THESIS ON CIVIL ENGINEERING F54

Water Distribution System Modelling and Pumping Optimization Based on Real Network of Tallinn

MARGUS KOOR

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TALLINN UNIVERSITY OF TECHNOLOGY
Faculty of Civil Engineering
Department of Mechanics

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Supervisors: Professor Tiit Koppel,
PhD Anatoli Vassiljev,
PhD Raido Puust,
Department of Mechanics,
Tallinn University of Technology

Opponents: PhD, P.E., D.WRE, F. ASCE Thomas M. Walski,
Senior Product Manager/Bentley Fellow
Bentley Systems, Inc.

PhD Peeter Ennet, Estonian Environmental Agency,
Chief Specialist

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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.

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EHITUS F54

Tallinna linna ühisveevärgi modelleerimine ja pumpamise optimeerimine

MARGUS KOOR

TTÜ KIRJASTUS

In memoriam

Professor Tiit Koppel (23.08.1945-30.07.2014)

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NOMENCLATURE

This chapter provides the description of all main symbols and units used in the thesis.

E – Energy consumed by the station
P – Pressure at the chosen point (N/m^2)
V – Flow speed (m/s)
z – Elevation head (m)
ρ – Density of the fluid (kg/m^3)
g – Acceleration due to gravity (m/s^2)
P_{wp} – Water power or useful power transferred to the fluid by the impeller (kW)
P_{shaft} – Power required at the drive shaft of the pump (kW)
$P_{shaft,\ nominal}$ – Pump shaft power at pump nominal rotational speed (kW)
P_{mip} – Power to the electric motor (kW)
P_{in} – Input power requirement for the whole system (kW)
$P_{ef,i}$ – Effective power of i^{th} pump
$P_{ef,i0}$ – Effective power of i^{th} pump at rated rotational speed N_i
η_p – Pump efficiency (%)
η_{VFD} – Variable frequency drive (VFD) efficiency (%)
η_m – Electric motor efficiency (%)
η_1 – Pump efficiency at nominal rotational speed N_1 (%)
η_2 – Pump efficiency at reduced rotational speed N_2 (%)
n – Ratio between nominal (rated) pump speed and actual pump speed
Q – Volumetric flow rate
H – Head developed in meters of fluid (m)
N – Rotational speed (rpm)
N_i – Relative rotational speed of the i^{th} pump (rpm)
$N_{nominal}$ – Pump nominal rotational speed (rpm)
S_a – Asynchronous rotational speed (rpm)
S_S –Synchronous rotational speed (rpm)
D – Impeller diameter (mm)
η_{BEP} – Pump efficiency at best efficiency point (%)
Q_{BEP} – Pump flow at best efficiency point (l/s)
m – Number of pumps
f – Electrical frequency changed by VFD (Hz)
p – Number of motor poles
s – Slip (%)
I_L – Current measured (A)
$I_{nameplate}$ – Current at full load of the motor (A)
V_L – Voltage measured (V)
$V_{nameplate}$ – Voltage at full load of the motor (A)

INTRODUCTION

Motivation

Optimization of pumping cost and reduction of energy consumption in water business have been the aim of many different experimental and theoretical studies during the last couple of decades. Optimization methods constantly evolve together with the development of faster, more accurate and more complex WDS (water distribution system) modelling methods and software. The benefits gained from that field of research are often not implemented in practice as fast as possible by smaller or medium size water companies, who do not have the resources to participate in research or do not fully realize the benefits. Fundamental questions, such as how to optimize and control the work of centrifugal pumps in WDS to reduce electricity usage and how to solve issues related to oversized pumps, have been the theme of discussion for some time. As energy becomes more and more expensive, the optimization of energy usage is inevitable for all Water Companies.

Research related to the current thesis gives a good opportunity to develop practical tools for optimization, concentrate knowledge and provide an idea what can be done and what the expected effect may be. Motivation of this thesis is to continue practical research and development cooperation between AS Tallinna Vesi and Tallinn University of Technology who have had a good productive partnership history in water distribution modelling and optimization research field already for many decades.

AS Tallinna Vesi is the only privately owned Water Company in Estonia. Therefore, the water utility business in Tallinn is driven by clear financial objectives as in any other business today - to achieve the compliance with required service levels using minimal resources. That means all operational costs have to be clearly justified and optimized. AS Tallinna Vesi is also the first listed company in the Baltics nominated by Nasdaq OMX as the company with the Best Investor Relations in 2013 and 2014. European Commission also presented AS Tallinna Vesi as a nominee of EMAS 2012 awards for recognising the Water Company's outstanding environmental performance. Thus, the optimization of energy usage is a very important task today and will be such in the future.

In order to carry out an optimization of pumping in WDS, the working near real-life mathematical hydraulic model of WDS is an essential tool for investigating further the hydraulic processes in the network and the efficiencies of pumps. For example, correction of pumps pressure control settings cannot be done without the hydraulic model of WDS. For the current research, AS Tallinna Vesi kindly provided permission to use WDS of Tallinn and all available monitoring data. The previous EPANET based version of the hydraulic model was more than 10 years old and therefore did not reflect the present day situation and had to be renewed or remade. To maximize the usage of the existing software the Water Company decided to move on with the creation of new hydraulic model that is based on *Bentley's WaterGEMS v8i* hydraulic modelling software (Bentley Systems, 2011). The idea was also to develop, simplify and automate the model renewal process for the future. That includes automated topology and consumption related data generation scripts in SQL database level. The usage of online Supervisory Control and Data Acquisition (*SCADA*) data for model calibration purposes and possible automation possibilities have been considered.

The first chapter of the thesis briefly outlines the studies and literature on how the pumping related efficiencies and energy usage can be calculated and assessed.

The second chapter describes how the hydraulic model of Tallinn WDS was created and calibrated to model different scenarios. The hydraulic model was calibrated based on real network data. Calibration was carried out with Darwin Calibrator using the genetic algorithm. Detailed instructions have been worked out how to create and renew the hydraulic model in the future as needed.

The third chapter focuses on detailed efficiency analysis of the booster pumping station “Punane” in Tallinn as an example of the efficiency evaluation process. The analysis is carried out based on the existing *SCADA* data. That enables to carry out similar efficiency analysis also for other pumping stations with minimal resources. To get a full understanding of how the existing pumping station performs, the pumps working conditions, energy usage, efficiencies and operational cost is analysed.

The fourth chapter introduces possibilities how to make previously analysed pumping station “Punane” more efficient using good engineering practice. The goal is to reduce energy usage and electricity cost. It includes replacement of oversized pumps with clearly achievable benefits and expected payback time calculation. The pumps pressure control settings are also reviewed using previously created *WaterGEMS* based hydraulic model.

The fifth chapter focuses on pump work related optimization. A new mathematical algorithm is proposed to optimize the work of identical centrifugal pumps working in parallel. Algorithm enables to calculate out optimal border conditions when to start or stop pumps to achieve maximal efficiency and to identify optimal pump switching points for different number of working pumps. To illustrate the usage of the proposed algorithm, two numerical case studies are carried out using the existing and newly proposed pumps.

The sixth chapter investigates theoretical possibilities how to optimize the work of centrifugal pumps working in parallel that have slightly or largely different performance characteristics. Alternative optimization method has been proposed, as polynomial approximation of efficiency surface based optimization implemented with identical pumps cannot be used when pumps performance parameters differ. Three theoretical scenarios are carried out to illustrate the method.

The final chapter summarizes the findings and recommends themes for possible research in the future.

Aim of investigation

The thesis will focus on WDS hydraulic modelling and pumping optimization based on real network of Tallinn city. Special focus is on the analysis of energy usage in booster pumping station "Punane" as an example and on the practical need to minimize energy usage and operational costs. The specific objectives of this thesis are defined as follows:

- To develop the creation process, create and calibrate close-to-real-life mathematical hydraulic model, which represents the WDS of Tallinn City. The goal is to maximize the practical usage of the Water Company's existing databases and introduce methodology how the renewal of the hydraulic model can be done with minimal effort and cost in the future. The accurate hydraulic model is a necessary precondition for further research.

- To carry out efficiency analysis of booster pumping station "Punane", based on the created hydraulic model and constantly collected *SCADA* data. To introduce ways how to save energy used for pumping in small or medium size WDS-s similar to Tallinn. Tallinn has no water towers, whereby water could not be pumped and stored at low energy cost period and used at peak hours. Therefore, the focus is on improving the efficiency of booster pumping stations with similar type of centrifugal pumps working in parallel controlled by variable frequency drives.

- To improve the efficiency of booster pumping station "Punane" as an example using good engineering practice. To propose more efficient pumps for replacement, define achievable benefits and to describe a way of calculating payback. To review and correct the pumps pressure control settings according to required service level using previously created hydraulic model of Tallinn WDS.

- To create a mathematical algorithm for calculation of optimal pumps switching points based on currently needed flow and pressure head. To carry out a numerical case study for the existing pumps and for the new proposed pumps.

- To investigate theoretical possibilities how to optimize work of pump system with centrifugal pumps working in parallel that have slightly or largely different performance characteristics.

Acknowledgements

I would like to commemorate and thank my supervisor Professor $\boxed{\text{Tiit Koppel}}$ for his great inspiration, guidance and advice throughout the PhD studies. His positive enthusiasm and ability to approach practical water distribution network related issues methodically in a scientific way provided valuable input and constant support.

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1. LITERATURE REVIEW

The history of pumping reaches back to the time of ancient Egyptians more than 2000 years BC, when the need to lift water from rivers to various irrigation channels was crucial to sustain agriculture. World Pumps (2009) has published an overall review article about the history of pumps and historical events related to pumping. Current thesis focuses only on centrifugal pumps.

The earliest form of centrifugal pump was invented at the University of Marburg by French mathematician Denis Papin (1647-1712) in 1687. Development and production started more than 130 years later in America in 1818. The pump was also known as Archimedean Rotor Blades. Pump had low head and was mainly used as a water positive displacement pump. As high-speed electric motors, steam turbines and internal combustion engines evolved, the centrifugal pumps became more and more popular. Since then, centrifugal pump has been an important component of most WDS-s at least for 100 years.

1.1. Principle of centrifugal pumps

The centrifugal pump is the most often used pump type in the world. Working principle of centrifugal pump is simple and therefore well described in many handbooks and articles. Example Centrifugal Pump Handbook, Third edition (2010) by Sulzer Pumps Ltd., The Centrifugal Pump by Grundfos and Hüdraulika ja Pumbad (1995).

Based on their working principle, all pumps are divided into hydrostatic and hydrodynamic pumps. A centrifugal pump, that has hydrodynamic working principle, consists of two main parts – rotating element called impeller and a stationary element called casing, housing or volute. Figure 1-1 gives a good principle overview of how a centrifugal pump works.

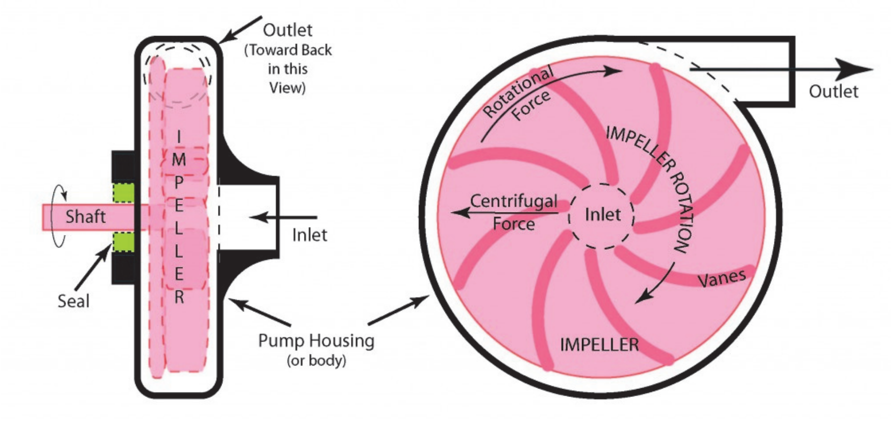

Figure 1-1 Working principle of centrifugal pump [Adapted from Fuchs, 2012)]

Rotation of impeller makes the water spin in the casing. Centrifugal force affects the water and pushes it away from central axis towards the walls of the casing. As fluid accelerates, a zone of low pressure is created near the centre of impeller. Due to the water not being able to continue to move away from central axis, it begins to climb upwards eventually overflowing the casing. By that process, kinetic energy is transferred from the impeller to water. Energy can be maintained if the overflowing water is caught and more water with sufficient energy is supplied to the pump suction nozzle. The principle of conservation of energy is first described by Daniel Bernoulli in 1738 as the Bernoulli Principle. For the incompressible, stationary flow it can be expressed as the Engineering Bernoulli equation given in Eq. (1), which describes mechanical energy balance in terms of energy per unit mass of pumped fluid

$$\frac{P_1}{\rho}+\frac{V_1^2}{2}+g\cdot z_1=\frac{P_2}{\rho}+\frac{V_2^2}{2}+g\cdot z_2 \qquad (1)$$

where

$\frac{P}{\rho}$ – Flow energy, $\frac{V^2}{2}$ – Kinetic energy and $g\cdot z$ – Potential energy.

Mechanical energy balance described by Daniel Bernoulli is often graphically expressed using different kind of heights of flowing fluid as

$$\frac{P_1}{\rho\cdot g}+\frac{V_1^2}{2g}+z_1=\frac{P_2}{\rho\cdot g}+\frac{V_2^2}{2g}+z_2 \qquad (2)$$

where

$\frac{P}{\rho\cdot g}$ – pressure head, $\frac{V^2}{2g}$ – velocity head and z – elevation head.

The Bernoulli equation given as Eq. (1) and Eq. (2) are the most fundamental relations in fluid mechanics.

Centrifugal pumps cannot suck or draw the liquid into pump housing very well, because pump cannot add energy to fluid that is not present. The pressure at suction side can still be slightly negative if it not exceeds vapor pressure of the pumped fluid and positive suction head (NPSH) requirement is met. To pump properly, the available NPSH (A) has to be higher than NPSH (R) required by the pump. NPSH (R) is a pump property and it is given by manufacturer. Available NPSH (A) is usually expressed as Eq. (3):

$$NPSH\,(A)=H_a+H_s-H_{vpa}-H_f \qquad (3)$$

where

NPSH(A) – Net positive suction head available
H_a – Absolute pressure head on the fluid

H_s – Static suction pressure head
H_{vpa} – Vapor pressure of the fluid
H_f – Friction losses in the suction pipe

NPSH deficiency can cause major problems – the pump will experience loss in efficiency, vibration and noise. Pump starts to cavitate, formation and collapse of bubbles can damage the impeller. That can be avoided example by relocating the pump, increasing the suction pipe size or cool the water. Best way is to use pumps that have lower NPSH (R).

1.2. What is pump efficiency?

The law of conservation of energy states that the energy cannot be created or destroyed, but it can be transferred or transformed from one form to another. Using centrifugal pumps or any other machines, efficiency depends on how well it can convert one form of energy into another. Due to the various energy losses, power in always exceeds power out and thereby efficiency is always less than 100%. Many medium and larger centrifugal pumps can achieve efficiencies of 75-90% and smaller ones in the range of 50-70%.

It exist different levels of efficiencies in pumping. As described in Walski (1993), the efficiency of the centrifugal pump h_p can be expressed as Eq. (4) – the ratio between produced water power P_{wp} (also known as useful hydraulic power) and power provided to pump P_{shaft} (also known as pump shaft power).

$$\eta_p = \frac{P_{wp}}{P_{shaft}} \tag{4}$$

Eq. (4) illustrates how effectively energy is transformed from pump to fluid. As power provided from motor to the pump P_{shaft} is very difficult to measure in the field, the electrical power provided to the motor P_{mip} can be measured instead. Based on that, motor and pump overall efficiency can be calculated based on following Eq. (5).

$$\eta_m \cdot \eta_p = \frac{P_{wp}}{P_{mip}} \tag{5}$$

Eq. (5) illustrates how effectively energy is transformed from motor and pump to fluid. If the motor is controlled by variable frequency drive (VFD), the efficiency of the VFD has to be taken into account also. When comparing the electrical power requirement provided to VFD P_{in}, the pump system overall efficiency (also called as wire-to-water efficiency) can be calculated based on following Eq. (8).

$$\eta_{VFD} \cdot \eta_m \cdot \eta_p = \frac{P_{wp}}{P_{in}} \tag{6}$$

Eq. (6) illustrates how effective is the pump system transforming electrical energy to fluid.

To improve pump system overall efficiency, the efficiencies of every component have to be investigated. The efficiency of the pump alone does not reflect how well separate components of the pump system work together. Evans (2012) defines the pump system overall efficiency as a combination of the pump efficiency; motor efficiency and variable frequency drive (VFD) efficiency. As an example, when pump runs at his best efficiency point with 85% efficiency, motor at 95% and VFD at 95% efficiency, the overall efficiency would then only be 77%. That shows how important it is to understand and take into account the effect of different efficiencies.

1.3. Why and when is efficiency important?

The process of treating and supplying the water to customers consumes a massive amount of electrical energy. As Coelho and Andrade-Campos (2014) pointed out in review paper, the energy consumption for water distribution is estimated as 7% of global energy usage. Feldman (2009) estimates that the global water loss in water distribution systems (WDS) is ca 30%. Based on that, proportionally ca 2% of global energy is wasted. Thus, the reduction of water losses always plays a very important role in providing a saving in overall energy consumption.

One of the main inefficiencies to lose energy in WDS-s is caused by an inefficient control of pumps or by historically oversized pumps because of intentionally added safety margin or drop in demand. As pointed out in Variable Speed Pumping — A Guide to Successful Applications, Executive Summary (2004) ca 75% of pumping systems today are oversized, many by more than 20% of pump best efficiency point (BEP) that provides substantial possibilities for the optimization of energy usage. Current research focuses on generally oversized centrifugal pumps. Ramos et al. (2012) describe that in most WDS-s all over the world the energy consumption can be reduced at least 25% through various performance improvements.

1.4. How is pump efficiency attained?

Numerous articles and pump books have been written about the ways of estimating or calculating efficiency of centrifugal pumps. In example, Evans (2012) describes, that the overall efficiency of the centrifugal pump consists of three individual efficiencies – mechanical (losses in bearings and seals), volumetric (losses due to leakages) and hydraulic (losses caused by liquid friction). Hydraulic efficiency is the largest factor affecting overall pump efficiency. Viholainen et al. (2012) proposed and described control strategy for carrying out a pump operation analysis for parallel pumps controlled by variable frequency drives with limited data and without additional measurements. Strategy bases on pump real time operation point estimation by Affinity Laws Eq. (7) and the selection of suitable operating area of parallel pumps.

1.5. Affinity laws

The Affinity Laws for centrifugal pumps Eq. (7) describe approximated relationships between rotational speed, impeller diameter, pump flow rate, pump head and power. Changes in pump geometry and speed can be approximated with high precision when geometrical, kinematical and dynamical similarities of comparable pumps are met

$$\begin{gathered} \frac{Q_1}{Q_2}=\frac{N_1}{N_2} \text{ or } \frac{Q_1}{Q_2}=\frac{D_1}{D_2} \\ \left(\frac{N_1}{N_2}\right)^2=\frac{H_1}{H_2} \text{ or } \left(\frac{D_1}{D_2}\right)^2=\frac{H_1}{H_2} \\ \left(\frac{N_1}{N_2}\right)^3=\frac{P_1}{P_2} \text{ or } \left(\frac{D_1}{D_2}\right)^3=\frac{P_1}{P_2} \end{gathered} \tag{7}$$

where

Q – Flow rate
D – Impeller diameter
N – Rotational speed
H – Head
P – Power absorbed

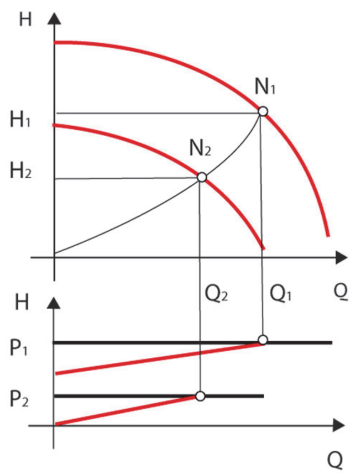

Figure 1-2 Graphical representation of Affinity Laws [Adapted from grundfos.com]

Previous Figure 1-2 is adapted from Grundfos White Paper, Pump Affinity Laws for centrifugal pumps. It is known, that Affinity Laws work well on pump impellers with low nominal speeds and give poor results on impellers with high nominal speeds because the accuracy of approximation decreases when rotational speed increases. It is also realized that Affinity Laws, described as Eq. (7), should not be used when rotational speed change forms more than 40-50% of the pump's nominal speed. As Walski et al. (2003) describes, the energy loss in VFD can cause significantly lower pump efficiency curves than predicted using the Affinity Laws. Manufactures of VFD-s often state, that the loss is VFD is very small comparing energy losses in pump or motor. Walski et al. (2003) shows clearly based on experiment, that losses in VFD need to be determined.

1.6. Estimation of variable speed pumps system efficiency

VFD-s to control pumps input power frequency have been widely used already for decades after the VFD cost became reasonable compared to pump purchase cost. Today most of the pumps used in water distribution system (WDS) are controlled by VFD-s. Lingireddy et al. (1998) pointed out and illustrated in addition to the obvious financial savings the main hydraulic benefits and disadvantages of using variable speed pump (VSP) system. Benefits and disadvantages are summarized and compared in following Table 1-1. It shows that variable speed pumps are not the magical cure for every problem in WDS. There are good benefits but also significant disadvantages that engineers have to think of.

Table 1-1 Benefits and disadvantages of VSP systems

Benefits	• Precise pressure management near to minimum required levels • Due to lower minimum pressures needed water losses and leakages will reduce • Control of pump operations is more flexible, more use and energy savings from off-peak pumping • Better control of flow into tanks for maintaining water quality • Better response to abnormal peak demands (fire) or pipe bursts • Elimination of transients caused by starting or stopping the pumps (except for sudden power loss) • More simple control of system flow rate
Disadvantages	• VSP-s are less energy efficient at low speed • Additional maintenance and equipment may be needed • Keeping of minimum pressure levels reduces overhead storage and thereby safety factor of WDS. • VFD's are expensive. Sometimes they cost almost as much as the pump itself.

	• Motors are designed to run at a given speed and modifying the speed puts and unexpected load on the bearings • VFD’s can push harmonics back into the electrical grid

As Bernier et al. (1999) described, there are four different power levels based on which VSP system efficiency can be evaluated (Figure 1-3). The most important indicator from the perspective of energy usage assessment point is the P_{shaft} that is often given by pump manufacturer together with pump characteristics curve. VFD, electric motor and pump all have their own efficiencies to be considered.

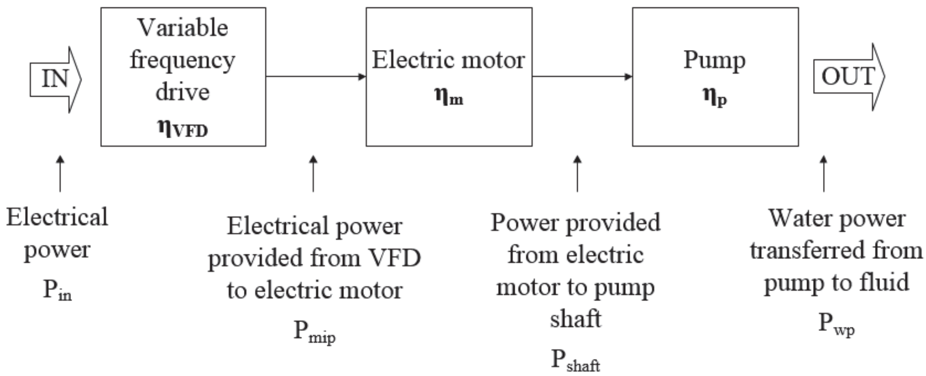

Figure 1-3 Typical pump-motor-VFD arrangement showing four power levels

As Centrifugal Pump Handbook, Third edition (2010) describes, the water power P_{wp} (useful hydraulic power transferred to the fluid by the pump) in kW can be defined as Eq. (8).

$$P_{wp} = \frac{\rho g Q H}{1000} \tag{8}$$

Where Q is the volumetric flow rate of the pump (m^3/s) that includes the useful flow rate at the pump discharge nozzle, possible leakage flow rate and flow rate for balancing axial thrust. Based on Ferman (2012), the axial thrust on a single stage pump is basically zero. H is pump head developed in meters of fluid and ρ is fluid density (kg/m^3). The power input required at the pump drive shaft P_{shaft} must always be higher than P_{wp} because it takes into account mechanical friction and shock between impeller and the fluid. P_{shaft} can be expressed as Eq. (9) – the ratio between water power P_{wp} (also often called as useful hydraulic power) and pump efficiency h_p.

$$P_{shaft} = \frac{P_{wp}}{\eta_p} \tag{9}$$

Electrical power required to run the electric motor P_{mip} is given as Eq. (10).

$$P_{mip} = \frac{P_{shaft}}{\eta_m} \tag{10}$$

where the η_m is the electric motor efficiency. As Bernier et al. (1999) describes, that usually motor efficiency is almost constant until motor is loaded over 50% and decreases rapidly when load drops below 25%. The electrical power requirement P_{in} at the inlet can be expressed as Eq. (11) using wire-to-water efficiency and water power.

$$P_{in} = \frac{P_{wp}}{\eta_{VFD} \cdot \eta_m \cdot \eta_p} \tag{11}$$

where the η_{VFD} is the variable frequency drive efficiency. Bernier et al. (1999) points out that if the pump motor is oversized, the actual power required at the inlet P_{in} can be significantly higher than Affinity Laws predict, especially at low flow rates.

There are many different best practice guides and methodologies for variable speed driven water-pumping systems to estimate efficiency of pumping and to save energy. The simplest way to analyse continuously-varying pump efficiency is to take pump flow versus time and compare it with the flow rate at the pump's best efficiency point as described by Walski et al. (2003). Results may be visualized by developing an inverted parabola equation based curve Eq. (12) that represents relationship.

$$\eta_p = \alpha_0 + \alpha_1 Q + \alpha_2 Q^2 \tag{12}$$

The α_0, α_1 and α_2 are coefficients and can be calculated as given in Eq. (13).

$$\alpha_0 = 0;\ \alpha_1 = \frac{2 \cdot \eta_{BEP}}{Q_{BEP}};\ \alpha_2 = -\frac{\eta_{BEP}}{Q_{BEP}^2} \tag{13}$$

where

η_{BEP} = pump efficiency at best efficiency point (%)

Q_{BEP} = pump flow at best efficiency point (l/s)

As described in The Centrifugal Pump by Grundfos, for variable speed pump system the principal is similar, but the function of total dynamic head has to be taken into account as well. The Eq. (12) takes the following Eq. (14) shape:

$$\eta_p = \alpha_0 + \alpha_1 \frac{Q}{n} + \alpha_2 \left(\frac{Q}{n}\right)^2 \tag{14}$$

where

n = ratio between nominal (rated) pump speed and actual pump speed

To describe pump efficiency, the Eq. (14) can also be defined in a more general form for higher degree of polynomials as Eq. (15).

$$\eta_p = \sum_{i=0}^{N} \alpha_i \left(\frac{Q}{n}\right)^i \tag{15}$$

Sárbu (1998) proposed the Eq. (16) how to scale pump efficiency η_p based on a change in pump rotational speed.

$$\eta_2 = 1 - (1 - \eta_1) \cdot \left(\frac{N_1}{N_2}\right)^{0,1} \tag{16}$$

where

η_1 – pump efficiency at nominal rotational speed N_1

η_2 – pump efficiency at reduced rotational speed N_2

To reduce error in pump efficiency estimation and improve efficiency assessment of variable speed pumps (VSP) at reduced rotational speeds, Simpson et al. (2013) proposed to use instead of Affinity Laws the relationship described by Sárbu (1998). The best result in reducing approximation error in the pump efficiency estimation is achieved especially for smaller pumps. As Simpson et al. (2013) demonstrate, the Eq. (16) gives reasonably good results if pump speed is reduced less than 70% of the nominal speed and point out that Eq. (16) can be easily implemented in hydraulic solver programs, because it uses the same input data as Affinity Laws.

For identical pump group Ulanicki et al. (2008) carried out interesting case study and considered two approaches on evaluating mechanical power (also known as shaft power) characteristics. In the first approach the power characteristic for a pump group of n identical pumps that all run at speed s is indirectly derived from hydraulic and efficiency curves provided by pump manufacturer. In the second approach, mechanical power characteristic is directly approximated based on mechanical power data points (zero flow, the peak efficiency flow and the cut-off flow) by using a cubic polynomial and then scaled by the speed of the pump and number of pumps. It was concluded that mechanical power derived from hydraulic and efficiency curves give a poor approximation by low flow, especially if the hydraulic curve is constructed by Affinity Laws but good results near peak efficiency flow.

1.7. Optimization of pump schedules and control settings

The working conditions of an ideal pump are achieved when the needed demands are met; pressure conditions satisfied and total pumping cost is minimum. Pump (Q, H) performance and efficiency characteristics are usually provided by pump manufacturers but better optimization results can be achieved using actually measured curves for particular pumps. In example, the curves of new and used pump can shift because of pump wear. Pump actual curves investigation requires dismantling of the pump and analysis in laboratory conditions. Because the process is expensive and time consuming, it is often avoided by operators who don't fully understand the benefits of knowing actual pump curves.

Many different algorithms and techniques are developed for simulation and solving of optimization and optimal pump scheduling tasks to minimize energy usage in different WDN-s with pumping stations. This chapter tries to give a very brief overview of research carried out in this field. The list is by far not complete as today there is considerable amount of methods.

First methods to solve complex water supply system optimization tasks came up with the appearance of the first digital computers already in 60`s. Since then the development of different optimization algorithms has rapidly increased. Example Coulbeck and Orr (1984) used dynamic programming (DP) technique to optimize the basic pump scheduling procedure and pump combination selection. Zessler and Shamir (1989) published an article about progressive optimality (PO) based on dynamic programming (DP) to find the decisions for pumps and valves for 24h time period. Ormsbee et al. (1989) focused on finding the optimal pump usage policy to minimize the pump operational cost in water distribution system according to variable electricity rate schedules and system demand schedules. This resulted in the development of an optimal water level trajectory in tank and pump operating policy to achieve it. Yu et al. (1994) used an algorithm based on nonlinear programming to determine optimal pump schedules. However, the complexity of water distribution systems limits the use of conventional optimization methods such as linear and nonlinear programming. The need to optimize more than one objective or goal led to the development of more complex multi-objective stochastic optimization methods such as simulated annealing, evolutionary algorithms and neural network. Simulated annealing method is based on a process similar to annealing of metals and glass, generic algorithms are search algorithms that are based on the mechanics of natural selection and genetics related with heredity. Neural network methods are based on a network of interconnected neurons. All previously mentioned sophisticated methods of mathematical optimization require significant scientific skills that most of smaller or medium size Water Companies usually do not have.

Therefore, practical and simpler instructions have been worked out for the optimization on a more general level. For example in terms of the overall practical estimation of the energy efficiency of a working pump, the best practice guide was prepared (Sustainability Victoria 2009) to help industries to improve pumping systems step by step. For pumps working in the oil industry, Crease (1996) showed that identical pumps work with the highest efficiency (running in parallel) if they operate in the same conditions and at the same rotational speed. Tianyi et al. (2012) also showed that identical pumps with variable frequency drives running in parallel are the most efficient when they run at the same speed. Both of these investigations were intended for pumps working outside water distribution systems (in the oil industry and in air-conditioning systems). The analysis of pumps in the water distribution systems (WDS) is more complex, as usually a WDS contains water towers and analysis must be made on the joint action of pumps and water towers. In this regard, the Tallinn WDS differs from other systems because it contains no water towers and the analyses of pumps covered by Crease (1996) and Tianyi et al. (2012) may be useful in this case.

