LAPPEENRANTA UNIVERSITY OF TECHNOLOGY
LUT School of Energy Systems
Degree Programme in Environmental Technology

Santeri Pöyhönen

VARIABLE-SPEED-DRIVE-BASED DETECTION METHODS FOR PROBLEMS IN FLUID HANDLING SYSTEMS

Examiners: Professor Risto Soukka
Professor Jero Ahola
Supervisor: D.Sc. Tero Ahonen

ABSTRACT

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Variable-speed-drive-based detection methods for problems in fluid handling systems

Master's Thesis

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81 pages, 35 figures and 1 table

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Keywords: variable-speed drive, pump, fan, compressor, fluid handling

Fluid handling systems account for a significant share of the global consumption of electrical energy. They also suffer from problems, which reduce their energy efficiency and increase life-cycle costs. Detecting or predicting these problems in time can make fluid handling systems more environmentally and economically sustainable to operate.

In this Master's Thesis, significant problems in fluid systems were studied and possibilities to develop variable-speed-drive-based detection methods for them was discussed. A literature review was conducted to find significant problems occurring in fluid handling systems containing pumps, fans and compressors. To find case examples for evaluating the feasibility of variable-speed-drive-based methods, queries were sent to industrial companies. As a result of this, the possibility to detect heat exchanger fouling with a variable-speed drive was analysed with data from three industrial cases.

It was found that a mass flow rate estimate, which can be generated with a variable speed drive, can be used together with temperature measurements to monitor a heat exchanger's thermal performance. Secondly, it was found that the fouling-related increase in the pressure drop of a heat exchanger can be monitored with a variable speed drive. Lastly, for systems where the flow device is speed controlled with by a pressure measurement, it was concluded that increasing rotational speed can be interpreted as progressing fouling in the heat exchanger.

TIIVISTELMÄ

Lappeenranta University of Technology
LUT School of Energy Systems
Ympäristötekniikan koulutusohjelma

Santeri Pöyhönen

Taajuusmuuttajapohjaiset havaitsemismenetelmät virtausjärjestelmien ongelmille

Diplomityö

2016

81 sivua, 35 kuvaa ja 1 taulukko

Työn tarkastajat: Professori Risto Soukka
Professori Jero Ahola
Työn ohjaaja: Tohtori Tero Ahonen

Hakusanat: taajuusmuuttaja, pumppu, puhallin, kompressori, virtausjärjestelmät

Pumppu-, puhallin- ja kompressorijärjestelmät kattavat merkittävän osan maailman sähköenergian kulutuksesta. Niiden toiminnassa voi esiintyä ongelmia, jotka vähentävät energiatehokkuutta ja kasvattavat elinkaarikustannuksia. Virtausjärjestelmien ongelmien havaitseminen ja ennustaminen voivat tehdä niiden käytöstä kestävämpää sekä ympäristön että talouden kannalta.

Tämän diplomityön tavoitteena oli etsiä virtausjärjestelmien ongelmia, joita varten on mahdollista kehittää taajuusmuuttajapohjaisia havaitsemismenetelmiä. Työssä suoritettiin kirjallisuusselvitys, jossa perehdyttiin virtausjärjestelmiin ja niiden komponentteihin yleisesti sekä kartoitettiin niissä esiintyviä ongelmia. Taajuusmuuttajapohjaisten havaitsemismenetelmien soveltuvuuden arvioimiseksi työssä otettiin yhteyttä teollisuuden yrityksiin. Yhteydenotot paljastivat kolme likaantuvaa lämmönsiirrintä kolmesta eri kohteesta, joista saadun datan avulla taajuusmuuttajapohjaisen havaitsemismenetelmän soveltuvuutta analysoitiin.

Työn tuloksena selvisi, että taajuusmuuttajan tuottamaa massavirtaestimaattia voidaan käyttää yhdessä lämpötilamittausten kanssa lämmönsiirtimen lämmönsiirron tehokkuuden tarkkailussa. Työssä selvisi myös, että likaantumisesta aiheutuvaa lämmönsiirtimen virtausvastuksen kasvua voidaan seurata taajuusmuuttajan avulla. Lisäksi työssä todettiin, että virtauslaitteelle, jonka pyörimisnopeutta säädetään painemittauksen perusteella, etenevä lämmönsiirtimen likaantuminen voidaan nähdä kasvavana pyörimisnopeutena.

PREFACE

This thesis has been carried out at the Laboratory of Digital Systems and Control Engineering in Lappeenranta University of Technology (LUT). The work was funded by the EFEU project coordinated by CLIC Innovation Ltd.

My first and foremost thanks go to my supervisor D.Sc. Tero Ahonen and examiner, Professor Jero Ahola for providing this work. The involvement and the plentiful support and guidance Tero offered was greatly appreciated. I would also like to thank Simo Hammo for his support, guidance, great ideas and interest in my work and its progress. I want to thank Simo and Tero together for somehow always finding work for me and to commend them for their excellent "marketing" of research topics, which may at first sound far-fetched for the less acquainted. Thanks to Professor Risto Soukka for accepting my thesis for examination, despite only hearing of it at a very late stage.

I would like to thank the people who responded to my inquiries and made the industrial case studies of my thesis possible. Special thanks go to D.Sc. Timo Talonpoika, who found the time to read and respond to my emails and provided huge amounts of relevant data for the first industrial case studied in this thesis. I would also like to thank Toni Tiainen and Henri Isbom for their cooperation.

My sincerest thanks and gratitude go to my parents and family for their unconditional support. Also, thanks for being smart. Having grown with you I could not have grown wrong, so to speak.

My deepest and utmost gratitude I send to my friends. The wonderful people who smile upon me with sincere eyes. You, who accept me as a part of your world. No names shall be named, (as cliché as it sounds) as the list would be a tad too long, but I think you know who you are. With you around, life is about more than just survival. Thank you.

Lappeenranta, April 27, 2016

Santeri Pöyhönen

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LIST OF SYMBOLS AND ABBREVATIONS

Roman letters

A	heat transfer surface area	(m^2)
c_p	specific heat capacity	(kJ/(kg K))
$cos\ \phi$	power factor	
D	diameter	(m)
f_D	loss coefficient	
g	standard gravity	9.81 m/s^2
H	head	(m)
h_f	head loss	(m)
I	electric current	(A)
L	length	(m)
n	rotational speed	(rpm)
P	power	(W)
P_e	electric power	(W)
Δp	pressure loss	(Pa)
$\dot{Q}$	heat flow rate	(W)
q_m	mass flow rate	(kg/s)
T	temperature	(°C, K)
U	overall heat transfer coefficient, voltage	($W/(m^2\ K)$), (V)
u	mean velocity	(m/s)

Greek letters

ΔT_{LMTD}	logarithmic mean temperature difference	(K)
ρ	density	(kg/m^3)

Abbreviations

GDP	Gross Domestic Product
HVAC	Heat, Ventilation and Air Conditioning
LCC	Life-Cycle Cost
VSD	Variable-Speed Drive

1 INTRODUCTION

The growing concern for the depletion of natural resources and the state of the climate drives communities towards rethinking the way energy is produced and consumed. Efficiency in the production and end-use of energy brings many benefits, including but not limited to, savings in fuel costs and reduction of emissions. In Europe, the European Union addresses the matter of energy efficiency with a set of legislative measures. A key element of these measures is the Energy Efficiency Directive, which includes binding targets for energy efficiency at all stages of the energy chain (European Commission 2015). In addition to the obligation to follow policies, the variety of undeniable benefits brought by energy efficiency serve as an incentive to pursue it (Figure 1).

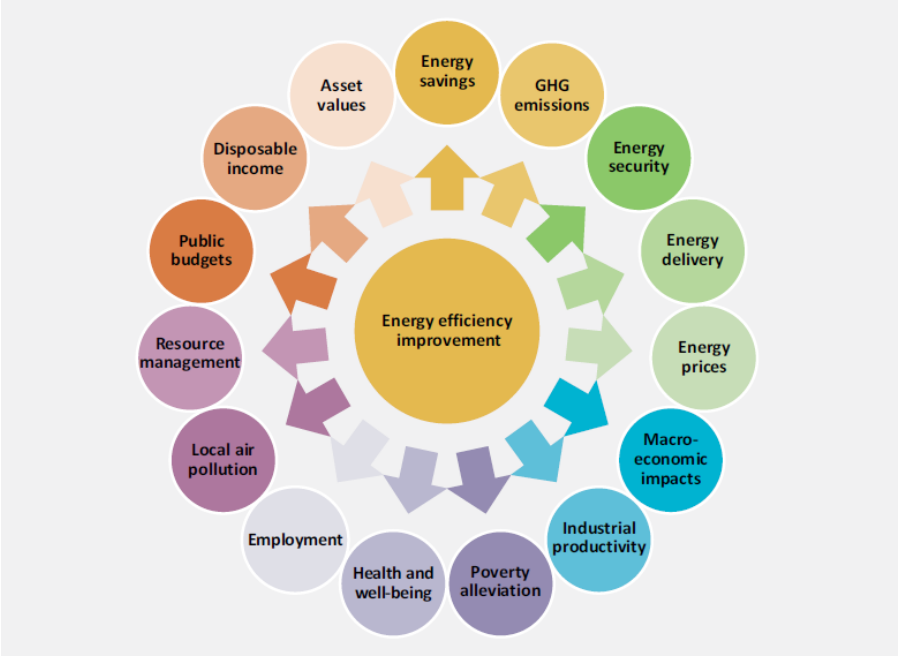

Figure 1. Benefits brought by energy efficiency (International Energy Agency 2014).

Fluid handling systems, which in this context refers to systems with pumps, blowers and compressors, are present in a wide variety of municipal and industrial applications. Not only are they a vital part of for example power generation systems, transferring the condensate and heat within the system, but their operation has great significance in terms of global energy consumption, too. In the European Union, electric motors account for 69 % of the industrial electricity consumption and for 38 % of the municipal and services

sector's consumption (Almeida et al. 2003). Within the industrial and the tertiary sectors, the electricity consumption is divided by end-use applications as shown in Figure 2.

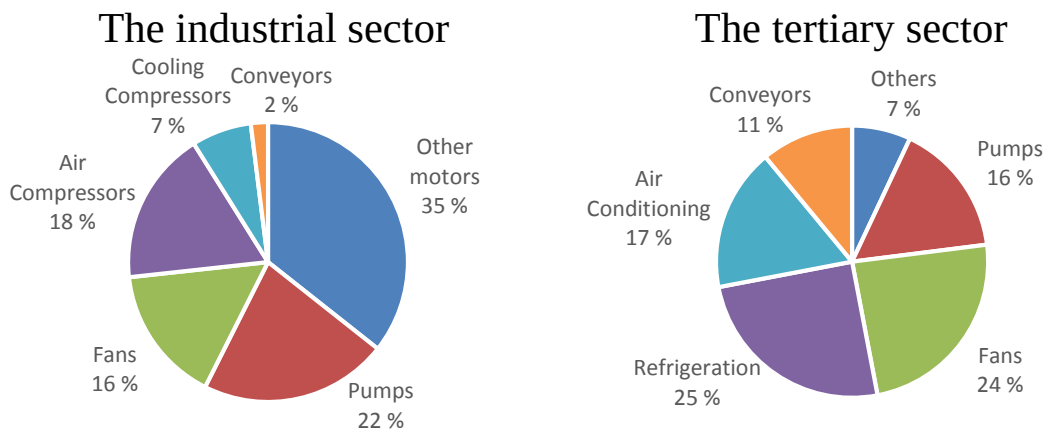

Figure 2. The shares of motor electricity consumption in the industrial and tertiary sectors (Almeida et al. 2003, 1–2).

The consumption of energy accounts for the greatest share of the costs of pumping, fan and compressed-air systems' life-cycles (Pump Systems Matter and Hydraulic Institute 2008, 63; Tamminen 2013, 16; Radgen et al. 2000, 89). Life-cycle cost (LCC) is the total cost of the ownership of a product or system. It covers costs beyond the initial investment costs and all the way until decommissioning. In addition to energy costs, other components included in a life-cycle cost analysis include initial costs, installation and commissioning costs, operating costs, maintenance and repair costs, downtime costs, environmental costs and decommissioning and disposal costs. In the most extreme of cases, for example for pumps, the energy cost can be 20 times the purchase price. (Pump Systems Matter and Hydraulic Institute 2008, 62-63.) Significant potential for improvement in the energy efficiency of fluid handling systems exists (Fleiter et al. 2011; Cleen Oy 2015; Zuercher 2015, 2). The significance of fluid handling systems as consumers of energy and the found potential for improvement in their energy efficiency together with governmental incentives and regulations have led to an increase in energy audits conducted for fluid handling systems.

In many fluid handling applications, the flow rate or pressure needs to be continuously regulated. Traditionally, the most common solution for flow regulation has been the use of

throttling valves. Pumps, for example, are often sized according to the highest demanded flow (Hydraulic institute 2004, 3). In practice, however, this maximum flow might be needed only very rarely. Thus, the flow may remain throttled for a large share of the operating time, which will lead to the system constantly wasting energy through the losses caused by the valve. Instead of throttling with control valves, variable-speed drives are increasingly applied to regulate the flow. The use of a variable speed drive (VSD) allows the adoption of a new pump or blower operating point by changing the rotational speed of the device. According to the affinity laws applicable to pumps and blowers, the power consumption of the device has a cubic relation to the rotational speed:

$$P = \left(\frac{n}{n_0}\right)^3 P_0 \tag{1}$$

P = power (W)
n = rotational speed (rpm)

It is worth noting that the affinity laws only apply for the flow device itself and do not take the surrounding system into account. The affinity laws enable for example a new pump curve to be defined for a new rotational speed. The pump curve alone will not reveal the performance of the pump in a system – only after adding the system curve together with the pump curve can the performance of the pump with the new rotational speed be properly predicted. (Nesbitt 2006, 132.) Still, the affinity law for power provides a theoretical basis for the energy savings achievable by decreasing the rotational speed and makes speed control a choice worth considering in many different kinds of systems.

The main purpose for installing VSD's is typically speed control. However, recent studies (Ahonen 2011; Tamminen 2013) show that a VSD is capable of providing accurate enough estimates of the motor shaft torque and rotational speed to be used in defining the operating point of a blower or a pump. This provides the possibility to detect changes in the system surrounding the flow device. Changes in the conditions of a fluid handling system can have an effect on the behaviour of the flow device and furthermore on the loading conditions of its motor.

1.1 Motivation of the study

Fluid handling systems are ridden with problems that reduce the energy efficiency and the lifetime of the systems. These issues can be divided into problems caused by mistakes in systems design and into problems, which occur during operation regardless of the application of the best possible design principles. In this Master's Thesis, a literature review along with surveys to companies will be conducted on significant problems of the latter mentioned type in flow systems. The surveys will reveal potential locations for case studies. The aim is to find potential targets for VSD-based detection methods, which can be developed later in the future.

The literature review and the case studies will reveal problems for which the development of VSD-based detection methods could be considered. The applicability of a VSD-based method to detect the problems occurring in the case studies will be discussed. The results of this study will act as groundwork for potential future research in developing VSD-based detection methods.

1.2 Outline of the thesis

First, as part of the results of the conducted literature review, a general overview of fluid handling systems will be given. The fluid handling devices and relevant system components will be introduced. In the following chapter, the common and significant problems revealed by the literature review will be presented and analysed.

In the second half of the thesis, the results acquired from the queries to companies will be presented case by case. The case environment will be described to a relevant detail and the problem for possible detection method development will be introduced. Measurements acquired from the studied system will be presented and the applicability of a VSD-based detection method will be discussed.

2 FLUID HANDLING SYSTEMS

Fluid handling systems comprise systems where a fluid, which can be of either gaseous or liquid form (possibly containing some solids), is moved through a system. The operating principles of the devices, which move the fluid, vary with systems. This study focuses on systems with rotating fluid handling devices. In the following chapters, relevant theory and typical applications of pump, blower and compressor systems are presented.

2.1 Pump systems

Pumps and the systems in which they operate are an essential part of process operations in the energy sector and in numerous other industries and municipal applications (Nesbitt 2006, III). Pumping systems are usually a part of a larger process, but the basic function of the pump is always the same: to transfer a fluid in a way that is required by the process. The fluid can be transferred for a variety of purposes: for example, in a municipal freshwater system, to deliver water to consumers or in a power plant to deliver feed water to the boiler. These are examples of some of the pumps of a larger scale, which can be several dozens of MW in size (LUT 2013a, lecture 3). Pumps of smaller sizes are utilized in applications such as housing potable water circulation and portable water transfer pumps, where their power is typically measured in some dozens of watts to a few kilowatts (Bell & Gossett 2015; Northern Tool + Equipment). In the European Union, pumps account for 15 % of the electricity consumption of industries. The corresponding share in the services sector is 6 % of the sector's total electricity consumption (Almeida et al. 2003, 1-2).

Pumping systems worldwide possess significant potential for energy efficiency improvements (Almeida et al. 2003, 3). In addition, energy efficiency is recognized as a sustainable means of saving energy and natural resources and cutting down emissions. As pumping accounts for a significant share of the world's electrical energy consumption, realizing improvements in pumping systems can be seen as a good target for investments in energy efficiency.

Pumping systems can be divided into open and closed loop systems. Exemplary schemes for the aforementioned systems are illustrated in Figure 3.

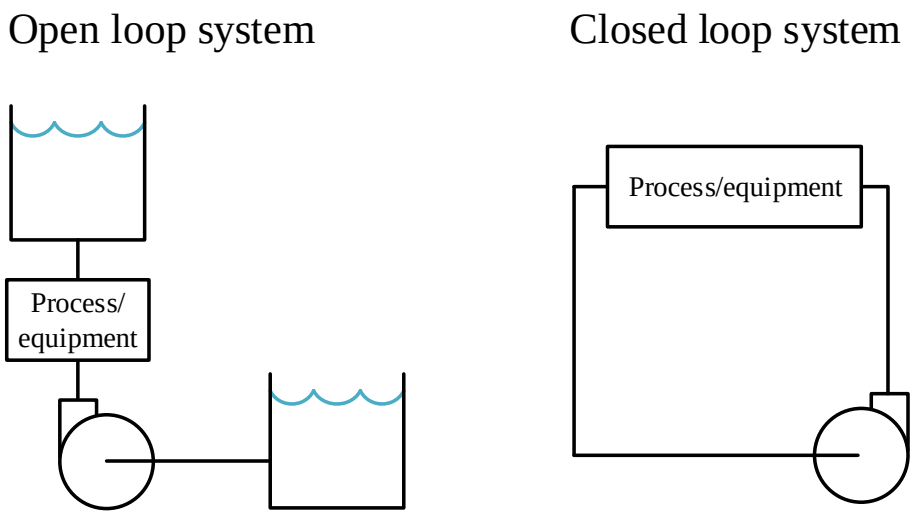

Figure 3. An illustration of the principles of the open and closed loop pumping systems. (Ahonen et al. 2011a.)

The purpose of open loop systems is usually to transfer fluid from one point to another. The function can for example be pumping fresh water from a reservoir to a higher one or to provide pressure, which enables water distribution to consumers in a water distribution network. Closed loop systems recirculate the fluid through the system. The fluid is often used to transfer heat from one point to another, for example in domestic and district heating systems, or to convert energy, for example heat energy into electrical energy in steam turbines coupled with generators.

A pumping system containing both an open and a closed loop is introduced in chapter 4.1. In the system, which will be explained in more detail in the said chapter, a closed loop is used to circulate cooling water through a heat exchanger and a generator and an open loop to transfer seawater through the heat exchanger.

2.1.1 Pump types

Pumps exist in a large variety of designs, which are often specific to the applications in which they serve. For an application, the best pump type depends on pumping system properties and attributes such as liquid properties, allowable leakage, driver type, installation arrangement, required operating efficiency, duty cycle, allowable noise level, operational safety and site facilities and the capabilities of the local staff (Nesbitt 2006, 3). A common way to classify pumps is to look at the way they add energy to the liquid. Following this approach, they can be divided into two major categories: dynamic and displacement pumps. In dynamic pumps, most of which are of the rotating type, energy is continuously added into the fluid, whereas in displacement pumps it is done periodically. (Karassik et al. 2008, section 1.2-1.3.) Pumps can be further classified according to the principle with which they handle the fluid and ever further according to their geometry. A classification of pumps presented by Karassik et al. is shown in Figure 4.

