kutt/ slag?	x				
Støy/trykk/ temperatur?		x			
Behandling av avfall?		x			
Behov for spesialverktøy ?		x			

Elektriske farer					
Strøm/ spenning/over 1000V?		x			
Støt/ krypstrøm?		x			
Tap av strømtilførsel?		x			

Området					
Behov for befaring?		x			
Merking/ skilting/ avsperring?		x			
Miljømessige konsekvenser?		x			
Sentrale fysiske sikkerhetssystemer					
Arbeid på sikkerhetssystemer?		x			
Frakobling av sikkerhetssystemer?		x			
Annet					

Appendix B

Density of air

Temperature T (°C)	Density of air (kg/m^3)
15	1,2250
10	1,2466
5	1,2690
0	1,2922
-5	1,3163
-10	1,3413
-15	1,3673

With interpolation, the density for:

9,4 °C – 1,2493 kg/ m^3

19,8 °C - 1,204 kg/m^3.

19,6 °C – 1,2051 kg/m^3

22 = 1,1948

Specific heat air

C_p= 1,0 kJ/kg K

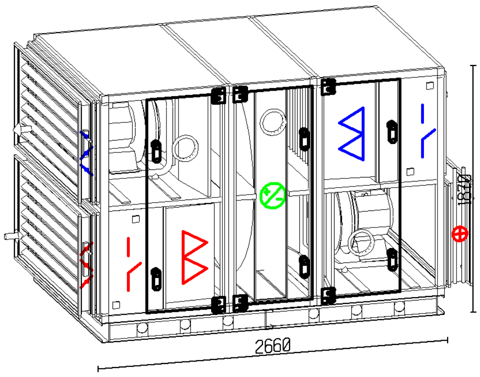
Appendix C

Produktutvalgsprogram | SystemairCAD Versjon C2014-11.05.D8 | 04.02.2015

Oversikt for aggregat nr: 1

DVCompact 40

Ordrenummer **0003410820**
Prosjekt **Øya Barnehage Trondheim 2015**
Anlegg **360.01 /**



DVCompact luftbehandlingsaggregat med fullintegrert automatikk ferdig kablet fra fabrikk

Luft-/viftedata	Tilluft	Fraluft	
Luftmengde (1,205 kg/m³)	11250	11250	m3/h
Lufthastighet i aggregat	2.45	2.45	m/s
Eksternt trykktap	200	200	Pa
Viftehastighet	1499	1487	o/min
Motor	3.90	3.90	kW
Spenning	3x230	3x230	V
Strøm, Amp., merket	18.7	18.7	A

Aggregatdata		
Aggregatbredde		1720 mm
Vekt		960 kg
Filter		Tilluft F7 - Fraluft, avtrekk F7
Varmeveksler		79.9 %
SFPv, ved rent filter inklusiv frekvensomformer		1.81 kW/(m³/s)
Varmebatteri	Luft	32.7 kW - 13.5/22.0°C
	Vann	60/40°C - 10.8 kPa - 0.40 l/s - 26.9mm / 26.9mm Rørtilkobling

Lydeffektnivå	Tilluft, innblåsning	Uteluft, inntak	Fraluft, avkast	Fraluft, avtrekk	Lyd, omgivelser
Total	84 dB(A)	69 dB(A)	86 dB(A)	69 dB(A)	59 dB(A)

Systemair AS

Telefon : +47 51969700 Org nr NO 929 387 384 MVA
Faks : +47 51969799
www.systemair.no
mailbox@systemair.no

Arbeidspunkt 2

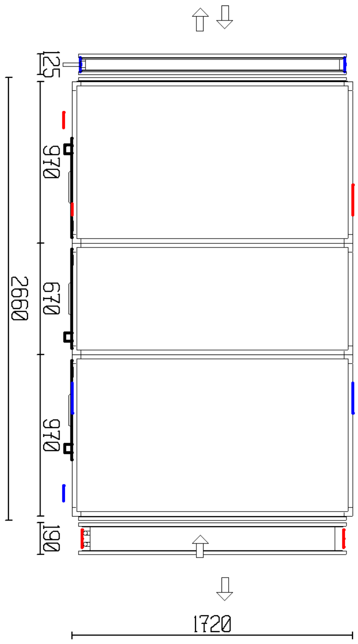
DVCompact luftbehandlingsaggregat med fullintegrert automatikk ferdig kablet fra fabrikk

Luft-/viftedata	Tilluft	Fraluft	
Luftmengde (1,205 kg/m³)	9000	9000	m3/h
Lufthastighet i aggregat	1.96	1.96	m/s
Eksternt trykktap	200	200	Pa
Viftehastighet	1302	1293	o/min
Motor	3.90	3.90	kW
Spenning	3x230	3x230	V
Strøm, Amp., merket	18.7	18.7	A

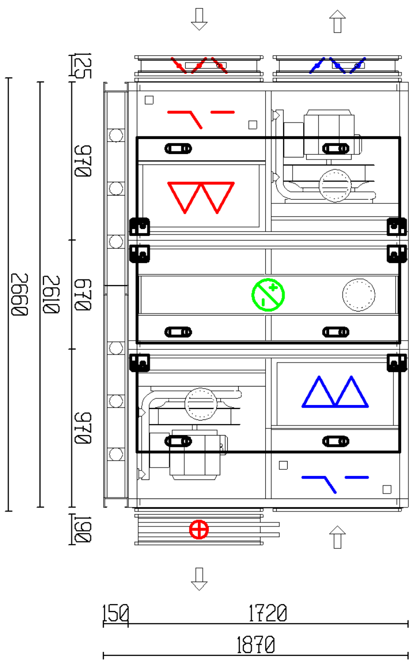
Aggregatdata		
Aggregatbredde		1720 mm
Vekt		960 kg
Filter		Tilluft F7 - Fraluft, avtrekk F7
Varmeveksler		83.5 %
SFPv, ved rent filter inklusiv frekvensomformer		1.46 kW/(m³/s)
Varmebatteri	Luft	21.3 kW - 15.1/22.0°C
	Vann	60/40°C - 5.0 kPa - 0.26 l/s - 26.9mm / 26.9mm Rørtilkobling

Lydeffektnivå	Tilluft, innblåsning	Uteluft, inntak	Fraluft, avkast	Fraluft, avtrekk	Lyd, omgivelser
Total	80 dB(A)	65 dB(A)	82 dB(A)	65 dB(A)	55 dB(A)

