THESIS ON POWER ENGINEERING,
ELECTRICAL ENGINEERING, MINING ENGINEERING D76

High-Efficiency Predictive Control of Centrifugal Multi-Pump Stations with Variable-Speed Drives

ILJA BAKMAN

TUT PRESS

TALLINN UNIVERSITY OF TECHNOLOGY
Faculty of Power Engineering
Department of Electrical Engineering

Dissertation was accepted for the defence of the degree of Doctor of Philosophy on May 16, 2016

Supervisor: Professor, Ph. D. Valery Vodovozov, Department of Electrical Engineering, Tallinn University of Technology

Opponents: Professor, Dr.-Ing. Dr. h. c. mult. Johannes Steinbrunn, University of Applied Sciences Kempten, Germany

PhD Margus Koor, AS Tallinna Vesi, Estonia

Defence of the thesis: June 15, 2016, 15:00, room NRG-422, Tallinn University of Technology, Ehitajate tee 5, Tallinn, Estonia

Declaration:

Hereby I declare that this doctoral thesis, my original investigation and achievement, submitted for the doctoral degree at Tallinn University of Technology, has not been submitted for any academic degree.

Ilja Bakman…………………………

Copyright: Ilja Bakman, 2016
ISSN 1406-474X
ISBN 978-9949-23-961-0 (publication)
ISBN 978-9949-23-962-7 (PDF)

ENERGEETIKA. ELEKTROTEHNIKA. MÄENDUS D76

Pumbajaamade muudetava kiirusega tsentrifugaalpumpajamite kõrgefektiivne ennetav juhtimine

ILJA BAKMAN

TTÜ KIRJASTUS

Contents

ACKNOWLEDGEMENTS

This research project was conducted at Tallinn University of Technology in 2012 – 2016. The project has been funded by TUT and supported by Estonian Archimedes Foundation, including European Social Fund for Doctoral Studies and International Programme DoRa, Project DAR8130 "Doctoral School of Energy and Geotechnology II", and Estonian Science Foundation Grants ETF 8020 and ETF 7572.

I am grateful to my supervisor, Professor Valery Vodovozov who have provided me an opportunity to complete this research project. His valuable comments and advice concerning this thesis and my other research work are highly appreciated.

During my research, I have had a pleasure to work with my fellow staff and young researchers of the Department of Electrical Engineering, who have helped me to realise ideas from discussions into actual results and scientific papers. My special thanks go to Associate Professors Raivo Teemets, Elmo Pettai, Zoja Raud as well as PhD student Levon Gevorkov. I would also like to thank all other personnel at our university who have helped me during my research. The fruitful help of Mare-Anne Laane is gratefully appreciated also.

Furthermost, I thank my family and my friends for their support and encouragement throughout my studies.

Ilja Bakman

June 15, 2016

ABBREVIATIONS

AC – Alternating current
ANSI – American National Standards Institute
AOR – Allowable operation region
BEP – Best efficiency point
BER – Best efficiency region
DC – Direct current
DTC – Direct torque control
EU – European Union
HI – Hydraulic Institute
HMI – Human-machine interface
HQ – Head-flow
LPM – Litres per minute
NPSH – Net positive suction head
OPC – Open platform communications protocol
PC – Personal computer
PI – Proportional-integral
PID – Proportional-integral-differential
PLC – Programmable logic controller
POR – Preferred operation region
PQ – Power-flow
PWM – Pulse width modulation
TUT – Tallinn University of Technology
VSD – Variable-speed drive
WP – Working point
SSU – Saybolt universal second

SYMBOLS

A – cross-sectional area of the pipeline
C – friction factor
E – energy
F – frequency
g – acceleration due to gravity
H – head
i – index
I – current
j – index
k – friction loss factor
n – speed
P – power
p – pressure
Q – flow rate

R – resistance
s – motor slip
t – time
T – motor torque
U – voltage
V – pumped volume
v – liquid velocity in a pipeline
z – elevation of the measuring point
Z – number of pumps
η – efficiency
ρ – liquid density

INTRODUCTION

I.1. Motivation and Background

Centrifugal pumping stations are applied worldwide for liquid distribution in industrial, commercial, and residential applications, providing from 80 to 90% of full water treatment. Because of relatively low efficiency, an increasing energy market and growing energy costs new solutions for pumping technologies are searched. To save energy and provide such functionalities as broad pumping speed adjustment and robust technology management, the variable-speed drives play an important role especially in the conditions of unstable energy supply and significant pipeline disturbances. Many recent studies address energy management problems in the field of water distribution.

Still, many problems of energy-efficient management for multi-pump stations remain open. One of the primary reasons is that the specific power losses in the motor drives and feasibility of high control of the contemporary drives by the logical controllers are overlooked.

The current work aims to implement an energy-efficient head and pressure management approach based on such drive abilities as operation above the rated speed and robust performance with pre-defined tabularized data. In the last case, collections of experimentally measured efficiency data are used to predict the best number of working pumps and to derive their performance speed.

This thesis presents the results of the research conducted at the Department of Electrical Engineering of Tallinn University of Technology (TUT) along with the activities of the most reputable world's professional associations related to electrical engineering. Major results of the thesis contribute to both the international and Estonian economy.

Besides TUT, this research was supported by European Social Fund for Doctoral Studies and International Programme DoRa, Estonian Archimedes Foundation Project DAR 8130 "Doctoral School of Energy and Geotechnology II", and Estonian Science Foundation Grants ETF 8020 and SF0140016s11

I.2. Objectives and Tasks

The aim of this thesis is to establish a novel control approach targeted to increase the efficiency of energy use and distribution across the multiple centrifugal pumping applications. The main research tasks of the study are as follows:

1. To offer a concept of the mutual load- and speed-dependent smart multi-pump system architecture

2. To develop a methodology of efficiency monitoring and intellectual predictive control as a basis for the subsequent hardware and software organisation
3. To design an algorithm for expeditious selection of the number of working pumps and their speeds optimal from the viewpoint of the process efficiency
4. To create a simulation toolbox and methodology for exploring pumping and preparing the software modules for the real control environment
5. To build an original experimental setup and methodology for verification and tuning of the developed tools
6. Based on the sensorless approach, to improve such auxiliary processes and operations as pressure measurement, liquid density estimation, and slip compensation
7. To develop a library of the ready-to-use tools for enhancement of existing pumping applications

I.3. Major Results

The **scientific contributions** of the thesis considered as novel can be summarised as follows:

- A concept of the mutual load- and speed-dependent smart multi-pump system architecture
- A methodology of efficiency monitoring and intellectual predictive control as a basis for the subsequent hardware and software organisation
- An algorithm for expeditious selection of the number of working pumps and their speeds optimal from the viewpoint of the process efficiency

The **practical value** of the thesis is as follows:

- A simulation model and a methodology for exploring pumping and preparing the software modules for the real control environment
- An original experimental setup and a methodology for verification and tuning of the developed tools.

The **direct practical outcomes** of the thesis are as follows:

- A set of improved applications for auxiliary processes and operations: sensorless pressure measurement, sensorless liquid density estimation, and slip compensation

- A library of the ready-to-use tools for enhancement of existing pumping applications

Confirmation and dissemination of results are based on both the theoretical and practical investigations. The mathematical apparatus used in the thesis includes the procedural analysis, analytical geometry, matrix derivation, statistics, and search methods based on the relational databases.

The profitability of the work has been confirmed by analytical exploration and computer simulation along with verification in the real pumping processes.

The results of the thesis were presented at 14 international conferences. 10 of the author's papers are presented in collections indexed by IEEE Explorer, Scopus and WoS.

I.4. Thesis Outline

Chapter 1 outlines the state of the art of pumping technologies, particularly in the field of electrical drives for the centrifugal pumping. Following the study of the current problems and major tendencies in pumping, contemporary challenges and improvements, solutions in the pump control, and advanced control techniques are summarised. Classification and ranging of applications are given. Current problems and possible solutions are shown. Recent advances and perspectives of pumping drives are discussed. Control methods and control topologies are reviewed. Common control and protection architecture, model-based control and its perspectives for pumping, predictive control, and its perspectives for pumping are described in detail. At the end of the Chapter, prospective review of pumping efficiency estimates from different vendors, current problems in pumping efficiency increase, and possible solutions are presented.

Chapter 2 describes the resources used in this project for the pumping study. First, models and simulation techniques are analysed. Here, an analytical, computer and physical modelling of pumping is discussed along with a review of modern simulation and experimentation techniques. The mathematical model of the pumping process is explained in detail. It includes pump performance characteristics, especially the performance characteristics of pumps with variable-speed drives, power and efficiency curves of pumps with variable-speed drives, and the pipeline system characteristics. The developed Matlab model of pumping is presented and the simulation methodology proposed is given, including the model composition and specification along with the methodology of pressure management simulation. To verify the solutions, an original multi-pump experimental setup and auxiliary equipment have been prepared in the frame of this project. They involve the benefits of ABB DriveStudio toolkit, the PLC for experimentation and control, and auxiliary measurement and sensing equipment.

Chapter 3 covers the designed methodology of efficiency monitoring and predictive control. First, a comprehensive analysis of pumping within and outside the best efficiency regions is provided. In this step, efficiency estimation in the simulation environment is explained along with multiple experimental efficiency estimates. Next, the novel algorithm of high-efficiency pumping management is given. It covers the pumping management procedure in the average-speed area and the pumping management procedure in the high-speed area. The methodology is proved by simulation the high-efficiency pumping. To this aim, the ABB DriveSize toolkit as the model of the pump variable-speed drive was introduced, the simulation results were analysed, and the benefits and drawbacks of the simulation study were assessed. At the validation stage, careful experimental study of high-efficiency pumping was realised. Using the proposed methodology of experimentation, a set of experiments was conducted, followed by an analysis of experimental results with benefits and drawbacks of the experimentation evaluation.

Chapter 4 introduces and explores a set of improvements in the auxiliary processes and operations. Among them, the sensorless pressure measurement and control system was developed. This section includes algorithmisation of pressure estimation and control, simulation, implementation, and testing of the obtained results. In addition, the sensorless estimation of liquid density is proposed, including algorithmisation of density estimation, simulation, implementation, and result testing. The slip compensation method of pump induction drives has been developed also. This part of the study too involves algorithmisation of slip compensation, simulation, implementation, and testing.

Lists of 93 references and 16 author's publications consummate the thesis.

CHAPTER 1. STATE OF THE ART AND RECENT ADVANCES IN PUMPING TECHNOLOGY

Pumps are sizable energy consumers worldwide. They work in all kinds of industry, communal setups, and households, providing liquids transfer in various applications. For that reason, problems in various fields related to pumping always remain relevant and require new solutions due to developing technologies in such areas as industrial electronics, computing, and industrial communication. New methods and products developed in these fields are able to improve significantly the performance of pumping systems in such areas as monitoring of operation, robust and rapid control, and variable productivity control.

Pumping systems suffer from such problems as oversizing on the design stage, obsolete control methods used for productivity control, monitoring problems and common wear of hardware. Most of these problems result in low overall efficiency of pumping stations and hence, excess energy consumption and premature wear of components. Solution of many problems in the field of pumping can be achieved by providing the variable productivity through the variable-speed pumping.

In (IEA, 2015) it is stated that the total world electricity installed capacity is around 5000 GW. About half of this volume is used by pumps and similar applications. Taking into account that electric drive consumes about 70% of full energy generated, even 1 % of energy economy amounts to 23 GW. It is remarkable that pumps are the least efficient components of variable pumping systems. Hence, they induce the largest share of losses into the system. It means that the overall efficiency of a pumping system can be significantly improved by increasing the efficiency of pumping operations. This can be achieved by introducing the modern methods of the pump productivity control.

1.1 Analysis of Contemporary Centrifugal Pumps

1.1.1 Pump classification and ranging on applications

Through the last century, pumping application has expanded from agricultural use to a wide variety of industrial, communal, and home domains. The following list includes a few main sectors that use pumps extensively:

- *Water supply*: To transport water to consumers.
- *Sewage*: To collect and process sewage.
- *Drainage*: To control the water level in a processed area.
- *Petroleum industry*: To transport the liquids on all stages of petroleum production and refinery.
- *Irrigation*: To supply water to agricultural areas for watering.
- *Construction*: Well-point draining, and general site pumping applications.

- *Chemical industry*: To transport liquids to various setups and processing tanks in the chemical plant.
- *Pharmaceutical and medical field*: To transfer chemical materials in the drug manufacturing process.
- *Steel mills*: To supply with cooling water.
- *Mining*: To wash water, dust control fines, to control ground water and heavy-duty construction.

Pumps are also applied in industrial processes that are not directly connected to transportation of liquids. In these applications, pumps are employed as instruments in such processes as metal sheet cutting where a gas flame torch is too hazardous.

Pumps are typically classified according to the way energy is provided to the liquid. The basic types are positive displacement pumps and kinetic pumps.

Pump classifications vary in their details in different sources. A commonly accepted classification is proposed by Hydraulic Institute of Standards (HI) accepted by American National Standards Institute (ANSI). A classification of pumps by type proposed by the HI is presented in Fig. 1.1. This classification of pumps is made regarding to the type rather than to the application (Nelik, 1999). Generally, based on personal experience, preference for a specific type of a pump emerges from a particular industry.

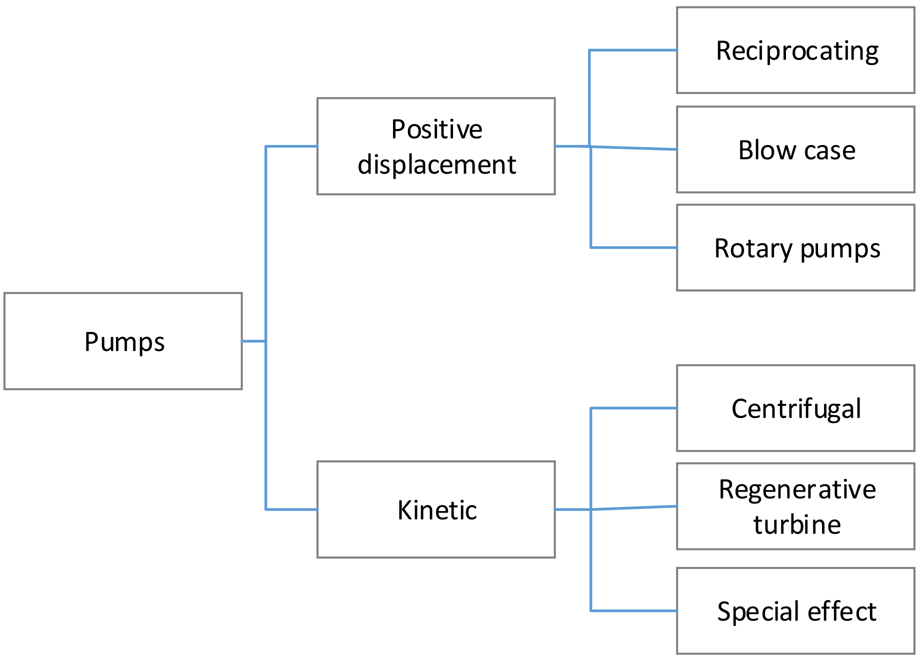


Fig. 1.1. Pump types.

In *positive displacement pumps*, the squeezing action of meshing gears or reciprocating action of one or several pistons, or lobes, displaces the liquid from suction to discharge. Generally, the pumped medium here is liquid. However, there exist designs that can handle solids in the forms of entrained or dissolved gas, suspension, paper pulp, slurries, mud, and other substances (Nelik, 1999). However, the behaviour of the medium must resemble the one of the liquid in order to be pumped. This means that resistance to tensile stress must be negligible. The positive displacement pump transports the liquid by taking in a fixed amount at the inlet, displacing that captured volume to the outlet. Displacement of the fixed volume per each cycle is a characteristic feature of this pump type. The transported volume is constant for each operation cycle. Another characteristic feature of the positive displacement pump is an ability to provide the same flow at a given speed, independent of the value of the discharge pressure, which makes this type of pumps constant flow machines. Positive displacement pumps have no shut-off head, so they cannot be set to operate against a closed valve. Operation against the closed discharge valve will damage the pump or the pipeline since the pump continues to generate the flow even in the shut off-state.

Unlike the centrifugal pumps that create pressure, the positive displacement pumps create flow. Here, the flow is generated by trapping a volume at the inlet, displacing it, and releasing at the discharge. The response of the system to the flow creates the pressure. Therefore, the use of displacement pumps is highly justified in systems that require constant flow with no regard to the value of pressure at the discharge. Such applications can be represented by oil pipelines that generally require a constant flow at various values of pressure.

Changes of viscosity and density would cause variations of the pressure. These changes are typical for applications in which pipelines transport liquids of different viscosity at different periods of a production cycle. Moreover, changing conditions would cause additional variations of processed liquid parameters. This can occur at production downtime. In such case, a centrifugal pump may not be able to generate a needed pressure to clear the pipeline. However, the positive displacement pump will produce enough pressure to restart the flow.

Fuel delivery systems at power plants require a constant flow of fuel to boilers or control turbines. In these systems, clogging of nozzles may cause variations of pressure. The constant flow feature of the displacement pumps is a significant advantage over the kinetic pumps.

In many cases, production lines include batch operations. The viscosity changes when the products or raw materials are being transported through the sectors or a new batch of product is added to the process. In case the constant flow condition is relevant for a given type of production line, the positive displacement pump is the type of machine able to meet the requirements. In addition, processing some types of liquids may require eliminating of the shearing action.

Compared to large centrifugal pump applications, with power range over 75 kW and viscosity over 50 SSU (a measure of kinematic viscosity, represented by time that 60 cm^3 of liquid takes to flow through a calibrated tube at 38°C), positive displacement pumps operate at high mechanical efficiencies over a wide range of viscosity values. In the overlap region, where both positive displacement and centrifugal pumps are able to operate, mechanical efficiency would be the key factor to make a choice.

Positive displacement pumps are capable of transferring a wide range of liquid types, providing the nearly constant flow without pulsation. The design of positive displacement pumps provides a low amount of shear to the liquid. Energy savings are provided by high mechanical efficiency at high loads.

In *kinetic pumps*, a centrifugal force of the impeller provides kinetic energy to the liquid, displacing it from the pump inlet to the discharge. The liquid is pressed out of the impeller by the centrifugal force and leaves the impeller chamber with increased velocity and pressure. In other words, the flow velocity is increased by adding the kinetic energy to the liquid.

This gain in energy is transferred to an increase in pressure (potential energy) when the liquid velocity is reduced at the flow exit to the discharge pipe. Kinetic pumps have several unique characteristics typical only of this type:

- Continuous energy.
- Transformation of added energy to kinetic energy increase (increase in velocity).
- Transformation of increased kinetic energy (velocity) to an increase in the pressure head.

Kinetic pumps can be safely operated under closed discharge valve conditions since they have a shut-off head and produce no continual pressure. So, at closed discharge, physical destruction of the pump or pipeline is less likely to happen.

The *centrifugal pump* is the most commonly used type of kinetic pumps. Share of centrifugal pumps in the industrial application is about 90 % (Girdhar, 2005).

The centrifugal pump construction is very simple as compared to other types of pumps. It essentially includes a volute and an impeller. The impeller is set on the shaft, which lies on bearings. The torque from the prime mover is typically transmitted through the coupling. Generally, an electrical motor is used as a prime mover.

A significant advantage of the centrifugal pumps over the positive displacement pumps is their ability to provide much higher flow. This advantage makes the centrifugal pumps especially attractive in many types of applications, such as municipal water supply, irrigation, construction, and petroleum industry.

1.1.2 Characteristics of a pumping process

Performance parameters of centrifugal pumps are generally described by pump characteristic curves that represent the connections between the efficiency, power on shaft, and head produced with a flow rate at rated speed (Karassik, 1998). The first of them is the pump performance characteristic (pump HQ curve)

$$H(Q) = H_0 - C_1 Q - C_2 Q^2 , \tag{1.1}$$

which links the flow rate Q with the total head H, where H_0 is the pump idle head and C_1, C_2 are the head factors.

These characteristic curves are provided by the pump manufacturer as a part of pump documentation. The head is represented by the total pressure difference across the pump. The location of the working point on the HQ graph pane depends also on the resistance of the pipeline to the pump. The pipeline resistance is described by the system characteristic

$$H(Q) = H_s + C_s Q^2 \tag{1.2}$$

with the definite system idle head H_s and system head factor C_s.

The working point, which is provided by the pump operating at the highest efficiency, is a Best Efficiency Point (BEP). Depending on the design, size, and application, the efficiency of the centrifugal pump varies. The efficiency of smaller centrifugal pumps is generally in the lower part (Europump, 1999).

The efficiency of the centrifugal pump is expressed by the location of the working point and the values of the input power and the density of the processed liquid. The exact relation is

$$\eta_{\text{pump}} = \frac{\rho \cdot g \cdot Q \cdot H}{P_{\text{pump in}}} \tag{1.3}$$

where ρ is the liquid density, g is the gravity acceleration, and $P_{\text{pump in}}$ is the pump input power.

Also, pump manufacturers usually provide the required net positive suction head ($NPSH_R$) vs. flow rate characteristic graph. This curve represents the head at the suction needed to avoid the cavitation in the pump. In (Sulzer, 2000) it is stated how the minimum allowable flow rate for continuous operation can be expressed by the $NPSH_R$ curve.

As an example, the characteristic curves of an Ebara CDX 120 series centrifugal pump are shown Fig. 1.2. The three HQ curves correspond to the different diameters of the impeller: 132, 157 and 176 mm for models 20, 12 and 07, respectively. The curves are defined for the constant operation at 3000 rpm. The BEP of the CDX 120/12 pump can be reached at the flow rate around 9.6 m^3/h and the efficiency of the pump in this case is 47 %.

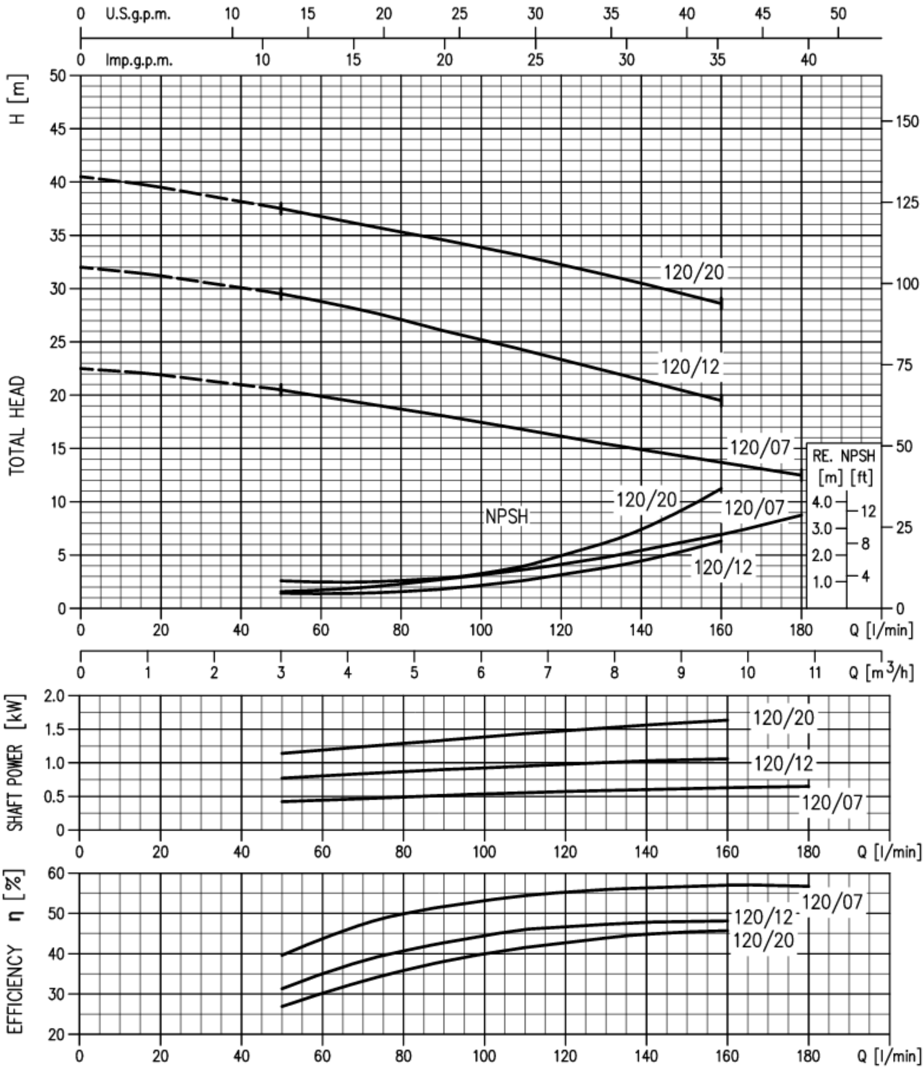


Fig. 1.2 Characteristic curves of an Ebara CDX 120 series pump provided in the datasheet.

One of the most important parameters for the evaluation of the pump operation is its rate of specific energy consumption E_s. It represents the (electric) input power per flow volume and can be expressed as

$$E_s = \frac{P_{in}}{Q}, \tag{1.4}$$

This parameter is especially useful for the determination of the energy efficiency at various methods of flow control (Europump, 2004). P_{in} is an electric power at the input of the pumping system.

Since the head is created by the rotating impeller, there exists a relationship between the peripheral velocity of the impeller and the produced head (Girdhar, 2005). For a constant impeller diameter, peripheral velocity is in the direct proportion to the rotational speed of the shaft. Therefore, varying rotational speed affects the performance of the pump. All the characteristic curves of the pump are shifting on the performance graph panes when the rotational speed varies. The displacement of characteristic curves of a pump is described by the following dependencies:

$$Q \propto n \tag{1.5}$$

$$H \propto n^2 \tag{1.6}$$

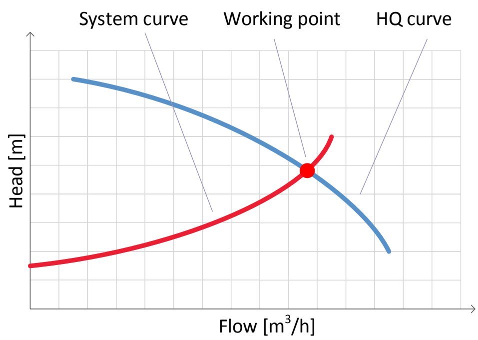
$$P \propto n^3 \tag{1.7}$$

Due to these dependences, HQ and PQ characteristic curves are shifting upwards and right when the speed increases and downwards and left when the speed decreases.

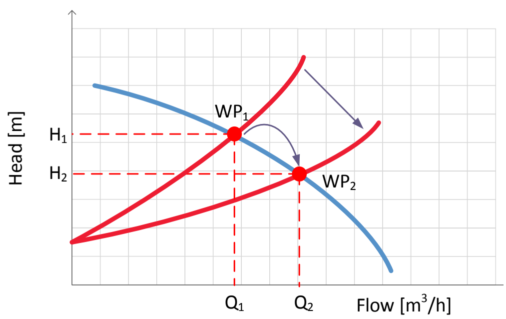
Cubical and squared relations of the head and power to speed mean that significant changes in the pump performance are caused by a relatively small change of the rotational speed (Europump, 2004).

1.1.3 Traditional speed-oriented scenario of pumping control

As Fig. 1.3 shows, the pumping working point (WP) lies in the intersection of the HQ characteristic curve and the system curve. The first one is defined by the physical characteristics of the pump to represent its performance. The second one is defined mainly by the physical characteristic of the pipeline. The intersection of these characteristics displays the current state of the system and expresses its working point.



(a)



(b)

Fig. 1.3 Location of the working point in the steady state (a) and in the case when the conductivity of a pipeline increases.

The shape of the system curve directly influences the performance of a centrifugal pump at various speeds. The system curve consists of two components: a dynamic head H_{dyn} and a static head H_{st}. This characteristic is expressed by a parabolic curve on the HQ graph. The static head component defines the flatness of the system curve. In case the distribution of the static head is high, the speed control capability of the head control applications tends to decrease since the significant displacement of the HQ curve leads to a relatively small change of the head.

In the ideal case, when the pump operates at its BEP, the main component of the system curve is the dynamic head. If the pump is run by the VSD, it will remain close to the BEP even if the speed varies. In the case when the system pipeline induces a significant amount of the static head, the working point of a pump can be displaced to the region of lower efficiency when the rotational speed changes.

The location of a working point can change with varying parameters of the system. It may shift in two dimensions:

a. Along the HQ curve when the conductivity of a pipeline changes
b. Along the system curve when the rotational speed of a pump changes

The case "a" is presented in Fig. 1.3 (b). The opening rate of the pipeline increases, making the flow rate to rise from a value Q_1 to a value Q_2. The increase of an opening rate itself is indicated by shifting the system curve down and to the right. The working point slides along the HQ curve together with the system curve. In the case of a decrease of the opening rate of the pipeline, the system curve deflects towards the H axis up and to the left making the working point to slide to the left along the HQ curve.

Fluctuations of the opening rate or conductivity of the pipeline can be caused by variations of the demand or process conditions. One example of demand variations in a city water supply system is represented in Fig. 1.4 (Koor, 2015).

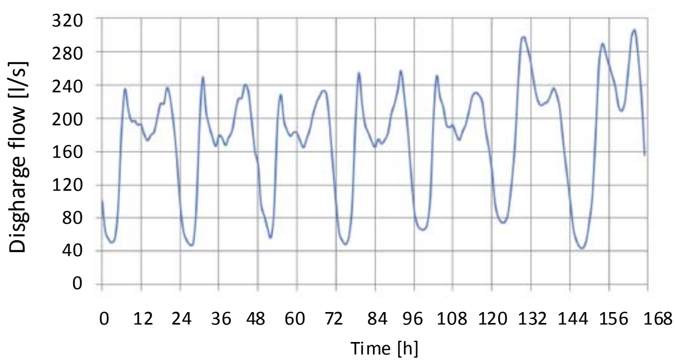


Fig. 1.4 Weekly graph of a discharge flow at "Punane" pumping station in Tallinn.

In order to keep the pressure on the constant level at the scenario shown in Fig. 1.3 (b), the working point should shift up to the level of H_1. This is possible due to an ability of the HQ curve to slide up and down correspondingly the rotational speed of the pump. In this situation, the control unit should increase the pump speed in order to react on an increase of the opening rate of the pipeline, increase of the flow, and pressure drop. This reaction of the control unit should compensate the pressure drop and provide an additional flow for fulfilling the growing demand.

The decreasing speed would make the HQ curve to shift down and to the left. The increasing speed would shift the HQ curve up and to the right. Fig. 1.5 shows the displacement of HQ curve in order to maintain the constant head in the system in the case when an opening rate of the pipeline increases.

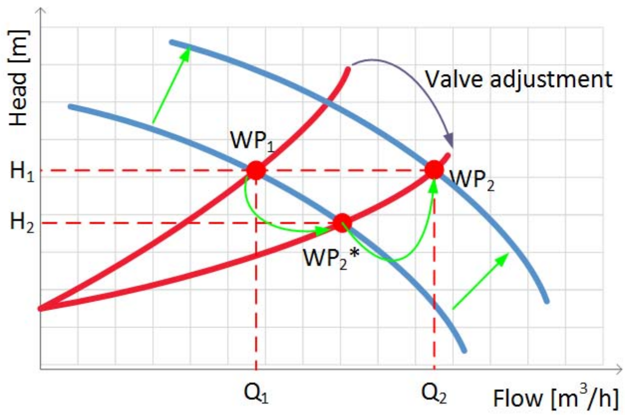


Fig. 1.5 Rotational speed of a pump increases making the HQ curve shift up and to the right. WP_2^* *is a transitional point.*

The new demand expressed by Q_2 is now being fulfilled with no pressure drop by an increase of the pump rotational speed. In the time domain, this process can be described by Fig. 1.6.

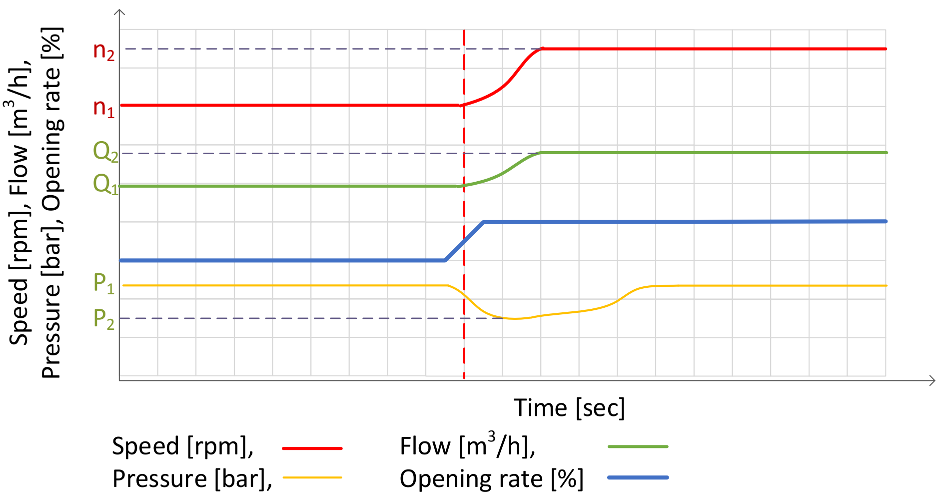


Fig. 1.6 Dynamics of pumping parameters at an increase of piping system conductivity.

Generally, in the existing pumping system control applications and firmware tools the programmable PID controller is responsible for the speed adjustment. In such applications, the PID controller adjusts the speed of a pump in order to compensate the deflection of the controlled parameter (pressure or flow) (Vodovozov, 2014). This makes the HQ curve slide along the HQ graph pane, displacing the working point of a pump.

When the productivity of a pumping system has to be increased but the speed cannot be increased due to the natural VSD limitations, staging up the number of working pumps is taking place. In other words, an additional pump (or several pumps) are being started. It should be noted that in most of existing pumping management software applications, the starting criterion for staging up is the growth of the pump rotational speed. In some applications, like Danfoss Aqua, the growing difference between the pressure of the flow and their reference values can be taken as a staging up criterion. In practice, the staging up looks like it is shown in Fig. 1.7.

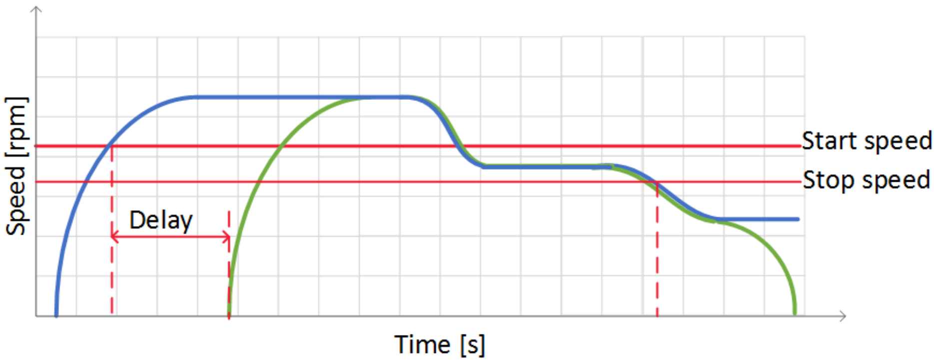


Fig. 1.7 Staging up the number of working pumps. Working application is ABB Pump Control software based on ACQ810 frequency converter. The graph is stored by ABB DriveStudio VSD monitoring tool.

It can be seen from Fig. 1.7 that staging up takes place when the speed starts exceeding the *Start speed* limit defined by the user. The time delay between overpassing the *Start speed* limit and a start of an additional pump prevents the system from an erratic staging in the case of sudden system fluctuations.

Generally, the value of *Start speed* limit is set in order to prevent the pump working at the speed, which is higher than the rated speed. Therefore, in these cases, one pump is capable of maintaining the needed demand and compensating an increase of consumption (before consumption becomes so high that the capacity of one pump becomes insufficient). Consequently, before the pump capacity limit is reached there are two options to fulfil the demand:

a. Stage up
b. Keep the increased speed

Naturally, an ability to operate at the high speed is defined by the system configuration and control considerations. If these abilities and considerations enable the pump to operate in the high-speed region, the optional operation method can be described by Fig. 1.8. This figure shows the possibility to keep the working point apply both options of fulfilling the above-mentioned demand.

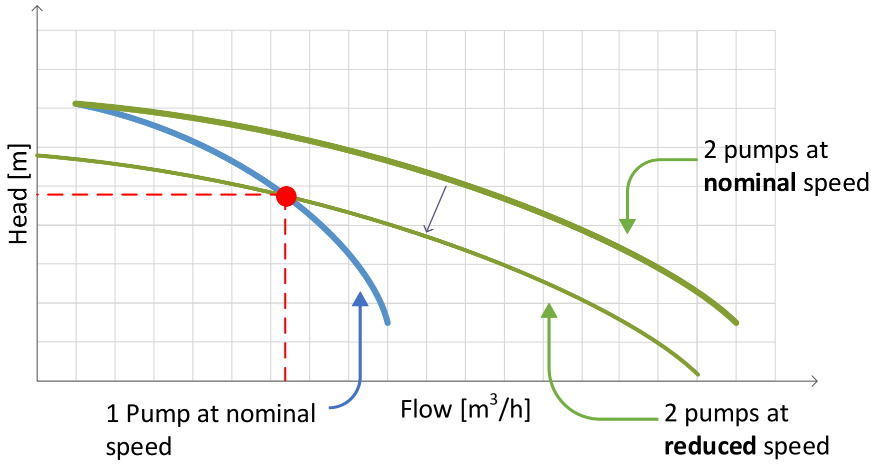


Fig. 1.8 Two options of keeping the pumping system at the working point: using one pump or two pumps at reduced speed.

Staging up the pumping system is implemented by starting an additional pump. In the case of synchronous speed application (when all pumps in the system run with identical speeds; this is a typical approach for a parallel pumping system), the speed of all previously working pumps would decrease in order to compensate the excess productivity.

The HQ curve for the double-pump case is generated based on the single-pump HQ curve, taking into consideration the principle of flow doubling in parallel pumping arrangements (Fig. 1.9). Therefore, double-pump HQ curve is a combined one expressing the performance of all the working pumps in the pumping system. The individual HQ curves for each working pump separately are shown in Fig. 1.9 as well. In fact, when pumps and their drives are identical, the curves are also identical and overlay.

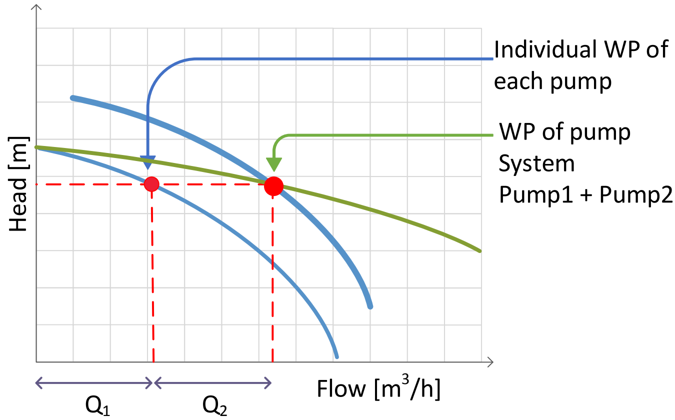


Fig. 1.9 Combined and individual HQ curves for pumping system when two pumps are running.

