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Airflow Measurements for Air Handling Units

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Executive summary

This technical report is intended as a preliminary study on airflow measurements as they are basic for the evaluation of the efficiency of air handling units (AHUs) in ventilation systems. The report targets readers intending to measure the field performance of the AHU. Its goal is to study solutions for airflow measurement of ducted air in ventilation systems.

The original purpose of this project was to measure the field efficiency of rotary heat recovery in buildings. For this purpose, measurements of the temperatures, airflows and relative humidity in all the ducts very close to the air handling unit were needed to calculate the field efficiency. The initial goal was to measure in nine different buildings and give some inputs regarding the discrepancy between rated and measured efficiency in buildings of different sizes and used for different purposes. However, it was soon unveiled the lack of suitable equipment to measure airflows in the ducted air for AHUs. For our purpose, it was necessary to have equipment that could measure accurately and that could be moved among the different buildings. It also was unveiled the lack of existing standards for measurements in the field of airflows in a reliable way. Thus, the purpose of this project was shifted to an overview and investigation of the existing knowledge and equipment for field airflow measurements, and these are the results presented in this report.

Heating, ventilation and air-conditioning (HVAC) play an essential role in achieving the desired indoor climate. The energy used by the HVAC in the building sector accounts for a large proportion of the total energy use in most countries around the world. Although extensive studies on energy conservation measures and innovations for the HVAC systems have been performed and implemented, the field performance of these measures in practice tends to be easily overlooked. The real performance can be considerably reduced from improper installation, system fault and lack of commissioning and in-field verification. IEA Annex 34 has shown that energy savings of 20-30 % can be reached by re-commissioning building HVAC system (Jagpal 2006). Monitoring airflow rates in ducted air in a ventilation system can indicate problems in the installation and operation of an AHU. The field performance of the AHU could imply whether the building energy use is in accordance with the design intent or is detrimentally affected.

However, it was not an easy task to perform airflow measurements in operating AHU. Most of the measurement recommendations of standards are defined for lab conditions where the air velocity profile in the duct is fully developed. In practice, it is difficult to follow the standards' requirements in most cases. Operating systems have bends or devices that disturb the flow, thus it is difficult to find stretches of ducts long enough or to get access to the straight parts of the ducts close to the AHU. Consequently, the air velocity profile at the measuring points is usually not fully developed which increases the measurement complexity and reduces the accuracy. Especially measuring the airflow rate close enough to the heat exchanger (to avoid branching of flows) was found to be a problem due to turbulence and distorted velocity profiles.

During this project, scientific literature, researchers, equipment suppliers and contractors around the world have been contacted in the search for suitable solutions. However, none of the sources had perfect solutions for field airflow measurements. This report provides an overview of relevant studies and standards concerning laboratory measurements in a building-configuration-manner of rotary heat exchangers. Additionally, it focuses on testing currently available methods and devices that could be used for reliable measurement of airflow rates. Airflow measurement techniques investigated in this project are summarized herein as follows:

1. Pressure differential technique

The typical airflow rate measuring device employs a pressure differential method (e.g. Pitot tube, Venturis or orifice plate) to determine the airflow. This method is relatively accurate, but it is fragile, expensive and introduces an additional pressure penalty, and their sensitivity

becomes low for low airflow rates (Yu, Li et al. 2011). Additionally, this method is mostly used for laboratory test instead of field test due to the extra pressure loss and the strong intervention to the practical systems. In this project, the orifice plate method to determine the airflow rate is regarded as the reference value considering its high accuracy and reliability.

2. Velocity traversal technique

Another method is to use velocity traversal which needs to place one or more velocity sensors to various designated positions in the duct to obtain the area-averaged velocity. The flow regime, the duct diameter and velocity profile in the duct influence the number of measuring points and the measuring accuracy. The turbulence caused by the bends, system devices such as fans, filters, coils and limited straight ductwork in the AHU may lead to inaccurate velocity results and thus a larger number of required measuring points, which also means more labour work. Moreover, the traverse method cannot capture the airflow variations. The measured airflows are assumed constant within the time period of measuring velocities in the different points across the cross-section of the duct. The sensor probe will interfere with the velocity profile which in turn decreases the measurement accuracy.

3. Ultrasonic airflow measurement technique

The third method is using ultrasonic airflow measurement device, which is considered as a superior alternative to the pressure differential and velocity traverse methods in this project. The measurement accuracy of the ultrasonic device is proven almost equally good as pressure differential device (e.g. orifice plate) without extra pressure loss and interference on velocity profile. The ultrasonic method can monitor constant and time-dependent airflows. This ultrasonic measurement device is able to measure a wide range of air velocities without degrading the sensitivity due to its linear response (time difference between the propagation of the ultrasonic waves) to flow velocity change (Olmos 2004). This especially benefits the low flow velocity measurement. The difficulty of connecting pre-fabricated ultrasonic flow measurement module with existing ductwork in the practical system may impede its application in existing AHU.

4. Tracer gas technique

Tracer gas method is widely used in ventilation to determine the airflow rates and air distributions. The difference between the tracer gas and aforementioned three methods is that the tracer gas method directly determines the airflow rate while the other methods measure air velocity firstly and calculate the airflow rate by interoperating with cross-section area. The tracer gas technique in the AHU also enables to reveal leakages and airflow recirculation in rotary heat wheels. However, the poor mixing of the tracer with the airflow in the AHU causes wrong airflow rates, which almost always occurs in the practical system. In addition, this method demands proper preparation and specific knowledge for interpretation of the tracer concentration. It is not intuitive to obtain the airflow and difficult to perform compared to other techniques. It has a larger cost that the previous measurement techniques.

The products and measuring techniques presented in this report are suitable for measuring airflow rates in ventilation ducts, provided they are installed according to their requirements. Based on this study, we have not found an ideal portable equipment available for measuring airflow rates in AHU as they all require minimum straight duct lengths to achieve fully developed flow profiles or good mixing between tracer gas and air. Sufficient lengths of straight ducts are normally not available in practice. When the real efficiency of the heat or energy recovery in the AHU is to be measured, the accuracy and reliability of the measurement are expected to be lower than in the laboratory measurements. The measuring equipment must be chosen based on the characteristics of each ventilation system, especially ductwork diameter and configuration, available space to perform the measurements and minimum straight duct lengths before and after disturbances. This study did not find one single device that could be universally used for all different systems.

Further investigation on this issue and new development for airflow measurement devices are encouraged, so that the real performance of rotary heat exchangers can be easily verified and continuously monitored when installed in buildings. The ideal field measuring equipment should be able to accurately measure airflow rates, temperatures and relative humidity. The measurements should be carried out automatically (requiring little manual labour) and the results should be logged and easily exported to a computer for analysis. The device should also be of minimal disturbance for the airflow, as disturbances will induce pressure drop and increase energy use for the fans.

Detailed information for the selected measuring techniques is included in the table below.

Airflow measurement method		**Velocity traverse**	**Ultrasonic airflow measurement**	**Pressure differential**	**Tracer gas**
Selected measuring equipment in this project		VelociCalc 9565-P	Lindab UltraLink	Orifice plate	CO_2 Tracer gas
Velocity	Range	0-50 m/s	0.2-15 m/s	Reynolds number ≤ 5000	NA
	Uncertainty	±3 % of reading or ±0.015 m/s, whichever is greater	±5 % of reading or ±X* l/s, whichever is greater	±0.5 % in general (Emerson Process Management 2010)	Airflow measurement uncertainty mainly depends on the mixing level
	Resolution	0.01 m/s	NA	NA	NA
Temperature	Range	-10 to 60 °C	-10 to 50 °C	NA	NA
	Uncertainty	±0.3 °C	±1.0 °C	NA	NA
	Resolution	0.1 °C	NA	NA	NA
Relative humidity		Yes, see Table 3	No	No	No
Automatic logging		Yes (in 1 point)	Yes	No (possible with advanced manometers)	For tracer concentration, yes
Duct size		25 to 12700 mm	Ø100-Ø315	Ø50-Ø1000 (according to ISO 5167-2:2003)	Any size
Strength		Easy to implement and understand	Accurate, low uncertainty for low air velocity, intuitive, no interference on the flow	Accurate	Able to determine the main and parasitic airflows at the same time, relatively accurate
Drawback		Measurement uncertainty is high for low velocity and developing flow, require manual labour	Difficult to install in the existing duct for a pre-fabricated ultrasonic airflow measurement device	Create extra pressure loss Fragile, expensive, not suitable for field tests	Need good mixing which is almost impossible in AHU, need specialized knowledge to calculate the airflows

*X equals the diameter in dm, for instance Ø100 => 1 l/s and Ø200 => 2 l/s

How to read this report?
The report is organized in a workflow as study questions → state-of-the-art → summary for testing equipment and measurements → discussion and conclusion → suggestion for future work → appendix.

The executive summary presents the main findings and the available airflow measurement techniques investigated of this project concisely.

The background for the project, definitions, relevant standards with test procedures and equations for calculating and measuring airflow rates in ventilation ducts and efficiency of the heat exchanger is presented in the *Introduction* section.

The *Literature study* section reviewed the state-of-the-art of the existing relevant airflow measurement technics for field measurements in the open literature.

The working principles and features of four different measuring technologies are addressed in *Measurement techniques.*

Available measuring equipment and their specifications for this project can be found in the section of *Measurement equipment for airflow rates in AHU*. Laboratory and field measurement performed in the project using four different techniques are available in the *Lab and field measurements* section. These two sections which include the detailed descriptions of the used measuring equipment and the measurements are placed in Appendix B and C, respectively. A summary of these two sections is available in chapter 4.

The discussion, conclusions and suggestions for future work are given at the end of the report.

Trondheim, 28.11.2018

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Table of Contents

1 Introduction

This chapter presents the background for the project, definitions, relevant standards with test procedures and equations for calculating and measuring airflow rates in ventilation ducts and ultimately for calculating heat exchanger's efficiency.

1.1 Background

The building sector represents a large share of the total energy use in Norway and thus contributes significantly to the country's greenhouse gas emissions. This has led to an increased focus on energy efficient buildings, which is reflected in the stricter building regulations. For new residential buildings, mechanical ventilation with heat recovery is, in practice, required in Norway.

In a heat recovery unit, heat is transferred from the extract air to the supply air in order to reduce the heat loss due to heating the ventilation air for cold climate. In dwellings without heat recovery or other forms of ventilation air preheating (for instance in older dwellings based on supplying air through vents in the walls or over windows), the occupants may experience cold draft from the outside air for many days of the year. Fans are often used in such buildings for sucking the used air out (exhaust ventilation) but most of the supply happens through windows or leakages. The ventilation rates are not well controlled in this situation, which may cause reduced thermal comfort. Therefore, using mechanical ventilation and heat recovery enables a better control of the thermal comfort and health IAQ.

Table 1 presents relevant energy requirements for new buildings built according to the energy measure (Norwegian: energitiltak) method. The requirements are stricter compared to previous regulations, and the same is the case for the energy frame calculation method. Heat recovery units need to have higher efficiencies than earlier to satisfy the requirements.

Table 1 Relevant requirements for ventilation from the energy measure method in the Norwegian building regulation, TEK 17. (DIBK 2017)

Energy measure	**Single-family houses and apartment blocks**
Annual temperature efficiency for heat recovery in air handling unit (%)	≥ 80
Specific fan power (SFP) in air handling units [kW/(m³/s)]	≤ 1.5
Air changes per hour (ACH) at 50 Pa pressure difference (h^{-1})	≤ 0.6

Measurements in occupied buildings often show much lower values than rated efficiencies. Heide (2012) collected several examples with deviations up to 37 % compared to the rated efficiency. Lassen, Fylling et al. (2009) refers to several studies in Norway, Switzerland and Sweden where the effective temperature efficiency for rotary heat exchangers typically is 10-20 % lower than nominal temperature efficiency. Lower efficiency is not primarily caused by incorrectly documented performance, but rather due to adjustment, installation and operation of the air handling units that deviate from the ideal conditions used during the product documentation (mainly due to recirculation and unbalanced airflows). Additionally, the testing points in the laboratory condition are constant, in contrast to occupied buildings where the outdoor and indoor conditions are dynamic with time. The airflow in AHUs will change with the system control for different heating or cooling loads, indoor air quality, indoor moisture level and frost protection. It is important to develop methods and protocols for rating and testing the real performance of the heat recovery in AHU for existing buildings as it significantly influences the thermal comfort and energy use. The big deviation between the unexpected practical and nominal performance may also result in incorrect design and estimation of the heat/energy recovery in AHU. The field performance of the recovery devices could be also an indicator to detect the malfunction and system fault.

To determine or monitor the field performance of heat/energy recovery systems, the airflow measurement in the AHU ducts is of importance as a prerequisite to calculate the performance of the heat recovery and to shed light on internal leakages in the heat recovery. However, measuring technics for field measurement are less studied than for the laboratories. This report has the intention to review and compare the relevant existing studies, standards, technologies, equipment for the efficiency of rotary heat recovery and the airflow measurement techniques in AHU.

1.2 Efficiency of rotary heat exchangers

As previously mentioned, the Norwegian building regulation, TEK, sets requirements regarding annual temperature efficiency. TEK currently refers to NS 3031:2014 *Calculation of energy performance of buildings – Method and data* for energy calculations and validation according to the building regulations. This standard has however been withdrawn from Standard Norge, so in this report, the definitions and calculation methods are instead collected from the supplement standard: SN TS 3031:2016 *Energy performance of buildings - Calculation of energy needs and energy supply*. The temperature efficiency of a heat exchanger at a constant and balanced supply and extract airflow rate is given by equation (1) and the reference points for temperature can be seen in Figure 1. It should be noted that equation (1) is only valid for ideally balanced ventilation without any internal and external leakages from/to the rotary heat wheel. In other words, this equation would most likely not be suitable for measuring the efficiency in the field as the airflows are seldom completely balanced and internal and external leakages occur.

$$\eta_T = \frac{\theta_2 - \theta_1}{\theta_3 - \theta_1} \tag{1}$$

where:

θ_1 is the air temperature before the heat exchanger on the supply side, in °C
θ_2 is the air temperature after the heat exchanger on the supply side, in °C
θ_3 is the air temperature before the heat exchanger on the extract side, in °C

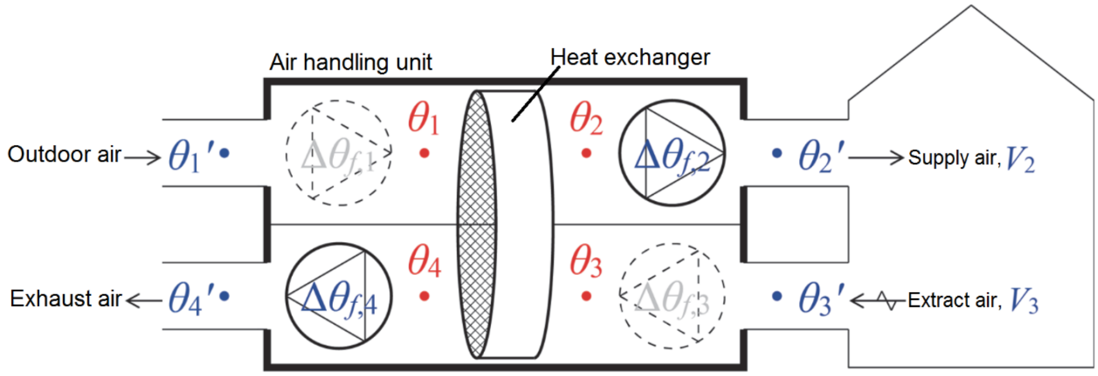


Figure 1 Reference points for temperatures in an air handling unit. Translated from SN TS 3031:2016. (Standard Norge 2017)

According to SN TS 3031:2016, suppliers of AHUs must document temperature efficiency and specific fan power (SFP) in five operating points; at 0,2 x V_{dim}, 0,4 x V_{dim}, 0,6 x V_{dim}, 0,8 x V_{dim} and V_{dim}. For large AHUs, the suppliers must document the heat exchanger's temperature efficiency at outdoor temperatures of +5 °C and without condensation, while for small AHUs for residential buildings, they must document the temperature efficiency at +7 °C without condensation.

The standard also presents equations for correcting the temperature efficiency for unbalanced airflow rates and frost control. If the extract flow rate is higher than the supply flow rate, the temperature efficiency in time step *i* must be corrected according to equation (2).

$$\eta'_{T,i} = \eta_{T,i} \times \min(1\ ;\ \frac{\dot{V}_{3,i}}{\dot{V}_{2,i}}) \qquad (2)$$

Because of periodic frost control, the temperature efficiency may vary throughout the year. Corrected temperature efficiency in time step *i* due to frost control can be found with equation (3), where all temperatures are expressed in °C.

$$\eta''_{T,i} = \min(\eta'_{T,i}\ ;\ \frac{\theta_{3,i} - \theta_{4,min}}{\theta_{3,i} - \theta_{1,i}}) \qquad (3)$$

where

$\eta_{T,i}$ is the heat exchanger temperature efficiency at time step *i*, found according to equation (1)

$\theta_{1,i}$ is the outdoor temperature for the local climate at time step *i*, eventually added the temperature increase over supply fan if placed in front of the heat exchanger

$\theta_{3,i}$ is the extract temperature at time step *i*, eventually added the temperature increase over extract fan if placed in front of the heat exchanger

$\theta_{4,min}$ is the frost control temperature, i.e. the lowest discharge temperature from the heat exchanger at the extract side to avoid frost inside the heat exchanger.

Equation (3) applies for the frost controls using discharge temperature at the extract side as the controlling criteria (e.g. bypass, reduced rotary speed, frost battery before the heat exchanger on the air intake side). The annual temperature efficiency for the operating period is calculated by the sum of equation (3) for all time steps divided by the number of time steps.

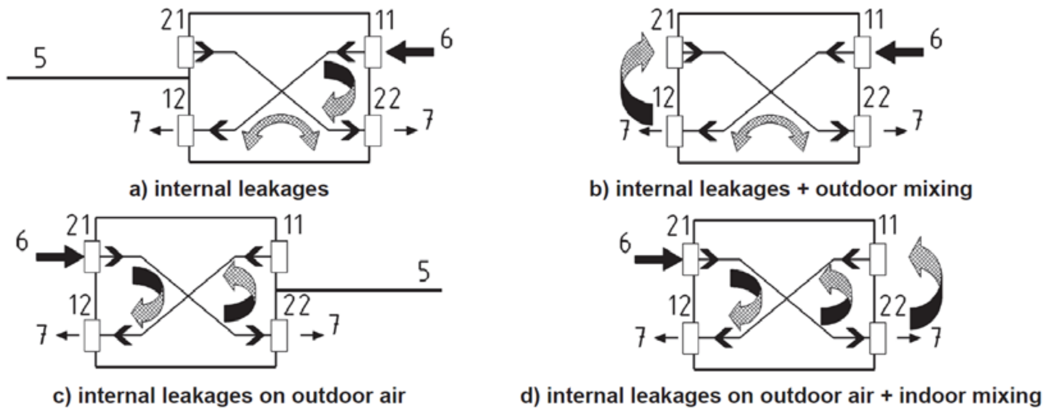
SN TS 3031:2016 also refers to NS-EN 308:1997 *Heat exchangers – Test procedures for establishing performance of air-to-air and flue gases heat recovery devices*. This standard specifies methods to be used in laboratory testing of air-to-air heat recovery. Input criteria, test requirements, and procedures for performing such tests are given to verify the performance data provided by manufacturers. The tests involve additionally, determining the air tightness by an external leakage test, internal exhaust air leakage, carry-over leakages, pressure drops, and temperature and humidity ratios.

What NS-EN 308:1997 refers to as the temperature ratio is the same as the temperature efficiency provided by SN TS 3031:2016. The standard also specifies that for each of the ducts connected to the unit, temperature measurements shall be arranged to determine air temperature at five points evenly distributed over the cross-section. The uncertainty of air temperature measurements shall not exceed ± 0.2 °C (dry bulb temperature) or 0.3 °C (wet bulb temperature) and the airflow shall be sufficiently mixed upstream of the measuring plane to avoid uneven temperature profiles. The distance between the measuring plane and the exchanger should not lead to changes in mean air temperatures above 0.1 °C. For airflow rates, the uncertainty of measurement shall not exceed ± 3 %. The standard also states that how the measurements and calculations have been carried out shall be clearly indicated in the test report, where the influence of frosting and defrosting on the heating capacity must be clearly pointed out.