There are many commercially available programs with integrated production tools (example *WaterGEMS* Darwin Sheduler, the Derceto Adapt program and Optimatics) to carry out pump scheduling optimization. Walski et al. (2010) analyses strengths and weaknesses that came up using Darwin Scheduler and GA for determining the optimal pump schedule for a realistic system. As Water Company owns *WaterGEMS* license that includes Darwin Scheduler, special attention is given to that paper.

Typically, consumed energy and maintenance cost of a medium-sized industrial pump will account for over 50-95% of a pump's life cycle cost (LCC). Efficiency can be best improved here through the reduction of energy consumed, maintenance recruitments and matching pumping system capacity more closely to actual production requirements. Improved efficiency of pumping systems also contributes to the reduction of greenhouse gas emissions and reduction of environmental impact of operations. In many cases, this process includes only limited financial resources whilst providing significant savings.

Two common issues to overcome in operation are unnecessary demand on the pumping system and oversized pumps. A pump is considered as generally oversized when it is not operated at or with 20% of its BEP as described in Sustainability Victoria (2009) best practice guide. Focus in this thesis is on the technique to keep identical centrifugal pump group running near the best efficiency point (BEP) and to identify when to start/stop lag pumps based on the flow and required pump head.

Next chapter focuses on the creation of essential tool to carry out optimization tasks – mathematical model of Tallinn water distribution system that will be used later as a hydraulic solver in current thesis to model various optimization scenarios. As current thesis is based on the real life WDS of Tallinn, the customized model creation process is described in detailed level. It has been acknowledged by author and Water Company that pumps performance graphs can shift during pump operation because of various reasons (example worn bearings, damaged impellers from cavitation, corrosion, etc.). Dismantling of the operational pumps and laboratory testing was not in the scope of current research and therefore hydraulic model was created based on new pump performance graphs given by pump manufacturers.

To estimate how much pump efficiency has been decreased during 16 year of operation, some investigation is done for pumping station ”Punane” pump based on test week data, after hydraulic model of Tallinn WDS was ready. Analysis is covered in chapter 3.8.

2. NETWORK HYDRAULIC MODEL

One of the main goals of the current research was to create a tool that enables to understand various hydraulic processes in a WDN. This chapter introduces the principles and methodology how to create a customized mathematical water model for city of Tallinn based on the existing data from Water Company. Good calibrated model reflects with reasonable accuracy the hydraulics in the network and enables to simulate various scenarios of optimization to be sure that pressure and demand requirements are met in every point of the network. In addition, the effect of tuning pumps pressure regimes cannot be assessed without understanding what will happen in the network during the possible scenarios.

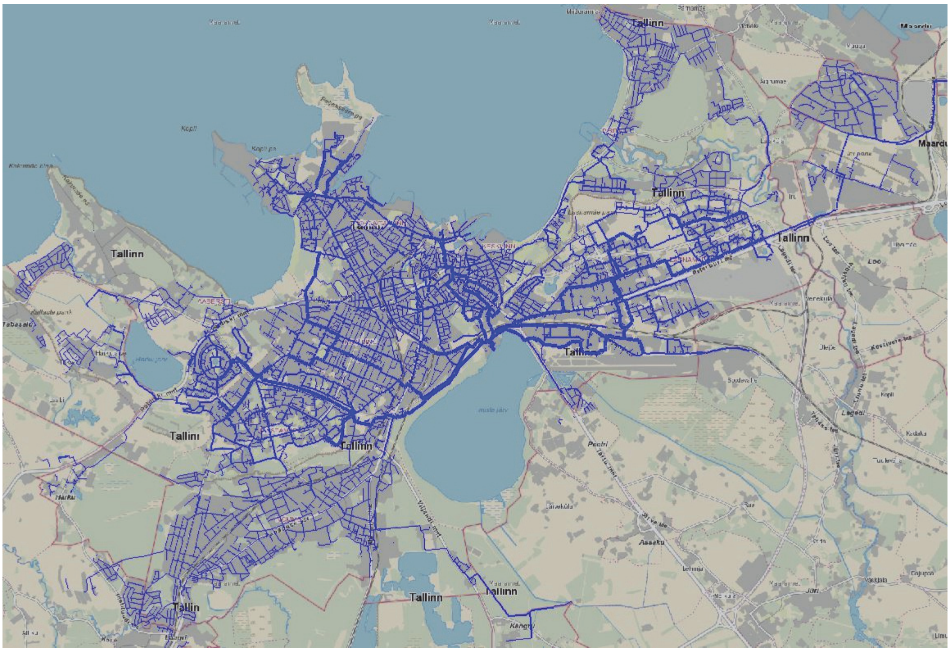

Figure 2-1 Scheme of water distribution network of Tallinn city

2.1. Review of Tallinn city water distribution system

The history of WDS of Estonia’s capital city Tallinn starts already in medieval period in 1345, when by the authority of the Danish king an open water channel was constructed from Lake Ülemiste to Tallinn bay. As the city grew, the open channels became polluted with sewer and sanitary issues caused diseases. Only after 1882, when the construction of Tõnismäe water tower was completed and wooden pipes where replaced by cast-iron pipes, the old city centre area was covered by relatively clean untreated lake water. In 1927, the modern water treatment plant of its time was built on the coast of Lake Ülemiste. The lake has been the primary water source since then. Today Tallinn WDS consists of ca 950 km of pipelines, 21 groundwater- and 11 pressure booster stations that all together provide service to ca 392 000 end consumers. The main source of water is the Lake Ülemiste that is supported by several ground water resources (Figure 2-2). Because 90-95% of consumed water comes from the lake, it has been acknowledged that the main source of water is very vulnerable against pollution and disasters. Especially because the airport is located near the lake, and therefore air traffic often goes over the lake. There have been already two accidents, where an aircraft made emergency landing on the ice. Ground water pumping stations are mainly kept in hot reserve. Ground water is supplied only to old residential district called Nõmme in south-west part of Tallinn and Merivälja in north-east.

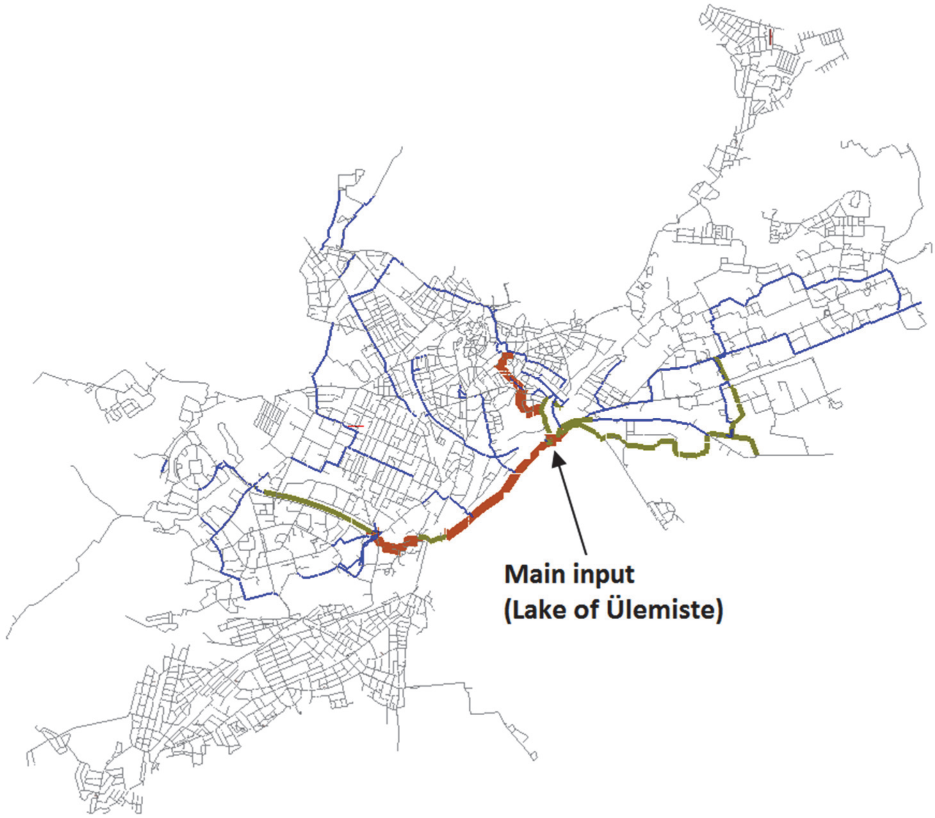

Figure 2-2 Hydraulic WDS model with transmission lines
[Adapted from Puust et al. (2014)]

The service is provided by a company named AS Tallinna Vesi (Water Company). WDS expands also beyond Tallinn municipality borders into neighbour municipalities.

2.2. Concept, methodology and process

The building of a WDS mathematical model started in 2011 when Water Company involved Tallinn University of Technology as a consultant to renew the old EPANET based water model. *WaterGEMS* v8i SELECTseries 3 was chosen as a platform because Water Company already owned the software licences and main GIS system was based on *Bentley's MicroStation* platform. Also popular freeware hydraulic solver EPANET2 was considered, but was rejected because of lack of existing functionalities. Hatchett et al. (2011) carried out an interesting research about the integration of network model with real measurement database to assess the accuracy of the model. As Marchi and Simpson (2013) also described in technical node EPANET2 was inaccurate in computing the efficiency of VSP pumps using Affinity Laws at lower pump rotational speeds.

One of the main goals in the hydraulic model creation process was to maximize the usage of existing Water Company databases and maximally automate hydraulic model preparation processes to the limit. For that, usage of *WaterGEMS* ModelBuilder tool was tested. During the process, some limitations came up. As Water Company GIS system did not supported exporting network elements as a shapefile, alternative solution was needed. Therefore, next step was to investigate possibility to use direct import from existing GIS database. It came up, that direct database import from SQL-based database allowed importing only straight pipes without vertexes (bending points). The need to have vertexes instead of dummy junctions (zero demand), enables to create simpler model with less nodes. *WaterGEMS v8i* skeletonization tool (Skelebrator) enables to simplify the model, but simplification routines (reducing the number of junctions) cannot be saved and automatically run after GIS database regular updates.

Considering the need to simplify the model with possibility to update it regularly, the simplification procedures were carried out at GIS database level and finally *EPANET *.inp* is generated using predefined database level scripts. Use of *EPANET *.inp* file, allows also to import pipes that have bending points without "braking" them and maintain elements Mslinks of pipes and junctions. *WaterGEMS* ModelBuilder tool is still used to import and update additional data (pump station sub-models, valves, hydrants, etc.) based on GIS element ID-s (Mslinks). If the pipes and nodes data had to be renewed, the new *EPANET *.inp* can be automatically generated using predefined database level scripts, thereafter opened in *WaterGEMS* and updated thru predefined ModelBuilder links.

Process of model creation mainly consisted of two phases. Phase 1 of the research lasted until May 2012 and was mostly dealing with database connections, model skeletonization and demand aggregation. Process has been fully described in Koor et al. (2012). Phase 2 started in summer 2012 and included the selection of pressure measurement points, field data gathering, data validation and model calibration that is described in Koor et al. (2014). In addition, an extended article Puust et al. (2014) about the whole modelling process and occurred challenges has been published. Since January 2013, the calibrated water model has been in everyday operational use. It has been used by Water Company to model various WDS related scenarios over 500 times.

The model creation procedure is based on a real network data that is constantly collected and stored. To update the model based on new data (topology and consumption changes etc.), it is highly advised to automate some routine steps already at database level and save time. That will also include data integrity verification and unification. Although database based information management is nothing new, the development of the procedure for creating a well-established hydraulic model is quite often the opposite. Usually at least one-time model creation stage has occurred in most cases. After that, every new update is commonly done manually. Is time-consuming and may introduce new errors that are hard to notice and solve because of the scale of the model/database. Following simplification procedure is created to automate in database level some processes related with data collection and optimization that are very time consuming and difficult.

Water Company uses in operation different Supervisory Control and Data Acquisition (*SCADA*) systems that have been implemented in different years as stand-alone solutions. Different types of raw data are kept in various databases (infrastructure, customer demand data, and 4 different *SCADA* systems). Databases have different structures and are thereby very hard to combine and use as a source for model build-up. Several approaches to combine those databases are described in Koor et al. (2012).

In the current study, geographical information system (GIS) and hydraulic network model are kept separately. Simplification exercise in 2011 was done in GIS database level as a workaround, to enable *WaterGEMS* to use Water Company GIS data, which turned out to be too big for *WaterGEMS* v8i SELECTseries 2 standard import functionality at this time.

The simplification workflow as a summary is illustrated on the following Figure 2-3. Simplification included in example pipe removal, pipe merging with demand aggregation, branch collapsing etc.

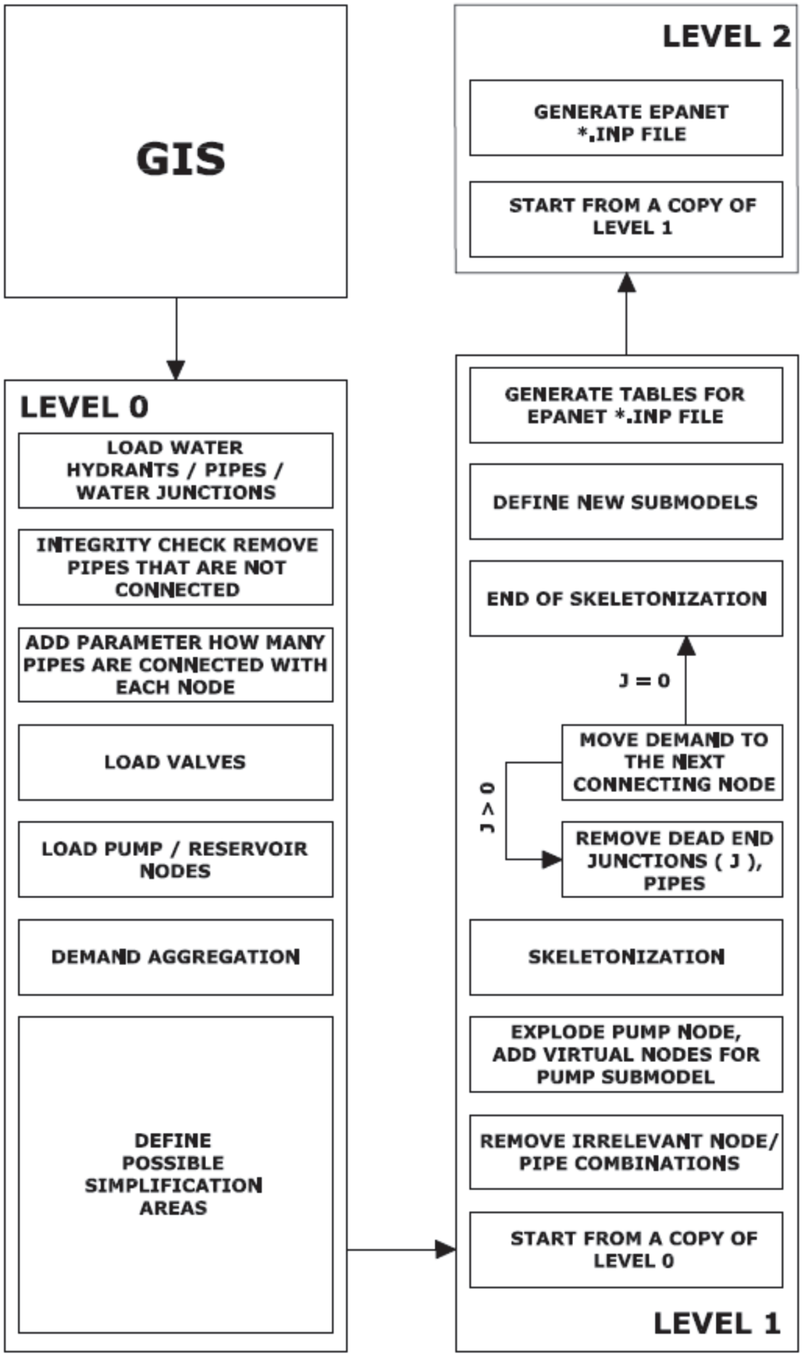

Figure 2-3 Main simplification procedures at SQL based database level [Adapted from Puust et al. (2014)]

After that, several database export views are used to generate *EPANET *.inp* file (using Microsoft SSIS package). In addition, several views are produced to prepare additional information of elements, which cannot be given *EPANET *.inp* file and therefore is *WaterGEMS* ModelBuilder (Pressure zone, material, diameter, installation year etc.). The results of current database level simplification are given in following Table 2-1.

Table 2-1 Number of main elements at various database levels [Adapted from Puust et al. (2014)]

	Level 0	**Level 1**	**Level 2**
Number of pump connections (input / output)	84	170	170
Number of hydrants	5 345	5 096	4 761
Client connections	22 239	18 566	17 576
Number of pipes	85 058	31 731	29 971
Number of isolation valves	36 837	8 265	7 740
Time to generate	~1h	~15min	~1min

Water Company uses *Bentley's WaterGEMS v8i* hydraulic modelling software (Bentley Systems 2011). Different approaches were discussed how to effectively build-up a model from GIS data. For network topology import *EPANET *.inp* file format described in Rossman (2000) was chosen and generated using *GIS SQL* database tools and custom queries. Before generated *EPANET *.inp* file was imported into *WaterGEMS* software package, all component libraries were added (11 material definitions, 6 main customer demand 24h patterns with daily and monthly coefficients).

The main reason to use in 2012 SQL database generated *EPANET *.inp* file was that Water Company's old GIS platform was not able to produce an industry standard *shapefile* format in 2012. In 2015, the GIS platform was finally updated and it includes now *shapefile* export/import functionality. That should improve the overall easiness of model updatability, but it needs further research and testing how to integrate usage of the *WaterGEMS* Skelebrator automation now, when shapefile of the network can be exported from GIS.

2.2.1. Additional data/model management

After the topology data import (pipes, nodes, hydrants, pumping stations and valves) into *WaterGEMS* software, additional pressure zones related information was added for every pipe/node element. In addition to zone information, all boundary data (flow inputs) was included into the model. All pumping stations were made as sub-models that enabled to carry out individual test-runs and to ensure the integrity of the model. During the model build-up, various pumping stations representations were considered as can be seen on following Figure 2-4, including also detailed ground water pumping stage if present at source or pressure booster stations. Real-life pump stations may consist of different pumping stages. Therefore, the model may describe the real network with various stages. Stage 1 describes pumping from the well into water tank. Stage 2 describes pumping from tank to network.

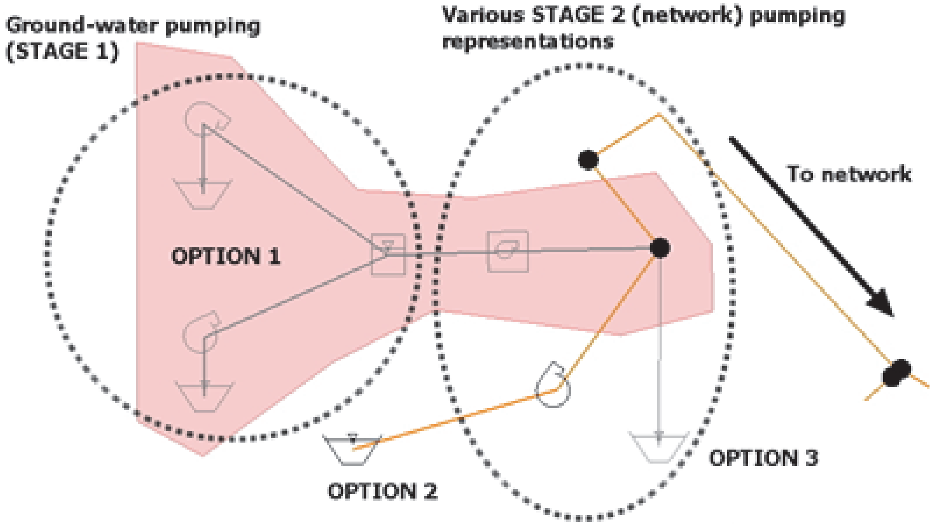

Figure 2-4 Various representations of pumping stations
[Adapted from Puust et al. (2014)]

The detail level of pumping stations models may have serious drawbacks on calculation (calibration) speeds and/or for the convergence of the simulation results. *OPTION 1* was the true representation of the pumping station, but it tremendously increases the calculation time and the *WaterGEMS* did not find the solution at some time steps at all, as described in Koor et al. (2012). Therefore, simpler pumping-station representations were used. For example, for calibration, *OPTION 3* is used where pumping station is represented by a simple reservoir element with a measured head pattern (derived from pumping station measurements). *OPTION 2* would be much more preferable for calibration but again, during the calibration, the preliminary hydraulic calculation might not converge and might cause a theoretical backflow through the pumping station. It was especially problematic in *Zone 11, Nõmme* (Figure 2-5) where a large number of pumping stations was present. Therefore, *OPTION 3* was chosen for calibration studies because of easier water balance calculations. *OPTION 2* was used with a calibrated model (including pump relative speed patterns or fixed pump head behaviour.

Several mathematical models exist to predict the initial pipe roughness values based on pipe age, material and soil type, example Duchesne et al., (2013). Because of Water Company pipes flushing strategies, it was decided that it is better to leave the estimation of pipe roughness into calibration stage, where pipe roughness and upper limits are defined according pipe material and age. Because Water Company carries out regular pipe flushing, a direct link between pipe age and roughness was not assumed before the calibration.

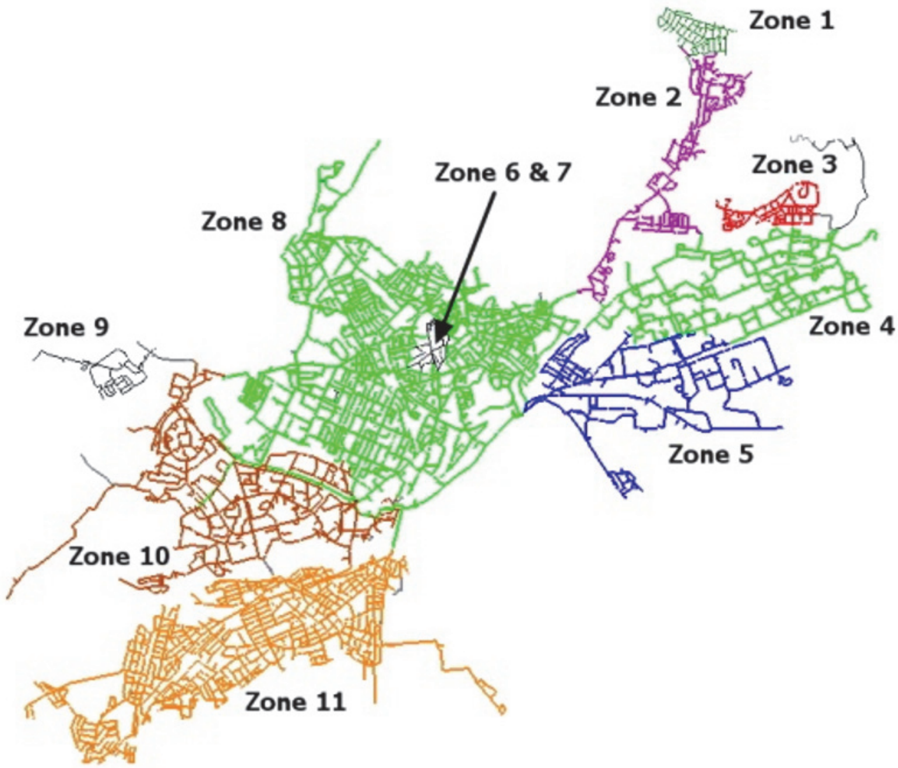

Figure 2-5 Pressure zones at Tallinn City water network
[Adapted from Puust et al. (2014)]

2.2.2. Selection of pressure measurement points

In addition to *SCADA* data gathered from pumping stations and stationary metering chambers, the extra field measurements were done for model calibration. Pressures in 10 different pressure zones were measured independently by additional set of temporally stationed pressure loggers with 1-minute time-step and zone inflows/outflows (Figure 2-5). Each pressure zone was monitored for minimum one week that included all the weekdays at least once. The size of the pressure zones varied from 110 pipes (smallest) to 11200 pipes (largest), but the number of additional pressure loggers was limited. Before the pressure measurements, sensitivity analyses were carried out at zone level to identify the best hydrant locations for pressure measurement.

The sensitivity analysis was carried out at a hydrant level and so-called *sensitivity coefficient* was calculated and added for each hydrant as a *User Defined* parameter. It shows how different pipe roughness values affect pressure head values at hydrant locations. *Sensitivity coefficient* consists from values calculated by two different approaches. Firstly, roughness sensitivity was searched at maximum demand hour at hydrants (08:30 AM). Different roughness multipliers are used with plastic/metal pipes. Thereafter the so-called leader board is created based on the difference of pressure head (at peak time) in between various pipe roughness values (initial vs multiplied). Then flow test (model based) was carried out for all hydrants in pressure zone by adding a simple fire flow (15 l/s) at hydrant locations. This is done using *WaterGEMS* fire flow analysis. Calculated values are then used to derive the pressure difference and again, a leader board is created. Those so-called sensitivity values are then used as *User Defined* parameter values for *Hydrants* inside *WaterGEMS* software. Thematic maps are created and suitable pressure measurement locations are then manually selected based on the sensitivity (larger values >> larger symbol) but also considering the fact, that all available pressure measurement sensors cover any given pressure zone. Sample results are given on following Figure 2-6 and Figure 2-7. Larger colour coding for node indicates more sensitivity.

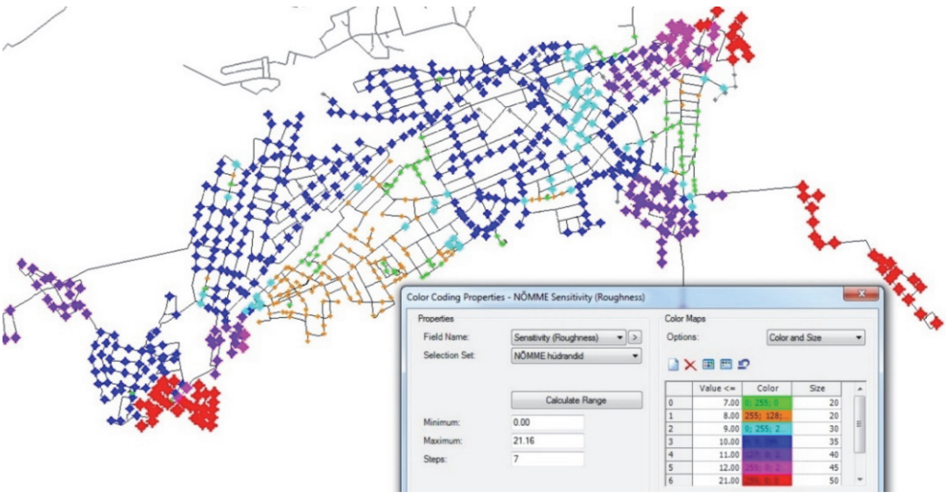

Figure 2-6 Roughness sensitivity map for Zone 11
[Adapted from Puust et al. (2014)]

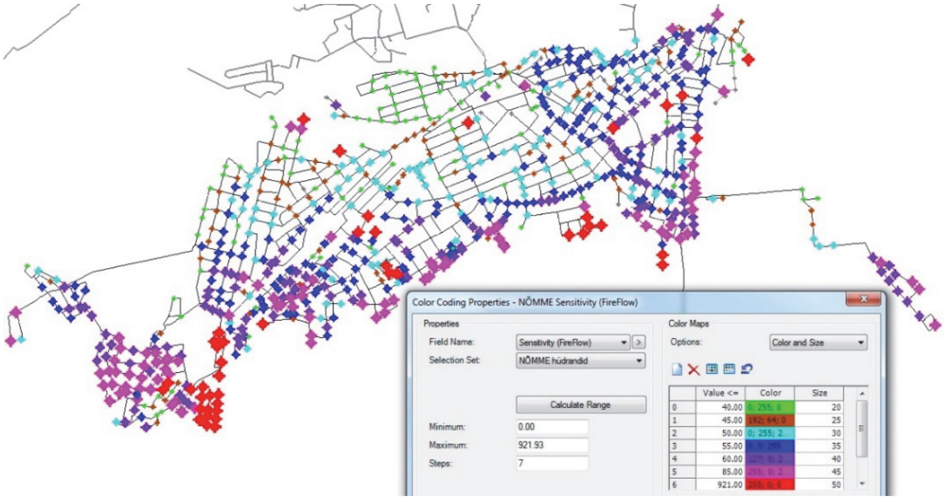

Figure 2-7 Fire flow sensitivity map for Zone 11
[Adapted from Puust et al. (2014)]

In result, from two different sensitivity analyses two different colour maps (Figure 2-6 and Figure 2-7) are derived using *WaterGEMS*. For example the most sensitive points with fire flow test are mostly located at dead-ends. Results of model calibration depend from number of pressure measurements done per pipe kilometre or per number of pipes/nodes/hydrants. Current sensitivity analysis will help to identify and concentrate attention into some particular sub areas in the network and measuring locations can be chosen so, that results will have positive effect on calibration results. The sensitivity analysis was carried out with each 11 zones. A colour maps as well as selection of exact measuring locations (example Zone 11, Figure 2-9) together with recommended measuring period was given to Water Company field measurements team as a project plan for execution. Water Company has its own experienced leakage detection department with necessary equipment (pressure sensors, flow meters and data loggers) who monitors daily the night flows and pressures. Their expertise has been used during monitoring period to get best results and to interpret the results. In addition, also the geodesic survey team is available in house for measuring the exact absolute heights of the pressure sensors. This ensured excellent data for further calibration process. Following Figure 2-8 introduces the technical solution that was used (Sensus pressure sensor, data logger, custom connection fitting) to measure pressures at underground hydrants.

Figure 2-8 Pressure measurement solution at underground hydrant

Hydrant manhole cover was closed and all the equipment was hidden and relatively protected from stealing. Figure 2-9 gives an overview of measurement points.

Figure 2-9 Pressure measurement points for Zone 11
[Adapted from Puust et al. (2014)]

Measurements were carried out in 2012 from spring to autumn. Measurement period varied from 7 – 20 days per pressure zone. As there was a risk, that previously planned period will be short to measure all zones, the smallest zone 9 Taela was chosen for last. *Zone 9* Taela was finally skipped at the end because shortage of time and because hydrants in that region were not in manholes. For security reasons, there was not possible to leave available pressure loggers on the street unguarded. That zone was also excluded from later calibration studies. Summary of measurement periods is given in the following Table 2-2.