1.1.4 Resume

1. Pumps are sizable energy consumers worldwide. Unfortunately, pumping systems suffer from such problems as oversizing on the design stage, obsolete control methods used for productivity control, monitoring problems and common wear of hardware. Most of these problems result in a low overall efficiency of pumping stations and hence, excess energy consumption and premature wear of components. It is shown in this study that many problems in the field of pumping can be solved by providing the variable productivity through the variable-speed pumping.

2. Centrifugal pumps are the most popular and common pumping objects keeping the major reserves for improvements. It can be concluded from the analysis of their features that there are some undetected resources of how to improve their properties.

3. Performance of centrifugal pumping stations is generally described by the characteristic curves of the pump and the system curves. As the need in operation in the BEP is the main force in the pumping process enhancement and its efficiency increase, the directions of the new analytical and experimental study were formulated in this section.

1.2 Variable-Speed Drives for Pumping Applications

1.2.1 Overview of pumping drives

Many types of prime movers are available for pumping applications. Among them are diesel engines, hydraulic and gas turbines. However, the majority of pumps is driven mainly by electrical motors (Viholainen, 2014). The use of electrical machines is justified with their simpler construction and ease of maintenance and control relative to all other types of movers. Selection of a motor for the particular application depends on such factors as power range, required rotational speed and available type of supply. The type of motor construction for the specific application depends on the cooling medium available and on the environment of the installation.

The most widely used type of drivers for residential markets and industrial pumping applications are induction motors. In most cases, they can be run by frequency converters at variable speed without any modification. By supplying the motor with appropriate variable voltage waveforms and variable frequency to the stator windings, the speed of the motor can be changed proportionally to the applied frequency.

Figure 1.10 shows the coverage of different types of pumping applications by various types of motors.

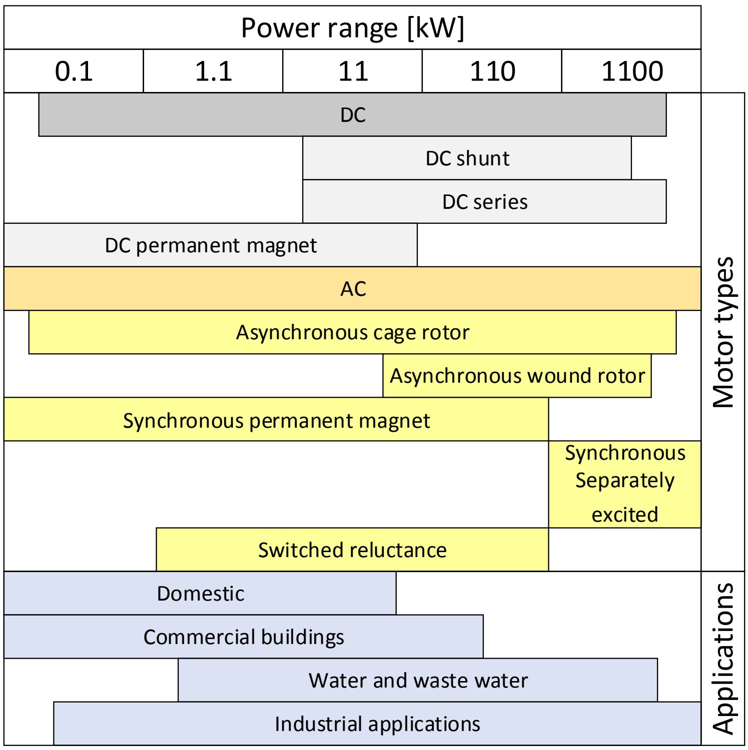


Fig. 1.10 Application fields of different motor types in pumping applications.

Synchronous motors are the most common solution for some specific low power applications, high power, typically greater than 5000 kW arrangements, and for the high-speed setups. For these types of applications, the separately excited synchronous motor is a good solution because of its relatively low rotor losses that result in simpler cooling. The permanent magnet generator can be attached to a synchronous motor in order to provide a wound exciter on the rotor or the rotor field. An important advantage of the synchronous motor is that these motors lack slipping. This can be especially important in variable-speed pumping when speed windows are defined in order to avoid the cavitation (Jones, 2013). Absence of slipping provides accurate operation in the limits of defined speed windows.

Switched reluctance motors are robust and simple in their construction. These motors are self-excited and use special geometry rotor for generation of a suitable field. The construction of the stator is also straightforward. It contains independent phase windings. The rotation is maintained by applying the energisation to the next pair of poles when the rotor poles align (Nelik, 1999). Operating of the switched reluctance motor requires a suitable converter unit. Most systems also require a position sensor or an encoder to provide the correct switching. The need in additional sensors and specific converters limits the use of switched reluctance motors in low power or fixed speed applications where the cost of additional equipment is difficult to justify (Nesbitt, 2006).

However, the use of AC permanent magnet motors seems more reasonable due to their high efficiency characteristics. The construction of these motors includes permanent magnets in the rotor. Unlike an induction motor, magnets of the rotor of the permanent magnet machine do not require electrical power. This makes it especially efficient at partial loads and reduced speed (Barns, 2003). Certainly, in addition, in the normal operational range, the permanent magnet motors are showing good efficiency rates. Traditionally, the permanent magnet motors have been used in many specialised applications like winding and spinning machines (Europump, 2004). The efficiency requirements made the manufacturers to turn to these machines in other types of applications like ventilation and pumping. The permanent motors are suitable for a wide range of high-speed small power applications as well as high power slow-speed setups. Performance of these motors is limited by the peripheral speed (Austin, 2013).

Asynchronous motors are most widely shared in today’s industry because of their relatively low price, simple construction, and ease of maintenance. The technological characteristics of these motors enable covering almost all power ranges. Additionally, low costs of these motors are critical for pumping applications, which are presented on water and wastewater market budgets that are smaller than the budgets of other industrial applications (Chazarra, 2014).

As proved by many sources (ABB, 2006), (Girdhar, 2005), (Viholainen, 2013), the variable-speed pumping is the most efficient way of liquid transportation. The variable-speed drives used in this type of transportation consist of motors and frequency converters providing the flexible speed control and ability to operate in a wide range of rotational speeds.

1.2.2 Variable-speed pumping

The most acceptable method of speed variation of induction motors is varying the supply frequency. Other methods, including the slip variation, induce unacceptable additional losses, which is outside the scope of this work.

Most of the variable-speed pumping applications utilize frequency converters based on the varying supply frequency of motors (Europump, 2004). The most popular type of the frequency converter modulation technique is pulse width modulation (PWM). In practice, the PWM principle is applicable to the majority of AC motor designs. This principle is usable for both permanent magnet synchronous motors and conventional synchronous motors, and with some adaptation to the switched reluctance motors. However, in the case of the conventional synchronous motors, an exciter control should be included. Figure 1.11 shows the distribution of frequency converter types among various applications and power ranges.

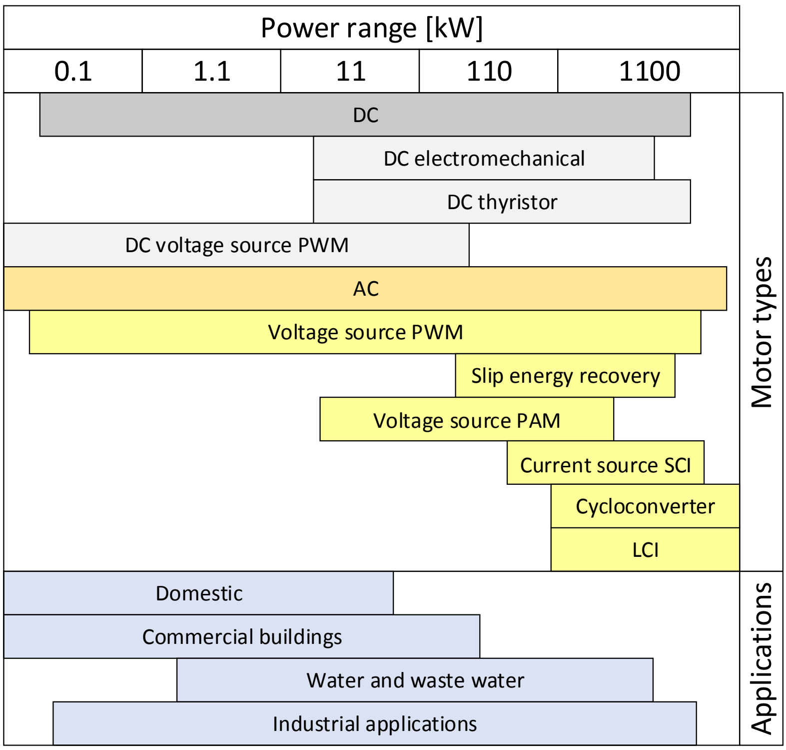


Fig. 1.11 Application fields and power ranges for VSDs.

The VSD provides an ability to define the preferred speed operation region for the rotating machinery and even allows operation at extreme speeds. In most of the applications, pumps are designed to operate with 50 – 60 Hz. The best efficiency can typically be achieved between these values (Rooks, 2004). With centrifugal equipment, variable-speed operation is extremely important from the point of view of operation close to BEPs, which can be impossible in constant speed setups with wrongly sized equipment.

1.2.3 Frequency converters for pumping applications

Reducing the energy consumption and increasing the reliability are among the main goals of pump control applications. Lack of sensing equipment and long distances between pumps in the field and control facilities impose significant difficulties for effective pump management. Automated self-adjustment solutions integrated into the control units of frequency converters can help to solve these problems (DOE, 2008). Employment of custom pump oriented control applications integrated into the firmware of frequency converters helps to improve the energy consumption and to increase the reliability of the whole pumping system (Petropol-Serb, 2014).

Increasing power consumption along with power economic requirements growth and rise in the energy price attract increasing attention to the optimisation of energy spending for rotating equipment. Improper sized pumps or those operating at inappropriate conditions induce a high amount of additional expenses. Sudden failures causing unplanned repair and low reliability increase company expenditures on additional maintenance and spare components. In addition, it is required to store the spare parts or the whole pumps in order to reduce the effect of sudden failures and downtime of equipment.

The intelligent pump management provides many advantages but its application is accompanied by such difficulties as distant location of pumps in the field, lack of sensors and individual features of different pumping setups (DOE, 2008). Moreover, some pumping sets are supplied with common sensing equipment for several pumps to be set up in series or parallel arrangements of several units (Bakman, 2012). In pumping systems like these, it is difficult to indicate the failed pump, type, and cause of failure remotely. The automated pump management systems, which are capable of monitoring and productivity adjustment, can be implemented based on the frequency converter control unit. In such a way, a pump management system can be integrated into the pump VSD platform, making the control and monitoring more effective.

Pump management software applications integrated into the VSD firmware have direct access to electrical and hydraulic parameters, providing rapid monitoring and adjustment of operation. Ease of acquiring the pumping parameters enables implementation of the intellectual algorithms of management that can keep the pumping system operation closer to the BEP where its reliability is highest, provide alerting in dangerous situations and redundancy.

Intellectual pump control software provides constant diagnostics of a pumping system to detect the harmful phenomena or cases when a pump is running in inappropriate region of operation. In the case of danger, the control program adjusts the parameters of pumping in order to prevent the destruction of the pump, pipeline or other equipment. Critically high pressure, cavitation, or dry run are dangerous occurrences that can be detected by modern digital signal processing controllers. Implementation of monitoring and diagnosis algorithms on the platform of frequency converter's control unit provides fast and technically easy data acquisition.

Variable-speed pumping provides serious advantages in the energy efficiency and process control in contrast to such control methods as throttling, bypassing, and on-off control. Possible energy optimisation methods by using the variable-speed drives were researched in last decades.

Liquid dynamics required to reveal the main causes of losses is addressed in (Hickok, 1985). In addition, an effort was made to estimate the possible savings when integrating the variable-speed drives into pumping systems and implementing the suitable control techniques.

Methods for evaluation of possible savings in the case when the frequency converters are used together with induction motors are presented in (Rice, 1988). Here, one of first proposals can be found how to apply the pumps and VSDs databases in order to optimize the process of selection and sizing of the pumping and ventilation equipment.

In (Carlson, 2000), it is emphasised that just integration of VSD into the application does not necessarily lead to significant savings. The need of correct sizing and investigation of process details and characteristics is required to make use of variable-speed pumping. A proposal is made to use the mathematical model of the pumping process to perform an accurate analysis of the target system.

In (Bortoni, 2008), a method is proposed to determine the shape of the system curve and the head-flow (HQ) curve parameters realising the common field measurements. The control method is implemented using polynomials characterising the process and the pump, so it is a model-based method. This method utilizes a program, which can be built and run in the programmable logic controllers (PLCs) in order to optimise the control of parallel pumping sets provided by multi-pump setups.

In (Schützhold, 2013), a guide for selecting an appropriate VSD was proposed in order to provide a better use of saving potential. Its purpose is to assist in choosing the best configuration of VSD and pump application on early design stages. The proposed method requires the use of free accessible data of the pumping equipment in order to estimate the energy efficiency of the pumping setup. To implement the method, the power flow model was proposed. Characteristics of the load process, pipeline system, centrifugal pump, induction motor, and frequency converter were taken into consideration in order to implement the model. The method provides selection of an energy efficient pump drive system and is suitable for multi-pump arrangements.

Generally, studies concentrate on two types of problems: justification of the VSDs use in various pumping applications and proposing new methods of productivity control and online estimation of pumping parameters (Viholainen, 2013).

Specific methodologies have been provided to manage the pumping arrangements due to changing operating conditions, which can be implemented on the basis of existing digital controllers. Thus, specific frequency converters have been developed for pumping applications. Effective solutions in the field of water and wastewater market were proposed by ABB (ABB ACQ, 2014), Danfoss (Danfoss, 2014) and ITT (ITT, 2007). Those companies manufacture frequency converters with integrated software containing pumping-oriented control algorithms. Their software contains utilities for monitoring pump operation, estimation of the pumping process parameters, protection of the pumping system as well as algorithms for productivity management for single-pump and multi-pump arrangements.

Solutions from ABB are represented by several generations of frequency converter families for pumping application: ACS800, ACQ810, and ACQ550. Each converter family provides frequency converters in the range from 1 kW to 500 kW, capable of providing the motor control running in the scalar or direct torque control (DTC) applications. The converters are equipped with high performance control units, enabling the use of demanding software applications for motor control as well as for pumping system management.

The pumping system management features include pump protection against overpressure, underpressure or dry run, impeller self-cleaning functionality, detection of impeller stacking, sensorless flow calculation, advanced ramping functionalities, staging management for single-drive multi-pump and for multi-drive multi-pump arrangements, productivity monitoring, and alerting functionalities. Common functionalities not directly connected to pump management are presented by adaptive programming environment, enabling the user to compose sophisticated control algorithms and to run them in parallel with native converters firmware, support of multiple industrial communication protocols, onboard analogue and digital inputs and outputs, flexible parameterisation of motor control, and monitoring functionalities. In fact, the control units of the converters of the families mentioned are provided with the PLC playing the role of programming and data acquisition.

The ACQ810 and ACS800 converters include the model-based functionalities, providing the sensorless flow estimation described in (Ahonen, 2010). Productivity adjustment is represented here by the functionalities providing the speed adjustment and staging of additional pumps. As a criterion for staging, the current speed of the pump is used. In the case of exceeding the user defined limits, the staging takes place. Both ACQ810 and ACS800 solutions provide control of the group of up to eight pumps via Modbus communication. The group functionality provides alternation of working pumps, acquiring their working parameters (in the multi-drive application) and hence, redundancy of the whole pumping system through ability to substitute the failed pump with a working one.

VLT AQUA of Danfoss is capable of driving the pumping systems of up to 2.5 MW. The frequency converter is also equipped with a powerful control unit providing the productivity management of a group of pumps and motor control of the main pump. It supports the single-drive multi-pump applications. Among the pump system management features, the staging cascade control, motor alternation function, pressure protection functionalities, flow calculation, and cavitation protection feature are used. The control unit supports several fieldbus protocols. Digital and analogue input/output interface is available. Productivity of the pumping system of up to eight pumps is controlled by the variable speed of the main pump and on-off control of auxiliary pumps unequipped with frequency converters.

The frequency converters of ITT PS200 family are capable of running the pumps in the range of 3 to 500 kW. Group of up to four pumps working in a parallel multi-drive arrangement can be run by the application. The application provides such monitoring and control features as flow and pressure protection against extreme levels, cavitation protection, and sensorless model based flow calculation. The staging productivity adjustment is capable of adjusting the speed of pumps and the amount of running pumps due to the temperature of the liquid, pressure, flow and main pump speed values.

Today, combined integrated designs of a motor and a frequency converter are available on the market. The combined setups consist of a motor and specially designed frequency converter manufactured as a single package with the converter mounted on the top or end side of the motor. These setups are generally available in the power range of up to 25 kW. Advantages of such solutions include, for instance, proper matching of the motor and the frequency converter provided by the manufacturer, ease of installation, since there are no additional cables, a combined cooling system of the motor and the converter. Designs of such kind is provided by many manufacturers, including the leaders in the water market (Grundfoss, 2015).

All the mentioned manufacturers declare the efficiency of frequency converters close to 98 %. The converters are capable of running a wide range of pump types, like horizontal and vertical centrifugal, submersible, positive displacement and other kinds of pumps.

1.2.4 Resume

1. According to the overview of pumping drives, the most widely used driver for residential markets and industrial pumping applications is the electrical motor, especially an induction motor. An increase of the pump productivity resulting from their specific features is described,

2. From the analysis of the application fields and power ranges of VSDs, new perspectives of variable-speed pumping are drawn.

3. Frequency converters used in pumping systems enable serious advantages in the efficiency control using the PLC and introducing appropriate regulators. Resulting from the comparison of the VSDs from different companies, their benefits and drawbacks were underlined.

1.3 Control Systems used for Pumping Management

1.3.1 Overview of control methods and control topologies

In industrial and communal pumping applications, the variable flow rate is a usual requirement. As it is stated in (Viholainen, 2014) there are four common approaches to control the output flowrate of a pumping system: throttling, bypassing, on-off, and variable-speed control. Variable-speed control may be achieved by running the pump motor with a frequency converter or by using hydraulic couplings and a gear.

The hydraulic power expressed by the rates of the output flow and the achieved head is indicated by the output power of different control methods. The relative power can be estimated by comparing the area limited by the values of the flow rate and the achieved head. Further, four most widely used flow control techniques will be presented and compared from the point of view of hydraulic power.

Throttling is a direct limiting of the pump system outlet. Typically, limiting is done by changing the conductivity of the pipeline by using the hydraulic valve. In case the flow reduction is needed, the valve-opening ratio is reduced. This means that the output flow is definitely lower than the rated flow of the pump.

The throttling flow control can be implemented by modulating a valve, which is located straight after the pump. In fact, throttling changes the system curve seen by the pump: the valve introduces the resistance and friction into the system. This makes the system curve to shift left, closer to the head axis so that it intersects with the HQ characteristic curve at a lower required flow rate.

Implementation of this method requires a low investment. However, throttling adjustment systems waste energy in two ways: forced capacity decrease makes the pump operate below its BEP and pressure drop occurs across the valve. Operating far from the BEP can cause destructive phenomena like cavitation or vibration and leads to premature wear of the pump.

In applications involving the heating of a pumped liquid, the energy spent because of throttling is not wholly wasted. From the point of view of energy efficiency, this approach in such kind of applications can be justified. However, due to the following reasons, the throttling appears to be inefficient and even harmful even in this type of applications:

- Higher maintenance costs. The pump operating far from BEP is a subject to intensive wear.

- The resolution of the valve may be insufficient to provide the accurate flow control. An oversized valve operates in a nearly closed state, which causes instability. Installing the oversized valve into the piping system can be caused by an intention to introduce the safety margin in the calculation of the pressure drop across the elements of the pipeline (Liptak, 2005)..
- Using the throttling approach in high-pressure systems can result in a significant wear of valve components (Lawrence, 1996).

Bypassing. In the bypass approach, the discharge flow of the pump is reduced by turning the part of the liquid from the pump outlet back to its suction. This method is typically used in circulation applications. The amount of liquid flowing backwards is adjusted by valves. In fact, this means that the discharge flow increases, but not all of the flow is going to the consumer.

A bypass approach can be wholly justified if the pump operates at lowered flow rates during extended periods. In this case, the pressure is not high enough to provide the desired flow in the piping system. This approach can also be useful in variable-speed pumping applications in order to protect the system from the low flow threats.

Flow recirculation causes the energy waste on bypassing the part of liquid (which does not reach the customer) back to the pump inlet. This kind of flow control may be used in condensate and boiler feed pumping systems of power plants in order to prevent overheating of the pump at low flow rates and to provide productivity control (Lawrence, 1996).

One more reason why a bypassed flow loop can be used is to keep the pump working at normal speed when consumption is low and to provide the readiness for an increased demand.

On-off control can be used where constant supply is unnecessary. These are typically wastewater applications or systems where keeping the specific level or pressure between the preset limits in the tank is a target. The adjustment of productivity is reached by simple alternation of start and stop states of predefined duration. The average productivity can be expressed as a relationship between the 'on' period and the total time.

Variable-speed control. The BEPs of several HQ characteristic pump curves of various speed values lie on a parabolic curve according to affinity laws. For a curve resulting from the BEPs, pressure values are proportional to the square of flow values.

The system curve of a low static head application has a shape similar to that defined by the BEPs at different speeds. Therefore, by varying the rotational speed of the pump, the working point can be shifted along the system curve (Ahonen, 2011). Hence, the working point can be kept close to the BEP.

The HQ characteristic curve of the pump is shifting on the flow-head graph pane, following the rotational speed of the pump. Generally, a manufacturer provides only the HQ characteristic for the rated speed. Each point on the HQ characteristic curve of the rated speed can be projected on the HQ curve of the arbitrary speed using the affinity laws (Bakman, 2014). The HQ curve shifts downwards and left when the rotational speed of the pump decreases. It shifts upwards and right when the speed increases. Using this principle, it is possible to predict the location and shape of the HQ curve at any speed. Also, once the shape and location of system curve is known, it is possible to predict the location of the working point taking into account that it lies on the intersection of the system and HQ curves. Hence, by varying the speed of the pump, it is possible to bring the working point closer to the BEP, at the same time, meeting the demand requirements (Hovstadius, 2005).

All of the above types of control, excluding the variable-speed control introduce friction and hydraulic losses, need in extra flow, pressure drop and risk of premature wear of the components to the system. Also, all the options, except the variable-speed control, are likely to require a mechanical reducing gear if the pump rated speed is higher than the motor rated speed. Reducing gears are efficient; however, they provide up to 1.5 % loss in the input shaft power.

Derived from the pump energy efficiency problem, many manufacturers provide readymade solutions for pump management. Generally, these solutions are implemented as narrowly focused digital controllers or PLCs containing custom-made pump oriented software applications (Grundfoss, 2015). Manufacturers of frequency converters also tend to embed the custom-made pump management applications into their products if enabled by the computing capacity (ABB ACQ, 2014), (Danfoss, 2014), (ITT, 2007).

1.3.2 Model-based control and its perspectives for pumping

One of the main requirements for the optimisation of energy consumption of the pumping system concerns the use of the real time information from the surrounding components. In the pumping systems, this information can be obtained from the pressure and flow sensors or from the frequency converters providing the parameters of the modulation. Also, such parameters of the modulation process as current speed of the motor and current power can be used in the sensorless calculation of hydraulic parameters like pressure or flow (Ahonen, 2008). The parameters of a frequency converter are typically accessible through industrial communication and can be used in the control equipment of a pumping station. The flow monitoring functionalities are available on contemporary frequency converters (Hammo, 2008).

Model-based control methods are applied in the control strategies especially in variable-speed pumping. The model-based methods enable easy estimation of the location of the working point using the current operation parameters of the frequency converter (Ahonen and Tamminen, 2012). Model-based methods are especially relevant for variable-speed parallel pumping systems. In these systems, estimation of the working point location requires only HQ characteristics of the pump and current operation parameters of the VSD. The HQ characteristics can be easily provided to the control equipment in the form of look-up tables and adopted for variable speed and variable number of running pumps. The model of a pump is expressed by the HQ and PQ characteristic curves of the pump provided by the manufacturer.

Another approach to the model-based control is based on the implementation of the polynomial equations that express the characteristic curves of the pump (Koor, 2014). A significant advantage of these models is the ability to represent the system curve as a polynomial. This method requires providing not the key points of HQ and PQ curves in the form of look-up table but the key constants of the polynomials describing these curves to the processing control unit. Estimation of the working point is implemented by solving the system of equations describing the pump characteristic curves and the system curve. At that, such parameters as speed reference and needed number of working pumps can be defined in the analytical way by solving the mentioned equations and mapping the results to the pump characteristics graph (Bogumil, 2008).

The model-based control methods can be easily implemented based on the contemporary control equipment like PLCs or programmable frequency converters. A significant advantage of the model-based methods is the ability to incorporate models of VSD together with models of pumps in order to take into consideration more operational parameters and to provide a wider overall view of the pumping system. This feature would be especially useful since the efficiency characteristics of various components of the pumping system are not homogeneous. Monitoring the efficiencies of all the components including the VSD would be useful in order to estimate the working point more accurately and take measures in order to improve the performance and efficiency in real time.

1.3.3 Predictive control and its perspectives for pumping

The model-based control approaches provide estimation of the working point location from common solutions of a system of polynomial equations or from an analytical determination of the intersection of the system curve with HQ curve. These methods utilize the relevant parameters of the pump characteristic curves and the system curve. The parameters are relevant for a real-time situation. The shape and location of the pump HQ characteristic curves depend on the current rotation speed and the number of running pumps (Ahonen, 2011). This means that the shape and location of these curves can be predicted for future situations.

Location of the HQ characteristic curve on the performance graph pane depends on the pump speed. Hence, once the desired speed and required flow parameters are known, the location of the working point for the variable-speed system may be predicted.

The shape of the HQ curve is changing depending on the number of running pumps. In the parallel pumping system, the value of flow in each point of curve is being incremented due to the contribution of each auxiliary pump, and the value of the head for the point stays constant. Therefore, the transformations of the HQ curve shape are also predictable. Once the future number of working pumps and speed needed to supply the desired flow or pressure is known, prediction of the location of the working point becomes possible. This is relevant for starting the new variable-speed auxiliary pumps or for stopping the working variable-speed auxiliary pumps.

In combination with the VSD model, the predictive control method is capable of estimating the overall efficiency of the pumping system in various situations. The use of the predictive method enables avoiding harmful phenomena in the pumping process by determining the change of operation towards the undesired areas.

Since predictive techniques are suitable to estimate the future working point, they are also capable of estimating the future rotational speed of the pump, needed to supply the required flow or pressure. An operation at low speed or low torque decreases the efficiency of each component in the pumping system. Therefore, for the multi-pump system, it is desirable to avoid operation in the low speed regions. As a result, the predictive control is capable of estimating the future changes of efficiency and avoiding the operation in the low efficiency regions.

The predictive control can be implemented on the contemporary control equipment. It utilizes the same parameters as other model-based methods.

1.3.4 Resume

1. Analysis of the pumping methods and control topologies revealed the benefits and resources of the variable-speed control. As a result, solution of the energy efficiency problem of a pump was chosen as the main task.

2. As a tool for the problem solving, the model-based approach was selected. It was shown how to move the working point in the optimal direction using this approach.

3. In this connection, the predictive control topology is emphasised. Since predictive techniques are suitable for the future working point estimation, they are also capable of estimating future rotational speed of the pump needed to supply the required flow or pressure.

1.4 Pumping Efficiency

1.4.1 Problem statement

A variable-speed pumping system typically consists of a centrifugal pump that is driven by an electric motor and a frequency converter (Ahonen, 2011). A simple single-pump pumping system consisting of a pump, a motor and a frequency converter is shown in Fig. 1.12.

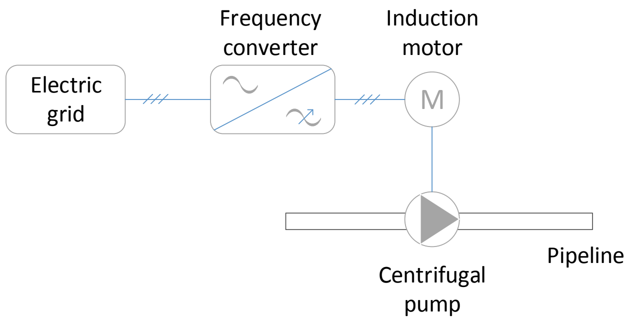


Fig. 1.12 Typical single-pump pumping system.

Figure 1.13 shows the types of losses and ranges of typical efficiencies of pumping system components.

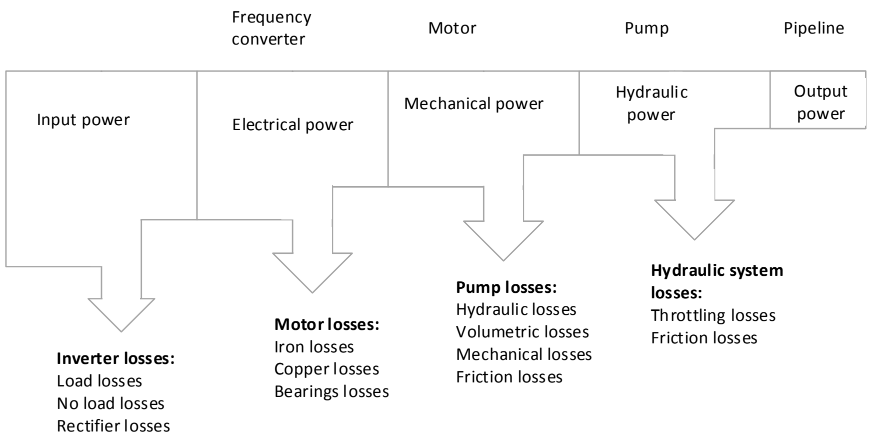


Fig. 1.13 Types of power losses in a pumping system.

The efficiency of a variable-speed pumping system is affected by the efficiencies of all its components. Every component of the system induces a corresponding type of power losses due to its efficiency.

The difference between the input power and the hydraulic power is caused by several types of losses.

Hydraulic losses represent the set of losses in the volute and impeller. These losses occur due to friction inside of the pump body, constant change of the flow direction inside the volute and recirculation of the liquid as it is being pushed by the impeller. Volumetric losses are represented by leakage of the small amount of liquid from the outlet of the centrifugal pump to the inlet side. The flow leakage through the vane front clearances of half-open impellers, wearing rings, and balancing holes defines the volumetric efficiency. Volumetric losses grow as internal clearances increases due to erosion and wear of pump components. Mechanical losses are represented by friction that occurs in the moving parts of the pump, such as bearings and seals or packing. Disc friction losses are the type of frictional losses. The reason is that the impeller, the dynamics of which can be described as the dynamics of a turning disk is rotating in very close proximity to the walls of casing. The losses are induced due to the frictional resistance occurring from disk turning in close proximity to the surface.

The relative share of the above losses varies from one pump type to another. The efficiency of pumps varies widely, making the pump in the pumping system, including an electrical drive, a component of lower efficiency.

In detrimental operation conditions, multiplication of efficiencies of all components can yield quite small numbers. Typically, an electric motor and a centrifugal pump are directly connected and supplied as a single unit by the manufacturer. The frequency converter provides a speed adjustment possibility for the system. The frequency converter may be a separate product or may be integrated into the pump-motor system. The pump is directly connected to the motor by hard shaft coupling. The frequency converter may be located in a separate cabinet or near the pump-motor unit.

It is important that the efficiency of every component is subjected to change depending on the conditions of operation.

Location of the working point defines the hydraulic power of a pump or its output power. In (1.3), the hydraulic power is represented by flow Q and head H. Efficiency of the pump is a relation of its hydraulic power to the input power as can be seen from (1.3) also. The efficiency of a pump itself can vary in the range of 35 – 88%.

As the losses are the variable properties of the system components, it is noticeable from the example in Fig. 1.14 that the efficiency of the pump is subjected to a significant change at the varying parameters of demand or the pipeline when the share of the static head is low.

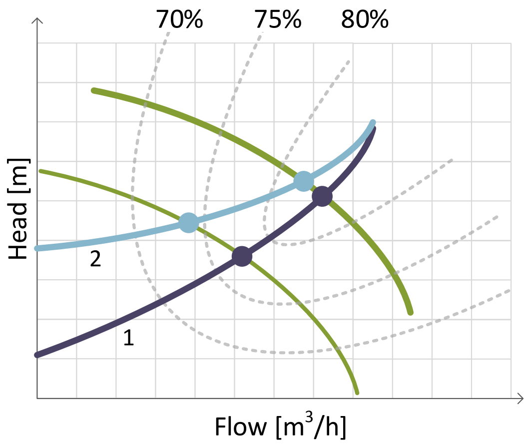


Fig. 1.14 Displacement of the working point of a pump as a result of varying rotational speed: 1 – for low static head, 2 – for high static head.

Electric motors and frequency converters are the high-efficiency devices. Nevertheless, efficiency of a frequency converter and a motor is subjected to change depending on the operation conditions as well.

As the efficiency of the pumping station varies along with the changes in loading and operational conditions, the problem of this thesis is in the arrangement of the control that would provide the system operation at the highest efficiency with minimal component losses. Further, this problem is discussed in detail.

1.4.2 Related work

Assuming that a pumping system consists of a pump driven by an asynchronous motor, which is run by the frequency converter, the assembly can be described by the following characteristics.

Figure 1.15 illustrates the performance curves of the 15 kW Ebara 3UB-65-200 centrifugal pump. As can be seen from the figure, the HQ curve is monotonically rising towards shutoff head with the pump developing the lower flow at higher head. The pump efficiency characteristic shows that the efficiency varies with the productivity rising to a maximum value known as the BEP.

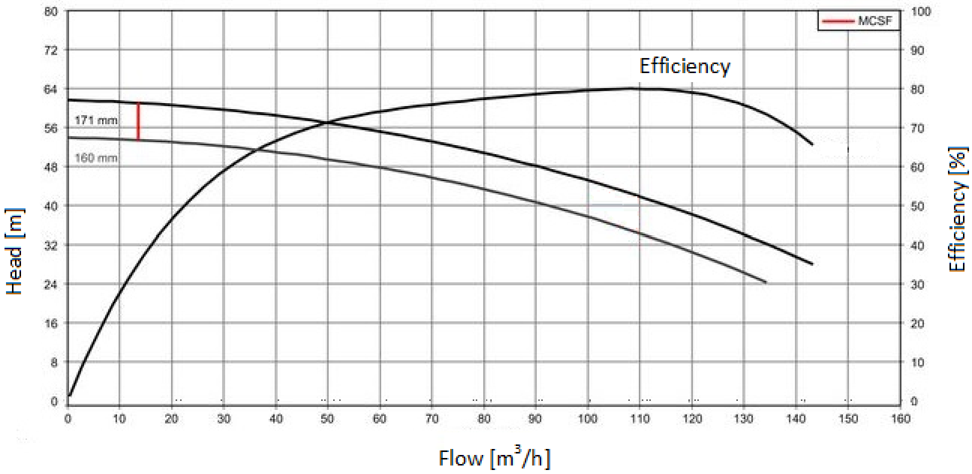


Fig. 1.15 Characteristic curves of Ebara 3UB-65-200 centrifugal pump for impellers of diameter 171 mm and 160 mm .

It can be seen that the efficiency of the pump is rising while the flow is increasing. At lower values of the flow (and typically low rotating speed of the pump), the efficiency is decreasing. The best efficiency can be obtained at higher flows and higher speed.

Decrease of efficiency is caused by such types of losses as mechanical, friction, volumetric and hydraulic losses. All of these types are varying with the rotational speed of the pump. However, the pump efficiency mainly depends on the speed change when the hydraulic losses increase dramatically. In its turn, hydraulic losses depend on the rotational speed and the productivity of the pump (Barringer, 2003).

The most widely used type of motors in the pumping applications is the asynchronous (induction) motor. Its low cost, simplicity and high efficiency make it attractive for a wide variety of applications.

Distribution of typical types of losses for a 15 kW induction motor are shown in Table 1.1.

Table 1.1 Typical induction motor losses (Europump, 2004).

Type of losses	Losses [kW]	Distribution [%]
Iron losses	0.48	43.75
Copper losses	0.20	18.75
Windage and bearing	0.22	20
Stray loss	0.19	17.5

The distribution of losses is varying for different types and sizes of motors. However, generally, the copper losses reduce as the load becomes lower. At the same time, the windage and bearing losses drop at lower speeds. The iron losses, which represent a greater part of the overall motor losses, are not so sensitive to the speed change. This means that these losses will not decrease with the speed decreasing. Thus, the distribution of iron losses at lower speeds will increase relative to other types of losses.

Figure 1.16 shows the efficiency vs. speed curves of a high efficiency motor driven by the frequency converter with a rated efficiency of 95 % at different torque loads. It can be seen that at a low torque, the efficiency decreases. This is a typical situation for the centrifugal pump operating at lower rotational speed, providing the low flow and hence running at high distance from the BEP.

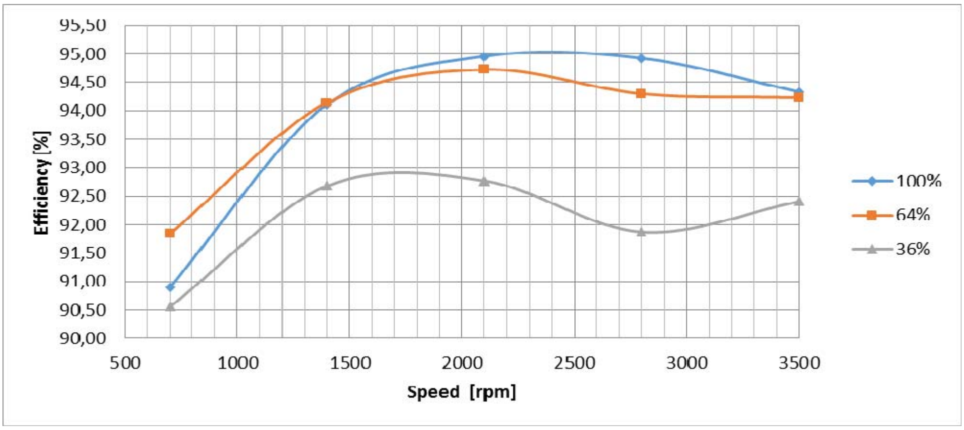


Fig. 1.16 Motor efficiency vs. speed of 18.5kW, 95 % efficient motor, M3BP 160MLC 2, ABB.

