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Automatic system identification and optimization of centrifugal pump operation
Automaattinen prosessin identifiointi ja optimointi keskipakopumppukäytössä

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Kandidaatintyö 14.11.2012
LUT Energia
Sähkötekniikan koulutusohjelma

TIIVISTELMÄ

Lappeenrannan teknillinen yliopisto
Teknillinen tiedekunta
Sähkötekniikan koulutusohjelma

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2012

Kandidaatintyö
Sivumäärä 30, kuvia 22, taulukoita 2, liitteitä 2.

Tarkastaja: TkT. Tero Ahonen

Keskipakopumput kuluttavat merkittävän osan teollisuuden käyttämästä sähköenergiasta. Tyypillisesti keskipakopumppua käytetään reservisäiliön täyttö- tai tyhjennysprosesseissa, joissa pumppua operoidaan vakionopeudella, kunnes prosessi on saatu päätökseensä. Taajuusmuuttajan asentaminen sähkömoottorin kontrolloimiseksi korvaa perinteisen vakionopeudella operoitavan pumppusysteemin, sallii pumppausprosessin pyörimisnopeusprofiilin optimoinnin sekä mahdollistaa oikosulkumoottorin pyörimisnopeuden ja vääntömomentin estimoinnin ilman lisämittauksia moottorin akselilta. Pyörimisnopeussäädön hyödyntäminen mahdollistaa pumppausprosessin kokonaisenergiankulutuksen pienentämisen.

Pumppausprosessin staattinen nostokorkeus saattaa muuttua prosessin aikana. Reservisäiliön täyttö- ja tyhjennysprosesseissa muuttuva staattinen nostokorkeus johtaa pumpun minimi pyörimisnopeuden vaihteluun. Tällöin pienintä mahdollista energiankulutusta ei voida saavuttaa vakiopyörimisnopeudella. Tässä kandidaatintyössä esitellään sulautettuja algoritmeja reservisäiliön täyttö- ja tyhjennysprosessin automaattista identifiointia, optimointia ja monitorointia varten. Lisäksi työssä arvioidaan staattisen nostokorkeuden muutokseen perustuvaa optimointimenetelmää.

ABSTRACT

Lappeenranta University of Technology
Faculty of Technology
Degree Programme in Electrical Engineering

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Automatic system identification and optimization of centrifugal pump operation
2012

Bachelor's Thesis
Pages 30, pictures 22, tables 2, appendices 2.

Examiner: D.Sc. Tero Ahonen

Centrifugal pumps are a notable end-consumer of electrical energy. Typical application of a centrifugal pump is the filling or emptying of a reservoir tank, where the pump is often operated at a constant speed until the process is completed. Installing a frequency converter to control the motor substitutes the traditional fixed-speed pumping system, allows the optimization of rotational speed profile for the pumping tasks and enables the estimation of rotational speed and shaft torque of an induction motor without any additional measurements from the motor shaft. Utilization of variable-speed operation provides the possibility to decrease the overall energy consumption of the pumping task.

The static head of the pumping process may change during the pumping task. In such systems, the minimum rotational speed changes during reservoir filling or emptying, and the minimum energy consumption can't be achieved with a fixed rotational speed. This thesis presents embedded algorithms to automatically identify, optimize and monitor pumping processes between supply and destination reservoirs, and evaluates the changing static head –based optimization method.

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SYMBOLS AND ABBREVIATIONS

Roman letters

E	energy
H	head
P	power
Q	flow rate
S	sum
Z	vertical distance
d	impeller diameter
g	acceleration due gravity
k	flow losses
n	rotational speed
p	pressure
t	time
v	flow velocity

Greek letters

η	efficiency
ρ	density

Subscripts

d	discharge
dyn	dynamic
est,1	estimate received in phase 1
est,2	estimate received in phase 2
f	friction
i	index
inu, out	drive output
mean	arithmetic mean
n	nominal value
s	suction
st	static
sys	system
tot	total
xy	intersection point

Acronyms

LH	limit, high
LL	limit, low
VI	virtual instrument
dt	drivetrain

1. INTRODUCTION

Typical application of a centrifugal pump is the filling or emptying of a reservoir tank where the pump is operated according to two surface level sensors located in the reservoir. In such systems, the high level sensor signals to start the pump, and the low level sensor, respectively, to stop. Fixed rotational speed pump is typically applied to these applications. Driving the pump at a constant, deliberately chosen speed results in the decrease of total energy consumption, as discussed in (Steffensen, 2010). The described method is based on the usage of an installed flow meter, which is used to quantify the specific energy consumption E_s (Wh/m^3) as a function of rotational speed n. The most advantageous rotational speed is then selected by minimizing the magnitude of E_s as illustrated in Fig. 1.1.

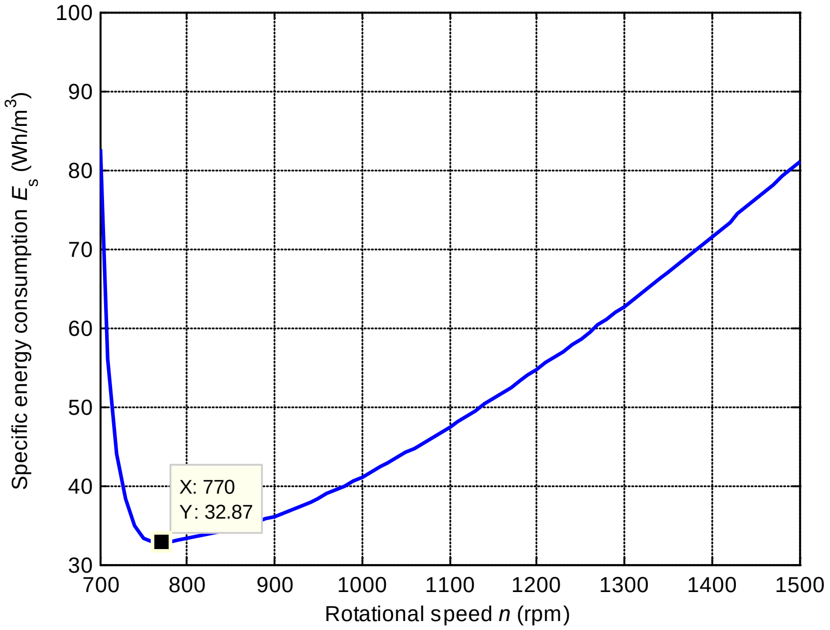

Fig. 1.1 Specific energy consumption E_s as a function of rotational speed n. In this example, the most optimal rotational speed would be 770 rpm, if this is allowed by the process requirements.

Energy efficiency of pumping system operation is often compromised, as the pump is typically oversized for the highest possible difference in liquid elevation. The over-sizing may lead to lower efficiency and to higher energy consumption. Furthermore, the process parameters, primarily the static head H_{st} (m), may change according to the difference in height of supply and destination reservoirs during the pumping. A change in the system characteristics affects the pumping system E_s curve and location of the E_s minimum. The method described in (Steffensen, 2010) doesn't take the effect of changing H_{st} into consideration. However, invention described in (Ahola, 2011) notices the changing process parameters and suggests a method to:

1) Automatically identify the system during the first run;
2) Automatically optimize and monitor energy consumption in a process with supply and destination reservoirs without any additional sensors.

Proposed variable-speed control method determines an optimum rotational speed table according to the identified process and utilizes the speed table to find the E_s minimum at a specific H_{st}. This is achieved by installing a frequency converter into the pumping system.

Centrifugal pumps are typically powered by induction motors, since they are simple and reliable. The alternating supply current of an induction motor results in fixed-speed operation, unless it’s connected across a variable frequency power supply. The proposed installation of a frequency converter serves as such, and the rotational speed can be adjusted to a selected level via controlling the supply frequency. Crucially for the proposed method, a frequency converter can estimate the rotational speed and the shaft torque of an induction motor without any additional measurements from the motor shaft (Vas, 1998). Therefore, the designated rotational speed n, formed energy in pressure H and flow rate Q can be estimated for the purpose to identify the inspected system without any additional sensors.

1.1 Objectives of the work

The main objective of this work is to build virtual instruments to automatically identify, optimize and monitor a given system – whose characteristics enable this type of approach – by using National Instruments’ LabVIEW™ system design software, and evaluate the virtual instrument via laboratory tests. LabVIEW™, graphical programming language, was chosen for its capabilities to relatively easily adjoin the interface between the measurement and the actual pumping system. The measurement system is illustrated in Fig. 1.2.