Table 2-2 Pressure zones monitoring periods
[Adapted from Puust et al. (2014)

Zone	**Zone name**	**Measurement period**	**Full days**
1	Merivälja	19.06.2012 - 08.07.2012	20
2	Pirita	07.06.2012 - 16.06.2012	10
3	Kose	12.07.2012 - 18.07.2012	7
4	Lasnamäe-III	26.05.2012 - 03.06.2012	9
5	Lasnamäe-II	04.05.2012 - 17.05.2012	14
6	Linna-III	25.08.2012 - 02.09.2012	9
7	Toompea	06.09.2012 - 12.09.2012	7
8	Linna-II	21.04.2012 - 01.05.2012	11
9	Taela	<not measured>	n/a
10	Mustamäe-Õismäe	11.04.2012 - 18.04.2012	8
11	Nõmme	18.09.2012 - 27.09.2012	10

The duration of measurement period was highly dependent on the size of pressure zone (in terms of nodes/pipes) and the number of available pressure loggers. In addition, the installation and dismantling time of sensors and loggers took considerably longer than was expected. Although minimum of 7 days measuring was planned for each pressure zone. Longer measuring periods happened at some zones simply because of summer vacation period when there was a shortage of work force. All the measurement season, assuming average 2 days for relocation was planned for 99 days, but it took altogether 105 days. In further planning, the measurement period could be shortened with more simultaneously available loggers but then also the relocation time may go up. All measurements were recorded with 1-minute time-step. Before importing the data into calibration module, the basic filtering and averaging was carried out. Final measurement data was with 1-hour time-step, if some anomalies were discovered within measurements, those were not accounted for final data selection. For example, one pressure logger was showing higher hydraulic grade in the network than at zone inflow point (pumping station) most of the time and therefore sensor data was not used. The cause of the problem could be faulty sensor as well as wrong sensor elevation measurement. Final measurement data was directly imported into *WaterGEMS* calibration module using the *ModelBuilder*. For online loggers (pump flows/pressures, pressure measurement stations) two different import alternatives where evaluated. At first, *WaterGEMS* v8i SELECTseries 3 module

SCADAConnect was tested to import the historical data. It turned out that the capabilities of that module were not suitable with current workflow: (a) there was no possibility to save the settings of signals for future updates because when model is rebuilt from scrach by *ModelBuilder* links into new file, there is no possibility to import pre-defined signal's definitions and (b) there was no possibility to combine various timesteps into one calibration study. The latter one was the most problematic for planned workflow as it can be seen in next chapters, where the overview of calibration module is provided. As an alternative, all online *SCADA* data was imported into calibration module in the same way as with off-line measurements. For that, additional data table was prepared in Excel that included flows, pressures from the pumping stations and from the fixed measurement stations. *ModelBuilder* was used to connect with that data table and all the needed measurement data was imported into *WaterGEMS* calibration module called *Darwin Calibrator*.

2.2.3. Model calibration procedure

At the beginning of the model creation process it was decided to use the out-of-the-box tools that are readily available for Water Company because of purchased *WaterGEMS* licences. In Koppel and Vassiljev (2009), Vassiljev and Koppel (2012), Vassiljev and Koppel (2013) has been shown, that *Levenberg-Marquardt* optimization algorithm works much faster than *genetic algorithm*. As the *Darwin Calibrator* calibration module is part of the *WaterGEMS* software, it was the most obvious choice for Water Company. Calibration speed was not so critical issue as usage of already owned software.

In addition to correct measurement data, various settings had be defined before start of model calibration. Calibration was carried out for pipe roughness values (*Darcy-Weisbach*) as well as for emitter coefficient values that represent leakages. Therefore, pipe and node groupings were created. Plastic and metal pipe groupings were created separately, using only the diameter aspect with 50mm increment. It was found that smaller increment did not add a value for calibration procedure considering calibration speed. Maximum roughness value for metal pipes was taken up to 75% of its diameter with the increment of 10% of its maximum value. Plastic pipe groups were separated from the metal pipes because of much lower maximum possible roughness (up to 52 mm with increment of 12 mm) and it was assumed that the maximum value does not depend on pipe diameter. The same principles were used in all zones to create pipe groups for further calibration.

Leakage node groups were purely based on measurement locations. Because pipes are mostly oversized in the whole city area and because of very low flows, there is no point to search a leakage far away from the measurement point. Therefore, the groups were created around the measurement point. The closest node to the measured hydrant location was considered as a leakage candidate. Although each node group has one single leak candidate, the results of leakage calibration should be expanded over all other nodes that are in that region. Various methods how to calibrate leakages have been reviewed by Puust et al. (2010). Leakage calibration in the current study was defined as a search for optimal emitter coefficient for each node grouping. The ranges for emitter coefficient were assumed so that any one maximum coefficient value can cause about 10% of additional outflow from the zone.

In addition to pipe/node groupings that help to keep the calibration times in reasonable timeframe, the increment values that drive all possible roughness/emitter coefficient values play an important part. Roughness increment has been chosen so that maximum 15 different values (metal pipes) are considered in one particular pipe group. Emitter coefficient increment has been selected so that 30 different values fall into the demand group. The number of all possible groupings and possible values greatly affects the speed of calibration. In the current study pipe roughness and emitter coefficients were calibrated in a separate calculation but multiple times with different initial values.

Before calibration, it is necessary to check the parameters of calibration algorithm. Most values were kept at their default values suggested by the software and only slight changes were made like increasing the maximum trials to 5,000,000 and population size to 200. After setting up parameters, the sensitivity analysis of measurements was carried out. Bowden et al. (2002) describes how to pick the most sensitive data for final calibration. In the current research, a different approach was used. Considering the fact that no real fire hydrant tests were carried out to get additional pressure data, the whole measurement cycle was imported into the calibration module and *all-data-calibration* was carried out at zone level. For example, if a particular zone was measured for 10 days, 10 days x 24 hours = 240 time steps were fed into calibration procedure. Of course, such calibration has a dramatic effect on calculation speed. The main purpose of that was to test the robustness of the calibration procedure itself and not to optimize the calculation time. *Virtual Machine* (64-bit, 4GB memory) was used to carry out calculations. The sensitivity calculation took about one hour (small zones) to two days (large zones). After calculation, the general model agreement with the measurements was drawn by *WaterGEMS* tools.

Based on the measurement sensitivity results the error of observed and simulated HGL values are ordered and divided into smaller groups. Additional pre-calibration studies are carried out to find out how the number of good measurement snapshots affects the result (in terms of error and calculation time). It was concluded that 6 – 12 different measurement snapshots are enough to carry out the final calibration study. In addition to pipe roughness evaluation over all measurement data, the analogous analysis were carried out with leakage studies to get the overall feeling, which data is good enough to use in final calibration. The final calibration was divided into three main stages.

Table 2-3 concludes the procedure of the calibration that was applied for each separate pressure zone.

Table 2-3 Three stages of calibration procedure [Adapted from Puust et al. (2014)]

Calibration Stage	Calibration Study	Method	Note
Stage 1	Roughness calibration	Pick a value between the boundaries	Roughness of the new pipes is assumed as a starting point
Stage 2	Leakage calibration	Pick a value between the boundaries	Results from "Stage 1" were used as a starting point
Stage 3	Roughness calibration	Multiply the roughness value in between 0.1 - 2.0	Results from "Stage 2" were used as a starting point

In general, the difference between measured and simulated values was reduced at every stage. As up to 12 measurement snapshots were used in every calibration run, the calculation time was reasonable, ranging from few minutes to half an hour depending on the zone size. Table 2-4 gives an overview of calibration results illustrating maximum difference between measured and simulated pressures.

Table 2-4 Main results of the calibration based on pressure zone [Adapted from Puust et al. (2014)]

Zone	Zone name	Mean square error (m)	Maximum error (m) (absolute)	Minimum error (m) (absolute)	Number of pressure measurements	Number of flow measurements
1	Merivälja	0.1	0.5	0	11	2
2	Pirita	0.1	0.7	0	13	3
3	Kose	0.1	0.8	0	11	2
4	Lasnamäe-III	1.4	4.3	0	22	2
5	Lasnamäe-II	0.1	0.6	0	14	3
6	Linna-III	0.3	1.4	0.2	8	1
7	Toompea	0.1	0.2	0	5	1
8	Linna-II	0.6	1.7	0.1	34	3
9	Taela	n/a	n/a	n/a	n/a	n/a
10	Mustamäe-Õismäe	1	3.2	0.1	18	3
11	Nõmme	0.2	1.4	0	28	15

As described before, three-stage calibration is used with a separate portion of measurement data for each calibration stage. The sensitivities within pipe roughness grouping might different from emitter coefficients in leak node grouping. Therefore, different data portions for those sub-calibrations were used. It is clearly noticeable that calibration results are better for smaller zones where more measurements per overall unit of pipes. Maximum errors pointed out in Table 2-4 are caused by particular faulty pressure logger and sensor elevation. Vassiljev et al. (2007) describes a method how to eliminate the elevation error but in *WaterGEMS* it was impossible to apply that approach. The most questionable calibration results were in *Zone 4* and *Zone 10*. Both zones are pressure booster zones where various pressure regimes are used. Larger errors were noticed at times when there was a change in pressure regime. It may indicate a false interpretation of *SCADA* data. Therefore, additional analyses are needed to understand the problematic side of those calibration results.

In general, the calibration results were satisfied the Water Company and mathematical model represented real-life WDS with reasonable accuracy. Pipe roughness and emitter coefficient values were exported into final model to create a calibrated model with proper pump components. Final model was ready for usage. More detailed overview of model creation and calibration process is given in Puust et al. (2014).

2.2.4. Usage of calibrated model

Next stage after the model calibration was the creation of whole WDS model. Main goal was to mimic a real-life network as closely as possible. That required defining and analysing of various model components such as reservoirs with a hydraulic grade pattern, pumps working in parallel with on-off timing controls and pumps with variable speed patterns or variable speed pump batteries. Soon it turned out that as a complexity of the model and sub-models representation increase, the choice of available pumping station representations decrease.

Although the WDS of Tallinn has 11 major pressure zones, not all are independent from each other on a daily basis. Therefore, zone flow inputs/outputs that were used at calibration stage are now swapped with proper, real-life network components, like pumps and valves. Depending on the source of water (surface or groundwater) and zone's connectivity with each other, the whole network can be divided into 4 main independent areas as shown on the following Figure 2-10 that can be run separately using *WaterGEMS Scenarios Manager*. While Area 2 and Area 3 are mostly on surface water source, Area 1 and Area 4 are on groundwater source.

Figure 2-10 Four main independent WDS areas
[Adapted from Puust et al. (2014)]

The WDS in most areas can be represented using real network components such as variable speed pumps and variable speed pump batteries, fixed head pumps and pressure reducing valves. The most challenging in *WaterGEMS* was to represent a variable speed pump that has fixed head setting that change throughout the day and/or weekday. As described in Koor et al. (2012), a true model component does not exist for those situations and some alternative ways had be worked out. To solve previously described issue, the multiple pump battery elements were used in model with timely controlled settings to represent different pressure regimes. That replicates the actual situation in WDS as close as possible. Due to heavy amount of various timely controlled pump switch settings for one week, it significantly affected the WDS hydraulic model calculation speed. Therefore, such representation of pumps is also not suitable for final pump optimization studies in the future. However, the pump speed patterns based WDS modelling principle is good at model creation and calibration phase, when real measurements are carried out. Correct pump speed patterns for specific period can be derived from the measurement data covering that period. Model calculation time will then be significantly shorter. In current study, the speed coefficients were calculated based on pump head measurements recorded in *SCADA*. As demands and appearing leakages constantly change in real WDS, the pump speed patterns have to be periodically reviewed.

Most WDS areas previously visualized on Figure 2-10 can be considered as independent, treated lake water based pressure booster zones. Only *Area 4* is based only on groundwater sources having 15 pumping stations (Figure 2-11).

Figure 2-11 Pumping stations in Area 4
[Adapted from Puust et al. (2014)]

It was impossible to pick out individual pump speed coefficients directly from *SCADA* data. Therefore, those control rules were developed from the flow and pump head measurements values done at pumping stations. Due to the high number of pumps, it was the most complicated task in representing that area. Although all pumps where able to pump into the same looped network, the lower left, right and central part of the *Area 4* could be temporally isolated in main model as sub-areas. That enabled to carry out separate modelling tasks for every sub-area of *Area 4* and afterwards unite the sub-areas into full model representing *Area 4*. It helped to avoid various conflicts and anomalies in main model that are caused by high numbers of simultaneously working pumps.

At the end, all four main areas of the WDS illustrated on Figure 2-11 were successfully modelled and calibrated. The model was ready be used as a tool for next stages – for example improving pump's pressure control settings to save energy.

The final calibrated model was used for more than 500 times in decision making since 2013 (ex. fire-flow capacity analyses, pipe rehabilitation studies, pressure regimes correction, valves closing, water quality estimation).

Table 2-5 Finished WDS model usage statistics since 2013

Activities / Scenarios	**count**
(Fire)flow and pressure capability analysis	422
District based pressures modelling	35
Flows modelling	32
New pipe diameter assessment	19
Pipe closure impact analysis	15
Total:	523

Although, the research related to current thesis involved working out specific solution for Tallinn Water Company how the hydraulic model can be built up, the results and lessons learned are useful for wider range of engineers/modellers who plan creation of the new WDS model using *WaterGEMS* modelling software package.

3. ANALYSIS OF THE MAIN BOOSTER PUMPING STATION

Goal of this chapter is to carry out an energy usage efficiency analysis of the biggest drinking water booster pumping station “Punane” in Tallinn city. Results and procedure can be used in the future as an example to investigate other pressure boosting pumping stations. Focus of current analysis is on reducing energy usage. Through that, also reduction of overall CO_2 production and cutting the operational cost in everyday pumping.

There are many overall best practice guides how to assess and improve performance and efficiency of a pumping station. For example, Sustainability Victoria (2009) guide provides good general understanding and steps how to optimize pumping energy usage. Current efficiency analysis is carried out from a Water Company’s perspective and is based on real measurements.

3.1. Background and concept of the analysis

The main practical reason to start the investigation of energy usage efficiency in WDS pumping is that annual water production in Tallinn city has been constantly decreasing during the last decades since the end of Soviet regime. Summarized annual production is visualized on the Figure 3-1.

Figure 3-1 Annual water production in Tallinn, 1999-2013

The main drastic fall in water consumption took place from year 1996 to 2004 when it dropped over 57%. During this period, many large industrial customers and factories moved out from the city or ceased to exist. Since 2008 annual consumption has been almost stable (around 23,5 million m^3/y). Today Water Company has to deal in everyday operation with challenges like long water retention times or water quality issues that are caused by oversized pipelines and pumping stations. Also, the optimal pumping costs are an issue. Current study focuses on the optimization of booster pumping stations. WDS pipelines related improvements are out of the scope of the current thesis but previously created and calibrated mathematical model of WDS gives good opportunity for that in the future. City population has been slightly increasing since 2004, but also the domestic consumers' awareness to save water has constantly increased. As visualized on the Figure 3-2 since 2004 the calculated average daily water consumption of the domestic customers has been decreased by more than 8% and since 2010 it has been almost stable (around 93,6 l/d per capita based on sales).

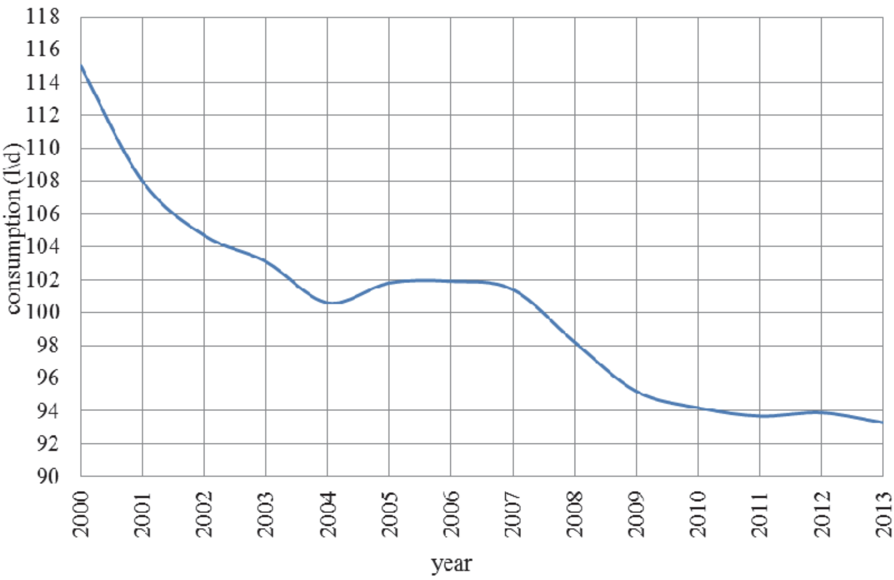

Figure 3-2 Average daily water consumption of domestic consumer 2000-2013

Water Company continuously rehabilitates water network according to financial possibilities to reduce water losses. Average leakage percentage over the years is visualized on the following Figure 3-3. Leakage is defined as unaccounted water (produced water – billed water), which does not include water used for Water Company in production (example pressure washing, cleaning of filters, etc.) and the water used for firefighting. Unaccounted water includes also water theft, meters under-registration etc. Leakage percentage is a theoretical value that has been calculated in Water Company on same principle since 90's. Therefore, it gives some good indication about trends. It shows that leakage level has been almost constant (around 17%) since 2011 that is considerably lower than 26% that is required by the Water Company's service agreement with local municipality. Today Water Company focuses on keeping that level.

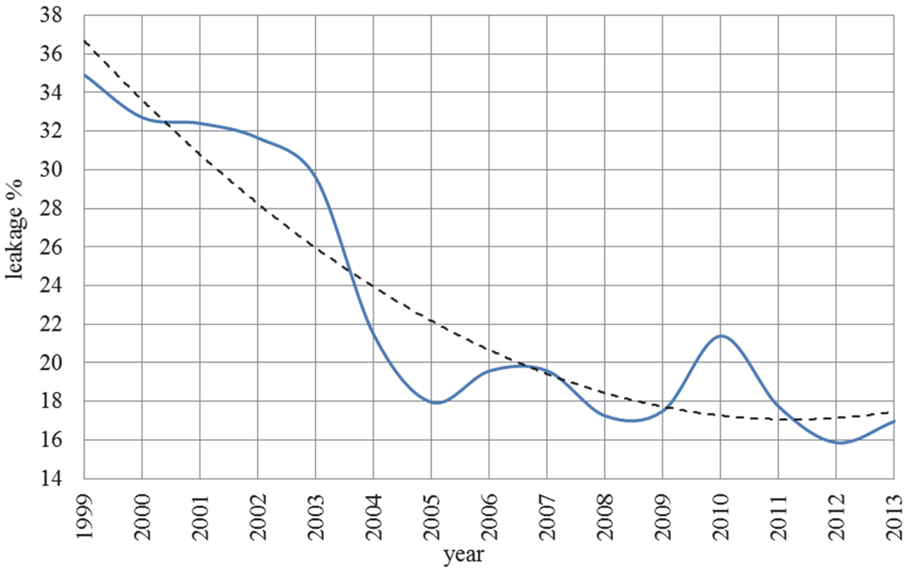

Figure 3-3 Average leakage percentage trend in Tallinn, 1999-2013

All those previously mentioned factors together have stabilized the annual water consumption of Tallinn city. Considering the economic, environmental and financial perspectives, the main large consumers – factories, are not likely to come back to Tallinn, because production in other parts of Estonia is more attractive. The expansion of development areas with new residential areas near the border of Tallinn had a little effect on the overall water consumption of the Tallinn city.

After the creation and calibration of near real-life mathematical model of the WDS was completed, the actual working conditions, efficiency and power consumption of the main booster pumping station "Punane" were taken under investigation as an example. To describe the efficiency assessment process, the main booster pumping station "Punane" was chosen, because it was renovated 16 years ago when water consumption was highest. Thereby it has today a great potential for the optimization of energy usage. The pumping station consist from four identical centrifugal pumps Ahlstrom APP 43-250 (rated flow 290 l/s at 27 m head) that work in parallel mode. All pumps are equipped with variable frequency drives (VFD-s) that programmed to operate based on predefined pressures at discharge point. Figure 3-4 shows the internal view of the booster pumping station "Punane". Station is in good condition and in everyday use.

Figure 3-4 Inside view of the pumping station "Punane"

In practical applications, operating a pump continuously at its BEP is uncommon in systems without storage, because pumping systems must adapt to the constantly changing flow rate and the system head. To save energy, improve reliability of the system and ensure needed fire flow during power failure situations, the usage of storage tanks is strongly advised. As mentioned before, WDS of Tallinn does not include any storage tanks. As there are also no back-up generators in booster pumping stations (except water treatment plant), during power outages customers will lose water service.

The results of current analysis depend heavily on monitored parameters and data quality. *SCADA* system in pumping station "Punane" was implemented in 1999 and therefore it has limited data gathering and storing capabilities. The main parameters are queried reasonably easy from *SCADA* database as it uses also the SQL database structure. Based on that data, weighted average hourly values are calculated for the period of one ordinary consumption week (from 7th of October to 13th of October 2013). Parameters are:

- Pressures in and out from the pumping station
- Discharge flow rate
- Frequencies of variable frequency drives (VFD-s)
- Electrical power and current
- Information when and how many pumps worked

The goal is to maximize the usage of data that is constantly been gathered by Water Company *SCADA* system and based on that assess working condition, energy usage and efficiencies of the pumps. All previously mentioned parameters have been prepared as weekly patterns. That enables better understanding of the dynamical changes and trends in power usage and efficiencies when flow or pump head changes. The following Figure 3-5 describes general process of the analysis and summarizes what parameters are analysed. The upper part shows the data measured and stored in Water Company *SCADA* system. Electrical power is provided directly from the Electricity Company. Other parameters are calculated based on those values.

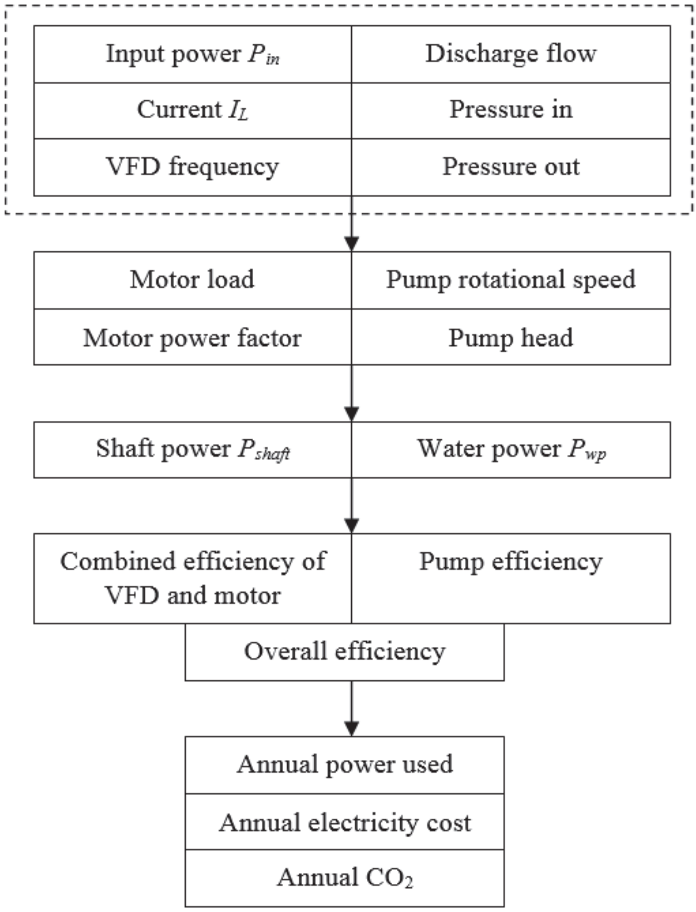

Figure 3-5 Process scheme of the pumping station analysis

Next chapter describes in detail how the efficiency assessment process related parameters are calculated and why. Analysis has been done as an example so that it can be used to assess the efficiencies of other Water Companies' booster pumping stations.

3.2. Measured discharge flow and head

As described in the previous chapter, pumping station "Punane" is equipped with stationary flow meters. Summarized graphs of discharge flow rate (Figure 3-6) and actually added pump head (Figure 3-7) provide a good visualization of demand and pump head fluctuations in time. Test week starts with Monday and ends with Sunday. As people are at home on weekends and using more water than on working day, it explains the flow increase for last two days. In current service area industrial customers practically missing. Therefore, the change in discharge has a common residential district pattern as expected.

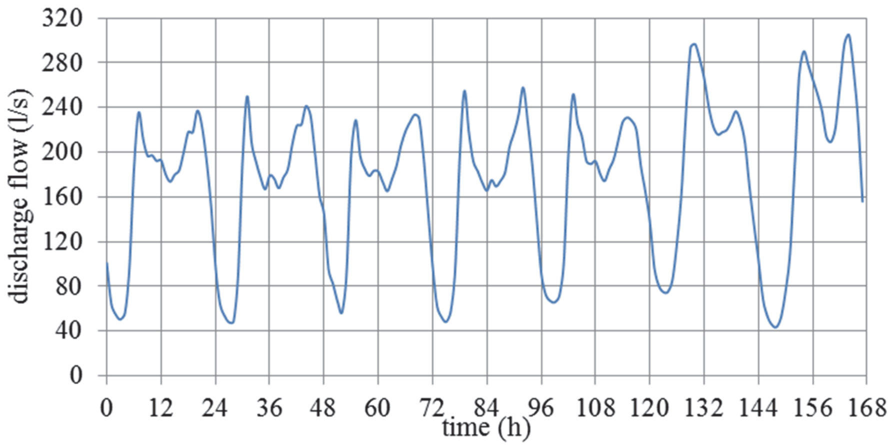

Figure 3-6 Weekly graph of discharge flow

Summarized discharge flow of investigated week from Monday to Sunday (from 7th of October to 13th of October 2013,) was in total 104 902 m^3 that makes ca 5 454 904 m^3 per year.

Because the pumping station "Punane" is a second stage booster pumping station, the actual added pressure depend from pressure regime settings of the water treatment plant main pumping station. That causes the temporal high peaks near midnight when in main pumping station pressure is lowered. Still Figure 3-7 shows a clearly visible pattern. Data presented in the Figure 3-7 is calculated based on *SCADA* data – difference between outgoing and incoming pressure heads.

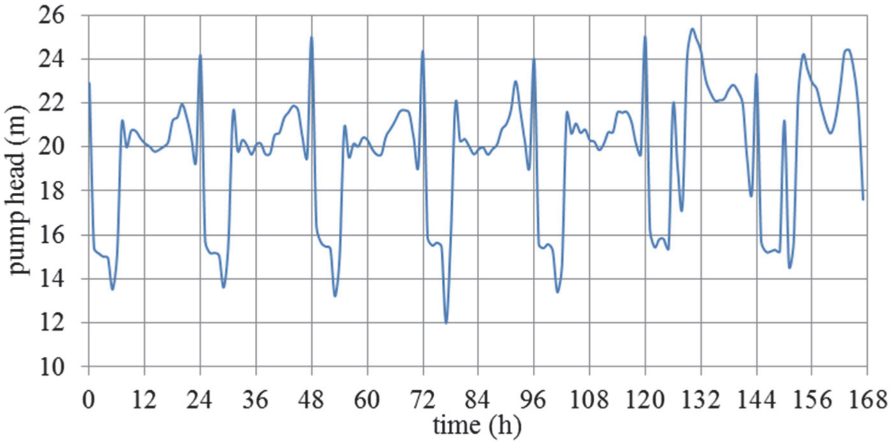

Figure 3-7 Weekly graph of actually added pump head

Current VFD-s are programmed to react to pressure head change every 15sec. Comparing pumps rated flow and head with actual flow and head, it is clearly visible that pumps never reach optimal working conditions and therefore running most of the time at very low efficiency. It is a typical case, when operators see usage of VFD-s as a magical tool to solve oversized pumps issue.

Pumping station "Punane" pressure regime weekly settings are visualized on Figure 3-8 and compared with actual measured data. It clearly shows that pumps generally hold the predefined pressure settings well.

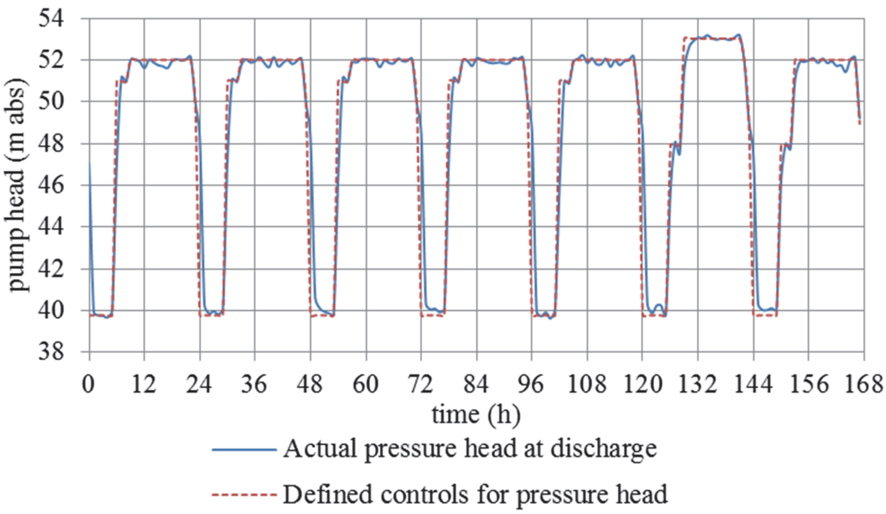

Figure 3-8 Predefined pump target head controls versus actual pressure out

Collected flow and pump head weekly data will be used afterwards to carry out pressure regime optimization in the chapter 4.2 where the detailed description of old and optimized pressure regime settings are given and compared. The data is also used to analyse pumps efficiencies and power usage.

3.3. Measured electrical power, reactive power and current

The electrical power P_{in} and reactive power are constantly measured before variable frequency drives by the Electricity Company. That provides very detailed data for power usage analysis. Only negative issue to overcome was that power is measured as total for all pumps together and not for every individual pump. As at the monitoring period only 1 pump from 4 was working, the measured total electrical power was considered as electrical power that is used by the working pump. To illustrate electrical power usage during the entire measured week, the results are given as a weekly graph in the following Figure 3-9.

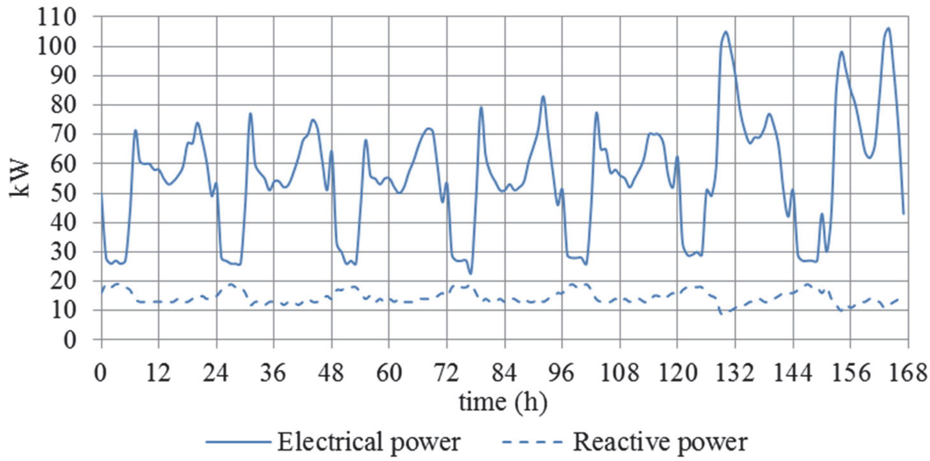

Figure 3-9 Weekly graph of measured electrical power and reactive power

Pumping station consumed 9312 kWh of electrical power per week that makes ca 484,224 kWh per year. In addition, the current I_L has been also continuously measured and stored in *SCADA*. Summarized hourly results are visualized on Figure 3-10.