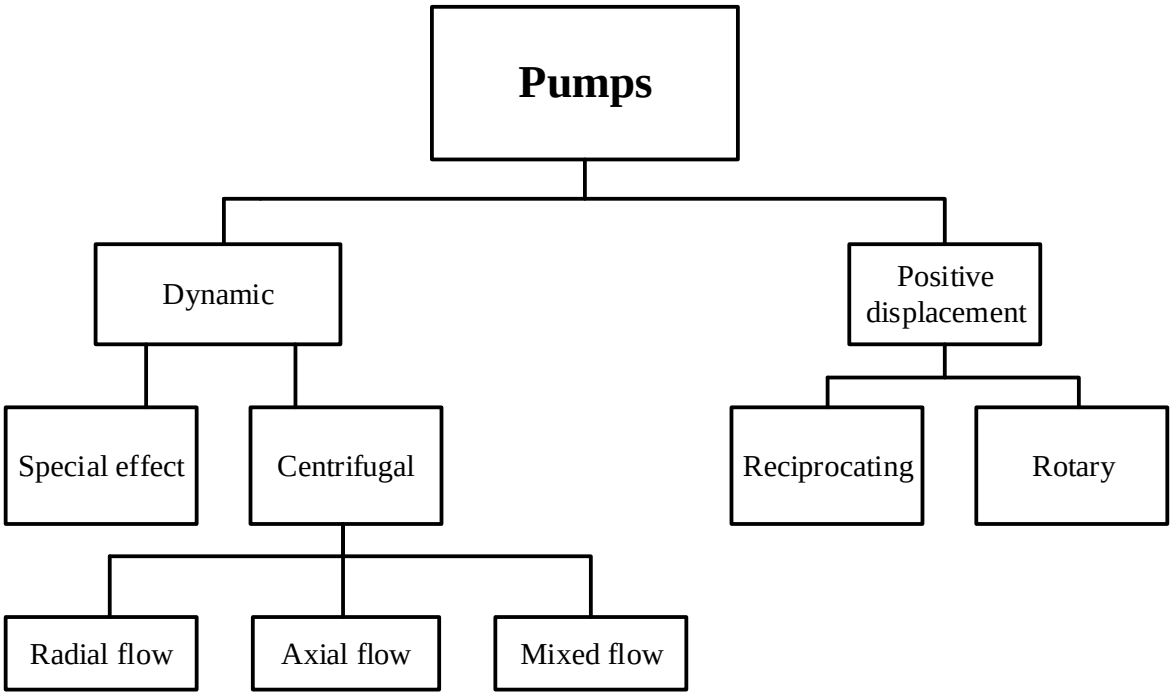

Figure 4. Classification of pumps (Karassik et al. 2008, section 1.3-1.4).

Thanks to their typically low cost, low maintenance requirements and long operating lives, centrifugal pumps are the most popular pump type used in industries. (Lawrence Berkeley National Laboratory 2006, 14.)

2.2 Fan systems

The fluid moved in fan systems is of the gaseous form. For the industries, they account for 11 % and for the services sector 9 % of the total electricity consumption of the respective sectors in the EU. (Almeida et al. 2003, 1-2.) Fans are present in a wide variety of industrial and municipal applications. For example, they are typically used to handle air in ventilation and air conditioning systems, to transfer product gases, to increase the heat transfer rate in cooling applications, to maintain pressure in spaces and to aerate process fluids.

The American Society of Mechanical Engineering has provided a definition for fans and blowers, which divides them by the ratio of the discharge pressure and the suction pressure. The ratio is referred to as the specific ratio. Devices handling air with a specific ratio of up to 1.11 are considered fans, and those with a specific ratio between 1.11 and 1.20 are considered blowers. (Government of India, Ministry of Power, Bureau of Energy Efficiency, section Fans and Blowers, 1). In some publications, the definition is followed, while in some the terms are used ambiguously. In this study, the term fan is used to refer to both blowers and fans.

2.2.1 Fan types

Nearly all fans are driven by electrical motors. Motors or different sizes are used according to the required flow, pressure and fan performance. Considering the impeller and the casing alone, fans exist in many different designs to meet the varying flow and pressure demands of different fluid handling applications (Radgen et al. 2008, 2-3). Fans can be characterized by the path of the flow in the fan. Such a characterization is presented in Figure 5.

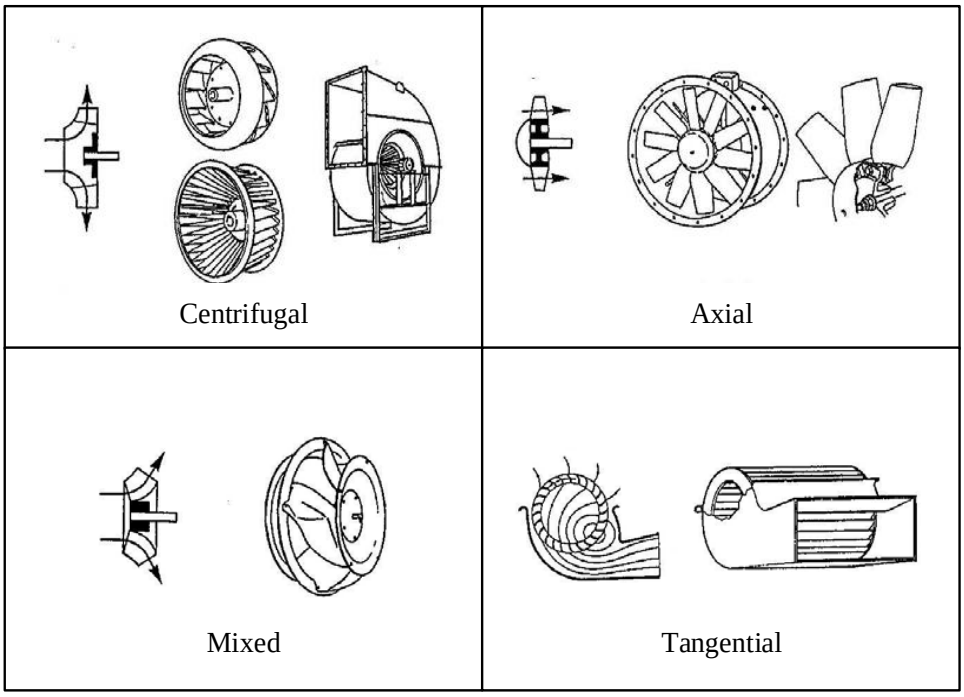

Figure 5. Classification of fans according to the flow path in the fan (Radgen et al. 2008, 3).

Different designs have different performance characteristics and they differ in terms of produced pressure and volumetric flow and efficient and reliable operational regions. The presented categories can further be classified based on the shape of the blades on the impeller. In addition, the shape of the impeller has an effect on the performance of the fan. For instance, a forward-curved impeller of a certain diameter will deliver a higher volumetric flow rate than a backward-inclined impeller of the same size. (Lawrence Berkeley National Laboratory 2003, 19.) The centrifugal fan is by far the most common fan type found in industrial and HVAC (Heat, Ventilation and Air Conditioning) fan systems (Lawrence Berkeley National Laboratory 2003a, 19, United States Environmental Protection Agency 2008, 7).

2.3 Compressed-air systems

Compressors are used in many systems where pressurized gas is needed by the end user. The end uses range from supplying pressurized breathable air in pressure tanks used in underwater diving to generating pressure with which brakes of trains or road vehicles can be operated. Compressed-air systems account for 10 % of industrial electricity

consumption in the EU (Almeida et al. 2003, 1-2). Most of the compressors used in compressed-air systems are powered by electric motors. Because of the significance and the potential applicability of VSD-based methods, this study focuses solely on compressed-air systems as far as compressors are concerned. Compressed-air systems provide pressurized air for the uses of various industries. The uses include but are not limited to tool powering, controls and actuators and conveying (Lawrence Berkeley National Laboratory 2003b 9, 15). The composition of a typical industrial compressed-air system is presented in Figure 6.

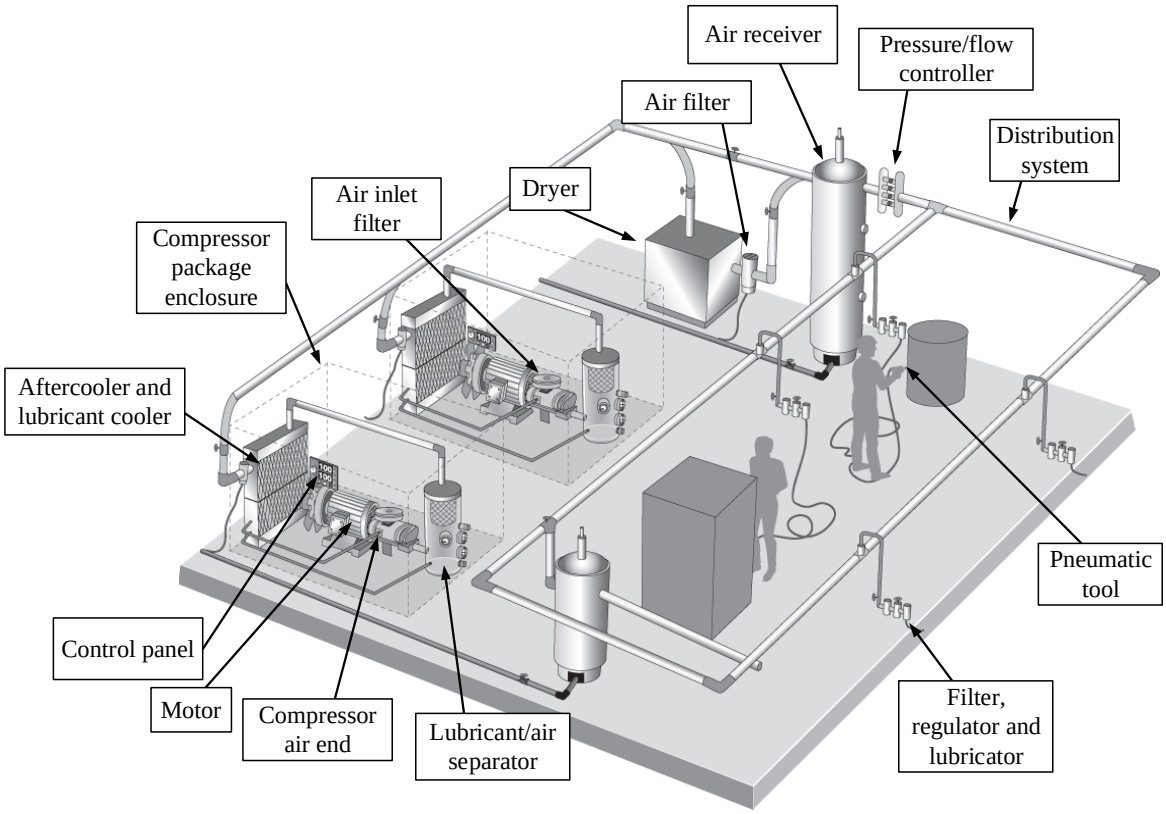

Figure 6. A typical industrial compressed air system and its components (Lawrence Berkeley National Laboratory 2003b, 4).

2.3.1 Compressor types

Like pumps, compressors can be divided into positive displacement and dynamic compressors. A diagram for the classification of compressors is presented in Figure 7.

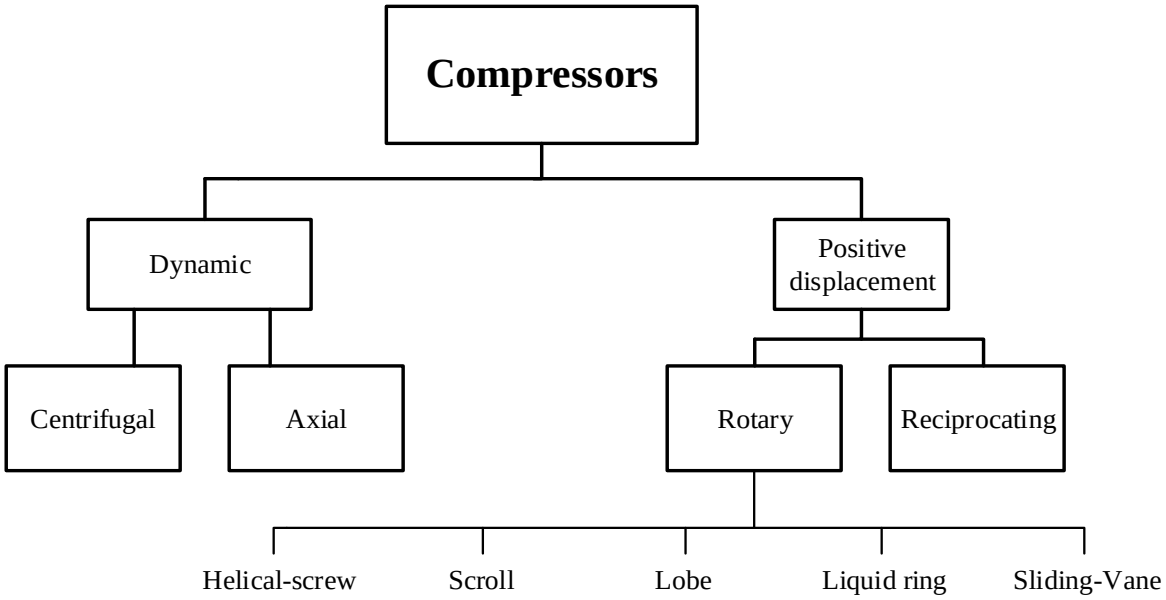

Figure 7. Classification of compressors (Lawrence Berkeley National Laboratory 2003b, 5).

In reciprocating compressors, the volume of air is reduced to increase its pressure. A piston is driven through a crankshaft with back-and-forth strokes. Single-acting reciprocating compressors compress the air with strokes in one direction, whereas double-acting do it in both directions. Rotary positive displacement compressors use rotating parts such as screws and lobes to reduce the volume of the air. Screw compressors are described as the "dominant compressor type" used in industrial compressed-air systems.

Dynamic compressors use impellers to convert the shaft energy of the motor into kinetic energy in the gas. The kinetic energy is further converted into pressure by slowing the airflow in a diffuser (Government of India, Ministry of Power, Bureau of Energy Efficiency, section Compressors, 48). Dynamic compressors can be divided into centrifugal and axial types.

A centrifugal compressor functions similarly to a centrifugal fan and a centrifugal pump. The rotating impeller imparts kinetic energy to the gas and generates a radial flow, which is directed through the collector scroll to the diffuser (Lawrence Berkeley National Laboratory 2003b, 8; Gorla and Khan 2003). Centrifugal compressors exist in single-stage

and multi-stage designs. Most centrifugal compressors have two to four stages (Lawrence Berkeley National Laboratory 2003b, 8).

Axial compressors include multiple stages of rotor blades with stator rows of vanes between them. The direction of the flow is primarily axial. Part of the velocity energy is converted into pressure in each of the stator stages. (Lawrence Berkeley National Laboratory 2003b, 8.)

2.4 Fluid handling system components

In addition to the properties of the pump, fan or compressor itself, the performance and the operating point of the device is defined by the surrounding system. In the following chapters, typical fluid handling system components and their effect on pump performance are described.

2.4.1 Piping and ductwork

To transfer and provide pressurized fluid in a system, piping or ductwork is required. The piping system constitutes individual pipe runs of possibly variable diameters and sizes, which connect system elements such as flow devices, control valves and heat exchangers together.

The length and quality of the piping plays a part in determining the performance requirements of the flow device. The fluid flowing through piping experiences friction, which causes a loss of head in the flow. The friction losses of piping increase as flow velocity increases and decrease as the pipe diameter increases. Also, the longer the piping and the rougher its inner surface, the greater the head loss will be. (Nesbitt 2006, 99.) Pressure loss in pipes with gaseous fluids and head loss in pipes with liquid fluid can be calculated with the following equations, respectively:

$$\Delta p = f_D * \frac{L}{D} * \frac{\rho * u^2}{2} \quad (2)$$

Δp = pressure loss (Pa)
f_D = loss coefficient
L = length of the pipe (m)
D = diameter of the pipe (m)
ρ = density (kg/m^3)
u = mean velocity (m/s)

(Kiijärvi 2011, 2.)

$$h_f = f_D * \frac{L}{D} * \frac{u^2}{2g} \quad (3)$$

h_f = head loss (m)
g = standard gravity 9.81 m/s^2

(Nesbitt 2006, 100.)

The loss coefficient, also referred to as the Darcy friction factor, is defined as a function of the Reynolds number for laminar flow and as a function of the Reynolds number and the ratio of the roughness of the pipe surface and the pipe diameter for turbulent flow. (Nesbitt 2006, 100.) The calculation values for the surface roughness of different materials are listed in numerous handbooks and often provided by pipe manufacturers (Nesbitt 2006, 99; OSTP; Karassik et al. 2008, section 11.32).

Ducts refer to passageways generally made out of sheet metal, which are suitable for low pressure. Pipes are sturdier, which allows fluids of higher pressures to be transferred through them. (Lawrence Berkeley National Laboratory 2003a, 12.)

2.4.2 Valves and dampers

Valves are used in fluid handling systems to regulate the flow rate or the pressure in the system and to direct the flow to and shut off certain paths in the piping system. Flow control with the valve is achieved by adding a resistance to the flow, which is higher than that of the system on its own. (Pump Systems Matter and Hydraulic Institute 2008, 26.)

Valves exist in numerous designs, depending on their task, required performance and the properties of the flowing fluid. Valves are used in pumping and compressed-air systems whereas in fan systems, they are often referred to as dampers.

Control valves are used to regulate the flow by reducing the flow area across the valve. This is done by using an actuator to adjust the position of the valve plug in relation to the valve seat. Globe valves and rotary valves are commonly used in flow control. Globe valves are named after their typically sphere-like outline of the body. Rotary valves typically use discs or balls to obstruct the flow. A cross-section through a single-ported globe valve and a rotary butterfly valve are shown in Figure 8.

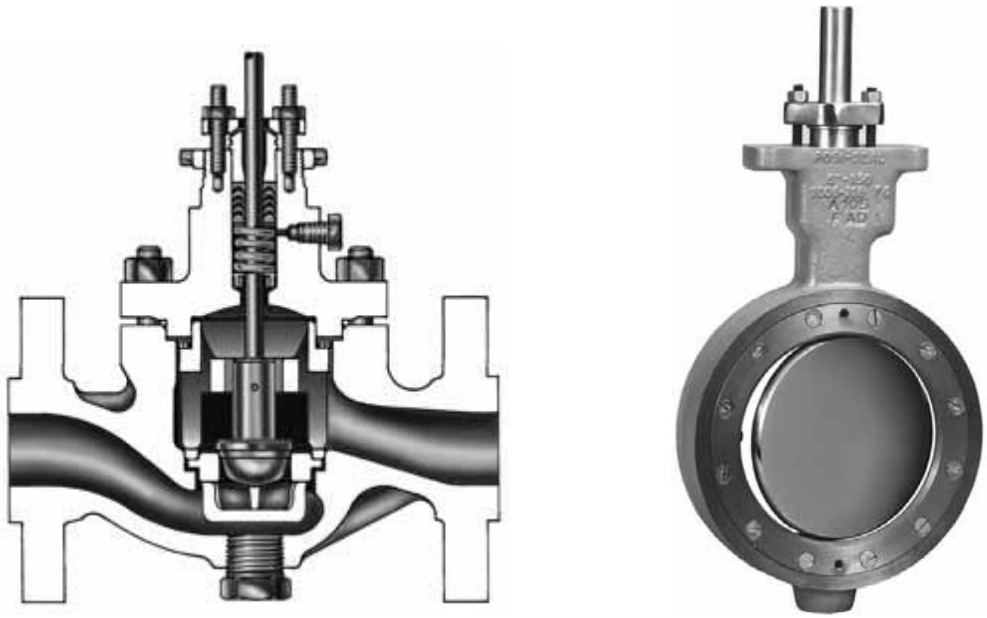

Figure 8. A single-ported globe valve and a rotary butterfly valve (Emerson Process Management. 2005, 42 & 45).

Control valves in fluid systems, which handle gas, also referred to as dampers, often comprise one or several blades with which the flow is obstructed. Some typical designs of dampers used in air handling applications are shown in Figure 9.

Figure 9. Dampers used in air handling systems (Fläktwoods 2014a; Fläktwoods 2014b).

Control valves exist in designs with more than one port. Three-way valves make it possible to divide a flow and direct it to two separate paths. Mixing two flows into one combined flow is also possible. Cross-sections through a three-way valve in diverging and converging constructions are shown in Figure 10.

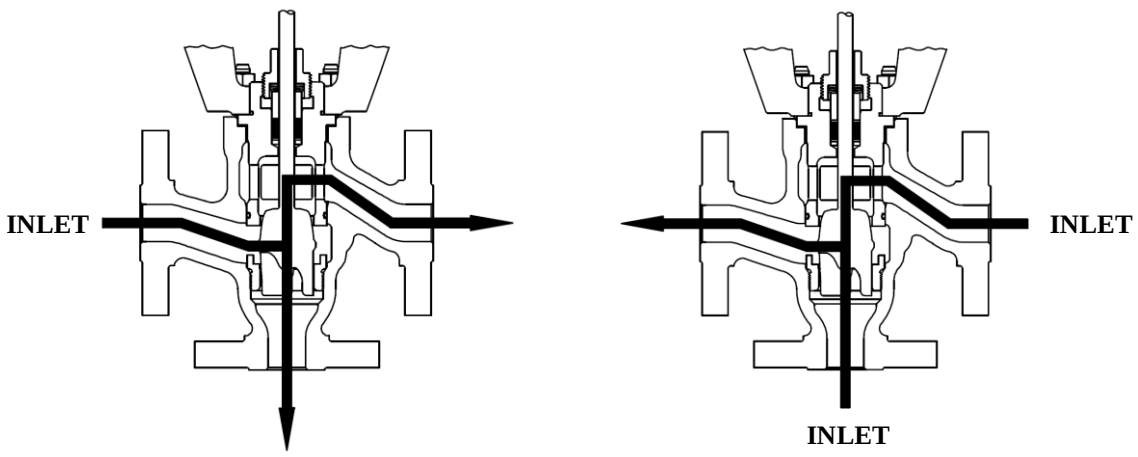

Figure 10. A three-way valve and the diverging and converging operational modes (Emerson Process Management 2015, 6 & 7).