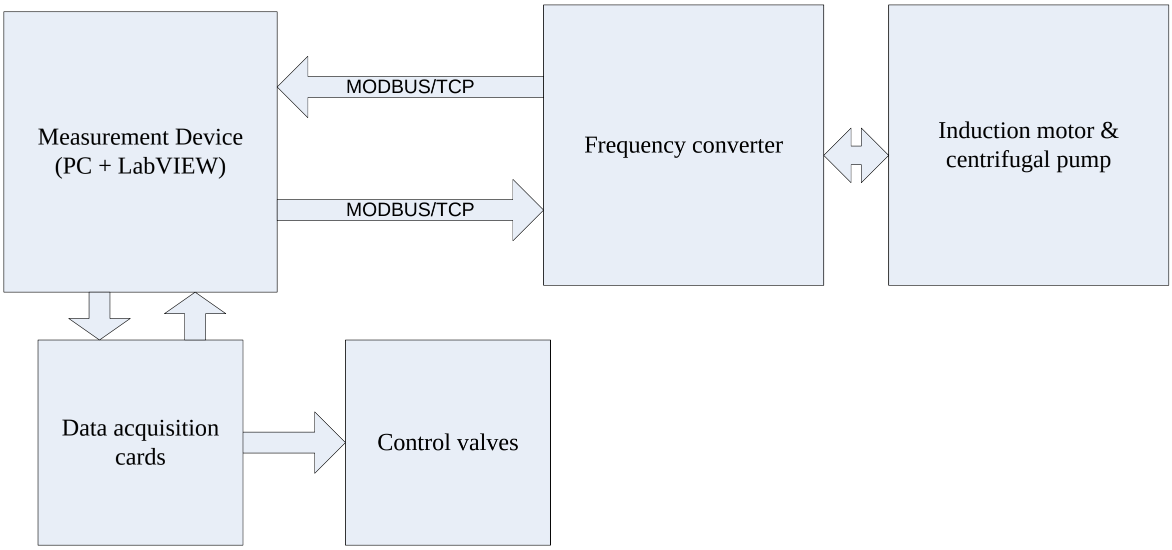

Fig. 1.2 Block diagram of the measurement system.

The measurement system includes a PC with installed LabVIEW™ software and data acquisition cards to send and receive information with control valves. Modbus protocol over TCP/IP was used for data transmissions between LABVIEW™ and ABB ACS850 frequency converter. Studies in this thesis focus on systems, in which liquid is transferred for the purpose of filling or emptying of a reservoir and static head changes during the pumping task, due to the change of liquid level in the reservoir tank. Exemplary pumping system is illustrated in Fig 1.3.

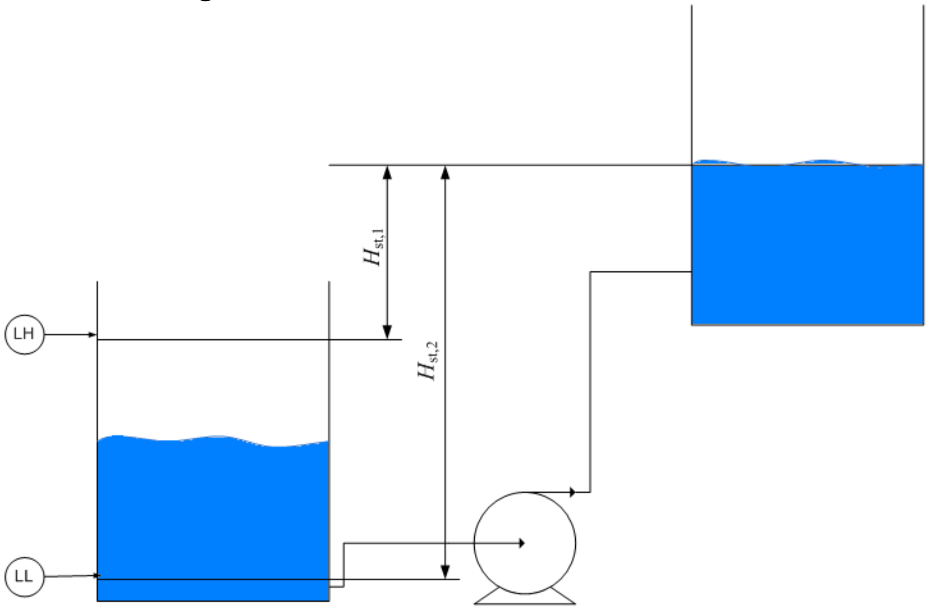

Fig. 1.3 Simplified pumping system, which includes supply and destination reservoirs and a centrifugal pump. Pump is started when the surface level in the supply reservoir rises to the higher limit (LH) and stopped when the low limit (LL) has been reached.

Surface level in the upper, destination reservoir is assumed to remain constant. Therefore, the system static head is only affected by the change of liquid level in the supply reservoir.

The proposed method is also applicable to other kind of reservoir systems, if the system has a typical operation range $H_{st,1}$… $H_{st,2}$ for the static head and the flow losses in the system remain approximately constant during normal operation.

1.2 Outline of the thesis

Chapter 2 explains the basic theory of pumping systems and the effects of speed variation in the form of affinity laws. The description of the automatic system identification method is presented with the built LabVIEW™ virtual instrument (VI).

Chapter 3 focuses on the automatic energy consumption optimization and monitoring. It presents a way to employ the identification method. The description of the optimization method is presented with the built LabVIEW™ virtual instrument.

Chapter 4 is dedicated to laboratory tests.

Chapter 5 summarizes the work and serves as an evaluation of laboratory tests and this thesis.

2. AUTOMATIC SYSTEM IDENTIFICATION

The basic concept in pumping is the addition of kinetic and potential energy to a liquid. In most cases the purpose is either to transfer liquid from a source to a required destination, or to circulate liquid around a system. Supplied energy will cause the liquid to work, such as flow through a pipe or rise to a higher level. A centrifugal pump transforms mechanical energy from a rotating impeller into the kinetic and potential energy required (Karassik 1998).

2.1 Hydraulic characteristics of a pumping system

In pumping, the kinetic and potential energy absorbed by the transferred liquid results in increase in liquid's pressure. This increase in pressure is referred to as head H, and is needed to overcome losses in the system. Heads can be measured in various units, such as feet of liquid or pounds per square inch of pressure, but in this thesis head is measured in terms of meters of liquid, since a change in pressure is directly proportional to the elevation of liquid (Wirzenius, 1978). Liquid capacity transferred per time unit is called flow rate Q.

There are two types of losses that a pump has to overcome: static and dynamic (friction) head (Europump, 2004). Static head H_{st} is the difference in height of the supply and destination reservoirs and is independent of flow rate. Dynamic head H_{dyn} consists of friction losses caused by pipes, valves and equipment in the system. Dynamic head is proportional to the square of the flow rate. This relation is often represented by using the variable k for flow losses. A system curve for a given system describes the head requirement set, the system head H_{sys}, for a centrifugal pump as a function of flow rate. System head can be calculated with:

$$H_{\mathrm{sys}} = H_{\mathrm{st}} + H_{\mathrm{dyn}} \quad (2.1)$$

$$H_{\mathrm{dyn}} = k \cdot Q^2 \quad (2.2)$$

An example of a system curve is presented in Fig. 2.1. Pump's operational characteristics can be described by QH and QP curves, which should be represented for a specific rotational speed.

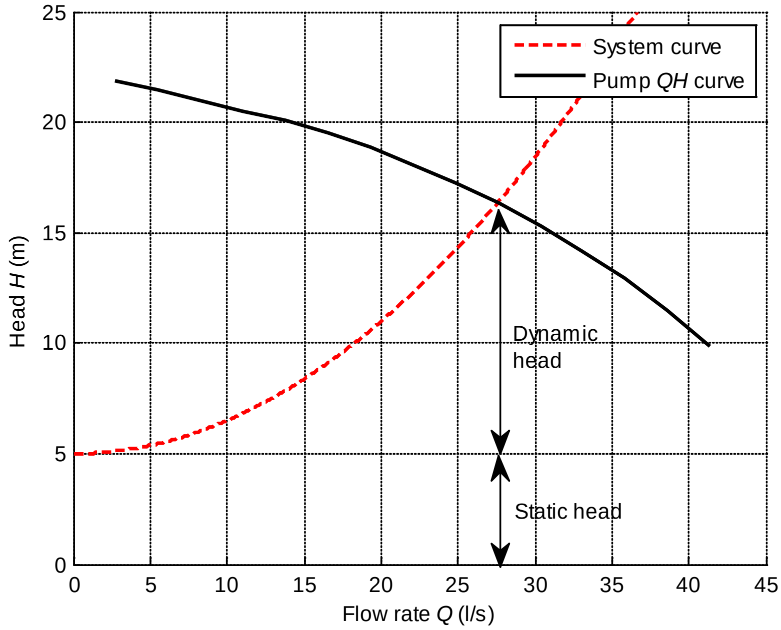

Fig. 2.1 A system curve and a pump *QH* curve (1450 rpm) for a centrifugal pump. Pump always operates at the intersection of these two curves. For centrifugal pumps, an increasing system resistance will reduce the flow rate to zero, but the maximum achievable head is limited by the pump performance.

