Jussi Tamminen

VARIABLE SPEED DRIVE IN FAN SYSTEM MONITORING

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Abstract

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Fan systems are responsible for approximately 10% of the electricity consumption in industrial and municipal sectors, and it has been found that there is energy-saving potential in these systems. To this end, variable speed drives (VSDs) are used to enhance the efficiency of fan systems. Usually, fan system operation is optimized based on measurements of the system, but there are seldom readily installed meters in the system that can be used for the purpose. Thus, sensorless methods are needed for the optimization of fan system operation.

In this thesis, methods for the fan operating point estimation with a variable speed drive are studied and discussed. These methods can be used for the energy efficient control of the fan system without additional measurements. The operation of these methods is validated by laboratory measurements and data from an industrial fan system.

In addition to their energy consumption, condition monitoring of fan systems is a key issue as fans are an integral part of various production processes. Fan system condition monitoring is usually carried out with vibration measurements, which again increase the system complexity. However, variable speed drives can already be used for pumping system condition monitoring. Therefore, it would add to the usability of a variable-speed-driven fan system if the variable speed drive could be used as a condition monitoring device.

In this thesis, sensorless detection methods for three lifetime-reducing phenomena are suggested: these are detection of the fan contamination build-up, the correct rotational direction, and the fan surge. The methods use the variable speed drive monitoring and control options for the detection along with simple signal processing methods, such as power spectrum density estimates. The methods have been validated by laboratory measurements.

The key finding of this doctoral thesis is that a variable speed drive can be used on its own as a monitoring and control device for the fan system energy efficiency, and it can also be used in the detection of certain lifetime-reducing phenomena.

Keywords: Fan system, control, energy efficiency, estimation, variable speed drive, condition monitoring

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I owe my deepest gratitude to the supervisor of this work, Professor Jero Ahola for his guidance and support on my research path. I would like to thank Dr. Tero Ahonen for his advice, fruitful cooperation, and his seemingly never-ending patience. I would also like to thank Mr. Juha Viholainen for the assistance in attaining our common academic and research goals.

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My warmest thanks go to Mr. Jukka Tolvanen and Mr. Juha Kestilä, both from ABB Oy, for providing me with this interesting research topic and comments in the course of the research. Most of all, I am grateful for the freedom you have given in carrying out the research.

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Jussi Tamminen
December 2013
Lappeenranta, Finland
“Ihmiselle kaikkein tärkein maailmassa on dägä”

M. A. Numminen
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Publications
List of publications

This doctoral thesis is based on the following papers. The rights have been granted by the publishers to include the papers in the thesis.


Nomenclature

Latin alphabet

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
<th>Unit</th>
</tr>
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<tbody>
<tr>
<td>$F$</td>
<td>Minimized system curve function</td>
<td>Pa</td>
</tr>
<tr>
<td>$n$</td>
<td>Rotational speed</td>
<td>rpm</td>
</tr>
<tr>
<td>$P$</td>
<td>Power</td>
<td>W</td>
</tr>
<tr>
<td>$p$</td>
<td>Pressure</td>
<td>Pa</td>
</tr>
<tr>
<td>$Q$</td>
<td>Flow rate</td>
<td>m$^3$/s</td>
</tr>
<tr>
<td>$T$</td>
<td>Torque</td>
<td>Nm</td>
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Dimensionless numbers

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
</tr>
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<tbody>
<tr>
<td>$i$</td>
<td>Index</td>
</tr>
<tr>
<td>$k$</td>
<td>Index</td>
</tr>
<tr>
<td>$n$</td>
<td>Index</td>
</tr>
<tr>
<td>$N$</td>
<td>Number of samples</td>
</tr>
<tr>
<td>$S$</td>
<td>Surge indicator</td>
</tr>
</tbody>
</table>

Greek alphabet

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>$\delta$</td>
<td>Uncertainty of the flow rate estimate</td>
<td>m$^3$/s</td>
</tr>
<tr>
<td>$\eta$</td>
<td>Efficiency</td>
<td>%</td>
</tr>
<tr>
<td>$\kappa$</td>
<td>Kernan affinity exponent</td>
<td></td>
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</tbody>
</table>

Subscripts

<table>
<thead>
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<th>Symbol</th>
<th>Description</th>
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<tbody>
<tr>
<td>0</td>
<td>Initial, start</td>
</tr>
<tr>
<td>1</td>
<td>Final, end</td>
</tr>
<tr>
<td>100%</td>
<td>Value at 100% of the nominal rotational speed</td>
</tr>
<tr>
<td>AC</td>
<td>Alternating current, alternating component</td>
</tr>
<tr>
<td>Aerodynamic</td>
<td>Aerodynamic value</td>
</tr>
<tr>
<td>Bias</td>
<td>Bias</td>
</tr>
<tr>
<td>Corrected</td>
<td>Corrected value</td>
</tr>
<tr>
<td>DC</td>
<td>Direct current, direct component</td>
</tr>
<tr>
<td>Est</td>
<td>Estimate</td>
</tr>
<tr>
<td>F</td>
<td>Fan</td>
</tr>
<tr>
<td>Final</td>
<td>Final value</td>
</tr>
<tr>
<td>Limit</td>
<td>Limit value</td>
</tr>
<tr>
<td>Lower</td>
<td>Lower value</td>
</tr>
<tr>
<td>Meas</td>
<td>Measured value</td>
</tr>
<tr>
<td>pf</td>
<td>Low-frequency power fluctuation</td>
</tr>
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## Nomenclature

<table>
<thead>
<tr>
<th>Abbreviation</th>
<th>Description</th>
</tr>
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<tbody>
<tr>
<td>QP</td>
<td>QP method</td>
</tr>
<tr>
<td>QpF</td>
<td>QpF method</td>
</tr>
<tr>
<td>Ref</td>
<td>Reference</td>
</tr>
<tr>
<td>SO</td>
<td>Shut-off</td>
</tr>
<tr>
<td>Start</td>
<td>Start value</td>
</tr>
<tr>
<td>Static</td>
<td>Static</td>
</tr>
<tr>
<td>Sys</td>
<td>System</td>
</tr>
<tr>
<td>Total</td>
<td>Total</td>
</tr>
<tr>
<td>Uncertainty</td>
<td>Uncertainty of value</td>
</tr>
<tr>
<td>Upper</td>
<td>Upper value</td>
</tr>
</tbody>
</table>

## Abbreviations

<table>
<thead>
<tr>
<th>Abbreviation</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>CFD</td>
<td>Computational fluid dynamics</td>
</tr>
<tr>
<td>DC</td>
<td>Direct component</td>
</tr>
<tr>
<td>DFT</td>
<td>Discrete Fourier transform</td>
</tr>
<tr>
<td>HVAC</td>
<td>Heating, ventilation, and air conditioning</td>
</tr>
<tr>
<td>LCC</td>
<td>Life cycle costs</td>
</tr>
<tr>
<td>PID</td>
<td>Proportional integral derivate</td>
</tr>
<tr>
<td>PSD</td>
<td>Power spectrum density</td>
</tr>
<tr>
<td>SFP</td>
<td>Specific fan power</td>
</tr>
<tr>
<td>VSD</td>
<td>Variable speed drive</td>
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</table>
1 Introduction

In this doctoral thesis, the option to use an electric variable speed drive (VSD), such as a frequency converter, as a monitoring device for a fan system is discussed. The thesis focuses on the use of VSD estimates for the motor rotational speed and shaft torque as a source of monitoring data. The estimates are derived from the internal voltage and current measurements of the VSD, and consequently, no additional instrumentation is needed to produce the information. The information derived from the estimates comprises the fan pressure and flow rate output, the fan energy efficiency, and the detection of three distinct lifetime-reducing phenomena.

This thesis presents methods that can be used for the monitoring of the fan system. The feasibility of these methods is evaluated through measurements in a laboratory test setup. In the test setup, axial and centrifugal fans with forward-curved airfoil profile blades are tested. However, the presented methods should also be applicable to other types of axial and centrifugal fans. In this chapter, the background and motivation of the study are presented together with the objectives and research methods. Finally, the outline of the thesis is given.

1.1 Background of the study

Fan systems are widely used in industrial and municipal applications (Almeida, 2003). Fan systems are responsible for 11% and 9% of the total electricity consumption in the industrial and municipal sectors, respectively (Almeida, 2003; EU Save 2000). It can be seen that this share of electricity consumption is significant when considering a single end-use device.

Figure 1.1 illustrates the whole energy transfer chain of a fan system (Tolvanen, 2013; Ahola, 2012). The figure shows that only a fraction of the primary energy (such as coal, oil, or gas) required for the electricity production is actually transferred to the moved fluid while the majority of the primary energy is already wasted before the fan system. In addition, it can be deduced that a joule of energy saved at the fan corresponds to four joules of primary energy saved, since the energy has to pass through the whole energy conversion chain. This emphasizes the importance of energy saving at the end-use device, such as the fan systems. Moreover, the energy-related products (ErP) directive (EU, 2009) imposes a minimum limit on the overall efficiency of fan systems. In a modern fan system, the overall efficiency comprises the fan, motor, and variable speed drive efficiencies. This further emphasizes the importance of energy efficiency in fan systems.
Two examples of fan life cycle costs (LCC) are given in Figure 1.2 as presented in (Tamminen, 2010). It can be seen that the energy costs cover the majority of the LCC. In addition, the maintenance costs and production losses caused by a fan failure have an impact on the LCC of a fan system. Thus, if the fan energy consumption can be decreased and the fan failures reduced or predicted, the LCC can also be cut down.

Figure 1.2: Two examples of life cycle cost distribution in fan systems: (a) provides an example of the LCC in a noncritical axial fan system, and (b) presents the LCC in a critical centrifugal fan system.
1.1 Background of the study

Considering the energy costs, there is often significant energy-saving potential in the existing fan systems because of component oversizing and inefficient control of the fan output (Almeida, 2003; Binder, 2008; EU Save 2000; Tamminen, 2013a; US EERE 2003; Waide 2011). Replacing an outdated throttle control or damping by a rotational speed control with the installation of a variable speed drive is the most common way to enhance the performance of a fan system (Al-Basaam, 2013; Almeida 2005; Binder, 2008; Ferreira, 2011; Waide 2011). As can be seen from Figure 1.3, the fan power consumption decreases significantly when the fan is rotational speed controlled. The reason for this is that in the throttle control, additional losses are generated in the control valve. In the rotational speed control instead, the fan operation is adjusted to meet the required flow. In this case, the throttle-controlled fan consumes 3.8 kW and the rotational-speed-controlled fan 2.4 kW; thus, the rotational speed control requires 38% less power.

![Graph of fan pressure vs. flow rate](a)

![Graph of shaft power vs. flow rate](b)

Figure 1.3: Effect of throttle control and rotational speed control on the fan power consumption, where ‘Original operating point’ refers to the operating point without control, ‘Throttle’ denotes the operating point after the throttle control, and ‘Rotational speed’ indicates the operating point after the rotational speed control.

In addition, it can be seen in Figure 1.3 that the efficiency of the fan improves with the throttle control. However, the efficiency alone does not directly indicate the total energy efficiency of the fan system operation, as can be seen from the difference between the power consumptions of the different control methods. The main purpose of fan systems is usually the transfer of gases. Thus, a more appropriate energy efficiency indicator is the specific fan power (SFP), which indicates the energy used per a moved volume of gas. In Figure 1.3, the resulting SFP after the control is 0.33 Wh/m$^3$ and 0.51 Wh/m$^3$ with the rotational speed control and the throttle control, respectively. Thus, by using this indicator, the least-energy-consuming control method can be found.

The fan operation at different rotational speeds can be estimated with affinity laws.
\begin{align}
Q &= \left( \frac{n}{n_0} \right) Q_0 \quad (1.1) \\
p_F &= \left( \frac{n}{n_0} \right)^2 p_{F,0} \quad (1.2) \\
P &= \left( \frac{n}{n_0} \right)^3 P_0 \quad (1.3)
\end{align}

where the subscript "0" denotes the initial conditions, \( Q \) is the fan flow rate, \( p_F \) is the fan pressure, and \( P \) is the fan shaft power. It can be seen that the power consumption has a cubic relation to the rotational speed, which makes the rotational speed control method a very attractive choice. However, it should be noted that even though the fan curve can be shifted by applying the affinity laws, the actual power consumption of the fan is also dependent on the surrounding system. Especially, if there is a static pressure difference between the inlet and the fan system piping, the resulting power consumption deviates considerably from the affinity laws.