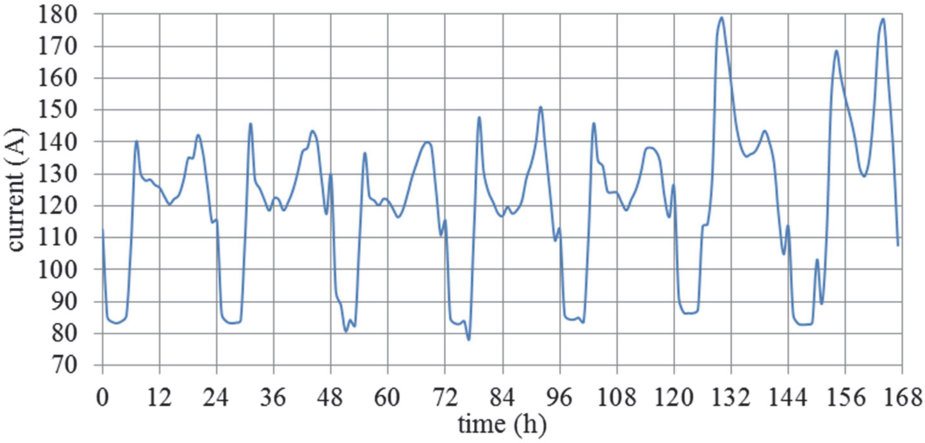

Figure 3-10 Weekly graph of measured current

Figure 3-9 and Figure 3-10 clearly show that power consumption is highest at weekends when also consumption is highest.

3.4. Pump rotational speed

To assess pump efficiency the main data source after discharge flow and pump head is pump rotational speed. As the frequency of all separate VFD-s was available from *SCADA* database, no direct pump rotational speed measurements were carried out during the current study. Data was consistent and usable to calculate the pump actual rotational speed *N* average hourly values as flow and head are also given as average hourly values. The pump average synchronous rotational speed S_S is calculated based on Eq. (17).

$$S_s = \frac{2 \cdot f \cdot 60}{p} \tag{17}$$

where

S_S – Synchronous speed (1500 rpm at 50 Hz)
f – Electrical frequency changed by VFD
p – Number of poles (4 by current motor of pump)

The synchronous speed S_S is theoretical and cannot be achieved by any loaded motor because of slip that reduces rotational speed. Slip value s for the current motor is 1.33 % and it is calculated based on Eq. (18).

$$s = \frac{S_s - S_a}{S_s} \tag{18}$$

s – slip (between 1-3 %)
S_a – asynchronous speed given by manufacturer, 1480 rpm

Actual pump rotational speed N is calculated using synchronous speed S_S that is calculated based on electrical frequency f recorded after VFD with equation Eq. (17) and then reduced by calculated 1.33 % slip value. To illustrate rotational speed changes during the monitored test week, the results are given as a weekly pattern of pump rotational speed N hourly values on the
Figure 3-11.

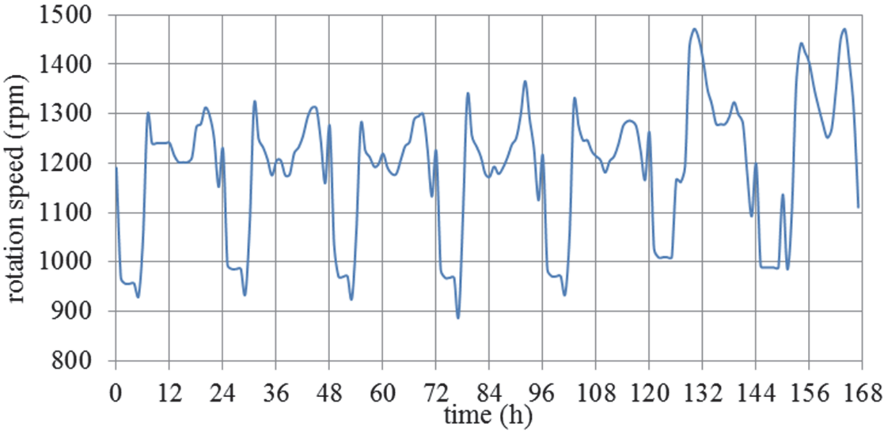

Figure 3-11 Pump rotational speed weekly pattern

We can see From Figure 3-11 that motor rotational speed (that is equal to pump rotational speed) is never reduced under 40-50% of the pump nominal speed (1480 rpm). So the Affinity Laws (7) based approximations can be used with sufficient accuracy to downside parameters like flow, pump head or power absorbed at nominal speed according to actual speed. Therefore actual pump rotational speed N is used in chapter 3.7 for the interpolation of actual pump shaft power P_{shaft}.

It is known, that VFD efficiency decreases with decreasing motor load, especially with drives that have smaller horsepower ratings. Efficiency VFD-s vary between different brands and models, but often the exact efficiency of VFD data from manufacturer is unavailable. As currently used VFD-s are powerful (rated power 110 kW), the efficiency of the VFD can be considered almost as stable (96-97%) when motor load is above 50% of rated power output of the VFD. Therefore, it is very important to assess the electrical motor load.

3.5. Motor load

To analyse the actual working condition of the electrical motor the load of the motor is calculated based on Eq. (19).

$$Load = \frac{I_L}{I_{nameplate}} \cdot \frac{V_L}{V_{nameplate}} \tag{19}$$

where

I_L – Current measured (A)
$I_{nameplate}$ – Current at full load of the motor (A)
V_L – Voltage measured (400 V)
$V_{nameplate}$ – Voltage at full load of the motor (A)

As voltage is considered as stable, the second component of the equation is taken as 1. The results are visualized on the following Figure 3-12.

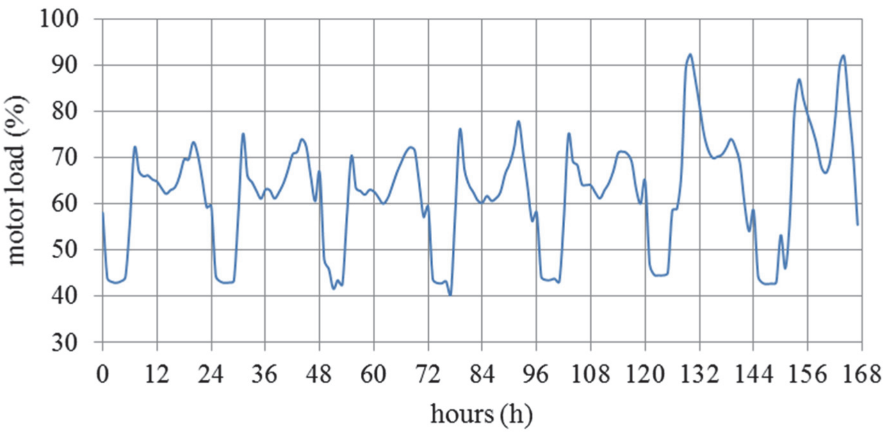

Figure 3-12 Motor load weekly pattern

As Chaurette (2003) describes, when motor is operated at more than 50% of the nominal load, the motor efficiency of big pumps (> 20 kW) can be taken almost as stable. As visible on the Figure 3-12, the motor load drops at night under 50% that indicates at light loads and not optimal working condition of the motor. Calculated motor load values are also used to analyse relationship with cos φ in the next chapter.

3.6. Motor power factor

To understand the working conditions of the motor and due to the manufacturer's graph for existing motor of cos φ not being available, the motor power factor or also known as cos φ was calculated. Only the electrical power P_{in} and current I_L readings were available for the current research calculations. Based on the principles described by Principles of Power System, Third Edition (2005) cos φ can be calculated based on Eq. (20). Results for investigated weekly period are visualized on the following Figure 3-13.

$$\cos\phi = \frac{P_{in} \cdot 1000}{\sqrt{3} \cdot V_L \cdot I_L} \tag{20}$$

where

P_{in} – Electrical power (kW)

I_L – Current measured (A)

V_L – Voltage measured (400 V)

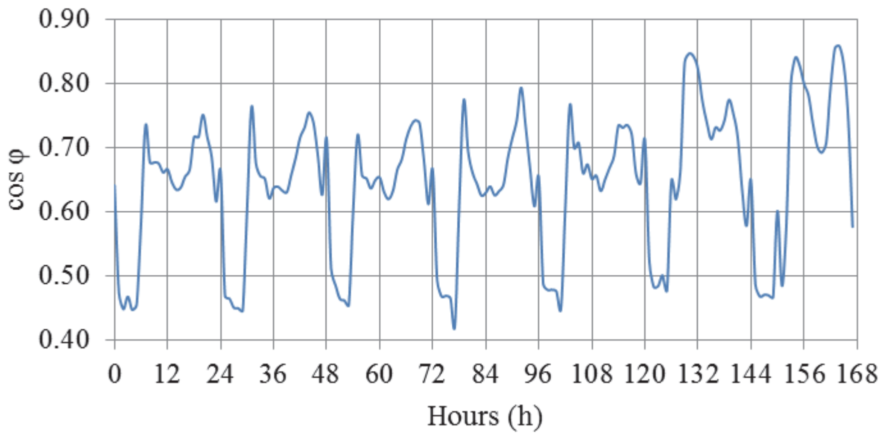

Figure 3-13 Motor power factor weekly pattern

The greater the motor cos φ, the higher is the energy delivered by the motor to the shaft of the pump. It is known that low cos φ causes poor motor efficiency. As visible on the Figure 3-13 the cos φ drops at night under 0,5 showing that the electric motor works at very light loads and in not optimal conditions. As Mehtla and Mehtla (2005) mentions, induction type motors work at cos φ which is extremely small at light loads (0.2-0.3) and rises to 0.8-0.9 at full load. The following Figure 3-14 illustrates the relationship between cos φ that is calculated with Eq. (20) with the motor load calculated previously with Eq. (19).

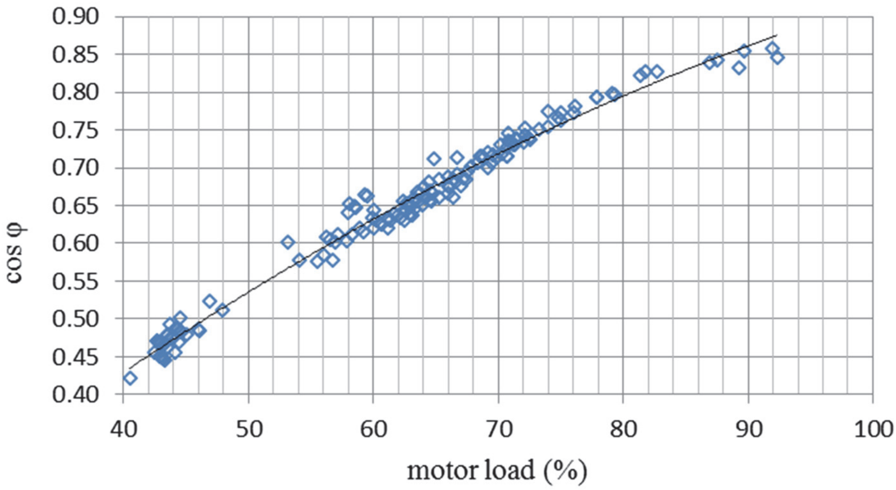

Figure 3-14 Relationship between motor load and power factor

As the Figure 3-14 shows, there is a clear correlation between current motor load and cos φ values. Because calculated cos φ points are mainly concentrated near the trend line and there are no big discrepancies in results, it can be assumed that the quality of electric power measurements provided by Electricity Company and current stored in *SCADA* are trustworthy and acceptable.

The author realized and proposed to install local independent electricity meters for each pump to simplify the analysis process for the future when several smaller pumps work at the same time, the value of electrical power P_{in} has to be measured separately for each pump. The Water Company realized that issue and has already started installing separate meters.

3.7. Pump shaft power

To understand power losses in different power levels described on the Figure 1-3 and to analyse efficiencies of the existing VFD, motor and pump the actual pump shaft power P_{shaft} is needed. Nominal shaft power at nominal speed is given by the pump manufacturer (Figure 3-15). As impelled diameter of the current pump was 355 mm, the nominal shaft power values at given flow where interpolated from the manufactures graph Figure 3-15.

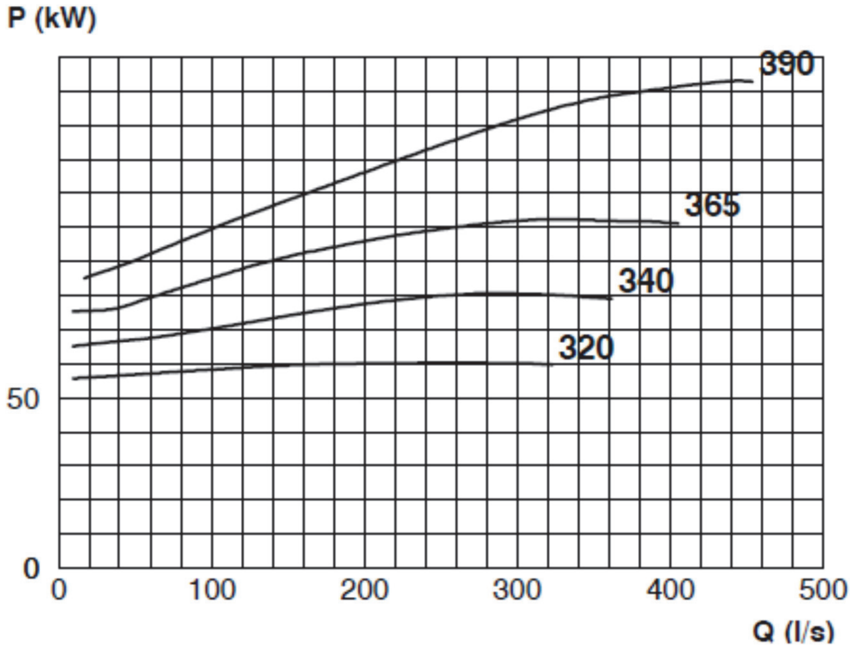

Figure 3-15 Ahlstrom APP 43–250 centrifugal pump shaft power at nominal speed based on impeller diameter

As mentioned before, shaft power P_{shaft} cannot be measured directly in operation, so the Affinity Laws Eq. (7) based approximation is used to downsize it based on pump actual rotational speed. For the test week, the average hourly values of actual pump shaft power P_{shaft} in kW are calculated based on Eq. (21), which is a derivation from the previously described Affinity Laws Eq. (7).

$$P_{shaft} = \frac{P_{shaft,nominal} \cdot N^3}{N_{nominal}^3} \tag{21}$$

where

$P_{shaft,\ nominal}$ – Pump shaft power at pump nominal rotational speed
$N_{nominal}$ – Pump nominal rotational speed
N – Pump actual rotational speed (calculated in chapter 3.4.)

Figure 3-16 illustrates the relationship between measured electrical power P_{in} described in the chapter 3.3 and actual pump shaft power P_{shaft}.

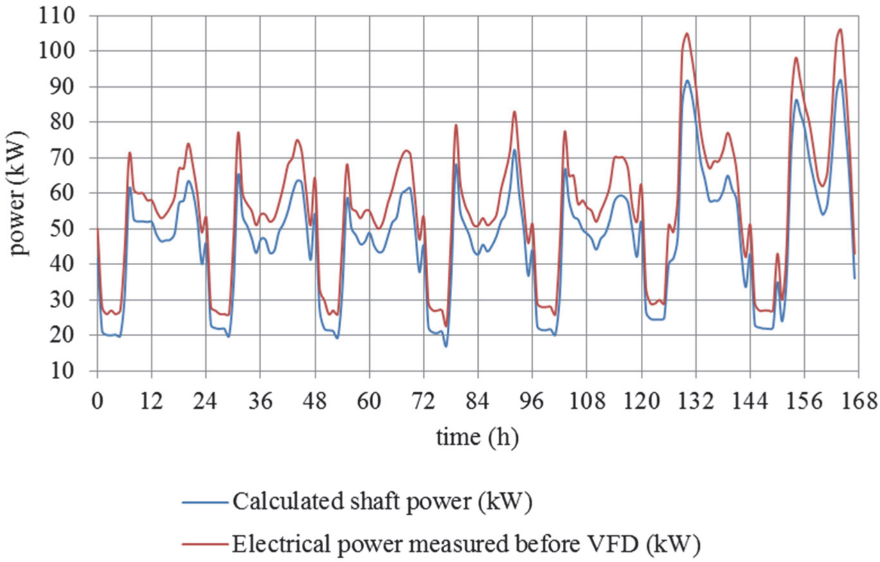

Figure 3-16 Electrical power P_{in} and shaft power P_{shaft} weekly patterns

3.8. Pump wear

The aim of current chapter is in addition to chapter 5.3, analyse the possibilities to optimize a group of similar pumps working in parallel. Even if the pumps are the same, the characteristics of identical pumps vary in everyday operation. A pump's efficiency usually degrades during normal operation due to wear by as much as 10% to 25% before replacement. Running a pump continuously under an extreme regime below or above its BEP will reduce pump's lifetime considerably. As investigated before, the pumps in pumping station "Punane" run most of the time at a very low efficiency. Following Figure 3-17 compares efficiencies of a new pump described by manufacturer's performance graphs and a same pump after 16 years of operation in the WDS of Tallinn. Efficiencies of a new pump are given as lines and actual efficiencies of the old pump are given as points.

It is practically very difficult to plan and carry out flow and pump head performance measurements on working pump that is already in active service. To cover the complete operational range of the pump with measured points, pump has to be dismantled first to carry out tests in laboratory. Sadly, the dismantling of the pump in pumping station "Punane" and actual pump performance testing was not in in the scope of current thesis.

Operational parameters were constantly measured and stored in the *SCADA* system, which enabled to analyse indirectly the changes in pump parameters in time. The set of data depends on the required discharge and the pump head that change constantly according to WDS requirements. To monitor specific flow and pump head cannot be achieved on live WDS. Therefore, the Figure 3-17 gives a summarized overview what are used pump efficiencies according to actual discharge flow and the pump head. Depending on predefined pressure regime, the points are concentrated into two main groups of points that correspond to night- and daytime. As it can be seen on Figure 3-17, the efficiency of the pump decreased slightly (by 7-10%) with 16 year in operation. This proves that the characteristics of a pump used for several years are not identical to the new one and has to be taken into account.

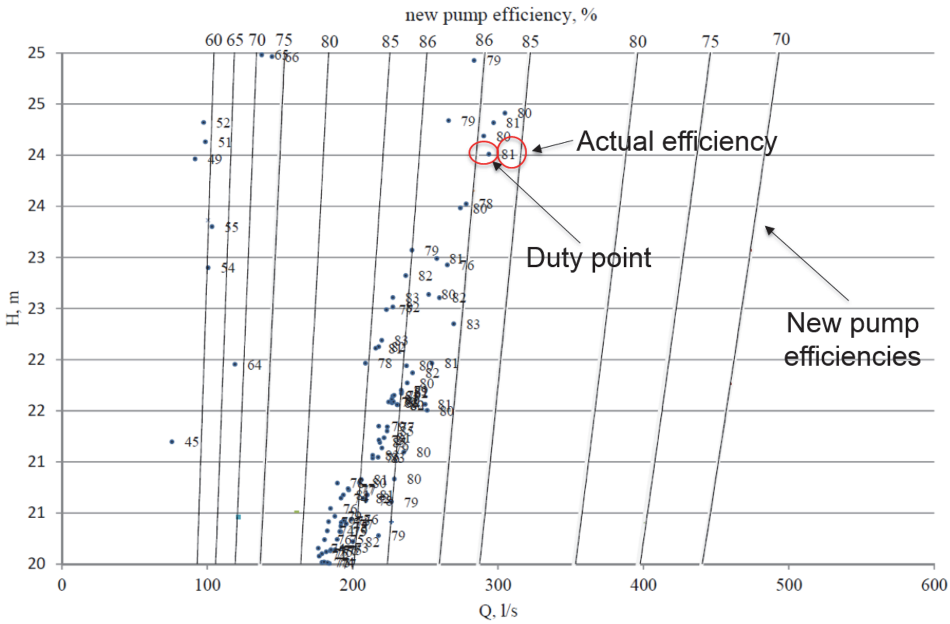

Figure 3-17 Pump efficiency of a new Ahlstrom APP 43–250 pump and one after 16 years of use

3.9. Variable frequency drive and motor combined efficiency

Because only electrical power P_{in} and calculated shaft power P_{shaft} were available for efficiency analysis, the efficiencies of the VFD and motor are handled together as one and calculated based on Eq. (22) that was derived from the Eq. (11).

$$\eta_{VFD} \cdot \eta_m = \frac{P_{shaft}}{P_{in}} \tag{22}$$

Relationship between P_{shaft} and P_{in} describes combined efficiency of the motor and the VFD that is visualized on the following Figure 3-19.

3.10. Pump efficiency

By Ahlstrom, the maximum achievable pump efficiency of existing APP 43-250 at full speed is 86,5%. Pump efficiency dependence from flow is given on following graph.

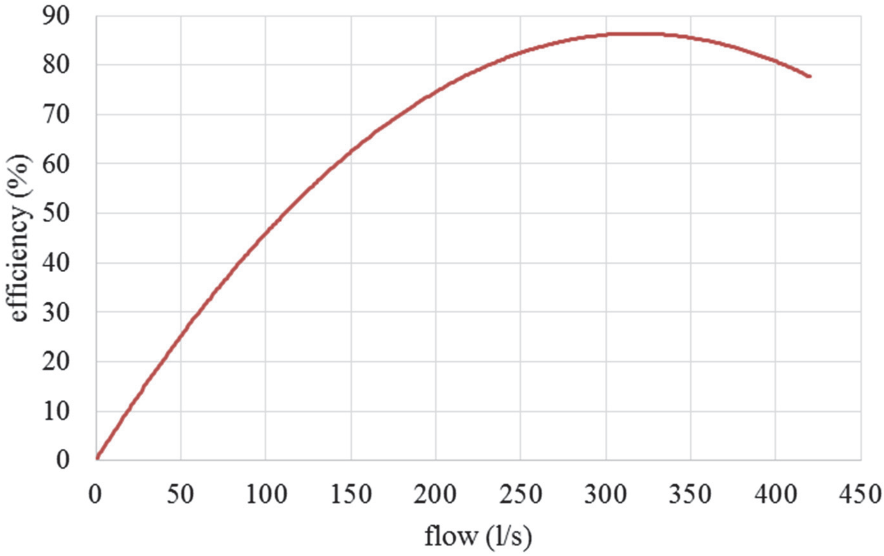

Figure 3-18 New Ahlstrom APP 43-250 pump efficiency vs. flow curve based on manufacturer data

The formula to calculate hourly pump efficiencies can be derived from the Eq. (9) and P_{wp} can be found based on the Eq. (8).

$$\eta_p = \frac{P_{wp}}{P_{shaft}} \tag{23}$$

Based on previously calculated P_{shaft} hourly values and P_{wp} hourly values for the test week, the pump efficiency pattern is calculated by Eq. (23) and visualized on the following Figure 3-19.

Every moving part of the pump wears and finally causes a drop in overall efficiency, which may reach to around 10-12.5% in case of unmaintained pumps. When pump wears, the BEP tends to shift to the left on the pump characteristic curves. Much of the wear occurs in the first few years so that regular pump maintenance is crucial to assure the expected performance parameters. The main things to look at when estimating pump weariness are:

- cavitation or internal recirculation
- pump impellers and casings that increase clearances
- between fixed and moving parts
- wear rings and bearings
- packing adjustment on the pump shaft

3.11. Overall efficiency of the VFD-motor-pump system

The overall efficiency of the VFD-motor-pump system can be calculated based on water power P_{wp} and electrical power P_{in} or by multiplying combined efficiencies of the VFD, motor and pump efficiency. To exclude any errors related with the calculation of separate efficiencies, the overall efficiency is calculated based on power data with Eq. (24) that is derived from Eq. (11).

$$\eta_{VFD} \cdot \eta_m \cdot \eta_p = \frac{P_{wp}}{P_{in}} \tag{24}$$

To understand relationships between VFD x motor, pump and overall efficiency the results are summarized on the following Figure 3-19.

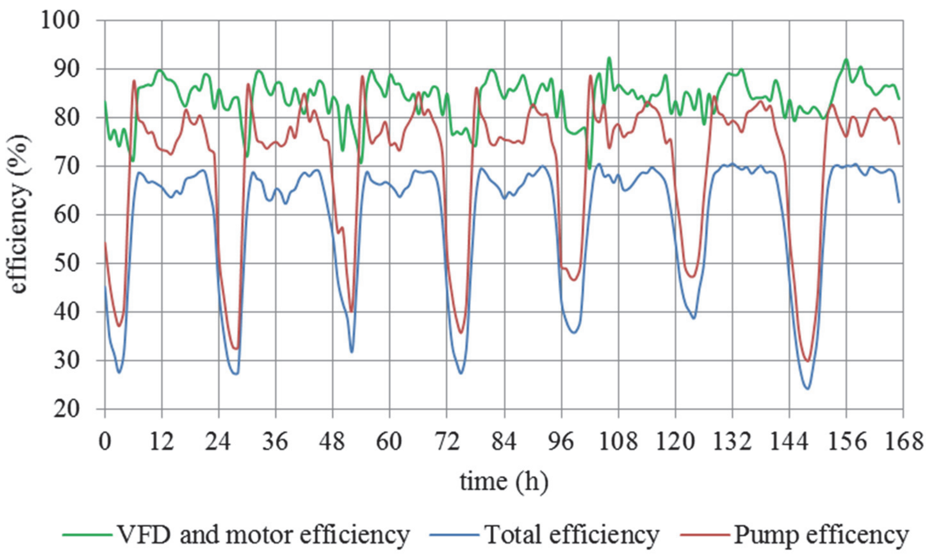

Figure 3-19 Existing VFD, motor and pump efficiencies weekly patterns for existing APP 43-250 pump

It helps to identify, which component has the lowest efficiency and thereby the biggest negative effect on the overall efficiency of pumping. Figure 3-19 shows that VFD and motor combined efficiency does not vary as much as pump efficiency. Low overall efficiency at night is mainly caused by low pump efficiency that drops below 40%. As identified in the previous chapter 3.5, the electric motor works at night with light load and thereby not in optimal condition.

The situation when pump efficiency drops but VFD and motor combined efficiency rises can be explained by the fact that both efficiencies have different dependence from flow as visualized on the following Figure 3-20. It shows that pump efficiency is more sensitive to flow change and drops at a much faster rate. At flows greater than nominal flow, the pump efficiency starts so drop again but motor and VFD combined efficiency is still rising.

As expected, running pump at very low flow and head comparing to nominal figures, will cause very low overall efficiency of pumping and main negative impact comes from low pump efficiency.

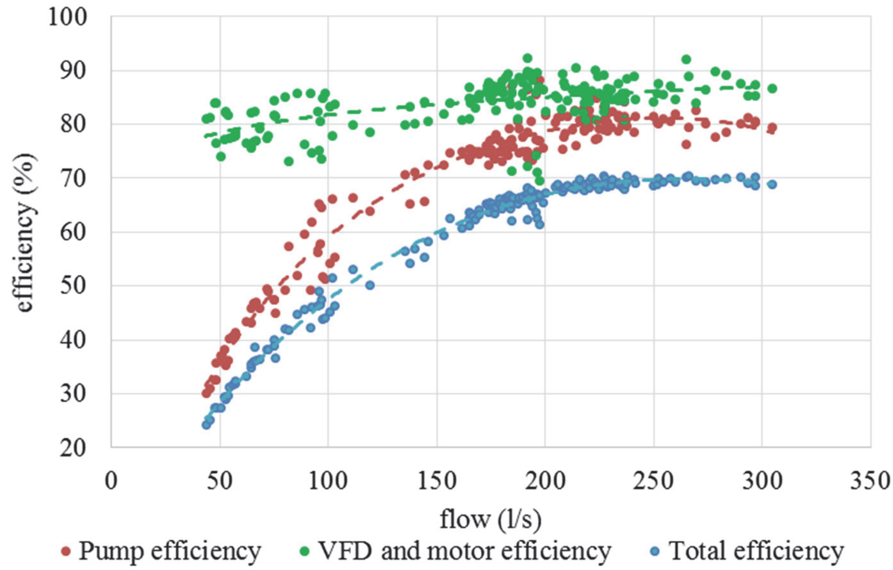

Figure 3-20 Efficiencies dependences on flow

3.12. Electricity cost of pumping

Walski and Hartell (2012) pointed out that in order to calculate energy cost according to complex tariffs, the energy use and cost calculations have to be handled separately.

As in January 2013, Estonia joined with the biggest electricity exchange market in region - Nord Pool Spot, the electricity pricing and calculation principles became very complex. Prices on the free market vary on an hourly basis making it nearly impossible to predict changes in operational cost. Clients can agree to fix electricity price for one year, but it gives no guarantee to have the best price.

The main components of electricity pricing in Estonia are:

- Network services (amperage charge, network service fee)
- Renewable energy fee
- Electricity excise tax
- Electricity price (day, night; winter, summer)
-

Electricity price for Water Company is not related with how much is used. Based on the electrical power P_{in} visualized in the Figure 3-16 and methodology of the kWh unit pricing in Estonia, the average hourly electricity cost is calculated for a weekly period. Total weekly electrical power consumed calculated in the chapter 3.3 was 9312 kWh. The electricity unit price for kWh at this week was 0.0913 eur/kWh. In that case, the cost of pumping for the monitored test week is 900.75 eur +VAT that makes ca 46 839 eur +VAT per year. The total pumping cost hourly pattern for monitored test week is given in following Figure 3-21.

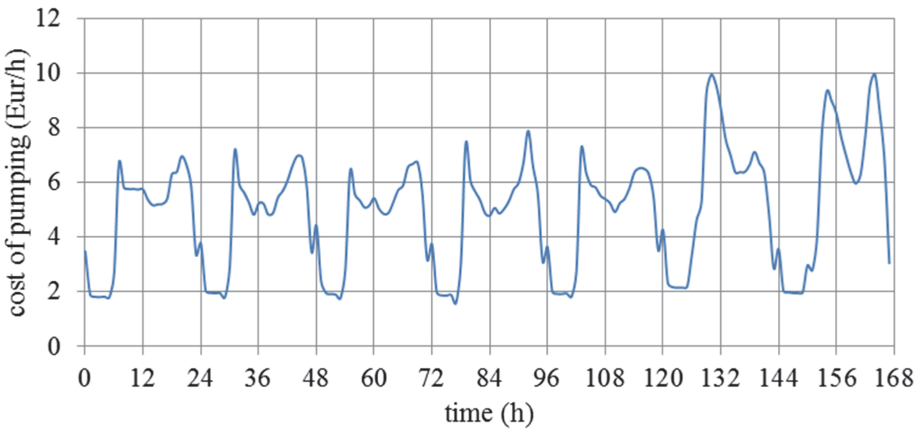

Figure 3-21 Average hourly cost of pumping in PS "Punane"

By dividing the total weekly electricity cost by total weekly volume of pumped water, we can calculate out the average unit cost of pumping (eur/m^3). That is indicative and can be used to compare different pumping stations unit costs to identify energy usage issues. The following Table 3-1 gives an overview about estimated annual unit cost of other Water Company booster pumping stations. Common trend is that pumping unit cost in smaller pumping stations is higher than in bigger ones.