Definitions of leakages connected to air handling units (Standard Norge 1997, Standard Norge 2014):

- External leakage is leakage to or from the air flowing inside the casing of the unit to or from the air external to the equipment under test.
- Internal leakage is leakage inside the unit between the exhaust and the supply airflows. Carry-over airflow is the term for when exhaust air is transferred into the supply air side in a heat recovery device at overpressure on the supply air side.
- Outdoor mixing is mixing of the two airflows external to the equipment under test between discharge and intake ports at outdoor terminal points caused by short circuiting.
- Indoor mixing is mixing of the two airflows under test between discharge and intake ports at indoor terminal points caused by short circuiting.

Standard test configurations for these leakages are shown in Figure 2.



Key

21	outdoor air	5	deflector
11	extract air	6	tracer gas introduction
22	supply air	7	tracer gas measurement
12	exhaust air		

Figure 2 Test configurations for internal leakages and mixing. (Standard Norge 2014)

In general, manufacturers of heat recovery units normally document the heat recovery rotor performance in laboratory tests, according to NS-EN 308 and NS-EN 13141-7:2010.

According to (Schild and Brunsell 2003) EN 308:1997 and prEN 13141-7:2003 did not explicitly define net recovery efficiency. These standards do not define how to account for system losses from fans, air leakage, defrosting, etc., when calculating net annual energy savings or net air exchange rate. The specified test conditions were considered neither realistic nor equally fair for all the different heat recovery types. An improved test method was developed by the Norwegian Building Research Institute (NBI) and it was accepted as a Nordtest method for Nordic countries. It is called NT VVS 130 – *Air/air heat recovery units: Aerodynamic and thermal performance testing and calculations (Nordtest 2011).* NT VVS 130 describes laboratory and calculation procedures for balanced ventilation systems and is valid for all air-to-air heat exchangers. The following characteristics are tested and rated:

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

The specimen for the sampling should preferably be picked randomly from the production line by a neutral body for testing.

Byggforskserien (Building Research Design Guides) has a relevant design guide; *552.340 Varmegjenvinnere i ventilasjonsanlegg* (Heat recovery in air handling units), explaining general principles for heat recovery units (Schild and Hestad 2002).

1.3 Uncertainty analysis for measurements

The general form of uncertainty for variable *r* is given by Coleman and Steele (1999)

$$U_r = \sqrt{\left(\frac{\partial r}{\partial X_1} U_{X_1}\right)^2 + \left(\frac{\partial r}{\partial X_2} U_{X_2}\right)^2 + \cdots + \left(\frac{\partial r}{\partial X_j} U_{X_j}\right)^2} \quad (4)$$

The uncertainty of the measured variable X_i is consisted of bias and precision errors, which is

$$U_{X_i} = \sqrt{\left(B_{X_i}\right)^2 + \left(P_{X_i}\right)^2} \quad (5)$$

The bias errors (B_{X_i}) are produced form every elemental error source from the measuring equipment. The error sources can be from a) calibration b) data acquisition c) date reduction.
The precision error $\left(P_{X_i}\right)$ is determined by

$$P_{X_i} = tS \quad (6)$$

Where

U is uncertainty
S is sample standard deviation
t is constant which depends on the degree of freedom

The confidence interval of 95 % with the assumption of Gaussian distribution for the readings of the measured variable is typically employed.

1.4 Relevant standards

The following collection of standards are selected as relevant for the measurement of airflows and testing efficiency of heat exchangers:

- NS-EN 308:1997 Heat exchangers - Test procedures for establishing the performance of air to air and flue gases heat recovery devices
- NS-EN 16211:2015 Ventilation for buildings - Measurement of air flows on site – Methods
- SN TS 3031:2016 Energy performance of buildings - Calculation of energy needs and energy supply
- NS 3031:2014 Calculation of energy performance of buildings – Method and data
- NS-EN 13141-7:2010 Ventilation for buildings - Performance testing of components/products for residential ventilation - Part 7: Performance testing of a mechanical supply and exhaust ventilation units (including heat recovery) for mechanical ventilation systems intended for single-family dwellings
- NS-EN 12599:2012 Ventilation for buildings - Test procedures and measurement methods to hand over air conditioning and ventilation systems
- NS-EN 14134:2004 Ventilation for buildings - Performance testing and installation checks of residential ventilation systems
- Nordtest method NT VVS 130, Air/air heat recovery units: Aerodynamic and thermal performance testing and calculations
- ISO 3966:2008 Measurement of fluid flow in closed conduits – Velocity area method using Pitot static tubes
- ISO 16494:2014 Heat recovery ventilator and energy recovery ventilators – Method of test for performance
- NS-EN ISO 5167-1:2003 Measurement of fluid flow by means of pressure differential devices inserted in circular cross-section conduits running full – Part 1: General principles and requirements.
- ANSI/AHRI Standard 1061. 2014 Standard for Performance Rating of Air-to-Air Exchangers for Energy Recovery Ventilation Equipment
- ASHRAE 84-2013 Method of Testing Air-to-Air Heat/Energy Exchangers
- ASHRAE 111-2008 Testing, Adjusting, and Balancing of Building HVAC Systems

2 Literature study

This chapter presents relevant studies where heat recovery efficiency of rotary heat exchangers has been investigated in field measurements. There were very few articles concerning field measurements and specifically the rotary type of heat exchangers. The articles were found through searching for the keywords "heat recovery, ventilation, field" in ScienceDirect and Google Scholar. The title of each article is given as heading for each subchapter which includes a short summary of significant findings.

2.1 Uncertainty analysis in testing of air-to-air heat/energy exchangers installed in buildings (Johnson, Simonson et al. 1998)

This paper presents results from field measurements for two types of heat recovery systems; heat-pipe heat exchanger and rotary energy wheel. They were installed in two retrofitted commercial buildings in Canada. The objective was to obtain field data on the performance of typical installed air-to-air heat/energy exchangers. The most accurate method for calculating efficiency should then be determined from uncertainty analysis.

According to Johnson, Simonson et al. (1998), it is difficult to measure airflow rates in ducting systems due to large uncertainties. This is partially related to large uncertainties with regards to pressure loss in ducting systems. In addition, it is difficult to measure the flow rate, as it requires long straight ducts near the recovery device, which is usually not the case for field installations. Non-uniform flow distributions are often caused by ducting elbows, transition pieces, dampers and fans, which increase the complexity and uncertainty in the flow measurement.

Only sensible energy efficiency is presented, as changes in absolute humidity were small. The uncertainty analysis focused especially on the non-uniform temperature distributions in the ducts. For one of the ducts, the temperature difference was as large as 4 °C in one supply duct and less than 0.1 °C in several other ducts.

The paper concludes that accurate calculations of bulk mean temperature and humidity by using local data will reduce errors in the calculation of efficiency. For non-uniform duct properties, it is recommended to install extra sensors for accurate measurements of bulk properties, as this will reduce the uncertainty of the calculated efficiency. Internal leakage rates between supply and exhaust were not measured with tracer gas tests, as it is "a very difficult measurement to perform in the field".

The average measured efficiency for the energy wheel was 74 % ± 8 %, during a 14-day test period. The study also involved a heat-pipe heat exchanger, where the average measured efficiency was 58 % ± 11 % for a 16-day test period. These uncertainties were larger than the typical uncertainties for laboratory testing, which is usually lower than ± 5 %.

2.2 Real heat recovery with air handling units (Roulet, Heidt et al. 2001)

In this study from 2001, measurements were performed on 30 air handling units located in 14 buildings in Switzerland and Germany. Results from 13 of the units were presented in the paper. Field heat recovery efficiency was evaluated, considering also infiltration and exfiltration through the building envelope. The authors claimed that nominal efficiency of the heat recovery unit leads to optimistic results when the air-handling unit has parasitic recirculation or when the buildings have in- or exfiltration, which was supported by their results. As infiltration rates have been significantly reduced in newer buildings, this study is most relevant for older non-refurbished buildings.

Measurements with the tracer gas dilution technique showed various malfunctions in several units. Parasitic shortcuts and leakages were particularly important, as they can dramatically decrease the efficiency of ventilation and heat recovery. The real global heat recovery efficiency defined in the paper was between 60 and 70 % for the three best units with 80 % nominal efficiency. For the three worst unit, however, the global efficiency was lower than 10 %, resulting in that the heat recovery system used more energy for fans to blow the air through the ducts than it saved. The authors questioned the economic viability and energy saving potential of some small ventilation systems with problematic installation or malfunction, due to poor performance for the air handling units and buildings.

2.3 Forhold tilknyttet bruk av roterende gjenvinnere i skoler (Petersen, Schild et al. 2009)

This is a report from a research project conducted by Erichsen & Horgen in cooperation with Undervisningsbygg & Utdannelsesetaten in Oslo commune, SINTEF Building and Infrastructure, SINTEF Energy, CAMFIL AS and Kaare Rustad AS. The background was an ongoing discussion on whether rotary heat exchangers should be used in schools. It involved an investigation of the general condition, hygienic condition, microbiology, leakage, dust concentrations and composition, humidity conditions, temperature conditions and temperature efficiency of five air-handling units with rotary heat exchangers in schools in Oslo.

Main conclusions from the report are as follows:

- Enforcement and establishment of follow-up and maintenance routines for conditions related to pressure, hygienics and airflow rates for air handling units are needed.
- There are strong indicators suggesting that purging sectors are in general inadequately installed and adjusted, in addition to pressurization of air handling units with rotary heat exchangers.
- There is a need for better solutions to account for the purging flow rate when calculating airflow rates in the air handling units.
- Measured temperature efficiency is in the same order of magnitude as provided by manufacturers. However, if the imbalance between supply and extract airflow rates are considered for these measurements, it is reduced by up to 14 %.
- It may be difficult to comply with the requirements for temperature efficiency in the prevailing Norwegian building regulation at the time, TEK07, without using rotary heat exchangers.
- It was not found any conditions related to hygiene or sanitary conditions that opposes using rotary heat exchangers in schools.
- Leakages from extract to supply are found to be 0.2-2.4 % for the air handling units when the air handling unit with faulty installation of purging zone is disregarded.
- The use of rotary heat exchangers is recommended in all premises where the extract does not contain fat, high moisture content, solvents, strong smells or large amounts of dust.

2.4 Analysis of Mechanical Ventilation System with Heat Recovery in Renovated Apartment Buildings (Kamendere, Zogla et al. 2015)

This study analysed two renovated apartment buildings in Latvia. The main objective was to assess the efficiency of the mechanical ventilation systems with heat recovery in the buildings, both for energy efficiency and indoor air quality. Both buildings were built in 1974 with the same construction principles and renovated in the same way. Field measurements were performed for one month (07.03.2014-07.04.2014). The buildings were occupied at that time, and the authors noted the challenges of in situ measurements in occupied buildings. Measurements are more complex as the boundary conditions are not controllable, and sensors must be placed where they do not disturb the residents in their daily life. For temperature and airflow measurements, they used four airflow transmitters (IVL10) with built-in temperature sensors, that were placed at the middle of the supply and exhaust air ducts (Ø200 mm) before and after the air handling unit.

The heat recovery energy efficiency was found to be 77 %, which is considered a good result compared to the given efficiency in the air handling units' Passive house certificate of 75 %. A regression analysis showed that the heat exchangers' thermal efficiency is dependent on ambient conditions.

2.5 Field tests of centralized and decentralized ventilation units in residential buildings – Specific fan power, heat recovery efficiency, shortcuts and volume flow unbalances (Merzkirch, Maas et al. 2016)

This study involved field tests with 20 centralized and 60 decentralized mechanical ventilation systems in single- and multi-family homes in Luxembourg. Main airflows, internal and external recirculation, sensitivity to differential pressure, specific fan power (SFP) and heat recovery efficiency were the investigated parameters. Measurements were performed during the heating seasons of 2013-2014, at outside temperatures between 0 and 4 °C on days with low wind speeds. Indoor temperatures were maintained between 20 and 22 °C. Volume flows in the ducts were measured with the constant emission tracer gas method, while air temperatures were measured with thermal wires placed as close to the heat exchanger as possible.

It is important to note that only one of the ventilation systems used a rotary heat exchanger, and it turned out that this heat exchanger was defected, leading to very high internal leakages. The findings are still considered relevant for this report.

Deviations between supply and exhaust flows of up to 60 % were found. Total mean external recirculation ratio was 6.5 % ± 12.5% for centralized and 13 % ± 6.2 % for decentralized devices, which led to the lower supply of fresh air. SFP was measured to be 0.475 ± 0.37 Wh/m^3 for centralized and 0.22 ± 0.023 Wh/m^3 for decentralized systems. The latter's lower value is due to reduced pressure loss as there is no ductwork. Heat recovery efficiencies were significantly lower than the values provided by manufacturers for all of the devices, with 65 % ± 24 % for centralized systems and 70 % ± 17 %for decentralized systems. In general, the study found that the overall energy efficiency of the evaluated ventilation devices was lower than expected under real working conditions.

The study concluded that good overall system performance can only be achieved with a well-balanced and well-installed system, where every single factor is taken into consideration, as the measured parameters showed strong interdependencies.

2.6 Analysis of the variable heat exchange efficiency of heat recovery ventilators and the associated heating energy demand (Choi, Song et al. 2018)

The performance of heat recovery units is determined in laboratory tests at certain indoor and outdoor conditions. Fixed heat recovery efficiency is typically used in building energy simulations. In contrast to this, this study sought to analyse the heat recovery efficiency under actual operating conditions. Field measurements were performed in a residential building in Korea over a period of 20 days during the winter (24.02.2016-14.03.2016).

It turned out that the enthalpy heat recovery efficiencies fluctuated between 25 and 70 % depending on the outdoor conditions. Sensible heat recovery efficiency varied between 30 and 65 %, proportional to the temperature difference between inside and outside. In contrast to the constant efficiency provided by manufacturers, the heat exchange efficiency fluctuated under actual operating conditions. The prescriptive sensible and enthalpy exchange efficiencies were also considerably higher, at 81 and 73 %, respectively.

Simulations with variable heat exchange efficiency showed that the heat load and energy demand for heating were 88 % and 69 % higher respectively than simulations with constant heat exchange efficiency. The heating energy demand may thus be underestimated if the heat exchange efficiency of the ventilation system is assumed to be constant in the simulation. The authors recommend that the variable heat/total exchange efficiency of a recovery ventilation system should be provided and used in building energy performance simulations in order to correctly predict the heating demand.

2.7 Conclusions from the literature review

There are very few papers and reports concerning field measurements of rotating heat exchangers. The papers presented here point out difficulties measuring the airflow rates with acceptable uncertainty during field measurements. The measurement uncertainties of the temperature efficiency range from ± 8 % to ± 24 %, which are higher than the typically accepted ± 5 %. The practical efficiency of the rotary heat wheel from field measurements is generally 10 – 20 % lower than the claimed efficiency from manufacturers or suppliers. In the extreme case, the temperature efficiency of the mounted rotary heat wheel is down to 10 % due to maladjustment. The presented literature also states that several measuring points are required to account for non-homogenous temperatures and velocities in the ducts. A protocol for field measurement of performance of the rotary heat recovery needs to be developed to guide the testing procedures for various ductwork configurations and different AHUs. The airflow in ducts for the AHUs should be measured in a feasible and accurate way, which is an important parameter to determine field performance of rotary heat recovery and direct indicator to show the leakages and recirculation in rotary heat recovery.

3 Measurement techniques

This chapter provides an overview of airflow measurement techniques based on information from literature and standards.

3.1 Pressure differential through an orifice plate

Pressure differential method is typically used to determine and monitor airflows and it is considered a relatively mature technique. A fluid passing through a reduction of the open area in the pipeline, such as an orifice, causes a static pressure drop. Bernoulli's equation for fluids can be used to calculate the flow rate based on the measured static pressure difference before and after the orifice plate. The principle is illustrated in Figure 3, while an example of an orifice plate and a lab setup is shown in Figure 4.

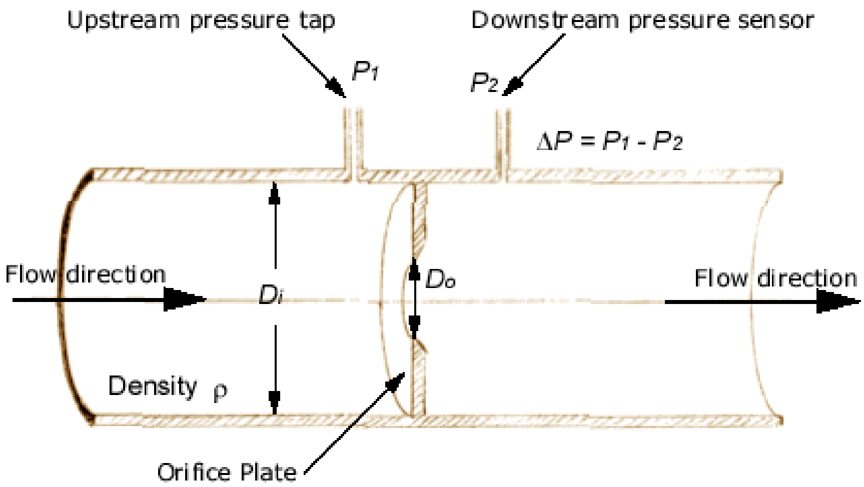


Figure 3 Principle of measuring air flow with an orifice plate. (eFunda 2001)

The standard for measurements with orifice plates is ISO 5167-1:2003 *Measurement of fluid flow by means of pressure differential devices inserted in circular cross-section conduits running full – Part 1: General principles and requirements.* The mass flow rate q_m is calculated according to equation (7), where the flow rate is proportional to the root square of the static pressure difference.

$$q_m = \frac{C}{\sqrt{1-\beta^4}}\,\varepsilon\,\frac{\pi}{4}\,d^2\sqrt{2\,\Delta p\,\rho_1} \tag{7}$$

where:

C is the discharge coefficient, defined for an incompressible fluid flow, which relates to the theoretical flow rate through a device

β is the diameter ratio, the ratio of the orifice to the diameter of the measuring pipe upstream of the orifice

ε is the expansibility factor used to take into account the compressibility of the fluid

d is the diameter of the orifice under working conditions

Δp is the static pressure difference between two taps properly located

ρ_1 is the fluid density at the upstream tapping plane at the temperature and pressure for which the volume is stated

Figure 4 Example of an orifice plate for 100 mm duct with 32 mm opening (left) and lab setup with orifice plate in duct and manometer for measuring the pressure difference (right).

ISO 5167 does not provide for the measurement of pulsating flows. Flow conditions should be constant or only varying slightly and slowly with time. The orifice plate shall be fitted between two straight sections of cylindrical pipe of constant diameter with specified minimum straight lengths without disturbances or branch connections both upstream and downstream. Minimum straight duct lengths both upstream and downstream are specified for different disturbances and diameter ratios (orifice diameter/duct diameter). By using a flow conditioner the use of shorter upstream pipe lengths is possible.

Additional information about general principles for measuring of the flow rate of a fluid flowing in a conduit by means of pressure differential devices (orifice plates, nozzles and Venturi tubes) can be found in ISO 5167-1:2003. This standard also specifies the general requirements for measurement, installation, and determination of the uncertainty of the flow rate.

This method provides relatively accurate results for moderate airflow rates. However, the measurement uncertainty of the pressure differential method for low flow rates is rather high due to the increased measuring uncertainty for the low pressure difference. The method is fragile, expensive and creates an additional pressure loss according to Yu, Li et al. (2011). The pressure differential method is only suitable for laboratory tests since it introduces extra high pressure loss to the ventilation system and it is normally impossible to connect it with the existing ductwork in AHUs.

3.2 Velocity traversal method

This technique is based on measuring a representative velocity for a sector of the duct. By multiplying the air velocity from the different measuring sectors with the cross-sectional area, the volume flow in the duct is found. For measuring the velocity using the traversal method, the air speed measurement device should be small enough to minimize disturbances in the airflow profiles (see Figure 5). Normally, hot wire or NTC anemometers, helix anemometers and Pitot tubes are used.