In addition to flow regulation, valves are used to fully prevent flow into certain directions or branches in piping networks. Isolation valves are used to prevent flow into certain parts of a system and to direct the flow to one or more of a variety of different flow paths. Non-return valves, also referred to as check valves, are used to prevent the fluid from flowing in the wrong direction.

Valves are a source of friction losses in fluid handling systems. Many valves bring hydraulic resistance even when they are in the fully open position. The reason for this is the structure of the body of the valve: as seen in Figure 8 and Figure 10, the flow path through the valve can contain turns, convergences and expansions. Disturbing the flow in these ways increases the hydraulic resistance of the component (Idel'chik 1960, 350-351).

2.4.3 Heat exchangers

Heat transfer is an important task in many pump, fan and compressor systems. Heat exchangers transfer heat between parts of processes either to supply heat energy where it is needed or to remove heat from a source that needs to be cooled down. In a heat exchanger, heat is transferred from the colder flowing fluid to the warmer one. Depending on the application, the heat transfer takes place between two or more fluids. Commonly, the heat transfer is done through a heat transfer wall with a high thermal conductivity, which separates the fluids. However, there are also heat exchanger designs where the fluids come into direct contact without a wall between (Kakaç and Liu 2002, 5). With the assumption, that heat losses into the environment are negligible, the heat flow rate of a concurrent and a counter current heat exchanger can be calculated with the equation

$$\dot{Q} = U * A * \Delta T_{LMTD} = q_{m,c} * c_p * \left(T_{c,2} - T_{c,1}\right) = q_{m,h} * c_p * (T_{h,2} - T_{h,1}) \quad (4)$$

$\dot{Q}$ = heat flow rate (W)

U = overall heat transfer coefficient (W/(m^2 K))

A = heat transfer surface area (m^2)

ΔT_{LMTD} = logarithmic mean temperature difference (K)

q_m = mass flow rate (kg/s)

c_p = specific heat capacity (kJ/(kg K))

T = temperature (°C or K)

(Springer 2010, 33-34 & 38.)

Furthermore, the logarithmic mean temperature difference is defined as

$$\Delta T_{LMTD} = \frac{\Delta T_1 - \Delta T_2}{\ln(\frac{\Delta T_1}{\Delta T_2})}, \quad (5)$$

where $\Delta T_1 = T_{hot,in} - T_{cold,in}$ and $\Delta T_2 = T_{hot,out} - T_{cold,out}$ for a counter current heat exchanger. (LUT 2013b, lecture “Lämmönsiirtimet”.)

A schematic example of a heat exchanger is presented in Figure 11.

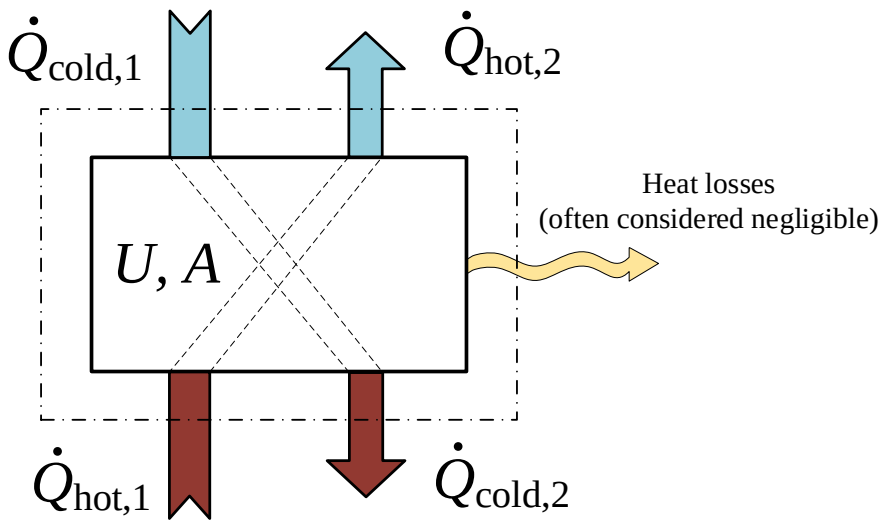

Figure 11. An exemplary model of a heat exchanger.

The deposition of materials on the heat transfer surfaces of heat exchangers is a common problem. This mechanism, referred to as fouling, adds thermal resistance, which lowers the heat transfer rate and makes the heat exchanger less efficient (Springer 2010, 80). To achieve a certain heat transfer rate, a dirty heat exchanger requires a greater volumetric flow rate than a clean one. In applications where the heat transfer rate should be kept at a certain level this means that as the heat exchanger experiences fouling, more power is required of the pump to increase the flow rate. In addition to its thermal performance, also the hydraulic performance of the heat exchanger is affected by fouling. The deposit build up will decrease the flow area in the device and thus increase the flow resistance. The increased flow resistance can lead to a higher requirement for the pumping power. Surveys have shown that more than 90 % of heat exchangers are subject to fouling (Springer 2010, 80).

Heat exchangers handle a variety of tasks in fluid handling systems. Cooling and heating of components and process fluids are typical tasks. Heat exchangers, which circulate cold water, are used, for example, to cool down generators in power plants. Additionally, aftercoolers are used in compressed-air systems to reduce the temperature of the compressed air to make it cold enough for the end use and to enable the removal of moisture in the form of condensate (Lawrence Berkeley National Laboratory 2003b, 10). In addition, for example in oil refineries, heat exchangers are used to preheat the product in distillation towers to reaction temperature (Kundnaney and Kushwaha 2015, 1).

Condensation and boiling of process fluids is accomplished with heat exchangers. For instance, in thermal steam power plants, where feed water after the turbine must be condensed, the condenser is often a shell-and-tube heat exchanger (Kundnaney and Kushwaha 2015, 1).

Heat exchangers are also used to recover heat that would otherwise be lost with heat losses through conduction or with fluids, which are released to the environment or atmosphere in an open cycle. For example, plate heat exchangers are used to recover the spare heat from

industrial plants' flue gases and in municipal air handling systems to recover heat from exhaust air.

Heat exchangers exist in a variety of designs. The right type of heat exchanger can be chosen when related process parameters and characteristics such as operating pressures and temperatures, size requirements and fluid fouling tendency are known (H&C Heat Transfer Solutions). There are many ways to classify heat exchangers. They can be divided into classes based on their construction, flow arrangements, transfer process, number of fluids, heat transfer mechanisms or surface compactness (Shah 1998, section 17.2). In the following, some of the most common types of heat exchangers are described.

Shell-and-tube heat exchanger is the most common type of multitubular heat exchanger used in industries (Walker 1982, 48). They are widely known and understood as a technology and allow a wide range of possible design pressures and temperatures (Walker 1982, 45). Shell-and-tube heat exchangers are available in varying constructions of its subassemblies, the front and rear ends, the tube bundle and the shell. Different constructions have varying accessibility for cleaning and accommodation to thermal expansion (Walker 1982, 54). An exemplary model of a shell-and-tube heat exchanger is presented in Figure 12.

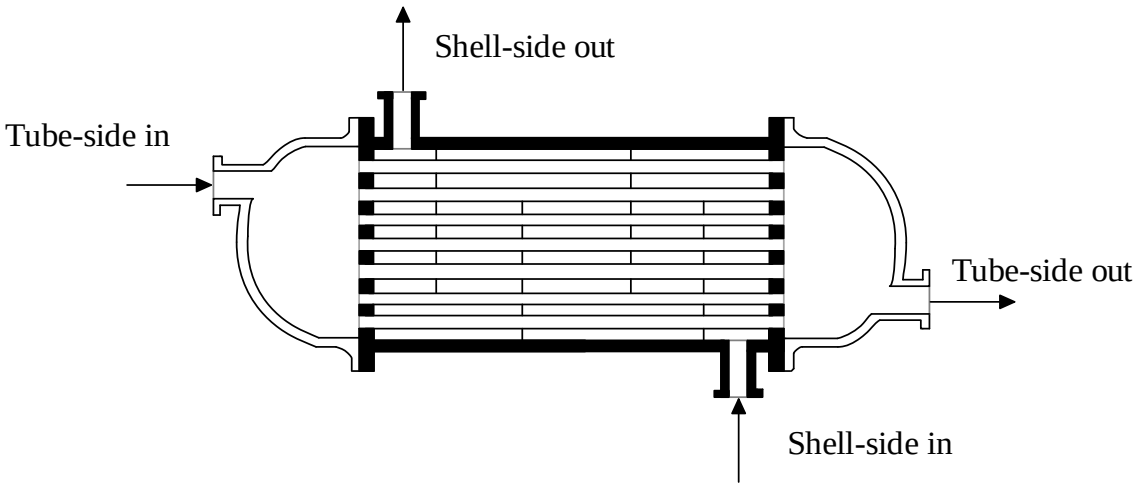

Figure 12. An exemplary model of a typical shell-and-tube heat exchanger (GNS Science).

The shell-and-tube heat exchanger consists of a shell and a set of tubes. The tubes go through the shell from the front end of the exchanger to rear end. The fluids engaged in heat transfer are separated by the tube walls. On the shell-side, disks called baffles are used to guide the flow perpendicularly to the tubes to enable crossflow heat transfer (Walker 1982, 58).

Plate heat exchangers are less common than their tubular counterparts are. They can be divided into four main groups - plate and frame, spiral, plate-coil and plate-fin heat exchangers. The operating principle of a plate-and-frame heat exchanger is illustrated in Figure 13.

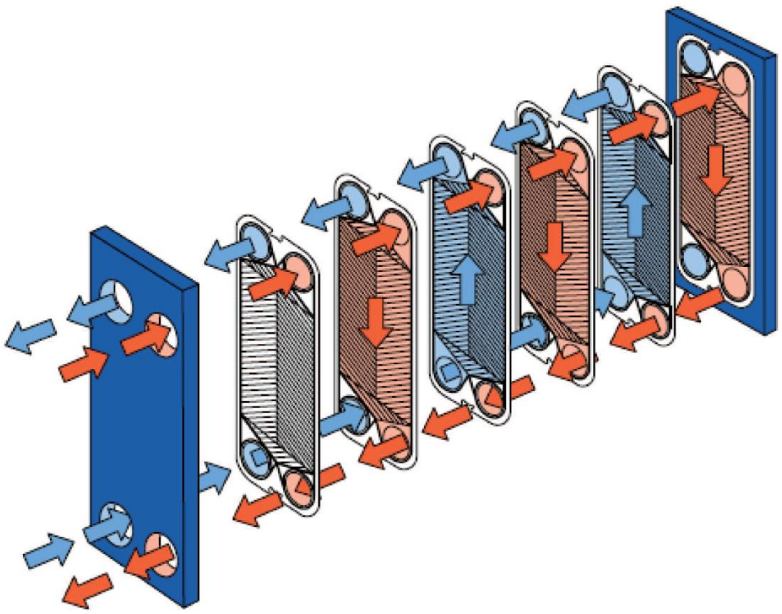

Figure 13. Schematic of a plate-and-frame heat exchanger (Alfa Brazing, 5).

Plate-and-frame heat exchanger is a type of plate exchanger, which has many advantages over shell-and-tube heat exchangers. Higher heat exchange effectiveness is reached due to counter current flow, reduced fouling and high turbulence, which is achieved with corrugated flow paths. Higher effectiveness brings more benefits. Less surface area is required in the heat exchanger, which leads to savings in the space required for the device. Operating costs are reduced because less pumping power is required due to a reduced flow rate requirement. In addition, effectiveness permits reduced gross mass of the heat exchanger, which further translates into lower installation, foundation and shipping costs.

Additionally, the capacity of the heat exchanger can be relatively conveniently altered by changing the amount of plates in the heat exchanger. (Walker 1982, 92.)

Plate heat exchangers in general have grown in popularity during the past few decades as engineering practices have developed. Due to their high effectiveness, they are used especially in applications where the temperature difference potential is relatively low. Heat from effluents such as exhaust air in air handling systems is often recovered with a plate-fin heat exchanger (Figure 14).

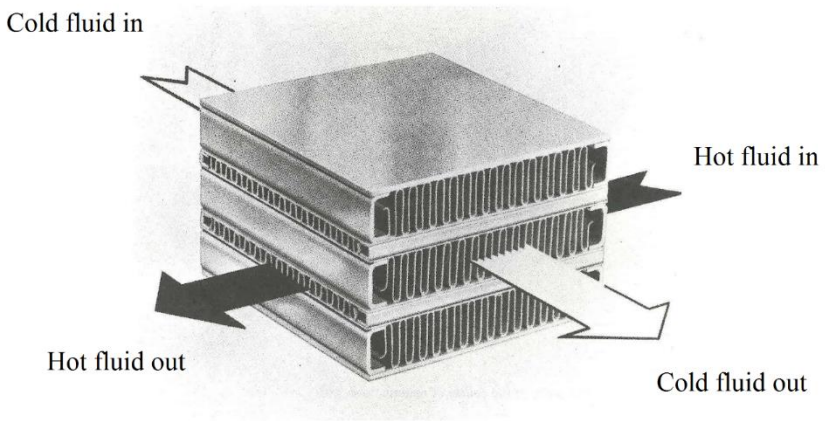

Figure 14. Flow configuration of the plate-fin heat exchanger (Walker 1982, 107).

The heat exchangers previously mentioned in this chapter are all classified as recuperative heat exchangers. Regenerative heat exchangers form another notable class of heat exchangers. They can be further divided into dynamic and static exchangers. In most cases, dynamic regenerative heat exchangers have a rotating hollow drum, which serves as a thermal storage. The hot flow heats up parts of the drum, which are moved over to the cold flow's way in a continuous rotational manner. The heat stored in the drum on the hot side is then released into the colder fluid. The flow principle of the rotary regenerative heat exchanger is illustrated in Figure 15.

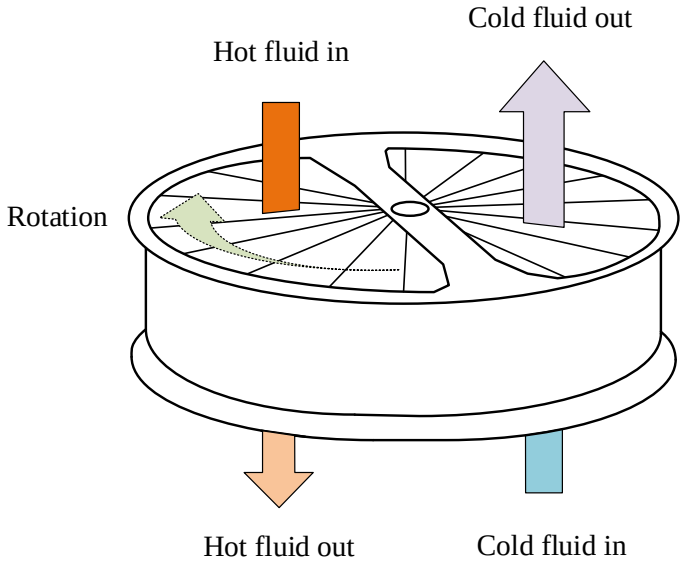

Figure 15. A rotating regenerative heat exchanger without the surrounding ductwork (Vulloju et al. 2014).

Rotating regenerative heat exchangers are used in electric power plants to transfer heat from flue gases to combustion air. They are also utilized in air conditioning systems to recover heat from exhaust air.

2.4.4 Filters and membranes

Filters and membranes are used in fluid handling systems to trap particles and impurities from flowing fluids and prevent them from continuing further along the system with the fluid. The matter trapped in the filter or membrane often ends up in suspension, which leads to clogging and fouling of the filter. Once a certain degree of fouling is reached, the filtration equipment will usually have to be either replaced or cleaned.

In flow systems, where the fluid handled is liquid, filters and membranes are used in wastewater treatment, in the production of clean water and in many processes of the chemical industry (Sutherland 2013, 23 & 25). One of the most common types of membrane filters is the spiral wound element type (Bódalo-Santoyo et al. 2004). The working principle and the construction of the spiral wound element membrane is illustrated in Figure 16.

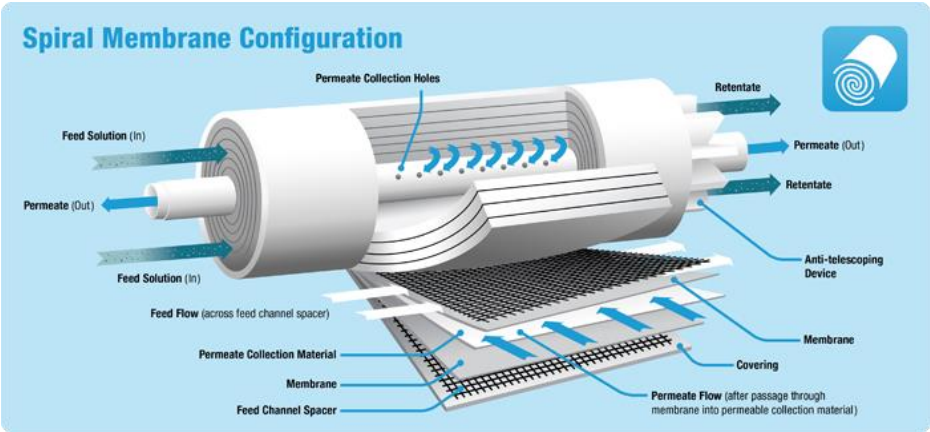

Figure 16. The configuration and flow principle of a spiral wound membrane element (Koch Membrane Systems).

Membrane filters are susceptible to fouling. The forming of a growing layer of foulant, also referred to as “cake”, has a detrimental effect on the membrane performance. Fouling of the membrane results in decreased permeate flux and quality and increases the hydraulic resistance of the membrane (Abdelrasoul et al. 2013).

Filters are used in both industrial and municipal gas handling applications. In air handling, they are used to remove impurities from air before delivering it to spaces and heat exchangers in the system. In industrial processes, filters are for example used to remove particles and other impurities from exhaust gases. Their purpose is to reduce the fouling effect of the gas on heat exchangers and to create cleaner emissions. Due to their function, which is to seize impurities and prevent them from continuing with the gas flow, filters tend to foul over time. The fouling increases the flow resistance caused by the filter, which again increases the required pressure from the fan.

In compressed-air systems, filters are often used both before and after the compressor. Inlet filters protect the compressor from impurities such as particles possibly contained in the atmospheric air (Lawrence Berkeley National Laboratory 2003b, 10). Depending on the requirements of the end user, filters are used after the compressor on the compressed air side to remove particulates, condensate and lubricant from the air (Lawrence Berkeley National Laboratory 2003b, 13). As in systems with pumps and blowers, also filters in compressed-air systems experience fouling, which leads to a greater pressure loss across the filter.

3 PROBLEMS IN FLUID HANDLING SYSTEMS

The need to enhance the financial and energy efficiency of fluid handling systems has led to research in efficient design and operation of flow systems. However, regardless of optimal design and operation, some systems and their components are prone to certain problems and faults. Reliable, automatic and fast detection of these faults can save energy and working hours as well as help avoid operation breaks and damage to process equipment, which can lead to significant production losses.

In the following, common problems of fluid handling systems found in the literature review are introduced and a classification of these problems is proposed. A way to divide the problems into categories may help in the application of the developed detection methods in the future. The solutions for a certain problem's detection method may be in some cases expanded to apply for other cases with a problem of the same category. Here, problems are presented based on their classification. The following chapters are named after the class and common practical examples found in the literature review are listed in the respective chapters.

3.1 System-related issues

Issues, which occur in the fluid handling system surrounding the flow device, are here referred to as system-related issues. This category does not include problems of the flow device itself or problems of its motor and drive. In the following sub-chapters, system-related issues found in the literature review are introduced.

3.1.1 Fouling and clogging

For system components and piping, unwanted deposition of material on flow surfaces increases the hydraulic resistance to the flow and, in the worst cases, can fully prevent the fluid from flowing. Fouling and clogging influence the system characteristics and have an

effect on the required flow device power. The change in flow device power depends on the control strategy of the flow device.

Heat exchangers are typically prone to fouling and clogging. The economic effect of fouling can be seen as increased life-cycle costs. As opposed to an ideal, although an unrealistic, completely fouling-resistant heat exchanger, fouling leads to over dimensioning of heat exchangers, which has an increasing effect on the investment and installation costs. Fouling also increases the fuel cost in systems, where the reduced heat transfer capacity leads to a need to consume more fuel. In addition, the pumping energy costs increase, as fouling increases the hydraulic resistance of the heat exchanger. Furthermore, cleaning the heat exchanger increases maintenance costs and production losses may occur when the heat exchanger is temporarily taken out of use. (Springer 2010, 85-86.)

Studies have shown that fouling may account for 2.5 % of global CO_2 emissions. In addition, the costs related to fouling have been estimated to be 0.25 % of the GDP of industrialized nations (Heat Exchanger Fouling and Cleaning a). Typical industrial heat exchangers that suffer from fouling include, for example, black liquor evaporators, power plant condensers and crude oil distillation preheater. Also heat exchangers in HVAC applications often experience fouling (Siegel 2002, 2).

Continuous effort is put into research on heat exchanger fouling. Studies usually focus on specific aspects of fouling: the fouling mechanism itself, cleaning methods and scheduling, fouling mitigation, fouling monitoring, fouling in a specific process, etc. A large biennial conference is held where research on the aforementioned topics is presented and discussed (Heat Exchanger Fouling and Cleaning b). The proceedings of two other notable conferences have been compiled and edited by (Somerscales and Knudsen 1981) and (Melo et al. 1988).