In 2006, 19 % of all electrical motors were used in pumping setups (Stoffel, 2015). Around 8 – 9 % of worldwide-generated electrical energy was utilized by pumping systems. One of the main reasons of power loss in pumping applications is the use of oversized pumps. Typically, pumps are oversized in order to provide the safety margin. In addition, a pump can become oversized because of constant decrease of demand. Up to 75 % of pumps in today's pumping systems are oversized (Europump, 2004). As shown in (Ostfeld, 2012), energy consumption in pumping systems worldwide can be reduced by 25 % through performance optimisation.

The induction motor is the most frequently used motor in the pump-motor combinations because of its reliability qualities and simple construction (Nelik, 1999). Also, this motor can be easily integrated with most of modern frequency converters that provide a good speed control capability. Due to the nature of pumping applications, there is often no need in precise speed and torque control or positioning.

Typically, when the load torque is around 75 % of its rated value, the induction motor operates at its maximum efficiency (Ahonen, 2011). For medium and high efficiency motors, the efficiency starts decreasing when the load torque approaches 25 – 35 %.

The motor efficiency also decreases when it is running at low speed. The reason is the presence of the constant-value losses. The motor efficiency will be seriously decreased when the pump is running at low speed. The reason is a squared dependency between the pump rotational speed and the pump load torque (Abrahamsen, 2000). Figure 1.16 shows how the decrease of the rotational speed influences on the motor efficiency when it is run by the frequency converter. The drastic decrease of efficiency at load torque lower than 40% should be noted.

As it was stated earlier, the frequency converter provides the speed control capability to the pump-motor combination. The values that affect most significantly the frequency converter efficiency when it is applied to an induction motor are the torque and the required speed. The maximum efficiency of modern frequency converters up to 400 kW output power range is typically 92 – 98 % depending on their size.

When the motor is running by the frequency converter at low rotational speeds, the efficiency of the frequency converter is also decreasing. A significant decrease in the efficiency of the frequency converter operating at a low speed can be seen in Fig. 1.17.

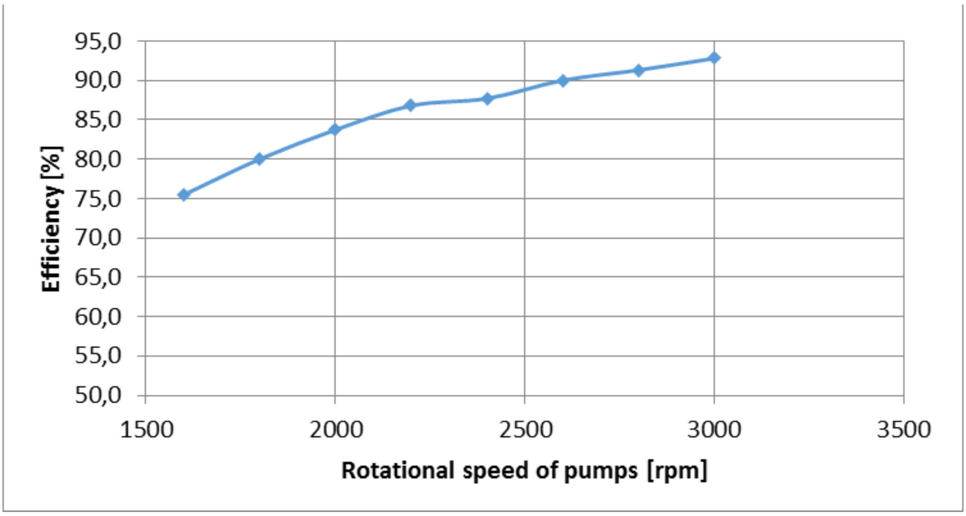


Fig. 1.17 Efficiency of 1.1 kW frequency converter (ACQ810-04-02A7-4) running the motor and the centrifugal pump (EBARA 120/12).

Total power loss of a frequency converter consists of the load and no-load losses. The no-load losses generally consist of losses in the discharge and snubber resistors, switching losses in the semiconductors, leakage in the capacitors and power spent on auxiliary supply equipment like fans and circuits of control board. The load losses are caused by the current passing in the components of the frequency converter, the corresponding smoothing components, and main circuit chokes. Table 1.2 shows the distribution of losses within the frequency converter.

Table 1.2 Typical losses in active rectifier frequency converters (Girdhar, 2005).

	Share of losses [%]	
Type of losses	**Active rectifier frequency converter**	**6-pulse frequency converter**
Fixed losses	20	10
Rectifier losses	35	30
Load losses	45	60

The load losses vary widely, depending on the operation conditions, corresponding to the drawn current. When running the low power factor motors, converter losses will rise disproportionally because of higher current. Also, since the low-efficiency motors are suffering from additional losses when operating with frequency converters, the converter can become a source of extra power loss in the drive train.

Another source of efficiency decrease at the drive side is the shape of the modulated signal at the output of the frequency converter. Additional losses in the motor are caused by the fact that the waveform of voltage generated by frequency is not purely sinusoidal (Bose, 2001). This phenomenon becomes more harmful at the low rotational speed.

1.4.3 Current problems and possible solutions

The efficiency of a centrifugal pump is merely a relation of the processed liquid power at the output to the input power or motor shaft power. The liquid power at the output of the pump is also called hydraulic power. The relation is expressed by the following equation:

$$\eta_{\mathrm{pump}} = \frac{P_{\mathrm{WP}}}{P_{\mathrm{pump\ in}}} \tag{1.8}$$

Here η_{pump} refers to the efficiency of the pump and $P_{pump\ in}$ to the input power of the pump and P_{wp} to the output power of the pump. Overall efficiency of the pumping system however, includes efficiencies of the pump motor and the frequency converter. Measuring the shaft power requires additional equipment and imposes specific technological difficulties, since the mechanical power is to be measured. Measuring the electrical power at the input of the motor is technologically easier. In case the motor input power is used in the calculations, (1.8) will transform as follows:

$$\eta_m \cdot \eta_{\text{pump}} = \frac{P_{wp}}{P_{mip}} \tag{1.9}$$

Where η_m refers to the motor efficiency, and P_{mip} to the motor input power.

However, calculation of the overall pumping system efficiency requires taking into consideration the frequency converter. The efficiency equation will become:

$$\eta_{\text{FC}} \cdot \eta_m \cdot \eta_{\text{pump}} = \frac{P_{wp}}{P_{in}} \qquad \eta_{\text{sys}} = \eta_{\text{pump}} \cdot \eta_{\text{m}} \cdot \eta_{\text{FC}} \qquad \eta_{\text{sys}} = \frac{Q \cdot \rho \cdot g \cdot H}{P_{\text{in}}} \tag{1.10}$$

Where η_{FC} refers to efficiency of the frequency converter, P_{in} to the electric power at the input of the system, η_m to the efficiency of motor and η_{sys} to the efficiency of pumping set containing frequency converter, motor and pump.

Equation (1.8) includes the losses of the motor and the efficiency of the frequency converter (Yang, 2010). It shows how efficiently pumping system transports the liquid at given process requirements and system conditions.

On the whole, Eqs (1.8) – (1.10) show that the overall efficiency of a pumping system is a combination of efficiencies of the frequency converter, the motor, and the pump. Each component of the pumping system influences the overall efficiency. Thus, in order to improve the efficiency of the pumping system, efficiencies of all the components must be reviewed in complex.

For example, if the pumping system consists of the components having quite good rated efficiency characteristics in their ranges – a frequency converter of 95 %, an asynchronous motor of 97 % and a centrifugal pump of 70 % – the maximal overall efficiency of the pumping system would be 64 %. Nevertheless, as these efficiencies change dependently on the loading and speed conditions, this value may appear as low as 50 or even 20 % at the inappropriate control.

Oversizing the pumps is typically expressed in introducing the safety margins for pump capacitances into the design. These safety margins lead to an increase of the physical parameters of the pumps. The decisions that lead to oversizing can be taken on different stages of the design. It can be a safety margin in order to prepare the spare capacity for a future demand growth or in order to prepare for future obsolescence of the piping and pumping systems. Also, the capacity of the system can become excessive when the consumption decreases by natural reasons; if, for example, some consuming systems or consumers become disconnected from the supply network.

There can be two types of additional costs from oversizing. First, an immediate cost increase may be caused by larger components, hence, a more expensive project design and physical implementation. The second type of additional costs will occur on the operation and maintenance stages of pump service life. It is caused by operation in low efficiency regions at reduced speed or by throttling in order to limit the supply for satisfying the demand at a desired level (Volk, 2005). Because of that, oversized pumps often run in low efficiency regions of operation.

In order to prolong the service life and make operation efficient, a pump must be run as close as possible to the BEP (ANSI/HI, 1997). Mechanical, hydraulic and thermal losses tend to their minimal values at the BEP. In case the pump operates in the regions far from the BEP, its service life can be significantly affected by such phenomena as cavitation, vibration, and shaft deflection. In addition, the process will be harmed by flow recirculation and thermal changes. Thus, operation outside the recommended region causes wastes related to the maintenance costs of the pump, decreased service life, and decreasing reliability resulting from its risk of failure. Therefore, it is important to define the limits of recommendable operating regions for the pump.

In (Ahola, 2009) the definitions for recommendable, allowable and avoidable operating regions were proposed. The regions are mapped onto a flow vs. a head characteristic graph of the pump. Mapping the regions was made in accordance with the recommendations for efficiency limits for the pump and VSD system and for fixed speed pumps (Fig. 1.18).

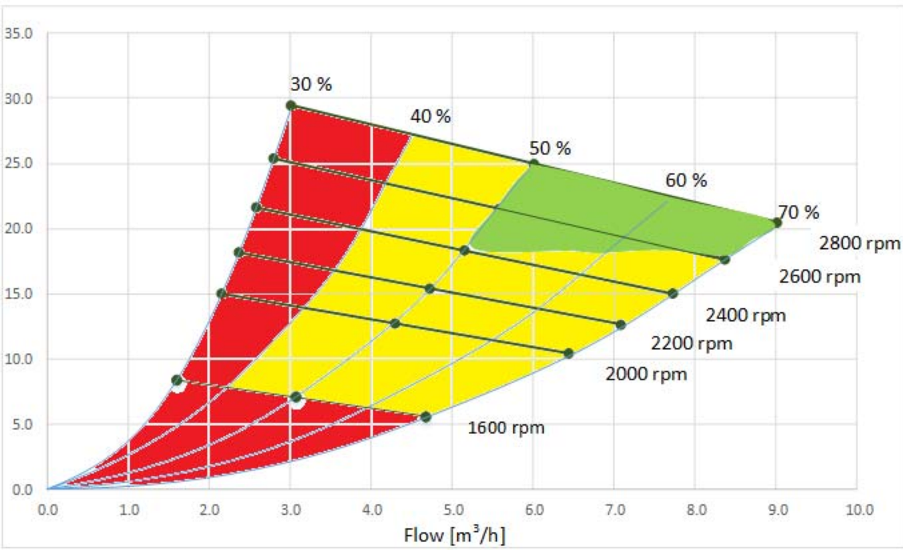


Fig. 1.18 Operating regions mapped on the HQ characteristic graph of Ebara 120/12 pump: recommendable (green), allowable (yellow), and avoidable (red).

The first step in avoiding the operation of the variable-speed pumping system in the low efficiency regions is raising the awareness. The operator should have appropriate feedback from the system and indication of current parameters. Modern frequency converters provide a wide variety of current parameters of the modulation process as well as information acquired from connected sensors or calculated through sensorless methods or estimation techniques.

Some of these methods are based on current signals collected from the sensors installed on the nodes of the pumping system. Others use predefined data entered to the control unit in the form of lookup tables. Typically, these lookup tables provide data concerning pump or motor performance characteristics.

The second step of avoiding the operation outside recommendable region is well-timed reaction in order to return the system into the appropriate state. In the case of multi-pump variable-speed pumping system, the speed of one or several pumps can be adjusted or the number of running pumps can be changed.

The PLC or a control unit of the frequency converter can provide sufficiently powerful data processing capacities for monitoring the state of the pumping system and adjusting its state using traditional, sensorless, estimation or prediction control methods.

The model-based method utilizing prediction algorithms is presented in this work. The method can be implemented using the typical pump system control equipment such as programmable units of frequency converters or PLC. Both control unit types are quite common in the pumping systems. They provide essential process control and hence, additional monitoring or control algorithms can be implemented with no need to purchase specific industrial control hardware.

1.4.4 Resume

1. All the pumping efficiency components, such as frequency converter, motor, and pump itself, were analysed in this chapter. As a result, different power losses were separated.

2. Based on the detailed study of Ebara 3UB-65-200 centrifugal pump, the change of the pumping efficiency with the flow and the speed was shown. Along with that, the distribution of losses in an induction motor and a frequency converter was explored.

3. Current problems and possible solutions were formulated. Using mapped operation regions, definitions for recommendable, allowable and avoidable operating areas were proposed.

1.5 Summary of Chapter 1

1. Resulting from the position of the centrifugal pumps as consumers in the energy market worldwide, the topicality of the thesis was formulated and the main tasks were stated. It was shown how most of the problems in the field of pumping could solved by providing the variable productivity through the variable-speed pumping.

2. Indepth study of the pump characteristics and the system curves showed the need in the BEP as the main force in the pumping process enhancement and its efficiency increase.

3. Analysis of the pumping drives presented showed how to increase the pump productivity using their specific features, VSDs characteristics, and intellectual control. As a result of the comparison of the VSDs from different companies, their benefits and drawbacks were underlined.

4. Study of the pumping methods and control topologies enabled the main thesis objective to be formulated as the solution to the pump energy efficiency problem. The model-based approach was selected as a tool for problem solving. Here, focus was on the predictive control topology and the estimation of the future rotational speed of the pump needed to supply the required flow or pressure.

5. All the pumping efficiency components were analysed, power losses were separated, their distribution of losses in a pump, an induction motor, and a frequency converter was explored. Using the mapped operation regions, the definitions for recommendable, allowable and avoidable operating areas were proposed.

CHAPTER 2. RESOURCES FOR PUMPING STUDY

One of the main goals of this research was to work out a methodology for predicting the location of the working point of the pumping system. Knowing the location of the current and future working point would help in decision making to take actions under changing pumping conditions. The methodology is required to locate the working point of the pumping system and to map it on the performance characteristic pane. To use the selected methods, it is necessary to collect the values of main parameters of the pumping system and to apply the predictive algorithms for effective decision making in the conditions of pumping parameter changes (i.e. increasing the number of working pumps in the system). To verify the methodology and justify its use, a series of model simulations and physical experiments were conducted.

The simulation was based on a set of the computer models of the pumping system operated in different regimes. The core model is built in the Matlab environment using Simulink. The modelling environment includes such subsystems as the Simulink model of a centrifugal pump, lookup tables providing the mathematical models of system resistance, and program blocks for locating the current and future working points of the pump. Real data concerning the power consumption and power losses of asynchronous motors and frequency converters were used for energy consumption and efficiency calculations. These data were generated by the Drive Size software tool from ABB (DriveSize, 2016). The database was integrated into the Simulink model as a lookup table.

Input parameters of the mathematical model were set in compliance with the physical parameters of the test stand used in the physical experiments. Also, internal parameters of all the components of the Simulink model (such as the centrifugal pump or the ball valve) were taken from the datasheet of real equipment.

The model of the pumping system serves as a simulation of the environment of the pump in the pumping system. Its main purpose is to verify the correctness of the algorithm which calculates the current and future locations of the working point and the efficiency.

Series of experiments were conducted in order to verify the benefits of the proposed control strategy. The purpose was to simulate situations in which predictive change of the pumping speed or the number of working pumps is needed to continue supply of the demand. In these situations, the electrical and hydraulic parameters of the pumping systems under various control techniques were compared. The control techniques included traditional speed control, efficiency control through the prediction of the working point location, and the efficiency control by selecting the number of working pumps required from the lookup tables.

The experiments were carried out on a test stand that emulates a pumping system. The stand contains five centrifugal pumps run by variable-speed drives, a pipe system, and a water tank. The test stand enables emulation of varying consumption, measuring the real electrical and hydraulic parameters of pumping, and logging the measured parameters. Experiments included the estimation of energy consumption or demand in order to make the working point of the pump travel along the pump characteristic pane.

2.1 Matlab Model for Pumping Simulation Study

2.1.1 Model description

The new model developed in this work is intended to help find new solutions and verify the correctness of the following assumptions:

1. Pressure at the outlet of the pump can be estimated based on an current flow, the Bernoulli principle, and other parameters of the pumping system.
2. Density of the pumped liquid can be estimated based on a current flow, pressure difference, the Bernoulli principle, and other parameters of the pumping system.
3. Efficiency of the working pumping system can be derived from the input power and hydraulic power. The hydraulic power can be calculated based on a lookup table derived from the pump HQ characteristic, current speed of the pump, and other parameters of the system. Real input power is derived from the power-speed characteristic of the variable-speed drive running the pump.
4. From the point of view of energy efficiency, it is more beneficial to run one pump at the speed that exceeds the rated than to run it at lower speeds.

Modelling was used to prove the technological solvency of the proposed methods of estimation and sensorless calculation. The results of modelling were compared with those from the experiments based on the real test stand consisting of a set of centrifugal variable-speed pumps.

The classification of modelling types from (Peierls, 1980) was used. A model type of capabilities demonstration was selected, which is supposed to demonstrate that the modelled method or system complies with the principles of a real system, phenomenon, or a process. Models of that type are used to confirm that the selected methods of implementation and hypotheses are valid according to the comparison of the results of model simulation and those from the experiment.

Pressure is one of the key parameters of industrial processes and liquid transportation systems. Pressure estimated from the natural parameters of the process can add extra redundancy to the pumping system. Parameters needed for the pressure calculation may be obtained from the frequency converter(Gevorkov, 2015). Connection between the pressure and other parameters of the process is provided by the Bernoulli principle:

$$H_T = \frac{v^2}{2g} + z + \frac{p}{g\rho} \tag{2.1}$$

$$v = \frac{Q}{A} \tag{2.2}$$

Here, H_T is the total head, p is the pressure, v is the liquid velocity in a pipeline, A – cross-sectional area of the pipeline, z – elevation of the measuring point.

Parameters of (2.1) and (2.2) can be obtained in the following way.

The flow rate must be taken from the flowmeter. The liquid density is considered as a constant, since typically conditions of a liquid remain unchanged during the process (Bakman, 2013). Naturally, wastewater applications need special consideration and they will be covered further. Elevation is considered as a constant; it depends on the physical parameters of the pipeline. The total head can be derived from the HQ characteristic of the pump at the current pump speed. The current speed is obtained from the frequency converter. In the model, it is considered as an input to the system.

The purpose of the discussed part of the model is to verify that the pressure at the outlet of the pump can be derived from the listed parameters. The following simplifications were assumed during the modelling:

- Flow rate is laminar.
- Pipeline is straight and does not represent any resistance.
- Friction in the pipeline, viscosity, bulk modulus, and vapour pressure were also neglected because of their insignificant influence under the given conditions.

The developed model of the pumping system shown in Fig. 2.1 was implemented in Simulink version 8.4 (R2014b). The pump was represented by the *Centrifugal Pump block*, a solution of which exists in SimHydraulics toolbox of Simulink. *Hydraulic flow rate sensor* was also represented by an element of SimHydraulics toolbox. It provides data for calculating the current head. The current head was then substituted into (2.1) to derive the density or pressure differences at the output of the model. The inputs of the model were the current speed of the pump and the opening rate of the valve (which represents the load or the system curve).

The model was used in three modes:

1. Sensorless calculation of the pressure difference. The model represents the system that calculates the pressure difference using the Bernoulli formula. The input parameters include the liquid density.

2. Sensorless calculation of liquid density. The model represents the system that calculates liquid density using the Bernoulli formula. The input parameters include the pressure difference.

3. The model can be used for both modes simultaneously. In this case, the parts for the calculation of both modes work separately, in parallel. Results of the calculation appear on the display indicators.

4. Calculation of the efficiency in the multi-pump multi-drive system.

In the case of sensorless pressure and density calculation, the model is acting due to the following algorithm:

1. Speed reference is defined for the pump.
2. Pump generates flow and pressure.
3. Current flow (Q_i) is indicated by a flowmeter.
4. Flow is used to determine the pump current head (H_i) from the HQ graph.
   - 4.1 HQ curve is adapted for current conditions according to the speed reference of the pump using affinity laws.
   - 4.2 Head is calculated from the HQ graph using the interpolation principle.
5. Bernoulli equation is solved according to the pressure difference or density.

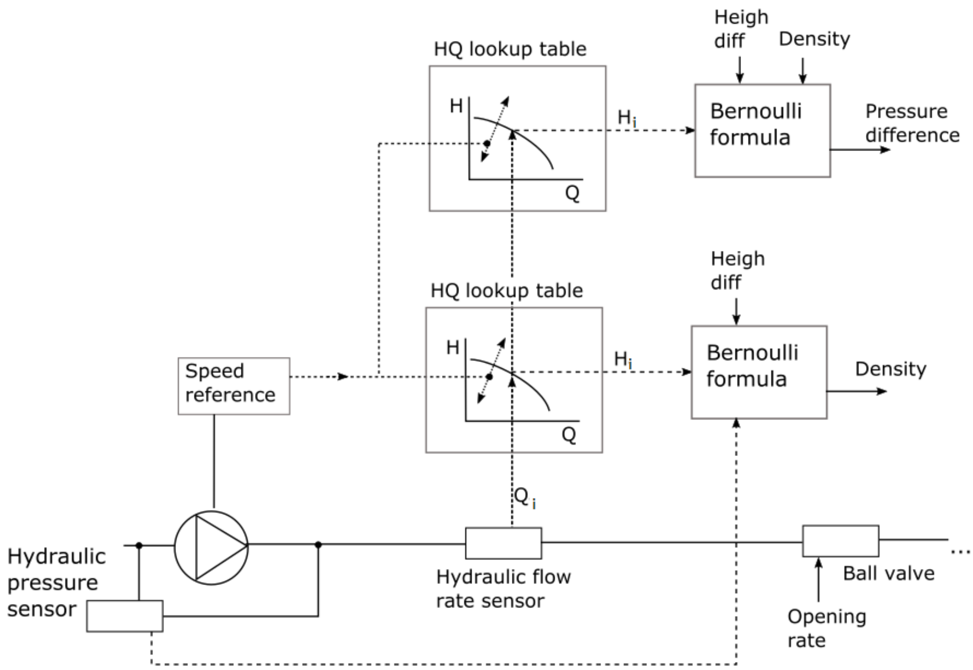


Fig. 2.1 Diagram of the model.

2.1.2 Model composition and specification

The main blocks of the developed model are as follows.

The centrifugal pump block represents a datasheet-based model of any type of a centrifugal pump. The block must be parameterised with values from a datasheet. These values are shown in Table 2.1

Table 2.1. Parameters of the centrifugal pump model

Type of parameterization	By two 1D characteristics: Q-P and H-Q	
Key points of pump Flow vs. head characteristic	[0 50 80 90 110 130 160] [3.01 2.45 2.25 2.15 2 1.8 1.55]	*LPM* Bar
Key points of pump input power vs. flow characteristic	[500 650 800 870 920 1000 1060 1100] [0 40 60 80 100 120 140 160]	W *LPM*
Rated speed of the pump	2760	*RPM*
Density of pumped liquid	1000	kg/m^3
Interpolation method	Cubic	
Extrapolation method	From last 2 points	

The values were taken from the datasheet of Ebara CDX120/12 pump. Pumps of these types were mounted on the test stand used for the experiments.

The block has three ports:

1. Port representing the inlet of pump; the hydraulic conserving port.
2. Port representing the outlet of pump; the hydraulic conserving port.
3. Port used for feeding the speed reference to the pump model; mechanical rotation conserving port, associated with motor-pump driving shaft.

Input for the pump is a constant number, representing the speed reference in rpm. Output is the flow and pressure that can be indicated by *Hydraulic pressure sensor* and *Hydraulic flow sensor* blocks.

The module belongs to the *Lookup tables* library. An approximate one-dimensional function uses the dynamic table. The block calculates an approximation of a curve described by the key points. The key points are defined as two 1-by-n vectors.

The selected lookup method is interpolation. If the input matches one of the key points, the output is the corresponding element in the output vector. If the input does not match any key point, the module implements the linear interpolation between two key points to determine the output. The range is limited by the first and last key points of the input vector (Bakman, 2015).

In both cases, the curve is represented by the input vectors containing key points that are represented by Q and H values. The input is a current flow (Q_i). The output is a current head of the pump (H_i). In practice, a current head is obtained by the control module of the frequency converter, using current flow coming from the flowmeter whereas the current speed is a natural parameter of the pump speed control.

The hydraulic flow rate sensor belongs to the *Hydraulic Sensors* library. The block represents an ideal flowmeter, i.e., some equipment that converts a volumetric flow through a hydraulic pipeline into a control signal according to this flow rate. The sensor is ideal since it does not take into account friction, inertia, pressure loss, delays, and other features.

The hydraulic pressure sensor belongs to the *Hydraulic Sensors* library as well. The block serves like an ideal hydraulic pressure sensor. This piece of equipment converts the difference of hydraulic pressures measured between two points into a control signal according to these pressures. The module is ideal since it does not take into account friction, inertia, pressure loss, delays, and other features.

The ball valve belongs to the *Flow Control Valves* library. The block represents a ball valve implemented by a spherical ball and a round sharp-edged orifice. The liquid flow through the valve is proportional to the orifice opening area and the pressure difference across the device. The opening rate of the device is defined via the input port. The liquid inertia is not taken into consideration. The flow passage area is taken as equal to the side surface of the frustum.

2.1.3 Pumping efficiency estimation via simulation

As it is stated in (Ahonen, 2011), the efficiency of the system containing a motor and a frequency converter can be expressed as relation of hydraulic power to pump input power, as it is shown in (1.10). Although indicating only the efficiency from the power supply line to the water, the system efficiency is frequently referred to as the total efficiency since it shows the total efficiency of the pump drive train. The power supply-to-water system efficiency can also be expressed as multiplication of efficiencies of all components of pump drive train.

At the same time, the effectiveness of a pumping setup can be expressed using the specific energy consumption, which describes the energy per pumped volume. The energy consumption is can be expressed as

$$E_s = \frac{P_{in} \cdot t}{V} = \frac{P_{in}}{Q} = \frac{\rho \cdot g \cdot H}{\eta_{sys}} \tag{2.3}$$

where E_s is energy consumption, P_{in} is the input power of the full setup, V is the pumped volume, and t is time.

In systems with the static head, the minimum value of energy is mainly dependent on the coefficients ρ and g and the amount of the static head. The energy consumption in the closed-loop systems or systems without static head mainly depends on the losses in the piping system and the efficiency of the system (Ahonen, 2011).

Thus, the efficiency of the pumping system can be calculated using natural parameters of the process that can be obtained from the frequency converter. On the frequency converter side, these parameters are obtained from external sensors (like current flow) or calculated (like head, from the lookup table representing the HQ characteristic of a pump). The motor power can be obtained from the frequency converter as a real time parameter of the modulation. The total consumed power of the setup can be calculated by summarising the power at the input of the pump and the power losses at the motor and frequency converter (ABB ACQ, 2015). The power losses can be calculated using the model of the VSD. The simplified model can be expressed as a lookup table of power losses vs. drive speed and implemented in the control module of the frequency converter or PLC.

Once the location of the working point is known, the efficiency of the pump setup can be predicted for pumping applications with a constant system curve (Volk, 2005).

The ratio of the predicted pumping system efficiency to the current efficiency is a valuable criterion for adjustments in the number of working pumps in the multi-pump system driven by the frequency converters. The adjustment can take place when the predicted efficiency after starting/stopping of the auxiliary pumps is higher than the current one.

The discussed model is shown in Fig. 2.2 and its user interfaces are shown in Figs. 2.3 and 2.4. Some experimental diagrams are shown in Figs. 2.5 and 2.6.

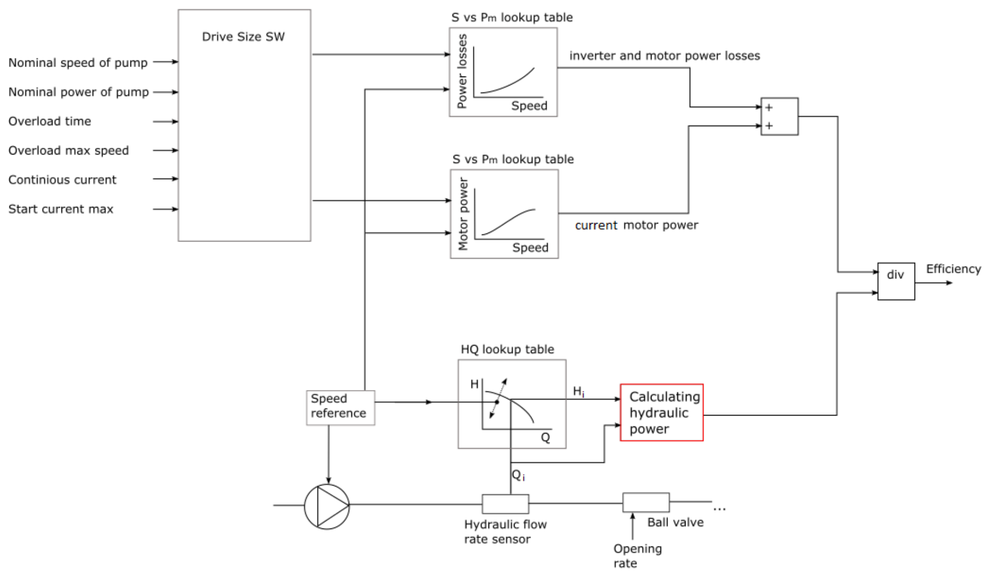


Fig. 2.2 Diagram of a model providing efficiency calculations.

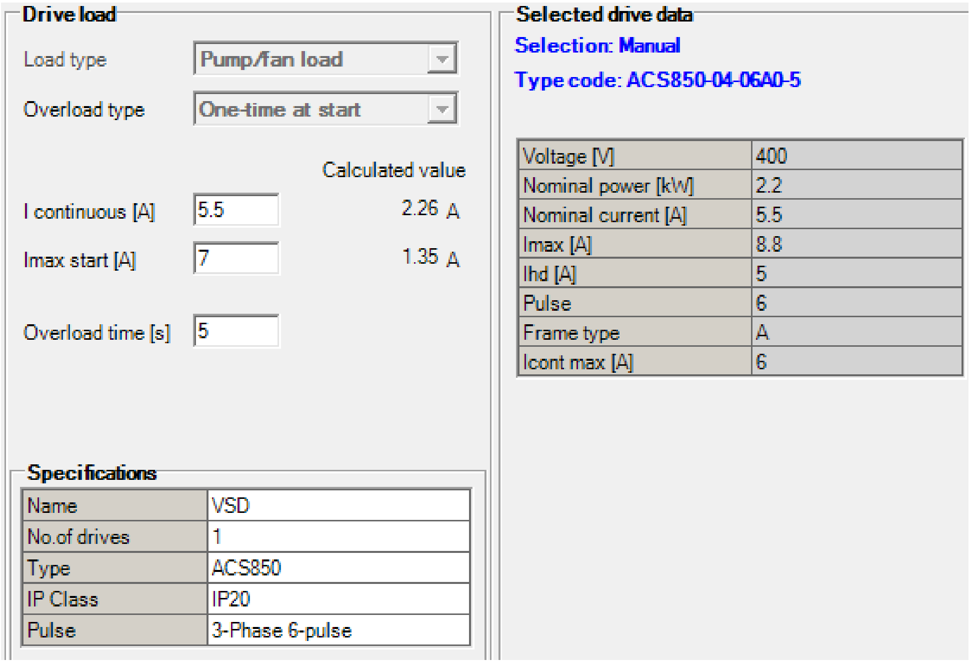


Fig. 2.3. The model of a frequency converter in ABB DriveSize utility.

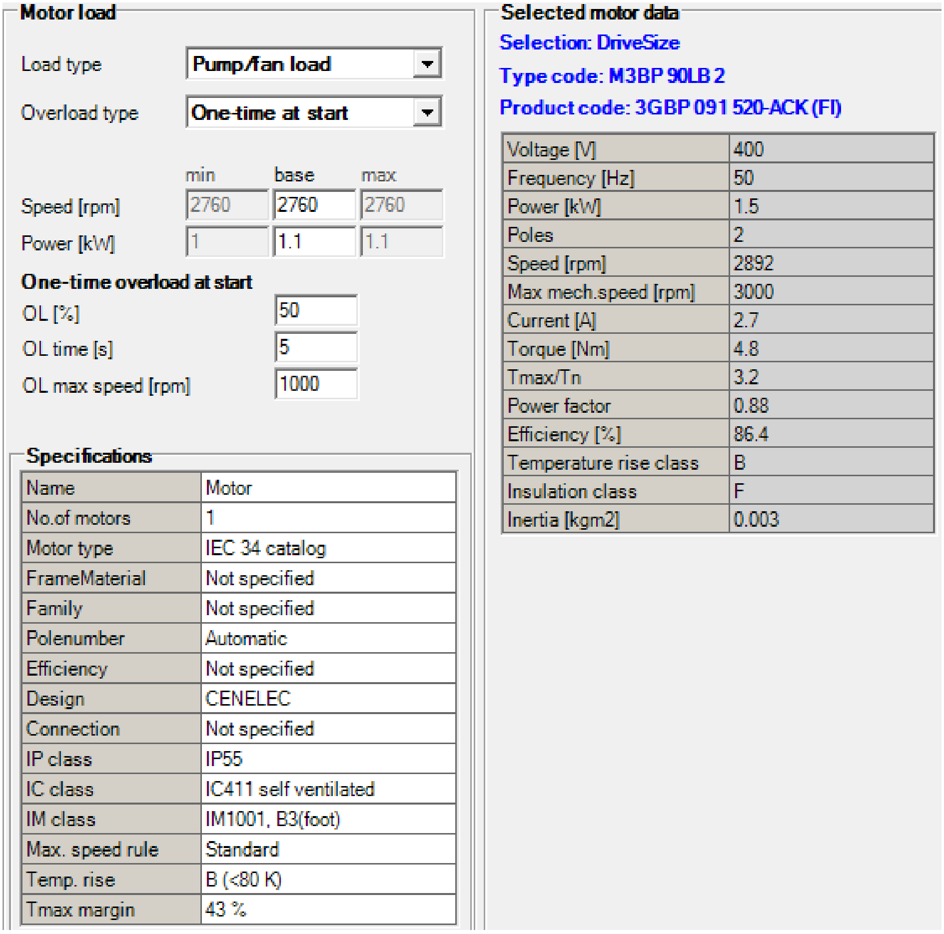


Fig. 2.4 The model of a motor in ABB DriveSize utility.

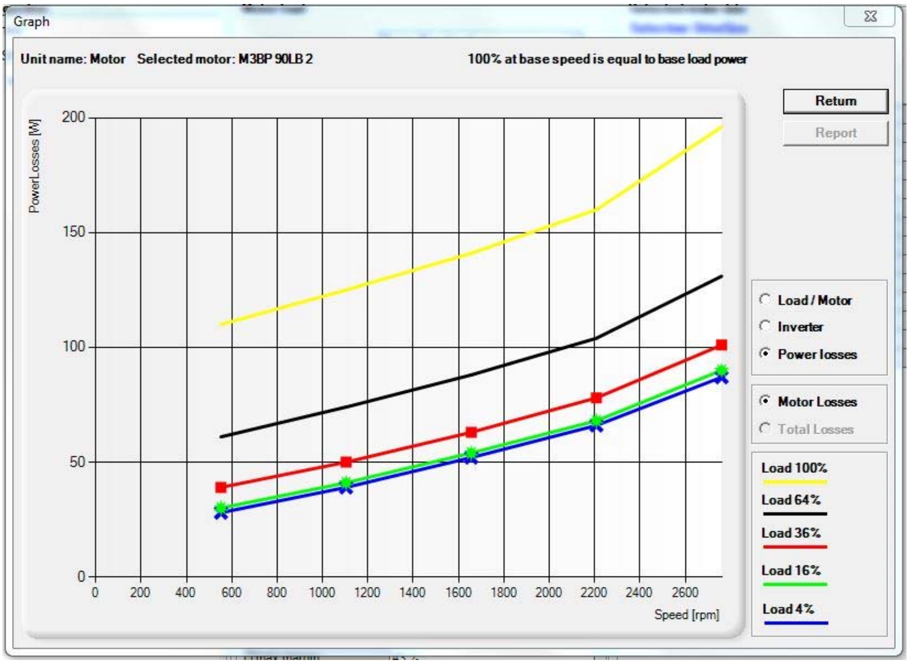


Fig. 2.5 Characteristics for obtaining the power losses of VSD. The graph is generated by the DriveSize tool.

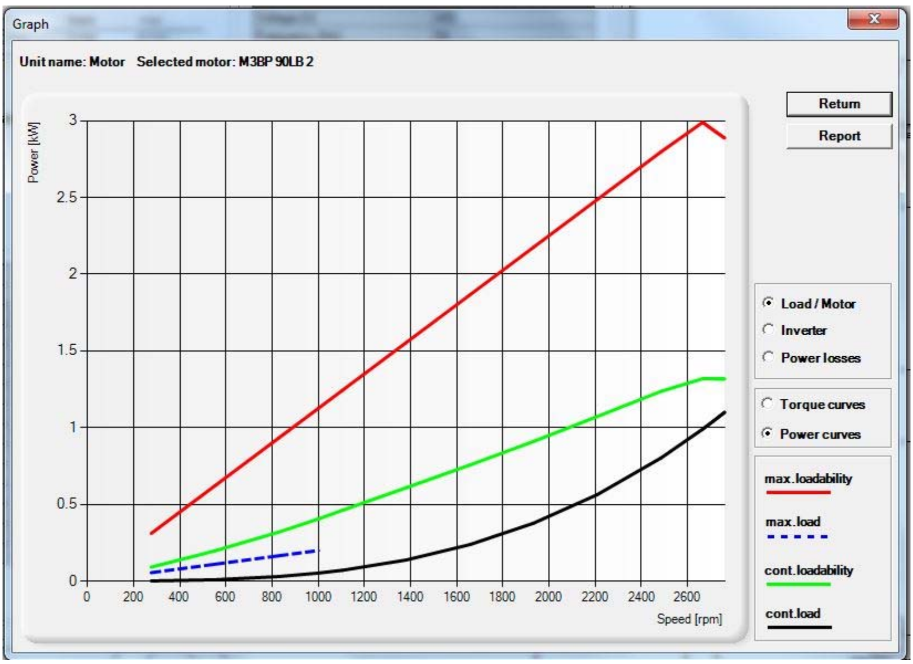


Fig. 2.6 Characteristics for obtaining the motor current power. The graph is generated by the DriveSize tool.

2.1.4 Resume

1. In this work, a new computer model of the pumping process was developed to help find new solutions and verify the correctness of the proposed methods, estimations, and sensorless calculations. The results of modelling were further compared with the results from the experiments obtained using the real test stand.

2. The designed model involves both the mathematical description of the studied process and the data obtained experimentally from sensors and databases.

3. The model is intended for usage in three different pumping modes: sensorless pressure, head, flow, and power; sensorless calculation of liquid density; calculation of the efficiency in the multi-pump multi-drive system. These three modes may be implemented jointly.