Efficiency of the pump changes according to the operational point's location. Therefore, the efficiency can be affected by controlling the pump either via pump or system curve. Pump should generally be operated near its nominal operating point (BEP).

2.2 Effects of speed variation

Affinity laws describe relations to estimate the performance of a centrifugal pump, if the rotational speed varies from the nominal value. These relations are:

$$Q = Q_{\mathrm{n}} \cdot \left(\frac{n}{n_{\mathrm{n}}}\right) \tag{2.3}$$

$$H = H_{\mathrm{n}} \cdot \left(\frac{n}{n_{\mathrm{n}}}\right)^2 \tag{2.4}$$

$$P = P_{\mathrm{n}} \cdot \left(\frac{n}{n_{\mathrm{n}}}\right)^3, \tag{2.5}$$

where P is power absorbed and subscript $_{\mathrm{n}}$ refers to nominal value. The squared and cubic relations of head and power implicate, that even small changes in speed result in significant changes in these parameters, as illustrated in Fig 2.2. The pump efficiency is the ratio between hydraulic power – the weight of liquid pumped in a period of time multiplied by the head developed – and power input.

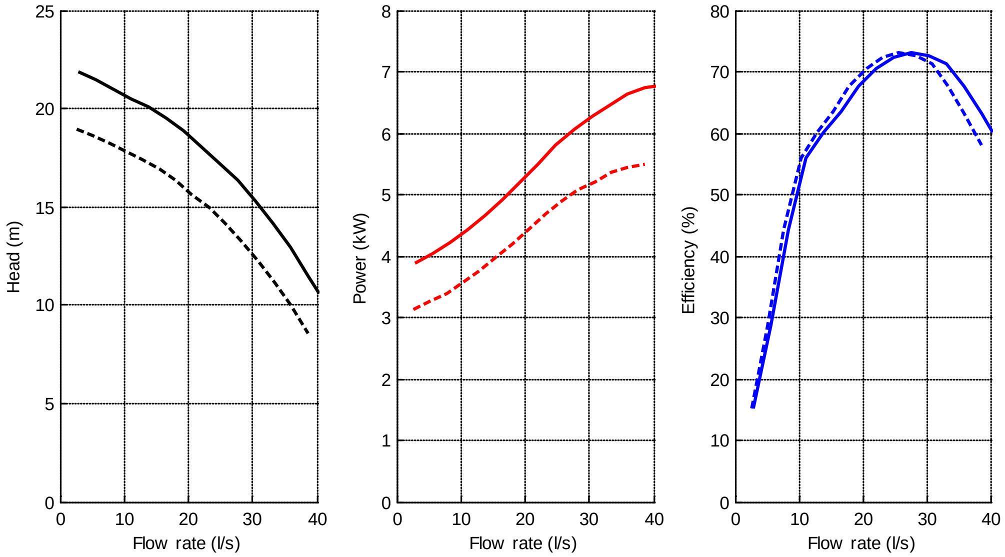

Fig. 2.2 Pump characteristics of head, power and efficiency as functions of flow rate with speed values of 1350 (dashed line) and 1450 rpm.

Affinity laws give a good approximation on pump performance for speed changes up to 2:1 (Karassik, 1998). The system curve must be taken into account in order to determine the actual performance.

2.3 LabVIEW™ implementation of the identification method

The automatic system identification method consists of two main phases: measurement and process identification phase. The procedure of measurement phase is specified in Fig 2.3.

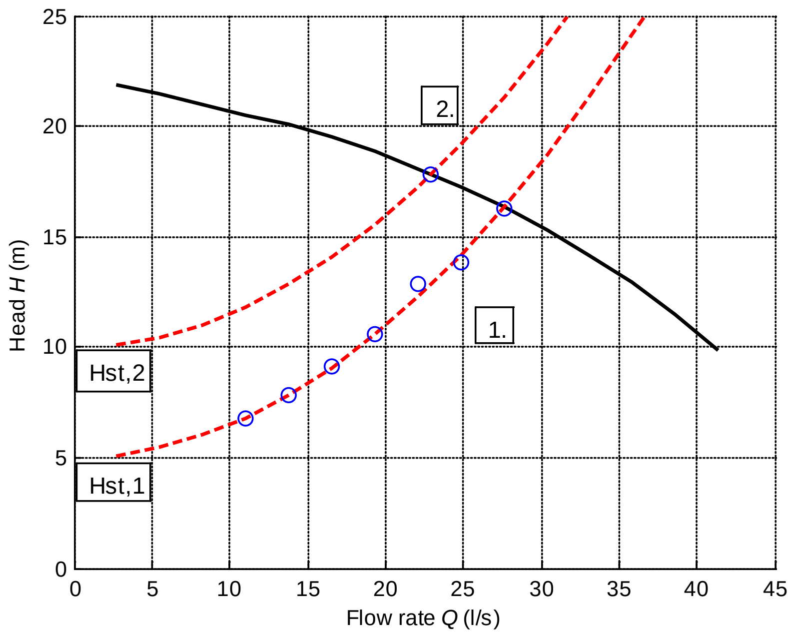

Fig. 2.3 Identification of the parameters of a given process during the first run. The measurement phase consists of two phases: 1. rotational speed ramp; 2. constant rotational speed drive.

During phase 1, the rotational speed is increased according to a specified linear speed ramp. Operating point locations are concurrently stored in an array in purpose of identifying the process curve with static head $H_{st,1}$. During phase 2, static head of the process changes as the surface level of the reservoir tank drops. When surface level of the reservoir is at its lowest, the process static head is as its highest ($H_{st,2}$). To identify $H_{st,2}$, the last operating state is saved in a variable. Actual calculation of process parameters k, $H_{st,1}$ and $H_{st,2}$ and is carried out in process identification phase.

A LabVIEW™ project file was created to process the program as a whole. Different VIs were created to serve as smaller functions of the identification method and a main VI to control all the subroutines. Some modifications had to be made to the frequency converter to enable external control. ABB FENA-01 Ethernet adapter was installed and necessary parameter changes were made for Modbus communication according to parameter setting examples for speed control in FENA-01/-11 user's manual.

Estimation of the pump operational state is carried out with a separate VI. For this purpose, the characteristic curves of the pump are stored in variables using built-in function, which separately loads values from text files. This VI is described in Fig 2.4.

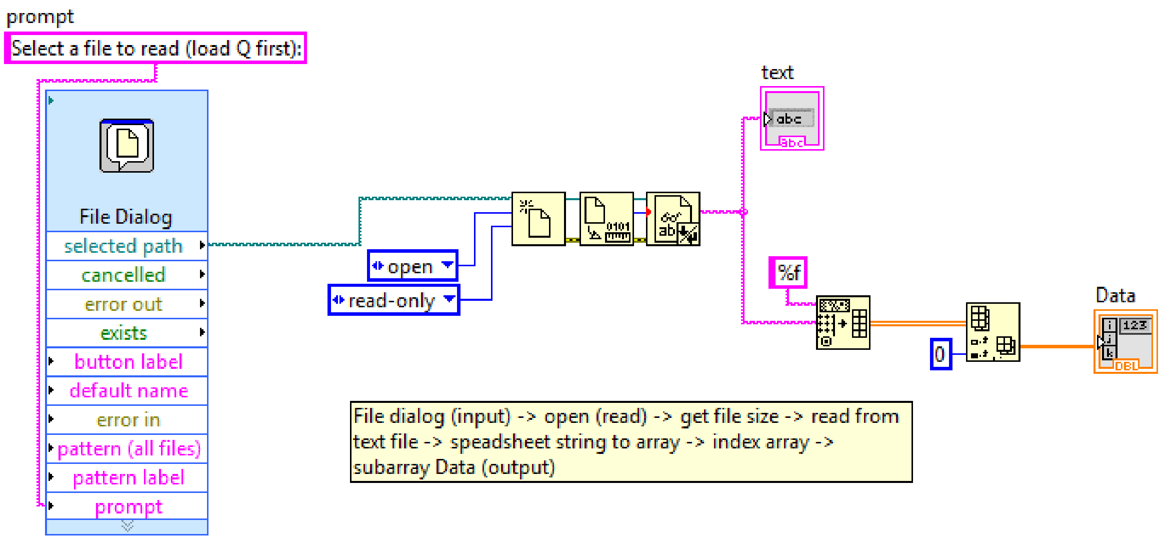

Fig. 2.4 Block diagram of load_data.vi. Text files that are used must include their data in vertical arrays.