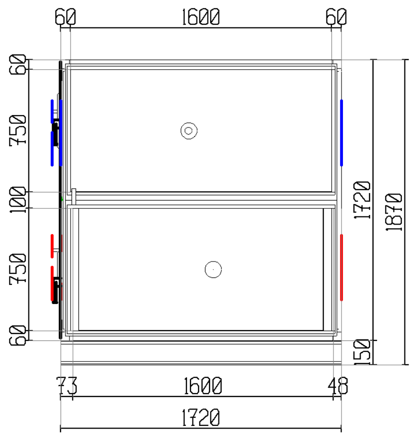
Plantegning



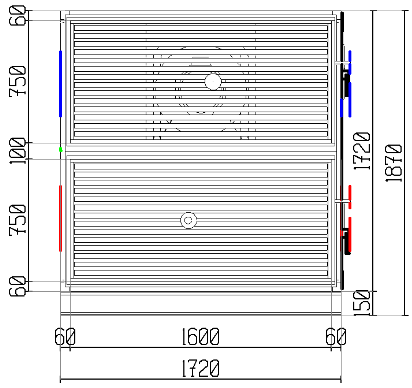
Inspeksjonsside



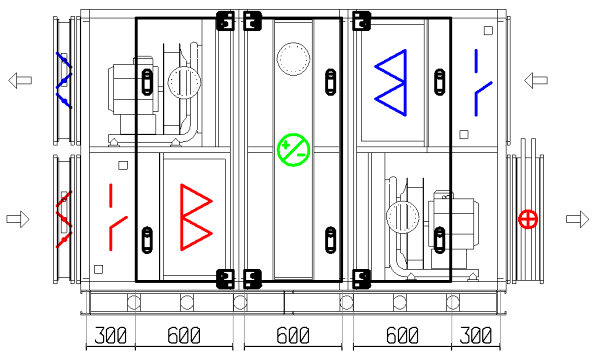
Høyre gavl



Venstre gavl



Mål på dører og panelser



Teknisk spesifikasjon

Aggregat

Frekvensbånd [Hz] Lydeffektnivå	63 [dB]	125 [dB]	250 [dB]	500 [dB]	1K [dB]	2K [dB]	4K [dB]	8K [dB]	Total [dB(A)]
Tilluft, innblåsning	73	83	82	82	82	71	69	68	84
Uteluft, inntak	64	75	74	66	63	53	48	47	69
Fraluft, avkast	74	85	83	83	84	74	72	72	86
Fraluft, avtrekk	64	75	74	66	63	53	48	47	69
Lyd, omgivelser	62	72	60	51	50	43	40	42	59

Automatikk

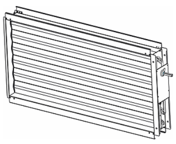
Sprog i controler menu	Norsk	
Ekstern kommunikasjon	Honeywell Ethernet BACnet type B-BC	
Viftestyring	CAV	
Spjeldmotor, tilluft	Motor fjærretur	
Spjeldmotor, avkast	Motor fjærretur	
Frostbeskyttelse, varmebatteri	Utvendig anleggsføler	
Batteri konfigurasjon	Varmebatteri	
Ventil for varme	3-veis ventil, Kvs 4.00, DN15 Innvendig koblet	
Trykkfall	13	kPa

Hovedtilførsel til automatikken

Tavledata	Spenning	3x230V + PE 50	Hz
	Maks. ik	16	kA
	Min. ik	650	A
	Maks. sikring	60	A
	Min. sikring	40	A

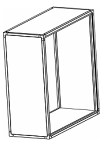
Tilluftsaggregat består av

Spjeld



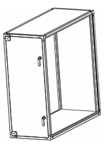
Trykkfall	8	Pa
Spjeldblad	Standard	

Tomdel



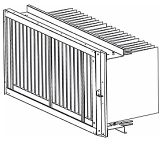
Trykkfall	4	Pa
Lengde	300	mm

Inspeksjonsdel



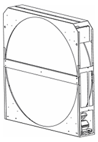
Trykkfall	4	Pa
Lengde	150	mm

Filter



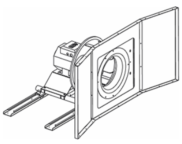
Dimensjonerende trykkfall	141	Pa
Start trykkfall/Slutt trykkfall	82/200	Pa
Lufthastighet, frontareal	2.86	m/s
Lufthastighet, filterareal	0.13	m/s
Filterklasse	F7	
Filterstørrelse	3x[490x490] + 2x[592x287] + 1x[287x287]	
Filterlengde	640	mm

Roterende varmeveksler



	Tilluft	Avtrekk	
Luftmengde	11250	11250	m3/h
Trykkfall	234	234	Pa
Lufttemperatur, før/etter	-20.0/13.5	22.0/-9.3	°C
Relativ luftfuktighet, før/etter	90/31	30/97	%
Effekt	148.51		kW
Temperaturvirkningsgrad	79.9		%
Varmegjenvinner type		Kondensatorvarmeveksler	
Temperaturvirkningsgrad		Høyeffekt	
Drivsystem for rotor		Variabel drift	
Elektriske data		1x230V, 100W, 1.3Amp	
Inspeksjonsvindu		1	Stk

Vifte, Kammer



Luftmengde	11250	m3/h
Eksternt trykktap	200	Pa
Trykkfall	36	Pa
Statisk trykk	646	Pa
Totaltrykk	712	Pa
Akseleffekt	2.77	kW
Viftehastighet	1499	o/min
Max viftehastighet	1735	o/min
Virkningsgrad ved statisk trykk	72.8	%
Virkningsgrad ved total trykk.	80.3	%
K-faktor	308	
Viftetype	ER56Cpro	
Direkte drift		

Motor

Motortype	IE4, PM-Motor	
Motor type-Normstr.	HPS 90 S-L	
Termosikring	Termistor	
Motoreffekt	3.90	kW
Omdreininger, merket	1700	o/min
Omdreininger, maksimum	1870	o/min
Strøm, Amp., merket	18.7	A
Virkningsgrad	92.3	%
Virkningsgrad i arbeidspunktet	91.8	%

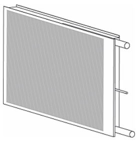
Frekvensomformer

Spenning	3x230	V
Frekvensomformer kablet fra fabrikk. (23.0 Amp)	1	Stk
Effektforbruk fra hovedtilførsel, inkl frekvensomformer	3.13	kW

Frekvensomformer IP 20 er montert ved siden av viften inne i viftedelen til aggregatet. Det er benyttet skjermet kabel mellom frekvensomformer og viftemotor. Alle nødvendige parametre er lagt inn på omformer, tilpasset til motor og viftehjulets karakteristikk.
Forsyningen til aggregatet skal ha jordfeilbryter ikke mindre enn 100 mA, og beregnet for frekvensomformere.