Table 3-1 Eur/m³ unit costs of other booster pumping stations

	annual		unit cost	
Name, year	**kWh**	**m³**	**(kWh/m³)**	**(eur/m³)**
Keldrimäe, 1997	10 829	47 495	0.228	0.0520
Sütiste, 2006	10 449	72 560	0.144	0.0257
Õismäe, 2009	5 054	58 764	0.086	0.0196
Sõle, 1997	2 693	43 430	0.062	0.0121
Punane, 1997	*484 224*	*5 454 904*	*0.089*	*0.0086*
Tuvi, 1998	29 320	424 933	0.069	0.0069
Tondi, 2006	378 132	5 109 891	0.074	0.0066
Kirsi, 1997	3 253	77 453	0.042	0.0064

The eur/m³ column also includes the cost for amperage usage. Total annual cost of electricity used to pump ca 11.3 million m³ is ca 87 600 eur +VAT forming an average unit cost of ca 0.0078 eur/m³ +VAT. All previously listed booster pumping stations, especially those that were renovated before 2004 when water consumption was high, have great potential for the optimization of energy usage. The pumping station “Punane” has the second biggest annual pumped volume after water treatment plant (WTP) main pumping station.

3.13. Summary of the analysis

Chapter 3 provides good instructions and methodology what parameters to analyse and how to assess the working condition and efficiency of the pumps in booster pumping stations based on the existing *SCADA* data. Analysis show that Ahlstrom APP 43-250 pumps currently in use in pumping station “Punane” are oversized and cause very low pump efficiency, especially at night hours. All that causes above average pumping unit cost.

Next chapter introduces good engineering practices as an example how to reduce energy usage in the main pumping station “Punane”. It includes replacement of pumps, correction of pumps hourly pressure control settings based on previously created hydraulic model of WDS and defines the achievable savings and benefits.

4. GOOD ENGINEERING PRACTICES TO SAVE ENERGY IN MAIN PUMING STATION

As previously mentioned in chapter 3.1, the annual water production in Tallinn has been decreased since 1999 by more than 57%. Therefore, further research has been carried out to understand how big is the negative effect of decreased production in terms of energy usage in pumping based on real-life WDS pumping station “Punane” example. Current chapter focuses on how to use less energy in booster pumping stations with identical centrifugal pumps.

4.1. Replacement of oversized pumps

Total production and necessary pumping head for the test week were previously analysed in the previous chapter 3 and results where visualized on the Figure 3-6 and Figure 3-7. Existing Ahlstrom APP 43-250 pumps are clearly oversized for current WDS. Walski (2001) examines based on small pumps (1.26-31.5 l/s) various alternatives of pumping configurations to identify the design that will produce the least cost and gives overall guidelines for design selection based on life-cycle costs.

Based on test week measurements, the average needed flow was 173.45 l/s and average needed pressure head was 19.66 m. Max peak flow was 304.79 l/s and peak pressure head 25.35 m. Dead end pumping configuration with 4 identical pumps with VFD-s is currently used.

The idea is to achieve higher efficiency of the pumping system and thru that save energy, especially at night flows when only one pump of minimal three identical pumps should run at nominal speed. Thereby, the new pump should also run at the highest efficiency compared to the current big pump running at drastically reduced rotational speed. Water Company has also set a precondition from the practical point of operation - all proposed pumps should be identical as they are today. That makes everyday operation simpler and eliminates the need to keep extra spare parts of different types of pumps in store. Using of two different kind pumps to handle the flow/head variations could be still justified when same type of pumps are used elsewhere in the system.

New pumps with smaller rated flow (115.56 l/s) and rated head (26m) are proposed based on the principle, that one pump can effectively cover average night-time requirements and three pumps can provide needed daily flow rate at required head. Fifth pump will be in hot reserve to cover fire flow. 100 l/s is a maximum fire flow requirement stated by rescue department. When more flow is needed, local water tank inside the properties are designed to provide additional water. As 100 l/s is near rated flow of proposed pumps, there is no need to use bigger pump. By regulation, in case of fire the pressure head can drop temporally until 10 m when maximum fire flow is taken.

There is always a risk to choose “to precise” pumps for network needs. Lingireddy and Wood (1998) pointed out with regard to precise optimization of VSP-s, the network reliability against big bursts or fire flows is decreasing. To overcome that risk, the Water Company is considering installing one extra pump for active reserve (4+1). There is no risk in increasing the overall capacity of the station in the future, because pressure pipes are oversized and the pumping station facility has sufficient extra space to add more pumps when needed.
In the current thesis, the new smaller, identical Grundfos 98274752 centrifugal pumps were proposed as an example, because they have good online software (WebCaps) that enables quick analysis of different parameters by various pump heads and discharges. In addition, Water Company has good experience and operational history and knowhow how to operate and maintain those pumps. The following Figure 4-1 presents the main technical characteristics of the proposed Grundfos 98274752 centrifugal pump.

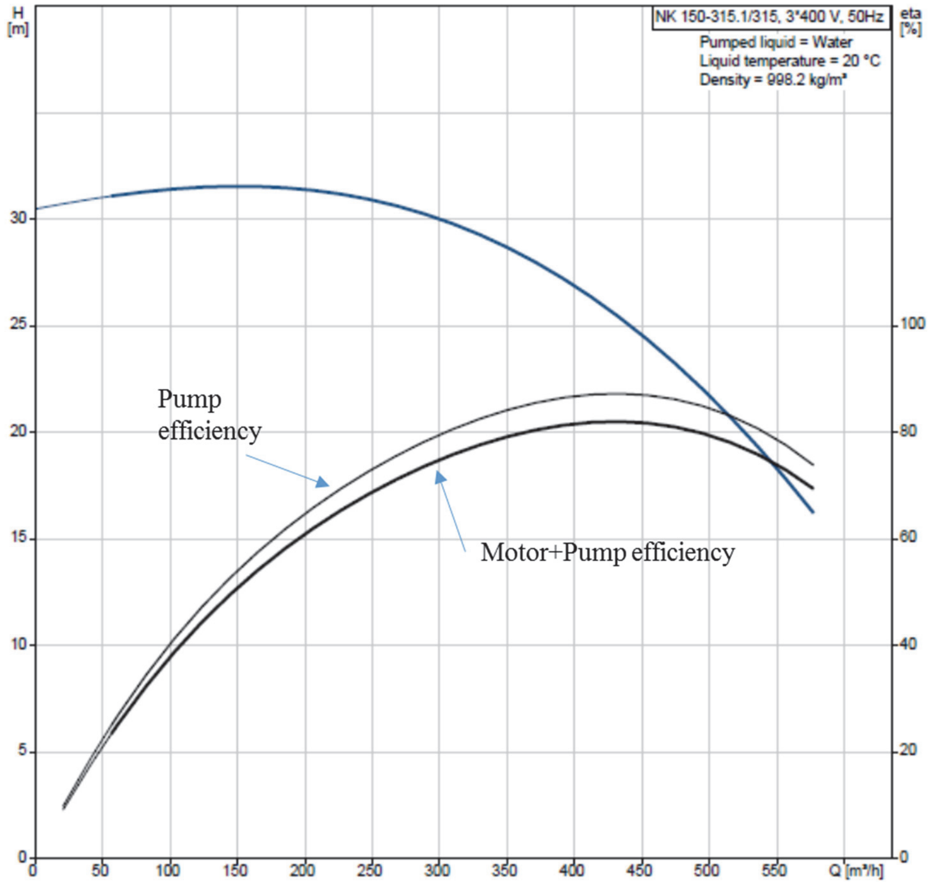

Figure 4-1 Characteristics of proposed new Grundfos 98274752 centrifugal pump

4.2. Choosing more optimal pressure control settings for pumps

In current study, the pressure regime is defined as a pump control setting that describes required pressure at discharge point versus time. The pressure control settings for pumping station “Punane” are overviewed based on 10 main critical points in WDS (Table 4-1). Points were chosen based on customer complaints history, ground elevation and height of the houses. The Water Company has ca 10-15 years of good statistics that enables to specify pressure critical customers.

Table 4-1 Pressure critical points

Nr	Junction Label	Number of floors in area	elevation (m abs)
1	J_13116	9	42.94
2	J_27948	9	42.40
3	J_116480	9	42.34
4	J_305822	9	41.94
5	J_28840	9	41.91
6	J_327888	8	41.58
7	J_256350	9	40.07
8	J_33159	9	39.99
9	J_295914	9	39.66
10	J_30230	9	39.58

Because the pipes in the current pressure zone are historically oversized, the main effect comes from terrain height change. As shown on the Figure 4-2, optimization points cover all area and locate almost evenly across the pressure zone.

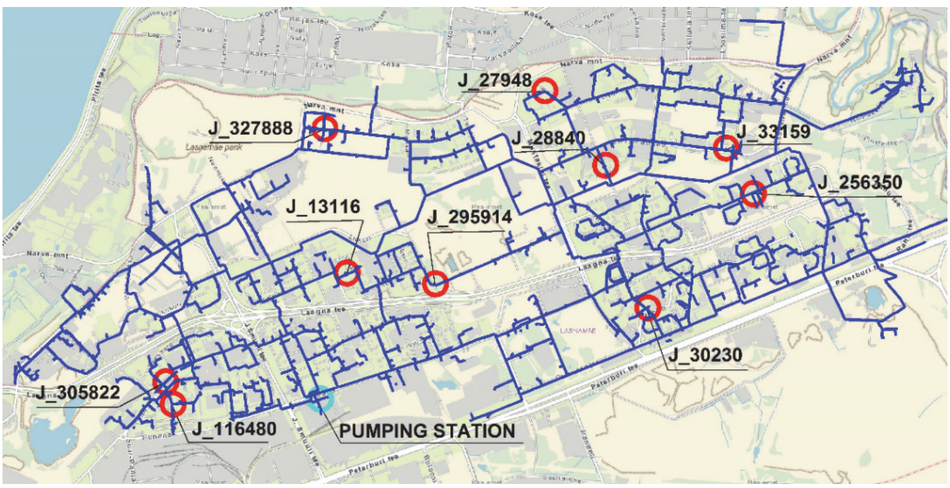

Figure 4-2 Location of pressure critical points

Special attention was given to identify periods and hours when actual pressure in critical points of the WDS can be lowered until the minimally recommended pressure. That ensures normal water supply availability in the houses (15 m H_2O at the highest floor of the building) and helps to avoid unnecessary pumping cost. Investigation and testing was carried out using previously created hydraulic model. To illustrate the results the summarized weekly patterns are visualized on Figure 4-3 (before) and Figure 4-4 (after).

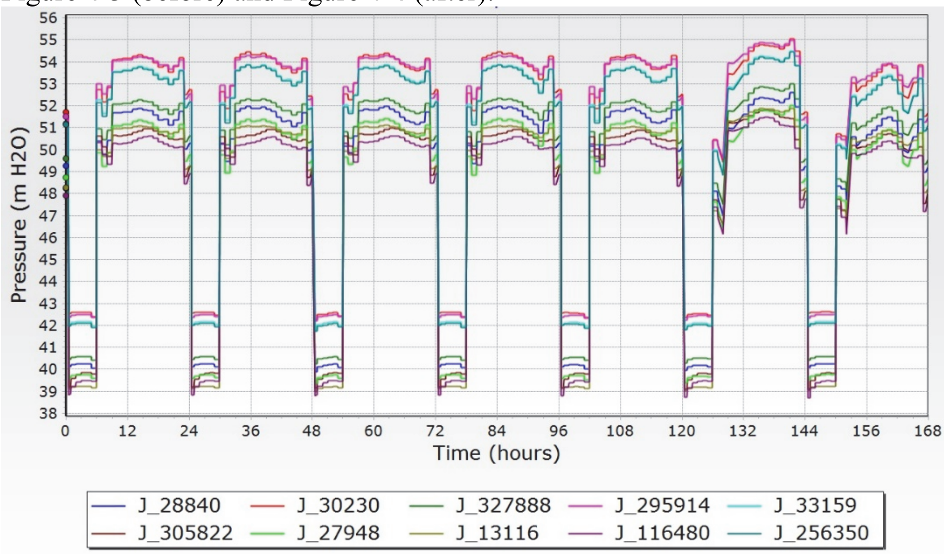

Figure 4-3 Pressures at critical points using old pressure control settings

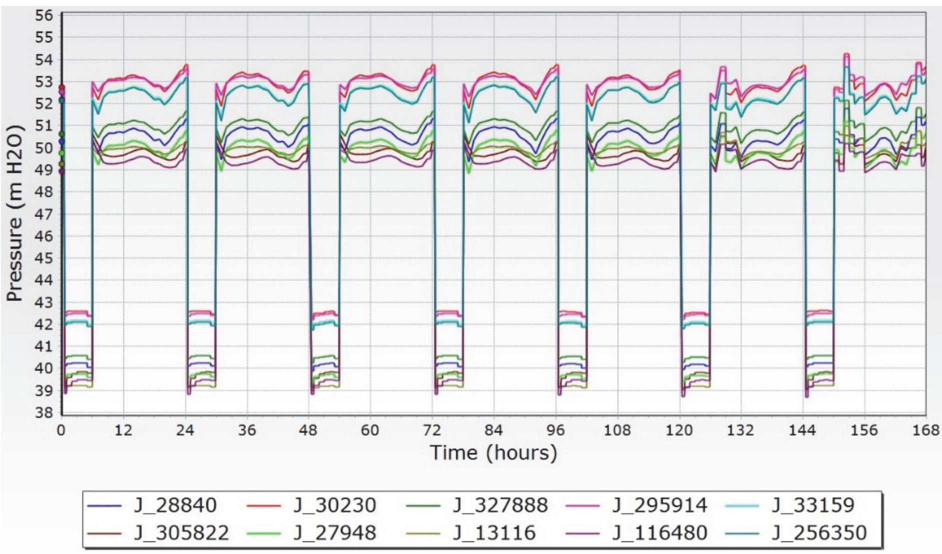

Figure 4-4 Pressures at critical points using new pressure control settings

It shows that on a typical working day the effect of changing pumps pressure control settings is not so high. Stronger effect has been achieved on weekends. Also, the following daily pressure patterns Figure 4-5, Figure 4-6, Figure 4-7 and Figure 4-8 show that pressure conditions are more unified and there are fewer unnecessary pressure fluctuations.

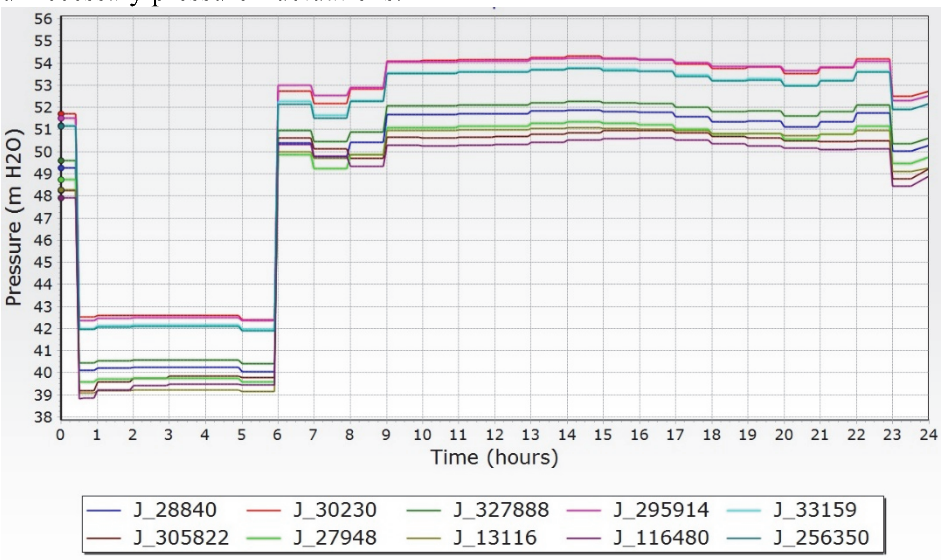

Figure 4-5 Typical working day pattern using old pressure control settings

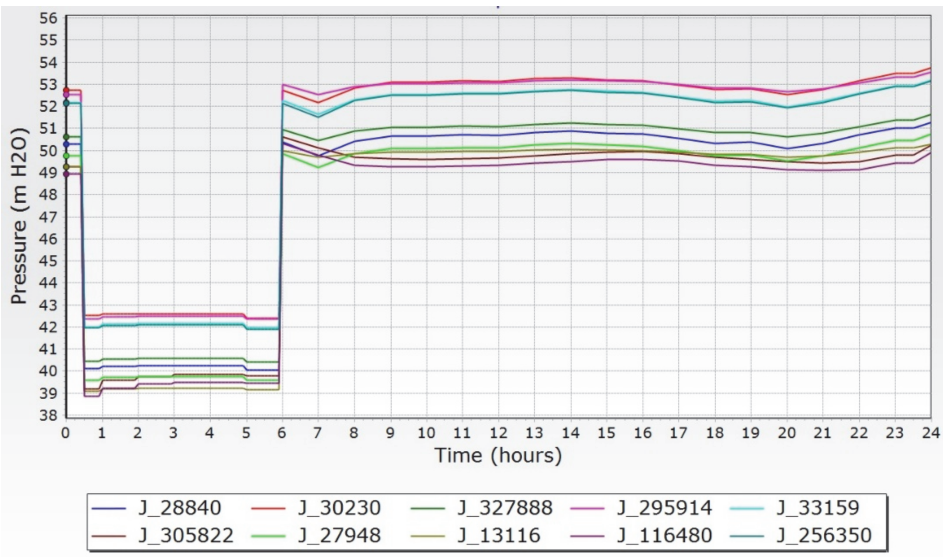

Figure 4-6 Typical working day pattern using new pressure control settings

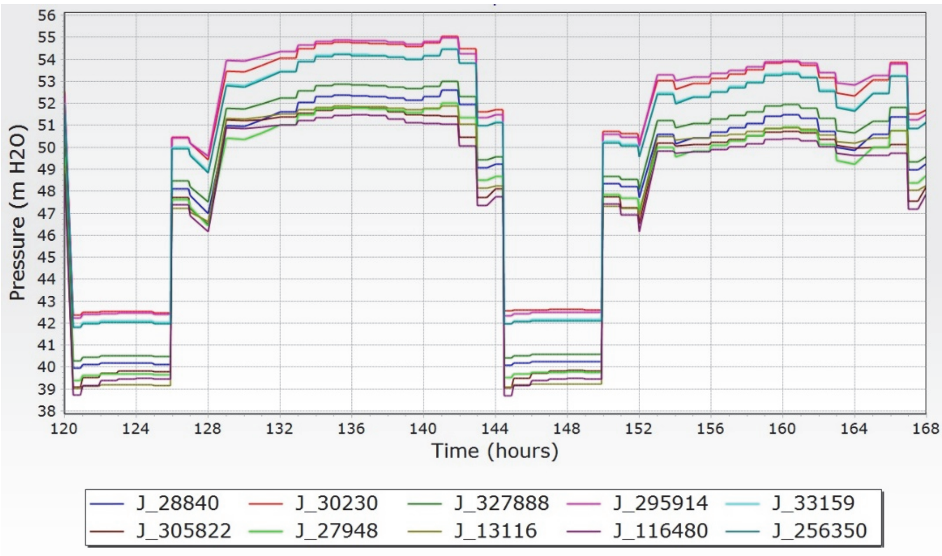

Figure 4-7 Typical weekend pattern using old pressure control settings

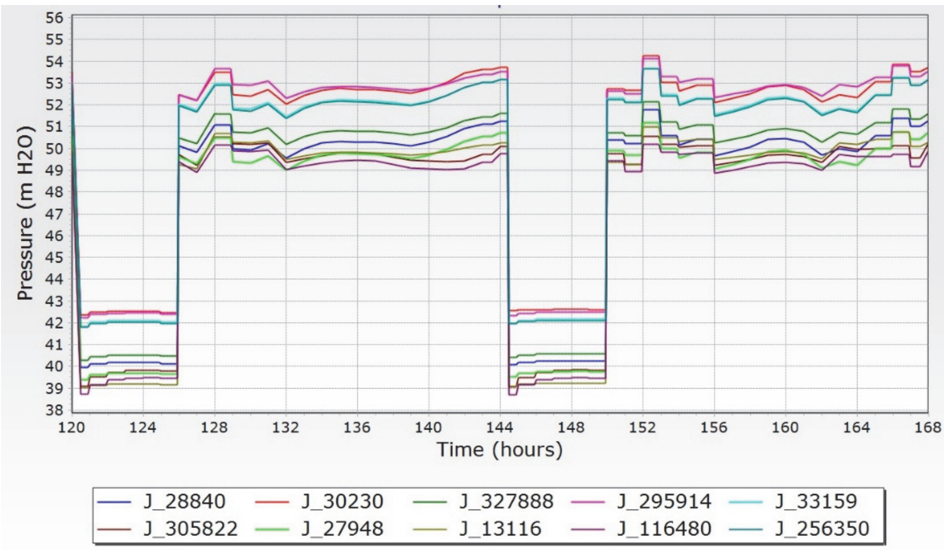

Figure 4-8 Typical weekend pattern using new pressure control settings

As shown on following Table 4-2, the pressure at night from 0:30 to 6:00 is kept at minimum. Based on operational statistics collected by Water Company there are no registered customer complaints due to low pressure at night. Current daily pressure control settings from 0:30 to 6:00 help to keep the leakage level low and already saves energy and therefore the Water Company has no intention to waste energy and boost the pressure over legally required pressure minimum.

Table 4-2 Old and new pressure control settings for pumps in "Punane"

<table>
<tr><th></th><th>Start time</th><th>End time</th><th>Pressure before (bar)</th><th>Pressure after (bar)</th></tr>
<tr><td rowspan="5">Working day</td><td>0:00</td><td>0:30</td><td>4.8</td><td>5.0</td></tr>
<tr><td>0:30</td><td>6:00</td><td>3.9</td><td>3.9</td></tr>
<tr><td>6:00</td><td>9:00</td><td>5.0</td><td rowspan="3">5.0</td></tr>
<tr><td>9:00</td><td>23:00</td><td>5.1</td></tr>
<tr><td>23:00</td><td>0:00</td><td>4.9</td></tr>
<tr><td rowspan="8">Saturday</td><td>0:00</td><td>0:30</td><td>4.9</td><td>5.0</td></tr>
<tr><td>0:30</td><td>6:00</td><td>3.9</td><td>3.9</td></tr>
<tr><td>6:00</td><td>8:00</td><td rowspan="2">4.7</td><td>4.9</td></tr>
<tr><td>8:00</td><td>9:00</td><td rowspan="2">5.1</td></tr>
<tr><td>9:00</td><td>12:00</td><td rowspan="2">5.2</td></tr>
<tr><td>12:00</td><td>22:00</td><td rowspan="3">5.0</td></tr>
<tr><td>22:00</td><td>23:00</td><td>5.1</td></tr>
<tr><td>23:00</td><td>0:00</td><td>4.8</td></tr>
<tr><td rowspan="8">Sunday</td><td>0:00</td><td>0:30</td><td>4.8</td><td>5.0</td></tr>
<tr><td>0:30</td><td>6:00</td><td>3.9</td><td>3.9</td></tr>
<tr><td>6:00</td><td>8:00</td><td rowspan="2">4.7</td><td>4.9</td></tr>
<tr><td>8:00</td><td>9:00</td><td rowspan="2">5.1</td></tr>
<tr><td>9:00</td><td>12:00</td><td rowspan="3">5.1</td></tr>
<tr><td>12:00</td><td>19:00</td><td>5.0</td></tr>
<tr><td>19:00</td><td>23:00</td><td>5.1</td></tr>
<tr><td>23:00</td><td>0:00</td><td>4.8</td><td>5.0</td></tr>
</table>

Because *SCADA* control system uses bars, the pressure setting in Table 4-2 are given as bars to avoid transformation of units. As previously described in chapter 2.2.1, the simplified pumping-station representations and controls were used in *WaterGEMS* to describe different pressure regimes. Therefore, the pressure settings and controls were manually lowered until minimal required pressure was still guaranteed at all critical points. As the pressure head varies at critical points all over the current pressure zone only ca 5m, the theoretical installation of pressure reduction valves will give only marginal effect.

4.3. Number of working pumps

Current chapter compares how many new proposed pumps would theoretically work at test week comparing to old pumps. As shown on the weekly pattern Figure 4-9, only one existing pump was always working. As at monitored week, there was no fire or major burst, so the second pump never turned on. Also further annual *SCADA* data analysis showed, that existing two or more pumps seldom need to run together.

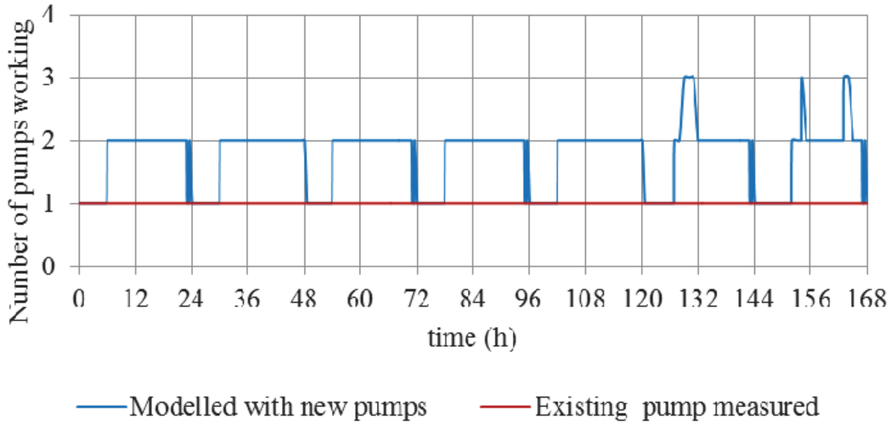

Figure 4-9 Weekly pattern of pumps working

By choosing new pumps with smaller rated flow as proposed in chapter 4.1, it is possible to achieve more flexible pump working regimes. It is now possible at night by low consumption to run only one smaller pump with considerably higher efficiency. In addition, the needed fire flow is also guaranteed by using four pumps.

4.4. Increasing the efficiencies

In this section, the pumps system different efficiencies are all analysed separately (pump, VFD, motor, total). That enables to understand better, the efficiencies of which components have the strongest effect on the overall efficiency of a pump system. It also gives opportunity to analyse the differences between old and new proposed pumping system. Efficiency values of the existing pumps and the new proposed pumps are modelled using *WaterGEMS.* Results are then compared with the calculated values that are based on real life measurements. Summarized results are presented as weekly patterns in Figure 4-10, Figure 4-11 Figure 4-12. Results show that the difference between calculated values and modelled values with existing pump is minor. This proves, that created *WaterGEMS* model reflects the real life situation on with sufficient accuracy. It is clearly visible from weekly patterns that the pump efficiency of new proposed pumps is much higher than with old pumps, staying for the majority of time above 80% and slightly drops only at night. The strongest efficiency increase is achieved at night hours when the old pumps efficiency dropped near 30% before, but with the new pumps work at ca 70% efficiency. This provides a great potential for energy saving and estimated benefit will be covered afterwards.

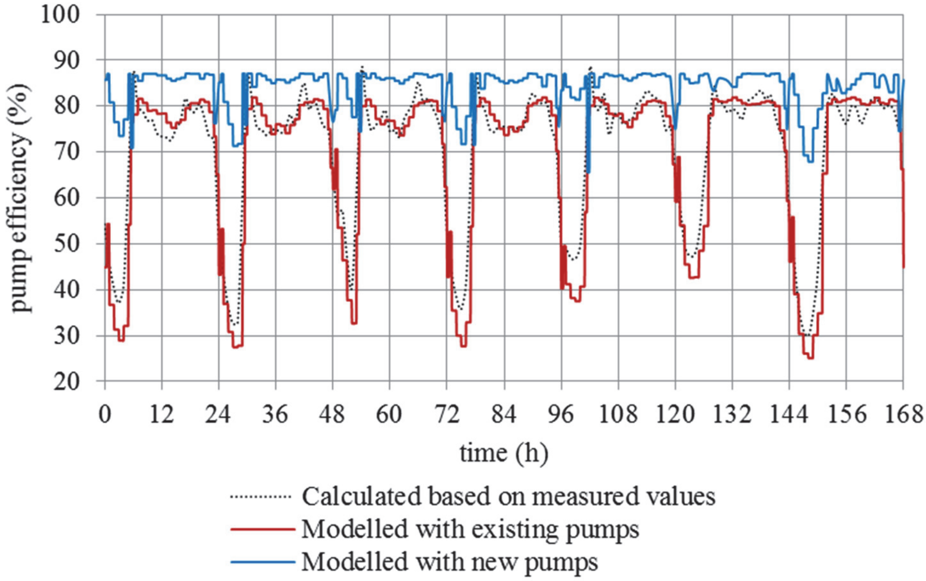

Figure 4-10 Comparison of pump efficiencies

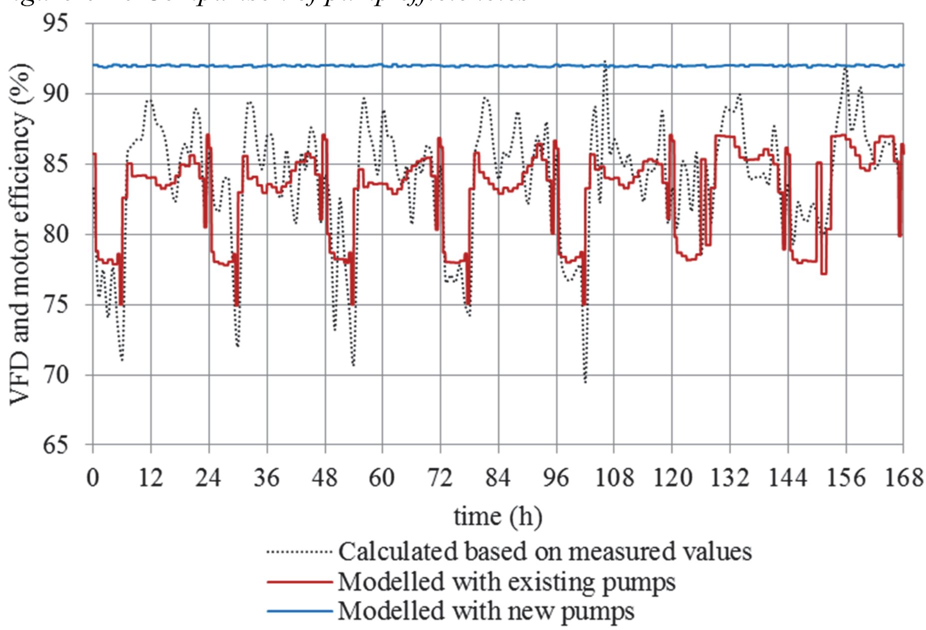

Figure 4-11 Comparison of VFD and motor efficiencies

Figure 4-11 shows that because existing pump is oversized the VFD and motor work also not in not optimal range that causes lower efficiency. As the motors of the new proposed pumps are smaller and more suitable for current working conditions, the motors work inside optimal rotational speed limit when motor and VFD efficiency is stable. This also causes higher and more stable overall efficiency (wire-to-water efficiency) of the pump system.