The frequency converter can be controlled and monitored externally by using Modbus messaging over TCP on an IP network. Operating is done by 16-bit control words using ABB Drives – Enhanced communication profile between PC (master) and frequency converter (slave). This profile provides register mapped access to the control, status, reference and actual values of the frequency converter.

2.3.1 Measurement phase

During first run the parameters of the system are identified. The procedure is the following: startup signal is sent to the frequency converter by writing control word “1151” to a single register to the slave. Before any further actions, the valves are opened and controlled using a specific VI. Frequency converter then momentarily, for approximately 10 seconds, drives the pump at a low fixed speed to fill the pipes between reservoirs with liquid. The VI which communicates with the frequency converter is illustrated in Appendix A. This VI sends speed references to the drive and gets rotational speed and power estimates from the frequency converter simultaneously. Control words are sent based on the actions of the user on the graphical interface.

After the brief fixed-speed operation, the frequency converter ramps up the flow rate of the pump up to the nominal speed in 30 seconds (phase 1). Concurrently, estimates received from the frequency converter for rotational speed and power ($n_{\text{est},1i}$, $P_{\text{est},1i}$) are sent to other VIs for further processing. Firstly, VIs “PQinterp” and “interp”, interpolate values for $Q_{\text{est},1i}$ and then, respectively, calculate values of head $H_{\text{est},1i}$ for each data point according to actual estimates (Fig 2.5). When the nominal speed has been reached, the pump is operated at a constant rotational speed until the user’s commands signal the pump to be stopped.

When stopped, five most recent values of ($Q_{\mathrm{est},2i}$, $H_{\mathrm{est},2i}$) are stored, and arithmetic mean values are calculated from those data points to form single estimates ($Q_{\mathrm{mean},2}$, $H_{\mathrm{mean},2}$).

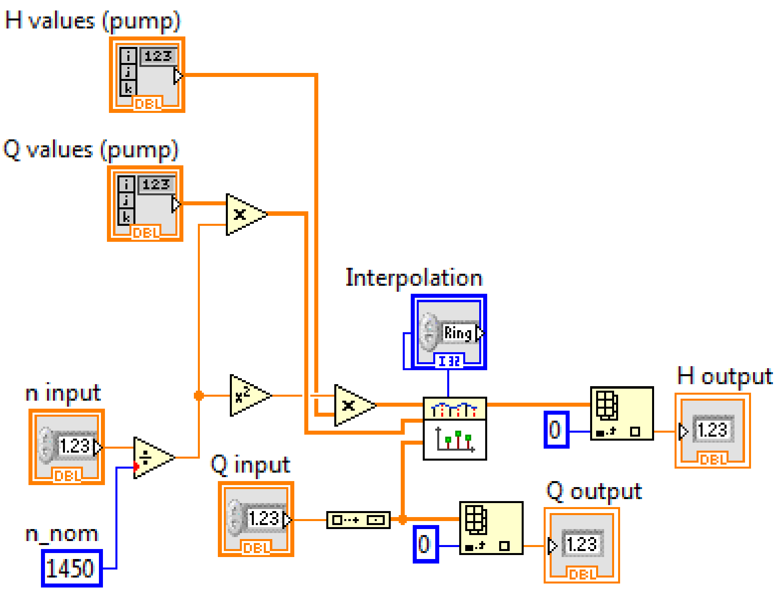

Fig. 2.5 Block diagram of interp.vi. This sub-level VI interpolates head estimates H_i utilizing pump *QH* curve, input values (Q_i, n_i) and affinity laws.

The VI represented in Fig. (2.5) runs concurrently with higher level VIs and interpolates values as long as input data is available. Interpolation for flow rate estimates is handled similarly.

2.3.2 Process identification phase

In the process identification phase, the process curves are estimated using previously loaded pump curves and estimates acquired and calculated in the measurement phase. From data points ($Q_{\mathrm{est},1i}$, $H_{\mathrm{est},1i}$) the process curve parameters $H_{\mathrm{st},1}$ and k are defined with utilizing least squares and Simplex methods, which result in equation:

$$S = \sum_{i=1}^{n} \left(H_{\mathrm{est},1i} - H_{\mathrm{st},1} - k \cdot Q_{\mathrm{est},1i}^{2}\right)^{2}. \tag{2.6}$$

According to equation (2.1) the process curve is of the second degree. The intersection of the process curve and *H*-axis defines $H_{\mathrm{st},1}$. For these calculations a VI was created in LabVIEW™, which is represented in Fig 2.6.

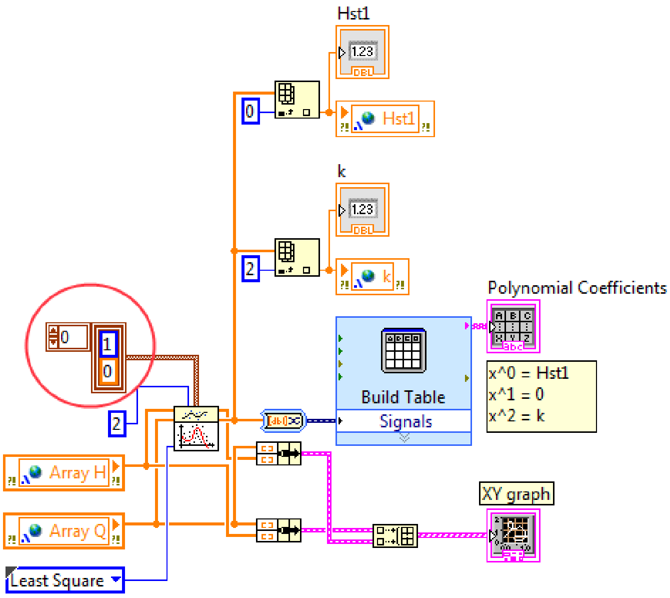

Fig. 2.6 Block diagram of sequentially the first part of solve.vi, which generates the best possible polynomial fit of the second degree for data set ($Q_{\mathrm{est},1i}$, $H_{\mathrm{est},1i}$). The first order coefficient is set to zero in the highlighted area. Therefore, an eligible fit can be achieved.

The flow friction parameter k is assumed to remain constant. Then, process static head as its highest $H_{\mathrm{st},2}$ can be determined with:

$$H_{\mathrm{st},2} = H_{\mathrm{mean},2} - k \cdot Q_{\mathrm{mean},2}^2 \tag{2.7}$$

As a result the pumping process is identified.

3. AUTOMATIC ENERGY CONSUMPTION OPTIMIZATION

Due to the energy efficiency optimization, the possible over-sizing of the pump is automatically fixed, and the rotational speed of the pump is adjusted according to the current static head. Procedure for this optimization is based on the formation of optimum rotational speed table.

3.1 LabVIEW™ implementation of optimum rotational speed table

After the pumping process has been identified, the program creates sufficient number of static heads ranging from the previously determined $H_{st,1}$ to $H_{st,2}$. For each generated static head, a process curve is calculated according to equation (2.1), where flow rate varies from zero up to a sufficient level. Process curve calculation is illustrated in Fig 3.1.