4. The most important feature of the model is that it involves real data from the manufacture databases. Particularly, the motor drive and converter losses are obtained from the ABB DriveSize toolbox depending on the current mode of the system operation.

2.2 Multi-Pump Experimental Setup

2.2.1 Experimental setup composition and specification

An experimental setup (Fig. 2.7) was designed to integrate testing of the embedded pump control software of frequency converters. The main purpose of the setup was to provide the emulation of the pumping system, its surrounding infrastructure, and to enable variation of the physical parameters of the pipeline and load. The test stand contains measuring and control devices and permits real-time process monitoring. The tested software involves pump VSD control programs containing the functionalities that are essential and useful for pumping applications. The software includes algorithms for monitoring and adjusting the work of a single pump or a group of pumps. Adjustment suggests various methods of productivity regulation, such as PID, bypass, and increase or decrease of the number of working pumps (Vodovozov, 2015), (Gevorkov, 2015). The purpose of integration testing is to verify the correctness of use of the algorithms and smooth work and interaction of software components. The test stand is intended to provide a possibility to emulate various situations and phenomena typical for pumping applications and systems. It was required to emulate challenging situations in order to identify the weaknesses of the control methods or malfunctions of the algorithms.

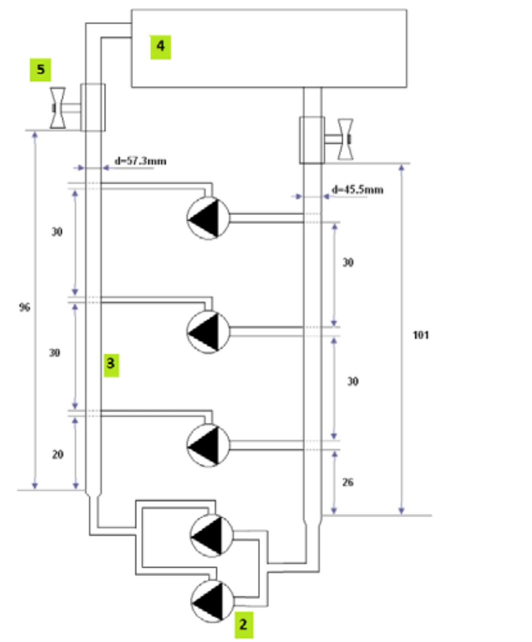
Diagram	Photo
	

Fig. 2.7 Experimental setup: 1 – frequency converter; 2 – centrifugal pump; 3 – pipeline; 4 – water tank; 5 - main valve; 6 – digital and analogue pressure sensors; 7 – switches for switching between the variable-speed and direct power supply modes.

The test setup contains five centrifugal pumps connected in a parallel network. Parallel connection enables emulation of a pressure boosting pumping system. All the pumps were identical. Each pump was equipped with a separate frequency converter. Specific switching circuitry enabled connection of each pump directly to the supply power line or to the corresponding frequency converter. In this way, the pumping systems consisting of variable-speed pumps and non-variable-speed pumps can be emulated. Also, there is a possibility to emulate mixed type pumping systems in which several pumps are connected directly to the supply power line, and a single pump is run by a frequency converter to provide smart productivity adjustment. Table 2.2 presents physical and electrical data of the pumps and frequency converters.

Table 2.2. Parameters of the centrifugal pump model

Data of pump motors		
Rated power	1.1	kW
Rated voltage	400	V
Rated current	4.5	A
Rated speed	2760	rpm
Number of poles	2	
Power rating	0.5 ÷ 2.5	
Protection degree	IP55	
Data of pumps		
Impeller	Closed centrifugal type	
Maximal flow	9.6	m^3/h
Maximal head	19.5	m

The test setup emulated successfully a typical boosting pumping station the purpose of which is providing the water supply with constant parameters, such as flow or pressure. Thus, keeping the constant value of flow or pressure is the main challenge.

The test stand is pressure oriented, since it is equipped with an analogue electronic pressure sensor, though a flowmeter is not installed there. The pressure sensor is connected to the analogue inputs of control boards of all frequency converters. It provides the real-time monitoring of pressure in bars. All the pumps are equipped with ABB ACQ810 frequency converters. Table 2.3 contains technical data of the converters.

Table 2.3. Technical data of converters

Rated power	1.1	kW
Rated voltage	400	V
Rated current	3	A
Maximum current	4.4	A

Each frequency converter is equipped with a control board running the modulation process. It can also be used as a powerful process controller. The board is built based on the Texas Instruments 2812 digital signal processor. It includes a circuitry for digital and analogue input/output signals conditioning, RS-232 and RS-485 communication adapters, and flash memory.

The main functions of the control board are as follows:

1. Control of the modulation process, running the speed control, and providing the correct reference to the power unit.
2. Control of the industrial process, such as pumping, and setting the correct reference for speed control functionality. This function keeps the

process value as close to the setpoint as possible. In case the process value is pressure, the purpose of the functionality control is to generate the speed reference, which would provide the stable pressure equal to the setpoint value (ABB, 2015). Generation of the speed reference is implemented based on the readings of the pressure sensor. The most typical speed controller in pumping applications is a PID controller (Gevorkov and Vodovozov, 2015). At that, the inputs of a PID controller are the current values (readings of pressure sensor) and the pressure setpoint, and the output is the speed reference, which is used for speed generation by speed control functionality.

The firmware of the converters keeps a wide range of parameters for tuning up and monitoring of the process. Parameters of the frequency converter enable tuning up of speed control and the modulation process, tuning up of the control tools that run the process of pumping, monitoring the current values of modulation and speed control, and monitoring the current process values.

All of these parameters are accessible through the following interfaces:

1. Monitoring and tuning up software, the "DriveStudio". This software enables real time monitoring and logging for all the parameters of the process control, speed reference generation, and modulation. The software is running on the PC and is connected to the frequency converter through the OPC server and RS-232 interface.
2. Various fieldbus protocols that enable acquisition of all the parameters from the frequency converter. The parameters can be read continuously, by a fieldbus master, and PLC. The control board of the frequency converter supports embedded Modbus RTU communication functionality. Also, various fieldbus adapters can be mounted onto the control board. ACQ810 supports such fieldbus protocols as Profiubus, Device NET, CanOpen. In this work, Profibus was used to acquire relevant parameters from the converter.

The fieldbus communication enables provision of the relevant parameters of the process and modulation to custom-made control algorithms on the PLC platform. The fieldbus communication also enables sending references and commands to a frequency converter from the custom-made control algorithms running on PLC.

The converter control board interfaces and connections are shown in Fig. 2.8.

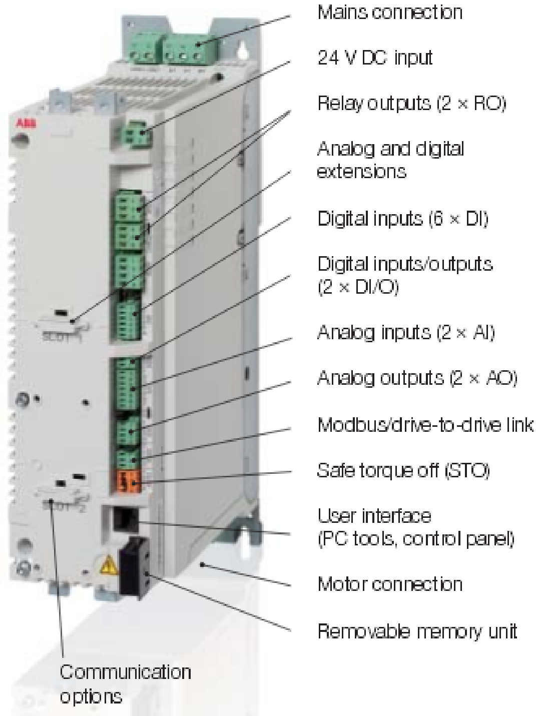


Fig. 2.8 Interface connections of ACQ810.

The test stand can emulate the performance of a typical pumping station. For example, the optional configuration of the test stand, which enables simulation of simple boosting pumping station work of three pumps, enables increasing of the flow rate and keeping the constant pressure. Each pump is run by a separate frequency converter. This enables adjusting the productivity through varying the speed of pumps. A valve makes it possible to emulate the variation of demand. Opening the valve emulates a growth of demand and a need in a greater flow from the pumps in order to satisfy it. The test stand is implemented as a closed loop system. Thus, the pipeline starts from a water tank and comes to the same tank. A water tank was installed on top of the stand. It enables considering the static head during the simulations. Due to the closed-loop architecture, the pressure at the inlet of the pipeline is always constant.

The control program, which runs the whole pumping process, is working inside the control board of one of the converters. The program is a firmware of ACQ810, which contains pumping specific functionalities. Hence, the pump that is run by the frequency converter is capable of working at variable speeds. In industry, it is typically called the “master pump”.

The readings of the pressure sensor enter the control program through an analogue input. These readings are used for productivity adjustment. In case the consumption increases (need in a greater flow), the pump must increase its speed in order to provide a greater flow and to satisfy the demand. When the master pump working at the maximal speed is incapable of satisfying the demand, the auxiliary pumps are started to help to keep the needed productivity.

Auxiliary pumps can be started or stopped by the control program of the master pump depending on the required productivity. These pumps are the "follower pumps". The control programs of the follower converters do not run the pumping process but follow the speed reference, start, and stop commands of the master. The commands and reference are sent via RS-485 communication from the converter of the master pump to the converters of the follower pumps.

There are two options of the speed reference selection for the follower pumps.

1. Follower can run with constant speed, as defined in its control program.
2. Follower can follow the master's speed reference, i.e. run with the same speed as the master.

The settings for the speed reference selection are made in the control program of each converter individually.

The statuses of the master and the followers can interchange, providing redundancy. Each follower converter can become a master if the original master becomes invalid for some reason.

Speed adjustment of pumps is implemented by a software PID controller, which is running inside the control board of the master converter. The feedback for PID from the system is the pressure at the outlet of parallel pumps set. The value of the pressure is entered to the PID through the analogue input of the converter, cascade of signal conditioning, and scaling program modules. The setpoint value can be assigned manually through one of interface options. It can be a user panel on top of the converter, analogue input, or PC connected via the RS-232 port. In the case of divergence between the feedback and the setpoint values, the PID generates the speed reference, which contains the corrective value in order to compensate the difference. The generated speed reference is supplied by the speed control functionality, which is in charge of the modulation process.

Data exchange between the master and follower converters is run by their firmware. The communication is implemented via the Modbus RTU protocol. The master converter of the pumping system is also a master of the Modbus network. The communication mean is a twisted pair cable. The data exchange is cyclic. During the exchange, the master reads statuses of the follower converters and sends start or stop commands and speed references. In the case of the master converter failure, one of the follower converters becomes a new master. Thus, the pumping system would continue its operation on the variable speed principle even in the case of failure of one or several pumps.

The firmware of converters provides some additional pump-related functionalities. It contains the program modules for monitoring the productivity and reacting on its extreme values, monitoring the inlet and outlet pressure, supervising the dry run and stack occurrences, and other functions.

The mechanical ball valve is used to emulate alternations of demand. Pumping system must react on the change of the demand with an increase of the pump speed (or number of working pumps) in the case of its growth or with a reduction of pump speed (or number of working pumps) at its decrease. The high opening rate of the ball valve emulates a high demand and makes the supply increase to enable the process to meet the setpoint. The ball valve is mechanical and is operated manually.

The follower pumps are started when the productivity of the master pump working at the maximum speed is not sufficient to satisfy the demand. Thus, the only criterion for starting the follower pump is the speed of the master pump (ABB, 2015). When the speed of the master pump exceeds the specified limit, the start command to the follower is generated.

When several pumps are working and the demand decreases, the productivity of a group of pumps starts to become excessive. The speed of pumping is falling in order to avoid the generation of the excess pressure. When the speed becomes lower than a specific low-speed limit, the master pump generates the stop command for one of the follower pumps. The command is sent to the follower via the Modbus network.

The low and high limit speeds for starting and stopping the followers are set by the user via one of the converter interface options.

2.2.2 Simulation with ABB DriveStudio toolkit

The DriveStudio toolkit is a software that represents the parameters of the frequency converter graphically on a PC. The values of parameters are taken from the frequency converter via RS-232 communication. Parameters are the values of the modulation process and of the process which is run by the control functionality of the frequency converter. Some parameter values are measured (like some parameters of modulation measured in the power unit or parameters of the process coming via analogue inputs) whereas others are estimated. The parameters are divided into parameter groups according to their meaning.

Parameters can be divided into the following groups:

1. Parameters for tuning up the speed control and modulation process. This group contains such subgroups like “Flux Reference” (parameters for tuning up the flux reference and U/f curve settings), “Motor control” (motor control settings such as performance/noise optimisation, slip gain, voltage reserve and IR compensation), “Voltage ctrl” (overvoltage and undervoltage control settings).
2. Parameters for tuning up the process control. The group contains such subgroups as “Process PID” (configuration of process PID control), “Pump protection” (settings for protecting the pump and pipeline against extremely

high or low pressure), “Pump logic” (settings for adjusting the productivity through varying the number of working pumps in the pumping system), “Setpoint selection” (scaling the setpoint and selection of its source).

3. Parameters for monitoring the values of modulation and speed control. The group contains parameters for observing the values of current and DC bus voltage of the converter, current motor power, estimated temperature of the power unit, various binary statuses reflecting the state of the power unit, control board, and modulation process.
4. Parameters for monitoring the current values of a process. This group contains such parameters as values at digital and analogue inputs (e.g. current pressure coming from the pressure sensor via analogue input), estimated flow, runtime of all pumps in the pumping system, number of running pumps and their state in the network, and binary statuses reflecting the state of the pumping process.

The values of all monitoring parameters are updated cyclically and continuously. The rate of refreshing can be set by the user.

The utility also provides a graphical view of the monitored parameters. The graphs are being represented and refreshed in real time. They can be saved in a csv-like format for further processing or research.

The data can be acquired from several frequency converters simultaneously, which is good for monitoring the work of a group of pumps. Each frequency converter must be connected to a PC via a separate RS-232 adapter.

The utility also enables monitoring data that are sent from the PLC to the frequency converters via various types of fieldbus communications. It is useful when the test runs include running the custom-made control algorithms or monitoring techniques. In these cases, the process control is partly pre-formed by the PLC. Values of parameters of the process, which are estimated by custom-made algorithms, can be sent to DriveStudio via fieldbus interface and represented graphically or numerically against other parameters of the test run.

The toolkit provides user friendly and usable interface for monitoring the test runs and adjusting their parameters.

2.2.3 Introducing the PLC into experimentation and control

PLC, a programmable logic controller, which is designed on basis of a digital signal processor utilizing especially robust industry oriented architecture. It is used for automation of industrial processes and setups, such as pumping systems, climate setups, or machinery on factory manufacturing lines.

. PLCs support large arrays of digital and analogue input and output modules. PLCs are designed for stable and independent operation in harsh conditions and hazardous environment. The main purpose of a PLC is to run the connected actuators or machinery corresponding to input signals and to control program logic. Another purpose is to collect the signals from connected sensors and to represent them to other control equipment via industrial communication networks.

The PLC runs the process by generating the control signals at the digital and analogue outputs. The control logic is defined by a program that is designed on a PC using the programming environment and is loaded into the non-volatile memory of the PLC (Fig. 2.9).

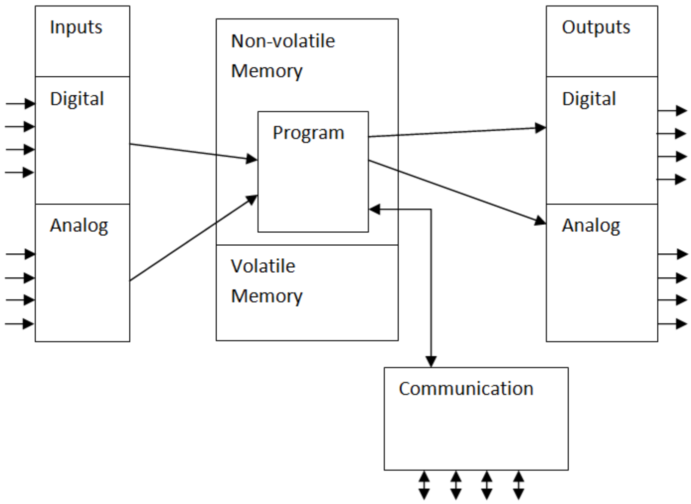


Fig. 2.9 Simplified diagram of the PLC.

The control program runs cyclically inside the PLC memory. Typically, it is possible to assign the call of program functions to various time levels. Normally, the time level ranges are defined between several milliseconds and several hundreds of milliseconds.

The PLC used in this research is an ABB AC500 PM583 controller with DA501 digital and analog inputs/outputs module. The controller is also equipped with a Profibus communication adapter for communicating with a frequency converter.

The control algorithms for AC500 are composed in the CodeSys environment. The environment supports various semantic software composition methods. In this research, functional block diagrams and structure text programs were used for the algorithm. The structure text method utilizes Pascal and C-programming language-similar syntax and logic.

Visualization functionality of the CodeSys enables representation of various parameters and values of the process in the form of simple animated graphics. This visualization is implemented in the CodeSys environment for observation. It enables representation of the signal values, plots, arrays, and tables of data and other visual instruments. The animated graphics act in real time so that it is possible to monitor ongoing processes via elements of visualization.

In this research, values of the pumping process estimated based on the data received from the frequency converter via Profibus were represented on visualization panels in the form of numerical values and plots. For example, the location of the working point could be monitored in real time and online on a plot of HQ characteristic of the pump implemented in the CodeSys visualization panel, as shown in Fig. 2.10.

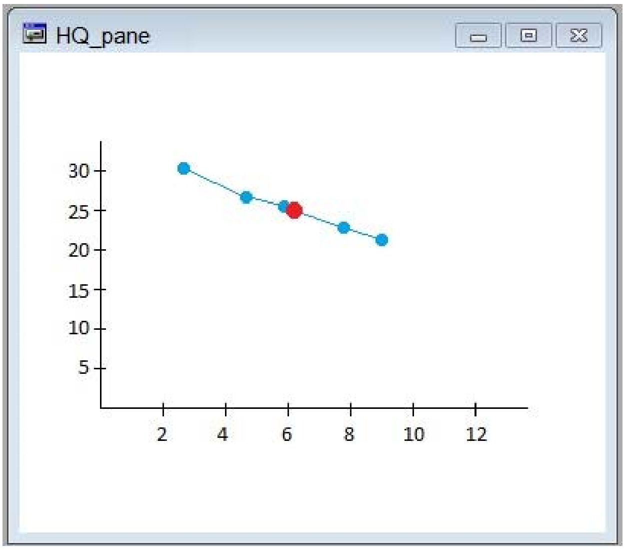


Fig. 2.10 CodeSys visualization panel showing the current location of the working point at 6 m^3/h.

In the given configuration, the PLC needs no data via digital or analogue inputs. The data used by the calculations for the pumping process can be taken from the frequency converter via the Profibus communication. The data from the sensors placed on the test stand are passed to each frequency converter in the pumping system as soon as the converter PID runs the process in traditional applications. Thus, pressure readings are essential for the converter PID. If there is a need in readings for pressure or in the calculations on the PLC side, these readings are taken via Profibus communication from the converter. In fact, the data can be taken from any Profibus enabled converter in the group.

In this configuration, the PLC plays the role of calculating equipment. It collects data from the frequency converters and uses them in estimations and prediction calculations. It is not required to run any circuitry or equipment at the output. The only type of commands it generates is the commands for frequency converters that can be sent via Profibus.

During the calculation of efficiency and data collection to create an efficiency map of the pumping setup, auxiliary measuring equipment was installed on the test stand. An ultrasonic flowmeter PORTAFLOW™ SE, a portable device designed by Micronics was introduced (Fig. 2.11). The flowmeter is intended for use on liquid flows in full pipes. The device utilizes Ultrasonic transit-time “Clamp-On” transducer technology. The readings of this flowmeter are presented in m^3/h. The sensing bodies are attached onto the pipe with no need in their cutting and introducing any new elements to the pipeline. Clamp-on transducers were used to measure the flow, enabling flowing liquid within a closed pipe to be measured without the need for any mechanical circuitry to be inserted through the pipe wall or protruded into the piping system. The meter is controlled by a DSP containing a wide range of application data that enable measuring flow in the pipe of any diameter from 13 mm up to 5000 mm and made from any material.

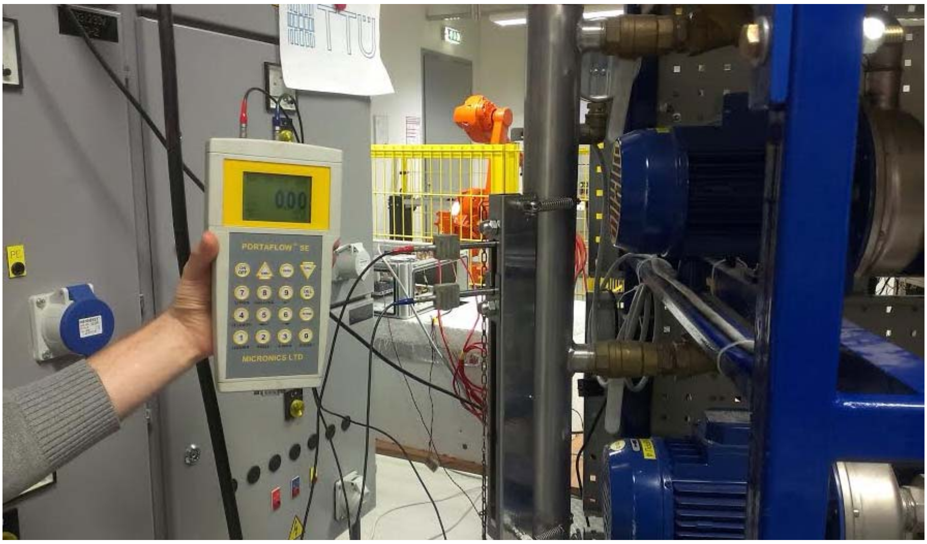


Fig. 2.11. PORTAFLOW™ SE in the test stand.

2.2.4 Resume

1. An original experimental setup was designed to integrate testing of the developed pump control software. The main purpose of the setup is to provide the emulation of the pumping system, its surrounding infrastructure, and to enable variation of the physical parameters of the pipeline and load.

2. The whole equipment used in this stand has real industrial firmware and software. Its converters keep a wide range of parameters for tuning up and monitoring the pumping process, pumping running and stopping, current values of modulation and speed control changes, and process value observation.

3. The simulation approach offered is based on the ABB DriveStudio toolkit. To this aim, four parameter groups were selected.

4. Taking into account the drawbacks of the DriveStudio firmware, the PLC was introduced to enhance the experimentation possibilities.

5. In addition, some auxiliary measuring equipment was installed on the test stand to obtain better results and to tune existing DriveStudio instruments.

2.3 Summary of Chapter 2

1. To design a new pumping management methodology, a solid resource basis is required. For that reason, an original simulation and experimental environment was created in this project.

2. A set of computer models of the pumping system operated in different regimes was prepared. The core model built in the Matlab environment includes such subsystems as the centrifugal pump, system resistance, and program blocks for locating the current and future pumping working points.

3. Real data of power consumption and power losses of asynchronous motors and frequency converters were generated by the Drive Size software tool from ABB. This database was integrated into the Simulink model as a lookup table.

4. The simulation methodology offered was used to verify the proposed control strategy. In these simulations, the electrical and hydraulic parameters of pumping systems at various control techniques were compared. The control techniques include the speed, pressure, flow, power, and efficiency management.

5. An original experimental setup with real industrial firmware and software was designed for testing the new pump control software. The PLC and auxiliary measuring equipment introduced into the setup enhances experimentation possibilities significantly.

Chapter 3. Methodology of High-Efficiency Pumping Management with Predictive Control

Keeping the pumps running in the best efficiency region (BER) of operation is one of the most important challenges of a control system. Major tasks of management include detecting the out-of-BER operation, preventing actions that can move the working point to undesired areas and adjusting the pumping parameters in order to keep operation in recommendable limits. Thanks to the process control architecture of the pumping systems that use PLCs and programmable frequency converters, new algorithms can be implemented on these devices.

This chapter presents proposals for data use from the sensors, internal parameters from a frequency converter, or pump performance data from characteristic graphs for novel control algorithms. To this aim, a methodology of predictive control and efficiency monitoring will be described. Pump operation outside the BER and accompanying phenomena will be reviewed. The proposed algorithm of high efficient pump management at specific speed ranges will be explained in detail. Simulations of the pump system test runs using computer modelling tools and VSD selection software will be presented. Also, experimental test runs on the laboratory setup emulating the pumping system will be introduced.

3.1 Analysis of Pumping Within and Outside the Best Efficiency Regions

3.1.1 Problem statement

As it was stated in (Barringer, 2003), from the viewpoint of efficiency, the most appropriate way to run the centrifugal pump is to keep its operation as close as possible to the BEP. Therefore, the region near the BEP is the preferred or recommendable pumping operating region (PSM, 2008). Performance in this area provides good reliability, efficiency, energy consumption, and long service life of a pump (ANSI/HI, 1997). Such harmful phenomena as hydraulic excitation forces on the impeller and cavitation are less likely to appear in this operating region, which is positively affecting the reliability of a pump (Gülich, 2008), (Nelik, 2005). Once the pump is operating far from the BEP, outside the recommendable region, its efficiency decreases and damaging phenomena such as cavitation and vibration are more likely to appear, reducing the reliability and service life of the pump. Moreover, risk of the phenomena harming the process, in which the pump is used, is growing. These phenomena are temperature rise (which can be critical for some types of processed liquids) and flow recirculation. Figure 3.1 demonstrates the relation between the reliability of the pump and the location of the working point on the flow vs. the head characteristic (Barringer, 2003). The data are relevant for pumps built according to the ISO 5199 standard for use in chemical and process applications.

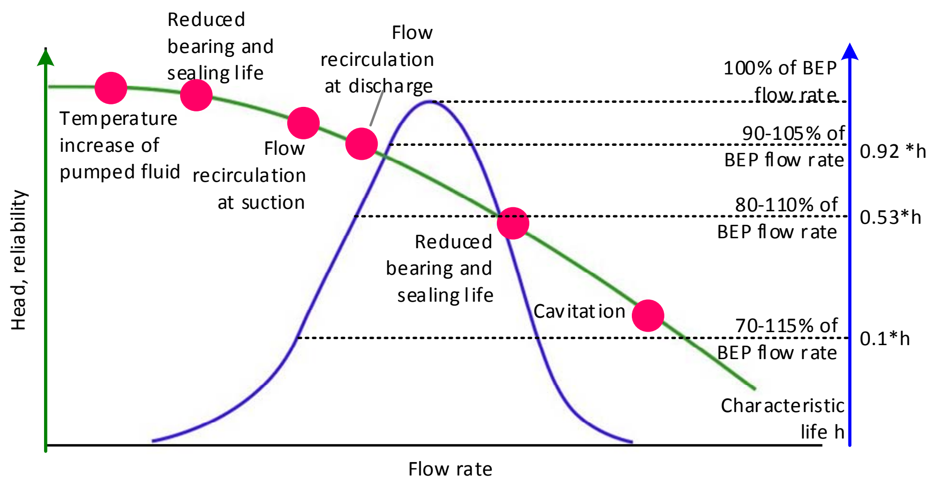


Fig. 3.1 Pump reliability relative to the head and the flow rate. Right axis shows the estimated decrease of the life of a pump. Destructive phenomena caused by operating outside the recommendable region are mapped on the graph..

3.1.2 Related work

A definition of allowable and preferred operation regions was proposed in (Kini, 2010) and (Wiegand, 1960). Those regions were mapped on the flow vs. head pump characteristic such that it was divided into two parts. The service life of a pump is not seriously reduced in a preferred operation region (POR) by vibration and hydraulic forces (Gülich, 2008). It was stated that POR could be defined in the limits of 70 – 120% of flow rate at the BEP for the radial flow centrifugal pumps. However, according to the pump life decrease estimated in Fig. 3.1, in this case, the estimated life of a pump would be reduced to 0.1 of service life. To achieve a lower reduction of service life, pump's operation should be kept in more narrow limits. The allowable operation region (AOR) was defined in a wider range of flow values within which exposure to risk of the service life of a pump is lower. Vibration and noise are also presented in this region and their magnitudes are higher than in the AOR. Bearing life is also reduced in this region. Therefore, the estimated service life of the pump is under the risk of reduction when operating in the AOR. However, running the pump in this region does not cause an immediate destruction of any of its parts.

In (Gülich, 2008) a similar definition of ranges was introduced. However, it is stated that the limits cannot be estimated theoretically because of non-similarity of separate applications. These limits are frequently derived from the experience and should not be taken as a strict definition. To delimit allowable performance areas, the following criteria could be taken into consideration:

• Type of pump and application
• Vibration profile of the pump and the system
• Stability of the HQ curve
• Power consumption of the motor
• Physical parameters of the liquid to be pumped and its temperature
• Risk of cavitation
• Risk of recirculation, including hydraulic excitation forces, noise, cavitation
• Energy costs
• Possible heating of the pumped liquid
• Power class of the pump

The limits of the allowable continuous operation may be defined for those conditions at which the pump can be operated for several thousand hours without damage or excessive wear. For example, the limits of a continuous allowable operation range could be defined such that the efficiency is kept between 80 and 85% of the maximum. Such limitation is justified with respect to the energy consumption. In addition, it tends to eliminate the situation when a pump operates with excessive part-load recirculation (which can occur below Q < 50% of the full Q) or with flow separation (occurring at high flows above the BEP).

In practice, for reasons discussed earlier, many pumps are oversized due to the margins applied on the head or flow on the design stage. As a result, these pumps often work with the sensible part-load recirculation. Considering robustness in design, this usually does not lead to noticeable damage at low or medium circumferential speeds if the pumps are operating within the range of allowed continuous operation.

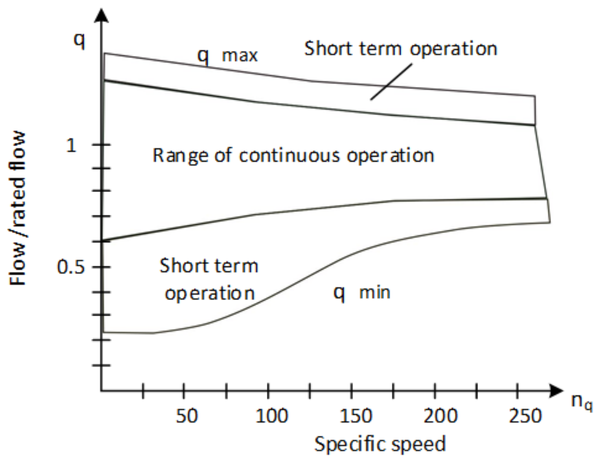


Fig. 3.2 Recommended ranges for continuous and short-term operation regarding the flow (Gülich, 2008).

Figure 3.2 shows the allowed continuous operation range of a pump as a function of the current speed, which is obtained by applying the efficiency criteria which were discussed earlier. This recommendation is relevant mostly for large pumps of 500 to 1000 kW. The higher the power and the head, the greater is a need to operate mainly near the BEP. Again, the recommended range could be modified by a detailed analysis of a given pump and a pumping system taking into consideration the mentioned criteria.

Small pumps, especially those designed to run at low speeds are even operated permanently at low loads (below 50% of full capacity) (Gülich, 2008).

The risk of cavitation at high flow rates is the main reason for the maximum flow rate restriction. However, the general rules for this case can hardly be defined since cavitation occurrence depends on the net positive suction head, suction impeller design, circumferential speed at the impeller inlet, physical properties of a liquid, and other parameters.

It would be correct to accept that the upper limit of the diagram in Fig. 3.2 is mostly relevant to the systems with a low cavitation risk. Because of relatively high robustness of low operating speed pumps, the AOR of these devices is often limited by:

- Stable region of the HQ curve
- Maximum power consumption of the motor
- Impact of cavitation observed at part-load and overload

Short-term operation may be defined as an abnormal condition which typically results in early wear of the pumps, whereas cumulative per year duration, as a rule, does not exceed 100 hours (Gülich, 2008).

The two approaches mentioned provide a general idea of defining the operation regions in which the pump should be operated. As was stated earlier, it depends strongly on the current conditions of the application. Thus, mapping the allowable and recommendable regions is very specific to different applications. Moreover, the mentioned approaches mainly concern mechanical reliability and energy consumption. The use of VSDs might make it more feasible to drive the pump at longer distant locations from the BEP with a lower risk of the physical damage of the pump.

According to these approaches, the limits of operating regions whether allowable or recommendable, must be known. Keeping a pump in these regions would reduce the risk of physical failure or premature wear of equipment. For that, techniques are required to determine the current location of the working point. Moreover, useful indication about the general mechanical reliability of the pump in a short and possibly long term can be acquired. Though setting the limits cannot exclude probability of unexpected fault occurrence, it could help prevent the operation of a pump in regions of poor energy efficiency or high risk of physical failure (Ahonen, 2011). Thus, knowing the current working state, efficiency and location of the pump's working point may provide a powerful monitoring and protecting tool for pumping system management. In the current approach, tools for detection of the mentioned parameters and risky situation are the key factors. Thanks to frequency converters integrated into the control systems, methods are provided to obtain required current parameters without a need to introduce new measuring equipment. As an example, parameters needed for the estimation of current efficiency can be obtained directly from the frequency converter.

3.1.3 Efficiency estimation in the simulation environment

To produce a desired total head H^*, the control system must run an appropriate number of pumps and set their required rotational speeds n^*.

In the traditional speed-oriented approach, the pumping system productivity is defined by the pump rated speed n_r. Typically, applications are programmed in such a way that when the current speed n_i for providing the head H^* at the requested level overcomes the rated one, the next Z pumps are being started one by one. Their speed reference n^* in this case may be found with the help of affinity laws (1.5-1.7) describing the head H, pressure p, flow Q, power P, and torque T changes on a system curve s (index r indicates rated parameter, and index i indicates current value of parameter):

$$\frac{Q_{is}}{Q_{rs}} = \frac{n_i}{n_r}, \tag{3.1}$$

$$\frac{p_{is}}{p_{rs}} = \left(\frac{n_i}{n_r}\right)^2, \tag{3.2}$$

$$\frac{H_{is}}{H_{rs}} = \left(\frac{n_i}{n_r}\right)^2. \tag{3.3}$$

$$\frac{P_{is}}{P_{rs}} = \left(\frac{n_i}{n_r}\right)^3, \tag{3.4}$$

$$\frac{T_{is}}{T_{rs}} = \left(\frac{n_i}{n_r}\right)^2. \tag{3.5}$$

as follows:

$$n^* = n_r\sqrt{\frac{H^*}{H_{rs}}} \tag{3.6}$$

$$n^* = n_r\sqrt{\frac{p^*}{p_{rs}}}. \tag{3.7}$$

Here, p^* represents the pump input power at desired speed n^*.

As the first approximation, it is favourable to have a matrix representation of the flow rate and total head (or pressure) values at the rated speed [H_{rs}, Q_{rs}] in the form of a lookup table stored in the memory of the control unit. Affinity transformations enable adoption of rated values from the table to current values corresponding to the state of the process. The data for populating the lookup table should be taken from essential characteristic graphs of the pump. The same graphs can be used for deriving the needed characteristic for the multi-pump system. The appropriate transformations should be applied to characteristic curves depending on the topology of the pump connections.

To illustrate, these data from Fig. 1.2 are displayed in Table 3.1 for a single pump ($Z = 1$).

Table 3.1. Lookup table of the rated pump characteristic.

System state	**I**	**II**	**III**	**IV**
Total head H_{rs}, m	28	26	25	23
Flow rate Q_{rs}, m^3/h	4.32	5.45	6.21	7.12

Using the multi-pump system characteristics, the same tables may be generated for each Z number. Nevertheless, the table restriction and the need in the data approximation between the table cells, which yields in reduces resolution of estimations, is the drawback of the lookup approach.

Another methodology is offered here where the efficiency criterion is imposed to calculate the required number of pumps.

Every working point may be reached using different number of pumps, from 1 to Z. As follows from (2.5), the pumping efficiency is the relation between the hydraulic power developed by the pump and the electrical power consumed by the pump:

$$\eta = \frac{P_{WP}}{P_{in}} = g\rho HQ\eta_{pump}\eta_{motor}\eta_{FC} \tag{3.8}$$

where η_{FC} refers to efficiency of frequency inverter, P_{WP} to the output power of pump and $\eta_{pump} = \frac{P_{WP}}{P_{pump\ in}}$ is the pump efficiency (2.4), which can be obtained from the pump documentation and $\eta_{motor}\eta_{FC} = \frac{P_{pump}}{P_{in}}$ is the efficiency of the pump motor drive.

Consumed power can be derived from the drive documentation where it is represented in the form of graphs and tables. In addition, it can be derived from the drive manufacturer's software for drive selection and emulation. Many manufacturers provide such software for sizing the drives on the system design stage.

Power-torque characteristic is represented by inverted parabolic curve. Its current shape and location is also dependent on speed. Therefore, this characteristic can be described by the polynomial of second order, a function of torque and speed. Usually, the losses ΔP have a non-linear dependence on the motor speed and torque:

$$P_{cons}(T,n) = P + \Delta P_0(n) - C_{\Delta 1}(n)T - C_{\Delta 2}(n)T^2 \tag{3.9}$$

where C are the friction factors.

The efficiency distribution shown in Fig. 3.3 was obtained applying this method. Here, the rated performance characteristics shown in Fig. 1.2 at 3000 rpm were replaced with the performance curves similar to those obtained at the increased speeds of 3750 rpm. The curved areas indicate constant efficiency regions.

As it follows from Fig. 3.3, the system consisting of five parallel centrifugal pumps has five BERs where the highest efficiencies of 33 – 35% were observed. They are located in the right upper parts of the operational areas slightly below and above the solid lines of the performance characteristics recorded at the rated speeds and dotted lines of the system characteristics recommended by the manufacturer. When the pipeline is narrowing and the speed is lowering, the efficiency drops. Therefore, to achieve relatively high-efficient operation, it is reasonable to choose a minimum number of pumps working at the highest possible speed.

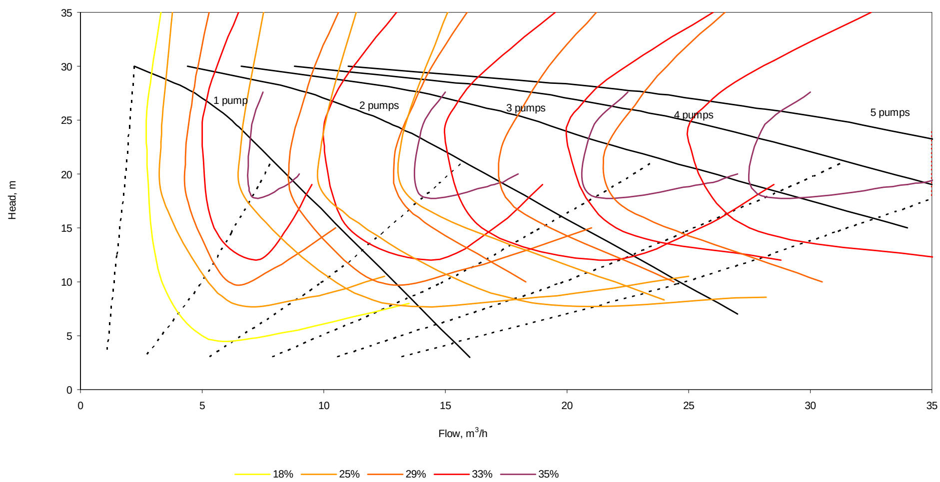


Fig. 3.3 Efficiency distribution across the performance and system characteristics of five parallel-connected pumps Ebara CDX 120/12.

