Lappeenranta
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Juha Viholainen

ENERGY-EFFICIENT CONTROL STRATEGIES FOR VARIABLE SPEED DRIVEN PARALLEL PUMPING SYSTEMS BASED ON PUMP OPERATION POINT MONITORING WITH FREQUENCY CONVERTERS

Thesis for the degree of Doctor of Science (Technology) to be presented with due permission for public examination and criticism in the room 1383 at Lappeenranta University of Technology, Lappeenranta, Finland on the 14th of March, 2014, at noon.

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Abstract

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The pumping processes requiring wide range of flow are often equipped with parallel-connected centrifugal pumps. In parallel pumping systems, the use of variable speed control allows that the required output for the process can be delivered with a varying number of operated pump units and selected rotational speed references. However, the optimization of the parallel-connected rotational speed controlled pump units often requires adaptive modelling of both parallel pump characteristics and the surrounding system in varying operation conditions. The available information required for the system modelling in typical parallel pumping applications such as waste water treatment and various cooling and water delivery pumping tasks can be limited, and the lack of real-time operation point monitoring often sets limits for accurate energy efficiency optimization. Hence, alternatives for easily implementable control strategies which can be adopted with minimum system data are necessary.

This doctoral thesis concentrates on the methods that allow the energy efficient use of variable speed controlled parallel pumps in system scenarios in which the parallel pump units consist of a centrifugal pump, an electric motor, and a frequency converter. Firstly, the suitable operation conditions for variable speed controlled parallel pumps are studied. Secondly, methods for determining the output of each parallel pump unit using characteristic curve-based operation point estimation with frequency converter are discussed. Thirdly, the implementation of the control strategy based on real-time pump operation point estimation and sub-optimization of each parallel pump unit is studied.

The findings of the thesis support the idea that the energy efficiency of the pumping can be increased without the installation of new, more efficient components in the systems by simply adopting suitable control strategies. An easily implementable and adaptive control strategy for variable speed controlled parallel pumping systems can be created by utilizing the pump operation point estimation available in modern frequency converters. Hence, additional real-time flow metering, start-up measurements, and detailed system model are unnecessary, and the pumping task can be fulfilled by determining a speed reference for each parallel-pump unit which suggests the energy efficient operation of the pumping system.

Keywords: variable speed drives, pumps, energy efficiency, fluid flow control
UDC 621.314.2:621.67:681.5.017:620.9

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Juha Viholainen
February 2014
Espoo, Finland

Contents

List of publications

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

I. Viholainen, J., Kortelainen, J., Ahonen, T., Aranto, N., and Vakkilainen, E. (2009). Energy efficiency in Variable Speed Drive (VSD) controlled parallel pumping. In: Bertoldi, P., Atanasiu, B. *Proceedings of the 6th International Conference eemods '09: Energy Efficiency in Motor Drives Systems*. 519–529. Luxembourg: Office for Official Publications of the European Communities.

II. Ahonen, T., Ahola, J., Viholainen, J., and Tolvanen, J. (2011). Energy-efficiency-based recommendable operating region of a VSD centrifugal pump. In: *Proceedings of the 7th International Conference eemods '11: Energy Efficiency in Motor Drives Systems*.

III. Viholainen, J., Tamminen, J., Ahonen, T., and Vakkilainen, E. (2011). Benefits of using multiple variable-speed-drives in parallel pumping systems. In: *Proceedings of the 7th International Conference eemods '11: Energy Efficiency in Motor Drives Systems*.

IV. Viholainen, J., Sihvonen, M., and Tolvanen, J. (2010). Flow control with variable speed drives. In: *Proceedings of ICIT 2010: International Conference on Industrial Technology*. 350–354.

V. Ahonen, T., Tamminen, J., Ahola, J., Viholainen, J., Aranto, N., and Kestilä, J. (2010). Estimation of pump operational state with model-based methods. *Energy Conversion and Management*. 51(6), 1319–1325.

VI. Tamminen, J., Viholainen, J., Ahonen, T., Ahola, J., Hammo, S., and Vakkilainen, E. (2013). Comparison of model-based flow rate estimation methods in frequency-converter driven pumps and fans. *Energy Efficiency*. Accepted for publication 2013.

VII. Viholainen, J., Tamminen, J., Ahonen, T., Ahola, J., Vakkilainen, E., and Soukka, R. (2012). Energy-efficient control strategy for variable speed-driven parallel pumping systems. *Energy Efficiency*. 6(3), 495–509.

Author's contribution

J. Viholainen is the principal author and investigator in Publications I, III–IV, and VII. Dr. T. Ahonen was the corresponding author in Publications II and V. J. Viholainen participated in the writing of Publication II. For Publication V, J. Viholainen participated in the background research and writing of the article. For Publication VI, J. Viholainen participated in the background research, methodology, and writing of the article. J. Viholainen was in the major role in the writing of Publications I, III–IV, and VII, with the help of the co-authors.

The author is also designated as a co-inventor in the following patent application considering the subjects presented in this doctoral thesis:

US Patent application 13/667,910 “Method and Controller for Operating a Pump System”. Application filed 2 November 2012, (Tamminen and Viholainen, 2012)

Nomenclature

Latin alphabet

A	area	m^2
d	diameter	m
E_s	specific energy consumption	Wh/m^3
f	frequency	Hz
g	acceleration due to gravity	m/s^2
H	head	m
H_0	shut-off head	m
H_r	head loss caused by friction	m
h	characteristic life	
k	coefficient	
l	length	m
T	torque	Nm
m	mass	kg
n	rotational speed	rpm
P	power	W
p	pressure	Pa
Q	flow rate	m^3/s
r	radius	m
t	time	s
U	estimation uncertainty	
V	volume	m^3
v	velocity magnitude	m/s

Greek alphabet

ζ	local loss-coefficient for piping components	
η	efficiency	
λ	loss-coefficient for certain pipe roughness	
ρ	density	kg/m^3

Subscripts

0	rated
a	outlet section
act	actual
base	selected base value
dyn	dynamic
e	inlet section
est	estimated
geo	geodetic difference
i	certain operation point, certain number of units

in	input
nom	nominal
max	maximum
meas	measured
min	minimum
motor	motor
pump	pump
QH	*QH* characteristic curve-based
QP	*QP* characteristic curve-based
ref	reference
rel	relative
st	static
sys	system
tot	total
VSD	variable speed drive

Abbreviations

BEP	best efficiency point
EU	European Union
GHG	greenhouse gas
HVAC	heating, ventilation and air conditioning
IE1	standard efficiency class (for electric motors)
IE2	high efficiency class (for electric motors)
IE3	premium efficiency class (for electric motors)
IE4	super-premium efficiency class (for electric motors)
LCC	life-cycle cost
NPSH	net positive suction head
$NPSH_a$	net positive suction head available
$NPSH_r$	net positive suction head required
MD	maximum demand
MTBF	mean time between failures
PLC	programmable logic controller
POA	preferred operation area
TOU	time-of-use
VSD	variable speed drive

1 Introduction

This thesis discusses the energy efficient control of parallel-connected pump units comprising a centrifugal pump, an electric motor, and a frequency converter. The main objective in this thesis is to present methods to increase the energy efficiency in variable speed controlled parallel pumping systems using control strategies which can be adopted without the instalment of additional metering or control devices and with less system information. The thesis focuses on radial flow end-suction centrifugal pumps as they are the most common pump type in various industrial and municipal parallel pumping applications. The studied methods consist of determining suitable operation conditions for variable speed driven parallel pumps, the use of performance curve-based operation point monitoring of the pumps, and the implementation of an energy efficient control strategy in targeted parallel pumping systems. In this chapter, the background and motivation of the study are presented. Also the studied research questions related to the objective of the thesis are shown. The outline of the thesis and the relation of the publications to the main research areas are also presented in this chapter.

1.1 Background of the study

The expected growth in energy use across the world due to the industrial development has increased the concentration of greenhouse gases (GHG), such as carbon dioxide, in the atmosphere (Abdelaziz et al., 2011). As the continued GHG emissions have shown to lead to a temperature increase in Earth's climate, there is a pressing necessity to reduce the emissions (IPCC, 2007). According to the Climate Action strategy by the European Commission, the EU countries target a 20% reduction of greenhouse gas emissions as well as a 20% increase in renewable energy in the total consumption (Forsström et al., 2011). In addition, the EU has set an objective to reduce its primary energy consumption by 20% compared with the projected 2020 energy consumption. Among various sectors contributing GHG emissions, the industrial sector is one of the most significant contributors of emissions. Thus, energy efficiency in the industrial sector is one of the key options to achieve the targeted GHG reductions (IPCC, 2007; Saidur, 2010; Abdelaziz et al., 2011; IEA, 2012). In 2013, the EU decided to accelerate the transition to more energy-efficient economy with the Energy Efficiency Directive, which will increase the emphasis of high-energy-efficiency performance also in the public sector investments (European Parliament, 2012).

Electric motors are responsible for a major part of the electrical energy use in industrial countries. In the EU, the electric motors use approximately 65–70% of the consumed electrical energy (Ferreira et al., 2011), which has made the electric motors a particularly attractive application for efficiency improvements. To reduce the energy consumption of electric motors, an international standard has been approved to promote the market transformation for higher efficiency motors (de Almeida et al., 2011). Efficiency levels for small and medium scale motors (0.75–375kW) have been defined for the standard efficiency (IE1), high efficiency (IE2), and premium efficiency (IE3) class in IEC 60034-30. In this standard, also super-premium efficiency class (IE4) is proposed, but only as an informative level. The given efficiency levels according to the motor-rated power at sizes 0.1.75–125 kW are shown in Figure 1.1

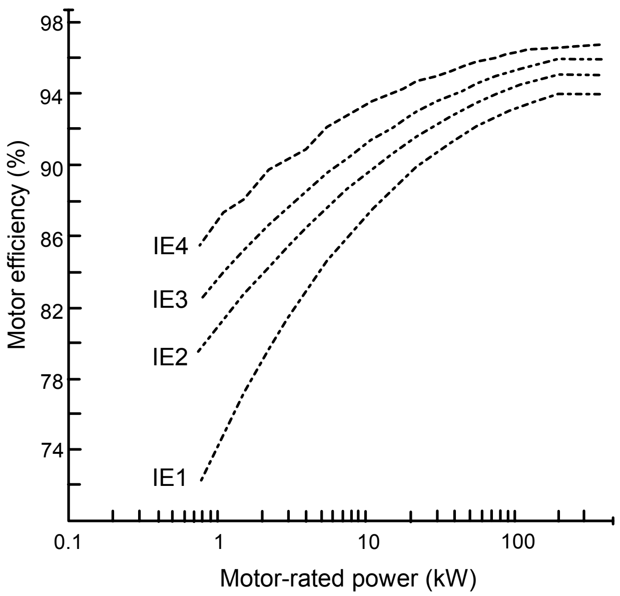

Figure 1.1. Efficiency levels of 50 Hz four-pole motors according to IEC 60034-30 standard (de Almeida et al., 2011).

In addition to conveyors, machine tools, lifts, and many other applications, electric motors are widely applied in fluid handling applications such as pumps, compressors and fans (de Almeida et al., 2005). In fact, the pumping systems can be responsible for 10–40% of the total energy end-use in industrial and service sectors (de Almeida et al., 2003; Kaya et al., 2008; Saidur, 2010), and the existing energy saving potential in pumping systems has been widely recognized (Carlson, 2000; Hovstadius et al., 2005; Ferreira et al., 2011). The energy costs also dominate the life cycle costs (LCC) of pumping. According to several studies, the energy costs can be up to 50–85% of the total LCC of pumping operations, although the share of energy, investment, and maintenance costs may vary depending on the pumping task (Ahonen et al., 2007; Pemberton and Bachmann, 2010; Augustyn, 2012).

The main purpose of using pumps is to move fluid from one place to another. The required energy for moving the liquid to the desired level can be only a fraction of the required primary energy when considering the whole energy chain. Figure 1.2 illustrates a typical energy flow from primary energy to the moved liquid.

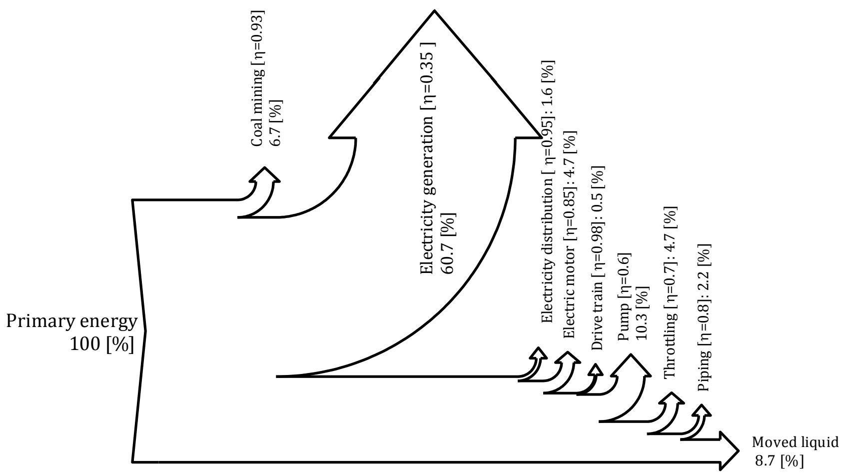

Figure 1.2. Typical energy flow – coal-fired energy is first converted to electricity, which is end-used by the pump to move liquid. The net gain of the energy efficiency improvement is the largest close to the end-use location (Ahola, 2012; Tolvanen et al., 2013).

The illustrated losses in Figure 1.2 are consequences of roughly categorized efficiency levels in the different phases of the energy flow. Naturally, the energy use of the pump unit is dictated by the surrounding system and process conditions. Because of this, usually the best results, when finding energy savings in pumping systems, can be obtained when the different options for improvements are evaluated systematically starting from the process needs (Hovstadius et al., 2005). Different options to increase the energy efficiency of pumping systems can be categorized for example into improvements in the efficiency of the system components, justified component selection and system dimensioning, and the energy efficient adjustment of the system output (Szychta, 2004a; Kaya et al., 2008). This categorization is illustrated in Figure 1.3 with some examples related to each research area.

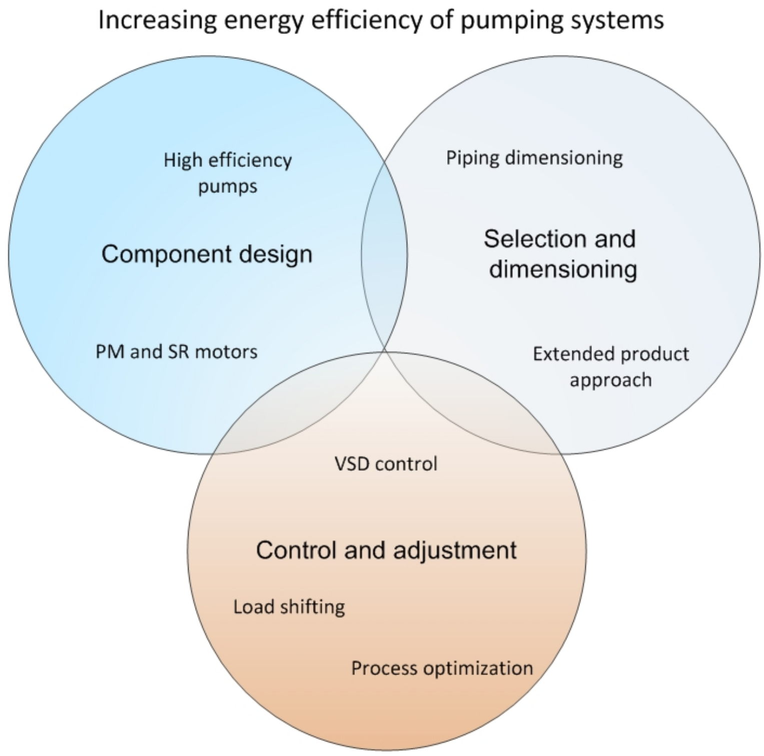

Figure 1.3. Example methods to increase the energy efficiency of pumping systems. The methods can be categorized into areas related to component design, system dimensioning, and adjustment methods.

Replacing inefficient components is probably the most explicit example of energy efficiency improvements in pumping systems. Switching to higher efficiency pumps as well as replacing the induction motor technology with high efficiency motors, such as permanent magnet motors and synchronous reluctance motors, has been shown to reduce the energy consumption in pumping systems with constant output (Kaya et al., 2008; Saidur and Mahlia, 2011; Marchi et al., 2012). The importance of the correct pumping system dimensioning has been discussed for example in the studies by Hovstadius et al. (2005), Bloch and Budris (2010), Pemberton and Bachmann (2010), and Augustyn (2012). In Europe, the requirements of the Minimum Efficiency Index (MEI) for water pumps were established in 2013, and the Energy Efficiency Index (EEI) for extend pump products, referring to a pump driven by a motor with or without a frequency converter, is also being prepared (Europump, 2013). In this standardization work, the aim is to ensure that for example in the component selection, more importance is given to the process load profile instead of a single design point. A significant potential for energy savings can also be found when optimizing the control of the pump system output (Szychta, 2004b; Kaya et al., 2008; Saidur, 2010; Ferreira et al., 2011). As shown previously in Figure 1.2, the net gain of the energy efficiency improvements is also the largest close to the end-use location since the saved energy when delivering the desired output can correspond ten times the need of primary energy (Ahola, 2012; Tolvanen et al., 2013).

A significant technology to increase the energy efficiency in pumping system has been identified in the variable speed control of pumps (Europump and Hydraulic Institute, 2004; de Almeida et al., 2005; Saidur et al., 2012). The use of variable speed control of pumps instead of valve adjustment has shown to reduce the energy consumption of pumps especially in systems having a wide range of flow (Bernier and Bourret, 1999; Pemberton, 2003; Europump and Hydraulic Institute, 2004). Variable speed technology can also reduce the energy costs in pumping systems without load variations by setting the operational speed to an optimum level (Marchi et al., 2012). The dominant technology in variable speed control is a frequency converter, often called the variable speed drive (VSD), coupled with a three-phase induction motor (de Almeida et al., 2005; Ferreira et al., 2011). The VSD allows the frequency control of the power applied to the motor, and the rotational speed of the induction motor can therefore be adjusted. Despite allowing the rotational speed control of motor-driven pumps, modern VSDs have become units with embedded pump intelligence, for example algorithms which can be used to monitor and control the pump unit operation (Pemberton, 2003; ABB, 2006). This allows a seamless integration of the pump into the process control via VSD.

It is justified to say that implementing the pumping system with high efficiency components and new control technologies can lead to significant energy savings in many industrial and municipal pumping tasks. Despite this, it is also important to evaluate the energy efficiency improvements with the right measures and concerning the process needs. So far, the effectiveness of the pumping system components has been evaluated mainly with product approach, which often leads to observing the efficiency of the individual components (Europump, 2013). Naturally, this can be a justified base from a supplier point of view, since the energy efficiency of the pump system component refers to the pump's ability to transfer the energy into liquid with minimum losses. From a pump user point of view, this is not necessary a suitable way to demonstrate the energy efficiency, since the user is usually interested in fulfilling the pumping task with minimum energy. It is essential to understand that these are not necessarily the same thing.

In this thesis, the studied energy efficiency improvements are related to the variable speed control of pumps using frequency converters. Hence, the energy saving methods related to high efficiency components and system dimensioning in pumping applications are excluded in this context. The scope and the objectives of the thesis are discussed further in the following sections.

1.2 Motivation of the study

Pumping systems having a widely varying flow rate demand are often operated with parallel-connected pumps (Hooper, 1999; White, 2003; Gülich, 2008). Parallel operation of pumps is widely used in several industrial and municipal sectors, for instance in waste water treatment, power plants, irrigation, and various cooling and water delivery pumping tasks (Jones, 2006; Moreno et al., 2007; Bortoni et al., 2008). The selected control method for parallel-connected pumps depends on the process requirements and the surrounding system conditions. In the simplest case, parallel-connected pumps can be operated with an on-off control method, in which additional parallel pumps are started and stopped according to the desired flow rate. In the systems in which more accurate flow regulation is needed the adjustment can be carried out by applying throttling or rotational speed control.

The energy saving benefits when using rotational speed control of pumps instead of throttling control have been broadly studied (Europump and Hydraulic Institute, 2001; Europump and Hydraulic Institute, 2004; de Almeida et al., 2005; Saidur et al., 2012). The experiences from energy saving benefits and also the political and ecological pressure have increased the number of VSDs in pumping systems (de Almeida et al., 2003; IEA, 2012). So far, less effort has been put into studying the energy-saving potential particularly in variable speed controlled parallel pumping systems.

Parallel pumps are typically used in processes having large variations in the delivered flow rate. Therefore, it is important to understand the effect of the varying conditions on the performance of each parallel-connected pump. Regardless of the way the flow rate is adjusted, the location of the operation point of parallel-connected pumps has a significant effect on both energy usage and the reliability of the pumping (Jones, 2006; Gülich, 2008). It is often suggested that pumps should be operated as near as possible to the Best Efficiency Point (BEP) (ANSI/HI, 1997; Barringer, 2003). At the BEP, the efficiency of the pump reaches its highest value. Also, it has been shown, that the reliability of pumping (e.g. the mean-time between failures), when the pump is run at the nominal speed, is at its highest close to the BEP. However, the varying rotational speed of the pumps affects, not only the energy consumption of the pumping, but also the pumping reliability (Stavale, 2008; Martins and Lima, 2010). It is important to take this into account in every variable speed pumping scenario.

Maximizing the benefits of variable speed control in pumping requires that the *control scheme* or *control strategy* of the VSD controlled pumps has to be determined in a systematic way (Hovstadius et al., 2005; U.S. Department of Energy, Hydraulic Institute, 2006). The control strategy in a pumping task can be described as a combination of methods and technologies which have been selected to fulfil the possibly varying process needs. The parallel use of variable speed controlled pumps gives opportunities to the pump user to fulfil the pumping task with several options (Zhao et al., 2012; Lamaddalena and Khila, 2013). In other words, it is possible to deliver the same output with a varying number of pumps and rotational speeds. The justified selection for the control strategy in these situations requires that the potential risks of harmful events, which can reduce the energy efficiency or reliability of the pumping, are avoided.

The information needed for optimizing the energy use of the VSD controlled parallel pumping system can be gathered by analyzing the available system data, using separate start-up measurements, or constantly monitoring the process output (Bortoni et al., 2008; Yang and Borsting, 2010; Lamaddalena and Khila, 2013). However, the required information for the optimization can be a limiting factor, since the pump user may be aware of the system details only to a limited extend (Kini et al., 2008). Correspondingly, the start-up measurements may have to be repeated if sudden changes occur in the pumping system characteristics. Also, the available real-time process information on the pumps is often limited since the direct measuring of the pump output is rarely available in pumping systems and monitoring the operation point of each parallel-connected pump with external equipment is even more exceptional (U.S. Department of Energy, Hydraulic Institute, 2006). As direct measurements are often excluded, there is a need for alternative, easily implementable output monitoring methods. Hence, studying the pump operation point monitoring applying VSDs is clearly justified.

The aim of the indirect measuring methods of the pump output or *model-based estimation methods* applied in VSDs is to use pump characteristic curves or system details and selected monitoring values (e.g. rotational speed, torque) in order to determine the operation point of the pump (Liu, 2002; Hammo and Viholainen, 2005). Studying these methods can reveal whether this kind of operation monitoring can be used for realizing energy efficiency improvements in pumping. It is also reasonable to find out whether the model-based operation point estimation can be used for control purposes in certain systems.

Utilizing model-based operation point estimation is interesting especially in variable speed controlled parallel pumping systems since it opens an opportunity to evaluate the operation point and the energy efficiency of each parallel-connected pump without the installation of flow meters on each parallel pumping unit. Hence, energy efficient control strategies for VSD controlled parallel pumps based on real-time operation point monitoring can be studied.

1.3 Objectives of the study

The main objective of this thesis is to determine suitable methods to increase the energy efficiency in variable speed controlled parallel pumping systems in which the typical parallel pumping set consist of at least two pump units comprising a VSD coupled with an induction motor and a centrifugal pump. In this thesis, the studied energy efficiency improvements concentrate on the control options of the parallel pumping set, excluding the solutions based on component level improvements (high efficiency pumps, motors, and VSDs) and system design (selection of optimal components according to the system characteristics). Therefore, the possibilities for energy savings are mainly sought in pumping systems which are already in use. Outlining the component level and design level outside this study can be justified since the aim in this case is to realize the possible energy efficiency improvements without installing additional devices or equipment to the system. The *control procedure* in this thesis means generating a speed reference to pump units, including the aim for justified energy efficiency and operational reliability of pump units.

The selected research questions are related to three main research areas. The selected research areas can also be seen as the main challenges when controlling variable speed driven pumps in a parallel pumping set. In this thesis, these challenges are studied in parallel pumping systems in which each parallel-connected pump unit is equipped with a VSD.

The research questions are described as:

1. What are the suitable operation conditions for variable speed controlled parallel pumps?

The first question is related to the possibilities to gain energy savings in parallel pumping systems using variable speed control of pumps. Naturally, finding energy-saving potential requires that certain indicators for the energy efficiency have to be selected. Related to this first research question, the recommendable operation region for variable speed driven centrifugal pumps is discussed from the energy efficiency point of view. Also, the utilization of the findings in the control procedures of parallel-connected pumps is studied.

2. How the operation point of each parallel pump can be monitored without metering applying variable speed drives?

The second question is related to the operation point monitoring of parallel-connected pump units without direct metering. In the studies related to this question, possibilities are sought for determining the flow rate of each pump and the total flow rate of the parallel pumping set using pump performance curve-based flow metering of VSDs. This research area presents also options for sensorless operation point estimation using VSDs and a possibility to increase the accuracy and usability of model-based operation point monitoring with the combined use of certain estimation methods. Based on these studies, the suitability of the model-based estimation methods for control purposes is discussed in variable speed controlled parallel pumping systems.

3. How energy efficient control procedures can be implemented in variable speed controlled parallel pumping systems?

The third research question focuses on the implementation of the energy efficient control procedures in variable speed controlled parallel pumping systems. The study of this research area focuses on implementing control schemes in parallel pumping systems with less system information and external measurements. Therefore, options for utilizing the model-based operation point estimation method for applying energy efficient control strategy in variable speed controlled parallel pumping systems are also studied.

1.4 Outline of the thesis

In this thesis, the parallel use of rotational speed controlled centrifugal pumps operated for instance in raw water, waste water, and industrial pumping are discussed. Based on actual measurements and simulated parallel pumping values, energy-efficient parallel pumping methods will be studied.

Chapter 1 introduces the research questions and motivation of the thesis. Chapter 1 also introduces the main focus areas of this study, the relation of the publications to focus areas and author's contribution in each publication.

Chapter 2 presents the theoretical background of this thesis. First, the basic concepts of parallel operated centrifugal pumps are discussed. Secondly, research related to the energy-efficient control of parallel pumps is studied. Chapter 2 also presents the application of variable speed drives in pump operation monitoring.

Chapter 3 presents the selected methods for studying the three main research areas of the thesis. The selected methods are discussed considering the suitable operation conditions of variable speed driven parallel pumps, model-based pump monitoring using variable speed drives, and the proposed energy efficient control strategy for parallel pumps. Chapter 3 presents also the basis and the most relevant findings and results of the calculations, simulations, and laboratory measurements related to three main research areas.

Chapter 4 presents the discussion part of the thesis reflecting the findings on the set objectives and research questions. The thesis is concluded in **Chapter 5.**

The thesis is based on the articles included in the List of Publications. The relation of the publications to the research areas is illustrated in Figure 1.4.

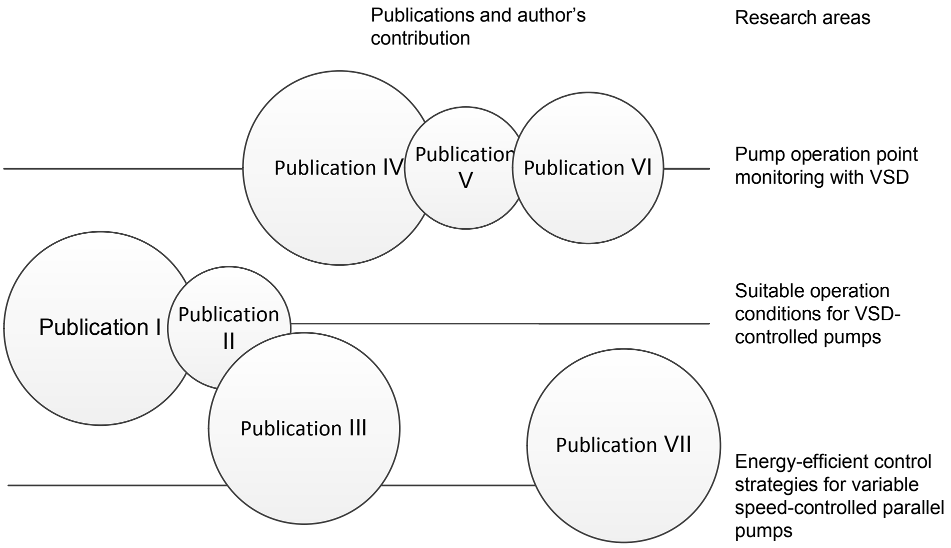

Figure 1.4. The relation of the publications to main research areas and the author's contribution in each publication.

The research questions are discussed in the publications as follows: Question 1 considering suitable operation conditions for variable speed driven pumps is discussed in Publications I–III and VII. Question 2 is related to the pump operation point monitoring applying VSD, which is discussed in Publications IV–VI. Question 3 is related to energy-efficient control strategies, which are discussed first in Publication III and more thoroughly in Publication VII. The author's contribution to each article has been discussed earlier in the List of Publications section, but the contribution can also be seen in Figure 1.4, in which the size of the article sphere corresponds the author's work in that specific publication.

In this thesis, only the main findings related to the research questions are presented. Detailed discussion of the results can be found in the attached publications shown in the Appendix section. In the case of Publications I, and II, some additional calculations and visualizations which are not included in the published papers have been added to this thesis. The aim of this additional material was to support the overall findings related to the first research area.

2 Energy efficiency in variable speed parallel pumping

In this chapter, the basic concepts of variable speed driven parallel-connected pumps and parallel pumping tasks are discussed. The solutions for the energy-efficient control of parallel pumping systems are studied.

2.1 Parallel operation of centrifugal pumps

The use of centrifugal pumps in parallel allows the production of a wider range of flow rates than would be possible with a single pump. In other words, the parallel connection of centrifugal pumps increases the flow rate capacity of a pumping system. Pumping tasks operated with parallel-connected pumps can be found in industrial processes such as petrochemicals, power plants, and raw water pumping (Bortoni et al., 2008). Parallel pumping applications are also common in water delivery, waste water treatment, irrigation, and also in various pumping systems related to cooling tasks (Jones, 2006; Moreno et al., 2007; Tang and Zhang, 2010; Zhao et al., 2012). A simplified example of a system consisting of two parallel-connected pumps and two water reservoirs combined by individual suction piping and common outlet piping section is illustrated in Figure 2.1.

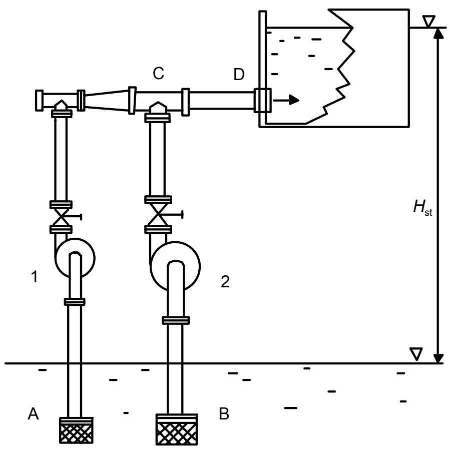

Figure 2.1. Two parallel pumps feeding a common outlet pipeline. The parallel pumps (marked with 1 and 2) have their individual piping parts between points A–C and B–C feeding the common pipeline between points C–D (Wirzenius, 1978).

The most common type of a pump unit in parallel pumping systems is a single-volute radial flow centrifugal pump attached to an induction motor. The interest for energy efficient flow adjustment of parallel-connected pumps has increased the use of frequency converters, also referred to as variable speed drives (VSDs) with induction motors, enabling the rotational speed control of the induction motor-driven pump. The combination of a pump, motor, and a VSD is often referred to as the pump drive train. The use of these terms in this study is illustrated in Figure 2.2 which shows a *pump unit* or *pump drive train* applied in a *pumping system.* In addition to possibly several pump units, the parallel pumping system includes the inlet and outlet reservoirs and the piping sections.

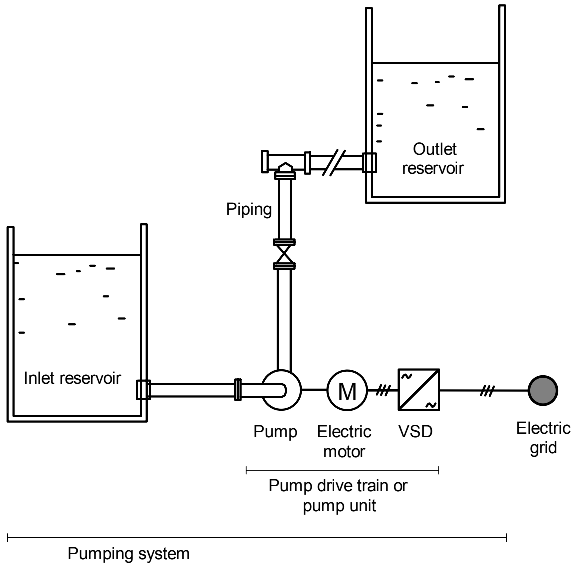

Figure 2.2. Pump drive train or pump unit in a pumping system. The pumping system consists of pump unit, inlet and outlet reservoirs, a piping network, and the pumped liquid. A VSD can be included in the pump unit to enable the rotational speed control of the pump driven by an electric motor.

2.1.1 Modelling the pump system head and pump performance

When centrifugal pumps are used in a system, the advancing elevation pressure to ensure a certain flow rate in piping is often called the system head, and it is marked with H. This resembles the head which the pump must overcome to deliver the certain amount of flow rate, often marked with Q, in the current system. The system head can be described as a sum of the geodetic difference between the suction and discharge fluid levels, the differential pressure in suction and discharge fluid levels, the friction head in piping, valves, fittings, etc., and the difference in the velocity heads in the inlet and outlet section of the system. Thus, the system head can be expressed with equation

$$H = H_{\mathrm{geo}} + \frac{p_{\mathrm{a}} - p_{\mathrm{e}}}{\rho g} + \frac{v_{\mathrm{a}}^{2} - v_{\mathrm{e}}^{2}}{2g} + \sum H_{\mathrm{r}} \quad (2.1),$$

where H is the head, p the pressure, ρ the density, g the gravitational constant, and v the velocity of the pumped liquid. Subscript $_{\mathrm{geo}}$ refers to the geodetic difference, $_{\mathrm{a}}$ to the outlet section, and $_{\mathrm{e}}$ to the inlet section of the system. Subscript $_{\mathrm{r}}$ denotes the head loss conducted by friction in the system (KSB Pumps, 1983). The system head can be divided into the static part and dynamic part; the static part includes the geodetic head and the pressure difference between the inlet and outlet sections of the system and the dynamic part consists of the velocity head difference and the friction head (Gülich, 2008). Thus, the static head H_{st} can be written as

$$H_{\mathrm{st}} = H_{\mathrm{geo}} + \frac{p_{\mathrm{a}} - p_{\mathrm{e}}}{\rho g} \quad (2.2).$$

Correspondingly, the dynamic head H_{dyn} is

$$H_{\mathrm{dyn}} = \frac{v_{\mathrm{a}}^{2} - v_{\mathrm{e}}^{2}}{2g} + \sum H_{\mathrm{r}} \quad (2.3).$$

The system head is usually illustrated with *system head curve* or just *system curve* in the *QH* axis. The pressure difference between the inlet and outlet sections is relevant in closed systems having pressurized reservoirs. In open-loop systems, this difference is insignificant and therefore usually ignored. Also, the velocity head in the dynamic part is often considered very small compared with the total system head. Therefore, the velocity head in equation (2.3) can be ignored to simplify the system modelling. The friction head (or the dynamic head in the simplified case) depends on the flow rate. In practice, the dynamic head in the system is

$$H_{\mathrm{dyn}} = k \cdot Q^{2} \quad (2.4),$$

where k can be described as a liquid-based coefficient depending also on the characteristics of the piping. The dynamic head can also be described as a sum of the head losses in each element of the piping. The dynamic head loss caused by friction for example in a pipeline section can be calculated using equation

$$H_{\mathrm{dyn}} = \left(\lambda \frac{l}{d} + \sum \zeta \right) \frac{v^{2}}{2g} \quad (2.5),$$

where λ is the loss-coefficient for a certain pipe roughness, l is the pipe length in meters, d is the pipe diameter in meters, and ς is the loss-coefficient for each elbow, valve, fitting, etc. (Wirzenius, 1978). Using equations (2.2) and (2.4), the system head is often simplified to

$$H = H_{st} + k \cdot Q^2 \tag{2.6}$$

The behaviour of the pump under different operation conditions is indicated with pump characteristic curves. Typically, the curves are given for the produced head H, power input P, and efficiency η as a function of flow rate Q at constant rotational speed n (Sulzer, 1998; Karassik and McGuire, 1998). In addition to the QH curve, QP curve, and $Q\eta$ curve, the pump manufacturers often present the $NPSH_r$ (Net Positive Suction Head) as a function of the flow rate. The $NPSH_r$ curve illustrates the required head at the suction of the pump to avoid harmful operation of the pump, such as cavitation.

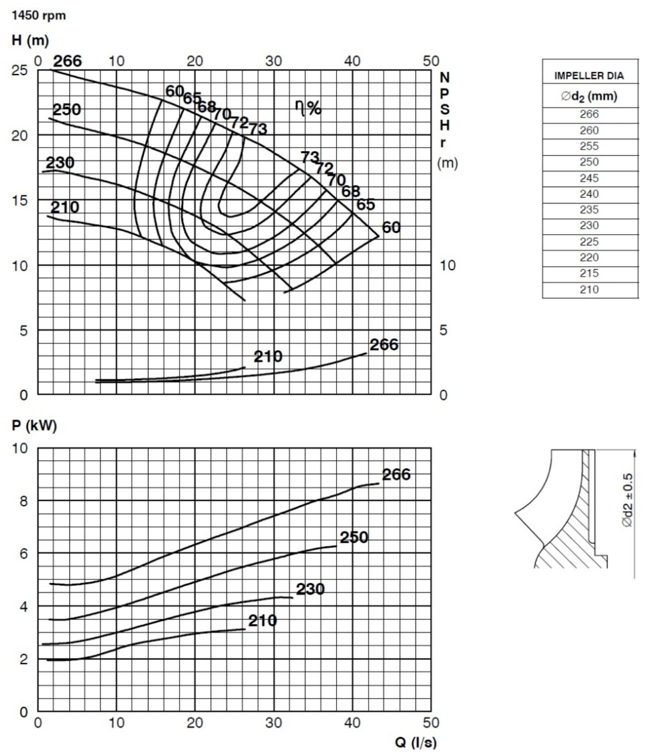

Figure 2.3. Characteristic curves for the centrifugal pump APP22-80 at the constant speed of 1450 rpm. The QH and QP curves are given in four different impeller diameters: 266 mm, 250 mm, 230 mm, and 210 mm. The $NPSH_r$ curve is illustrated for 266 mm and 210 mm impeller diameters (Sulzer, 2006).

Figure 2.3 illustrates the characteristic curves for the single-stage centrifugal pump APP22-80 for impeller diameters 210–260 mm. Based on the curves in Figure 2.3, the observed pump would gain the highest efficiency (approx. 73%), in the case of a 250 mm impeller, when operating at the flow rate of 26 l/s and a 16 m head. This point is traditionally referred to as the Best Efficiency Point (BEP).

The pump performance with varying impeller diameters and rotational speeds are estimated using affinity laws. Using the affinity laws it is possible for example to generate a series of performance curves for a certain range of different rotational speeds. According to the affinity laws, if the pump rotational speed *n* differs from the nominal rotational speed n_{nom}, the relations between the pump flow rate, head, input power, and pump rotational speed can be written as

$$\frac{Q}{Q_0} = \frac{n}{n_{\mathrm{nom}}} \tag{2.7}$$

$$\frac{H}{H_0} = \left(\frac{n}{n_{\mathrm{nom}}}\right)^2 \tag{2.8}$$

$$\frac{P}{P_0} = \left(\frac{n}{n_{\mathrm{nom}}}\right)^3 \qquad (2.9),$$

where $_{\mathrm{nom}}$ denotes the operational value in the nominal rotational speed (Volk, 2005; Sulzer, 1998). The use of the affinity laws is based on the assumption that the efficiency of the pump remains constant regardless of the pump speed (Gülich, 2008). Figure 2.4 shows the characteristic curves for the centrifugal pump A P61-600, which have been generated using the affinity laws (2.7−2.9). The *QH* and *QP* curves are given at nominal rotational speed 1000 rpm and varying rotational speeds 400−800 rpm. The lines of constant pump efficiency according to affinity laws are also plotted in Figure 2.4 (a).

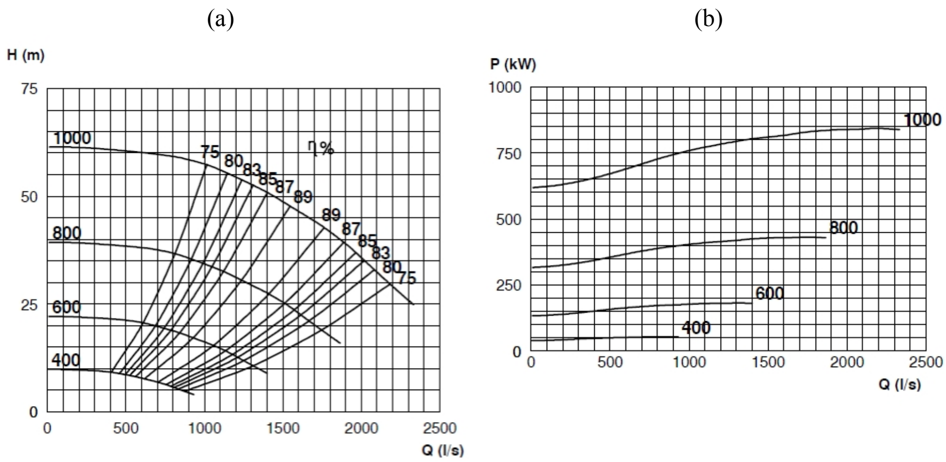

Figure 2.4. Characteristic curves for the centrifugal pump A P61-600 at varying rotational speeds 400−1000 rpm. The *QH* (a) and *QP* (b) curves at constant impeller diameter 700 mm are generated using affinity laws (Sulzer, 2008)

When running a pump in a pumping system, the pump operation point can be found in the intersection of the pump *QH* curve and the system curve. The pump *QH* curve, system curve, and the resulting operation point are illustrated in Figure 2.5.

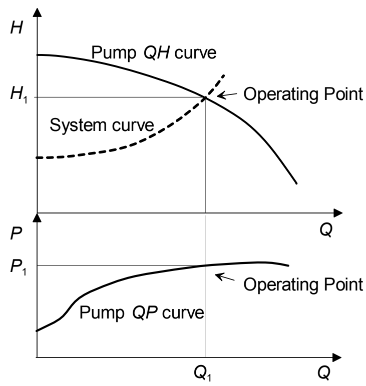

Figure 2.5. Pump operation point. The pump operation point is located in the intersection of the system curve and the *QH* curve (Q_1,H_1). The pump power in this operation point is P_1.

2.1.2 Measures for pumping system effectiveness

The effectiveness of a single pump is often observed with the pump efficiency

$$\eta_{\mathrm{pump}} = \frac{Q \cdot \rho \cdot g \cdot H}{P_{\mathrm{pump}}} \tag{2.10}$$

where P_{pump} refers to the power input of the pump. If the total input power including the motor's and drive's losses is observed in equation (2.10), the system efficiency (Yang and Borsting, 2010) is

$$\eta_{\mathrm{sys}} = \frac{Q \cdot \rho \cdot g \cdot H}{P_{\mathrm{in}}} \tag{2.11}$$

where P_{in} represents the total input power to the pump drive train. Although indicating only the efficiency from wire to the water, the system efficiency is sometimes referred to as the total efficiency as it shows the combined efficiency of the pump drive train. The wire-to-water system efficiency can also be written:

$$\eta_{\mathrm{sys}} = \eta_{\mathrm{pump}} \cdot \eta_{\mathrm{motor}} \cdot \eta_{\mathrm{VSD}} \tag{2.12}$$

where η_{motor} is the motor efficiency and η_{motor} the efficiency of the VSD (Szychta, 2004a; Marchi et al., 2012). In addition to the system efficiency, the effectiveness of a pumping task can be evaluated using the specific energy consumption, which describes the energy used per pumped volume (Europump and Hydraulic Institute, 2004). The specific energy consumption is given by

$$E_{\mathrm{s}} = \frac{P_{\mathrm{in}} \cdot t}{V} = \frac{P_{\mathrm{in}}}{Q} = \frac{\rho \cdot g \cdot H}{\eta_{\mathrm{sys}}} \tag{2.13}$$

where E_{s} is the specific energy consumption, P_{in} the input power, t time, and V the pumped volume. Since the delivered flow rate is often the control variable in parallel pumping, the specific energy can be seen as a justified metrics to evaluate the energy efficiency in different parallel pumping control strategies. The specific energy consumption for parallel-connected pump units can be written

$$E_{\mathrm{s}} = \frac{P_{\mathrm{in,tot}}}{Q_{\mathrm{tot}}} = \sum_{i=1}^{n} \frac{\rho g \cdot H_i}{\eta_{\mathrm{sys},i}} \tag{2.14}$$

where $P_{\mathrm{in,tot}}$ is the total input power of parallel-connected pump units, Q_{tot} the total flow rate and the subscript i represents the number of parallel-connected pumps in the system.

The specific energy consumption in closed-loop systems or systems without static head is primarily dependent on the losses in the piping system and the system efficiency. In systems with the static head, the minimum value of specific energy is also dependent on the amount of static head and the coefficients ρ and g (Europump and Hydraulic Institute, 2001).

2.1.3 Relation between pump operation point and pump reliability

In general, a centrifugal pump should be driven as close as possible to its best efficiency point (BEP), in the range often referred to as the preferred operating area (POA) or region (ANSI/HI, 1997; Barringer, 2003). This is the most effective way to ensure that the pump operates with a good efficiency and reliability (Gülich, 2008; Bloch and Budris, 2010). In the case of fixed-speed pumps, the preferred operation region can be limited, as can be seen in Figure 2.6, which illustrates the relation between the pump relative output and pump reliability (Barringer, 2003). The reliability curve is illustrated as a function of the relative flow rate and the characteristic life *h* based on MTBF (Mean Time between Failures). In this case, *h* has the highest value in BEP. Correspondingly, when the fixed-speed pump is operated in the region of 80–110% of the BEP flow rate, the reliability can be considered to be only 53% of its highest value. Thus, running pumps far away from the POA can lead to events which can accelerate the pump wearing and decrease the reliability of the pump.

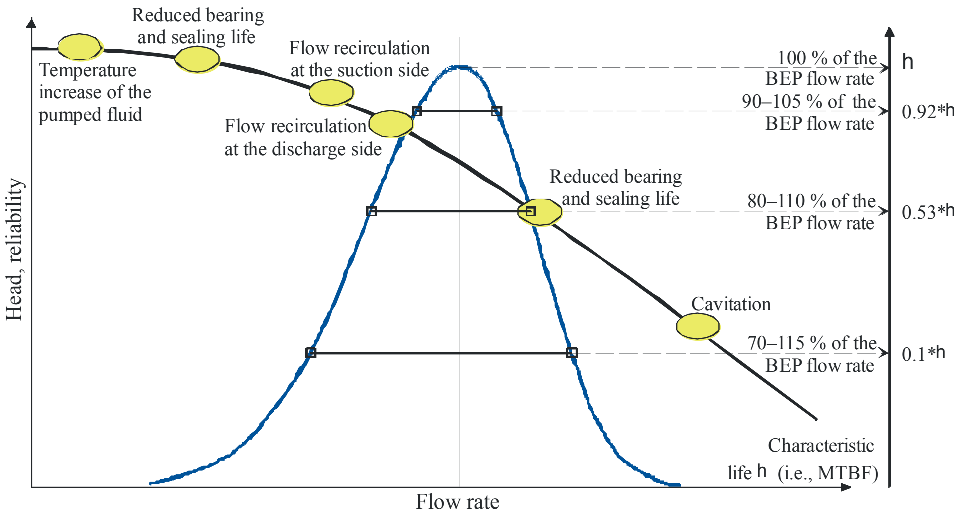

Figure 2.6. Relation between the relative flow rate and reliability. In this case, reliability is observed as a characteristic life based on the MTBF (secondary axis on the right) (Barringer, 2003; Ahonen, 2011). Also the possibly occurring harmful events are plotted on the pump *QH* curve.

As shown in Figure 2.6, moving away from the BEP can expose the pump to several harmful events, such as flow recirculation, reduced bearing and sealing life, shaft deflection, and cavitation, all of which can affect negatively the pump reliability. The illustrated relation is applicable for fixed-speed pumps, but it has been shown that there is also a clear connection between the pump reliability and pump rotational speed (Stavale, 2008; Bloch and Budris, 2010; Martins and Lima, 2010). The more acceptable operation state for a pump can thus be achieved by reducing the rotational speed, as the speed reduction typically reduces the level of vibrations and radial loads to pump (Kaya et al., 2008; Martins and Lima, 2010). In a study by Stavale (2008), it was demonstrated that the reliability of the pump does not necessarily drop when the location of the operation pump is away from the BEP when operating the pump at 50…62.5% of the nominal speed. Instead, the relative flow region in which high reliability can be achieved in terms of vibration levels seems to be significantly wider compared to the operation at the 100% pump speed. The effect of the operation point on the pump reliability has been discussed extensively already in the study by Ahonen (2011).

Operating centrifugal pumps at regions in which the service life of the pump is reduced can increase the pump life cycle costs due to the increased maintenance costs, additional investments, and production losses. Therefore, selecting the right pump in accordance with the process needs in the system design phase is essential to maintain justified operation. It is also important to take the relation between the pump operation point and reliability into account in the pump control, as there might be a risk of unwanted operation conditions when adjusting the output of the pumping system. Hence, the suitable adjustment method and the overall control strategy should be selected in such a way that the possibly hazardous events could be avoided when delivering the desired output to the process.

2.1.4 Operation point of parallel-connected pumps

The operation point of parallel pumps is found at the intersection of the system curve and the combined *QH* curve of all pumps in operation. The combined characteristics of parallel pumps are obtained by adding the flow rates of the operated pumps at the constant head. The parallel pumping system of two different parallel-connected pumps is illustrated in Figure 2.1. In the illustrated example, the pump units have individual piping parts between points A−C and B−C feeding the common pipeline between points C−D. The operation of the parallel pumps in the *QH* axis is shown in Figure 2.7.

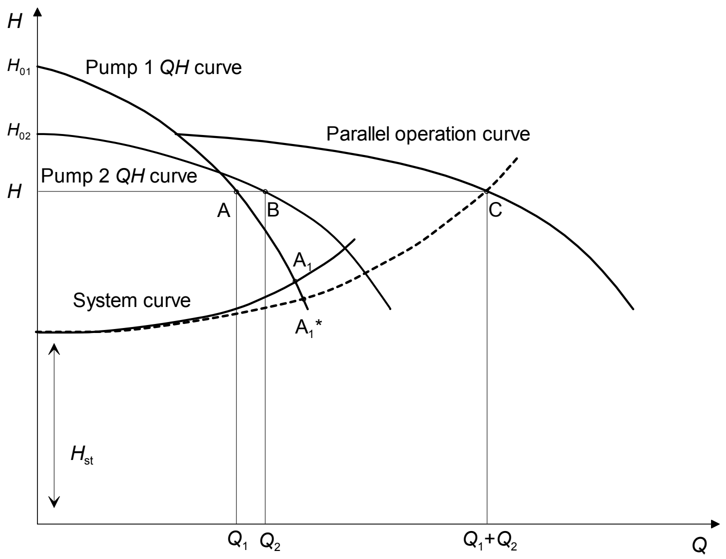

Figure 2.7. Parallel operation of two different pumps. The point C represents the total flow rate Q_1+Q_2 delivered through the system, and the points A and B show the flow rates of pumps 1 and 2. If a single pump (Pump 1) is operating, its flow rate can be located at point A_1 instead of point A_1* if there is significant friction head in individual piping sections. H_{01} and H_{02} represent the shut-off heads of pumps 1 and 2 (Gülich, 2008).

The operation point of the parallel pumps in Figure 2.7 is at the intersection of the system curve (point C) and the parallel operation curve which adds the flow rates of pumps 1 and 2 at the same head H. Correspondingly, the individual operation points are found at points A and B resulting in the total flow rate Q_1+Q_2 to the system. If only a single pump is operated (Pump 1), its flow rate is given by point A_1 (Figure 2.7). The difference between the system characteristics for the operation with a single pump and with two pumps depends on the contribution of the friction losses in the individual piping branches (e.g. piping section A−C in Figure 2.1). Often the friction head in individual branches can be regarded insignificant, which would result in the operation of Pump 1 at point A_1* (Figure 2.7) in a single pump operation. Although in the illustrated system, the example has a relatively high portion of dynamic head, the parallel operation of pumps is often most justified if the system curve is flat. In the case of too steep system curves, the amount of combined flow gained by adding the second or third pump to the system can be so incremental that the parallel operation is not sensible (Volk, 2005; White, 2003).

The maximum head for the pump at the set operational speed is often referred to as the shut-off head of the pump. In Figure 2.7, the shut-off head for the illustrated parallel pumps are marked with H_{01} and H_{02}. Usually, parallel pumps having a steadily falling

QH curve are selected for parallel use to avoid operation near the shut-off head since prolonged operation at zero flow can result in pump damage (Shiels, 1997). The shut-off head of the operated parallel pumps may vary if the pumps are not identical or they are not operated at the same speed. The shut-off head can also be different than expected due to manufacturer tolerances (Gülich, 2008).

As shown in Figure 2.7, starting and stopping of the parallel pumps can shift the operation point of the pumps remarkably in the *QH* axis. Besides avoiding the shut-off, the parallel pumps should be selected and operated taking the suction characteristics into account. Figure 2.8 shows another example which illustrates the operation of parallel pumps in the *QH* axis. In this example, the selected three parallel pumps are identical, and the system conditions are plotted into the characteristics of one pump.

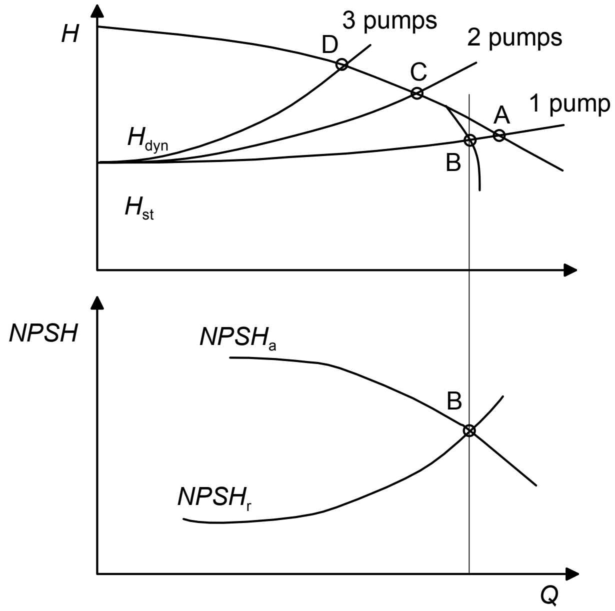

Figure 2.8. Relation between the flow rate and NPSH in a parallel pumping system. The plotted system characteristics show the operation points when a different number of parallel pumps are operated. At points D and C, the normal operation is possible because the $NPSH_a$ is above the $NPSH_r$. At point A, the $NPSH_a$ is below the $NPSH_r$, and the pump is exposed to cavitation, causing that the flow will be limited to point B (Shiels, 1997; Volk, 2005; Gülich, 2008) .