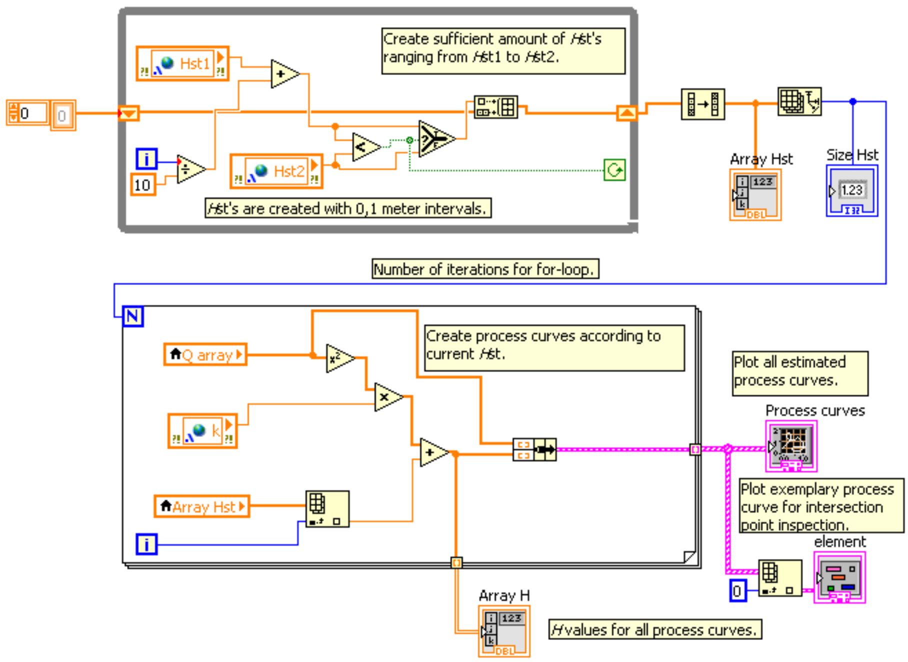

Fig. 3.1 Part of optimum_table.vi, which creates process curves, whose static heads range from $H_{st,1}$ to $H_{st,2}$ with 0.1 m intervals.

Then, the characteristic pump *QH*-curve is calculated for specific rotational speeds according equations (2.3) and (2.4). Rotational speeds were selected to vary from the minimum speed used in the identification ramp startup up to the respective final speed with 5 rpm intervals. The program then finds the intersection points (Q_{xy}, H_{xy}) between each estimated process curve and calculated *QH*-curve. This search is implemented in LabVIEW™ by minimizing the difference in head for each pair of curves and is illustrated in Appendix B. For estimated intersection points, or operating points, values of efficiency are interpolated. The method for estimating operating point locations is described in Fig. 3.2.

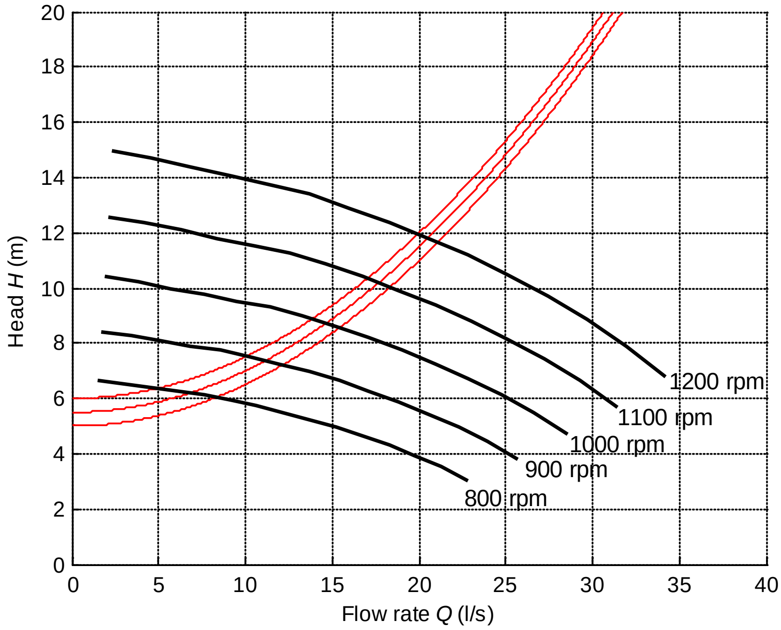

Fig. 3.2 Process curves, whose static heads are 5.0, 5.5 and 6.0 m, and characteristic *QH*-curves for rotational speed values ranging from 800 to 1200 rpm. In the LabVIEW™ program, the actual intervals for intersection point estimation are smaller.

In Fig. (3.2), as the rotational speed of the centrifugal pump increases, the operating point (intersection) moves towards the center of the *QH*-curve. With these exemplary values, this results in increased pump efficiency.

Then, according to intersection points (Q_{xy}, H_{xy}), specific energy consumption curves as a function of flow rate and rotational speed are formed according to (Ahonen, 2011):

$$E_s = \frac{P}{\eta_{dt} \cdot Q} = \frac{\rho \cdot g \cdot (H_{st} + H_{dyn})}{\eta_{dt} \cdot \eta}, \tag{3.1}$$

where ρ is fluid's density, g the prevailing acceleration due gravity, η_{dt} the combined efficiency of the motor and frequency converter and η pump's efficiency at a given operating point. The latter form of equation (3.1) allows the calculation of specific energy according to the current static head and efficiency. The effect of changing static head on specific energy is illustrated in Fig. 3.3.

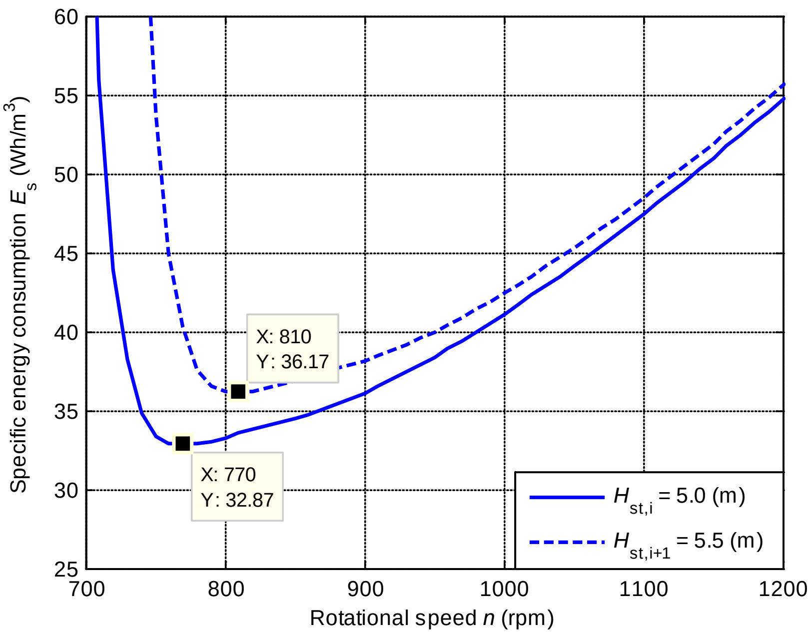

Fig. 3.3 Exemplary specific energy curves as a function of rotational speed for two different static heads. The most optimal rotational speed for static head $H_{st,i}$ isn't favorable throughout the reservoir filling or emptying process.

Even smaller changes in static head affect the overall energy consumption. Therefore, driving the pump at a constant rotational speed shouldn't lead to the best possible solution, if the change in static head is considerable. Calculation of E_s-curves is described in Fig. 3.4.

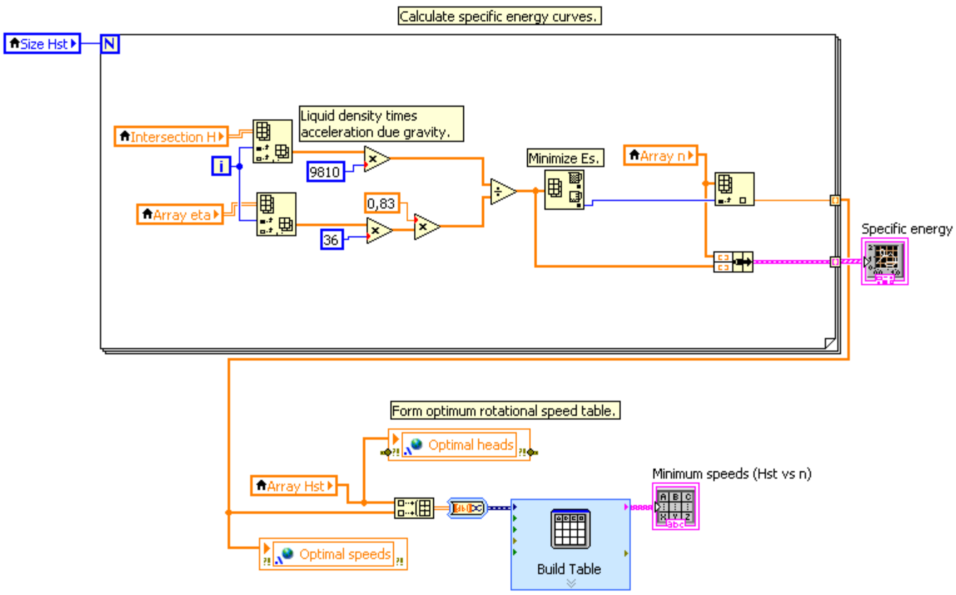

Fig. 3.4 The last part of optimum_table.vi. The number of process curves estimated defines the number of specific energy curves created according to equation (3.1). Each magnitude of E_s is minimized and a matching rotational speed is pointed out. Finally, minimum rotational speeds are associated with corresponding values of static head.