Rotational speed control is nowadays implemented with variable speed drives, also known as frequency converters (Waide, 2011). However, the most energy efficient operation of a fan system is not guaranteed only by the addition of the rotational speed control. This is because the efficiency target is not reached simply by reducing the rotational speed, but rather by using a rotational speed that optimizes the whole system energy efficiency (Bortoni, 2008; Tamminen, 2013b).

Traditionally, the most energy efficient operation of a fan system is found based on the specifications or initial data of the system (Bortoni, 2008), or by measurements of the fan flow rate or head, or moreover, the fan operating point (Steger, 2011). However, fan systems seldom provide these measurements, and the initial data may be inaccurate or the measurement equipment may fail. Thus, methods are needed that can produce the information about the fan operating point and the surrounding system without adding components to the system.

1.2 Motivation of the study

Modern variable speed drives are capable of monitoring the motor shaft rotational speed and torque (Nash, 1997). Modern VSDs apply internal voltage and current measurements, and an adaptive motor model to estimate the motor rotational speed and shaft torque (Ananth, 2012; Wolbank 2004). As the aerodynamic behavior of the fan changes the loading conditions of the motor, the VSD estimates could be used to produce information about the fan operating point. It has been noted in (Ahonen, 2011b; Ahonen, 2013) that the variable speed drive estimates are accurate enough to estimate the operating point of a pump, and can thus also be used for the operating point estimation of fans. The methods used for the operating point estimation usually have the
1.3 Objectives of the study

The main objectives of this study are to investigate the use of a variable speed drive in fan system operation monitoring and to produce tools for the optimization of the fan system LCC in the system operation phase. The research concentrates on the use of the rotational speed, shaft torque, and shaft power estimates of the VSD in the assessment of the fan operating conditions. The research focuses on centrifugal and axial fans with forward-curved blades with an airfoil profile. However, the same methods should also be applicable to other types of fan blade geometries.

A further objective of the study is to analyze the determination of the fan operating point by model-based estimation methods that use the VSD estimates as inputs and the
fan characteristic curve as a model. The limitations and error sources of these methods are discussed. In addition, application of these methods to energy efficient control is presented.

Detection of three specific lifetime-reducing phenomena is investigated. An option to use the variable speed drive as a source of information about the contamination build-up on a fan impeller is studied. Methods that do not require additional sensors for detecting the correct rotational direction of a fan and a method for detecting surge in fans are examined.

All of the presented methods share the same objectives: determination and reduction of fan system life cycle costs without additional sensors. The model-based determination of the fan operating point can reveal the possible oversizing of the fan system and be used as a source of control information. The sensorless detection of lifetime-reducing phenomena diminishes the need for maintenance, unexpected maintenance actions, and downtime of the fan.

The key research questions of the doctoral thesis are:

- Is it possible to use the shaft torque and rotational speed estimates provided by the variable speed drive in the determination of the fan output together with the fan characteristic curves and possible additional information? Can this information be used in the energy efficient control of the fan system?

- Is it possible to use the shaft torque and rotational speed estimates provided by the variable speed drive to detect lifetime-reducing phenomena, such as in the detection of the fan contamination build-up, reverse rotation, and fan surge? How reliable are these detection methods?

1.4 Research methods

In the course of the research, a literature study is first made on the subject under consideration. Then, simulations of the problem are made. However, most weight is given to the laboratory measurements. An axial fan system presented in Appendix A and a centrifugal fan system presented in Appendix B are used in the study as sources of experimental data. For example, the fan systems are operated in various operating points, and the performance of the model-based operating point estimation methods are tested as presented in Section 2.2. The estimates of the proposed methods are compared with the actual measurements of the system and with previously known model-based estimation methods. In addition, proof-of-concept studies are made from the data gathered from an industrial fan system as shown in Section 2.3.

Further, certain physical phenomena, such as a fan surge condition presented in Section 3.3, are introduced to the fan system, and the resulting characteristics of the condition are compared with the previous methods and the methods presented in this thesis. In
1.5 Scientific contributions

The scientific contributions of this doctoral dissertation are:

- A method for enhancing the accuracy of flow rate estimation in pump and fan systems. The method applies a known model-based method for the estimation of the fan operating point. The operating point estimates are used to form an estimate of the system curve. This system curve estimate is used when the rotational speed of the system is changed significantly compared with the pump or fan nominal rotational speed. The method is based on the rotational speed and shaft torque estimates of the variable speed drive and the pump or fan characteristic curves.

- A method that can be used to calculate the flow rate estimation uncertainties of two known model-based methods for the fan operating point estimation and to improve the flow rate estimation accuracy based on the estimated uncertainties. The method that has the least uncertain flow rate estimate is used as the preferred method to produce the operating point estimate. The presented method can also combine the two flow rate estimates to provide a more reliable and accurate flow rate estimate when the two estimation methods are equally uncertain. The method is based on the variable speed drive estimates of the rotational speed and the shaft torque, the measured pressure, and the characteristic curves of the pump or fan. The method improves the accuracy of the separate model-based estimation methods.

- A method to detect the increasing mass of a fan impeller by a variable speed drive without additional sensors. The method is based on the rotational speed and shaft torque estimates available of the motor and the option to start the motor with a desired torque reference. The increasing mass of a fan impeller is detected from the decrease in the impeller acceleration when a constant torque reference is used at the fan start-up.

- A method that can be used to detect the correct rotational direction of a centrifugal fan. The method is based on the capability to rotate the fan in both directions and compare the available rotational speed and shaft torque estimates. An analysis of the low-frequency content and the electrical frequency content of the rotational speed and torque estimates is performed. The correct rotational direction is determined when the studied features of one rotational direction are compared with the opposite direction.

- A method to detect surge in a variable-speed-driven fan. The method is based on the detection of the increased low-frequency shaft power fluctuation. This
fluctuation is detected from the shaft torque and rotational speed estimates available.

The author also has other publications on closely related topics. These publications are listed here, but are not appended to this thesis and are therefore not discussed in detail. These 16 publications are:


1.5 Scientific contributions


The author is also designated as a coinventor in the following patent applications concerning the subjects presented in the doctoral thesis or closely related to the topic:

**European Patent application 10153168.9** “Method in Connection with a Pump Driven with a Frequency Converter and a Frequency Converter.” Application filed 10 February 2010.


**European Patent application 11160232.2** “Method of detecting wear in a pump driven with a frequency converter.” Application filed 29 March 2011.

**European Patent application 11160573.9** “Stall detection in fans utilizing frequency converter.” Application filed 31 March 2011.

**European Patent application 11160574.7** “Method and arrangement for estimating flow rate of pump.” Application filed 31 March 2011.

**European Patent application 11179147.1** “Method and apparatus for determining change in mass of fan impeller.” Application filed 29 August 2011.

**Finnish Patent application 20116080** “Method and controller for operating a pump system.” Application filed 2 November 2011.

**European Patent application 11189925.8** “Method for detecting the correct rotational direction of a centrifugal apparatus, and a centrifugal apparatus assembly.” Application filed 21 November 2011.

**European Patent application 11195777.5** “Method and apparatus for optimizing energy efficiency of pumping system.” Application filed 27 December 2011.

**European Patent application 12151397.2** “Method for detecting the correct rotational direction of a centrifugal apparatus, and a centrifugal apparatus assembly.” Application filed 17 January 2012.

**European Patent application 12153017.4** “Method and apparatus for monitoring air filter condition.” Application filed 30 January 2012.

**European patent application 12166510.3** “Method for tuning a ventilation system.” Application filed 03 May 2012.

**European Patent application 12192713.1** “Method for approximating a static head of a fluid transfer system.” Application filed 15 November 2012.
1.6 Outline of the thesis

The doctoral thesis studies the opportunity to use a variable speed drive as a monitoring device for the fan system operation. Background and motivation of the work are first provided in this introductory chapter. Then, the use of model-based fan operating point estimation methods is presented. Seven model-based estimation methods are discussed, and their operation is validated through laboratory measurements. An industrial case is also presented in the discussion of the usability of the model-based methods for the fan operating point estimation. In addition, the use of these methods in the energy efficiency control is considered. Then, three separate methods to detect lifetime-reducing phenomena are presented. The application of these methods is validated by laboratory measurements. The conclusions, key findings, and suggestions for future work are presented in Chapter 4. The relations between the chapters and the publications are outlined in Figure 1.4.

The rest of the thesis consists of the following chapters:

**Chapter 2** introduces the operating point estimation of a fan system. First, the basic model-based methods are presented and discussed. These methods apply the fan characteristic curves as a model of the fan. The inputs to the model-based methods are usually the rotational speed of the fan and the shaft power or fan pressure. Seven model-based methods are compared with each other. The model-based methods are also examined with data from the laboratory test setup and from an industrial fan system. The use of a model-based method in the control of parallel pumps is discussed and evaluated by laboratory measurements.
Chapter 3 concentrates on the detection of lifetime-reducing phenomena in the fan systems. The detection of an impeller mass increase, reverse rotation, and surge is addressed more closely and validated by laboratory measurements.

Chapter 4 is the final chapter before the appended seven publications. It presents the conclusions and makes suggestions for future work.

In the following, the contents of the publications are summarized, and the contributions of the author and the coauthors to them are reported. The coauthors not listed below have participated in the project cooperation. In addition, they have contributed to the preparation of the publications by revision comments and suggestions. These publications comprise four articles published in international journals and three international conference papers. Publications I–V are related to the model-based estimation of the fan operating point and are given in the order in which they are presented in the thesis. Publications VI–VIII relate to the detection of lifetime-reducing phenomena in a fan system, and they are also in the same order as in which they are presented in the thesis.

Publication I addresses a known sensorless operating point estimation method of a variable-speed-driven fan system. The model-based method uses the variable speed drive estimates of the shaft rotational speed and torque and the fan characteristic curves to estimate the fan operating point. The paper discusses the error sources and limitations of the presented estimation method. A simple example of the effect of the fan characteristic curve tolerances and the variable speed drive estimates on the flow rate estimation is given. The presented sensorless operating point estimation is validated both with axial and centrifugal fan systems. The main contribution of this paper is to show the factors affecting the sensorless operating point estimation and determine its usability by laboratory measurements.

The laboratory measurements setup was designed by the author. The measurements were conducted and analyzed by the author and Dr. Ahonen. The article is written in cooperation with the coauthors.

Publication II introduces and compares model-based pump operating point estimation methods. Two different methods are compared, and the error sources and limitations are discussed. Laboratory measurements are conducted to compare the methods in a laboratory pumping system, and one of the model-based operating point estimation methods is used to analyze two industrial pumping systems. The main contributions of the paper are the analysis of the different model-based pump operating point estimation methods, validation of their operation by laboratory measurements, and the use of a model-based method in industrial pumping systems.

Dr. Ahonen and the author have conducted the measurements and analyzed the data. Dr. Ahonen is the main author of the paper, and the author of this thesis has participated in the writing and commenting of the paper with the other coauthors.
1.6 Outline of the thesis

Publication III presents and analyzes an improvement to the sensorless operating point estimation. The method uses the traditional sensorless operating point estimation near the nominal rotational speed of the fan. These points are used to form a system curve estimate. This system estimate curve is then used to estimate the fan operating point when the fan is operated at a rotational speed significantly deviating from the fan nominal rotational speed. The operation of this invention is validated by laboratory measurements of a pumping system. The main contribution of this paper is to present and discuss an improvement to the known model-based methods and to validate the operation of the presented method by laboratory measurements.

The publication is based on a European patent application 10153168.9. The author is one of the three inventors of the presented method. The measurements and analysis were made by the author of this thesis and Dr. Ahonen. Dr. Ahonen is the main author of the paper, but the author has participated in the writing and commenting of the paper together with the other coauthors.

Publication IV provides an improvement to the model-based methods for estimation fan or pump operating point. The method calculates the assumed uncertainty of two basic model-based estimation methods and combines the flow rate estimates to ensure more reliable flow rate estimation. The presented estimation method is compared with other model-based estimation methods by laboratory measurements. The method to enhance the accuracy of the known model-based and the supporting laboratory measurements are the main contributions of this paper.

A European Patent application 11160574.7 has been filed for the invention. The author is the main inventor of the method presented in the paper. The author has conducted the laboratory measurements with Dr. Ahonen. The analysis is made by the author, and the coauthors have commented on and written the paper together with the author.