- The anemometers measure the temperature drop of a heat wire or resistor, which is directly related to the airflow's temperature and velocity. Velocities between 0.05 and 5 m/s can be measured, which make the devices suitable for the typical range in ventilation ducts of 1-5 m/s.
- The helix anemometer and Pitot tube are most accurate for velocities above 10 m/s, and are therefore not suitable for measurements in air ducts. (Roulet 2008)

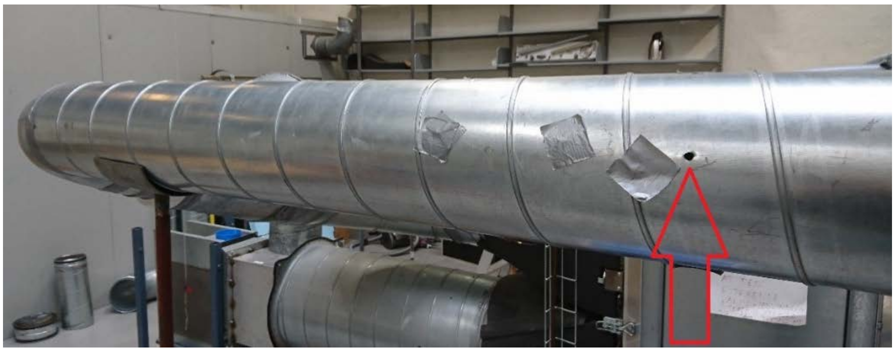
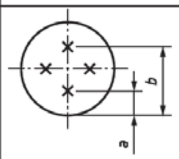
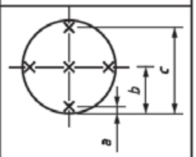
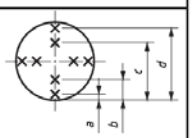


Figure 5 Example of a small hole for measurements with the velocity traversal method.

NS-EN 16211:2015 suggests that the position and the number of measuring points, depends on the duct diameter, as shown in Table 2 for circular ducts. A similar table exists for rectangular ducts. The standard states that the uncertainty for point measurement using hot-wire anemometer or mechanical anemometer is 4 or 6 % depending on the angle of the cross-section measurements. If the measurements in the cross- section are perpendicular to the axis of the duct (horizontal/vertical relative to the duct, as shown in the figures in Table 2), the uncertainty is 4 %, while if the measurement is angled relative to the duct, the uncertainty is 6 %. The reason for the increase in uncertainty is that the flow profile is different and that accurate location of the probe is more difficult compared to the first case. The standard further specifies minimum straight duct lengths for the method (5·D before and 2·D after the measuring plane) as well as other criteria for approving the measurement plane.

Table 2 Measurement points for circular ducts from NS-EN 16211:2015. (Standard Norge 2015)

Nominal Diameter [a] D mm	Position of measurement points	a mm	b mm	c mm	d mm	Figure
100 125 160	$a = 0{,}29 \cdot D$ $b = 0{,}71 \cdot D$	29 36 46	71 89 114			
200 250 315 400	$a = 0{,}10 \cdot D$ $b = 0{,}50 \cdot D$ $c = 0{,}90 \cdot D$	20 25 32 40	100 125 158 200	180 225 283 360		
500 630 800 1 000 1 250	$a = 0{,}043 \cdot D$ $b = 0{,}290 \cdot D$ $c = 0{,}710 \cdot D$ $d = 0{,}957 \cdot D$	22 27 34 43 54	145 185 230 290 360	355 445 570 710 890	478 603 766 957 1 196	

[a] According to duct standard.

TSI (2014) suggests measuring at least 7.5 straight duct diameters downstream and at least 3 duct diameters upstream from any turns or flow obstructions. Traverse measurements with as little as 2 duct diameters downstream and 1 duct diameter upstream from the obstruction can be performed, but this affects the measurement accuracy for circular ducts. For rectangular ducts, the application note from TSI refers to the following equation to find the equivalent diameter:

$$Equivalent\ diameter = \sqrt{\frac{4 \times H \times W}{\pi}} \tag{8}$$

It also refers to the log-Tchebycheff method, where the duct is divided into three traverses (instead of two as above), where the traverses are oriented with 60° angles from each other. As in Table 2, the measurement points depend on the duct size. For this method, the average velocity of all measurements is found and multiplied with the total area of the duct.

Fluke (2012) says that the preferred location of the traverse in a supply duct should be in a straight section of the duct with 10 straight duct diameters downstream and 3 diameters upstream from disturbances. Adequate results may also be achieved with a minimum of 5 duct diameters downstream and 1 duct diameter upstream from disturbances. A minimum of 25 measuring points is also recommended, regardless of duct size. See the reference for more information.

The number of needed measurement points also increases with increasing duct diameter, as shown in Table 2. Air properties may be non-uniform spatially across one or more ducts and vary with time. Ideally, all points should be measured simultaneously in case of fluctuating velocities or temperature levels, but this is not possible without disturbing the airflow significantly. If measuring with one device, the holes in the ducts that are not in use for that specific measurement point should be taped, to avoid leakages, which will affect the measurements.

The velocity traversal method is easy to implement and understand. The measurement accuracy depends on the velocity sensor and the velocity profiles. For very low velocities and developing flows, the testing uncertainty may become quite high. The sensor probe always interferes with the flow profile which will affect the measurement results. Traversing the sensor needs to be conducted manually. This means large amounts of labour and the airflow results cannot be recorded automatically.

3.3 Ultrasonic measurements

Ultrasonic flow rate sensors have been around for at least 60 years according to Lynnworth and Liu (2006). Ultrasonic sensors are commonly used for liquid flow measurement in pipelines. It has been used to measure the flow of natural gas since the 1970s (Conrad and Lynnworth 2002). The historically high cost has restricted its application for HVAC systems. More recently, commercially available ultrasonic products for airflow measurement with low cost in HVAC systems are emerging (FläktGroup 2017, Lindab 2017).

The theory of ultrasonic devices is based on the principle that sound waves propagate faster in the direction of the flow than against it (Conrad and Lynnworth 2002). The air velocity can be calculated using the transit times of two beams of ultrasonic waves. A schematic view of the single path ultrasonic airflow measurement device is shown in Figure 6.

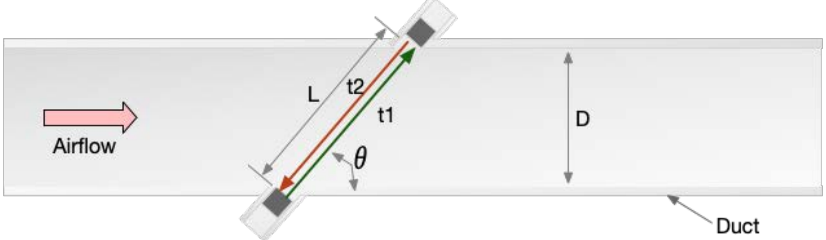


Figure 6 Single path ultrasonic waves propagate in a duct.

The mean airflow velocity u_m across the ultrasonic path is calculated by the equation below.

$$u_m = \frac{L}{2cos\theta}\left(\frac{1}{t1} - \frac{1}{t2}\right) \tag{9}$$

When the air velocity is sufficiently smaller than one Mach which is always valid for air in HVAC systems, the mean air velocity can be approximated as

$$u_m \approx \frac{c^2 \Delta t}{2L cos\theta} \quad (10)$$

It can be seen that the mean air velocity across the ultrasonic path is proportional to the transit time difference. The ultrasonic method has a linear response to mean velocity, which provides low measurement uncertainty for very low air velocity. As it measures the mean velocity across the ultrasonic wave path, the ultrasonic sensor is preferred to be placed at the position where the airflow is fully developed with symmetry velocity profile.

Two products for airflow measurements, Lindab UltraLink and Optivent Ultra are presented in this project. When analysing air movement with ultrasound waves, several methods can be used. The measurement range for the two products, Lindab UltraLink and Optivent Ultra is approximately 0.5-15 m/s.

The ultrasonic measurement device can provide comparable measuring accuracy with pressure differential methods without causing any additional pressure loss. Another advantage using ultrasonic devices is that they have a linear response to flow velocity changes, so their sensitivity does not degrade with low airflow velocity, as opposed to with pressure differential airflow measurement devices. A certain straight length of duct in front of the ultrasonic measurement device is needed to achieve high measurement accuracy with the symmetry fully developed air velocity profile. For the airflow measurement close to the turbulence inlet with asymmetry air velocity profile, multiple wave path will improve the accuracy. Normally the ultrasonic device is pre-fabricated in a section of duct, which leads to difficulty to incorporate with existing ductwork or different sizes of new ducts in AHUs.

A type of clamp-on ultrasonic measurement device without using any open nozzle is presented by Conrad and Lynnworth (2002). This clamp-on solution to gas airflow measurement can measure the airflow from outside of the duct, which means it can be applied for existing ductwork without any intervention. It is also possible to use multiple cross paths to improve measurement accuracy for asymmetric velocity profiles. In other words, it could be used to measure the airflows which are very close to bends, dampers, coils and fans in the AHU with low uncertainty.

3.4 Tracer gas method

Tracer gas dilution techniques are effective and well-known for assessing airflow rates and patterns within buildings and ventilation systems. The air is "marked" with one or various tracer gases, which mix well with the air and is easy to analyse the amount of tracers. By interpreting the evolution of tracer concentrations, it is possible to calculate airflow rates, the age of air, ventilation efficiency, leakage flow rates (Mundt, Mathisen et al. 2004). The tracer gas technique can be used to determine the main and parasitic airflows in rotary heat wheel. For the application in this study, the constant tracer injection rate is employed. Information presented in this chapter is based on the book *Ventilation and Airflow in Buildings, Methods for Diagnosis and Evaluation* by Roulet (2008).

A tracer gas system consists of the following components:

- Tracer gas container with a pressure reduction valve
- Injection system that allows for the constant and measured volumetric flow of tracer gas to the designated locations
- Mixing system that ensures good mixing between the tracer gas and the air
- Gas sampling system, for bringing the sample to the analyser
- Gas analyser, for measuring the concentration of tracer in the air

For airflow measurement in AHUs, the following properties are desirable for the tracer gas:

1. Easy to analyse, preferably at low concentrations to reduce cost and side effects such as toxicity or change in density of the gas/air mixture
2. Low background concentration, so that a low concentration can be used
3. Non-flammable nor explosive at practical concentrations
4. Non-toxic at the used concentration, especially in inhabited buildings
5. Density close to the air density to ensure good mixing
6. Non-absorbable by furnishings, decomposable or reactive with air or building material
7. Low price

Some tracer gases are very potent greenhouse gases, such as SF_6 with the highest given global warming potential (GWP) of 23 900 (IPCC 2007). Freons and halons have high ODP and destroy the ozone layer. These gases are also regulated by the F-gas directive, and Norway is obligated (Regjeringen 2017) to reduce the emissions of such gases. In light of effects of GWP, CO_2 could be a superior alternative to these tracer gases.

When using the tracer gas dilution technique, the tracer gas is injected at a known constant flow rate, I. The air is then analysed downstream from the injection port, sufficiently far to achieve a good mixing of the tracer into the air. The ductwork of AHU is modelled by nodes connected by ducts. The assessment procedure of main and parasitic airflows using tracer gases in AHU is in principle similar to its application for multi-zone buildings. The node by node method is applied to airflow and tracer gas, giving all airflow rates entering in this note. Multiple tracers could be injected at designated locations for different ducts. Sequential measurement with one single tracer is also possible to determine the main and parasitic airflows in different ducts. The air with mixed tracers is delivered to the tracer analyser. The concentration of the tracer is measured by the LumaSense (LumaSense Technologies Inc. 2018) photoacoustic gas monitor in this study. At steady state, based on the tracer gas mass conservation,

$$-I_{ik} = \sum_{j=0}^{N} (C_{jk} - C_{ik})\, Q_{ji} \tag{11}$$

where:

I_{ik} is the injection rate of tracer gas k in node i,
C_{jk} is the concentration of tracer gas k in node j,
Q_{ji} is the airflow rate from node j to node i.

Steady state is assumed, so that airflow rate and injection flow rate are constant, and the concentration is recorded only when a constant concentration is reached. Each system can be rewritten in a matrix form using airflow and tracer gas conservation equations:

$$\vec{I}_i = C_i \vec{Q}_i \tag{12}$$

A good mixing of tracer gas with airflow is crucial to assess airflow rates in each duct branch. To achieve sufficient mixing, several practical criteria on the distance between injection points and air sampling locations are suggested by Roulet (2008):

- 10 diameters (or duct widths) in straight ducts;
- 5 diameters if there is a mixing element such as bends, droplet catcher or a fan between injection points and the air sampling nodes.

If the above criteria for straight length to mix the tracer and air cannot be fulfilled due to the limited space, the multiple injections with evenly distributed tiny openings on perpendicular tubes can be applied to enhance the mixing of tracer and airflow.

Unlike the other techniques described in this study, the tracer gas method can directly measure the volumetric airflow rates instead of measuring air velocity. The mixing level is a key factor determining the measuring accuracy of the tracer gas method. Compared to other presented methods, the tracer gas method is able to determine the airflows and recirculation in different ducts at the same time. However, it normally needs more efforts and time to get the measuring system ready. It also needs specialize knowledge to calculate the airflow rate and its uncertainty from the measured tracer concentrations. The complexity and possible poor mixing in AHUs of tracer gas method may limit its application in the field airflow measurement.

4 Summary of available airflow measuring equipment and measurements performed in this project

4.1 Differences between measuring in the lab and in the field

Most of the measurement techniques presented in chapter 3 are addressed for laboratory measurements that unfortunately are not easily transferred to measurements in occupied buildings. Measurements of rotary heat exchangers in the lab are done under almost steady-state conditions, but this is not the case for measurements in the field. Temperature variations occur and the airflows may vary, especially for newer installations with demand-controlled ventilation (DCV). Bends, reducers, or T-pieces lead to straight lengths shorter than the minimum required to achieve accurate measurements. The velocity profile is also disturbed by dampers and before and after a fan.

4.2 Project needs

The ideal field measuring equipment for our project should be portable so that it can be used in several locations for existing buildings, and flexible so that it is suitable for different duct sizes. The measurements should be carried out automatically (requiring little manual labour) and the results should be logged and easily exported to a computer. Note that automatic measurements will be difficult to achieve in practice and acquiring more accurate measurements than what is possible today will be a big progress in its own. If the equipment is placed directly in the flow, this will introduce pressure losses, which will lead to extra energy use for the fans if it is mounted in the system for a long term to monitor the airflow. In our case, as we were working with existing buildings, it was required to be possible to install the equipment without too large intervention. This was what we requested to the suppliers. Other solutions not described here could have been ok for the case of fixed equipment, but we needed to move the equipment as we planned to measure in 9 different locations.

4.3 Contact with suppliers of measuring equipment

In the pursuit of the most suitable measuring equipment, several instrumentation suppliers were contacted, in addition to looking for products online. Below is a list of companies contacted by e-mail or phone:

- Dwyer
- Pro-instruments
- Elma
- Impex
- Omega
- Senmatic
- Nortelco
- Instrumentcompaniet
- Lindab
- Ventistål
- Trox

At the time of the survey (Autumn 2017), most of the suppliers only had simple devices that could measure temperature, velocity and relative humidity in one point only. It was requested to measure in multiple points in the cross-section to achieve sufficient accuracy, and to account for bends and other disturbances. No commercially available equipment was found. A custom-made device would have been necessary and it was too expensive to develop within this project. Additionally, the difference of duct diameters targeted posed an extra challenge.

The detailed descriptions for some equipment tested and evaluated in this project, which includes VelociCalc ventilation meter, Lindab Ultralink and Optivent Ultra airflow measurement device and DEBIMO airflow measuring blades, can be found in Appendix B.

Several measurements have been carried out and addressed in Appendix C during the duration of the project in order to test both different equipment in Appendix B and methods for measuring addressed in Chapter *Measurement techniques*. Most measurements have been performed in the laboratory at NTNU's Department of Energy and Process Engineering in Trondheim. During these lab experiments, we tried to find accurate methods for measuring airflow rates in ventilation ducts and in an air handling unit. Some of the devices were also been tested under difficult conditions similar as to what can be expected for field measurements. Additionally, field measurements have been performed at Øya kindergarten.

4.4 General conclusions from the measurements

To sum up common findings from the tests, the measurements with VelociCalc 8388 are in general more accurate than UltraLink. Note that the tested setups are mostly outside producer's recommendations due to too short ducts before and after disturbances. The main drawback with using the velocity traversal method compared to using UltraLink is that it requires manual measurements, which are time-consuming and that measurements cannot be taken continuously. For unstable flow conditions, either due to fan variances or not fully developed flow after disturbances, it will also be a drawback that the measurements do not happen simultaneously in the different points of the cross-section. Another important observation is that the deviations for VelociCalc 8388 are highest for low airflow rates, and in general higher than the deviations found with the UltraLink oriented towards the side. For higher airflow rates, the VelociCalc usually had lower deviations from the orifice plate than the UltraLink. Thus, the choice of measurement equipment is also dependent on the size and range of the airflow rate to be measured.

For the tracer gas measurement, the cross-compensation function of adsorption spectrums for different tracers should be activated in the gas analyser to reduce the interference between various tracer gases. If the function is not available, the sequent testing with single tracer should be used. The tracer gas method is able to present the main and parasitic airflows in the AHUs. However, the good mixing between tracer and airflow is a big challenge in the tested ducts even when the multiple injections have been used.

The velocity traversal method was applied to the field test for the AHU at Øya kindergarten. The layout and measurements of the field test present a representative example of how difficult and uncertain to perform the airflow measurements for the limited space for inserting measuring equipment and insufficient straight duct for testing.

5 Discussion

The real efficiency of rotary heat exchangers taking recirculation and leakage in the air handling unit into account is a topic that up until recently have been poorly investigated. This is also confirmed with the literature study, where only a few relevant articles for field measurements, most of them published during the past couple of years are available. There is also a lack of relevant measuring techniques and standards focusing on field measurement procedures. Equipment suitable for finding the airflow rates close to the heat recovery unit could not be found, though this is needed for field measurements before any possible branching of the flows. There has however been a lot of progress lately for measuring velocities for VAV dampers in demand-controlled ventilation systems, but this is not directly transferable to measuring closer to the air handling unit.

The devices based on ultrasound technology, Lindab UltraLink and Optivent Ultra are promising tools, especially due to their automated measurements and the fact that they are not disturbing the airflow in the duct. They are however primarily intended to improve the energy efficiency of the duct network in the ventilation system, especially Optivent Ultra which is used as a VAV damper. Neither of the devices is intended to be mounted close to the air handling unit. Neither are they by today, available for diameters of duct size larger than Ø315mm. They are also most suitable for new installations, as their installation would otherwise require cutting the duct, action considered too invasive and expensive, if it even is sufficient room for the device, at least if they are intended for temporary measurements as the project was targeting.

The measurements with the velocity traverse method showed that this method is highly accurate and can be used to determine the airflow rates, even close to disturbances. A major drawback, however, is that it requires larger labour. Our measurements also showed that there are large differences between the different measuring points in the cross-section of the duct, proving the necessity of measuring at multiple points to find the correct airflow rate. To determine the real efficiency of rotary heat exchangers, measurements must also be performed continuously over a certain time span, which is not possible with the VelociCalc device. To conclude, the method is good, but another instrument that can log multiple points in the cross-section over time with minimal disturbance for the airflow is needed.

For our tests, the airflow rates have been constant, but as demand-controlled ventilation is becoming increasingly popular as ventilation control, the airflow rates are expected to be more fluctuating. If the airflow rates and temperatures are not constant during the measurement period for the different points in the cross-section, the velocity traverse method will not be accurate. But solutions like the Optivent with control of VAV damper may be even more interesting. Ideally, the measurements in the different points should take place simultaneously, but that is not possible with VelociCalc without significantly disturbing the airflow by introducing multiple devices. NS-EN 16211:2015 also states that the overall diameter of the anemometer device obstructing the duct passage area should not exceed 1/10 of the duct diameter. This excludes the possibility of using multiple devices simultaneously (at least for small ducts) and limits the design of new devices with multiple measuring points in the cross section. A solution for air handling units with more fluctuating airflow rates could be to use another device than VelociCalc with the same accuracy but lower influence on the airflow. It does, however, seem like such solutions are not yet available., at least it was not found during the project time.

For the velocity traversal method, the standard NS-EN 16211:2015 and other sources in chapter 3.2 suggest finding the airflow rates through multiplying the *average velocities* with the *total area* for the cross-section of the duct. If the airflow is not fully developed (which is often the case, except for long straight ducts), it will be more accurate to find the airflow rates by using area-weighted velocities, i.e. distribute the cross-section into sections as illustrated in Figure 7 and multiply the velocities with their corresponding areas. In that way, velocities representing different sized areas will be accounted for. For our measurements, airflow rates were calculated using both methods, and the simple method of using the average velocity and multiply with the total area of the duct resulted in a significantly higher deviation from the values found with the orifice plate. This was especially the case for appendix C1, where the measurements were performed further from the duct wall than what was specified in the standard. Thus, the outer measurement points (A1, A4, A5 and A8 in Figure 7) were representing larger areas than the inner measurement points. We, therefore, recommend the method of dividing the cross-section into area-weighted sectors representing the different velocities, to account for differences in area sizes.