The combined efficiency of motor and frequency converter was assumed to remain constant through the pumping process (η_{dt} = 0.83). As a result, the optimum rotational speed table is created.

3.2 Optimized variable-speed operation

When the process parameters are estimated and the optimum rotational speed table is created, the program utilizes this calculated information to drive the pump as efficiently as possible at all times. Current static head is estimated concurrently when the pump is operated. A LabVIEW™ VI then searches the optimum rotational speed table for the closest value for static head and chooses a corresponding minimum speed. Alternative option is to fill or empty the reservoir using a linear speed ramp.

4. LABORATORY EVALUATION

The system transforming electrical power into liquid's kinetic energy consists of an ABB ACS850 frequency converter, an ABB M3BP160M4 induction motor and a Sulzer APP22-80 centrifugal pump and is illustrated in Fig. 4.1. The nominal rotational speed of the Sulzer pump was 1450 rpm, and the impeller diameter d = 255 mm. Flow rate and pressure could be measured via additional sensors to validate the real operation of the system.

Fig. 4.1 The measurement system used in laboratory evaluation. The core components are the measurement device (PC with installed LabVIEW™), a frequency converter, an induction motor and a centrifugal pump.

The destination reservoir, which is 1 m^3 of liquid capacity, was placed approximately 5 meters above the lower, supply reservoir. A small plastic tube was placed to connect the actual piping and the destination reservoir and to increase the duration of the reservoir filling process. This resulted in decreased pump efficiency and increased flow losses in the system. Coupling to the upper reservoir is illustrated in Fig. 4.2.

Fig. 4.2 View next to the destination reservoir. The flexible tube increases the flow losses in the system and chokes the liquid supply.

Due to the low liquid capacity of the destination reservoir, the liquid level could only vary approximately 50 cm in the supply reservoir during the test process. This limits the variation of the minimum magnitude of E_s down to adverse level for evaluating purposes.

4.1 Modifications to the measurement device

In the identification phase, a linear speed ramp of 800 to 1200 rpm was used. The starting speed was barely high enough to elevate and move liquid in the system. The pump was briefly operated at the chosen starting speed before the actual identification. This should reduce the otherwise rapid change in head developed during startup, as the pipes are already filled with liquid.

In purpose for identifying the test process, the flow rate interpolation from rotational speed and motor power estimates proved to be unreliable. ABB ACQ810 frequency converter was used to check, whether the flow rate estimation could be improved, albeit with little success. Characteristic pump curves were measured for rotational speed of 1100 rpm, which reduced the subtract between the actual and estimated flow rate. The actual head measurements were carried out by determining the pressure difference over the operating centrifugal pump. Using Bernoulli equation for incompressible fluids, the head could be calculated according to (Ahonen, 2011):

$$H = \frac{p_{\mathrm{d}} - p_{\mathrm{s}}}{\rho \cdot g} + \frac{v_{\mathrm{d}}^2 - v_{\mathrm{s}}^2}{2 \cdot g} + \frac{k_{\mathrm{f,d}} \cdot v_{\mathrm{d}}^2 + k_{\mathrm{f,s}} \cdot v_{\mathrm{s}}^2}{2 \cdot g} + (Z_{\mathrm{d}} - Z_{\mathrm{s}}), \tag{4.1}$$

where p is the fluid static pressure, v the flow velocity, k_{f} the friction loss factor between the measurement point and the corresponding pump flange and Z the vertical distance between pressure sensor and reference plane. The subscripts $_{\mathrm{d}}$ and $_{\mathrm{s}}$ denote the discharge and suction sides of the pump. The principle of measuring the generated head is demonstrated in Fig. 4.3.

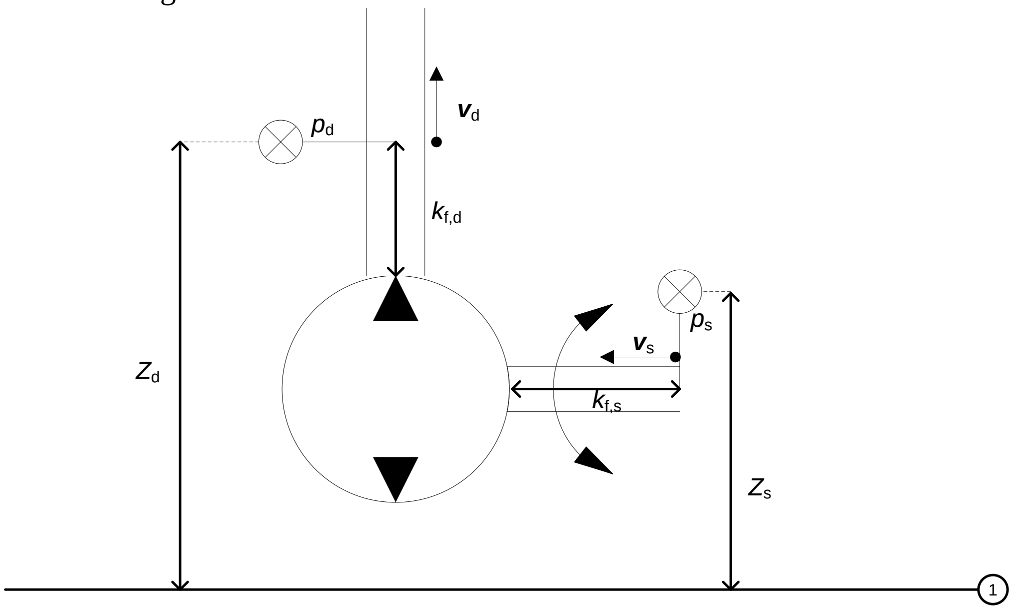

Fig. 4.3 The measurement of head produced by the pump. In the ideal case, p_{d} and p_{s} should be measured at the discharge and suction flanges of the pump.

The flow friction factor k_{f} was assumed to remain as zero during pump operation. The measured pump curve is presented in Table 4.1.

Table 4.1 Measured characteristic pump values for a rotational speed of 1100 rpm and impeller diameter of 255 mm. The measurement was carried out by adjusting the control valve in the system. In proper conditions, Sulzer APP22-80 centrifugal pump should achieve an efficiency of 73 %. * Values were extrapolated.

Q **(l/s)**	***H*** **(m)**	***P*** **(kW)**	***η*** **(%)**
15 (*)	10,4 (*)	2,51 (*)	60,90
12 (*)	11,3 (*)	2,45 (*)	54,23
8,90	12,03	2,36	44,45
8,70	12,09	2,36	43,66
7,95	12,17	2,33	40,70
6,70	12,38	2,27	35,81
5,10	12,47	2,21	28,20
1,35	12,51	2,00	8,27

Even though the measured pump characteristic values improved the estimation of flow rate, the proposed identification method couldn't be executed consistently. Estimation using motor power and acquired rotational speed resulted in erroneous estimates for flow rate, and therefore, especially for head developed. Accelerating the pump more gradually didn't improve the estimation. Test process identification couldn't be realized via estimates ($Q_{est,1i}$, $H_{est,1i}$). Comparison between measured and interpolated flow rate is presented in Fig. 4.4.

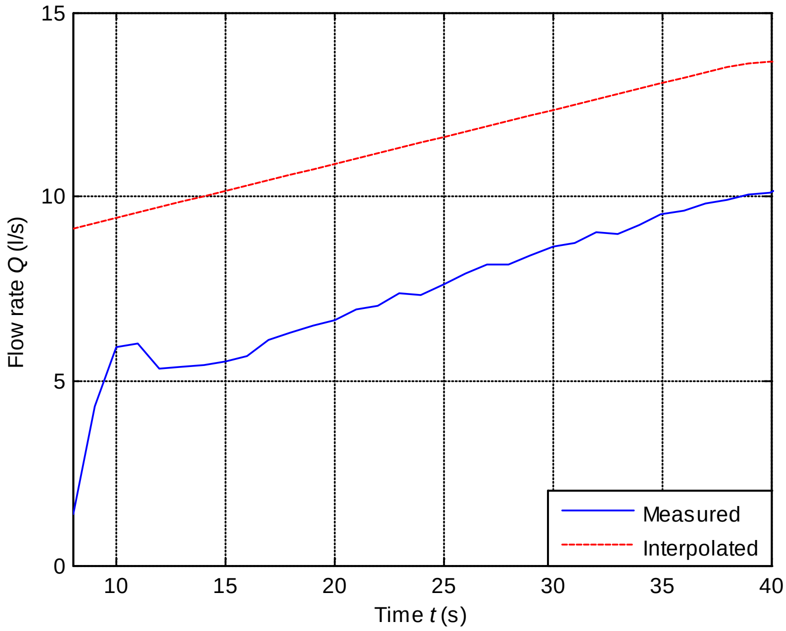

Fig. 4.4 Measured and interpolated flow rate as a function of time. Measured 1100 rpm pump *QP*-curves were used in the interpolation. Values were acquired during identification ramp-shaped drive (800 to 1200 rpm).