In the illustrated example (Figure 2.8), justified operation is achieved when operating the system with three or two parallel pumps since the $NPSH_a$ is well above the $NPSH_r$ at the operation points C and D. However, when only one pump is operated, the $NPSH_a$ is clearly lower than the $NPSH_r$, and the output of the pump is limited to the point B because of cavitation (Gülich, 2008). Cavitation is caused by the formation and collapse of vapour bubbles, and besides degraded efficiency, it can cause severe damage to the pump impeller (Karassik et al., 2001; Volk, 2005).

2.1.5 Adjusting the output of parallel pumping system

In general, the selected control method of the pumping system should be based on the process needs. The available solutions like throttling, by-pass, and rotational speed control can be evaluated for example in terms of a possibly varying flow or head requirements, the necessity of accurate flow adjustment, and the spread of the output requirements in the operating time span (Hovstadius et al., 2005).

Typical methods to control the output of pumping systems are for instance the on-off method, throttling, and rotational speed control. The use of the on-off method is usually justified for applications having a tank or a reservoir and no need for accurate control of the flow rate. In pumping systems with a regular need for flow adjustment, the throttling method or rotational speed control are commonly used. The effect on the pump operation point when adjusting the output with throttling, rotational speed control, and on-off control is illustrated in Figure 2.9, which plots the *QH* curve of the pump and the system curve in the *QH* axis.

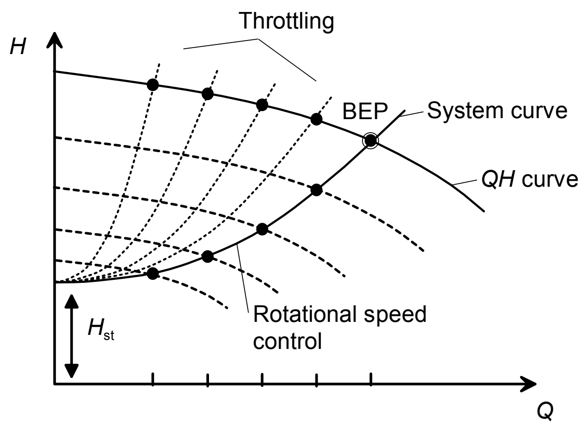

Figure 2.9: Operation points using basic flow adjustment methods. In the on-off control, the pump is operating at the BEP. Reducing the output of the pump with throttling shifts the operation points along the *QH* curve. Correspondingly, when the output is adjusted with rotational speed control, the operation points are found along the system curve. (Europump and Hydraulic Institute, 2001)

The pump is operated only at a single operation point when using the on-off control, which in this example is located at the BEP (Figure 2.9). Adjusting the output of the pump with throttling valve changes the dynamic head in the system causing that the system curve intersects the *QH* curve in new operation points. Thus, throttling the output of the pump reduces the flow rate, but the pump is forced to produce increased head. If the output of the pump is reduced with rotational speed control, the piping system characteristics remain unchanged. Instead, the pump characteristics moves down in accordance with the speed change and the operation points are found at new intersections with system curve (Figure 2.9).

The benefit of the rotational speed control is that the efficiency of the pump usually drops much less compared to the throttling adjustment, and also less energy is wasted at increased head when reducing the output of the pump (Rossmann and Ellis, 1998; Carlson, 2000; Europump and Hydraulic Institute, 2004; Hovstadius et al., 2005). This has a positive effect on the energy efficiency of pumping, as the amount of hydraulic losses in the piping system decreases with the lower flow velocity. The amount of static head has a strong influence on the pump energy efficiency when using rotational speed control. Figure 2.10 illustrates the operation points of the rotational speed controlled pump in two different system scenarios: in both cases, the pump operation point is located in BEP when the pump is running at nominal speed, but the static head of the system is different. As it can be seen from Figure 2.10, in the case of high static head H_{st1}, the pump has to overcome more head compared with the low static head case H_{st2}. Also, the operation points in the low static head system are located close on the line of constant best efficiency (marked *q*' in Figure 2.10), and as a result, the efficiency of the pump remains on a good level during the adjustment. It should be noted that the line *q*' is a theoretical efficiency line based on the affinity laws. (Gülich, 2008; Europump and Hydraulic Institute, 2001)

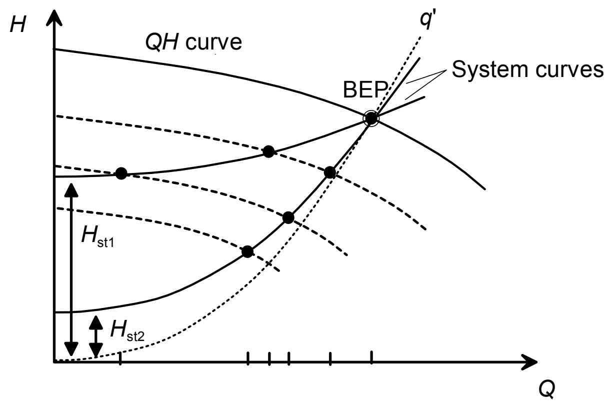

Figure 2.10. Operation points when the output of the pump is adjusted with rotational speed control in high static head and low static head systems. In both system scenarios, the pump operation point is located in BEP when running the pump at nominal rotational speed. The plotted line *q*' illustrates the line of constant efficiency according to affinity laws. (Gülich, 2008; Europump and Hydraulic Institute, 2001)

The impact of the flow adjustment methods on the specific energy consumption is shown in Figure 2.11 which plots the example E_s curves as a function of the flow rate. As it can be seen from Figure 2.11, the rotational speed control is feasible from the energy efficiency point of view, especially in systems having relatively low static head. Using rotational speed control in systems with high static head is usually a more energy efficient control method than throttling, although the difference in E_s can be much smaller than in low static head systems. In the flow region in which the resulting E_s of the rotational speed controlled pump is below the E_s line of the on-off control, the rotational speed control is saving energy compared to the on-off control (Europump and Hydraulic Institute, 2001).

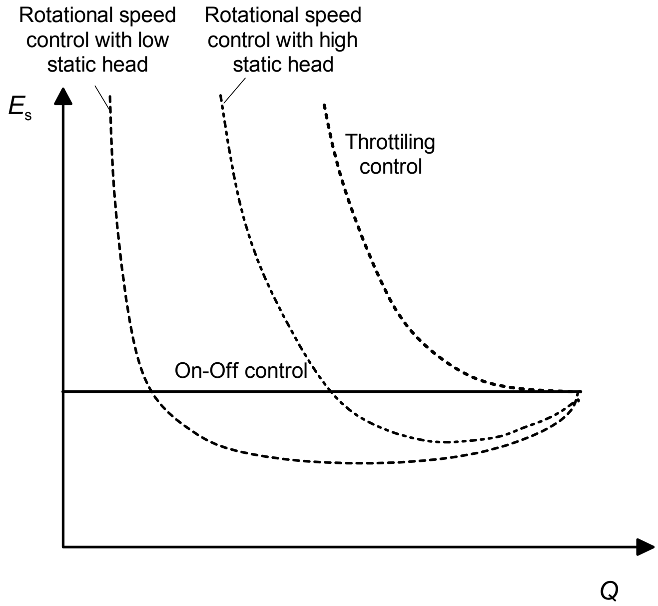

Figure 2.11. Specific energy consumption of pumping in the case of different flow adjustment methods as a function of the flow rate. Examples of the specific energy consumption are plotted using throttling control and rotational speed control in the case of relatively low static head and relatively high static head. The specific energy consumption of the on-off control is constant since the pump is operated only at a single operating point. (Europump and Hydraulic Institute, 2001)

As shown in Figure 2.11, at a certain point, the specific energy of pumping is rising radically when the flow is adjusted towards zero in rotational speed control and in throttling control. This is caused by the moving of the operation point towards shut-off head. Near the shut-off, the pump will have very poor pumping efficiency causing a high rise in the specific energy consumption (eq. 2.13). When using the rotational speed control in pumping systems with high static head, attention should be paid to ensure that the reducing of the rotational speed will not expose the pump to the operation near the shut-off.

The parallel operation of rotational speed controlled pumps ensures a broad flow range for the system (KSB, 2004). The change in the pump's speed has a similar effect on the pump operation point also when the pumps are parallel connected; the operation point can be found at the intersection of the system curve and the parallel *QH* curve of the operated pumps at the given rotational speed. Figure 2.12 plots the *QH* characteristics of a parallel pumping system with three identical centrifugal pumps operating at the nominal speed n_{nom} and at the reduced speed n'. The operation point and the specific energy consumption of the system depend on the number of pumps in use and the set rotational speed. As shown in Figure 2.12, delivering the maximum flow in the *High flow* range of the system, three pumps must be operated at the nominal speed. Here, the operation point of the individual pumps in the illustrated system is located slightly left from the BEP on the pump *QH* curve when the pump is running at the nominal rotational speed.

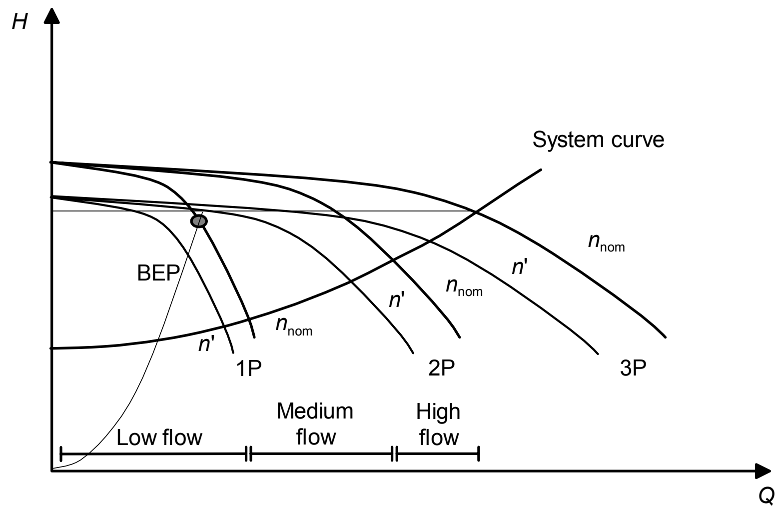

Figure 2.12. *QH* performance curves for three similar parallel-connected pumps (1P, 2P, 3P). The pumps are operating near the BEP when all three pumps are running at the nominal speed n_{nom}. The figure illustrates also the performance of the pumps when the rotational speed of each pump is reduced to n'.

Figure 2.13 shows the specific energy consumption with the changing rotational speed in the same system. The E_s curves demonstrate that, especially at part-loads, there is a clear difference in the energy efficiency when looking for the available options to deliver the same amount of flow to the system with a different combination of parallel pumps and rotational speeds. As illustrated in Figure 2.13, the lowest specific energy consumption E_s in the case of low flow requirement can usually be achieved when using only one rotational speed controlled pump. In the case of medium flow, using two parallel pumps instead of three is more energy efficient. Notably, when three pumps are required to deliver the maximum flow in the illustrated system, all pumps are running very close to the BEP (Figure 2.12). Although the pumps are operated with high pumping efficiency, the specific energy consumption of the parallel pumps can be significantly higher in this case compared with the situation in which the pumps are operating at the reduced flow (Figure 2.13).

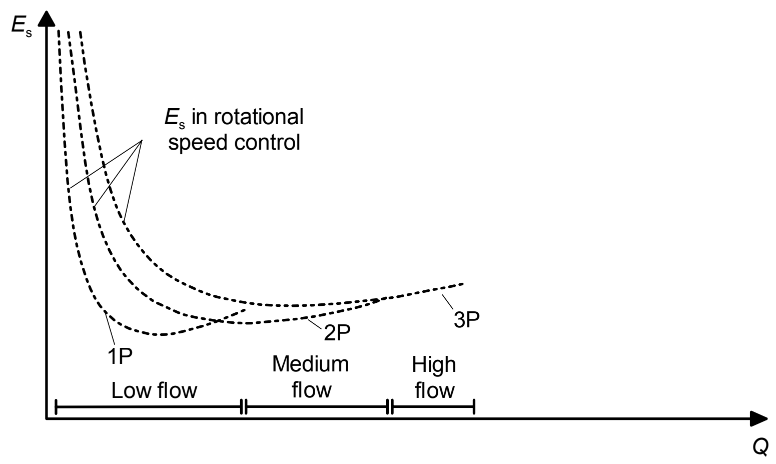

Figure 2.13. Specific energy consumption as a function of flow rate of variable speed controlled parallel pumps. The illustrated curves represent the specific energy consumption at the changing rotational speed when a single pump (1P), two parallel pumps (2P), or three parallel-connected pumps (3P) are adjusted with the rotational speed control with common speed reference.

2.1.6 Staggered and load-sharing operation of parallel pumps

In general, the rotational speed control for operated parallel pumps can be divided into two base-line alternatives: *load-sharing* and *staggered* control methods (KSB, 2004; Jones, 2006). In the staggered operation, the rotational speeds of the pumps are adjusted individually to set the total output of the system to the desired level. In the load-sharing operation, all operated pumps are run at the same speed reference to adjust the system output. Usually, both methods can be used to deliver a very broad adjustment range in various system cases, although there might be a notable difference in the operation conditions and the required energy use of the controlled pumps.

If the flow required by the process is divided for instance between two parallel pumps, the load-sharing operation of pumps can be applied by using a single rotational speed controlled pump in the low flow range, and if more output is necessary, an additional pump can be started before the maximum operation speed is reached. Thus, all pumps can be operated at the same speed reference. In the staggered operation, the rotational speed of the primary pump is adjusted to the nominal value before additional pumps are started. Figure 2.12 presents an example of the impact of the flow adjustment on the pump power use in the staggered and load-sharing operation.

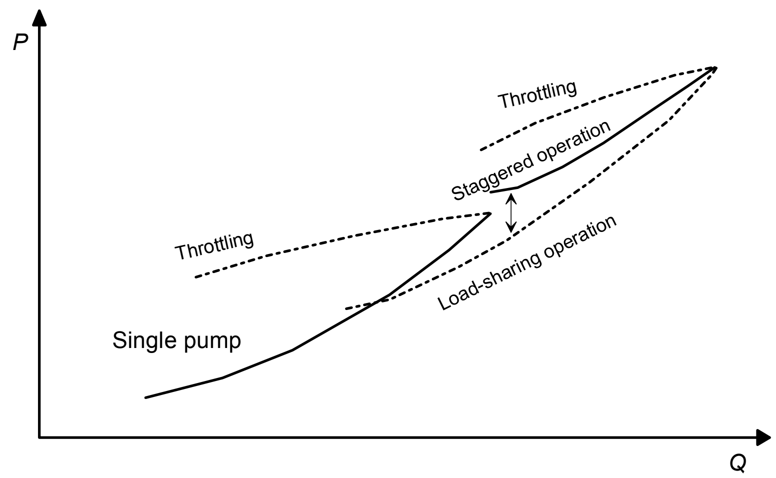

Figure 2.14. Pump power in the case of two parallel pumps when using either staggered or load-sharing operation. (KSB, 2004; Jones, 2006)

The staggered operation of parallel pumps consumes more energy at high flow rates (Figure 2.14), and therefore, it is not typically recommended at variable speed driven pumping stations (Jones, 2006). However, the justified load-sharing operation of parallel pumps requires that all pump units are equipped with VSDs. The staggered operation for rotational speed controlled parallel pumps may be selected for example because of lower investments costs. The pump user may also be satisfied with the fact that the selected control method is saving energy compared with throttling (KSB, 2004; Lamaddalena and Khila, 2013) and the process requirements are met without interruptions. It is also worth noting that the optimal load-sharing operation of the pumps requires that the point at which the additional rotational pumps shall be stopped/started has to be determined in the control strategy (Jinguo, 2008; Vacon, 2012). Although this task may seem simple in theory, implementing the control strategy in real-life systems can set up unexpected challenges. As shown in previous examples, the energy efficiency of pumping in varying operation conditions is dictated by both the pumping unit and system characteristics, and this information required for the optimization may be available only to a limited extent (Yang and Borsting, 2010).

These issues are often the reason for the neglecting of the full optimization of rotational speed controlled parallel pumping systems.

2.2 Increasing the energy efficiency in parallel pumping systems

It is not unusual that pumping stations consist of several different sized parallel pumps. New parallel pumps may be installed because at the normal state the required flow is much less than the maximum needed, or the pumping system has been expanded and the cheapest and easiest option to get the wanted flow level is to add more pumps (Wirzenius, 1978). Therefore, the most energy efficient control strategy to operate the parallel pumps may not be so simple to achieve.

There is a great variety in the system characteristics and selected control methods related to parallel pumping tasks. The energy saving potential in a pumping system can be identified with energy audits and modelling the system operation in varying process conditions (Moreno et al., 2007; Aranto et al., 2009; Moreno et al., 2010; DeBenedictis et al., 2013; du Plessis et al., 2013). For example, DeBenedictis et al. (2013) have presented a method for the estimation of energy savings in municipal water pumping. Using such models it is possible to determine the potential pumping tasks in which the improvements in the energy efficiency of pumping are both realizable and cost effective. The energy efficiency of a parallel pumping station in variable-flow operation is strongly related to the dimensioning of the parallel pumps in line with the system characteristics and process needs.

In the optimization of the pumping control based on the system efficiency, it should also be noted that operating at the highest system efficiency does not necessary mean the lowest energy consumption at the desired flow rate. As demonstrated by Marchi et al. (2012), optimizing the control of a pumping system can reduce the system efficiency of a water delivery scenario when decreasing the pump output using VSDs. As a result, the pump is running more hours with lower system efficiency to deliver the same amount of water to the reservoir, but the control is more energy efficient because less energy is wasted to dynamic head. This can be often demonstrated by calculating the specific energy use as shown in equation 2.13

Even though it has been shown that there is a major potential for achieving energy saving when using variable speed controlled pumps, a significant number of parallel pumping stations is still operated with fixed-speed pumps. Savings in the operational costs of fixed-speed pumps is therefore often striven with load-shifting of parallel pumps, in which the emphasis of operations is shifted from high energy demand periods to low demand periods.

Besides pump scheduling, energy efficient operation of parallel pumping station can be obtained using control schemes based on variable speed use of pumps. In fact, if only energy efficiency issues are considered, the optimal control of pumps is often achieved when all parallel pumping units are equipped with VSDs (Szychta, 2004b).

2.2.1 Optimized control using load-shifting of parallel pumps

The load-shifting of parallel pumps has been studied broadly, and suggestions for the optimized control in fixed-speed pumping systems have been presented for example for the municipal and industrial water supply and waste water treatment sector (Tang and Zhang, 2010; van Staden et al., 2011; H.Zhang et al., 2012; Harding, 2012; Zhuan and Xia, 2013).

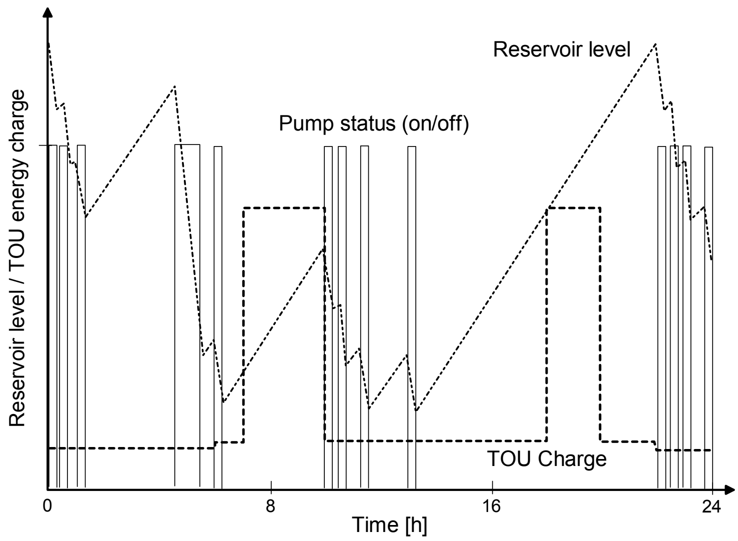

Figure 2.15. Optimized load-shifting control based on TOU and MD tariffs. The pump units can be started and stopped according to energy tariffs, reservoir level, and estimated future output. (van Staden et al., 2011).

The load-shifting control of parallel pumps can mean that parallel pumps are scheduled to operate according the valid energy tariffs such as TOU (time-of-use) and MD (maximum demand) (van Staden et al., 2011; H.Zhang et al., 2012) and/or according to suitable system conditions. For instance, if there is a change in the static head of the system, it can be more energy efficient to schedule the emphasis of the pumping in the time periods in which the static head is relatively low (Tang and Zhang, 2010; Harding, 2012). Figure 2.15 illustrates an example on the operation of the pump units according to energy tariffs. In this example, the TOU energy charge is the highest between daily hours of 7−11 and 18−20. Correspondingly, the lowest TOU charge is between daily hours of 22−6. During high demand periods, for example the daily hours from 7 to 11, a certain number of parallel pumps is switched off and started again when the lower demand period starts (Figure 2.15). The number of operating pumps can also be set based on the total power, so that the maximum power demand (MD) will not be exceeded.

Shifting the operation of parallel pumps from the peak demand to low demand can naturally decrease the energy costs of pumping, but it typically requires large reservoirs to stop the operations during high electricity price periods. Also, when using load-shifting control, the presumption is that it is possible to estimate the future demand of the process with justified accuracy.

As mentioned, the benefits of using load-shifting control with fixed-speed pumps instead of optimized variable speed control should be evaluated based on process needs and system details, since sometimes the system characteristics do not support the idea of variable speed control in pumping (Harding, 2012). Also, the extra investment costs of equipping each parallel pumping unit with VSD may direct the pump user's interest towards the fixed-speed operation of parallel pumps (H.Zhang et al., 2012).

2.2.2 Optimizing system efficiency in VSD controlled parallel pumping

Besides enabling energy efficient flow adjustment, the energy-saving potential of using VSD controlled parallel pumps is caused by the fact that the majority of the pumping stations is not properly designed in agreement with the process needs and actual operational conditions (Hovstadius et al., 2005; Moreno et al., 2007; Kaya et al., 2008; Pemberton and Bachmann, 2010). Having multiple VSD controlled pumping units allows that the output of the pumping system can be set to the desired level in a very broad operational range with high energy efficiency.

In studies related to the optimal control schemes in VSD controlled parallel pumping systems, the energy efficient operation is often indicated with system efficiency (Szychta, 2004b; Bortoni et al., 2008; Yang and Borsting, 2010; Zhao et al., 2012). Typically, the output of the parallel pumps in varying conditions is modelled using pump characteristic curves and affinity rules. By monitoring the output, the most suitable combination of pumps is then chosen based on the system efficiency in each operation point. An example of the control strategy implementation in a pumping system is illustrated in Figure 2.16.

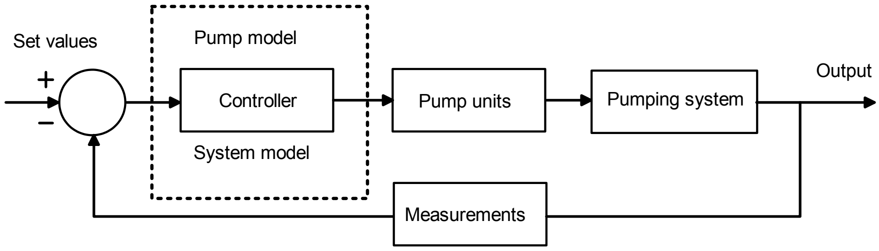

Figure 2.16. A simplified example of a control loop in a pumping system. The optimization is based on the modelling of the pump operation in varying system conditions. The control algorithm gives the reference to pump units, and the output to the system is monitored with separate measurements (Pemberton, 2003; Ma and Wang, 2009).

In the study by Bortoni et al. (2008), the optimal rotational speed for parallel pumps for example in water treatment processes is predicted using a mathematical-optimization-based tool. The optimization is conducted based on the obtaining of minimum power usage to fulfil the required discharge (Q, H), respectively. The mathematical-optimization tool searches for the optimal flow allocation for each pump to meet the total discharge required by the process. The given options are switching the pumps on or off and adjusting the speed of each pump. The optimization tool is based on the Bellman principle of optimality, which in this context suggests that the minimum power usage at the desired discharge is found when the location of the operation point of each pump is as close as possible to the rated value. Hence, the optimization tool aims to minimize the pump power usage by determining the shortest distance to the rated operating point, which is usually equal to the pump's Best Efficiency Point (BEP).

A method for increasing energy efficiency in multi-pump boosting systems is discussed in the study by Yang and Borsting (2010). Also in this solution, the output of the pump is modelled in varying conditions using pump characteristics and affinity laws. The suggested method calculates the optimal number of pumps and optimal speed for each operating pump at the desired level of output. Since the method concerns water delivery in boosting systems, the suggested control aims to deliver the required head to consumers with optimized energy consumption. Energy efficient control strategy for enabling the required head in municipal water delivery using multiple VSD controlled pumps has also been discussed in the study by Szychta (2004b). This study presents a method for determining the optimal rotational speed and justified range of operating speeds to maintain acceptable efficiency (Szychta, 2004b).

The control strategies for multiple VSD controlled pumps in HVAC (Heating, Ventilation and Air Conditioning) systems have been studied for example by Ma and Wang (2009) and Zhao et al. (2012). The control method studied by Zhao et al. (2012) focuses on the application of parallel VSD controlled pumps in a pressure-control loop of a central air-conditioning system. In this case, the pressure-control refers to the situation in which the speed of the operated pumps is adjusted to maintain the pressure at the required set point. In the observed closed-loop pumping task, there is no static head in the system, but the system curve varies according to the supported process. The suggested method uses the extreme value analysis to investigate the optimal combination of speed ratios and number of operating pumps depending on the automatically calculated operational zones for a single pump. According to Zhao et al. (2012), the optimal state is achieved when the desired flow rate is delivered with the best system efficiency, which is again determined using a model for the pump performance in varying conditions and actual output monitoring.

Energy efficient control strategy of variable speed parallel pumps in a waste water treatment plant is discussed in the study by Zhang et al. (2012). In this method, neural network algorithm is utilized to develop data-driven models for predicting the energy consumption and waste water flow rate in different configurations for the observed parallel pumps. Instead of using pump characteristics and system modelling, the ideal starting and stopping points for each pump and the rotational speed are calculated based on the collected operational data.

2.2.3 Challenges for implementing optimal control strategy

As mentioned earlier, the required information for optimizing the control of a VSD controlled parallel pumping system is typically obtained by modelling the pump and the system characteristics in varying operation conditions (Bortoni et al., 2008; Yang and Borsting, 2010; Zhao et al., 2012). The pump model is often determined using pump performance curves and affinity laws shown in equations (2.7)–(2.9). Correspondingly, the system model is often based on equation (2.1). The referred studies show that these models can be utilized to estimate the operation of a VSD controlled pumping system in a great number of system scenarios and pumping schemes. The possible limitations of the used models may result for example from unexpected differences in the system characteristics or inaccuracies of pump performance curves. Also, estimating the behaviour of the entire pump drive train in varying conditions can be a difficult task with the available data.

The required head for a VSD controlled pump is often illustrated with a system curve illustrating a constant amount of static head and dynamic head, which is dependent on the flow velocity as shown in equation (2.6). In this case, the only required information to solve the system characteristics are the static head H_{st} and the coefficient k for friction losses, although the identifying typically requires the start-up measurements or other operational data (Bortoni et al., 2008; Yang and Borsting, 2010). As mentioned by Yang and Borsting (2010), the assumption of constant and stabile system curve can be misleading since it is not true in many practical solutions. The shape of the system curve can vary for example because of a slow increase in friction losses of the piping sections and changes in the supporting process. There may also be significant changes in the static head of the system for example in reservoir filling pumping tasks. Also, in booster pumping cases, the pressure of the inlet section may vary between certain tolerances. Correspondingly, the tolerances between the actual and published pump characteristic curves (ISO 9906, 2012) can reduce the accuracy of the pump model. It should also be noted that the affinity laws used in many pump models are based on the assumption that the pump efficiency remains constant regardless of the pump rotational speed. The relation between the pump speed and pump efficiency in the pumping system operation has been studied for example by Szychta (2004a), Muszynski (2010), and Marchi et al. (2012). According to Marchi et al. (2012), the affinity laws are usually still justified, if the rotational speed of the pump changes less than 33% of the rated value.

The impact of the pumping system model inaccuracies on the pump control is usually reduced with a feedback-signal (Yang and Borsting, 2010; Zhao et al., 2012). This is typically applied by using the output monitoring of the pumping system, for instance by monitoring the total flow rate and output head. However, the suggested parallel pumping models may be applicable only if all parallel pump units are identical and there is no difference between the system characteristics of individual pump units (Yang and Borsting, 2010; Zhao et al., 2012). In the case of dissimilarities between parallel pumping units, the optimized control may require additional information on the behaviour of an individual pump operation point. Alternatively, start-up measurements may be used to gather the information on the pumping system for the optimized control (Bortoni et al., 2008; Zhang et al., 2012). In the control study presented by Zhang et al., (2012), the optimized operation of VSD controlled pumps in a waste water treatment was based on gathered operational data, thus eliminating the possible inaccuracies in pump performance curves. Naturally, this can enable an accurate basis for both optimal scheduling and speed control of parallel pumps in the observed process plant, but start-up measurements may not be feasible in every pumping scenario, or they may have to be repeated if there are changes in the pumping system.

2.2.4 Motor and drive effectiveness in variable speed pumping

Although the pump efficiency is often highlighted, the efficiencies of the motor (η_{motor}) and variable speed drive (η_{VSD}) also affect the combined system efficiency of pumping as shown in equation 2.12. Modelling the behaviour of the combined motor and VSD efficiency in variable speed pumping can be yet difficult (Bernier and Bourret, 1999; Ferreira et al., 2011; Marchi et al., 2012). In practice, the efficiency of the motor and drive varies according to the motor rotational speed and torque, but the behaviour of the combined efficiency of motor and VSD is also a sum of many system factors related to for example motor type, size, and control strategy (Bernier and Bourret, 1999; Kaya et al., 2008; Ferreira et al., 2011).

Typically, the efficiency of an induction motor is equal to the maximum when the motor is operated at loading values of 75% or higher of the rated value, but the efficiency may drop radically if the loading values are well below 50% of the rated value (Kaya et al., 2008). The impact of varying load and frequency on the motor efficiency is illustrated in Figure 2.17.

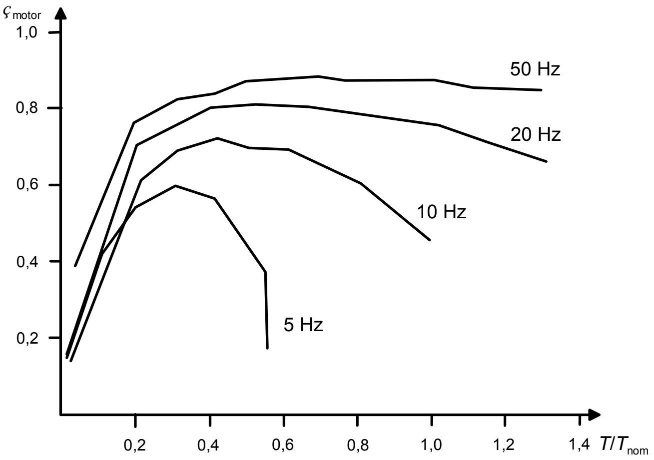

Figure 2.17. Efficiency of a 7.5 kW induction motor as a function of relative torque. Usually induction motors can maintain high efficiencies in a wide range of relative loads at the rated frequency (Lukasik and Szychta, 2007).

The losses of the electric motor can be roughly categorized to the resistance and iron losses of the rotor and stator and the ventilation and friction losses in the rotor (de Almeida et al., 2005; Stockman et al., 2010; de Almeida et al., 2011). The use of VSD affects negatively the efficiency of the motor by causing additional losses, but modern VSDs are often capable of maximizing the combined efficiency of the motor and VSD by means of proper magnetizing flux regulation, thus compensating the added losses to the drive train (Ebrahim et al., 2010; Ferreira et al., 2011).

Although a standardised method is not available, the combined efficiency of the motor and VSD has been studied to some extent (Stockman et al., 2010; Aarniovuori et al., 2012). In the study by Stockman et al. (2010), several measured Iso-Efficiency maps were presented for different motor-drive combinations. Using Iso-Efficiency curves or maps it is possible to determine the impact of the motor and VSD losses on the system efficiency of a VSD controlled pumping scenario. An example of the combined efficiency of a 7.5 kW Siemens induction motor and ABB ACS 850 industrial drive at varying motor load and frequency is illustrated in Figure 2.18. The efficiency of the motor-VSD unit in this case was determined using direct measuring of input and output power.

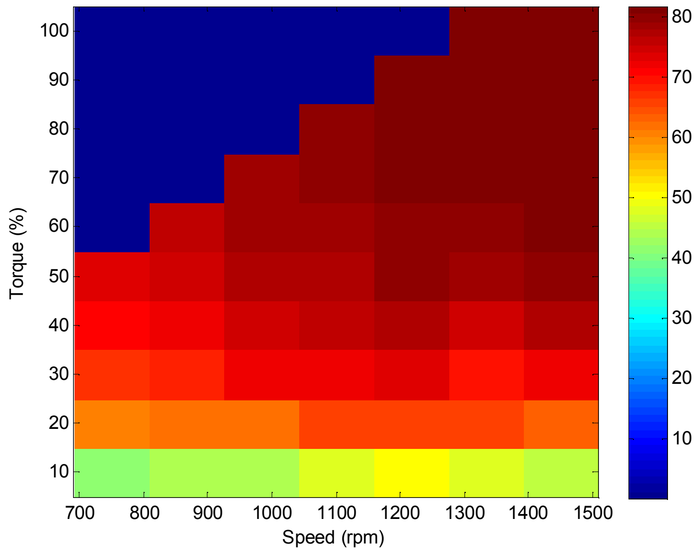

Figure 2.18. Combined efficiency of a Siemens 7.5 kW Siemens induction motor driven by ABB ACS 850 industrial drive at varying load and rotational speed. The efficiency was determined by measuring the input and output power using power analyzers and shaft torque transducer. The load was adjusted using a DC-load machine between 10–100% of nominal torque (50 Nm).

According to Lukasik and Szychta (2007), Stockman et al. (2010), and Gong et al. (2012), a typical induction motor and VSD combination used in a pumping system can achieve a high efficiency in a broad torque range. In general, motors having a higher efficiency class, including permanent magnet and synchronous reluctance motors can also enable better efficiency characteristics in a low load and rotational speed range (Kaya et al., 2008; Stockman et al., 2010; Saidur, 2010; Marchi et al., 2012).

From a process optimization point of view, relative changes in the system efficiency are more relevant than the exact values for the efficiency in variable-load process scenarios. In many instances, the efficiency of the motor and drive is ignored, or they are considered as constant values regardless of the pump operation point when modelling the VSD pumping system for optimized pump control (Carlson, 2000; Ma and Wang, 2009; Yang and Borsting, 2010; Zhao et al., 2012). According to Zhao et al. (2012), the efficiency variation of the motor and VSD is affected by the relative rotational speed n_{rel} which is the ratio of the operating speed and the rated speed. According to Zhao et al. (2012), if $0.4 < n_{\mathrm{rel}} < 1$, both η_{motor} and η_{VSD} are considered less affected by n_{rel}. Correspondingly, if $n_{\mathrm{rel}} < 0.4$, a larger impact on the system efficiency can be expected, but it is also considered that 0.4 can be set to the minimum value for n_{rel}, and that the system characteristics in the studied case (closed-loop HVAC parallel pumping task with typically flat *QH* performance curves) support the assumption of a small difference between the parallel-multiple pump and single-pump speed ratios (Zhao et al., 2012). Hence, the variation of η_{motor} and η_{VSD} is neglected, and the optimization target is simplified to pumping efficiency.

Similar considerations are also given in water delivery parallel pumping cases. According to Szychta (2004a), the frequency of a pump drive train in the water delivery systems of Europe is typically between 35–50 Hz, and operating a VSD controlled pump between these limits would rarely cause more than a 5% relative change in η_{motor} of a voltage feeding induction motor. However, a method for estimating η_{motor} in a pump drive train is proposed in the study by Lukasik and Szychta (2007), where the effect of varying η_{motor} is included in pump modelling. Despite this, it should be taken into account that in many instances, the over sizing of motors and drives may increase the risk of lowered system efficiency in low load situations (Bernier and Bourret, 1999; Kaya et al., 2008; Saidur, 2010; Bortoni, 2009; Ferreira and de Almeida, 2012). From this point of view, it is justified to estimate also the combined efficiency of the motor and VSD at each pump operation point (Marchi et al., 2012).

2.2.5 Control strategy based on specific energy consumption

In previous sections, methods suitable for controlling parallel pumping systems based on the optimized system efficiency were studied. Optimizing the pump output according to the system efficiency is justified especially in closed-loop pumping systems and systems without static head (e.g. HVAC systems) since the energy efficiency is dependent on the friction losses in piping and the system efficiency of the pump drive train. In the pumping systems with static head, specific energy consumption can offer a justified base to determine the energy efficiency of the operation. For instance, the system efficiency of a pump unit may be maximized in situations in which the pump is operating near the BEP, but feasible and more energy efficient operation conditions can be obtained at lower output levels in terms of specific energy consumption as shown in Figure 2.11 and Figure 2.13. Thus, the pumping task may be fulfilled with higher energy efficiency, even though the system efficiency would decrease notably.

A simple option to determine the specific energy consumption in a pumping system is to monitor the flow rate and input power of the pump drive train. Therefore, a justified solution to control such a system would be a control strategy which aims to minimize the specific energy consumption E_{s} based on the monitored output. An example of a simple method to optimize the specific energy consumption of a pumping system with a step-wise manner is introduced by Steger and Pierce (2011). In this method, the pump units are controlled based on the instantaneous E_{s} in the system. If the monitored E_{s} in the system is different to the previous one, different control options (starting/stopping pumps, rotational speed change) can be executed to determine the operation state that results in a lower E_{s}.

The benefit of optimizing the specific energy in pumping is that for example the losses caused by the motor and drive and also the effect of increased friction losses in piping can be taken into account without modelling since only a simple feedback system is required to determine whether the achieved operation point is more energy efficient than the previous. Hence, the possibly inefficient operation caused by oversized pump system components at a low load situation can be avoided. In the suggested step-wise strategy (Steger and Pierce, 2011), the required data from the system is limited to minimum since the optimization requires only feedback signals from the process.

When optimizing a parallel pumping system, the right moment to start and stop new additional pumps has to be determined (Yang and Borsting, 2010). In practice, the control strategy suggested by Steger and Pierce (2011) is justified when optimizing the energy efficiency of the operating pumping units, but additional control logic may be required to determine the optimal number of operating pumps in a parallel pumping system. Another challenge of implementing such a control strategy to a parallel pumping system, or any other control strategy based on the monitored output, is to offer adequate metering of the pumping system output. As pointed out by Steger and Pierce (2011), typically the VSDs supplying the motors in a pumping system are capable to determine the input power to the pump unit. However, additional flow metering is often required to determine the total flow rate of the system, and monitoring the flow rate of each parallel connected pump drive train can be even more difficult to attain.

2.3 Applying VSDs in pump operation point monitoring

The location of the operation point is essential information when evaluating the effectiveness of the pump in variable-flow conditions. Measuring the pump flow, total head, and power consumption using field testing gives the pump user a chance to create a set of pump curves in the observed system. Correspondingly, the on-line monitoring of the pump output can give valuable information related to the pumping system effectiveness since even simple electric and hydraulic measurements at the pumping stations can improve the maintenance and identify possible inefficient operations (U.S. Department of Energy, Hydraulic Institute, 2006; Moreno et al., 2007; Lamaddalena and Khila, 2013). As discussed in the previous section, also the optimal control of VSD controlled parallel pumping systems often requires on-line monitoring of the system output.

In general, the best conditions for field tests in pumping systems are obtained if the pressure and velocity distributions in the measuring sections are uniform. The recommendations for accurate measurements, for example the minimum distance of straight pipeline between the measuring points, can be difficult to meet in many pumping system cases. Also, the required investments for accurate field tests, especially for flow metering, can reduce the pump user's interest on pump monitoring with traditional measurements. Since in many occasions the direct measuring of the flow rate of the pump is not provided due to unwanted investment costs or system limitations, methods to determine the pump flow with indirect methods have been developed.

2.3.1 Measuring the operation point in a parallel pumping system

The field test for measuring the pump operation can be conducted with various methods. There can also be a notable difference in the accuracy, cost, and complexity of the methods as many are also useful for a specific range of flows or for certain types of liquid (Volk, 2005). Typical methods for flow rate measuring in the water supply and industrial pumping sector can be the venturi, orifice plate, pitot tube, and magnetic flow meter, all of which are installed directly in the piping. Portable ultra-sonic meters are also often used to determine the flow in numerous locations. The disadvantage of the methods in which the measuring equipment is mounted inside the pipe is the possible inaccuracy due to wear or contamination in the piping system.

The total head of the pump is commonly measured using pressure gauges or differential pressure transmitters. Depending on the system, the total head may be determined by measuring the inlet head and outlet head separately or measuring the differential pressure between the inlet and outlet sections of the pump and adding the difference in the velocity head (Sulzer, 1998). An example of the measuring of the pump output using pressure transmitters and flow meter installation in a system is shown in Figure 2.19.

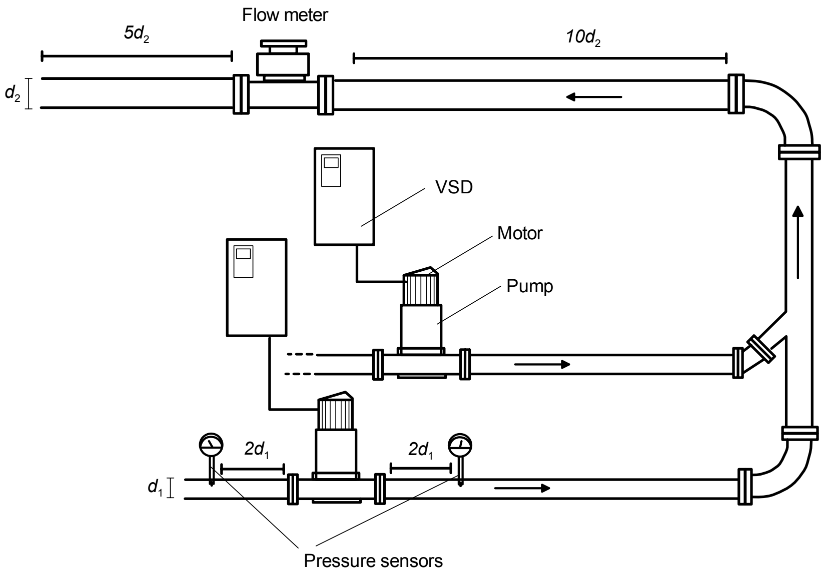

Figure 2.19. Measuring the pump output using common field test equipment. A flow meter is installed into the common outlet piping having a straight section of 10 pipe diameters (d_2) upstream and 5 diameters downstream. Pressure sensors are installed in the distance of two pipe diameters (d_1) from the pump inlet and outlet flange. (ISO 9906, 2012)

To determine the output in parallel pumping systems, the individual flow rates of each pump is usually not measured since the investment cost would be multiplied according to the number of parallel pumps. In theory, this problem could be solved with portable flow metering. In practice, however, the requirement for a uniform velocity distribution for example in the case of ultrasonic flow meters can be limited by valves, bends, and fittings in individual piping parts of a parallel pumping system (U.S. Department of Energy, Hydraulic Institute, 2006). For justified flow metering, it is recommended that the piping upstream and downstream from the flow meter should be straight and have the same diameter as the flow meter and a length at least 10 times the pipe diameter to the upstream direction and 5 times the pipe diameter to the downstream direction (ISO 9906, 2012). These requirements are often not achievable in the individual parts of the parallel pumps and can be difficult to be met even in the common outlet piping section.

The field test for measuring the pump total head can be seen more accurate than the flow metering with practically attainable methods (Sulzer, 1998). Also, the investment cost of the pressure metering is typically lower compared to the costs of most flow meters. Thus, implementing pressure sensors for each pump in a parallel pumping system is clearly more sensible than acquiring flow meters for the individual pumps.

In addition to flow rate and pressure, power consumption, rotational speed, and NPSH are often important variables for a pump user. The pump power input can be measured using for example dynamometers or torque meter. NPSH conditions can be determined with measuring the $NPSH_a$ at the suction of an operating pump. The measured $NPSH_a$ can be compared with the $NPSH_r$ published by the pump manufacturer. In practice, the pump power consumption is often observed with electric power measurements from the motor coupled to the pump. To get accurate results for the pump power in electric power metering, the efficiency of the motor should be determined. In many fixed speed pumping systems, the pump users may observe only the electric power consumption instead of determining the pump power consumption. Since the most typical method to implement rotational speed control in pumping systems nowadays is using the variable speed drives, the pump user can obtain power consumption and rotational speed values directly from the VSD's estimates. NPSH measurements are seldom conducted in field due to the complexity of the test setup and difficulties in the accurate measuring of the required parameters. They are usually used only in situations in which cavitation is suspected. (Volk, 2005; Sulzer, 1998)

2.3.2 Basic methods for monitoring the flow rate with VSDs

In situations in which field tests or measurements are not available, the flow rate of the pump unit can be determined with model-based estimation methods. The pump model is typically composed of liquid properties, pump characteristic curves, and rotational speed signal, allowing a number of new pump curves to be generated using the affinity laws. The characteristic curves at nominal rotational speed can be expressed with a selected number of points from the curve. Thus, the flow rate of the pump can be interpolated from the transformed curves based on the head or power signal.

Figure 2.20 presents the estimation of the pump flow rate using either the head or power signal and the pump model. The head of the pump (H) can be measured using the pressure measurement across the pump. In variable speed pumping systems, the pump axial power P_{pump} and the rotational speed signal n can be acquired from the VSD (Nash, 1997; Ahonen, 2011). In this study, the method using the head or pressure signal for flow estimation is referred to as the *QH method*. Correspondingly, the method using the pump power signal for flow estimation is referred to as the *QP method*.

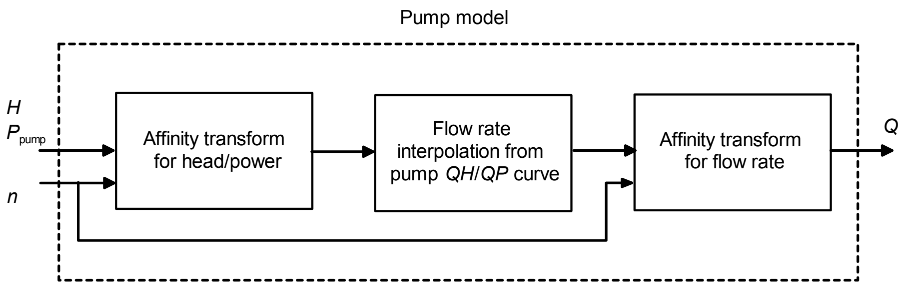

Figure 2.20. Model-based pump flow metering. The flow rate can be determined based on the pump head or power signal. The model can generate a series of characteristics curves using the measured rotational speed and affinity laws. The flow rate can be interpolated from the transformed *QH* or *QP* curve (Ahonen, 2011).

As shown in Figure 2.20, both *QH* and *QP* can determine the flow rate with similar steps: first the input signal is transformed into the rated speed with the affinity laws. After the affinity transform, the flow rate can be interpolated from the characteristic curves. The calculated value for the flow rate is then transformed to match the actual rotational speed. The same estimation procedure is illustrated in Figure 2.21, which plots the *QH* curves (Figure 2.21 a) and *QP* curves (Figure 2.21 b) at the actual rotational speed and rated speed. The flow rate can be calculated using the head signal (H_{meas}) or the power signal (P_{meas}). For example, if the head signal value H_{meas} is 11 m as shown in Figure 2.19 a, the value after affinity transform would be 16.8 m. The corresponding flow rate at the nominal *QH* curve would then be 27.5 l/s. After another affinity transform, the *QH* method can then return the actual estimated flow rate $Q_{est,\,QH}$ of 22.8 l/s (Figure 2.19 a).

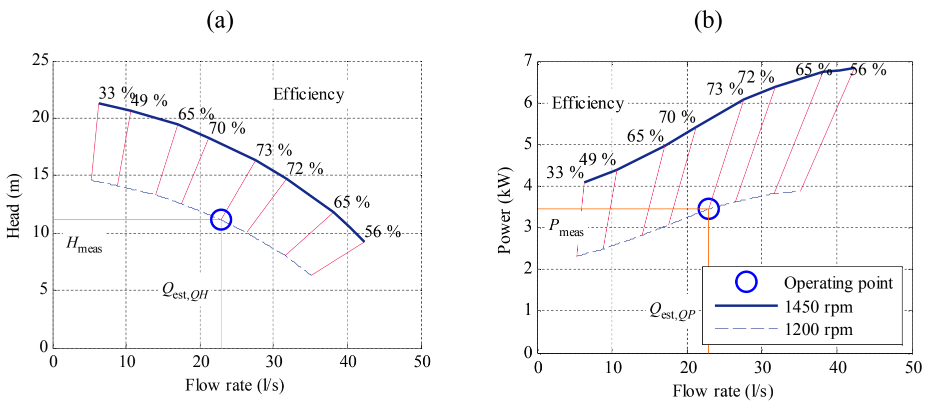

Figure 2.21. Estimating the pump flow based on the measured head (a) or power (b).

Since the pump model is based on the pump characteristic curves and liquid properties, the applicability of the presented methods is clearly connected to the exactness of this data. In addition to the pump model, also the accuracy of the input signals affects the uncertainty of the model-based flow metering. The functionality of the basic model-based flow metering methods has been discussed already by Liu (2002), Wang and Liu (2005), Hammo and Viholainen (2005), and Ahonen (2011).

The pump model based on the characteristic curves and affinity laws is widely utilized to determine the pump performance in varying operation conditions (Bortoni et al., 2008; Yang and Borsting, 2010). The affinity laws are based on the assumption that the efficiency of an operating pump stays constant regardless of the rotational speed. Usually, it is not taken into consideration that this is not exactly true since the change in the rotational speed also affects the pumping efficiency (Szychta, 2004a; Muszynski, 2010). This change, however, is often considered irrelevant if the rotational speed differs less than 20–33% from the rated rotational speed n_0 (Muszynski, 2010; Marchi et al., 2012). Correspondingly, the accuracy of the pump model can be reduced in situations in which the rotational speed is adjusted more drastically. An important factor related to the accuracy of the model-based methods is the accuracy of the characteristic curves at the rated speed. The curves given by the pump manufactures are typically measured according to grade 2 in the ISO 9906 standard, which gives tolerances 3.5%, 4%, and 2% for the total head and flow rate, for the pump power input, and for the rotational speed, respectively (ISO 9906, 2012). In practice, the pump performance may also be affected by the wear of the pump or other unexpected system settings, which can also cause the difference between the performance curves and actual operation (Zhang et al., 2012).

Another major limiting factor in the use of the basic *QH* and *QP* methods is related to the shape of the characteristic curves, since the methods may not be applicable in the case of too flat curves. If the applied characteristic curve is flat, even a small error in the model or the input can have a significant effect on the estimated operating point. In some situations, the characteristic curve can also be non-monotonic resulting in several flow rate values that match the same head or power signal, and therefore, the interpolation cannot be used.

2.3.3 Alternatives and improvements in model-based pump operation point monitoring

Besides the basic *QH* and QP methods, some other alternatives and improvements in the model-based estimation have been suggested to determine the operational state of the VSD controlled pump in a system. A method suitable for improving the accuracy and applicability of the basic *QP* method has been studied by Ahonen et al. (2012). In the suggested method, the *QP* curve-based method is used to identify the system curve near the pump nominal rotational speed. The identified system curve can then be used to determine the flow rate in the operation range in which the *QP* method is not accurate. This identification phase is illustrated in Figure 2.22, which plots the resulted *QH* curves of the pump in the operation region in which the basic *QP* method can be used. Using the estimated operation points based on the *QP* method results at the rotational speeds (n_1, $n_2 \ldots n_6$), the values for the static head H_{st} and the coefficient k in equation (2.6) can be solved.

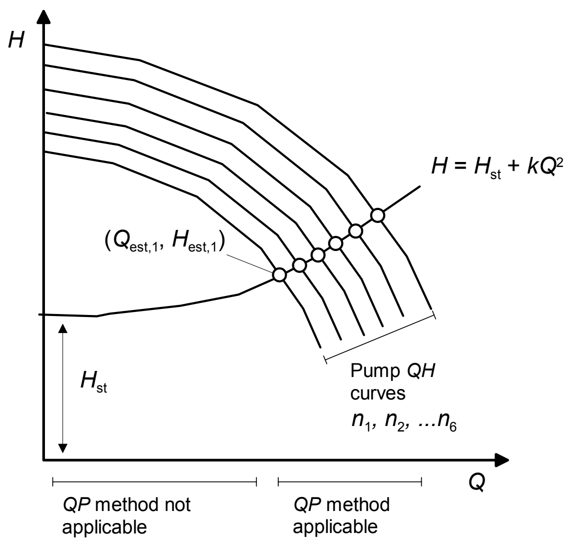

Figure 2.22. Identifying the system curve using the *QP* curve-based operation point estimation. The *QP* method is used to determine a set of operation points at different rotational speeds (n_1, $n_2 \ldots n_6$). The resulting operation points are utilized to solve the values for H_{st} and k (Ahonen et al., 2012).

The suggested method to determine the pump operational state with VSD in the study by Ahonen et al. (2012), the *Hybrid method*, has shown to improve the accuracy of the operating point estimation at low rotational speeds. Partly, this happens because the method is able to reduce the occurring error in the affinity laws caused by a substantial change in the rotational speed. Still, the usability of the Hybrid method is limited in a similar way as the basic *QP* method: if the pump *QP* characteristic curve is inaccurate or unsuitable for the interpolation at the observed operating range, the system curve cannot be identified accurately. The method is also applicable only in systems, where the system curve can be assumed constant and stabile.

In a patent by Kernan et al. (2011), the accuracy of the pump *QP* characteristic curve is improved with a closed valve test. This test can identify the actual pump shut-off power at different sets of rotational speeds and eliminate the impact of possibly inaccurate affinity transforms for *QP* curve.

The drawback in the suggested *Kernan method* is the requirement to operate the pump in abnormal operation conditions which can harm the pump. Moreover, because the pump *QP* curve is only verified in a zero-flow situation, the operation at the rest of the flow range remains still unknown. Alternatively, the Kernan method can be utilized with pressure measurements. Thus, the *Kernan-QH method* uses the shut-off pressure measurements instead of measuring the power in the shut-off conditions to correct the *QH* curve set at variable rotational speeds. Despite this, there can be system scenarios in which the interpolation results are not accurate because of flat or non-monotonic characteristic curves.

3 Selected methods and the main results of research focus areas

The selected methods suitable for studying the research questions are discussed in this section. The focus of this thesis is divided into three main research areas. Because of this arrangement, the selected methods and the main results of each focus area are presented in this same section. The focus areas are further divided into article-based studies, presenting the methods and the most relevant results of each publication.

3.1 Arranging the studies into focus areas

The main focus of this thesis is limited to parallel pumping systems having certain typical characteristics:

- Parallel pumps are delivering the output into common outlet pipeline
- The parallel use of pumps is always required to deliver the maximum output of the process
- The parallel use of the pumps in not necessarily required to deliver the minimum output of the process
- Either the process conditions or the system characteristics may alter, as a result of which the system curve of the process is not completely stabile
- A notable part of the total head which the parallel pumps must overcome is caused by the static head of the system

The aim of the introduced methods is to ensure the energy-efficient operation of parallel pumps without exposing the pumps to events which can cause a significant decrease in the reliability of the pumping. The emphasis of the suggested methods is to offer such control in parallel pumping systems in which all parallel pumps are equipped with VSDs. Suitable applications for the methods can be found for example in systems in which the previous control method of the pumping system is replaced with the rotational speed control of pumps.