The fouling and clogging of filters and membranes is a common issue in systems where they exist. In membranes, fouling causes a decrease in the permeate flux, i.e. the flow rate

is decreased because of the added hydraulic resistance. Also the permeate quality is negatively affected by fouling.

As the importance and acceptance of membranes has grown, the amount of research conducted to study their fouling has also increased. The topics of research related to membrane fouling include, for example, membrane- and substance-specific fouling mechanisms, the reduction of fouling and cleaning of fouled membranes. These topics are covered in scientific journals such as “Separation and Purification Technology”, “Journal of Membrane Science” and “Desalination”.

In HVAC systems, filters are used to ensure good indoor air quality and to prevent heat exchanger fouling by preventing air impurities from travelling with the flow to heat transfer equipment and to building occupants. Fouling increases the pressure drop across the filter, which leads to higher energy costs for the fan. Additionally, as the filter’s flow resistance increases, more of the unfiltered flow will bypass the filter through air gaps between the filter and the duct wall, which are common in HVAC systems (Ward and Siegel 2005; Walker et al. 2013).

The effect of filtering efficiency on the energy efficiency of HVAC systems has been extensively studied (Stephens et al. 2010a; Stephens et al. 2010b; Noh and Hwang 2010; Nassif 2012). The efficiency of filters has been quantified by ASHRAE (The American Society of Heating, Refrigerating and Air-Conditioning Engineers) as “MERV” (Minimum Efficiency Rating Value) (Zaatari et al. 2014). The airflow pressure drop caused by a high-efficiency (e.g. MERV 13-16) air filter is greater than that of a filter of a lower efficiency category (e.g. MERV 1-8) (Walker et al. 2013). As increasing the filter media’s efficiency and the loading of a filter both increase the flow resistance of the filter, the magnitude of the effect of fouling can be predicted from the results of the aforementioned studies.

In addition to energy efficiency, fouling also affects the indoor air quality through at least two effects. Firstly, fouling may increase the amount of unfiltered bypass air ending up in occupied spaces. Secondly, the deposit gathering on a filter may serve as a platform for the

accumulation and growth of biological contaminants, from which they may be released into the airflow (Waring and Siegel 2008).

The piping of a fluid handling system can also suffer from clogging. In some systems, the solids contained in the fluid may end up in suspension, or scaling may occur when the solubility limit of a substance is reached. These phenomena may lead to problems of clogging of piping. The deposits or scale accumulating on the surface of piping reduce the cross sectional area of the pipe, which increases flow friction losses as seen from equations (2) and (3). In some cases, there is a threat of complete clogging, where the deposits can accumulate to a point where they fully prevent the flow of the fluid.

In pulp and paper mills, green liquor is produced in a tank by dissolving molten smelt with weak wash. Scaling occurs in components that handle green liquor, including piping through which green liquor is transferred. (Zakir et al. 2013.) Scale starts to form when the saturation of the scale components exceeds the solubility limit (Sitholé; Zakir 2011, 7). The rate of scaling in green liquor lines is relatively fast. Complete clogging can occur in dozens of hours. Periodical backflush in the line is used to keep the line clean enough for the green liquor to flow. (Vakkilainen, interview 26 June 2015.)

3.1.2 Leaks

Leaks can occur in any fluid handling system with pressures higher than that of the atmosphere. In addition to material loss, leaks cause energy losses in some systems. In steam systems, for instance, the steam is valuable because of its high energy content. Compressed air is also a relatively expensive utility, where leakages occur. Compressed-air systems are especially prone to leaks, with leakage rates of 10 - 50 % of the produced compressed air being common for poorly managed systems (United States Department of Energy 2000, 1; Carbon Trust 2012, 12).

3.2 Flow device

Problems in the flow device itself can have a negative impact on the energy efficiency of operation and on the reliability of the device. For pumps, for example, problems such as bearing and sealing failure and impeller wear are usually caused by adverse operating conditions. Instead of detecting and fixing these problems as they occur, attention should be directed towards detecting the operating conditions, which create the problems.

Conditions and circumstances, which lead to problems in the flow device often lie outside the device itself. For example harmful consistency of the flowing fluid, such as dirt or debris, can lead to impeller fouling in pumps and fans or even clogging in pumps. Furthermore, for pumps, running the device far from its Best Efficiency Point (BEP) will bring about risks of different kinds of adverse effects. At the BEP, the uniformity of the average static pressure in the volute around the impeller is at its highest (Barrio et al. 2011, 1). Deviating from the BEP will disturb the flow and pressure patterns in the volute and create a radial thrust load, which will increase the load on the bearings and cause movement of the shaft seal. Due to this effect, the service life of the bearings and the shaft seal will be reduced. (Budris.) A pump can also suffer from cavitation damage when it is run too far to the right or to the left on its *QH* curve (Barringer 2003, 8).

A pump can operate far away from its BEP for two reasons: because of poor system design and because of changes in the surrounding system's characteristics. Oversizing a pump to make sure its output is enough to supply the rest of the process is a typical system design mistake, which will make the pump run further away from its BEP than would be possible. In addition to poor sizing, the surrounding system and its characteristic curve can change over time in such a way that even a pump, which initially operated at BEP, may end up running at a non-preferred region on its curve. This is due to the fact that the pump's operating point is defined by the intersection of the pump and system curves. For example, build-ups of deposits or wearing of piping or system components can alter the process characteristics.

Somewhat like with pumps, also fans have what is referred to as unstable operating regions. Fans, when operated too far on the left of their performance curve, are prone to the effects of surge and stalling. These adverse effects can greatly increase the noise of the fan and damage the fan and the ductwork through vibration and drastically changing loading. (Aerovent 2012, 1).

3.3 System control and sensor failures

Sensors are used to keep track of process variables such as, for example, pressure, flow rate and temperature in industrial and municipal fluid handling systems. Data and signals produced by sensors are used in the control of process devices. For example, to maintain a specific pressure in a duct or piping, a fan or a pump can be speed controlled according to a pressure measurement. Similarly, in the case of a fixed-speed flow device, the pressure could be regulated with a valve or a damper, which could be controlled according to a measurement of the pressure.

In automatically controlled systems, sensors are trusted to guide the control elements to keep the process performing in desired ways. However, when sensors fail and no other methods to detect the failure and to provide a substitutive control signal exist in a system, the system can run end up running erroneously. Bad operation can further create safety, efficiency and reliability related problems for the process.

A common method to detect faulty sensors is to include additional redundant metering. Measuring the same process variable with multiple sensors makes it possible to detect the failure of an individual sensor by comparing with each other the signals produced by each of the sensors. Other, more analytical methods, which use a model of the process in question to find sensor faults have also been developed and studied. (Afonso 1998.)

3.4 VSD in system operation monitoring

The growing general desire to operate systems energy efficiently and the discovered energy saving potential of fluid handling systems together increase operators' interest in the energy efficient operation of fluid handling systems. Flow and pressure sensors, for instance, are traditionally used to determine the output of the system, which is crucial in finding out how energy efficiently the system operates. However, research has shown that the system output can be monitored with the application of model-based methods, which require less metering in the system (Tamminen et al. 2013). VSD's can provide estimates for the rotational speed, the torque and ultimately the shaft power of a fluid handling device. The developed methods allow defining the operating point of the flow device with the application of these estimates and the flow device's performance curves.

In addition to monitoring the energy efficiency, variable-speed drives could be used to detect the fluid handling system problems, which were described in the preceding chapters. Problems in system components other than the flow device itself can lead to a change in the system characteristics, which will change the operating points of the flow device. The change in the operating point can be monitored with a model-based estimation method. Faults and problems occurring in the flow device could also possibly be detected with such a method. In this study, the possibility to use a variable-speed drive's estimates to detect the problems of the case locations with a reduced amount of metering is studied. In a recent study done by Tamminen et al., it was found that contamination build-up on the impeller of a fan could be detected with the use of the torque estimate of a variable-speed drive (Tamminen et al. 2015). Methods have also been developed and tested for sewage pumping, where build-up of debris in the inlet of the pump and on the impeller is detected and cleaned with the help of a VSD (Water & Wastewater Treatment 2013; Moore 2011). The methods can work in a passive manner by only collecting and evaluating process data during normal operation or more actively by introducing an identification run into the process, which will enable the detection of a change in an examined process variable.

Traditionally, problems are detected through either automatic or manual monitoring with sensors, during scheduled maintenance or simply when the problem causes operations to

cease. The early detection of a problem can help minimize the life-cycle costs related to it. For example, for heat exchangers, monitoring data about the progress of fouling can be used together with the results of an optimal cleaning cycle study to optimize the operating and cleaning schedule of a fouling heat exchanger. Moreover, in cases where for example a fan suffers from contamination build-up on its impeller, early detection of the phenomenon can prevent flow device failure, which could potentially occur due to the impeller imbalance caused by the build-up. Many fluid handling systems use VSDs and more are installed as time passes, often as a result of energy audits. Therefore, once developed, methods for problem detection can be easily implemented in a large and constantly increasing number of systems.

In some cases, VSD-based methods can replace metering, and in some work as an alternative means of monitoring, with which the measured performance values can be verified. Replacing metering with estimation and detection methods will bring savings, as the price for acquiring and installing sensors will be avoided. Where metering already exists, VSD-based estimation and detection methods can be used alongside with sensor-based monitoring. In the case of, for example sensor failure, a VSD-based method can continue to provide reliable monitoring data on the system performance.

4 INDUSTRIAL CASE STUDIES OF PROBLEMS IN FLUID HANDLING SYSTEMS

For the case analysis, industrial companies with fluid handling systems as part of their processes were considered. Contact was made to companies supposedly using pumps, fans and compressors as vital parts of their processes. During the search for suitable case study locations, three cases of fouling heat exchangers emerged. The three heat exchangers are of distinct importance. The first heat exchanger, a nuclear power plant generator cooler, has significance from a process safety and plant availability point of view. The second case comprises a heat exchanger, the fouling of which has an effect on the environmental performance of a pulp and paper mill. The third and final case presents a heat exchanger, which is important energy-efficiency-wise. In the following subchapters, the cases are presented and the possibility to use a VSD to monitor the state of fouling is discussed.

4.1 Power plant generator cooler

Earlier communication with the staff of the Olkiluoto 2 nuclear power plant revealed a case example of a fouling heat exchanger in the plant's generator cooling circuit. With further queries, more information about the cooling system and data from the measurements of the system were acquired. An illustration of the cooling system with exemplary temperatures of the fluids is presented in Figure 17.

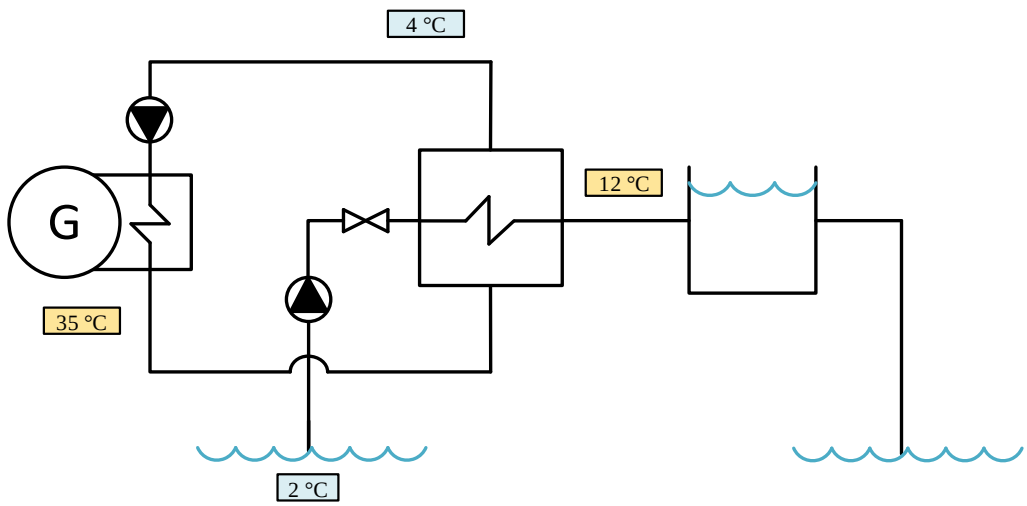

Figure 17. The cooling system of the nuclear power plant's generator.

The cooling demand, which is usually between 6 and 7 MW, is satisfied with a heat exchanger, which transfers heat between two water flows. Here, the cold water is pumped from its source, the sea, through the heat exchanger by a pump, which is located upstream of the heat exchanger. Between the heat exchanger and the pump, there is a manual control valve, which is used to keep the flow rate within a certain range. Downstream of the heat exchanger, there is a reservoir, from which the water is released back into the sea. The level of water in the reservoir remains constant. The cold seawater side of the cooling system can be considered an open loop system.

In the heat exchanger, the cold seawater receives heat from water in another circulation loop. In this closed loop, the cooling water, which has been cooled in the heat exchanger, is pumped through parts of the generator, which require cooling. Heat is again transferred from the generator into the circulating water. The water warmed up by the generator then returns back to the heat exchanger to be cooled by the seawater. This side of the generator cooling application can be considered a closed loop system, as the water is taken through a path with matching start- and end-points. In reality, the closed loop side consists of branches and includes three pumps for cooling the respective parts of the generator. However, as the general idea can be understood from a single branch loop, and because measurement data was acquired mainly from the open loop side of the system, a simplification has been made in the previous figure for the sake of clarity.

4.1.1 Operating principles

On the seawater side, two identical pumps are used, of which only one is running at a time. To achieve equal wear, the pumps are operated alternatively in three-month periods. The pumps are driven with induction motors running at equal constant rotational speeds. The pumps have their own branches in the piping and manual control valves downstream of them.

Two heat plate-and-frame heat exchangers are used periodically. The heat exchanger to be used is changed annually during the downtime maintenance, which lasts from the last week

of May until the first week of June. During the maintenance, the fouled heat exchanger is decommissioned and cleaned and the previously cleaned heat exchanger is taken into use. Annually, from July until October, sodium hypochlorite is released into the seawater upstream of the heat exchanger. It is done to reduce the fouling rate.

The manual control valves after the pumps are adjusted very seldom, and the moments when they were adjusted can easily be distinguished from the data. In addition, the seawater side of the cooling system is relatively simple and does not contain significant unknown process variables. Therefore, it is possible to find timespans where the increasing hydraulic resistance of the heat exchanger due to fouling can be assumed the greatest change in the system characteristics.

4.1.2 Acquired data

Data from various measurements in the cooling system was acquired. The measurement results, which were included in the data analysis, are

- Cooling water temperature before and after the heat exchanger (°C)
- Seawater temperature before and after the heat exchanger (°C)
- Volumetric flow in the open seawater loop (l/min)
- Electric current input to the motors of pumps 1 and 2 (% of nominal value)

The measurement results were recorded on an hourly basis and the acquired data ranges from July 2011 until August 2015. Additionally, the results of a weekly measurement of hydraulic pressure in the piping before and after the heat exchanger were provided.

For the pumps, test measurement results at the operating rotational speed were provided. Based on these results, the pump curves were defined by fitting third degree polynomials on the measurement data. The test measurement results and the fitted pump curves are shown in Figure 18. From the figure it can be seen that the *QP* curve's d*P*/d*Q* is sufficient for flow rate estimation based on the known shaft power. Also the *QH* curve can be considered suitable for head estimation due to its monotonicity, which will yield

unambiguous results when the head is estimated based on a flow rate estimate or measurement.

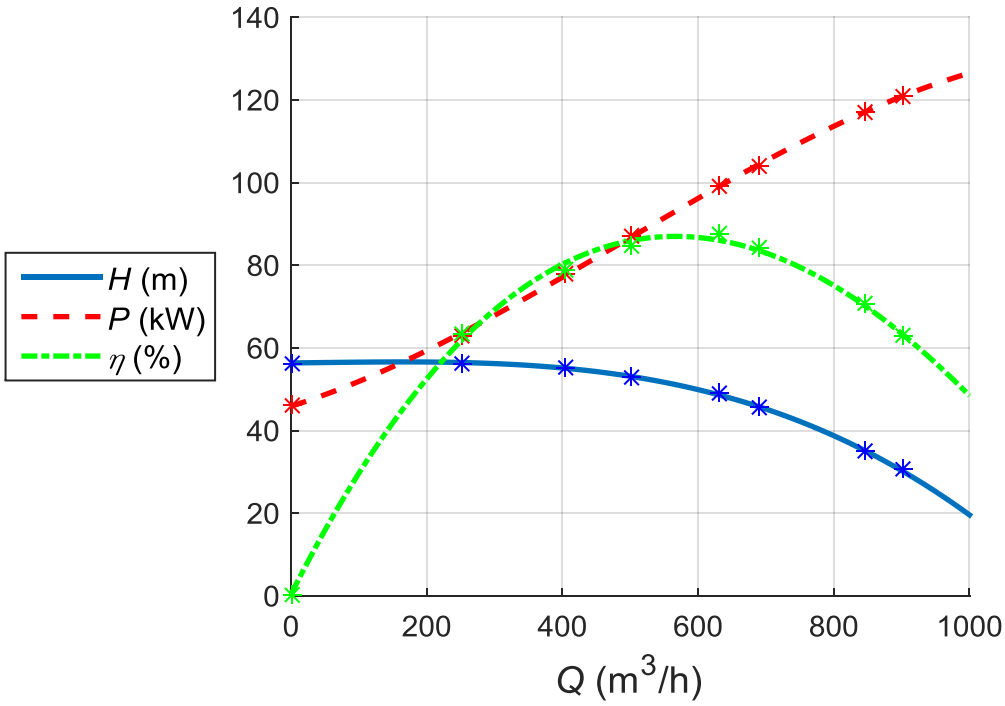

Figure 18. The fitted pump curves based on the given test measurement results.

4.1.3 Applicability of a VSD-based method

The fouling of the heat exchanger can be detected in two variables, which can be derived from the knowledge of other process variables. Fouling can be seen in the decreased heat transfer ability of the heat exchanger and the increased pressure drop across it. In this chapter, the possibility to replace the need for measurement instruments with estimates, which a VSD can provide is discussed. First, the possibility to use VSD estimates to detect changes in the heat transfer ability of the heat exchanger is studied. Then, the possibility to use the estimates to detect the increasing pressure loss of the heat exchanger is discussed.

The heat transfer rate of a heat exchanger was presented in equation (4). From the equation, a value that well describes the heat exchanger's ability to transfer heat can be derived:

$$UA = \frac{q_m * c_p * (T_{c_{out}} - T_{c_{in}})}{\Delta T_{LMTD}} \tag{6}$$

The *UA* value, in kW/K, describes the heat transferred by the heat exchanger per a degree of temperature difference. Fouling results in the generation of a thermally resistive layer onto the heat transfer surface, which decreases the heat transfer coefficient *U*. Therefore, as a variable, *UA* can be used to monitor the performance of the heat exchanger. The *UA* value was calculated based on the acquired measurement results. It was found that the rate of fouling was significant every year from August until April. The *UA* value from August 2012 until April 2014 is shown in Figure 19.

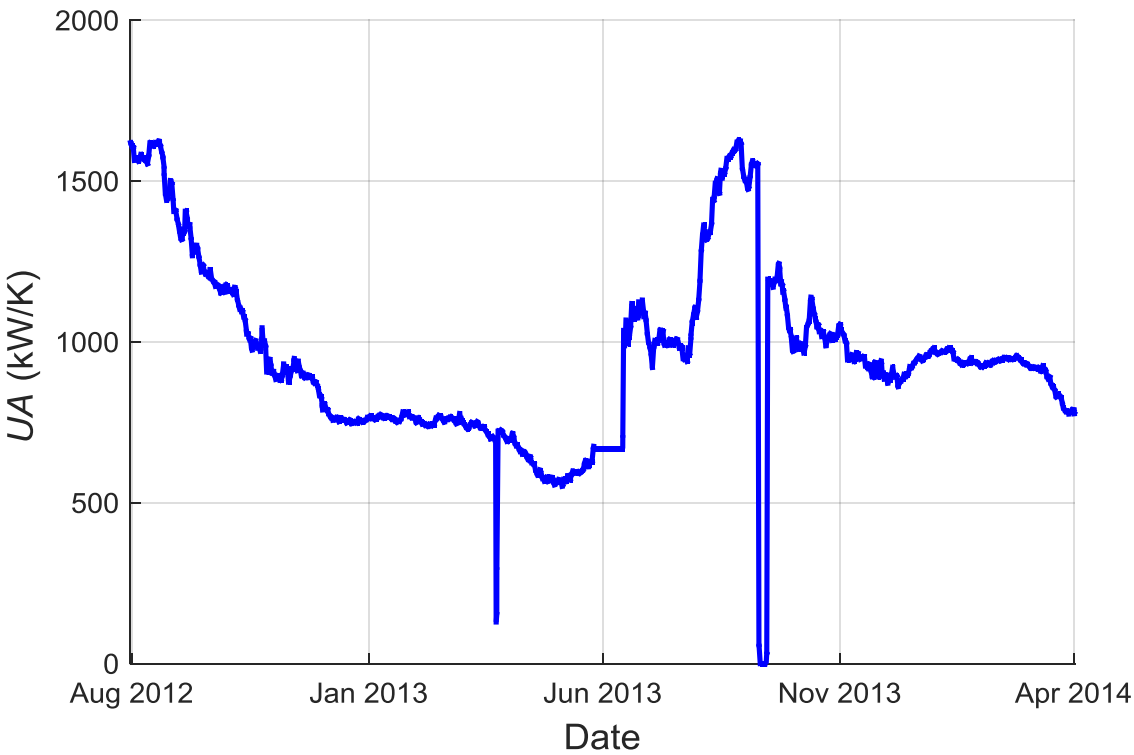

Figure 19. The *UA* value of the heat exchanger during August 2012 - April 2014.