Inspeksjonsvindu	1	Stk

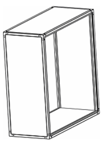
Varmebatteri MAX, Væske



Luftmengde	11250	m3/h
Trykkfall	19	Pa
Lufttemperatur, inn/ut	13.5/22.0	°C
Relativ luftfuktighet, før/etter	31/18	%
Effekt	32.74	kW
Lufthastighet	2.60	m/s
Væskemedie	Vann	
Væsketemperatur inn/ut	60.0/40.0	°C
Væskemengde	0.40	l/s
Væske trykktap	10.8	kPa
Væskehastighet	0.79	m/s
Batterivolum	3.5	l
Anslutningsside	Service side	
Tilkoblingsmål inn/ut	26.9mm / 26.9mm	
Rørmateriale	Cu	
Lamellmateriale	Al	
Lamellavstand	2.1	mm
Rørrader	1	
Overflate areal	22.9	m2
Vekt, batteri	14.4	kg
Batteri type	M 25x22-3/8 CS 30 T 1 R 1600 A 2.1 P 7 NC	

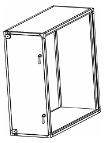
Fraluftsaggregat består av

Tomdel



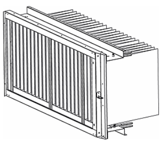
Trykkfall	4	Pa
Lengde	300	mm

Inspeksjonsdel



Trykkfall	4	Pa
Lengde	150	mm

Filter



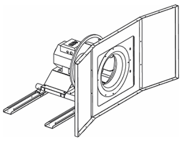
Dimensjonerende trykkfall	141	Pa
Start trykkfall/Slutt trykkfall	82/200	Pa
Lufthastighet, frontareal	2.86	m/s
Lufthastighet, filterareal	0.13	m/s
Filterklasse	F7	
Filterstørrelse	3x[490x490] + 2x[592x287] + 1x[287x287]	
Filterlengde	640	mm

systemair

Roterende varmeveksler

Data vises på tilluft		

Vifte, Kammer



Luftmengde	11250	m3/h
Eksternt trykktap	200	Pa
Trykkfall	36	Pa
Statisk trykk	627	Pa
Totaltrykk	693	Pa
Akseleffekt	2.70	kW
Viftehastighet	1487	o/min
Max viftehastighet	1735	o/min
Virkningsgrad ved statisk trykk	72.6	%
Virkningsgrad ved total trykk.	80.3	%
K-faktor	308	
Viftetype	ER56Cpro	
Direkte drift		

Motor

Motortype	IE4, PM-Motor	
Motor type-Normstr.	HPS 90 S-L	
Termosikring	Termistor	
Motoreffekt	3.90	kW
Omdreininger, merket	1700	o/min
Omdreininger, maksimum	1870	o/min
Strøm, Amp., merket	18.7	A
Virkningsgrad	92.3	%
Virkningsgrad i arbeidspunktet	91.8	%

Frekvensomformer

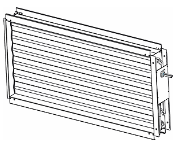
Spenning	3x230	V
Frekvensomformer kablet fra fabrikk. (23.0 Amp)	1	Stk
Effektforbruk fra hovedtilførsel, inkl frekvensomformer	3.04	kW

Frekvensomformer IP 20 er montert ved siden av viften inne i viftedelen til aggregatet. Det er benyttet skjermet kabel mellom frekvensomformer og viftemotor. Alle nødvendige parametre er lagt inn på omformer, tilpasset til motor og viftehjulets karakteristikk.

Forsyningen til aggregatet skal ha jordfeilbryter ikke mindre enn 100 mA, og beregnet for frekvensomformere.

Inspeksjonsvindu	1	Stk

Spjeld



Trykkfall	8	Pa
Spjeldblad	Standard	

Andre komponenter

Seksjoner

Produkt	Dimensjoner (Bredde x høyde x lengde)	Vekt
CS-40-0-970-1-2	1720 x 1720 x 970 mm	
CS-40-0-670-1-2	1720 x 1720 x 670 mm	
CS-40-0-970-1-2	1720 x 1720 x 970 mm	

Bunnramme

Produkt	Dimensjoner (Bredde x høyde x lengde)	Vekt
DVZ-40-3-150-2640	1720 x 150 x 2610 mm	
Bunnramme leveres med løst for sammemmontering på byggeplass før aggregat plasseres oppå.		

Fast forbindelse, 20mm LS profil

Produkt	Dimensjoner (bredde x høyde)	
Friskluft	1600x750 mm	
Tilluft	1600x750 mm	
Avtrekk	1600x750 mm	
Avkast	1600x750 mm	

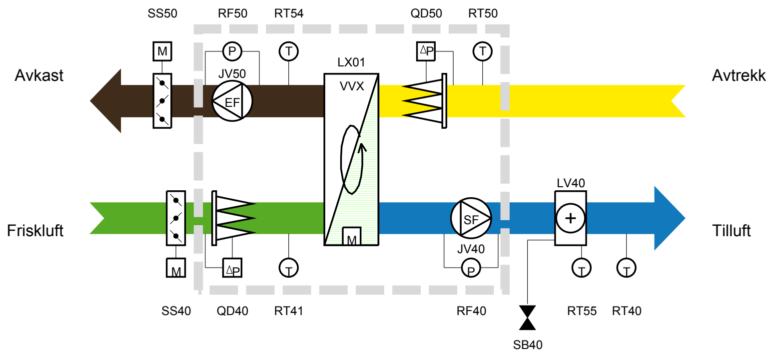
Produktkode	Beskrivelse	Antall	
18210	DVCompact-40-R-3,9-230-m/aut	1	
16569	Spjeld DVC-40 kl.3	2	
16589	Varmebatteri DVH-40-1R-7NC	1	
7433	R3015-4-S1+LR24A-SR	1	
15146	Bunnramme DV-40-R	1	
33688	Honeywell Ethernet BACnet type B-BC	1	
33691	Spjældmotor tilluft, Motor fjærretur SF24A	1	
33691	Spjældmotor fraluft, Motor fjærretur SF24A	1	
7899	TG-AH1/PT1000 Utvendig anleggsføler	1	

Integrert kontrollsystem Systemair E56

DVCompact luftbehandlingsaggregat er bygget med et komplett og fullintegrert kontrollsystem - basert på Systemair E56 regulator. Alle kablene mellom el-tavle og komponentene i DVCompact luftbehandlingsaggregatet er installert.

Aggregatet er prøvekjørt i fabrikken og har bestått en sluttest uten feil og fabrikkinnstillinger er lagret i regulatoren. Aggregatet leveres delt i tre separate deler for enkel transport. Etter at aggregatet er sammenmontert på byggeplass, må kabelskjøtene mellom aggregatdelene settes sammen.

Flytskjema



JV50	Avtrekksvifte	QD40	Trykkvakt, tilluftsfilter
JV40	Tilluftsvifte	QD50	Trykkvakt, avtrekksfilter
RT40	Temperaturføler, tilluft	SS40	Spjeld, inntak
RT50	Temperaturføler, avtrekk	SS50	Spjeld, avkast
RT41	Temperaturføler, inntak/uteluft	LX01	Varmegjenvinner
RT55	Temperaturføler, frostvakt	LV40	Vannvarmebatteri
RF40	Trykkgiver, tilluftsvifte	SB40	Motorventil
RF50	Trykkgiver, avtrekksvifte		

El-tavle og tilførsel

El-tavle med rekkeklemmer, releer, sikringer, 24V AC transformator og regulator er integrert i aggregatet foran avtrekksfilter.
Strømtilførsel skal kobles til hovedbryter på toppen eller fronten av aggregatet.
Elbatteri og DXMatic kjøledel skal ha egen strømtilførsel.