The first focus area considers the suitable operation conditions for VSD controlled parallel pumps. The selected methods are based on the known relation between the operation point of the pump and the energy-efficiency of the pumping. In this focus area, the energy saving potential in VSD controlled parallel pumping, the energy-efficiency-based suitable operation region for the VSD controlled pump, and the benefits of avoiding the unwanted operation of parallel pumps when selecting the control scheme, are studied.

The second focus area presents the methods suitable for the operation point monitoring of the parallel pumps. The selected methods are based on the modelling of the pump performance in varying conditions with the affinity laws and the estimation of the output of the pump using pump performance curves and internal estimates of the VSD. These model-based methods using indirect monitoring can be applied in pumping systems without additional sensors or with pressure sensors to determine the head of the pump more accurately. In this focus area, the basic options for monitoring the flow rate of each parallel pump in a system, the sensorless methods for monitoring the operation point of a pump in a system, and the possibilities of combining the model-based estimation methods to achieve more accurate pump operation monitoring are studied.

In the third focus area, the methods aiming for the suggested energy efficient control are proposed. The introduced methods are based on the operation point estimation of the pumps applying VSDs in the pumping systems. In this focus area, a method for energy efficient and reliable control strategy for VSD controlled parallel pumps is studied.

3.2 Suitable operation conditions for variable speed driven parallel pumps

The benefits of the rotational speed control are directly related to the resulting operation points of the controlled pump. As mentioned in section 2.1, the energy saving possibilities when replacing valve controlled pumping system with a VSD controlled system are widely recognized. Realizing the energy savings in variable speed controlled pumping requires the observation of all process components: pumps, motors, drives, and system conditions. Naturally, this makes the study on a parallel pumping system a more complex case compared with single pump unit applications.

The control type of the pumping system is generally dictated by the process demands, but there can be several different options to fulfil the requirements (Hovstadius et al., 2005; Zhao et al., 2012). This is very true especially in parallel pumping systems in which the number of options to deliver the desired output is increased also by the number of additional pumps. Energy efficient operation can thus be achieved if the required output is delivered with minimum energy usage.

The focus of this section is to study the suitable operation conditions for VSD controlled parallel pumps. First, the energy saving potential in a parallel pumping system is demonstrated with example calculations. In the calculations, the energy use and efficiency of the whole pumping system are studied. The energy efficiency-based recommended operation range for a VSD controlled pump is discussed in the second research study. In this case, the selected criterion for energy efficiency is based on the specific energy consumption of the pump unit. In the third research study, the benefits of avoiding unwanted operation conditions in VSD controlled parallel pumping are studied.

3.2.1 Example of energy-saving possibility in variable speed driven parallel pumping system (Publication I)

The energy efficiency improvements in parallel pumping systems using variable speed drives were studied in a real-life parallel pumping system comprising of two surface condenser pumps in a forest industry power plant. In this research case, the energy-saving potential of replacing valve control with rotational speed control of pumps was calculated.

In the example case, the surface condenser pumps of the plant are used to pump cooling water from an open water tank to the condenser. The two identical parallel pumps are driven by 315 kW motors, and the delivered flow is adjusted using a throttling valve. The static head of the system is approximately 32 meters. The 700 mm piping from the tank located under the cooling towers leads to the condenser. The approximated length of the pipeline is 170 meters.

At the current state, water is delivered to the condenser using both parallel pumps at the nominal speed. The flow rate of the system is adjusted with a control valve near the condenser. The studied alternatives for the valve control are the rotational speed control of only one pump and the rotational speed control of both pumps. The energy consumption of the example system was evaluated using a simulation model shown in the study by Viholainen et al. (2009). The annual energy consumption in each adjustment case was calculated using simulated drive input power values, including the drive and motor losses, and the approximated duration curve.

As mentioned in the first chapter of this thesis, the findings related to this section contain additional calculations and visualizations. In this particular case, the presented findings are based on the updated approximation of the duration curve and the motor and the VSD efficiency in varying operation conditions. Thus, the updated values have also affected the results of the calculated energy-saving potential.

In the new calculations, the operating hours of the pumping system and the estimated flow range is based on monitored operating values between January 1st 2007 and December 31st 2007. This data was not available during the original calculations. Another modification to the original calculations is related to motor and drive train efficiency. The used efficiency matrix of the motor and the VSD in original calculations was based on the measurements of a 7.5 kW motor and variable speed drive. Since the efficiency of the 315 kW motor and variable speed drive can be presumed higher, new values for the combined efficiency were used.

3.2.2 Findings of Publication I

The estimated annual energy consumption of the pumping system with different control strategies can be calculated when the working hours of the pumps are known. The duration curve used in the calculations of this section is based on the approximated operation hours and delivered output in 2007 (Kortelainen, 2008). The estimation of the delivered annual output is shown in Table 3.1.

Table 3.1. The annual delivered output of the parallel-connected pumps in the surface condenser pumping system. Most of the time the required flow rate is between 1130–940 l/s. The pumps are out of operation approximately 340 hours during the year.

t (h)	*Q* (%)	*t* (%)	*Q* (l/s)	*V* (m^3)
40	100	0.5	1231	177244
350	96	4.0	1181	1488667
1230	92	14.0	1132	5012933
1250	88	14.3	1083	4872222
1450	84	16.6	1033	5394000
1550	80	17.7	984	5490444
1000	76	11.4	935	3364444
600	72	6.8	885	1912000
400	68	4.6	836	1203556
250	64	2.9	786	707778
250	60	2.9	737	663333
50	53	0.6	650	117000
0	47	0.0	580	0
0	32	0.0	400	0
340	0	3.9	0	0
total (h)			avg. (l/s)	total (mil.m^3)
8760		100	830	30.4

Table 3.1 shows that the annual pumped volume in this case is 30.4 million cubic meters and the flow rate can vary between 650–1230 l/s, but at 80% of the operation, the flow rate in the pumping system is 885–1130 l/s. The pumps are 340 h out of operation annually.

The characteristics of the pumping system and the emphasis of the simulated operation point locations are illustrated in Figure 3.1.

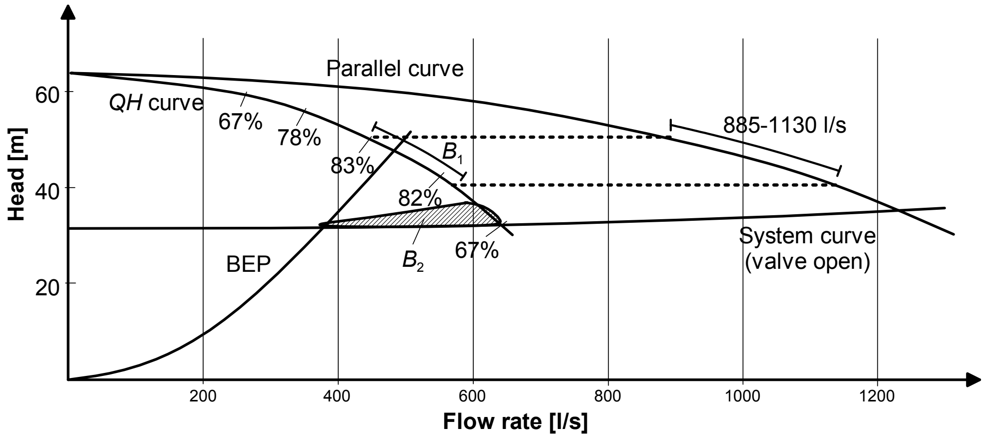

Figure 3.1. Characteristics of the surface condenser pumping system in the *QH* axis. The figure plots the pump *QH* curve and parallel curve for 1500 rpm rotational speed. The system curve, in a situation when the control valve is open, is also shown. Typically, the parallel pumps are operated between 885−1130 l/s as a result of which the emphasis of the operation in a valve controlled system is located in region B_1 on the pump *QH* curve for each pump. In a rotational speed controlled system, the emphasis can be found in region B_2.

Figure 3.1 shows the *QH* curve of a single pump and the parallel curve for two parallel-connected pumps. The system curve, when the control valve is fully open, is also shown in Figure 3.1. If the flow demand is typically between 885−1130 l/s, using valve adjustment in the system results locates the emphasis of the operation points for both pumps in region B_1 on the pump *QH* curve. Instead, if the flow is adjusted using rotational speed control, the emphasis of the operation points is found in region B_2. The location of the operation points in rotational speed control is affected by the selected control strategy: are the pumps running at the same rotational speed, is the first pump operated at fixed speed and the second one at reduced speed, etc.

The location of the operation point determines also the pumping efficiency. It can be seen from Figure 3.1 that in region B_1, in which the pumps are operated at fixed speed (1500 rpm) and valve adjustment is used, the pumping efficiency is approximately 81−83%. The pumping efficiency in region B_2 can be expected to vary between 67−83% according to the affinity laws. The main purpose of Figure 3.1 is to illustrate the indicative operation conditions of the parallel pumps in the *QH* axis when using different control schemes in this system.

To include the losses of the motor and the frequency converter in the case of rotational speed adjustment in the energy use, the efficiency of the drive train system was evaluated. As a rough approximation, the motor efficiency in fixed-speed operation when the output of the pumps is varied between 650−1230 l/s was estimated between 93.5−94.6%. Correspondingly, the drive train efficiency in variable speed operation in the same output range was estimated to vary between 84−92% due to lower motor load and drive losses.

The power use including the motor and drive losses as a function of the total flow rate was calculated using three different control strategies. In these control strategies, the total flow rate of the parallel-pumps was adjusted using either the control valve in the common pipeline or using rotational speed control with VSDs. The rotational speed control was studied with two different options: using speed regulation in the first pump (1070−1500 rpm) while the second pump was operated with a fixed speed (1500 rpm), and using speed regulation in both pumps with the same reference speed (1190-−1500 rpm). Figure 3.2 shows the calculated power use of the pumps when the flow rate is varied between 680-−1220 l/s.

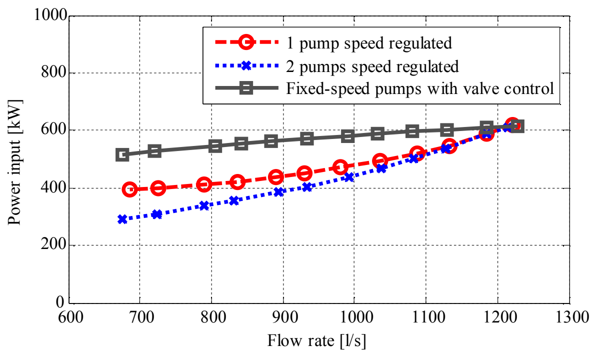

Figure 3.2. Total power use as a function of the total flow rate with different control scenarios.

Figure 3.2 shows that the lowest power use as a function of flow rate is achieved by using the rotational speed control of both pumps. Using one fixed speed pump and one speed regulated pump in this system also results in lower power usage compared with valve adjustment. Delivering the maximum flow rate (approx. 1220 l/s) requires that the pumps have to be operated at the nominal rotational speed (1500 rpm).

The energy efficiency of the control strategies in the observed system was studied in terms of the system efficiency and specific energy consumption. The results are shown in Figure 3.3 as a function of the total flow rate.

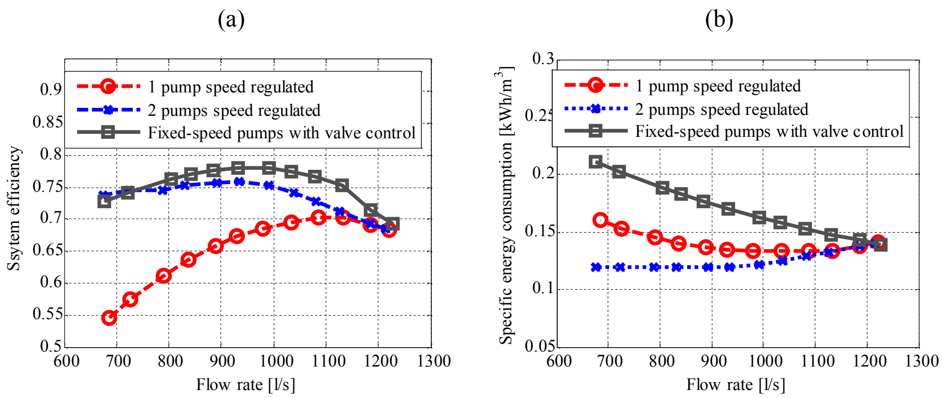

Figure 3.3. Simulated system efficiency (a) and specific energy consumption (b) of the surface condenser pumps when using valve adjustment or speed regulation of pumps. System efficiency when using valve adjustment is higher compared with speed regulation options. The energy efficiency of the speed regulation can be seen in the specific energy use which is lower in the case of rotational speed control than in valve adjustment when operating in the 675−1140 l/s range.

The calculated system efficiency of the parallel pumps shown in Figure 3.3 (a) suggests that the pumps are generating flow and pressure most efficiently when using valve adjustment. The system efficiency describes how the pump drive train transfers the electric energy into the liquid. Hence, operating pumps at the nominal speed and near BEP results in a higher combined efficiency of the drive train and pumping efficiency in the observed operating points compared with the system efficiency in the rotational speed control options. The energy efficiency of the rotational speed control options can be seen when observing the specific energy consumption of the pumps shown in Figure 3.3 (b). Rotational speed adjustment enables lower specific energy consumption compared with valve adjustment in this case, since less energy is wasted to the increased head. This can be seen from the operation point analysis shown in Figure 3.1. Thus, specific energy use when using rotational speed control is lower when operating below 1140 l/s flow rate. The difference in specific energy usage between two rotational speed control strategies results from the system efficiency divergence since in both cases, the pumps are operated nearly at the same head range. Operating the first pump at a fixed speed and the second with a reduced speed to deliver the observed flow range results in a different set of operation points compared with the situation in which both pumps are operated at the same speed reference. In other words, reducing the speed of both pumps shifts the operation point of each individual pump near the BEP curve shown in Figure 3.1, and if only one pump is speed regulated, the pump running at the fixed speed must still operate at the lower pump efficiency, caused by the operation point location at the right end of the pump QH curve in Figure 3.1.

Based on the calculated performance, the annual energy costs of different control strategies can be evaluated. The total energy consumption of the parallel pumping system using different control strategies is shown in Table 3.2.

Table 3.2. Energy consumption and costs of surface condenser pumps using different control strategies

	1 speed regulated pump	2 speed regulated pumps	valve adjustment
Energy consumption (MWh/a)	4101	3826	4903
Energy savings (MWh)	802	1078	
Energy savings (%)	16.4	22.0	
Energy costs* (€)	205073	191284	245169
Cost savings (€)	40096	53886	
*Estimated energy price is 50 €/MWh			

Based on the calculated performance and operating hours shown in Table 3.1, the total energy costs of the surface condenser pumps can be reduced 40000–54000 € with rotational speed control (Table 3.2).

3.2.3 Energy efficiency-based recommendable operation conditions of variable speed driven pumps (Article II)

The effectiveness of the pumping system is often evaluated using pumping efficiency (2.10) or system efficiency (2.11–2.12), which both describe how well the given energy is transferred to the liquid. Compared with the pumping efficiency and system efficiency, the specific energy consumption E_s (2.13) considers also the effects of rotational speed and system losses on the pumping energy efficiency. This makes E_s a feasible indicator of the pumping system energy efficiency and of the recommendable operation conditions for variable speed driven pumps.

The drawback of using specific energy consumption as a criterion for energy efficiency is that the attainable level of E_s is affected by the system characteristics: the amount of static head and dynamic head. Therefore, E_s levels in different pumping systems cannot be compared with each other, as the minimum E_s depends on the system characteristics. If the system characteristics are known, the E_s of a pumping system can be evaluated using

$$E_s = \frac{\rho \cdot g \cdot H}{\eta_{\text{pump}} \cdot \eta_{\text{motor}} \cdot \eta_{\text{VSD}}} = \frac{\rho \cdot g \cdot (H_{\text{st}} + H_{\text{dyn}})}{\eta_{\text{sys}}} \quad (3.1),$$

where H_{st} is the static head and H_{dyn} is the dynamic head of the system. According to equation (3.1), the minimum E_s when transferring the fluid to the required elevation can be determined based on the system static head and maximum system efficiency. The theoretical minimum attainable specific energy $E_{s,min}$ in a pumping system can be written

$$E_{s,min} = \frac{\rho \cdot g \cdot H_{st}}{\eta_{sys,max}} \tag{3.2}$$

The energy efficiency of a pumping system can therefore be studied by comparing the actual E_s to $E_{s,min}$, or any other base value of E_s, for instance E_s achieved in the pump BEP. Thus, pump energy efficiency can also be studied as a relative value for specific energy consumption, for instance with

$$E_{s,rel}(Q) = \frac{E_{s,base}}{E_{s,i}} \qquad (3.3),$$

where $E_{s,rel}$ is the relative specific energy consumption, $E_{s,base}$ is the selected base value for specific energy consumption, and $E_{s,i}$ is the specific energy consumption in a certain operating point of the pump.

In this research case, the justified limits for the recommendable energy efficiency for rotational speed controlled pumping were studied by calculating the relative specific energy use of a variable speed controlled pump unit. E_s values were formed for a Sulzer APP22-80 centrifugal pump attached to an 11 kW ABB induction motor and ABB ACS800 industrial drive module. The resulting relative E_s for the pump unit was used to demonstrate energy efficiency-based recommendable operation conditions, which can be seen also as the Best Efficiency Area (BEA) for the pump unit.

3.2.4 Findings of Publication II

The specific energy characteristics of the APP22-80 pump were created using a base value 61.1 Wh/m^3, which is achieved when the pump is operated in the BEP. To indicate the energy efficiency as the minimum relative E_s, the equation (3.3) is used in inverse order. Hence, the lower the $E_{s,rel}$, the better the energy efficiency of the pumping. In this case, only the pump input power was studied, and therefore, the motor and drive losses are not included. The results for the relative specific energy consumption E_s in varying operation conditions can be seen in Figure 3.4.

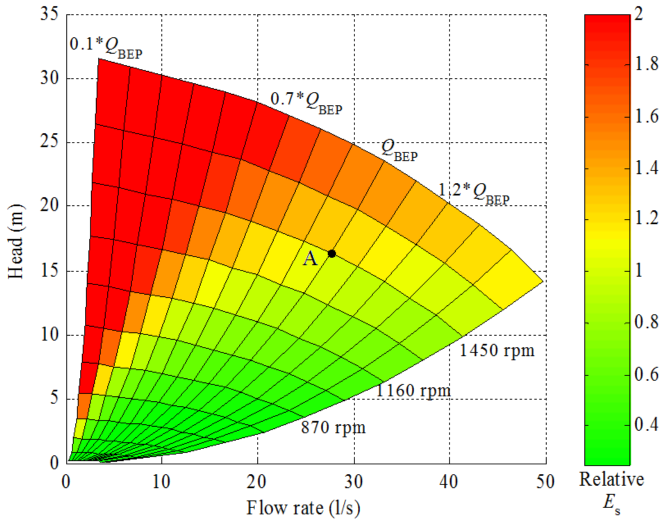

Figure 3.4. Relative E_s of the APP22-80 pump in varying operation conditions. The base value is selected according to point A, in which the pump is operated at the nominal speed 1450 rpm and in the BEP. The base value for E_s in this point is 61.1 Wh/m^3.

The results shown in Figure 3.4 illustrate the operating regions in which the pump E_s exceeds the chosen base value having an increasing effect on the pump energy consumption per pumped volume. For example, at 70% flow rate and operating speed of 1450 rpm, the relative E_s is 1.23. At the rotational speed of 1595 rpm and 70% flow rate, the relative E_s is 1.48, indicating clearly higher energy consumption per pumped volume.

Figure 3.4 also shows operating regions in which the relative E_s is the same as or lower than the selected base value. For instance, when operating at the speed of 1160 rpm and 50 % flow rate, the relative E_s is 1, indicating that the pump output is delivered with justified energy usage. In general, Figure 3.4 demonstrates that in terms of the selected energy efficiency criteria, reducing the rotational speed of the pump extends the recommendable operating region compared with the HI guidelines.

As shown in Figure 3.4, the relative E_s tends to decrease at the fixed speed when moving towards the increased flow rate on the pump *QH* curve due to the decreasing head. Because of this, E_s can be seen as a good criterion to evaluate the recommendable maximum rotational speed and minimum flow rate for the studied pump. When operating the pump at relatively high flow rates, the risk of occurring cavitation increases, and hence, the maximum recommendable flow rate should be evaluated also from the pump reliability point of view.

The effect of the drive and motor efficiency on the specific energy consumption of pumping was also calculated. The available data for the ACS 800 variable speed drive and the 11 kW ABB induction motor efficiency was included in the APP22-80 pump relative E_s calculations. The resulting relative E_s for the pumping system is illustrated in Figure 3.5. The base value for the relative E_s is again 61.1 Wh/m^3, for the sake of clarity. Figure 3.5 shows the exemplary limit for the BEA based on the relative E_s threshold of 1.25.

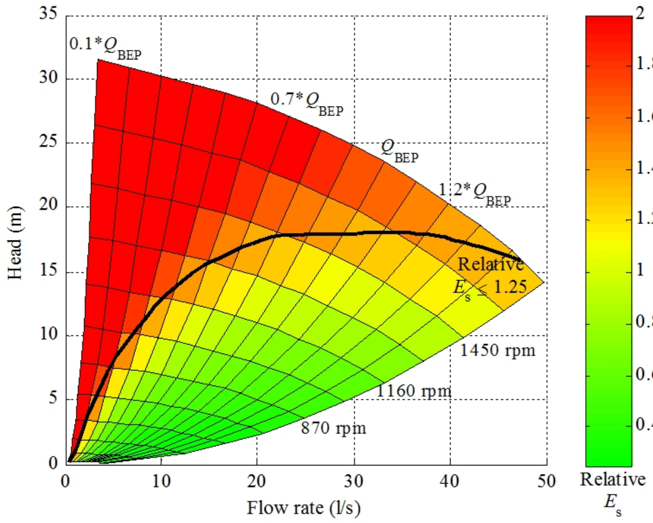

Figure 3.5. Relative E_s of the APP22-80 pump including the drive train losses. Figure shows the exemplary line for the energy efficiency-based recommendable operating region for pump drive train.

Comparing the results shown in Figure 3.4 and Figure 3.5 illustrates that the effect of drive and motor losses is notable at low rotational speeds and low relative flow rates. In this region, the drive train efficiency is considerably lower compared with the maximum attainable efficiency, increasing therefore the relative E_s. Otherwise Figure 3.5 is similar with Figure 3.4; again the decrease in the rotational speed seems to extend the energy efficient operating area of the pump on the basis of the pumping system E_s.

The calculations show that suitable conditions for a VSD controlled pump unit can be determined with the relative specific energy consumption. To compare different pump units with each other, a certain base value has to be given for the relative E_s. The shown calculations are presented and discussed also in the study by Ahonen (2011).

3.2.5 Suggestions for preferred operation of VSD controlled parallel pumps (Publication III)

Besides the energy efficiency considerations, also the reliability issues should be taken into account when determining suitable operation conditions for VSD controlled parallel pumping. As discussed in section 2.1, it is often recommended that centrifugal pumps should be driven close to the BEP. Correspondingly, operating outside the BEP can increase the risk of reduced service life. Reducing the rotational speed of the pump can, however, improve the pumping reliability compared with the operation on the nominal rotational speed.

In parallel pumping, the most hazardous events can be the operating in the shut-off or in the area with a cavitation risk. Near the shut-off, the pump will have very poor pumping efficiency and typically decreased motor and drive efficiency, causing a high rise in the specific energy consumption (see equation 2.13). Operating the pumps in both cases affects negatively the pump wear and energy efficiency of pumping. For this reason, operating variable speed controlled parallel pumps in areas with a risk of shut-off or cavitation should be avoided.

Sharing the load of parallel pumps can ensure their operation in conditions in which the energy efficiency and reliability of the pumping is justified (KSB, 2004; Jones, 2006; Jinguo, 2008). As an example, operating two VSD controlled pumps with reduced speed can result in the same output with less energy use than operating only one of the pumps with a higher speed in the same system. Hence, including the load sharing of the parallel pumps should be taken into account when implementing the control strategies for VSD controlled parallel pumps. If the characteristics of the system are known, it is possible to calculate the preferred moment when additional parallel pumps should be started or stopped to deliver the desired output with the best available energy efficiency. This is, however, a major challenge for the control strategy, since information from the pumping process is often available only to a limited extent.

3.2.6 Findings of Publication III

In parallel pumping, the most avoidable operation conditions can be operating at the shut-off or in an area with cavitation risk, both of which also affect not only the pump wear but also the energy efficiency of pumping (Shiels, 1997). Especially in systems having a large variation in the required flow rate, the risk of these harmful operation conditions can increase if the variable speed controlled pumps are not similar sized or the pumps are used at different rotational speeds. Hence, avoiding these operation conditions suggests that parallel-connected pumps are operated with justified energy efficiency and reliability when delivering the required output for the process.

The resulting control scheme in a parallel pumping system is illustrated in Figure 3.6, which plots the *QH* characteristics of two variable speed driven pumps in a system. The operation is studied in a case in which the total flow rate of the pumping system is increased from zero to maximum; in low flow rates, the total flow is delivered using the primary pump (Pump 1), and to enable higher flow rates, the secondary pump (Pump 2) is started.

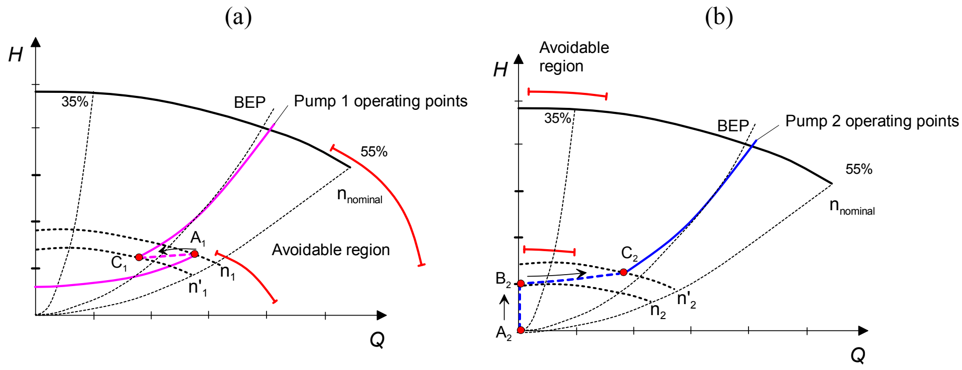

Figure 3.6. Using rotational speed balancing of parallel pumps to avoid unwanted pump operation areas. Pump 2 should be started when there is a risk that Pump 1 is entering an undesirable operating region. After Pump 2 has started to deliver flow at point B_2, the speed of both pumps can be balanced to meet the same head at points C_1 and C_2.

To ensure that the rotational speed controlled parallel pumps do not operate in undesirable operation regions, the speed of the pumps is balanced as illustrated in Figure 3.6. The speed of the primary pump (Pump1) is not increased further if there is a risk of entering the avoidable region. Thus, the speed is kept constant at n_1 (point A_1). Consequently, the secondary pump (Pump 2) is started (A_2), and when Pump 2 starts to deliver flow at speed n_2 at point B_2, the speeds of both pumps are balanced to meet the same head level. This procedure causes the operating point of Pump 1 to shift from A_1 to C_1 and the operation point of Pump 2 from B_2 to C_2 (Figure 3.6). Balancing the rotational speed of the parallel pumps has been suggested already by Hammond (1984), although not from the perspective of energy savings but to even out the pump working hours and wearing. Adding pumps and performing the balancing procedure on the right moment can also result in a lowered specific energy compared with traditional speed control, in which the rotational speeds of individual pumps are increased to nominal before adding more parallel pumps to the process. Similar control steps can also be applied to systems with a higher number of parallel pumps. For instance, the on-going pumps can be seen as a unit representing the primary pump (Pump 1) while the next pump in turn represents the secondary pump (Pump 2).

In other words, a base-line for energy efficient control strategy can be set if the pump user determines a preferred operation area (POA) for each parallel pump. The major drawback of implementing the exemplary control strategy in parallel pumping systems is the lack of pump operation monitoring, which is necessary to determine the relation between the individual operation point of each parallel pump and the set POA for the pumps. Also, each parallel pump unit must be equipped with variable speed control to apply speed balancing and operation below the nominal rotational speed of the motor.

3.3 Model-based pump monitoring using variable speed drives

Real-time information from the pumping process is one of the requirements when optimizing the energy use of the pumping. In variable speed pumping applications, this information can be acquired from VSDs by utilizing the internal estimates of the motor. In fact, flow monitoring applications using for example basic *QH* and *QP* methods are already available in modern VSDs (Hammo and Viholainen, 2005). Using the model-based estimation methods to determine the operation point of the pump is relevant especially when implementing control strategies in parallel pumping systems, since using separate flow metering for each parallel pump can be seen unrealizable in many real-life systems. The basic idea of using flow estimation in variable speed parallel pumping system is illustrated in Figure 3.7.

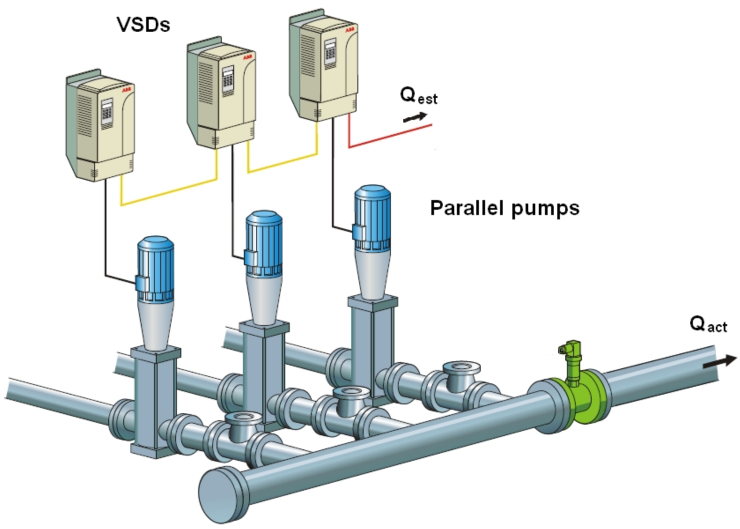

Figure 3.7. Providing flow estimation with VSDs for each parallel-connected pump. Three VSD controlled parallel pumps are delivering a total flow Q_{act} to the common outlet pipeline. The VSDs provide the estimated total flow Q_{est} to the pump user based on the individual operation point of each pump.

In the example system shown in Figure 3.7, three VSD controlled parallel pumps are feeding a common outlet pipeline. The operation point of each parallel pump can be monitored without separate metering using sensorless flow estimation, for example the

QP method provided by the VSDs. In the case of the *QH* method, pressure transmitters for both the inlet and outlet section of each pump should be installed.

The focus on this section is to study the model-based operation point estimation using VSDs especially in parallel pumping systems. First, determining the individual flow rate of each parallel-connected pump is studied using *QH* and *QP* methods in a VSD controlled parallel pumping system built in a laboratory. Enabling the sensorless flow estimation in parallel pumping is studied further in a second research study. In this second case, the studied model-based methods are the *QP* method and Process curve method. The usability and accuracy of these methods are evaluated based on laboratory tests on a single centrifugal pump in a system. Thirdly, the combined use of *QH* and *QP* is proposed to improve the accuracy and usability of the basic flow estimation methods. The usability and accuracy of the presented *QH*/*QP* method is tested in a blower laboratory together with several other methods: *QH* method, *QP* method, Kernan method, and Hybrid method and combinations.

3.3.1 Testing the *QH* and *QP* method in a parallel pumping system (Publication IV)

Estimation of the flow rate of individual pumps in a parallel pumping system was tested with the *QH* and *QP* method available in ABB industrial drive module ACS800. The inputs for applying the *QH* method in the VSD are the *QH* characteristic curve at the nominal speed of the pump and pressure signals from the inlet and outlet section of the pump. In the QP method, the only input is the *QP* characteristic curve as discussed already in section 2.3.

The flow estimation methods were tested in a laboratory consisting of two parallel-connected pump drive units. The first unit (Pump 1) comprises a Serlachius DC 80/255 centrifugal pump, a four-pole 15 kW Strömberg induction motor, and an ABB ACS 800 industrial drive. The second unit (Pump 2) consists of an integrated 5.5 kW Grundfos LP 100–125 centrifugal pump with an induction motor and an ABB ACS 800 industrial drive. The laboratory setup is illustrated in Figure 3.8.

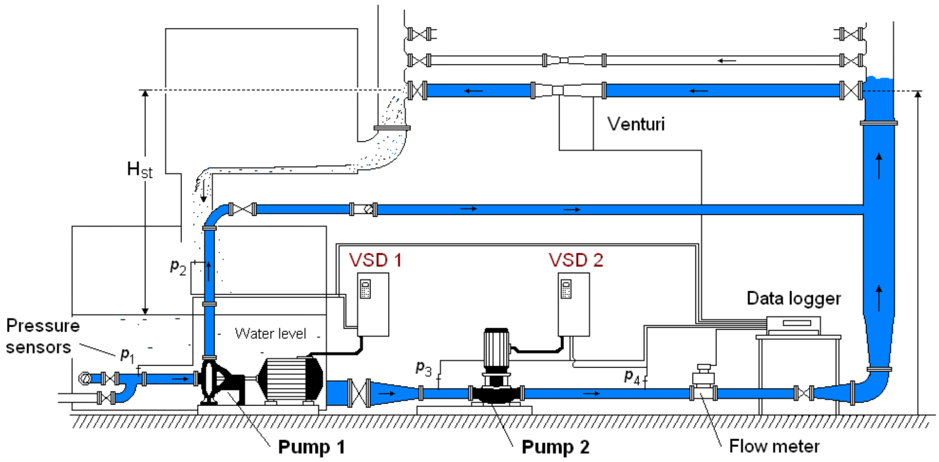

Figure 3.8. Laboratory setup for testing the flow estimation in a parallel pumping system. The actual flow rate of the pumps can be measured using a magnetic flow meter and venturi tube installed in the system. The pressure sensors are used to determine the head of each pump and the signals are used as inputs for the *QH* method.

The flow rate in the parallel pumping system was measured with a magnetic flow meter and venturi tube. The head of both pumps was measured using pressure sensors installed to the pump input and output tappings. The data from the metering equipment was gathered to a data logger. The static head in the system was approximately 2.5 meters.

3.3.2 Findings of Publication IV

The flow rate estimation of the VSD was tested in a parallel pumping system. The accuracy of the flow estimation and factors affecting the estimation results were studied in varying operation conditions. The flow rate of each parallel-connected pump was changed during the measurement runs using rotational speed control or valve adjustment.

Figure 3.9 plots the estimated flow rate using *QH* curve-based calculation and the measured flow for parallel pumps when the rotational speed of each pump was varied between 84–100% of the nominal rotational speed. For example, the flow estimation values for Pump 1 shown in Figure 3.9 (a) differ 13% from the measured ones at the 1200 rpm rotational speed. At the rotational speeds of 1302 rpm, 1350 rpm, and 1425 rpm, the difference is only 1–5%. The estimated flow rates for Pump 2 were also very close to the measured values during the same measurement run, as can be seen from

Figure 3.9 (b). The difference between the estimated and measured flow was 2−7% at the flow rates of 2526 rpm, 2742 rpm, 2640 rpm, and 3000 rpm.

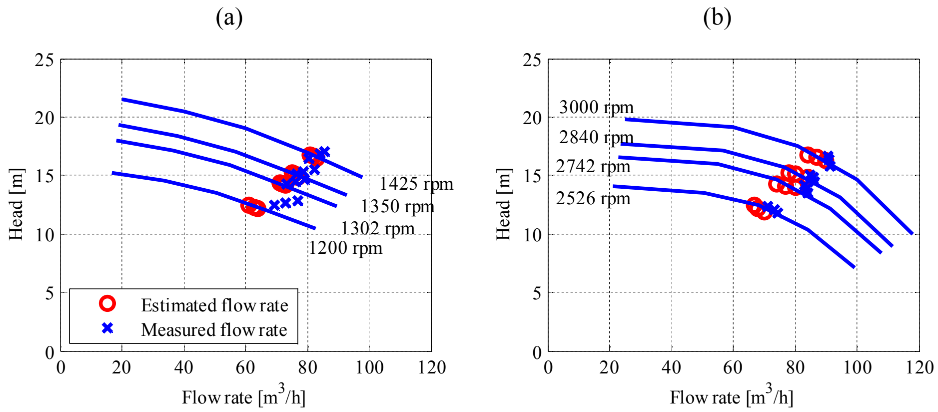

Figure 3.9. Exemplary results of the flow rate estimation of individual pumps in a parallel pumping system using *QH* curve-based calculation. The difference between the estimated and measured flow for Pump 1 (a) is 0−13 % at 84−100% rotational speed range. For Pump 2 (b), the difference between the estimated and measured flow rate values was 1−7 % at the same rotational speed range.

Similar results were also obtained when the flow resistance of the piping was varied separately for the two pumps, and therefore, the pumps had different heads to achieve. In this case, the laboratory tests indicate that the *QH* curve-based flow metering can determine the individual flow rate of parallel pumps and the total flow in varying system conditions with 5−10% accuracy.

An example of the results using *QP* curve-based flow estimation is shown in Figure 3.10. The figure plots the estimated and measured flow rate at varying rotational speed (70−100% of the nominal speed). Based on the measurements shown in Figure 3.10, *QP* curve-based estimation gives very similar results as the laboratory flow metering equipment at rotational speeds 91−100% of nominal; the difference between the measured and estimated flow for Pump 1 (Figure 3.10 a) and Pump 2 (Figure 3.10 b) was 1−6%. However, there was a clear decrease in the flow estimation accuracy when the rotational speed of the pumps was adjusted to 70% of the nominal speed. The rapid decrease in the estimation accuracy is probably caused by the shape of the *QP* curve in the current flow range; when operating at 70% speed, the curve is very flat, especially for Pump 2, as illustrated in Figure 3.10 (b).

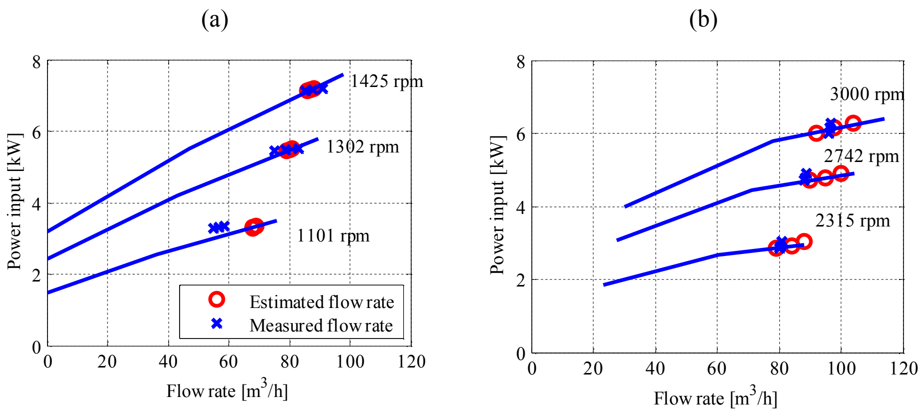

Figure 3.10– An example of the flow rate estimation of individual pumps in parallel pumping system using *QP* curve-based calculation. The difference between the measured and estimated flow rate for Pump 1 (a) and Pump 2 (b) is approximately 1–6% at rotational speeds 90–100% of the nominal speed. When operating at 70% rotational speed, the *QP* curve is very flat, which decreases the accuracy of the flow rate estimation especially at relatively high flow rates.

The *QP* curve-based flow estimation gave similar results when the system head was adjusted using valves. Noteworthy, the difference between the measured and estimated values was even smaller in the case of Pump 2 when the system head was increased with valves, since the *QP* is steeper in low flow regions.

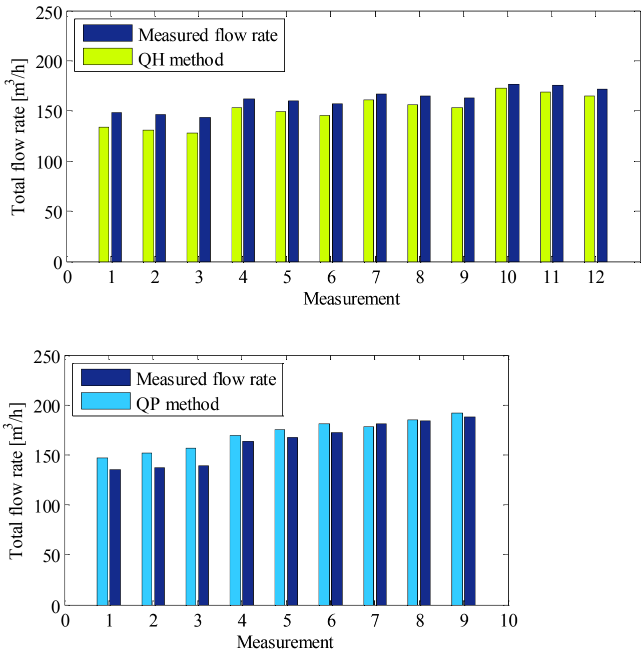

Figure 3.11. Examples of the total flow rate values using the *QH* and *QP* methods. The total flow rates are the sum flow rates shown in Figure 3.9 and Figure 3.10. The difference between the measured total flow rates is 3–16 m^3/h in the case of *QH* method and 0–18 m^3/h in the case of *QP* method.

The total flow rate of the parallel pumps using the *QH* method and *QP* method in the same tests is shown in Figure 3.1. The total flow rate in this case is the sum flow given by both parallel pumps during the test sequences. The results show that the difference between the measured total flow rate and estimated flow is 3–16 m^3/h in the case of *QH* method and 0–18 m^3/h in the case of *QP* method. Thus, the highest relative difference in the measured flow rate in both cases is only 10–13%.

3.3.3 Testing the sensorless operation point monitoring in a pumping system (Publication V)

The pump operating point can be estimated without external measurements of the pump flow rate *Q* and the pump head *H*. In this research case, two sensorless methods for estimating the flow rate and head of the pump were tested in a pumping system. The tested methods are the QP method and Process curve method.

In this context, the *QP* method was used not only to determine the flow rate of the pump, but also the head of the pump was estimated based on the flow rate. The estimation procedure was similar as explained in previous sections, and the only inputs for the method are the *QP* and *QH* characteristic curves.

The Process curve method is based on the fact that the pump is operated in the intersection point of the pump *QH* curve and system curve. When the *QH* curve is transformed to the current rotational speed using the affinity laws (equations 2.7–2.9), the location of the operation point can be determined by calculating the intersection point of the transformed *QH* curve and system curve. As shown in equation (2.6), the only required input parameters, besides the pump performance curves, are the static head H_{st} and the coefficient for dynamic head k. The values for the required input parameters can be calculated using start-up measurements with for example a portable flow meter or by simply calculating the friction losses according to equation (2.5).

As mentioned in section 2.3, the justified operation point estimation using the model-based methods depends on the accuracy of the characteristic curves and the accuracy of the internal measurements. To eliminate the effect of the inaccurate characteristic curves in the estimation, the pump characteristic curves were measured separately, and the measured curves were applied to the estimation methods instead of manufacturer curves.

The accuracy of internal estimates was determined with a test measurement for the ABB ACS 800 industrial variable speed drive which applies the direct-torque control method. The VSD was feeding an ABB 11 kW induction motor attached to a Sulzer APP 22-80 centrifugal pump. The rotational speed and torque of the motor were measured with a dataflex 22/100 measurement shaft over the entire operation region of the Sulzer pump. The tested pump drive is shown in Figure 3.12.

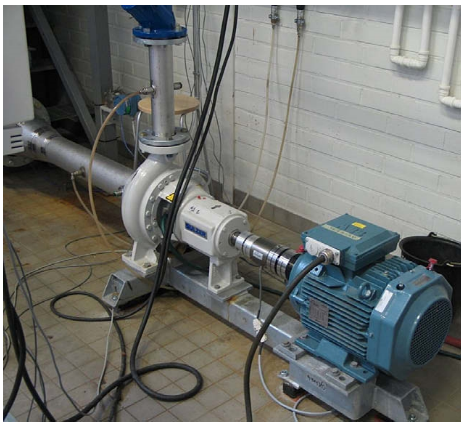

Figure 3.12. Pump drive attached to the piping system. The pump drive consists of a centrifugal pump, an 11 kW induction motor, and an industrial drive.

The process curve and QP curve-based estimation methods were tested with the same laboratory setup. Different operation points for the pump drive were generated using a different valve setting for each measuring sequence and varying the rotational speed of the pump.

3.3.4 Findings of Publication V

Relative errors of the estimated values (shaft power, shaft torque, and rotational speed) were calculated, and they are presented in Figure 3.13 (a).

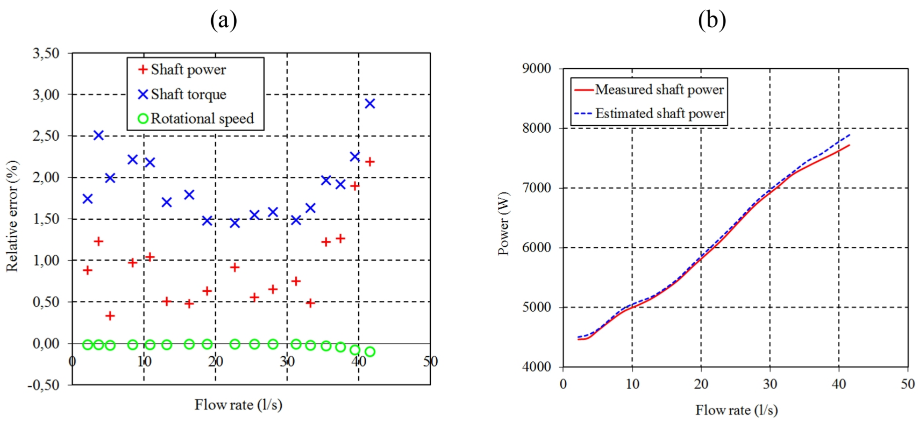

Figure 3.13. Measurement results for the estimates of the ABB ACS 800 variable speed drive. (a) Relative errors of the shaft power, the shaft torque, and the rotational speed estimates. Accuracy of these estimates is within 3%. (b) The estimated shaft power as a function of flow rate and the measured shaft power.

Based on test measurements results, the accuracy of the rotational speed estimate is within 0.25%. The relative error of the shaft torque exceeds 2% at partial and excessive flow rates, which weakens the shaft power estimations in these regions. However, a shaft power estimation error has only a limited effect on the estimated power, which can be seen from Figure 3.13 (b) which shows the estimated power as a function of flow rate. When estimated power values are applied to determine the flow rate from the measured QP curve, the estimation accuracy is within 2 l/s, excluding the high flow rate region (~35–42 l/s) in which the difference between the estimated and measured shaft power starts to increase (Figure 3.13). The results indicate that in the major part of the operating range, the accuracy of the pump characteristic curves has a larger effect on the estimation accuracy of the QP curve-based method than the accuracy of the internal estimates of the VSD.

The process curve and QP curve-based estimation methods were tested with the same laboratory drive comprising of an ABB ACS 800 variable speed drive, an ABB 11 kW induction motor, and a Sulzer APP 22-80 centrifugal pump with a 255 mm impeller. Different operation points for the pump drive were generated using a different valve setting for each measuring sequence. During the sequence, the rotational speed of the pump was varied from 1020 rpm to 1560 rpm. The pump characteristic curves were measured separately, and the measured curves were applied to the estimation methods instead of manufacturer curves. The reason for this was to eliminate the effect of inaccurate pump characteristic curves on the estimation results.

The results for the measurement sequence with the nominal flow rate valve setting are shown in Figure 3.14. When the pump is driven in the nominal flow region, the estimation error of the *QP* method for head and flow rate is less than 6% compared with the measured values, and in the case of the process curve method, the estimation error of the head and flow rate is within 3%. The results of the measurement sequence for 50% of the nominal flow and 140% of the nominal flow are shown in Figure 3.15.

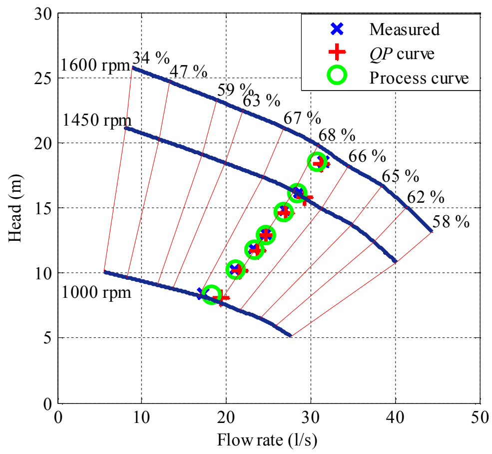

Figure 3.14. Result of the laboratory test for the process curve and *QP* curve-based estimation methods with the nominal flow rate valve setting.

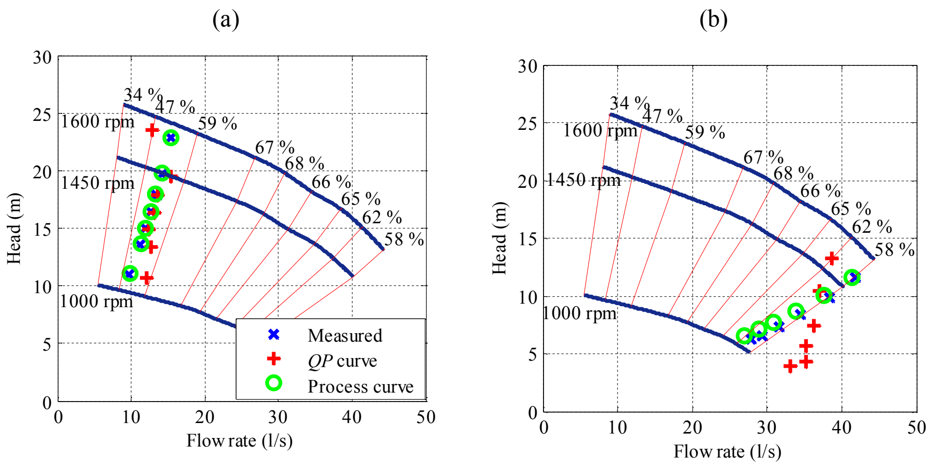

Figure 3.15. Results of the laboratory test for the process curve and *QP* curve-based estimation methods. (a) Measurement sequence with 50% of the nominal flow rate. (b) Measurements with 140% of the nominal flow rate.

In the case of 50% of the nominal flow, the relative error of the process curve estimation method is less than 2%, and with the *QP* estimation method, the relative error is within 8% as shown in Figure 3.15 (a). In this operating region, the *QP* estimation method is most accurate at rotational speeds 1230−1410 rpm. When the motor is driven at the speed of 1000−1200 rpm or over 1500 rpm, the estimations differ from the measured operating points. This is mainly caused by the inaccurate affinity transforms and the flat *QP* curve shape at the flow rates 10−17 l/s (Figure 3.15 b).

When operating the pump in the 140% of the nominal flow region, the *QP* estimation results in erroneous estimates (Figure 3.15 b). This is caused by the decreasing d*P*/d*Q* at the flow rates above 30 l/s and the increased error in the power estimates: in this measurement sequence, the shaft power estimation error was 4−8%. The relative error for the head and flow rate values of the process curve method in this region is within 3%.

3.3.5 Combined use of *QH* and *QP* method (Publication VI)

The proposed *QH*/*QP* estimation method selects and combines two known operating point estimation methods (the *QH* and *QP* methods) in a novel way to determine the operating point location of a pump or a blower with improved accuracy and in a wider selection of system conditions. The *QH* method estimation typically has its best accuracy when operating approximately at the nominal flow rate (or in the BEP) or flow rates above it, due to the distinctive increase of d*H*/d*Q* in the pump *QH* characteristic curve. Correspondingly, the *QP* method can typically give the most accurate estimation at the nominal flow rates or below it, again because of the inherent shape of the *QP* characteristic curve (Sulzer, 1998; Kernan et al., 2011; Tamminen et al., 2011).Thus, the overall accuracy of the model-based operating point estimation can be improved by the selective use of the basic *QH* and *QP* methods in appropriate regions.

The performance curve regions which would cause uncertainty to the basic flow estimation methods can be automatically calculated based on the characteristic curve data and the estimated error in the pressure measurement and power estimate. For example, the derivative of the characteristic curve gives a rough estimate of the error which can be produced by the erroneous pressure measurement or power estimate. Because of this, it is justified to use the characteristic curves to determine the usable regions associated with each estimation method. An example of the calculation of this flow rate estimation uncertainty U_{QH} for the *QH* method is

$$U_{QH}(Q) = \frac{\mathrm{d}p\left(Q_{\mathrm{est},QH}, n_{\mathrm{Est}}\right)}{\mathrm{d}Q} \Delta p_{\mathrm{meas}} \tag{3.4}$$,

where Δp_{meas} is the expected error in the pressure measurement, and the subscripts $_{est}$ and $_{QH}$ denote the estimated value and the *QH* method, respectively. Thus, the flow rate estimation uncertainty can be calculated as a function of flow rate and rotational speed.

The procedure of the *QH*/*QP* method is as follows. First, both the *QH* and *QP* methods are used to estimate the flow rate, and the uncertainty in the estimates is calculated. Secondly, the method which has a lower uncertainty is selected as the estimate of the flow rate. If the uncertainties in the estimates are approximately the same, the flow rate estimates can be weighted so that both flow rate estimates are taken into account. A simple solution to weight the calculated uncertainties can be written as

$$Q_{\mathrm{est}} = \frac{U_{QP} \cdot Q_{\mathrm{est},QH} + U_{QH} \cdot Q_{est,QP}}{U_{QP} + U_{QH}} \tag{3.5}$$,

where the subscript $_{\mathrm{Est}}$ denotes the estimation result, and *QP* and *QH* denote the *QP* and *QH* methods, respectively. To demonstrate why the uncertainty of one method is multiplied by the flow rate estimate of the other method, an example is given: if U_{QP} is 0.5 $\mathrm{m^3/s}$ and U_{QH} is 0.3 $\mathrm{m^3/s}$, the *QH* method is less uncertain. The flow rate estimate of the *QH* method has now a weight of five-eighths and the *QP* method flow rate estimate only three-eighths, and thus, the less uncertain method has a higher weight.

The combined *QH/QP* method requires the rotational speed, shaft power, and pressure as inputs from which the rotational speed and shaft power inputs are directly available from the frequency converter. However, the net pressure measurement is not always installed in pumping systems, and this is naturally a drawback in the commissioning of the *QH/QP* estimation method compared with the normal *QP* method, the Hybrid method, and the Kernan method, all of which require only the rotational speed and shaft power as inputs. In general, the realization of the pressure measurement according to ISO measurement standards can also be challenging in real-life pumping systems. This results from the requirement of straight pipeline parts both before and after the pressure measurement installation. On the other hand, the tolerances of the *QH* curve are 0.5 percentage units smaller than the *QP* curve tolerances in the ISO 9906 standard. Hence, the flow estimation based on the *QH* curves can be seen more accurate in this point of view, and therefore justify the installation of the pressure sensors.

The proposed method can be further improved by using the *QH/QP* method with the Hybrid method presented by Ahonen et al. (2012). This can be executed as follows: the system curve is estimated near the pump nominal rotational speed, but instead of using the *QP* method only, the *QH*/*QP* estimation method is used to identify the system curve. Like mentioned previously, the identifying of the system curve in the Hybrid method reduces the effect of the relative error in the pressure and power estimation, in the case of a substantial change in the rotational speed from the initial value, and can therefore improve the estimation accuracy at low rotational speeds (Ahonen et al., 2012). Thus, studying the use of the *QH/QP* method together with the hybrid method is also justified.

The introduced methods are applicable to both pumping and blower systems. In this research case, the proposed *QH*/*QP* method and other model-based flow estimation methods were tested with a blower system consisting of a FläktWoods Centripal EU 630 MD radial blower (nominal values: 2.9 m^3/s, 1190 Pa, 4.6 kW, 1446 rpm), an ABB induction motor (nominal values: 400 V, 21 A, 7.5 kW, 1450 rpm), and an ABB ACS850 industrial drive module (nominal current: 35A). The observed system contains six meters of piping on the blower inlet and outlet. The blower static pressure was measured using pitot static tubes and a Rosemount 2051C differential pressure meter. The produced flow rate was measured using an Elridge Series 9800MPNH thermal flow meter. The shaft power and rotational speed were observed using internal estimates of the VSD. The laboratory test setup is presented in Figure 3.16.