As seen in equation (5), the inlet and outlet temperatures of the water on both sides of the heat exchanger and the mass flow rate must be known in order to determine the *UA* value. Temperature measurements are in this and many other cases required by other parts of the process. However, it may be possible to replace the flow measurement with an estimate of

the flow rate derived from the power consumption estimate and the pump curve with the *QP* method.

Here, the pump's *QP* curve was generated by fitting a third degree polynomial on the pump operating points included in the pump's test measurement results. The power consumption of the pump was derived using the measured electrical current. The three phase electric power supplied to the pump's motor can be calculated with equation

$$P_e = \sqrt{3} * U * I * \cos\phi \tag{7}$$

P_e = electric power (W)
U = voltage (V)
I = electric current (A)
$cos\ \phi$ = power factor

No information about the voltage or the power factor of the motor was available. To simplify the estimation of the power consumption, they were assumed constant. In addition to this, the motor efficiency was assumed constant. The simplification should not cause a significant error in the estimate, as the pumps' motors were run within a relatively narrow range of loads, as seen in Figure 21. With these assumptions, the given value for electric current as a percentage of the nominal current is equal to the power of the motor as a percentage of the nominal power. With the power consumption estimate, the volumetric flow rate was estimated with the *QP* method, as shown in Figure 20.

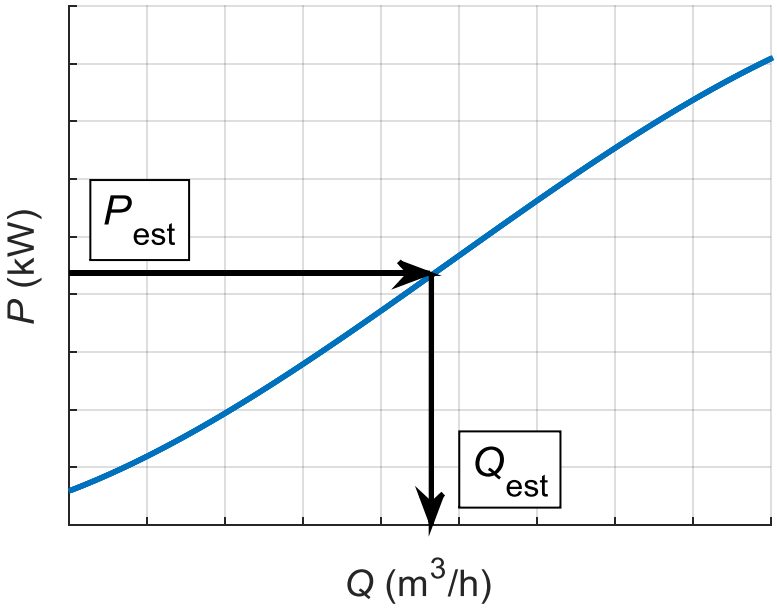

Figure 20. Estimates of the power consumption were used to derive an estimate for the volumetric flow rate.

The same *QP* curve was used to estimate the flow rates of both pump 1 and pump 2, as they were assumed to be identical and running at the same rotational speed. From both the measured and estimated values of volumetric flow rate, mass flow rate was derived using an assumed constant water density of 1000 kg/m^3.

The estimated and measured mass flow rates, the relative error of the mass flow rate estimate and the pump motors' power consumption are shown in Figure 21. The relative error of the estimate as a percentage of the measured mass flow rate is calculated with equation

$$error = \frac{q_{m,estimated} - q_{m,measured}}{q_{m,measured}} * 100\ \% \tag{8}$$

error = relative error of the estimated mass flow rate (%)

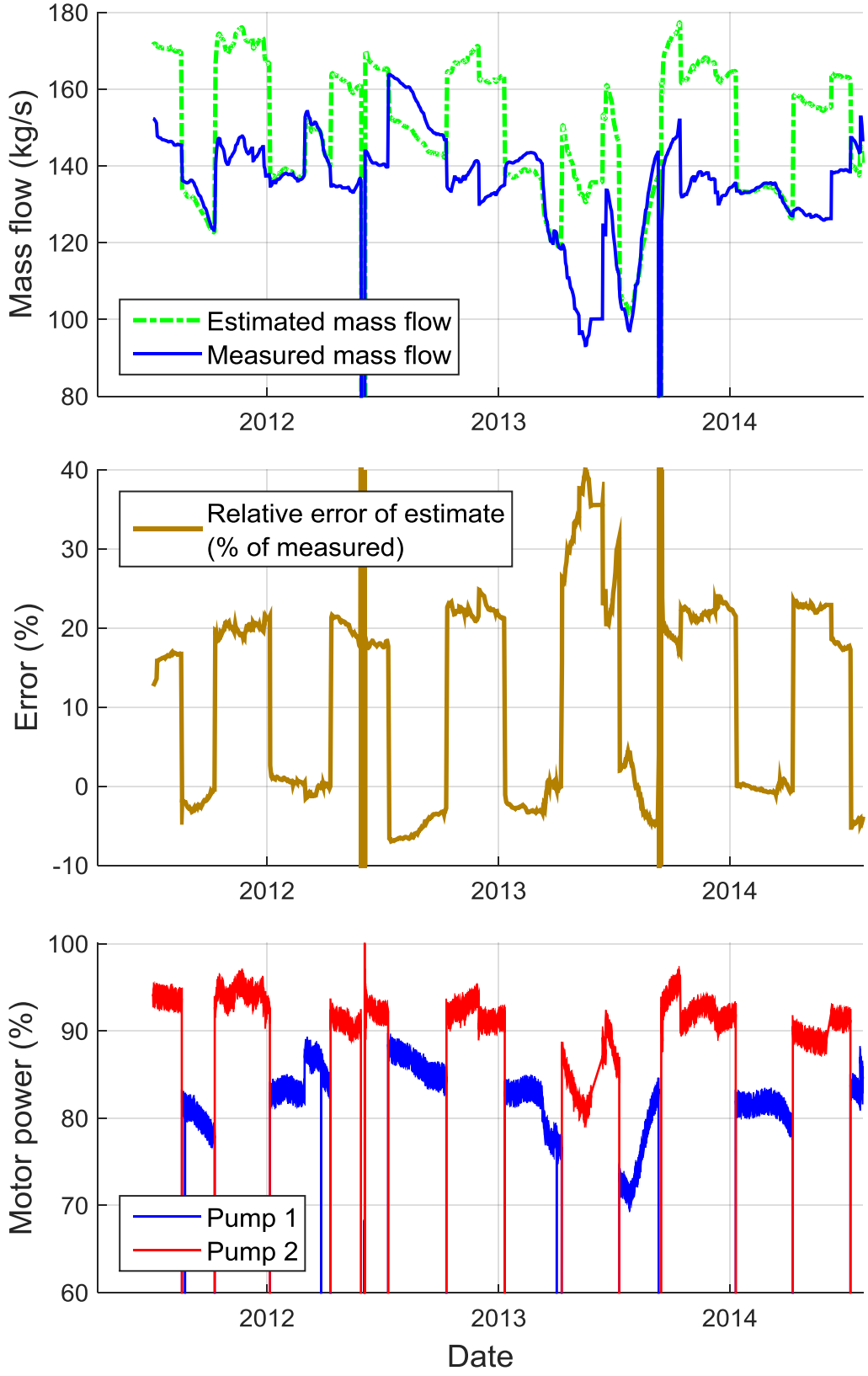

Figure 21. The measured and estimated mass flow rates, the relative error of the estimate and the pumps' motor power consumption.

As seen in Figure 21, the relative error of the estimate depends on which pump is in operation. For pump 1, the error is relatively small: the greatest deviation from the measured value is approximately -7 %. For pump 2, the error is significantly greater. However, it can be seen that the error for pump 2 does not deviate much from the value of 20 %. In addition, variations in the measured mass flow rate are clearly seen correspondingly in the estimated values. Therefore, for this case, a linear correction could be applied to the *QP* curve of pump 2 to reduce the error. A correction factor was determined by trying which value would correct the estimate to match the measured value. The reference date for setting the offset for the estimate was October 19th 2011 (Figure 22).

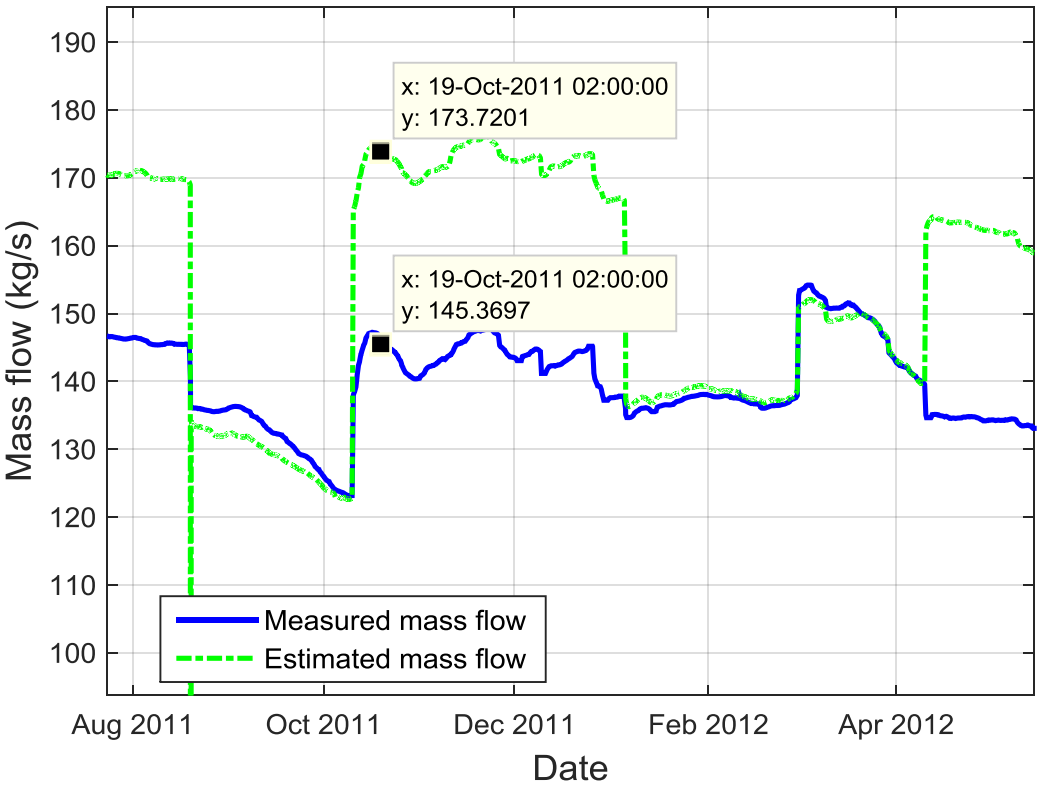

Figure 22. 19th of October 2011 was chosen as the date from which the values for correcting the flow estimate were taken.

The suitable value for the correction factor, by which the pump's *QP* curve's equation was multiplied, was found out to be 1.11. The resulting corrected pump curve, corrected mass flow rate estimate, the measured values and the relative error of the estimates are shown in Figure 23.

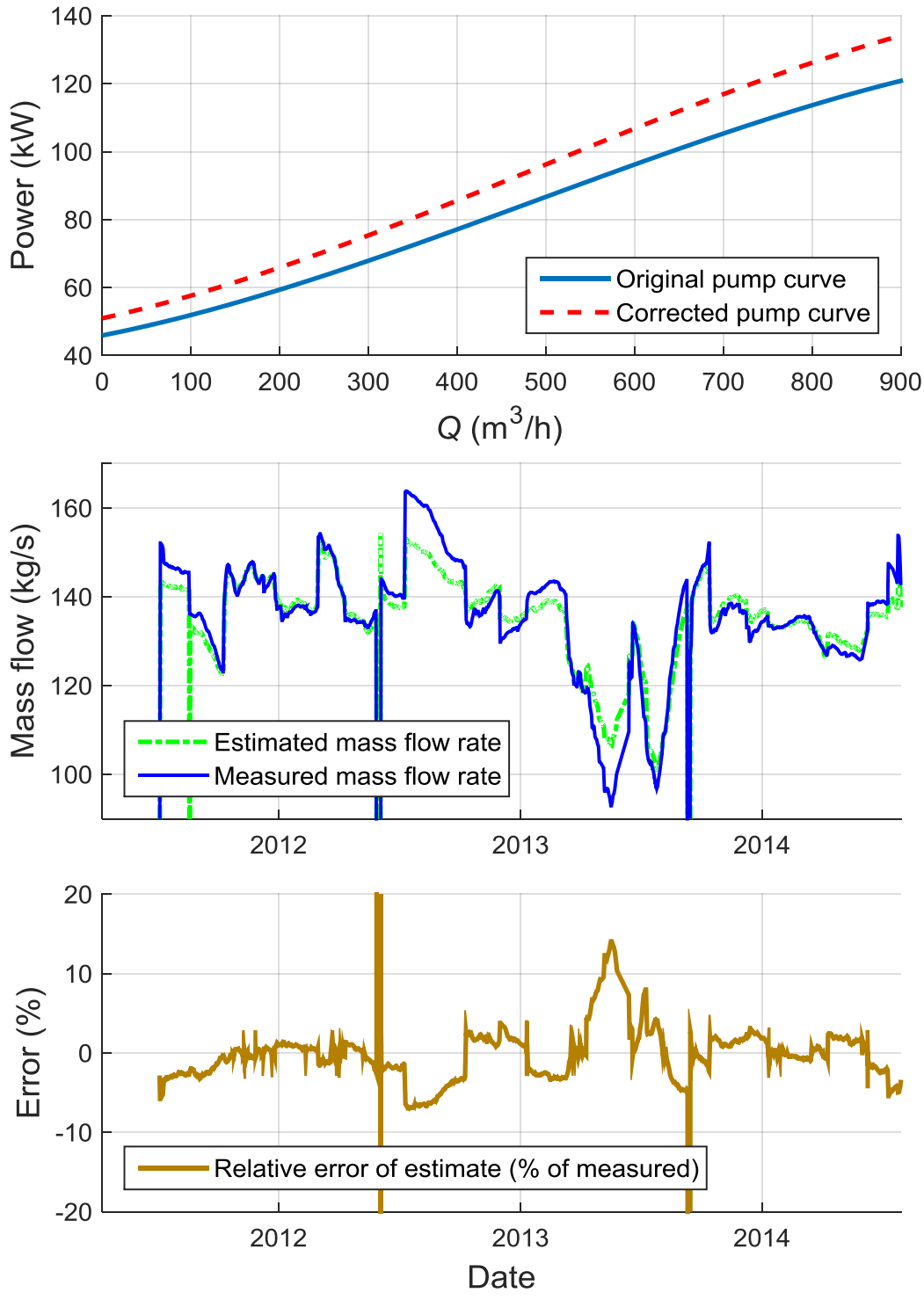

Figure 23. Corrected estimate for mass flow rate, measured mass flow rate and the relative error of the corrected estimate.

Figure 24 presents the *UA* values calculated using the measured mass flow rate and the estimated mass flow rate with and without the correction for the *QP* curve of pump 2.

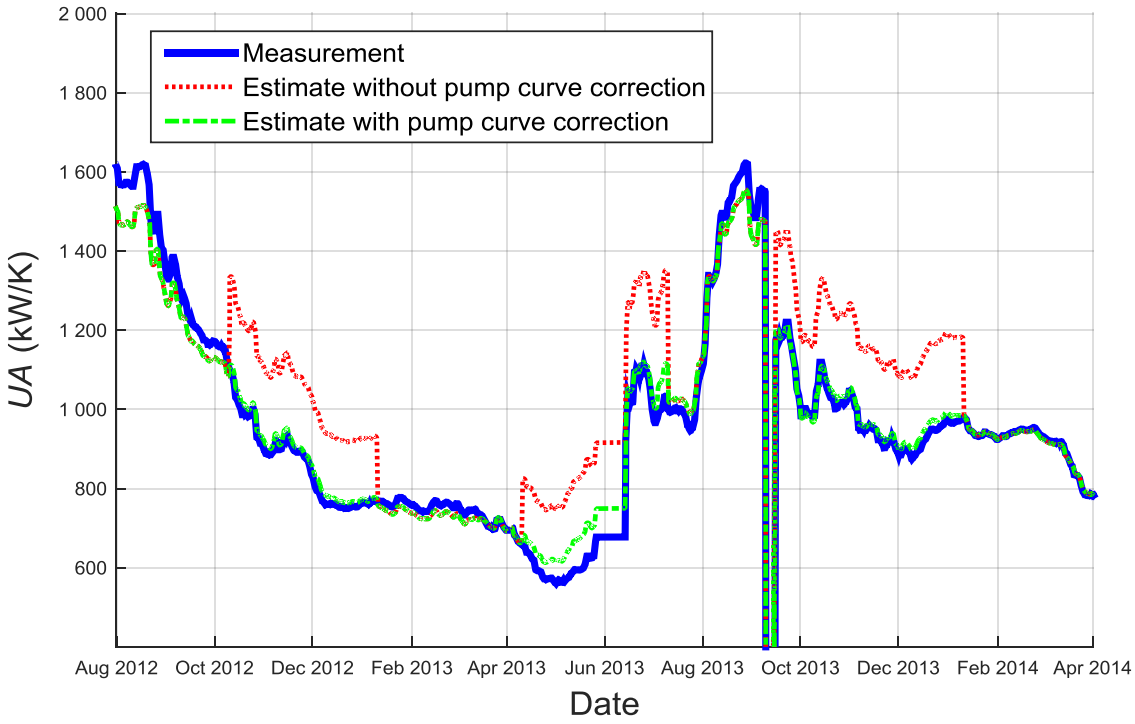

Figure 24. *UA* values calculated using estimated and measured values of the mass flow rate.

As can be seen in the figure above, the estimation method where the *UA* value is calculated using the estimated mass flow rate (the data set with the green plot), is relatively accurate when the pump curve can be assumed sufficiently accurate. Even with the error, which exists for the mass flow estimation for pump 2 (and included in the data set with the red plot), the trend of the *UA* value estimate follows that of the *UA* value, which is based on measurements.

The accuracy of the method, with which the *UA* value was estimated can be improved by taking into account the change in the rotational speed of the induction motor, which occurs as the electric current supplied to the motor changes (Ahonen et al. 2012). Due to the lack of the motors' performance characteristics, such a model could not be used in the presented analysis. In addition, some error occurs when the power consumption estimate for the

pump is generated, as the voltage and power factor of the electric motors were assumed constant.

For the flow estimates of both of the pumps, some error may be caused by a difference between the rotational speed for which the pump curve is defined and the rotational speed at which the pumps are actually driven. The flow estimation method was used assuming that the aforementioned rotational speeds are equal. Thus, difference in the rotational speeds would bring about a systematic error in the estimated values.

If pumps 1 and 2 are identical in terms of pump model and wear and they are driven with identical motors, as has been assumed, the error in the estimate for pump 2 could be caused by a systematic error in the measurement of its motor's power consumption. Presently, the correction for the flow estimate of pump 2 has been done by applying an offset for the pump curve directly. In the case of there being systematic error in the electric current measurement of the pump's motor, the correct would better be applied directly on the electric current measurement result.

The increased pressure drop across the heat exchanger can be possibly seen in the change of the pump's operating point. Figure 25 illustrates a typical pump *QH* curve and two system resistance curves.

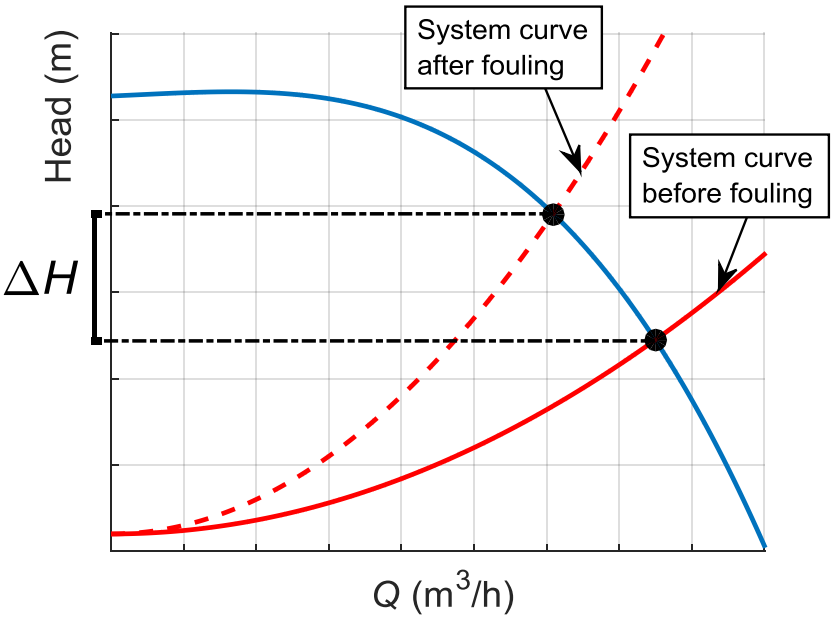

Figure 25. The operating points of a pump before and after heat exchanger fouling.