Due to the erroneous estimates for flow rate and head developed, the measurement device was modified to acquire signals from an additional flow rate meter and pressure sensors. Therefore, the actual values for flow rate and head could be reliably measured, and the proposed system identification method (Ahola, 2011) evaluated.

4.2 Test results

Using additional measurements, the identifications became more accurate and consistent. The linear speed ramp of 800 to 1200 rpm resulted in a starting static head of approximately 5 m. Calculated static heads, identified flow losses and corresponding minimum rotational speeds are represented for two independent identification drives in Table 4.2.

Table 4.2 Static heads H_{st}, minimum rotational speeds n and flow losses k of two different measurements.

H_{st} [m]	n [rpm]	k[m/(l/s)^2]	H_{st} [m]	n [rpm]	k[m/(l/s)^2]
5,08	875	0,089	5,11	880	0,086
5,18	885	0,089	5,21	885	0,086
5,28	885	0,089	5,31	895	0,086
5,38	885	0,089	5,41	900	0,086
5,48	915	0,089	5,51	910	0,086
5,58	920	0,089	5,61	915	0,086
5,68	930	0,089	5,71	920	0,086
5,78	935	0,089	5,81	920	0,086
5,88	945	0,089	5,91	920	0,086
5,90	940	0,089	6,01	955	0,086
			6,05	955	0,086

Overall energy consumed during reservoir filling process was estimated according to equation:

$$E_{tot} = \int_{t_{\mathrm{LH}}}^{t_{\mathrm{LL}}} P_{\mathrm{inu,out}} dt, \tag{4.2}$$

where t_{LH} is the time indicated by higher liquid level indicator, t_{LL} time when pumping is stopped and $P_{\mathrm{inu,out}}$ the frequency converter's output power. Energy consumption measurements were carried out for different fixed rotational speeds, VSDs utilizing formed minimum rotational speed tables and linear speed ramps. Energy consumption and duration of reservoir filling for fixed rotational speed drives are illustrated in Fig. 4.5 and Fig. 4.6.

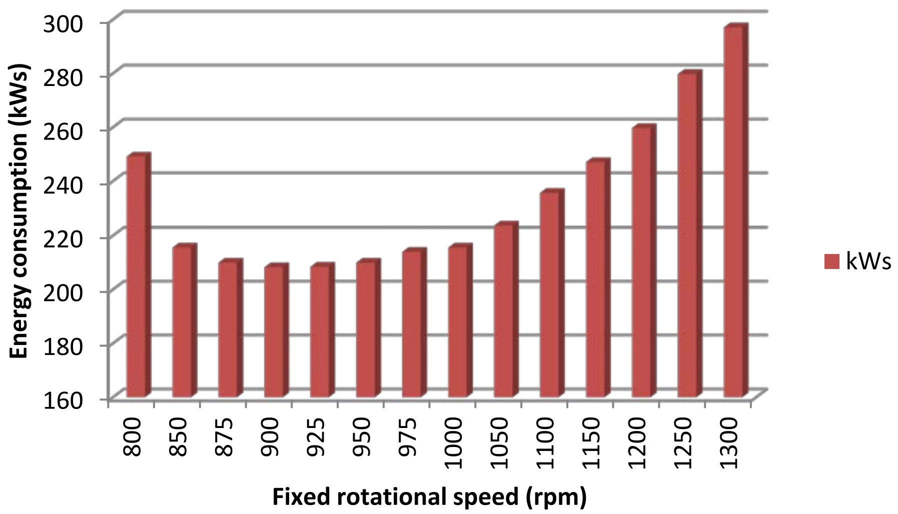

Fig. 4.5 Overall energy consumption for different fixed rotational speeds.

Resulting energy consumptions indicate that the minimum fixed speed is around 900 rpm for the test process. Identified minimum rotational speeds (Table 4.2) support these calculations.

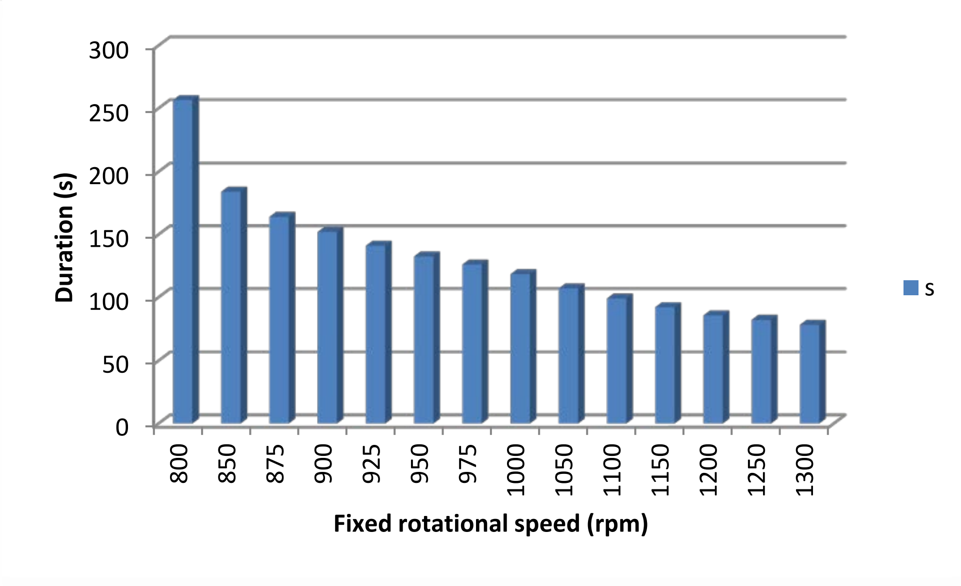

Fig. 4.6 Duration of reservoir filling for previous fixed rotational speeds.

As the rotational speed increases, the duration of the reservoir filling clearly decreases. A too low fixed rotational speed results in high energy consumption due to the duration of the process. Rotational speeds over 1000 rpm require increasingly more power and result in unfavorable energy consumption, despite the faster execution.

Then, the effect of driving the pump at a variable rotational speed on energy consumption was tested. Linear, ramp-shaped drive and speed control according to current estimated static head and minimum rotational speed table are compared to the lowest fixed speed drives in Fig. 4.7 and Fig. 4.8.

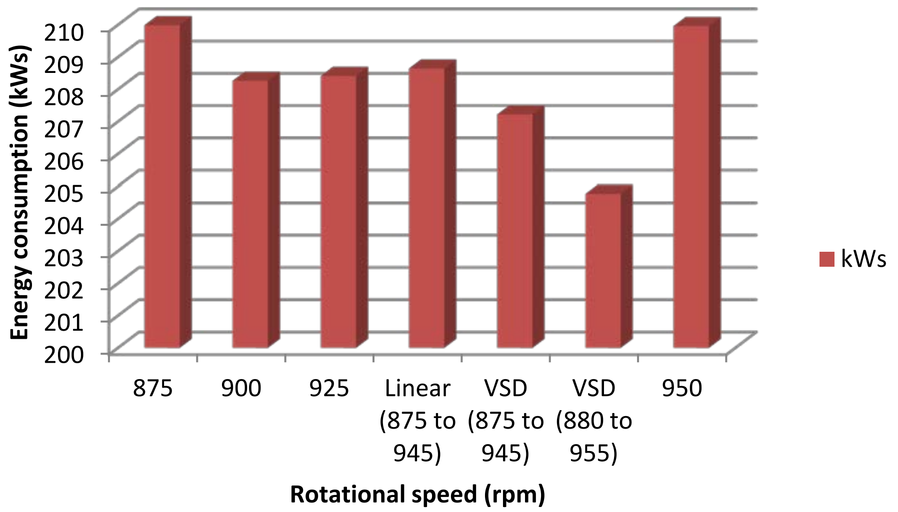

Fig. 4.7 Energy consumptions for lowest fixed rotational speeds, linear ramp drive and VSD drives according to the minimum rotational speed tables (Table 4.2).