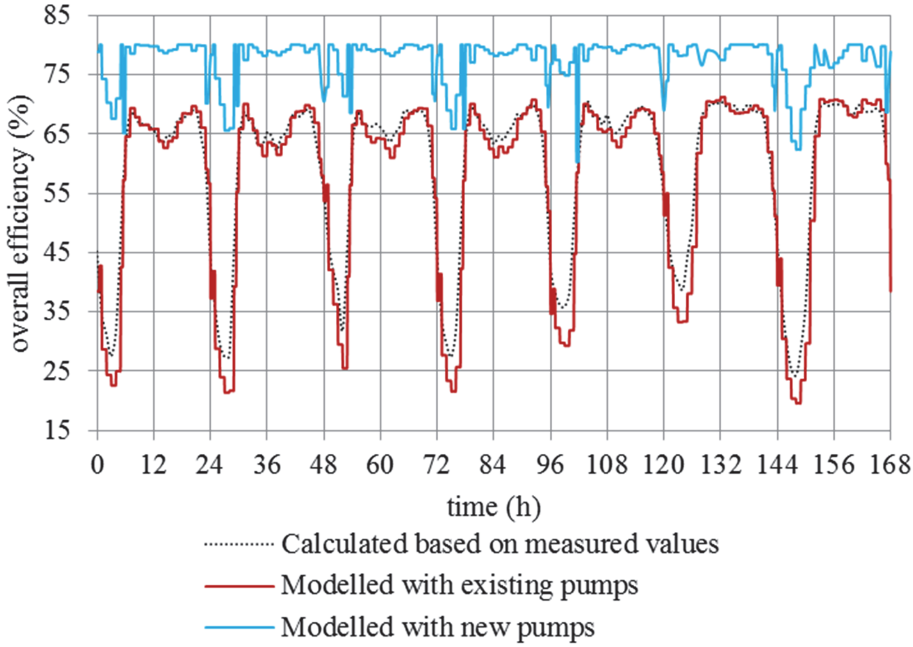

Figure 4-12 Comparison of overall efficiencies of existing and new pump systems

As presented on Figure 4-12 there is a very good correlation between calculated and modelled overall efficiency of the existing pump system.

4.5. Pump rotational speeds

To analyse and compare fluctuations in pumps rotational speeds with old and proposed pumps, again modelling task was carried out for both pumps as described previously in chapter 3.4. The results are visualized on following Figure 4-13. When more than two new pumps are working, it is considered that they run at the same rotational speed.

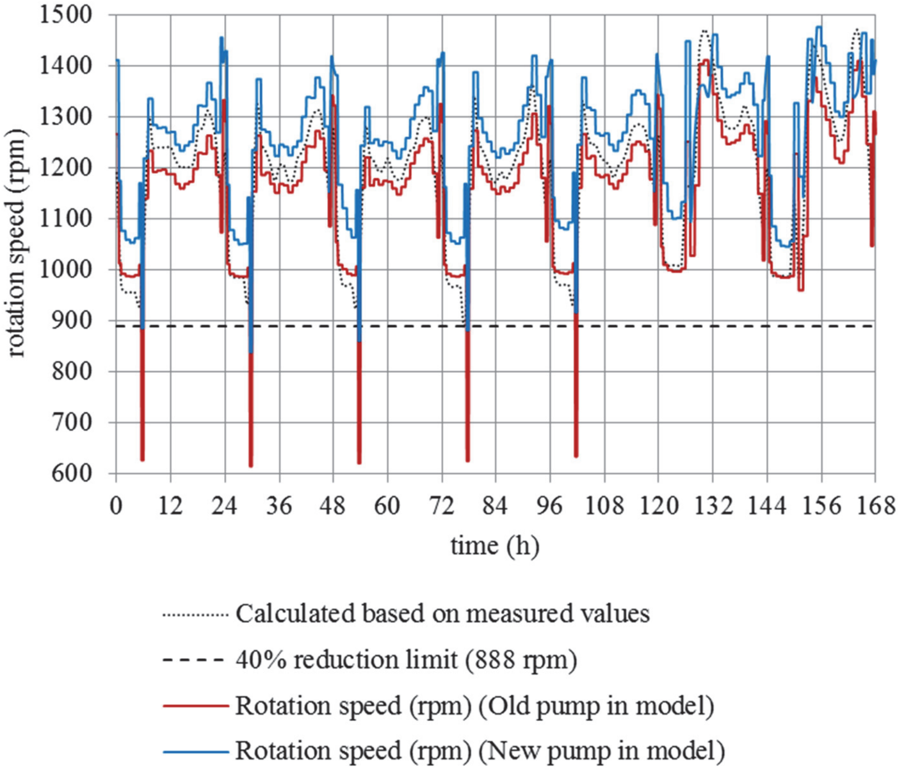

Figure 4-13 Comparison of pump rotational speeds

Big spikes in modelled rotational speeds are not data errors but are caused by a deviation in the water treatment plant and in the pressure regimes of the current 2nd stage booster pumping station. Outgoing pressure in the pumping station is defined to drop ca 30 minutes before the water treatment plant.

4.6. Electrical power usage

To compare modelling results of the electrical power used by the old pump and the new proposed pump the Figure 4-14 was generated to illustrate hourly changes. To simplify the graph, both the old and proposed new pumps are visualized using current pressure regime. Weekly results also reflect the optimization effect of pressure regimes mentioned in chapter 4.2. Final summarized result is given as Table 4-3. This gives a better understanding and further basis for payback time analysis. Generally, ca 25% reduction in energy usage can be achieved with smaller pumps and pressure regimes optimization.

Table 4-3 Comparison of electrical power used by various scenarios

Description	**weekly, kWh**	**annual, kWh**
Old pumps, old pressure regimes	9 312	484 224
New proposed pumps, old pressure regimes	7 138	371 176
New proposed pumps, optimized pressure regimes	6 935	360 620
Estimated saving	2 377	123 604

Booster pumping station provides water to Lasnamäe city distinct where monthly average consumption is almost stable. Therefore, the average annual consumption can be estimated based on measured week data with sufficient accuracy.

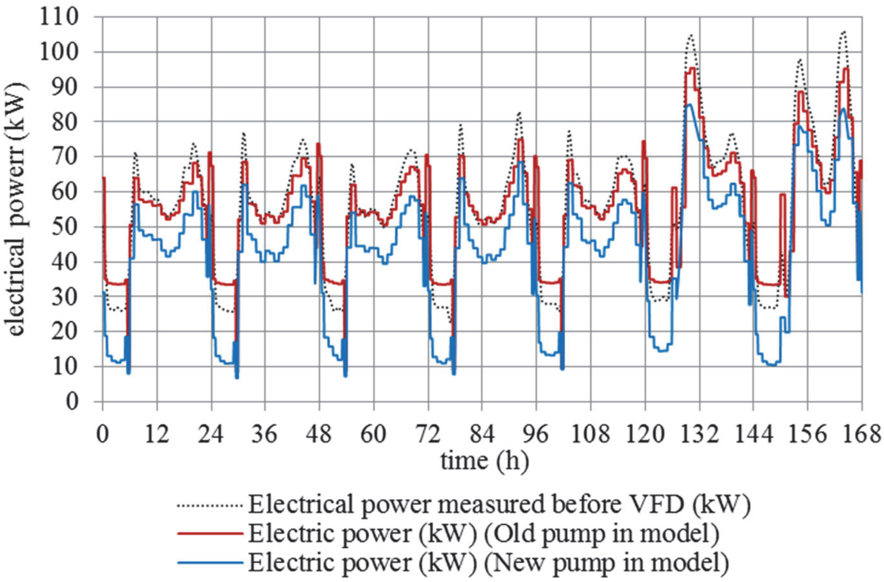

Figure 4-14 Comparison of weekly electrical power usage trends

Figure 4-14 demonstrates a reasonably good correlation between measured and modelled hourly values of electrical power used by the current pumps. This assures that the *WaterGEMS* model is working properly and reflecting reasonably well different scenarios in WDS and pumping station.

Based on IEA Statistics (2013) the average annual use of electricity in Estonia in 2011 was 3465 kWh per household. The estimated annual saving that can be achieved through pumping optimization in pumping station "Punane" alone is for example equal to annual electricity consumption of ca 35 households.

4.6.1. Reduction of electricity cost

The main interest of every Water Company, after providing customers with excellent level of service, is usually to optimize its operational and production costs. As pumping in WDS of Tallinn consumes ca 8 % of the total annual electricity consumption of a Water Company, the priority of optimization is high. To be more focused, research related to current thesis deals only with pressure boosting stations. Pumping station "Punane" is chosen as an example to give an overview how to carry out the efficiency analysis for other pumping stations and how to estimate theoretically achievable savings. As research bases on 2013 consumption data and electricity unit price, the actual benefit assumption may vary in the future. Therefore, the all process gives general idea what the rough saving may be.

Following Figure 4-15 gives cumulative overview how the test week cost of electricity builds up.

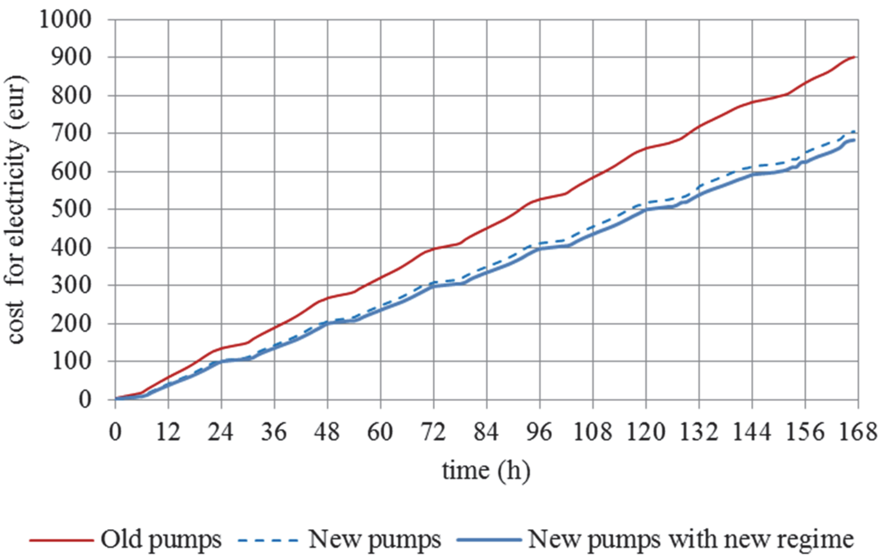

Figure 4-15 Cumulative weekly cost of used electricity

To give concentrated input for further cost saving and payback analysis, the following Table 4-4 gives an overview of weekly and estimated annual electricity cost.

Table 4-4 Comparison of electricity cost by various scenarios

Description	**Weekly (eur+ VAT)**	**Annually (eur+ VAT)**
Old pumps, old pressure regimes	901	46 839
New proposed pumps, old pressure regimes	704	36 591
New proposed pumps, optimized pressure regimes	683	35 499
Estimated saving	218	11 340

As visualized on the Figure 4-15 and summarized in Table 4-4, a considerable saving in electricity cost (ca 21.9%) can be achieved by replacing existing oversized pumps without changing the old pressure regimes of pumps. After pressure regimes optimization with *WaterGEMS* software based model, the weekly cost of pumping can be reduced even further, which enables to achieve an estimated saving of ca 24.4%. As modelling shows, the current pressure regimes of pumps were near optimum because the Water Company has constantly tested and adjusted it in everyday operation for many decades.

4.7. Reduction of CO_2 emissions

Today the majority of electricity in Estonia is produced by burning the fossil fuel - oil shale. According to IEA Statistics (2013) the CO_2 emissions per kWh from electricity generation in Estonia in 2011 was 1086 g of CO_2 per kWh, which is the highest value in European Union states and ca two times higher than world's average. Based on that statistics it is very important for the environment that the usage of energy, especially fossil fuel based energy, reduces. Estonia has made great efforts together with other Baltic States and Finland, to establish alternative power connections and to get access to Nordic hydro energy that is more environmental friendly. Since 2006, Estlink 1 was opened with 350 MW power but it does not cover all needs so oil shale based electricity production continued. In 2014, the Estlink 2 connection was opened with an extra 1000 MW that ensures working electricity market between Baltics and Nordic states. Therefore, there are possibilities to reduce the proportion of oil shale based electricity usage. Next analysis focuses on pumping optimization and effect.

As visualized on following Figure 4-16 and summarized in Table 4-5, considerable saving in CO_2 emission (ca 22.8%) can be achieved in pumping station "Punane" by replacing the existing oversized pumps and using the old pressure regimes. After replacing the pumps and optimization of pressure regimes, the weekly CO_2 emission can be reduced by ca 25.7 %.

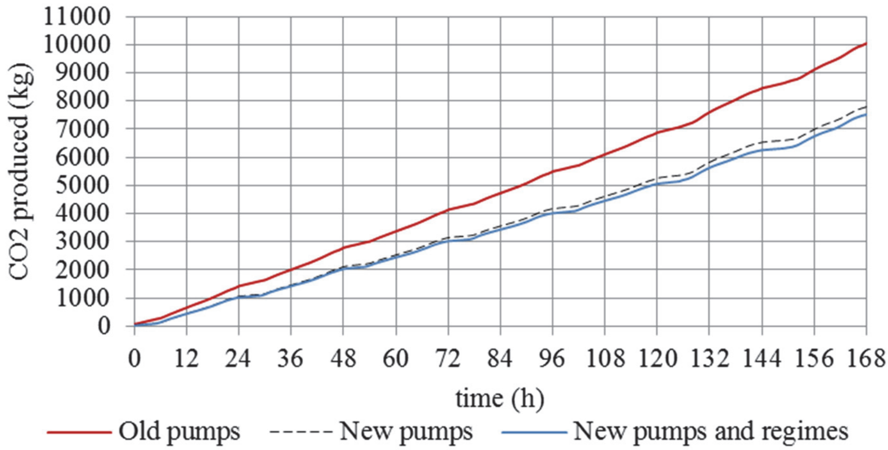

Figure 4-16 Cumulative weekly CO_2 production

Table 4-5 Comparison of cumulative CO_2 emission by various scenarios

Description	**Weekly (tons of CO_2)**	**Annually (tons of CO_2)**
Old pumps, old pressure regimes	10.1	525.2
New proposed pumps, old pressure regimes	7.8	405.6
New proposed pumps, optimized pressure regimes	7.5	390
Estimated saving	2.6	135.2

For example, to realize the environmental effect of optimization in pumping station “Punane”, as well as to compare the annual saving in CO_2 emission, a small car Toyota Yaris (gasoline engine, 123 g CO_2 per km) produces theoretically the same amount of CO_2 as having driven 27 times around the world’s equator. That is only one booster pumping station. There is a huge potential in reducing the CO_2 emissions only by optimizing all pressure booster stations.

4.8. Calculation of LCC and investment payback time

The first thing for every company is to be sure, that the planned investment has a positive payback and financial benefit can be achieved with reasonable time. Thereby even a rough business case and Life Cycle Cost (LCC) analysis is always necessary. LCC analysis comparing the alternatives will identify the lowest cost solution. Pumping systems often a have lifespan of 15 to 20 years. In most cases lifetime energy and maintenance cost will dominate over the initial purchase and installation cost. There are many guides how to determine and compare LCC of different scenarios, but the basic principle is the same. Example HI/Europump Guide (HI 2001) defines LLC calculation equation as Eq. (25).

$$LCC = C_{ic} + C_{in} + C_e + C_0 + C_m + C_s + C_{env} + C_d \tag{25}$$

where

C_{ic} = initial cost, purchase price (pump, system, pipe, auxiliary)
C_{in} = installation and commissioning
C_e = energy costs
C_o = operating cost (labour cost of normal system supervision)
C_m = maintenance cost (parts, man-hours)
C_s = down time, loss of production
C_{env} = environmental costs
C_d = decommissioning

LCC analysis is carried out with 15 years lifespan. The calculation of the cost of energy usage is based on the modelling results and detailed analysis described in chapter 3.12 and 4.6. Results of the LCC costs calculations for existing and new proposed pumps are summarized in the following Table 4-6.

Table 4-6 Comparison of existing and new pump system 15 years of LCC

Life Cycle costs	**4 new smaller pumps (Eur + VAT)**	**4 existing oversized pumps (Eur + VAT)**
Initial purchase cost	32 000 €	0 €
Installation cost	30 000 €	0 €
Energy cost	541 920 €	714 419 €
Maintenance costs	91 592 €	120 390 €
Total:	**695 513 €**	**834 809 €**

The estimated annual cost for energy usage of the existing pumps with maintenance cost for 2015 is ca 50 585 euros + VAT and it will increase ca 2.5% per annum because of the inflation. The maintenance cost of existing pumps is expected to increase even more because old pumps will need more maintenance every year.

At the same time new smaller Grundfos 98274752 pumps work most of the time near BEP, therefore they consume less energy and require less maintenance. The expected annual cost for energy usage of new pumps together with maintenance cost for 2015 is ca 41 604 euros + VAT that is ca 18% less than with old pumps.

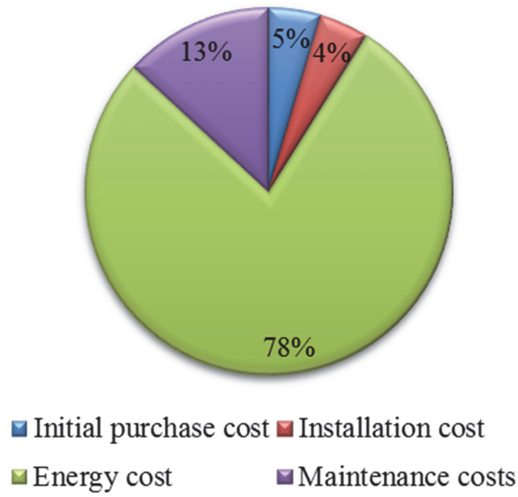

Figure 4-17 Proportions in the LCC analysis components based on new smaller Grundfos 98274752 pumps

As visualized on the Figure 4-17, the majority of LCC cost is coming from energy usage costs and therefore optimisation of energy usage it the main priority. Due to the replacing of pumps and optimization of pressure regimes, the annual saving in electricity and maintenance cost can be achieved. Annual saving together with comparison of remaining investment into new smaller pumps is given in Table 4-7.

Table 4-7 Summary of payback calculation

Year	Operational cost		Annual saving	Remaining investment cost
	Existing pumps	New pumps		
2015	50 585 €	41 500 €	9 085 €	-52 915 €
2016	50 711 €	41 604 €	9 108 €	-43 807 €
2017	50 914 €	41 708 €	9 207 €	-34 601 €
2018	51 197 €	41 812 €	9 385 €	-25 216 €
2019	51 569 €	41 917 €	9 652 €	-15 564 €
2020	52 042 €	42 021 €	10 021 €	-5 543 €
2021	52 639 €	42 126 €	10 512 €	4 970 €

Payback time is calculated by comparing annual saving with initial investment. Results are visualized in Figure 4-18.

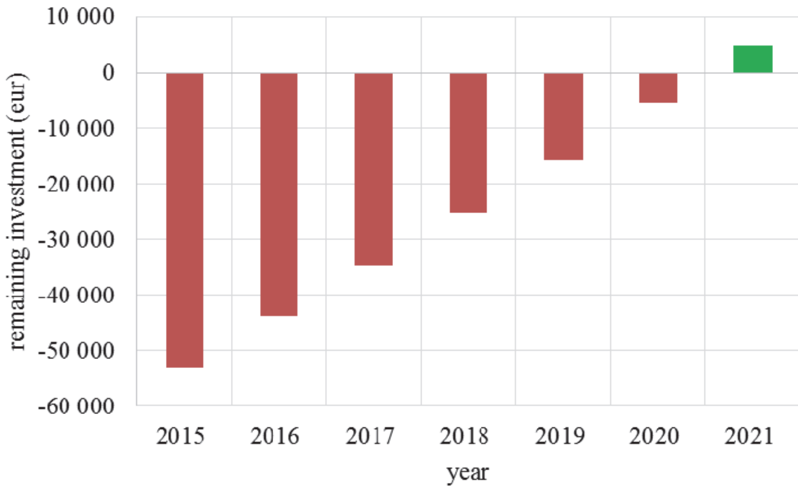

Figure 4-18 Visualization of new smaller pumps payback analysis results

As shown on Figure 4-18, the estimated payback time to cover initial investments for replacement of oversized pumps is ca 6.5 years that is considered acceptable in most utility companies.

4.9. Summary of using the good engineering practices

Chapter 4 introduced the good engineering practices how to make pumping station "Punane" (that was previously analysed in the chapter 3) more energy efficient. Proposal for replacement of oversized pumps is made together with changing of pumps pressure control settings. Benefits are defined together with expected investment payback time. Using good engineering, ca 25.5 % lower energy usage has been achieved providing ca 25.5 % less CO_2 emissions and ca 21.9 % lower electricity cost. Analysis shows, that investment payback time for the replacement of oversized pumps is ca 6.5 years.

Next chapter introduces an algorithm, which can be used to optimize pump system work and keep identical centrifugal pumps working at their highest efficiency. Based on the proposed new algorithm the optimal number of running pumps can be calculated in live operation based on flow and pressure head needed in WDS. Two numerical case studies have been carried out.

5. OPTIMIZATION OF PUMP SYSTEM WITH IDENTICAL PUMPS

This chapter concentrates on identical pumps work related optimization. An algorithm is proposed, that enables to keep identical variable speed pumps (VSP) working close to the best efficiency point (BEP) given by pump manufacturer.

It is common in operation that pumps controls are defined so, that extra pump will be switched on only when working pump(s) cannot provide necessary flow at predefined head. In some other cases, flow is fixed with one numerical value when extra pump will be started. Both ways are easy to program but do not ensure maximal pump efficiencies.

Proposed algorithm enables to calculate out (based on efficiency graph, actual flow and pressure head required by the WDS) switching points for the pumps when is more efficient to use an extra pump. The efficiency graph can include only pump efficiency or pump+motor, or pump+motor+VFD efficiency.

Using proposed algorithm in PLC-s, the extra pump can be switched on/off based on flow and pressure head required by the WDS that ensures higher efficiency. Figure 5-1 illustrates the principle and pumps switching point based on new proposed Grundfos 98274752 pump efficiency graph at H=16m.

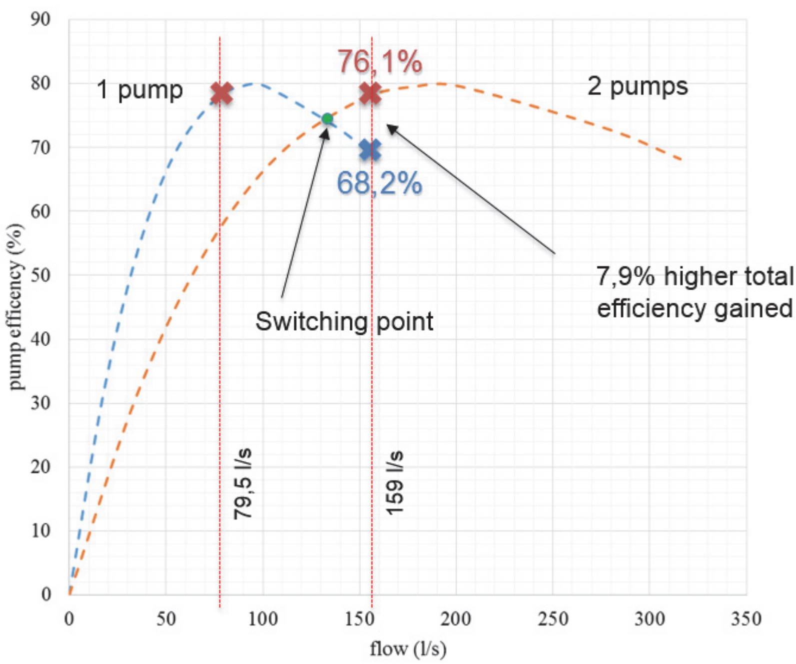

Figure 5-1 Idea of using proposed algorithm with new Grundfos 98274752 centrifugal pump at 16m.

As shown on Figure 5-1, the higher pumps efficiency can be achieved when WDS requires an example 159 l/s total flow and 16m pressure head. Using two pumps (with same rotational speed) gives higher total pump efficiency even when flow and pressure head can be provided by one pump.

In general, the proposed algorithm for PLC-s bases on polynomial approximation of efficiency surface. It does not matter in this case what kind of efficiency surface is used. The efficiency surface can include only pump efficiency or pump+motor, or pump+motor+VFD efficiency.

It is known, that sometimes the polynomial type approximation is less accurate to describe efficiency surface at all possible pressure heads and flows (even with identical pumps) but reasonably good fit can be achieved near maximal efficiency (rated flow and head) as shown later in case study. This enables to derive equations for programmable logic controllers (PLC-s) that control the pumps. As optimal number of identical pumps is determined and switched on based on currently needed flow and pressure head, the VFD-s will automatically adjust the rpm of all pumps so, that flow and pressure head required by the WDS is provided.

The pumps switching points can be calculated when pump efficiency, using extra pump, is higher than with currently working number of pump(s). That means combination of actual flow, actual pressure head and number of pumps *m* is most efficient.

The usage of the proposed algorithm is illustrated by two case studies based on an existing pumping station "Punane". More detailed overview of proposed algorithm is given in Koor et al. (2014).

As pump performance changes during operational years, so the calculated switching points for pumps have to be recalculated after some period. Technically, it means only correcting the polynomial coefficients in PLC equations and the algorithm will work based on new determined pump efficiency. Problem is that calculating out new polynomial coefficients will require special testing of the pump to determine actual pump efficiency.

5.1. Description of the optimization task

The task is to optimize the work of a pumping station equipped with identical centrifugal pumps running in parallel mode. It is necessary to estimate the parameters derived from the network characteristics and the working condition when electrical power consumption is minimal. Let us assume that number of pumps in the pumping station is *m,* all the pumps can be switched on and off and their revolutions per minute (rpm) are adjustable. In such case, a complex optimization task should be resolved with the objective function Eq. (26), which minimizes energy consumed by the station for pumping:

$$E = \sum_{i=1}^{m} \varepsilon_i P_i\left(Q_i, N_i\right) \tag{26}$$

where relative rotational speed of the i^{th} pump N_i and can be described as Eq. (27).

$$N_i = \frac{N_i^*}{N_{i0}} \tag{27}$$

where

N_i^* – new rotational speed of the i^{th} pump
N_{i0} – nominal rotational speed of the i^{th} pump

Constant $\varepsilon_i = 1$ if the i^{th} pump is operational, and $\varepsilon_i = 0$ if the pump is not operational. It is necessary to find such values of ε_i and N_i where the value of the objective function Eq. (26) is minimal. If the following conditions have been met, condition derived from the network characteristic can be calculated with following function Eq. (28):

$$H = H(Q) \tag{28}$$

where flow can be expressed as function Eq. (29).

$$Q = \sum_{i=1}^{m} \varepsilon_i Q_i \tag{29}$$

Conditions derived from the pump pressure characteristics can be described as Eq. (30).

$$H_i = N_i^2 H_{i0}\left(\frac{Q_i}{N_i}\right) \tag{30}$$

Usually the pumping head H_i at relative rotational speed of the i^{th} pump is expressed as Eq. (31)

$$H_i = a_{i1} + a_{i2}Q_i + a_{i3}Q_i^2 \tag{31}$$

in which case H_i can be defined as Eq. (32).

$$H_i = a_{i1}N_i^2 + a_{i2}N_iQ_i + a_{i3}Q_i^2 \tag{32}$$

Conditions derived from the performance characteristic of the pumps are described as follows. Effective power of i^{th} pump $P_{ef,i}$ at relative rotational speed N_i is calculated as Eq. (33).

$$P_{ef,i} = N_i^3 P_{ef,i0}\left(\frac{Q_i}{N_i}\right) \tag{33}$$

Effective power of i^{th} pump $P_{ef,i0}$ at rated rotational speed N_{i0} can be expressed as Eq. (34).

$$P_{ef,i0} = b_{i1}Q_i + b_{i2}Q_i^2 + b_{i3}Q_i^3 \tag{34}$$

So the Eq. (33) takes following shape:

$$P_{ef,i} = b_{i1}N_i^2Q_i + b_{i2}N_i\,Q_i^2 + b_{i3}Q_i^3 \tag{35}$$

Relationship between the effective power $P_{ef,i}$ of i^{th} pump at relative rotational speed N_i and the electrical power P_i consumed by the i^{th} pump can be expressed by the following Eq. (36):

$$P_i = \frac{P_{ef,i}}{\eta_i} \tag{36}$$

Restrictions of the pump's rotational speed can be defined as:

$$N_{i\,\min} \le N_i \le N_{i\,\max} \tag{37}$$

The reason for the latter is the need to avoid motor overloads. The optimization task will be resolved in two stages. First, it is necessary to optimize the pumping station operations with a constant number of pumps. Then, the optimal number of pumps will be established.

5.2. Optimization with a constant number of pumps

Mathematically, the problem set leads to solving of a non-linear planning task. It involves finding non-negative values of f_i, which satisfy the equalities and inequalities given as Eq. (28), (29), (35), (36), (38) and provide the minimum value of the objective function Eq. (26) at these following conditions. Let us assume that there are *m* identical pumps at the pumping station. In such case ε_i=1 and *i* =1,2...m. The objective function Eq. (26) takes the form:

$$E = \sum_{i=1}^{m} P_i\left(Q_i, N_i\right) \tag{38}$$

If the pumps are identical, and their pressure characteristic is described as Eq. (39),

$$H_0 = H_0(Q_0) \tag{39}$$

Where

Q_0 – nominal (rated) flow and H_0 – nominal (rated) head

the pumping head H_i and effective power $P_{ef,i}$ functions take the following shapes:

$$H_i = N_i^2 H_0\left(Q_0\right);\ P_{ef,i} = N_i^3 P_{ef,i0}\left(Q_0\right) \tag{40}$$

With the given pumping head *H*, the efficiency of the pump $\eta_{p,i}$ can be expressed as

$$\eta_i = f(Q_i) = f\left(Q_0 \frac{N_i}{N_0} \right) \tag{41}$$

Therefore, in case of identical pumps and accordance with the Eq. (36), the electrical power consumed by the i^{th} pump P_i can be calculated as

$$P_i = \frac{\rho g Q_i H_0}{\eta_i} \tag{42}$$

and i^{th} pump flow at relative rotational speed Q_i can be defined as

$$Q_i = N_i Q_0 \tag{43}$$

Taking the Eq. (29), we will now find that

$$Q = \sum_{i=1}^{m} N_i Q_0 \tag{44}$$

Let us find the objective function Eq. (38) minimum using the additional condition described with Eq. (44). To solve the task the Lagrange function is used, which, using Eq. (42) and Eq. (44), can be written as following Eq. (45).

$$L = \sum_{i=1}^{m} \frac{Q_0 N_i H_0}{N_0 \eta_i} + \lambda\left(Q_0 - \sum_{i=1}^{m} N_i Q_0 \right) \tag{45}$$

Equating the partial derivatives of the Lagrange function to zero

$$\frac{\partial L}{\partial N_i} = 0 \quad ; i= 1,2,\ldots m \tag{46}$$

the equations for establishing the N_i-s take the following shapes:

$$\frac{Q_0 H_0 - N_0 \frac{\partial \eta_i}{\partial N_i}}{N_0 \eta_i^2} - \lambda Q_0 = 0 \quad ; i= 1,2,\ldots m \tag{47}$$

$$Q - Q_0 \sum_{i=1}^{m} N_i = 0 \tag{48}$$

This is a system of m+1 nonlinear equations that can be solved numerically.

System has a solution, if we take $N_i = N$ for each *i*.

Eq (48) will have the form of

$$Q - Q_0 m \cdot N = 0 \tag{49}$$

and can be used for finding *f* values as:

$$N = \frac{Q}{mQ_0} \tag{50}$$

From here, we can calculate our Lagrange multiplier. However, this solution may not be unique. Therefore, we tried to use real pump characteristics given on Figure 5-3 to find the efficiency of three pumps working with different frequencies at H_0 equal to 27 m. Let us suppose that a WDS needs 1200 l/s and we use random flows for two pumps. The third pump flow is calculated according to Eq. (48) as a difference between 1200 l/s and the sum of previous two pumps. The efficiency of each pump at every flow between 380 l/s and 410 l/s and power consumption according to flow is derived from Figure 5-3. The Figure 5-2 visualizes the surface that describes the total power consumed by pump system. It has one minimum only and it is at point 400,400,400 where all pumps run at same rotational speed.