Two groups of areas attract highest attention in Fig. 3.3.

When the low-speed operation is needed where it is impossible to choose less pumps working at the high speed, the areas to the right of the dotted lines are of interest. In these areas, the efficiency decreases and the risk of premature mechanical wear and cavitation increases and the service life shortens (Nesbitt, 2006).

Fortunately, Fig. 3.3 demonstrates several zones where, due to the additional running pumps, an efficiency higher than at a lower number of pumps running can be achieved. For example, to maintain a total head below 15 m at the flow rate above 7 m^3/h, it seems preferable to run two pumps instead of one. In the same way, to support the low total head at the flow rate above 17 m^3/h, it is better to run three pumps instead of two whereas at the flow rates above 27 m^3/h, four pumps instead of three look more efficient, etc.

The areas above the rated speed deserve some attention as well. In (Vodovozov, 2014) it is stated that VSDs provide optimal operation in the so-called "constant power region" running at the speeds above the rated level at the limited motor torque. Hence, pumping in this region under the accurate torque control will probably provide many benefits by running a lower number of pumps.

3.1.4 Efficiency map

The diagram in Fig. 3.4 represents the pump system efficiency together with the number of operating pumps. The diagram was generated on the basis of the efficiency map shown in Fig 3.3.

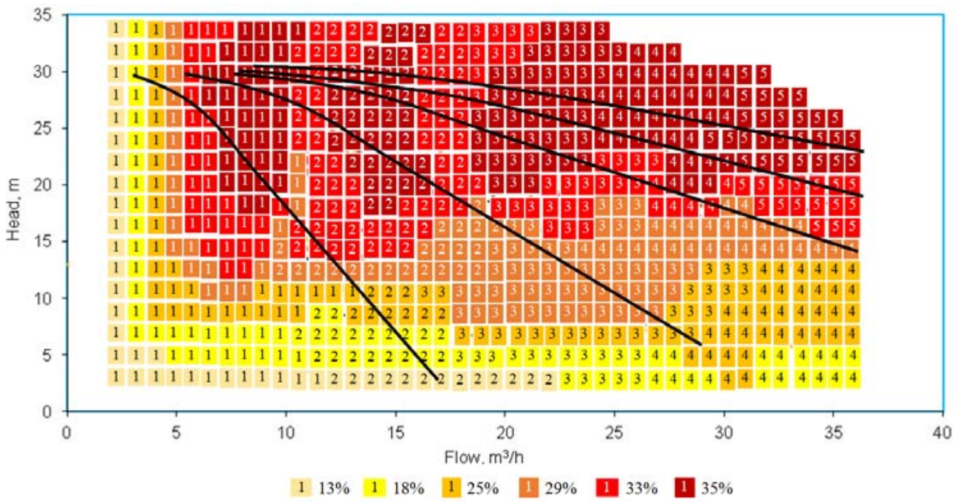


Fig. 3.4 Efficiency map for five parallel connected pumps Ebara CDX 120/12.

Here, the coloured HQ cells show the recommended number of pumps for keeping the working point in a specific location.

An efficiency map shown in Fig. 3.4 may be used for both the pump system control and the analysis on the design stage.

Based on these possibilities, a new approach to pump system management can be proposed. The main idea is to provide the predefined values of a recommended number of pumps for a wide range of locations on the performance graph. For this purpose, the graph shown in Fig. 3.4 must be stored into the control unit memory as a HQ lookup table. The lookup table may be implemented as a three-dimensional array containing all definable [H_i, Q_i] working points together with a recommended number of working pumps for each point.

The purpose of the control algorithm designed on the basis of such a table is to select the optimal number of working pumps Z and to calculate the required speed reference n^* for these pumps.

Any cell in Fig. 3.4 may be treated from the viewpoint of an expected efficiency. The resulting value of efficiency in this cell is an average of the composing cell efficiencies. In general, for the overall head-flow area of Fig. 3.4, this value is 29%. Providing a 9 m total head in the full range of flow rates would result in an average efficiency of 24%. While providing a 30 m total head in the full range of flow rates, an estimated average efficiency is 33%.

For comparison, an efficiency map for the traditional speed-oriented approach is shown in Fig. 3.5. The average efficiency level for the overall head-flow area in this case equals 26%. Providing a 9 m total head in the full range of flow rates would give an average efficiency of 8% and providing a 30 m total head in the full range of flow rates would result in an average efficiency of 25%.

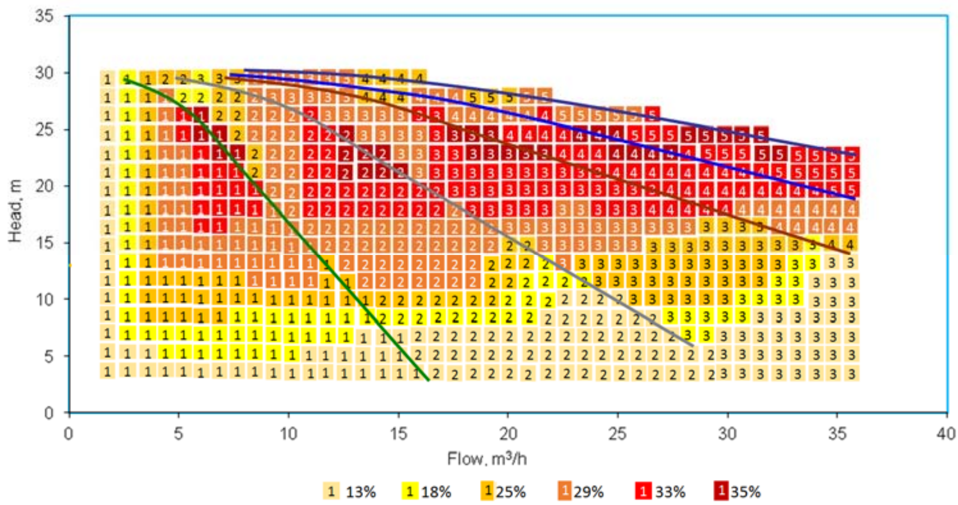


Fig. 3.5 Efficiency map for the traditional speed-oriented approach.

3.1.5 Experimental efficiency estimates

In practice, monitoring and control of centrifugal pump operation is hardly possible without information about the current location of the working point of the pump. The working point of the pump can be determined if the produced flow rate or head is measured. In this case, the working point location is indicated by the head produced by the pump and its flow rate. Generally, external measuring devices are providing the data needed for pump control. Usually, in many pumping applications only the pressure at the outlet of the pump is measured. Therefore, alternative estimation methods are needed to determine the location of the working point.

In (Ahonen, 2011) it is proposed that location of the working point of a centrifugal pump can be directly obtained if the head *H* and flow rate *Q* produced by the pump are measured. The flow rate can be obtained by various methods. Most frequently differential pressure and flow velocity methods are used. The pump head equals the total pressure difference across the pump. Hence, it can be estimated by measuring the static pressure difference and taking into consideration the effect of liquid characteristics, flow losses, and velocity head, which is caused by the liquid velocity difference across the pump. With these direct measurements, it is possible to map the working point on the HQ characteristic graph of the pump. These measurements provide information regarding the hydraulic or output component of the pump efficiency. The remaining part or the input component of the efficiency is an input power. It can be determined by measuring the mechanical power at the pump shaft, which requires the torque and speed measurements. In the case, when the pump motor is driven by the frequency converter, it is possible to obtain the inlet power component from the converter current parameters.

Another approach to determine the efficiency is based on estimation methods. The model-based sensorless flow calculation method serves here as a good example. The model utilises centrifugal pump modes based on the HQ pump characteristic supplied by the manufacturer and on the Bernoulli principle (Bakman, 2015). The model is tuned with such values like liquid density, geometrical parameters of the pipeline, and rotational speed of the pump. The HQ characteristic of the pump is supplied to the model as a lookup table, providing the key points of the curve. The model-based approach is highly dependent on such input parameters as the key points of the HQ curve. The fact that the performance parameters of the pump can change with time is a drawback of this method. On the other hand, this approach provides an opportunity to observe the operational state of the pumping system by using the internal measurements of a frequency converter.

In this approach, the location of the pump working point is estimated from the HQ curve of the pump. The pump head can be defined as the total pressure difference across the pump. It can be measured with two separate pressure sensors located at the inlet and outlet of the pump (Ahonen, 2011). The rotational speed of the pump is taken from the relevant parameters of frequency converter. The HQ curve shifts on the pump characteristic graph pane depending on the value of the current speed. So, to utilise it in the estimation of the flow, it must be adopted corresponding to the current rotational speed of the pump. Affinity transformations provide such adoption. Figure 3.6 describes the estimation process of a current flow and the head of the pump.

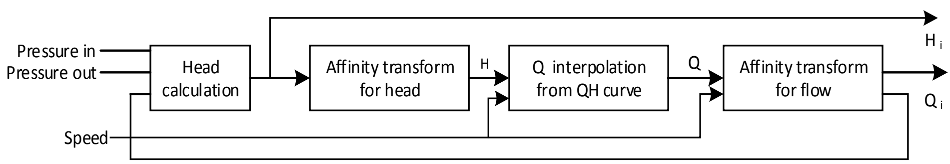


Fig. 3.6. Steps of the HQ graph based estimation method (Ahonen, 2011).

Typically, calculations are implemented inside the PLC or programmable control unit of the frequency converter. The head calculation is based on the Bernoulli equation for incompressible liquids, which can be expressed as:

$$H = \frac{p_\mathrm{d} - p_\mathrm{s}}{\rho \cdot g} + \frac{8}{\pi^2 \cdot g} \cdot \left(\frac{1 + k_{\mathrm{f,d}}}{d_\mathrm{d}^4} - \frac{1 - k_{\mathrm{f,s}}}{d_\mathrm{s}^4} \right) \cdot Q^2 + \left(Z_\mathrm{d} - Z_\mathrm{s} \right), \tag{3.10}$$

where, p is the liquid static pressure, k_f is the friction loss factors between the measurement point and the corresponding pump section, and z is the vertical distances between pressure sensors at the inlet and outlet. Indexes d and s indicate the discharge and the suction side of the pump, respectively. There can be valves and other piping system components causing flow losses between the pump. The friction loss factor k_f enables these factors to be taken into account (Nesbitt, 2006).

Having completed the pump head calculation, the corresponding flow rate can be estimated from the HQ characteristic curve of the pump using the interpolation method. The published HQ characteristic curve is relevant to the rated speed case. Therefore, if the current speed of the pump is different from the rated one, the affinity transformation (3.3) should be applied to estimate the value of the flow rate regarding the current speed n_i.

The flow value rate should be applied to the calculation of the head in (3.10) as soon as it has been estimated for the first time. Working point estimations should be performed with a proper time interval that provides the instant detection of the varying flow rate. The time interval is mostly dependent on the dynamic rate and the state of the process. The minimum recommended time interval can vary from tens of milliseconds to several seconds (ABB, 2006). In practice, the modern control units provide flexible task timing and triggering and traditionally, process control applications are implemented in tens of milliseconds periods.

This method can partially be considered as a measurement-based estimation approach since it requires external measurements. However, the measuring pieces of equipment like pressure sensors required in this method are typically installed to the pumping system since they are essential elements of it.

3.1.5 Resume

1. From the viewpoint of efficiency, the most appropriate way to run the centrifugal pump is to keep its operation as close as possible to the BEP. Though this region is the recommendable pumping operating area, often it is required to overcome its borders for a short or continuous time. According to the literature review, some useful ideas are available of how to avoid such operations; however, most of those cannot be accepted without a decrease in efficiency.

2. To reach a new solution, the system efficiency was estimated in the developed simulation environment with a built-in lookup table. Efficiency distribution across

the performance and system characteristics of five parallel-connected pumps Ebara CDX 120/12 were obtained.

3. The two areas most attractive here are as follows: low-speed operation and high-speed operation. In the former, due to additional running pumps, the efficiency that can be achieved is higher than that with a lower number of pumps running. In the latter case, pumping under an accurate torque control will provide many benefits by running a lower number of pumps than that with a higher number of pumps.

4. As a result, an efficiency map has been produced where the coloured HQ cells show the recommended number of pumps for keeping the working point in a specific location. This map may be used for both the pump system control and the analysis on the design stage. After storing into the control unit memory, it becomes a core of the new control approach suitable for selecting an optimal number of working pumps and calculation of the required speed reference for these pumps.

3.2 Novel Algorithms of High-Efficiency Pumping Management

3.2.1 Problem statement

Many studies have concentrated on the energy management challenges in the field of water distribution. As shown above, typically, the major factor is set on the energy efficiency (Carlson, 2000), (Szychta, 2012), (Arribas, 2002).

The energy efficiency of the multi-pump stations was studied, for example, in (Viholainen, 2013) where an optimal management of a water boosting station based on the parallel-connected VSD driven pumps is discussed. In (Yang, 2010), an alternative control method based on the pump models and calculation of future power consumption of the system was proposed.

Nevertheless, multiple problems of optimal and energy efficient control for multi-pump systems are still open and deserve attention.

In most of studies mentioned above, the pump efficiency was calculated whereas the drive factor stays beyond the calculations. As in the case of induction motors, the VSD efficiency is defined by the motor and converter size as well as by the torque developed. When the motor is driven at a low rotational speed, it has a decreasing effect on the VSD efficiency (ITT, 2007). Neglecting that factor results either in the drive oversizing or in the system efficiency drop.

Many studies concerning variable-speed pumping concentrate on the efficiency of pumps and pump motors. At the same time, frequently, the efficiency of the frequency converter is outside their scope. However, overall energy efficiency of a pumping system depends highly on the specific power losses in the frequency converters (Europump, 2004). Moreover, the efficiency of the VSD is highly affected by the rotational speed of the pump and its torque. The varying speed of the pump is one of the main tasks in the productivity regulation of the pumping system, which is a subject of most control algorithms. Hence, adjusting the pump speed can seriously affect the efficiency of the frequency converter and the full pumping unit.

Among several attempts to estimate the motor drive efficiency in real time, the study (Vodovozov, 2015) is addressed to the drive manufacturers' datasheets where the motor and frequency converter losses are collected for their part-loading operation at different speeds. Though this approach looks prospective, it needs accurate torque measurement and manufacturer's data extrapolation at every estimation step.

Varying number of working pumps is a core of productivity regulation. However, changing the number of working pumps in the system causes a need in speed balancing or adjustment in order to keep the desired head or flow level. This is also a widespread cause of setting the frequency converter into an undesirable mode of operation from the point of view of energy efficiency.

In addition, some problems today have more fine-drawn solutions due to high control and data processing opportunities of the frequency converters that can provide more possibilities to integrate the high-efficiency control algorithms straight into the pump drive.

3.2.2 Related works

In (Yang, 2010), a multi-pump control system based on the mathematical model of pumps was proposed. This method was applied to boost the system equipped with multiple variable-speed pumps in parallel. Its purpose is to minimise energy consumption of the pumping system by adjusting the number of running pumps and their corresponding speed references in a real-time manner. The method was implemented such that the system is a subject to potential changes of setpoints and operating conditions. Following the estimation of a number of static models for different pump combinations, a table of optimal scheduling algorithms was generated. The Branch and Bound method has been used to cope with the table generation problem, and the Lagrangian Multipliers method has been applied to process the corresponding nonlinear programming within each program cycle. A feedback control mechanism was introduced into the proposed control system in order to handle the potential modelling errors. In the case of undefined operating conditions, a detection algorithm was introduced to estimate unknown system coefficients online. The model based on the pump performance characteristics has expressed the control signals in the form of polynomials. The model was adjusted with current process parameters in order to represent the correct current state. The polynomials used the data provided in the pump manufacturer's datasheet. The table was constantly updated due to current process indications taking into account the model. The table of possible system states was populated with data by applying the current process indications to the pump model. In this way, rating of possible working system configurations was created in the control unit memory. The most optimal configuration was then selected from the rating list. The main criterion for selection was an expected power consumption.

The research described in (Viholainen, 2013) concentrates on finding a solution to energy efficiency improvements in a variable-speed parallel pumping system with possibly less input data without start-up measurements and utilising a sensorless flow calculation. The research proposes a new control strategy for the variable-speed parallel pumping system based on the flow rate estimation and pumping operation monitoring based on the VSDs. The advantage of the new control method is the opportunity to run the variable-speed parallel pumping system in a region, which provides improved energy efficiency and reduced risk of mechanical failure of the pumps compared with the traditional control approach. This control aimed to prevent the variable-speed parallel pumping system from operating in regions with low energy efficiency and high risk of mechanical failure based on the pumping working point estimation and preferable operating area, which could be limited by the pump operator. Implementing this control strategy does not require any mathematical optimisation tools. The control program used a simple feedback from the pump output. Information about the working points and individual rotational speed values was collected from the frequency converter. Such a control algorithm tends to keep the working points of all pumps in their preferable operating areas.

This feature is also available for pumping sets consisting of different pumps unequal in their sizes. Keeping the pumps in preferable operating areas is implemented through the speed adjustment. In some cases, this control strategy causes the pumps to run at a reduced speed. The speed adjustment is based on a balancing principle described in (Hammond, 1984). When the additional pump is running and starting to generate a flow, the rotational speed of both pumps can be balanced to provide the same head value (Viholainen, 2013). The next additional pump is started when the working pump is about to leave its preferable operation area towards an increase of the flow rate.

Industrial solutions on the water market are typically represented readymade software utilities embedded into the frequency converters or PLCs. Among multiple useful features, these solutions provide productivity adjustment functionalities. Typically, these functionalities utilize two methods of productivity adjustment: speed control and staging of additional pumps.

The speed control is frequently implemented using PI regulation (Danfoss, 2014), (ITT, 2007). The current value of the process can be freely defined and brought to the converter controller through the analogue input. This can be a pressure, flow, temperature, or any other signal transmitted by the sensors. The speed is adjusted in order to keep the current value of the process as close to the reference value as possible.

The staging of additional pumps is typically based on the rate of demand satisfaction. In particular, the pump array is staged up when the current working pumps are not able to provide sufficient productivity. The array is staged down when the productivity of working pump array is excessive. Some manufacturers provide a possibility to select the criteria for staging up. Typically, the criterion of the staging up or down is an event, such as exceeding the specific or defined limit by the process value (or other value brought to the control unit through an analogue input or estimated by control algorithms). In the ITT PS200 pumping solution (ITT, 2007), there is a possibility to select the signal for staging up or down criteria. The value can be selected among such values as flow, pressure or temperature. In the pump control software of Danfoss Aqua for a frequency converter, only current pressure is available as a criterion for staging up or down (Danfoss, 2014).

Therefore, the criteria of productivity adjustment of the pumping system available in many industrial solutions do not take into account limitations of pump efficiency. In addition, efficiency-monitoring approaches typically do not exist in the solutions available on the market.

Most of the existing pumping control solutions for water and wastewater applications and pump management neglect the current and future efficiency of the pump and motor in their productivity adjustment. No means for keeping the working point of the pump in a preferable operation area are represented in these solutions or can be possible only by setting the static limits via speed limitations.

Traditional pump management approaches overlook the efficiency of the VSD and, in particular, the frequency converter when implementing the adjustment of the pump system productivity. However, taking into account the frequency converter losses in overall pump system efficiency is quite significant.

3.2.3 Scenario of high-efficiency predictive control

It can be stated that each staging up is shifting the individual working points of pumps to the left, moving them away from the BEP. This principle is fully justified when the pump is working at its limit performance or near the limit of AOR before staging. However, if the limit capacity of a pump is still not reached, it may be beneficial to operate at high speed, not staging up the system if the AOR conditions are met.

The estimations and decisions described above can be implemented on a platform of the industrial process control equipment commonly used. The following diagram represents the architecture of a program for these estimations and preparation of the data for decision-making.

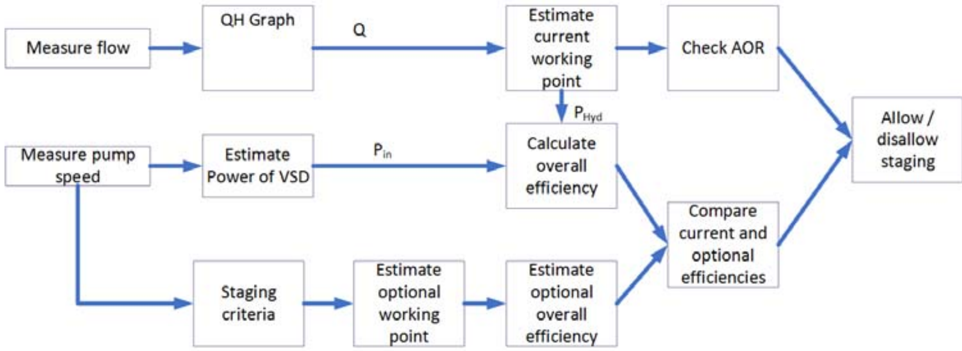


Fig. 3.7. Program architecture for implementation of the efficiency-based control method on a PLC platform.

The proposed algorithm provides an ability to estimate the real-time efficiency of the pump together with its drive. Moreover, the optional efficiencies for the following cases are constantly re-estimated and compared with the current efficiency:

a. Efficiency for the case when a pumping system is staged up (next, an additional pump is started)
b. Efficiency for the case when a pumping system is destaged (one of several pumps is stopped)

The mentioned estimations predict the state of the system in the case of changing the number of working pumps and provide information for comparing the current state with some optional (predicted) state. Figs. 3.8 and 3.9 show the current and predicted working points for both cases.

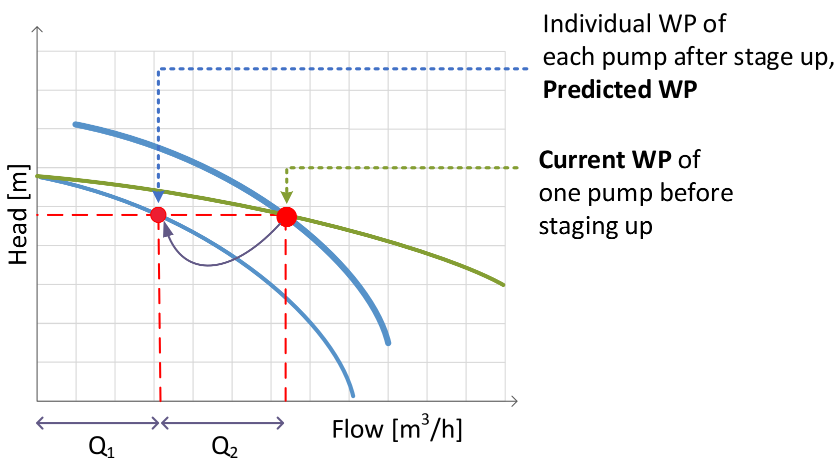


Fig. 3.8. Graphical representation of the predicted state for the staging up.

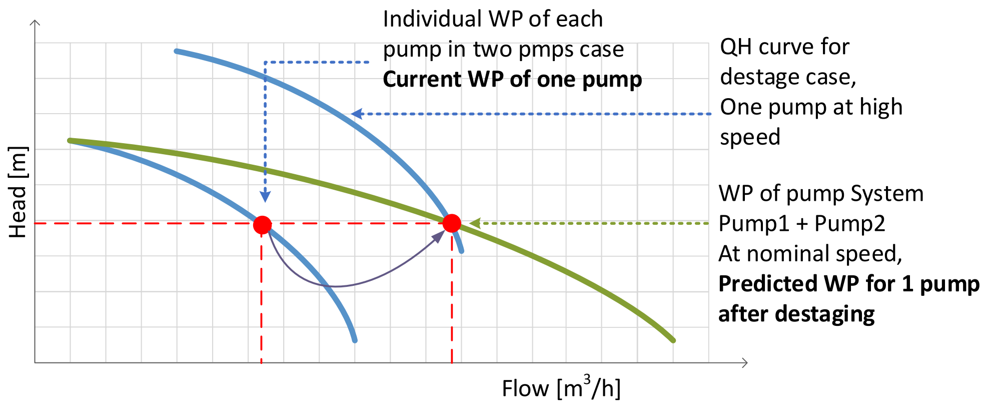


Fig. 3.9 Graphical representation of the predicted state for the destage case.

Both predicted working points are being re-estimated during each program cycle. Therefore, in the case the staging criteria are met, the predicted working point is ready to be used for decision-making. The steps of working points, overall efficiencies estimation, and prediction as well as making a decision are described further.

The staging or destaging is required when the corresponding condition is met. This condition is an increase of speed or increase of a difference between the process variable and its reference. The first step of the algorithm is to detect the active criterion of staging. In the current research, the criterion check was implemented by the standard firmware of ACQ810 frequency converter based on the current speed comparison to the *Start speed limit* and *Stop speed limit*. The staging delay is taken into consideration. If the speed exceeds the mentioned limits and stays outside the limits for a time that is longer than the staging delay, the staging trigger will be activated.

In order to enable the staging, the current efficiency must be compared with the predicted after-staging efficiency. After-staging efficiency includes the values of efficiency of a pump together with a VSD in the case of staging up and destaging. Hence, three types of efficiency must be estimated on this step:

a. Current efficiency
b. Efficiency for the staging up case
c. Efficiency for the destaging case

Calculation of efficiency is divided into two stages: estimation of the pump working point (which is expressed by hydraulic power) and calculation of the VSD efficiency.

The model-based predictive method is applied to estimate the current and the after-staging working points. To estimate the current working point, the value of the current flow must be obtained. This value can be retrieved from an external flowmeter or calculated through the sensorless estimation methods (Ahonen, 2008).

The value of the current flow is applied to the HQ curve to obtain the value of the current pump head. The numerical representation of the HQ curve in the form of the lookup table containing coordinates of key points is used to calculate the pump head. The calculation itself is implemented through the interpolation method. Naturally, the accuracy of calculation depends on the number of key points in the lookup table and the curvature rate of the HQ curve.

To estimate the working point for a staging up case, the HQ curve based on the corresponding (after-staging) number of working pumps is used. The example of the HQ curve for two working pumps is shown in Fig. 3.9. The after-staging HQ curve for the rated speed is estimated using the parallel pumping flow distribution principle applied to the key points of the active HQ curve rated speed shape. The resulting curve is then adopted to keep the desired flow using the affinity transformation. The value of the after-staging rotational speed of each pump is calculated from the location of an individual predicted working point and the HQ curve of the pump.

The same principle is used to determine the working point in the destaging case. The parallel pumping flow distribution principle is applied considering the reducing number of the working pumps. The resulting predicted working point and the value of the after-destaging speed are used in further calculations.

Obtained coordinates of individual working points provide data for calculating the output power of the pump. The input power is to be calculated considering the power losses in the frequency converter and the motor. The on-shaft power is obtained from the current parameters of the frequency converter (as in the case when ACQ810 is used).

Data concerning the power losses are obtained from the VSD model. In this research, the model provided by the ABB DriveSize tool was used. The data concerning the power losses should be obtained from of graphs and integrated into the program as a multidimensional lookup table. The current power losses of the VSD were extracted from the model by applying the interpolation method for the current speed. For the after-staging cases, the predicted speeds were used. The input power was estimated by summarising the on-shaft power obtained from the frequency converter and the values of power losses obtained from the VSD model. The current and after-staging efficiency was calculated applying the values of hydraulic power and input obtained.

In this stage, the current and optional efficiencies may be compared. The result of the comparison provides optional criteria for staging or destaging of the pumping system. In this comparison, it is possible to increase or decrease the number of working pumps depending on the efficiency for a staged up or destaged option. However, this method of staging criteria definition is out of scope of this research.

Once the predicted efficiency for the after-stage case is known, it is possible to allow or disallow staging by triggering the firmware of the VSD. In the case of staging disabling, the consideration of AOR must be taken into account. As it is stated in (Gülich, 2008), the pump is capable of operating in the AOR during a specific period without harming the hardware components.

Generally, the efficiency is falling in the case of staging up. It is caused by the form of the efficiency map. Particularly, from the efficiency map of Ebara 120/12 run by with ACQ810-04-02A7-4 it can be seen that an increase of the efficiency is only possible when the working point is moving to the right and up the HQ performance graph pane. This is possible when the rotational speed of the pump is increasing. In the case of the speed decrease, which happens when a new pump is started, the working point (and hence the efficiency) is shifting to the left. Taking into account the direction of the working point shift and the frequency converter efficiency drop at low speed, it is possible to state that starting a new pump results in a loss when the system is safely operating in the AOR.

3.2.4 Prediction of the location of the working point

In order to prevent the pump management program from making the decisions that would harm the process, reduce the efficiency of the full pumping system, and move the working point to undesired operational areas, possible consequence of productivity adjustment should be taken into consideration. To provide the possibility of foreseeing the consequences of changing the number of working pumps and its influences on the overall efficiency of the pumping system, the predictive control method is implemented in this section using the polynomial approach.

The proposed method provides the efficiency prediction of the pumping system together with the frequency converter. The approach is based on the mathematical model of the pumping system that takes into consideration the power consumed by the pump itself as well as power losses of the VSD. This prediction algorithm is implemented on a typical industrial PLC platform, which provides high flexibility in tuning up and monitoring. The models of the pump and the VSD are stored into the memory of the PLC as lookup tables. These lookup tables contain key points of characteristics of the VSD and the pumps published by the manufacturer of the pump.

The characteristic HQ curves (1.1) keep information about the performance capabilities of a single pump at rated operation conditions. The performance capabilities of the pumping system consisting of Z similar pumps can be derived from the single-pump HQ curve graph taking into account that the total capacity of all the parallel-connected pumps is a combined capacity of each one of them ($Q_1 + Q_2 + \ldots$). The resulting graph is shown in Fig. 3.10 (a).

Adopting the published rated HQ characteristic for their dynamic use is achieved through employment of affinity laws (3.1) – (3.5). This adaption is reflected in shifting the HQ curve right-up and left-down on the performance graph pane depending on the pump rotational speed. The combined HQ curve for several pumps slides on the graph pane in a similar way. The speed-related dynamics of the combined HQ curve is shown in Fig. 3.10 (b).

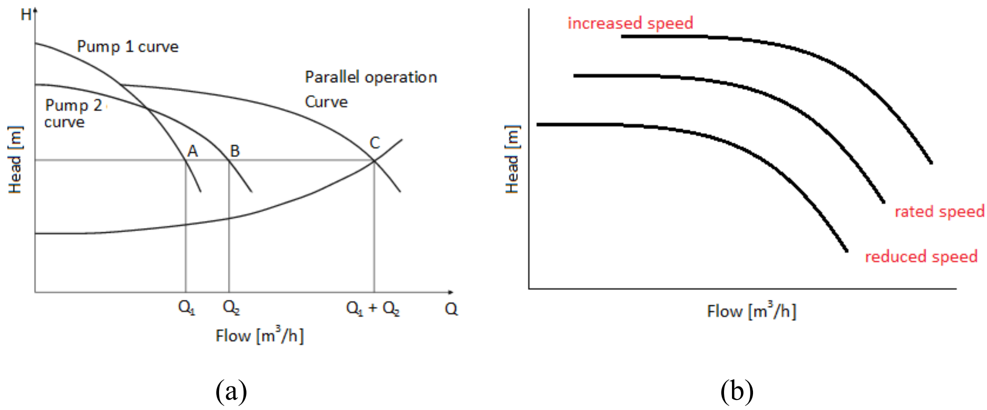


Fig. 3.10. Parallel operation of two pumps (working points A and B).The combined curve and resulting operating point location C with the total flow rate $Q_1 + Q_2$ (a) (Bortoni, 2008) and shifting correspondingly to speed dynamics (b).

As it was stated above, the model of the VSD was obtained from the ABB DriveSize tool. The input data for that tool are parameters of the motor, type of the load, and its dynamics, information regarding the parameters of extreme motor operation, and parameters of power supply. The output data in the form of VSD power losses vs. pump rotational speed are shown in Fig. 3.11.

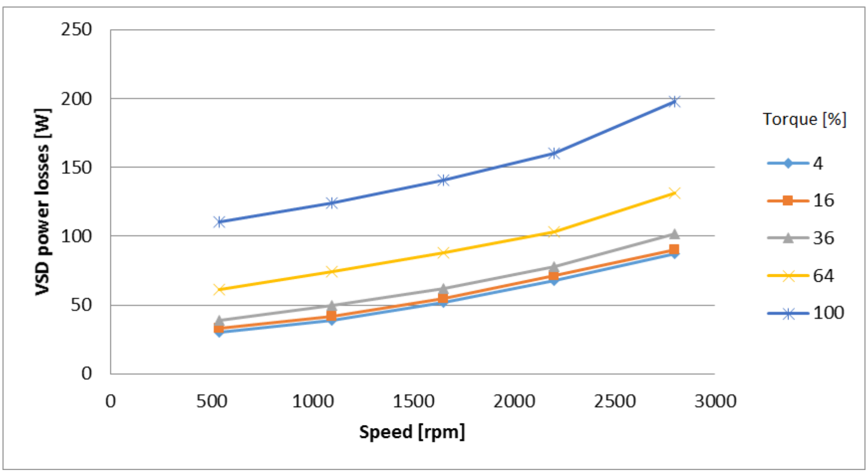


Fig. 3.11 Characteristics of the VSD, representing the dependence between pump speed and drive losses. The graph shows power losses of ABB ACQ810-04-06AD-5, Pnom = 1.1 kW and M3BP90LB2 2 pole asynchronous motor with P_r = 1.5 kW and n_r = 2892 rpm.

The graph showing drive power losses vs. generated speed enables derivation of the power consumed by the VSD and calculation of the total power consumption of the pumping system, including the motor and the frequency converter. This is relevant to both the single-pump and the multi-pump systems. The input data are stored in the control unit as a lookup table containing the key points.

The analytical representation of the proposed control method was introduced in Section 3.1. In the given case, the algorithm is utilised for keeping the constant head application.

The represented head control algorithm shown in Fig. 3.12 is used in the following way. First, the desired total head H^* is defined together with the parameters of the pump performance characteristic (1.1).

In every iteration, the current flow rate Q_i and the total head H_i readings are acquired. Next, the current total head H_i is compared with the reference head H^*. To reduce their difference, a suitable flow rate Q^* is calculated using the affinity laws (1.5 - 1.6) as follows:

$$Q^* = Q_i \sqrt{\frac{H^*}{H_i}} . \tag{3.11}$$

These Q^* and H^* data are used for calculating the optimal number of pumps Z from the lookup Table 3.1.

Substitution of H^*, Q^* and H_i, Q_i to the system characteristic (1.2) brings the system head H_s and head factor C_s as a solution of two equations with two unknown variables:

$$C_s = \frac{H^* - H_i}{Q^{*2} - Q_i^2} , \tag{3.12}$$

$$H_s = H^* - C_s Q^{*2} . \tag{3.13}$$

A working point at the intersection of the system curve (1.2) with the current HQ curve (1.1) yields the flow rate

$$Q_{rs} = -\frac{a}{2} + \sqrt{\frac{a^2}{4} - b} \tag{3.14}$$

where

$$a = \frac{C_1 Z}{C_s Z^2 + C_2} , \; b = z^2 \frac{H_s - H_0}{C_s Z^2 + C_2}$$

at the suitable total head H_{rs} obtained from (1.2). The required speed n^* is obtained from (3.6).

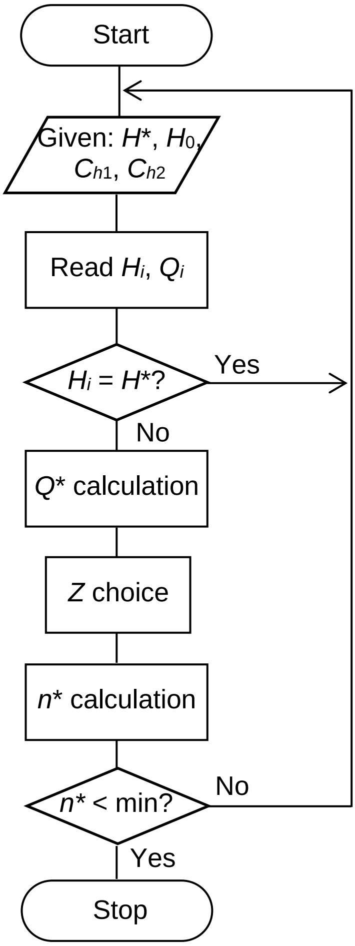


Fig. 3.12 Control algorithm.

3.2.5 Resume

1. The criteria of productivity adjustment of a pumping system available in many industrial solutions overlook the limitations for pump efficiency. In addition, typically, efficiency-monitoring approaches are absent in the solutions available on the market. Today's leading production companies do not supply any devices for keeping the working point of the pump in a preferable operation area.

2. The developed scenario of high-efficiency predictive control provides step-by-step instructions for the working point motion inside and outside the BER.

3. The proposed methodology provides an ability to estimate the real-time efficiency of the pump taking into account drive losses. To this aim, the efficiencies are constantly re-estimated and compared with the current efficiency. It concerns

both the efficiency for the pumping system staged up (next additional pump is started) and that destaged (one of several pumps is stopped).

4. An algorithm for the prediction of the location of the working point is implemented on the typical industrial PLC platform. The models of the pump and the VSD are stored into the memory of the PLC as lookup tables containing the key points of characteristics of the VSD and the pumps published by the manufacturer of the pump.

3.3 Simulation of High-Efficiency Pumping

3.3.1 Simulation resources and methodology

To verify operability of the predictive method, computer simulation was used. The simulation was based on the pump and VSD models. The purposes of the simulation include the following: to demonstrate the distribution of losses between all devices in the pumping system, to compare the energy consumption of a traditionally controlled system and a system operated in the high-speed zone, and to emphasise the importance of the VSD efficiency. The simulation included emulating the multi-drive multi-pump station consisting of two pumping units. Each pumping unit consists of a centrifugal pump run by the VSD including a frequency converter and an induction motor.

In order to implement the models of pumping units, two separate modelling environments were used. The centrifugal pump and all hydraulic equipment including the pressure and flow sensors, pipeline, and tanks were modelled in Matlab/Simulink. The model of the VSD consisting of an induction motor and a frequency converter was prepared in the ABB DriveSize software tool. Both models were brought together in the Simulink environment. Fig. 3.13 demonstrates the simulation architecture in brief.