Publication V proposes an energy efficient method to control parallel pumps. The method applies the model-based estimation of the pump operating point, and uses this information to ensure that the pumps are operated in an energy efficient way and in an operating region where the presence of any lifetime-reducing phenomenon is unlikely. Simple efficiency restrictions are used as a limitation on the pump operating point. In addition, a head balancing state is included in the method to facilitate the start of additional parallel pumps in the system. This is done by reducing the head of the running pump so that the added pump does not have to overcome as much head to produce flow. The main contribution of the article is to introduce new control logic of operating parallel pumps and show how this logic can be implemented applying model-based pump operating point estimation methods. In addition, the laboratory validation of the presented control logic contributes to the article.

The publication is based on a Finnish patent application 20116080. The author is one of the inventors of the presented parallel pump control. The author has implemented the pump control logic in the dSpace environment used in the laboratory measurements.
The measurements were conducted in cooperation with Mr. Viholainen. Mr. Viholainen is the main author of the paper, and the author has participated in the writing and commenting of the article together with the other co-writers.

**Publication VI** concentrates on a method to detect the increasing mass of a fan impeller without additional sensors. The increasing mass of the fan impeller can be a sign of contaminant building up on the impeller blades, which may cause imbalance in the impeller. Thus, the method can be used to prevent imbalance in the fan. The presented method uses a constant torque at the start-up of the fan, and a rotational speed profile at the fan start-up is used to detect the increasing mass. Laboratory measurements have been conducted to ensure the applicability of the presented method in different hydraulic conditions for the fan and in different thermal conditions for the motor. The main contributions of this paper are the method for detecting the increased mass of a fan impeller and the laboratory measurements to validate the method.

The author is the main inventor of the presented method. A European patent application 11179147.1 has been filed for the method. The author has made the measurements with the co-writers Mr. Tahvanainen and Mr. Potinkara. The analysis has been made by the author. The co-authors have participated in the writing and commenting of the paper.

**Publication VII** introduces a sensorless method that can be used to detect the correct rotational direction of a centrifugal device. The method compares the correct and incorrect rotational direction frequency spectra of the variable speed drive shaft torque and rotational speed estimates. The examined frequency components are the electrical supply frequency component and the low-frequency fluctuation between 0–4 Hz. A lower frequency component in both of the estimates and frequency bands indicates a correct rotational direction. An incorrect rotational direction significantly reduces the device efficiency and leads to a reduced service life. The application of the method to industrial systems is discussed. The method is based on the capability of the variable speed drive to estimate the shaft torque and the rotational speed in connection with controlling the motor to rotate in both directions. Laboratory measurements have been conducted to ensure the usability of the presented method in fan and pump systems. The main contributions of the paper are the method for detecting the correct rotational direction and the laboratory measurements to validate the method.

The author is the main inventor of the method. A European patent application 12151397.2 has been filed for the invention. The laboratory measurements were conducted by the author and Dr. Ahonen. The analysis was made by the author. The co-authors have participated in the writing and commenting of the paper.

**Publication VIII** relates to a European patent application 11160573.9, where the author is the main inventor. The paper presents a surge detection method that can be applied to a variable speed drive. Surge is an unwanted phenomenon in fan systems reducing the system efficiency and service life. The method detects the increase in the low-frequency
shaft power fluctuation from the rotational speed and torque estimates of a variable speed drive. An increase in the fluctuation of the shaft power indicates a surge in the fan. The surge detection method is compared with other surge detection methods with data gathered from laboratory measurements. The main contributions of the paper are the method for sensorless detection of surge and the comparative laboratory measurements with other known detection methods.

The author has conducted the measurements and made the analysis presented in the paper. The coauthors have participated in the commenting and writing of the paper.
2 Model-based fan operating point estimation

It is important to know the fan operating point, since it is a key parameter on which almost every control and efficiency estimation is based. The operating point can be used for the energy efficiency analysis and efficient control of the fan, as well as to the detection of such operation points in which there is, for example, a risk of a surge. Traditionally, the fan operating point is acquired through direct measurements of flow rate and pressure. However, not all fan systems are instrumented. In addition, if temporary measurements of flow rate and pressure are installed, they may not be permanently available to acquire measurements from the whole process cycle, and hence, some operating conditions during the process cycle may go unnoticed. Furthermore, after modifications to the ventilation process, the instrumentation may not be reinstalled to ensure the proper operation of the fan system. This leads to the need for easily implementable methods that can monitor the fan operating point continuously and without additional instrumentation.

Model-based operating point estimation methods provide a solution for easy operating point monitoring in fan systems. Typically, these methods use the \( Q_{pF} \) and \( QP \) characteristic curves as a model of the fan. Terminologically, the methods can be referred to as characteristic-based methods. However, the term ‘model-based’ is used to generally cover methods for the fan operating point estimation that do not directly use the fan characteristic curves as a model of the fan. Such a method is presented in (Bortoni, 2003) and discussed in Publication IV.

In this section, model-based fan operating point estimation methods are introduced. The limitations and error sources of these methods are discussed. The operation of certain model-based methods is examined by laboratory measurements and data from an industrial fan system. The application of model-based methods in an energy efficient control of a parallel pumping system is given as an example. From an LCC point of view, the fan operating point is needed to determine the current operation of the fan and assess if there is any energy savings potential in the fan system.

2.1 Model-based estimation of the fan operating point

Model-based fan operating point estimation methods provide a solution to constant monitoring of fan systems without additional measurement equipment. There are two basic operating point estimation methods: the \( Q_{pF} \) method (Liu, 2003; Hammo, 2005) and the \( QP \) method (Wang, 2005; Ahonen 2009).

The \( QP \) method uses the affinity-law-corrected fan \( QP \) curve as a model of the fan as mentioned in Publication I. The inputs to the model are the rotational speed and the shaft power, which can be estimated with the variable speed drive. The estimation procedure is as follows: first, the fan \( QP \) curve is shifted to the current rotational speed applying the affinity laws (1.1)–(1.3). The flow rate corresponding to the shaft power
estimate is found from the shifted $QP$ curve. A graphical representation of the $QP$ method is given in Figure 2.1.

Figure 2.1: Graphical illustration of the $QP$ method for the estimation of the fan operating point.

The $Qp_F$ method uses the actual pressure measurement and the rotational speed estimate as the inputs to the model. The $Qp_F$ curve is used as a model of the fan. The estimation procedure of the $Qp_F$ method is as follows: first, the $Qp_F$ curve is rotational-speed-corrected by applying the affinity laws (1.1)–(1.3). Second, the flow rate corresponding to the measured pressure is found from the corrected $Qp_F$ curve as shown in Figure 2.2.

Figure 2.2: Graphical representation of the $Qp_F$ method.
In addition, the fan ideal $Q_{p_F}$ and $Q_P$ curves can be modeled with the well-known Euler’s equations derived from the impeller blade velocity triangle (White, 2003). However, as these equations are for an ideal system, they do not take into account the losses in the whole fan assembly. Thus, the characteristic curves given by the manufacturer can be considered to provide more reliable information of the actual performance of the fan, because they are based on measurements of the prototype fan or calculations by computational fluid dynamics (CFD). Hence, the given characteristic curves are used as a model in the estimation methods.

There is one distinct limitation in the estimation methods: the shape of the characteristic curve as given in Publications I and II. If the applied pressure or shaft power input produces two results for the flow rate, then the methods cannot be used as such. Thus, it is preferred that the fan characteristic curves are monotonic. This is a problem in fan systems where the characteristic curves tend to be nonmonotonic, as can be seen in Figure 2.2. A simple yet efficient way to combat this problem is to use only the dominant monotonic part of the characteristic curves, for example the above 1 m$^3$/s region at 1450 rpm of the $Q_{p_F}$ characteristic curve in Figure 2.2. This leads to loss of information, and therefore, one has to be sure that the fan is operated in the selected monotonic region.

Error sources in the estimation methods are the accuracy of the model and the inputs. The accuracy of the fan characteristic curves is given by the ISO 13348:2007 standard (ISO 2007). The standard defines that the pressure and the flow rate must be within ±5% of the given characteristic curve, and the shaft power must be within +8% of the given curve. The lack of a negative limit for the shaft power basically means that there is no limit on how much the efficiency of the fan can exceed the given efficiency. In addition, the tolerances change when the fan is operated in a flow rate region where the fan efficiency has decreased below 90% of the nominal efficiency.

The inputs to the model-based methods have an impact on the accuracy of the flow rate estimates. The pressure measurement can be affected by piping elbows, tees, and other modifications to the flow pattern, and there may not be enough straight piping to ensure an even flow profile. This may affect the accuracy of the pressure measurement. In addition, the rotational speed, shaft torque, and shaft power estimates are not absolutely accurate. In (Ahonen, 2011b; Ahonen, 2013) it has been measured that the rotational speed estimate is within 0.2% of the actual rotational speed, but the shaft torque and power estimates can deviate even by 4% from the actual measured value. Thus, the accuracy of the model inputs can reduce the accuracy of the flow rate estimates. Together, the input inaccuracy and the fan characteristic curve produce a great uncertainty to the estimation of the flow rate as shown in Publication I.

The affinity laws cause error to the estimation methods, especially to the $Q_P$ method, as the change in the fan efficiency as a function of rotational speed is not considered in the affinity laws (Muszynski, 2010). In centrifugal fans, this traditionally results in a positive-signed error of the flow rate estimate, as there is a higher shaft power
requirement than the model expects. In addition, the properties of the medium, such as the density, can be different from what the characteristic curves are given for, or the properties of the medium can change during the operation of the fan. The change in the medium density can be compensated in the model-based estimation methods when the properties such as temperature and ambient pressure are known (Bleier, 1998). However, these measurements require additional instrumentation.

A number of procedures have been introduced that improve the model-based operating point estimation methods and combat the error sources presented above. The improvements are given below.

The level correction method uses a temporary reference measurement of the fan flow rate, the rotational speed, and the shaft power to fix one point of the actual fan QP curve. After the reference measurement, the difference between the model and the actual fan operation $P_{\text{Bias}}$ can be calculated by

$$ P_{\text{Bias}} = P(Q_{\text{Meas}}, n_{\text{Meas}}) - P_{\text{Meas}}, $$

(2.1)

where the subscript "Meas" denotes the values acquired by the reference measurement, and $P(Q_{\text{Meas}}, n_{\text{Meas}})$ is the power given by the QP characteristic curve with $Q_{\text{Meas}}$ and $n_{\text{Meas}}$. The $P_{\text{Bias}}$ can then be used to correct the shaft power input to the QP method. The corrected input can be calculated by

$$ P_{\text{Corrected}} = P_{\text{Est}} + P_{\text{Bias}} \left( \frac{n}{n_{\text{Meas}}} \right)^3. $$

(2.2)

The reference measurement and estimation method are illustrated graphically in Figure 2.3.
2.1 Model-based estimation of the fan operating point

The level correction method increases the accuracy of the model used in the estimation thereby also improving the accuracy of the flow rate estimation. Besides, it reduces the effect of the shaft power input error since the curve is calibrated to the applied shaft power estimate, and the repeatability of the shaft power is more certain than the absolute value of the estimate (Nash, 1997; Ahonen, 2011; Ahonen, 2013). However, the affinity laws are not corrected, and there is no solution to the nonmonotonic QP curve cases.

The **Kernan method** is the same as the QP method, but with a correction measurement (Kernan, 2011). The correction measurement is made by closing the pressure side valve, and rotating the fan at different rotational speeds. The shut-off power consumption at different rotational speeds is acquired with this additional measurement. These shut-off power consumption and rotational speed pairs are used to enhance the accuracy of the affinity laws. The correct exponent in (1.3) is calculated by

$$
\kappa = \frac{\ln\left(\frac{P_{SO,1}}{P_{SO,2}}\right)}{\ln\left(\frac{n_1}{n_2}\right)}
$$

(2.3)

where the subscript "SO" denotes the shut-off conditions, the subscript 1 the first measurement at the rotational speed $n_1$, and the subscript 2 the second measurement at the rotational speed $n_2$. Thus, the affinity law in (2.3) is rewritten as
In addition, the power level correction of the \( QP \) curve is calculated with the shut-off power consumption at the nominal rotational speed by

\[
P = P_a \left( \frac{n}{n_0} \right)^\kappa. \tag{2.4}
\]

where the subscript "SO,100%" denotes the measured shut-off power at the nominal rotational speed and "SO" the shut-off power given by the untreated model at the nominal rotational speed. \( P_{\text{Bias}} \) is used as in the level correction method to treat the input to the model as given in (2.2).