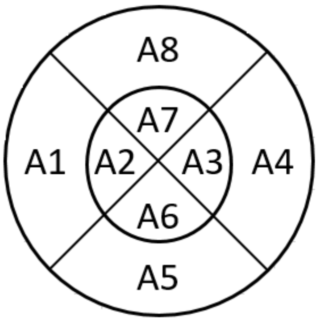


Figure 7 Illustration of how a cross-section with eight measuring points for velocities can be divided into areas to find the area-weighted airflow rate.

The tracer gas method is able to reveal the main and parasitic airflow rates in the heat wheel. The recirculation between the supply and extract sides can be then determined. A key factor for a successful use of tracer gas technique is the good mixing of tracers with the airflow in the duct. This can be achieved by applying multiple injections, sufficient straight length of duct for sampling and injecting and airflow mixer or using turbulators. Poor mixing may lead to wrong results of concentrations and airflows. Multiple tracer gases can be injected and sampled to calculate the airflows simultaneously. The interference between different tracer gases in the analyzer should be compensated for the raw signal in order to obtain the correct resulting concentration. When a single tracer gas is applied, the airflow rate and fan frequency should be maintained constant. The time interval between different sequential runs of the tracer injections should be adequate to eliminate the influence of the previous injection. With the information of the main and parasitic airflows for the rotary heat wheel, the real recovery efficiency of the heat wheel can be calculated. However, a good mixing of tracers and airflows is usually difficult to achieve in practice due to the limited space for the air handling unit. The preparation, implementation, interpretation of the tracer gas tests for the ducts close to the AHU often need much time and labour. It is not so intuitive compared to other techniques such as ultrasound technology and velocity traverse, which may difficult its application.

Location is an integral part of this project, as the measurements ideally should be taken as close as possible to the heat exchanger, so that all branches are included, but also due to heat transfer with the surroundings and possible leakages. In real air handling units, there is often limited space for measurements inside the actual air handling unit, while bends, ventilation silencers, heating batteries, fans, etc. will make it necessary to have a certain straight duct distance downstream to make accurate measurements with the currently available equipment and methods. In appendix C2 we performed measurements closer to a bend than the recommended minimum distances. It was found that the VelociCalc still was quite accurate, although the deviation from the reference value from the orifice plate increased, while the UltraLink was accurate for a certain orientation (first sensor at the "side"), while not for the other orientations (first sensor at inner and outer radius). These findings may be relevant to similar configurations with bends, but for other disturbances, such as measurements closer to fans, the results may be different. Further work is required to verify whether the same accuracies can be achieved for other disturbances.

6 Conclusion

The field measurement of airflows in AHUs is of significance to determine the real efficiency of rotary heat exchangers, and ensure energy efficiency in accordance with the design. Limited space in technical rooms, bends and other disturbances as well as the influence of fans, the rotary heat exchanger and other components in the air handling unit makes it difficult to measure the airflow rates close to the heat exchanger in real buildings.

The products and measuring techniques presented in this report are suitable for measuring airflow rates in ventilation ducts, provided they are installed according to their requirements, for instance regarding minimum straight duct lengths. However, neither of the devices and methods evaluated in this report are suitable for accurate measurements. The real efficiency of rotary heat exchangers could not be calculated automatically over a certain time span. None of the tested solutions was suitable to be moved to several buildings without incurring in large labour. Based on this study, there is no ideal equipment available for measuring both airflow and temperatures close to the air handling unit, as they all have requirements regarding minimum straight duct lengths. If the real efficiency is attempted measured, the accuracy and reliability of the measurement are expected to be low. The measuring equipment must be chosen based on the characteristic of each ventilation system, especially duct diameter, available space to perform the measurements and minimum straight duct lengths before and after disturbances.

We would like to encourage manufacturers and research institutions to further investigate this issue so that more suitable airflow measuring equipment for field measurement in AHU will be developed in the future. The ideal field measuring equipment should be able to accurately measure airflow rates, temperatures and relative humidity as close as possible to the rotary heat exchanger. The measurements should be carried out automatically (requiring little labour) and the results should be logged and easily exported to a computer. The device should also be of minimal disturbance for the airflow, as disturbances will induce pressure drop and increase energy use for the fans. We would also recommend the following measurement uncertainties for the different parameters:

- Temperature: ±0.2 °C
- Airflow rate: ±3 %
- Relative humidity: ±5 %

7 Further work

Due to limitations on the hardware to measure airflows the field efficiency of rotary heat exchangers could not be calculated in the nine existing buildings. This work is still of high relevance as systems installed in buildings are proven to normally have lower efficiency. Field measurements of airflows, temperatures and humidity are recommended. They are relevant to reveal the potential for improving component performance and to understand the effect of frost formation when heat recovery units are requested to have higher temperature efficiency. Knowledge about deviations between measured and estimated efficiency would also be useful for revising guidance material, standards and energy calculation methods for heat recovery.

To determine the real efficiency of rotary heat exchangers, new measuring devices are needed. We encourage suppliers of measuring equipment and research institutions to develop solutions suitable for this purpose. The equipment should have the properties described in the conclusion, and if possible be able to measure velocities, temperatures and relative humidity according to Figure 8.

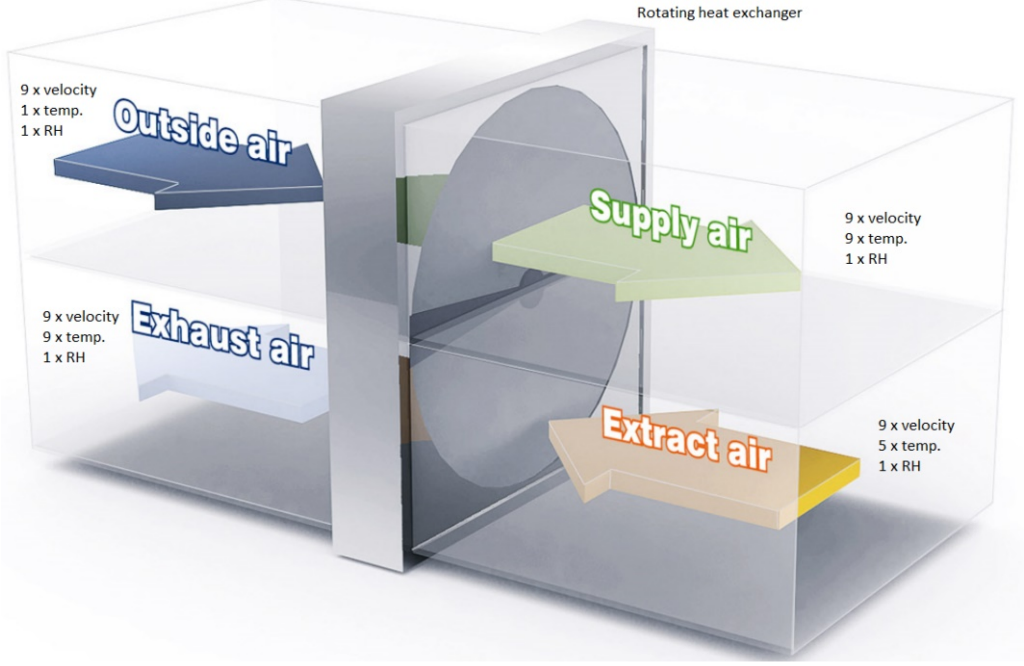


Figure 8 Illustration of a rotary heat exchanger and desired number of measuring points for velocity, temperature and relative humidity. Illustration retrieved from Klingenburg (2018) and edited to include the number of measuring points.

The number of required measurement points may be slightly adjusted based on the size of the air handling unit. Note that the velocity and temperature ought to be measured in multiple points. For the intake of outside air, it can be assumed that the outdoor temperature is uniform, so here only one measuring point is needed.

8 Acknowledgements and intellectual property notice

The project has been financed by Husbanken, and the authors would like to thank Husbanken represented by Vegard Heide for making this research project possible. Hans Martin Mathisen, professor at NTNU, has been supervising the work, and have contributed through invaluable guidance and discussions as well as quality assurance of the report. Synne Kathinka Bertelsen at Trondheim municipality has also been very helpful with supplying pilot buildings for measurements. In addition, we would like to thank Lindab for providing us with measuring equipment (Lindab UltraLink) and discussion regarding the results, as well as Instrumentcompaniet for discussion and follow-up regarding measurement techniques and their products, especially DEBIMO air flow measuring blades. Many measurements have been carried out at the lab at NTNU, which would not have been possible without the help from many people, but especially from Håvard Rekstad for mounting the test rig and supplying us with tracer gas and other necessary equipment.

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Appendix A Definition of real energy recovery efficiency

Field energy recovery efficiency for heat wheels used on this report is defined in this section. The real efficiency for a heat wheel is defined in the boundary as shown in Figure 9. The real energy recovery efficiency of the heat wheel is defined as the ratio of recovered energy by heat wheel to the maximum possible heat energy recovery potential.

The recovered energy from a heat wheel is

$$\phi_R = (1 - R_e)\dot{m}_{12}(h_3 - h_2) = (1 - R_s)\dot{m}_{12}(h_4 - h_5) \tag{13}$$

Where:

ϕ_R is the recovered energy from a heat wheel (J).

R_e is the extract air recirculation rate (defined as the ratio of leaked volumetric airflow from extract air towards supply air to actual extract volumetric airflow).

$\dot{m}_{12}$ is air mass flow rate flowing through nodes 1 and 2.

h is air enthalpy at different nodes (J/kg).

The maximum possible energy recovery potential is

$$\phi_{max} = min\{\dot{m}_{12}\,, \dot{m}_{12}\,\}(h_4 - h_2) \tag{14}$$

The real energy recovery efficiency for the heat wheel is

$$\eta_{real} = \frac{(1 - R_e)\dot{m}_{12}(h_3 - h_2)}{min\{\dot{m}_{12}\,, \dot{m}_{12}\,\}(h_4 - h_2)} \tag{15}$$

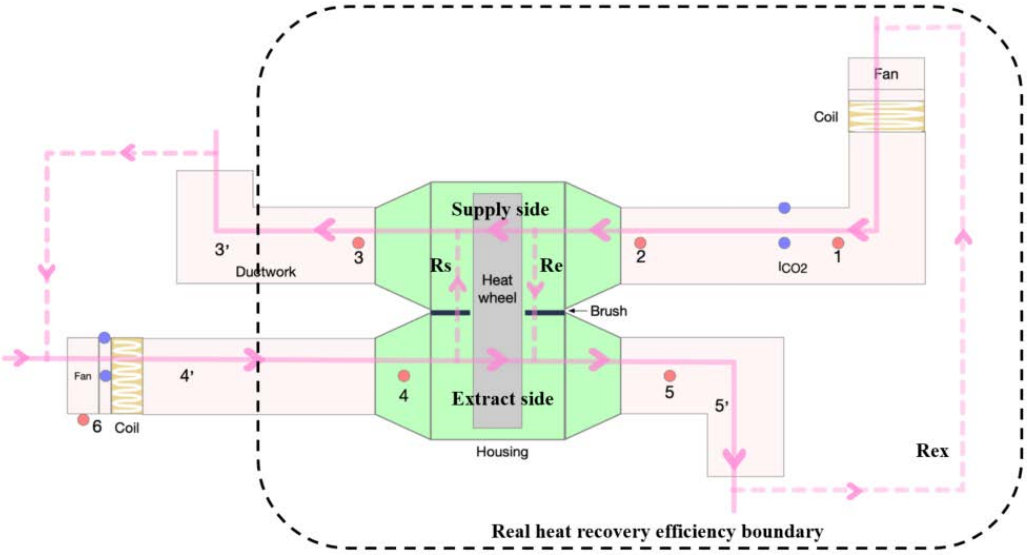


Figure 9 Illustration of the boundary of real heat recovery efficiency.

Appendix B Tested measuring equipment for airflow rates in AHU

The following equipment were tested or evaluated though they could not comply with the requirements specified before.

Appendix B1 VelociCalc Multi-Function Ventilation Meter

VelociCalc is a type of hot wire anemometer. It is normally used in the velocity traverse method, where multiple points are measured over the cross-section of a duct. In addition to velocity, this device measures temperature and relative humidity simultaneously. Figure 10 shows a picture of the instrument along with the extendable measurement probe.

Figure 10 VelociCalc 9565-P along with extendable thermoanemometer probe.

There are multiple models of VelociCalc available, and the newest models have the possibility of logging and exporting measurement data to computers. One drawback of using such devices for traverse measurements is that it is either necessary to mount multiple devices with logging in the air duct, thus disturbing the airflow significantly. Otherwise, one device could be moved manually to measure the airflow in the selected points. The latter solution makes it impossible to measure the velocities in different sections simultaneously, so this method is dependent on stable flow conditions.

In this project, the models VelociCalc 9565-P and VelociCalc 8388 have been used. Some relevant specifications for them are listed in Table 3.

Table 3 Specifications for VelociCalc 9565-P and 8388 with thermoanemometer probe.

Specification	VelociCalc 9565-P (TSI 2018)	VelociCalc 8388 (TSI 1998)
Velocity		
Range	0-50 m/s	0.15 to 50 m/s
Uncertainty	±3 % of reading or ±0.015 m/s, whichever is greater	±3 % of reading or ±0.02 m/s, whichever is greater
Resolution	0.01 m/s	-
Temperature		
Range	-10 to 60 °C	-10 to 60 °C
Uncertainty	±0.3 °C	±0.3 °C
Resolution	0.1 °C	0.1 °C
Relative humidity		
Range	5 to 95 % RH	0 to 95 % RH
Uncertainty	± 3% RH	± 3% RH
Resolution	0.1 % RH	0.1 % RH
Duct size, range	2.5 to 1270 cm in increments of 0.1 cm	1 to 100 cm in increments of 0,5 cm, 100 to 255 cm in increments of 1 cm

Data logging and export to PC?	Yes	No
Data sheets for more information	http://www.tsi.com/uploadedFiles/_Site_Root/Products/Literature/Manuals/9565-VelociCalc-6004851-web.pdf	http://www.tsi.com/uploadedFiles/_Site_Root/Products/Literature/Manuals/1980253c.pdf

Due to the placement of the sensors on the anemometer probe, it is not possible to measure "just inside" the air duct. This is explained in the datasheet for VelociCalc 9565-P in the following manner: *"Note. For temperature and humidity measurements, make sure that at least 7,5 cm of the probe is in the flow to allow the temperature and humidity sensors to be in the air stream."* (TSI 2018). The datasheet of VelociCalc 8388 did not have such a note, but the minimum distance is found to be approximately 3-4 cm. For easily available ducts (with sufficient space on all sides) a solution could be to drill four holes (top, bottom and both sides) and tape over the holes that are not in use. Then it is possible to access the short distance measuring points from the other side. This will however not be possible in most field installations, as walls/roof/floor, other ducts and technical installations will be blocking the measurements.

Appendix B2 Lindab UltraLink

Lindab UltraLink (see Figure 11) measures airflow and temperature with an angled ultrasonic beam, without posing an obstacle in the airstream that would lead to pressure drop. In addition, the device allows for measuring velocities in a wide range: 0.2-15.0 m/s, while maintaining a high accuracy. Temperature is also measured, while it does not measure relative humidity. Some other specifications can be found in Table 4.

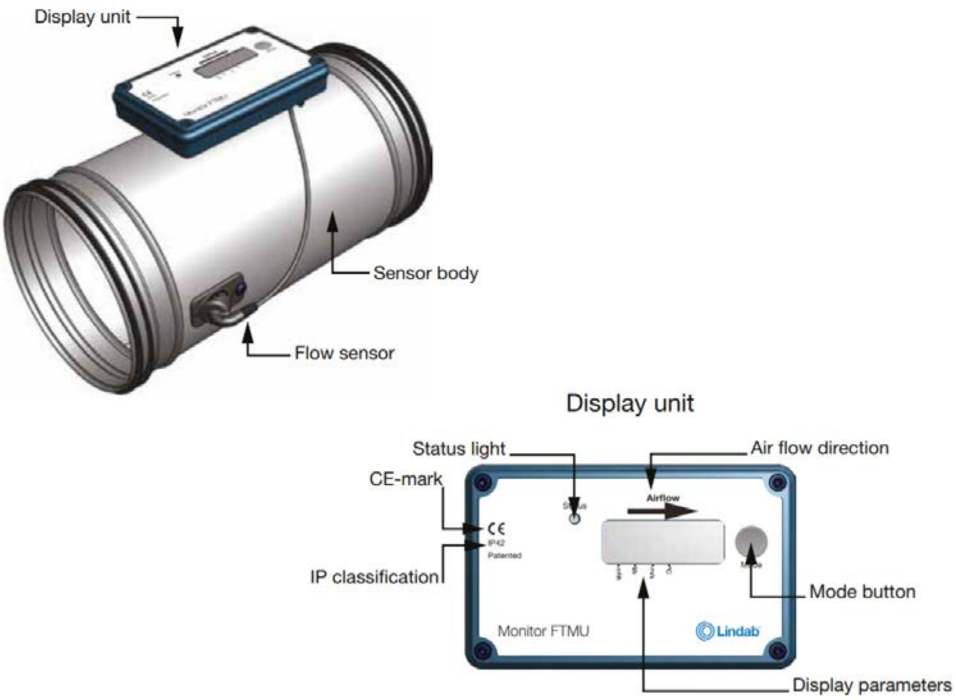


Figure 11 Illustration of Lindab UltraLink. (Lindab 2017)

Table 4 Specifications for Lindab UltraLink. (Lindab 2017, Lindab 2018)

Specification	Lindab UltraLink
Velocity Range Uncertainty for flow	 0.2-15.0 m/s ±5 % of reading or ±X* l/s, whichever is greater
Temperature	

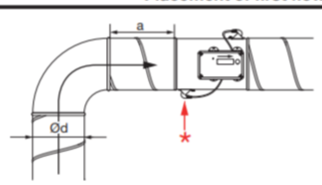
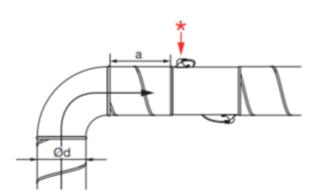
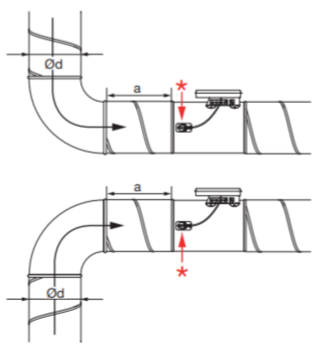
Range Uncertainty	-10 to 50 °C ±1 °C
Duct size	Ø100 to Ø315 (mm)
Data logging and export to PC?	Yes
Data sheet for more information	https://itsolution.lindab.com/LindabWebProductsDoc/PDF/Documentation/ADS/Lindab/Technical/Technical-FTMU.pdf

*X equals the diameter in dm, for instance Ø100 => 1 l/s and Ø200 => 2 l/s

The main drawback of Lindab UltraLink for performing field measurements is that the measuring equipment is mounted on a 500 mm long duct. For existing buildings, this makes it necessary to either dismantle or cut the ductwork in order to insert the measuring device. This is a relatively large intervention if not planned, that would require qualified service personnel, and this may be expensive. Another drawback is that the device is by today only available in diameters up to Ø315, which excludes measurements on larger air handling units.

The highest measurement accuracy is achieved with long straight ducts before the device. The technical data sheet specifies measurement uncertainty for different orientations and distances of the device relative to disturbances. Table 5 shows the uncertainty with regards to bends, while other disturbances (reducers and T-pieces) can be found in the data sheet (see link in Table 4). It is not recommended to mount the device so that the first flow sensor (*) is placed on an outer radius of a fitting. A minimum straight distance of 1·D after the monitor is required. The UltraLink should never be used downstream of a duct fan. It should be placed on the inlet side or in worst case use a flow conditioner if it must be placed on the outlet side.

