MSc: Sustainable Engineering - Energy Systems and the Environment

Cameron L. Smith

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"Energy Efficient Operation of Submersible Borehole Water Pumps Using Electrical Variable Speed Drives"

Department of Mechanical Engineering

Faculty of Engineering

University of Strathclyde

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Abstract

Traditionally, water pumping processes have utilised mechanical control devices to regulate the flow rate delivered by a pump. The most widely used of these devices are flow control valves. These valves function by inserting additional resistance at the outlet of a pump. This additional resistance results in lost energy. Over recent years more interest has been taken in the energy efficiency of water pumping processes. Flow control is an area which has seen significant development, particularly in the field of electrical control. Electrical control devices have been developed, and established in industry, that can perform flow control for many water pumping applications. These electrical control devices, known as 'electrical variable speed drives', function by regulating the speed of a pump in order to deliver the required flow rate. For many water pumping applications electrical variable speed drives provide better energy efficiency than the equivalent mechanical flow control device. This project is a study into the application of electrical variable speed drives within water pumping systems. The focus of the study is the energy efficiency achievable through the use of electrical variable speed drives in comparison to flow control valves. The study looks at various types of water pumping system, in relation to specific flow control requirements. A case study is presented of a water pumping station which uses flow control valves to maintain constant flow rates delivered by each pump in response to variations in system pressure. An analysis is made in order to determine the economic viability of installing electrical variable speed drives in the system to replace the flow control valves. For a given set of flow conditions, it is shown that energy savings ranging from 9% to 25% can be achieved on each pump that is operated by an electrical variable speed drive.

Chapter 1. Introduction

This project is an investigation into the energy efficiency of pump operation at the Spey Wellfields Water Pumping Station. The pumping station, which is operated by Scottish Water, has been in operation since early 1995, and supplies water to the local public water distribution system. The pumping station is situated on the banks of the river Spey, in the north east of Scotland. The map below depicts the location.

Figure 1.1 *location of the Spey Wellfields Water Pumping Station*

The site consists of 36 submersible borehole pumps and two control buildings, the locations of which are detailed in figure 1.2.