As the system's resistance increases because of fouling, the system curve steepens. When the pump is run at a constant rotational speed, it gets a new operating point on its *QH* curve. Periods of time can be distinguished from the data, during which the only change in the system characteristics is the increasing flow resistance due to fouling. The change in the pump's produced head should therefore depend on the increasing pressure drop across the heat exchanger.

To determine the head generated by the pump, the pump's *QH* curve was used. The corrected estimate of the volumetric flow rate was be used to derive an estimate for the pump head, as shown in Figure 26.

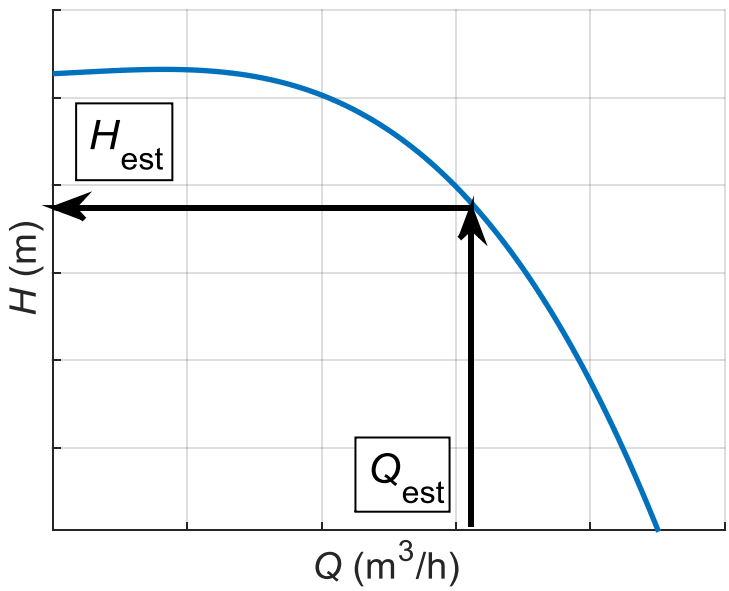

Figure 26. Estimates for volumetric flow rate were used to estimate the pump's head.

Using the aforementioned method, the differential head of the pump was estimated. The estimated differential head and the measured pressure loss across the heat exchanger are shown together in Figure 27. In the figure, fouling can be seen through an increasing differential head produced by the pump. However, there are periods, during which the head can be seen to decline. This is very likely due to the cleaning effect of sodium hypochlorite fed into the cooling water before the heat exchanger. This assumption is supported by the

observation that, in addition to the decreasing hydraulic resistance, the measured *UA* value recovers during these periods, which is the desired effect of sodium hypochlorite.

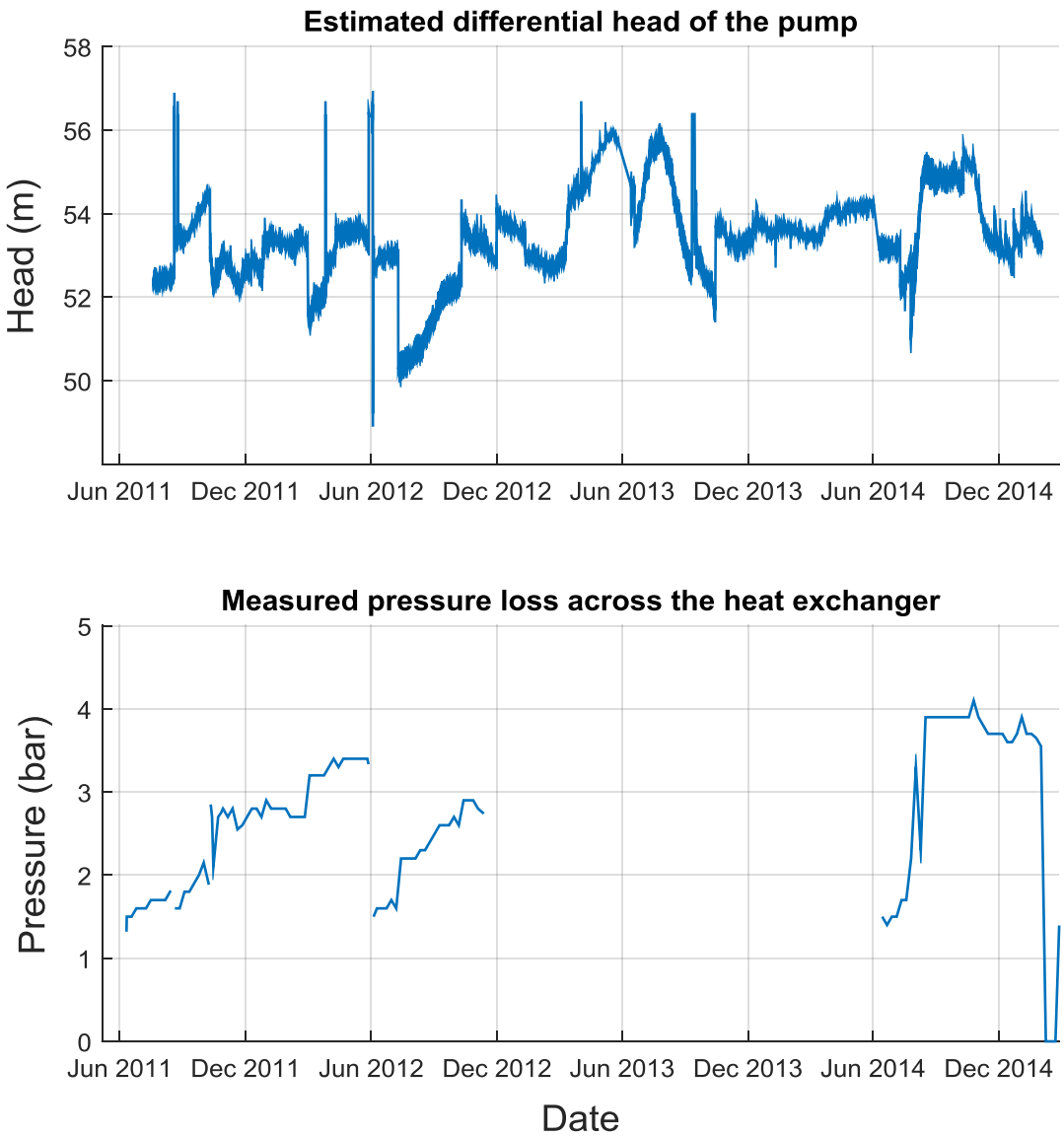

Figure 27. The estimated differential head of the pump and the measured pressure drop across the heat exchanger.

For a quantitative analysis, i.e. to see if the change in the pump's differential head corresponds with the measured increase in the pressure loss across the heat exchanger, a time period was chosen for the estimation, where the only change in the system could be assumed to have been the fouling effect of the heat exchanger. Also, because the pump

curve for pump 1 was found to be sufficiently accurate, a time period should be chosen during which pump 1 was operating. The period chosen for the analysis ranges from July 2012 until October 2012. During this time, the fouling created a constantly increasing pressure loss in the heat exchanger. Other periods with such visible constant increase in the hydraulic resistance of the heat exchanger could not be found, thus this period was chosen. It has to be noted, however, that during this time, the error of the flow estimate was at its highest, reaching a maximum deviation of approximately 7 % of the measured value. Therefore, in the following, an estimate of the pump's differential head derived also from the measured flow rate will be provided, and it can be compared to the head estimate, which is generated using the estimated flow rate. The estimated pump head from July 2012 until October 2012 is shown in Figure 28.

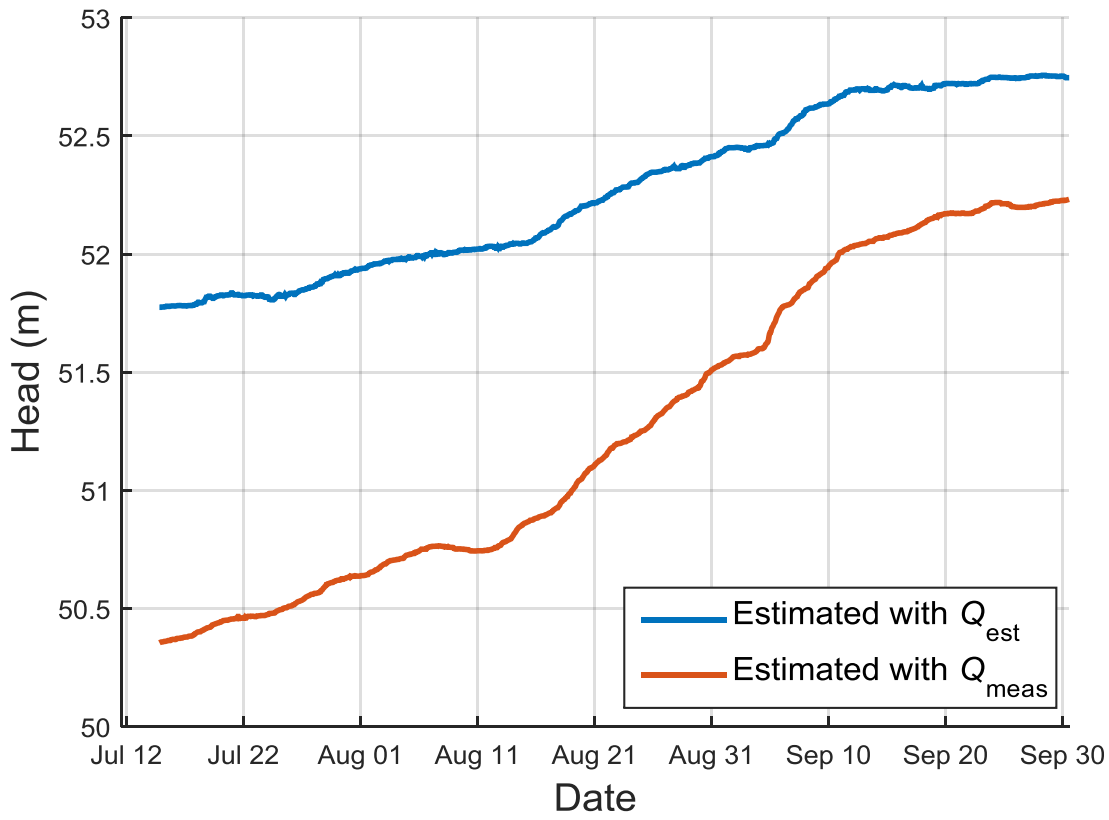

Figure 28. The estimated differential head of the pump during July 2012 - October 2012.

As can be seen from the graph and as can be expected (see Figure 25), the head produced by the pump increases as the heat exchanger fouls and its hydraulic resistance increases. A

comparison can then be made between the estimated pump head increase and the measurements of the pressure drop across the heat exchanger. The change in the actual pump head, expressed as pressure, should correspond with the increase in the measured pressure drop across the heat exchanger during the examined time period.

Head is expressed as a function of pressure in equation

$$H = \frac{p}{\rho * g} \tag{9}$$

H = head (m)
g = gravity of Earth (kg/s^2)

From equation (9), pressure in can be derived:

$$\Delta p = \Delta H * \rho * g. \tag{10}$$

The equation above was used to convert the increase in the pump's head into pressure. The reference point, from which on the increase in the estimated head and in the measured pressure drop across the heat exchanger was calculated, was chosen to be the 17th of July 2012. A comparison of the pressure loss increase derived from the pump's head estimate and of the measured pressure drop increase is shown in Figure 29.

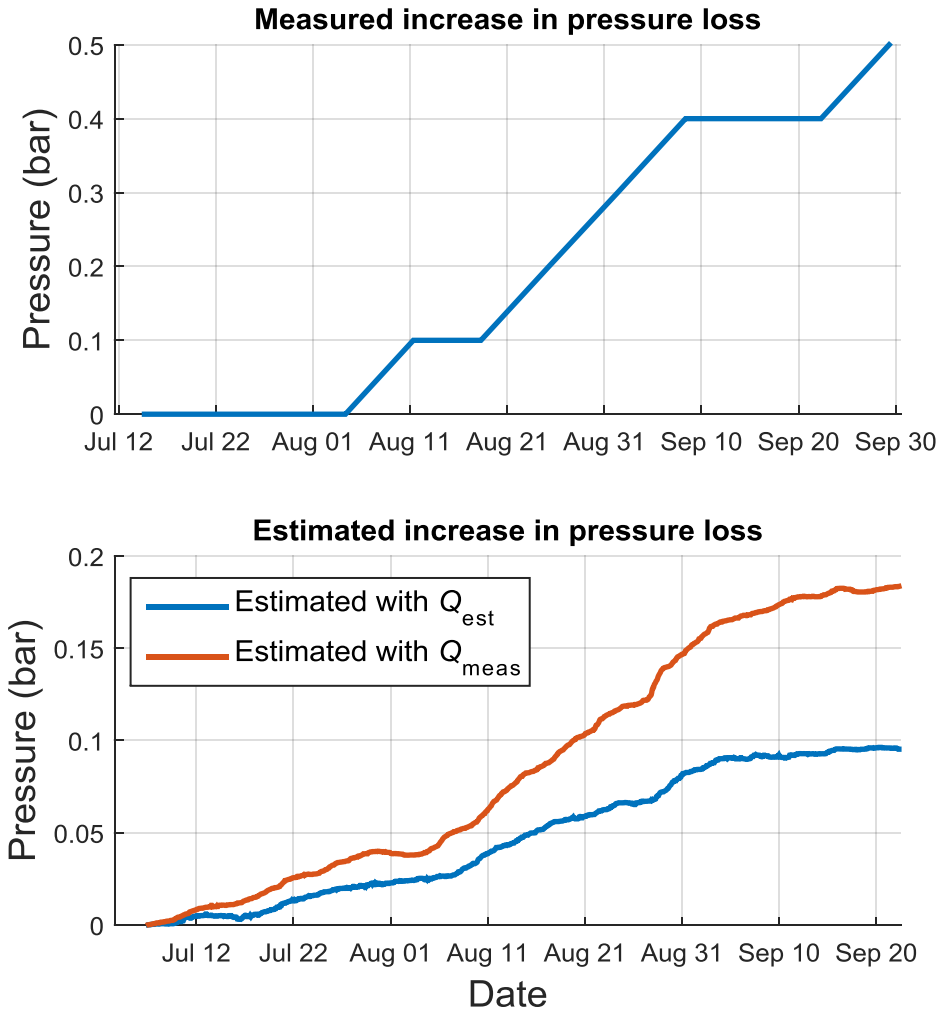

Figure 29. A comparison of the estimated increase in pressure lost and the measured pressure drop across the heat exchanger.

The analysis shows that the quantity of the estimated increase in the pressure drop does not correspond with the increase in the pressure drop of the heat exchanger seen in the measurements. During the examined time period, the estimation method suggests the increase in the heat exchanger's pressure drop had been approximately 0.1 bar. The measurements show that the actual increase in the pressure drop had been approximately 0.4 bar. The rising trend in the estimated change in the pressure drop, however, is similar to that of the measured values.

4.2 Paper and pulp mill vent gas cooler

Another case example of a fouling heat exchanger was discovered at a UPM pulp and paper mill in the Kaukas district of Lappeenranta. A shell-and-tube heat exchanger is used to cool vent gases coming from a causticizer and a slaker tank. The vent gases are used as secondary combustion air in the lime kiln. For the gas to be suitable to be used as combustion air, it is cooled and the moisture contained in it is condensed. The cleaning, which is required because of the fouling of the heat exchanger, does not cause downtime for the rest of the plant. A flow diagram of the heat exchanger and its surrounding system is shown in Figure 30.

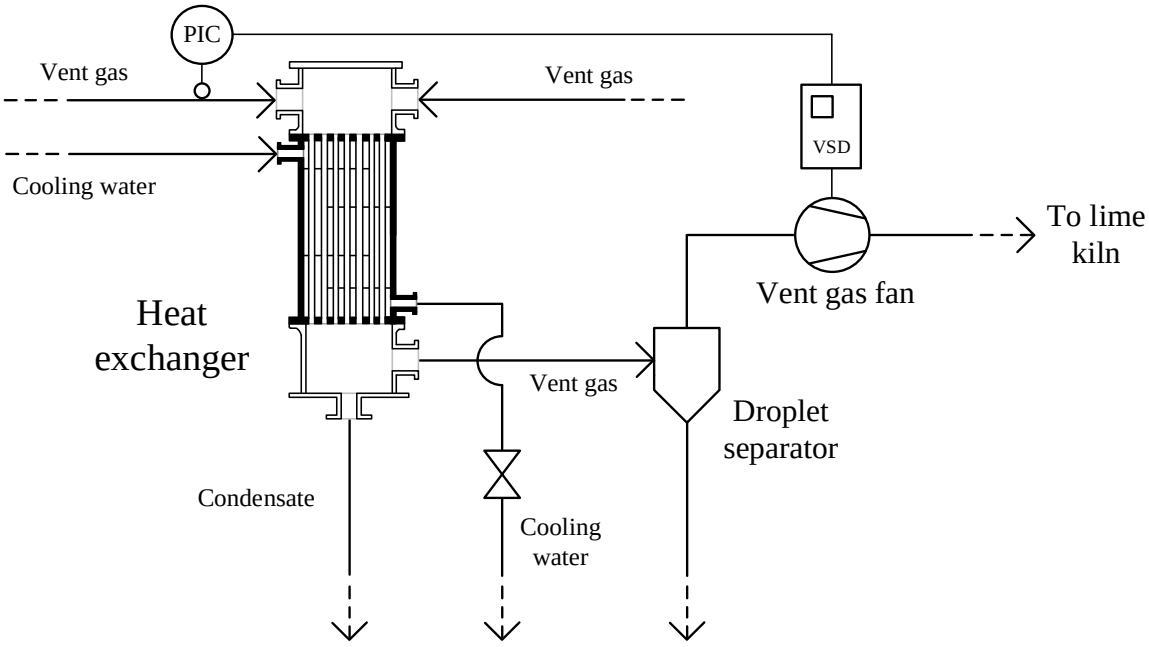

Figure 30. The vent gas cooler and the surrounding system. Here, PIC denotes the pressure measurement, which is used in the speed control of the vent gas fan.

The vent gas is taken to the heat exchanger's tube side through two individual pipe runs and they enter the heat exchanger through individual inlets, after which they are mixed in the front-end bonnet. The vent gas and the condensate exit the heat exchanger in the rear end through separate outlets. After the heat exchanger, a droplet separator and the fan, the vent gas is taken to the lime kiln to be used as secondary combustion air. Cold water from lake Saimaa is pumped through the shell side.

The fouling occurs on the tube side of the heat exchanger. Cleaning is required when the degree of fouling is great enough to significantly reduce the heat transfer capacity of the heat exchanger and the vent gas is no longer condensed. The frequency of the need to perform a clean-up is approximately once per three months.

4.2.1 Operating principles

The fan used to transfer vent gas is a centrifugal fan driven by an electric motor with a belt drive. The motor is speed controlled and the speed reference to the motor is defined by the pressure measurement in the pipe before the heat exchanger. As the heat exchanger experiences fouling, a higher rotational speed is required to maintain sufficient negative pressure in the piping before the heat exchanger.

The flow rate of the cooling rate is adjusted with a valve, as seen in Figure 29. The valve is controlled according to the temperature of the vent gas before the fan. As fouling occurs in the heat exchanger and its heat transfer capacity decreases, a higher cooling water flow rate is required to keep the vent gas temperature at the set value. Thus, as fouling occurs, the valve gradually opens.

4.2.2 Acquired data

Measurement data acquired from the system surrounding the heat exchanger includes:

- Pressure before the heat exchanger (kPa)
- Vent gas temperature after the cooler (°C)
- Cooling water valve position (%)
- Electric current supplied to the fan (%)
- Rotational speed of the fan (%)
- Volumetric flow rate of the vent gas after the fan (l/s)

For the aforementioned measurements, data on an hourly basis was acquired from January 2014 until September 2015.

4.2.3 Applicability of a VSD-based method

In this case, because of the lack of temperature measurements for all the inputs and outputs of the heat exchanger, the *UA* value cannot be calculated. Therefore, this analysis focuses on recognizing existing process parameters, in the values of which the fouling could be visible.

In the control scheme of the vent gas cooling system, the fouling of the cooler affects the fan's operation by increasing the rotational speed required to maintain the negative pressure before the cooler. In a control scheme such as this, the fouling could be detected by monitoring when the rotational speed requirement exceeds a certain level.

In addition, because of the control strategy, the cooling water valve is gradually opened as fouling occurs. However, the opening of the valve is not caused by the fouling alone, as also the incoming cooling water temperature has an effect on the heat transfer rate of the heat exchanger. For example, during summertime, the cooling water is significantly warmer than during wintertime. A warmer cooling water inlet temperature will require a more open valve. Therefore, the progress of fouling cannot be followed by monitoring the cooling water valve position alone.

Because fouling causes the fan to run at a higher rotational speed, it may be possible to see from its value, when the state of fouling is significant enough to require a clean-up operation. Figure 31 shows the rotational speed of the vent gas fan during a period, which encompasses multiple fouling-clean-up cycles. In the figure, the triangle symbols above the x-axis show when the heat exchanger was cleaned.