Table 5 Measurement uncertainty for Lindab UltraLink for different orientations and distances after bends. (Lindab 2018)

Disturbance	* Placement of first flow sensor		Measurement uncertainty ± % or X l/s depending wich is the greatest, where X equals the diameter in dm, see table on page 13.		
			a		
			2-4·Ød	>4-5·Ød	>5·Ød
Bend		Inner radius	5	5	5
Bend		Outer radius (Not recommended)	20	10	5
Bend		Side	10	5	5

Appendix B3 Optivent Ultra (FläktGroup)

Optivent Ultra (see Figure 12) is a VAV damper, meaning that it regulates and controls airflow rates for either supply or extract air. Dampers can be used to control room temperature and air quality. In addition to the damper function, it also performs airflow and temperature measurements. It is similar to Lindab UltraLink, it also performs the measurements using ultrasound technology, i.e. no disturbance of the airflow. For the damper function, low noise levels and maintenance free operation have been emphasized as considerable advantages. Specifications can be found in Table 6.

Figure 12 Illustration of Optivent Ultra. ULSA is a non-insulated casing, while ULDA is insulated. (FläktGroup 2017)

Table 6 Specifications for Optivent Ultra. (FläktGroup 2017)

Specification	**Optivent Ultra**
Velocity Range Uncertainty for flow	 0.5-15 m/s See Figure 13
Temperature* Range Uncertainty	 5 to 50 °C ±0.5 °C
Duct size	Ø100 to Ø315 (mm)
Data logging and export to PC?	Yes
Data sheet for more information	http://resources.flaktwoods.com/Perfion/File.aspx?id=248d7d6e-f301-47ac-aff4-85849ac262a9

*Not mentioned in data sheet, values acquired by contacting Fläkt Group.

One drawback for this product is that it is relatively new, and only became available for mainland Europe in November 2017, thus difficult to test during the project period. We contacted Ventistål, the Norwegian supplier, and they also said it was not suitable for our purpose. Another drawback is, as for Lindab UltraLink, that it is only available in diameters up to Ø315 mm, thus excluding larger installations. It does not measure relative humidity either.

Figure 13 shows the measurement uncertainty for the airflow based on the installation and different velocities in the duct. These are valid when the damper blade opening is larger than 30 %. In addition, the installation parameter must be set according to the separate commissioning instructions. For other installations, Fläkt Woods' technical support should be contacted for support. Optivent Ultra is developed to be able to install VAV dampers for demand-controlled ventilation (DCV) closer to disturbances like bends and T-pieces, and still be able to monitor and control them in an accurate manner. The damper is installed so that the sensor is placed in front of the damper blade in the flow direction. The reference to > 2 x D is the distance from the bend or T-piece to the connection of the damper.

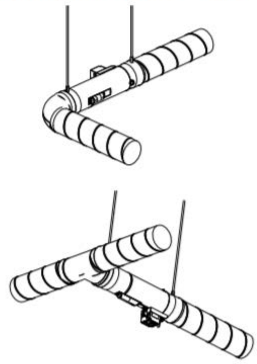
Installation	Velocity in the duct (m/s)		
	0,5 - 1	> 1	> 4
After disturbance (safety distance = 0 x D) 	± 10% or 1 l/s	± 8%	± 6%
In the straight tube (safety distance > 2 x D)	± 8% or 1 l/s	± 5%	± 4%

Figure 13 Measuring uncertainty for Optivent Ultra. (FläktGroup 2017)

Appendix B4 DEBIMO air flow measuring blades

DEBIMO air flow measuring blades (see Figure 14) averages the pressure drop over multiple points (6 per blade) in the cross section to find the airflow rate. Equipment for measuring temperature and moisture must be mounted in addition but could possibly be placed on the blades. Multiple blades can be mounted, depending on the required accuracy. The accuracy typically ranges from 3 to 5 % depending on the installation. It is a small intervention on the existing ductwork to mount the device, as it is only necessary to drill two holes on each side of the cross-section, but the blades will, however, disturb the airflow. They can be used for both circular and rectangular ducts and are available for a wide range of duct diameters (Ø100-Ø3000). The working principle is illustrated in Figure 15.

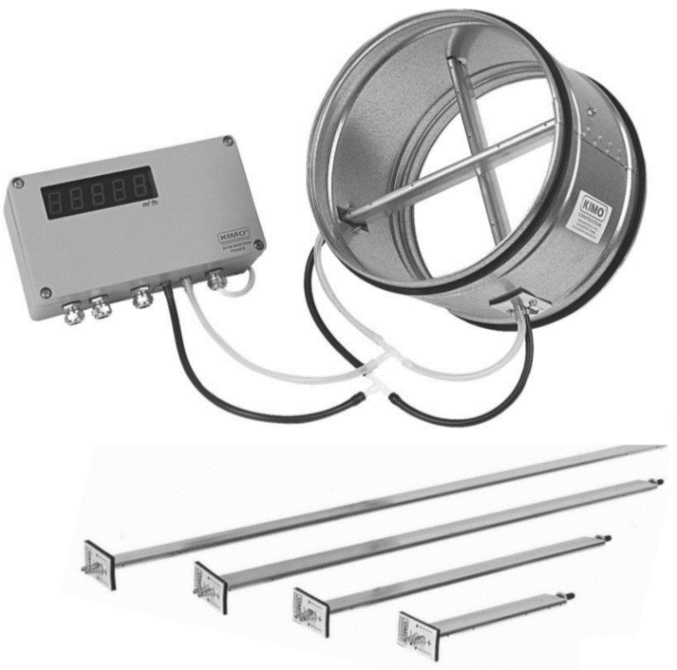


Figure 14 Picture of DEBIMO air flow measuring blades. (Kimo Constructeur)

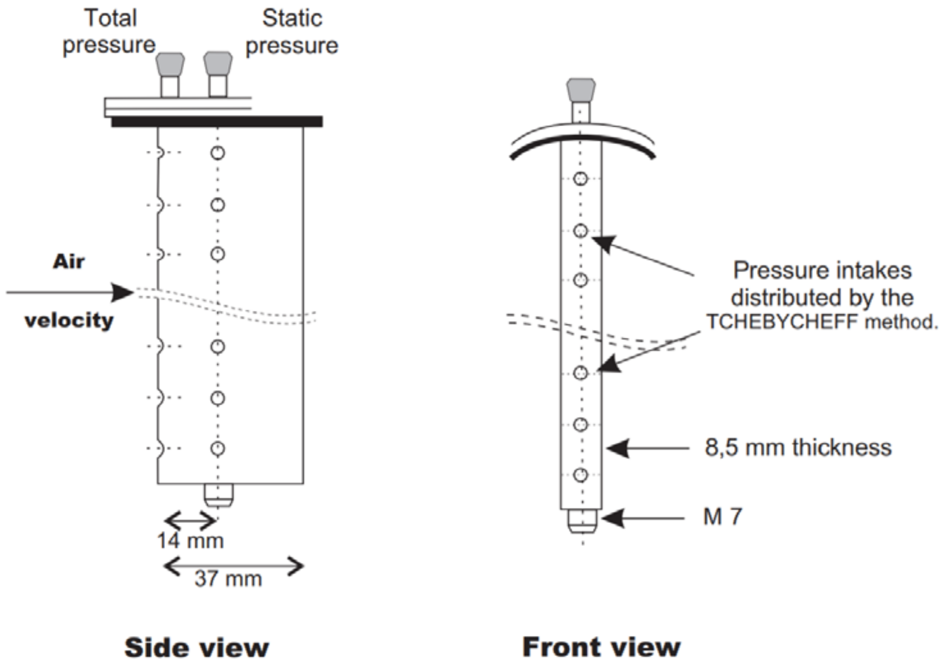


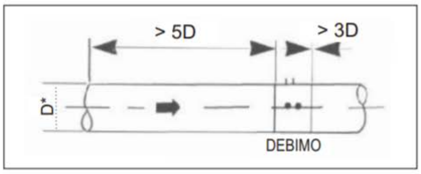
Figure 15 Working principle of Debimo air flow measuring blades. (Kimo Constructeur)

Specifications for the device can be found in Table 7. The main drawback for this device is the necessary minimum straight length from disturbances (see Figure 16), which makes it difficult to use in field measurements close to the air handling unit. The longer length of straight unobstructed duct before the DEBIMO blades the greater the accuracy. If the flow profile in the duct is not fully developed, the method of measuring the average pressure drop over 6 points instead of with individual measuring points will yield larger inaccuracy, so the minimum straight length requirements must be followed. Another drawback is that it only permits measurements of air velocities from 3 to 100 m/s. This will exclude newer installations, designed for low SFP (Specific Fan Power), with velocities lower than 3 m/s. The device also introduces pressure loss in the duct, as the blades are placed physically in the flow.

Table 7 Specifications for DEBIMO air flow measuring blades. (Kimo Constructeur)

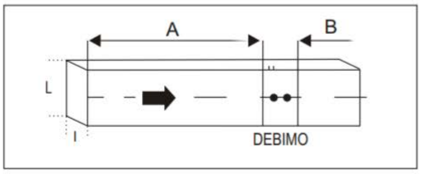
Specification	**DEBIMO air flow measuring blades**
Velocity Range Uncertainty for flow	 3-100 m/s ±3-5 % depending on the installation
Temperature	Not measured
Duct size	Ø100 to Ø3000 (mm)
Data logging and export to PC?	Yes, with a differential pressure measuring instrument (in addition to the blades)
Data sheet for more information	http://www.instrumentcompaniet.no/files/KIMO_Databla d/DEBIMO.pdf

Circular duct



Mounting of a DEBIMO measuring system in a horizontal duct.
Before the DEBIMO, safety distance : 5 x D*
After the DEBIMO, safety distance : 3 x D*
Following NF X 10-114 norm.
*D = duct diameter in m.

Rectangular duct



Mounting of a DEBIMO measuring system in a horizontal duct.
Before DEBIMO, safety distance :

$$A > 5 \times \sqrt{\frac{4 \times L \times l^*}{\pi}}$$

After DEBIMO, safety distance :

$$B > 3 \times \sqrt{\frac{4 \times L \times l^*}{\pi}}$$

Following the NF X 10-114 norm.

* with L and l in m (length and width of duct).

Figure 16 Necessary minimum straight length for DEBIMO air flow measuring blades. (Kimo Constructeur)

Appendix B5 Products discovered too late to be properly evaluated

Air Monitor seems to be a relevant supplier of measuring equipment. They were discovered very close to the deadline for the report, so it was not enough time to evaluate their products properly within this project. Some information is still included in this subchapter, so that interested readers can investigate the products further on their own.

For measuring airflow at or inside the air handling unit, they have two instruments suitable for fan inlet measurements:

- ELECTRA-Flo/FI Thermal Fan Inlet Airflow Measurement Probe
  Dual point thermal dispersion technology sensors mounted directly to the fan's inlet bell mouth.
- VOLU-Probe/FI Pitot Fan Inlet Airflow Measurement Probe
  Pair of offset traverse probes mounted directly to the fan's inlet bell mouth.

These products can be found here for more information:
https://www.airmonitor.com/application/fan-inlet-measurement-inside-air-quality/

For ducted airflow measurements, they have these five options:

- ELECTRA-Flo Thermal Airflow Measurement Probe Array
  Utilizes thermal dispersion technology in multi-point probes to measure average airflow and temperature.
- ELECTRA-Flo/CM Thermal Airflow Measurement Station
  Combines thermal dispersion measurement technology with an integral honeycomb cell air straightener.
- VOLU-Probe Pitot Airflow Measurement Traverse Probe
  Multiple Pitot total and static pressure sensing ports to traverse each duct cross-section.
- VOLU-Probe/VS Pitot Airflow Measurement Traverse Station
  Utilizes one or more VOLU-probes, factory mounted in a welded, galvanized casing.
- FAN-E Pitot Airflow Measurement Station
  Multi-point, self-averaging Pitot traverse airflow measurement station with an integral air straightening honeycomb cell.

More information on these products can be found here:
http://www.airmonitor.com/application/ducted-airflow-measurement/

As demand-controlled ventilation is becoming increasingly popular in new buildings, several VAV dampers have been put on the market. Trox LVC (link: https://www.trox.de/en/vav-terminal-units/type-lvc-216c413ab44ae408). is available in Ø125-250, has a case length of 310 mm and is optimised for airflow velocities from 0.6 to 6 m/s. A nozzle is measuring the pressure on each side of the damper blade (both upstream and downstream), then a controller determines the resulting differential pressure and compares it to the stored characteristic to find the volume flow rate. The upstream conditions do not have to meet any special requirements, i.e. the device can be mounted close to disturbances. There is a development towards such products that can measure low airflow rates in ducts with high accuracy and no maintenance need. These products are usually mounted far from the air handling unit, and are currently only available in small duct dimensions as they are meant for control at the zone level of demand-controlled ventilation. The measurement techniques are probably transferable for potential new products for determining the real efficiency of rotary heat exchangers.

CTF878 clamp on gas meter is using the ultrasonic flowmeter principle, which is suitable for metal and plastic ducts and pressure as low as atmospheric. It has no wetted or moving parts and it is non-invasive to the duct. The CTF878 flow meter measures the upstream and downstream disturbance with two pairs of ultrasonic transducers. These clamp-on gas transducers can produce signals that are five to ten times more powerful than the traditional ultrasonic transducers. The CTF878 is able to measure the air velocity with the range from 1.1 to 46 m/s with ±2 % of the reading. The moisture in the air influences the propagation of ultrasonic waves. The effects of moist or saturated air on the airflow measurement should be accounted for. More information about the clamp-on ultrasonic airflow measuring equipment can be found at http://www.rshydro.co.uk/flow-meters/clamp-on-gas-flow-meters/digitalflow-ctf878-flowmeter/

Appendix B6 Summary of available measuring equipment

Performing the field measurements proved to be more difficult than expected, due to a lack of suitable measuring equipment. Table 8 presents the same specifications as the previous chapters for comparison, but here only the newest model of VelociCalc is included.
Equipment from Appendix B5 is not included.

Table 8 Specifications for the different measurement devices.

		VelociCalc 9565-P	Lindab UltraLink	Optivent Ultra	Debimo blades	Tracer gas
Airflow measurement method		Velocity traverse	Ultrasonic airflow measurement	Ultrasonic airflow measurement	Differential pressures	Measure concentrations and calculate from mass balance
Velocity	Range	0-50 m/s	0.2-15 m/s	0.5-15 m/s	3-100 m/s	NA
	Uncertainty	±3 % of reading or ±0.015 m/s, whichever is greater	±5 % of reading or ±X* l/s, whichever is greater	See Figure 13	±3-5 %	Airflow measurement uncertainty mainly depends on the mixing level
	Resolution	0.01 m/s	NA	NA	NA	NA
Temperature	Range	-10 to 60 °C	-10 to 50 °C	5 to 50 °C	NA	NA
	Uncertainty	±0.3 °C	±1.0 °C	±0.5 °C	NA	NA
	Resolution	0.1 °C	NA	NA	NA	NA
Relative humidity		Yes, see Table 3	No	No	No	No
Automatic logging		Yes (in 1 point)	Yes	Yes	With additional equipment	For tracer concentration, yes
Duct size		25 to 12700 mm	Ø100-Ø315	Ø100-Ø315	Ø100-Ø3000	Any size
Tested in this project		Yes	Yes	No	No	Yes
Strength		Easy to implement and understand	Accurate, low uncertainty for low air velocity, intuitive, no interference on the flow		Available for all duct sizes, easy to implement and understand	Able to determine the main and parasitic airflows at the same time, relatively accurate
Drawback		Measurement uncertainty is high for low velocity and developing flow, require manual labour	Difficult to install in the existing duct for pre-fabricated airflow measurement device		Fully developed flow is very important, disturbs the airflow	Need good mixing which is almost impossible in AHU, need specialized knowledge to calculate the airflows

[1]The uncertainty of VelociCalc cannot be directly compared with the others, as the probe is only part of it. The method uncertainty of ±4 or ±6 % must be added.

[2]X equals the diameter in dm, for instance Ø100 => 1 l/s and Ø200 => 2 l/s

VelociCalc can be used for a broad range of measurements, as it can measure velocity, temperature, and relative humidity within large ranges and with high accuracy. The main drawbacks of this device are that it disturbs the airflow and that it only measures in one point in the duct at the time. Thus, several measurements are needed to achieve accurate measurements with the duct traverse method, as the velocities will vary through the cross-section of the ducts (aside from long straight ducts). If the measurements are performed with the same instrument, the different points will be measured at different times, which then require stable flow and temperature conditions, while if the measurements are performed with several devices; this will affect the airflow increasingly for each device introduced in the duct.

The ultrasonic measurement devices, Lindab UltraLink and Optivent Ultra, seem to be highly accurate even at small distances from disturbances and the measurements can be logged without manual labour once mounted. Another important advantage is that they do not disturb the airflow, not incurring on pressure drop and increased energy use for fans. However, these devices are still under development and they are not available in all sizes. Additionally, they require relatively large interventions in existing ductwork. As Optivent Ultra is primarily used as a damper, it is also likely not to be installed close enough to the heat exchanger to be used to determine the efficiency.

The Debimo blades are available for virtually all duct sizes (Ø100-Ø3000) and can log the measurements with additional equipment installed. Fully developed flow is, however, a prerequisite for this device to be accurate, so the minimum straight length requirements must be followed. The blades will disturb the airflow, which is a drawback, but the blades may be suitable for mounting sensors for temperature and relative humidity, which is an advantage.

Tracer gas dilution techniques is also an accurate method for determining airflow rates and especially internal leakage rates. It could also be a good method if it is difficult to get access to the system. One disadvantage is that it is difficult to find gases with good properties for the measurements that also are environmentally friendly. This method also requires a minimum straight length of the duct to ensure proper mixing, unless measures are introduced to enhance the mixing process. In addition, these measurements are not automated and require manual labour, which is time-consuming and the evaluation of data could be complex.

Neither of these devices is suitable for directly measuring the field efficiency of rotary heat exchangers. Based on our study, we could not find any ideal equipment that can be used and moved in different air handling units. However, we see a development in new products in the market and we hope soon new products may be available. The measurement equipment must thus be chosen based on duct diameter, available space to perform the measurements and minimum straight duct lengths before and after disturbances.

Appendix C Lab and field measurements

Appendix C1 Preliminary lab measurements with VelociCalc

The purpose of these measurements was to test the velocity traverse method, and answer the following questions:

- What is the minimum number of points that need to be measured when the flow is fully developed to measure accurately? What is the minimum number of measuring points if the flow is not fully developed?
- Will the precision of the velocity traverse method increase by introducing additional measuring points? Will it be reduced by removing measuring points?

Methods

The measurements were performed with two equipment at the same time:

- One measuring device, VelociCalc 9565-P, in Figure 10 was installed in the middle of the duct. This measures constantly logging velocity and temperature.
- A second device, VelociCalc 8388, in Figure 17 was moved within cross section of the duct to measure in the multiple points. The cross section was a couple of centimetres upstream of the fixed measuring device.

The measurements were performed with two different equipment as only one device of each type was available. However, both equipment was calibrated against each other with a minimum error. The instrument uncertainty is ±3 % of reading or ±0.02 m/s (VelociCalc 8388) or ±0.015 m/s (VelociCalc 9565-P), whichever is greater. The method uncertainty is ±4 %. For the temperature measurements, both VelociCalc 9565-P and VelociCalc 8388 have uncertainties of ±0.3 °C.

The duct diameter was 315 mm, and the airflow faces downward at the measuring point. The setup is shown in Figure 17. There is a bend 650 m upstream the measurement point, thus there is roughly 2·D straight length duct before the measuring point, which means that the flow is probably not fully developed. The duct was perforated with two holes to introduce the measurement equipment. The wholes are kept taped when not used to reduce the error in the measurement.

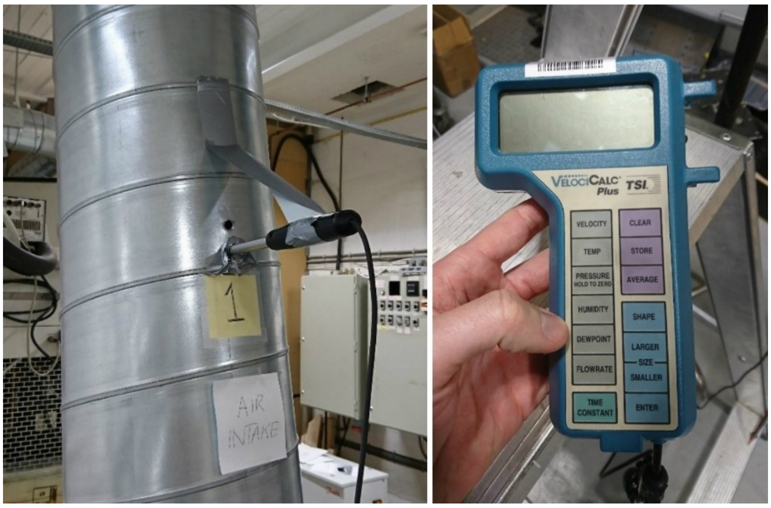
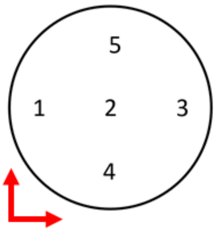
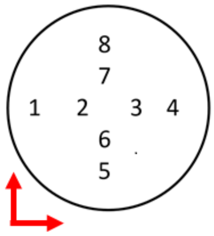


Figure 17 Picture of the lab setup (left) with the VelociCalc 9565-P mounted in the centre of the duct and picture of the VelociCalc 8388 (right).