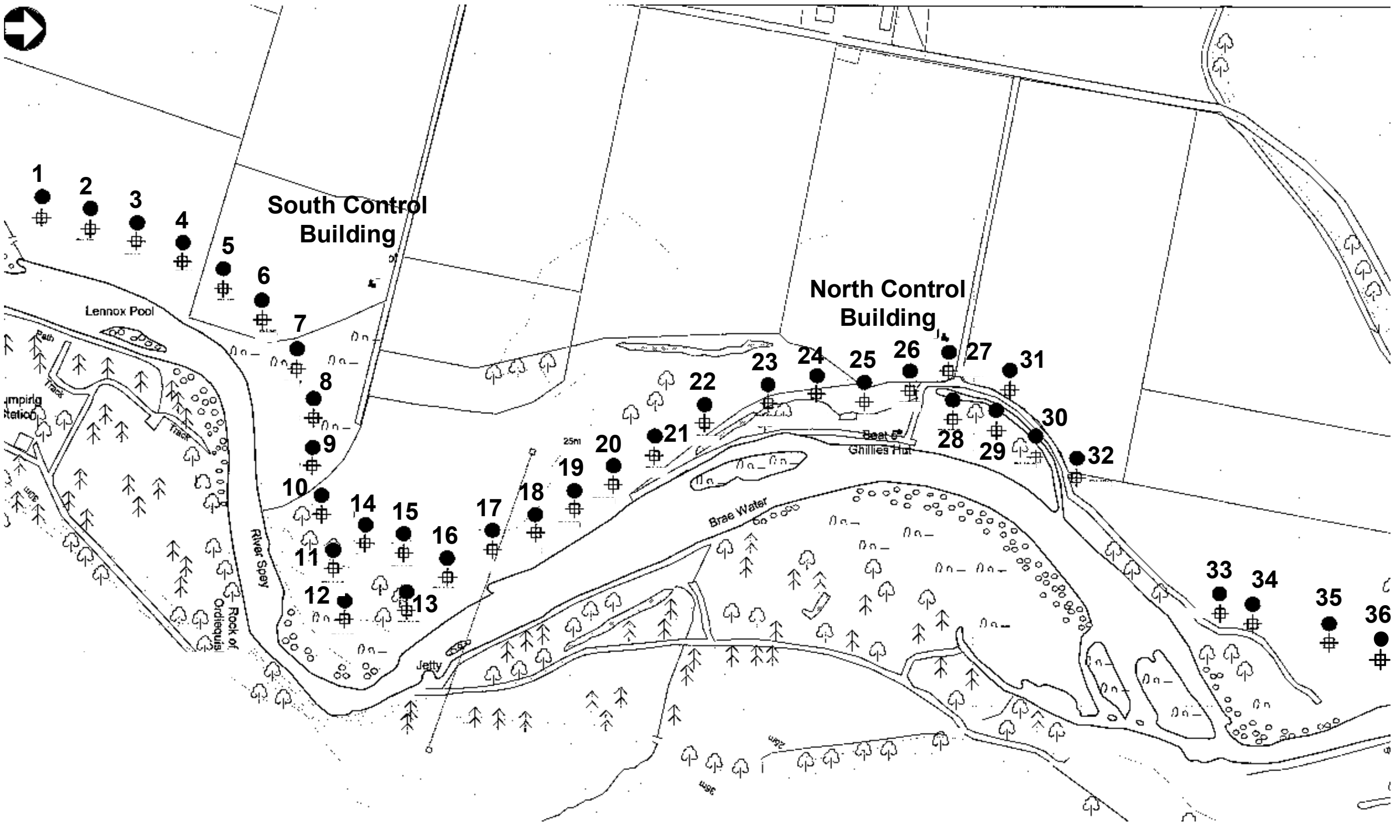


Figure 1.2 *site plan*

The plant currently operates up to 36 pumps, each controlled by a throttling valve to maintain a constant pump flow rate. When a pump is operating, its motor is running constantly at full load. Many of the motors are operated in excess of their rated power capacity. Energy is wasted by throttling the pump discharges. Regular pump failures are experienced. An investigation was proposed into the excessive power consumption of the motors, to see if electrical variable speed drives could be used to economically reduce power consumption, and consequently provide better energy efficiency and reduce pump failures. This project aims to analyse the pumping system at the Spey Wellfields Water Pumping Station in order to determine the potential power reduction and energy savings achievable through the use of electrical variable speed drives.

Chapter 2. Water Pumps

There are several types of water pump. The two main categories are 'rotodynamic' and 'positive displacement'. These are defined by distinctive modes of operation. Within each of these categories there is a further breakdown into many types of pump. Figure 2.1 shows the main categories of pump.