After the shut-off power measurements, the estimation procedure in the Kernan method is as follows: first, the \( QP \) characteristic curve is shifted to the rotational speed used by (1.1) and (2.4). Then, the corrected power is used to determine the corresponding flow rate from the shifted \( QP \) curve as in Figure 2.3 (b).

The Kernan method improves the accuracy of the applied model and the affinity laws. However, since the shut-off condition is not a normal operating condition for a fan, the curve accuracy in the normal operating point may be left uncorrected. In addition, the limitation of the nonmonotonic curve is left untreated.

The \( \frac{p_F}{P} \) method can reject the changes in the medium density. This is because of the applied model, which is the fan pressure divided by the shaft power. The \( \frac{p_F}{P} \) method takes the shaft power, the pressure measurement, and the rotational speed as the inputs and the \( \frac{p_F}{P} \) curve as the model within the method. First, the model is rotational-speed-corrected applying the affinity laws (1.1)–(1.3). Then, the flow rate corresponding to the measured \( \frac{p_F}{P} \) value is found from the corrected \( \frac{p_F}{P} \) curve. A graphical example of the estimation method is given in Figure 2.4. The resulting curve shows that the \( \frac{p_F}{P} \) model (the fan pressure divided by the shaft power) changes significantly less compared with the \( QP \) and \( Qp_F \) curves even with a significant 17% change in the rotational speed. Thus, it may be assumed the \( \frac{p_F}{P} \) method is less influenced by the error in the affinity laws than for example the \( QP \) method.
2.1 Model-based estimation of the fan operating point

The $p_F/P$ is the only model-based estimation method presented here that is not affected by the change in the medium density. However, the estimation method needs an additional measurement of the fan pressure and leaves the model or the affinity laws untreated.

The hybrid method is presented in Publication IV. In the hybrid method, the system curve is estimated by the $QP$ method. The system curve is estimated in an operating region where the $QP$ method has preconditions to provide accurate flow rate estimates. This region is determined by the rotational speed that should be within 20% of the nominal rotational speed, and by the $QP$ curve derivate where the flow rate uncertainty is lower than for instance 10%. This system curve is used in the flow rate estimation when there is a significant change in the rotational speed of the fan, for example over 20% of the nominal rotational speed.

The system curve parameters that best suit the estimated operating points can be found by minimizing the sum of least squares. The equation to be minimized is

$$ F = \sum_{i=1}^{m} (p_{\text{Est},i} - p_{\text{Stat}} - k Q_{\text{Est},i}^2)^2, $$

(2.6)

where $i$ is the index of an estimated operating point, $p_{\text{Stat}}$ and $k$ are the parameters of the system curve, and the subscript "est" denotes the estimated values. The estimated system curve for a pumping system is presented in Figure 2.5.

Figure 2.4: Graphical example of the estimation procedure of the $p_F/P$ method.
2 Model-based fan operating point estimation

Figure 2.5: Example of the identification of the system curve based on the QP method for a pumping system.

The improvement this method brings is the more accurate flow rate and pressure estimation at lower rotational speeds. The power consumption of a fan does not follow the affinity laws when the rotational speed of the fan has changed significantly (for example over 20%) from the nominal rotational speed. However, the affinity laws related to the fan pressure are not affected as much by a change in the rotational speed as the respective affinity law for the shaft power.

There is one distinct issue with the hybrid estimation method that has to be taken into account. If the system curve changes during the fan operation, it has to be identified again. The limit of the repetition of the system curve identification can be set to be automatic when the QP method estimates start to deviate significantly from the previously identified system curve (e.g. a flow rate estimation difference of 10%). In addition, the initial accuracy of the QP method is an issue with the hybrid estimation method. Thus, the method cannot accurately estimate the fan operating point, if the QP method produces erroneous results.

The combined QpF/QP method, presented in Publication IV, selects from the QpF and QP methods the one that is assumed to be more certain. The selection of the more accurate estimation method is made with the fan characteristic curves and the estimated uncertainty of the pressure and shaft power measurements. For example, the uncertainty of the QpF method can be calculated by

\[
\delta_{QpF}(Q) = \frac{\Delta Q}{\frac{dp_F}{dQ}} p_{F, \text{Uncertainty}},
\]

(2.7)

where \( p_{\text{Uncertainty}} \) is the pressure measurement uncertainty, and the subscript "Meas" denotes the measured values.
2.2 Laboratory evaluation of the introduced model-based method

When there is a significant difference between the uncertainties of the flow rate estimates of the two methods, for example one is two times larger than the other, then the more certain estimation method is used solely for the estimation purposes. When the difference in the flow rate estimation uncertainty is lower, the flow rate estimates can be combined. This can be accomplished, for example, by using the estimated uncertainties and weighting the estimates accordingly by

$$Q_{Est} = \frac{\delta_{QP} \cdot Q_{Est,QP} + \delta_{PF} \cdot Q_{Est,PF}}{\delta_{QP} + \delta_{PF}},$$  (2.8)

where $\delta$ is the uncertainty $Q$ is the flow rate, the subscript “est” denotes the estimated value, and the subscripts "QP" and "PF" denote the QP estimation method and the QP method, respectively.

The benefit of the combined QP/QP method is that it can reduce a grossly erroneous flow rate estimation by selecting the appropriate estimation method for the present operating conditions. In addition, the combined QP/QP method is feasible in the whole operating region of the fan. This is because usually one characteristic curve is monotonic in the region where the other is nonmonotonic. However, the combined QP/QP method does not treat the accuracy of the characteristic curves or the affinity laws.

2.2 Laboratory evaluation of the introduced model-based method

Experimental results of the introduced methods are presented for a centrifugal fan in the following. The laboratory system is run in an open loop so that an operating point is set manually and no feedback of the measured or estimated flow rate is used. The flow rate estimation methods are treated as observers of the fan flow rate and their accuracy is studied accordingly. Nine different operating points were examined as presented in Figure 2.6. It can be seen that the actual model of the fan does not correspond well to the given fan model, as the measured fan curve differs significantly from the manufacturer’s fan curve. However, this is a good case to demonstrate the benefits and drawbacks of the model-based fan operating point estimation methods. Figure 2.6 demonstrates that the ‘Test point’ measurements deviate slightly from the measured characteristic curve ‘Measured’. This is because these measurements were carried out on different occasions.
When the fan is operated at the 120% nominal flow rate, the introduced estimation methods yield flow rate estimates presented in Figure 2.7. The $Q_P$ method gives an accurate estimate of the flow rate at 900 rpm, because the error in the incorrect model at the nominal speed is corrected by the error in the affinity laws, as a result of which the rotational-speed-shifted model is accurate. However, the flow rate estimates at 1500 and 1800 rpm are erroneous because of the shape of the fan curve and the model inaccuracy. The Kernan method is also inaccurate because of this model inaccuracy. The hybrid method is inaccurate since the system curve estimated with the $Q_P$ method is erroneous. Furthermore, the level correction method corrects the model in the flow-producing area and is thus more accurate than the other methods that are based on the $Q_P$ curve. The $Q_{pF}$ method gives accurate estimates of the flow rate, while the error is less than 0.4 m$^3$/s throughout the three rotational speeds. The model for the $p_F/P$ method is accurate, and it provides as accurate estimates as the $Q_{pF}$ method. In addition, the combined $Q_{pF}/Q_P$ method is able to select the $Q_{pF}$ method for the estimation, which provides more accurate estimation results than the $Q_P$ method.
2.2 Laboratory evaluation of the introduced model-based method

Figure 2.7: Flow rate estimates of the presented estimation methods when the fan is operated with a system curve resulting in a 120% nominal flow rate at the nominal rotational speed.

In Figure 2.8, the estimation is carried out for a system curve where a 70% of the nominal flow rate is acquired with the fan nominal rotational speed. It can be seen that the estimates of the \( Q_p \) method are inaccurate; the error is 0.5–1 m\(^3\)/s. This is a result of the inaccuracy of the \( Q_p \) curve. The level correction method is more inaccurate than the \( QP \) method as the model is accurate in this flow rate region without the bias power. Thus, the bias power reduces the model accuracy and the estimation accuracy. The Kernan method gives more accurate estimation results at 900 rpm than the \( QP \) method as there is a shut-off power consumption difference in the model and the measurement. The hybrid method is able to compensate the error in the affinity laws by forming the system curve from the \( QP \) method estimates. As the \( QP \) method is accurate in the nominal rotational speed region, so is the hybrid method throughout the rotational speed range. The combined \( Q_p/QP \) method is able to select the \( QP \) method as the sole provider of estimation as it is also more accurate than the \( Q_p \) method. The \( p_v/P \) method is more inaccurate at 900 rpm than at other rotational speeds, which is due to both the initial model accuracy and the inaccuracy of the affinity laws.
When operating at nominal flow rate, the model-based fan operating point estimation methods work as given in Figure 2.9. Both the $QP$ method and the $QpE$ method yield equally accurate flow rate estimation results as the model for both of the estimation methods is erroneous. Again, the Kernan method and the hybrid method reduce the accuracy of the $QP$ method with 900 rpm because of the erroneous system curve estimate and the shut-off power correction at 900 rpm. The level correction method improves the model accuracy in this operating region and thus provides a more accurate flow rate estimation than the $QP$ method at 1500 and 1800 rpm, but yields a more inaccurate estimate at 900 rpm. The combined $QpE/QP$ method combines the $QpE$ and $QP$ method flow rate estimates into one with 1500 rpm and 1800 rpm. This leads to a more accurate flow rate estimation than with the individual estimation methods. At 900 rpm, the $QpE$ method is calculated to have a significantly lower uncertainty, and thus, at this rotational speed the $QpE$ method is used solely to produce the flow rate estimate. The $pE/P$ method provides a good estimation accuracy at all of the rotational speeds, while the error is less than 0.2 m$^3$/s.
Figure 2.9: Estimation results carried out with a system where approximately a nominal flow rate was acquired at the nominal rotational speed.

The estimation results presented in Figure 2.7–Figure 2.9 show that the basic $QP$ and $Qp_F$ methods have regions where they are extremely erroneous and regions where they are accurate. For example, the $Qp_F$ method is accurate at high flow rates and erroneous at low flow rates as can be seen in Figure 2.7 and Figure 2.8. The Kernan method and the hybrid method improve the accuracy of the basic $QP$ method, but still have a region where they are erroneous usually in the same flow rate region as the $QP$ method. The level correction method can improve the accuracy of the $QP$ method across all flow rates. However, the $p_F/P$ method and the combined $Qp_F/QP$ method that use both the power and the pressure to estimate the flow rate are the most accurate ones. In addition, they do not have regions where the estimates are as erroneous as with the basic $QP$ or $Qp_F$ methods.

2.3 Pilot case

The model-based estimation methods were tested in a fan system in a Finnish pulp factory. The data gathered from the system were the total pressure at the fan outlet, the volumetric flow rate, the gas temperature, and the rotational speed, shaft torque and shaft power estimates of the variable speed drive. The data were gathered from a six-month period with a five-minute time interval. The studied fan system consists of a
centrifugal fan, an induction motor, and a variable speed drive. The fan is located in a system where there is an inlet box before the centrifugal fan, and there is a 523 Pa pressure drop at 43.6 Nm\(^3/s\) at the inlet of the fan because of the inlet box and inlet ducts.

Table 2.1: Motor and fan nominal values in the pilot fan system.

<table>
<thead>
<tr>
<th>Fan data for 40 °C</th>
<th>Motor data</th>
</tr>
</thead>
<tbody>
<tr>
<td>Rotational speed</td>
<td>Rotational speed</td>
</tr>
<tr>
<td>1021 rpm</td>
<td>994 rpm</td>
</tr>
<tr>
<td>Flow rate</td>
<td>Shaft power</td>
</tr>
<tr>
<td>50.2 m(^3/s)</td>
<td>355 kW</td>
</tr>
<tr>
<td>Static pressure</td>
<td>Shaft torque</td>
</tr>
<tr>
<td>3675 Pa</td>
<td>3 410 Nm</td>
</tr>
<tr>
<td>Total pressure</td>
<td></td>
</tr>
<tr>
<td>4247 Pa</td>
<td></td>
</tr>
<tr>
<td>Shaft power</td>
<td></td>
</tr>
<tr>
<td>285.9 kW</td>
<td></td>
</tr>
</tbody>
</table>

The average, maximum, minimum, and standard deviations of the gathered data are presented in Table 2.2. It can be seen that the fan is operated rather steadily since the standard deviation of the rotational speed and the flow rate is 5% or less than the average value.

Table 2.2: Key figures of the data gathered from the pilot case.