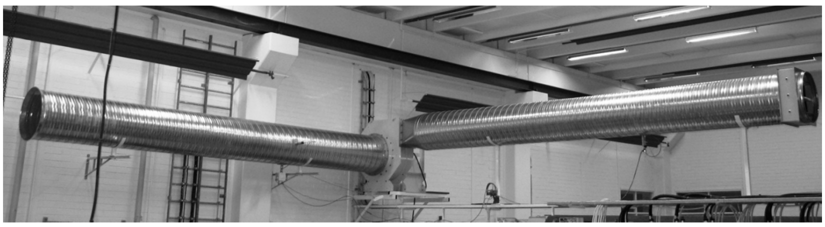

Figure 3.16. Radial blower system used in test measurements.

3.3.6 Findings of Publication VI

To verify the blower characteristics in the laboratory setup, the output of the blower was measured at 1500 rpm and with different valve settings. The measured operation points and blower characteristics are presented in Figure 3.17. Based on these results, the measured *QH* and *QP* curves have some inaccuracy compared with the manufacturer characteristics. Although similar difference can be sometimes expected in real-life system conditions, it plainly causes error when using presented estimation methods based on the use of characteristic curves.

The blower characteristics also indicate that there are flow regions in characteristic curves in which basic *QP* and *QH* estimation would give erroneous results. For instance, based on the shape of the manufacturer curves, the basic *QH* method would give two different flow rate estimates for each measured pressure value within a range of about 0–2 m^3/s when operating at 1500 rpm as seen from the manufacturer's curve. Correspondingly, the *QP* method would give two different flow value results for each measured power value when operating in a range above 3 m^3/s flow rates at 1500 rpm rotational speed. Thus, the flow rate range in which the basic *QP* or *QH* methods are usable is limited.

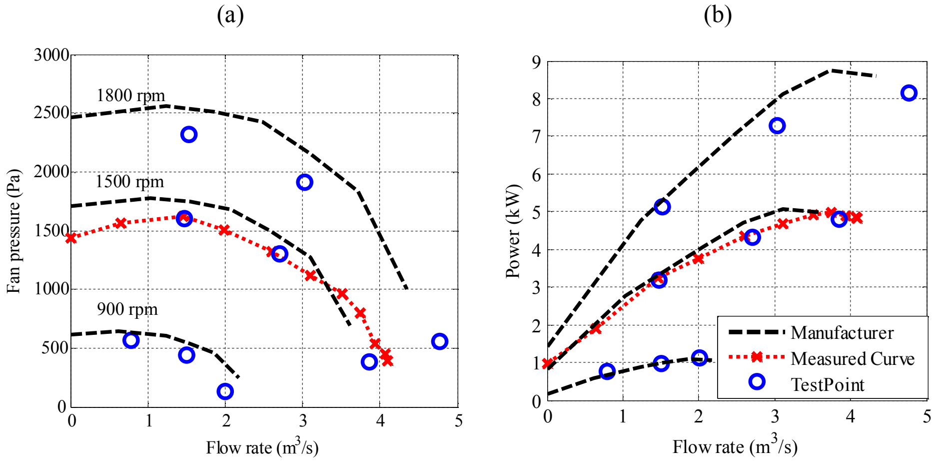

Figure 3.17. Blower characteristics given by the manufacturer and the measured output. (a) Output pressure as a function of the flow rate. (b) Blower power as a function of the flow rate. The TestPoints marked in (a) and (b) indicate the measured operation points when comparing the different flow rate estimation methods. The difference in the test point location compared with the measured curve results from the measurement equipment repeatability and the effect of ambient conditions.

Despite the difference between the manufacturer and measured performance curves in a laboratory system, the manufacturer curves are selected for the estimation. By doing this, no start-up measurements are needed in the commissioning of the introduced *QH/QP* method. Also, the effect of inaccurate manufacturer curves on the flow rate estimation is included in this study. If the measured curves are used as a model in the estimation methods instead of the manufacturer curves, the flow rate estimation accuracy of several methods can be expected to increase significantly.

The estimation methods were tested with several different rotational speeds and system curve settings. As an example, the location of nine different tested operation points can be found from Figure 3.17, which plots the exemplary TestPoints in both the *Qp* and *QP* axis. These operation points result in 50%, 95%, and 130% relative flow regions at the studied rotational speeds of 900, 1500, and 1800 rpm. The studied flow rate estimation methods were:

- Basic *QH* curve-based method (Liu, 2002)
- Hybrid method (Ahonen et al., 2012)
- Kernan method (Kernan et al., 2011)
- Basic *QP* curve-based method (Wang and Liu, 2005)
- Kernan-*QH* method (Kernan et al., 2011)
- *QH*/*QP* method
- Hybrid *QH*/*QP* method

As expected, the test results demonstrated that the basic *QH* and *QP* methods gave highly erroneous results in flow regions having very low d*p*/d*Q* or d*P*/d*Q*. Also, the difference between the measured and actual characteristic curve had a strong influence on the accuracy of the flow estimation. Exemplary results for flow estimation methods when the blower is operating in the 95% flow region are shown in Figure 3.18. The measured flow rate is represented as 'True' values.

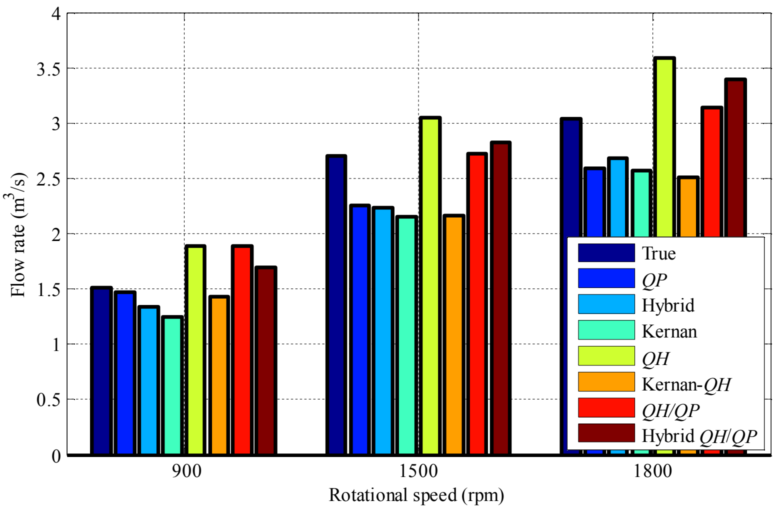

Figure 3.18. Flow rate estimation results for the blower in the 95% flow region at rotational speeds 900, 1500, and 1800 rpm.

In the presented flow region (Figure 3.18), the *QH* and *QP* methods have approximately the same accuracy, since the uncertainty of the curve-interpolation is nearly at the same level. The *QH*/*QP* method combines these estimation methods with weighting (3.5) and gives the most accurate estimation at the rotational speeds 1500 and 1800 rpm. At 900 rpm, the Hybrid *QH*/*QP* estimation compensates the error produced by the rotational speed change resulting in improved estimation accuracy compared with the *QH*/*QP* method. The Hybrid method and Kernan method give approximately the same estimation results in this flow region. Combining the Kernan method with the *QH* estimation does not increase the accuracy of the estimation in this measurement sequence, because the bias in the *QH* curve seems to be overcompensated, and thus, the estimated flow rates are too low compared to the measured values (Figure 3.18).

To evaluate the overall accuracy of the estimation methods compared with the measured values, the average error and standard deviation of the error was calculated including all test sequences. The results for the average error and standard deviation are shown in Figure 3.19.

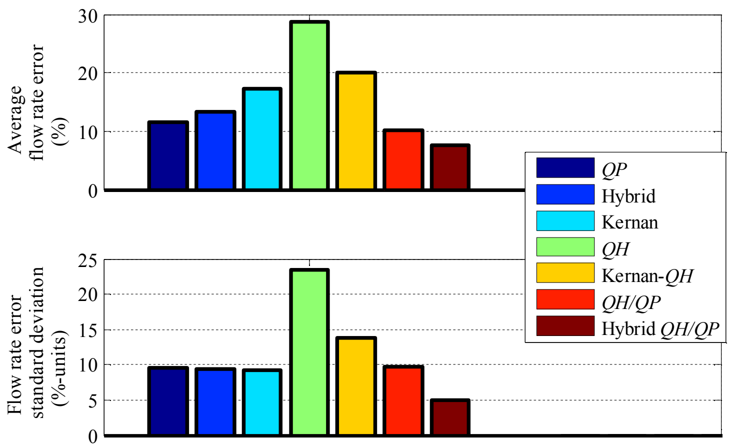

Figure 3.19. Average error and standard deviation of the error in the flow rate estimation of the studied methods.

Based on the measurements, the *QH*/*QP* method and Hybrid *QH*/*QP* method have the lowest average error across the blower operation area compared with other estimation methods tested in the laboratory system. The results in the standard deviation of the error demonstrate that there is no significant variation in the average error throughout the blower operation range (Figure 3.19). In this case, flow rate estimation using the *QH* method had the highest average error, which is the result of poor estimation accuracy especially in low flow regions. As mentioned previously, the reasons for this are the low dp/dQ and the difference between the measured operation and characteristic curves given by the manufacturer.

3.4 Energy efficient and reliable control strategy for parallel pumps

The aim of this section is to introduce a new control strategy for variable speed controlled parallel-connected centrifugal pumps (later referred to as parallel pumps) in a system in which the pumps in individual piping parts feed a common outlet pipeline. The suggested control strategy can offer a justified base for the energy efficiency improvement in variable speed controlled parallel pumping systems, even in such cases in which the information on the pumping system is limited or changing. The suggested control strategy is based on the simple use of the existing pumping monitoring solutions of a modern VSD and a known relation between the preferred operation area of a centrifugal pump and the energy efficiency of the pumping. By implementing the suggested control strategy in the control procedure, the flow rate of the parallel pumping system can be adjusted with improved energy efficiency compared with traditional rotational speed control. The control strategy can be applied for instance to parallel pumps located in water stations, waste water pumping stations, and industrial plants.

The relation between the pump operation point location and pump reliability and energy efficiency has been discussed in many occasions (ANSI/HI, 1997; Ahonen et al., 2011). In the suggested new control strategy, the preferred operating area (POA) of each pump represents only the selected operating area between the set mark-ups in the pump performance curve. The mark-up points are selected based on the pump efficiency data and parallel pumping system details.

3.4.1 Control strategy and implementation to parallel pumping systems (Publication VII)

In this study, the improved energy efficiency of the variable speed controlled parallel pumps compared with the traditional control is striven by introducing a new control strategy for the parallel pump control. The introduced control strategy of parallel-connected pumps was designed based on the following requirements:

- The suggested control strategy should be able to work with as little amount of initial information as possible, even without additional sensors in the pumping system.
- Compared with the existing and known flow adjustment methods, the suggested control strategy should be able to reduce the energy consumption of the pumping system.

- The suggested control strategy should also prevent the inefficient or harmful operation with a higher risk of reduced pump service life of an individual pump when a certain flow rate is produced with parallel-connected pumps.

The possible harmful and inefficient operation in parallel pumping can be avoided if the preferred operation area (POA) of each parallel pump is taken into account in the control strategy. Thus, these risks can be controlled by preventing the pumps from operating outside the selected region during the rotational speed control if possible. Figure 3.20 plots the POA between the efficiency mark-ups at different pump rotational speeds according to the affinity laws. The area outside the flow limits in the *QH* axis can be described as High-H and High-Q range areas. The flow rate limits for the POA can be set using only the pump characteristics. To select the flow rate limits, the pump efficiency can be seen as a justified variable for limiting values, since centrifugal pump performance curves usually contain efficiency data (Sulzer, 1998; Karassik et al., 2001).

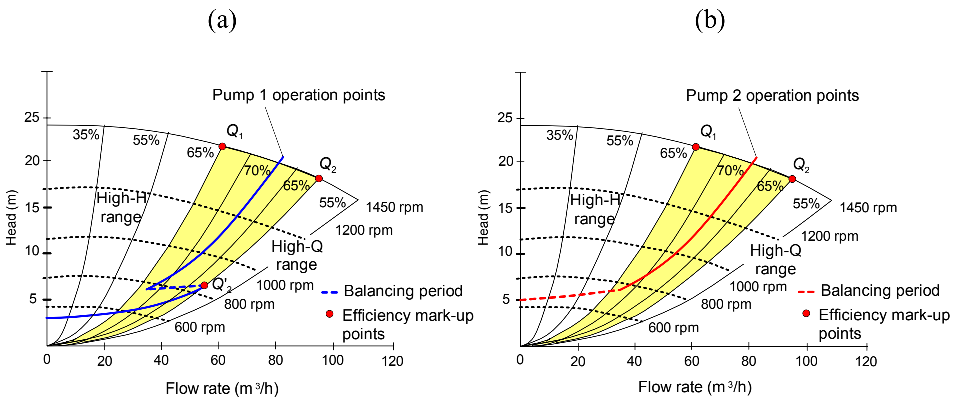

Figure 3.20. Operation points of Pump 1 (a) and Pump 2 (b) in the suggested control strategy when the flow rate of the system is increased to nominal. The rotational speed balancing of the parallel pumps is started when the operation point of Pump 1 reaches the set flow limit (Q_2) at (Q'_2). The area between the limit values Q_1 and Q_2 in the *QH* axis according to the affinity rules is the set preferred operating area (POA). The area outside the flow limits can be described as High-H range and High-Q range areas.

Observing the output of parallel pumps during operation is usually limited by the lack of metering in the pumping systems. However, information from the parallel pump operation can be gathered by utilizing the pump operation point estimation available in a modern VSD. Adequate flow metering of individual pumps applying VSDs in the suggested control strategy allows the adjusting of the pumped volume according to the process changes, and each pump can be monitored to operate in the selected POA. Therefore, a separate flow meter installation or start-up field measurements are unnecessary. The estimation of the operating point of the pump with VSDs in the suggested control strategy can be done either using only the pressure sensors for the inlet and outlet pressure measurements or utilizing the sensorless option based on the motor power estimate.

Since the suggested control strategy is based on the pump operation point estimation and POA, which in this case can be limited by the pump user based on pump efficiency data and process conditions, implementing the control strategy does not require any mathematical-optimization tools. Instead, the control can be set with a simple feedback control based on the pump output shown in Figure 3.21. As illustrated in Figure 3.21, the information on each pump's operation point and individual rotational speed data is gathered from the VSD supporting it. The VSD's flow monitoring can supply the head, flow rate, and power values of each pump based on the input power reference or measured head. Using the reference value for the total flow rate and selected preferred operating area for each parallel-connected pump in the current system, the control algorithm returns the reference speed for each pump drive.

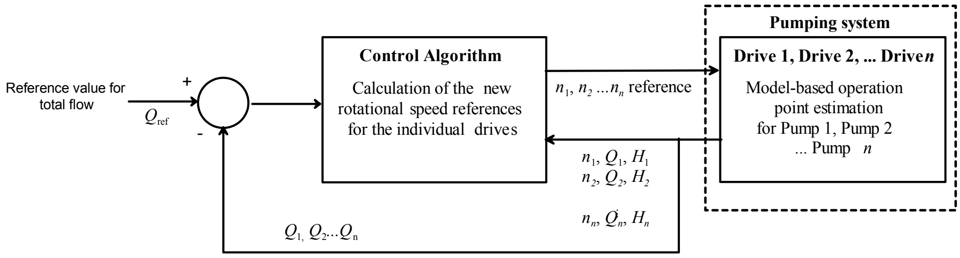

Figure 3.21. Implementing the control strategy to a parallel pumping system. The pump operation point estimation gives the pump output values to the control algorithm. The control algorithm determines the reference speed to each pump drive based on the monitoring data and the set preferred operating area of each parallel pump.

An example of the control algorithm to provide the suggested control strategy in parallel pumping systems was created. The block diagram of the prototype algorithm is illustrated in Figure 3.22.

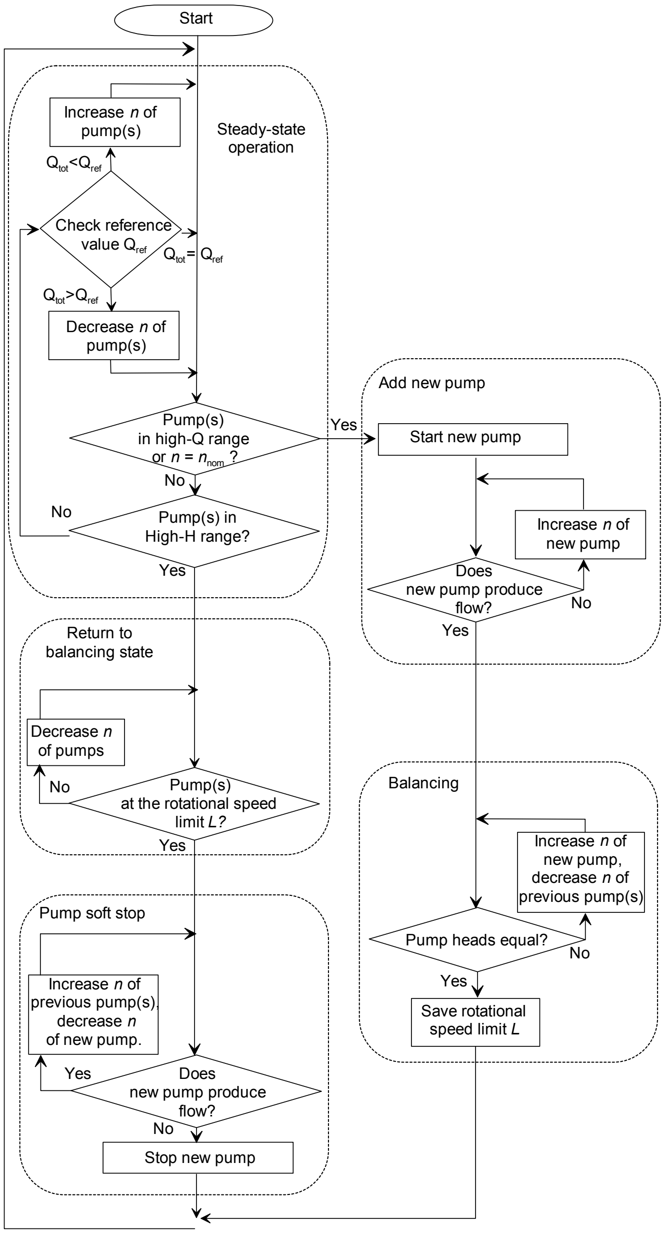

Figure 3.22. Block diagram for the control algorithm which can provide the suggested control strategy in a variable speed controlled parallel pumping system. The control step should be decided based on the pump nominal speed. The point marked as *Start* is the point when the first parallel pump is started.

The block diagram illustrates the control procedure in the case of increasing or decreasing the total flow of the parallel pumps according to the required output marked as Q_{ref} in Figure 3.22. In the block diagram, the system is started with just one pump regardless of the required flow rate. After starting, the output of the parallel pumps is adjusted to meet the process requirements based on Q_{ref}. In this case, the operation point of each pump (Q, H) is determined by utilizing the VSDs' *QH* curve-based flow estimation, although the head of the pump could also be determined with the *QP* curve-based method.

As mentioned previously, the mathematical-tool based optimization of rotational speeds of the parallel pumps may require start-up measurements and detailed system data (Bortoni et al. 2008.), which may have to be repeated or re-evaluated if there are changes in the system conditions, for example in the amount of the static heads or the shape of system curve. The implementation of the suggested control strategy into a parallel pumping system does not require start-up measurements, additional flow meters, or data related to the piping system characteristics, although the system conditions should be considered when selecting the preferred operating area in the pump *QH* axis. Thus, changes in the pumping system do not result in the re-editing of the control setup. Instead of the optimization of the speed of each parallel-connected pump, the energy efficiency and reliability is obtained by ensuring that the pumps are operated in the selected operating area, if possible. Including the POA as a control factor in the parallel pump control strategy can also ensure that the pump user does not have to decide whether the efficiency or reliability should have more value in varying conditions. Because of these qualities, the suggested control strategy can be justified in parallel pumping systems in which the requirements for a complete energy-optimization are not met.

The control strategy was tested in a laboratory system which contains two pump units; both of them include a single-stage centrifugal pump and a VSD connected to a three-phase motor. The primary pump (Pump 1) unit consists of a Serlachius DC 80/255 centrifugal pump, a four-pole 15 kW Strömberg induction motor, and an ABB ACS 800 frequency converter. The secondary pump (Pump 2) unit consists of a Sulzer APP 22-80 centrifugal pump, an ABB 11 kW induction motor, and an ABB ACS 800 frequency converter. Both VSDs estimate the individual flow rates using pump head measurements. The total flow rate is also measured using a Venturi tube. The pumps were connected in parallel, and the basic layout of the measurement setup is presented in Figure 3.23.

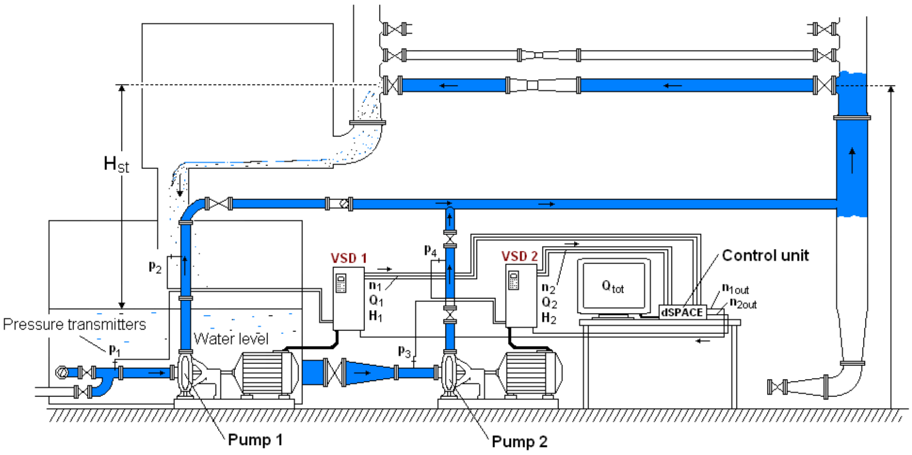

Figure 3.23. Test setup used in the laboratory measurements. The pressure transmitters are installed into the inlet and outlet section of each pump, and the pressure signals are wired to the frequency converters to enable the flow calculation. The control board, a dSPACE system, is attached to both VSDs. The values from VSDs' pump monitoring application; speed (n), flow rate (Q), and head (H) signals; are led from the VSDs to the dSPACE system. The determined speed commands (n_{out}) are transmitted to the VSDs from the dSPACE unit.

The control algorithm is implemented into a dSPACE DS1103 PPC controller board which was used as a separate platform for the control strategy in this prototype testing. The dSPACE board has analogue voltage inputs and outputs; the inputs for the controller board are the rotational speeds, heads, and flow rates of the individual pumps, and the total flow rate from the VSDs. The outputs of the controller board are the rotational speed references for the individual pumps. The sample time for the control algorithm was one second. In the laboratory measurements, the flow rate is controlled based on the requirement for more flow, less flow, or no change in the flow rate.

The static head of the piping system was 2.5 meters, and the system curve was set using valves located in individual piping branches so that both pumps would gain a reasonable efficiency when operating parallel at the nominal rotational speed. This illustrates a case in which a parallel pumping system is dimensioned according to the highest flow rate. The operating values of the parallel pumps in the test setup system are shown in Table 3.3. Since the pump systems have separate piping parts causing individual friction head to each pump, the head levels are not equal in parallel use (Table 3.3).

Table 3.3. Parallel pumping system in the laboratory setup.

		Pump 1 Serlachius DC 80/255	Pump 2 Sulzer APP 22-80
BEP *			
Speed (rpm)		1425	1450
Flow rate (m^3/h)		76	90
Head (m)		17.4	15
Efficiency (%)		69	73
Parallel operating point**			
Speed (rpm)		1448	1449
Flow rate (m^3/h)		91	83
Head (m)		16.3	17.6
Efficiency (%)***		68	70
Selected POA (%BEP flow)		70–130	70–130
*Operating values in a rated efficiency point according to the characteristics curves given by pump manufacturer ** Operating values (measured) of the parallel-connected pumps in a test setup system *** Based on the pump characteristic curves			

The operation of the presented control methods is simulated for the laboratory pumping system with a Matlab Simulink model. The model is constructed to enable the energy efficiency calculations of pumping systems and has been reported by Viholainen et al. (2009). A similar simulation model has been utilized to characterize the hydraulic systems also by Pannatier et al. (2010). In the simulation of this study, the performance, the combined power consumption, and the specific energy consumption of two parallel-connected pumps, having the same characteristics as the introduced pumps in the laboratory setup, are evaluated in a case in which the total flow of the pumping system is increased using either the traditional rotational speed control strategy or the presented new control strategy. In the following section the operation based on the new control strategy is represented as the alternative control.

3.4.2 Findings of Publication VII

The simulation was conducted with the flow rates from 0 to 189 m^3/h. The rotational speeds of the individual pumps using both control methods during a simulation sequence (0–1200s) are given in Figure 3.24.

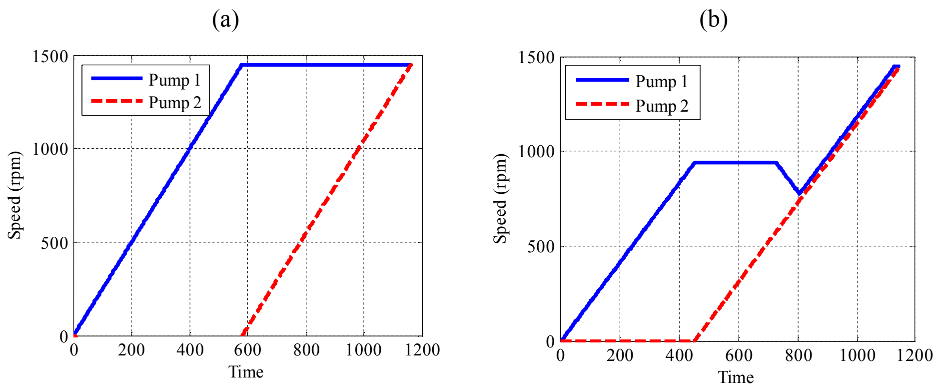

Figure 3.24. Operation speeds of two parallel-connected pumps in a system in the case of the traditional speed control (a) and the alternative control (b). In both cases, the pumps were operated to deliver the total flow rate from 0% to 100%. The time axis shows the direction of the increasing flow demand.

It can be seen that in the traditional control, the rotational speed of the primary pump (Pump 1) is increased to 1450 rpm, after which the secondary pump (Pump 2) is started and run towards the nominal rotational speed (Figure 3.24 a). When using the alternative control, the secondary pump is started before the primary pump reaches the nominal rotational speed because the primary pump hits the set flow limit (point A in Figure 3.25). This means a smaller flow rate difference at the secondary pump's starting point compared to the traditional control scheme. The simulated operation points of both parallel-connected pumps using either the traditional or alternative control are illustrated in Figure 3.25. The figure also shows the chosen flow rate limits for the alternative control algorithm based on the pump data given by the pump manufacturers.

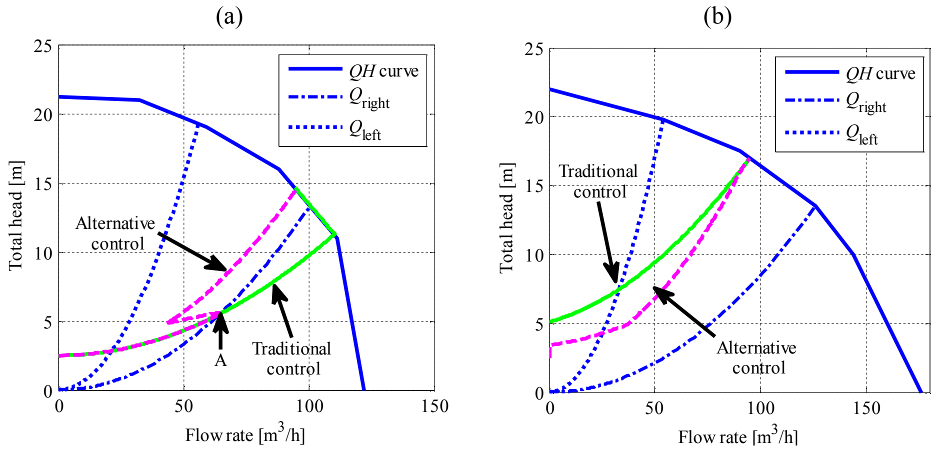

Figure 3.25. Simulated operation points of Pump 1 (a) and Pump 2 (b) using either the traditional or the alternative control. With the alternative control, Pump 2 is started when the operation point of Pump 1 reaches the set flow rate limit (Q_{right}) in point A. When Pump 2 starts to deliver flow, the speeds of both pumps are balanced to have the same head.

Figure 3.25 shows that even though traditionally controlled parallel pumps are operating in the same operation point as in the alternative control when both pumps have reached their nominal rotational speed, the alternative control enables the continuous operation between the set flow rate limits. Therefore, the operating points, especially in the case of Pump 1 (~65–90 m^3/h) shown in Figure 3.25 (a), are located in a better efficiency area compared with the traditional rotational speed control. Because of the balancing, the duty point of the secondary pump is located only temporarily in an unwanted region, and the actual operation (~40–90 m^3/h) takes place between the set limits (Figure 3.25 b). During the balancing period, the primary pump is always delivering flow and head, and hence, the secondary pump (Pump 2) can generate a flow rate only when it has exceeded the required head (~4 m). However, the required head for the secondary pump can be smaller than the primary pump's total head, since the friction head values for both pumps are not necessarily equal during the control.

The benefit of the alternative control can be seen best when observing the total pump power consumption and the specific energy consumption of both parallel pumps in the same simulation (Figure 3.26).

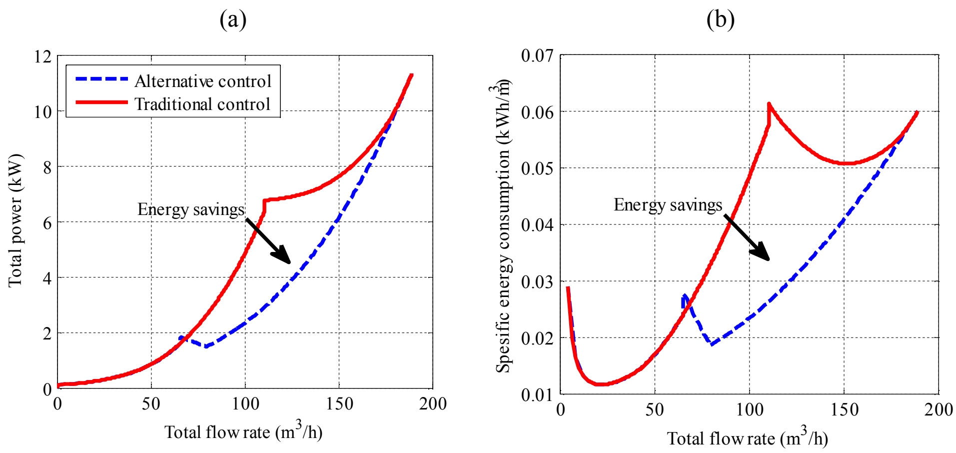

Figure 3.26. Simulated total pump power (a) and specific energy consumption (b) of both pumps according to the total flow. The figure plots the simulated values in the case of alternative control and traditional control. Energy savings using alternative control can be found when operating in a flow region in which the electric power use and specific energy consumption is lower compared with traditional control.

The results suggest that in this particular case, the alternative control enables much lower power usage and specific energy consumption in the flow range of 70–175 m^3/h compared with the traditional control. Outside this range, the energy consumption was equal (Figure 3.26). However, the difference in the energy use seems to be more than 50% at the highest point (110–120 m^3/h) between the alternative control and the traditional rotational speed control.

The new parallel pump control strategy was tested in an actual pumping setup using measuring sequences in which the flow rate was increased using the rotational speed control of parallel pumps. The total flow of both pumps varied from 0 to 175 m^3/h during the sequences. These values represent the minimum and maximum total flow rate values of the parallel pumps in the used system conditions. The measured operation points of each pump represent the average values gathered manually from the data control unit and the measuring equipment.

Figure 3.27 plots the test results of the sequences in which the flow rate is increased from zero to maximum using either the traditional or alternative control. Figure 3.27 (a) shows the measured operation points of the primary parallel pump when the total flow of the system is increased from 0 to 175 m^3/h. The balancing of Pump 1 starts when the flow rate reaches the set mark-up line (Q_{right}) in the alternative control. When the traditional control is used, the speed of Pump 1 is adjusted to the nominal rotational speed before Pump 2 is started. Figure 3.27 (b) shows the Pump 2 operation points.

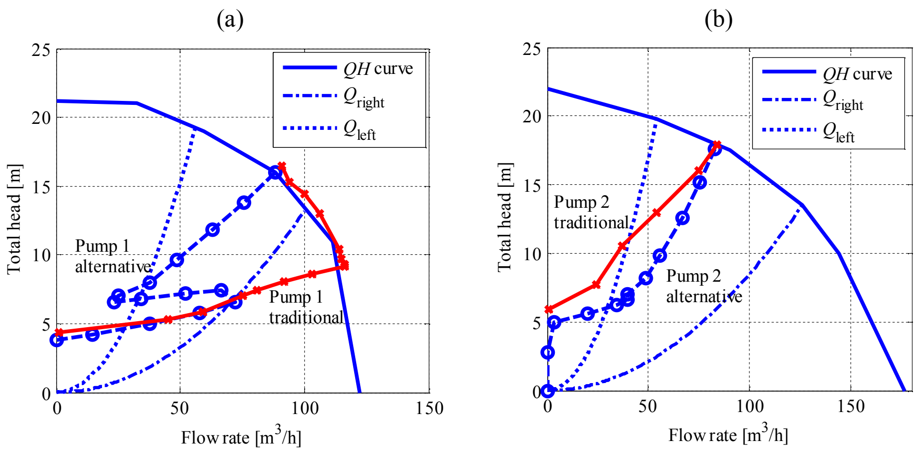

Figure 3.27. Operation points of Pump 1 (a) and Pump 2 (b) during the alternative control and traditional control measuring sequences.

It can be seen from Figure 3.27 that Pump 1 and Pump 2 are not operated on the same head in a point when Pump 2 starts to deliver flow. The reason for this are the losses in the elbows and valves in the system which cause that the dynamic head losses in individual piping parts are significantly different for Pump 1 and Pump 2 in this particular parallel operation point. However, a brief look at Figure 3.27 shows that the alternative control is operating the parallel pumps as simulated.

Since the laboratory equipment used in this study does not include the measurement of the pump shaft power, only the consumed total input power to each drive during parallel pumping was estimated using the input power reference of the VSDs. The results of the estimated total input power of both drives during the traditional and alternative control measurement sequences are illustrated in Figure 3.28. Figure 3.28 shows that in contrast to the simulations, the measured total flow rate is not increasing during the balancing period (~75 m^3/h). Despite this, the advantage of the alternative control compared with the traditional control can be seen in the total power consumption and in the specific energy use.

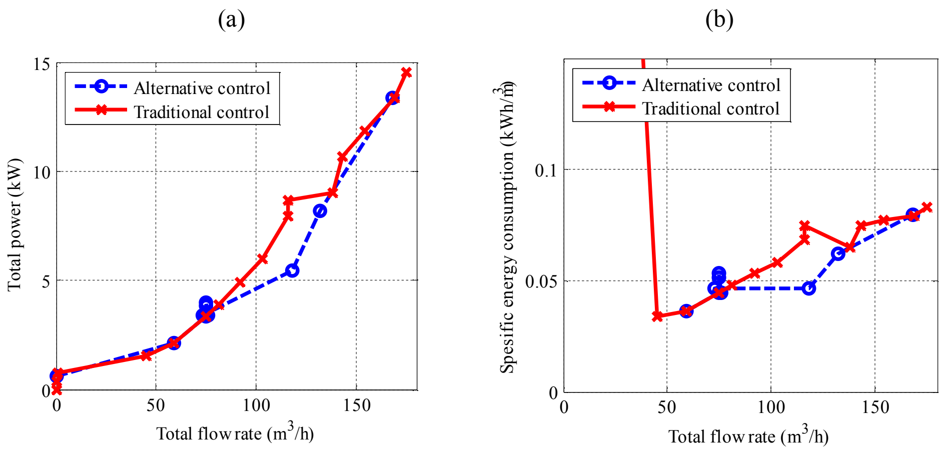

Figure 3.28. Estimated total input power (a) and specific energy consumption (b) of parallel-connected pump drives in the alternative control and the traditional control.

Even though the estimated total input power rates during different control schemes are not directly comparable with the simulated pump shaft power values, the measured results seem to agree with the simulations. The results suggest that in this case the alternative control seems to reduce the combined input power consumption and the specific energy use up to 20–25% on the flow rates from 80 to 160 m^3/h (Figure 3.28). The benefits of using the alternative control can also be seen in the higher system efficiency of parallel pump drives illustrated in Figure 3.29.

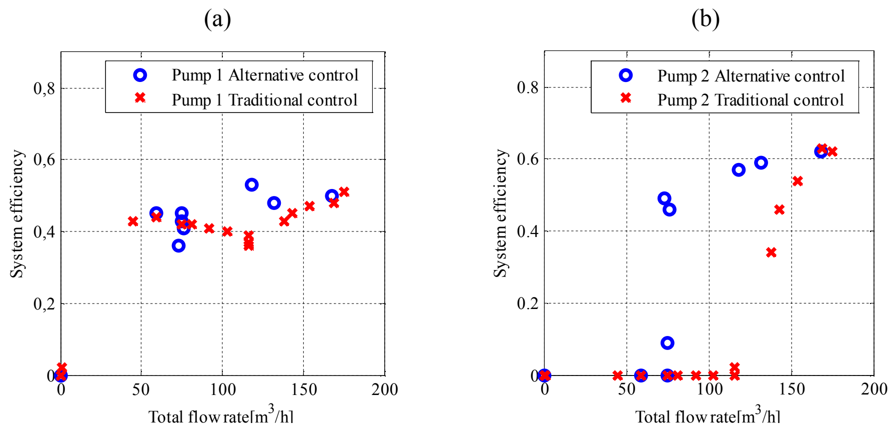

Figure 3.29. Estimated system efficiency of parallel-connected pump drives in alternative control and traditional control. The system efficiency of Pump 1 (a) and Pump 2 (b) drives are shown according to total flow rate of both pumps, showing the variation of system efficiency during measuring sequences.

4 Discussions

In this section, the findings of the research are discussed. The conducted research and main results are collated with other studies related to the focus areas. The practical benefits of the findings are also presented in this section, as well as the recommended research in future.

4.1 Suitable operation conditions for VSD controlled parallel pumps

A great variety of variable speed controlled parallel pumping systems can be easily found in water delivery, waste water treatment, petrochemicals, irrigation, etc. (Szychta, 2004b; Bortoni et al., 2008; Yang and Borsting, 2010; Zhang et al., 2012). Even though using VSDs has shown significant potential to reduce the energy consumption in pumping, this potential is always bound to the system details of each pumping task. One major reason for the energy saving potential in VSD controlled pumping lies in the fact that in many instances the pumping system is not optimally dimensioned according to the process needs (Hovstadius et al., 2005; Moreno et al., 2007; Kaya et al., 2008; Pemberton and Bachmann, 2010). Another main reason for the energy efficiency potential of using VSDs is the broadly studied fact that rotational speed control of pumps can often enable flow adjustment of the pumping system output with less energy use compared with other adjustment methods (Europump and Hydraulic Institute, 2004; de Almeida et al., 2005).

The findings related to the first research study (Publication I) demonstrate an example of the energy saving potential in a parallel pumping system. The illustrated example shows the energy calculations of two condenser pumps driven by 315 kW electric motors. In this study, the energy saving potential of using the VSD control instead of valve adjustment is a consequence of achieving more energy efficient control of the system output. The calculations show that the fixed-speed operation of parallel pumps enables noteworthily the system efficiency of approximately 73−78% when using the valve adjustment to deliver the flow range of 670−1140 l/s. Correspondingly, the system efficiency in VSD control can be 5−15 percentage units smaller, but energy is still saved when delivering the same output range. The benefits of the VSD control can be seen when determining the specific energy use of both adjustment scenarios, which suggest smaller energy use per pumped volume when using the VSD control compared with the valve adjustment (Figure 3.3). This occurs, since less energy is wasted on the increased head compared to the valve control.

As mentioned previously, the studied pumping system is just one example of achieving energy efficient operation of pumps with VSDs, but it supports the importance of a systematic approach when revealing the energy saving potential in pumping. As concluded in the study by du Plessis et al. (2013), examples of the energy saving potential are also important to ensure that the decision makers will consider the implementation of the VSDs in pumping systems. Although it is still sometimes remarked that pumping system optimization is all about maximizing the efficiency (Pemberton and Bachmann, 2010; Kini and Bansal, 2010), the main point should be in fulfilling the pumping task with reduced energy consumption instead. This is demonstrated with this example of an industrial pumping case in quite a similar way as in the study by Marchi et al. (2012). In the referred study (Marchi et al., 2012), the energy efficiency improvements were achieved with the VSD control compared with the on-off control scheme, even though there was a decrease in the system efficiency.

Observing the energy efficiency of VSD controlled pumps with specific energy consumption was discussed in the second research study (Publication II). In this study, the effectiveness of the pump drive train (pump, motor, and VSD) is not evaluated with the occurring losses when transferring the electric energy into the liquid, but by evaluating the consumed energy when delivering a certain output in the flow rate. In theory, the major advantage compared with the system efficiency evaluation is that also the energy used in the friction head is taken into account when evaluating the pumping system effectiveness. The disadvantage, however, is that the specific energy use is bound to the system details in every pumping scenario, and the difference in the effectiveness between the pumping systems is hard to evaluate.

The shown exemplary limit for the BEA shown in Figure 3.5 does not consider the reliability issues of the pumping. Based on known studies (ANSI/HI, 1997; Stavale, 2008; Bloch and Budris, 2010), operating the pump at the nominal speed and high relative flow rate decreases the pump reliability, as the magnitude of vibration and pump suction pressure increases. As a rough estimate, applying these findings in the studied example could limit the operation regions having a high relative flow rate (≥120%) and rotational speed (≥1450 rpm) outside the recommendable conditions; not because of poor energy-efficiency, but the high risk of harmful events (cavitation, flow recirculation) which can reduce the reliability of the pumping.

The results of the second research study can nevertheless point out the relevant factor in the VSD controlled parallel pumping systems: in certain conditions it is more energy efficient to operate with a higher number of pumps with reduced speed than a smaller number of pumps with increased speed. This can be seen as one of the major motivations for optimizing the operation of VSD controlled parallel pumps in general. In varying system conditions, the optimal operation should not be determined only in terms of energy efficiency, since the operational reliability affects the life-cycle costs of the pumping (Ahonen et al., 2007; Pemberton and Bachmann, 2010; Augustyn, 2012). Even though optimizing the reliability of the VSD controlled pump is not considered as a target in this study, it is justified to take into account the suggestion for avoiding certain operation regions in the pump *QH* axis (ANSI/HI, 1997; Martins and Lima, 2010). The results also show that there can be a risk of reduced reliability (e.g. cavitation) if the pumping is optimized with the specific energy consumption only. From this perspective, the pumping efficiency has a better connection to the pump reliability than specific energy use, as the risk of pump failures is typically reduced when the pump efficiency is optimized (Bloch and Budris, 2010; Martins and Lima, 2010).

The results of the first and the second research studies (Publication I and Publication II) clearly suggest that from the available options to deliver a certain output with VSD controlled parallel pumps, some operation conditions are more recommendable than the others, especially from the energy efficiency point of view. The findings clearly indicate that when the pump output has to be adjusted to meet the process needs, the pump efficiency or system efficiency should not necessarily be maximized, since energy efficient operation conditions can be achieved also when operating the pumps at lower rotational speeds.

The main results of the third research study (Publication III) are not related to the calculations of this issue, but to present a base-line for the control strategy which can ensure the operation of VSD controlled pumps with justified energy efficiency and pump reliability. The benefits of using rotational speed controlled pumps with load-sharing operation instead of staggered operation has already been recognized (KSB, 2004; Jones, 2006), and the calculations of the second study (Publication II) also support the idea of running for example two parallel pumps with a lower rotational speed rather than one pump with a higher rotational speed in certain situations. The real challenge in realizing the energy saving potential of the load-sharing in VSD controlled pumps is creating a control strategy which can execute the desired operation: starting and stopping the pumps at the right moment and setting the output according to the process needs. In the presented base-line for such a strategy, the load-sharing operation of VSD controlled parallel pumps is executed by determining a certain preferred operation area for each parallel pump.

The preferred operation area (POA) can be determined based on the pump efficiency, as it is information which is typically available for pump users. However, the process demands should also be considered when selecting the desirable operating range. From a different perspective, the pump user may alternatively select the area of unwanted operation at lower flow rates where there may be a risk of poor energy efficiency and shut-off. In addition, operating at high flow rates can increase the risk of cavitation, and therefore, the operation can be limited in this range as well. As a result, the POA is located between these set limits (Figure 3.20). Hence, running pumps at the selected POA suggests that parallel-connected pumps are operated with justified energy efficiency and reliability when delivering the required output for the process. If only the pump characteristics are considered, the rough estimate for the POA can be for example the recommendable operation region of 70−120% of the BEP suggested by Martins and Lima (2010), but the system characteristics should be taken into account to ensure the justified operation. In this case, the concept of the POA of each parallel pump represents the selected operating area for the current pumping task, not a universal recommendation for the pump operational region.

4.2 Monitoring the pump output with model-based methods

Monitoring the pump operation point is an effective base to evaluate the overall performance and reliability of the pumping in variable-flow conditions (Moreno et al., 2007; U.S. Department of Energy, Hydraulic Institute, 2006). Still, direct measuring of the operation point of the pump is often not provided in pumping applications due to unwanted investment costs or system limitations. In VSD controlled pumping tasks, it is possible to determine the pump output utilizing indirect measurements and a model for the pump performance in variable-conditions. In this thesis, such indirect pump monitoring methods are referred to as model-based methods.

The aim of the second research area was to investigate the usability of the model-based methods to determine the output of each parallel pump in a VSD controlled parallel pumping system. In the first study of this research area (Publication IV), the functionality of the basic model-based methods *QH* and *QP* were observed in a pumping laboratory comprising two parallel pump units and a piping network with water tanks. The results given by the model-based flow rate monitoring were compared with basic laboratory flow meter results. In the second study (Publication V), a similar experiment was conducted with two sensorless operation point estimation methods (*QP* method, Process curve method). These methods were tested in the same laboratory with a single pump drive train. The results were again compared with laboratory equipment measurements. In the third study (Publication VI), several model-based methods (*QH*, *QP*, Kernan, Hybrid, *QH*/*QP*) and their combinations were tested in a blower system comprising an industrial blower, induction motor, and a VSD installed in a pipeline system. The flow rate results given by the methods were compared to the measured values of the flow meter. The reliability of the results can be regarded as sufficient, as the tested equipment included two different pump units and a separate blower system. Also the flow estimation results were compared with the results given by several different flow meter types.

The main conclusion of the first research study (Publication IV) is that based on the results, the basic *QH* method was able to determine the flow rate of the individual VSD controlled parallel pumps with a reasonable accuracy (5−10%) compared with the measured flow rate. In this study, the *QP* method showed more variation in the flow metering accuracy. The results suggest that some factors can be linked to possible inaccuracies of these model-based estimation methods. Probably the most relevant factor is the shape of the *QH* and *QP* characteristic curves since the dH/dQ and dP/dQ ratio affects the accuracy of the interpolation of the values. If the curves are flat, even a small error in the determined head or power value can result in a significant divergence in the estimated flow rate. The accuracy of the head and power estimate is naturally linked to this same issue. In this case, the performance curves must also be monotonic; for instance, if there are two flow rate values for the same head or power value, the presented estimation cannot give reliable results. In addition, the allowed tolerances for the characteristic curves given by the manufacturer can decrease the accuracy of these methods. Based on this study, the *QH* calculation can be seen a more accurate and versatile solution in parallel pumping systems since the pressure metering caused less oscillation in the flow estimation compared with power-based interpolation. Also, the allowed tolerances for the *QH* curves in ISO 9906 standard are 0.5 percentage units smaller than in the *QP* curve case (ISO 9906, 2012).

The overall conclusion based on the results of the second research study (Publication V) is that both tested sensorless estimation methods (*QP* and Process curve) are applicable to auditing and pump operation monitoring purposes. The tests in this second study indicate that the operation of the motor can be monitored with the VSD's internal estimates with reasonable accuracy especially when the pump drive is operating in the 100% flow region. This leads into the conclusion that inaccuracies in the pump characteristic curves have probably a more severe effect on the *QP* curve-based operation point monitoring than the inaccuracies in the VSD's internal estimates. Still, the relative error of the estimates increases when operating outside the 100% flow region, which reduces the accuracy of the *QP* curve-based estimation. As mentioned earlier, another drawback in the *QP* curve-based estimation is the limiting effect of d*P*/d*Q*, which can exclude the use of the *QP* method in certain pump types and operating regions. The results indicate that the process curve-based estimation can give an accurate estimation suitable for control purposes. The suggested method, however, can be seen justified only in the pumping systems in which no changes in the process curve are expected.

In the third research study (Publication VI) case, a similar test as in the previous studies of this research area was made in a variable speed controlled blower system. The measurements were conducted in a laboratory system with an adequate straight duct before and after the blower. In real-life blower systems, such ideal conditions are rarely available, which can have an effect for example on the accuracy of the pressure measurements. In this study, there was a clear difference between the measured operation and characteristic curves given by the manufacturer in certain flow regions, which reduces the congruency between the studied pump/blower model and actual system. Despite this limitation, the test results indicate that the accuracy of the characteristic curve-based flow rate estimation methods can be improved with the combined use of different methods. As the studied flow rate estimation methods in a blower system can be applicable also in pumping systems, similar results can be expected when studying the suitability of the methods for controlling variable speed driven parallel pumping processes. As a conclusion, calculating the uncertainty of the *QH* and *QP* method and selecting the most suitable method for flow estimation, and combining both methods in certain cases, can enable justified pump operation point monitoring in systems in which the applicability of other studied model-based methods is limited.

The tests of these three research studies (Publication IV, V and VI) show that important information related to the energy efficiency and reliability of the VSD controlled pumps can be acquired using model-based output estimation methods. The biggest challenge in accurate output monitoring is not related to indirect measurements (pressure metering, power estimation) but to possible inaccuracies of the pump model. As the pump model is based on the pump characteristic curves at rated speed and using the affinity laws to generate a series of new curves at a varying speed, the error in the affinity transforms can decrease the accuracy of the output monitoring. In parallel pumping systems, however, the system curve is typically flat, and a significant part of the total head of the pumps is caused by the static head (White, 2003; Volk, 2005). This often indicates that even a small change in the rotational speed of the parallel-connected pumps can have a large impact on the delivered flow rate. Thus, operating in a wide range of rotational speeds for pumps is less likely in parallel pumping systems, even though there might be a large variation in the process output. This has also been stated in the studies by Szychta (2004a) and Zhao et al. (2012). From this point of view, inaccurate affinity transforms in characteristic curve-based estimation can be seen less fundamental in variable speed controlled parallel pumping. The use of the pump model can also be justified only if there is no significant difference between the manufacturer curve and actual operation. On the other hand, the tolerances of the curves are standardised in ISO 9906 (ISO 9906, 2012), but it is clear that the shape of the curves can be affected by the pump wear. In these situations, the only available choice may be the measurements on the pumping site with for example a portable flow meter and the determination of the performance curve with such measurements.

Based on the gathered results, many of the tested methods can be seen justified for control purposes, but with clear limitations. Hence, applying for example sensorless methods especially in parallel pumping systems can be seen sensible as supporting methods along with possibly more adaptable methods. Combining these methods can therefore extend the usability of the VSD's operation point monitoring to a wider range of pumping systems. As there is typically a high probability that there is a variation in the values of the static head or friction losses in many parallel pumping applications (water delivery, waste water pumping), the applied operation point estimation methods must be able to adapt to the changing system characteristics (Yang and Borsting, 2010). Because of this, pressure metering-based flow estimation, in addition to the sensorless flow metering methods, can be recommended in pumping systems in which the flow estimation is used also for control purposes.

Motivations for using model-based flow estimation methods in control purposes especially in variable speed driven parallel pumping systems can be described:

- A chance to monitor the output of each parallel-connected pump without separate metering
- VSDs already contain the required indirect measurements with sufficient accuracy (power, rotational speed) excluding the pressure measurements
- The functionality of the utilized pump model is not affected by inaccurate affinity transforms, since the required rotational speed range is often relatively small

These factors support the idea that, utilizing model-based flow estimation methods, the operating areas suggesting harmful or inefficient operation conditions in parallel pumping can be identified and furthermore avoided based on the selected control strategy.

4.3 Implementing energy efficient control strategy in parallel pumping systems

In the third research area, the implementation of the energy efficient control strategy in VSD controlled parallel pumping systems was studied (Publication VII). A prototype for a control logic based on the POA of each parallel pump and the utilization of VSDs flow monitoring abilities was created. The functionality was evaluated with simulations and laboratory measurements.

The simulation results show that using the suggested new control strategy (*alternative control*) results in the improved energy efficiency in pumping compared with the traditional rotational speed control strategy since the same flow rate can be delivered with a lower energy use (in the flow range of 70−175 m^3/h). In the illustrated examples, the alternative control enables parallel pumps to operate in the set POA on the *QH* axis (Figure 3.25), which in this case was defined simply as an area between the set efficiency limits based on the pump characteristics.

The benefits of the suggested control strategy were verified also by laboratory measurements with an actual parallel pump setup. In the laboratory measurements, the amount of saved energy was 20–25% at highest (in the flow range of 80−160 m^3/h). The measurement results showed (Figure 3.28, Figure 3.29) that in that flow range (80−160 m^3/h) when using alternative control, the parallel pumps are located in an area which results in the improved system efficiency compared to the situation in which the same total flow is delivered with traditional control.

In the laboratory measurements, the control procedure including the prototype control algorithm based on the suggested control strategy was tested using a separate controller board (Fig. 8), but the introduced method could also be implemented in the VSD software. Although the measurements indicate quite a similar operation as simulated, differences were found in the shut-off heads of the pumps and in the point in which Pump 1 reaches the set efficiency limit. The reasons for these differences can be the inaccuracies in the pressure metering, resulting in a further error in the flow metering of the VSDs. Despite this, the collected data support the assumption that the presented control strategy could be implemented in VSD controlled parallel pumping systems without separate flow metering devices or field measurements except the pressure sensors for the flow metering of the VSDs. The results also showed that the parallel pumps do not have to be identical, although significantly dissimilar pumps may need further considerations.

The tests support the conclusion that model-based flow metering by VSDs can be utilized for control purposes. The prototype testing was performed using VSDs which are known to have applications to estimate the flow rate of the controlled pump. In this case, *QH* curve-based flow estimation was used. Without providing similar monitoring of the pump output with VSDs, the use of the introduced method would need additional metering of the pump flow rate. It is clear that because the presented parallel pump control strategy is based on the VSD's pump system monitoring applications, its adequate operation depends on the monitoring accuracy. Hence, the limitations of the used model-based flow estimation can exclude the implementation of such control strategy in the case of certain pump types or system scenarios.

It should be noted that the energy saving potential of a certain control strategy in VSD controlled parallel pumping is always system-specific. Hence, using a similar strategy in a different system scenario would not necessary result in the same savings in energy use. Despite this, the results indicate that when using the introduced strategy, it is possible to run pumps in a region which suggests justified energy efficiency and pump reliability. The actual energy saving potential in a certain system is, however, possible to determine with energy calculations and audits (Aranto et al., 2009; DeBenedictis et al., 2013; du Plessis et al., 2013). Based on the results, the introduced control strategy for parallel pumps can be justified especially in parallel pumping systems with a varying flow need, relatively flat system curve, and when the pumping systems are dimensioned according to the highest flow rate.