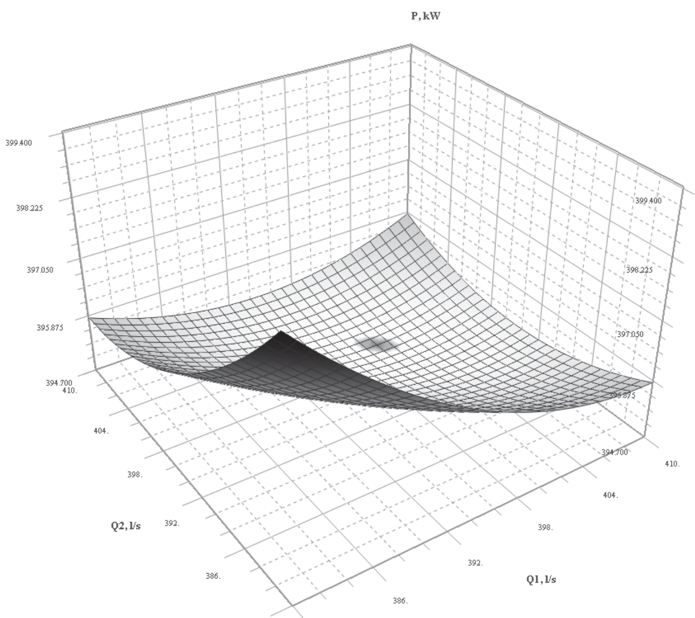

Figure 5-2 Summarized power consumption of pumps at different flows

Therefore, with the constant number of pumps in the pumping station, the most optimal solution is to keep the pumps running with the same rotational speed.

5.3. Establishing the optimal switching points of pumps

Let us assume that the pumping station is equipped with identical pumps that run at identical rpm. The optimal number of running pumps for the given pumping head H and the flow rate Q can be calculated as follows. Let the efficiency of a single pump be

$$\eta_p = \eta_p(Q,H) \tag{51}$$

in which case the efficiency of each m identical pump is

$$\eta_{p,m} = \eta_{p,m}\left(\frac{Q}{m}, H\right) \tag{52}$$

Accordingly, the overall electrical power consumed by the pumps can be described as Eq. (53).

$$P = \frac{\rho g Q H}{\eta_{p,m}\left(\frac{Q}{m}, H\right)} \tag{53}$$

In order for the electrical power to be minimal, the pump efficiency $\eta_{p,m}$ must be maximal. Consequently, it is necessary to find $\eta_{p,m}$ for all m-s, and to establish at which m the $\eta_{p,m}$ will obtain its highest value. The respective m will give us the optimal number of pumps. Let us assume that the number of pumps m is a constantly changing value, which is found from the condition of the maximum of the function Eq. (54).

$$\frac{d\eta_{p,m}\left(\frac{Q}{m}, H\right)}{dm} = 0 \tag{54}$$

Let the efficiency surface of the pump be described as

$$\eta_p(Q,H) = c_0 + c_1 Q + c_2 Q^2 + c_3 H + c_4 QH + c_5 H^2 + c_6 Q^2 H + c_7 QH^2 \tag{55}$$

where c_0 –c_7 are coefficients found with the non-linear least squares method from Figure 5-3. In order to obtain number of pumps m, the Eq. (55) can be rewritten as

$$\eta_{p,m} = c_0 + c_1\frac{Q}{m} + c_2\frac{Q^2}{m^2} + c_3 H + c_4\frac{Q}{m}H + c_5 H^2 + c_6\frac{Q^2}{m^2}H + c_7\frac{Q}{m}H^2 \tag{56}$$

Replacing m by x and taking into account Eq. (54) we will now find that the optimal number of pumps must be an integer around the value x that can be calculated by the following Eq. (57).

$$x = -\frac{(2c_2 + 2c_6 H)Q}{(c_1 + c_4 H + c_7 H^2)} \tag{57}$$

Based on the optimal number of pumps, the *(Q, H)* area can be divided into subareas. Boundaries separating subareas can be found by using the following equalities:

$$\eta\left(\frac{Q}{m-1}, H\right) = \eta\left(\frac{Q}{m}, H\right) \tag{58}$$

Using the efficiency given as Eq. (56) and the condition given in Eq. (58), the boundaries for Q when it is necessary to increase the number of running pumps from *n-1* to *n* can be calculated with following Eq. (59).

$$Q_{m-1,m} = -\frac{(c_1 + c_4 H + c_7 H^2)m(m-1)}{(c_2 + c_6 H)(2m-1)} \tag{59}$$

The approximation of efficiency surface Eq. (56) has been checked and it showed quite good accuracy near rated flow and head where efficiency is highest. It may be possible to obtain better results with the cubic approximation as described in Ulanicki et al. (2008) but in this research, only approximation Eq. (56) is used.

5.3.1. Case study based on the existing pumps

This case study is targeted to optimize the work of a pumping station equipped with four identical Ahlstrom APP 43–250 (300-250-315) centrifugal pumps with a closed impeller, running in parallel. The pressure and efficiency characteristics of pumps are presented in following Figure 5-3. The motor efficiency and VFD efficiency are not accounted in case of old pumps, due to lack of information.

Using the least squares method, the efficiency surface and polynomial coefficients are calculated based on data obtained from Figure 5-3 and using polynomial approximation Eq. (55). These coefficients are further used in Eq. (60). The fit between data used to determine polynomial coefficients (Figure 5-3) and Eq. (55) based approximation is presented Figure 5-4.

$$\begin{aligned}\eta_{p,m} &= 65.85 + 0.316\frac{Q}{m} - 0.00097\frac{Q^2}{m^2} - 2.224H + 0.0075\frac{Q}{m}H + 0.023H^2 \\ &+ 0.0000085\frac{Q^2}{m^2}H - 0.000115\frac{Q}{m}H^2\end{aligned} \tag{60}$$

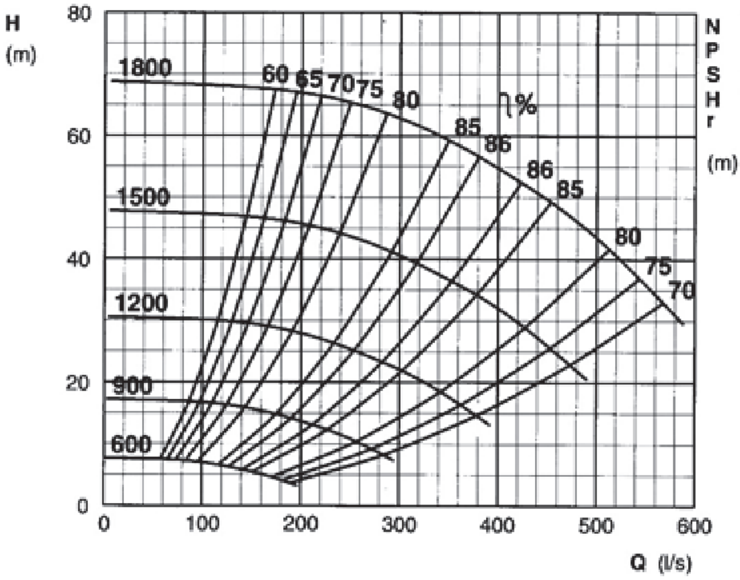

Figure 5-3 Characteristics of existing Ahlstrom APP 43–250 centrifugal pump

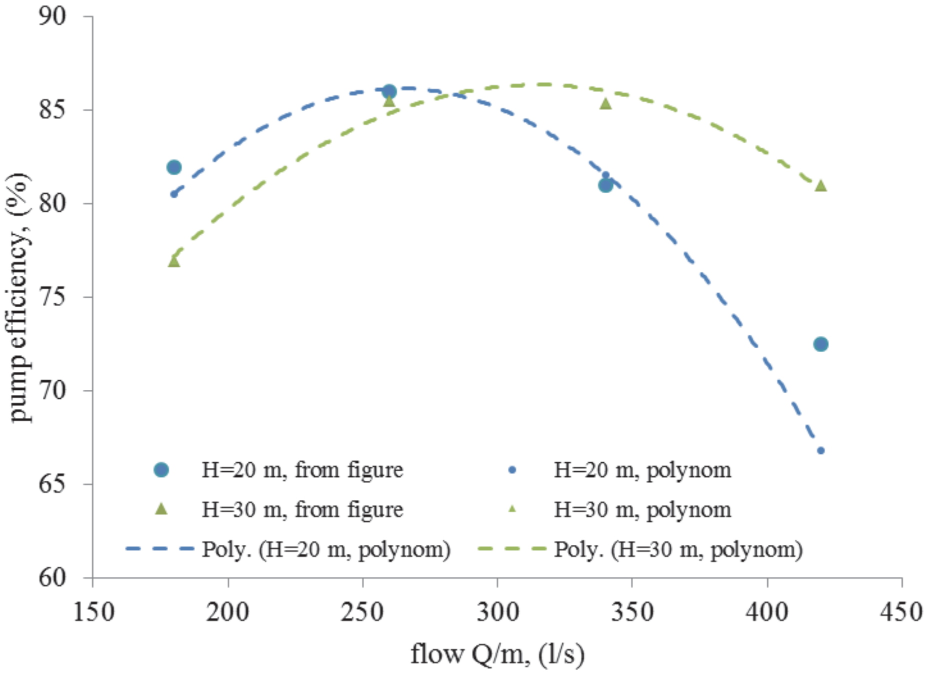

Figure 5-4. Fit between (Figure 5-3) and polynomial based approximation when m=1 using Ahlstrom APP 43–250 pump.

One can see from previous graph that fit of polynomial approximation is quite good in the area near maximal pump efficiency and rated H (15-30m).

Let us now use the efficiency expression given in Eq. (60) to establish the optimal number of pumps. Using Eq. (57), the value of x can be calculated using Eq. (61).

$$x = -\frac{Q \cdot (-0.00193 + 0.000017 \cdot H)}{0.3166 + 0.0075 \cdot H - 0.000115 \cdot H^2} \tag{61}$$

If x is an integer and $x = m$, then it ensures from the Eq. (61) that the pumps are running in the most optimal regime. The most efficient combination between Q and H with different m can be found using Eq. (62).

$$Q = -\frac{m \cdot (0.3166 + 0.0075 \cdot H - 0.000115 \cdot H^2)}{-0.00193 + 0.000017 \cdot H} \tag{62}$$

According to Eq. (59) and (60), the pump switching points with m-1 and m number of pumps can be calculated as

$$Q_{m-1,m} = -\frac{(0.3167 + 0.0075 \cdot H - 0.000115 \cdot H^2) \cdot m(m-1)}{(-0.00097 + 0.0000085 \cdot H)(2m-1)} \tag{63}$$

Eq. (62) and Eq. (63) allow mathematically calculate the switching point values based on any H, but the realistic results can be obtained when H is between 15-30 m. Figure 5-5 presents the results of calculations based on Eq. (62) and Eq. (63).

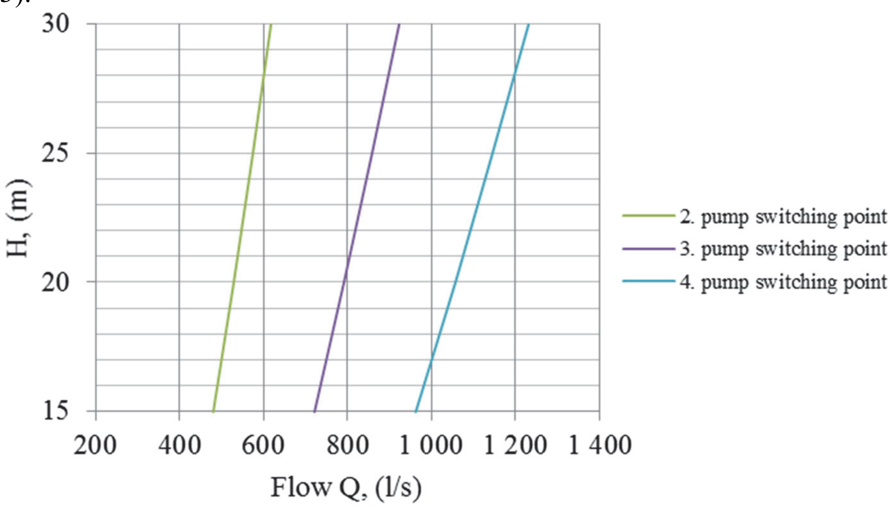

Figure 5-5. Optimal pump switching points for existing Ahlstrom APP 43–250 pumps

If to compare Figure 5-5 with weekly graph of discharge flow given on Figure 3-6, it is clearly visible that to achieve higher efficiency of the pumping station the pumps would have to be replaced by pumps with smaller nominal flow rate or water consumption should somehow be increased (example by expanding WDS service area).

5.3.2. Case study based on new proposed pumps

This case study is carried out to optimize the work of a pumping station equipped with four smaller identical Grundfos NK 150-315.1/315 centrifugal pumps. The new pump performance and efficiency characteristics are given on following Figure 4-1. The efficiency graph principle is not similar to graph for old pump (Figure 5-3) used in previous case study when we had possibility to obtain efficiency data in order to construct surface of efficiency η=f(Q,H).
In this case, we have to calculate data to construct efficiency surface using principle given in Pump Handbook (2004) and previously described Eq. (16).

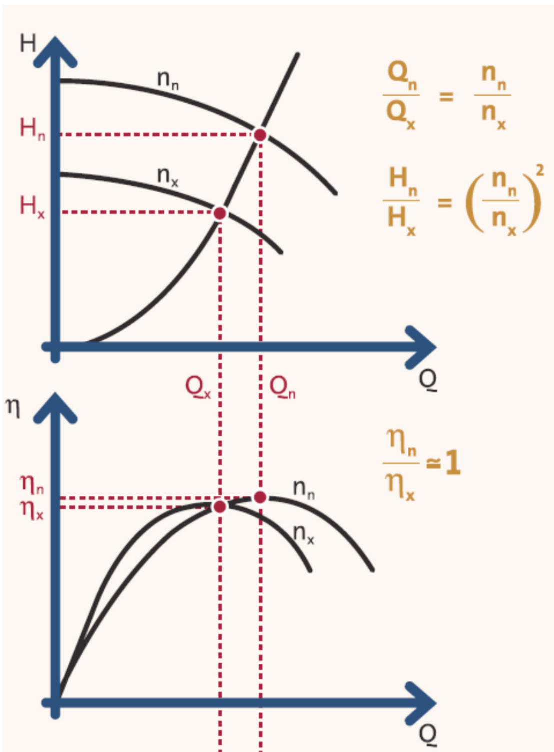

Figure 5-6 Pump efficiency calculation principle given by Grundfos [Adapted from Pump Handbook (2004)]

This enables us to construct the efficiency surface *η=f(Q,H)*. As pump and motor combined efficiency is given as a graph on Figure 4-1, then current case study takes into account also motor efficiency.

Using the least squares method, the efficiency surface of the new proposed pumps can be approximated using previously obtained data and approximation Eq. (55). Using the same coefficients, the Eq. (55) can be written for different number of pumps *m*.

$$\begin{aligned}\eta_m &= 32.61 + 1.589\frac{Q}{m} - 0.0130\frac{Q^2}{m^2} - 3.008H + 0.0266\frac{Q}{m}H + 0.0769H^2 \\ &+ 0.000296\frac{Q^2}{m^2}H - 0.00147\frac{Q}{m}H^2\end{aligned} \tag{64}$$

The approximated efficiency surface of the pumps Eq. (64) can be used now to calculate the optimal number of pumps according to needed pump head and flow. Based on Eq. (57) the value of x can be calculated using Eq. (65).

$$x = -\frac{Q \cdot (-0.02596 + 0.000593 \cdot H)}{1.5892 + 0.0266 \cdot H - 0.001469 \cdot H^2} \tag{65}$$

As described before, if *x* is an integer, *x* = *m*, then it ensures from the Eq. (65) that the pumps are running in the optimal regime. The most efficient combination between Q and H with different *m* can be calculated using Eq. (66).

$$Q = -\frac{m \cdot (1.5892 + 0.0266 \cdot H - 0.001469 \cdot H^2)}{-0.02596 + 0.000593 \cdot H} \tag{66}$$

According to Eq. (59) and Eq. (64), the boundaries of the (Q, H) areas with *m*-1 and *m* number of pumps can be calculated using Eq. (67).

$$Q_{m-1,m} = -\frac{(1.5892 + 0.0266 \cdot H - 0.001469 \cdot H^2)m(m-1)}{(-0.0130 + 0.000296 \cdot H)(2m-1)} \tag{67}$$

The following Figure 5-7 presents the results of calculations based on Eq. (66) and Eq. (67) and illustrates the optimal pumps switching points using new Grundfos NK 150-315.1/315 pumps as a graph.

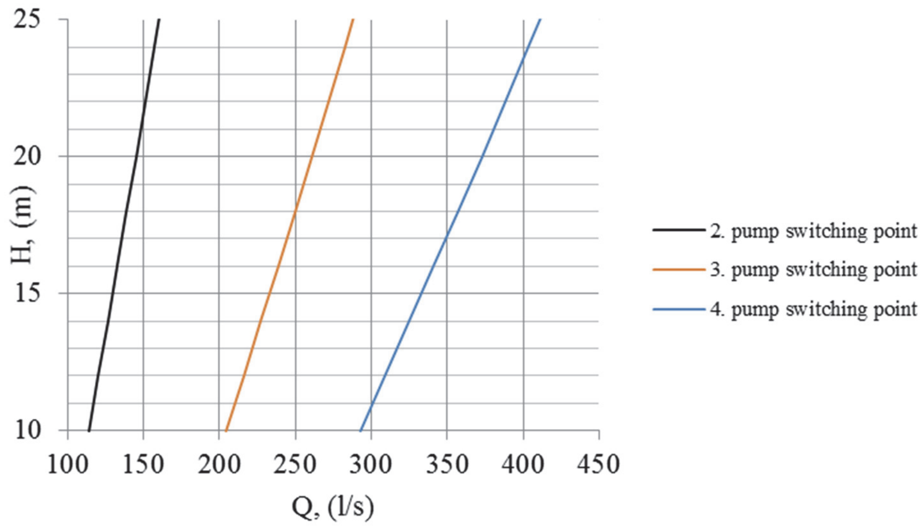

Figure 5-7 Optimal pump switching points for new Grundfos NK 150-315.1/315 pumps

Figure 5-7 gives a good understanding of optimal pump switching points – at what flow and pump head what number of pumps must be turned on to achieve maximum efficiency.

5.4. Calculation of optimal pump work areas based on pump switching points

To demonstrate and analyse effectiveness of proposed algorithm, the pumping station "Punane" is further investigated based on the test week real data. First, the graph Figure 5-7 describing the optimal pump switching points is improved by adding the pumps (Q,H) characteristic's for one, two and tree pumps working. That will set upper border of optimal work area because of pump performance limits. Now the actual flow and pump head measured at test week can be compared with calculated pump switching points, Results are given in the following Figure 5-8 were green dots are marking the average hourly (Q, H) values measured during the test week period.

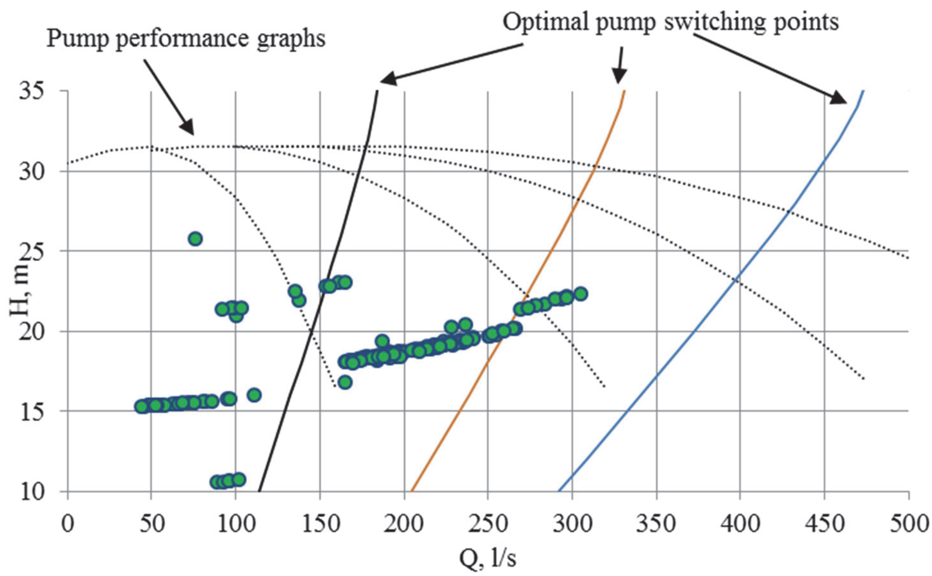

Figure 5-8 Test week Q, H values with new Grundfos NK 150-315.1/315 pump switching points

As described, if needed flow Q is less than pump switching point for two pumps working and the needed pump head H is below the maximal performance graph of one pump, then the maximal efficiency can be achieved using one pump. If needed flow Q exceeds the switching point for two pumps working, but remains below two pumps maximal performance graph, two pumps are most effective to run. It is the same with three and four pumps. Same principle applies with 3 and 4 pumps. The control logic can be programmed into pump controllers (PLC-s).

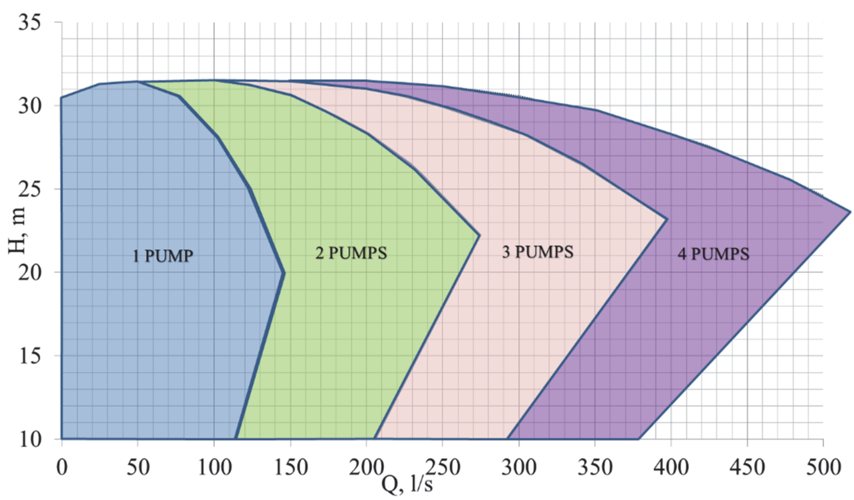

Figure 5-9 Optimal pump work areas for new Grundfos NK 150-315.1/315 pumps

Sometimes the efficiency surface approximation using polynom gives not so good fit comparing to raw data that is used to calculate polynomial coefficients as illustrated on Figure 5-10 as an example.

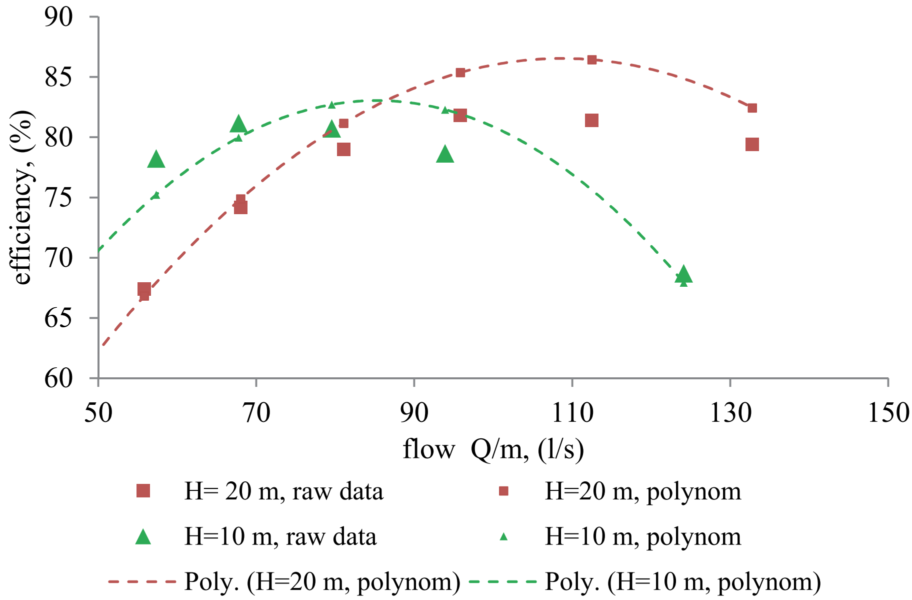

Figure 5-10 Fit between raw data and polynomial based approximation when m=1 using new Grundfos NK 150-315.1/315 pump.

Because of that, the additional approach is also proposed. Only difference is, that in case we not try to approximate all the efficiency surface, but only at set of given pressure heads (In example H = 6, 8, 10, 12, 14, 16, 18, 20 m). It gives us possibility to find optimal pump switching points for m pumps.
This approach is a little more complex but gives results that are more accurate if polynomial approximation of all efficiency surface is not too good. Following Figure 5-11 compares the Grundfos NK 150-315.1/315 pump switching points that are calculated based on all efficiency surface approximation (solid lines) and based on approximation carried out only at some given heads (dashed lines).

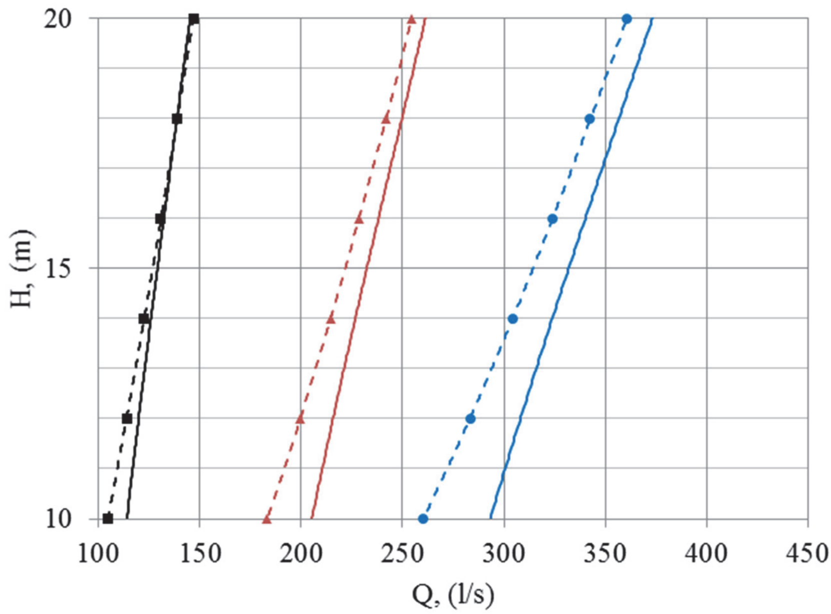

Figure 5-11 Pump switching points based on approximation of all the efficiency surface and only surface calculated based on given head values.

5.5. Summary of the proposed algorithm

In general, approximation of pumps efficiency characteristics by mathematical equations allow creating of analytical formulas and alternative ways for the selection of most efficient number of pumps to work based on needed flow and pressure head.

To achieve maximal pump efficiency at currently needed flow and pressure head, the optimal switching point of the pumps can be calculated live using proposed algorithm in programmable link controllers (PLC-s) that control pumps and additional pumps can be switched on or off as needed. To simplify the algorithm logic, the demands are taken as given and are not considered as pressure dependent.

Some investigation is carried out in MS Excel using help of ABB Estonia specialists, to test the algorithm logic and usage possibilities in PLC-s. Live input data from flow and pressure sensors were simulated. Results show, that common PLC-s are capable to coop with algorithm logic and it will not require any special type PLC.

If analysis show, that if polynomial approximation of efficiency surface will not give a good fit, then second proposal can be used.

6. OPTIMIZATION OF PUMP SYSTEM WITH DIFFERENT PUMP CHARACTERISTICS

If the previous chapter was based on the use of identical pumps, the current chapter focusses on the optimization of total efficiency of a pump system, in which the pumps can have slightly or large differences in performance parameters – a situation similar to that common in WDS-s. Therefore, three theoretical scenarios were investigated.

Let us assume that the number of pumps in the pumping station is *n*; all the pumps can be switched on and off and their revolutions per minute (rpm) are adjustable. In such cases, a complex optimization task must be solved with the objective function of maximizing pump system total efficiency that will eventually minimize energy usage.

Initially, the use of several identical pumps with the same performance characteristics was considered in scenario 1. Then in scenario 2, several pumps with slightly different characteristics were examined. That describes situation, when characteristics of initially identical pumps mentioned in scenario 1 will change during operation because of wear. Pumps working more often show faster degrading rate in terms of their efficiency and so they form a slightly different pumping system compared to new identical pumps. Therefore, one or more pumps were replaced with pump of slightly different characteristics. Finally, a scenario 3 with quite large differences in pump parameters was analysed. For example, because the pump to replace the broken one was out of stock.

6.1. Scenario 1 – identical pumps

First, we consider the new pump with the characteristics presented in Figure 5-3. For example, a WDS needs Q =1100 l/s at the pump head H=25m. Our task is to find the combination of discharges at which the pumps work with maximal total efficiency. In case of m pumps, the total efficiency can be formulated as Eq. (68).

$$\eta_{tot} = \frac{\rho \cdot g \cdot H \cdot Q}{\rho \cdot g \cdot H \cdot \left(\sum_{i=1}^{m} \frac{Q_i}{\eta_i} \right)} \tag{68}$$

where Q_i – discharge from the *i*-th pump; Q – sum of discharges from all pumps; m – number of pumps; η_i– efficiency of the i-th pump (%); ρ – density and g – gravitational coefficient. Taking into account that ρ, g, H, Q do not change, the task may be formulated as a search of a minimum of denominator $\sum_{i=1}^{n} Q_i / \eta_i$ component. Efficiency of the pump at different discharges at the same head H=25 m can be obtained from Figure 5-3 and represented as the polynomial function given as Eq. (69).

$$\eta = 1.758E - 0{,}6 \cdot Q^3 - 0.00213 \cdot Q^2 + 0.7691 \cdot Q + 0 \qquad (69)$$

The following Table 6-1 presents the results of discharge calculations for a different number of pumps when the pumps efficiency is highest. To maximize efficiency of the pumps, individual discharge Q_i for each working pump is calculated. Individual flow of every pump can be different during optimization.

Table 6-1 Water discharge at the highest pump efficiency for a different number of identical pumps.

Number of pumps	*Pump1 l/s*	*Pump2 l/s*	*Pump3 l/s*	*Pump4 l/s*	*Pump5 l/s*	*Total flow, l/s*	*Total efficiency, %*
3	367	367	367	-	-	1100	81.9
4	275	275	275	275	-	1100	86.7
5	220	220	220	220	220	1100	84.6

Table 6-1 shows that in all cases the highest pumps efficiency can be achieved by optimization when all pumps produce the same discharge flow. These results coincide with the findings of Crease (1996), Tianyi et al. (2012) and Koor et al. (2014) that identical pumps work at the highest efficiency if they operate at the same rotational speed and with same discharge.
It is easy to find discharge flow and corresponding rotation speed for each pump by dividing the total flow by number of pumps. The question how many pumps should work to deliver the arbitrary total flow at needed pump head is issue that is more complex. The fact that all identical pumps must work at the same rotational speed makes it easier to calculate out the boundary flows when there is optimal to change the number of working pumps to maintain maximal efficiency of pumps. Since all pumps are identical, it is possible to generate pump efficiency dependencies on discharge flow with the different number of pumps working using following Eq. (70). Let us assume, pump system consists from 5 identical pumps.

$$\eta_{p,m} = c_0 + c_1 \frac{Q}{m} + c_2 \frac{Q^2}{m^2} + c_3 \frac{Q^3}{m^3} \qquad (70)$$

where m – number of pumps; coefficients c_0=0, c_1=0.7691; c_2= –0.00216; c_3= –1.758E–06; Q – discharge.