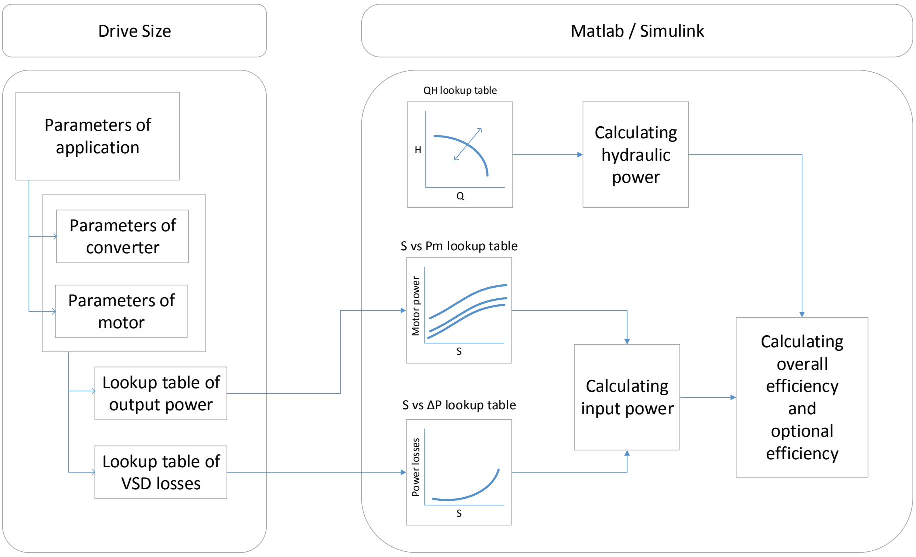


Fig. 3.13 Architecture for simulation the multi-drive multi-pump station.

The simulation included running the pumping station under the traditional speed-oriented control scenario and under the proposed efficiency-oriented control method in the high-speed zone. Operation under the traditional control was expressed by the staged up system. In this case, two pumps were operating at the reduced speed. The readings of the output (hydraulic) power of the pumping station, input power of the whole system, and values of power at all stages were monitored.

Running in the high-speed zone at the proposed control was expressed by the single-pump operation at the speed, which was higher than the rated one. The consumed energy readings were monitored for both cases. The energy was calculated for a one-hour operation period, but could be extrapolated for longer periods.

The speed of operation was selected to enable comparison of the benefits of operation under the traditional control method (staging up depending on current speed) and the proposed method. Hence, the operating region was selected where the flow provided by the single high-speed pump or by two pumps running at the reduced speed. Simulations for both control methods were implemented in order to provide the same amount of flow by staged up and high-speed systems. The models of pumping units were selected in order to express the real setup used in the experiments. Therefore, the references of the flow for simulation were set to $8m^3/h$ and $10m^3/h$. During the simulation, these references were fulfilled by the staged up system and by the high-speed system.

The models of the centrifugal pumps were taken from the Simulink Sym SimHydraulics toolbox.

During the simulation, the pumps were set to operate with a defined speed, depending on the test case. The current speed of pumps was applied to the VSD model expressed in the form of the lookup tables derived from the characteristic graphs. These lookup tables provide the current losses and input power depending on the pump speed and torque for each test case.

The special functionality was computed from the hydraulic power at the output of the pumping system based on the provided flow and pump head. The value of the current flow was measured by the metering functionality of SimHydraulics. The achieved pump head was calculated by the pump model implemented as the lookup table, which expresses the characteristic HQ curves of the pump. The input power was estimated using the hydraulic power, pump input power, and VSD losses obtained from the motor and frequency converter model.

3.3.2 ABB DriveSize toolkit as the model of the pump variable-speed drive

DriveSize is a PC software enabling the user to select a frequency converter and a motor, based on the specific requirements of the application. The tool is especially applicable in cases of no simple selection from a catalogue. Also, it can be used to compute network harmonics, currents, and to compile reports describing the sizing based on the current load. DriveSize contains ABB frequency converter and motor catalogue data. It is also possible to import own databases containing the motor data. The default values provide straightforward motor/frequency converter selection. However, the user is also provided with an option to select the drive manually from available databases.

DriveSize consists of product databases, user interface, and computing functionality. The database contains around 60000 catalogue items for motors and frequency converter types. Application-specific motors can be selected based on the ABB catalogues compiled by ABB Oy / Machines. The program was developed for OS Microsoft Windows, following the common user interface traditions.

The DriveSize software tool enables selection of the components of the VSD according to application needs. An appropriate VSD can be selected automatically by the software tool based on the application parameters defined by the user, or it can be selected manually from the proposed list. The list is represented by the selected components of the VSD from the database of motors and frequency converters. The database contains all motor and frequency converter models manufactured by ABB.

VSD component selection process is accompanied by the generation of the characteristic graphs of the motor and the frequency converter. The model of the VSD used in this simulation is based on the speed vs. VSD power losses graph and on the speed vs. motor power graph. These characteristic graphs are transformed into the lookup tables in order to integrate them into the Simulink part of the simulation model. The lookup table values are represented by the key values of the characteristic graphs. The third option of Simulink lookup tables enables considering the various load cases.

DriveSize enables selection between the different mechanical load types of the motor. For this simulation, squared torque typical for pumps and fans was selected.

The simulation includes motor and frequency converter models that are identical to those installed on the experimental setup. Thus, the real values were set as input to the DriveSize tool. The parameters of the selected devices are shown in Table 3.2.

Table 3.2 Parameters of selected motor and frequency converter

Motor			Frequency converter		
M3AA 80C 2			**ACS850-04-03A0-5**		
DOL Catalogue data			**Catalogue data**		
Product code		3GAA 081 313-ASE (ES)			
Voltage	[V]	400	Voltage	[V]	400
Frequency	[Hz]	50	Rated power	[kW]	1.1
Power	[kW]	1.1	Rated current	[A]	2.8
Poles		2	Imax	[A]	4.4
Speed	[rpm]	2875	Ihd	[A]	2.5
Max mech.speed	[rpm]	6000	Pulse		6
Current	[A]	2.4	Frame type		A
Torque	[Nm]	3.6	Icont max	[A]	3
Tmax/Tn		3.5			
Power factor		0.8			
Efficiency	[%]	80.5			
Temperature rise class		B			
Insulation class		F			

3.3.3 Analysis of the simulation results

Below the simulation involving the operation of the pumping system in order to provide the desired flow at the staged up and high-speed mode will be described. Table 3.3 describes the parameters of the simulation.

Table 3.3 Simulation cases.

Control type	**Staged up (2 pumps)**		**High-speed zone (1 pump)**	
Flow reference [m^3/h]	8	10	8	10
Speed of pump [rmp]	2500	2630	2800	3000

Fig. 3.14 shows the location of the rated speed curves of the modelled pump. The characteristic curves at various speeds are expressed for the 2500 rpm case of 1 and 2 running pumps.

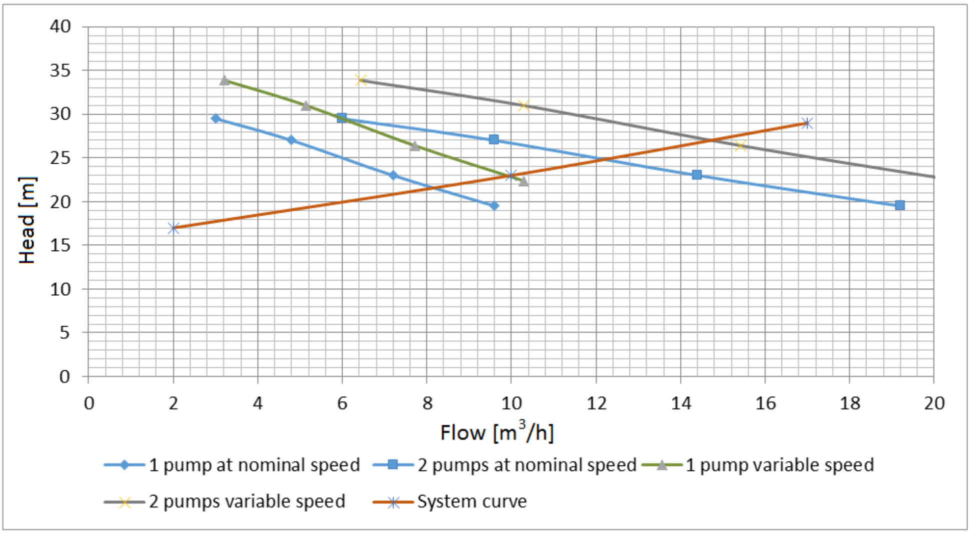


Fig. 3.14 Pump characteristic curves for 1 pump and 2 pumps cases. The graph shows that the flow of 8 m^3/h can be provided by 1 pump running at 2800 rpm or by 2 pumps when each is running at 2500 rpm (the intersection of HQ curves with the system curve at 8 m^3/h).

Figure 3.18 demonstrates the operation of the staged up system at 10 m^3/h and operation in the high-speed zone. In the first case, two pumps provide the flow operating at 2630 rpm, which is lower than the rated speed. In the second case, one pump provides the desired flow at 3000 rpm, which is higher than the rated speed.

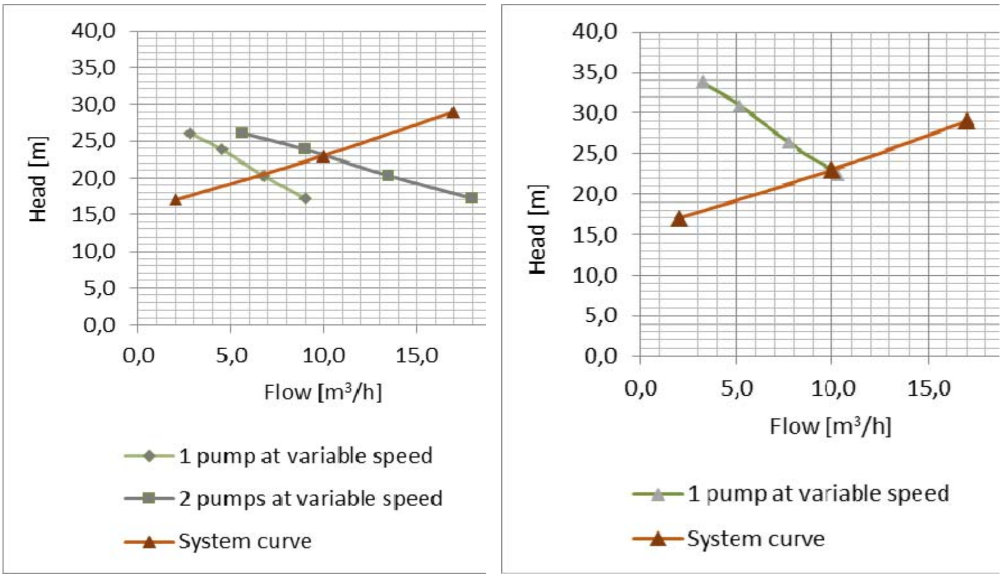


Fig. 3.15 HQ characteristic curves for providing 10 m³/h in high-speed case (right) and staged up case (left).

The readings were collected by running the pump system using both control techniques in similar conditions. Table 3.4 shows readings of both simulations.

Table 3.4 Readings of simulations for two control techniques

1 pump operating in high-speed zone		
Speed [rpm]	2800	3000
Q [m^3/h]	8	10
H [m]	23	23
$P_{Hyd\ total}$ [kW]	0.50	0.62
$P_{in\ total}$ [kW]	1.43	1.73
Pumping set efficiency [%] (incl. VSD)	0.35	0.36
Pump power [kW]	1.14	1.34
Pump efficiency [%]	0.44	0.47
VSD efficiency [%]	0.80	0.77

2 pumps, staged up system		
Speed [rpm]	2500	2630
Q [m^3/h]	8	10
H [m]	23	23
$P_{Hyd\ total}$ [kW]	0.50	0.62
$P_{in\ total}$ [kW]	1.92	2.15
Efficiency of each pumping set [%] (incl. VSD)	0.26	0.29
Power of each pump [kW]	0.8	0.94
Each pump efficiency [%]	0.31	0.33
Efficiency of each VSD [%]	0.83	0.87

Energy consumption was also calculated for both cases for a one-hour period. Figure 3.19 shows the energy consumption of pumping sets during one-hour operation.

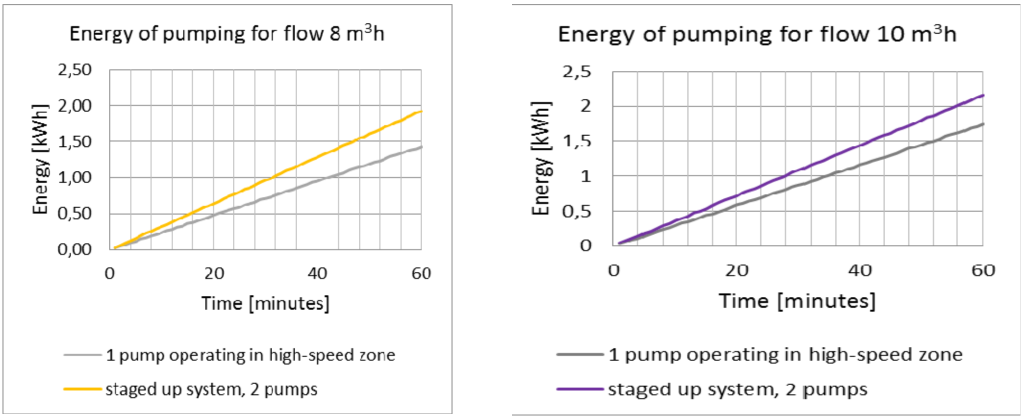


Fig. 3.16 Energy consumption of pumping system providing flow of 10 m^3/h and 8 m^3/h.

Conducted simulations show that the efficiency of the pumping system running under the proposed control strategy in high speed-zone is higher. The energy consumption in this case is lower. The efficiency of a centrifugal pump is also higher in this case, which means lower damage to mechanical parts and the liquid processed is less likely to be harmed by accompanying phenomena like heating.

3.3.4 Resume

1. Ability to combine models of the VSD with those of a centrifugal pump provides a significant benefit to the simulation system. The motor and frequency converter models can be generated in DriveSize utility and imported into the Simulink environment. This enables experimentation with various VSD models.

2. The DriveSize tool includes tens of thousands combinations of motors and frequency converters. It is also possible to import the motors and frequency converters into existing databases. Diversity of possible solutions enables a wide range of simulation cases to be implemented.

3. The Matlab/Simulink environment enables transformation of the model into C-codes or functional blocks for CodeSys and use of mathematical models of control algorithms in machine codes and their transfer into the PLC.

4. Among the disadvantages of the simulation model is its narrow focus on specific applications. Some physical phenomena were kept outside of the scope of simulation for simplification.

5. Resolution of the output values depends on the amount of the key points in the lookup table. However, this disadvantage is natural for all lookup-based models. It can be improved by inducing more key points into the characteristic curves of the

model. Moreover, accuracy of calculation is highly dependent on the number of real parameters taken into consideration. For simplification, some physical parameters and phenomena were kept outside the scope of this simulation.

3.4 Experimental Study of High-Efficiency Pumping

3.4.1 Experimental hardware and software resources

The purpose of the test stand was to conduct the experiments for comparison of the energy consumption in two modes of operation of the pumping station:

- Multi-pump mode where one or two pumps are running depending on the desired productivity
- Single-pump running even at the high productivity setpoints

The general diagram of the stand is shown in Fig. 3.17.

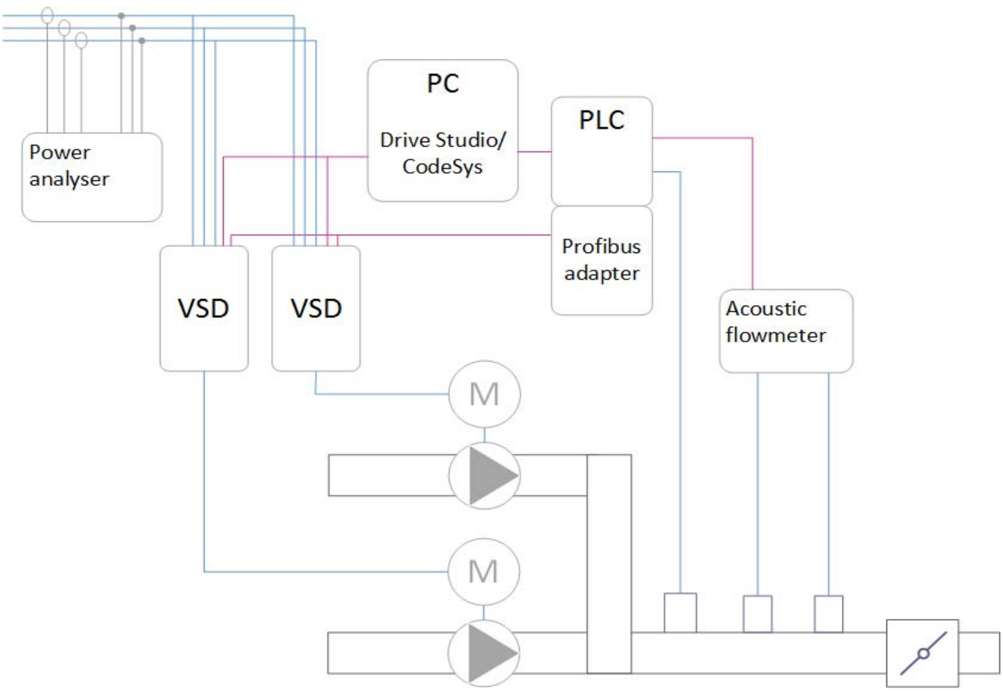


Fig. 3.17 General diagram of the test stand.

For logging the energy and power readings, Fluke 434 II Energy Analyser was used. A current flow rate was obtained from the flow calculation functionality of the pump control software of the frequency converter. The data needed for the flow calculation were acquired from the pressure sensors on the test stand. A current speed of the pump was acquired as a working parameter accessible from the frequency converter via HMI.

The roles of the equipment were as follows:

1. *Frequency converter*. For experimentation, two pumps were used from a 5-pumping system. Each one was run by a separate frequency converter. Therefore, the group of drives consisted of two. The first converter fed the main pump working constantly (Pump 1). The second one fed the additional pump (Pump 2), which was started when an increase of productivity was required. Converters provided the following functions:
    a. *PID control of the pump speed*. The process value is flow.
    b. *Value of the current flow is calculated* by Flow Calculation functionality of the embedded Pump Control program. The calculation is based on the inlet/outlet pressure difference and the Bernoulli principle. Inlet and outlet pressure values are taken from the pressure sensors before and after the pumps. Calculation is implemented on one converter running Pump 1. Therefore, this is a master converter in the converter network.
    c. *Productivity adjustment* by starting/stopping of Pump 2. The converters communicate via the Modbus network. Converter of Pump 1 is a master converter. When the speed of the master converter exceeds the specific predefined value, it sends the start command to the converter of Pump 2 (follower).
    d. *Sending the speed reference to a follower* drive when it is running.
    e. Providing the *readings of current speeds* of both pumps and current flow. Also, sending these data to the PLC for efficiency calculation and efficiency control.
2. The PLC provides the efficiency control for a single-pump part of the experiment. The PLC can suppress the start signal from the master converter if the predicted efficiency (efficiency after taking Pump 2 into use) is lower than the current efficiency. The efficiency is calculated based on the pump HQ characteristic taken as a lookup table, current flow and speed of the pump. The predicted efficiency is calculated using a value of the desired flow and pump characteristic curves adjusted in accordance with the current and future numbers of working pumps. Data needed for calculation are obtained from the master frequency converter via the Profibus connection.
3. The energy analyser logs the power and energy readings. It obtains values of the voltage and current from each of the three phases of the supply cable. Values of the current are obtained through the current clamps. Voltage is measured by the direct contact with cables.
4. The acoustic flowmeter is used to measure the common output flow of the pumps. Its output is a value of the flow rate in m^3/h. The output value is further processed in the controller.

The ABB DriveStudio software tool was used to parameterise the frequency converters and to monitor their operation. The tool enables connecting with the frequency converter through the RS-232 communication interface. With the DriveStudio tool, it is possible to browse and edit the parameters of the frequency converter represented in the form of a table. It is also possible to log and monitor the current parameters and signals in the form of graphs. The graphs can be saved into the PC memory. Relevant signals can be sent to the PLC via the Profibus connection for more advanced processing and graphical representation.

The PLC programming and configuring was implemented through the CodeSys 2.3 software. The software enables composing of the control algorithms and storing them to the memory of the controller. The programming is implemented in the structure text or functional blocks programming methods. The controller supports the 61131-3 industrial programming standard.

3.4.2 Methodology of experimentation

The main target of the pumping systems and applications is to maintain the needed flow at constant pressure in the water supply system. Naturally, on the design stage, prospective values of pressure and flow are used to calculate the mechanical and electrical parameters of pumps. It is accepted that the service life of the pumping system will be longer when operating at the parameters close to rated (BER). Naturally, such harmful phenomena as cavitation, vibration, recirculation of flow, and temperature rise occur when working outside the BER. Long-term operation outside the BER reduces the service life of pumps and hence, running the system in those modes is undesirable.

Allocation of efficiency zones is represented by the following principle. The efficiency of the pumping system is falling when the working point is moving towards increased flow, to the right from the BER. Furthermore, working in zones at the right from the BER is possible only at high speeds.

In practice, when the speed of the main pump becomes insufficient to support the desired productivity, additional pumps should be taken into use. This would help to keep the desired productivity and avoid operation at speeds that are higher than the rated.

Typically, in the multi-pump system, several pumps are run at a synchronous speed (all pumps with the same speed reference). The speed reference is generally generated by the PID. When the speed of the main pump becomes insufficient and an additional pump is taken into use, it results in the productivity rise of the whole system. PID reduces the speed reference in order to avoid the excess productivity. As a result, all pumps are working at the reduced speed. In such case, the new, reduced speed is lower than the rated one.

Because of the speed change, the working point of each pump shifts to a new location. In some cases, the new location of the working point can be more distant from the BEP than the previous one.

The productivity range of the pump can be mapped on the characteristic graph of the pump. It can be seen that the productivity zones of one and two pumps are overlapping. The same is true to the productivity zones of two and three pumps and so on.

Operation in the overlapping zone can be achieved on high speed by one pump or on low speed by two pumps. For both the multi-pump and the single-pump applications, the PID control was used as a main speed generation method. The speed generating functionality and signal conditioning was implemented with the embedded software of frequency converters running the pumps.

The next stage of the productivity control concerns the adjustment of a number of running pumps. This functionality is a part of embedded pump control software of frequency converters ACQ810, which was used in experimentation.

In the multi-pump part of the experiment, productivity of the pumping station was adjusted by taking an additional pump into use. The number of working pumps depends of the desired productivity at the given conditions. When the system curve of the pipeline shifts to the right, a higher speed is needed for keeping a constant flow. Exceeding the specific speed limit is the criterion for taking the additional pump into use. Thus, an additional pump is started when the speed of the main pump is becoming too high in order to provide the desired flow.

In the single-pump case, starting of an additional pump at the same principle as in the multi-pump case was suppressed by the efficiency control functionality. The functionality is implemented in the PLC. It includes the algorithm of monitoring the location of the working point and comparing its current and predicted distance from the BEP. The predicted distance is a distance from the BEP after an increase of a number of running additional pumps.

In this experiment, the working zone of the pump was defined in a region where an increasing number of running pumps is not beneficial (i.e. taking additional pumps into use would shift the working point to a region of lower efficiency).

The speed is to be changed because of varying parameters of the pipeline. Emulation of pipeline parameters alternation was implemented via shifting of the system curve representing the state of the pipeline. In this way, the main changing of the opening rate of the main valve was used to alternate the curve. Increasing of the opening rate of the valve emulates the increase of demand and hence shifts the curve to the left. Reducing of the opening rate of the valve emulates the decrease of demand and shifts the system curve to the left. Naturally, location of the working point changes as a result of displacement of the system curve. The PID adjusts the speed of the pump (or pumps) in order to keep the flow on a desired level.

The opening rate of the valve was adjusted by the same profile in both the single-pump and the multi-pump cases.

As can be seen from the diagram of the profile (Fig. 3.18), the state in which two working pumps are needed for keeping the desired pressure is achieved on the last stage of the experiment.

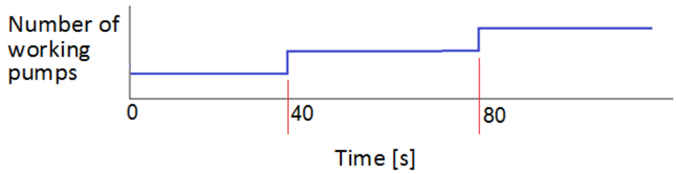


Fig. 3.18 Profile of alternation of the system curve.

3.4.3 Analysis of experimental results

The results obtained from the power analyser, DriveStudio tool, and sensors connected to the PLC were combined and PC processed. The energy consumption as the main criterion of the effectiveness of the proposed method is represented in Fig. 3.19. The readings of experimental pumping providing 8m^3/h are presented in Fig. 3.20 and Table 3.5.

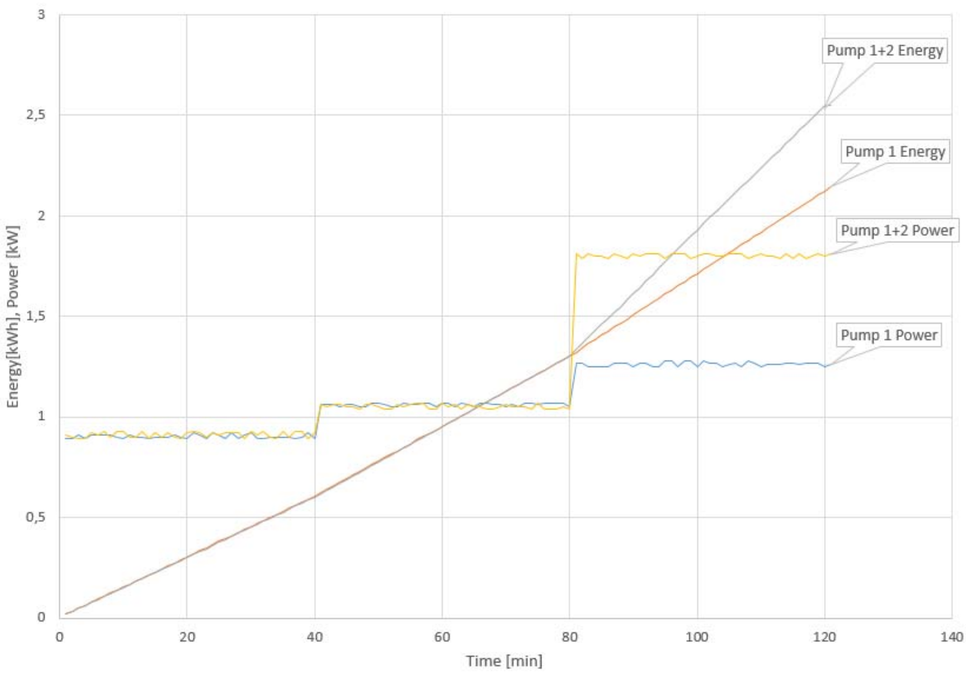


Fig. 3.19 Energy consumption of the experimental pumping system

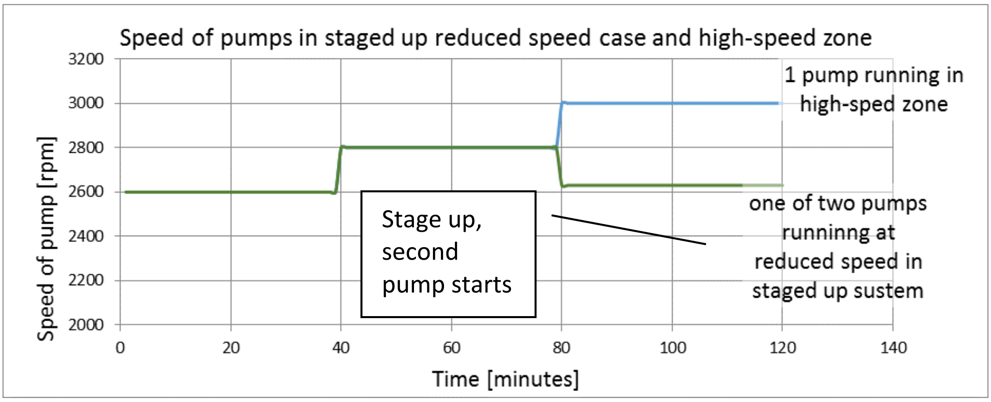


Fig. 3.20 Readings of energy according to the stages of the experiment and the speed of operation.

Table 3.5 Readings of power and efficiency per stage of experiment

1 pump operation			
Speed [rpm]	**2800**	**2600**	**3000**
P_{in} [kw]	1.29	1.1	1.56
P_{FC} [kW]	1.21	1.06	1.48
$P_{pump\ in}$ [kW]	1.03	0.9	1.27
Overall efficiency [%]	0.37	0.36	0.36

2 pump operation on the last stage	
Speed [rpm]	**2630**
P_{in} [kw]	1.84
P_{FC} [kW]	0.89
$P_{pump\ in}$ [kW]	0.77
Overall efficiency [%]	0.26

It can be seen from the results obtained that the pumping system is consuming more power when running two pumps at lower speed than in the case of running one pump at higher than rated speed.

Naturally, running a pump at higher than the rated speed is harmful for the motor and pump mechanics, but for short periods and low-torque operation, it can be beneficial for such specific conditions as limitations of temperature, mechanical stress, and possible harm to the process (rise of temperature of pumped liquid and recirculation).

3.4.4 Resume

1. The most significant benefit of the experimental study is the possibility to run the pumping process close to the real conditions. Real hardware provides running the same physical phenomena, which can be observed in a real pumping system. The rated power of pumps is low, which emphasises the harmful effect of running the system at low speed. The reason is that the efficiency of low-power pumps is

relatively small. In undesirable regions of operation, efficiency drops are especially noticeable.

2. All the components of the test system are similar to those used in real pumping systems. The pumping process itself emulates the real processes that can be observed at boosting pumping stations. The ball valve enables simulation of variations of pipeline conductivity and hence, the real phenomena from the field of real pumping. Variations of flow demand and conductivity of the pipeline can be emulated manually due to the needs of the experiment.

3. The pump test stand enables real time observation of pumping parameters and reading of measuring equipment. The monitored values can be easily saved into the memory of measuring devices.

4. One significant disadvantage of the discussed pump test stand is its limited performance. The maximum achievable flow is quite low and the stand itself is quite narrowly focused. The configuration of the pipeline is not changeable, which limits the scope of possible experiments. The variation of the static head is impossible for that reason. The short pipeline enables no emulation of the phenomena of real long-piping systems. The dynamics of the system is simplified due to the length of the pipeline and absence of real components natural for large pumping systems. The static nature of the test stand allows no wide range load scenarios to be performed.

3.5 Summary of Chapter 3

1. Following the system efficiency estimation, two areas are attracting attention: the low-speed operation area and the high-speed operation one. It was shown that in the former, due to the additional running pumps, higher efficiency can be achieved than in the case of running a smaller number of pumps. In the latter, pumping under accurate torque control will provide many benefits by running a smaller number of pumps than in the case of running a larger number of pumps.

2. As a result, an efficiency map was produced intended for both the pump system control and the analysis on the design stage. After storing into the control unit memory, it becomes a core of the new control approach suitable for selection of an optimal number of working pumps and calculation of the required speed reference for these pumps.

3. The scenario of high-efficiency predictive control developed provides step-by-step instructions for the working point motion inside and outside the BER.

4. The new methodology proposed provides an ability to estimate the real-time efficiency of the pump taking into account drive losses. It is shown how an algorithm for the prediction the location of the working point can be implemented on the typical industrial PLC platform.

5. The pumping model developed is capable of combining models of the VSD with those a centrifugal pump in the common simulation environment with the ABB DriveSize. This enables experimentation with various VSD models.

6. The designed experimental setup provides the possibility to run the pumping process close to the real conditions. Real hardware imitates the same physical phenomena, which can be observed in a real pumping system.

CHAPTER 4. IMPROVEMENT OF AUXILIARY PROCESSES AND OPERATIONS

Pumping systems are widely used in all kinds of industry. In some applications, they operate under aggressive conditions and are subject to multiple harmful effects. System analysis of operation conditions show that such problems as dynamical stresses in mechanical parts, hydraulic hammer effects, high starting currents in driving motors are encountered in many cases (Stewart, 1977), (Ionel, 1986). In many types of industries, external conditions of pump operations are frequently changed, so that variations of liquid velocity and pressure in the pipeline alter significantly. In such conditions, the purpose of control system is to adjust the productivity of the pump in order to balance the process, at the same time providing the maximal pumping efficiency and minimizing the consumed power (Ahonen, 2011). In some cases, it requires reducing the dynamic torques and currents in the driving motor (Ali, 2009).

In variable-speed pumping applications, the pump control is implemented by continuous adjustment of the pump rotational speed and therefore, modification of the location of pump characteristic curve. Generally, pumping speed is adjusted due to the current process needs by means of a closed loop. The variable-speed pump is a final controlled object in such systems. Changing pressure or liquid velocity is recognised by control system as a deviation of control variables. When the error induced by the deviation exceeds the specific predefined value, the speed of pump is adjusted in order to maintain the required productivity (Dorf, 2001). A typical feature of a closed-loop system is that altering conditions require adjustment of the actuator.

In this chapter, some novelties are introduced to the auxiliary processes related to pumping based on the sensorless methods and prediction control environment described above.

4.1 Sensorless Pressure Measurement and Control

4.1.1 Problem statement

Readings of the flow rate and produced head enable to determine the operating point of a pump. There are several methods to measure the flow rate. They include flow velocity and differential pressure methods (Karassik, 1998). Pressure difference sensors across the pump enable measurements of the pump head while the rotational speed of the pump, liquid characteristics and the value of flow losses are known. This type of direct measurement using pressure sensors is most typical for pump control systems.

However, especially in small pumping applications, costs of flow rate and head sensors are comparable with those of the pump or driving motor (Pechenik, 2011). This is the reason of high interest in research and development of sensorless pumping systems, which require no flow or head sensors (Ahonen and Tamminen, 2012). In these systems, location of the working point can be estimated by the models of pumps tuned for specific operation. For this purpose, flow rate can be calculated based on the measured values of the rotational speed and the head of the pump. The measured values are to be applied to the pump model based on the characteristic curves of the pump. The characteristic curves are provided by the manufacturer of the pump. The advantage of this method lies in the use of the internal measurements provided by a frequency converter with no need to adopt additional measuring equipment.

An additional feature, which sensorless methods can provide, is a redundancy of the system, which can be provided by an additional pressure estimation method. As it was mentioned previously, various effects of the pumping process are able to damage the equipment, including the measuring devices. Failure of sensor is a significant damage, which leads to stopping of the whole application. In this situation, failure of sensor can be detected by protection mechanisms of the pumping application. However, even a worse situation can occur when the sensor is not totally broken but is damaged and continues to transmit the signal. In this case, there is a deviation between the real value and the data transmitted to the control system. Failure of the sensor cannot be detected in this situation, since the signal is continued to be transmitted to the control system. Operation based on a deviated feedback value is especially harmful for the process and equipment. Deviated feedback data are capable of bringing the system to undesirable regions of operation and providing an incorrect speed adjustment of the pump. In this case, only comparing the data received from the sensor to a signal coming from the backup sensor or sensorless calculation functionality can help to detect the deviation.

4.1.2 Scenario for the pressure estimation and control

In pressure control applications, a system tends to keep the pressure on the reference level (Viholainen, 2014). The speed of the pump is adjusted according to the current pressure and reference pressure relation. Typically, the PID controller is responsible for the rotational speed adjustment in pumping applications. The feedback for PID is a current pressure coming from the sensor installed on the pipeline. The value generated by a PID controller is an output value for the speed controller.

The total head H_T of the flowing liquid is expressed by the Bernoulli equation (2.1). This equation can be applied to estimate the pressure at any arbitrary point of the liquid flow (Finnemore, 2002). However, some assumptions and simplifications are made in order to use the equation. Particularly, it is assumed that the liquid induces no friction, characterised by zero viscosity, is laminar and incompressible. These assumptions are quite well justified for water applications.

The liquid velocity is defined by (2.2). Control units of contemporary frequency converters enable monitoring multiple parameters of the pumping process and the motor operation. Dependencies between such parameters of modulation as current speed of the motor and output power of the motor in combination with characteristics of the pump enable calculation of the specific parameters of a liquid flow.

The relation of modulation parameters and a flow are defined by pump characteristic curves. Particularly, the combination of pump on-shaft power vs. flow (PQ) curve and total head vs. flow (HQ) curve provide the connection between the output power of the motor and the hydraulic parameters. From the characteristic curves of the pump, the joint relation between the total head, flow and on-shaft power can be obtained. Therefore, as the on-shaft power of the pump is monitored, the total head can be estimated.

Generally, in many types of frequency converters, on-shaft power is among current parameters of the VSD, available for monitoring through user interface or acquiring through communication ports. When the value of the on-shaft power is known, the flow rate can be derived from the PQ curve by applying the interpolation method. The total head can be calculated applying the Bernoulli equation in which the flow and pressure are directly interconnected.

Since the method is applicable for variable-speed pumping, it must be considered that PQ and HQ curves are shifting on the graph pane depending on the speed. The affinity laws describe shifting of characteristic curves on the graph pane. Therefore, these laws must be taken into consideration when calculating the current flow from the PQ graph and the total head from the HQ graph.

4.1.3 Simulation of the method

The test bench equipped with a pressure sensor was used to verify the proposed methodology. The pressure sensor generates an electrical signal corresponding to the imposed pressure. Piezoresistive strain gauge is the most commonly used sensing technology in general-purpose measurements. The Piezoresistive effect of the formed strain gauges is used here to detect the strain due to induced pressure. In the test system, the strain gauges are connected to provide the Wheatstone bridge circuit, which reduces the sensitivity to errors and maximises the output signal of the sensor. The most typical pressure sensors in pumping applications provide 4 to 20 mA or 0 to 10 V at the output. Depending on the sensor type, from 2 to 4 wires are needed to connect the pressure sensor to a controller. The ACQ810 frequency converter was used in the testing. It includes all the required computing and signal conditioning software and hardware to pass the signal from the pressure sensor to the control program.

The main input variables for the pressure application are an process value (feedback from pressure sensor) and a setpoint (desired pressure). The control functionality adjusts the rotational speed of the pump in order to keep the process value as close to the setpoint as possible. The frequency converter estimates such variables as a current power and a current speed of motor. Additionally, the application control functionality collects and updates data needed for the pressure calculation. Depending on the process characteristics, requirements of the control system and operational state of the pump, minimal time interval for calculations can vary in the range of hundreds of milliseconds. Contemporary controllers provide time levels of program execution in the range of several milliseconds. For the changing flow rate estimation, an execution time of 10 ms is sufficient (ABB Drives, 2006).

To implement the calculations required for pressure estimation, a PLC was used. The algorithm of calculation is shown in Fig. 4.1.