<table>
<thead>
<tr>
<th>Variable</th>
<th>Average</th>
<th>Maximum</th>
<th>Minimum</th>
<th>Standard deviation</th>
</tr>
</thead>
<tbody>
<tr>
<td>Power (kW)</td>
<td>84.5</td>
<td>132.8</td>
<td>40.5</td>
<td>11.7</td>
</tr>
<tr>
<td>Torque (%)</td>
<td>36.0</td>
<td>49.0</td>
<td>22.3</td>
<td>3.3</td>
</tr>
<tr>
<td>Rotational speed (rpm)</td>
<td>654.3</td>
<td>744.6</td>
<td>516.0</td>
<td>33.4</td>
</tr>
<tr>
<td>Flow rate (Nm(^3/s))</td>
<td>23.5</td>
<td>28.0</td>
<td>18.4</td>
<td>0.9</td>
</tr>
<tr>
<td>Total pressure (Pa)</td>
<td>2243.7</td>
<td>3027.2</td>
<td>1343.8</td>
<td>272.9</td>
</tr>
<tr>
<td>Temperature (°C)</td>
<td>27.0</td>
<td>36.9</td>
<td>17.4</td>
<td>2.6</td>
</tr>
</tbody>
</table>

The distribution of the operating points on the fan \(Q_p\) curve is depicted in Figure 2.10. This also indicates that the fan is operated in a narrow flow rate range.
2.3 Pilot case

Figure 2.10: Distribution of the operating points of the pilot case on the fan characteristic curves.

The data gathered from the pilot case were used to study the operation of the following methods:

- \( Q_p \) method
- \( p_v/P \) method
- Power level correction method

The hybrid estimation is left out of the study because of the narrow rotational speed region in which the fan is operated. In 90% of the cases the rotational speed is within 20% of the nominal rotational. Thus, there would be no satisfactory reason to estimate the system curve, since the \( QP \) method would be used in a vast majority of the estimates. In addition, the combined \( Qp_v/QP \) method is left out of the study since the operation of the fan is in such a narrow flow rate region, and in this region the uncertainty of the \( Qp_v \) method is less than half of the uncertainty of the \( QP \) method. Thus, the \( Qp_v \) method is used as the only source of flow rate estimation, resulting in that the combined \( Qp_v/QP \) method is as accurate as the \( Qp_v \) method. Finally, the Kernan method is not considered since there are not measurements of the no-flow situation.

The accuracy of the characteristic curves is illustrated in Figure 2.11. The operating points and the characteristic curves are both transformed to represent the average conditions of 654 rpm and 27 °C, and the pressure drop on the inlet of the fan is taken into account. It can be seen that the fan \( Qp_v \) characteristic curve corresponds to the measured values indicating that the \( Qp_v \) curve is accurate enough to be used in the model-based estimation methods. However, there is a 15 kW (20%) difference in the estimated shaft power and the characteristic curve shaft power. This indicates that the \( QP \) curve is not accurate enough to be used in the model-based estimation methods as such. The basic \( QP \) method is left out of the pilot case study, because it is apparent that it does not function properly.
Figure 2.11: Accuracy of the fan characteristic curves according to the measured values. The operating point and the characteristic curves are transformed to consider the suction pressure losses.

The accuracy of the $Q_{PF}$ curve method can be seen in Figure 2.12. The flow rate estimate follows the changes in the actual value of the flow rate. However, between 2600 and 2700 h there is a static difference in the estimate and the actual value, which indicates that some properties have not been taken into account in the estimation method. It can be seen in Figure 2.11 that the $Q_{PF}$ curve goes through the mean value, and thus the average error of the flow rate estimate is low. However, the actual pressure at 25 m$^3$/s and above is lower than the model predicts, which leads to too high flow rate estimates at these pressures.

Figure 2.12: Time domain representation of the measured flow rate and the flow rate estimate produced by the $Q_{PF}$ method.
2.3 Pilot case

The resulting curve when the fan $QP$ curve is level-corrected to correspond to the mean value of the data set is illustrated in Figure 2.13. It can be seen that even with the level correction, some of the power estimates are so high that they do not meet the corrected $QP$ curve. This results in a problem that in a mathematical sense, no flow rate estimate can be made. The analysis applies a simple method of limiting the estimate to the flow rate that corresponds to the maximum power. The maximum power at 654 rpm is approximately 87 kW corresponding to $35 \text{ m}^3/\text{s}$. This flow rate is then shifted to correspond to the rotational speed used. For example, if the power estimate were 90 kW with 654 rpm, the flow rate estimate would be limited to $35 \text{ m}^3/\text{s}$. If this were not done, the comparison of the three methods would not be reasonable.

There are some explanations for why the shifted flow rate vs. shaft power points do not form a curve when they are shifted to the same rotational speed with the affinity laws. The time constants of the flow rate measurement and the shaft power estimates may have a difference of several decades. The time constant of the flow rate measurements can be in seconds, but the variable speed drive shaft power estimate reacts instantly when the load current of the motor is affected (Nash, 1997). Moreover, the shaft power reflects the acceleration and deceleration of the motor and the fan. As the rotational speed is not affected as fast as the power because of the accelerating or decelerating torque, the power estimate does not follow the fan laws during transients, and thus, does not directly correspond to the hydraulic requirements of the fan. The scatter in the estimates was not filtered or compensated, because median filtering, for example, caused a significantly higher average error in the estimates. This is most likely due to the long 5-minute sampling interval. Thus, the filtering should be carried out at a higher frequency so that the unwanted effects of acceleration and deceleration can be avoided, as presented in (Leonow, 2013).

The effect of the above-mentioned nonidealities in the power estimate can be seen in Figure 2.13b, where the scatter of the flow rate estimates is much greater compared with the $Qp_F$ method and the $p_F/P$ method in Figure 2.12 and Figure 2.14b, respectively. In addition, the standard deviation of the flow rate estimation error of the power level correction method is much greater compared with the other two estimation methods presented in Table 2.3.
The results acquired and the accuracy of the model in the $p_F/P$ method can be seen in Figure 2.14. The model for the $p_F/P$ method has been corrected with the power bias as in the level correction method. Otherwise, the model would be completely erroneous and there would be no reason to analyze the estimation results. It can be seen in Figure 2.14 that the model corresponds well to the measured values. Thus, when applying the level correction in combination with the $p_F/P$ method, the estimation results are reliable and accurate. In addition, the model also corresponds to high and low flow rate ends of the operating point set, resulting in an accurate estimation even in these regions.

Figure 2.14: $p_F/P$ method model and the saved operating points transformed to correspond to a 654 rpm rotational speed.
The flow rate estimation error of the different methods is given in Table 2.3 as values of average, maximum, and minimum error and as the standard deviation of the error. It can be seen that the \( p_v/P \) method is the most accurate one and has a low standard deviation of the error. The second best one is the \( Q_p \) method where the estimation accuracy is still good. The level correction method has the lowest accuracy. However, the estimates of the level correction method are still accurate enough to give an indication of the fan operating point location for energy efficiency audition purposes.

<table>
<thead>
<tr>
<th>Error in the method</th>
<th>Average</th>
<th>Maximum</th>
<th>Minimum</th>
<th>Standard deviation</th>
</tr>
</thead>
<tbody>
<tr>
<td>( Q_p ) method</td>
<td>-0.13</td>
<td>2.65</td>
<td>-8.66</td>
<td>0.97</td>
</tr>
<tr>
<td>Level correction method</td>
<td>0.51</td>
<td>11.85</td>
<td>-4.98</td>
<td>2.33</td>
</tr>
<tr>
<td>( p_v/P ) method</td>
<td>0.02</td>
<td>2.07</td>
<td>-1.35</td>
<td>0.41</td>
</tr>
</tbody>
</table>

Table 2.4 gives the flow rate estimation results when taking into account the effect of temperature in the density of air and using this information to treat the power and pressure measurements. The average error of the \( Q_p \) method and the power level correction method decreases compared with the case of the untreated temperature. Moreover, the standard deviation of the error is decreased, which is expected. The \( p_v/P \) method remains unaffected by the correction, since the change in density is already eliminated from the estimation procedure. The reason why the average error and the standard deviation of the error with the \( Q_p \) method are higher compared with the \( p_v/P \) method may lie in the ambient pressure. Since the ambient pressure is not measured, it can have an effect on the estimates of the \( Q_p \) method. The average ambient pressure is 1013 hPa, and the normal range of the ambient pressure is 920–1080 hPa (FMI, 2013). Thus, there is approximately a difference of 15% in the densities of air in the upper and lower ranges of the normal ambient pressure.

Table 2.4: Error in the flow rate estimates of the different model-based estimation methods with temperature correction.

<table>
<thead>
<tr>
<th>Error in the method</th>
<th>Average</th>
<th>Maximum</th>
<th>Minimum</th>
<th>Standard deviation</th>
</tr>
</thead>
<tbody>
<tr>
<td>( Q_p ) method</td>
<td>-0.11</td>
<td>2.38</td>
<td>-7.46</td>
<td>0.78</td>
</tr>
<tr>
<td>Level correction method</td>
<td>0.39</td>
<td>11.92</td>
<td>-5.40</td>
<td>1.89</td>
</tr>
<tr>
<td>( p_v/P ) method</td>
<td>0.02</td>
<td>2.07</td>
<td>-1.35</td>
<td>0.41</td>
</tr>
</tbody>
</table>
2.4 Model-based estimation in parallel pump control

Model-based methods can be used as a source of information in the control of fans and pumps. As an example of this, a parallel pump control, as presented in Publication V, was implemented by the $Q_pF$ method. One of the main ideas behind the control algorithm was to prevent the pumps from entering an operating region where the lifetime of the pump would be reduced. This was simply determined by giving the pump efficiency limits within which to operate. The efficiency limits can be converted into rotational-speed-dependent flow rate limits represented by the higher flow rate limit $Q_{\text{right}}$ and the lower flow rate limit $Q_{\text{left}}$.

Figure 2.15 provides a graphical example of the parallel pump control. In the traditional control, the rotational speed of pump 1 is increased until it reaches its nominal value. At this point, the rotational speed of pump 2 is increased to produce additional flow rate. In the alternative control, pump 1 is operated until the $Q_{\text{right}}$ limit is reached at operating point A, as can be seen in Figure 2.15(a). At this point, if the rotational speed of pump 1 is increased, it will be operated in an unwanted region. This leads to the inclusion of the second pump, which is called balancing. As the second pump starts to produce flow rate, the flow rate of the first pump can be decreased without reducing the total flow rate. After the balancing, the pumps are operated approximately at the same head, and when an increase in the total flow rate is needed, the rotational speed of both of the pumps is increased.

In the traditional control, pump 1 is operated in an unwanted region, and pump 2 has to overcome more head to start producing flow than in the alternative control. As can be seen in Figure 2.15, the traditional speed control does not keep the pump operating in the preferred operating region between $Q_{\text{left}}$ and $Q_{\text{right}}$.

![Figure 2.15: Parallel pump control to keep the pump operating in an optimal operating range. The operation of pump 1 is given in (a) and the operation of pump 2 in (b).](image-url)
2.4 Model-based estimation in parallel pump control

The control algorithm was tested in a laboratory setup with a flow rate ramp-up test where the flow rate requirement was steadily increased. The $Q_p F$ method was used to estimate the pump operating point. In the test, the algorithm keeps the pump operating in the correct operating region most of the time. However, it can be seen that during the balancing, pump 1 falls into the unwanted region. This is mostly because of the stepwise increase in the rotational speeds, so that the rotational speed of pump 1 is reduced, and partly because of the inaccuracy of the $Q_p F$ method.

![Figure 2.16: Pump operating points during the flow rate ramp-up measurement with the laboratory setup.](image)

It can be seen from Figure 2.17 that the presented control method provides energy savings compared with the traditional speed control of parallel pumps. This can be seen from the total power consumption of the pumps and in the pumping system specific energy consumption, which corresponds to SFP in fan systems. Both pumps consume less or an equal amount of power compared with the traditional control. This is accomplished by the balancing of the pump heads. The conducted measurements indicate that the model-based methods can be used in the energy efficient control of pumping systems.
Model-based fan operating point estimation

Figure 2.17: Power consumption of the pumps during the alternative control and the traditional control. The alternative control requires less power than the traditional control and is thus more efficient.

2.5 Summary

Model-based methods are required to remove the need to measure the actual operating point of the fan. In addition, the easily implementable operating point estimation methods can reveal inefficiently operating fan systems. These model-based methods were presented in Section 2.1, tested with laboratory measurements in Section 2.2, and with a pilot case in Section 2.3. A summary of the methods is given in Table 2.5.