Results are visualized on the following Figure 6-1.

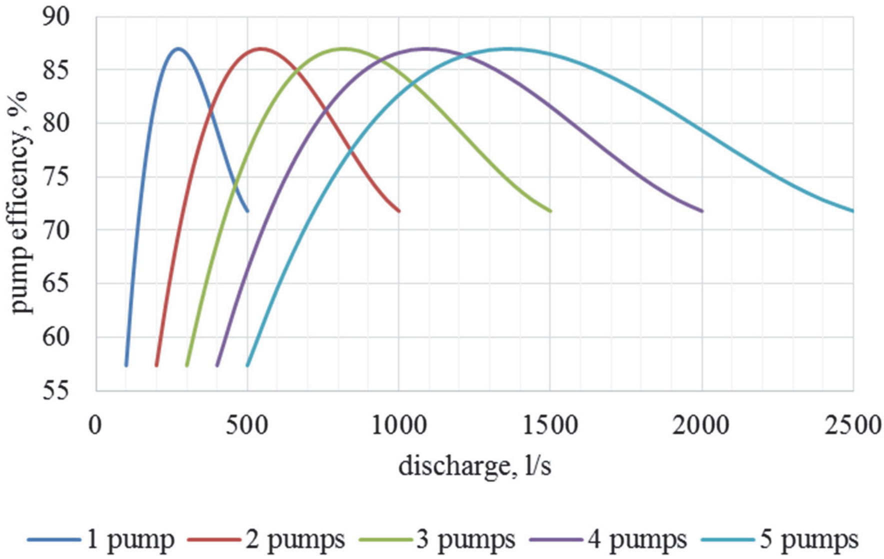

Figure 6-1 Pump efficiencies of pump system with different number of identical pumps at H=25m.

As visualized on the Figure 6-1, one pump works with highest efficiency at 280 l/s and then efficiency starts to drop when flow increases further. Pump system with two identical pumps work with highest total efficiency at 560 l/s. The cross point of those two total efficiency curves gives the discharge value at when usage of two pumps will give higher pumps efficiency. One way to find this flow is by solving the following Eq. (71).

$$c_0 + c_1 \frac{Q}{m-1} + c_2 \frac{Q^2}{(m-1)^2} + c_3 \frac{Q^3}{(m-1)^3} = c_0 + c_1 \frac{Q}{m} + c_2 \frac{Q^2}{m^2} + c_3 \frac{Q^3}{m^3} \tag{71}$$

where m – number of pumps; c0, c1, c2, c3 – coefficients; Q – discharge for different number of pumps.

After removing c_0 that is the same in left and right sides of Eq. (71) and dividing both sides by discharge Q, the Eq. (71) can be simplified as Eq. (72), which can be solved easily.

$$c_1 \frac{1}{m-1} + c_2 \frac{Q}{(m-1)^2} + c_3 \frac{Q^2}{(m-1)^3} = c_1 \frac{1}{m} + c_2 \frac{Q}{m^2} + c_3 \frac{Q^2}{m^3} \tag{72}$$

Second way is to calculate out the flow at minimal difference between two pump efficiency curves. In example, the task can be solved with optimization software using the Levenberg–Marquardt algorithm (LMA) as it was done during current research. Boundary conditions will be the flows when efficiencies of both pumping systems are highest. The second way is more preferable, since it allows usage of different type of approximation to assess the pumps efficiency dependence from flow.

6.2. Scenario 2 - Pumps with slightly different characteristics

As previously described in chapter 3.8, the efficiency of pumps degrades during years of service. In this scenario 2, two alternatives of the calculations were investigated for the pump system consisting from five pumps (pump 1 to pump 5). Although all the five pumps are the same type, some pumps are older and worn-out that causes slightly different performance and efficiency curves. Let us assume that pumps 1 and 4 are new, pumps 2 and 5 have several years of service and pump 3 is the oldest. The pumps switch on priority order is also given. Pump 1 has the highest priority and pump 5 the lowest. That simplifies the calculations. In alternative 1, the efficiency of pump 1 and 4 were taken as new, given by pump manufacturer (Figure 5-3). To simulate pump wear, the efficiency of the pump 2 and pump 5 were decreased by 5% and pump 3 by 10%. The flows when the efficiencies are highest are not changed. Following Figure 6-2 gives an overview about efficiency dependencies of pumps with different age that can be represented by using the following three different polynoms given as Eq. (73), Eq. (74) and Eq. (75). Let us assume, pump system consists from 5 pumps.

Pump 1 and 4 are identical

$$\eta_{1,4} = 1.758 \cdot E^{-6} \cdot Q^3 - 2.13 \cdot E^{-3} \cdot Q^2 + 0.7691 \cdot Q \tag{73}$$

Pump 2 and 5 are identical

$$\eta_{2,5} = 1.670 \cdot E^{-6} \cdot Q^3 - 2.03 \cdot E^{-3} \cdot Q^2 + 0.7306 \cdot Q \tag{74}$$

Pump 3 is unique

$$\eta_3 = 1.583 \cdot E^{-6} \cdot Q^3 - 1.92 \cdot E^{-3} \cdot Q^2 + 0.6922 \cdot Q \tag{75}$$

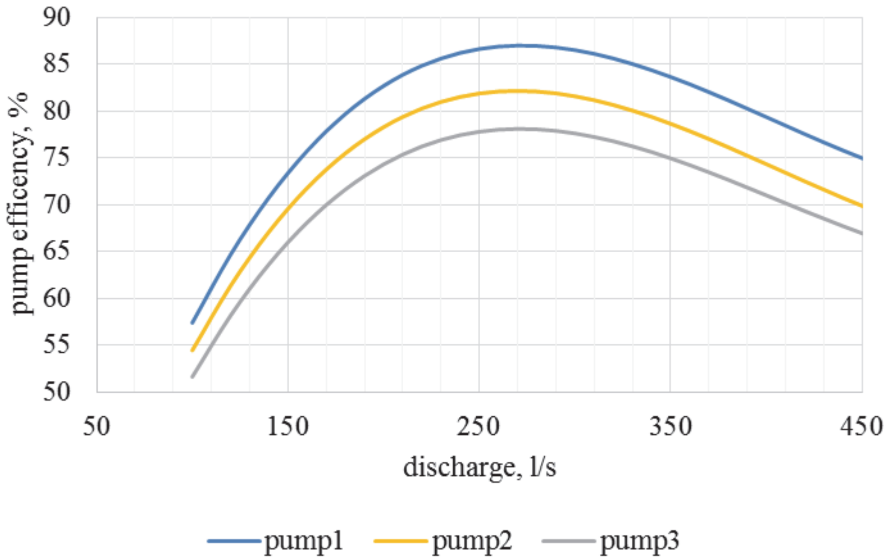

Figure 6-2 Pump efficiencies of slightly different pumps at H=25m.

In the alternative 2, some discharges of pumps at maximal efficiency were also changed in addition to decreased efficiency to test the optimization. The efficiency curves of pump 1 and pump 4 were taken as new, given by pump manufacturer (Figure 5-3). To simulate changed performance, the efficiency curves of the pump 2 and pump 5 were shifted to left by 5% and pump 3 to right by 10%. Individual efficiency dependencies of pumps with different age that can be represented by using the following three different polynoms given as Eq.(76), Eq. (77) and Eq. (78). Pump system still consists from 5 pumps.

Pump 1 and 4 are identical

$$\eta_{1,4} = 1.758 \cdot E^{-6} \cdot Q^3 - 2.13 \cdot E^{-3} \cdot Q^2 + 0.7691 \cdot Q \tag{76}$$

Pump 2 and 5 are identical

$$\eta_{2,5} = 2.412 \cdot E^{-6} \cdot Q^3 - 2.63 \cdot E^{-3} \cdot Q^2 + 0.8545 \cdot Q \tag{77}$$

Pump 3 is unique

$$\eta_3 = 1.321 \cdot E^{-6} \cdot Q^3 - 1.76 \cdot E^{-3} \cdot Q^2 + 0.6992 \cdot Q \tag{78}$$

Polynoms are graphically given also on following Figure 6-3.

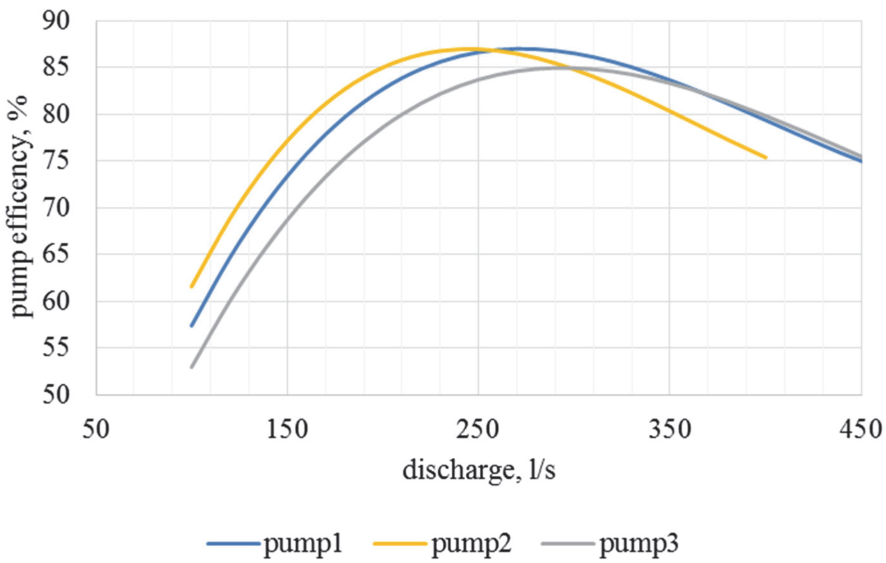

Figure 6-3 Pump efficiencies of slightly different pumps at H=25m with shifted discharges.

Table 6-2 presents the results of calculations. Comparing scenarios 1 and 2, the discharges at the highest combined efficiency differ in this scenario 2. As expected, optimal water discharge is higher for pumps that work with higher efficiency.

Table 6-2 Water discharges at the highest pump efficiency of pumping system with slightly different pumps (scenario 2).

Number of pumps	*Pump1 l/s*	*Pump2 l/s*	*Pump3 l/s*	*Pump4 l/s*	*Pump5 l/s*	*Total flow, l/s*	*Total efficiency, %*
Alternative 1							
3	383	364	353	-	-	1100	78.0
4	283	272	261	284	-	1100	83.4
5	230	218	204	230	218	1100	81.2
Alternative 2							
3	366	344	390	-	-	1100	81.9
4	275	260	289	276	-	1100	86.7
5	222	212	231	222	213	1100	84.9

The calculations showed that optimization software works successfully also in the case with slightly different pumps.

6.3. Scenario 3 - Pumps with large different characteristics

The pumps with quite large differences in characteristics were used in this scenario to mimic situation when some pumps are intentionally replaced with smaller or larger pumps. Efficiency curves of different kind of pumps were represented by the following polynoms given as Eq. (79), Eq. (80) and Eq. (81). Pump system still consists from five pumps.

Pump 1 and 4 are identical

$$\eta_{1,4} = 1.758 \cdot E^{-6} \cdot Q^3 - 2.13 \cdot E^{-3} \cdot Q^2 + 0.7691 \cdot Q \tag{79}$$

Pump 2 and 5 are identical

$$\eta_{2,5} = 2.412 \cdot E^{-6} \cdot Q^3 - 2.63 \cdot E^{-3} \cdot Q^2 + 0.8545 \cdot Q \tag{80}$$

Pump 3 is unique

$$\eta_3 = 1.321 \cdot E^{-6} \cdot Q^3 - 1.76 \cdot E^{-3} \cdot Q^2 + 0.6992 \cdot Q \tag{81}$$

Polynoms are graphically given also on following Figure 6-4.

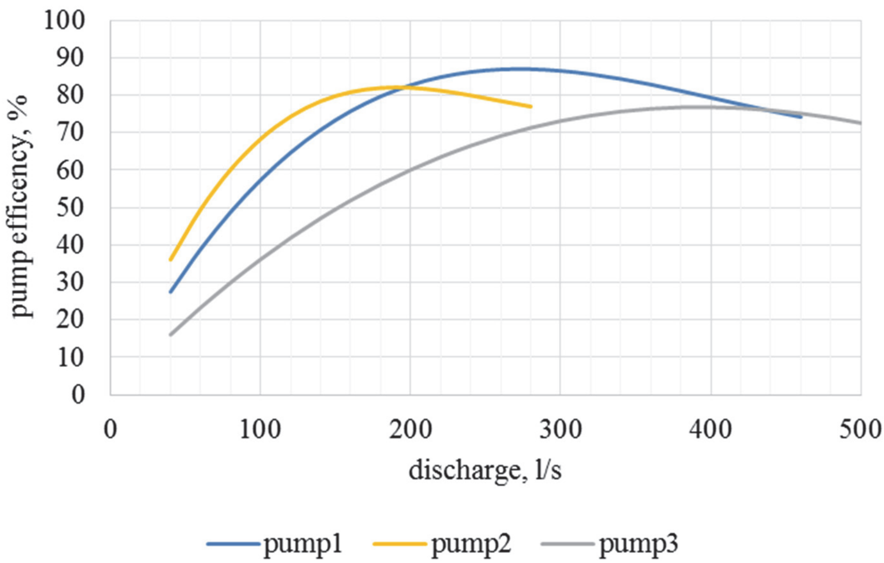

Figure 6-4 Pump efficiencies of largely different pumps at H=25m with shifted discharges.

Optimization was accomplished for a different number of pumps. As in previous scenario 2, three different kind of efficiency curves were used. The pumps switch on priority order was same as in scenario 2. Pump 1 and 4 are identical. Pump 2 and 5 have identical efficiency curves. Pump 3 is different. Table 3 presents the results of optimization.

Table 6-3 Water discharges at the highest efficiency of pumping system with largely different pumps (scenario 3).

Number of pumps	*Pump1 l/s*	*Pump2 l/s*	*Pump3 l/s*	*Pump4 l/s*	*Pump5 l/s*	*Total flow, l/s*	*Total efficiency, %*
3	362	288	450	-	-	1100	77.9
4	271	191	367	271	-	1100	82.2
5	234	153	325	234	154	1100	80.7

The highest water discharge was found during optimization to correspond to the pump with the highest discharge at BEP, as expected. Discharges differ significantly because of high differences in the characteristics of the pumps. For better visualization of results, the surface of pump efficiency was also calculated as an example for the case with three pumps (Figure 6-5).

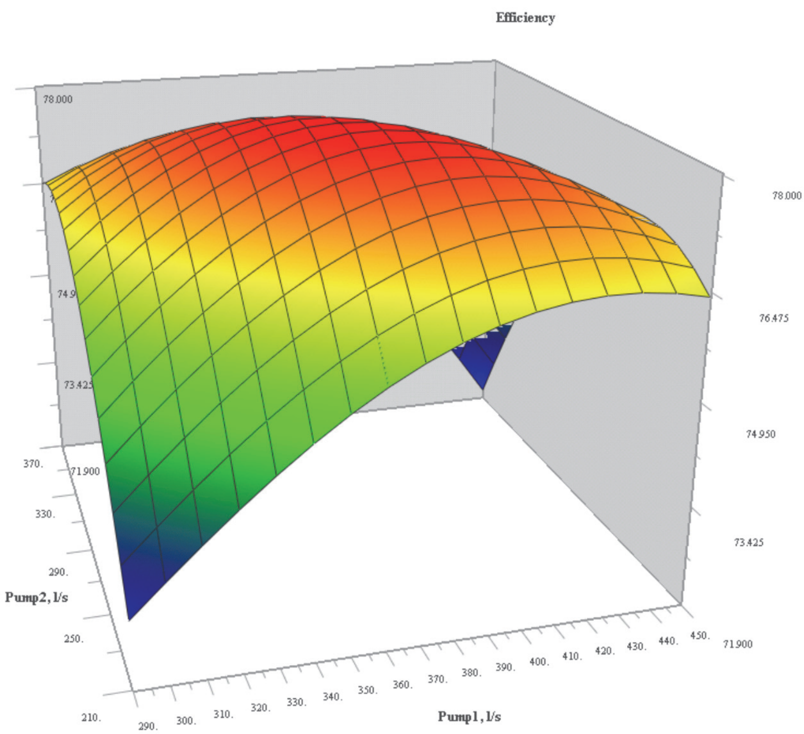

Figure 6-5 Surface of total efficiency of three pumps with quite large differences at H=25m.

Calculations were made for different combinations of discharges. Water discharge was changed from 290 to 450 l/s for pump 1 and from 210 to 370 l/s for pump 2 in this case. Water discharge for pump 3 was estimated as the difference between 1100 l/s and the sum of discharges for pumps 1 and 2. Results of the calculations visualized on Figure 6-5 show, that even with significantly different pumps, the combined efficiency surface of the pump system at constant pressure head is quite simple and allows using of LMA algorithm for optimization.

6.4. Summary of pump system optimization

In addition to chapter 5, current chapter 6 focussed on more theoretical optimization possibilities of pump efficiencies, in which the pumps can have different performance parameters. The previously proposed optimization based on polynomial approximation of pump efficiency surface cannot be used to solve complex optimization challenge as every pump has its own different surface. Second way is to calculate out the flow at minimal difference between two pump efficiency curves. The task is solved with optimization software using the Levenberg–Marquardt algorithm (LMA). Boundary conditions will be the flows when efficiencies of both pumping systems are highest. It allows usage of different type of approximation to assess how the pump system efficiency depends from flow.

Sometimes the polynomial type approximation is less accurate to describe all possible pressure heads even if identical pumps used. Therefore, alternative solution is proposed how to calculate the boundaries that indicate water discharge and pressure head when additional pump should be switched on to ensure the highest pumps efficiencies of the pump system.

7. CONCLUSIONS

The focus of this thesis is on creation of the hydraulic model of Tallinn WDS and proposals how to reduce the energy usage for pumping. The findings of current thesis have immediate practical usage and implementation in Water Company.

7.1. Summary of findings

Chapter 1 gives a brief historical review of the studies and literature how the pumping related efficiencies and energy usage can be calculated and assessed. This thesis focusses only on pumping stations with centrifugal pumps working in parallel.

Chapter 2 introduces new ways, how to develop a hydraulic model based on Tallinn WDS example using *WaterGEMS* v8i SELECTseries 3. The main emphasis lies in developing practical instructions for the Water Company, how to create and renew the hydraulic model in the future to keep it close to real-life. The specific SQL based simplification procedure in database level has been developed and used to decrease the number of network elements and to gather consumer and consumption related data from various Water Company databases. Justification and reasons are given why EPANET *.inp file import had to be used instead of *WaterGEMS* v8i SELECTseries 3 built-in skeletonization tool (Skelebrator). Simplification produces a correct EPANET *.inp file, which can be then imported into the hydraulic model at any time when renewal is needed. Model creation process includes also the optimal selection of pressure measurement points for calibration and field measurements at 10 pressure zones during 105 days in total.

Chapter 3 proposes the efficiency analysis process based on pumping station "Punane" as an example. The working conditions, pressure control settings, energy usage, efficiencies, motor loads, production of CO_2 and operational cost of the existing pumps were analysed. After having the full understanding of the existing situation, the pumping stations efficiency can be improved.

Chapter 4 covers the usage of good engineering practice to make previously analysed pumping station "Punane" more efficient. During the process, the proposal for replacement of the pumps is made with clearly defined benefits. Analysis also includes the estimation of expected investment payback time. Pumps pressure control settings have been reviewed and adjusted, using previously created *WaterGEMS* based hydraulic model of Tallinn WDS. At the end, the considerable financial and environmental saving is achieved – ca 25% less energy used, ca 24.4% lower electricity cost and ca 25.7% reduction in CO_2 emission.

Chapter 5 presents a new mathematical algorithm to optimize the work of identical centrifugal pumps working in parallel. Based on currently needed flow and pressure head, optimal pump switching points can be calculated using the proposed algorithm. Two numerical case studies were carried out to demonstrate the usage of proposed algorithm and to calculate optimal pump switching points for existing Ahlstrom APP 43–250 centrifugal pump and new proposed Grundfos NK 150-315.1/315 centrifugal pumps. Some investigation is carried out in MS Excel using help of ABB Estonia specialists, to test the algorithm logic and usage possibilities in PLC-s. Live input data from flow and pressure sensors were simulated. Results show, that common PLC-s are capable to coop with algorithm logic and it will not require any special type PLC. Usage of proposed algorithm will help to achieve higher pump efficiencies in operation.

Chapter 6 focuses on theoretical optimization possibilities of pump efficiencies when the pumps have different performance parameters. Previously proposed optimization algorithm cannot be used as every pump can have its own different efficiency surface. Therefore, optimization software using the Levenberg–Marquardt algorithm (LMA) is used to calculate out the flows at minimal difference between different pump efficiency curves. Boundary conditions can be the flows when total efficiencies of both pumping systems are highest. It allows usage of different type of approximation to assess the pump efficiency dependence from flow. Optimization showed that it is theoretically possible to achieve higher pump efficiencies with different pumps using flow control.

7.2. Recommendations for further research

The algorithm proposed in chapter 5 has today only theoretical value, because it has not been tested to control in-service pumps programmable link controllers (PLC-s). Today most of the PLC-s are programmed in a very basic way with minimal effort and knowledge about achieving the maximum efficiency of the pumps. The Water Company has a clear objective to move on with the pump work optimization in all booster-pumping stations. That also includes the review and reprogramming of existing PLC-s. Therefore, the main recommendations for further research are as follows:

- To implement proposed algorithm in real PLC-s that control the work of new proposed pumps in the pumping station "Punane". That requires further detailed investigation of PLC programming and ABB *SCADA* system possibilities.

- During the longer operational period, monitor and analyse the power usage and efficiencies of algorithm controlled pumps that will enable to confirm the expected energy saving.

- To investigate further the possibilities how to adopt and expand the usage of the algorithm with pumps that have different characteristics

- To review the hydraulic model creation and simplification procedure proposed in 2011, as today the shapefile can be produced from Water Company GIS database and improved *WaterGEMS* v8i SELECTseries 6 modelling software will be out soon. Generation and usage of *EPANET *.inp* file may not be the best solution anymore.

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ABSTRACT

Current thesis studies the aspects of WDS model creation, calibration and renewal possibilities based on Tallinn city network and develops a method to optimize switching between pumps. The research is driven by practical need to minimize energy usage for pumping and to find new ways how to control the pumps to achieve the highest efficiency of the pumping system.

Novelty of this thesis lies in GIS database level topology simplification procedure developed during the model creation process and in proposed mathematical algorithm for the optimization of the work of identical centrifugal pumps running in parallel. The algorithm makes it possible to calculate out the optimal pumps switching points when to start or stop pumps to achieve the highest pump efficiency. Two numerical case studies were conducted to introduce the usage of the algorithm. The pump switching logic can be programmed into PLC-s controlling the pumps. During the 2015 all pumps, VFD-s and PLC-s in pumping station "Punane" will be replaced. That provides a great opportunity to test algorithm in real life system and assess its effectiveness.

Additionally, an optimization methology is also proposed to optimize the work of centrifugal pumps that have slightly and largely different performance characteristics using flow control. Three theoretical scenarios were investigated to illustrate the concept.

In the research, Bentley *WaterGEMS* V8i software based WDS hydraulic model of Tallinn city is created to enable further network related research. Model is calibrated based on real network data. Calibration was carried out with Darwin Calibrator using the genetic algorithm. Detailed built-up instructions have been worked out how to create and renew the model in the future. The completed and calibrated model was taken into operational use immediately and since that has been used to model more than 532 different practical scenarios.

Efficiency analysis of pumping station "Punane" is carried out as an example to give guidelines to assess other similar pumping stations. Good engineering practice is used to make pumping station "Punane" more efficient. New pumps have been chosen to replace the currently oversized pumps. Pumps pressure control settings have been reviewed and adjusted, using previously created *WaterGEMS* based hydraulic model of Tallinn WDS. It is possible to reduce annual electricity usage by ca 25% that will lower the cost of pumping by ca 24.4% and CO_2 emissions ca 25.7%.

KOKKUVÕTE

Käesolevas doktoritöös uuritakse Tallinna linna reaalselt toimivat veevõrku iseloomustava hüdraulilise mudeli loomise, kalibreerimise ning tulevase uuendamise võimalusi. Teadustöö on ajendatud vee-ettevõtte praktilisest vajadusest vähendada pumpamisel tarbitava energia kogust ja leida uusi võimalusi, kuidas juhtida pumpade tööd selliselt, et pumpade süsteemi kasutegur oleks maksimaalne.

Töö uudsus väljendub SQL andmebaasi põhises topoloogia lihtsustamise protseduuris, mis on välja töötatud hüdraulilise mudeli koostamise protsessi käigus ja matemaatilises algoritmis, millega on võimalik optimeerida identsete, paralleelselt töötavate tsentrifugaalpumpade tööd. Algoritm võimaldab vajaliku vooluhulga ja tõstekõrguse põhjal igal hetkel välja arvutada optimaalse töötavate pumpade arvu ja täiendava pumpa sisse/välja lülitamise tööpunkti. Algoritmi olemust ja kasutamist on tutvustatud kahe juhtumiuuringu kaudu – leides pumpade lülitamise optimaalsed tööpunktid olemasolevatele ja uutele välja pakutud pumpadele. Algoritmi loogika kastutamise võimalikkust pumpade tööd juhtivates kontrollerites on koostöös ABB Eesti spetsialistidega testitud Exceli keskkonnas.

Samuti on käesoleva teadustöö mahus uuritud, kuidas optimeerida paralleelsete tsentrifugaalpumpade koostööd, kui töötavate pumpade efektiivsuse karakteristikud on erinevad. Metoodikat on tutvustatud läbi kolme juhtumiuuringu.

Teadustöö tulemusel on valminud *WaterGEMS* V8i tarkvara põhine, Tallinna linna veevõrku kirjeldav hüdrauliline mudel, mis on eeltingimuseks veevõrgus toimuvate protsesside modelleerimiseks. Hüdrauliline mudel on kalibreeritud reaalsete mõõtmistulemuste põhjal, kasutades Darwin Calibrator moodulit ja GA algoritmi. Välja on töötatud detailsed juhised, kuidas hüdraulilist mudelit tulevikus uuendada. Tänaseks on mudelit kasutatud juba üle 500 erineva stsenaariumi modelleerimiseks.

Doktoritöö raames teostati ka rõhutõstepumpla „Punane“ pumpade efektiivsuse hindamise näidisanalüüs. Välja on pakutud uued pumbad, mille kasutusele võtmisega ja algoritmi rakendamisega on teoreetiliselt võimalik saavutada aastas elektri kokkuhoid *ca* 25% ja vähendada CO_2 emissioone *ca* 25,7%. Samuti väheneb pumpamise elektrikulu *ca* 24,4%.

CURRICULUM VITAE

1. Personal data

Name: Margus Koor
Date and place of birth: 09.09.1975, Kärdla, Estonia
Nationality: Estonian
E-mail address: marguskoor@msn.com

2. Education

Educational institution	Graduation year	Education (field of study/degree)
Tallinn University of Technology	1999	Engineer
Kärdla High School	1992	Secondary

3. Language competence/skills (fluent, average, basic skills)

Language	Level
Estonian	Native language
English	Fluent
German	Basic Skills
Russian	Basic Skills

4. Professional employment

Period	Organisation	Position
2011- ...	AS Tallinna Vesi	Networks development manager
2001-2011	AS Tallinna Vesi	Design supervision manager
2000-2001	AS Tallinna Vesi	IT Project manager
1997-2000	OÜ Veka Inseneribüroo	Water and sewer solutions designer

5. Research activity, including honours and thesis supervised

M. Koor, A. Vassiljev and T. Koppel, "Optimization of Pump Efficiency in a Water Distribution System", – *Proceedings of the Ninth International Conference on Engineering Computational Technology*, P. Iványi and B.H.V. Topping, (Editors), Civil-Comp Press, Stirlingshire, United Kingdom, paper 91, 2014. DOI:10.4203/ccp.105.91

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Defended theses:
Reconstruction of water distribution system of town Kärdla and assessment of city's sanitary status, Engineer's degree (equivalent to Master's degree today), 1999

Honours and awards:
The ESF DoRa programme activity 3, "Research cooperation between universities and businesses" funding, 2011

ELULOOKIRJELDUS

1. Isikuandmed

   Ees- ja perekonnanimi: Margus Koor
   Sünniaeg ja -koht: 09.09.1975, Kärdla, Eesti
   Kodakondsus: eestlane
   E-posti aadress: marguskoor@msn.com

2. Hariduskäik

Õppeasutus (nimetus lõpetamise ajal)	Lõpetamise aeg	Haridus (eriala/kraad)
Tallinna Tehnikaülikool	1999	Ehitusinsener
Kärdla Keskkool	1992	Keskharidus

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

Keel	Tase
Eesti	Emakeel
Inglise	Kõrgtase
Saksa	Algtase
Vene	Algtase

4. Teenistuskäik

Töötamise aeg	Tööandja nimetus	Ametikoht
2011- ...	AS Tallinna Vesi	Võrkude arenduse juht
2001-2011	AS Tallinna Vesi	Projekteerimise järelevalve juht
2000-2001	AS Tallinna Vesi	IT projektjuht
1997-2000	OÜ Veka Inseneribüroo	Projekteerija

5. Teadustegevus, sh tunnustused ja juhendatud lõputööd

Avaldatud artiklid ja konverentsi ettekanded on ära toodud inglise keelses elulookirjelduses.

Kaitstud lõputöö:
Kärdla linna veevarustussüsteemi rekonstruktsioon ja linna sanitaarse olukorra hinnang, 5.a õppekava järgne inseneritöö (võrdsustatud magistritööga), 1999

Teaduspreemiad- ja tunnustused:
ESF DoRa tegevuse 3 „Teadusalane koostöö ülikoolide ja ettevõtete vahel” toetus, 2011

DISSERTATIONS DEFENDED AT TALLINN UNIVERSITY OF TECHNOLOGY ON *CIVIL ENGINEERING*

1. **Heino Mölder**. Cycle of Investigations to Improve the Efficiency and Reliability of Activated Sludge Process in Sewage Treatment Plants. 1992.

2. **Stellian Grabko**. Structure and Properties of Oil-Shale Portland Cement Concrete. 1993.

3. **Kent Arvidsson**. Analysis of Interacting Systems of Shear Walls, Coupled Shear Walls and Frames in Multi-Storey Buildings. 1996.

4. **Andrus Aavik**. Methodical Basis for the Evaluation of Pavement Structural Strength in Estonian Pavement Management System (EPMS). 2003.

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10. **Ott Koppel**. Infrastruktuuri arvestus vertikaalselt integreeritud raudtee-ettevõtja korral: hinnakujunduse aspekt (Eesti peamise raudtee-ettevõtja näitel). 2006.

11. **Targo Kalamees**. Hygrothermal Criteria for Design and Simulation of Buildings. 2006.

12. **Raido Puust**. Probabilistic Leak Detection in Pipe Networks Using the SCEM-UA Algorithm. 2007.

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52. **Artem Rodin**. Propagation and Run-up of Nonlinear Solitary Surface Waves in Shallow Seas and Coastal Areas. 2015.