Regulator og håndterminal

Regulatoren er montert i el-tavlen. Programmering og normal betjening utføres med separat kabeltilkoblet håndterminal med knapper og display av typen - Systemair Control Panel - SCP. Den medfølgende kabelen til håndterminalen er 10m lang og håndterminalen har kapslingsklasse IP 44.

Oppstart

Oppstartsprosedyren er beskrevet i installasjons- og igangkjørings- veiledningene. Fabrikkinnstillinger vises i displayet og disse kan justeres med knappene på håndterminalen.

Tidsinnstillinger

Regulatoren har individuelle innstillinger for start, stopp og luftmengder for hver dag og i tillegg for helligdager. Regulatoren har automatisk veksling mellom sommer- og vintertid.

Temperaturregulering

Temperaturregulering er basert på avlesning fra to temperaturfølere:
- En føler i avtrekksdelen gir en gjennomsnittstemperatur fra alle rommene.
- En føler installert i tilluftskanalen.

Tilluftregulering har blitt valgt som standard reguleringsform fra fabrikk. Settpunkter for både tilluftsregulering og avtrekksregulering kan justeres med håndterminalen. Varmegjenvinner, varmebatteri og kjølebatteri reguleres i sekvens av en PI-regulator. Roterende gjenvinner reguleres trinnløst av en innebygget frekvensstyring.

Luftmengderegulering (CAV)

Luftmengden reguleres separat for tilluft og avtrekk og settpunktene for de to luftmengdene stilles separat i m3/h med håndterminalen. Det faktiske trykket over viftene blir målt med trykkgivere på viftene og luftmengden blir regnet ut vha en K-faktor i regulatoren. Luftmengdene blir sammenlignet med settpunktene i regulatoren og viftene reguleres av hver sin frekvensomformer.

Adgangsrettigheter - passord

Det er tre forskjellige adgangsnivåer.
- Grunnivå (ikke passord) - kun lesetilgang til alle innstillinger og parametere.
- Operatørnivå (passord) - lese-/skrivetilgang til alle innstillinger og parametere, men ikke tilgang til konfigurering av systemet.
- Systemnivå (høyest autoritet) (Passord) - full lese-/skrive-tilgang til alle innstillinger og parametere (også konfigurering av hele systemet)

Alarmer og sikkerhetsfunksjoner

Hvis en alarm oppstår, vil alarm LED på håndterminalen begynne å blinke. LED vil fortsette å blinke så lenge det finnes ukvitterte alarmer. Hvis det er flere aktive alarmer legges disse i en alarmliste. Alarmlisten viser type alarm, tid og dato og alarmklasse - A, B eller C:
- Alarmklasse A - stopper vifter og lukker spjeld eller setter aggregatet i en spesiell modus iht konfigurasjon.
- Alarmklasse B - er kun for å informere brukeren om en feil, men aggregatet vil fremdeles gå så normalt som mulig.
- Alarmklasse C - Kun som informasjon om at aggregatet er satt i manuell modus.

Fleksibelt system

En servicetekniker vil kunne tilpasse reguleringen videre etter brukerens ønsker:
- Luftmengdereguleringsform kan endres fra CAV til VAV, dette inkluderer andre innstillinger for regulering, settpunkter for Pa ved VAV (Variable Air Volume) regulering og montering av trykkgivere i tilluftskanal og avtrekkskanal.
- Temperaturreguleringsform kan endres fra tilluftsregulering til avtrekksregulering. Softwaren er klargjort for denne endringen.
- Istedenfor start og stopp via det innebygde urprogrammet, kan aggregatet startes når det er personer i rommet vha en PIR føler (bevegelsesdetektor) eller med et opptrekksur. (begge tilgjengelig som tilbehør fra Systemair)

Spjeld - tilluft, med spjeldmotor

Spjeldet skal geides på utsiden av aggregatet på tilluft før filter. Spjeldet åpnes og lukkes med spjeldmotor.

Spjeld - avkast, med spjeldmotor

Spjeldet skal geides på utsiden av aggregatet på avkast før filter. Spjeldet åpnes og lukkes med spjeldmotor.

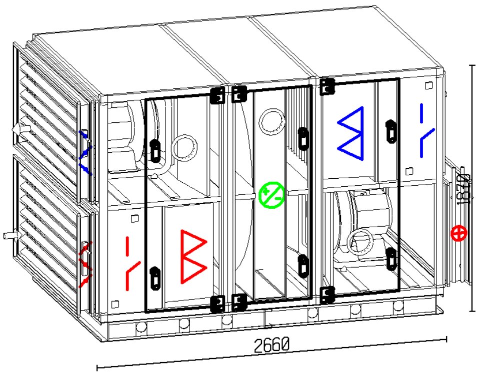
Vannvarmebatteri - frostvakt for beskyttelse varmebatteri

Kontrollsignal til shuntventil er 0-10V DC og tilførsel er 24V AC. Kabelen mellom regulator og ventilmotor er installert og testet av Systemair - klar for oppstart.
Regulatoren gir alltid et signal til shuntventil for å sikre tilstrekkelig gjennomstrømning av varmt vann for å beskytte batteriet mot frost. Temperaturen på returvannet sendes til regulatoren fra en frostføler. Beskyttelsessekvens er aktiv også når aggregatet er stoppet, dette gir en ekstra beskyttelse av batteriet. Hvis returvanntemperaturen faller under settpunktet for frostvakt, stopper viftene og spjeldene stenger.