Driving the pump according to the minimum rotational speed determined by the current static head was competitive in energy efficiency compared to the lowest consuming fixed speed drives. Constant speed of 900 rpm results in an energy consumption of approximately 208.3 kWs, and VSD (880 to 955) in 204.8 kWs.

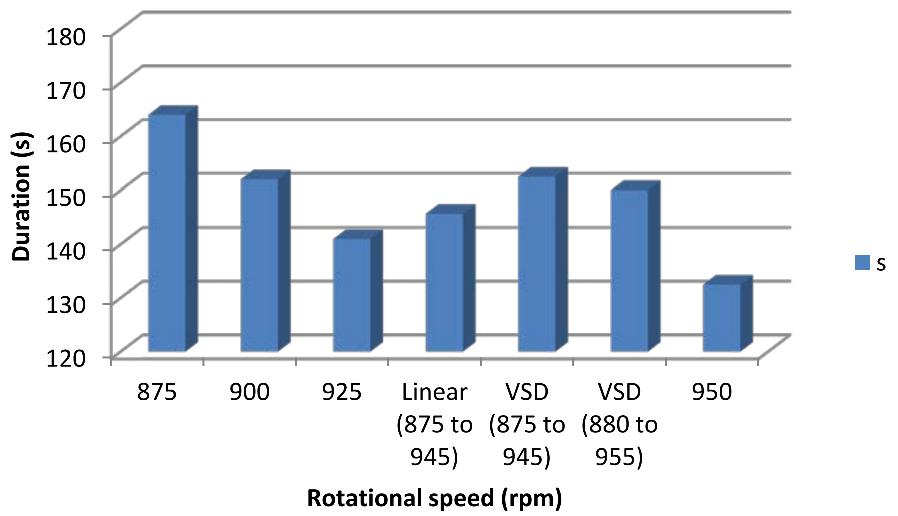

Fig. 4.8 Duration of reservoir filling for previous drives.

Duration of reservoir filling for these drives differs considerably more than the total energy consumption. The difference between two of the lowest consuming drives, fixed 900 rpm and VSD (880 to 955), was also small in this comparison. Closer inspection on the different optimized drives is illustrated in Fig. 4.9.

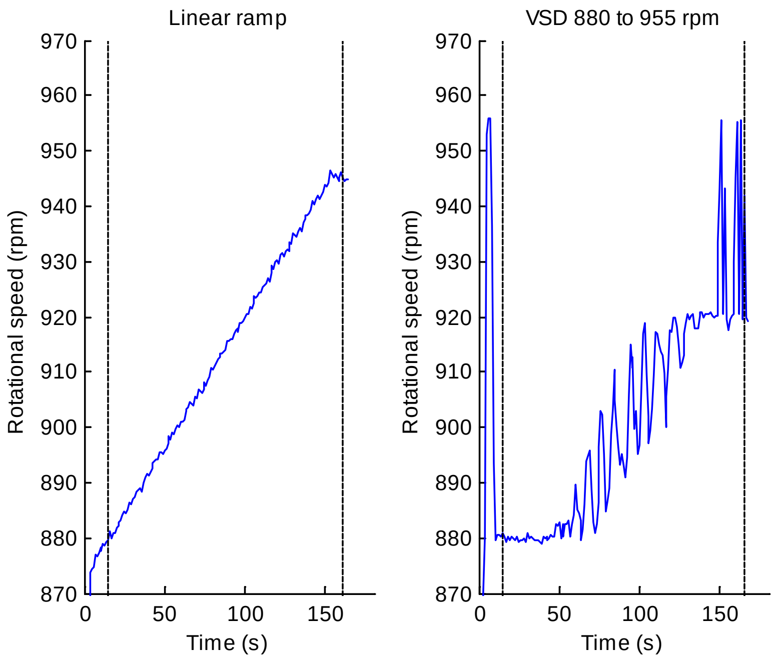

Fig. 4.9 Rotational speed as a function of time for linear speed ramp and variable-speed operation according to minimum rotational speed table. The dashed lines represent the duration of the actual reservoir filling – time indicated by the high and low liquid level indicators.

Linear speed ramp – although consuming a bit more energy than the operation according to the minimum rotational speed table – is considerably less aggressive solution to drive the pump. The fluctuating liquid level in the upper reservoir causes the calculated static head to vary erroneously. This results in notable fluctuation in the reference rotational speed. But as the change in static head was so minor, the momentarily required power didn't increase to unfavorable level during the test process.

5. CONCLUSIONS

LabVIEW™ virtual instruments were created to identify the inspected system – to solve process parameters *k*, $H_{st,1}$ and $H_{st,2}$ – during the first run, and to optimize and monitor energy consumption in a pumping process between supply and destination reservoirs. The proposed system identification method (Ahola, 2011) couldn't be realized in its original form, since the method had to be evaluated using additional sensors to measure flow rate and pressure in the test system. Using additional sensors, the created measurement device was capable of identifying the test system and able to optimize energy consumption. Driving the pump according to the formed minimum rotational speed table resulted in as low energy consumption during reservoir filling as the most energy-efficient fixed-speed drives. Due to the low liquid capacity of the destination reservoir, the conditions to evaluate the proposed method weren't optimal, since the change in static head was insufficient for proper evaluation. The full potential of savings in energy should be further evaluated by using a reservoir tank with higher liquid capacity. Finally, evaluation of sensorless estimation of flow rate and applicable filtering of motor power estimates acquired from frequency converter is recommended for future study.

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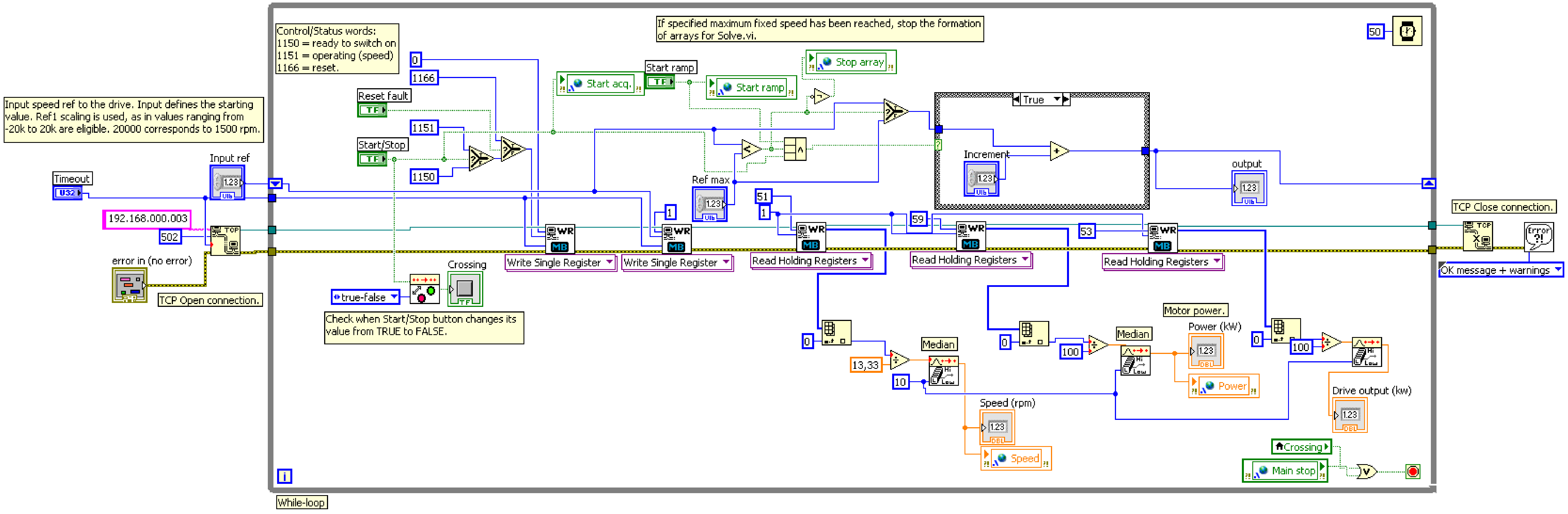
APPENDIX A

Block diagram of Ethernet_comm.vi



APPENDIX B

LabVIEW™ implementation of intersection point estimation