Five and eight points were measured according to Figure 18. In order to reduce random measurement error, the velocities and temperatures were found as an average of five consecutive measurements of steady-state values.

Measurement point	Coordinates (x,y)
1	(50,158)
2	(158,158)
3	(265,158)
4	(158,50)
5	(158,265)





Measurement point	Coordinates (x,y)
1	(50,158)
2	(104,158)
3	(211,158)
4	(265,158)
5	(158,50)
6	(158,104)
7	(158,211)
8	(158,265)

Figure 18 Principal sketch of the measuring points distributed in the cross-section of the duct.

NS-EN 16211:2015 suggests five measuring points for a 315 mm duct, but in slightly different positions, at 32, 158 and 283 mm from the edge of the duct. For VelociCalc 8388, there was a limitation on the placement of the sensor (it couldn't be placed closer than 40 mm from the duct wall) and thus the values stated in Figure 18 were used so that the measurements were 50 mm from the duct wall.

Results

The reference airflow rate was calculated by measuring the pressure difference over an orifice plate with an orifice opening of 215 mm. The pressure difference was measured with a TT series micromanometer from DP Measurement. Table 9 presents the results from the measurements, which were conducted at three different airflow rates.

Table 9 Result of the three measurements tests. Measured velocities and temperatures from the lab for three different airflow rates and five and eight measurement points.

Flow rate number		1		2		3	
5 measuring points		Vel. [m/s]	Temp. [°C]	Vel. [m/s]	Temp. [°C]	Vel. [m/s]	Temp. [°C]
1	(50,158)	2.00	25.7	1.69	25.1	1.51	26.5
2	(158,158)	1.64	26.1	1.63	25.0	1.32	26.6
3	(265,158)	2.30	26.1	2.21	24.8	1.85	26.7
4	(158,50)	2.41	25.8	2.14	26.1	1.83	26.3
5	(158,265)	1.57	25.8	1.34	26.1	1.22	26.3

8 measuring points

1	(50,158)	2.00	25.7	1.69	25.1	1.51	26.5
2	(104,158)	1.70	26.0	1.55	26.1	1.41	26.8
3	(211,158)	1.62	26.0	1.84	26.2	1.55	27.0
4	(265,158)	2.30	26.1	2.21	24.8	1.85	26.7
5	(158,50)	2.41	25.8	2.14	26.1	1.83	26.3
6	(158,104)	2.04	26.0	1.88	26.2	1.61	26.8
7	(158,211)	1.52	26.1	1.33	26.3	1.09	26.9
8	(158,265)	1.57	25.8	1.34	26.1	1.22	26.3

For the first measurement with the highest velocity, the flow is not fully developed. There is a difference of 35% between the largest and smallest air velocity in the cross section. For lower airflow rates (test 2 and 3) the difference is even larger. The distribution of maximums and minimums on the three cases is not equal for the different airflow rates. Thus, the flow is not fully developed for any of the measurements.

The velocities, temperature and relative humidity measured from the VelociCalc 9565-P in the centre of the duct also showed large variations over time. presents the results from the first two flow rates, but unfortunately, data from the third measurement is not available. It is clear that the flow conditions are not stable in the test rig. Measuring in only one point would have not been enough and in order to account for the large differences in the airflow rates, but it is proven that constant measuring is needed.

Table 10 Results from VelociCalc 9565-P (single point) logging in the centre of the duct.

	Flow rate 1			Flow rate 2		
	Velocity (m/s)	Temp. (°C)	RH (%)	Velocity (m/s)	Temp. (°C)	RH (%)
Average	1.70	25.6	29.1	1.61	25.8	28.5
Minimum	1.35	24.7	28.7	1.31	25.2	28.0
Maximum	2.18	25.8	30.5	2.00	26.1	29.6

In a long straight section of the duct, an orifice plate was mounted according to standard test conditions to find the reference airflow rate. These values were compared to the measurements with the traverse method. In the traverse, the airflow rates were calculated multiplying the measured air velocity in the different points, by their corresponding areas in the sector of the cross-section of the duct. The results are presented in Table 11.

Table 11 Calculated airflow rates and deviation compared to airflow rate calculated based on pressure difference over orifice plate.

Flow rate number	1	2	3	
	Calculated airflow rates [m³/h]			
Orifice plate	582	533	442	
VelociCalc 5 points	567	511	440	
VelociCalc 8 points	555	503	436	
VelociCalc centre point	477	452	-	
	Deviation from orifice plate			Average deviation
5 points	-2.7 %	-4.2 %	-0.4 %	-2.4 %
8 points	-4.7 %	-5.6 %	-1.4 %	-3.9 %
VelociCalc centre point	-18.1 %	-22.4 %	-	-20.2 %

When comparing the calculated airflow rates found with the velocity traverse method to the ones measured with the orifice plate, the average deviation is -2.4 % for the five measuring points, -3.9 % for the eight measuring points and -20.2 % for the VelociCalc mounted in the centre of the duct. The VelociCalc systematically measures lower values than the orifice plate, but the deviation is quite low when using 5 or 8 points. Based on these results, the precision of the velocity traverse method did not increase by increasing the number of measuring points from 5 to 8 points. A bit surprisingly, the deviation rather increased by increasing the measuring point from 5 to 8. The placement of the eight points does not touch the centre where the velocity is highest thus underestimating the airflow rates. Based on this test, at least one point should measure in the centre to compensate for the low velocities near the duct. It is clear that only measuring the velocity in the centre of the duct is insufficient to calculate the airflow rate accurately.

Appendix C2 Lab measurements with Lindab UltraLink and VelociCalc 8388

The aim of this test was primarily to test Lindab UltraLink under different conditions and compare the measurements with a reference airflow measured with an orifice plate. The velocity traverse method was also tested for comparison in the same setup.

Methods

The tests were performed with one intake fan and measurements on one the supply air duct for the test rig in the lab. The fan impulses air through Ø500 mm duct, to the rotary heat wheel, then through a bend, reducer of the duct size to Ø200 mm and at last a 90° bend before a long straight duct with the UltraLink and an orifice plate (80 mm) mounted as shown in Figure 19. The velocity traverse measurements with VelociCalc 8388 were performed at this long straight section.

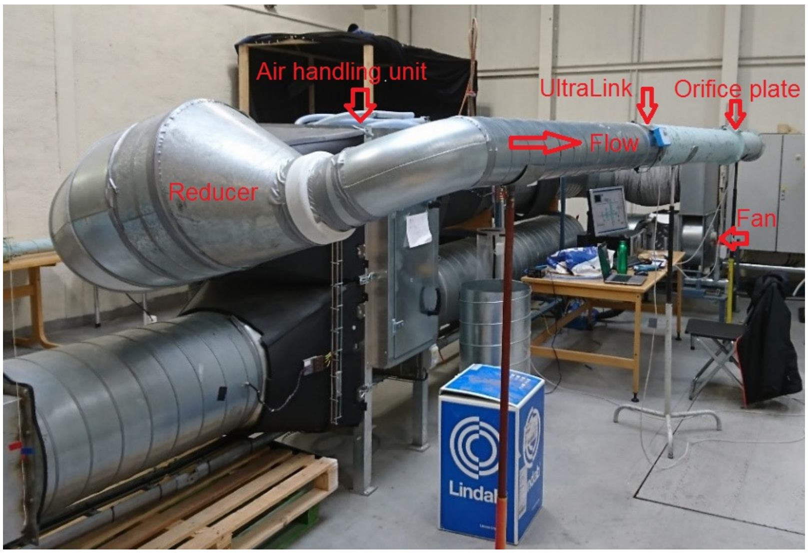
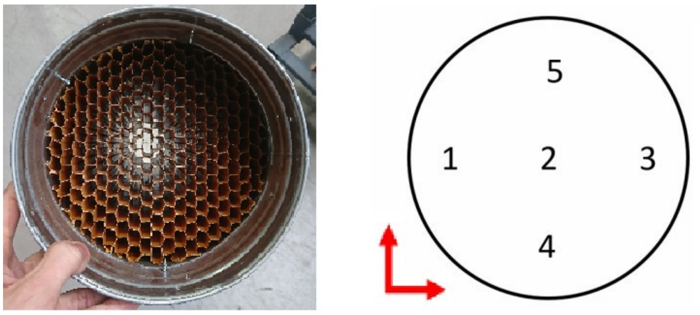


Figure 19 Picture of the lab setup for testing Lindab UltraLink and VelociCalc 8388 to compare to calculated values found from the pressure difference over an orifice plate.

Tests were performed for five different airflow rates, with different distances to the measuring equipment, different orientations of the UltraLink and with and without flow straighter[1] (see Figure 20). The different test configurations and procedures are listed below, where the provided distances are in the long straight duct measured from the bend:

- Test 1: 1.27 m to VelociCalc, 1.50 m to UltraLink, 3.75 m to the orifice plate, measurements both with and without additional honeycomb between bend and VelociCalc.
- Test 2: 0.85 m to VelociCalc, 0.35 m to UltraLink and 4.12 m to the orifice plate.
- Test 3: 0.28 m to VelociCalc, 0.35 m to UltraLink and 4.12 m to the orifice plate.
- Test 4: 0.05 m to VelociCalc, 0.12 m to UltraLink and 4.00 m to the orifice plate.



Measurement point	Coordinates (x,y)
1	(20,100)
2	(100,100)
3	(180,100)
4	(100,20)
5	(100, 180)

Figure 20 Left: Flow straighter (honeycomb) and right: measuring points for the velocity traverse method.

For the traverse method with VelociCalc 8388, five measurement points were used, according to Figure 20. Point 2 is in the centre, while points 1, 3, 4 and 5 are all situated 20 mm from the perimeter of the Ø200 duct. To measure closer for the duct, four holes were drilled, to be closer to the opposite duct wall. The average value was taken from several constant measurements. The hot wire sensor is always ninety degrees to the flow. As it is not possible to measure in all

[1] The use of the flow straighter can shorten the required straight length.

points simultaneously, and the flow is not 100 % constant, an extra uncertainty needs to be accounted for in the measurements. This uncertainty was reduced by reading the pressure difference over the orifice plate both before and after the velocity traverse measurements and control that it had not changed. The pressure difference over the orifice plate was very stable at low airflow rates, while varying more for higher airflow rates. The results from Lindab UltraLink were more stable.

The terms for UltraLink orientation; *side*, *inner radius* and *outer radius* refer to the placement of the first flow sensor on the UltraLink as shown in Table 5. Table 5 shows the specified uncertainties for the three orientations as well though these only apply at the provided distances from bends. Test 2-4 in this chapter are performed with lower than recommended distances from the bend, so they are not directly comparable. It is assumed that the two options for the orientation *side* would give the same result. For each orientation of the UltraLink, new measurements have been performed with the orifice plate and VelociCalc. The measurements are specified in Table 12-Table 15.

For each of the four tests, measurements were performed at five different airflow rates. For easier visualisation of the results, the deviations for UltraLink and VelociCalc 8388 relative to the orifice plate have been colour-coded:

- Deviations between 0-10 % are coded green
- Deviations between 10-20 % are coded orange
- Deviations above 20 % are coded red

Additionally, the dark green background shows which of the two devices has the lowest deviation compared to the orifice plate.

As the lowest airflow rate, i.e. measurement #1 in the tables below is in the range 0.40-0.60 m/s, the instrument uncertainty for these measurements is ±0.02 m/s, while measurement #2-5 will have an instrument uncertainty of ±3 % of the reading. The UltraLink measurements have the instrument uncertainties provided in Table 5 depending on the orientation of the first sensor and distance from the bend. For measurements with a minimum of 5 diameters straight duct length (in this case 1 meter), the uncertainty is either ±5 % of reading or ±2 l/s (i.e. ±7.2 m³/h) for Ø200 duct. For test nr 1, this results in an instrument uncertainty of ±7.2 m³/h for measurement #1-2 and ±5 % of the reading for measurement #3-5. For test 2-4, the distance from the bend to the UltraLink is shorter than specified in the data sheet, so the instrument uncertainty cannot be quantified, but it will at least be higher than the specified values for the shortest distance, i.e. >±5 % for the orientation at the *inner radius*, >±10 % for the *side* and >±20 % for the *outer radius*. As ultrasound measurements is a relatively new method, we have not found documentation determining the method to calculate its uncertainty. The system error for the orifice plate ranged from ±0.5-1.7 % for all measurements.

Results

Table 12 presents the results from test 1. Test 1 compares the measurements of the UltraLink and VelociCalc 8388 to the orifice plate. It also studies the effect of the honeycomb. The distances from the bend were 1.27 m to VelociCalc, 1.50 m to UltraLink, 3.75 m to the orifice plate, and measurements were performed both with and without an additional honeycomb between the bend and VelociCalc. For both of these tests, the UltraLink was oriented to the "side" relative to the duct.

Table 12 Results from test 1, showing calculated flow rate over orifice plate, measured values for UltraLink and VelociCalc 8388 and deviation from calculated value over orifice plate.

UltraLink orientation*	Measurement #	1	2	3	4	5	Mean	Standard deviation
	Calc. flow rate [m³/h]	58.2	129.8	201.8	273.8	346.8	-	
Side	Lindab UltraLink [m³/h]	58	131	205	280	354	-	
	VelociCalc 8388 [m³/h]	60.1	128.1	196.3	269.4	343.5	-	
with	Deviation UltraLink	-0.3 %	0.9 %	1.6 %	2.3 %	2.1 %	1.3 %	1.1 %
honeycomb	Deviation VelociCalc 8388	3.2 %	-1.3 %	-2.7 %	-1.6 %	-1.0 %	-0.7 %	2.3 %
	Calc. flow rate [m³/h]	59.9	133.7	206.3	281.2	354.1	-	
Side	Lindab UltraLink [m³/h]	56	126	196	267	337	-	
	VelociCalc 8388 [m³/h]	62.7	136.4	205.4	281.3	355.1	-	
without	Deviation UltraLink	-6.5 %	-5.8 %	-5.0 %	-5.0 %	-4.8 %	-5.4 %	0.7 %
honeycomb	Deviation VelociCalc 8388	4.8 %	2.0 %	-0.4 %	0.0 %	0.3 %	1.3 %	2.1 %

*The terms for UltraLink orientation refer to the placement of the first flow sensor on the UltraLink as previously described in Table 5

The results show that measurements with the UltraLink and VelociCalc agree very well ones with the ones with orifice plate when the honeycomb was used and there was enough duct length to have a fully developed flow. In general, the UltraLink slightly overestimated the airflow rate (+1,3%), while the VelociCalc slightly underestimated the airflow rate(-0.7 %).

When the flow straighter was removed, the airflow rate over the orifice plate increased. The average deviations for the UltraLink and VelociCalc increased compared to the orifice plate. The UltraLink underestimated the airflow rates in average -5.4 %, though this deviation may still be considered acceptable. The traverse method with VelociCalc overestimated the airflow relative to the orifice plate. The deviations were highest (+4.8 %) for the lowest airflow rate, and extremely low for the three highest airflow rates (±0.4 %). Regardless of whether the honeycomb was used or not, the VelociCalc overestimated the airflow rate at the lowest airflow rate, while the deviations were lowest and quite accurate at higher airflow rates.

Table 13 provides results from Test 2, where the additional honeycomb from test 1 was not used. The goal of this test was to evaluate the effect of the different possibilities to install the UltraLink. The distances were 0.85 m to VelociCalc, 0.35 m to UltraLink and 4.12 m to the orifice plate. The measurements with the VelociCalc is for this case not horizontal/vertical, as the holes for the velocity traverse method were slightly rotated (around 20°). It was assumed that this would not matter, but NS-EN 16211:2015 says that the method uncertainty will increase from ±4 to ±6 % for these cases.

Table 13 Results from test 2, showing calculated flow rate over orifice plate, measured values for UltraLink and VelociCalc 8388 and deviation from calculated value over orifice plate.

UltraLink orientation*	Measurement #	1	2	3	4	5	Mean	Standard deviation
Side	Calc. flow rate [m³/h]	59.9	132.1	205.8	277.5	352.6	-	
	UltraLink [m³/h]	59	126	195	267	336	-	
	VelociCalc 8388 [m³/h]	65.5	137.4	210.1	283.4	359.0	-	
	Deviation UltraLink	-1.5 %	-4.6 %	-5.2 %	-3.8 %	-4.7 %	-4.0 %	1.5 %
	Deviation VelociCalc 8388	9.3 %	4.0 %	2.1 %	2.1 %	1.8 %	3.9 %	3.2 %
Inner radius	Calc. flow rate [m³/h]	59.9	132.1	205.3	278.6	350.9	-	
	UltraLink [m³/h]	50	112	179	244	307	-	
	VelociCalc 8388 [m³/h]	64.0	135.9	212.0	283.6	359.9	-	
	Deviation UltraLink	-16.5 %	-15.2 %	-12.8 %	-12.4 %	-12.5 %	-13.9 %	1.9 %
	Deviation VelociCalc 8388	6.8 %	2.9 %	3.3 %	1.8 %	2.6 %	3.5 %	1.9 %
Outer radius	Calc. flow rate [m³/h]	59.9	132.1	205.3	277.9	351.4	-	
	UltraLink [m³/h]	52	110	172	237	300	-	
	VelociCalc 8388 [m³/h]	62.1	136.1	205.3	280.3	358.4	-	
	Deviation UltraLink	-13.2 %	-16.7 %	-16.2 %	-14.7 %	-14.6 %	-15.1	1.4 %
	Deviation VelociCalc 8388	3.8 %	3.0 %	0.0 %	0.9 %	2.0 %	1.9	1.5 %

*The terms for UltraLink orientation refer to the placement of the first flow sensor on the UltraLink as previously described in Table 5.

For Test 2, the distance from the bend to the UltraLink is lower than specified in the data sheet (Table 5). The results show that the deviation for the UltraLink compared to the orifice plate is lowest when the UltraLink is oriented towards the *side*. This contrasts with the data sheet specifications for longer distances from bends, whereas the orientation of the first sensor on the inner radius is the most accurate. When oriented towards the *side*, the average deviation was -4.0 %. When oriented towards the *inner radius* and *outer* radius respectively, the average deviation was -13.9 % and -15.1 %, i.e. considerably higher. For all measured orientations, the UltraLink underestimates the airflow compared to the orifice plate.

The deviation for the VelociCalc to the orifice plate measurements was lower than for UltraLink, except for the lowest airflow rate when the UltraLink was oriented towards the side. In these tests, The UltraLink was placed 0.35 m away from the bend and followed by the holes for the VelociCalc 0.85 m from the bend. Thus, the comparison of both equipment is not completely fair as the distance from the disturbances is not the same for both measurements. The deviation for the VelociCalc is highest at the lowest airflow rates, and quite accurate at the higher airflow rates. The VelociCalc systematically overestimates the airflow rate compared to the orifice plate measurements. The deviations indicate that the measurements with the VelociCalc are acceptable for this setup.

Table 14 provides results from Test 3 (see the setup in Figure 21), where the distances were 0.28 m to VelociCalc, 0.35 m to UltraLink and 4.12 m to the orifice plate, i.e. reduced distances for VelociCalc.