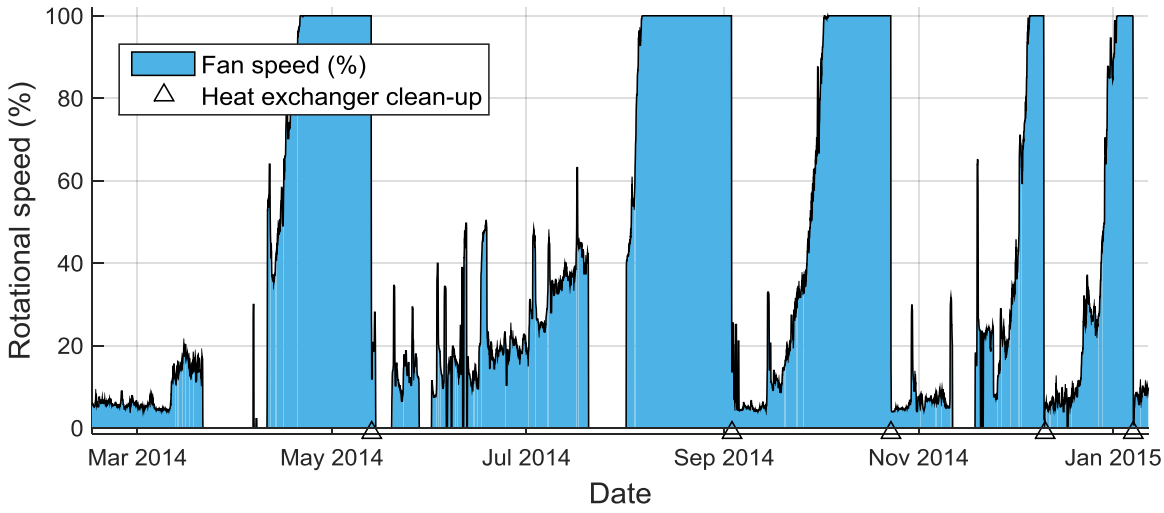

Figure 31. The rotational speed of the vent gas fan and the conducted clean-ups of the heat exchanger.

As a consequence of fouling, the fan will end up running at 100 % of its allowed rotational speed. As seen in the previous figure, the fan runs at full rotational speed until a clean-up operation is undertaken. During the examined time period, the fan was driven at full speed from 4 to 30 days before the clean-up of the heat exchanger.

In this case, a clean-up is needed when the fan fails to maintain a sufficient negative pressure in the piping before the heat exchanger. A state of operation where the fan is running at full speed and the pressure measurement indicates an insufficient negative pressure should indicate that the heat exchanger should be cleaned. Figure 32 shows the measured pressure before the heat exchanger and the rotational speed of the fan from a clean-up on the 3rd of September 2014 until the 23rd of October 2014, when the following clean-up took place.

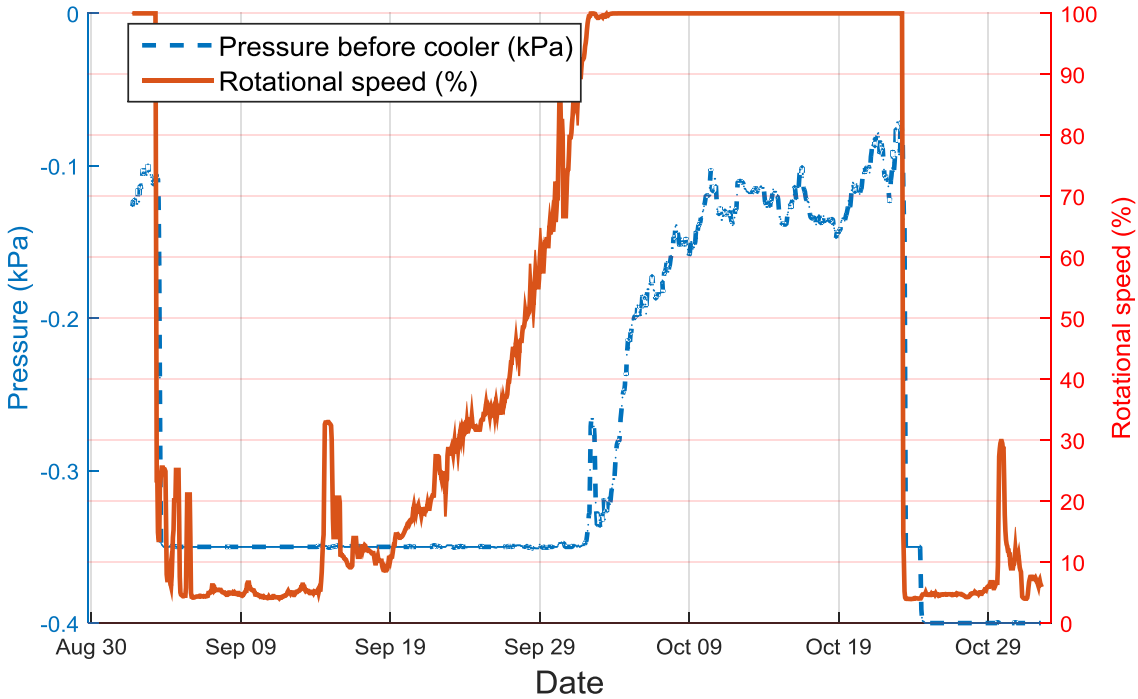

Figure 32. The rotational speed of the fan and the pressure before the cooler during a clean-up-fouling-clean-up cycle.

As can be seen from the figure, when maximum rotational speed is reached, the continuing fouling and the increasing flow resistance cause the pressure to deviate from its set value. In other words, the now fixed and no longer increasing rotational speed is not enough to maintain the desired negative pressure. An analysis of the times surrounding the moments at which the fan starts running at maximum rotational speed shows that it takes approximately 7 days from the moment the fan starts running at maximum rotational speed to the point where the pressure has diminished from a set value of -3.5 kPa to a value of approximately -0,1 kPa.

In conclusion, insufficient negative pressure is soon reached after the fan reaches its maximum rotational speed. This knowledge can help in determine an optimum cleaning cycle for the heat exchanger. Once notified of the fan reaching the maximum rotational speed, maintenance personnel can use the information to schedule a timely clean-up of the heat exchanger. Being able to predict the imminent state of insufficient negative pressure can help minimize the time when the system operates inefficiently.

4.3 Heat exchanger at a cement factory

An interview at a Finnsementti Oy cement factory in Lappeenranta revealed a case example of a fouling heat exchanger. A plate heat exchanger is used to transfer heat from hot flue gases to water. The recovered heat energy is transferred into the district heating network, where it replaces some of the energy that would otherwise have to be purchased from elsewhere. A flow diagram of the system surrounding the heat exchanger is shown in Figure 33.

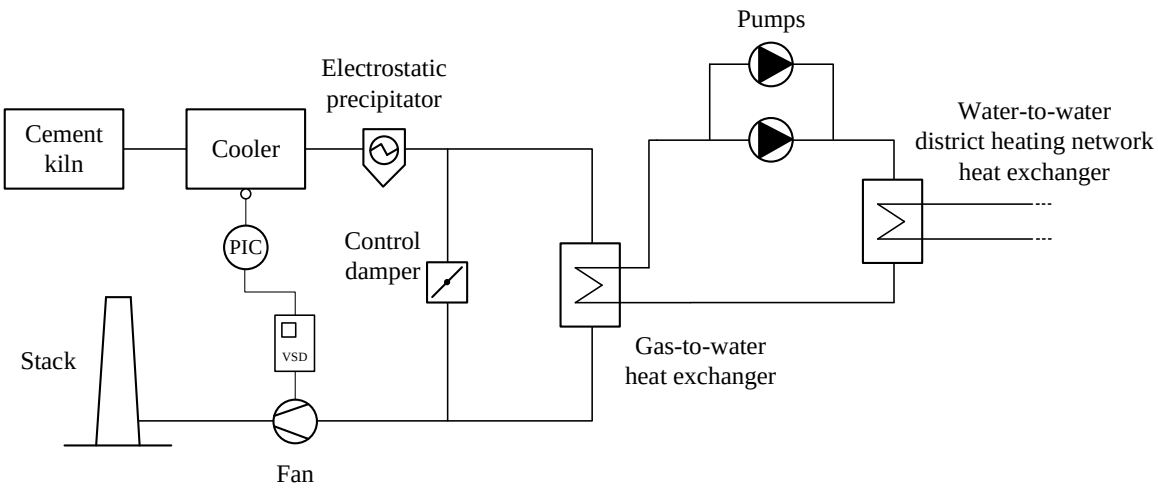

Figure 33. The heat exchanger used for flue gas energy recovery and its surrounding system. Here, PIC denotes the pressure measurement, which is used in the speed control of the fan.

The flue gas originates from a cement kiln. After the kiln, it passes through a cooler, where the cement product is cooled down. Then, it is taken through an electrostatic precipitator. Next, the gas flows to the heat exchanger. Some of the flow bypasses the heat exchanger through a branch before the heat exchanger. The fan, which induces the flow is located downstream of the heat exchanger and the bypass branch. The flue gas is released into the atmosphere through a stack after the fan.

The heat exchanger is cleaned whenever the kiln is run down for maintenance or breaks in production for other reasons. In 2015, the heat exchanger was cleaned in January, March and July.

4.3.1 Operating principles

The bypass flow rate is controlled with a control damper, whose position is adjusted according to the temperature in the secondary circuit of the heat exchanger. The bypass control damper is at all times kept at least 5 % open, to ensure that the negative pressure in the cooler remains sufficient. Too high a pressure could lead to equipment damage in the cooler due to an increased temperature. The fan is speed controlled according to a pressure reference acquired from the cooler. The control is set to maintain a certain negative pressure. Thus, as fouling occurs in the heat exchanger, the negative pressure in the cooler should decrease, which should lead to the control running the fan at a higher speed to maintain the negative pressure. In this case, however, there is a bypass parallel to the fouling heat exchanger. The position of the control damper of the bypass will also have an effect on the amount of negative pressure before the heat exchanger.

4.3.2 Acquired data

The measurements, from which data was acquired are:

- Temperatures for the gas and water before and after the heat exchanger (°C)
- Negative pressure before and after the heat exchanger (mbar)
- Water flow rate (l/s)
- Rotational speed of the fan (rpm)
- Shaft torque of the fan (% of nominal)
- Electric current to the fan’s motor (A)
- Gas bypass damper position (%)
- Kiln discharge funnel pressure (mbar)

The data includes measurements recorded once per half and hour and they range from January 2013 until October 2015. Some of the measurements were not recorded before October 2014. At that time, the data logging system of the factory was renewed. The capacity of logging was increased and since then more measurements have been included in the logged data.

4.3.3 Applicability of a VSD-based method

In this case, the existing temperature measurements allow estimating the *UA* value if an estimate for the flow either on the gas or water side of the heat exchanger can be generated. At the time of writing, neither the fan curves for the fan on the gas side nor the pump curves for the pump on the water side were available. Therefore, estimates for the flow rate could not be derived. As fouling occurs on the gas side of the heat exchanger, the change in the system resistance caused by the fouling could have possibly been detected through a method, which estimates the operating point of the fan.

However, to see the magnitude of the fouling, the *UA* value was calculated using the water flow measurement data. The *UA* value during the year 2015 is shown in Figure 34.

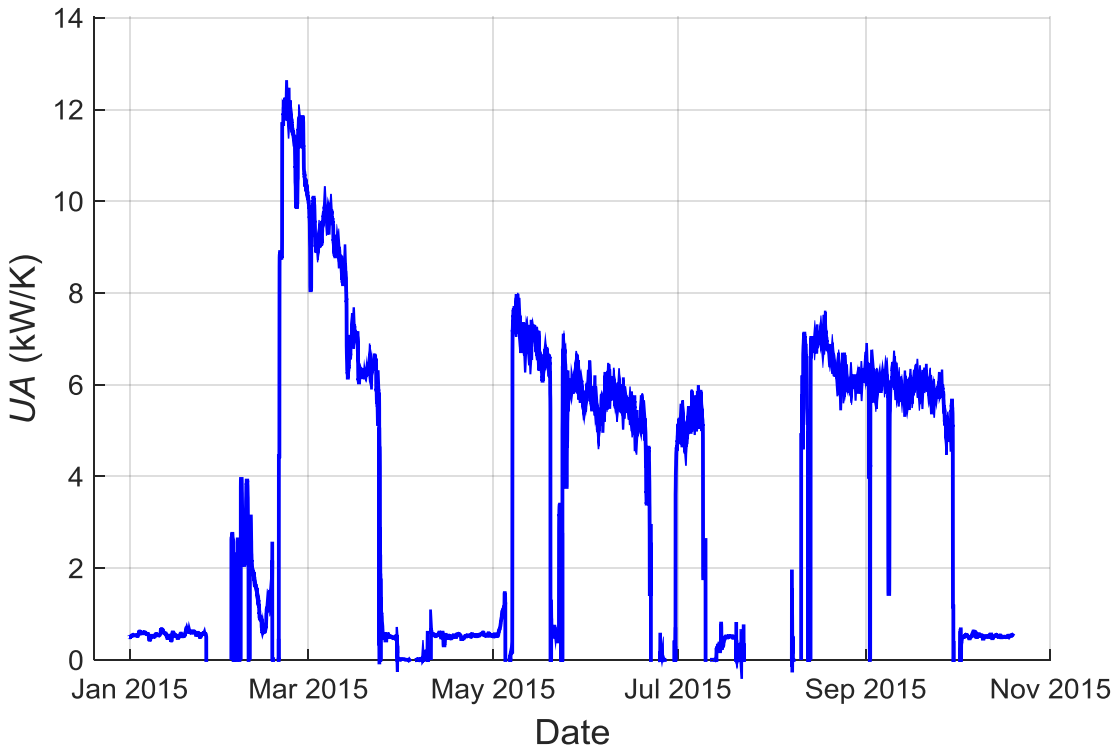

Figure 34. *UA* value of the heat exchanger during the year 2015.

A continuous decline can be seen in the *UA* value. With accurate specific curves for the water pump, the *UA* value could possibly be estimated. Estimating it with the mass flow of

the gas may prove more difficult, even with the addition of a proper fan curve, because some of the gas flows through the bypass instead of the heat exchanger. To see if the change in the *UA* value could be seen in as a change in the rotational speed of the fan, the calculated *UA* value was plotted in a graph together with the rotational speed, as seen in Figure 35. No clear correlation between the rotational speed of the fan and the *UA* value can be observed.

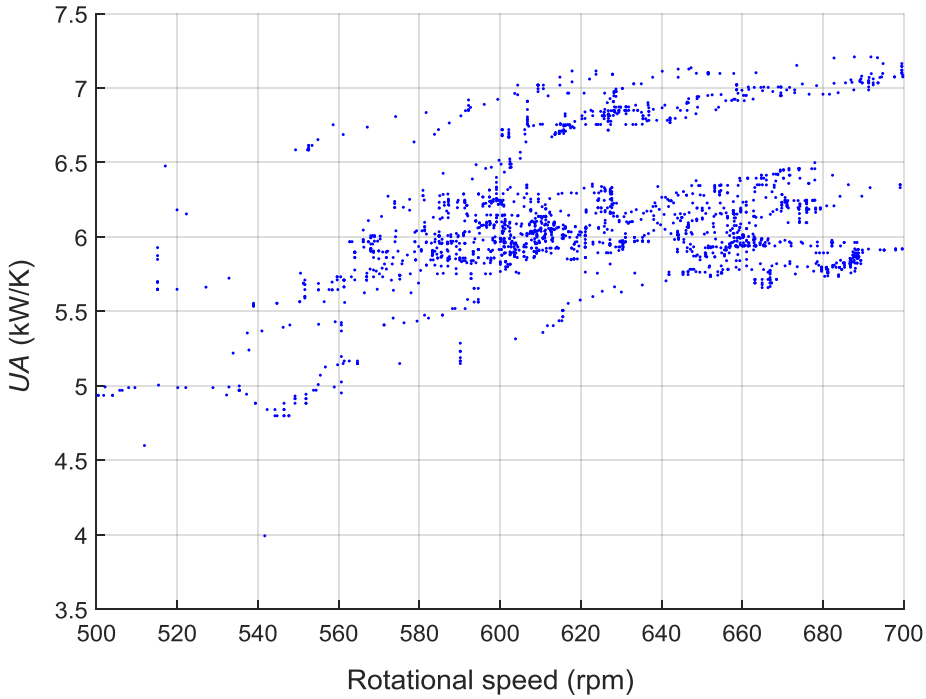

Figure 35. The *UA* value of the heat exchanger as a function of rotational speed.

5 CONCLUSIONS

Being able to monitor the fouling of a heat exchanger and plan optimal clean-up schedules can help the heat exchanger serve its purpose better. The three heat exchangers examined all experienced relatively aggressive fouling and in each of the cases, being able to tell when a clean-up is due will bring benefits tied to the purpose of the heat exchanger. In the case of the nuclear power plant's generator cooler, being able to monitor the fouling can decrease the risk of the generator overheating. Therefore, a method to detect fouling will here be beneficial from a plant availability and process safety point of view. On the other hand, in the case of the vent gas cooler at the pulp and paper mill, it is worthwhile to monitor fouling because of environmental reasons: when the clean-up schedule of the heat exchanger is optimized, the time during which odorous vent gases are released into the atmosphere is minimized. Finally, in the case of the flue gas heat recovery at a cement factory, the fouling reduces the energy recovered from hot flue gases, the concern there being energy efficiency.

As part of a heat transfer system, a variable-speed drive can, by providing an estimate for the device's power consumption, make it possible to estimate the operating point of a fluid handling device, such as a pump or a fan. When the power consumption and the characteristic *QP* and *QH* curves of the flow device are known, the *QP* and *QH* methods can be used to estimate the flow rate and the differential head or pressure of the flow device. Based on the results, when the input and output temperatures of a heat exchanger are measured in the system, the *UA* value of the heat exchanger can be estimated with the application of a flow estimation method, where the flow device's *QP* curve is used to derive an estimate for the flow rate.

In the case where the *UA* value was estimated (chapter 4.1.3), some uncertainty was present in the power consumption estimation of the flow device because only an electric current measurement was available. VSD's have been found to provide relatively accurate estimates for shaft power (Ahonen et al. 2011b). Therefore, the accuracy of the method with which the *UA* value was estimated would be better when a VSD would be used as the source of the shaft power estimate.

An attempt was also made to estimate the increase in a heat exchanger's pressure drop as fouling occurred. The estimated increase in the pressure drop did not coincide with the measured values. The estimated accumulated pressure drop at the end of the examined time period was less than half of the measured value. The reason of the failure to correctly quantify the change in the pressure drop through estimates remains unknown. The accuracy of the pressure measurement across the heat exchanger may have contributed to the difference of the estimated and measured values. However, the trend seen in the estimated differential head of the pump followed the progress of fouling qualitatively.

In the analyses of the cases where the examined flow device was a fan, a similar approach could have been taken to estimate the pressure drop increase caused by fouling. Without the characteristic fan curves, however, estimating the fans' operating points was not possible. In the case of the vent gas cooler system, it was suggested that when a fan is speed controlled to maintain negative pressure in the piping before a heat exchanger, reaching the maximum rotational speed indicates that the heat exchanger should be cleaned.

The applicability of the presented fouling monitoring methods depends on the measurements available in the heat exchanger system, the availability of the characteristic performance curves of a pump or a fan in the system and the control strategy of the pump or fan. The requirements of the methods are summarized in Table 1. In the table, the methods are referred to in the same order they were discussed in this chapter.

Table 1. Process requirements of the suggested VSD-based fouling monitoring methods.

	Required (yes/no)			
Heat exchanger fouling monitoring method	Temperature measurements around the heat exchanger	*QP*-curve available	*QH*-curve available	Rotational speed controlled by pressure
UA-method	yes	yes	no	no
Detection through increased heat exchanger pressure drop	no	yes	yes	no
Rotational speed monitoring method	no	no	no	yes

Out of the suggested fouling monitoring methods, the *UA*-method provides the most valuable information. The method yields information that can be utilized in calculations, which aim, for example, to define the energy balance of a heat exchanger or to determine optimum clean-up cycles.

The so-called rotational speed monitoring method cannot reach such accuracy. Once a level of fouling, which requires cleaning and the rotational speed associated with it are determined, the method can be used to serve as an indicator of when a clean-up should be conducted.

The third monitoring method, in which the pressure drop of the heat exchanger is the estimated variable, does not currently seem like a viable option for at least two reasons. Firstly, as the results in chapter 4.1.3 and Figure 29 show, it can only indicate that fouling is ongoing, but cannot determine the severity of the fouling or the actual performance of the heat exchanger. Secondly, in a system where the said method can be applied, merely adding temperature measurements would enable the use of the *UA*-method. Since temperature measurements often exist around heat exchangers (and if not, they are relatively easily and cheaply acquired and installed), the pressure-drop estimating method is not highly useful as such, as the more valuable *UA*-method can be utilized instead.

6 SUMMARY

A significant share of the world's electric energy consumption is accounted for by pumps, fans and compressors. Because of a discovered energy saving potential and the oncoming threat of depleting natural resources, the energy efficiency of systems where these devices operate is often investigated. Problems in fluid handling systems can reduce their energy efficiency and increase the life-cycle costs of these systems.