Table 2.5: Summary of the models, inputs, benefits, and drawbacks of the presented model-based methods.

<table>
<thead>
<tr>
<th>Method</th>
<th>$Q_p F$ method</th>
<th>$QP$ method</th>
</tr>
</thead>
<tbody>
<tr>
<td>Model</td>
<td>$Q_p F$ curve</td>
<td>$QP$ curve</td>
</tr>
<tr>
<td>Input</td>
<td>Pressure measurement</td>
<td>Shaft power</td>
</tr>
<tr>
<td></td>
<td>Rotational speed</td>
<td>Rotational speed</td>
</tr>
<tr>
<td>Benefits</td>
<td>Actual process measurement</td>
<td>$Q_p F$ curve tolerance is smaller</td>
</tr>
<tr>
<td></td>
<td>$Q_p F$ curve tolerance is smaller</td>
<td>No additional instrumentation</td>
</tr>
<tr>
<td>Drawbacks</td>
<td>Reliable pressure measurement is an issue</td>
<td>Greater curve tolerances than with the $Q_p F$ curve</td>
</tr>
<tr>
<td></td>
<td>Additional sensors</td>
<td></td>
</tr>
</tbody>
</table>

0
50
100
150
Total flow rate (m³/h)

0
5
10
15
Total power (kW)

0
0.05
0.1
0.15
Specific energy consumption (kWh/m³)

0
50
100
150
Total flow rate (m³/h)
## 2.5 Summary

<table>
<thead>
<tr>
<th>Method</th>
<th>Level correction method</th>
<th>Kernan method</th>
</tr>
</thead>
<tbody>
<tr>
<td>Model</td>
<td>( QP ) curve</td>
<td>( QP ) curve</td>
</tr>
<tr>
<td>Input</td>
<td>Shaft power</td>
<td>Shaft power</td>
</tr>
<tr>
<td></td>
<td>Rotational speed</td>
<td>Rotational speed</td>
</tr>
<tr>
<td></td>
<td>Reference measurement of ( Q, P, ) and ( n )</td>
<td>Reference measurements of shut-off power with different rotational speeds</td>
</tr>
<tr>
<td>Benefits</td>
<td>Corrected ( QP ) curve at ( Q, P, n )</td>
<td>Known ( QP ) model at shut-off</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Estimation of the real affinity power equation</td>
</tr>
<tr>
<td>Drawbacks</td>
<td>Reference measurement</td>
<td>Model accuracy untreated in flow production region</td>
</tr>
<tr>
<td></td>
<td>Shaft power estimation incorporates acceleration and deceleration</td>
<td>Shaft power estimation incorporates acceleration and deceleration</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Method</th>
<th>( p_0/P ) method</th>
<th>Hybrid method</th>
</tr>
</thead>
<tbody>
<tr>
<td>Model</td>
<td>( Qp_0 ) curve</td>
<td>( Qp_0 ) curve</td>
</tr>
<tr>
<td></td>
<td>( QP ) curve</td>
<td>( QP ) curve</td>
</tr>
<tr>
<td>Input</td>
<td>Pressure measurement</td>
<td>Shaft power</td>
</tr>
<tr>
<td></td>
<td>Shaft power</td>
<td>Rotational speed</td>
</tr>
<tr>
<td></td>
<td>Rotational speed</td>
<td></td>
</tr>
<tr>
<td>Benefits</td>
<td>Built-in density correction</td>
<td>Better accuracy at rotational speed below 80% of nominal rotational speeds</td>
</tr>
<tr>
<td>Drawbacks</td>
<td>Additional instrumentation</td>
<td>Relies on the accuracy of the ( QP ) method</td>
</tr>
<tr>
<td></td>
<td>Shaft power estimation incorporates acceleration and deceleration</td>
<td>Shaft power estimation incorporates acceleration and deceleration</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Method</th>
<th>Combined ( Qp_0/QP ) method</th>
</tr>
</thead>
<tbody>
<tr>
<td>Model</td>
<td>( Qp_0 ) curve</td>
</tr>
<tr>
<td>Input</td>
<td>Pressure measurement</td>
</tr>
<tr>
<td></td>
<td>Shaft power</td>
</tr>
<tr>
<td>Benefits</td>
<td>Basic model-based methods used in their accurate regions</td>
</tr>
<tr>
<td>Drawbacks</td>
<td>Model accuracy untreated</td>
</tr>
</tbody>
</table>
There are several factors affecting the estimation accuracy of the presented model-based estimation methods. The error sources, their effect on the flow rate estimates, and an opportunity to reduce the error source are summarized in Table 2.6.

Table 2.6: Summary of the error sources of the model-based methods and their effect on the flow rate estimation.

<table>
<thead>
<tr>
<th>Error source</th>
<th>Effect on flow rate estimate</th>
<th>Opportunity to reduce</th>
</tr>
</thead>
<tbody>
<tr>
<td>Fan characteristic curves</td>
<td>High</td>
<td>High</td>
</tr>
<tr>
<td>Shaft power estimate</td>
<td>High</td>
<td>Medium</td>
</tr>
<tr>
<td>Pressure measurement</td>
<td>Medium</td>
<td>Low</td>
</tr>
<tr>
<td>Changes in gas properties</td>
<td>Medium</td>
<td>High</td>
</tr>
<tr>
<td>Affinity laws</td>
<td>Medium</td>
<td>Medium</td>
</tr>
<tr>
<td>Rotational speed estimate</td>
<td>Low</td>
<td>Low</td>
</tr>
</tbody>
</table>

The greatest source of error is the fan characteristic curves. However, the accuracy can be improved easily with a reference measurement. The variable speed drive estimate of the shaft power can also have a significant effect on the flow rate estimate. Even though there is room for improvement in the shaft power estimation accuracy, it may be more difficult to improve than the accuracy of the characteristic curves. The rotational speed estimate is within a few rpm of the actual rotational speed in the current state (Ahonen, 2011; Ahonen, 2013), so the resulting error in the flow rate estimate is negligible and there is not much room to improve the estimate.

Moreover, the accuracy of the pressure measurement can have an effect on the flow rate estimate. However, it may be demanding to improve a pressure measurement, as it would, in general, require an additional straight duct or a flow straightener. The gas properties are simple to measure and can be used to correct the possible moderate error in the flow rate estimate. Corrections to the affinity laws have been suggested in (Muszyński, 2010) and (Marchi, 2003), but an ambiguous correction method has not been found. Thus, it may be challenging to improve the accuracy of the affinity laws, since the correct correction method is unknown.

Improvements have been suggested that try to combat the above problems related to the model-based estimation methods. Moreover, the most accurate estimation methods were the \( p_F/P \) method and the combined \( Qp_F/QP \) method since they use both the \( Qp_F \) and \( QP \) curves as sources of the model used in the estimation and the measured fan pressure and the estimated shaft power as inputs.

However, as the laboratory measurement setup was run open loop, there were no adverse effects of the unreliable flow rate estimates present. As could be seen in the pilot case measurements in Section 2.3, the model-based methods can be extremely erroneous if the accuracy is not validated, as was the case with the \( QP \) method. If the flow rate estimate had been used in a closed-loop manner to control the process flow rate, the results could have been disastrous. Further, the desired operation of the process
and the reference flow rate would never have been acquired. Therefore, because of the possible problems with the estimation accuracy, the flow rate estimation methods cannot yet be recommended to be used as such in a closed-loop control. However, if the accuracy of the model-based methods is validated, then the use of the flow rate estimates in a closed-loop control may be reasonable as can be seen in Section 2.4.

The operating point information provided by a model-based method can be used in the energy efficient control of fan systems as shown for parallel-connected pumps in Section 2.4. In the example, parallel pumps were controlled so that operation in a lifetime-reducing operating region was avoided. In addition, a pump-head-balancing step was included in the starting of an additional pump, which leads to a significant reduction in the specific energy consumption compared with the traditional speed control of parallel pumps.
Detection of lifetime-reducing phenomena in a fan system

Similarly as any other mechanical devices, fan systems have lifetime-reducing abnormal operating conditions. These operating conditions are for example the aerodynamic instability or surge, foreign bodies such as dirt on the fan impeller, mechanical unbalance, and resonance (Drbal, 1995). Traditionally, the fan condition monitoring is carried out by using vibration measurements (Drbal, 1995) or possible monitoring of the fan pressure (Guzel’baev, 2006). However, these require additional measurement equipment.

Analysis of the load torque has also been used in condition monitoring for example in (Wiedenbrug, 2002). In addition, in pumping systems, the VSD shaft torque estimate has been used in the detection of cavitation (Ahonen, 2011a; Harihara, 2006). Thus, it can be reasoned that the VSD torque estimate should also be applicable to control monitoring of a fan system.

In this chapter, the use of a VSD as a source of condition monitoring data for specific applications is discussed. Three phenomena are studied: the detection of fan contamination build-up, the fan reverse rotation, and the fan surge. Methods to detect these phenomena are presented and discussed in this chapter.

3.1 Detection of the impeller mass increase

Fan systems are responsible for the transfer of many types of gases. Some gases may have contaminants that build up on the fan impeller. This can eventually lead to imbalance of the impeller when some of the contaminants detach from the blades. This imbalance can cause a failure of the fan. Traditionally, the contamination build-up is detected by visual inspection. However, this requires operations from the maintenance personnel, and therefore, the decision is subjective.

The imbalance of a fan impeller can be traced by vibration measurements or even by measurements of the motor current and voltage (Blödt, 2008; Concari, 2010; Kim, 2003; Kral 2004). However, this already indicates that there is an unwanted phenomenon present in the fan. Thus, if the root cause for the imbalance is found, then the imbalance can be avoided. The increased mass of the fan impeller is a clear indication of the contamination build-up. The increased mass of a fan impeller is directly correlated to the inertia of the impeller. This change in the fan inertia can be detected without additional measurements. However, traditional methods for detecting the inertia do not take into account the aerodynamic effects of the fan, which degrades the performance of these methods. As can be seen in Figure 3.1, the fan has first a constant acceleration when a constant torque is applied. The linear acceleration is due to the fact that the rotational inertia requires the dominant part of the accelerating torque. When the fan gains rotational speed, the torque required by the aerodynamics of the fan
becomes more dominant. This results in a reduction in the fan acceleration. Thus, the rotational inertia of the fan can be deduced from the linear part of the acceleration.

Figure 3.1: Start-up of a fan with a constant torque reference, where ‘Linear acceleration’ demonstrates the fan acceleration, if the aerodynamic effects are not taken into account.

The procedure of the detection is as follows. First, the fan is accelerated with a constant torque reference and the rotational speed estimate is saved from the start-up. Second, the rotational speed estimate is filtered to remove clearly erroneous estimates. Third, the angular acceleration is calculated from a certain rotational speed range. The impeller inertia and the impeller mass can be calculated from the angular acceleration.

The rotational speed region, from which the angular acceleration is calculated, is selected so that 95% of the required torque is due to the acceleration of the fan. A simple approximation of the torque requirement of the aerodynamic performance $T_{\text{Aerodynamic}}$ can be calculated based on the affinity laws and the final rotational speed $n_{\text{Final}}$ acquired with the start-up torque $T_{\text{Start}}$

$$T_{\text{Aerodynamic}} = \left( \frac{n}{n_{\text{Final}}} \right)^2 T_{\text{Start}},$$  \hspace{1cm} (3.1)

If the 95% limit for the torque ratio of the aerodynamic torque and the accelerating torque is selected, then the upper rotational speed limit $n_{\text{Limit,Upper}}$ at which the detection is made can be calculated by

$$n_{\text{Limit,Upper}} = \sqrt{(1 - 0.95)} n_{\text{Final}}.$$

(3.2)
3.1 Detection of the impeller mass increase

Figure 3.2 presents a flow chart of the detection method where multiple start-ups are used, and the difference in the angular acceleration of the start-ups is calculated and used as a means of detecting the increased mass of the impeller. In Figure 3.2 there is a lower limit in the rotational speed range of the detection. This is due to the challenges in the rotational speed estimation close to zero rotational speeds and physical phenomena such as stiction. Because of these, the non-zero rotational speed of 30 rpm is used as the starting rotational speed for the detection, as explained in Publication VI.