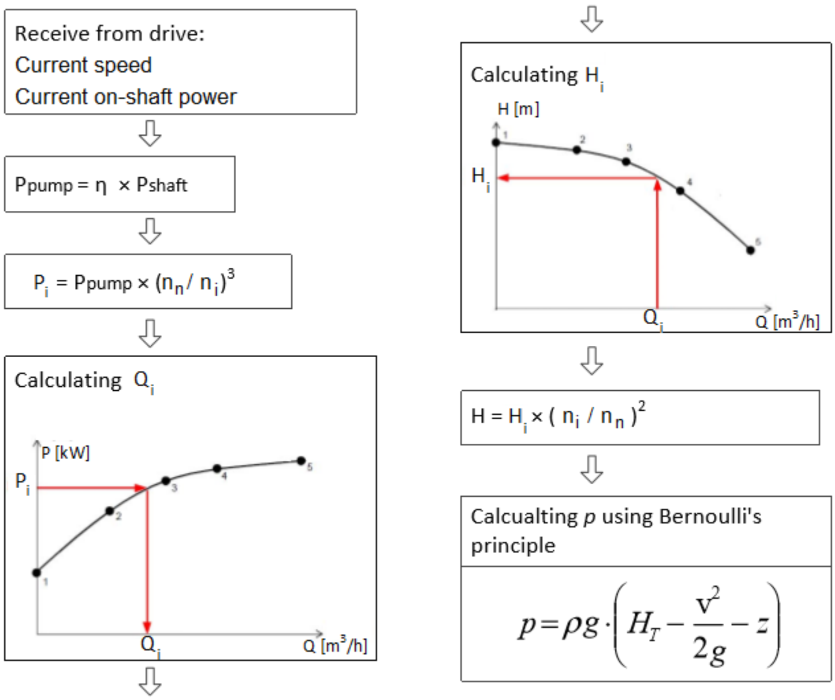


Fig 4.1 Algorithm of pressure estimation implemented in the PLC.

A test bench for implementation and verification of proposed method was arranged. It included the following components:

- *Frequency converter ABB ACQ810-04-02A7-4* 1.1 kW. The converter was equipped with the powerful control unit providing the overall control of the pumping process. The firmware of the converter contains the pump control functionalities and enables running the pressure control applications.
- *PLC ABB AC500,* PM 571. The execution time of the program was set to 10 ms. The duration of implementation of floating point operations was 6 μs. The data exchange between the PLC and the frequency converter was implemented via Profibus communication.

- *Pressure sensor Danfoss* 0-10 bar, 4-22mA.
- *Pump Ebara CDX 120/12.* Rated power 0.9 kW, rated current 3 A, rated voltage 400 V, and rated speed 2760 rpm.

The following data were obtained by the pressure estimation functionality (Fig. 4.2): on-shaft power of the pump, current speed of the motor, cross-sectional area of the pump inlet and outlet, efficiency of the pump-motor combination and density of the pumped liquid. The model of the pump is required for calculations. This model was implemented in the control functionality as a lookup table containing the key points of the pump HQ and PQ curves. Each curve was described by five key points. The coordinates of key points were hardcoded in the memory of the PLC. The points are relevant for the rated shape of PQ and HQ characteristic curves but can be transformed by the program using affinity transformation laws.

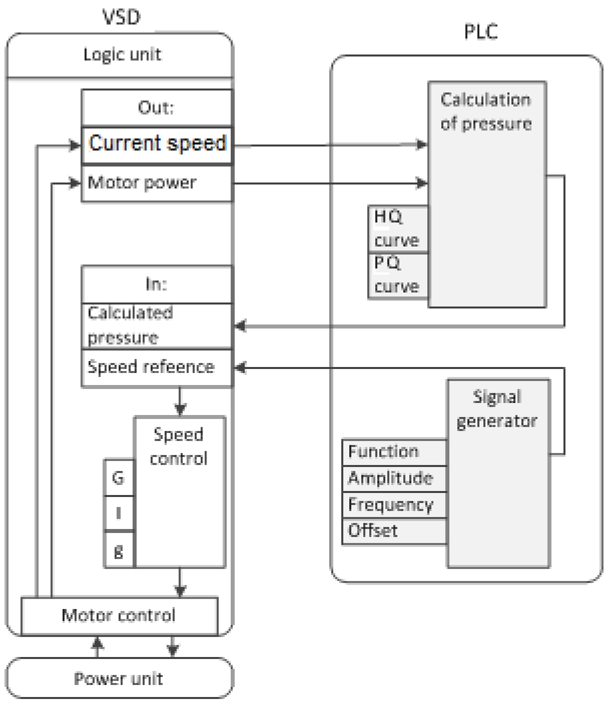


Fig. 4.2 Data flow between the PLC and the control unit of frequency converter.

The algorithm performs the following actions:

1. Retrieve the current speed and current on-shaft power from the frequency converter.
2. Transform the PQ curve according to the current speed of the pump.
3. Transform the HQ curve according to the current speed of the pump.
4. Find the current value of the flow rate corresponding to the current on-shaft power using the interpolation method applied to the PQ curve.

5. Calculate the total head based on the flow rate estimated in the previous step. The interpolation method applied to the HQ curve is used to estimate the value of the total head.
6. Calculate the values of pressure using the Bernoulli formula. The values of the total head, liquid density, and cross-sectional areas of the pump inlet and outlet were used in this calculation.

4.1.4 Experimental testing of the method

The testing system shown on Fig. 4.3 was arranged to test the performance of the proposed method. The test system is capable of running the testing sequences as described below.

During the testing, the value of the pressure obtained from the sensorless estimation method was compared against the value obtained from the common industrial pressure sensor. Both values were logged and plotted in the monitoring software tool of the frequency converter DriveStudio 3.2. Several test sequences were run in order to compare the readings of the calculated and the measured pressure in different operation conditions. In each test sequence, the pressure in the pipeline was varied according to a specific principle. Using this test method, all pressure values in the range provided by the given test setup were tested. This method enables detecting the calculation errors, their magnitudes, and nature. Occurrence of errors can be caused by an assumption that the pumping system is time invariant and linear (Europump, 1999).

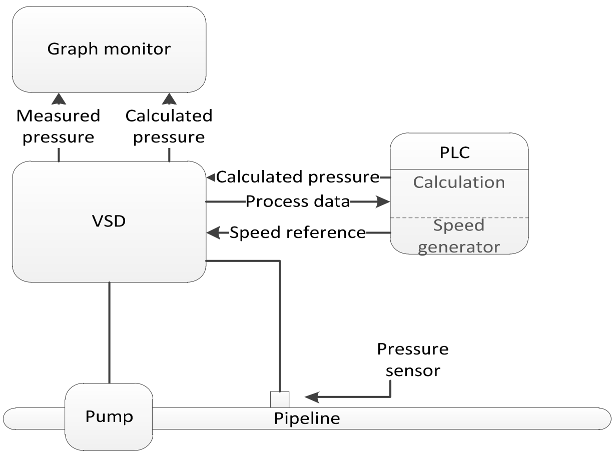


Fig. 4.3 VSD response analysis system.

The first-order response analysis system was used in order to evaluate the errors. Typical input signals were provided to implement the system analysis. The nature of the proposed signals corresponded to most frequently occurring pipeline phenomena in the common pumping operation:

- Step function which describes the hammering effect imitating a sharp pressure increase.
- Ramp function which describes processes occurring at start of the pumping and pipe filling also imitates the pressure growth fall during normal operation.
- Sine wave.

Although the function cannot describe exactly any of the phenomena occurring in the pumping system, it helps to reveal the errors caused by phase shifts from possible mistakes in the algorithm, lags of calculation, and integrated errors.

The variation of pressure was implemented by varying the pump rotational speed. The speed reference providing the pump speed variation was generated in the PLC by a standard software signal generator. The speed reference was sent from the PLC to the frequency inverter through the Profibus communication and was used as a VSD speed reference.

The test sequences containing generated speed references and corresponding responses are shown in Figs. 4.4, 4.5, and 4.6. It can be seen from the traces that the value of the calculated pressure has the same dynamics as that measured. The calculated value is stable, its form is similar to that measured. The value of the calculated pressure rises straight after the implementation of the step speed reference and the increase of the measured pressure value.

The step signal shown in Fig. 4.4 imitates a sudden sharp rise of pressure.

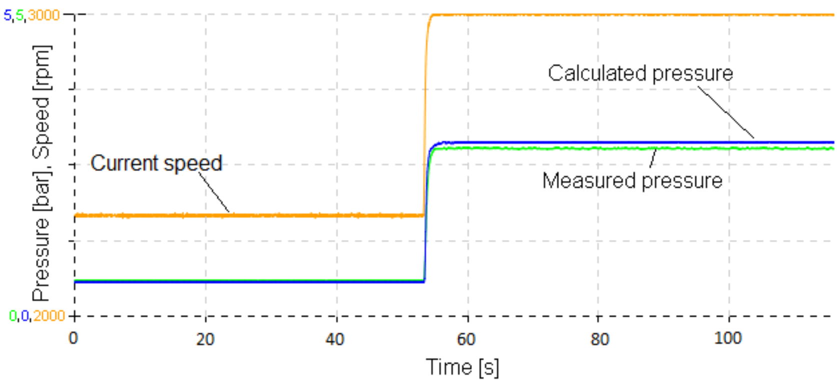


Fig. 4.4 System response to the step input.

The ramping test sequence is shown in Fig. 4.5. It simulates a slow increase of pressure during normal operation. The speed amplitude of this sequence is 1200 rpm.

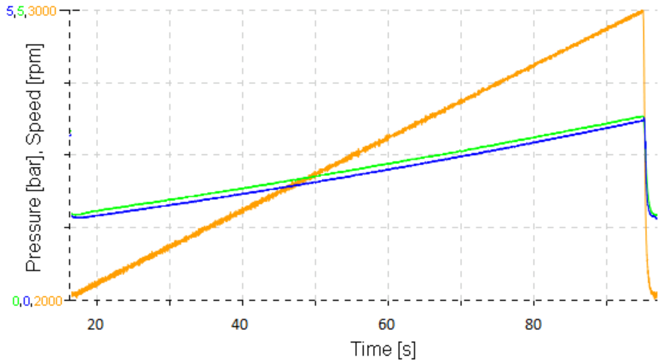


Fig. 4.5 System response to the ramp input.

The monitored signals show that the calculated pressure has the same dynamics as the measured one and is stable. The shape of the calculated pressure is similar to that measured though the linear error can be noticed. Also, a relatively high value of ripping was found at low calculated values.

The sine input (Fig. 4.6) was applied to reveal the phase errors, integrated errors and shifts. The speed variation of 1200 rpm was implemented in this test sequence.

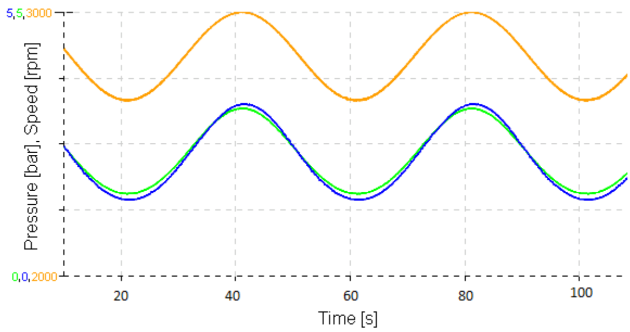


Fig. 4.6 System response to the sinewave input.

The traces indicate that the dynamics of the measured pressure and of that calculated are similar. The shape of the calculated pressure repeats the shape of that measured. However, a relatively high value of rippling is presented in the calculated value of pressure. In addition, a linear error is noticeable. Neither integrated nor the phase errors were detected.

The obtained results of the test runs indicate that the dynamics of the calculated pressure is similar to that measured. Rippling, which can be noticed at low values of calculated pressure, is most probably caused by an inaccuracy during the calculation when affinity transformations were implemented. The operations including 2nd and 3rd power involutions can be the reason of these inaccuracies. Alternation of current on-shaft power and current speed combined with almost flat PQ characteristic can also be a reason of rippling. The nature of errors indicates that filtering and a higher resolution of pump characteristic curves defined in the lookup-table are required.

4.1.5 Resume

1. The proposed sensorless pressure calculation algorithm successfully estimates the value of the pressure in various operation conditions.

2. The sensorless pressure control provides such advantages as increasing reliability in the case of failure of a primary sensor and due to the absence of such pump system elements like cables, connectors and transducers that can be a reason of malfunction. Also, additional cost savings are predicted due to possible exclusion of a pressure sensor from the architecture.

3. The algorithm was implemented based on the pump model. Such algorithm can be realised on the platform of contemporary controllers widely used in pumping applications for overall process control. The tune up of the model enables adjusting the algorithm to various types of applications and architectures.

4. However, the use of pump characteristic curves for the algorithm implementation induces the need to update the key points due to obsoleting of pump components and transformation of curves.

4.2 Sensorless Estimation of Liquid Density

4.2.1 Problem statement

As the water supply management is an important question, numerous of studies cover the developments in this area. Moreover, there are strict regulations for the parameters of the water flowing into the pipelines. It concerns the centrifugal pumps capable of transporting high-viscosity liquids and solid–liquid mixtures used in various industrial settings, including cement plants, sewage treatment plants, food plants, and multiple medical fields (Saito, 2012).

Parameters of the water flowing into these systems are generally below the permitted threshold value. During pumping, external conditions are usually alternated, so the liquid velocity and the pressure in the pipelines are changed.

Although centrifugal pumps can operate over a wide range of capacities, they commonly encounter difficulty at increased liquid density (Karassik, 1985). In general, high-density problems are worse for large high-energy pumps, for pumps handling hot or abrasives-laden liquids, and for pumps designed for high efficiency at best efficiency point. The source of the pump distress at increased liquid density is fourfold – thermal, hydraulic, mechanical, and abrasive wear. The thermal source results in inescapable energy conversion loss in the pump that warms the liquid. Because of the hydraulic source, when the flow decreases far enough, the impeller encounters suction or discharge recirculation, or perhaps both. Flashing, cavitations, and shock occur, often with vibration and serious damage. Because of the mechanical source, both constant and fluctuating loads in the radial and axial directions increase as pump capacity falls. Bearing damage, shaft and impeller breakage, and rubbing wear on casing, impeller, and wear rings can occur. As an abrasive wear is a concerned, liquid containing a large amount of abrasive particles, such as sand or ash, must flow continuously through the pump. In the case of their increased density, the particles can circulate inside the pump passage and quickly erode the impeller, casing, and even wear rings and shaft.

Resulting from the system analysis of working conditions in pumping systems, many other problems are encountered, such as hydraulic hammers, dynamic stresses in the mechanical parts, high starting currents in the driving motors, energy saving and other problems (Stewart, 1977), (Ionel, 1986).

To improve the situation, an on-line measuring equipment is often applied, the purpose of which is a dynamic water assessing and management to estimate the temporary state of the pipeline and to make a decision of its use (Hajnal, 2012). In these situations, the safe pump control must exclude pump damage (Chenghu, 2011).

This research describes a new method of on-line liquid density control for prevention of abnormal pumping performance. The density calculation is implemented by software running in the PLC, which acquires data from the VSD via the Profibus link. Density is estimated based on the Bernoulli's principle. The input data for the calculation are the speed and the input power of the pump, and the reading of the pressure sensor at the outlet of the pump.

4.2.2 Algorithmisation of density calculation

Using the relation between the on-shaft power and the current flow defined by the performance curves represented at Fig. 1.2, the system is capable of calculating the total head at the outlet of the pump. To this aim, the on-shaft power can be acquired as an output power measured among other relevant modulation parameters of the VSD.

The current flow rate can be obtained from the PQ curves. In the described system, the points obtained from the PQ and HQ curves were stored in the lookup table inside the calculating software. The flow rate was acquired by the interpolation method utilizing the triangles similarity rules. This approach is illustrated in Fig. 4.7.

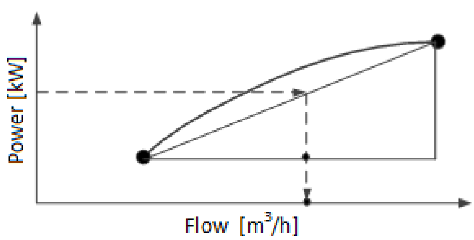


Fig. 4.7 Interpolation method using the triangles similarity rule.

The algorithm developed for the sensorless density calculation is presented in Fig. 4.8. To derive density, the current speed and on-shaft power of the pump are required as the input data. This data were acquired from the VSD, as mentioned above.

In this schema, the PQ curve is defined by five key points whereas those coordinates are hardcoded in the program. The HQ curve is defined in the similar manner. Using this approach, the lookup tables containing five points of the PQ curve and five points of the HQ curve are created in the program. The algorithm consists of the following steps:

1. Acquire the current speed and on-shaft power from the VSD.
2. Calculate the current input power of the pump using the affinity transformation and the efficiency coefficient of the pump-motor combination.
3. Estimate the value of the flow rate corresponding to the current power from the PQ graph using an interpolation principle.
4. Based on the HQ graph, calculate the total head being in line with the current flow found in the previous step. It is the value of the total head referred to the rated speed, which should be brought to the conformity with the current speed using the affinity laws.
5. Calculate the density using the Bernoulli's principle.

The following notations are used in the description of the algorithm:

P_i	current pump input power
P_{pump}	output power of pump/motor combination
η	pump efficiency
n_n	rated speed of pump
n_i	current speed of pump
Q_i	current flow
H_i	current head

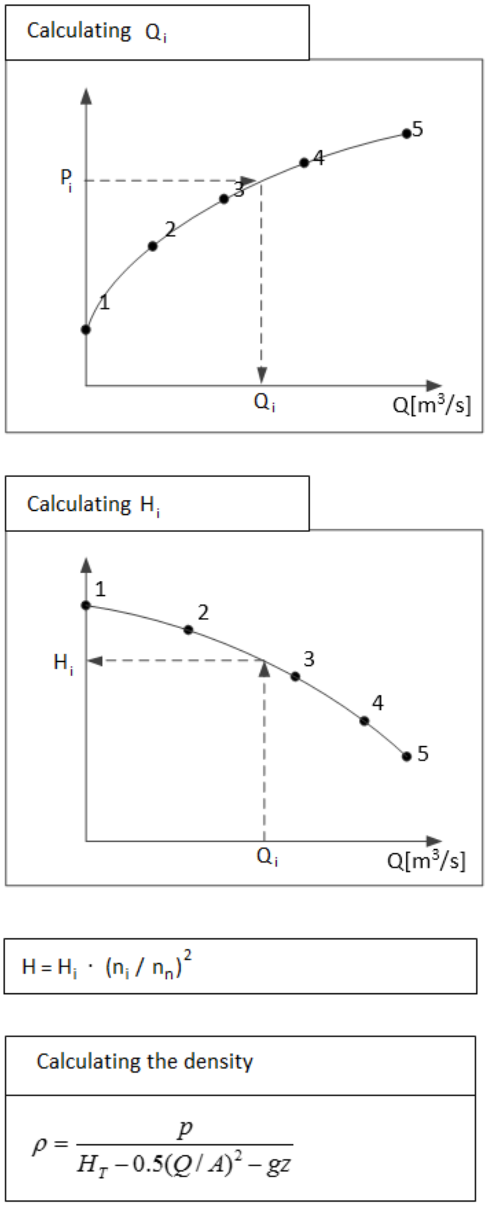


Fig. 4.8 Algorithm of density calculation.

4.2.3 System implementation and testing

For the density estimation, a PLC is used along with the Profibus link to obtain the current speed and current power from the VSD. The calculation data flow is shown in Fig 4.9.

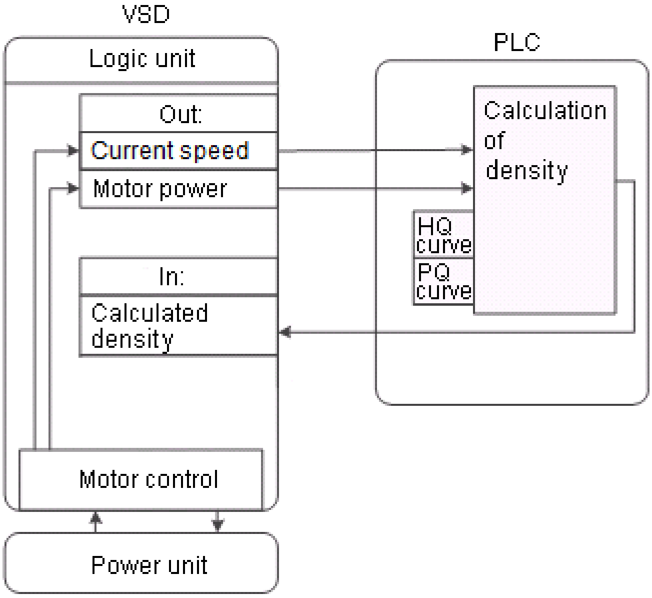


Fig. 4.9. Data flow between the VSD and the PLC.

To test the developed methodology, the testing sequence shown in Fig. 4.10 was arranged.

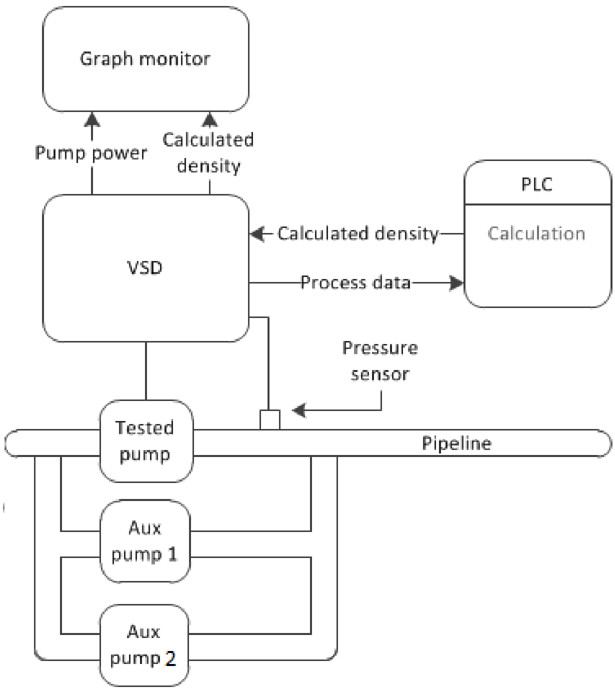


Fig. 4.10 Sequence of the VSD response analysis.

Calculations were implemented using the PLC. The values of calculated density were then transmitted to the VSD. The transmitted values were logged by means of the VSD monitoring functionality. During experimentation, the power alternation of the tested centrifugal pump was simulated via starting and stopping of two auxiliary pumps. This affected the loading of the tested pump, resulting in a changed input power.

The tested pump rotated at a constant speed. The auxiliary pumps were located in parallel with the tested pump. Both of them rotated with a constant speed. The switch-on of the auxiliary pump decreased the load at the tested pump. In this way, a liquid density grow was imitated. The switch-off of the auxiliary pump increased the load at the tested pump. In this way, a liquid density drop was imitated.

The auxiliary pumps were started gradually, in a cascade. They were stopped in the same way. In such a way, the complex loading at different levels of pumped liquid density was imitated.

The results of testing are shown in Fig. 4.11. Two stepping load alternations were simulated. First, the high load level was assigned (all auxiliary pumps were stopped), thus simulating the high density of the liquid. Herein, the input power of the tested motor was about 0.4 kW. Then, the auxiliary pumps were started one by one, which resulted in a two-step fall of density (and also the input power of the tested pump). Next, both auxiliary pumps were started, simulating the load grow at the tested pump, at which the power increased. According to the analysis, the designed software was properly reacting on the load change of the tested pump.

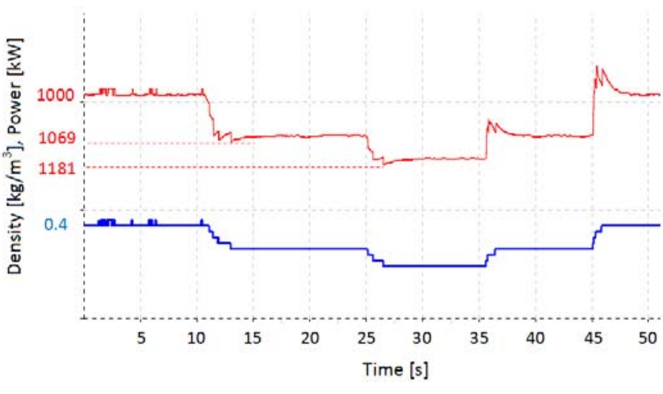


Fig. 4.11 Measuring the density. Simulating the density change by increase / decrease of the tested pump loading.

4.2.4 Resume

1. The new method of online liquid density control is intended to prevent an abnormal pumping performance. The proposed approach is suitable for pumping

systems, consisting of centrifugal pumps based on induction motors running by variable-speed drives.

2. The density calculation was implemented by software running in the PLC, which acquires data from the drive via the Profibus link.

3. The control algorithm was validated using the multi-pump test bench.

4.3 Slip Compensation of Pump Induction Drives

4.3.1 Problem statement

Pumps are variable load machines since their torque varies as a function of speed. They develop an increased torque as the speed grows and reduced torque as the speed drops below the rated level. The motor load which pumps represent is often discussed regarding the pump start-up behaviour, drive design, response to pump overload as well as losses of processed liquid (Stewart, 1977), (Karassik, 1985). However, controllability of the pumping drives relatively rarely becomes a subject of proper analysis.

The speed change is a reason of load alternations for pump drives. For this reason, traditional techniques of speed deviation compensation are less effective for pump and fan applications. In this study, a new way to adjust the speed of pump motors under conditions of variable speed and torque is proposed.

4.3.2 Related works

As it was stated in (Bose, 2001), the torque is proportional to some power of the rotational speed. For pumps, it is true that it is a square dependence between the speed and the torque. There is also a cubed dependence between the power and the speed. In (Ionel, 1986) it was stated that the load of centrifugal pumps represents a complex combination, including constant load components and a component which is periodically changing and depends on the speed or the angular position of the shaft. Hence, a load can be expressed as a value which alternates with respect to time. This variation can be repetitive and periodic.

Quite frequently, pumps operate under cyclically alternating loads (Smirnov, 2007), (Peretti, 2009). It is possible to divide the cyclically operating pumping applications into the following groups:

- Impact loads presented by apparent, repetitive and regular load pulses or peaks. These phenomena generally occur in wastewater applications (Vodovozov, 2012).
- Pulsating loads, represented by compressors and reciprocating pumps (Saito, 2012).
- Short-time intermittent loads, such as loads occurring at multiple forms of abnormal operations (Krutzsch, 1985).

This classification is not applicable to some types of pumps that are used for transporting high viscosity liquids and solid-liquid mixtures like cement, sewage water and food plants (Saito, 2012). In some cases, it is difficult to distinguish the impact loads from pulsating loads since the nature of both types is periodic.

In pumping applications, the accuracy of speed adjustment at load alternation is highly dependent on the control strategy of a frequency converter. The motor speed control in such applications is generally based on the scalar voltage-frequency technique since the fast torque response is not a crucial issue (Mohan, 2003), (Vodovozov, 2012). The speed adjustment capacity remains quite low because of open-ended topology of VSD-pump combination. Online measuring equipment can be applied in order to assess the load and evaluate the temporary liquid rate. However, the effect of this measurement is relatively low (Hajnal, 2012), (Ebrahim, 2010).

Such harmful phenomena in pumping applications like hydraulic hammer effects, overheating in driving motors and dynamic stresses in mechanical parts occur due to inaccuracies at speed-loaded alternations.

Risk of cavitation and magnitudes of hydraulic excitation forces on the impeller are minimised when a pump operates in the preferable region. Therefore, the reliability of a pump is at risk if speed-load variations become too high. The risk of mechanical damage and premature wear grows when the pump runs outside the preferable operation region. In (Ahonen, 2011) it is shown that operation at 70 or 115% of the BEP decreases the service life of pump for ten times. Naturally, location of the pump working point and its distance from the BEP is highly dependent on the pump speed.

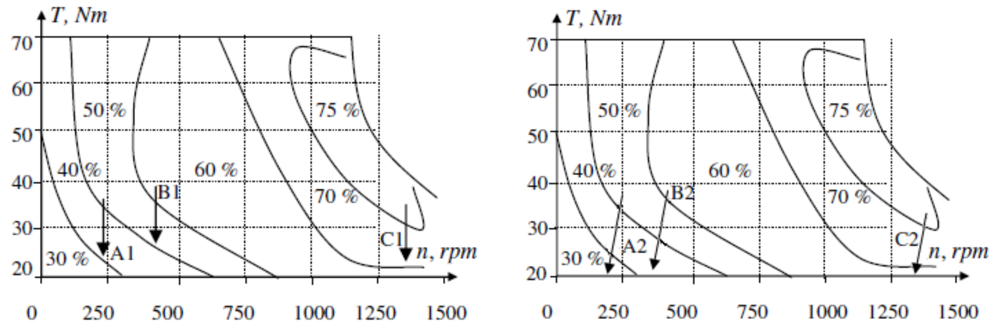
The speed errors also reduce the accuracy of sensorless calculations of such parameters as flow, pressure and density (Vodovozov, 2013), (Bakman, 2013).

Generally, 75% of rated value of torque load provides pump motor operation at maximum efficiency (Ionel, 1986). As it is shown in (Angers, 2009), the efficiency of a pump can drop significantly because of speed inaccuracy. Decrease of the motor speed is accompanied by an efficiency drop due to growing losses. Also, decreasing effect on the efficiency of a frequency converter feeding the motor is induced due to low speed operation. The overall efficiency tends to be decreased significantly when a pump is running at low speed since the load torque curve has a squared relation with the rotational speed of motor.

The efficiency maps are especially convenient for indicating the impact of speed inaccuracy of pump motors. The example of their use in order to express the speed inaccuracy problems can be found in (Angers, 2009).

Two situations are shown in Fig. 4.12. Both represent the 30% drop of torque. The situation in Fig. 4.12 (a) represents the load-independent speed levels of 250, 400 and 1350 rpm, respectively. The situation in Fig. 4.12 (b) relates to the load-dependent case where the speed drops by 10% due to the speed control inaccuracy.

Arrows A1, B1, C1 and A2, B2, C2 represent shifts of efficiency. It can be seen that in the first case, the efficiency is reduced by 10% at low speed (A1, B1) and by 5 % at high speed (C1). In the second case, the efficiency is reduced by 20% at low speed (A2, B2) and by 15% at high speed (C2). This analysis shows the destructive influence of speed inaccuracy on the overall efficiency of the application. The necessity of proper speed stabilisation is clear.



a. b.

Fig. 4.12 Efficiency maps of an induction motor in the torque-speed reference frame. The source data are taken from (P. Angers, 2009). The arrows indicate an efficiency change after the torque drop for stable speed case (a) and 10% speed inaccuracy case (b).

According to (Bose, 2001), (Vodovozov, 2012), (Böcker, 2007), a difference called absolute slip exists between the desired and current speeds of the motor. This absolute slip is defined as:

$$s = n_0 - n \tag{4.1}$$

where n_0 is the no-load motor speed. The slip occurs because of inability of the motor rotor to convert completely its magnetic field into an induced magnetic field in the rotor.

Traditional slip compensation methods proposed by many manufacturers of frequency converters are hardly applicable to pumps. In the majority of frequency inverters, the compensation is implemented on the basis of a constant parameter which a customer must predefine. In pumping applications, such compensation would be better expressed as a function of speed and torque.

4.3.3 Algorithmisation of slip compensation

The following methodology for slip compensation in load-speed changeable sensorless pump applications is proposed here.

The method is based on the predefined speed-torque lookup table, which is populated at the system tuning stage. On this stage, the speed sensors are temporarily attached to the pump motors. The experimental run is executed when the pumps accelerate from zero speed to maximal speed. One test run obtained during such a tuning stage is shown in Fig. 4.13.

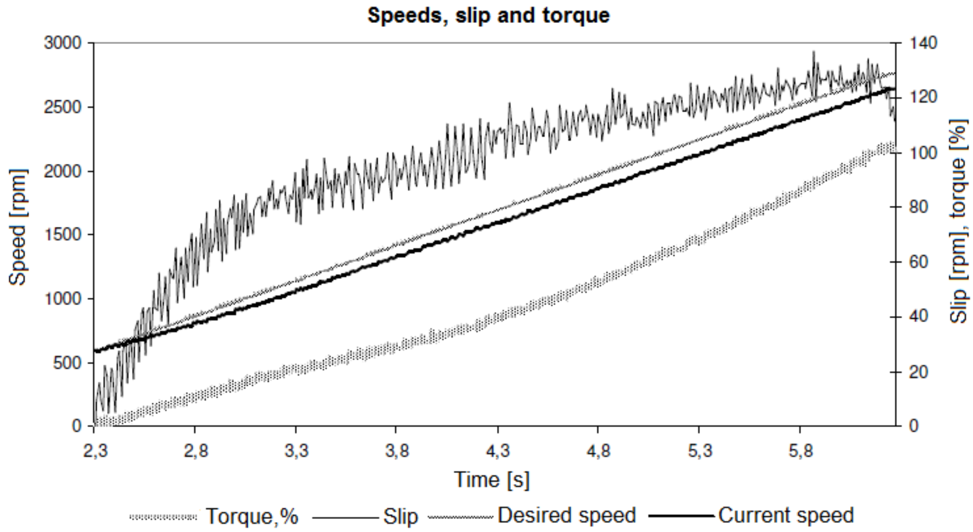


Fig. 4.13 Accelerating a pump during the tuning stage.

The speed and torque readings are logged during the tuning stage. From the logged readings, the speed-slip relations are estimated for different speed values and are registered in the lookup table. In order to implement that, the full speed range is divided into sectors. The average slip values are calculated for each sector. Fig. 4.14 shows the compensation values of different speed range sectors. These compensation values are stored to the lookup table of ten cells. Each compensation value represents an average slip of the particular sector.

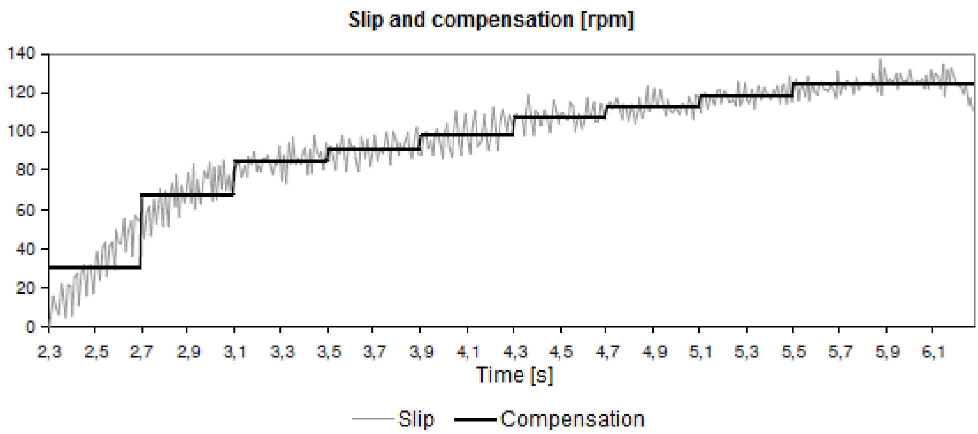


Fig. 4.14 Compensating the slip with compensation values from the lookup table.

Such a slip compensation method is effective only in the steady state mode. Each time when the reference changes rapidly, compensation must be disabled. It concerns the speed changes that occur at startup, breaking, or compensating the pressure fluctuations.

4.3.4 Experimental study of slip compensation

For an experimental study and verification of the proposed method, the test bench described earlier was used.

During the normal operation, compensating value (an average slip for the given speed) is retrieved from the lookup table and added to the speed reference in order to compensate the slipping. The diagram demonstrating the principle of such compensation is shown in Fig. 4.15.

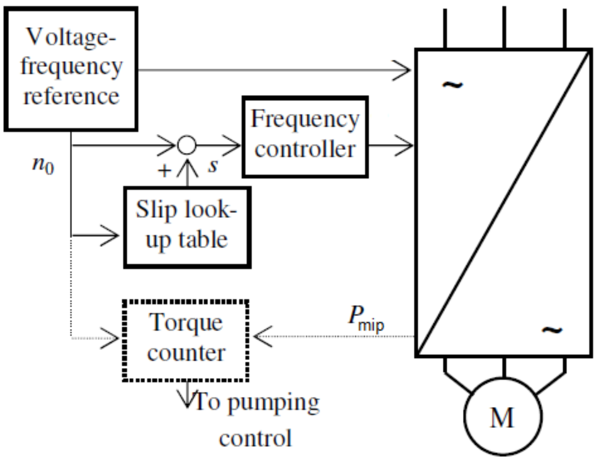


Fig. 4.15 Diagram of the sensorless slip compensation. P_{mip} is motor input power.

Figure 4.16 represents the slip and speed after compensation, using the proposed method. It shows that the average slip has decreased from 96 rpm (6% in Fig. 4.13) to 15 rpm (0.9% in Fig. 4.16).

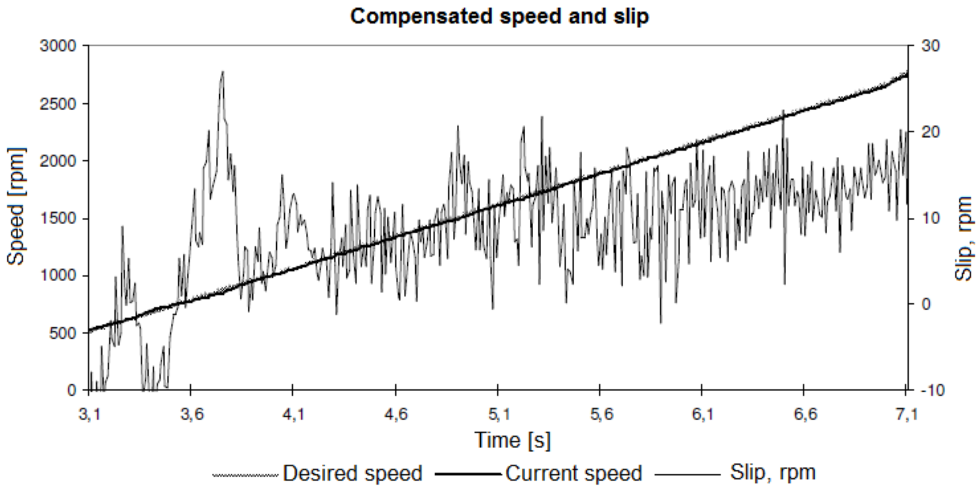


Fig. 4.16 Logged readings of slip and speed under the proposed compensation method.

In pumping applications operated in the scalar mode, this approach results in more stable operation closer to the BEP. It can keep the working point in the desired region of operation and hence avoid phenomena, which accompany the operation outside the preferred working region.

In contemporary frequency converters, the load torque T can be estimated for any speed as a parametric function electromagnetic torque T_2, electric power P_{mip} and idle torque ΔT as can be expressed in:

$$T = T_2 - \Delta T; \quad T_2 = \frac{P_{\text{mip}}}{n_0} \tag{4.2}$$

Where n_0 refers to synchronous (no-load) speed.