In the case of many referred optimization strategies (Bortoni et al., 2008; Yang and Borsting, 2010; Zhao et al., 2012), the optimization of VSD controlled parallel pumps is based on the already discussed pump model and a system model, often illustrated with a system curve. The structural basis of the presented strategy of this study is very similar to the method presented by Bortoni et al. (2008), in which the pumps are controlled to operate as close as possible to the BEP. The major difference between these methods is that the control steps are based in the mathematical-optimization presented by Bortoni et al. (2008), as the presented strategy in this research utilizes the determined POA.

In the theoretical perspective, the control strategies studied for example by Bortoni (2008) and Zhao (2012) are actual optimization methods, as the presented strategy based on the POA can be seen more as a sub-optimization method. It should be noted that the sub-optimization of parallel pumps based on the POA can be beneficial in pumping tasks, in which the presumption of the constant and stabile system curve is not valid. In such cases, the system characteristics cannot be described as a constant curve, and optimizing the operation of parallel pumps may require additional measurements to determine suitable operation conditions (Yang and Borsting, 2010).

A challenge for the justified control is to set the POA based on the pump data only since the efficiency data is not the only relevant factor when determining the preferred operation region of the pump. The pumping process can set limitations for instance to the minimum flow and pressure rate, and the dimensions of the system and the parallel pump units naturally dictate the available operating range. Also, higher pump rotational speeds which can increase radial and axial forces in the pump and thereby affect the pump mechanical reliability, should be taken into account for a more systematic approach. Importance should also be given to the run-time balance of the parallel pumps to avoid unnecessary starting and stopping of the pump units. Determining a general recommendation for the POA in VSD controlled pumping is not considered a target of this study since the pump design issues are considered to be out of the scope of this study.

4.4 Connective implications of the results

The presented results indicate that the energy efficiency of the VSD controlled parallel pumping systems can be improved by selecting the most suitable control strategy for the pumping task. The illustrated examples demonstrate that lower specific energy use and decreased risk of pump failure can be achieved if all parallel pumps are rotational speed controlled compared with the situation in which only certain pumps are VSD controlled.

The studies point out that the modelling of the pump performance in varying operation conditions using performance curves and affinity rules (Bortoni et al., 2008; Yang and Borsting, 2010; Zhao et al., 2012) can also be utilized to pump output estimation. This estimated output can also be used as an important feedback signal in the control strategy, allowing energy efficient operation in varying system conditions. The concept of the proposed control strategy is illustrated in Figure 4.1.

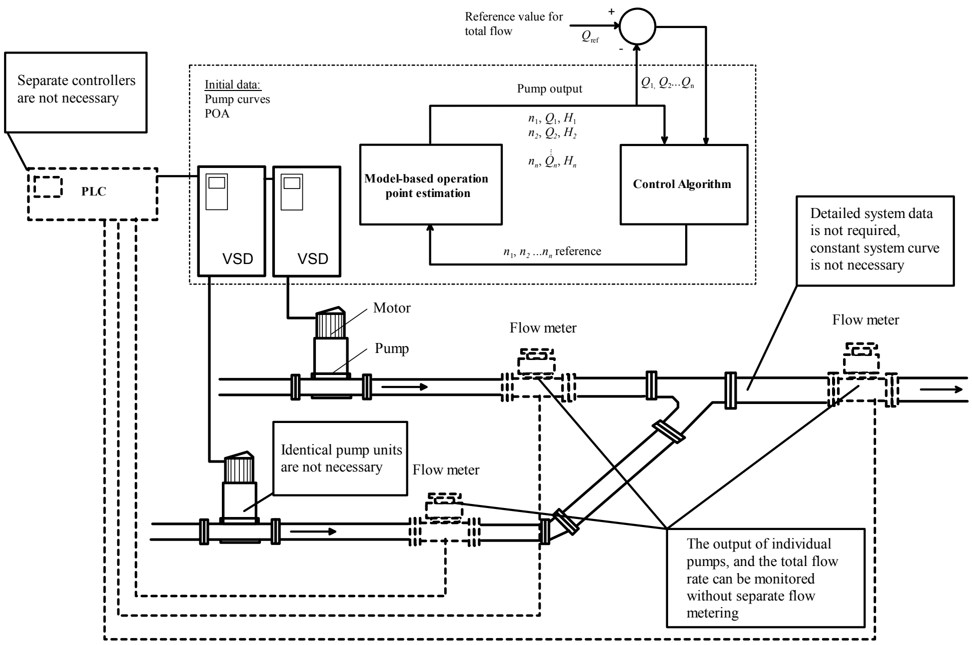

Figure 4.1. The findings related to the proposed control strategy for VSD controlled parallel pumps.

The major differences of the proposed control strategy compared with other available options (Bortoni et al., 2008; Zhao et al., 2012) are also highlighted in Figure 4.1. The proposed control strategy does not require start-up measurements or detailed system data (static head, system curve) or any kind of separate flow metering to be implemented in VSD controlled parallel pumping systems. Also, the identical pump units are not necessary in the proposed control strategy; a benefit that results from the fact that the output of the individual pumps can be monitored with model-based estimation methods. Because of these differences, the proposed control strategy can be justified in parallel pumping systems in which the available data or system specifications do not support the methods for complete energy-optimization. Additional benefit in the presented concept is that the control logic is possible to be implemented directly to VSDs without using separate Programmable Logic Controllers (PLCs), although this can be done also with other control strategy options. In other words, if the VSDs are already provided in the parallel pumping system, the conducted results support the idea that energy efficiency improvements are possible only by switching to the suitable control strategy and without installing additional devices.

Despite all the energy saving potential in VSD controlled pumping systems, energy cannot be saved with any control strategy unless the adoption of the strategy is easy and feasible enough. From this perspective, utilizing already available devices and solutions with more intelligence can be seen more attractive from the pump user's point of view. Thus, easily implementable control schemes with as less initial data as possible can direct the energy efficiency improvements to the industrial and municipal field with less effort.

4.5 Recommended research

Even though the benefits of the control strategy could be verified with laboratory measurements, more results of realizing energy-savings in real-life pumping systems could be gathered with simulated and experimental system scenarios. Especially case examples for pumping systems having more than two parallel pumping units would give interesting information of the benefits of the proposed strategy. In this study, the functionality of the presented control strategy was tested using *QH* curve-based pump flow rate estimation. To allow a completely sensorless control of parallel pumps, *QP* curve-based methods should be used.

The accuracy of the used pump operation point estimation methods was evaluated to be sufficient for the presented sub-optimization of VSD controlled parallel pumps. However, the model-based methods could also be applied in similar optimization strategies referred to the studies by Bortoni et al. (2008), Zhao et al. (2012), and Steger and Pierce (2011). It would be interesting to see whether the specific energy consumption of the parallel pumping system could be optimized with the sensorless flow estimation of individual pump units. The introduced control strategy does not concern the efficiency of the motor and VSD, but the losses of the motor-drive package could be taken into account by optimizing the specific energy consumption of a pump drive train. Thus, the effect of oversized motors and drive could be studied from the control perspective in such a study.

5 Conclusions

In this study, an energy efficient control strategy for variable speed driven parallel pumps is presented. The study focuses on pumping systems in which the pump units comprising a pump, induction motor, and a frequency converter are operated in parallel for achieving a wide range of flow for the process.

In variable speed controlled parallel pumping systems, the required output for the process can often be delivered with several options. The requirements for the justified control strategy in these situations include the energy efficient operation and also the avoidance of conditions which can reduce the service life of the pump. The major challenge when determining energy efficient control strategy for the variable speed controlled parallel pumping systems is the lack of required information for energy optimization. Even though the operation of the variable speed pumps in varying conditions can be estimated by adaptive pump models, the pump user may be aware of the system details only to a limited extent. Also, the system characteristics tend to change in many parallel pumping applications. Correspondingly, real-time information from the process output is rarely available in parallel pumping systems, and direct flow measuring of each parallel pump unit is even more exceptional. These factors decrease the chance of determining the most energy efficient operation conditions for the pump units, and there is a distinct possibility that the requirements for justified system optimization are not met.

In this thesis, the determining of energy efficient control strategies for VSD controlled parallel pumping systems is discussed by focusing on three main research areas. The first focus area considers the suitable operation conditions for VSD controlled parallel pumps. In this first focus area, the potential for achieving energy savings in variable speed controlled parallel pumping, the energy-efficiency-based suitable operation regions for VSD controlled pumps, and the benefits of avoiding unwanted operation of parallel pumps when selecting certain control scheme, are studied.

The second focus area presents the available methods to determine the individual output of parallel-connected pump units using performance curve-based pump operation monitoring applying VSDs. In this focus area, the basic options for monitoring the flow rate of each parallel pump in a system, the sensorless methods for monitoring the operation point of a pump in a system, and the possibilities of combining model-based estimation methods to achieve more accurate pump operation monitoring are studied.

In the third focus area, methods for implementing energy efficient control strategy in a VSD controlled parallel pumping system are discussed. The aim in this focus area is to introduce an option which can ensure the operation of the variable speed controlled parallel pumps with justified energy efficiency without the use of separate flow metering devices or detailed system information.

The gathered results on these research areas indicate that the connection between the pump operation point and energy efficiency of the pumping can be utilized in the control strategy for VSD controlled parallel pumps. Thus, determining the possibly inefficient or harmful operation regions and selecting the preferred operation area (POA) as a base-line for the control strategy can ensure the operation which suggests justified energy efficiency and reduced risk of pump failure. The results of the second research demonstrate that the output of VSD controlled parallel pumps can be estimated with performance curve-based methods. This estimated output can also be used as the required feedback signal in the suggested control strategy based on the POA. According to the results of the third research area, the suggested control strategy can be implemented in a VSD -controlled parallel pumping system with limited system data and without separate flow metering devices. The introduced strategy is based on the sub-optimization of the parallel pumps; therefore the suggested method does not require a separate model for the system conditions, which is often the base for complete optimization.

The benefits of energy efficient control can be seen best when evaluating the life-cycle costs (LCC) of pumping. As the energy costs usually dominate the LCC in pumping applications, fulfilling the pumping task with reduced energy consumption is one of the main interests of the pump user. The discussed methods in this thesis concentrate on the systems in which all parallel pumps are equipped with VSDs. Therefore, a suitable application for the presented control strategy can be found for instance in systems in which the previous control method is updated to the rotational speed control of pumps.

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Publication I

Viholainen, J., Kortelainen, J., Ahonen, T., Aranto, N., and Vakkilainen, E.
Energy efficiency in Variable Speed Drive (VSD) controlled parallel pumping

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Energy efficiency in Variable Speed Drive (VSD) controlled parallel pumping

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Abstract

Energy losses in pumping processes can be categorized into adjustment losses and dimensioning losses. Energy savings can be realized, if at least one of these losses can be diminished. In many instances, adjusting the pump speed has proven to be a good solution in reducing adjustment losses in pumping. Likewise, using speed adjustment with Variable Speed Drive (VSD) can be one possible solution to decrease dimension losses in pumping systems.

Realizing the energy saving potential from both losses requires the observation of all the pumping process components - pumps, motors, drives and system conditions. The efficiency of a pump is usually quite easy to estimate, even when it is operating in off-design conditions. The efficiencies of the motor-VSD combinations in pumping processes may not be as easy to take into account. Even though the efficiencies of motors and frequency converters are arguably complicated characteristics, certain estimates of motor-VSD combination efficiencies are necessary in order to attain energy savings in pumping processes, hence this study. The aim of this research is to find an efficient way to use parallel pumps and prepare the way for developing new, dynamic controlling methods for VSD-controlled parallel pumps.

The results show that the defined simulation environment enables observation and comparison of on-going and planned adjusting methods in parallel pumping processes, and gives opportunities to create adjustment methods that can decrease the adjustment losses in parallel pumping. Simulated estimates for VSD-motor combination efficiencies with different speed and load values can point out possible, and yet unexplained reasons for dimension losses.

1 Introduction

The purpose of this study is to determine more efficient and versatile ways to operate variable speed driven parallel pumps and seek possibilities to lower the energy consumption of parallel pumping. In one perspective, energy losses in pumping systems can be categorized into adjustment losses and dimensioning losses [1]. The adjustment losses are the consequence of the need to set the pump performance along the process needs. The dimension losses are dependent on how well the pumping process components are selected and sized. Naturally, the dimension losses are at minimum, when motors and pumps are operating at their best efficiency points (BEP).

Using variable speed drives (VSDs) in adjusting the pump speed has proven to be an energy efficient way to decrease the adjustment losses in pumping processes [1], [2]. Simulating the performance of the pumps in case of different adjustment methods ensures that the energy efficiency of the parallel pumping can be studied. Simulating the operational values of the parallel pump process components can point out possible reasons for dimensioning losses due to oversized motors and pumps, and can help creating new ways to operate VSD controlled parallel pumps with lower energy costs.

A simulation environment that can be used to observe parallel pumping characteristics is described briefly in this study. The defined simulation environment can be used to determine efficiencies of the motor-drive package attached to the pumps and therefore help calculating the total energy consumption of the parallel pumps. Examples of finding energy savings with simulating variable speed driven parallel pumps are also presented in this research.

1.1 Parallel operation

The main reason to operating pumps in parallel is to allow wider range of flow than it would be possible with single pump. Parallel operation is often most justified, if the system curve is flat. In case of too steep system curves, the amount of combined flow gained by adding the second or third pump to the system can be so incremental that parallel operation is not sensible [3], [4]. Examples of applications for parallel pumping can be found in municipal water supply and wastewater pumping, process pumps in variable capacity process plant and condensate pumps in power plants [3].

It can be quite common that pumping station consist of several different sized parallel pumps. New parallel pumps may be installed because, at normal state, the required flow is much less than the maximum needed, or the pumping system has expanded and the cheapest and easiest option to get to the wanted flow level is to add more additional pumps [5]. Therefore the most efficient way to operate the parallel pumps in traditional way in this kind of situations may not be so simple.

To ensure better ways to observe parallel pumping applications, a simulation environment for determining pumping operation values was built in Lappeenranta University of Technology in co-operation with ABB Drives [6]. The purpose of this was to create a tool, which could help finding energy savings in parallel pumping and ensure the developing of new, more sophisticated methods to operate pumps.

1.2 Variable speed pumping

Variable speed pumping has become more justified in recent years due to improvements in the technology for achieving speed control of pumps and the reduction in the initial cost of such devices [3]. Very often a centrifugal pump is run with AC induction motor, which is a single speed device, because the frequency of the power applied to the motor is fixed. Variable speed drive allows the frequency of the power to be controlled and the speed of the motor’s shaft to be adjusted.

The energy savings gained by variable speed pumping depend on among other things the system curve. The benefit of speed adjustment is that efficiency of the pump usually drops much less compared to the throttling adjustment and, depending on system curve, less energy is wasted compared to increased head when producing reduced flow with throttling adjustment [3], [7], [8]. In many cases variable speed pumping has been shown to be an effective way to reduce total pumping costs for systems that require a wide range of flow [9].

1.3 Oversizing problems in pumping processes

There are often rather good reasons to dimension the design parameters of pumping systems bit higher than calculated capacity and head. It’s possible that the capacity of the designed system may become too small in few years, or piping system requires increased head due to for example corrosion products on the pipe walls. Adding safety factors may happen severaö times during different stages of design process and cause unwanted oversizing to pumping system components. The direct negative consequences of oversizing can be increased capital costs because of bigger components, increased energy costs and possible throttling losses when the quantity is adjusted to the wanted level [3].

1.4 Preferred operating region

In general, centrifugal pumps should be driven as close as possible to their best efficiency point (BEP), At the BEP, the hydraulic, mechanical, and thermal losses attain their minimum values, and service life of the pump is maximized. If pumps are driven outside their recommended operating region, their service life may be seriously affected by the flow recirculation, high flow cavitation, and shaft deflection at reduced flow rates [10]. Since these events are typically related to a decreased pumping efficiency and reliability, they have a strong effect on the pumping life cycle costs. Hence, it is important to apply limits for the region, in which pumps should be driven.

Applying the method presented in [11] pump characteristic curve (i.e. *QH* curve) can be divided into recommendable, allowable, and avoidable operating regions as shown in Fig. 1: by applying available recommendations for fixed-speed pumps and simple efficiency limits for the pump and drive system (i.e. a frequency converter and an electric motor), manufacturer's *QH* curve for Serlachius DC-80/260 was divided into different regions. In this case, the decrease in the drive system efficiency at speeds below 1200 rpm has resulted in allowable operating region in the area where the pump efficiency is above 65 %. Correspondingly, at speeds 0-600 rpm the pump operation should be avoided because of the system static head and notably decreased drive system efficiency, which will be discussed more in chapter 3. The limiting effect of mechanical resonances, cavitation, unstable operating region, and other factors on the pump operation are considered in the analysis.

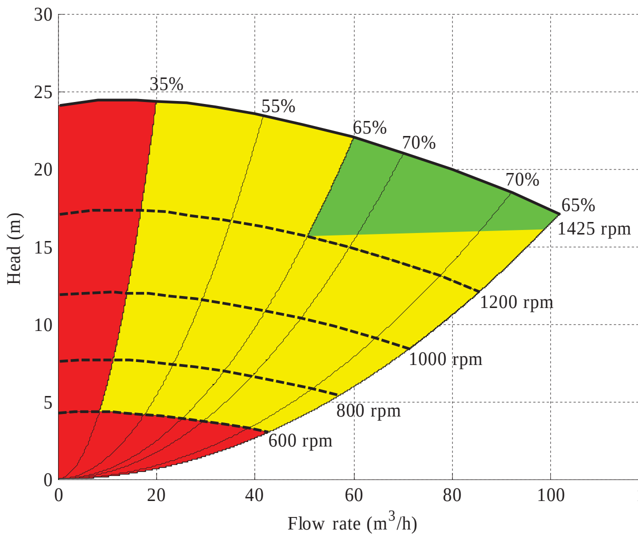

Figure 1. Recommendable (green), allowable (yellow), and avoidable (red) operating region of Serlachius DC-80/260. *QH* characteristics curve can be easily divided into different regions on the basis of pump and drive system efficiency. In addition, the limiting effect of various other factors is considered in the analysis [11].

The preferred or recommendable operating region of the pumps is important to take into account especially in variable speed controlled parallel pumping so that speed adjustment doesn't cause mechanical wearing of the pump due unwanted operating region.

2 Simulation model for parallel pumping systems

In general, the benefit of using software tools when calculating pumping system characteristics is not to achieve more accurate results than in manual calculation. The advantages of software calculation are saved time and the possibility to compare and analyze the gathered results on easy and versatile way [12]. In this case, the simulation environment ensures the comparison of pump operating values with different adjustment and system conditions. This ensures the observation of the energy consumption level and the overall state of the parallel pumps.

The simulated operation values are determined using input data based on observed pumping system variables. This data must contain, for example, pump performance curves, piping characteristics such as loss coefficients and friction factors, and other relevant information to simulate the operation of parallel pumps [6]. An example layout picture of the simulation model is illustrated in figure 2.

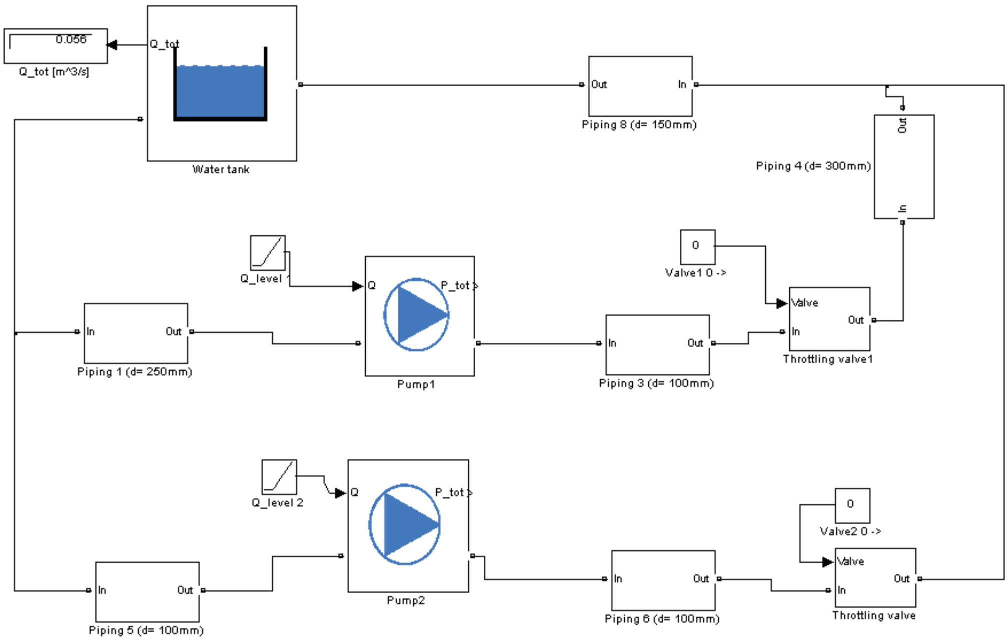

Figure 2. Layout picture of parallel pumping simulation

In the illustrated case, the operation of the pumps can be studied in accordance with target flow. The simulation model calculates results to pump operational values at different flow quantities using for example speed adjustment. The simulation environment can determine the delivered pressure and flow values with various pumps and show the energy use of each pump in case of different adjustment methods [6]. With the help of simulating the behavior of pumps it is possible to compare different adjusting methods and estimate if there are possibilities to gain energy savings using variable speed pumping instead of throttling.

3 Simulating the drive system efficiency

Along the pump characteristics observation, the simulation environment ensures versatile opportunities for energy use monitoring of the whole pump-motor-VSD combination. The motor and VSD characteristics can be inserted to simulation environment and operation efficiency of the motor-VSD combination or drive system can be included in the calculation of different pump adjustment simulations.

There are various factors that affect the efficiency of the drive system. The losses of the electric motor can be roughly categorized to resistance and iron losses of rotors and stators, and ventilation and friction losses in rotor [13], [14]. The efficiency of the electric motors is usually better on larger motors than on smaller scale (1-11kW) electric motors. The drive losses can be categorized into semiconductor and coupling losses, junction, conductor and choke losses, and losses of control circuit. It's known that the efficiency of the drive system is dependent of motor speed and motor load [14]. The efficiency of the system drops when approaching lower motor load and lower speed.

Since the drive system losses are a sum of innumerable different factors, it's extremely challenging to simulate operation of each part. Yet, the exact value of the efficiency is dependent of many system settings, like for example switching frequency. Even the measurement of the power consumption and power output in test conditions can rarely give exact and dependable values to system efficiency [13]. To solve the total energy consumption of the pumping processes in simulation environment, the motor-VSD system efficiency has to be taken into notice. Therefore it is justified to estimate the drive system efficiency even in roughly categorized way.

Figure 3 illustrates a possibility to determine drive system efficiency when simulating the pump operation. The figure shows efficiency matrix to motor-VSD combination depending on motor nominal torque and motor speed. The efficiency matrix is based on lab measurements of 7,5 kW motor-VSD combination at Lappeenranta University of Technology [13].

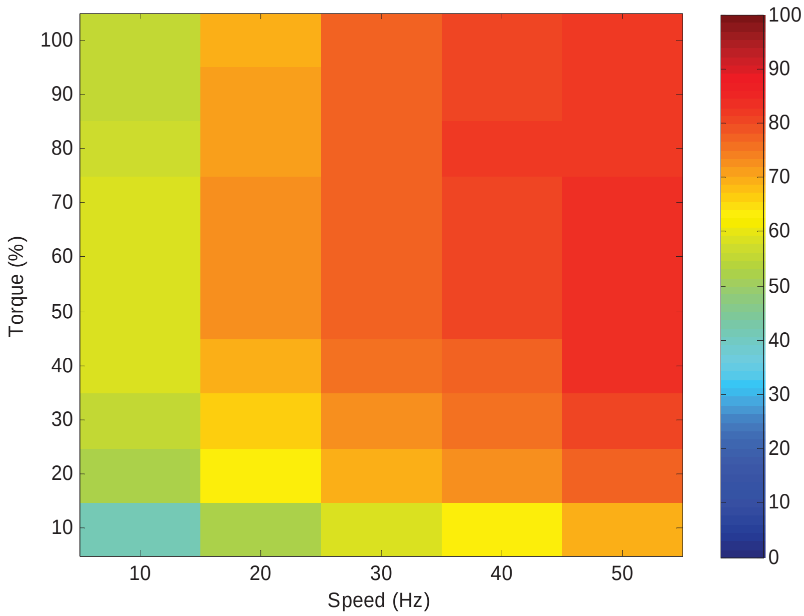

Figure 3. Measured total efficiency of motor and drive system

As it can be seen from the figure 3, the combined efficiency of the drive system will drop as the speed or the load is decreased. The simulation environment can estimate the electric energy consumption at different pump speeds using the information from pump performance curves and efficiency matrix. This ensures the possibility to determine total energy consumption and calculation of total energy costs. If more detailed information on motor-VSD system efficiency is available, new data can be inserted to simulation environment.

During pumping system design, many process components can be oversized, since under sizing usually must be avoided by any means [1]. Oversized pumps may lead to too large flow quantity or increased adjustment losses and oversized motors to decreased motor load during pump operation. If the pumps or motors are oversized, it is not hard to see how the combined efficiency of the motor-drive package can drift to lower efficiency area.

Likewise, the combined efficiency is a relevant factor in actual pumping, even when the process components are chosen correctly. Using the speed adjustment in pumping processes in order to gain energy savings affects the combined efficiency of the drive system. When the rotational speed of the pump is decreased, the load of the motor naturally drops depending on pump and piping characteristics. This decreased load and motor speed can cause a significant efficiency drop, which can’t be taken into account when only the pumping efficiency is observed. The effect can be even more severe, if the pumping components are clearly oversized. Even though this possible efficiency drop is not always even avoidable, it may be useful to note in order to achieve more energy efficient ways to operate variable speed pumping systems. The efficiency of the motor-VSD combination affects directly the energy consumption and energy costs of the pumping processes.

4 Example of VSD controlled parallel pumping in industrial sector

Lappeenranta University of Technology took part to an Energy Auditing case in a forest industry power plant located in Finland. The target of the research was to study electricity use in the power plant and find potential energy savings in selected pumping processes [15]. The studied pumping processes included feed water pumping-, cooling water pumping-, raw water pumping- and concentrated liquor pumping applications. The energy use of pumping processes with different kinds of flow adjustment methods were estimated using the simulation environment mentioned earlier. The study pointed out significant possibilities to gain energy savings using VSD control in pumping processes. The results of the parallel pumping process, which delivers cooling water to surface condenser, are presented briefly in this chapter.

4.1 Surface condenser pumps

The surface condenser pumps of the plant are used to pump cooling water from open water tank to the condenser. The identical parallel pumps are driven by 315 kW motors each, and the delivered flow is adjusted using throttling valve. The static head of the system is approximately 32 meters. The 700mm piping from the tank located under the cooling towers leads to the condenser. The approximated length of the pipeline is 170 meters. The observed parallel pumping system is illustrated in figure 4.

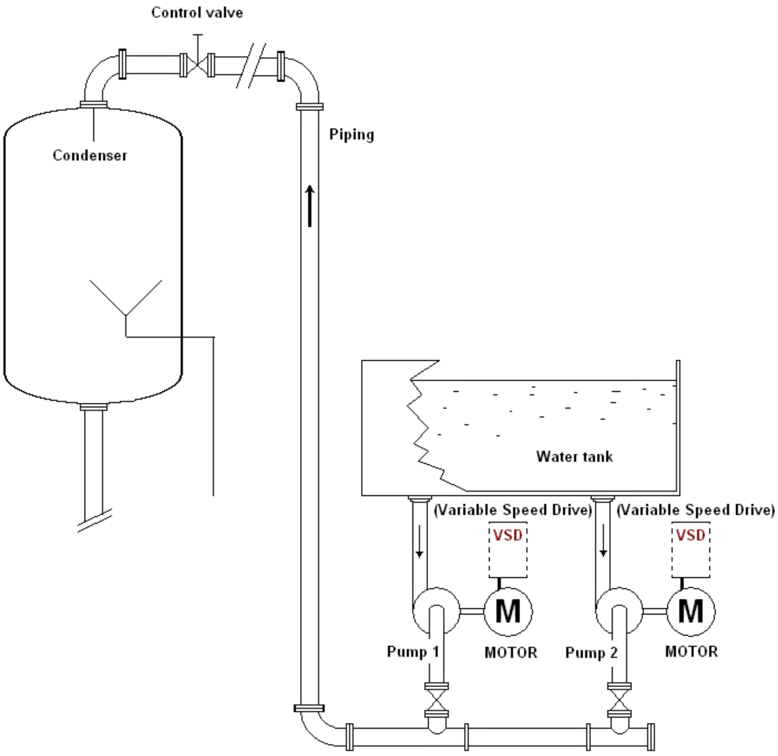

Figure 4. Surface condenser pumps

On the pumping plant the water is delivered to condenser using both pumps at nominal speed, and on current state there are no Variable Speed Drives attached to the system. The flow rate of the system is adjusted with control valve near the condenser. In the research study [15], the energy consumption of this system was evaluated using the defined simulation environment and simulating the operating values of pumping process on valve adjustment, single pump variable speed control and also when both pumps were on speed control with VSDs. In this research, to estimate the electric energy consumption, the efficiency of the motor-VSD combination was simulated using similar efficiency matrix as presented earlier. The annual electricity energy consumption in each adjustment case was calculated using simulated drive input power values and approximated duration curve.

4.2 Simulated results

The results of the simulated power use in observed case revealed clear possibilities to lower the energy consumption of the pumping process by using variable speed control instead of valve adjustment. The simulated specific energy consumption with different flow adjusting methods can be seen in figure 5.

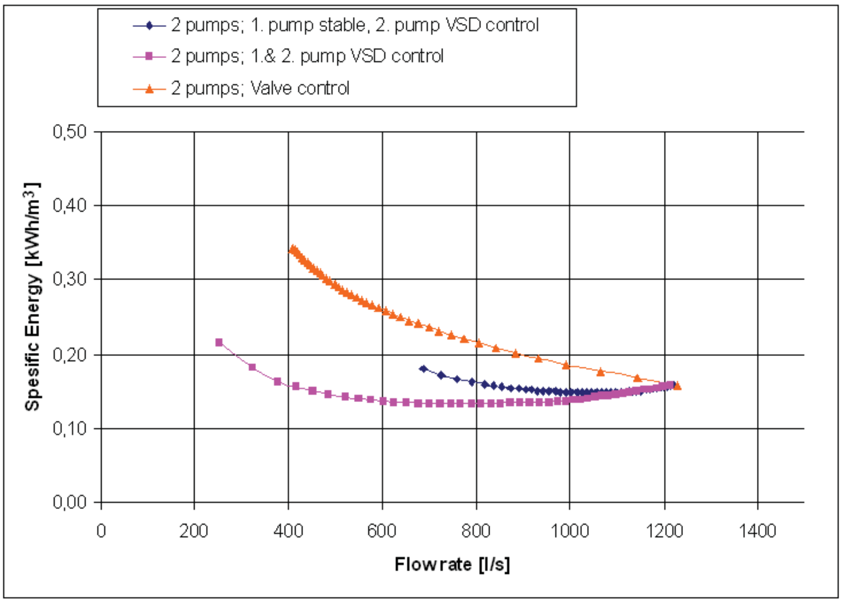

Figure 5. Specific energy consumptions with different flow adjustment ways

The figure 5 illustrates, that energy consumption per volume unit in the observed case is clearly higher than in VSD control. The simulated specific energy is at lowest level when flow rate of both pumps is adjusted with VSDs, but using VSD control with only one pump, when second one is at stable speed, can help gaining notable energy savings compared to valve adjustment. When the results are proportioned to approximated duration curve, the savings in annual electricity use are 12 % when using VSD in one pump and 15 % when both pumps are adjusted with VSDs. According to [15], this saved energy in observed power scale means that the annual energy cost savings could be approximately 35 000 €/a - 43 000 €/a.

The simulated results were calculated using gathered information on pump performance curves and system characteristics. Some of the gathered data weren't available in detailed level, so some values are only rough approximations. These approximations were made using high caution and are intentionally somewhat pessimistic. The efficiency matrix that was used to determine electric energy use of the pump drives is based on measured data of 7,5 kW motor and variable speed drive. The total efficiency of the 315 kW motor-VSD combination can be presumed higher, since the efficiency values are usually higher on larger motors. This phenomenon should make the annual energy savings even higher.

Once again, there is no guarantee that using variable speed pumping energy savings can be achieved in every pumping case. The benefits are always dependent on many pumping process factors. There can many applications where variable speed pumping cannot be adapted due process reliability requirements or system conditions. This case clearly points out that there still is a lot of energy saving potential in industrial parallel pumping systems. Even though many municipal and industrial plants are nowadays equipped with more and more sophisticated controlling and monitoring applications. Ways to lower energy costs via variable speed pumping can be found using energy auditing procedures.

5 Energy efficient flow control

It's known that energy consumption is often one of the larger cost elements in pumping processes and may dominate the life cycle costs of the pumping [8]. This obviously increases the interest to evaluate more energy efficient ways to control parallel pumps as well as pumping processes in general. In this chapter, some aspects to find out energy saving potential at parallel pumping stations are considered.

Figure 6 illustrates a pumping station, which consists of three similar parallel pumps, two water tanks and piping network. Again, the operation of the parallel pumps in illustrated system can be observed using the defined simulation environment. The characteristics of the pumps and piping network can be inserted into simulation software, and the results of the simulations enable observation of the operation in various adjustment methods.

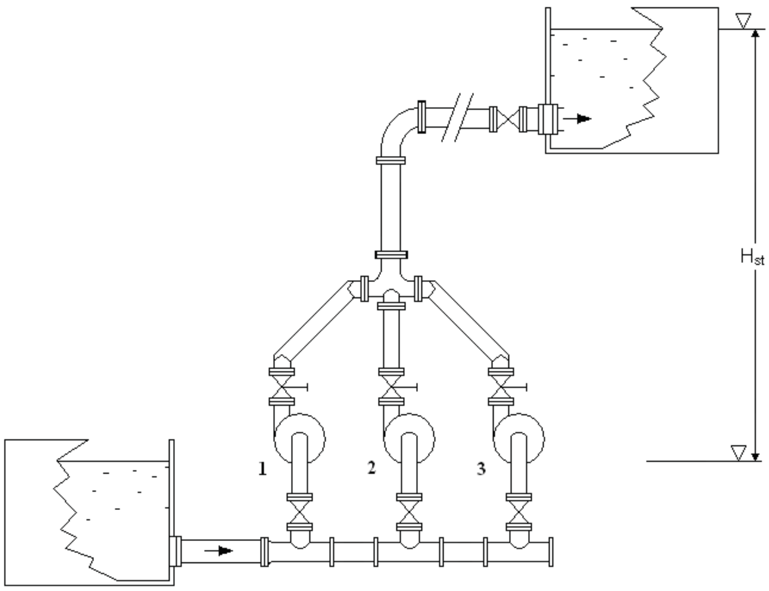

Figure 6. Example of a pumping station with three similar parallel pumps

One solution to carry out variable speed drive controlled flow adjustment in this fictional pumping case is to optimize the speed between phases, where additional pumps are brought to pumping process due to increased flow demand [16]. This means that the speed of the first pump is not increased to its nominal value, but instead, at optimized point, the speed of the first pump is kept stable while the speed of the second pump is increased in order to produce more total flow. Again, at optimized point, the speed of the pumps can be balanced, and due to more flow demand, both pumps can be adjusted closer to nominal speed. In this case, the third pump can be added to process using same balancing phases.

With the help of the simulation environment it's possible to see how the operation point shifts in QH-axis during speed adjustment of the pumps. The operation points of each pump during the previously explained adjustment are illustrated in figure 7. In addition, the figure shows the simulated pumping efficiency on observed flow rate. The final diagram represents the speed of each pump during adjustment.

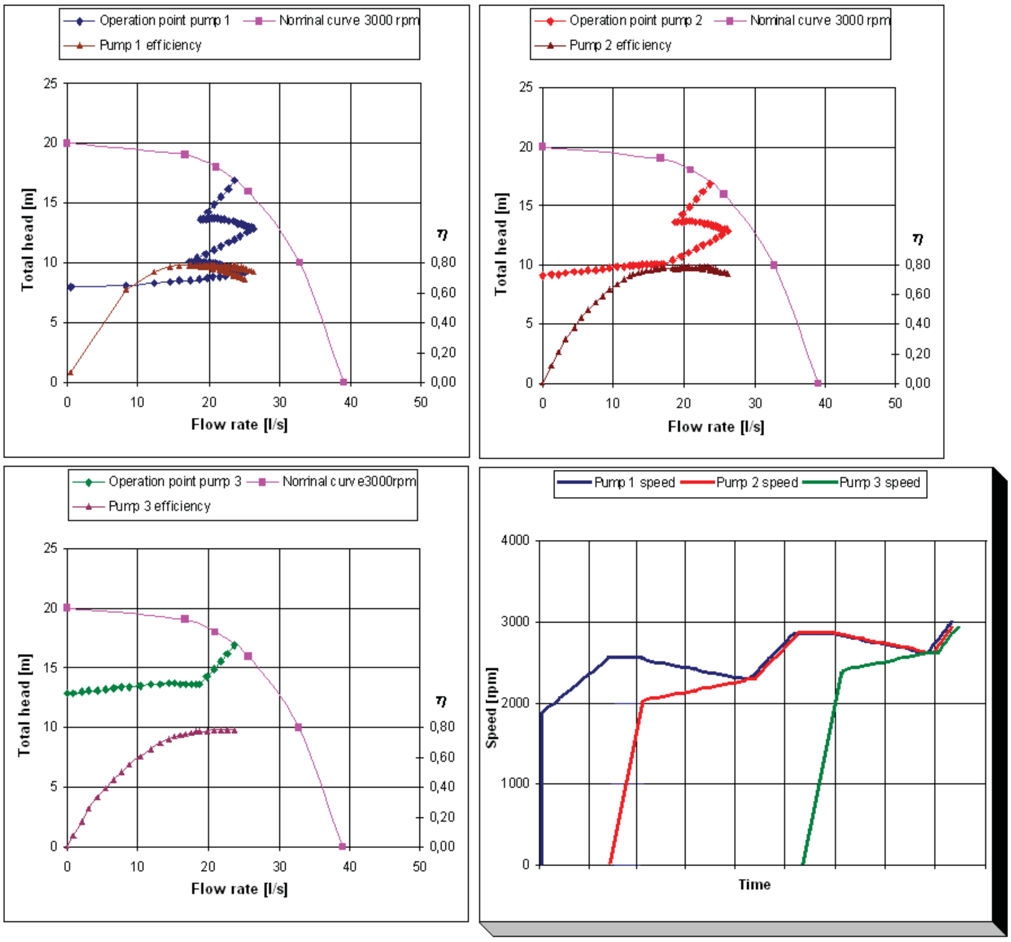

Figure 7. Observing the operation of parallel pumps during adjustment

Since the pumps are identical and the friction head of the separate piping sections are similar for each pump, the total head of the simultaneously operating pumps is at the same level when producing flow (figure 7). In this case, the adjustment seems to be quite effective in the means of pump efficiency as well as reasonable operating region. Yet there are some points during adjustment where each pump is operating at an unwanted region and can therefore suffer the disadvantages of unstable state.

6 Conclusions

The aim of this paper is to study energy efficient methods to operate variable speed controlled parallel pumps. Simulating the behavior of pumps is a reasonable way to compare different flow adjustment methods in order to find savings in energy consumption. When calculating the energy use of the pumping process, it is justified to observe the behavior of all pumping process components, including motors and drives attached to pumps. The efficiency of the motor-VSD combination is related to the total energy consumption of the pumping process and therefore it is an important factor to take account when calculating the energy use of the pumps.

Solving the parallel pump performance and energy consumption with simulation software is justified especially in cases of energy auditing. So far the presented simulation environment has played a major role in several research cases at Lappeenranta University of Technology. The goals of these research cases were to study the energy consumption of the raw-water pumping in cement and lime industry plant and to solve energy efficient ways in parallel pumping at forest industry power plant. The brief results from the surface condenser energy audit case at forest industry power plant are presented in this paper. The most essential conclusion of these results is that there still are possibilities to lower the energy consumption in various pumping processes in Finnish industry using sophisticated variable speed control.

To run parallel pumps in an energy efficient way, monitoring the pumping state via specified measuring equipment such as flow calculation or pressure and vibration measurements can be used to control and pilot the pumps to operate on preferred operating region and with minimized energy use. These techniques, added to variable speed pumping, can help to develop more sophisticated and intelligent methods to run parallel pumping processes at lower energy costs and yet compliance with process needs.

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Publication II

Ahonen, T., Ahola, J., Viholainen, J., and Tolvanen, J.
Energy-efficiency-based recommendable operating region of a VSD centrifugal pump

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2011

Energy-Efficiency-Based Recommendable Operating Region of a VSD Centrifugal Pump

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Abstract

Centrifugal pumps are traditionally advised to be driven near their best efficiency point (BEP), which should result in the maximum efficiency and reliability for the pump. These guidelines generally hold true for fixed-speed centrifugal pumps, but unfortunately they do not consider positive effects of driving the pump at a lower rotational speed. For instance, Hydraulic Institute has published the generic guideline for centrifugal pumps that they should be driven at the 70–120 % of the BEP flow rate Q_{BEP}, but there is no mention concerning the effect of rotational speed on these limits.

The speed control method can significantly reduce the energy consumption of a pumping system compared with other flow control methods, which can be quantified by specific energy consumption E_s (kWh/m^3). For this reason, energy-wise it is often more feasible to drive the pump at a lower rotational speed and away from the pump BEP than at a higher rotational speed with the BEP efficiency. This seems true also in the reliability sense: it is studied that a lower rotational speed improves the pump mechanical reliability, resulting in wider ranges where the pump operation can be considered recommendable or at least allowable. Consequently, there should be rather a best efficiency area (BEA) informed for a variable-speed-driven pump than just a single BEP.

This paper proposes energy-efficiency-based determination of the pump recommendable operating region. It is based on the calculation of the specific energy consumption at different operating point locations. Besides the energy-efficiency-based BEA region determination, the proposed method could be utilized in the frequency converters for analysis and monitoring purposes. As the proposed method does not directly consider the effect of static head and other limitations, they should be separately considered in the determination of the recommendable operating region limits.

Introduction

Pumps are widely used in industrial and municipal applications, and pumping systems account for 15 % of the total electrical energy consumption in the European industry [1]. Often still, pumping systems operate with poor energy efficiency, resulting in unnecessary energy consumption and increased energy costs during their life time. A typical reason for inefficient pumping system operation is the use of throttle control method to adjust the pump output flow rate and pressure [2]–[3]. The throttle control method is a traditionally used and simple, but often the least recommendable method to control the pump operation, as the same pump output could often be attained by driving the pump at a lower rotational speed. In the case of a pumping system consisting of a centrifugal pump and an induction motor, variable speed operation is nowadays typically realized by a frequency converter (later simply referred as a variable speed drive or VSD). Benefits and also possible pitfalls of variable speed operation of centrifugal pumps have been studied in several references, see for instance [4]–[6].

Although VSDs have been available for some time, the tradition of driving a centrifugal pump at a constant speed is still affecting the standards and recommendations given for centrifugal pumps. An example of this is the concept of recommendable pump operating region, for which guidelines have been determined with no consideration for the pump variable speed operation. Namely, ANSI/HI 9.6.3-1997 standard defines the preferred and allowable operating regions of a centrifugal pump, and also lists the factors affecting the limits of these two operating regions [7]. However, the standard contains no mention on the variable speed operation of a centrifugal pump and its effect on the preferred operating region limits. Nearly a corresponding situation is also with the ISO 13709:2009 standard, which however states that the pump should be able to be driven over its rated speed and within its specified speed range without exceeding the given vibration limits [8].

As the severity of pump vibration, cavitation, and flow recirculation phenomena are greatly affected by the pump rotational speed [9], guidelines given for centrifugal pumps should more clearly recognize the effects of variable speed operation on the pumping system energy efficiency and reliability. For variable-speed-driven (VSD) pumps the recommendation of driving the pump at its best efficiency point (BEP) may not be the most feasible approach, if the pump can alternatively be driven at a lower rotational speed. Hence, the concept of best efficiency area (BEA) is proposed to be used with VSD pumping systems.

This paper studies the BEA determination for a VSD centrifugal pump. Primary focus of this paper is in the energy-efficiency-based limits of the BEA, although the effect of rotational speed on the pump reliability is also discussed. Firstly, known guidelines for the recommendable pump operating region are introduced. The use of relative specific energy consumption as an indicator of the pump BEA is then evaluated with a laboratory pumping system. Also the analysis of the pumping system operation according to its specific energy consumption by a variable speed drive is introduced.

Concept of recommendable operating region

It is traditionally advised to drive a centrifugal pump at its best efficiency point in order to optimize the pump energy consumption based on its efficiency. In addition, the risk of cavitation and the amount of hydraulic excitation forces caused by the pressure distribution on the impeller are typically minimized near the BEP, meaning maximized pump reliability in this operating region [10]–[13]. Hence, the range around the pump BEP is often called the recommendable or preferred operating region of a centrifugal pump [2], [7]. If the pump is driven outside this operating region, the pump efficiency decreases, and it may be susceptible to harmful phenomena as shown in Fig. 1. This figure also shows the reliability curve given in [10] for an ANSI centrifugal pump.

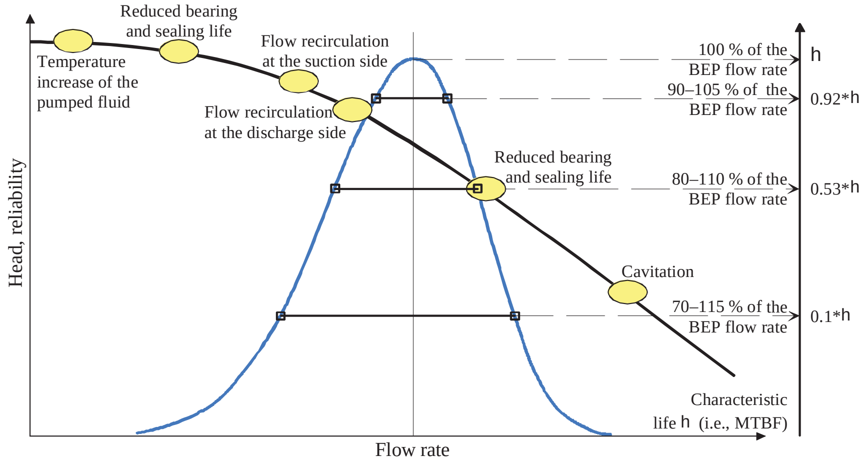

Fig. 1. Pump head and reliability curves as a function of flow rate for an ANSI pump [10].

The concept of recommendable operating region is discussed in the ANSI/HI 9.6.3-1997 and ISO 13709:2009 standards, and by J.F. Gülich [7], [8], [12]. According to [7], the pump curve can be divided into the preferred and allowable operating regions. In the preferred operating region (POR), the service life of a centrifugal pump is not significantly reduced by hydraulic forces, vibration or flow separation. The general guideline for the POR of a typical radial flow and fixed-speed centrifugal pump is 70–120 percent of the BEP flow rate Q_{BEP}. Based on the reliability curve shown in Fig. 1, this can be considered an optimistic recommendation, as the characteristic life • of an ANSI centrifugal pump may decrease to one tenth of its ideal value within this flow rate region. A better guideline could be for instance 80–110 percent of the BEP, which should be used as the range of rated flow rate according to the ISO 13709:2009 standard for petroleum industry pumps [8].

J.F. Gülich has introduced a similar classification with ranges for the continuous and short-term operation [12]. The range of continuous operation allows the use of a centrifugal pump for many thousand hours without damages or excessive wear. This region can be defined for instance by utilizing limits for the decrease in the pump efficiency. As the risk of cavitation and the amount of hydraulic excitation forces increase in the operating region with a lower pump efficiency, this approach can exclude those operating points that decrease the pump service life from the allowable range of continuous pump operation. In the range of short-term operation, a centrifugal pump is susceptible to abnormal operating conditions, which may result in premature wear of the pump. However, a centrifugal pump may be driven in this region for a short period of time without a pump failure.

These two guidelines only provide a general view on the operating regions, in which a centrifugal pump should be driven. For this reason, these guidelines may not correctly determine recommendable and allowable operating regions of the pump. In addition, these references and reliability data given in [10] do not consider the effect of a variable rotational speed on the energy efficiency and reliability of the pump operation, which is why their applicability to VSD centrifugal pumps is doubtful: it is apparent that a variable rotational speed affects the pump mechanical reliability and energy consumption, and therefore, it may be more feasible to drive the pump at a lower rotational speed and slightly off the pump BEP than at a higher rotational speed and exactly in the pump's best efficiency point. This is addressed in the following section with the concept of best efficiency area of a centrifugal pump.

Determination of the pump best efficiency area

Adjustment of the pump rotational speed is often the most energy efficient method to control the pump output flow [4]. However, this cannot be directly seen from the pump efficiency that may remain unaffected by the rotational speed change. Depending on the pump and system characteristic curves, the pump efficiency may decrease, although the speed decrease would be feasible in the terms of pumping life cycle costs. For this reason, the energy efficiency of the pump operation should be quantified by calculating its specific energy consumption E_s with

$$E_s = \frac{P_{tot}}{Q} = \frac{\rho \cdot g \cdot H}{\eta_{dt} \cdot \eta_p}, \tag{1}$$

where P_{tot} is the power consumption of the pumping system, Q the flow rate, ρ the fluid density, g the acceleration due gravity, H the pump head, η_{dt} the drive train efficiency, and η_p the pump efficiency, respectively. In practice, E_s shows the pumping system energy consumption per pumped volume (e.g. kWh/m^3) [5].

Compared with the sole use of the pump efficiency η, specific energy consumption E_s considers also the effects of rotational speed, drive train efficiency, and system losses on the pumping energy efficiency. This makes the E_s a feasible indicator for the pumping system energy efficiency, and thus for the pump best efficiency area BEA, if limits for the allowable E_s can be determined.

In practice, the attainable level of E_s is affected by the process function set for the pumping system as discussed in [14]. If the system characteristics are known, base value for E_s can be solved with

$$E_s = \frac{\rho \cdot g \cdot \left(H_{st} + H_{dyn}\right)}{\eta_{dt} \cdot \eta_p}, \tag{2}$$

where H_{st} is the static head and H_{dyn} the dynamic head of the process.

Equation (2) allows determination of the minimum E_s that is required to transfer the fluid from one place to another based on the system static head and maximum efficiencies of the pump and drive train [15]. By comparing the actual specific energy consumption with this or some other base value of E_s, the energy efficiency of pump operation can be expressed as a relative value for the specific energy consumption, which is proposed to be used as the main variable in the BEA determination.

Best efficiency area of a variable-speed-driven centrifugal pump

As an example, an E_s value graph has been formed for a Sulzer APP22-80 centrifugal pump shown in Fig. 2. Calculation of E_s values was conducted by using Equation (1) for the pump specific energy consumption (i.e., $\bullet_{dt}$ was set to 1) and published characteristic curves of the pump. BEP values for the Sulzer pump are 28 l/s for the flow rate and 16 m for the head at the rotational speed of 1450 rpm.

Fig. 2. Sulzer laboratory pumping system that comprises a Sulzer APP22-80 centrifugal pump, an 11 kW ABB induction motor, and an ABB ACS 800 variable speed drive.

The resulting E_s of the pump in different operating point locations is illustrated as relative values in Fig. 3. The chosen base value and also a possible criterion for E_s is 61.1 Wh/m^3, which is consumed at the pump BEP, when the Sulzer pump is driven at the rotational speed of 1450 rpm. The resulting figure shows the operating regions in which the pump E_s exceeds the chosen base value having an increasing effect on the pump energy consumption. For instance, at the 70 % relative flow rate and at 1450 rpm, the relative pump E_s is 1.23, and so at this rotational speed the HI guideline for the POR seems reasonable. At the rotational speed of 1595 rpm, the corresponding scaled E_s is 1.48, indicating a clearly higher energy consumption.

The figure also shows the operating regions where the pump usage can be considered energy efficient based on the chosen E_s threshold. Compared with the previously introduced guidelines for fixed-speed centrifugal pumps, these results clearly demonstrate how the decrease of the rotational speed extends the recommendable operating region of the pump based on the pump E_s.

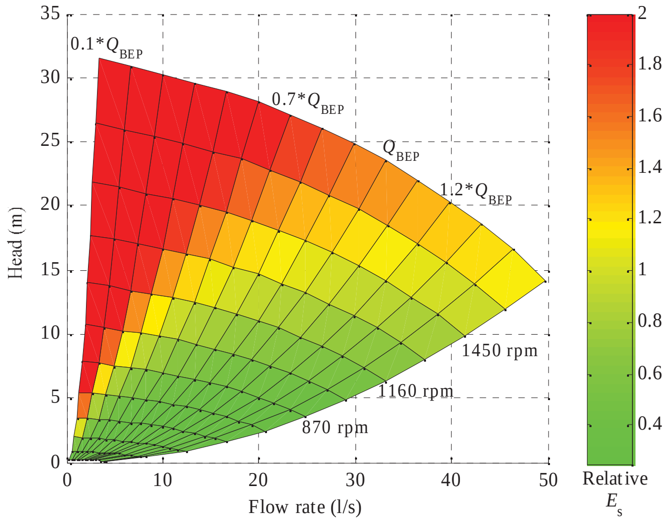

Fig. 3. Relative E_s of the Sulzer pump at different operating point locations. The chosen base value for E_s is 61.1 Wh/m^3.

It is worth noting that E_s tends to decrease at a fixed rotational speed, when the flow rate increases and the head decreases on the known part of the pump *QH* characteristic curve. For this reason, the E_s criterion primarily results in a minimum recommendable flow rate and maximum recommendable rotational speed limits for this VSD pump. As the cavitation phenomenon may occur in the pump with a sufficiently high relative flow rate, E_s-based BEA determination should be accompanied by the determination of pump reliability at different operating point locations.

Best efficiency area of a VSD pumping system

Also the possible effect of the drive train efficiency on the pumping energy efficiency has been determined with the efficiency data available for the 11 kW ABB induction motor and the ABB ACS 800 frequency converter, which have been used as the drive train of the Sulzer laboratory pumping system. The resulting E_s figure for the Sulzer pumping system is illustrated in Fig. 4. For the sake of clarity, 61.1 Wh/m^3 is again applied as the base value for E_s. In addition, an exemplary limit for the BEA is shown in the figure based on the relative E_s threshold of 1.25.

Comparison of Figs. 3 and 4 shows that the effect of drive train efficiency is most notable at low rotational speeds and relative flow rates, in which the lower drive train efficiency increases the relative E_s. For instance, at the rotational speed of 870 rpm and at the 30 % relative flow rate, the relative E_s of the pumping system is 1.18, while the relative E_s of the pump is only 0.83. Otherwise the figures are similar, meaning that a decrease in the rotational speed extends the best efficiency area of the pump on the basis of the pumping system E_s. More generally, the proposed method seems applicable to the determination of the energy-efficiency-based recommendable operating region for a VSD centrifugal pump or a pumping system.

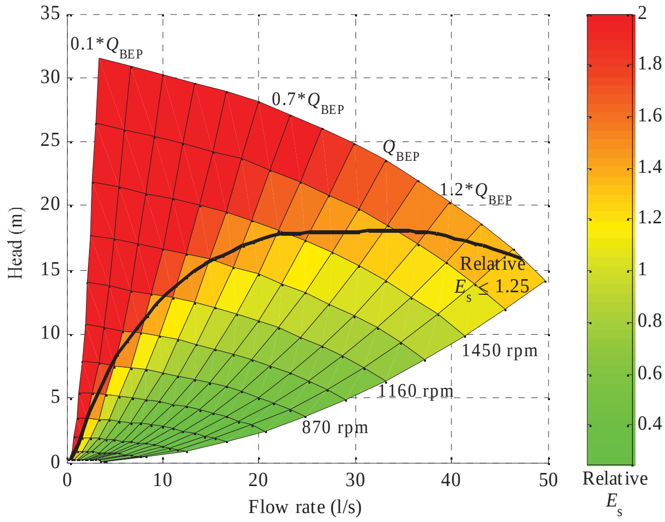

Fig. 4. Relative E_s of the Sulzer pumping system including the drive train efficiency at different operating point locations. Exemplary limit for the BEA is also given in the figure.