![Flow chart of the algorithm to detect the increasing mass of the fan impeller.](image)

The presented method was tested with a centrifugal fan system. The build-up of contaminants was emulated by fastening bolts on the fan impeller. The added weights were 50, 104, and 150 g, which correspond to 0.4, 0.8, and 1.2% changes in the impeller rotational inertia, respectively. The measurements were conducted with two different torque references and with three different system settings. The system settings were such that 120%, 100%, and 70% flow rates were acquired with the fan nominal rotational speed. Each torque reference, system setting, and added weight was tested three times to ensure repeatability. The results of the estimated increases in the impeller inertia are given in Table 3.1 and Table 3.2. In each row, the three repetitive measurements and different system settings are combined as one. Thus, nine different measurements are averaged and examined in each row. It can be seen from the standard deviation that regardless of the system setting and the repeated measurements, the detection method gives approximately the same angular acceleration for each weight. However, the estimated inertia is slightly different with different start-up torques. Thus, to ensure the most reliable results, the results with the same start-up torque should be compared with each other.
Table 3.1: Estimated increase in the fan impeller inertia, where each row contains information about three different system settings and three repetitive measurements. A torque reference of 30% of the nominal torque was used in the startup.

<table>
<thead>
<tr>
<th>Mass increase (g)</th>
<th>Mean of $\alpha$</th>
<th>Standard deviation of $\alpha$</th>
<th>Calculated $J_{Fan}$ (kg·m²)</th>
<th>Indicated change in the moment of inertia ($\Delta J_{Fan}/J_{Fan,0g}$)</th>
<th>Actual change in the moment of inertia</th>
</tr>
</thead>
<tbody>
<tr>
<td>0</td>
<td>115.4</td>
<td>0.15</td>
<td>1.225</td>
<td>0.00 %</td>
<td>0.00 %</td>
</tr>
<tr>
<td>50</td>
<td>115.0</td>
<td>0.17</td>
<td>1.231</td>
<td>0.41 %</td>
<td>0.40 %</td>
</tr>
<tr>
<td>104</td>
<td>114.5</td>
<td>0.10</td>
<td>1.236</td>
<td>0.81 %</td>
<td>0.80 %</td>
</tr>
<tr>
<td>151</td>
<td>114.1</td>
<td>0.16</td>
<td>1.240</td>
<td>1.17 %</td>
<td>1.20 %</td>
</tr>
</tbody>
</table>

Table 3.2: Estimated increase in the fan impeller inertia where a torque reference of 50% of the nominal torque is used. Each row contains information about three different systems settings and three repetitive measurements.

<table>
<thead>
<tr>
<th>Mass increase (g)</th>
<th>Mean of $\alpha$</th>
<th>Standard deviation of $\alpha$</th>
<th>Calculated $J_{Fan}$ (kg·m²)</th>
<th>Indicated change in the moment of inertia ($\Delta J_{Fan}/J_{Fan,0g}$)</th>
<th>Actual change in the moment of inertia</th>
</tr>
</thead>
<tbody>
<tr>
<td>0</td>
<td>190.5</td>
<td>0.16</td>
<td>1.238</td>
<td>0.00 %</td>
<td>0.00 %</td>
</tr>
<tr>
<td>50</td>
<td>189.4</td>
<td>0.29</td>
<td>1.245</td>
<td>0.58 %</td>
<td>0.40 %</td>
</tr>
<tr>
<td>104</td>
<td>189.0</td>
<td>0.27</td>
<td>1.248</td>
<td>0.81 %</td>
<td>0.80 %</td>
</tr>
<tr>
<td>151</td>
<td>188.2</td>
<td>0.32</td>
<td>1.253</td>
<td>1.20 %</td>
<td>1.20 %</td>
</tr>
</tbody>
</table>

3.2 Detection of reverse rotation of a centrifugal fan

The direction of flow in a centrifugal fan or pump is independent of the rotational direction of the impeller. However, the efficiency and output decrease dramatically when the fan is rotated in a reverse direction. This is due to the rotational-direction-dependent shape of the fan volute and impeller and their interaction as can be seen in Figure 3.3.
3.2 Detection of reverse rotation of a centrifugal fan

Traditionally, the correct rotational direction is detected by visual inspection. However, visual inspection requires skilled personnel and is not possible in all fan systems. Thus, an automated method should be found. The correct rotational direction can be ensured for example by a phase order detector connected to the motor terminals. In addition, two vibration sensors can be added to the fan. They measure the vibration in the x and y directions of the fan, and from the phase shift of the vibration, the rotational direction of the fan can be deduced. However, these methods require additional sensors.

In (Knudsen, 2010), the correct rotational direction of a centrifugal fan or pump is detected from steady-state flow rate, pressure, and shaft power values when the device is rotated in the correct and reverse directions. In the correct rotational direction, the produced flow rate and pressure are significantly higher than with the reverse rotational direction. For example, in the laboratory measurements, a fan produced 2.68 m$^3$/s and 1079 Pa when rotated in the correct rotational direction, but only 0.73 m$^3$/s and 77 Pa when the fan was rotated in the reverse direction. Thus, the difference is clear when considering the pressure and the flow rate. However, additional sensors are needed to measure these quantities.

If the additional sensors are to be left out, the variable speed shaft power estimate can be used in the comparison of the two rotational directions as mentioned in (Knudsen, 2010). In the previous example, the shaft power consumption was 4.68 and 4.70 kW in the correct and reverse directions, respectively. This may not be a sufficient difference to make a reliable decision of the correct rotational direction.
When a fan or a pump is rotated in a reverse direction, there are phenomena present that can be used as indicators of the reverse rotational direction. These phenomena are for example surge and cavitation in the fans and pumps, respectively. These phenomena can be detected from the shaft torque and rotational speed estimates of the variable speed drive.

The surge of a fan can be detected from the low-frequency fluctuation of the shaft power. The low-frequency fluctuation is present approximately in a 0–4 Hz band. The shaft power fluctuation is present in the rotational speed and the shaft torque, and can thus be detected from the variable speed drive estimates of these variables. In addition, the electrical supply frequency component of the shaft power increases when the fan is rotated in a reverse direction. When the fan is rotated in the correct and reverse directions and these two features are studied, the correct rotational direction can be deduced. The difference in the frequency spectra of correct and reverse rotational directions can be seen in Figure 3.4. In the frequency spectrum, the difference in the low-frequency band of 0–4 Hz and in the electrical supply frequency is clear.

![Figure 3.4: Variable speed drive shaft torque and rotational speed frequency spectra of a centrifugal fan rotated in forward and reverse directions. The electrical supply frequency is approximately 44 Hz.](image)

The method by which the correct rotational direction can be found is presented by a flow chart in Figure 3.5. The fan is rotated in both directions, and the rotational speed and shaft torque estimates are saved from the tests. The four indicators are calculated and compared with each other. The correct rotational direction is then deduced from the comparisons. It can be seen that the fan has to be rotated in both directions. Thus, the detection method is not applicable to systems where the reverse rotational direction is not allowed even for a short period of time.
3.2 Detection of reverse rotation of a centrifugal fan

The detection of the correct rotational direction was tested with a centrifugal fan and a centrifugal pump system. The fan system was rotated at five different rotational speeds against four different dynamic flow resistances. The rotational speeds were 60, 70, 80, 90, and 100% of the nominal rotational speed, and the dynamic flow resistances were such that 0, 70, 100, and 120% flow rates were acquired at the nominal rotational speed. A 10 s long sample with a 10 ms sampling rate of the variable speed drive shaft torque and rotational speed estimates was taken from each operating point.

In the pumping system, the rotational speeds were 40, 60, and 80% of the nominal rotational speed, corresponding approximately to 580, 870, and 1160 rpm. The system settings correspond to a situation where 0, 70, and 100% of the pump nominal flow rate are acquired at the pump nominal rotational speed. The frequency converter estimates were saved at a 5 ms interval for 30 s at each of the nine operating points.

In the experimental tests with the fan and pump system, three measurements were made in every operating point to ensure repeatability. A Welch spectrum estimate was calculated from the samples. In the Welch spectrum estimate, the sample was divided into 1024 point long segments with an overlap of 50%. In addition, the length of the discrete Fourier transform (DFT) was 1024 points.

A graphical representation of the fan system and pump system measurements can be seen in Figure 3.6. Clearly, in the majority of cases, the correct rotational direction is detected. However, in a small number of cases, the correct rotational direction has not been determined or a false detection is made, especially if the indicators are used separately.

![Flow chart for the detection of the correct rotational direction of a centrifugal fan or a pump.](image-url)
3 Detection of lifetime-reducing phenomena in a fan system

3.3 Detection of surge with a variable speed drive

Surge or stall in fans is an unwanted phenomenon present especially in axial fans. Fan surge is caused when there is insufficient flow through the fan and the flow detaches from the fan blades (US EERE, 2003). This causes a dramatic reduction in the fan total pressure (Forsthoffer, 2005; US EERE, 2003). When the fan is in a surge condition, the total pressure of the fan can fluctuate randomly with sudden drops or rises in the fan total pressure. Commonly this fluctuation in the fan total pressure is referred to as pressure surges. The fan surge condition leads to increased noise, vibration and mechanical stress in the fan system.

Fan surge is detected by a pressure measurement to reveal a drop in the fan total pressure. In addition, the pressure surges can be used as information about the fan surge. In (Mertens, 2011), a rotational-speed-dependent shaft power limit is used as a surge limit for the compressor. If the shaft power of the compressor is below the limit, the compressor is assumed to surge and shut down. However, there may be cases where the compressor does not surge even if these conditions are met, and the compressor is shut down unnecessarily.

The changing loading conditions of the fan during surge conditions caused by the pressure surges and “breathing” can be used as an indicator of the fan surge. These indicators are visible in the driving motor shaft and thereby in the variable speed drive shaft torque and rotational speed estimates. The indicator of surge can be extracted from the rotational speed and shaft torque estimates by first removing the DC component.
from the set of estimates so that only the AC component remains. This can be done for example for the shaft torque estimate with

\[ T_{AC}(n) = T(n) - \frac{1}{N} \sum_{k=1}^{N} T(k), \]  

(3.3)

Further, the desired frequency component of the estimate \( T_{AC} \) can be extracted by filtering or forming a power spectrum density (PSD) estimate. The feature extraction by filtering can be done with a low-pass filter with a cut-off frequency of 4 Hz. Then, a root-mean-square (RMS) value is calculated for the signal. This can be done in the same way for the rotational speed and shaft torque estimates. Feature extraction from a PSD can be done by first forming a PSD from the signal, for example with a Welch spectrum estimate. Then, the signal power in the 0–4 Hz frequency band is integrated from the PSD estimate.

Surge is present in both the rotational speed and the shaft torque. The ratio in which the fluctuation is present in the estimates is control scheme related. Thus, the indicator in these estimates can be combined to make one surge indicator to facilitate decision making. First, the rotational speed and torque fluctuation is made to have the same weighting because in absolute values 1 Nm indicates a greater fluctuation than 1 rpm. These equally weighted values are then summed geometrically. The indicator \( S \) can be calculated by

\[ S = \sqrt{\left( \frac{n_{pf}}{n_{pf,Ref}} \right)^2 + \left( \frac{T_{pf}}{T_{pf,Ref}} \right)^2}, \]  

(3.4)

where the subscript "pf" denotes the low-frequency shaft power and "Ref" denotes the reference value for the low-frequency shaft power fluctuation measured when the fan is known not to surge. The surge of a fan can be detected when the change in the indicator \( S \) is observed. It has been noticed in the laboratory measurements that a limit value of 2 for the variable \( S \) can be considered a limit for surge.

The surge detection method was studied by laboratory measurements and compared with other surge detection methods. First, the traditional indicator of a dramatic pressure drop and excessive heating of the transferred fluid was studied. Second, a reduced dispersion method presented in (Guzel’baev, 2006) is studied, where a reduced dispersion value is calculated from the pressure signal. This dispersion value reveals the pressure surges caused by the surge conditions. Fan surge is detected when the dispersion value is above a threshold value. Third, an increase of 43% of the rotational frequency component in the total pressure has been used as an indicator of the fan surge (Galindo, 2009; Smith, 1998).
The method was tested with an axial fan system. The measurements were conducted using rotational speeds of 2400, 2700, and 2900 rpm. With each rotational speed, the control valve was set from open to close with 12 discrete valve settings. In these operating points, the pressure signals were saved for a period of 10 seconds at a sampling frequency of 1 kHz. In addition, the frequency converter torque and rotational speed estimates were saved for 6.4 seconds at a sampling frequency of 500 Hz allowed by the drive. In the laboratory test, the measurement number 1 indicates the lowest measured flow rate and the number 12 the highest measured flow rate (10% and 120% of nominal flow rate, respectively) in the measurement set. It was deduced from the measured pressure, flow rate, and gas temperature that the fan is surging at measurements 1–2 and 10–12.