Taking into account (4.2), the lookup table must contain information about the torque for each speed. To populate such a table in this test run, two, three, four and five pumps were running step-by-step. From each run, the developed torque T and desired speed n_0 were retrieved. The diagram obtained in the test run for lookup table population is shown in Fig. 4.17.

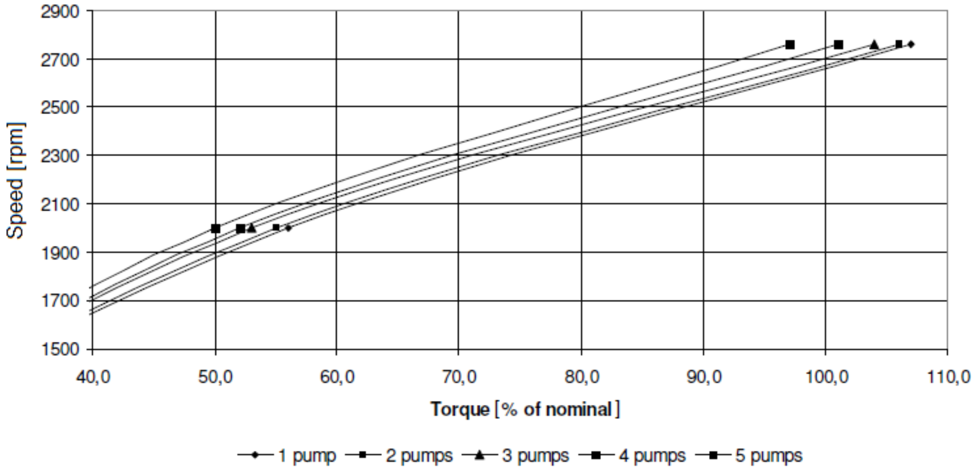


Fig. 4.17 Speed-torque characteristics of five pumps Ebara CDX 120/12.

4.3.5 Resume

1. Traditional techniques of speed deviation compensation are of low efficiency for pumping applications, which is shown here with the help of efficiency maps.

2. A new way to adjust the speed of pump motors under the conditions of variable speed and torque is proposed based on the predefined speed-torque lookup table, which is populated at the system tuning stage.

3. Experimental study proved the suitability of the offered method for pumping applications with a scalar control.

4.5 Summary of Chapter 4

1. The sensorless pressure control system developed provides such advantages as increased reliability in the case of failure of a primary sensor and absence of such pump system elements as cables, connectors and transducers that can be a reason of malfunction. Further, additional cost savings can be predicted due to possible exclusion of a pressure sensor from the architecture.

2. The new method of online liquid density control prevents abnormal pumping performance. The proposed approach is suitable for pumping systems consisting of centrifugal pumps based on induction motors running by variable-speed drives.

3. The new way to adjust the speed of pump motors under the conditions of variable speed and torque is based on the predefined speed-torque lookup table, which is populated at the system tuning stage. The approach is suitable for pumping systems consisting of centrifugal pumps with induction motors running by the scalar controlled VSDs.

References

ABB Product Notes. (2006). Flow Calculation in ABB Industrial Drives. Helsinki, Finland : ABB.

ABB. (2006). Using Variable Speed Drives (VSDs) in Pump Applications, ABB, Helsinki, Finland.

ABB. (2014) ABB Drives for Water and Wastewater ACQ810. ABB Oy.

ABB. (2015) ABB ACQ810 Firmware Manual: ABB Oy.

ABB. (2016). DriveSize. [WWW] http://new.abb.com/drives/software-tools/drivesize (20.11.2016).

Abrahamsen, F. (2000). Energy Optimal Control of Induction Motor Drives: Doctoral Dissertation. Aalborg, Aalborg University.

Ahola, J., Ahonen, T. and Tamminen, J. (2009). Consideration of recommended operating region for inverter-driven pumps. − *Proceedings of 6th International Conference on Condition Monitoring and Machinery Failure Prevention Technologies. Dublin, Ireland: CM 2009*, 951-962.

Ahonen, T. (2010). Estimation of pump operational state with model-based methods.− *Energy Conversion and Management*, 51 (6), 1319–1325.

Ahonen, T. (2011). Monitoring of Centrifugal Pump Operation by a Frequency Converter. Doctoral Thesis. Lappeenranta University of Technology, Finland.

Ahonen, T., R. Tiainen. (2008). Pump operation monitoring applying frequency converter. − *Proceedings of the 19th International Symposium on Power Electronics, Electrical Drives, Automation and Motion. Lill: Symposium on Power Electronics, Electrical Drives, Automation and Motion 2008*, 184-189.

Ahonen, T., Tamminen, J., Ahola, J. (2012). Frequency-converter-based hybrid estimation method for the centrifugal pump operational state.− *IEEE Trans. Ind. Electron*, 59 (12), 4803-4809.

Ali, A., Al-Soud, E., and Abdallah, E. (2009). Water pumping system with PLC and frequency control. − *Jordan Journal of Mechanical and Industria Engineering*, 3 (3), 216-221.

Angers, P. (2009). Variable Frequency Drive Testing. Motor Efficiency Performance Standards, Sydney, Australia.

Arribas, J. R. and González, C. M. V. (2002). Optimal vector control of pumping and ventilation induction motor drives.− *IEEE Trans. Ind. Electron*, 49 (4) , 889-895.

Austin, H. and Drury, B. (2013). Electric Motors and Drives, 4th edn. Oxford: Elsevier.

Bakman, I. (2013). Implementation and testing the sensorless pressure measurement of centrifugal pumps. − *Proceedings of 13th International Symposium "Topical Problems in the Field of Electrical and Power Engineering." Doctoral School of Energy and Geotechnology II. Pärnu, Estonia : 13th International Symposium "Topical Problems in the Field of Electrical and Power Engineering."*, 132-138.

Bakman, I. (2014). Control of electric drives in the best efficiency region of pumping system.− *The Scientific Journal of Riga Technical University: Electrical, Control and Communication Engineering*, 6 (4), 26-31.

Bakman, I. and Vodovozov, V. (2012). Multi-pump control applications for ABB ACQ810 industrial drive. − *Proceedings of 12th International Symposium "Topical Problems in the Field of Electrical and Power Engineering.". Kuressaare, Estonia: Topical Problems in the Field of Electrical and Power Engineering*, 37-44.

Bakman, I. Gevorkov, L. and Vodovozov, V. (2015). Efficiency control for adjustment of number of working pumps in multi-pump system. − *Proceedings of 9th International Conference-Workshop on Compatibility and Power Electronics CPE 2015. Caparica, Portugal: CPE 2015*, 396-402.

Barns, M. (2003). Practical Variable Speed Drives and Power Electronics. Oxford: Elsevier.

Barringer, P. (2003). A Life Cycle Cost Summary. − *Proceedings of International Conference of Maintenance Societies 2003. Perth, Australia : ICOMS-2003*, 1-10.

Böcker, J. and Mathapati, S. (2007). State of the art of induction motor control. − *Proceedings of IEEE International Electric Machines & Drives Conference IEMDC 2007. Antalia, Turkey : IEMDC 2007*, 1459-1464.

Bogumil, U. and Kahler, J. (2008). Modelling the efficiency and power characteristics of a pump group.− *Journal of Water Resources Planning and Management*, 88, 88-93.

Bortoni, C (2008). Optimization of parallel variable-speed-driven centrifugal pumps operation.− *Energy Efficiency*, 1 (3), 167-173.

Bose, B. K. (2001). Modern Power Electronics and AC Drives. New Jersey: Prentice Hall.

Buksnaitis, J. (2011). Analytical determination of mechanical characteristics of asynchronous motors by varying the electric current frequency. − *Proceedings of Electronics and Electrical Engineering. Kaunas : Technologija*, 6, 3-6.

Carlson, R. (2000). The correct method of calculating energy savings to justify adjustable-frequency drives on pumps.− *IEEE Trans. Ind. Appl.,* 36 (6), 1725-1733.

Carlson, R. (2000). The correct method of calculating energy savings to justify adjustable-frequency drives on pumps.− *IEEE Trans. Ind. Appl*, 36 (6) , 1725-1733.

Centrifugal and vertical pumps for allowable operating region. (1997). ANSI/HI 9.6.3: 1997. Parsippany, NJ: Hydraylic Institute.

Chazarra, M. (2014). Optimal operation of variable speed pumped storage hydropower plants participating in secondary regulation reserve markets.− *Proceedings of the 11th International Conference on the European Energy Market (EEM14), Krakow, Poland*, 1-5.

Chenghu, Z., Haiyan, W. and Dexing, S. (2011). Flow resistance and energy analysis of urban sewage delivery heat transfer system. − *Proceedings of 3rd International Conference on Measuring Technology and Mechatronics Automation ICMTMA 2011. Shanghai, China: ICMTMA 2011*, 234-237.

Danfoss. (2014). Danfoss AQUA Drive FC 202, Programming guide: Danfoss A/S.

DOE. (2008). Improving Motor and Drive System Performance: A Sourcebook for Industry. U.S. Department of Energy's (DOE).

Dorf, R.C. (2001). Modern Control Systems. Prentice Hall.

Ebrahim O. S., (2010). ANN-based optimal energy control of induction motor drive in pumping applications.− *IEEE Trans. Enr. Cons*., 25 (3), 652-660.

Europump and Hydraulic Institute. (1999). Attainable Efficiencies of Volute-Casing Pumps: A Reference Guide. Kidlington, Oxford, UK: Elsevier Advanced Technology.

Europump and Hydraulic Institute. (2004). Variable Speed Pumping – A Guide to Successful Applications. Kidlington, Oxford, UK: Elsevier.

Finnemore, J. E. (2002). Fluid Mechanics with Engineering Applications. NY, USA : McGraw Hill.

Gevorkov, L. (2015). Pressure Regulation in Centrifugal Pumping System with PLC. *15th International Symposium.− Proceedings of Topical Problems in the Field of Electrical and Power Engineering. Pärnu: Doctoral school of energy and geotechnology II*, 105−109.

Gevorkov, L., Vodovozov, V. and Lehtla, T. (2015). PLC-based flow rate control system for centrifugal pumps.− *2015 56th International Scientific Conference on Power and Electrical Engineering of Riga Technical University (RTUCON): Latvia, Riga:* 239-243.

Girdhar, P and Moniz. O. (2005). Practical Centrifugal Pumps. Oxford, Elsevier.

Grundfoss. (2015). CM, CME Horisontal Multistage Centrifugal Pumps. Product brocure : Grundfoss A/S.

Gülich, J.F. (2008). Centrifugal Pumps. Berlin, Germany : Springer-Verlag.

Hajnal, E., Lakner, G. and Ivanics, P. (2012). Real time control system for industrial waste water management. − *Proceedings of 16th International Conference on Intelligent Engineering Systems. Lisbon: ICIES*, 429-433.

Hammo, S. and Viholainen, J. (2008). Testing the accuracy of pump flow calculation without metering.− *World Pumps*, 2005 (471), 26-39.

Hammond P, W. (1984).A universal controller for parallel pumps with variable-frequency drives.− *IEEE Trans. Ind. Appl*, IA-20 (1), 203-208.

Hickok. H. (1985). Adjustable speed − a tool for saving energy losses in pumps, fans, blowers, and compressors.− *IEEE Trans. Ind. Appl.,* IA-21 (1), 124-136.

Hovstadius, G., Tutterow, V., and Bolles, S. (2005). Getting it right. Applying a system approach to variable speed pumping. − *Proceedings of ," 4th International Conference on Energy Efficiency in Motor Driven Systems EEMODS 2005. Heidelberg, Germany : EEMODS*, 19-27.

IEA. (2015), Electricity Information 2015, IEA, Paris: International Energy Agency.

International Energy Statistics, , [Online]. Available: http://www.eia.gov/cfapps/ipdbproject/IEDIndex3.cfm?tid=2&pid=2&ai d=7 (12.10.2013)

Ionel, I. I. (1986). Pumps and Pumping with Particular Reference to Variable Duty Pumps. Amsterdam: Elsevier.

ITT. (2007). PumpSmart Control Solutions PS200 v5: ITT.

Jones, W. V. (2013).Motor selection made easy: Choosing the right motor for centrifugal pump applications.− *IEEE Industry Applications Magazine*, 19 (6) , 36-45.

Karassik, I. J. and McGuire, T. (1998). Centrifugal Pumps, 2nd edn. New York: Chapman & Hall.

Karassik, I. J., Krutzsch, W. C. and Fraser, W. H. (1985). Pump Handbook: McGraw Book Company.

Karassik, I. J. and Messina, P. (2008). Centrifugal Pumps, 4nd edn. USA: McGraw-Hill.

Kini, P. G. and Bansal, R. , C. (2010). Effect of voltage and load variations on efficiencies of a motor-pump system.− *IEEE Trans. Energy Convers*, 25 (2) , 287-292.

Koor, M, Vassiljev. A (2014). Optimal pump count prediction algorithm for identical pumps working in parallel mode. − *Proceedings of 12th International Conference on Computing and Control for the Water Industry. Perugia, Italy: CCWI2013,* 951-958.

Koor, M. (2015). Water Distribution System Modelling and Pumping Optimization Based on Real Network of Tallinn. Doctoral Thesis. Tallinn University of Technology, Estonia.

Krutzsch, W. C., Fraser, W. H. and Messina, J. P. (1985). Pump Handbook. NY : McGraw Book Company.

Lawrence, D. (1996). Power Plant Engineering. New York : Chapman & Hall.

Liptak, B. (2005). Instrument Engineers' Handbook, Fourth Edition, Volume 2: Process Control and Optimization. Suite : CRC Press.

Mohan, N. (2003). First Course on Power Electronics and Drives. Minneapolis: MNPERE.

Nelik, L. (1999). Centrifugal and Rotary Pumps, 1st edn. New York: CRC Press LLC.

Nelik, L. (2005). What happens when a pump no longer operates at optimum conditions?.− *Pumps & Systems*, 54-63.

Nesbitt, B. (2006). Handbook of Pumps and Pumping. Burlington, USA: Elsevier Science.

Ostfeld, A. (2012). Water Supply System Analysis. New York: InTech.

Pechenik, M., Kiselychnyk, O., Buryan, S. (2011). Sensorless Control of Water Supply Pump Based on Neural Network Estimation. − Kiev Polytechnic Institute, 462-466. [WWW] www.nbuv.gov.ua/portal/natural/emeo/2011_79/462- 466.pdf (12.10.2013)www.nbuv.gov.ua/portal/natural/emeo/2011_79/462- 466.pdf ()

Peierls, R. (1980). Model-making in physics. — *Contemporary Physics*, 21, 3-17.

Peretti, L. and Zigliotto, M. (2009). Identification of mechanical load for electrical drives commissioning − Labelling machine case study. − *Proceedings of International IEEE Conference EUROCON 2009. St.-Petersburg. Russia: International IEEE Conference EUROCON 2009*, 797-803.

Petropol-Serb, G and Petropol-Serb, I. (2014). Control systems applied to analyze the environmental aspects in the activity of a wastewater treatment plant. - *Proceedings of International Conference on Applied and Theoretical Electricity. Craiova: ICATE 2014*, 1-5.

Pump Systems MatterTM and Hydraulic Institute. (2008). Optimizing Pumping Systems: A Guide for Improved Energy Efficiency, Reliability & Profitability. New York, USA : Elsevier.

Rice, D. (1988). A suggested energy-savings evaluation method for ac adjustable-speed drive applications.− *IEEE Trans. Ind. Appl.*, 24 (6), 1107-1117.

Rooks, J. A. and Wallace, A. K. (2004). Energy efficiency of VSDs.− *Electrical, Control and Communication Engineering*, 10 (3) , 57-61.

Saito, K., Hirayama, Y., Kimura, Y. and Nakamura, T. (2012). Development of a peristaltic pump based on bowel peristalsis improvement of closing area rates and suction pressure measurement. − *Proceedings of the 4th IEEE RAS/EMBS International Conference on Biomedical Robotics and Biomechatronics. Roma: IEEE RAS/EMBS*, 949-954.

Sarhan, H., Issa, R. and Alia, M. (2011). Slip compensation in efficiency-optimized three-phase induction motor drive systems. − *Procedings of the 9th World Congress on Intelligent Control and Automation. Taipei, Taiwan : WCICA 2011*, 95-99.

Sarlin. (1995). Submersible Pumps Handbook : Sarlin.

Schützhold, J. (2013). Design criteria for energy efficient pump drive systems. − *Proceedings of European Conference on Power Electronics and Applications. Lill: EPE 2013*, 1-10.

Smirnov, Y. V. (2007). Improving asynchronous electric drives of machines with a periodically varying load.− *Russian Electrical Engineering*, 78 (6), 287-290.

Stewart, L. (1977). Pumps. Indianapolis: Sams.

Stoffel, B. (2015). Assessing the Energy Efficiency of Pumps and Pump Units. New York: Elsevier.

Sulzer Pumps. (2000). AHLSTARTM Process Pumps, Intended Use. Mänttä, Finland: Sulzer Pumps Finland Oy.

Szychta, L. and Figura, R. (2012). Analysis of efficiency characteristics of squirrel-cage induction motor for pump applications. − *Proceedings of 20th International Conference on Electrical Machines ICEM 2012. Perth, Australia : ICEM 2012*, 73-78.

Viholainen, J., Tamminen, T. and Ahonen, J. (2013). Energy-efficient control strategy for variable speed-driven parallel pumping systems.− *Energy Efficiency*, 6, 495-509.

Viholainen, J. (2014). Energy-efficient control strategies for variable speed driven parallel pumping system based on pump operation point monitoring with frequency converters, Doctoral Thesis. Lappeenranta University of Technology, Finland.

Vodovozov, V. (2012). Electric Drive Systems and Operation. London: Bookboon.

Vodovozov, V. (2014). Electrical Drive: Performance, Design and Control. Saarbrücken, Germany: LAP Lambert Academic Publishing.

Vodovozov, V. and Bakman, I. (2013). Control of liquid density to prevent abnormal pumping performance.− *Proceedings of the 7th WSEAS International Conference on Waste Management, Water Pollution, Air Pollution, Indoor Climate (WWAI '13). Lemesos, Cyprus: WWAI '13*, 217−222.

Vodovozov, V. and Bakman, I. (2014). Sensorless pressure calculation for parallel redundancy in pumping systems. – *Proceedings of 2014 16th European Conference on Power Electronics and Applications: CPE 2014*.

Vodovozov, V., Gevorkov, L. and Raud, Z. (2015). PLC-based pressure control in multi-pump applications.− *Electrical, Control and Communication Engineering*, 9 (1) , 23-29.

Volk, M. (2005). Pump Characteristics and Applications. NW, Suite: CRC Press.

Wiegand, H. A. And Lee, B. (1960). Characteristics of centrifugal pumps and compressors which affect the motor driver under transient conditions.− *American Institute of Electrical Engineers, Transactions of the Applications and Industry*, 79 (3) , 150-156.

Yang, Z. and Borsting H. (2010). Energy efficient control of a boosting system with multiple variable-speed pumps in parallel. − *Proceedings of 49th IEEE Conference on Decision and Control. Atlanta: CDC 2010*, 2198-2203.

Publications of the Author

Bakman, I., Gevorkov, L. and Vodovozov, V. (2015). Efficiency control for adjustment of number of working pumps in multi-pump system.− *Proceedings of the 9th International Conference on Compatibility and Power Electronics (CPE).* Lisbon: CPE 2015, 396-402.

Vodovozov, V., Raud, Z., Bakman, I., Lehtla, T. and Gevorkov, L. (2015). Hardware-in-the-loop simulator of vessel electric propulsion drive.− *Proceedings of the 9th International Conference on Compatibility and Power Electronics (CPE)*. Lisbon: CPE 2015, 425-430.

Gevorkov, L., Vodovozov, V., Lehtla, T. and Bakman, I. (2015). PLC-based flow rate control system for centrifugal pumps.− *Proceedings of the 56th International Scientific Conference on Power and Electrical Engineering of Riga Technical University (RTUCON)*. Riga: RTUCON 2015, 239-243.

Bakman, I. and Gevorkov, L. (2015). Speed control strategy selection for multi-pump systems.− *Proceedings of the 56th International Scientific Conference on Power and Electrical Engineering of Riga Technical University (RTUCON)*. Riga: RTUCON 2015, 353-356.

Bakman, I. (2014). Control of electric drives in the best efficiency region of pumping system.− *The Scientific Journal of Riga Technical University: Electrical, Control and Communication Engineering*, 6 (4), 26-31

Gevorkov, L., Bakman, I. and Vodovozov, V. (2014). Hardware-in-the-loop simulation of motor drives for pumping applications.− *Proceedings of the 2014 Electric Power Quality and Supply Reliability Conference (PQ)*. Rakvere, Estonia: PQ 2014, 203-209.

Bakman, I., Gevorkov, L. and Vodovozov, V. (2014). Optimization of method of adjustment of productivity of multi-pump system containing directly connected motors.− *Proceedings of the 2014 Electric Power Quality and Supply Reliability Conference (PQ)*. Rakvere, Estonia: PQ 2014, 209-215.

Bakman, I., Gevorkov, L. and Vodovozov, V. (2014). Predictive control of a variable-speed multi-pump motor drive.− *Proceedings of the IEEE 23rd International Symposium on Industrial Electronics (ISIE 2014)*. Istanbul: ISIE 2014, 1409-1414.

Vodovozov, V. and Bakman, I. (2014). Sensorless pressure calculation for parallel redundancy in pumping systems.− *Proceedings of the EPE'14 ECCE Europe. Lappeenranta, Finland: EPE'14 ECCE Europe*, 1-9.

Bakman, I. (2014). Using efficiency readings for choosing the number of running pumps in multi-pump system.− *Proceedings of the 11th International Symposium "Topical Problems in the Field of Electrical and Power Engineering". Tallinn: Topical Problems in the Field of Electrical and Power Engineering*, 132-138.

Bakman, I. and Vodovozov, V. (2013). Sensorless pressure control of centrifugal pumps.− *Proceedings of the 8th International Conference-Workshop on Compatibility and Power Electronics CPE 2013. Ljubljana, Slovenia: CPE 2013*, 304−309.

Vodovozov, V. and Bakman, I. (2013). Performance improvement of pumps fed by the variable speed drives. − *The Scientific Journal of Riga Technical University - Electrical, Control and Communication Engineering*, 4 (7), 45-51.

Vodovozov, V. and Bakman, I. (2013). Control of liquid density to prevent abnormal pumping performance.− *Proceedings of the 7 th WSEAS International Conference on Waste Management, Water Pollution, Air Pollution, Indoor Climate (WWAI '13). Lemesos, Cyprus: WWAI '13*, 217−222.

Bakman, I. (2013). Implementation and testing the sensorless pressure measurement of centrifugal pumps.− *Proceedings of the 11th International Symposium "Topical Problems in the Field of Electrical and Power Engineering". Doctoral School of Energy and Geotechnology II. Pärnu, Estonia: Topical Problems in the Field of Electrical and Power Engineering*, 132-138.

Vodovozov, V. and Bakman, I. (2013). Slip compensation in sensorless scalar electric drive of a pump.− *Proceedings of the 54th International Scientific Conference of Riga Technical University. Riga: 54th International Scientific Conference of Riga Technical University*, 27.1-27.4.

Bakman, I. and Vodovozov, (2012). Multi-pump control applications for ABB ACQ810 industrial drive.− *Proceedings of the12th International Symposium "Topical Problems in the Field of Electrical and Power Engineering". Doctoral School of Energy and Geotechnology II. Kuressaare, Estonia: Topical Problems in the Field of Electrical and Power Engineering*, 37-44.

ABSTRACT

High-Efficiency Predictive Control of Centrifugal Multi-Pump Stations with Variable-Speed Drives

Focus in this doctoral thesis is on centrifugal pumping stations for liquid distribution in industrial, commercial, and residential applications. Because of relatively low efficiency, an increasing energy market and growing energy costs, new solutions for pumping technologies are searched. The aim is to establish a novel control approach targeted to high efficiency of energy use and distribution across multiple centrifugal pumping applications. The scientific contribution of the thesis lies in the following: a concept of system architecture for mutual load- and speed-dependent smart multi-pump control, a methodology of efficiency monitoring and intellectual predictive control as a basis for subsequent hardware and software organisation, and an algorithm for expeditious selection of the number of working pumps and their speeds optimal from the viewpoint of process efficiency. The practical value of the thesis is in a simulation model and a methodology for exploring pumping and preparing software modules for the real control environment, an original experimental setup and a methodology for the verification and tuning of the developed tools. The direct practical outcomes of the thesis represent a set of improved applications for auxiliary processes and operations: sensorless pressure measurement, sensorless liquid density estimation, and slip compensation as well as a library of ready-to-use tools to enhance existing pumping applications.

Keywords: centrifugal pump, variable-speed drive, predictive control, energy efficiency

KOKKUVÕTE

Pumbajaamade muudetava kiirusega tsentrifugaalpumpajamite kõrgefektiivne ennetav juhtimine

Käesolevas doktoritöös keskendutakse tsentrifugaalpumpadega pumbajaamade kasutamisele tööstuses, kaubandussektoris ja elamute veevarustuses. Suhteliselt madala efektiivsuse, üha suureneva energiaturu ning kasvavate energiahindade tõttu otsitakse pumpamisel uusi tehnoloogilisi lahendusi. Eesmärgiks on luua uudne kontrollsüsteem, mille fookuses on võimalikult kõrge energiakasutuse efektiivsusaste erinevates tsentrifugaalmasinate pumbarakenduste tüüpides.

Doktoritöö peamine teaduslik panus on uudne pumbasüsteemi juhtimise arhitektuuri kontseptsioon, mis pakub mitme pumba töö koormusest ja kiirusest sõltuva vastastikku targa kontrolli. Samuti on suurt tähelepanu pööratud efektiivsuse jälgimise metoodikale ja intellektuaalsele ennustavale kontrollile, mis on aluseks tulenevale riist- ja tarkvara rakendamisele, ning algoritmile, mis teeb kiire valiku töötavate pumpade töösoleku arvu ja nende kiiruse kohta, lähtudes protsessi efektiivsusest optimaalsuse vaatepunktist.

Väitekirja praktiliseks väärtuseks on pumpamise simulatsioonimudel, pumpamise uurimise ning tarkvaramoodulite valmistamise metoodika reaalse kontrollkeskkonna tarbeks, originaalne eksperimentaalne disain ja väljatöötatud vahendite kontrollimise ja häälestamise metoodika.

Lõputöö praktilised tulemused sisaldavad täiustatud lisatoiminguid ning tsentrifugaalpumba töösüsteemi rõhu mõõtmise ja vedeliku tiheduse hindamise metoodikat ilma rõhuanduriteta, pumba libistuse kompensatsioonimeetodit ning tarkvaralisi teeke olemasolevate pumpamissüsteemide efektiivsuse parandamiseks.

Märksõnad: tsentrifugaalpump, elektriajam, ennustav kontroll, energiaefektiivsus

ANNEX

Curriculum Vitae

1. Personal data

Name	Ilja Bakman
Date and place of birth	01.08.1981, Tallinn, Estonia

2. Contact information

Phone	+372 56 801 408
E-mail	ilja.bakman@ttu.ee

3. Education

Educational institution	Graduation year	Education (field of study/degree)
Tallinn University of Technology	2012	Electric drives and power electronics / Master of Science
Negev Academical College of Engineering	2005	Electrical and Electronics Engineering / Bachelor of Science
Jewish Gymnasium of Tallinn	1999	Basic

4. Language competence/skills (fluent; average, basic skills)

Language	Level
Estonian	Good
Russian	Fluent
English	Good
Hebrew	Good

5. Professional employment

Period	Organisation	Position
2008 - ...	ABB AS	Automation engineer
2008 - 2008	AMS Electronics	Process engineer
2005 - 2008	IDF, Israel	Instructor

6. Defended thesis

“Pump control application for ACQ810 variable speed drive”. Master thesis, TTU, 2012.

Design of DC-DC Buck convertor with non-linear capacitor applying the limit control method. Bachelor thesis, Negev Academical College of Engineering, 2005.

7. Main areas of scientific work

Natural Sciences and Engineering, Electrical Engineering and Electronics

LISA

Elulookirjeldus

1. Isikuandmed

Ees- ja perekonnanimi	Ilja Bakman
Sünniaeg ja -koht	01.08.1981, Tallinn, Eesti
Kodakondsus	Eesti

2. Kontaktandmed

Aadress	
Telefon	+372 56 801 408
E-posti aadress	ilja.bakman@ttu.ee

3. Hariduskäik

Õppeasutus (nimetus lõpetamise ajal)	Lõpetamise aeg	Haridus (eriala/kraad)
Tallinna Tehnika Ülikool	2012	Elektriajamid ja jõuelektroonika/ Tehnikateaduse magister
Negev Akadeemiline Inseneeria Kõrgkool	2005	Elektrotehnika ja Elektroonika
Tallinna Juudi kool	1999	Põhiharidus

4. Keelteoskus (alg-, kesk- või kõrgtase)

Keel	Tase
Eesti	Kõrgtase
Vene	Emakeel
Inglise	Kesktase
Heebrea	Kõrgtase

5. Teenistuskäik

Töötamise aeg	Tööandja nimetus	Ametikoht
2008 - ...	ABB AS	Automaatika insener
2008 - 2008	AMS Electronics	Tehnoloog
2005 - 2008	IDF, Iisrael	Instruktor

6. Kaitstud lõputööd

“Pumpade juhtimise rakendustarkvara sagedusmuundurile ACQ810”. Bakalaurusetöö, TTU, 2012.

“DC-DC Buck konverteri mittelieaar kondensaatoriga projekteerimine kasutades piir kontrolli meetodi“. Magistritöö, Negev Akadeemiline Inseneeria Kõrgkool.

7. Teadustöö põhisuunad

Loodusteadused ja -tehnika, energeetikaalased uuringud, energeetika

DISSERTATIONS DEFENDED AT TALLINN UNIVERSITY OF TECHNOLOGY ON *POWER ENGINEERING, ELECTRICAL ENGINEERING, MINING ENGINEERING*

1. **Jaan Tehver**. Boiling on Porous Surface. 1992.
2. **Aleksandrs Cars**. Woodhips Combustion Technology. 1992.
3. **Endel Risthein**. Electricity Supply of Industrial Plants. 1993.
4. **Tõnu Trump**. Some New Aspects of Digital Filtering. 1993.
5. **Vello Sarv.** Synthesis and Design of Power Converters with Reduced Distortions Using Optimal Energy Exchange Control. 1994.
6. **Ivan Klevtsov**. Strained Condition Diagnosis and Fatigue Life Prediction for Metals under Cyclic Temperature Oscillations. 1994.
7. **Ants Meister**. Some Phase-Sensitive and Spectral Methods in Biomedical Engineering. 1994.
8. **Mati Meldorf**. Steady-State Monitoring of Power System. 1995.
9. **Jüri-Rivaldo Pastarus**. Large Cavern Stability in the Maardu Granite Deposit. 1996.
10. **Enn Velmre**. Modeling and Simulation of Bipolar Semiconductor Devices. 1996.
11. **Kalju Meigas**. Coherent Photodetection with a Laser. 1997.
12. **Andres Udal**. Development of Numerical Semiconductor Device Models and Their Application in Device Theory and Design. 1998.
13. **Kuno Janson**. Paralleel- ja järjestikresonantsi parameetrilise vaheldumisega võrgusageduslik resonantsmuundur ja tema rakendamine. 2001.
14. **Jüri Joller**. Research and Development of Energy Saving Traction Drives for Trams. 2001.
15. **Ingo Valgma**. Geographical Information System for Oil Shale Mining – MGIS. 2002.
16. **Raik Jansikene**. Research, Design and Application of Magnetohydrodynamical (MHD) Devices for Automation of Casting Industry. 2003.
17. **Oleg Nikitin**. Optimization of the Room-and-Pillar Mining Technology for Oil-Shale Mines. 2003.
18. **Viktor Bolgov**. Load Current Stabilization and Suppression of Flicker in AC Arc Furnace Power Supply by Series-Connected Saturable Reactor. 2004.
19. **Raine Pajo**. Power System Stability Monitoring – an Approach of Electrical Load Modelling. 2004.

20. **Jelena Shuvalova**. Optimal Approximation of Input-Output Characteristics of Power Units and Plants. 2004.

21. **Nikolai Dorovatovski**. Thermographic Diagnostics of Electrical Equipment of Eesti Energia Ltd. 2004.

22. **Katrin Erg**. Groundwater Sulphate Content Changes in Estonian Underground Oil Shale Mines. 2005.

23. **Argo Rosin**. Control, Supervision and Operation Diagnostics of Light Rail Electric Transport. 2005.

24. **Dmitri Vinnikov**. Research, Design and Implementation of Auxiliary Power Supplies for the Light Rail Vehicles. 2005.

25. **Madis Lehtla**. Microprocessor Control Systems of Light Rail Vehicle Traction Drives. 2006.

26. **Jevgeni Šklovski**. LC Circuit with Parallel and Series Resonance Alternation in Switch-Mode Converters. 2007.

27. **Sten Suuroja**. Comparative Morphological Analysis of the Early Paleozoic Marine Impact Structures Kärdla and Neugrund, Estonia. 2007.

28. **Sergei Sabanov**. Risk Assessment Methods in Estonian Oil Shale Mining Industry. 2008.

29. **Vitali Boiko**. Development and Research of the Traction Asynchronous Multimotor Drive. 2008.

30. **Tauno Tammeoja**. Economic Model of Oil Shale Flows and Cost. 2008.

31. **Jelena Armas**. Quality Criterion of road Lighting Measurement and Exploring. 2008.

32. **Olavi Tammemäe**. Basics for Geotechnical Engineering Explorations Considering Needed Legal Changes. 2008.

33. **Mart Landsberg**. Long-Term Capacity Planning and Feasibility of Nuclear Power in Estonia under Certain Conditions. 2008.

34. **Hardi Torn**. Engineering-Geological Modelling of the Sillamäe Radioactive Tailings Pond Area. 2008.

35. **Aleksander Kilk**. Paljupooluseline püsimagnetitega sünkroongeneraator tuuleagregaatidele. 2008.

36. **Olga Ruban**. Analysis and Development of the PLC Control System with the Distributed I/Os. 2008.

37. **Jako Kilter**. Monitoring of Electrical Distribution Network Operation. 2009.

38. **Ivo Palu**. Impact of Wind Parks on Power System Containing Thermal Power Plants. 2009.

39. **Hannes Agabus**. Large-Scale Integration of Wind Energy into the Power System Considering the Uncertainty Information. 2009.

40. **Kalle Kilk**. Variations of Power Demand and Wind Power Generation and Their Influence to the Operation of Power Systems. 2009.

41. **Indrek Roasto**. Research and Development of Digital Control Systems and Algorithms for High Power, High Voltage Isolated DC/DC Converters. 2009.

42. **Hardi Hõimoja**. Energiatõhususe hindamise ja energiasalvestite arvutuse metoodika linna elektertranspordile. 2009.

43. **Tanel Jalakas**. Research and Development of High-Power High-Voltage DC/DC Converters. 2010.

44. **Helena Lind**. Groundwater Flow Model of the Western Part of the Estonian Oil Shale Deposit. 2010.

45. **Arvi Hamburg**. Analysis of Energy Development Perspectives. 2010.

46. **Mall Orru**. Dependence of Estonian Peat Deposit Properties on Landscape Types and Feeding Conditions. 2010.

47. **Erik Väli**. Best Available Technology for the Environmentally Friendly Mining with Surface Miner. 2011.

48. **Tarmo Tohver**. Utilization of Waste Rock from Oil Shale Mining. 2011.

49. **Mikhail Egorov**. Research and Development of Control Methods for Low-Loss IGBT Inverter-Fed Induction Motor Drives. 2011.

50. **Toomas Vinnal**. Eesti ettevõtete elektritarbimise uurimine ja soovituste väljatöötamine tarbimise optimeerimiseks. 2011.

51. **Veiko Karu**. Potential Usage of Underground Mined Areas in Estonian Oil Shale Deposit. 2012.

52. **Zoja Raud**. Research and Development of an Active Learning Technology for University-Level Education in the Field of Electronics and Power Electronics. 2012.

53. **Andrei Blinov**. Research of Switching Properties and Performance Improvement Methods of High-Voltage IGBT based DC/DC Converters. 2012.

54. **Paul Taklaja**. 110 kV õhuliinide isolatsiooni töökindluse analüüs ja töökindluse tõstmise meetodid. 2012.

55. **Lauri Kütt**. Analysis and Development of Inductive Current Sensor for Power Line On-Line Measurements of Fast Transients. 2012.

56. **Heigo Mõlder**. Vedelmetalli juhitava segamisvõimaluse uurimine alalisvoolu kaarleekahjus. 2012.

57. **Reeli Kuhi-Thalfeldt**. Distributed Electricity Generation and its Possibilities for Meeting the Targets of Energy and Climate Policies. 2012.

58. **Irena Milaševski**. Research and Development of Electronic Ballasts for Smart Lighting Systems with Light Emitting Diodes. 2012.

59. **Anna Andrijanovitš**. New Converter Topologies for Integration of Hydrogen Based Long-Term Energy Storages to Renewable Energy Systems. 2013.

60 **Viktor Beldjajev**. Research and Development of the New Topologies for the Isolation Stage of the Power Electronic Transformer. 2013.

61. **Eduard Brindfeldt**. Visually Structured Methods and Tools for Industry Automation. 2013.

62. **Marek Mägi**. Development and Control of Energy Exchange Processes Between Electric Vehicle and Utility Network. 2013.

63. **Ants Kallaste**. Low Speed Permanent Magnet Slotless Generator Development and Implementation for Windmills. 2013.

64. **Igor Mets**. Measurement and Data Communication Technology for the Implementation in Estonian Transmission Network. 2013.

65. **Julija Šommet**. Analysis of Sustainability Assessment in Carbonate Rock Quarries. 2014.

66. **Tanel Kivipõld**. Real-Time Electricity Tariff System for Retail Market. 2014.

67. Priit Uuemaa. Industrial CHP Optimal Management Model in the Energy Market under Incomplete Information. 2014.

68. **Anton Rassõlkin**. Research and Development of Trial Instrumentation for Electric Propulsion Motor Drives. 2014.

69. **Toomas Vaimann**. Diagnostics of Induction Machine Rotor Faults Using Analysis of Stator Signals. 2014.

70. **Aivar Auväärt**. Development of Energy Reserve Optimization Methodology for Households with Renewable Power Systems. 2014.

71. **Raivo Attikas**. Modelling of Control Systems and Optimal Operation of Power Units in Thermal Power Plants. 2014.

72. **Liisa Liivik**. Semiconductor Power Loss Reduction and Efficiency Improvement Techniques for the Galvanically Isolated Quasi-Z-Source DC-DC Converters. 2015.

73. **Victor Astapov**. Technical-Economic Analysis of Distributed Generation Units in Power Systems. 2015.

74. **Tiit Hõbejõgi**. Possibilities to Optimize Low Voltage Network Investments in Rural Areas. 2016.