Effect of rotational speed on the pump reliability

As shown in Fig. 1, the relative flow rate of a centrifugal pump has an effect on its mechanical reliability that can be quantified for instance with the mean time between the failures (MTBF) value. However, only a few papers have been published focusing on the reliability of a centrifugal pump at different rotational speeds and flow rates, although the possible benefits and pitfalls of pump variable speed operation have otherwise been discussed for instance in [4]–[6]. As practical examples, variable speed operation may lead to the cavitation occurrence in the pump, or to the overloading of the motor, if no limits have been determined for the pump rotational speed range.

The pioneering work has been carried out by H.P. Bloch and F.K. Geitner, who have proposed a quantitative method to determine the effect of rotational speed, relative flow rate and impeller diameter tip clearance on the pump reliability [16]. In their model, pump reliability is assumed to be inversely affected by the pump rotational speed via the reliability factor R_n (reliability vs. speed). A.E. Stavale has published vibration-velocity-based R_q (reliability vs. flow) curves for an ANSI centrifugal pump at four different rotational speeds [9], which show the extending and increasing effect of a lower rotational speed on the R_q factor (see Fig. 5). Based on these results, lower rotational speed should widen the recommendable and allowable operating regions of the pump. Naturally, increase in the rotational speed has an opposite effect on the pump reliability, as it increases the magnitude of pump vibration and the pump suction energy, which describes the severity of cavitation and flow recirculation phenomena [11].

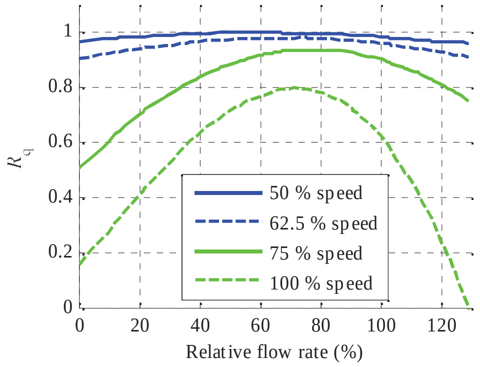

Fig. 5. R_q curves as a function of relative flow rate at four different rotational speeds according to [9].

Analysis of the pump operation by a variable speed drive

As studied in [17]–[20], centrifugal pump operation can be effectively monitored by a variable speed drive. The sensorless flow rate calculation found in modern frequency converters allows also the calculation of the pump specific energy consumption: if the *QP*-curve-based estimation method is used by the variable speed drive (see Fig. 6), then the VSD readily estimates the pump flow rate for certain pump power consumption. With the estimated pump shaft power and flow rate, specific energy consumption of the pump can be calculated according to Equation (1).

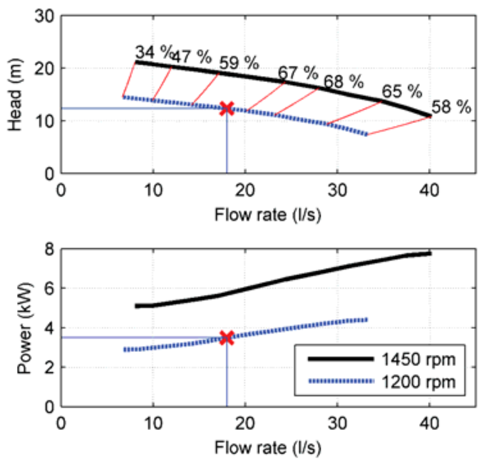

Fig. 6. Estimation of the pump operational state by the *QP*-curve-based method [19]. Shaft power and rotational speed estimates provided by the variable speed drive are used to determine the pump flow rate and head, respectively.

Monitoring and analysis of the pumping energy efficiency could be implemented into the variable speed drives by utilizing the previously introduced BEA concept. Besides the generic information whether the pump is within or outside the BEA, a variable speed drive could also inform the instantaneous relative E_s of the pumping system. Just like the gas mileage and fuel consumption quantities used with vehicles, relative E_s provides easily understandable information on the pumping energy efficiency. As a non-dimensional variable it could also be applied to the quantitative analysis of the pump operation together with the reliability factors discussed in [9], [11], and [16].

Summary

Variable speed drives can provide additional benefits and also pitfalls to the operation of centrifugal pumps. For this reason, guidelines given for centrifugal pumps should more clearly recognize the effects of variable speed operation on the pump energy efficiency and reliability. As a lower rotational speed can widen the pump recommendable operating region, there should be rather the best efficiency area (BEA) informed for a variable-speed-driven pump than just a single best efficiency point.

This paper studied the BEA determination for a VSD centrifugal pump. The use of relative specific energy consumption as an indicator of the pump BEA was successfully evaluated with a laboratory pumping system. Also the analysis of the pumping system operation according to its specific energy consumption by a variable speed drive was discussed in the paper.

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Publication III

Viholainen, J., Tamminen, J., Ahonen, T., and Vakkilainen, E.
Benefits of using multiple variable-speed-drives in parallel pumping systems

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Benefits of using multiple variable-speed-drives in parallel pumping systems

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Abstract

In general, parallel-connected centrifugal pumps can be controlled using either ON-OFF, throttling or speed control methods. The use of ON-OFF method is usually applicable to systems having a tank or a reservoir and thus no need for accurate control of the flow rate. In situations, where more precise flow adjustment is necessary, the most energy efficient solution is often the speed control method using e.g. variable-speed-drives (VSDs). In the simplest case, speed control for parallel-connected pumps adjusts the rotational speed of only one pump at the time, for example according to desired total flow/pressure rate. Although this kind of solution can improve the pumping efficiency compared to throttling control, there is no guarantee that the parallel pumps are not driven in near shut-off head or at situations where cavitation or other harmful operation with higher risk of reduced pump service life can occur.

Compared to traditional speed control, where speed of only one pump is controlled at once, better energy efficiency is possible if all parallel-connected pumps are speed regulated. In addition to saved energy, using speed control in multiple parallel pumps gives a chance to avoid situations where parallel pumps are operating in shut-off or in a region, where pump service life may be affected by the flow recirculation, high flow cavitation and shaft deflection.

Installing variable speed drives to parallel-connected pumping systems can be a major step towards energy-savings from pumping processes. However, energy efficient control with lower risk of mechanical failure in parallel pumping requires that non-traditional possibilities to execute speed control are evaluated. In this paper, the possible drawbacks of traditional speed control of parallel-connected pumps are discussed and alternative solutions using simultaneous speed control of parallel pumps are studied.

1 INTRODUCTION

The increased number of variable speed drives in pumping applications has generated the potential to improve the energy efficiency in pumping processes. The energy saving benefits of using rotational speed control using variable speed drives (VSDs) has been widely recognized and utilized in various pumping facilities [1],[2],[3]. Implementing rotational speed control of a single centrifugal pump can be a simple procedure, but controlling multiple parallel-connected pumps is often a more complex case.

Installing VSDs to parallel pumping systems means that different options to execute rotational speed control of the parallel-connected pumps are possible. The improved energy efficiency of rotational speed control compared with throttling has been widely studied [4],[5],[6]. So far, less effort has been put to optimize the rotational speed control of parallel pumping. The differences in energy efficiency between different control schemes for parallel pumps have been recognized [7], [8], but so far it seems that the trend still is to execute the speed control in the simplest way, without estimating if feasible energy savings are available. It is also important to take into account that there are risks in speed control of pumps, e.g. if the pumps are operating near shut-off or area with cavitation risk. These situations are harmful in both pumping efficiency and pump reliability point of view [9]. Choosing the most suitable controlling scheme can improve both reliability and the overall energy efficiency of the parallel pumping process.

In this paper, the rotational speed control of parallel-connected pumps is being studied. The aim of this study was to demonstrate that choosing suitable rotational speed control scheme for parallel-connected pumps can reduce the risk of harmful operation that can increase the probability of mechanical failure in pump. This study also aimed to show that implementing the introduced rotational speed control scheme can be justified in terms of energy efficiency.

2 PARALLEL-CONNECTED CENTRIFUGAL-PUMPING SYSTEMS

Connecting centrifugal pumps in parallel allows a wider range of flow rates than would be possible with a single pump. In other words, parallel connection of centrifugal pumps increases the ability to meet flow rate variability of a pumping system [10]. Parallel-connected pumps are commonly used in municipal water supply and wastewater pumping applications, in process applications having a variable flow demand, and in the condensate and feed-water pumping systems found in power plants [10] [11]. An example of two parallel-connected centrifugal pumps installed in a pumping system is given in Figure 1.

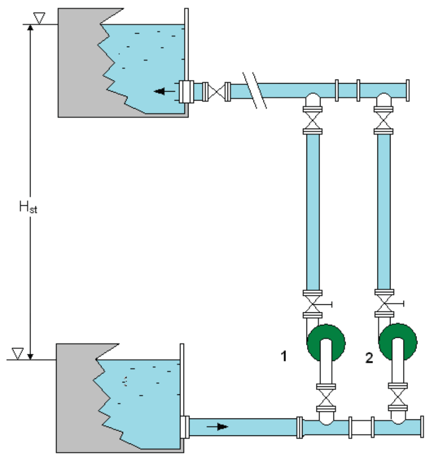

Figure 1. Two parallel-connected centrifugal pumps in a pumping system.

This illustrated open system consists of two parallel pumps, inlet and outlet tanks and piping, in which the static head of the pumps H_{st} is the elevation difference of tank water surfaces. The operation of parallel-connected pumps is illustrated in Figure 2.

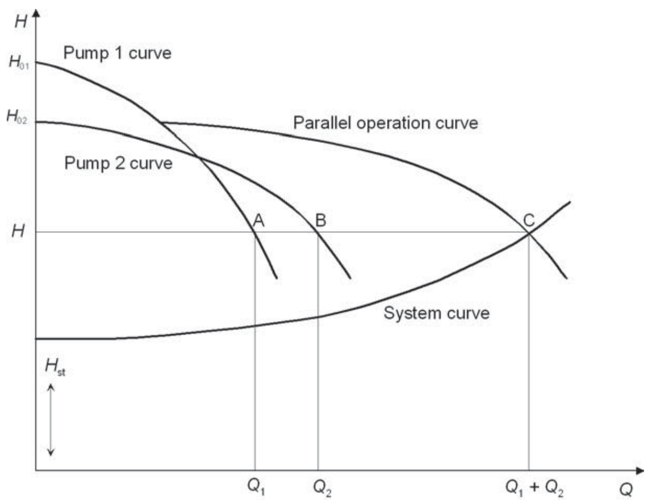

Figure 2. Parallel operation of pumps 1 and 2 (points A and B) and the resulting operating point location C with the total flow rate Q_1+Q_2

The parallel pump curve is the sum flow rate of individual pump curves. The individual operating point locations (A and B) of pumps 1 and 2 can be determined with the flow rates Q_1 and Q_2. In this case, the parallel-connected pumping system provides the sum of flow rates Q_1 and Q_2 of pumps 1 and 2 respectively with a common amount of head. The operating point location of this parallel-connected pumping system (marked C in Figure 2) is the intersection point of system curve and parallel operation curve [10]. Different sized pumps, as in this illustrated case, can operate parallel only below the shut-off head of the smaller pump (H_{02})

2.1 Evaluating the energy efficiency of parallel pumps

Since the delivered flow rate is often the control variable in parallel pumping, a justified way to evaluate the energy efficiency of pumping is the specific energy, which describes the energy used in relation to pumped volume [4]. Specific energy is given by:

$$E_s = \frac{P_{in} \cdot t}{V} = \frac{P_{in}}{Q}, \tag{1}$$

where E_s specific energy [kWh/m^3]

P_{in} input power to pump drives [kW]

t time [h]

V pumped volume [m^3].

An example of specific energy consumption in speed regulated parallel pumping is illustrated in figure 3. The figure plots the specific energy consumption of two parallel-connected pumps in on-off control and rotational speed control E_s curves for single pump and two parallel-connected pumps.

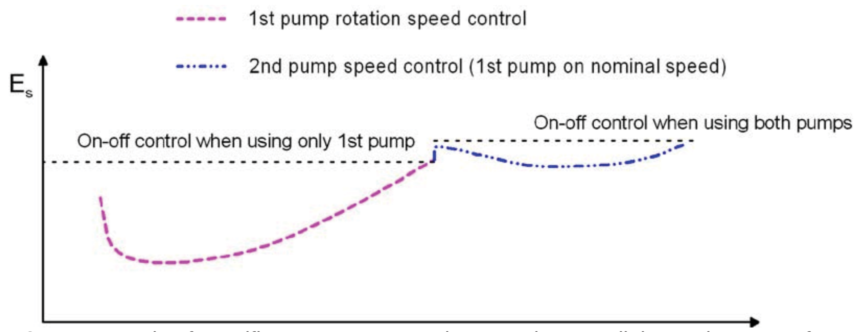

Figure 3. An example of specific energy consumption rates in a parallel pumping case of two pumps.

The example (Figure 3) shows a case, where using speed control of pumps can be seen as more energy efficient alternative, since the specific energy use is lower in most flow rates compared with on-off control method. The benefits of the rotational speed control depend on pumping system characteristics; the performance of the pumps, motors and drives as well as the system conditions.

2.2 Different control schemes for parallel pumps

Parallel-connected centrifugal pumps can be controlled, for example, with ON-OFF, throttle, or speed control methods. The use of the ON-OFF method is usually justified for applications having a tank or a reservoir and no need for accurate control of the flow rate. Correspondingly, the throttle control method can be used to regulate the flow rate produced by the pump, but because of its poor energy efficiency, it is rarely justified. Speed control, on the other hand, can allow the flow rate control with a lower energy use compared with the throttling method. Examples of using different control methods in parallel pumping systems are illustrated in Figure 4.

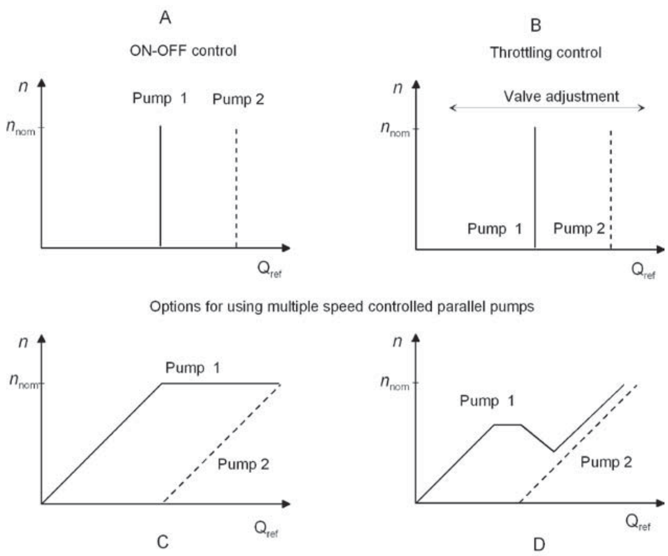

Figure 4. Different control methods for parallel-connected pumps. The Q_{ref} shows the direction of increasing flow demand. Scheme A illustrates the use of ON-OFF control; more flow is acquired only by starting and stopping pumps. Scheme B illustrates the throttling control, in which the flow rate can be set using throttling valves. Schemes C and D are examples of options for rotational speed control.

As can be seen from Figure 4, the on-off control (scheme A) can't enable accurate flow regulation, since the flow rate is adjusted by starting and stopping parallel pumps. In general, the throttling control (scheme B) differs from on-off solution only because the flow rate can be set more accurately using control valves.

Speed control, however, can allow the flow rate control with a lower energy use compared with the throttling method [4]. The basic version of speed control for parallel-connected pumps, the traditional rotational speed control method, is based on the adjustment of the rotational speed of only a single pump at a time (scheme C in Figure 4). At low flow rates, only the primary pump is used, and the secondary pump is started when the Pump 1 reaches its nominal speed and still more flow rate is required [8]. In scheme D the rotational speed of the primary pump is not increased to nominal flow, but instead, secondary pump is started and the speed of Pump 1 is balanced to meet similar flow circumstances with Pump 2 [12].

3 RISKS OF TRADITIONAL SPEED CONTROL

Several studies suggest that centrifugal pumps should be driven close to their best efficiency point (BEP) [9][13]. Based on [14], at BEP, the hydraulic, mechanical, and thermal losses of the pump attain their minimum values. If pumps are driven outside their recommended operating region, their service life may be seriously affected by the flow recirculation, high flow cavitation, and shaft deflection. However, running pumps on reduced rotational speed instead of nominal value can widen the reliable pump operating range [15]. These observations suggest that running pumps near BEP-curve, which can be approximated for different pump speeds using affinity rules, is beneficial for pump service life.

In parallel pumping, the most hazardous events can be operating at near shut off or in an area with cavitation risk, both of which also affect not only pump wear but also the energy efficiency of pumping [16]. Extra care has to be taken especially when different sized parallel pumps are operated using traditional speed control [17]. Figure 5 illustrates these avoidable areas in pump *QH* curve, where operating close to shut off or exposed to cavitation are possible [13][14].

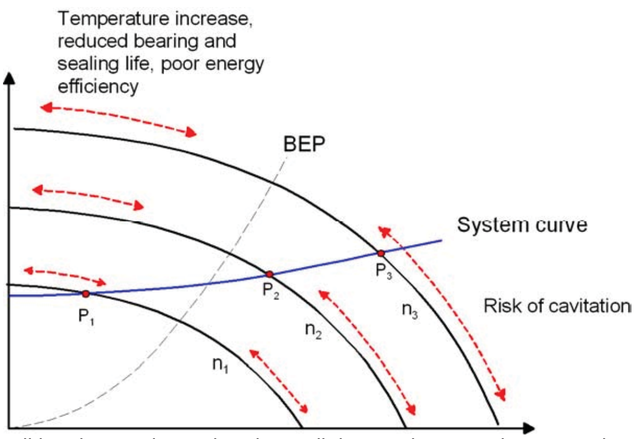

Figure 5. Traditional speed regulated parallel pumping can increase the risk of individual pumps operating inside potentially damaging operating ranges.

The figure also plots a flat system curve, which can be presumed in parallel pumping systems. The figure illustrates the operation of individual pumps in a speed regulated parallel pumping system on rotational speeds n_1, n_2 and n_3. The BEP-curve determined using affinity laws can also be seen in the figure. When the object of individual pump is to increase the flow by only a small amount; speed (n_1) may be used which can cause the operation point P_1 to be located near shut off. This can expose the pump to e.g. reduced bearing and seal life and very often to poor energy efficiency. Better operating conditions can be expected at point P_2, where the location of the operating point is also closer to BEP-curve. If the system is dimensioned so that high pumping efficiency is reached when all parallel pumps are running at nominal speed, using only a single pump at nominal speed can again cause cavitation and a higher risk of mechanical failure at point P_3 [14].

A more practical example of a preferable option compared with the traditional speed control can be demonstrated, if the operation of two identical raw water pumps is observed in a system with a 15 m static head. In this example, the system curve is chosen so that both pumps (Ahlström P-X80X-1) will have a high pumping efficiency when they are operated at the nominal speed [8]. The adjustment of the process to a lower flow rate using either the traditional rotational speed control or reducing the rotational speeds of both pumps is illustrated in Figure 6.

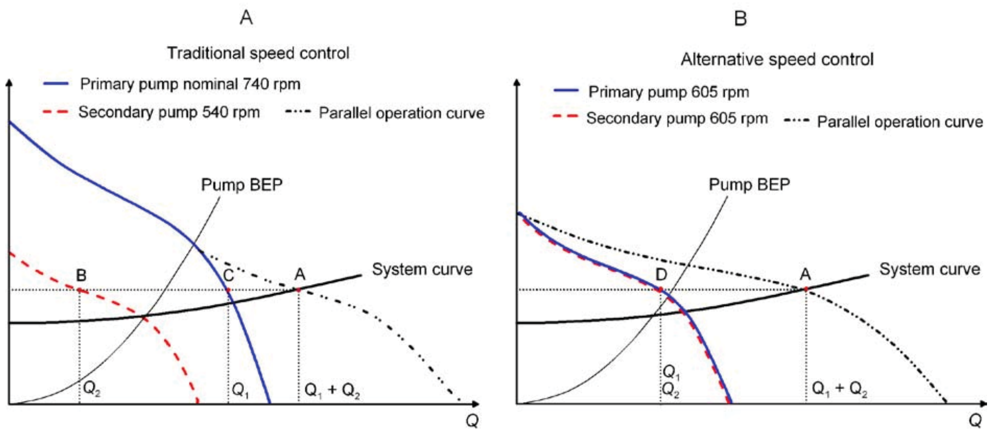

Figure 6. Speed-regulated parallel pumping using the traditional rotational speed control (graph A) and when both pumps are running at a reduced speed (graph B). Adjusting the flow rate by using traditional speed control delivers the desired flow rate (Q_{1+2}), but the operation points are located far from the BEP curve. Adjusting the flow rate by reducing the speed of both pumps results in the same flow rate, and the operation points can be located in a region of better energy efficiency and mechanical reliability.

Graph A in Figure 6 plots the *QH* curves of the parallel pumps: the first pump operating at the nominal 740 rpm speed and the second pump at a 540 rpm speed, the system curve, and the combined parallel pump curve. Graph B shows the *QH* curves when both pumps are operating at the same reduced speed (605 rpm) delivering the same total flow Q_{1+2}. In the traditional speed control, it is quite common that parallel pumps are not operating (points B and C) near the pump BEP curve, which in this figure represents the justified operating region at different pump speeds rather than just the location of the best pump efficiency [9][14]. If the same flow rate is delivered using a decreased rotational speed for both pumps, the operation points of the pumps (point D) are closer to the BEP curve, which in this case suggests a higher energy efficiency and mechanical reliability.

4 ENERGY SAVING POSSIBILITIES

The energy saving benefits of using multiple simultaneously speed controlled parallel pumps compared with traditional rotational speed control can be studied by evaluating the specific energy use of observed control schemes.

Figure 7 plots the operation of three similar parallel-connected pumps in *QH* axes. The figure shows the nominal *QH* performance curve of a single pump and operation curves when either two or three pumps are operated in parallel. The system curve is also illustrated, and it can be seen that if three parallel pumps are running at nominal speed, the operation point of each pump would be at BEP.

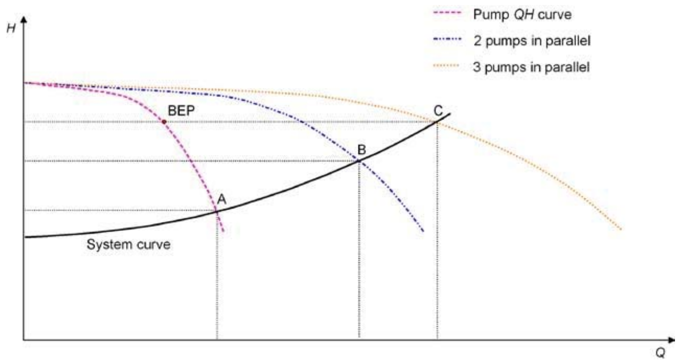

Figure 7. QH curves of three parallel pumps and a system curve.

The points A and B show the operation point location if the desired flow rate is delivered; using only one parallel pump at nominal rotational speed (point A) or using two parallel pumps at nominal speed (point B). In point C all parallel pumps are operating at nominal speed and system delivers the maximum flow rate. An example of the specific energy use of the illustrated system in traditional speed control is shown in Figure 8. In this study, the traditional speed control means that the speed of only one pump is controlled at a time. The specific energy use includes, besides pump power, also the estimated energy use of the motors and VSDs of the pump drive trains.

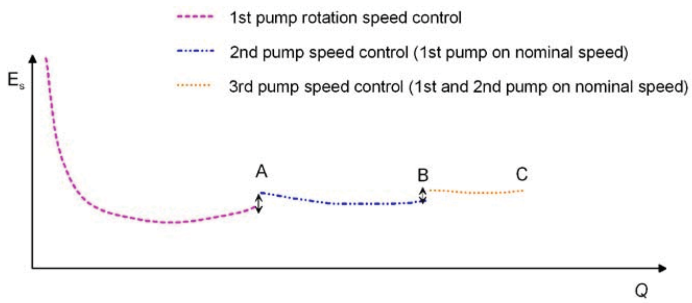

Figure 8. The specific energy consumption in traditional rotational speed control in parallel pumping.

At point A (Figure 8), the first pump has reached its nominal speed. To generate more flow, the second parallel pump is started, which increases the specific energy to a higher level. After reaching a rotational speed that overcomes the required head, more flow is delivered as the rotating speed of the second parallel pump is increased. Usually after this, if pumps are dimensioned as shown in Figure 7, the specific energy of pumping starts to decrease slightly, but starts to increase again as the second pump is nearing its nominal speed at point B. Starting another parallel pump at point B again increases the specific energy to a new level. At point C all three parallel pumps are operating at nominal speed.

If the rotational speed of multiple parallel pumps is being controlled at the same time, the specific energy use of the system is different as illustrated in Figure 9. The figure shows the same specific energy curves as in traditional rotating speed control scheme and also alternative curves if the speed of two or three parallel pumps is being controlled simultaneously.

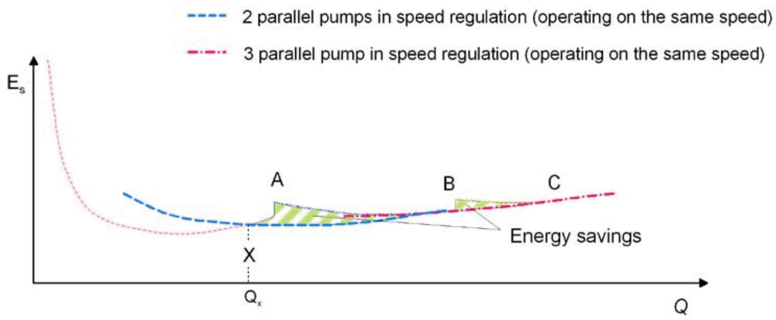

Figure 9. Specific energy use of parallel pump control schemes where two or three pumps are controlled simultaneously.

Two parallel pumps, having the same rotational speed will end up with the specific energy use illustrated in Figure 9. As can be seen, running two pumps instead of one will result a higher specific energy use at low flow rates. However, when the speed of the pumps is further increased, using two pumps with lowered rotational speed becomes more energy efficient than delivering the same flow using only one speed regulated pump. In this case, using two parallel pumps with simultaneous speed control is clearly advantageous compared with traditional speed control just after point A, where the first pump is running at nominal speed but the second pump is operating at lower speed. Correspondingly, at higher flow rates, using three parallel pumps at the same lowered rotational speed results in lower specific energy consumption compared with traditional speed control that uses two parallel pumps at nominal speed and a third pump with reduced speed to deliver the same flow rate. The energy savings are achieved after point B. At point C all pumps are running at the same, nominal speed, which results in equal specific energy consumption using both traditional speed control and multiple rotational speed control of parallel pumps.

The main point of this example is that it is not always justified to increase the speed of the first pump to nominal at low flow rates. Instead, secondary pumps should be added to process, and both pumps should be operated at lowered rotational speed to deliver the same flow. As can be seen from Figure 9, the specific energy curves of alternative control schemes cross at point X, which in this case determines the flow rate (Q_x) after which two simultaneously speed controlled parallel pumps should be operated.

5 HOW TO TAKE SHUT-OFF OR CAVITATION INTO ACCOUNT WHEN OPERATING PUMPS WITH VSDS

Figure 9 showed that energy savings can be achieved compared with traditional speed control if additional parallel pumps are started and operated at lowered speed to deliver the same flow as using less pumps. The point, when to start using more parallel pumps can be determined with specific energy curves, as illustrated in the previous section. On the other hand, running parallel pumps in a situation where one or several pumps are running at a higher speed, there can be a risk that pumps are operating in an undesirable region that can accelerate pump wear, as explained in section 3.

To prevent unwanted operation, the rotational speed of the parallel pumps should be balanced after new pumps are added to the system. Balancing the rotational speed up towards higher speed shifts the additional pump's operation point to an area with lower risk of shut-off. Consequently, balancing the speed of already ongoing pump(s) to lesser speed can reduce the risk of harmful events at the right hand side (high flow region) of the QH-curve. After speed balancing, all ongoing pumps can be controlled simultaneously to higher speed if more flow is required. An example of this procedure is illustrated in Figure 10.

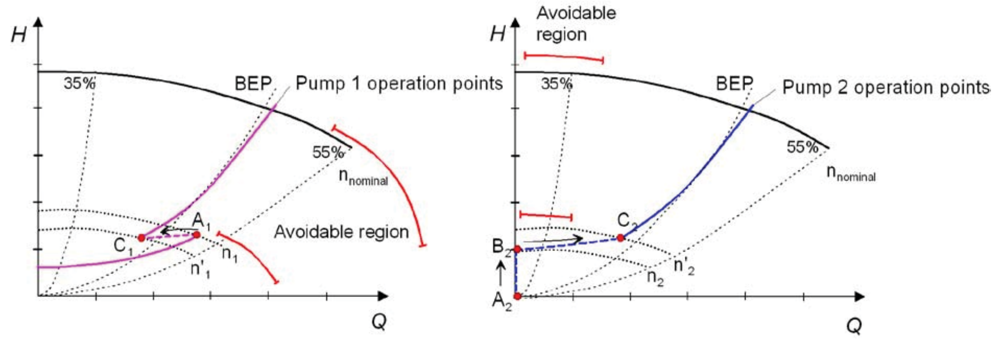

Figure 10. Using rotational speed balancing of parallel pumps to avoid unwanted pump operation area. Pump 2 should be started when there is a risk that Pump 1 is entering an undesirable operating region. After Pump 2 has started to deliver flow at point B_2, the speed of both pumps can be balanced to meet the same head at points C_1 and C_2.

To ensure that rotational speed controlled parallel pumps do not operate in undesirable operation regions, the speed of the pumps can be balanced as illustrated in Figure 10. The speed of the primary pump (Pump1) is not increased further if there is risk of entering avoidable region (A_1). Thus, the speed is kept stable at n_1. Consequently, the secondary pump (Pump 2) is started (A_2), and when Pump 2 starts to deliver flow at speed n_2 at point B_2, the speeds of both are balanced to meet the same head level. This procedure causes the operating point of Pump 1 to shift from A_1 to C_1 and the operation point of Pump 2 from B_2 to C_2. Adding pumps and performing the balancing procedure on the right moment can also result a lowered specific energy compared with traditional speed control, where the rotational speed of individual pumps are increased to nominal before adding more parallel pumps to the process.

The preferred point when to use several speed regulated parallel pumps instead of only one pump can be determined if the individual operating point of each parallel-connected pump is being observed. VSDs can be utilized to pump monitoring as well as control [18] and flow metering using e.g. pressure sensors and pump characteristics can be used for control purposes. In that case, the undesirable operating regions, such as areas near shut-off or increased cavitation risk should be selected based on pump characteristics. This data could be implemented to VSD-control procedure and based on flow metering information, parallel pumps could operated with lesser risk of shut-off and cavitation in addition to reduced energy consumption.

6 CONCLUSION

Installing variable speed drives to parallel pumping system enables the use of rotational speed control to adjust the pumping process flow rate with better energy efficiency compared with many other control methods. The rotational speed control of parallel pumps can be seen most beneficial if the control scheme is fitted for parallel pumping and existing system characteristics. This can be done using rotational speed balancing of ongoing parallel pumps and adding and cutting parallel pumps at the right moment. It is possible to determine the justified point by calculating the specific energy use of different pump control alternatives but also by defining the unwanted operating regions of each parallel pump, where there is a risk of reduced service life and poor energy efficiency.

Improving the speed control of parallel pumps does not necessary mean investments in new process components; more advantageous operation can be achieved in any parallel pumping systems where pumps are already equipped with VSDs. Utilizing the VSDs for process monitoring can open the possibility to perform justified control schemes with very little initial data. This can reduce the pressure to select alternative control schemes to execute speed control of parallel pumps instead of the traditional way, which can be simple but far from optimal. The result of improving the speed control of parallel pumps can be energy efficient operation of pumping process with lower risk of mechanical failure in pumps.

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Publication IV

Viholainen, J., Sihvonen, M., and Tolvanen, J.
Flow control with variable speed drives

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Flow control with variable speed drives

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***Abstract* - This paper focuses on how integrated flow measurement works on variable speed drives (VSD's) that are used to control the speed of pumps. With the measurement feature it is possible to build a pumping system without using separate flow-meters. Having a system with fewer parts makes it more reliable and also less expensive. The flow can be measured with sufficient accuracy for most purposes. The integrated approach can also help in finding leaking spots in the system.**

I. Introduction

Monitoring the performance of pumping systems gives important information on the process state to the pump user. The head, flow and power consumption values are key figures in finding out if the pumping system is really running efficiently and according to process needs. The head of the pump is usually monitored using pressure transmitters installed to pump inlet and outlet sections. Flow calculation can be executed using flow-meters, such as magnetic flow-meters or venturi tubes installed to piping systems. The installation of flow metering equipment to existing piping network can be both difficult and expensive [1][2].

ABB industrial drive module ACQ810 contains a flow control function that enables the calculation of flow without a separate flow-meter. The function is initially meant for single pump applications but it can also be used in parallel pump systems if the parallel-connected pumps share the same inlet and outlet conditions.

II. Different flow calculation methods

The software can calculate the flow either by measuring the total head of the pump (QH measurement) using two pressure sensors, or by measuring the power input of the pump (QP measurement). It can also use a combination of the QH and QP measurements. To calculate the flow, the pump characteristics, pump inlet and outlet diameters and the height difference of the pressure sensors must be entered as parameter values in the software. The pump characteristics set as parameter values are based on the pump performance curves supplied by the pump manufacturer. The flow calculation in the ACQ810 drive is based on Bernoulli's equation. The widely accepted Bernoulli's equation describes the balance between pressure, velocity and elevation of a fluid moving along a streamline. Based on Bernoulli's equation, the head of the system is often stated as follows where total head (H) can be calculated from elevation z, pressure difference between inlet and outlet tanks Δp, liquid density ρ, mean velocity v, gravity g and friction head $\sum H_r$ [3][4].

$$H = z + \frac{\Delta p}{\rho g} + \frac{v^2}{2g} + \Sigma H_r \tag{1}$$

A. Flow calculation based on the QH curve

The total head of the pump (QH) is measured using two pressure sensors installed at the pump inlet and outlet tappings.[2] When the total head has been calculated, the program interpolates the flow using the pump's QH curve set as parameter value in the software. The actual flow is calculated using the affinity equations (2) and (3) where H stands for head and Q for flow rate and n for pump speed [5][6].

The relation between measured head at actual speed and head at nominal speed is determined using equation (2). Again, the relation between interpolated flow at nominal speed and actual flow at actual speed is determined using equation (3). The software measures the head and iterates the velocity of the flow 100 times per second.

$$\frac{H'}{H} = \left(\frac{n'}{n}\right)^2 \tag{2}$$

$$\frac{Q'}{Q} = \frac{n'}{n} \tag{3}$$

B. Flow calculation based on the QP curve

The software can also calculate the flow by measuring the input power of the pump (QP calculation). The pump input power can be evaluated on the basis of the motor efficiency set as a parameter value. When the pump input power has been calculated, the software interpolates the flow using the pump's QP performance curve entered as parameter value. The actual flow is calculated in a similar way as in QH calculation using the affinity equations (3) and (4) where P stands for power and n for pump speed [5][6].

978-1-4244-5697-0/10/$25.00

$$\frac{P'}{P} = \left(\frac{n'}{n}\right)^3 \tag{4}$$

Flow calculation using the QP measurement is the simplest of the three flow calculation alternatives available in the ACQ810.

With the combination of the QH and QP measurements, the flow calculation starts with the QP measurement and at the breakpoint changes to the QH measurement. This is based on the fact that usually centrifugal pump QH-curves can be described flat on lower quantities and steeper when approaching larger flow values. At the same time QP-curves are often steeper on low flow values but flat when reaching more quantity. In such operational states, where the performance curve is flat, the presented flow calculation methods tend to be more inaccurate, mainly because of interpolation in calculation procedure.

III. Flow calculation in parallel pumps

In some applications several pumps are connected in parallel to enable a higher flow rate. The flow calculation methodology has been developed already for a number of years at Lappeenranta University of Technology (LUT). A prior version of the calculation that uses the same principle was reviewed at LUT in 2005 – 2006. The current version is more sophisticated and predicts the flow more reliably.

A Laboratory equipment to test accuracy of flow calculation

The information presented in this paper is extracted from a research report on the use of VSDs to provide flow measurement in parallel pumping systems. The research project was undertaken at Lappeenranta University of Technology (LUT), Finland, in 2006 [7].

Two centrifugal pumps were installed in the system, as shown in fig. 1. Pump 1 was driven by a separate 15 kW electric motor while pump 2 was run by an integrated 5.5 kW motor. Both pumps were operated by ABB industrial drives for pump control. Pressure transmitters were installed in the pump inlet and outlet tappings to enable the program to calculate the flow, and the transmitters also ensured that pressure values could be obtained for the two pumps individually. The values were shown on the data logger.

The actual flow rate through the pumps could be measured using a magnetic flow-meter and venturi tube installed in the system. All data output by the measuring equipment was fed to the data logger. With the measuring equipment the operating point of the pumps could be studied both when they were running separately and simultaneously in parallel. The capacity of the laboratory piping system was relatively low when compared to the ratings of the pumps. However, the total head of the system could be adjusted with valves provided both in the piping for the individual pumps and in the common piping.

B Results

The flow rate calculated by the program was observed when the operating points of the pumps were set at a certain point on the QH and QP performance curves. The accuracy of the flow measurement and factors affecting the calculation were studied by varying the total head of the system with the valves and by varying the rotational speed of the pumps with the VSDs. In this way the flow calculation function could be studied in variable system conditions and flow situations.

C. QH calculation in parallel pumping systems

Fig. 2 and 3 show an example of the QH calculation mode of the program when pumps 1 and 2 are operating in parallel. The flow rate and total head for each pump are shown at selected

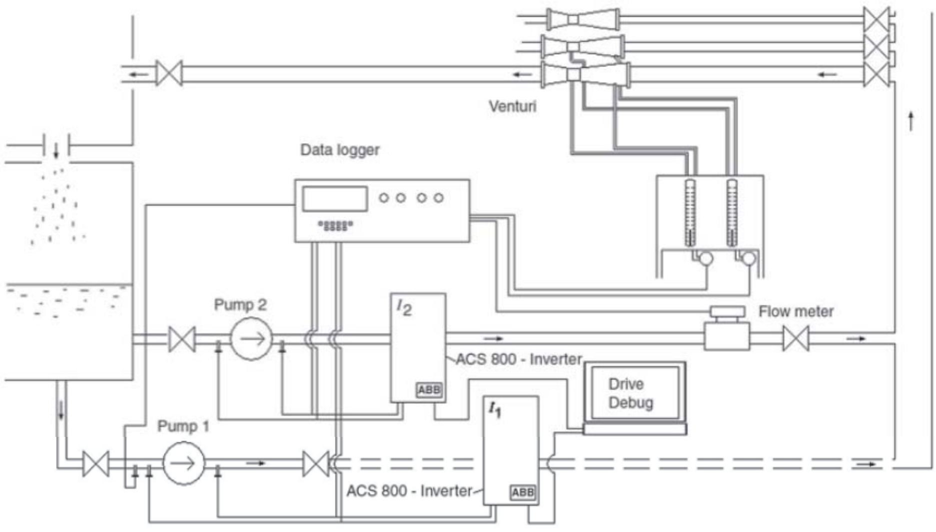

Fig. 1. The measuring equipment used in the project consists of two pumps and the attached variable speed drives.

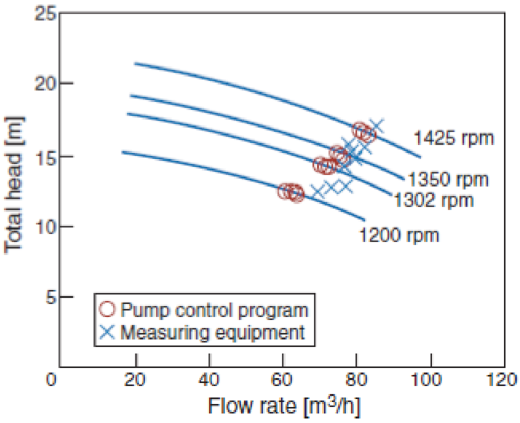

Fig. 2.With the QH calculation of pump 1 the measured values show a deviation of 0 – 5% with the higher speeds and of 13% with a lower speed.

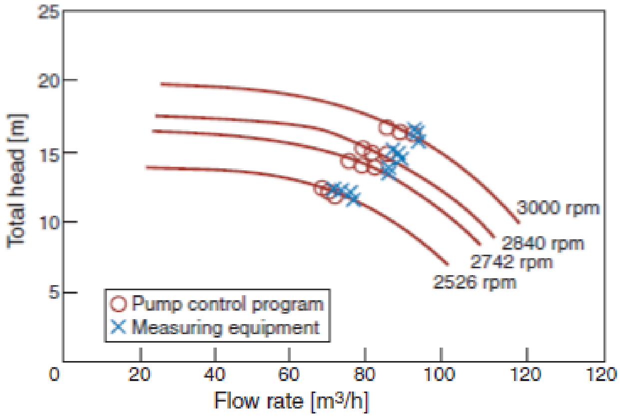

Fig. 3. With the QH calculation of pump 2 the measured values show a deviation of 1 – 7% for each selected rotational speed.

rotational speeds. As the measurements were conducted, the rotational speed of each pump was varied from 84% to 100% of nominal.

For pump 1, the values provided by the measuring equipment and the program's QH calculation mode differed by 0-5%. At a speed of 1200 rpm the difference increased to 13%. In the case of pump 2 the calculated flow rates show a difference of 1-7% for each selected rotational speed.

The same example of QH calculation is shown in figure 4, where the results can be seen in parallel QH curves. The results show that the sum flow provided by the QH calculation for each pump corresponds quite closely to the measured total flow. The difference between the measured total flow and QH calculated sum flow was 2-7% at rotational speeds 91-100% of nominal, and 10% when the rotational speed of the pumps was 84% of nominal.

Similar results were also obtained when the flow resistance of the piping was varied separately for the two pumps, and the pumps therefore had different total heads to achieve. The tests show that the program's QH calculation and the measuring equipment react in a similar way to changes in the piping system, for example if the flow resistance of the piping increases or the rotational speed of the pump varies. The QH calculation mode does not require stable system conditions for the total head, and the program's flow measurement function can provide reasonably accurate flow rates even in variable conditions.

D. QP calculation in parallel pumping systems

Flow measurement with parallel pumps was also tested using the program's QP calculation mode. Figures 5 and 6 illu-

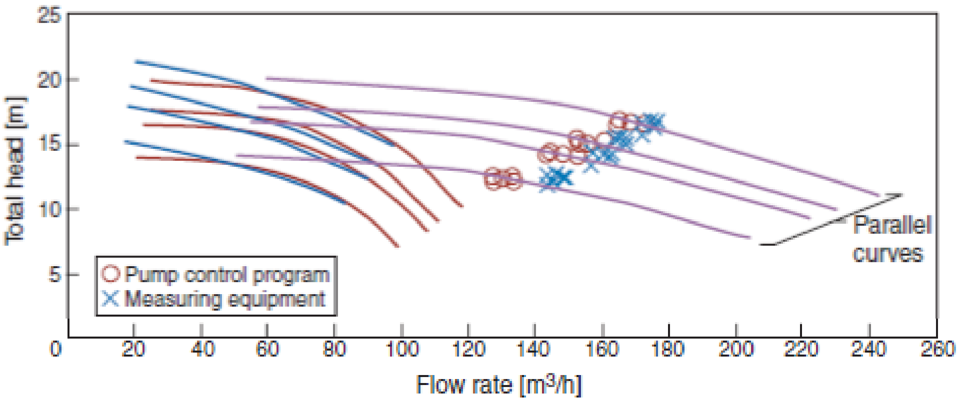

Fig. 4. The results show that the sum flow provided by the QH calculation for each pump corresponds quite closely to the measured total flow.

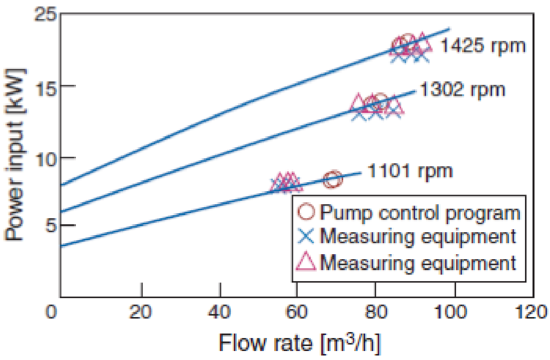

Fig. 5. The difference between the flow rate figures given by the measuring equipment and QP calculation was approximately 1-6% at each operating speed, pump 1.

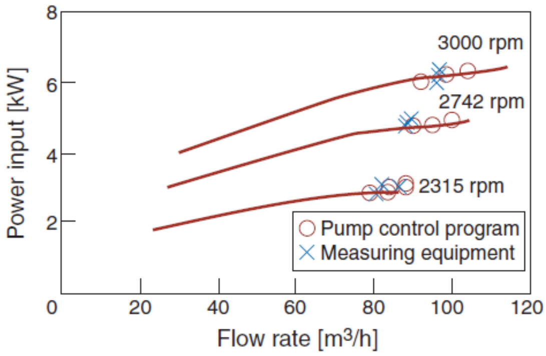

Fig. 6. The difference between flow rate figures from the measuring equipment and QP calculation was approximately 1-6%, pump 2.

strate an example of the QP calculation with two parallel pumps.

The graphs show the calculated flow rate and pump power input provided by the measuring equipment and program QP calculation mode with the pumps operating simultaneously in parallel. The difference between the flow rate figures given by the measuring equipment and QP calculation was approximately 1-6% at each operating speed.

However, there was a clear decrease in the accuracy of the calculated flow rate when the rotational speed of the pump was set at approximately 70% of nominal. This is mainly due to a flattening of the performance curves in accordance with the affinity laws.

The QP calculation gave similar results when the system head was adjusted using valves. In fact, the accuracy of the QP calculation increased for pump 2 when the total head was increased using valves, because the QP performance curve of the pump is steeper at low flow rates.

IV. Drawbacks of VSD based flow calculation

A major drawback of the QH calculation in parallel operation relates to the position of the operating point on the QH nominal curve defined in the parameter data. If the total head measured by the program does not lie between the parameter values, then the program will give the nearest known flow value as the calculated flow.

If the pumps are over dimensioned because of the piping system used, the actual operating point can easily be outside the QH performance curve provided by the pump manufacturer. This happens because manufacturers often specify pump performance only for the ideal operating range of the pump. This is a significant drawback, especially in parallel pump systems where it is more likely that the system curve is flat.

Pump manufacturers usually supply QP performance data for the same range as the QH performance data, and the QP calculation cannot give accurate flow rates if the operating point of the pump lies outside that range.

In some cases a centrifugal pump will have a QP performance curve that is very flat or bends, so that there are two separate flow rates for the same pump input power. The program's QP calculation function cannot be used in such conditions.

In certain circumstances air may enter a centrifugal pump. These types of pumps can handle liquids containing air or gas only to a limited extent, however, without a decrease in their performance. Air or gas accumulates at the impeller hub and the flow may collapse. At low inlet pressures, the pump pressure and power performance characteristics will fall even when the volume fraction of air or gas is only around 2% [3].

The deviation in the pump output caused by the gas means that the precision of the calculation deteriorates the more gas there is.

V. Conclusion

The flow calculation mode of the ABB industrial drive for pump control was studied and compared to theoretical models of parallel pumping. The flow measurement program of the drive was tested in a real parallel pumping system and the calculated flow rates were compared to those provided by separate laboratory measuring equipment.

The program's flow calculation modes represent an effective method for measuring flow without a separate flow-meter, even in parallel pumping systems. The study showed that calculating flow on the basis of pump performance can provide reasonably accurate (±0...7%) flow rates for each pump and for the sum flow. The accuracy of the flow calculation in parallel pumping - as in single pump installations - depends to a great extent on the accuracy of the pump characteristics provided by the manufacturer. The QH calculation mode of the program was more accurate and flexible in this case, mainly due to the characteristics of the tested pumps.

A drawback of both the QH and QP calculation modes relates to the position of the operating point on the system curve. If the actual operating point does not lie between the performance values defined in the parameter data, the flow calculation

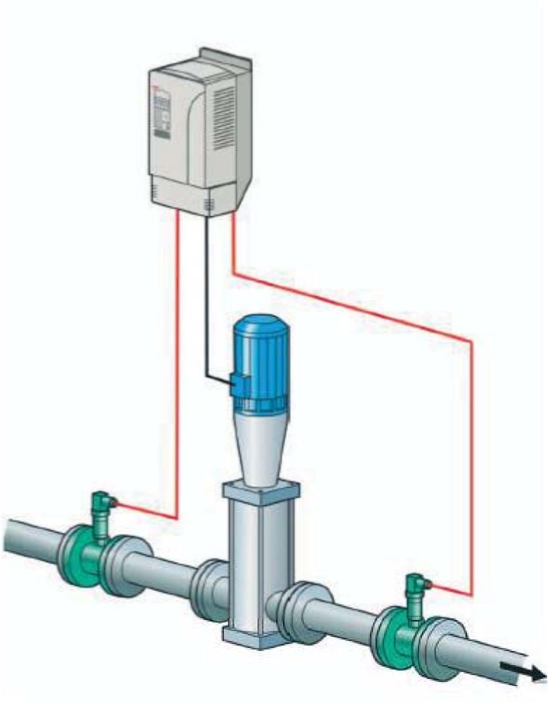

Fig 7. The VSD unit acquires the required data for flow calculation from the input and output valves of the system.

cannot give accurate flow rates. The shape of the system curve is also a significant factor.

The tests performed provided a good perspective on flow calculation using VSDs. The tests were made with the flow calculation software of ACS800. They can be also applied to the newer version ACQ810 that is more sophisticated and predicts the flow more reliably.

Using the ACQ810 flow calculation eliminates the need for a control valve and expensive flow-meter in single and parallel pump applications where the flow data is not required for invoicing purposes. A pump system including fewer electrical components is more reliable, particularly in the harsh environment typical of wastewater pump stations. Potential leakages within the same multi-pump system can be detected by comparing the calculated flow values at different points of the system. This requires that all pumps are controlled with the same kind of drives and software.

VI. References

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Publication V

Ahonen, T., Tamminen, J., Ahola, J., Viholainen, J., Aranto, N., and Kestilä, J.
Estimation of pump operational state with model-based methods

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Estimation of pump operational state with model-based methods

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ABSTRACT

Pumps are widely used in industry, and they account for 20% of the industrial electricity consumption. Since the speed variation is often the most energy-efficient method to control the head and flow rate of a centrifugal pump, frequency converters are used with induction motor-driven pumps. Although a frequency converter can estimate the operational state of an induction motor without external measurements, the state of a centrifugal pump or other load machine is not typically considered. The pump is, however, usually controlled on the basis of the required flow rate or output pressure. As the pump operational state can be estimated with a general model having adjustable parameters, external flow rate or pressure measurements are not necessary to determine the pump flow rate or output pressure. Hence, external measurements could be replaced with an adjustable model for the pump that uses estimates of the motor operational state. Besides control purposes, modelling the pump operation can provide useful information for energy auditing and optimization purposes.

In this paper, two model-based methods for pump operation estimation are presented. Factors affecting the accuracy of the estimation methods are analyzed. The applicability of the methods is verified by laboratory measurements and tests in two pilot installations. Test results indicate that the estimation methods can be applied to the analysis and control of pump operation. The accuracy of the methods is sufficient for auditing purposes, and the methods can inform the user if the pump is driven inefficiently.

1. Introduction

In industry, the major part of electric energy is consumed by electric motor applications, which are typically pump, fan, and compressor drives [1]. Hence, increased electricity prices and social awareness of environment have increased the public interest in the energy efficiency of these devices [2–4]. According to Karassik and McGuire [5], high energy efficiency can also be linked with improved system reliability stating the importance of operating electric motor applications with high efficiency.

Traditionally, a centrifugal pump is driven with a constant-speed induction motor, and the pump flow rate and head (i.e., pressure increase across the pump) is controlled by throttling the flow with control valves in the pipeline [5]. However, throttling causes additional losses as a part of the fluid's hydraulic energy is consumed in the control valve in order to achieve the required operational value. As the pump flow rate and head can be controlled without additional losses by varying the rotational speed of the pump, modern pump drives are equipped with a frequency converter that enables the speed control of an induction motor [6]. Rotational speed of the motor and the pump is then varied according to the external process command and respective measurement of the flow rate, output pressure, or other process variable.

Modern frequency converters are operated with sensorless control methods such as the direct-torque control (DTC) of an induction motor: by applying an internal motor model and optimum switching logic, the rotational speed of an induction motor can be controlled accurately without an external measurement sensor on the motor shaft [7]. In addition, the air-gap flux can be controlled in order to minimize losses in the motor and the frequency converter at reduced loads, which is discussed in [8].

However, the above works do not take the load machines into consideration. As the location of the pump operating point directly affects the pump efficiency and the shaft power requirement of the motor, it has a paramount impact on the total efficiency of a pump drive. This is demonstrated by the fact that the maximum practically attainable efficiency of a centrifugal pump is below 90%, when the nominal efficiency of a frequency converter is 95–98% and the nominal efficiency of an Eff1 induction motor is over 90%. Pump efficiency is also highly dependent on the operational state of the drive [5]. Correspondingly, information provided by a frequency converter is often limited to the motor shaft, although the user is more interested in the flow rate and pressure difference produced by the pump. Hence, the operational state of a pump (i.e., flow rate, head, and efficiency) should be estimated as well as the motor operation.

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Estimating the pump operating point in a frequency converter can enable a truly sensorless control of a pump drive: the pump drive operation can be controlled on the basis of flow rate and pressure estimates of a centrifugal pump without external measurements. Correspondingly, estimates for the pump operating point location can be used in the control of parallel-connected pumps [9–11]. These estimates can also be utilized in the energy audits and optimization of a pump drive components.

In this paper, two model-based estimation methods of the pump operational state (i.e., pump models) are presented. The methods are tuned with parameters describing the pump and the process characteristics. Estimates for the operational state of the motor are utilized with the presented methods. The main factors affecting the accuracy of the methods are analyzed. The estimation methods are evaluated by laboratory measurements. Their applicability is also tested with pilot installations in a paper mill.

2. Estimation methods

As shown in [12], the pump operating point can be estimated without external measurements of the pump flow rate Q and the head H. Then, the frequency converter's estimates on the rotational speed n_{est} and the shaft power P_{est} of an induction motor are applied to determine the operational state of the pump.

Next, two methods for estimating the flow rate and the head produced by the pump are described. The QP curve estimation method is based solely on the pump characteristic curves, while the process curve method also uses the characteristic curve of the process, in which the pump is located. The presented methods provide an opportunity to control the pump flow rate or the output pressure p_{out} without external measurements (Fig. 1): the estimated flow rate Q_{est} can be directly used in the control applications. However, as the head produced by the pump equals the pressure increase of the fluid across the pump, the fluid density and the fluid pressure value at the pump inlet are required to estimate the output pressure of the pump.

$$H = \frac{p_{out} - p_{in}}{\rho g} \tag{1}$$

$$p_{out} = p_{in} + \rho g H \tag{2}$$

where p is the fluid total pressure, ρ is the fluid density, g is the gravitational acceleration, and the subscripts in and out denote values at the pump inlet and outlet, respectively.

2.1. QP-curve-based estimation method

This estimation method is based on the flow rate Q of the pump vs. the power consumption P curve, which is typically given at the nominal speed of the pump (e.g. 1450 rpm). For unambiguous estimation results, the QP curve should be constantly increasing

$$\frac{dP}{dQ} > 0 \tag{3}$$

If the rotational speed of the pump differs from the nominal value, it affects the pump characteristics, which can be demonstrated by affinity equations:

$$\frac{Q}{Q_{nom}} = \frac{n}{n_{nom}} \tag{4}$$

$$\frac{H}{H_{nom}} = \left(\frac{n}{n_{nom}}\right)^2 \tag{5}$$

$$\frac{P}{P_{nom}} = \left(\frac{n}{n_{nom}}\right)^3 \tag{6}$$

where n is the instantaneous rotational speed of the pump, and the subscript nom denotes the value at the nominal speed [5].