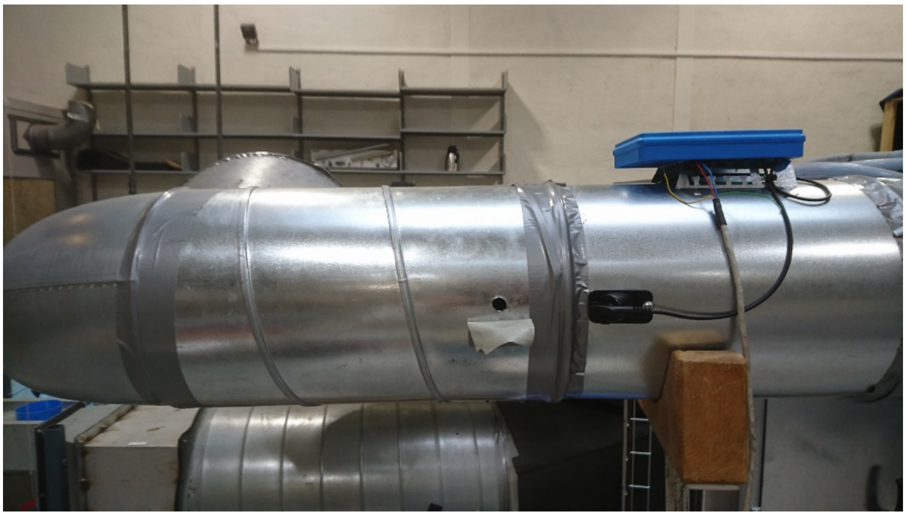


Figure 21 Picture of the setup for test 3. The holes for VelociCalc 9565-P is closest to the bend and the first UltraLink sensor is oriented on the "outer radius" relative to the bend.

The main difference between test 2 and test 3 is that the VelociCalc measurements are performed closer to the bend for test 3, at 0.28 m, instead of 0.85 m as for Test 2. The traverse measurements were performed horizontally/vertically (not angled) so that the method uncertainty is reduced. The reduced distance from the bend has led to an increase in the average deviation for all the measurements with the VelociCalc. Only one out of the 15 measurements had a slightly lower deviation, which indicates that the VelociCalc is systematically overestimating the airflow rate at shorter distances from the bend. The measurements with UltraLink oriented towards the *side* are already "close enough" to the measured value, while the measurements with the *inner radius* are quite stable (-11.6 to -13.5 %), but the measurements with the *outer radius* are more unstable and unreliable (-12.8 to -19.9 %) and should be avoided for small distances to disturbances.

Table 14 Results from test 3, showing calculated flow rate over orifice plate, measured values for UltraLink and VelociCalc 8388 and deviation from calculated value over orifice plate.

UltraLink Orientation*	Measurement #	1	2	3	4	5	Mean	Standard deviation
Side	Calc. flow rate [m³/h]	58.2	130.6	202.8	274.9	347.6	-	
	UltraLink [m³/h]	56	124	198	268	340	-	
	VelociCalc 8388 [m³/h]	63.6	140.3	215.0	285.8	354.1	-	
	Deviation UltraLink	-3.8 %	-5.1 %	-2.4 %	-2.5 %	-2.2 %	-3.2 %	1.2 %
	Deviation VelociCalc 8388	9.2 %	7.4 %	6.0 %	4.0 %	1.9 %	5.7 %	2.9 %
Inner radius	Calc. flow rate [m³/h]	58.2	130.6	202.8	275.3	347.3	-	
	UltraLink [m³/h]	51	113	178	243	307	-	
	VelociCalc 8388 [m³/h]	63.6	138.2	211.4	280.4	361.9	-	
	Deviation UltraLink	-12.4 %	-13.5 %	-12.2 %	-11.7 %	-11.6 %	-12.3 %	0.8 %
	Deviation VelociCalc 8388	9.2 %	5.8 %	4.3 %	1.9 %	4.2 %	5.1 %	2.7 %
Outer radius	Calc. flow rate [m³/h]	59.9	130.6	202.8	275.3	347.6	-	
	UltraLink [m³/h]	48	108	173	237	303	-	
	VelociCalc 8388 [m³/h]	67.2	139.3	207.6	292.1	367.4	-	
	Deviation UltraLink	-19.9 %	-17.3 %	-14.7 %	-13.9 %	-12.8 %	-15.7 %	2.9 %
	Deviation VelociCalc 8388	12.3 %	6.7 %	2.4 %	6.1 %	5.7 %	6.6 %	3.6 %

*The terms for UltraLink orientation refer to the placement of the first flow sensor on the UltraLink as previously described in Table 5.

For test 3, the deviations for VelociCalc are varying for the different orientations of the UltraLink, even though they are made at the same distance from the bend. There might have been small differences in the angle of the measurements, as the duct might have been slightly rotated when rotating the UltraLink for the different orientations. The variations are still considered to be mostly caused by fluctuating airflow rates, which will reduce the accuracy of the velocity measurements. Especially the measuring points 2 and 3 (i.e. centre and inner radius of the bend) had a lot of turbulence and fluctuating velocities. Also, for this test, new holes were drilled in the duct for VelociCalc measurements. These were 11 mm, slightly larger than the previous holes of 8 mm, resulting in that more air can exit through the hole where the measurement is being performed. In general, when using the velocity traverse method, the holes for the thermoanemometer should be as close as possible to the diameter of the measurement probe. As the probe is extendable, and the diameter of the probe varies with the level of extension (i.e. low diameter at short range and a larger diameter at longer ranges), this will however not be possible for short ranges in large ducts.

Table 15 provides results from Test 4, where the distances were 0.05 m to VelociCalc, 0.12 m to UltraLink and 4.00 m to orifice plate (see Figure 22). This is close to the limit of how close it is physically possible to measure from the bend for both the VelociCalc and the UltraLink. For this test, the airflow will be highly turbulent, and unevenly distributed over the cross-section of the duct, making it very difficult to achieve accurate measurements.

Table 15 Results from test 4, showing calculated flow rate over orifice plate, measured values for UltraLink and VelociCalc 8388 and deviation from calculated value over orifice plate.

UltraLink Orientation*	**Measurement #**	**1**	**2**	**3**	**4**	**5**	**Mean**	**Standard deviation**
Side	Calc. flow rate [m³/h]	59.9	132.1	204.8	277.9	351.4	-	
	UltraLink [m³/h]	62	141	222	298	376	-	
	VelociCalc 8388 [m³/h]	64.3	143.4	218.6	293.5	363.3	-	
	Deviation UltraLink	3.5 %	6.7 %	8.4 %	7.2 %	7.0 %	6.6 %	1.8 %
	Deviation VelociCalc 8388	7.4 %	8.6 %	6.7 %	5.6 %	3.4 %	6.3 %	2.0 %
Inner radius	Calc. flow rate [m³/h]	59.9	132.1	204.8	277.9	351.4	-	
	UltraLink [m³/h]	48	98	156	215	275	-	
	VelociCalc 8388 [m³/h]	64.1	144.3	215.5	285.0	361.3	-	
	Deviation UltraLink	-19.9 %	-25.8 %	-23.8 %	-22.6 %	-21.7 %	-22.8 %	2.2 %
	Deviation VelociCalc 8388	7.0 %	9.3 %	5.2 %	2.6 %	2.8 %	5.4 %	2.8 %
Outer radius	Calc. flow rate [m³/h]	59.9	132.1	204.8	277.9	350.9	-	
	UltraLink [m³/h]	50	109	171	233	295	-	
	VelociCalc 8388 [m³/h]	62.5	143.0	212.0	279.3	358.6	-	
	Deviation UltraLink	-16.5 %	-17.5 %	-16.5 %	-16.2 %	-15.9 %	-16.5 %	0.6 %
	Deviation VelociCalc 8388	4.4 %	8.3 %	3.5 %	0.5 %	2.2 %	3.8 %	2.9 %

*The terms for UltraLink orientation refer to the placement of the first flow sensor on the UltraLink as previously described in Table 5.

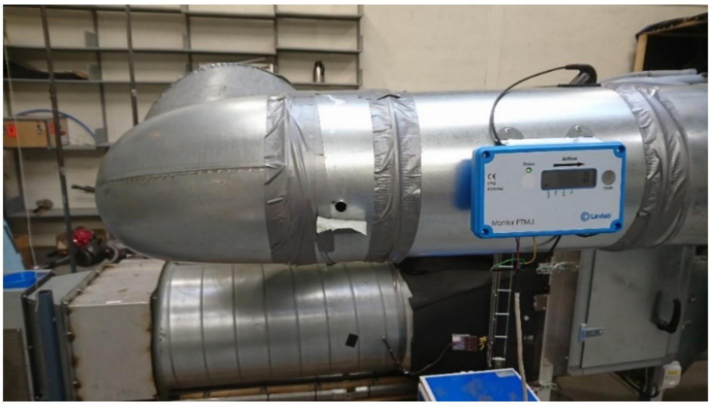


Figure 22 Picture of the setup for test 4. The holes for VelociCalc 8388 is closest to the bend and the first UltraLink sensor is oriented on the "side" relative to the bend.

The results show that the measurements with the UltraLink oriented towards the *side* are quite good (average deviation 6.6 %), albeit higher than for the previous measurements further from the bend. For this case, the UltraLink rather overestimated the airflow rate. For this case, the deviation for UltraLink is highest when the first sensor is oriented on the *inner radius* (-22.8 %), not on the *outer radius* (average deviation -16.5 %). The deviation for the UltraLink was really close to the VelociCalc when oriented towards the *side*, whereas the lowest deviation for the UltraLink was for the lowest airflow rate, while the deviation was lowest for the VelociCalc at the highest airflow rate. For the VelociCalc, the average deviation ranged from 3.8 to 6.3 %, while the deviations for every single measurement ranged from 0.5 to 9.3 %. This is very good for such a small distance to the bend.

The following documentation and information are collected from Lindab (Berg 2018). Figure 23 shows their measurements with Lindab UltraLink that form the basis for the measurement uncertainties provided in Table 5 (taken from datasheet). The tabulated values for maximum deviations after a bend for the different orientations of the first ultrasonic sensor, is based on the assumption that the lowest velocity is found at the inner radius relative to the bend. In reality, the instruction should not be that the sensor should be placed on the inner radius, but rather at the position where the velocity profile has its minimum value. Normally, the latest disturbance source, in this case the bend, is the product that decides how the disturbance of the air looks. For a case with a single bend this position coincides with the inner radius and that is the reason to why it is written like this (to simplify for the installer).

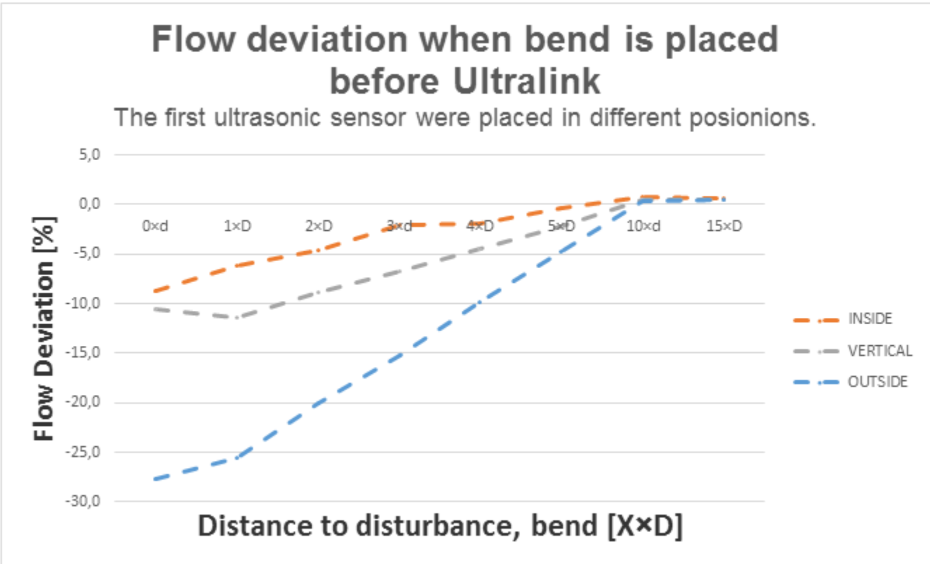


Figure 23 Documentation of flow deviations for Lindab UltraLink at different orientations of the first ultrasonic sensor and different distances from a bend. (Berg 2018)

The results presented in this chapter showed that the flow deviation of the UltraLink was lowest when the first sensor was oriented relative to the *side* of the bend, not the *inner radius* according to Lindab's experience. This deviation may be caused by that the test rig used here has a reducer (additional disturbance) right before the bend, as shown in Figure 19. In addition, there is a slight upwards angle before the bend, so the UltraLink could have been slightly rotated from the horizontal/vertical position to account for this. However, as the minimum velocities found with the velocity traversal method were found for point 3, corresponding to the position used for the *inner radius* orientation of the UltraLink, this is not considered to be of significant importance for the results.

Conclusions

The measurements performed here with the UltraLink cannot be directly compared to the uncertainties in the data sheet, as the distances from the bend were shorter than the minimum specified distance for test 2-4, and the UltraLink was only tested for one orientation in test 1. Based on our measurements at shorter distances from the bend, the best orientation for the UltraLink is at the *side* with regards to the bend, not the *inner radius* which is most accurate for longer distances according to the data sheet and Lindab's experiences.

As previously mentioned in chapter 3.2 describing the velocity traversal method, the standard NS-EN 16211:2015 specifies minimum straight duct lengths for the method of 5·D before and 2·D after the measuring plane as well as other criteria for approving the measurement plane. This means that the measurements with the VelociCalc were also below the minimum distance that for this case would be 1 meter from the bend. These test configurations are not recommended but were chosen for this project to investigate the deviations in cases similar to those that would happen in a real case and not in the lab.

Appendix C3 Tracer gas measurements at NTNU laboratory

The purpose of the tracer gas measurements is to investigate the feasibility and complexity of using tracer gas technic to determine the main and parasitic airflow (leakage) rates in AHU. The real efficiency of the rotary heat wheel (defined in Appendix A) can be calculated and the influence of recirculation can be considered. The tracer gas is studied as an alternative to the previously reported techniques as it does not need a fully developed velocity profile (though it needs fully mixing of the tracer gas). Airflows are calculated based on the diffusion of the tracer gas.

Methods

NTNU test rig is employed for the tracer gas measurement as well. The supply and extract airflows and the rotary speed of the heat wheel are controlled with LabView and kept constant during each test scenario. The test rig and tracer gas analyser are shown in Figure 24. The measurements were conducted under two different ductwork layouts: a) the heat wheel with branch duct connecting orifice plate and UltraLink device as shown in Figure 19; b) the heat wheel without branch duct connecting orifice plate and UltraLink device (see Figure 26).

Figure 24 Heat wheel test rig setup: heat wheel and duct with dosing and sampling tubes on the left; the LumaSense tracer gas analyser with sampling tubes.

Multiple tracers can be applied in tracer gas method to determine the airflows in different ducts at the same time. When the multiple tracers are not available, it is possible to use a single tracer sequentially for different ducts. The time slots between two tests should be sufficiently long to eliminate the influence from the last test. For the multiple tracer gas method, various tracers are simultaneously injected into different ducts. For this project, carbon dioxide (CO_2) and nitrous oxide (N_2O) were injected into each side of the ductworks at constant flow rates as shown in Figure 26. For the single tracer method, Carbon dioxide (CO_2) as the single-tracer was injected into the corresponding designated spots as implied in Figure 26.

A good mixing of the tracer gas is crucial to assess airflow rates in each duct branch. To achieve sufficient mixing, the distance between injection points and air sampling locations are suggested by Roulet (2008):

- 10 diameters (or duct widths) in straight ducts;
- 5 diameters if there is a mixing element such as bends, droplet catcher or a fan between injection points and the air sampling nodes.

However, the above criteria cannot be fulfilled in the test rig due to space limitations. To enhance the mixing of tracer and airflow, two-tracer gas jets injections for each point, perpendicular into the cross-section of the duct are used (see Figure 25). These tubes also have multiple holes to ensure proper mixing of the tracer gas.

Figure 25 Picture of the duct where the tracer gas is released. Two tubes with multiple holes are inserted perpendicular to the duct as marked with red circles.

The injecting and sampling nodes of carbon dioxide and nitrous oxide are illustrated in Figure 26. The sampling points are arranged as far as possible from the injection point to pursue a sufficient mixing. The flow regime during the measurements is turbulent. The swirl flow from the turbulence may transport some tracer gas upwind of the injection point. The sampling point 1 is hence used to track this upwind transportation influence. Other assumptions used in the tracer gas test are: 1) leakages through the ducts are negligible between the dosage point and the AHU; 2) External leakage through the air handling unit box is negligible compared to the recirculation flow rates.

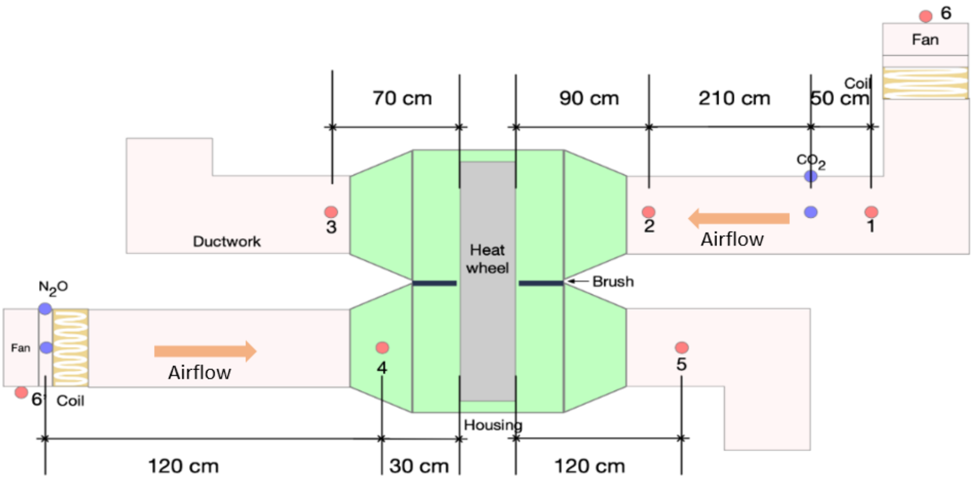


Figure 26 Principal sketch of the lab setup with ducts, injection points in purple and measuring points in red (the number of sampling points is corresponding to the number on the gas analyser channels).

The main ducts connected to the rotary heat wheel have a diameter Ø500. To verify that the airflow rates were balanced in both sides of the ductwork, the orifice plate, UltraLink and *testo 420 Flow hood* are used to measure the flow rates. The airflow rates are adjusted to be balanced by tweaking the fan frequency on both sides.

The LumaSense tracer gas analyser is used to measure the concentration of CO_2 and N_2O in multiple and single tracer methods. The analysis of tracer gases is achieved by infrared spectroscopy. CO_2 and N_2O, have some overlapping wavelength adsorption interval. The overlapping wavelength will interfere the raw signal and lead to incorrect concentration measurements. To get the correct concentration when both gases are analysed, the cross-compensation for the raw signals should be previously implemented in the tracer gas analyser. The cross-compensation function for multiple tracer gas method was not activated for the measurement in this project. Consequently, the resulting concentrations of the CO_2 and N_2O are wrong due to the interference of multiple gas measurements. Thus, only the single tracer gas method is employed to calculate the airflow rates and the recirculation in the rotary heat exchanger.

The test with installed orifice plate and UltraLink to measure the main airflow at the supply side has ducts of different length and thus pressure drop with extract side. These have to be accounted for in the balancing of the airflows. Therefore, the static pressures between the extract and supply sides are rather different. As a result, a large fraction of supply air penetrates to the exhaust side through the gaps as indicated in Figure 27. This causes malfunction of the heat wheel. It also indicates the balanced duct resistance and balanced static pressure in the extract and supply sides should be pursued in practice, to avoid big leakages in rotary heat recoveries.

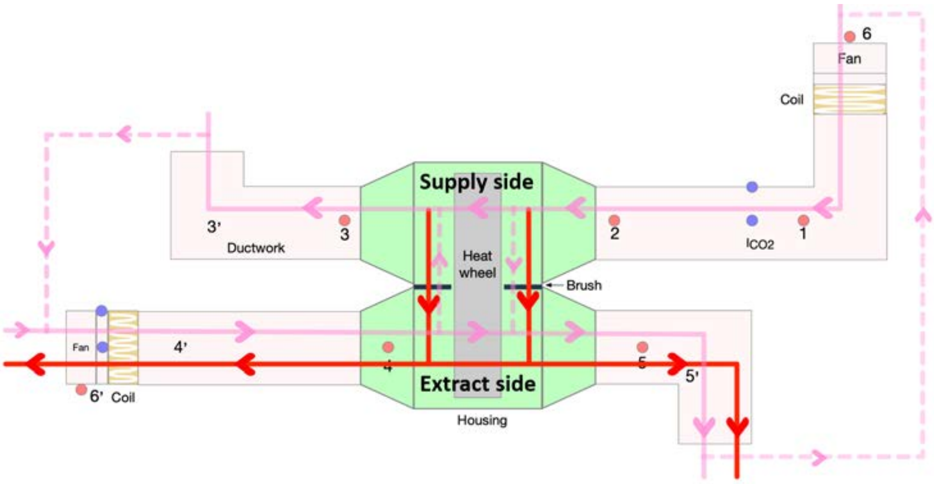


Figure 27 Schematic view of airflows in the unbalanced duct resistance. The air penetrates from the supply side to the extract side as shown in red.