Oversikt for aggregat nr: 2

DVCompact 40

Ordrenummer **0003410820**
Prosjekt **Øya Barnehage Trondheim 2015**
Anlegg **360.02 /**



DVCompact luftbehandlingsaggregat med fullintegrert automatikk ferdig kablet fra fabrikk

Luft-/viftedata	Tilluft	Fraluft	
Luftmengde (1,205 kg/m³)	11250	11250	m3/h
Lufthastighet i aggregat	2.45	2.45	m/s
Eksternt trykktap	200	200	Pa
Viftehastighet	1499	1487	o/min
Motor	3.90	3.90	kW
Spenning	3x230	3x230	V
Strøm, Amp., merket	18.7	18.7	A

Aggregatdata		
Aggregatbredde		1720 mm
Vekt		960 kg
Filter		Tilluft F7 - Fraluft, avtrekk F7
Varmeveksler		79.9 %
SFPv, ved rent filter inklusiv frekvensomformer		1.81 kW/(m³/s)
Varmebatteri	Luft	32.7 kW - 13.5/22.0°C
	Vann	60/40°C - 10.8 kPa - 0.40 l/s - 26.9mm / 26.9mm Rørtilkobling

Lydeffektnivå	Tilluft, innblåsning	Uteluft, inntak	Fraluft, avkast	Fraluft, avtrekk	Lyd, omgivelser
Total	84 dB(A)	69 dB(A)	86 dB(A)	69 dB(A)	59 dB(A)

systemair

Systemair AS

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Faks : +47 51969799
www.systemair.no
mailbox@systemair.no

Arbeidspunkt 2

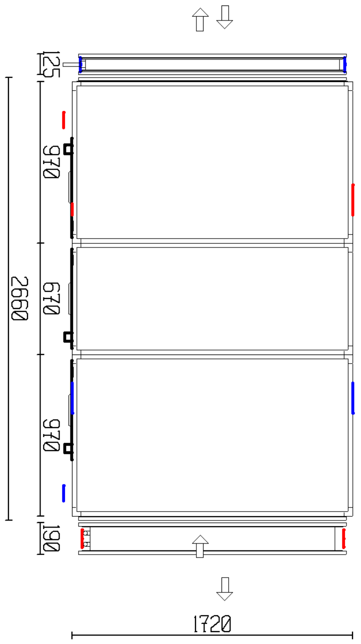
DVCompact luftbehandlingsaggregat med fullintegrert automatikk ferdig kablet fra fabrikk

Luft-/viftedata	Tilluft	Fraluft	
Luftmengde (1,205 kg/m³)	9000	9000	m3/h
Lufthastighet i aggregat	1.96	1.96	m/s
Eksternt trykktap	200	200	Pa
Viftehastighet	1302	1293	o/min
Motor	3.90	3.90	kW
Spenning	3x230	3x230	V
Strøm, Amp., merket	18.7	18.7	A

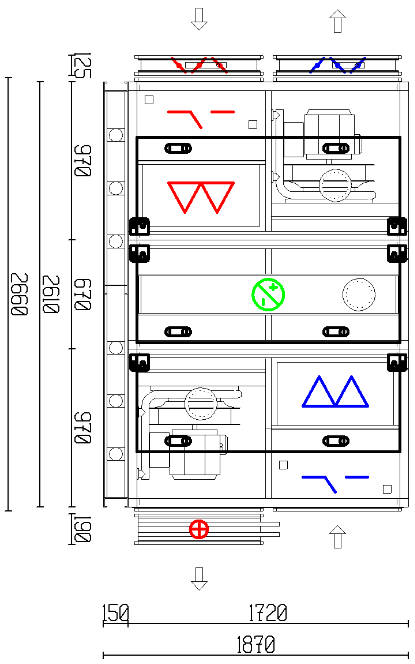
Aggregatdata		
Aggregatbredde		1720 mm
Vekt		960 kg
Filter		Tilluft F7 - Fraluft, avtrekk F7
Varmeveksler		83.5 %
SFPv, ved rent filter inklusiv frekvensomformer		1.46 kW/(m³/s)
Varmebatteri	Luft	21.3 kW - 15.1/22.0°C
	Vann	60/40°C - 5.0 kPa - 0.26 l/s - 26.9mm / 26.9mm Rørtilkobling

Lydeffektnivå	Tilluft, innblåsning	Uteluft, inntak	Fraluft, avkast	Fraluft, avtrekk	Lyd, omgivelser
Total	80 dB(A)	65 dB(A)	82 dB(A)	65 dB(A)	55 dB(A)

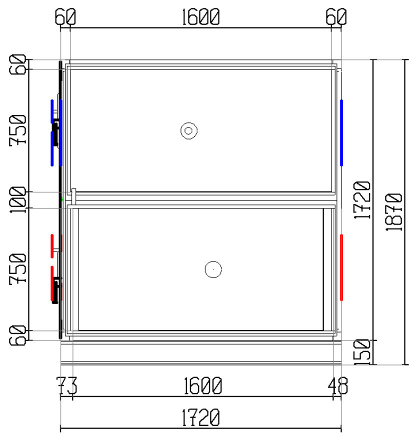
Plantegning



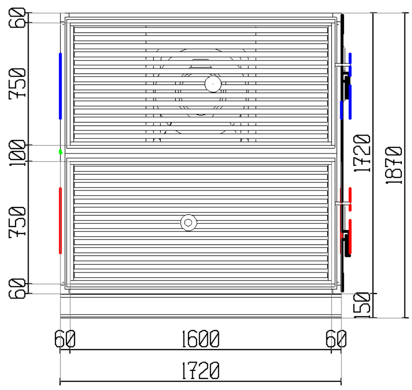
Inspeksjonsside



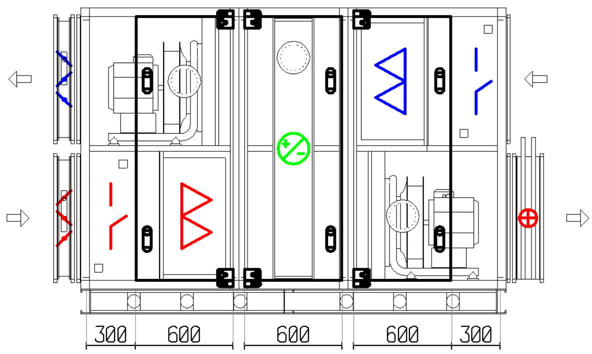
Høyre gavl



Venstre gavl



Mål på dører og panelser



Teknisk spesifikasjon

Aggregat

Frekvensbånd [Hz] Lydeffektnivå	63 [dB]	125 [dB]	250 [dB]	500 [dB]	1K [dB]	2K [dB]	4K [dB]	8K [dB]	Total [dB(A)]
Tilluft, innblåsning	73	83	82	82	82	71	69	68	84
Uteluft, inntak	64	75	74	66	63	53	48	47	69
Fraluft, avkast	74	85	83	83	84	74	72	72	86
Fraluft, avtrekk	64	75	74	66	63	53	48	47	69
Lyd, omgivelser	62	72	60	51	50	43	40	42	59

Automatikk

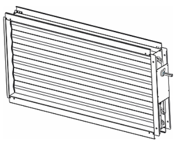
Sprog i controler menu	Norsk	
Ekstern kommunikasjon	Honeywell Ethernet BACnet type B-BC	
Viftestyring	VAV	
Spjeldmotor, tilluft	Motor fjærretur	
Spjeldmotor, avkast	Motor fjærretur	
Frostbeskyttelse, varmebatteri	Utvendig anleggsføler	
Batteri konfigurasjon	Varmebatteri	
Ventil for varme	3-veis ventil, Kvs 4.00, DN15 Innvendig koblet	
Trykkfall	13	kPa