The laboratory test results for the three different surge detection methods at 2900 rpm are depicted in Figure 3.7. It can be seen that all of the detection methods detect the surge at measurements 1–2. However, the surging conditions at measurements 10–12 are not detected by all methods. The low-frequency method does not detect the surge at measurement 10, the 43% rotational frequency method does not detect surge at measurement 11, and the reduced dispersion methods are unable to detect the surge at measurement 12. In addition, the low-frequency fluctuation falsely indicates surge at measurements 7 and 9, and the 43% rotational frequency method indicates falsely surge at measurement 4.

Figure 3.7: Surge indicators when the fan was rotated at 2900 rpm. “LFF” indicates the low-frequency fluctuation, “Red” the reduced dispersion, “Rot” the 43% rotational frequency, and “Act” the actual surge conditions.
3.3 Detection of surge with a variable speed drive

The tests were repeated with a 2700 rpm rotational speed, and the results are shown in Figure 3.8. The reduced dispersion method is unable to detect the surge at measurement 1. In addition, the low-frequency fluctuation method is unable to detect the surge at measurement 10, and the 43% rotational frequency method does not detect surge at measurement 11. There are false indications of surge with the low-frequency fluctuation at measurements 3 and 4. Moreover, there is a false indication of surge with the reduced dispersion method at measurement 5.

Figure 3.8: Surge indicators when the fan was rotated at 2700 rpm. “LFF” indicates the low-frequency fluctuation, “Red” the reduced dispersion, “Rot” the 43% rotational frequency, and “Act” the actual surge conditions.
The measurements were repeated with 2400 rpm, and the results are shown in Figure 3.9. The reduced dispersion method does not indicate surge at measurements 1–2. In addition, the low-frequency fluctuation does not indicate surge at measurement 2, and the 43% rotational frequency method does not indicate surge at measurement 1. At 2700 and 2900 rpm, the surge indicators indicate surge more strongly at measurements 1–2. This may be explained by the fact that at low flow rates and low rotational speeds the power and pressure fluctuations caused by the surge are not as severe as with the higher rotational speeds and flow rates. At higher flow rates (measurements 10–12), the low-frequency fluctuation is unable to detect the surge at measurement 10, and the 43% rotational frequency method does not detect surge at measurement 11. Otherwise, the surge at high flow rates is detected.

Figure 3.9: Surge indicators when the fan was rotated at 2400 rpm. “LFF” indicates the low-frequency fluctuation, “Red” the reduced dispersion, “Rot” the 43% rotational frequency, and “Act” the actual surge conditions.

All of the three examined surge detection methods detect 11 of the 15 surging conditions. However, the low-frequency fluctuation method falsely indicates surge five times out of 21 when the reduced dispersion and the 43% rotational frequency method only two and one time(s) out of 21, respectively. The difference in the false detections may be explained partly by the difference in the sample length. The sample length in the low-frequency fluctuation method was 6.7 seconds while the sample length was 10 seconds in the other methods. In addition, the pressure is a more direct indication of fan surge, and thus, inherently more accurate. However, the pressure measurement requires an additional metering device. The low-frequency power can be considered an adequate source of information about the fan surge condition, since it does not require additional instrumentation and yields roughly the same detection accuracy as the other examined surge detection methods.

In addition, the test measurements show the challenges related to the accurate detection of the surge conditions of a fan; in each of the methods, this is indicated/demonstrated
3.4 Summary

Fan system condition monitoring is traditionally carried out by visual inspection, vibration measurements, or pressure measurements. These require additional instrumentation and increase the complexity of the fan systems. In this chapter, sensorless methods for the fan condition monitoring were presented.

In Section 3.1, a sensorless method to detect the fan contamination was introduced. It uses the capability of a variable speed drive to control the motor torque and estimate the rotational speed. When contaminants build up on the fan impeller, the rotational inertia will increase. This increase is detected when a suitable rotational speed range is used in the detection that eliminates the fan aerodynamic properties. Even an increase of 0.4% in the rotational inertia was detected with the presented method.

In Section 3.2, a sensorless method to detect the correct rotational direction of a fan or a pump was presented. In the method, the device is rotated in both directions. The characteristics (low-frequency fluctuation, supply frequency component) are calculated from the steady state of the rotational speed and shaft torque estimates, and the correct rotational direction can be determined. The correct rotational direction was detected in 95% of the cases. However, no false detections were made.

A method to detect surge in fans was presented in Section 3.3. The low-frequency fluctuation of shaft power caused by the fan surge was used as a feature for the detection of surge. The results indicate that the presented method gives approximately the same indications of surge as the three methods used for comparison.

These examples show that the variable speed drive can be used as a condition monitoring device for the fan system; at least in the specific cases of lifetime-reducing phenomena presented in this chapter.
4 Conclusion

Fan systems account for approximately 10% of the total electricity consumption in the industrial and municipal sectors. It has been found that there is energy-saving potential in the existing fan systems because of incorrect component sizing and inefficient control. Variable speed drives have been adopted in fan systems to improve the energy efficiency of the systems. However, the full potential of the variable speed drive is seldom seized. The variable speed drive can be used as a measuring device, a signal source, and a signal analyzer, and thus, there is potential in the variable speed drive in itself to improve the operation of the fan system. In this doctoral thesis, the opportunities to use a variable speed drive as a monitoring device for the fan system were discussed and analyzed. This chapter presents the key results of the thesis and suggestions for future work.

4.1 Key results of the thesis

A key result of the doctoral thesis is that the variable speed drive can be used in the estimation of the fan operating point. This can be done by applying model-based fan operating point estimation methods. The methods normally use the variable speed drive rotational speed estimate and the shaft power estimate or the measured fan pressure as an input to the model. Traditionally, the model comprises the fan characteristic curves of pressure as a function of flow rate and the shaft power as a function of flow rate.

Several previously known methods were presented and two novel methods were introduced to improve the performance of the known fan operating point estimation methods. In addition, it was shown that the estimates provided by the model-based estimation methods are accurate enough to be used in the energy efficient control of a fluid handling system.

The second key finding of the thesis is that a variable speed drive can be used as a condition monitoring device for the fan system. This is because the aerodynamic phenomena have an effect on the motor shaft torque and the rotational speed, and the variable speed drive can estimate the rotational speed and shaft torque of the motor.

Three separate fan lifetime-reducing phenomena were examined in this thesis; the fan impeller contamination build-up, the correct fan rotational direction, and the fan surge. The contamination build-up was detected from the increased acceleration time of the fan when started with a constant torque. The smallest increase in the impeller inertia tested and detected was 0.4% of the initial fan impeller inertia. The correct rotational direction is detected from the difference in the shaft torque frequency content between the correct and reverse rotational directions. Fan surge is detected when the low-frequency power fluctuation of the fan increases.

VSDs can be used to cut down the life cycle costs of fan systems. The majority of the fan system LCC consist of energy costs. The estimation of the fan system operating
points and possible energy efficient control based on the operating point save energy thereby reducing costs. Secondly, the fan maintenance and possible production losses caused by a fan failure can increase the LCC of a fan system. Consequently, using the VSD as a source of condition monitoring data can prevent an unexpected failure of the fan, and the VSD can be used as a source of information for preventive maintenance.

In addition, both the model-based fan operating point estimation methods and the methods for fan condition monitoring are software based. Hence, they do not add to the instrumentation of the fan system and are easily replicable to other fan systems.

4.2 Suggestions for future work

In the course of study, certain topics of further research have arisen.

The acceleration and deceleration of the motor have an effect on the accuracy of the model-based estimation methods, especially on the accuracy of the estimation methods based on the fan $Q_P$ curve. Thus, it would be interesting to find a way to eliminate this problem. This would enhance the usability of the estimation methods in systems where there is a constantly changing rotational speed reference as in most of the remotely controlled process-related fan systems.

An ambitious research topic would be the correction of the fan characteristic curves without additional sensors. This would dramatically improve the accuracy and usability of the model-based estimation methods in fan systems where it is not possible to use a reference measurement.

In this thesis, only a few specific fan system lifetime-reducing phenomena were discussed. There are still other important lifetime-reducing phenomena that are not detected with a VSD. These phenomena include for example the fan resonance. In addition, the surge detection method presented in Publication VIII requires that the fan has a constant rotational speed reference during the detection. It would be interesting to find methods that can be used to separate the effect of the control signal on the fan torque fluctuation from the fluctuation caused by the surge. This would improve the usability of the detection method in industrial settings.
References


(Al-Bassam, 2013) E. Al-Bassam, and R. Allasseri, “Measurable energy savings of installing variable frequency drives for cooling towers’ fans,
compared to dual speed motors,” *Energy and Buildings*, Vol. 64, December 2013, pp. 261–266.


References


(Vacon, 2013) Vacon, *Vacon 100 HVAC ac drives application manual*, 120 p.


Appendix A: Axial fan measurement setup

The axial fan measurement setup comprises the axial fan and the induction motor presented in Table A.1 and an ABB ACS 850 frequency converter with a nominal current of 16A.

Table A.1: Nominal values of the axial fan and the induction motor used in the measurement setup.

<table>
<thead>
<tr>
<th></th>
<th>Axial fan</th>
<th>Motor</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>FläktWoods Axipal BZI VA 630 4P</td>
<td>ABB M3AA 132 SC-2</td>
</tr>
<tr>
<td>Rotational speed (rpm)</td>
<td>2900</td>
<td>2880</td>
</tr>
<tr>
<td>Power (kW)</td>
<td>4.54</td>
<td>11.0</td>
</tr>
<tr>
<td>Flow rate (m³/s)</td>
<td>2.4</td>
<td>21.0</td>
</tr>
<tr>
<td>Fan total pressure (Pa)</td>
<td>900</td>
<td></td>
</tr>
<tr>
<td>Impeller diameter (mm)</td>
<td>630</td>
<td></td>
</tr>
<tr>
<td>Rotational speed (rpm)</td>
<td>2880</td>
<td></td>
</tr>
<tr>
<td>Power (kW)</td>
<td>11.0</td>
<td></td>
</tr>
<tr>
<td>Current (A)</td>
<td>21.0</td>
<td></td>
</tr>
<tr>
<td>cosφ</td>
<td>0.91</td>
<td></td>
</tr>
</tbody>
</table>

The measurement setup has static pressure difference, total pressure, and flow rate measurements as presented in Table A.2.

Table A.2: Measurement equipment in the axial fan system.

<table>
<thead>
<tr>
<th>Measurement</th>
<th>Method</th>
<th>Sensor</th>
</tr>
</thead>
<tbody>
<tr>
<td>Static pressure difference</td>
<td>Pitot static tube on the suction and discharge sides of the fan</td>
<td>Rosemont 2051 C, 0-3000 Pa</td>
</tr>
<tr>
<td>Total pressure at fan discharge</td>
<td>Pitot tube on the discharge side of the fan</td>
<td>Wika SL-1, 0-4000 Pa</td>
</tr>
<tr>
<td>Flow rate</td>
<td>Thermal mass flow meter on the suction side of the fan</td>
<td>Elridge 9800 Series FAT, 0-6 m³/s</td>
</tr>
</tbody>
</table>

The axial fan laboratory test setup is presented in Figure A.1. Sensor data were acquired with a National Instruments data acquisition system NI E-NET 9205 capable of a 250 kHz sampling frequency of the voltage inputs.
Figure A.1: Axial fan laboratory test setup.
Appendix B: Centrifugal fan measurement setup

The centrifugal fan measurement setup comprises the centrifugal fan and the induction motor presented in Table B.1 and an ABB ACS 850 frequency converter with a nominal current of 16A.

Table B.1: Nominal values of the centrifugal fan and the induction motor used in the measurement setup.

<table>
<thead>
<tr>
<th>Centrifugal fan</th>
<th>Motor</th>
</tr>
</thead>
<tbody>
<tr>
<td>FläktWoods Centripal EU MD 630 LG 90</td>
<td>ABB M2AA 132 M-4</td>
</tr>
<tr>
<td>Rotational speed (rpm)</td>
<td>Power (kW)</td>
</tr>
<tr>
<td>1449</td>
<td>4.59</td>
</tr>
</tbody>
</table>

The centrifugal fan laboratory test setup is presented in Figure B.1.

The measurement equipment and the data acquisition device are the same as with the axial fan measurements setup presented in Appendix A.