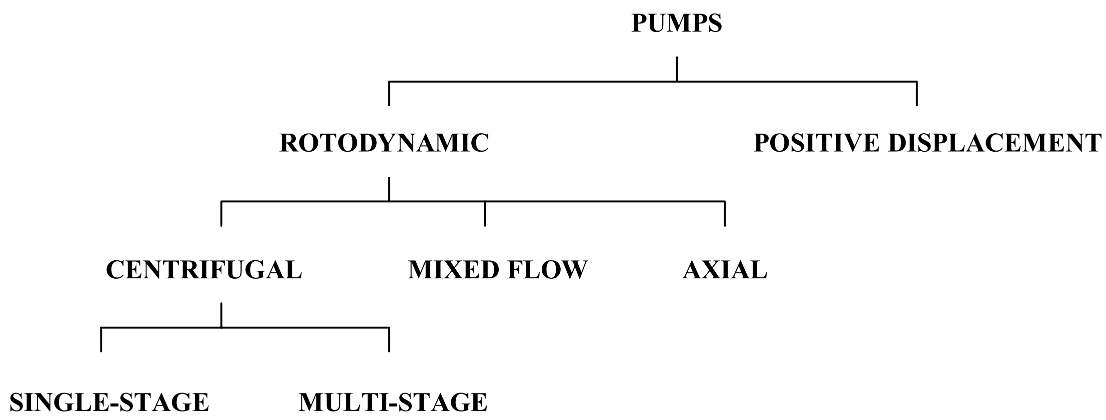


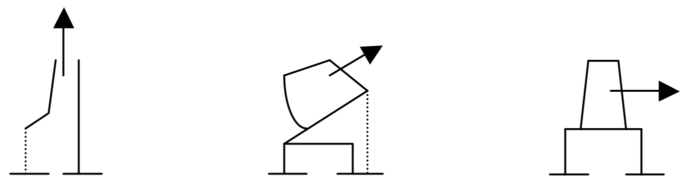
Figure 2.1 *pump categories*

2.1 Positive Displacement Pumps

Positive displacement pumps work on the principle of decreasing the volume of the boundaries containing the fluid, and so increasing the pressure at the outlet valve, and allowing the fluid to move into the discharge line. Positive displacement pumps are best suited to applications for pumping high viscosity fluids, pressure control applications, and for high pressure, low flow applications.

2.2 Rotodynamic Pumps

Rotodynamic pumps utilise a rotational part, known as an 'impeller', to displace fluid by generating a higher pressure. The direction in which the fluid flows through the impeller defines the pump type: radial flow (or centrifugal); axial flow; and mixed flow (for intermediate directions of flow). Figure 2.2 illustrates these types of impeller, used with rotodynamic pump.



a. *radial flow* ***b.*** *mixed flow* ***c.*** *axial flow*

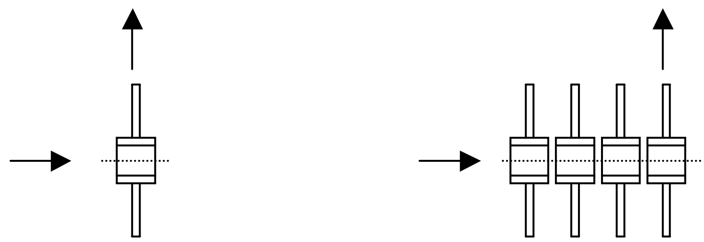
Figure 2.2 *types of rotodynamic pump impeller*

2.2.1 Centrifugal Pumps

For almost all water pumping applications in industry, it is centrifugal pumps that are used.

2.2.1.1 Pump Stages

Centrifugal pumps are either single-stage or multi-stage. Single-stage pumps have only one impeller. Multi-stage pumps can have any number of impellers connected in series. A greater number of impellers (or stages) will generate a higher pressure for a given flow rate. Figure 2.3 illustrates the arrangement of impellers in single-stage and multi-stage pumps.



a. *single-stage* ***b.*** *multi-stage*

Figure 2.3 *impeller arrangements in centrifugal pumps*

2.2.1.2 Pump and System Characteristics

The characteristics of a centrifugal pump are often described by graphs, known as the 'characteristic curves'. These curves show the head generated by the pump and the efficiency of the pump, across the range of flow rates at which it can operate. An example of the characteristic curves of a centrifugal pump is shown in figure 2.4. The plot showing the head generated by the pump against pump flow rate is known as the 'pump curve'.