In the following testing, the branch with UltraLink and orifice plate is removed to enable flows that are more balanced. As justified before, single tracer gas CO_2 is utilized. The injection and sampling configuration for single tracer are similar to the multiple tracer gas method. The difference is that the CO_2 is injected into the two injection locations successively on each side. Care is taken to maintain constant airflow rates and constant fan frequency (100 RPM) in the system. The time interval between the first run of tracer CO_2 in supply side and second run of tracer CO_2 in extract side is 24 hours in order to avoid their interference on each side. The injection rate of tracer CO_2 is a constant at 4 l/min.

The concentrations of tracer CO_2 at the six sampling points in supply and extract side are shown in Figure 28 for heat wheel with rotation speeds of 10 and 20 RPM, respectively.

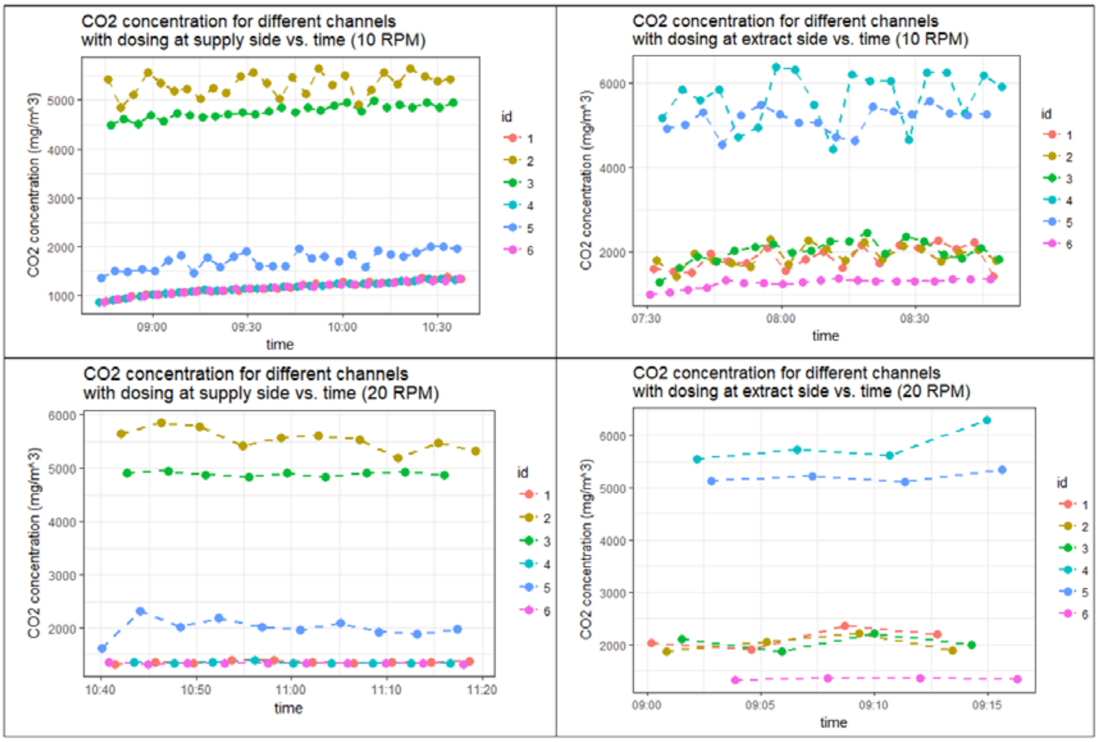


Figure 28 CO_2 concentration at various sampling points with heat wheel rotation of 10 and 20 RPM.

The concentration fluctuation of tracer CO_2 at some sampling point is rather high e.g. for point 4 with dosing at extract side with a wheel rotation speed of 10 RPM. This indicates that the mixing at these points is not good due to the insufficient mixing length. The mass conservation equations for tracer CO_2 and airflows for the measuring system can be established based on the airflows as shown in Figure 29 for each injecting and sampling point.

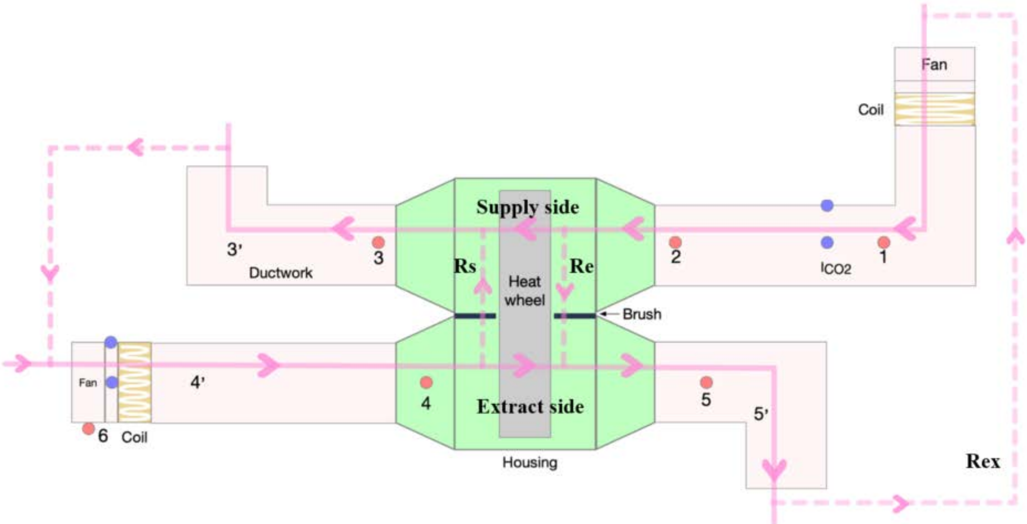


Figure 29 Nodes and airflows in the heat wheel duct with balanced duct resistance.

$$Q_{12} = \sum_{i=1}^{n} \frac{1}{n} \frac{I_{co_{2,supply}}}{C_2 - C_1} \tag{16}$$

$$Q_{64} = \sum_{i=1}^{n} \frac{1}{n} \frac{I_{co_{2,extract}}}{C_4 - C_6} \tag{17}$$

$$Q_{12} = Q_s + Q_e \sum_{i=1}^{n} \frac{1}{n} \frac{(C_{4,supply} - C_{3,supply})}{(C_{3,supply} - C_{2,supply})} \tag{18}$$

$$Q_{64} = Q_s \sum_{i=1}^{n} \frac{1}{n} \frac{(C_{5,extract} - C_{2,extract})}{(C_{4,extract} - C_{5,extract})} + Q_e \tag{19}$$

Where:

Q_{12} is volumetric airflow rate through node 1 and 2 (m^3/h)
Q_{64} is volumetric airflow rate through node 6 and 4 (m^3/h)
n is sampling times for constant tracer concentration
C is tracer concentration at different sampling points (mg/m^3)
Qs is supply recirculation flow rate (m^3/h)
Qe is supply extract flow rate (m^3/h)
R_s is supply air recirculation rate. ($Q_{43}/Q_{4'4}$)
R_e is extract air recirculation rate. (Q_{25}/Q_{12})
R_{ex} is external recirculation rate. (defined as a ratio of exhaust airflow returning to the heat recovery in outdoor air to the exhaust airflow)

When the CO_2 concentration and injection rate are substituted into the mass conservation equations, the flow rates for each duct and recirculation flow rates are calculated as shown in Figure 30.

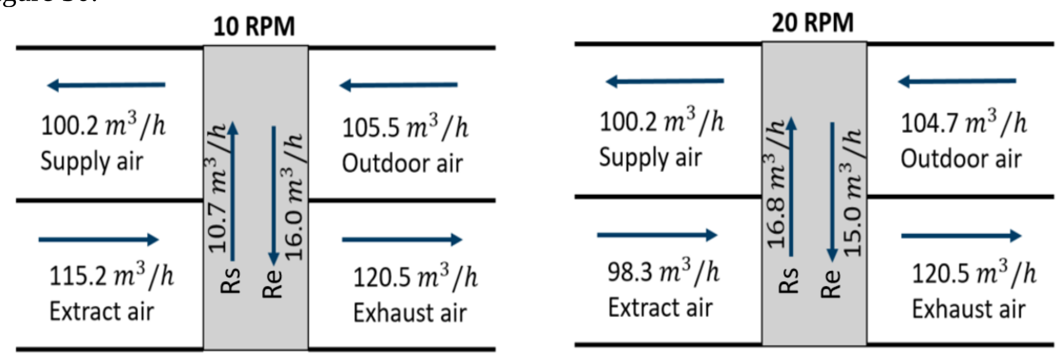


Figure 30 Calculated airflow rates and recirculation in the ducts close to heat wheel for wheel rotation of 10 and 20 RPM.

The extract and supply recirculation rates for 10 RPM are 15.2 % and 9.3 %, respectively. For wheel rotation speed of 20 RPM, the corresponding values are 14.3 % and 16.8 %. The high speed of wheel rotation (20 RPM) increases the extract air recirculation rate, which means the high rotation speed introduces more polluted air into the supply air and may degrade the performance of the heat recovery and reduce indoor air quality in practice.

Appendix C4 Field measurements at Øya kindergarten

The purpose of the measurements at Øya kindergarten was to evaluate the velocity traverse method on an air-handling unit installed in an occupied building. The goals of the testing are responding to the same research questions as in appendix C1 applied to the four ducts connected to the air-handling unit.

Methods

Measurements with the velocity traverse method have been performed (date 30.08.2017) on one of the air handling units, type DVCompact 40 (see Figure 31), in Øya kindergarten. The nominal airflow rate is 9000 m³/h and the rated efficiency of the rotary heat exchanger 83.5 % at the operating point. The fans are VAV controlled, but during the measurements, the system was controlled to supply and exhaust constant airflow rate. The outside air duct has a diameter of Ø800, while the ducts for supply and extract have diameters of Ø500. During measurements, one VelociCalc 9565-P was fixed to log continuously in the middle of the duct, while an additional VelociCalc 9555-P was used to measure over the cross-section of the ducts. Velocity, temperature and relative humidity were measured. The VelociCalc 8388 (without logging function) was used to perform measurements close to the holes drilled in the duct, due to disturbance of the sensors previously explained. It was not possible to drill holes on four sides, due to limited space. The exhaust duct could not be measured as there was no straight access to the duct.

Figure 31 Picture of the air handling unit at Øya kindergarten.

The technical room for this kindergarten illustrates the difficulties of measuring the airflow rates close to the air-handling unit. Figure 32 shows the limited space between supply and extract ducts and between the air handling unit and a concrete wall. The distance from the air-handling unit to the measuring points is also too short for the airflow to be fully developed. Most measuring equipment require a length typically five times the diameter of straight ducts for measurements to be valid. In this case, five diameters would correspond to a minimum 2.0 meters for the Ø400 ducts.

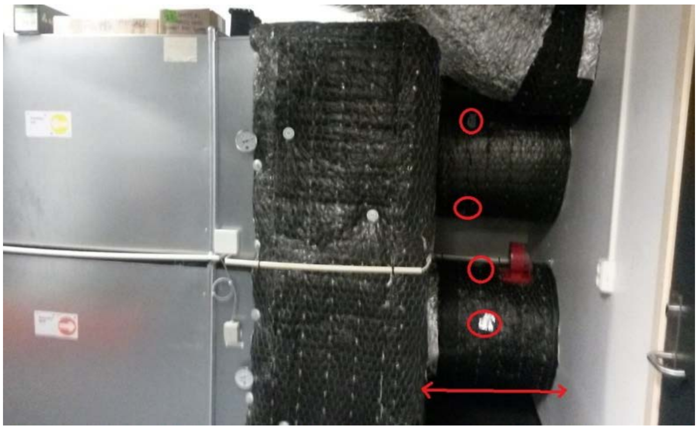


Figure 32 Picture of parts of the air handling unit, with limited area for measurements on the supply and extract duct on the right side (less than 1 m). Red circles indicate where the holes for measurements with the velocity traverse method are located.

Figure 33 shows the other side of the air-handling unit, where one can see that the supply air (bottom duct) has to go through a 90° bend just before the air handling unit. The exhaust air (top duct) is going through a rectangular duct and is then merged with the exhaust air from the other air-handling unit and released just outside the building.

Figure 33 **Picture of the air intake and exhaust duct from one of the AHUs at Øya kindergarten. Red circles indicate where the holes for measurements with the velocity traverse method are located.**

Measurements were performed at two different airflow rates, to see whether the shape of the velocity profile is identical for different flow rates. NS-EN 16211:2015 (see Table 2) specifies that ducts of diameters Ø500-Ø1250 should have eight measuring points for the velocity traverse method. The number of measuring points used in this project for the different ducts are provided in Table 16. The reason for the large number of measuring points was to calculate the airflow rate with different variations of measuring points, then reducing the total number, and comparing the deviations to the reference airflow rate monitored by the air-handling unit. Thus, how many measuring points, and which measuring points, are required to get an accurate result could be determined. The airflow rates were calculated based on the velocities in the different points, and each of their corresponding areas in the cross-section of the duct instead of the method using average velocity and total duct area.

Table 16 **Number of measuring points in the different ducts at Øya kindergarten.**

	Horizontally	Vertically
Outdoor air duct (Ø800)	22 points	22 points
Supply air duct (Ø500)	18 points	18 points
Extract air duct (Ø500)	18 points	18 points

Results

In this case it was not possible to install an orifice plate and the system BMES was not logging the airflow rates. Fan power was logged with an attempt to determine the airflow (Falke 2014). This method involves measuring the fan speed (or revolutions per minute, rpm) and operating static pressure, then inserting this to the manufacturer's fan table to plot the fan airflow. Unfortunately, the results were not reasonable, so this method did not pan out either. Table 17 presents the calculated airflow rates for outdoor, supply and exhaust ducts for the two tested airflow rates. The reason behind the large differences between outdoor air and supply/extract air, is that there are two additional ducts for supply and extract air branching through the wall behind the air handling unit. Values for temperature and relative humidity will not be presented in this chapter.

Table 17 Calculated airflow rates for each duct at the two different airflow rates.

Airflow rate number	1	2
Outdoor air duct (Ø800)	9331 m³/h	7830 m³/h
Supply air duct (Ø500)	4062 m³/h	3705 m³/h
Extract air duct (Ø500)	3288 m³/h	3225 m³/h

The velocity traversal method has a method uncertainty of ±4 % for the outdoor air duct and ±6 % for the supply and extract air ducts, as it was not possible to measure horizontally and vertically due to limited space around the ducts. The instrument uncertainty is ±3 % of reading for all three VelociCalc devices. The average of five values (1 measurement per second) for each measuring point across the cross-section was presented here.

Figure 34-Figure 36 present the graphs that were made for the first airflow rate, illustrating the measured velocities in the different points both horizontally (red line) and vertically (blue line). These show the differences in measured velocities across the cross-section of the ducts. For Figure 34 for the duct with outdoor air, the velocities are quite stable over the cross-sections, both horizontally and vertically. For such stable velocity profiles, fewer measuring points may be needed to determine the true airflow rate. The same is the case for Figure 36.

Figure 34 Graph showing the measured velocities in different points across the horizontal and vertical cross-section of the duct for outdoor air and the first airflow rate.

Figure 35 shows the results for the measurements on the supply air duct. This duct is following a bend in the air handling unit, which is why the horizontal velocity profile is so uneven. The velocity distribution will also have been affected by the fan and AHU. Note that this is the measured velocities only in two straight lines across the duct. Here it could probably have been introduced a third traverse to more accurately determine the airflow rates for each sector of the duct.

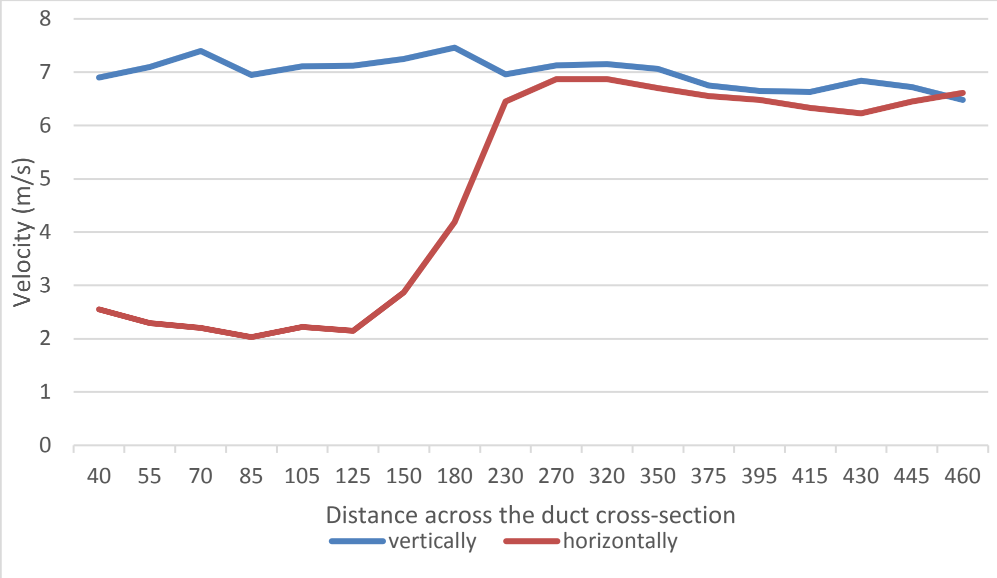


Figure 35 Graph showing the measured velocities in different points across the horizontal and vertical cross-section of the duct for supply air and the first airflow rate.

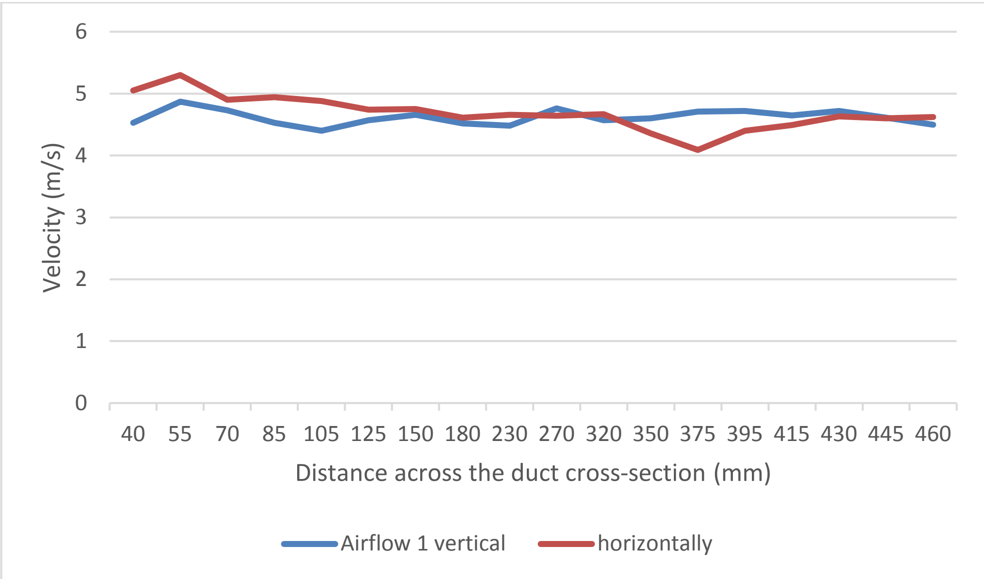


Figure 36 Graph showing the measured velocities in different points across the horizontal and vertical cross-section of the duct for extract air and the first airflow rate.

The planned analysis was not possible to be performed for this building. It is, however, a good example to show the challenges with performing field measurements close to air handling units. Many of the ducts were difficult to access and the ducts had different diameters and shape (rectangular exhaust duct).

AIRFLOW MEASUREMENTS FOR AIR HANDLING UNITS

This technical report is intended as a preliminary study on airflow measurements, as they are basic for the evaluation of efficiency of air handling units (AHUs) in ventilation systems.

The report targets readers intending to measure the field performance of the AHU. The purpose of this project is to investigate the existing knowledge and equipment for field airflow measurements. This report provides an overview of relevant studies and standards concerning laboratory measurements in a building-configuration-manner of rotary heat exchangers. Additionally, it focuses on testing currently available methods and devices that could be used for reliable measurement of airflow rates.

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