Hovedtilførsel til automatikken

Tavledata	Spenning	3x230V + PE 50	Hz
	Maks. ik	16	kA
	Min. ik	650	A
	Maks. sikring	60	A
	Min. sikring	40	A

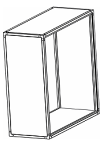
Tilluftsaggregat består av

Spjeld



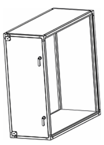
Trykkfall	8	Pa
Spjeldblad	Standard	

Tomdel



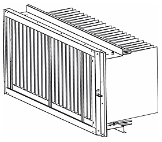
Trykkfall	4	Pa
Lengde	300	mm

Inspeksjonsdel



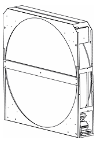
Trykkfall	4	Pa
Lengde	150	mm

Filter



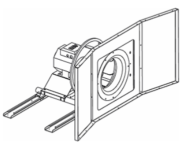
Dimensjonerende trykkfall	141	Pa
Start trykkfall/Slutt trykkfall	82/200	Pa
Lufthastighet, frontareal	2.86	m/s
Lufthastighet, filterareal	0.13	m/s
Filterklasse	F7	
Filterstørrelse	3x[490x490] + 2x[592x287] + 1x[287x287]	
Filterlengde	640	mm

Roterende varmeveksler



	Tilluft	Avtrekk	
Luftmengde	11250	11250	m3/h
Trykkfall	234	234	Pa
Lufttemperatur, før/etter	-20.0/13.5	22.0/-9.3	°C
Relativ luftfuktighet, før/etter	90/31	30/97	%
Effekt	148.51		kW
Temperaturvirkningsgrad	79.9		%
Varmegjenvinner type		Kondensatorvarmeveksler	
Temperaturvirkningsgrad		Høyeffekt	
Drivsystem for rotor		Variabel drift	
Elektriske data		1x230V, 100W, 1.3Amp	
Inspeksjonsvindu		1	Stk

Vifte, Kammer



Luftmengde	11250	m3/h
Eksternt trykktap	200	Pa
Trykkfall	36	Pa
Statisk trykk	646	Pa
Totaltrykk	712	Pa
Akseleffekt	2.77	kW
Viftehastighet	1499	o/min
Max viftehastighet	1735	o/min
Virkningsgrad ved statisk trykk	72.8	%
Virkningsgrad ved total trykk.	80.3	%
K-faktor	308	
Viftetype	ER56Cpro	
Direkte drift		

Motor

Motortype	IE4, PM-Motor	
Motor type-Normstr.	HPS 90 S-L	
Termosikring	Termistor	
Motoreffekt	3.90	kW
Omdreininger, merket	1700	o/min
Omdreininger, maksimum	1870	o/min
Strøm, Amp., merket	18.7	A
Virkningsgrad	92.3	%
Virkningsgrad i arbeidspunktet	91.8	%

Frekvensomformer

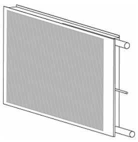
Spenning	3x230	V
Frekvensomformer kablet fra fabrikk. (23.0 Amp)	1	Stk
Effektforbruk fra hovedtilførsel, inkl frekvensomformer	3.13	kW

systemair

Frekvensomformer IP 20 er montert ved siden av viften inne i viftedelen til aggregatet. Det er benyttet skjermet kabel mellom frekvensomformer og viftemotor. Alle nødvendige parametre er lagt inn på omformer, tilpasset til motor og viftehjulets karakteristikk.
Forsyningen til aggregatet skal ha jordfeilbryter ikke mindre enn 100 mA, og beregnet for frekvensomformere.

Inspeksjonsvindu	1	Stk

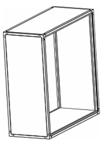
Varmebatteri MAX, Væske



Luftmengde	11250	m3/h
Trykkfall	19	Pa
Lufttemperatur, inn/ut	13.5/22.0	°C
Relativ luftfuktighet, før/etter	31/18	%
Effekt	32.74	kW
Lufthastighet	2.60	m/s
Væskemedie	Vann	
Væsketemperatur inn/ut	60.0/40.0	°C
Væskemengde	0.40	l/s
Væske trykktap	10.8	kPa
Væskehastighet	0.79	m/s
Batterivolum	3.5	l
Anslutningsside	Service side	
Tilkoblingsmål inn/ut	26.9mm / 26.9mm	
Rørmateriale	Cu	
Lamellmateriale	Al	
Lamellavstand	2.1	mm
Rørrader	1	
Overflate areal	22.9	m2
Vekt, batteri	14.4	kg
Batteri type	M 25x22-3/8 CS 30 T 1 R 1600 A 2.1 P 7 NC	

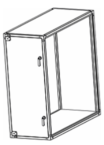
Fraluftsaggregat består av

Tomdel



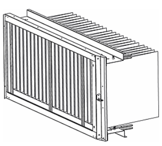
Trykkfall	4	Pa
Lengde	300	mm

Inspeksjonsdel



Trykkfall	4	Pa
Lengde	150	mm

Filter

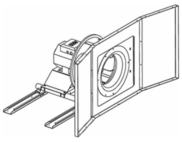


Dimensjonerende trykkfall	141	Pa
Start trykkfall/Slutt trykkfall	82/200	Pa
Lufthastighet, frontareal	2.86	m/s
Lufthastighet, filterareal	0.13	m/s
Filterklasse	F7	
Filterstørrelse	3x[490x490] + 2x[592x287] + 1x[287x287]	
Filterlengde	640	mm

Roterende varmeveksler

Data vises på tilluft		

Vifte, Kammer



Luftmengde	11250	m3/h
Eksternt trykktap	200	Pa
Trykkfall	36	Pa
Statisk trykk	627	Pa
Totaltrykk	693	Pa
Akseleffekt	2.70	kW
Viftehastighet	1487	o/min
Max viftehastighet	1735	o/min
Virkningsgrad ved statisk trykk	72.6	%
Virkningsgrad ved total trykk.	80.3	%
K-faktor	308	
Viftetype	ER56Cpro	
Direkte drift		

Motor

Motortype	IE4, PM-Motor	
Motor type-Normstr.	HPS 90 S-L	
Termosikring	Termistor	
Motoreffekt	3.90	kW
Omdreininger, merket	1700	o/min
Omdreininger, maksimum	1870	o/min
Strøm, Amp., merket	18.7	A
Virkningsgrad	92.3	%
Virkningsgrad i arbeidspunktet	91.8	%

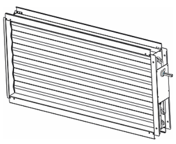
Frekvensomformer

Spenning	3x230	V
Frekvensomformer kablet fra fabrikk. (23.0 Amp)	1	Stk
Effektforbruk fra hovedtilførsel, inkl frekvensomformer	3.04	kW

Frekvensomformer IP 20 er montert ved siden av viften inne i viftedelen til aggregatet. Det er benyttet skjermet kabel mellom frekvensomformer og viftemotor. Alle nødvendige parametre er lagt inn på omformer, tilpasset til motor og viftehjulets karakteristikk.
Forsyningen til aggregatet skal ha jordfeilbryter ikke mindre enn 100 mA, og beregnet for frekvensomformere.