The aim of this thesis was to find significant fluid handling system problems, which could be detected using a variable-speed drive. First, a literature review was conducted to find problems, for which VSD-based detection methods could be developed. The found problems were presented in chapter 3. A classification for the problems was proposed, which could enable a developed detection method to apply to other problems of the same class.

To further assess the applicability of VSD-based methods, industrial case examples of fluid handling system problems were sought out. Inquiries were made to companies and the surveys revealed three cases of fouling exchangers, each with a distinctive task. The cases were presented in the subchapters of chapter 4. Being able to monitor the rate of fouling and quantify its effect can help process management to come up with optimal cleaning cycles for fouling heat exchangers. The possibility to use a VSD to monitor fouling in each of the cases was discussed. In the following, the cases and summaries of the conclusions made for each case are presented.

The first case to be analysed comprised a nuclear power plant generator cooling system, where seawater is used to cool water in a closed circuit, to which the generator is connected. The heat exchanger experiences aggressive fouling on the seawater side, which requires an annual clean-up of the heat exchanger during summer maintenance. The heat exchanger in this case has specific importance in terms of process safety and availability.

In the analysis of the first case, it was concluded that the flow estimate provided by a VSD could replace a flow sensor in determining the heat transfer capability of a heat exchanger,

the *UA* value. Therefore, if the input and output fluid temperatures of the heat exchanger are measured and a VSD is used to run the pump, with which a fluid is transferred through the heat exchanger, the *UA* value of the heat exchanger can be constantly estimated without additional metering.

In addition, the possibility to estimate the increased flow resistance caused by fouling was discussed for the first case. In the analysis, the quantity of the estimated pressure drop increase did not coincide with the amount of the measured increase. However, the estimated differential head of the pump, from which the estimate for the increase in the heat exchanger's pressure drop was derived, did follow a trend, which is typical to fouling. As fouling occurred and the hydraulic resistance of the system increased as a result, the pump's differential head increased.

The second case encompassed a heat exchanger at a pulp and paper mill, with which vent gases are dried to enable their use as combustion air. Here, fouling reduces the heat exchanger's ability to condense moisture from the gas and increased the flow resistance of the heat exchanger to a point where sufficient negative pressure upstream of it can no longer be maintained. In this case, fouling leads to an environmental issue, the release of odorous vent gases in the atmosphere.

For the second case, a method to monitor fouling was suggested. In a system, where a pressure-controlled fan is used to transfer gas through a heat exchanger, approaching the maximum rotational speed of the fan can serve as an indicator of a state of fouling where a clean-up of the heat exchanger is necessary.

The third and final case concerned a flue gas heat recovery heat exchanger used at a cement factory. The heat exchanger is used to recover heat from the flue gas originating from the cement kiln. The recovered heat is transferred into the factory's district heating network, where it replaces some of the heat bought from elsewhere. Thus, here the fouling has an effect on the energy efficiency of the process as a whole.

In this case, similarly to the other two cases, it was possible to detect the progress of fouling within the available measurement data from the system. However, due to the lack of information about the characteristics of the fan in the system, the effect of fouling on the performance of the fan could not yet be determined.

Targets for future research may be suggested as a result of this thesis. The analysis in the third case could be extended by acquiring the fan curve. Applying the fan curve would make it possible to estimate the operating point of the fan with the *QP* and *QH* methods and to see how it changes during fouling. Further research effort could also be put into studying the suitability of the *QH* method to monitor the change in a system's flow resistance. Being able to reliably estimate the fouling-induced increase in a heat exchanger's pressure drop without sensors would help system operators plan optimal cleaning schedules. Furthermore, more queries into industrial and municipal actors with significant pump, fan and compressor applications could be made. Additional communication may reveal more fluid handling system issues, for which VSD-based detection methods could be developed.

REFERENCES

Abdelrasoul, A, Doan, H, Lohi, A. 2013. *Fouling in Membrane Filtration and Remediation Methods*. In: Nakajima, H: Mass Transfer - Advances in Sustainable Energy and Environment Oriented Numerical Modeling. Intech. Chapter 8, pp. 195-218.

Aerovent. 2012. *Surge, Stall, and Instabilities in Fans* [e-document]. Retrieved October, 2015. From: http://www.aerovent.com/docs/fan-engineering-topics/surge-stall-and-instabilities-in-fans---fe-600.pdf?Status=Master

Afonso, P, Ferreira, J, Castro, J. 1998. *Sensor Fault Detection and Identification in a Pilot Plant Under Process Control*. Chemical Engineering Research and Design, vol. 76, issue 4, pp. 490-498.

Ahonen, T, Kortelainen, J, Ahola, J, Hammo, S, Soukka, R. 2011a. *Generic Unit Process Functions Set for Pumping Systems.* EEMODS'11 - 7th International Conference on Energy Efficiency in Motor Driven Systems.

Ahonen, T, Kortelainen, J, Tamminen, J, Ahola, J. 2012. *Centrifugal pump operation monitoring with motor phase current measurement.* International Journal of Electrical Power & Energy Systems, vol. 42, issue 1, pp. 188-195.

Ahonen, T, Tamminen, J, Ahola, J, Niemelä, M. 2011b. *Accuracy study of frequency converter estimates used in the sensorless diagnostics of induction-motor-driven systems*. 14th European Conference on Power Electronics and Applications (EPE).

Ahonen, T. 2011. *Monitoring of centrifugal pump operation by a frequency converter*. Thesis for the degree of Doctor of Science (Technology). Lappeenranta University of Technology, Department of Electrical Engineering, Institute of Energy Technology. Lappeenranta. 134 pages. ISBN: 978-952-265-075-7.

Alfa Brazing. *Plate Heat Exchanger Technology* [e-document]. Retrieved June, 2015. From: http://www.alfabrazing.com/pdf_down/12p.pdf

Almeida, A, Fonseca, P, Falkner, H, Bertoldi, P. 2003. *Market transformation of energy-efficient motor technologies in the EU.* Energy Policy, vol. 31, pp. 563-575.

Barringer, H.P. 2003. *A Life Cycle Cost Summary*. International Conference of Maintenance Societies (ICOMS®-2003). Retrieved September, 2015. From: http://www.barringer1.com/pdf/LifeCycleCostSummary.pdf

Barrio, R, Fernández, J, Blanco, E, Parrondo, J. 2011. *Estimation of radial load in centrifugal pumps using computational fluid dynamics*. European Journal of Mechanics - B/Fluids, vol. 30, issue 3, pp. 316-324.

Bell & Gossett. 2015. *ecocirc® e^3 Series - WHOLE HOUSE POTABLE WATER CIRCULATORS* [e-document]. Retrieved October, 2015. From: http://documentlibrary.xylemappliedwater.com/wp-content/blogs.dir/22/files/2012/07/A-133C.pdf

Bódalo-Santoyo, A, Gómez-Carrasco, J.L, Gómez-Gómez, E, Máximo-Martín, M.F, Hidalgo-Montesinos, A.M. 2004. *Spiral-wound membrane reverse osmosis and the treatment of industrial effluents*. Desalination, vol. 160, issue 2, pp. 151-158.

Budris, A. *Operating Pumps Close to Best Efficiency Flow Rates Extends Life* [e-document]. WaterWorld. Retrieved September, 2015. From: http://www.waterworld.com/articles/print/volume-23/issue-12/departments/pump-tips-techniques/operating-pumps-close-to-best-efficiency-flow-rates-extends-life.html

Carbon Trust. 2012. *Compressed air - Opportunities for businesses* [e-document]. Retrieved August, 2015. From: https://www.carbontrust.com/media/20267/ctv050_compressed_air.pdf

Cleen Oy. 2015. *Remarkable energy savings potential in pumping systems* [e-document]. Retrieved August, 2015. From: http://www.cleen.fi/en/news?type=n&item=96&sourceUrl

Emerson Process Management. 2005. *Control Valve Handbook*. Fourth edition. From: http://www.documentation.emersonprocess.com/groups/public/documents/book/cvh99.pdf

Emerson Process Management. 2015. F*isher GX 3-Way Control Valve and Actuator System* [e-document]. Retrieved June, 2015. From: http://www.documentation.emersonprocess.com/groups/public/documents/bulletins/d103305x012.pdf

European Commission. 2015. *Energy Efficiency Directive* [e-document]. Updated October 23, 2015. Retrieved April, 2015. From: https://ec.europa.eu/energy/en/topics/energy-efficiency/energy-efficiency-directive

Fläktwoods. 2014a. *BDEP Damper* [e-document]. Retrieved June, 2015. From: http://www.flaktwoods.com/products/air-management-/dampers/rectangular-1/bdep/

Fläktwoods. 2014b. *SPC Multi Leaf Damper* [e-document]. Retrieved June, 2015. From: http://www.flaktwoods.com/products/air-management-/dampers/circular-1/spc3/

Fleiter, T, Eichhammer, W. 2011. *Energy efficiency in electric motor systems: Technology, saving potentials and policy options for developing countries* [e-document]. United Nations Industrial Development Organization. Retrieved May, 2015. From: http://www.unido.org/fileadmin/user_media/Publications/Research_and_statistics/Branch_publications/Research_and_Policy/Files/Working_Papers/2011/WP112011%20Energy%20Efficiency%20in%20Electric%20Motor%20Systems.pdf

GNS Science. *Shell and tube heat exchanger* [e-document]. Retrieved June, 2015. From: http://www.gns.cri.nz/Home/Learning/Science-Topics/Earth-Energy/Extracting-the-Heat/Heat-Exchangers/Shell-and-tube-heat-exchanger

Gorla, S.R, Khan, A. 2003. *Turbomachinery - Design and Theory*. Marcel Dekker. ISBN: 0-8247-0980-2.

Government of India, Ministry of Power, Bureau of Energy Efficiency. *Guide books* [e-document]. Retrieved May, 2015. From: http://www.beeindia.in/energy_managers_auditors/guide_books.php

H&C Heat Transfer Solutions. *Heat Exchanger Types and Selection* [e-document]. Retrieved June, 2015. From: http://www.hcheattransfer.com/team.html

Heat Exchanger Fouling and Cleaning a. *Heat Exchanger Fouling and Cleaning Conference Series* [e-document]. From: http://www.heatexchanger-fouling.com

Heat Exchanger Fouling and Cleaning b. *Conferences by Year* [e-document]. From: http://heatexchanger-fouling.com/conferences.htm

Hydraulic Institute. 2001. *Pump Life Cycle Costs: A Guide to LCC Analysis for Pumping Systems - Executive Summary*. From: http://www1.eere.energy.gov/manufacturing/tech_assistance/pdfs/pumplcc_1001.pdf

Hydraulic Institute. 2004. *Variable Speed Pumping - A Guide to Successful Applications - Executive Summary*. From: https://www1.eere.energy.gov/manufacturing/tech_assistance/pdfs/variable_speed_pumping.pdf

Idel'chik, I.E, 1960. *Handbook of Hydraulic Resistance - Coefficients of Local Resistance and of Friction*. Jerusalem: Israel Program for Scientific Translations. 517 pages.

International Energy Agency. 2014. *Capturing the Multiple Benefits of Energy Efficiency - Executive Summary.* From: https://www.iea.org/Textbase/npsum/MultipleBenefits2014SUM.pdf

Kakaç, S, Liu, H. 2002. *Heat Exchangers: Selection, Rating and Thermal Design.* Second edition. CRC Press. ISBN: 0-8493-0902-6.

Karassik, I, Messina, J, Cooper, P, Heald, C.. 2008. *Pump Handbook.* Fourth edition. McGraw-Hill. ISBN: 978-0-07-146044-6

Kiijärvi, J. 2011. *Darcy Friction Factor Formulae in Turbulent Pipe Flow* [e-document]. Lunowa Fluid Mechanics Paper 110727. Retrieved October, 2015. From: http://www.kolumbus.fi/jukka.kiijarvi/clunowa/fluid_mechanics/pdf_articles/darcy_friction_factor.pdf

Koch Membrane Systems. *About Spiral Membranes* [e-document]. Retrieved June, 2015. From: http://www.kochmembrane.com/Learning-Center/Configurations/What-are-Spiral-Membranes.aspx

Kundnaney, N, Kushwaha, D. 2015. *A Critical Review on Heat Exchangers used in Oil Refinery*. Afro - Asian International Conference on Science, Engineering & Technology. Akshar Publication. ISBN: 9-780993-909238.

Lawrence Berkeley National Laboratory. 2003a. *Improving Fan System Performance - a sourcebook for industry.* United States Department of Energy, Air Movement and Control Association International. From: https://www1.eere.energy.gov/manufacturing/tech_assistance/pdfs/fan_sourcebook.pdf

Lawrence Berkeley National Laboratory. 2003b. *Improving Compressed Air System Performance - a sourcebook for industry*. U.S. Department of Energy, Compressed Air Challenge. From: https://www1.eere.energy.gov/manufacturing/tech_assistance/pdfs/compressed_air_sourcebook.pdf

Lawrence Berkeley National Laboratory. 2006. *Improving Pumping System Performance - A Sourcebook for Industry*. Second edition. United States Department of Energy, Hydraulic Institute. Retrieved June, 2015. From: https://www1.eere.energy.gov/manufacturing/tech_assistance/pdfs/pump.pdf

LUT. 2013a. Course: Voimalaitosopin perusteet 2013.

LUT. 2013b. Course: Lämmönsiirron perusteet 2013.

Melo, L.F, Bott, T.R, Bernardo, C.A. 1988. *Fouling Science and Technology*. Dordrecht, Netherlands: Kluwer Academic Publishers.

Moore, G. 2011. *Intelligent control of sewage pumps*. World Pumps, vol. 2011, issue 6, pp. 26-28, 30-33.

Mulder, M. 1996. *Basic Principles of Membrane Technology*. Second edition. Kluwer Academic Publishers. 564 pages. ISBN: 0-7923-4248-8.

Nassif, N. 2012. The impact of air filter pressure drop on the performance of typical air-conditioning systems. Building Simulation, vol. 5, pp. 345-350.

Nesbitt B. 2006. *Handbook of Pumps and Pumping*. Elsevier. 470 pages. ISBN: 978-1-85617-476-3.

Noh, K.-C, Hwang, J. 2010. *The effect of ventilation rate and filter performance on indoor particle concentration and fan power consumption in a residential housing unit*. Indoor Built Environment, vol. 19, pp. 444–452.

Northern Tool + Equipment. *Pacer Electric Drive Self-Priming Centrifugal Water Pump* [e-document]. Retrieved October, 2015. From: http://www.northerntool.com/shop/tools/product_31620_31620

OSTP. *Surfaces* [e-document]. Retrieved October, 2015. From: http://www.ostp.biz/Tubes-and-Pipes-Surfacec.aspx

Pump Systems Matter, Hydraulic Institute. 2008. *Optimizing Pumping Systems - A Guide For Improved Energy Efficiency, Reliability & Profitability*. Parsippany, New Jersey: Hydraulic Institute, Pump Systems Matter. 254 pages. ISBN: 978-1-880952-83-2.

Radgen, P, Blaustein, E. 2000. *Compressed Air Systems in the European Union - Energy, Emissions, Savings Potential and Policy Actions*. ISBN: 3-93228-16-0.

Radgen, P, Oberschmidt, J. 2008. EuP Lot 11*: Fans for ventilation in non residential buildings*. Fraunhofer Institute Systems and Innovation Research, Karlsruhe, Germany. From: http://www.energimyndigheten.se/globalassets/energieffektivisering/produkter-med-krav/industriflaktar/lot11_fans_finalreport.pdf

Shah, R.K. 1998. *Heat Exchangers*. In: Handbook of Heat Transfer. New York: McGraw-Hill. Chapter 17.

Siegel, J. 2002. *Particulate Fouling of HVAC Heat Exchangers*. Doctoral dissertation. University of California, Berkeley, Mechanical Engineering. From: http://www.camfil.us/FileArchive/_10_Camfil_Message_CamTab/Air%20Quality/Particle%20Fouling%20of%20Heat%20Exchangers.pdf

Sitholé, B. *Scale deposit problems in pulp and paper mills* [e-document]. Technical Association of the Pulp and Paper Industry of Southern Africa. Retrieved August, 2015. From: http://tappsa.co.za/archive/APPW2002/Title/Scale_deposit_problems/scale_deposit_problems.html

Somerscales, E.F.C, Knudsen, J.G. 1981. *Fouling of Heat Transfer Equipment.* Washington, United States: Hemisphere Pub. Corp.

Springer. 2010. *VDI Heat Atlas*. Second edition. Berlin, Germany: Springer-Verlag. 1584 pages. Multiple authors. ISBN: 978-3-540-77876-9. ISBN (online): 978-3-540-77877-6.

Stephens, B, Novoselac, A, Siegel, J.A. 2010a. *The effects of filtration on pressure drop and energy consumption in residential HVAC systems (RP-1299)*. HVAC&R Research, vol. 16, pp. 315–329.

Stephens, B, Siegel, J.A, Novoselac, A. 2010b. *Energy implications of filtration in residential and light-commercial buildings*. ASHRAE Transactions, vol. 116, pp. 346–357.

Sutherland, K. 2013. *50th Anniversary: Half a century of developments in filtration - Part 3*. Filtration + Separation, vol. 50, issue 3, pp. 23-27.

Tamminen, J, Ahonen, T, Ahola, J, Pöyhönen, S. 2015. *Variable Speed Drive-Based Fan Impeller Contamination Build-Up Detection: Industrial Case Study*. 17th Conference on Power Electronics and Applications, EPE'15-ECCE Europe.

Tamminen, J, Viholainen, J, Ahonen, T, Ahola, J, Hammo, S, Vakkilainen, E. 2013. *Comparison of model-based flow rate estimation methods in frequency-converter-driven pumps and fans*. Energy Efficiency, vol. 7, issue 3, pp. 493-505.

Tamminen, J. 2013. *Variable speed drive in fan system monitoring*. Thesis for the degree of Doctor of Science (Technology). Lappeenranta University of Technology, Department of Electrical Engineering, LUT School of Technology. 81 pages. ISBN: 978-952-265-532-5

The New York Blower Company. *Applications* [e-document]. Retrieved May, 2015. From: http://www.nyb.com/applications.asp

United States Department of Energy. 2000. *Energy Tips - Minimize Compressed Air Leaks* [e-document]. Retrieved July, 2015. From: http://www.energystar.gov/ia/business/industry/compressed_air3.pdf

United States Environmental Protection Agency. 2008. *ENERGY STAR® Building Upgrade Manual* [e-document]. Retrieved September, 2015. From: http://www.energystar.gov/ia/business/EPA_BUM_Full.pdf

Vakkilainen, E. 2015. Professor, Lappeenranta University of Technology. Lappeenranta. Interview, 26 June 2015.

Vulloju, S, Kumar, E, Kumar, M, Reddy, K. 2014. *Analysis of Performance of Ljungstrom Air Preheater Elements*. International Journal of Current Engineering and Technology, pp. 501-505.

Wagner, J. 2001. *Membrane Filtration Handbook - Practical Tips and Hints*. Second edition. From: http://www.massey.ac.nz/~ychisti/MHandbook.pdf

Waide, P, Brunner, C. 2011. *Energy-Efficiency Policy Opportunities for Electric Motor-Driven Systems*. International Energy Agency. Retrieved June, 2015. From: https://www.iea.org/publications/freepublications/publication/EE_for_ElectricSystems.pdf

Walker, G. 1982. *Industrial heat exchangers - a basic guide*. Hemisphere Publishing Corporation. 408 pages. ISBN: 0.89116-259-3.

Walker, I, Dickerhoff, D, Faulkner, D, Turner, W. 2013. *System Effects of High Efficiency Filters in Homes*. ASHRAE Annual Conference.

Ward, M, Siegel, J. 2005. *Modeling Filter Bypass: Impact on Filter Efficiency*. ASHRAE Transactions, vol. 111, issue 1.

Waring, M, Siegel, J. 2008. *Particle loading rates for HVAC filters, heat exchangers, and ducts*. Indoor Air, vol. 18, pp. 209-224.

Water & Wastewater Treatment. 2013. *Anti-ragging routine slashes maintenance costs* [e-document]. Retrieved September, 2015. From: http://wwtonline.co.uk/features/anti-ragging-routine-slashes-maintenance-costs#.VitysX4rKUn

Zaatari, M, Novoselac, A, Siegel, J. 2014. *The relationship between filter pressure drop, indoor air quality, and energy consumption in rooftop HVAC units*. Building and Environment, vol. 73, pp. 151-161.

Zakir, T, Tran, H, Papengelakis, V. *Formation of pirssonite in green liquor handling systems*. TAPPI JOURNAL, July 2013.

Zakir, T. *Evaluation and Control of Pirssonite Scale Formation in Green Liquor Systems of the Kraft Process* [e-document]. Toronto: Graduate Department of Chemical Engineering and Applied Chemistry, University of Toronto, 2011. Master's Thesis. From: https://tspace.library.utoronto.ca/bitstream/1807/33786/1/Zakir_Tasnuva_201111_MASc_Thesis.pdf

Zeng, H, Meng, J, Li, Z. 2012. *Numerical study of a power plant condenser tube arrangement*. Applied Thermal Engineering, vol. 40, pp. 294-303.

Zuercher, J. 2015. *Estimating the Energy Savings Potential in Compressed Air Systems - Using the Efficiency Quotient Standard* [e-document]. Quincy Compressor. Retrieved August, 2015. From: http://www.quincycompressor.com/images/pdf/eq%20whitepaper-001.pdf