When the rotational speed and the power consumption of the pump are estimated by the motor model [7], QP and QH curves can be transformed into the instantaneous rotational speed, and the location of the operating point can be solved from the transformed characteristic curves as shown in Fig. 2.

2.2. Process-curve-based estimation method

Since a centrifugal pump operates in the intersection of the pump and process QH curves (Fig. 3), the process curve provides basic information on the pump operating point location. When the pump QH curve is transformed into the current rotational speed of the pump by affinity Eqs. (4) and (5), the pump operating point can be determined by calculating the intersection location of these curves.

The process curve can be obtained from the knowledge of the piping and the static head by simple laws of hydraulics, or it can be determined by test measurements. In general, the process curve shape is defined by equation:

$$H_{process} = H_{st} + k \cdot Q^2 \tag{7}$$

where H_{st} is the static head of the process and k is the coefficient for the dynamic head (i.e., friction losses).

Thus, only two parameters are required to determine the process curve shape. However, the process curve shape may alter because of varying process conditions, which decreases the accuracy of this estimation method.

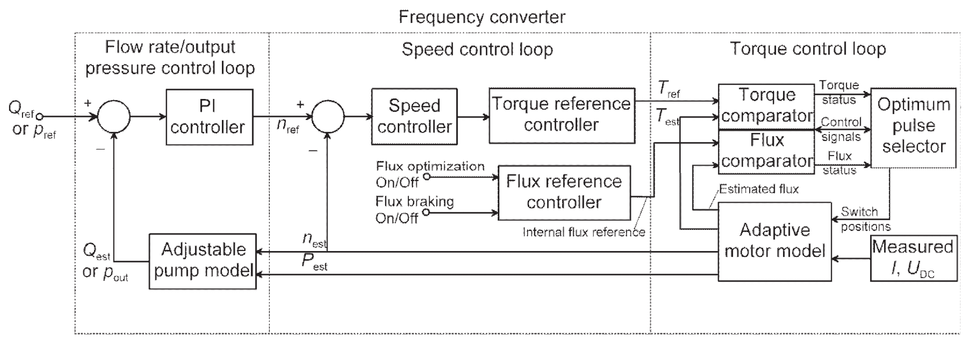

Fig. 1. Sensorless pump control system implemented in a DTC frequency converter [7]. Pump operation can be directly controlled by applying an adjustable pump model instead of external measurement sensors.

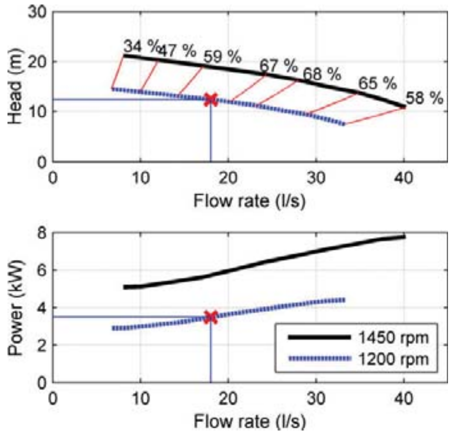

Fig. 2. Pump operating point estimation applying the *QP*-curve-based estimation method.

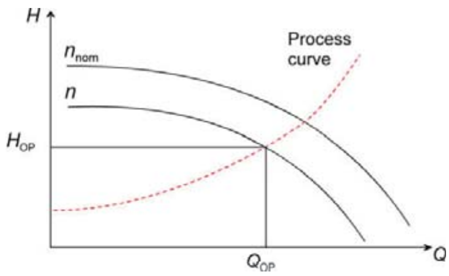

Fig. 3. Process-curve-based method for defining the pump operating point. Pump operates in the intersection of the pump and the process characteristic curves (Q_{OP}, H_{OP}).

3. Accuracy analysis of the methods

Accuracy of the both methods depends on the operational state of the pump, the accuracy of the pump characteristic curves, and the accuracy of the estimates provided by the motor model. In addition, the interpolation methods applied to produce the pump characteristic curves from a limited number of measurement points and to calculate the intersection locations may affect the estimation accuracy, but their impact can be considered insignificant, and therefore they are not discussed further in this paper.

3.1. Operational state of the pump

In general, the estimation methods may not provide accurate results in the operating region, where cavitation or flow recirculation may occur [12]. These hydraulic phenomena have a decreasing effect on the head produced by the pump, and hence they change the shape of the pump *QH* curve [5]. They also cause pressure fluctuations and in the worst case intermittent fluid flow, which is not considered in the estimation methods. On the other hand, near the best efficiency point (BEP) of the pump the risk of cavitation and flow recirculation is minimized. Hence, the estimated location of the pump operating point describes also the accuracy of the estimation method.

3.2. Pump characteristic curves

The pump characteristic curves provided by the manufacturer are usually measured to match with the grade 2 accuracy criteria defined in the ISO 9906 standard [13]. It determines the amount by which the real operating point can differ from the published pump curve: for the flow rate and the head the allowed tolerance is 3.5%, for the power consumption 4.0%, and for the rotational speed 2.0%, respectively (Fig. 4). Grade 1 accuracy criterion is typically applied with large (0.5–10 MW) pumps and the allowed tolerances are 2.0% for the flow rate, 1.5% for the head, 2.0% for the power consumption, and 0.5% for the rotational speed, respectively. In practice, pumps may be worn or the pumped fluid has different properties than water that also decrease the accuracy of the published pump characteristic curves. Hence, identification tests may be needed to determine the actual pump characteristic curves.

Besides the accuracy of the published characteristic curves, also the shape of the *QP* curve affects the accuracy of the *QP* estimation in combination with the imprecise estimations of the frequency converter; if the *QP* curve has a slow clime, even a slight inaccuracy of the power consumption estimate causes a notable error in the location of the pump operating point. Hence, the *QP* curve estimation method is typically applicable with radial flow pumps, which have a steep and constantly increasing *QP* curve shape (1).

The accuracy of the published pump characteristic curves was determined for a Sulzer APP 22-80 centrifugal pump (Q_{BEP} = 28 l/s, H_{BEP} = 17 m, n_{nom} = 1450 rpm), which pumps 20 °C water. The pump head was measured with absolute pressure sensors (0.25% accuracy), and the flow rate was measured applying a pressure difference measurement across the venturi tube. The pressure difference sensor was calibrated before the measurements. The rotational speed and torque of the motor were measured with a Dataflex 22/100 measurement shaft. The published torque accuracy of the measurement shaft is 1 Nm, which equals a 2–3.4% relative accuracy in the measurements [14].

The measured pump performance values and published characteristic curves with grade 2 accuracy are presented with their allowed tolerance limits in Fig. 5. For some reason, there is a systematic difference between the measured and published *QP* curve that cannot be solely explained by the allowed 4.0% tolerance and the uncertainty of the torque measurement. Most likely, the difference is caused by mechanical losses in the bearings and seals, as the pump *QH* curve is very accurate and shows no marks of pump deterioration. This difference can cause erroneous estimation results especially at reduced flow rates where the absolute flow rate estimation error may be even 7–10 l/s. Otherwise, the absolute estimation error of the flow rate is within 5 l/s, and the *QP* curve estimation method can produce accurate results for the operating point location in the steep region of the *QP* curve.

A simple method to correct the inaccuracy of the *QP* curve is to measure the power consumption against a closed discharge valve

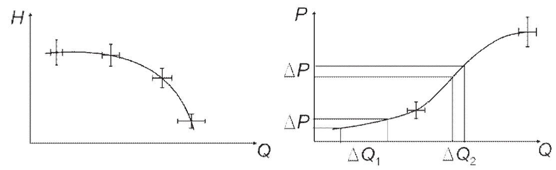

Fig. 4. Effect of the allowed tolerances and the *QP* curve shape on the estimation accuracy. Because of differences between the published and actual characteristic curves, the *QP* curve estimation method may produce inaccurate results. The *QP* curve should also be steep enough to ensure sufficient estimation accuracy. In this case, ΔQ_2 is notably smaller than ΔQ_1 because of the larger d*P*/d*Q* value at the flow rate Q_2.

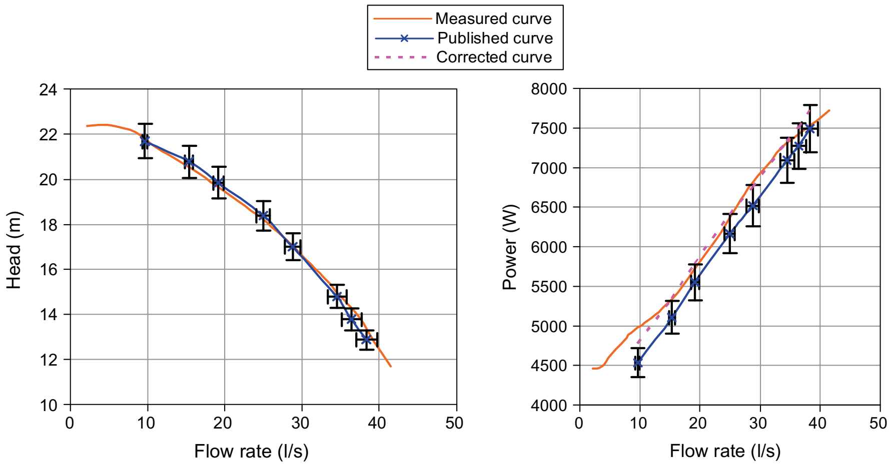

Fig. 5. Measured and published characteristic curves of a Sulzer APP 22-80 centrifugal pump. The difference between the measured and the published *QP* curve affects the estimation accuracy especially at small flow rates. The closed discharge valve test method notably improves the accuracy of the published *QP* curve.

at various speeds, and correct the location of the published *QP* curve on the basis of these results: according to [15], the difference between the actual and the published *QP* curve is often caused by mechanical losses in the bearings and seals of a centrifugal pump. These losses have a constant effect on the actual *QP* curve, and therefore the slope of the power curve stays relatively constant. By applying a constant 240 W correction on the published *QP* curve, its accuracy is notably improved against the measured *QP* curve. Then, the estimation accuracy of the absolute flow rate is within 3 l/s.

3.3. Internal estimates of the frequency converter

The frequency converter estimates the motor operational state on the basis of current and voltage measurements [7]. Since the accuracy of motor estimates is crucial for successful operation without a motor shaft speed measurement, frequency converters designed for sensorless applications are also applicable to pump operation estimation. According to [16], a typical estimation accuracy of a direct-torque-controlled (DTC) frequency converter is 10% of the induction motor nominal slip for speed (e.g. 0.3%), and 2% for torque, respectively. A specific accuracy for a motor shaft power estimate (i.e., pump power consumption) has not been presented, but on the basis of speed and torque accuracy values, the shaft power estimation error is 2.25% in the case of a four-pole ABB 1465 rpm induction motor. For comparison, the admissible total uncertainties of grade 2 pump speed and input power measurements are ±2% and ±4%, respectively. Hence, an estimation error in the power consumption may affect the accuracy of the *QP* curve estimation method. On the other hand, the speed estimate can be considered very accurate for the process curve estimation method.

The accuracy of internal estimates was determined with test measurements for an ABB ACS 800 frequency converter, which applies the direct-torque control method. The frequency converter fed a four-pole ABB 11 kW induction motor. The rotational speed and torque of the motor were measured with a Dataflex 22/100 measurement shaft over the entire operating region of the Sulzer pump. Converter estimates for the rotational speed, the shaft torque, and shaft power were stored utilizing ABB DriveDebug software. Relative errors of the estimated values were calculated, and they are presented in Fig. 6a. As expected, the accuracy of the rotational speed estimate is within the 0.25% tolerance. Relative error of the shaft torque exceeds the 2% limit at partial and excessive flow rates, which weakens the shaft power estimations in these regions. However, it should be noted that the relative torque estimation error equals with the published accuracy of the torque measurement shaft [14]. A shaft power estimation error has only a limited effect on the *QP* curve method accuracy, which can be seen from Fig. 6b. When the estimated shaft power values are applied to determine the flow rate from the measured *QP* curve, the estimation accuracy is within 2 l/s, which may be sufficient even for control purposes. By comparing these results with the test results shown in Fig. 5, one can see that the accuracy of the pump characteristic curves has a larger effect on the estimation accuracy of the *QP* curve method than the accuracy of the internal estimates of a frequency converter.

4. Laboratory tests

Laboratory tests were conducted to analyze the applicability of the both estimation methods, when the pump is driven in different operating points. Because of the varying process conditions and reference values for the pump flow rate or output pressure, a pump cannot work all the time near its best efficiency point. Hence, there is a risk that the estimation methods produce inaccurate results because of the adverse operating point location.

The both methods were tested with a laboratory drive comprising of an ABB ACS 800 frequency converter, an ABB 11 kW induction motor, and a Sulzer APP 22-80 centrifugal pump with a 255 mm impeller (Q_{BEP} = 28 l/s, H_{BEP} = 17 m, n_{nom} = 1450 rpm). The pump drive is equipped with a Dataflex 22/100 torque and speed measurement shaft (Fig. 7). The pump drive is attached to a closed-loop system, which consists of two water tanks. The shape of the process curve can be modified with control valves. The head produced by the pump was measured with two absolute pressure sensors across the pump (0.25% accuracy), and the flow rate was measured utilizing a pressure difference measurement across the venturi tube. The pressure difference sensor was calibrated before the measurements. The accuracy of the venturi system has been previously verified with an ISOIL electromagnetic flow meter to be within 2 l/s over the entire pump operating region. The rotational speed and torque of the motor were measured with the

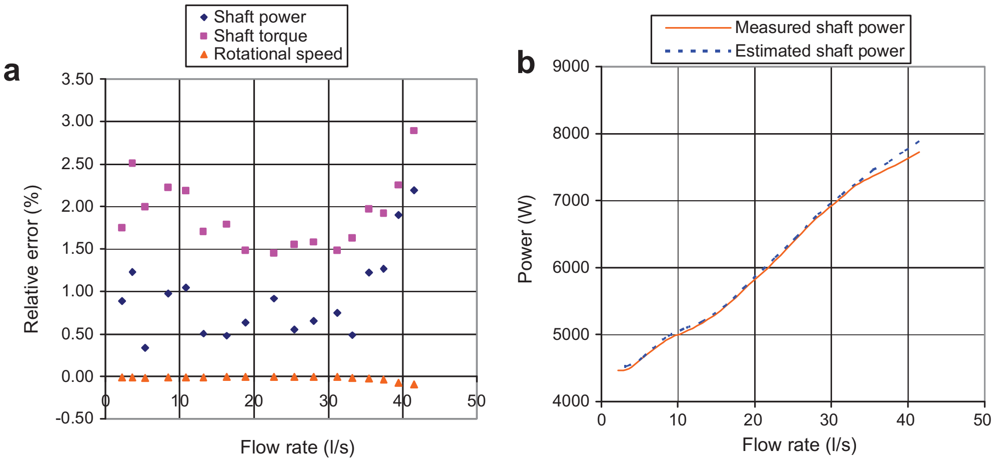

Fig. 6. Measurement results for the estimates of an ABB ACS 800 frequency converter. (a) Relative errors of the shaft power, the shaft torque, and the rotational speed estimates. Accuracy of the frequency converter estimates is within 3%. (b) Effect of the shaft power estimation on the pump *QP* curve estimation method. Accuracy of the shaft power estimates is sufficient even for control purposes.

Fig. 7. Pump drive applied in the laboratory tests.

Dataflex measurement shaft. Operation of the frequency converter was monitored with ABB DriveDebug software.

Different operating points were generated by using a different valve setting for each measuring sequence. The valves were adjusted so that the relative flow rate was 50%, 70%, 100%, 120%, and 140% of 28 l/s at 1450 rpm. With these valve settings, the pump was driven at rotational speeds from 1020 rpm to 1560 rpm. The rotational speed, the shaft torque, and the shaft power estimates of the frequency converter were stored, and the actual operating point was measured with pressure measurements. Also the pump rotational speed and shaft torque were measured to analyze the accuracy of the frequency converter's estimates. The pump characteristics curves were measured separately and they were applied to the estimation methods. This eliminates the effect of inaccurate pump characteristic curves on the estimation results. For each measuring sequence, the process curve shape was generated based on the knowledge of the piping and valve positions.

The results for the measuring sequence with the nominal flow rate valve setting are presented in Fig. 8a. With the *QP* curve method, the estimation error of the head and the flow rate is less than 6% compared with the measured values, and in the case of the process curve estimation method, the estimation error of the head and the flow rate error is within 3%. The accuracy of the speed estimate is within 0.25% and the shaft power estimate error is less than 4%.

In the case of 50% of the nominal flow rate, the pump operating point estimates were located as seen in Fig. 8b. The relative flow rate error of the process curve estimation method is less than 2%, and with the *QP* estimation method it is within 8%. In this case, the *QP* curve method is the most accurate at rotational speeds of 1230–1410 rpm. When the motor is driven at the speed of 1000–1200 rpm or over 1500 rpm, the estimations differ from the measured operating points. This is mainly caused by the inaccurate affinity transforms and the flat *QP* curve shape at the flow rates 10–17 l/s (Fig. 6b), since the accuracy of the shaft power estimate is within 2%. However, also in this region the results given by the *QP* curve estimation method are accurate enough to determine the operational state of the pump.

When operating the pump in the far-right region of the pump *QH* curve, where the risk of cavitation is increased, the *QP* estimation method results in erroneous estimates (Fig. 8c). This is caused by the decreasing dP/dQ at flow rates above 30 l/s and the increased error in the power estimates: in this measurement sequence, the shaft power estimation error was 4–8%. However, the location of the estimated points shows that the pump is driven inefficiently, which is crucial information for monitoring and energy auditing purposes. The relative flow rate error of the process curve method is within 3%.

According to the test results, both methods are applicable to auditing and pump operation monitoring purposes. Since the accuracy of the presented methods depends on the pump operating point location and the accuracy of pump characteristic curves, they cannot be recommended for control purposes, where strict accuracy in the flow rate or pressure control is required.

5. Pilot tests

The applicability of the *QP* curve estimation method has also been tested in a paper mill with two separate pump drives. The

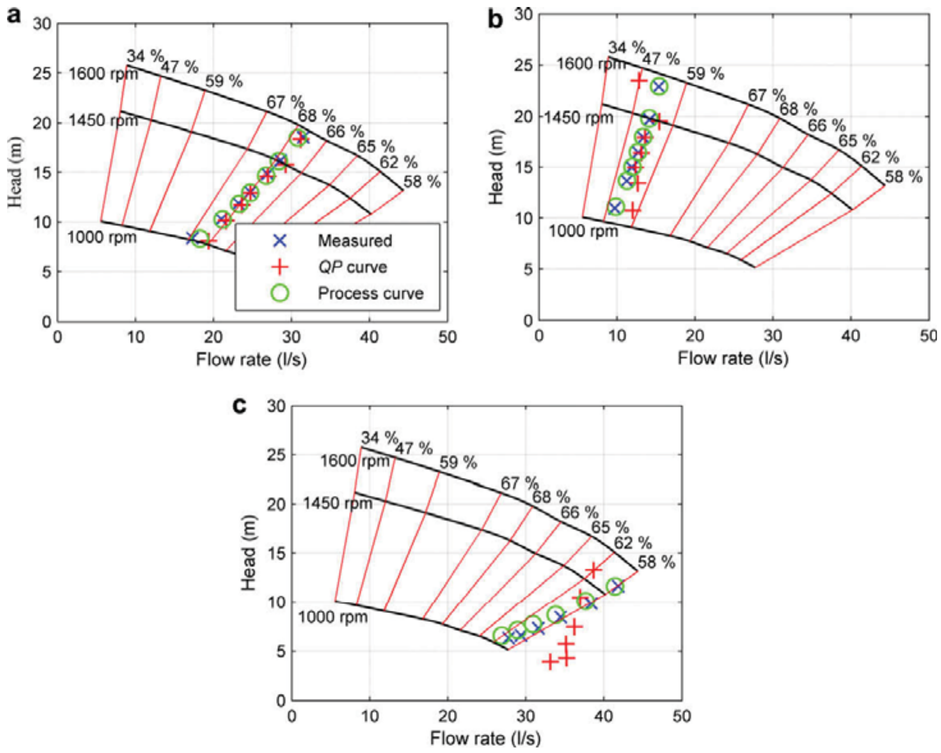

Fig. 8. Results of the laboratory tests for the estimation methods. (a) Measurement sequence with the nominal flow rate valve setting. (b) Measurements with the 50% of nominal flow rate. (c) Measurements with a 140% of nominal flow rate.

operation of the primary water pump and the mass pump drive was monitored using a frequency converter. The primary water pump drive consists of an Ahlström P-X80X-1 vertical wet pit pump, a Strömberg 550 kW induction motor, and an ABB ACS 800 frequency converter. The nominal values of the vertical wet pit pump are 1.85 m^3/s for the flow rate, 22 m for the head, 85% for the efficiency, and 740 rpm for the rotational speed, respectively. The water pump transfers cold water from the water station further to the mill, which enables the use of original pump characteristic curves for analysis. However, there is no way to define a constant process curve for the water pump, which excludes the use of the process curve estimation method in this case.

The mass pump drive comprises a Sulzer ARP 54-400 centrifugal pump, an ABB 400 kW induction motor, and an ABB ACS 600 frequency converter. The nominal operating values of the mass pump are 675 l/s, 24 m, 85%, and 990 rpm, respectively. The pumped fluid consists of water and tree fibers. However, fluid consistency is only 1.5% and it has the same density as water, which enables the use of original characteristic curves for analysis also in this case. The mass pump is located next to the paper machine 3, and it transfers the mass from a reservoir to the tanks near the paper machine. The process curve method has not yet been tested in this case.

In both cases, the published pump *QP* curves are more flat than the curve in Fig. 6b, and there is no certainty on their accuracy. This limits the use of *QP* curve estimation method for control purposes, and naturally decreases the reliability of a single estimate for the pump operating point location. In addition, both pumps are pressure controlled, but information provided by the pressure measurements was not available for the verification of estimation results.

Hence, estimations of the pump operating point locations were carried for a period of 6 months in order to determine all the typical operational states for both cases. Although a longer estimation period does not improve the accuracy of the estimated results, it gives a picture on the most typical operational states of the pump. The rotational speed and shaft power estimates provided by the frequency converter were fetched and stored in every 5 min to a personal computer, which was connected to the frequency converter. The resulting figures for operating point locations are presented in Fig. 9. The most typical operating point location is indicated with an asterisk. The normal distribution of operating points (about 68% of all points) around the most typical operating point is demonstrated with an ellipse.

The estimation results for the primary water pump drive are shown in Fig. 9a. According to the results, the pump operating point locations vary all over the *QH* curve. The major part of the operating point locations lie at partial flows, thereby resulting in pump operation with weak efficiency. Compared with the nominal efficiency, 85%, the pump efficiency is typically 55% and the power consumption is about 285 kW, which may result even in a 100 kW difference in the pump power consumption. Hence, the sizing of the pump should be reconsidered.

The estimation results for the mass pump drive are shown in Fig. 9b. Its process function is to provide the mass delivery into tanks, and hence it has a more uniform operating point location distribution than the primary water pump. Also in this case, the

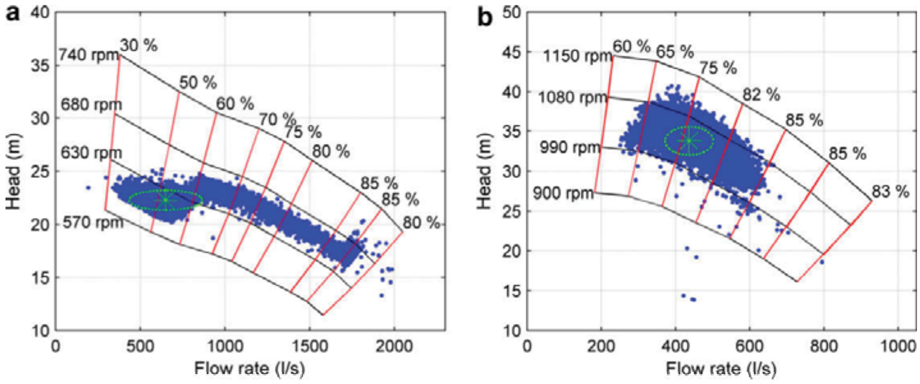

Fig. 9. Estimation results for the pump drives in a paper mill. (a) Estimation results of the primary water pump drive. Since the pump duty varies greatly depending on the water consumption, the operating points are distributed over the *QH* curve. However, over 70% of the points are located in the region with reduced efficiency. Hence, pump operation causes additional energy costs. (b) Estimation results of the mass pump drive. Process variations are considerably smaller compared with the primary water pump drive. Neither in this case, the pump is not typically working in the best efficiency point.

most typical operating point locations are not in the pump's best efficiency point, which affects negatively the total efficiency of the pump drive. The typical power consumption of the mass pump is 240 kW. However, the difference between the nominal and the most typical efficiency is less than 10% units, which results in a 30 kW saving potential.

6. Conclusion

Model-based methods for pump operation estimation are presented. They provide a cost-efficient way to control and analyze the operation of the pump drive. Applicability of the methods is tested in a laboratory. The test results show that the *QP* and process curve estimation methods are applicable, if the available characteristic curves describe the pump and system operation accurately enough. The accuracy of the methods is sufficient for auditing purposes, and they can inform the user if the pump is driven inefficiently. This is demonstrated with two pilot cases, in which the pumps are operating with reduced efficiency.

Role of funding source

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Publication VI

Tamminen, J., Viholainen, J., Ahonen, T., Ahola, J., Hammo, S., and Vakkilainen, E.
Comparison of model-based flow rate estimation methods in frequency-converter driven pumps and fans

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ORIGINAL ARTICLE

Comparison of model-based flow rate estimation methods in frequency-converter-driven pumps and fans

Jussi Tamminen · Juha Viholainen · Tero Ahonen · Jero Ahola · Simo Hammo · Esa Vakkilainen

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Abstract Fluid handling systems such as pump and fan systems are found to have a significant potential for energy efficiency improvements. To determine the energy-saving potential, there is a need for easily implementable methods to monitor the system output. This essential information provides an opportunity to identify inefficient operation of the fluid handling system and to control the output of the pumping system according to process needs. Model-based pump or fan monitoring methods implemented in variable-speed drives have proven to be able to give information on the system output without additional metering; however, the present model-based methods may not be usable or sufficiently accurate in the whole operation range of the fluid handling device. To apply model-based system monitoring in a wider selection of systems and to improve the accuracy of the monitoring, this paper proposes a new method for pump and fan output monitoring with variable-speed drives. The method uses a combination of already known operating point estimation methods. Laboratory measurements are used to verify the benefits and applicability of the improved estimation method, and the new method is compared with five previously introduced model-based estimation methods. According to the laboratory measurements, the new estimation method is the most accurate and reliable of the model-based estimation methods.

Keywords Energy efficiency · Fans · Fluid flow measurement · Pumps · Variable-speed drives

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Introduction

Fluid handling systems, such as pump and fan systems, are widely used in industry, and they are responsible for a significant part of electrical energy use. It is estimated that a quarter of all electricity consumption in the industrial sector is caused by pump and fan applications (de Almeida et al. 2005). Nowadays, the energy efficiency of these systems has become more important as the political and ecological pressure to reduce energy consumption is increasing (European Parliament 2009). In variable-flow fluid handling systems, the use of rotational speed control instead of throttle control has shown to improve the system energy efficiency, and many studies make a strong argument to increase the number of frequency converters, often called simply variable-speed drives (VSDs) in fluid handling systems (de Almeida et al. 2003, 2005; Binder 2008; European Parliament 2009; Ferreira et al. 2011).

The energy efficiency of a fan or a pump is dictated by its operating point location, which represents the pump or fan output of the system in terms of flow rate, pressure and power consumption. Traditionally, the operating point of a certain pump or fan in a system is studied using direct flow rate and pressure measurements. However, such direct metering is seldom available or can be difficult to implement in industrial systems, and thus, pump and fan users are often satisfied when the delivered output serves the process without unexpected interruptions. However, as the location of the fan or pump operating point is essential information when evaluating the energy saving potential of the system, it has become increasingly relevant for the users to find out how the pumps and fans are actually operated, as this information can be used as a basis for energy efficiency improvements.

To avoid the direct metering of the system output, some easily implementable output monitoring methods have been developed. VSDs can nowadays estimate the mechanical power and rotational speed of the motor with a good accuracy as model-based motor control methods have evolved over the years. This makes it possible to estimate the pump or fan operating point with the help of the VSD without using any additional measurements of the fluid handling system. The idea of model-based operating point estimation used in the VSDs is to apply the pump or fan characteristic curves or system details and selected monitoring values (e.g., pressure in the inlet and outlet sections of the pump, measured input power or torque) to estimate the pump operating point. Giving information of the pump operating point, model-based methods can be used in the energy-efficient control of pumping systems, for instance, in the parallel pump control strategy proposed by Viholainen et al. (2011, 2013).

The simplest examples of model-based methods that are suitable to find the pump and fan online monitoring are the basic *QH*-curve-based method in Fig. 1a and the *QP*-curve-based method in Fig. 1b. These methods use the VSD rotational speed estimate (Nash 1996; Ahonen et al. 2011) and the measured pressure or the VSD power estimate as inputs in the *QH*- and *QP*-curve-based methods, respectively. The head as a function of flow rate (*QH*) and power as a function of flow rate (*QP*) characteristic curves, given by the manufacturer (Sulzer Pumps 2010), are used as the model in these methods in combination with the well-known affinity laws. The affinity laws are as follows:

$$Q = \left(\frac{n}{n_0}\right) Q_0 \tag{1}$$

$$H = \left(\frac{n}{n_0}\right)^2 H_0 \tag{2}$$

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

where the subscript $_0$ denotes the initial values, Q is the flow rate, P is the mechanical power, n is the rotational speed, and H is the total head, consisting the total head difference between the pump suction and pressure sides. In fan systems, the fan output is typically observed with total pressure p comprising the total pressure difference between the inlet and outlet of the fan in pascals instead of total head in meters. Thus, the fan characteristic curve of the total pressure as a function of flow rate (*Qp*) is referred to as the *QH* curve, for the sake of convenience. In variable-speed operation of pumps and fans, affinity laws are widely used to generate new *QH* and *QP* curves for a pump or a fan running at a different speed than published or tested for.

Usually, it is not taken into consideration that the affinity laws are not exactly correct when the rotational speed n differs by more than 20 % from the initial rotational speed n_0, because the efficiency of the fan is affected by changing rotational speeds (Muszýnski 2010). The estimation methods presented in the references of this paper do not take into account the effect of an efficiency change on the accuracy of the affinity laws; however, this speed-related change in pump efficiency has been modeled, for instance, by Muszýnski (2010) and Marchi et al. (2012). Since the use of efficiency-based corrections in the affinity laws yields ambiguous characteristics according to the studies referred to in this paper, they are left untreated in this article also. Instead, the effect occurring on the model-based estimation accuracy in the studied methods is reduced by other solutions.

The major limiting factor in the use of the basic *QH*- and *QP*-curve-based methods is related to the shape of the characteristic curves. Depending on the shape of the curve, the basic estimation methods may not be applicable to the whole operating region of the pump or the fan. For example, if the fan in Fig. 1 is operated at 1,450 rpm, and it produces a 1,600 Pa pressure, it cannot be said whether the fan produces 0.2 or 1.8 m^3/s resulting in that the basic *QH*-curve-based method cannot give adequate results on the operating point location. Moreover, if the applied characteristic curve is flat, even

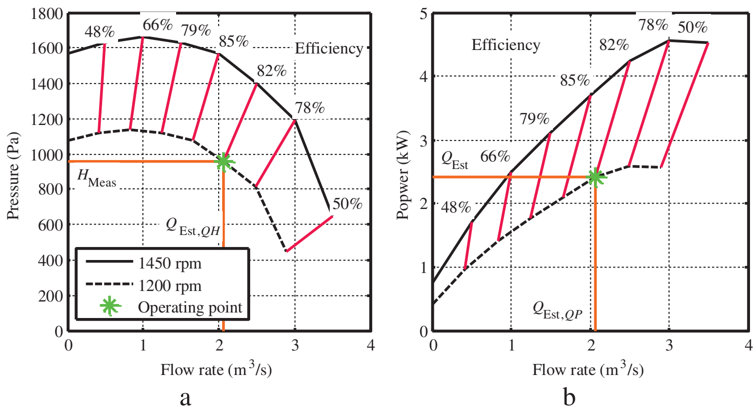

Fig. 1 Application of model-based pump operating point estimation methods in estimating the flow rate. The characteristic curves are shifted to the instantaneous rotational speeds by using affinity laws and the flow rate corresponding to the measured total pressure (**a** *QH* method), or the estimated power (**b** *QP* method) is defined based on the shifted curve

a small error in the model or the input can have a significant effect on the estimated operating point. The functionality of these basic estimation methods and the associated problems are discussed extensively by Ahonen et al. (2010), Hammo and Viholainen (2006), Tamminen et al. (2011), Liu (2002), Wang and Liu (2005), and Kernan et al. (2011). In this paper, these estimation methods are referred to as the *QH* method and the *QP* method.

An improvement in increasing the accuracy and applicability of the basic *QP* method has been studied by Ahonen et al. (2012). In this study, the *QP* method is used to identify the system curve near the pump nominal rotational speed. The resulting information on the shape of the system curve can be used together with the VSD rotational speed estimate and the fan *QH* curve to estimate the operating point of the pump. This hybrid method has shown to improve the accuracy of the operating point estimation at low rotational speeds. This is partly due to the fact that the method is able to reduce the effect of the speed-related change in the blower efficiency. Still, the usage of the hybrid method is limited in a similar way as the basic *QP* method: if the pump *QP* characteristic curve is inaccurate or unsuitable for this estimation method, the hybrid method cannot improve the estimation accuracy compared with the normal *QP* method, since the system curve cannot be identified.

In a patent by Kernan et al. (2011), the accuracy of the pump *QP* characteristic curve is improved with a closed valve test. This test gives the actual pump shut-off power with different rotational speeds, and the characteristic curve can be treated with this information. However, the method requires that the pump must be operated in abnormal operating conditions. Moreover, because the fan *QP* curve is only verified in a no-flow situation, the operation at other flow rates remains unknown, and thus, operation, for example, at nominal flow rates cannot be verified. In addition to the *QP* method, the same limitations apply to the Kernan method as well. Alternatively, the Kernan method can also be used with a shut-off pressure measurement. This is done in the same manner as in the normal Kernan method, where instead of power in shut-off conditions, the shut-off pressure is measured at different rotational speeds, and these measurements are used to correct the *QH* curve. This estimation method is referred to as the Kernan–*QH* method. However, this method also has the same limitations as the normal *QH* method and suffers from the same deficiencies as the Kernan method.

da Costa Bortoni et al. (2008) suggested a model for the pump that can be used to estimate the pump output without exact pump curves. The model of the pump is generated from the measured shut-off head and the measured nominal point of the pump. It can be used in the same manner as the *QH* method in the estimation of the pump flow rate, when the pump total head is measured. However, if startup measurements are unavailable, the model can be constructed using only the rated shut-off head and the nominal pump operation point, which can reduce the accuracy of the model compared with the use of characteristic curves. The method of da Costa Bortoni et al. (2008) for the flow rate estimation is later called the Bortoni method.

The aim of this paper is to present a solution that allows the model-based methods to be applied to a wider selection of systems and to increase the accuracy of the model-based methods compared with the existing methods. Since the introduced solution is based on the selective use of basic estimation methods, the suggested new method is referred to as the *QH*/*QP* method. The suggested method uses the calculated uncertainties of the *QH* and *QP* methods to find the most suitable estimation method. In addition, if the methods are equally uncertain, the method combines the *QH* and *QP* flow rate estimates to ensure more accurate flow rate estimation. The benefits and usability of the proposed method are validated by exemplary laboratory measurements and compared with other known model-based estimation methods.

The structure of the paper is as follows. In Combined QH/QP estimation method section, a method to improve the flow rate estimation of pumps and fans with the suggested combined *QH*/*QP* estimation is studied. The laboratory measurements to confirm the benefits of the suggested estimation method are presented in Laboratory measurements section, and the results are discussed in Results section. Conclusion section concludes the paper.

Combined *QH*/*QP* estimation method

The proposed *QH*/*QP* estimation method selects and combines two known operating point estimation methods (the *QH* and *QP* methods) in a novel way to determine the operating point location of a pump or a fan more reliably than the previously introduced methods. The *QH* method estimation typically has its best accuracy when operating approximately at the nominal flow rate or above it. Correspondingly, the *QP* method can typically give the most accurate estimation at nominal flow rates or below it. This is because of the inherent shape of the fan and pump characteristic curves (Ahonen et al. 2011; Tamminen et al. 2011; Kernan et al. 2011; Sulzer Pumps 2010). Thus, the overall accuracy of the model-based operating point estimation can be improved by the selective use of the basic *QH* and *QP* methods in appropriate regions.

The performance curve regions, which would cause uncertainty to the basic flow estimation methods, can be automatically calculated based on the characteristic curve data and the estimated error in the pressure measurement and power estimate. For example, the derivative of the characteristic curve gives a rough estimate of the error that can be produced by the erroneous pressure measurement or power estimate. Because of this, it is justified to use the characteristic curves to determine the usable regions associated with each estimation method. An example of the calculation of this flow rate estimation uncertainty U_{QH} for the *QH* method is as follows:

$$U_{QH}(Q) = \frac{\mathrm{d}p\left(Q_{\mathrm{Est},QH}, n_{\mathrm{Est}}\right)}{\mathrm{d}Q} \Delta p_{\mathrm{Meas}} \tag{4}$$

where Δp_{Meas} is the expected error in the pressure measurement; and the subscripts $_{\mathrm{Est}}$ and $_{\mathrm{QH}}$ denote the estimated value and the *QH* method, respectively. Thus, the flow rate estimation uncertainty can be calculated as a function of flow rate and rotational speed. In addition, a more complex equation for the estimation uncertainty can be formulated using the ISO 13348 standard that gives the tolerances of the fan characteristic curves. For example, in standard industrial fans, the accuracy is given with the ISO 13348 class AN3 accuracy (ISO 2012). The standard defines the tolerance for the flow rate and the fan pressure to be within ±5 % and the power +8 % of the given curves. In addition, the tolerances change when the fan is operated at flow rates where the fan efficiency has decreased to less than 90 % of the best efficiency. Therefore, the allowed tolerances for the performance curves are often significant error source in system output monitoring.

The procedure of the *QH*/*QP* method is as follows: first, both the *QH* and *QP* methods are used to estimate the flow rate, and the uncertainty in the estimates is calculated. Secondly, the method that has a lower uncertainty is selected as the estimate of the flow rate. If the uncertainties in the estimates are approximately the same, the flow rate estimates can be weighted so that both flow rate estimates are taken into account. There are several options to do this, for instance, the uncertainties calculated by (5) can be used in weighting with

$$Q_{\mathrm{Est}} = \frac{U_{QP} \cdot Q_{\mathrm{Est},QH} + U_{QH} \cdot Q_{\mathrm{Est},QP}}{U_{QP} + U_{QH}} \tag{5}$$

where the subscript $_{\mathrm{Est}}$ denotes the estimation result; and *QP* and *QH* denote the *QP* and *QH* methods, respectively. To demonstrate why the uncertainty of one method is multiplied by the flow rate estimate of the other method, an example is given: if U_{QP} is 0.5 m^3/s and U_{QH} is 0.3 m^3/s, the *QH* method is less uncertain. The flow rate estimate of the *QH* method has now a weight of five-eighths and the *QP* method flow rate estimate, only three-eighths, and thus, the less uncertain method has a higher weight.

There can be certain characteristic curves where the *QH* or *QP* methods would produce two or more estimates for the flow rate based on a single head or power value, causing the estimation methods to be unusable in a certain part of the curve. In these cases, the characteristic curve is divided into monotonic parts, and the unambiguous estimation method is used to choose the correct monotonic part for the estimation method. Then, the previously unusable estimation can be used to estimate the flow rate. When both methods have produced an estimate of the flow rate, the method with a lower uncertainty is chosen or the flow rate estimates are combined. An example of this can be seen in Fig. 2, where the *QH* method estimation produces several flow rate estimates, and the *QP* method is used to select the correct monotonic part of the *QH* curve. The *QH* method is then used with the selected monotonic part to estimate the flow rate. In the last step, the flow rate estimate with the lowest uncertainty is chosen as the flow rate estimate of the *QH*/*QP* method.

The combined *QH*/*QP* method requires the rotational speed, shaft power, and pressure as inputs, of which the rotational speed and shaft power inputs are directly available from the frequency converter. However, a total pressure measurement is not always installed in pump and fan systems. This is naturally a drawback in the commissioning of the *QH*/*QP* estimation method compared with the normal *QP* method, the hybrid method, and the Kernan method, which only require the rotational speed and the shaft power as inputs. In general, performing the pressure measurement according to the ISO measurement standards may also be challenging in actual pump and fan systems. This results from the requirement of straight duct parts both before and after the pressure measurement installation. On the other hand, the tolerances of the *QH* curve are much smaller

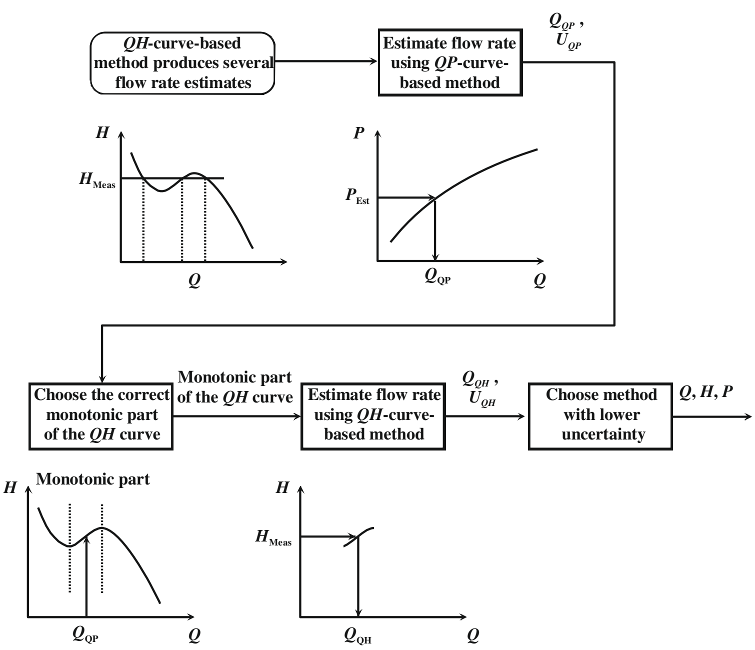

Fig. 2 Method to apply the *QH*/*QP* method in cases where the other estimation method produces several flow rate estimates, while the other one produces only one estimate. The subscript *QP* refers to the estimate of the *QP* method; the subscript *QH* to the estimate of the *QH* method; P_{Est} is the power estimate; and H_{Meas} is the measured head

than the QP curve tolerances in the ISO 13348 standard. Hence, the pressure-based methods should provide more accurate estimates of the flow rate and thus justify the installation of the pressure sensors.

Combined use of the QH/QP and hybrid methods

The proposed method can be further improved by using the QH/QP method with the hybrid method presented in (Ahonen et al. 2012). This can be executed as follows: similarly as by Ahonen et al. (2012), the system curve is estimated near the fan nominal rotational speed, but instead of using the QP method only, the QH/QP estimation method is used to identify the system curve. As stated above, the identification of the system curve in the hybrid method reduces the effect of the relative error in the pressure and power estimation, especially in the case of a substantial change in the rotational speed from the initial value and can, therefore, improve the estimation accuracy at low rotational speeds (Ahonen et al. 2012). Thus, applying the QH/QP method together with the hybrid method is clearly justified.

Laboratory measurements

The presented method was tested with a fan system consisting of a FläktWoods Centripal EU 630 MD radial blower (nominal values: 2.9 m^3/s; 1,190 Pa; 4.6 kW; 1,446 rpm), an ABB induction motor (nominal values 400 V; 21 A; 7.5 kW; 1,450 rpm), and an ABB ACS850 variable-speed drive (nominal current 35 A). The observed system contains 6 m of piping on the blower inlet and outlet. The blower static pressure difference between the fan suction and pressure sides was measured using pitot static tubes and a Rosemount 2051C differential pressure meter. The produced flow rate was measured using an Eldridge Series 9800MPNH thermal flow meter. Together with the static pressure difference and flow rate measurements, the fan total pressure was calculated. The VSD estimates of the mechanical power and rotational speed are used in the results. The laboratory test setup is presented in Fig. 3.

The laboratory blower is located in a system in which there is an adequate straight duct before and after the blower (Fig. 3.), meaning that the characteristic curves should hold true within the ISO 13348 accuracy, and the conditions for pressure measurement are sufficient. Naturally, in industrial applications such ideal conditions may not exist. Consequently, there may be a difference between the performance curve given by the manufacturer and the actual operation in the system, for example, because the flow impact angle to the blade is changed because of the piping bends. In addition, the wear in the blower may reduce the accuracy of the blower model.

Blower characteristic curves

To verify the blower characteristic curves in the laboratory setup, the output of the blower was measured at 1,500 rpm and compared with the QH and QP curves given by the blower manufacturer. The blower characteristics and the measured values can be seen in Fig. 4. Based on these results, both the measured QH and QP curves have some difference compared with the manufacturer's curve. Although a similar difference can sometimes be expected also in real-life system conditions, it causes error when using the presented estimation methods. For instance, based on the shape of the performance curves, the basic QH method would give two different flow rate estimates for each measured pressure value within a range of about 0–2 m^3/s when operating at 1,500 rpm as seen from the manufacturer's curve. Correspondingly, the QP method would give two different flow value results for each measured power value when operating in a range above 3 m^3/s flow rates. Thus, the flow rate range where the basic QP or QH methods are usable is limited.

The decision on which estimation method is used as the QH/QP method flow rate estimate is based on the estimation uncertainty calculated by (4). The power estimation uncertainty is assumed to be 0.2 kW, and the pressure measurement uncertainty, 50 Pa. The estimation uncertainties in the QH and QP methods are given for 1,500 rpm in Fig. 5. The flow rate region where both basic flow rate estimation methods are combined as a single estimate was selected to be the flow rate region where the uncertainties are within two times from each other ($U_{QH} \leq 2U_{QP}$ and $U_{QP} \leq 2U_{QH}$). It can be seen that in the flow region of 0–1.9 m^3/s, the uncertainty of the QH method is more than two times the uncertainty of the QP method Therefore, the QP method is used as the only estimation method in this flow region. At flow rates above 2.8 m^3/s, the estimation uncertainty of the QP method is more than two times the uncertainty of the QH method. Thus, at these flow rates, only the QH method is used. At flow rates

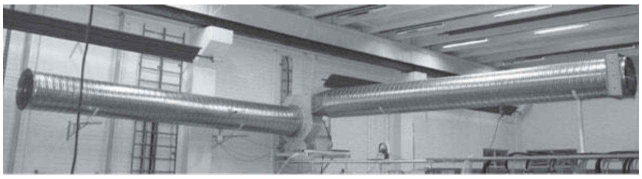

Fig. 3 Radial blower system used for the test measurements

between 1.9 and 2.8 m^3/s, both estimation methods are used, and the flow rate estimates are combined according to (5). It has to be noted that the regions are a function of rotational speed as can be seen from (4).

The uncertainty in the estimation methods is calculated solely on the manufacturer's characteristic curves and can thus be misleading. In this case, the measured *QH* curve has a clear difference (8–10 %) compared with the manufacturer's curve up to ~3.3 m^3/s where the curves intersect (Fig. 4). In the region of 2–3.3 m^3/s, the *QH* method seems to be more accurate, even though this may not be the case because of the erroneous *QH* curve. This case demonstrates that it is not always the optimal estimation method that is selected in the *QH*/*QP* method.

Despite the difference between the manufacturer and measured performance curves in the laboratory system, the manufacturer curves are selected for the estimation. Consequently, no startup measurements are needed in the commissioning of all of the examined model-based methods, except the shut-off power and shut-off head measurements of the Kernan and Kernan–*QH* methods. Thus, the effects of the difference between the manufacturer curves and the actual operation in the system, in combination with the curve shape, can be examined in this laboratory system. If the measured curves are used as a model in the estimation methods instead of the manufacturer curves, the flow rate estimation accuracy of each method can be expected to increase significantly.

Test points and applied estimation methods

The estimation methods were tested with seven different rotational speeds and five different system curve settings. In this paper, the operation of the estimation methods is studied in nine operating points in more detail; the points are shown in Fig. 4. These operating points result in 50, 95, and 130 % relative flow rates when operating at 1,500 rpm, and the studied rotational speeds are 900, 1,500, and 1,800 rpm. In the "Results" section, the measured values given by the presented laboratory equipment are referred to as "true values". The flow rate estimation methods examined are the following:

- Basic *QH*-curve-based method (Liu 2002)
- Hybrid method (Ahonen et al. 2012)
- Kernan method (Kernan et al. 2011)

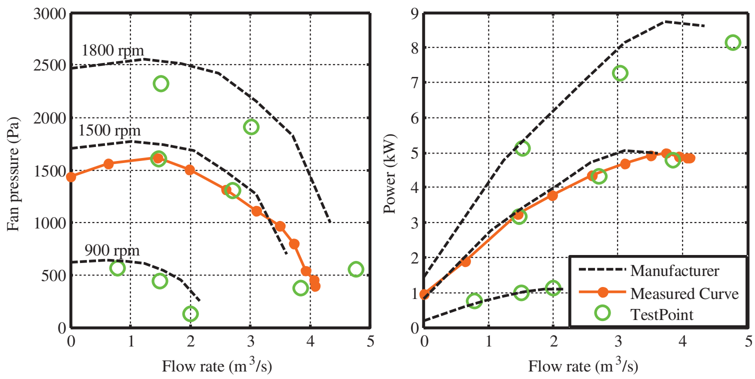

Fig. 4 Measured blower characteristics and the characteristics given by the manufacturer. The TestPoints indicated by a *circle* are the actual measured values used in the "Results" section to compare the different flow rate estimation methods. The difference in the test point compared with the measured curve results from the measurement equipment repeatability and the effect of ambient conditions

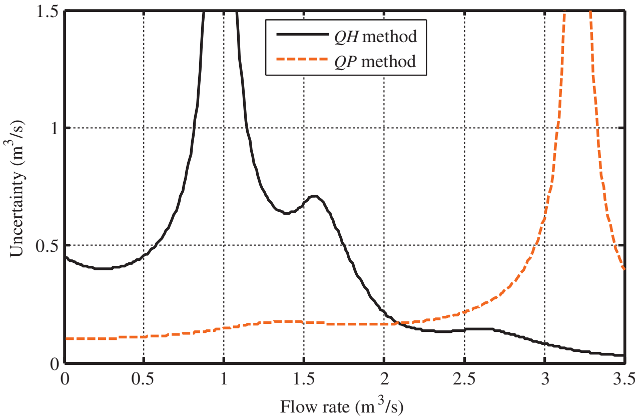

Fig. 5 Resulting uncertainty in the estimation methods with the given power estimate uncertainty of 0.2 kW and the pressure measurement uncertainty of 50 Pa calculated from the manufacturer's characteristic curves by (4) and for the nominal rotational speed of the blower

- Basic *QP*-curve-based method (Wang and Liu 2005)
- Kernan–*QH* method (Kernan et al. 2011)
- Bortoni method (da Costa Bortoni et al. 2008)
- *QH*/*QP* method
- Hybrid *QH*/*QP* method

Results

In Fig. 6, the flow rate estimates of the various methods are presented for a system curve that produces a 50 % flow rate at the nominal rotational speed. It can be seen from Figs. 4 and 5 that the *QH* method is unusable at this flow rate region as it has a much larger estimation error than the *QP* method. The *QH* method is significantly more accurate at all of the three rotational speeds than the *QP* method.

As can be seen in Fig. 6, the Kernan method gives more accurate estimates than the *QP* method when the blower is rotated at 900 rpm, the reason being that the error in the affinity laws is corrected with the shut-off power measurement. When operated at 900 rpm, the hybrid method is able to compensate the rotational speed better than the normal *QP* method, and it also gives about the same accuracy in the 1,500- and 1,800-rpm rotational speeds as the normal *QP* method.

The Bortoni method is able to estimate the fan flow rate point accurately at this low flow region. This is a result of the accurate model used in the estimation method, which is formed on the basis of shut-off and nominal conditions.

The *QH*/*QP* method selects the *QP* method as the sole estimation method to base its estimates on, because it is has a much lower uncertainty in this flow rate region. Thus, the *QH*/*QP* method gives the same results as the *QP* method. The hybrid *QH*/*QP* method gives slightly different results than the hybrid method; this is because the actual pressure measurement is used in the definition of the system curve and not the estimated value for pressure.

The Kernan–*QH* estimation method significantly improves the basic *QH* estimation method. The low estimate of the flow rate in this system curve setting is a result of the fact that the shapes of the actual and given characteristic curves differ from each other. Thus, even though the shut-off pressure is corrected, the *QH* curve is not corrected in the normal operating region of the fan.

The accuracy of the flow rate estimates of the model-based methods is clearly different when the fan is operated in a system where there is 130 % flow at the nominal rotational speed, as can be seen in Fig. 7. Compared with the *QP* method, the *QH* method gives accurate flow rate estimates at the nominal rotational speed and above it. The *QP* method is the more accurate one at 900 rpm, because in this case, the two erroneous models of the affinity laws and the *QP* curve seem to compensate for each other's errors. Moreover, the hybrid method and the Kernan method also give significantly erroneous results because they use the *QP* curve, and the shape of the *QP* curve suggests infeasible accuracy for estimation in this flow region, as mentioned in the previous chapter.

Correspondingly, the *QH*/*QP* uses only the *QH* method in this high flow rate region, because the estimation

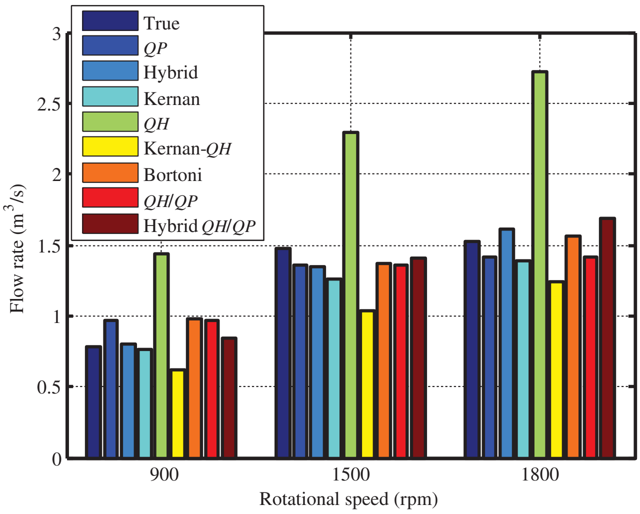

Fig. 6 Flow rate estimation results with a system curve that results in approximately a 50 % flow rate at 1,500 rpm

uncertainty is much lower than the estimation uncertainty of the *QP* method. Hence, the flow rate estimates of the *QH*/*QP* method are equal to the *QH* method. As it can be seen in Fig. 7, the hybrid *QH*/*QP* method does not significantly improve the estimation accuracy compared with the basic *QH*/*QP* method. All in all, both the *QH*/*QP* and hybrid *QH*/*QP* methods can give considerably more accurate results than the *QP* method and its modifications. The Kernan–*QH* method is less accurate in this flow rate region than the basic *QH* method. This is because the manufacturer's curve suggests already too low fan output pressures for this flow rate region, and this is corrected to be even lower with the shut-off pressure measurements. However, the Kernan–*QH* method is more accurate than the *QP* methods in this flow region.