Inspeksjonsvindu	1	Stk

Spjeld



Trykkfall	8	Pa
Spjeldblad	Standard	

Andre komponenter

Seksjoner

Produkt	Dimensjoner (Bredde x høyde x lengde)	Vekt
CS-40-0-970-1-2	1720 x 1720 x 970 mm	
CS-40-0-670-1-2	1720 x 1720 x 670 mm	
CS-40-0-970-1-2	1720 x 1720 x 970 mm	

Bunnramme

Produkt	Dimensjoner (Bredde x høyde x lengde)	Vekt
DVZ-40-3-150-2640	1720 x 150 x 2610 mm	
Bunnramme leveres med løst for sammemmontering på byggeplass før aggregat plasseres oppå.		

Fast forbindelse, 20mm LS profil

Produkt	Dimensjoner (bredde x høyde)	
Friskluft	1600x750 mm	
Tilluft	1600x750 mm	
Avtrekk	1600x750 mm	
Avkast	1600x750 mm	

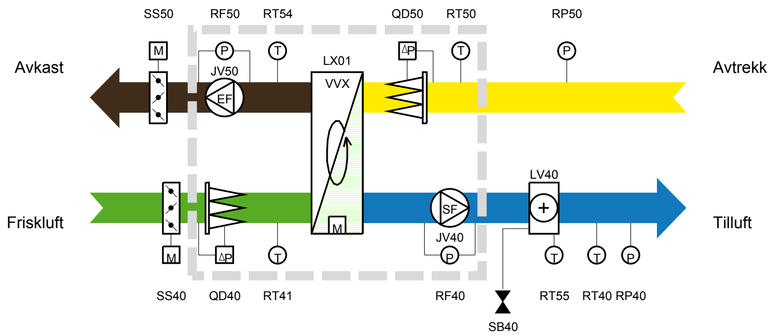
Produktkode	Beskrivelse	Antall	
18210	DVCompact-40-R-3,9-230-m/aut	1	
16569	Spjeld DVC-40 kl.3	2	
16589	Varmebatteri DVH-40-1R-7NC	1	
7433	R3015-4-S1+LR24A-SR	1	
15146	Bunnramme DV-40-R	1	
33688	Honeywell Ethernet BACnet type B-BC	1	
16396	VAV	1	
33691	Spjældmotor tilluft, Motor fjærretur SF24A	1	
33691	Spjældmotor fraluft, Motor fjærretur SF24A	1	
7899	TG-AH1/PT1000 Utvendig anleggsføler	1	

Integrert kontrollsystem Systemair E56

DVCompact luftbehandlingsaggregat er bygget med et komplett og fullintegrert kontrollsystem - basert på Systemair E56 regulator. Alle kablene mellom el-tavle og komponentene i DVCompact luftbehandlingsaggregatet er installert.

Aggregatet er prøvekjørt i fabrikken og har bestått en sluttest uten feil og fabrikkinnstillinger er lagret i regulatoren. Aggregatet leveres delt i tre separate deler for enkel transport. Etter at aggregatet er sammenmontert på byggeplass, må kabelskjøtene mellom aggregatdelene settes sammen.

Flytskjema



JV50	Avtrekksvifte	QD40	Trykkvakt, tilluftsfilter	RP40	VAV trykkgiver tilluft
JV40	Tilluftsvifte	QD50	Trykkvakt, avtrekksfilter	RP50	VAV trykkgiver avtrekk
RT40	Temperaturføler, tilluft	SS40	Spjeld, inntak		
RT50	Temperaturføler, avtrekk	SS50	Spjeld, avkast		
RT41	Temperaturføler, inntak/uteluft	LX01	Varmegjenvinner		
RT55	Temperaturføler, frostvakt	LV40	Vannvarmebatteri		
RF40	Trykkgiver, tilluftsvifte	SB40	Motorventil		
RF50	Trykkgiver, avtrekksvifte				

El-tavle og tilførsel

El-tavle med rekkeklemmer, releer, sikringer, 24V AC transformator og regulator er integrert i aggregatet foran avtrekksfilter.
Strømtilførsel skal kobles til hovedbryter på toppen eller fronten av aggregatet.
Elbatteri og DXMatic kjøledel skal ha egen strømtilførsel.

Regulator og håndterminal

Regulatoren er montert i el-tavlen. Programmering og normal betjening utføres vha separat kabeltilkoblet håndterminal med knapper og display av typen - Systemair Control Panel - SCP. Den medfølgende kabelen til håndterminalen er 10m lang og håndterminalen har kapslingsklasse IP 44.

Oppstart

Oppstartsprosedyren er beskrevet i installasjons- og igangkjørings- veiledningene. Fabrikkinnstillinger vises i displayet og disse kan justeres med knappene på håndterminalen.

Tidsinnstillinger

Regulatoren har individuelle innstillinger for start, stopp og luftmengder for hver dag og i tillegg for helligdager. Regulatoren har automatisk veksling mellom sommer- og vintertid.

Temperaturregulering

Temperaturregulering er basert på avlesning fra to temperaturfølere:
- En føler i avtrekksdelen gir en gjennomsnittstemperatur fra alle rommene.
- En føler installert i tilluftskanalen.
- Tilluftregulering har blitt valgt som standard reguleringsform fra fabrikk. Settpunkter for både tilluftsregulering og avtrekksregulering kan velges i håndterminalen. Varmegjenvinner, varmebatteri og kjølebatteri reguleres i sekvens av en PI-regulator. Roterende gjenvinner reguleres trinnløst av en innebygget frekvensstyring.

Trykkregulering (VAV)

Luftmengden reguleres separat for tilluft og avtrekk. Settpunktene for ønsket trykk stilles inn separat på håndterminalen. Det faktiske totale trykket blir målt av trykkgivere montert i tilluft- og avtrekkskanalene. Disse verdiene blir sammenlignet med settpunktene og hastigheten på begge viftene justeres samtidig med separate frekvensomformere.

Adgangsrettigheter - passord

Det er tre forskjellige adgangsnivåer.
- Grunnivå (ikke passord) - kun lesetilgang til alle innstillinger og parametere.
- Operatørnivå (passord) - lese-/skrivetilgang til alle innstillinger og parametere, men ikke tilgang til konfigurering av systemet.
- Systemnivå (høyest autoritet) (Passord) - full lese-/skrive- tilgang til alle innstillinger og parametere (også konfigurering av hele systemet)

Alarmer og sikkerhetsfunksjoner

Hvis en alarm oppstår, vil alarm LED på håndterminalen begynne å blinke. LED vil fortsette å blinke så lenge det finnes ukvitterte alarmer. Hvis det er flere aktive alarmer legges disse i en alarmliste. Alarmlisten viser type alarm, tid og dato og alarmklasse - A, B eller C:
- Alarmklasse A - stopper vifter og lukker spjeld eller setter aggregatet i en spesiell modus iht. konfigurasjon.
- Alarmklasse B - er kun for å informere brukeren om en feil, men aggregatet vil fremdeles gå så normalt som mulig.
- Alarmklasse C - Kun som informasjon om at aggregatet er satt i manuell modus.

Fleksibelt system

En servicetekniker vil kunne tilpasse reguleringen videre etter brukerens ønsker:
- Luftmengdereguleringsform kan endres fra VAV til CAV, dette inkluderer andre innstillinger for regulering, settpunkter for m3/t ved CAV (Constant Air Volume) regulering og montering av trykkgivere på tilluftsvifte og avtrekksvifte.
- Temperaturreguleringsform kan endres fra tilluftsregulering til avtrekksregulering. Softwaren er klargjort for denne endringen.

Istedenfor start og stopp via det innebygde urprogrammet, kan aggregatet startes når det er personer i rommet vha en PIR føler (bevegelsesdetektor) eller med et opptrekksur. (begge tilgjengelig som tilbehør fra Systemair).

Spjeld - tilluft, med spjeldmotor

Spjeldet skal geides på utsiden av aggregatet på tilluft før filter. Spjeldet åpnes og lukkes med spjeldmotor.

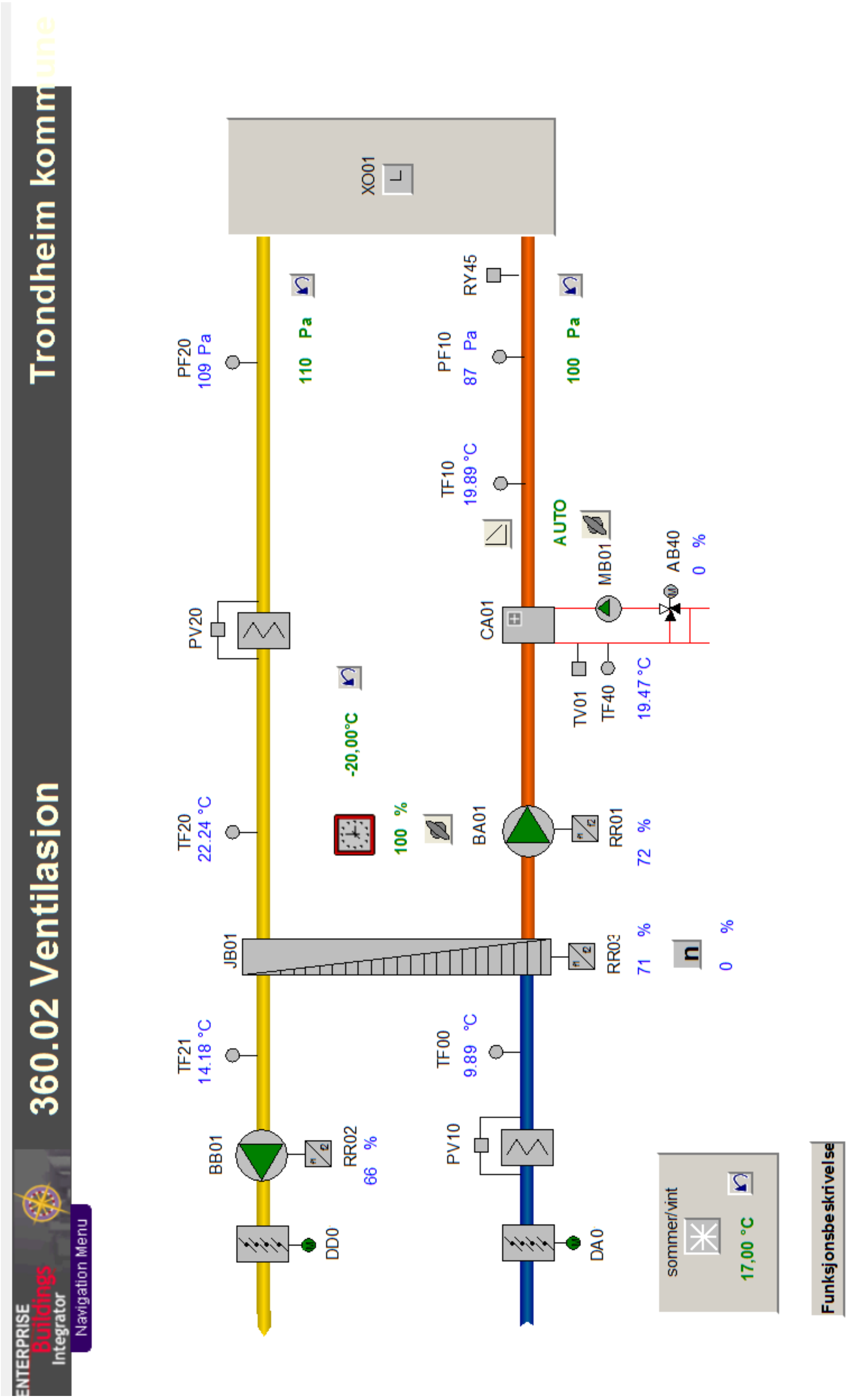
Spjeld - avkast, med spjeldmotor

Spjeldet skal geides på utsiden av aggregatet på avkast før filter. Spjeldet åpnes og lukkes med spjeldmotor.

Vannvarmebatteri - frostvakt for beskyttelse varmebatteri

Kontrollsignal til shuntventil er 0-10V DC og tilførsel er 24V AC. Kabelen mellom regulator og ventilmotor er installert og testet av Systemair - klar for oppstart.
Regulatoren gir alltid et signal til shuntventil for å sikre tilstrekkelig gjennomstrømning av varmt vann for å beskytte batteriet mot frost. Temperaturen på returvannet sendes til regulatoren fra en frostføler. Beskyttelsessekvens er aktiv også når aggregatet er stoppet, dette gir en ekstra beskyttelse av batteriet. Hvis returvanntemperaturen faller under settpunktet for frostvakt, stopper viftene og spjeldene stenger.

Appendix D



Trondheim kommune
360.02 Ventilasjon
ENTERPRISE
Buildings
Integrator
Navigation Menu
XO01
PF20
109 Pa
110 Pa
RY45
PF10
87 Pa
100 Pa
TF10
19.89 °C
AUTO
MB01
AB40
0 %
PV20
CA01
TV01
TF40
19.47 °C
-20,00°C
TF20
22.24 °C
100 %
BA01
RR01
72 %
JB01
RR03
71 %
0 %
TF21
14.18 °C
TF00
9.89 °C
BB01
RR02
66 %
PV10
DD0
DA0
sommer/vint
17,00 °C
Funksjonsbeskrivelse