When operating at over the nominal flow rates, the Bortoni method estimates significantly higher flow rates than the actual flow rate. This is because the actual fan pressure drops significantly after the nominal operating

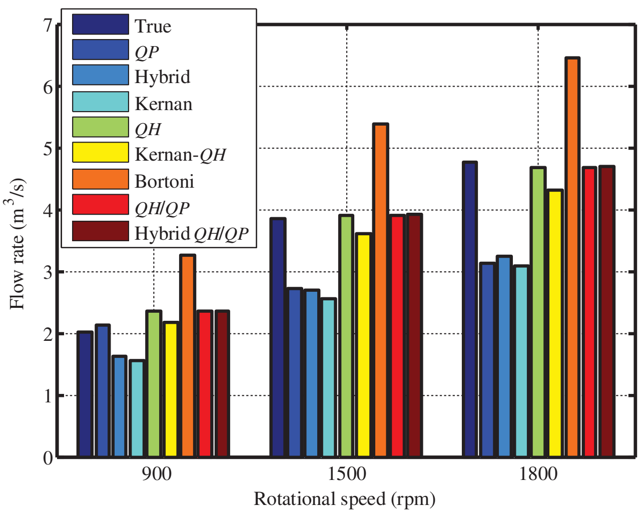

Fig. 7 Flow rate estimation results with a system curve that results in a 130 % flow at 1,500 rpm

point, and the model in the Bortoni method does not take that into account.

The estimation results with a system curve that produces approximately the nominal flow at the fan nominal rotational speed are presented in Fig. 8. In this flow rate region, both of the *QH* and *QP* methods produce approximately the same uncertainty to the estimation results, as shown in Fig. 5. Therefore, the accuracy of these basic estimation methods is approximately the same in this flow region, as shown in Fig. 8. Thus, the *QH*/*QP* method combines these estimation methods with the weighting equation given in (5) as explained above. At 900 rpm, the uncertainty of the *QH* method is much less than the uncertainty of the *QP* method, and thus, only the *QH* method is used in the *QH*/*QP* estimation.

The results show that at 1,500- and 1,800-rpm rotational speeds, the combined *QH*/*QP* method estimates the flow rate closest to the measured values. The hybrid *QH*/*QP* estimation compensates the error produced by the rotational speed change at 900 rpm resulting in a flow rate closer to the measured value compared with the *QH*/*QP* method. The difference between the *QH*/*QP* and hybrid *QH*/*QP* methods at 1,500 and 1,800 rpm can be explained by the fact that the hybrid *QH*/*QP* method also includes the estimated system curve in the estimation.

The hybrid method and the Kernan method give approximately the same estimation results as the *QP* method. The bias in the *QH* curve is compensated too much in the Kernan–*QH* method. Hence, the Kernan–*QH* method gives too low estimates of the flow rate, and the accuracy is approximately the same as in the *QP* method in this flow region.

The nominal flow rate is estimated accurately with the Bortoni method. This is because the rated nominal point does correspond to the measured nominal point of the fan.

Analysis of the flow rate estimation error in the studied methods

The average error and the standard deviation of the error in the whole test series with the six different system curves ranging from 50 to 130 % of the fan nominal flow rate and nine rotational speeds ranging from 600 to 1800 rpm are presented in Fig. 9. Thus, 35 different operating points are examined. The average error of the *i*th measurement is calculated by

$$Q_{\mathrm{Error},i} = \frac{\left|Q_{\mathrm{Est},i} - Q_{\mathrm{Measured},i}\right|}{Q_{\mathrm{Measured},i}} 100\ \% \qquad (6)$$

It can be seen from Fig. 9 that the mean error of the *QH* method is the greatest, because in the low flow region, the error is significant in each rotational speed. It can clearly be seen that the Kernan–*QH* method improves the estimation accuracy of the *QH* method across the fan operation, even though the total accuracy is still worse than with the methods

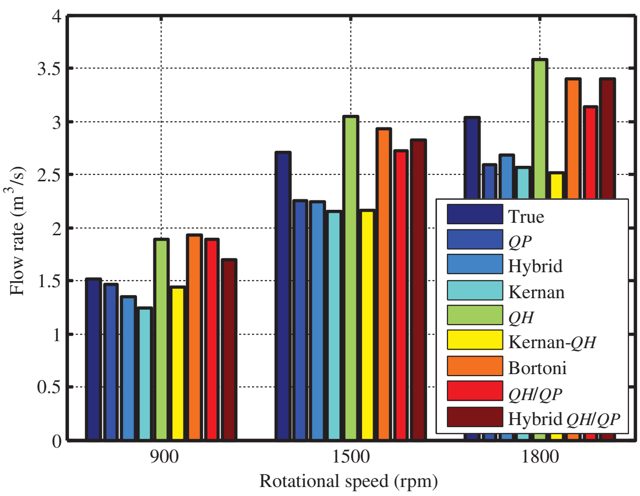

Fig. 8 Flow rate estimation results with a system that produces approximately the nominal flow at 1,500 rpm

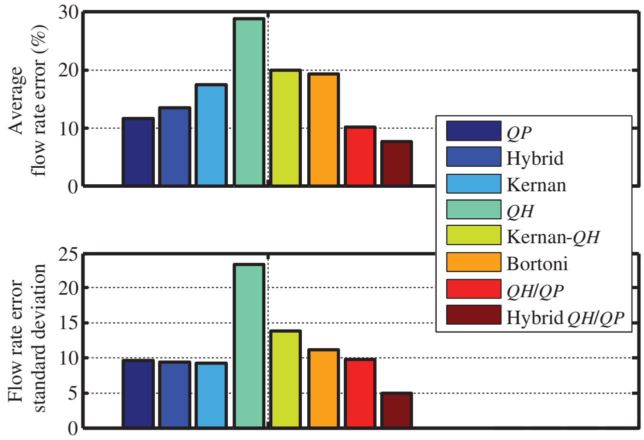

Fig. 9 Average error and standard deviation in the flow rate estimation errors of the different methods

that are based on the *QP* curve. Moreover, the high standard deviation of the error is also due to the high difference in the of the flow rate estimation accuracy across the whole fan operation.

In the *QP*-curve-based estimations, the error is lower than in the *QH*-curve-based methods, because the reduced rotational speed corrects the otherwise highly erroneous model at the high flow rates. This is also the reason why the improvements seem to have a negative effect on the basic *QP* method. At high flow rates and nominal rotational speed where the basic estimation has a high uncertainty with the hybrid method, the model is improved, and thus, the error is more significant also at lower rotational speeds than with the *QP* method. As it happens, the *QP* method compensates in the right direction with the erroneous models. In the Kernan method, the shut-off power also corrects the erroneous *QP* curve to be more erroneous and is thus more inaccurate than the *QP* method. The deviation of these methods is approximately the same, which shows that the accuracy of the methods does not change over the fan operating compared with each other, and also the deviation is half of the deviation of the *QH* method.

The Bortoni method is able to accurately estimate the flow rate when the fan is operated at flow rates that are lower than the fan nominal flow rate. This is because in this case, the fan model, based on rated shut-off and nominal operation point, compensates the error between the manufacturer *QH* curve and measured *QH* curve. The total pressure of the fan drops significantly when flow rate is above the nominal value, and the *QH* curve approximation is no longer sufficiently accurate in this region. This can be seen also in high average flow rate error and error deviation.

Moreover, the *QH*/*QP* and hybrid *QH*/*QP* methods have the best accuracy across the fan operation, indicating that the methods are improvements to the previous methods. The standard deviation of the error is also approximately 5 % units, which indicates that there is no significant change in the error over the fan operation. However, according to the laboratory measurements, the hybrid *QH*/*QP* method enhances the estimation method most.

Conclusion

Pump and fan systems are responsible for a quarter of the European industrial electricity consumption. As the studies have shown, there is a significant energy-saving potential in pumping applications. To identify the inefficient pumping processes, there is a clear need for easily implementable methods that can be used to estimate the pump or fan operating point by a frequency converter and to give information that can be used in the energy-efficient control of fluid handling systems. Even though there already are model-based methods suitable for frequency-converter-driven pump and fan operating point estimation, these models have limitations and error sources that can produce erroneous results and render the methods unusable. Consequently, these methods may even reduce the system efficiency if used for control purposes.

In this paper, a method that combines two existing model-based pump operating point estimation methods is presented aiming to improve the usability and accuracy of the previously introduced methods. The method automatically selects the more accurate one from the *QP* or *QH* methods to be used for the operating point estimation, hence reducing the risk of erroneous estimation. In addition, a method that combines the *QH*/*QP* method and the previously introduced hybrid method is presented. These methods are compared with the previously introduced methods.

The accuracy of the previously introduced and suggested model-based flow rate estimation methods was tested in a laboratory system consisting of a radial blower, a motor, a frequency converter, and piping. Even though the test setup could be considered a nearly ideal system, there was a clear difference between the manufacturer and measured performance curves. This difference has a clear effect on the model-based estimation accuracy, but similar drawbacks for accurate estimation can be expected also in actual systems.

Based on the measurement results, the hybrid and hybrid *QH*/*QP* methods are able to compensate for the uncertainties produced by the rotational speed change at lower rotational speeds, but may overcompensate at higher than nominal rotational speeds. In addition, if one characteristic curve is known to be more accurate, for example based on previous measurements, it should be selected as the primary basis of the fan model. This can be seen for example in the Kernan–*QH* method, which could achieve sufficient accuracy in regions where the basic methods are extremely erroneous. The measurement results indicate that the suggested *QH*/*QP* method is able to select the most suitable characteristic curve for the flow rate estimation and, thus, provide accurate flow rate estimates. In addition, when both of the methods are equally uncertain, the method can combine the estimates to include both estimation methods to give a more reliable and accurate estimate of the flow rate. Combining the *QH*/*QP* method with the hybrid method enhances the performance at low rotational speeds and is thus the most reliable and accurate operating point estimation method.

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ORIGINAL ARTICLE

Energy-efficient control strategy for variable speed-driven parallel pumping systems

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Abstract The article aims to find a solution for the energy efficiency improvements in variable speed-controlled parallel pumping systems with lesser initial data and without additional flow metering and start-up measurements. This paper introduces a new control strategy for variable speed-controlled parallel pumps based on flow rate estimation and pump operation analysis utilizing variable speed drives. The energy-saving potential of the new control strategy is studied with simulations and laboratory measurements. The energy consumption of the parallel pumps using the new control strategy is compared with the traditional rotational speed control strategy of parallel pumps. The benefit of the new control strategy is the opportunity to operate variable speed-controlled parallel pumps in a region which suggests improved energy efficiency and lower risk of mechanical failure of the controlled pumps compared with traditional control. The article concludes by discussing the implications of the findings for different applications and varying system conditions.

Keywords Variable speed drives · Pumps · Energy efficiency · Process control · Fluid flow control

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Introduction

Pumps are widely used in industrial and service sector applications. They consume approximately 10–40 % of the electricity in these sectors (de Almeida et al. 2003). Pumping systems are found to have a significant potential for energy efficiency improvements (Binder 2008; Kaya et al. 2008; Ferreira et al. 2011; Pemberton and Bachmann 2010; Zhang et al. 2012). The pressure for energy efficiency improvements has led to an increasing number of variable speed drives (VSDs) in pumping applications, since in many instances, variable speed pumping has been shown to be an effective way to reduce the total pumping costs, especially in systems that require a wide range of flow (Bernier and Bourret 1999; Pemberton 2003; Europump and Hydraulic Institute 2004).

Pumping systems with a widely varying flow rate demand are often implemented using parallel-connected pumps (Hooper 1999; Volk 2005). There are several control methods available for operating the parallel-connected pumps. In the simplest case, parallel-connected pumps are operated with an on–off control method, where additional parallel pumps are started and stopped according to the desired flow rate. In the systems where more accurate flow regulation is needed, the adjustment can be carried out by applying throttling or rotational speed control for a single pump, while other pumps are controlled with the on–off method. The benefits using rotational speed control instead of the throttling control method are widely studied (Rossmann and Ellis 1998; Carlson 2000; Europump and Hydraulic Institute 2004; Hovstadius et al. 2005) and are therefore excluded from this study.

Energy optimization of parallel-connected, rotational speed-controlled pumps has been studied to some extent (Izquierdo et al. 2008; Bortoni et al. 2008; Yang and Borsting 2010), and the results have shown that there is a major energy-saving potential in the sector of parallel pumping. In Bortoni et al. (2008), the optimal rotational speed for parallel pumps is predicted, in order to gain energy savings, using a mathematical optimization-based tool suitable for programmable logic controllers. However, the suggested optimized control method requires adequate information from the system curve, including start-up field measurements using pressure sensors and flow meters. In many parallel pumping cases, sufficient data for energy optimization from continuously changing systems are available only to limited extent (Kini et al. 2008; Aranto et al. 2009). An alternative approach to obtain the required system information can be the model-based flow monitoring method in which the flow rate of each parallel pump is estimated without separate flow meters based on pressure metering or torque and rotational speed estimates of the VSD (Hammo and Viholainen 2006; Ahonen et al. 2012).

The aim of this study is to introduce a new control strategy for variable speed-controlled parallel-connected centrifugal pumps (later referred to as parallel pumps) in a system where pumps in individual piping parts feed a common outlet pipeline. The suggested control strategy can offer a justified base for the energy efficiency improvement in variable speed-controlled parallel pumping systems, even in such cases where the information on the pumping system is limited or changing. The suggested control strategy is based on the simple use of existing pumping monitoring solutions of a modern VSD and a known relation between the preferable operation area of a centrifugal pump and the energy efficiency of the pumping. By implementing the suggested control strategy in the control procedure, the flow rate of the parallel pumping system can be adjusted with improved energy efficiency compared with traditional rotational speed control. The control strategy can be applied for instance to parallel pumps located in water stations, wastewater pumping stations, and industrial plants.

The relation between the pump operation point location and pump reliability and energy efficiency has been discussed in many occasions (ANSI/HI 1997; Ahonen et al. 2011). In the suggested new control strategy, the preferable operating area (POA) of each pump represents only the selected operating area between the set markups in the pump performance curve. The markup points are selected based on pump efficiency data and parallel pumping system details.

The structure of the article is as follows. First, the basics of rotational speed-controlled parallel pumping systems are discussed and the idea of the new parallel pump rotational speed control strategy is introduced. Next, the operation of the proposed strategy is verified by simulations using two parallel pumps in a system. After this, the performance of the proposed control strategy is demonstrated with laboratory measurements.

Control of parallel-connected pumps

The use of centrifugal pumps in parallel allows the production of a wider range of flow rates than it would be possible with a single pump. In other words, the parallel connection of centrifugal pumps increases the flow rate capacity of a pumping system (Hooper 1999; White 2003; Volk 2005). A simplified example of a pumping system consisting of two parallel pumps and two water reservoirs combined by individual suction piping and common outlet piping section is illustrated in Fig. 1. An example illustrating the operation of the parallel-connected pumps in a system is given in Fig. 2.

A parallel-connected pumping system can provide the sum flow rate Q_1+Q_2 of individual pumps 1 and 2 with a common amount of head H as shown in Fig. 2. In practice, the individual head value of each parallel pump connected to common outlet pipe can also vary according to system characteristics, especially if there are valves on individual piping parts or the pumps are

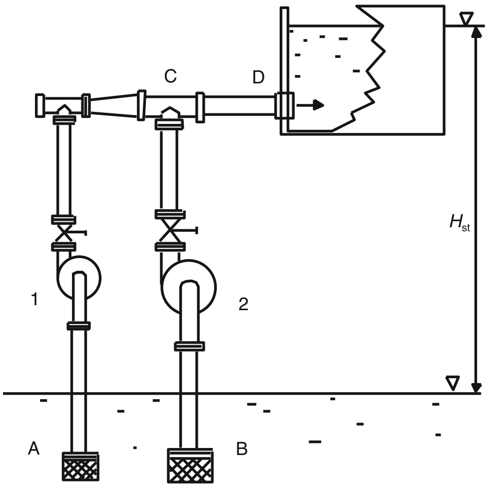

Fig. 1 Two parallel pumps feeding a common outlet pipeline. The parallel pumps (marked *1* and *2*) have their individual piping parts between points *A*–*C* and *B*–*C* feeding the common pipeline between points *C* and *D*

operated with different rotational speeds. In this simplified case, the operating point location of this parallel-connected pumping system (marked with C in Fig. 2) is in the intersection of the system and parallel operation curve which is the sum of individual pump characteristic curves. The individual operating point locations (A and B) of pumps 1 and 2 can also be determined with the flow rates Q_1 and Q_2, respectively (Volk 2005). In many instances, the pumps are

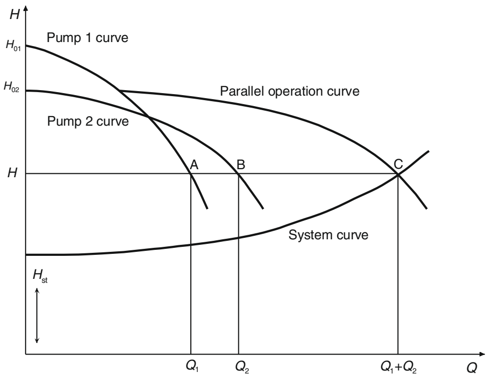

Fig. 2 Parallel operation of pumps 1 and 2 (points *A* and *B*) and the resulting operating point location *C* with the total flow rate Q_1+Q_2 (Bortoni et al. 2008)

selected according to system so that the operation point would be near pump's best efficiency point (BEP) during normal use. This is justified, since when operating considerably afar from BEP, the pumping efficiency can decrease rapidly and the pump service life may be affected by the flow recirculation, high flow cavitation, and shaft deflection (ANSI/HI 1997; Karassik and McGuire 1998; Ahonen 2011).

The output of the parallel-connected centrifugal pumps in a system can be adjusted, for example, with an on–off, throttle, or rotational speed control methods. The use of the on–off method is justified for applications having a tank or a reservoir and no need for accurate control of the flow rate. Correspondingly, the throttle control method can be used to regulate the flow rate produced by the pump, but because it can have a negative effect on the pumping efficiency, it is not always justified. In many pumping systems, the rotational speed control of pumps can allow the flow rate adjustment with a lower energy use compared with the throttling method. In some cases, the rotational speed control can be used in an on–off control scheme to fix the rotational speed lower than nominal, thus gaining more energy-efficient operation. The basic version of the rotational speed control for parallel-connected pumps, the *traditional rotational speed control* strategy, is based on the adjustment of the rotational speed of only a single pump at a time. This is illustrated in Fig. 3 in the case of two parallel pumps. Before the additional pump is started, the rotational speed n of the primary pump is increased to the nominal rotational speed n_{nom} (Shiels 1997; Karassik et al. 2001; Volk 2005; de Almeida et al. 2005; Jones 2006).

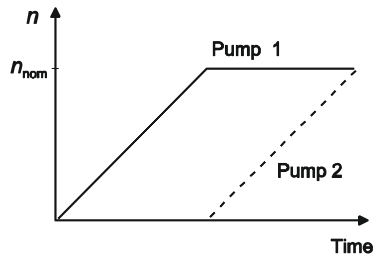

Fig. 3 Traditional rotational speed control of two parallel-connected pumps as a function of time. The flow need is increasing when moving to the right on the time axis. When the primary pump (pump 1) reaches its nominal speed, more flow is striven by starting the secondary parallel pump (pump 2)

A higher energy efficiency compared with the traditional rotational speed control can be achieved if both parallel pumps are rotational speed-controlled (Viholainen et al. 2009a). In addition to the saved energy, using rotational speed control in multiple parallel pumps provides an opportunity to avoid situations where parallel pumps are operating in shut-off or in a region where the risk of reduced pump service life is higher (ANSI/HI 1997; Karassik and McGuire 1998; Ahonen 2011). An example of a preferable option compared with the traditional rotational speed control can be demonstrated if the operation of two identical raw water pumps (Ahlström P-X80X-1) is observed in a system of a 15-m static head. In this example, the system curve is chosen so that both pumps will have a high pumping efficiency when they are operated at the nominal rotational speed (Jones 2006). An example case of adjusting the output of the pumps to a lower flow rate using the traditional rotational speed control or delivering the same flow rate by reducing the rotational speeds of both pumps is illustrated in Fig. 4.

Figure 4a plots the *QH* curves of the parallel-connected pumps, the system curve, and the combined parallel pump curve. In Fig. 4a, the first pump is operating at the nominal 740-rpm rotational speed and the second pump at a 540-rpm rotational speed. Figure 4b shows the corresponding *QH* curves when both pumps are operating at a reduced rotational speed (605 rpm) delivering the same total flow Q_1+Q_2. In the traditional rotational speed control, it is quite common that parallel-connected pumps are not operating near their BEP curve (see points B and C) which in this figure represents the rough estimate of justified operating region at different pump rotational speeds, rather than just the location of the best pump efficiency (ANSI/HI 1997; Barringer 2003; Martins and Lima 2010). If the same flow rate is delivered using a decreased rotational speed for both pumps, the operation points of the pumps (point D) are closer to the BEP curve which in this case suggests a higher energy efficiency and mechanical reliability (Fig. 4b). This kind of solution is not possible in parallel pumping systems, unless all pumps are rotational speed controlled.

The effectiveness of a single pump is often observed with the pump efficiency

$$\eta_p = \frac{Q \cdot \rho \cdot g \cdot H}{P_p} \qquad (1)$$

where Q refers to the flow rate of the pump (in cubic meter per second), ρ is the fluid density (in kilogram per cubic meter), g the gravitational constant (in meter per square second), H the head of the pump (in meter), and P_p the power input of the pump (in watt). If the total input power including the motor's and drive's losses is observed in Eq. (1), the system efficiency (Yang and Borsting 2010) is

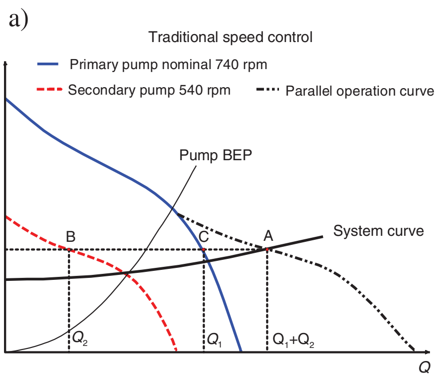

Fig. 4 Speed-controlled parallel pumping using the traditional rotational speed control (**a**) and when both pumps are running at a reduced speed (**b**). Adjusting the flow rate by running the primary pump at the nominal speed (740 rpm) and decreasing the secondary pump's speed to 540 rpm delivers the desired flow rate (Q_1+Q_2), but the operation points are located far from the best efficiency point (*BEP*). Adjusting the flow rate by reducing the speed of both pumps to 605 rpm by using VSDs results in the same flow rate, and the operation points can be located in a region of better energy efficiency and mechanical reliability

$$\eta_s = \frac{Q \cdot \rho \cdot g \cdot H}{P_{in}} \tag{2}$$

where P_{in} represents the total input power to the pump drive (in watt). The energy efficiency of parallel pumping can be evaluated using specific energy which describes the energy used per pumped volume (Europump and Hydraulic Institute 2004). The specific energy is given by:

$$E_s = \frac{P_{in} \cdot t}{V} = \frac{P_{in}}{Q} \tag{3}$$

where E_s is the specific energy (in kilowatt-hour per cubic meter), P_{in} the pump drive power (in kilowatt), t time (in hour), V the pumped volume (in cubic meter), and Q the flow rate (in cubic meter per hour). Since the delivered flow rate is often the control variable in parallel pumping, the specific energy can be seen as a justified metrics to evaluate the energy efficiency of parallel pumping system instead of the pumping efficiency or the system efficiency.

Control strategy based on preferable operation area and pump operation point estimation

In this study, the improved energy efficiency of the variable speed-controlled parallel pumps compared with the traditional control is striven by introducing a new control strategy for the parallel pump control. The introduced control strategy of parallel-connected pumps was designed based on the following requirements:

- The suggested control strategy should be able to work with as little amount of initial information as possible, even without additional sensors in the pumping system.
- Compared with the existing and known flow adjustment methods, the suggested control strategy should be able to reduce the energy consumption of the pumping system.
- The suggested control strategy should also prevent the inefficient or harmful operation with a higher risk of reduced pump service life of an individual pump when a certain flow rate is produced with parallel-connected pumps.

The possible harmful and inefficient operation in parallel pumping can be avoided if the POA of each parallel pump is taken into account in the control strategy. Thus, these risks can be controlled by preventing the pumps from operating outside the selected region during the rotational speed control if possible.

In the case of two similar parallel pumps, the rotational speed of the primary pump is not necessarily increased to its nominal value, but instead, at the determined point, the rotational speed of the primary pump is kept constant while the rotational speed of the second pump is increased in order to produce flow. When the secondary pump has started to produce flow, the rotational speed of the pumps can be balanced to the same pump head value, and in the case of more flow demand, both pumps can be controlled closer to their nominal rotational speeds. Balancing the rotational speed of the parallel pumps has been suggested already by Hammond (1984), although not from the perspective of energy savings but to even out the pump working hours and wearing. Especially, if parallel pumps are dimensioned according to the flow rate at the nominal rotational speed, the balancing procedure should enable a lower specific energy consumption compared to the traditional rotational speed control of parallel pumps, and both pumps can be kept closer to each pump's best efficiency area during the control.

Figure 5 plots the POA between the efficiency markups at different pump rotational speeds according to the affinity laws. The area outside the flow limits in the QH axis can be described as high H and high Q range areas. The flow rate limits to start the balancing of the rotational speeds of parallel pumps can be set using only the pump characteristics. To select the flow rate limits, the pump efficiency can be seen as a justified variable for limiting values, since centrifugal pump performance curves usually contain efficiency data (Sulzer 1989; Karassik et al. 2001). As illustrated in Fig. 5, balancing the rotational speeds shifts the operation point of pump 1 to a higher efficiency region at the same time when pump 2 is being run towards the same head level. Consequently, both pumps are running in a region that can be considered beneficial from the perspectives of energy efficiency and reliability. Similar control steps can also be applied to systems with higher number of parallel pumps. In this case, the ongoing pumps are seen as a unit representing the primary pump (pump 1), while the next pump in turn represents the secondary pump (pump 2).

Observing the output of parallel pumps during operation is usually limited by the lack of metering in the

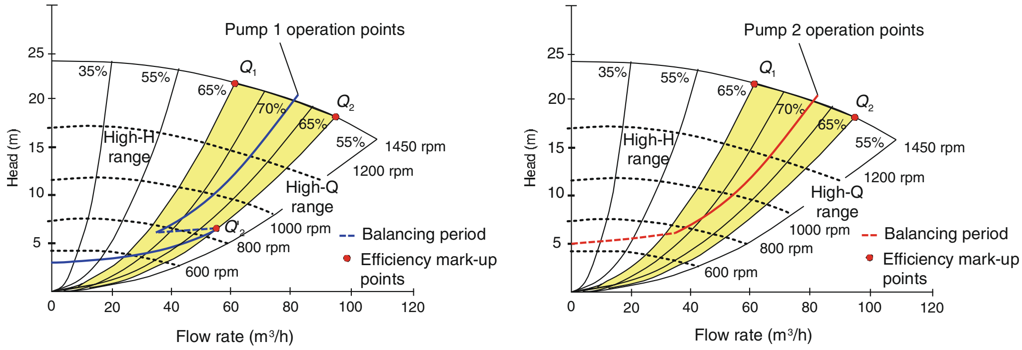

Fig. 5 Operation points of pump 1 (on the *left*) and pump 2 (on the *right*) in the suggested control strategy when the flow rate of the system is increased to nominal. The speed balancing of the parallel pumps starts when the operation point reaches the set flow limit (Q_2) at Q'_2. The area between the limit values Q_1 and Q_2 in the QH axis according to the affinity rules is the set preferable operating area (POA). The area outside the flow limits can be described as high H range and high Q range areas

pumping systems. However, information from parallel pump operation can be gathered by utilizing the pump operation point estimation available in a modern VSD. The operating point of individual variable speed-driven pumps can be monitored using the pump characteristic curves and the measured head or estimated power of the pump (Hammo and Viholainen 2006; Ahonen et al. 2010). In these estimation methods, the pump characteristic curves are shifted to the used rotational speed with the affinity equations

$$Q = \left(\frac{n}{n_0}\right) Q_0 \quad (4)$$

$$H = \left(\frac{n}{n_0}\right)^2 H_0 \quad (5)$$

$$P = \left(\frac{n}{n_0}\right)^3 P_0 \quad (6)$$

where H is the pump head (in meter), P is the pump power (in watt), n is the pump rotational speed (in revolutions per minute), and the subscript 0 denotes the initial values given by the characteristic curves. In the flow rate estimation, the flow rate corresponding to the measured head is found on the shifted pump QH curve in the case of the QH curve-based method. Correspondingly, QP curve-based method determines the pump flow rate by using the estimated pump shaft power and the shifted QP curve.

Adequate flow metering of individual pumps in the suggested control strategy allows the adjusting of the pumped volume according to the process changes, and each pump can be monitored to operate in the selected POA. Therefore, a separate flow meter installation or start-up field measurements are unnecessary. The estimation of the operating point of the pump with VSDs in suggested control strategy can be done either using only the pressure sensors for inlet and outlet pressure measurements or utilizing the sensorless option based on the motor power estimate. VSDs not containing such flow estimation features are excluded from this study, since the requirement of not to use additional flow meters would not be met.

Implementing the control strategy

The objective of the introduced control strategy is to prevent the variable speed-controlled parallel pumps from operating in regions with poor energy efficiency and increased risk of mechanical failure. Since the suggested control strategy is based on pump operation point estimation and POA, which in this case can be limited by the pump user based on pump efficiency data and process conditions, implementing the control strategy does not require any mathematical optimization tools. Instead, the control can be set with a simple feedback control based on pump output (Fig. 6). As illustrated in Fig 6, the information on each pump's operation point and individual rotational speed data are gathered from the VSD supporting it. The VSD's flow monitoring can supply the head, flow rate, and power values of each pump based on the input power

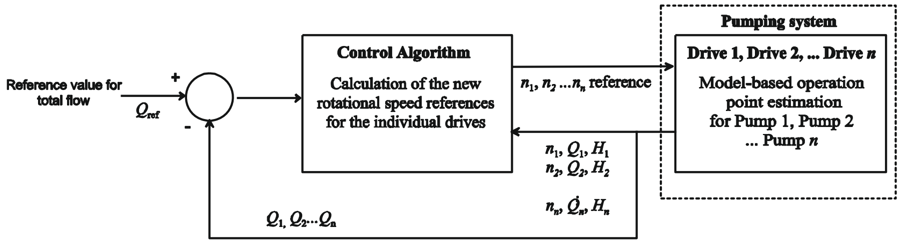

Fig. 6 Implementing the control strategy to a parallel pumping system. The pump operation point estimation gives the pump output values to the control algorithm. The control algorithm calculates the reference speed to each pump drive based on the monitoring data and the set preferable operating area of each pump

reference or measured head. Using the reference value for the total flow rate and selected preferable operating area for each parallel-connected pump in the current system, the control algorithm returns the reference speed for each pump drive.

An example of the control algorithm to provide the suggested control strategy in parallel pumping systems was created. The block diagram of the prototype algorithm is illustrated in Fig. 7. The block diagram illustrates the control procedure in the case of increasing or decreasing the total flow of the parallel pumps according to required output marked as Q_{ref} in Fig. 7. In the block diagram, the system is started with just one pump regardless of the required flow rate. After starting, the output of the parallel pumps is adjusted to meet the process requirements based on Q_{ref}. In this case, the operation point of each pump (Q, H) is determined by utilizing the VSDs' *QH* curve-based flow estimation, although the head of the pump could also be determined with the *QP* curve-based method.

As mentioned, mathematical tool-based optimization of rotational speeds of the parallel pumps may require start-up measurements and detailed system data (Bortoni et al. 2008), which may have to be repeated or reevaluated if there are changes in the system conditions, e.g., in the amount of the static heads or the shape of system curve. The implementation of the suggested control strategy to a parallel pumping system does not require start-up measurements, additional flow meters, or data related to the piping system characteristics, although the system conditions should be considered when selecting the preferable area in the pump *QH* axis. Thus, changes in the pumping system do not result in the reediting of the control setup. Instead of the optimization of the speed of each parallel-connected pump, the energy efficiency and reliability are obtained by ensuring that the pumps are operated on the selected operating area, if possible. Including POA as a control factor in the parallel pump control strategy can also ensure that the pump user does not have to decide, whether the efficiency or reliability should have more value in varying conditions. Because of these qualities, the suggested control strategy can be justified in parallel pumping systems in which the requirements for a complete energy optimization are not met.

Simulations and measurements

Since the aim of the suggested control strategy is not to optimize the energy efficiency of the variable speed-controlled parallel pumps, but to ensure that they are operated in an area of justified efficiency and pump reliability, the comparison of such control with an optimization-based control schemes can be seen controversial. Instead, the benefits of the suggested control strategy are compared with the explained traditional rotational speed control strategy.

The comparison is made using a simulation tool for the pumping system observation. The simulated operation is verified by laboratory measurements in a parallel pump setup. Differences between the control methods are evaluated in terms of power consumption and specific energy use.

Laboratory setup

The laboratory contains two pump systems; both of them include a single-stage centrifugal pump and a VSD connected to a three-phase motor. The primary pump (pump 1) system consists of a Serlachius DC 80/255 centrifugal pump, a four-pole 15-kW Strömberg induction motor, and an ABB ACS 800 frequency

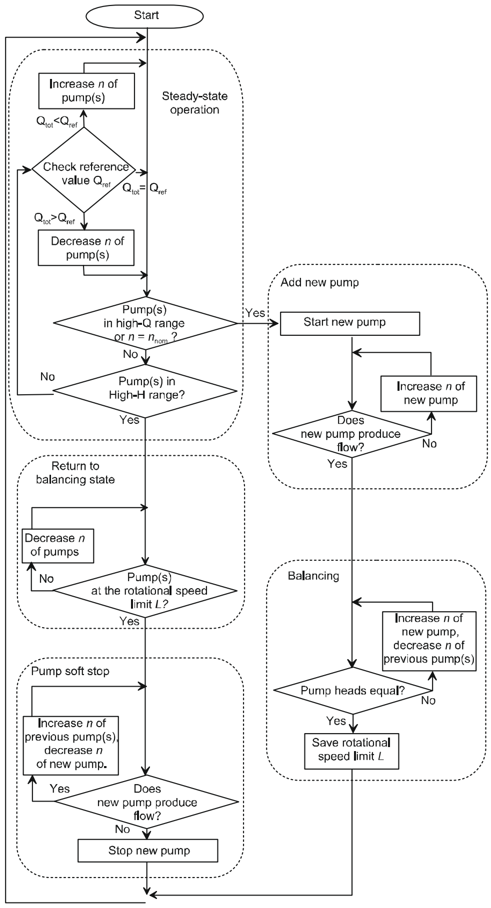

Fig. 7 Block diagram for the control algorithm which can provide the suggested control strategy for parallel pumps. The control step to increase or decrease pump speed should be decided based on the pump nominal speed. The point marked as *Start* is a point when the first parallel pump is started

converter. The secondary pump (pump 2) system consists of a Sulzer APP 22-80 centrifugal pump, an ABB 11-kW induction motor, and an ABB ACS 800 frequency converter. Both VSDs estimate the individual flow rates using pump head measurements. The total flow rate is also measured using a Venturi tube. The pumps were connected in parallel, and the basic layout of the measurement setup is presented in Fig. 8.

The control algorithm is implemented in a dSPACE DS1103 PPC controller board which was used as a separate platform for the control strategy in this prototype testing. The dSPACE board has analog voltage inputs and outputs; the inputs for the controller board are the rotational speeds, heads, and flow rates of the individual pumps, and the total flow rate from the VSDs. The outputs of the controller board are the rotational speed references for the individual pumps. The sample time for the control algorithm was 1 s. In the laboratory measurements, the flow rate is controlled based on the requirement for more flow, less flow, or no change in the flow rate.

The static head of the piping system was 2.5 m, and the system curve was set using valves located in individual piping branches so that both pumps would gain a reasonable efficiency when operating parallel at the nominal rotational speed. This illustrates a case where a parallel pumping system is dimensioned according to the highest flow rate. The operating values of the parallel pumps in the test setup system are shown in Table 1. Since the pump systems have separate piping parts causing individual friction head to each pump, the head levels are not equal in parallel use (Table 1).

Simulation sequences

The operation of the presented control methods is simulated for the laboratory pumping system with a Matlab Simulink model. The model is constructed to enable energy efficiency calculations of pumping systems and has been reported by Viholainen et al. (2009b). A similar simulation model has been utilized to characterize hydraulic systems also by Pannatier et al. (2010). In the simulation of this study, the performance, the combined power consumption, and the specific energy consumption of two parallel-connected pumps, having the same characteristics as the introduced pumps in the laboratory setup, are evaluated in a case where the total flow of the pumping system is increased using either the traditional rotational speed control strategy or the presented new control strategy. In the "Results" section, the operation based on the new control strategy is represented as the *alternative control*.

Results

Simulation results

The simulation was conducted from the flow rates 0 to 189 m^3/h. The rotational speeds of the individual

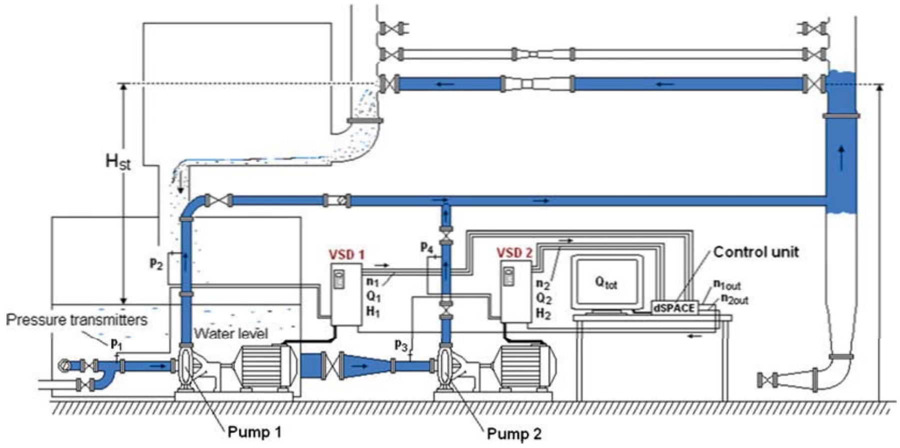

Fig. 8 Test setup used in the laboratory measurements. The pressure transmitters are installed to the inlet and outlet section of each pump, and the pressure signals are wired to the frequency converters to enable the flow calculation. The control board, a dSPACE system, is attached to both VSDs. The values from VSDs' pump monitoring application; speed (*n*), flow rate (*Q*), and head (*H*) signals are led from the VSDs to the dSPACE system. The determined speed commands (n_{out}) are transmitted to the VSDs from the dSPACE unit

pumps using both control methods during a simulation sequence (0–1,200 s) are given in Fig. 9.

It can be seen that in the traditional control, the rotational speed of the primary pump (pump 1) is increased to 1,450 rpm, after which the secondary pump (pump 2) is started and run towards the nominal rotational speed (Fig. 9). When using the alternative control, the secondary pump is started before the primary pump reaches the nominal rotational speed because the primary pump hits the set flow limit (point A in Fig. 10 a), as described in the previous section. This means a smaller flow rate difference at the secondary pump's starting point compared to the traditional control scheme. The simulated operation points of both parallel-connected pumps using either the traditional or alternative control are illustrated in Fig. 10. The figure also shows the chosen flow rate limits for the alternative control algorithm based on the pump data given by the pump manufacturers.

Figure 10 shows that even though traditionally controlled parallel pumps are operating in the same operation point as in the alternative control when both pumps have reached their nominal rotational speed, the alternative control enables the continuous operation between the set flow rate limits. Therefore, the operating points, especially in the case of pump 1 (~65–90 m^3/h) shown in Fig. 10a, are located in a better efficiency area compared with the traditional rotational speed control. Because of the balancing, the duty point of the secondary pump is located only temporarily in an unwanted region, and the actual

Table 1 Parallel pumping system in laboratory setup

Type	Pump 1 Serlachius DC 80/255	Pump 2 Sulzer APP 22–80
	BEP[a]	
Speed (rpm)	1,425	1,450
Flow rate (m^3/h)	76	90
Head (m)	17.4	15
Efficiency (%)	69	73
	Parallel operating point[b]	
Speed (rpm)	1,448	1,449
Flow rate (m^3/h)	91	83
Head (m)	16.3	17.6
Efficiency (%)[c]	68	70
Selected POA (% BEP flow)	70–130	70–130

[a] Operating values in a rated efficiency point according to the characteristics curves given by pump manufacturer

[b] Operating values (measured) of the parallel-connected pumps in a test setup system

[c] Based on pump characteristic curves

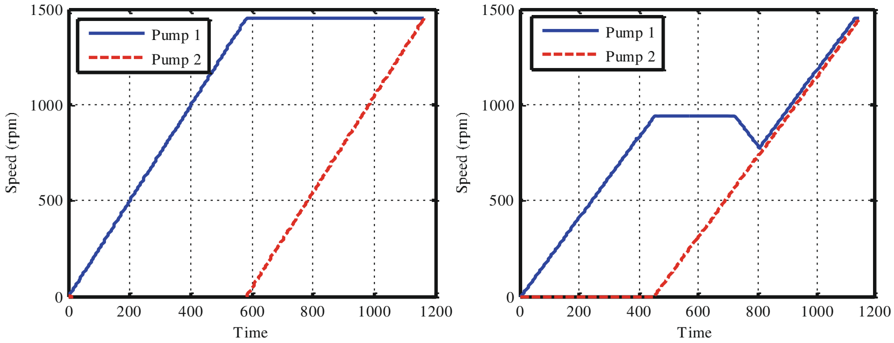

Fig. 9 Operation speeds of two parallel-connected pumps in a system in the case of the traditional speed control (on the *left*) and the alternative control (on the *right*). In both cases, the pumps were operated to deliver the total flow rate from 0 to 100 %. The *time axis* shows the direction of increasing flow demand

operation (~40–90 m^3/h) takes place between the set limits (Fig. 10b). During the balancing period, the primary pump is always delivering flow and head, and hence, the secondary pump (pump 2) can generate a flow rate only when it has exceeded the required head (~4 m). However, the required head for the secondary pump can be smaller than the primary pump's total head, since the friction head values for both pumps are not necessarily equal during the control.

The benefit of the alternative control can be seen best when observing the total pump power consumption and the specific energy consumption of both parallel pumps in the same simulation (Fig. 11). The results suggest that in this particular case, the alternative control enables much lower power consumption and specific energy consumption in the flow range of 70–175 m^3/h compared with the traditional control. Outside this range, the energy consumption was equal. However, the difference in the energy use seems to be more than 50 % at the highest point (110–120 m^3/h) between the alternative control and the traditional rotational speed control.

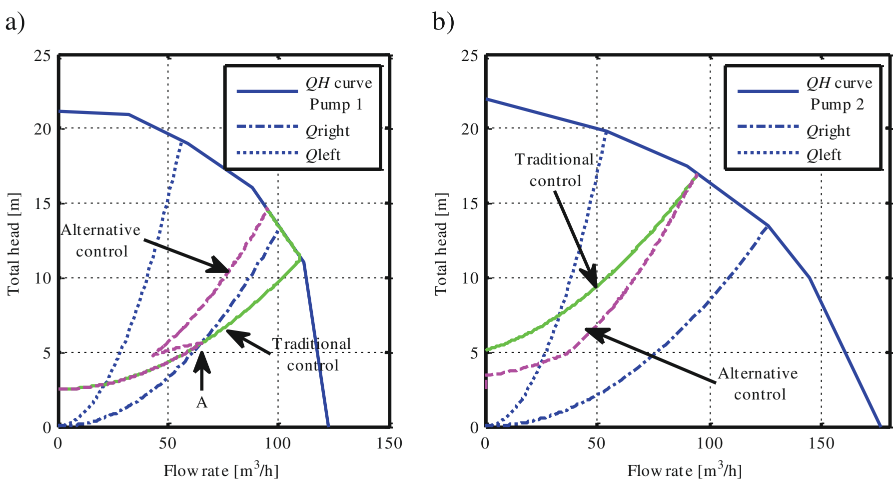

Fig. 10 Simulated operating points of pump 1 (**a**) and pump 2 (**b**) using either the traditional or the alternative control. With the alternative control, pump 2 is started when the pump 1 operating point reaches the set flow rate limit (Q_{right}) in point A. When pump 2 starts to deliver flow, the speeds of both pumps are balanced to have the same head

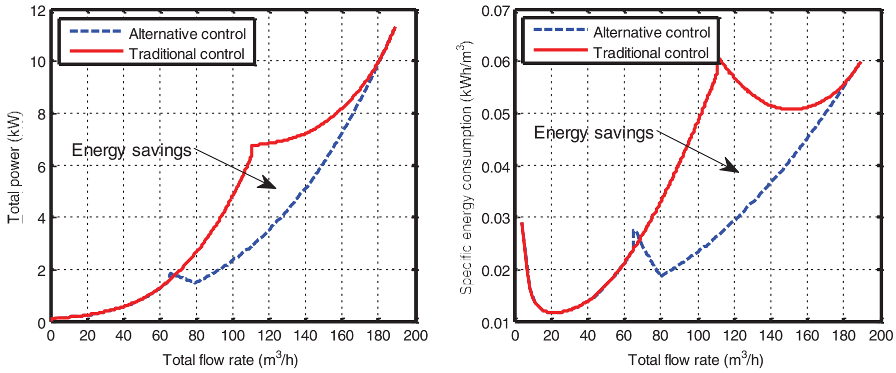

Fig. 11 Simulated total pump power (on the *left*) and specific energy consumption (on the *right*) of both pumps according to the total flow. The figure plots the simulated values in the cases of alternative control and traditional control. Energy savings using alternative control can be found when operating on a flow range, where the electric power use and specific energy consumption are lower compared with traditional control

Experimental results

The new parallel pump control strategy was tested in an actual pumping setup using measuring sequences where the flow rate was increased using the rotational speed control of parallel pumps. The total flow of both pumps varied from 0 to 175 m^3/h during the sequences. These values represent the minimum and maximum total flow rate values of the parallel pumps in the used system conditions. The measured operation points of each pump represent the average values gathered manually from the data control unit and the measuring equipment.

Figure 12 plots the test results of the sequences where the flow rate is increased from zero to maximum using either traditional control or alternative control. Figure 12a shows the measured operation points of the primary parallel pump when the total flow of the system is increased from 0 to 175 m^3/h. The balancing of pump 1 starts when the flow rate reaches the set markup line (Q_{right}) in alternative control. When traditional control is used, the speed of

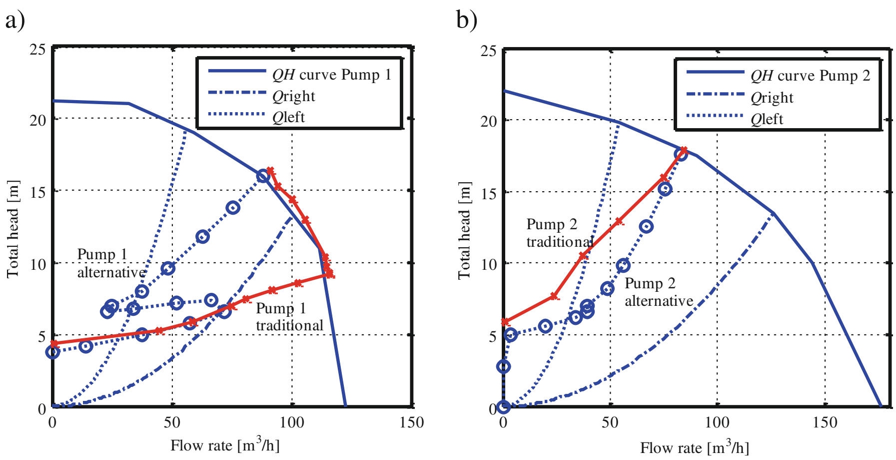

Fig. 12 Operation points of pump 1 and pump 2 during the alternative control and traditional control. The graph on the *left* shows the measured operation values for pump 1, and the graph on the *right* shows the pump 2 operation points

pump 1 is adjusted to nominal rotational speed before pump 2 is started. Figure 12b shows the pump 2 operation points. It can be seen from Fig. 12 that pump 1 and pump 2 are not operated on the same head in a point when pump 2 starts to deliver flow. This is because of the losses in elbows and valves in system, causing that dynamic head losses in individual piping parts are significantly different for pump 1 and pump 2 in this particular parallel operation point. However, a brief look at Fig. 12 shows that the alternative control is operating the parallel pumps as simulated. Since the laboratory equipment used in this study does not include the measurement of the pump shaft power, only the consumed total input power to each drive during parallel pumping was estimated using the input power reference of the VSDs. The results of the estimated total input power of both drives during the traditional and alternative control measurement sequences are illustrated in Fig. 13. The first look at Fig. 13 shows that in contrast to simulations, the measured total flow rate is not increasing during the balancing period (~75 m^3/h). Despite this, the advantage of the alternative control compared with the traditional control can be seen in the total power consumption and in the specific energy use.

Even though the estimated total input power rates during different control schemes are directly not comparable with the simulated pump shaft power values, the measured results seem to agree with the simulations. The results suggest that in this case, the alternative control seems to reduce the combined input power consumption and the specific energy use up to 20–25 % on the flow rates from 80 to 160 m^3/h. The benefits of using alternative control can also be seen in the higher system efficiency of parallel pump drives (Fig. 14).

Discussion

The simulation results showed (Fig. 11) that using the suggested new control strategy (*alternative control*) resulted in the improved energy efficiency in pumping compared with the traditional rotational speed control strategy, since the same flow rate could be delivered with a lower energy use (in the flow range of 70–175 m^3/h). In the illustrated examples, the alternative control enabled parallel pumps to operate in the set POA on the *QH* axis (Fig. 10) which in this case was defined simply as an area between the set efficiency limits based on the pump characteristics. The improved energy efficiency was verified by observing the simulated power consumption and the specific energy use of parallel pumps during the control procedure.

The benefits of the suggested control strategy were verified also by laboratory measurements with an actual parallel pump setup (Figs. 12, 13, and 14). In laboratory measurements, the amount of saved energy was 20–25 % at highest (in the flow range of 80–160 m^3/h). The measurement results showed (Figs. 12 and 14) that in that flow range (80–160 m^3/h) when using alternative control, the parallel pumps are located in an area which results in improved system efficiency compared to the situation, where the same total flow is delivered with traditional control.

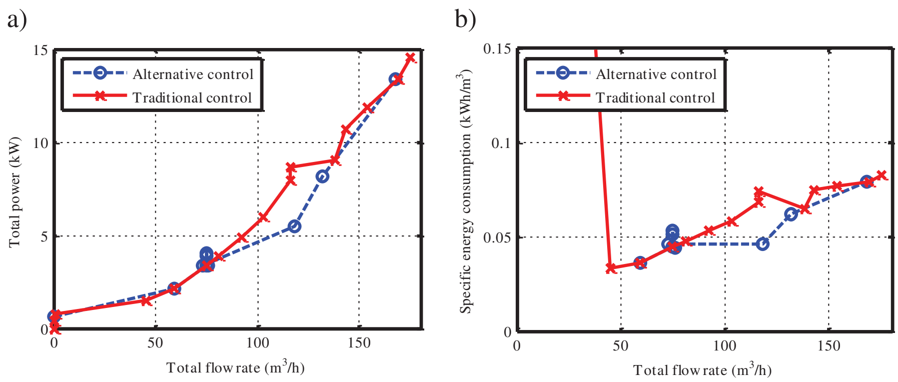

Fig. 13 Estimated total input power (on the *left*) and specific energy consumption (on the *right*) of parallel-connected pump drives in the alternative control and the traditional speed control

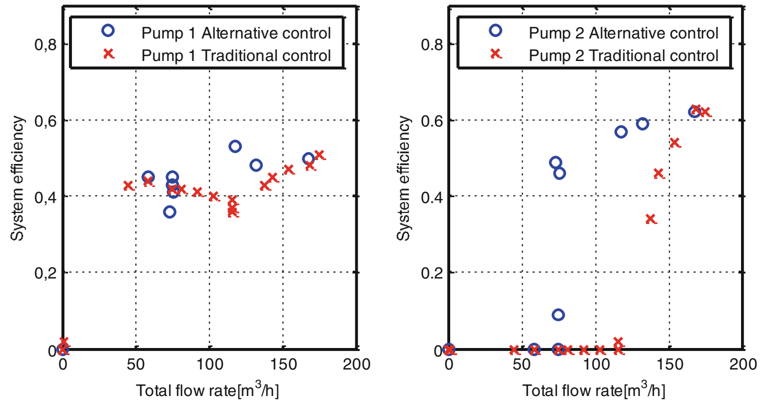

Fig. 14 Estimated system efficiency of parallel-connected pump drives in alternative control and traditional control. The system efficiency of pump 1 (on the *left*) and pump 2 (on the *right*) drives are shown according to total flow rate of both pumps, showing the variation of system efficiency during measuring sequences

Although the measurements indicated quite a similar operation as simulated, differences were found in the shut-off heads of the pumps and in the point where pump 1 reaches the set efficiency limit. The reasons for these differences can be the inaccuracies in the pressure metering, resulting in a further error in the flow metering of the VSDs. Based on the measurements, the oscillation of reference values (pressure, flow rate, pump rotational speed) can also disturb the control algorithm. Despite this, the collected data support the assumption that the presented control strategy could be implemented in VSD-controlled parallel pumping systems without separate flow metering devices or field measurements except the pressure sensors for the flow metering of the VSDs. The results also showed that the parallel pumps do not have to be identical, although significantly dissimilar pumps may need further considerations.

The prototype testing was performed using VSDs which are known to have applications to estimate the flow rate of the controlled pump. Without providing a similar monitoring of pump output with VSDs, the use of the introduced method would need additional metering of pump flow rate.

In the laboratory measurements, the control procedure including the prototype control algorithm based on suggested control strategy was tested using a separate controller board (Fig. 8), but the introduced method could also be implemented in VSD software.

It is clear that because the presented parallel pump control strategy is based on the VSD's pump system monitoring applications, its adequate operation depends on the monitoring accuracy. It is also known that the model-based pump monitoring cannot provide accurate flow metering in certain pump types and this may exclude the implementation of the introduced control strategy in some pumping systems.

A challenge for a justified control is to set the POA based on pump data only, since the efficiency data are not the only relevant factor when determining the preferable operation region of the pump. The pumping process can set limitations for instance to the minimum flow and pressure rate. Also, higher pump rotational speeds that can increase radial and axial forces in the pump and thereby affect the pump mechanical reliability should be taken into account for a more systematic approach. If the POA is chosen according to the pump efficiency data, but the parallel pump system is dimensioned so that the reasonable efficiency cannot be achieved when operating at the nominal rotational speed, it is likely that the operation points of parallel pumps can be located outside the defined flow limits. In addition, depending on the amount of static head and the shape of the system curve, the primary pump may operate mainly between the flow limits regardless of its rotational speed. In these situations, the alternative control operation greatly resembles the discussed traditional rotational speed control. Based on the results, the introduced new rotational speed control strategy for parallel pumps can enable higher energy efficiencies compared with the traditional rotational speed control, especially in parallel pumping systems with a varying flow need, relatively flat system curve, and when the pumping systems are dimensioned according to the highest flow rate.

Conclusions

In addition to the energy-efficient flow control in pumping, the rotational speed control using VSDs for each parallel-connected pump can open new opportunities for the advanced control of pumping processes. Utilizing VSDs to both system monitoring and system control provides opportunities not only to meet the varying requirements of the parallel pumping process, but also help in operating pumps with a lower energy consumption and reduced risk of mechanical failure.

The paper introduced a new control strategy for parallel pumps, which can improve the energy efficiency in variable speed-driven parallel pumping systems. The introduced control strategy is based on real-time pump operation point estimation and the selection of the preferable operating area of parallel pumps in a system, making it suitable for different applications and varying system conditions. The suggested control strategy can be implemented using the sensorless flow rate estimation of parallel pumps excluding separate flow meters and additional field measurements. The presented control was compared with a traditional rotational speed control strategy, and both the simulations and laboratory measurements showed that a lower energy consumption could be achieved using the introduced new control strategy. Further on, the discussed method showed to be able to run parallel pumps in the determined operating range, which suggests lower risks of reduced pump service life.

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