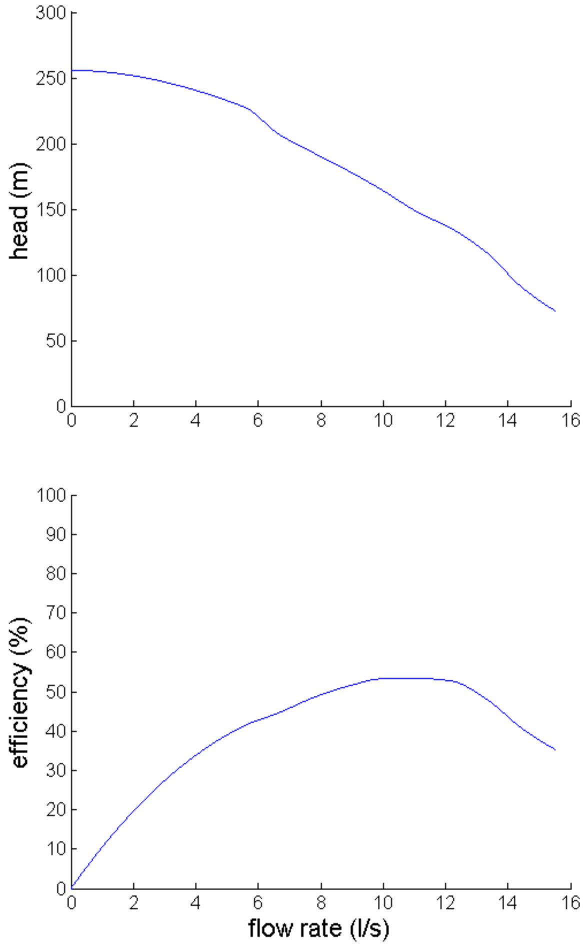


Figure 2.4 *example of the characteristic curves of a centrifugal pump*

The power absorbed by a pump can be found from the characteristic curves by using the following equation:

power absorbed, $P = \dfrac{\rho.Q.g.H}{1000.\eta}$ (W) **(equn. 2.1)**

where ρ is liquid density ($^{kg}/_{m^3}$), Q is the pump flow rate ($^{l}/_{s}$), g is acceleration due to gravity ($^{m}/_{s^2}$), H is the head generated by the pump (m) and η is the wire-to-water efficiency (%)

For water at standard conditions, $\rho = 1000 \,{}^{kg}\!/_{m^3}$. Assuming this value for density is constant, then the power absorbed by a water pump can be written as:

$$P = \frac{Q.g.H}{\eta} \quad (W) \qquad \textbf{(equn. 2.2)}$$

A pump would normally be selected such that the head requirement and desired flow rate occur at the maximum efficiency point. The head requirement is dependent upon the piping and valve systems through which the output must flow. This can be broken down into two components of head loss: the dynamic head loss - due to frictional resistance in the system; and static head loss - the height by which the water must be raised. The static and dynamic head losses are normally plotted against the pump curve. This curve, describing total head loss, is referred to as the 'system curve', and represents the resistance of the system. Figure 2.5 shows an example of a system curve, plotted against a pump curve.

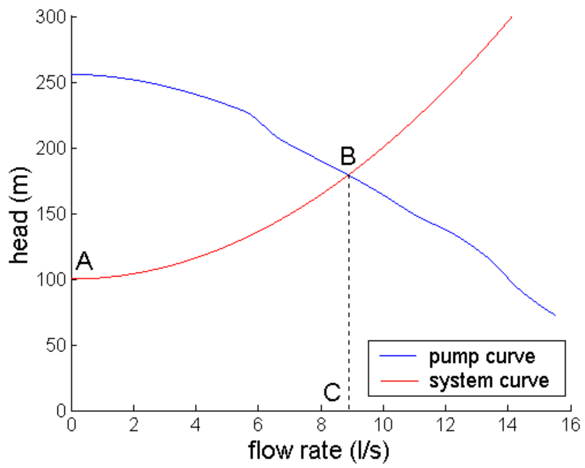


Figure 2.5 *example of pump curve and system curve*

The intersection of the system curve with the *H*-axis (point A) is the value of static head loss. The static head loss is independent of the pump flow rate. The curve from point A to point B represents the dynamic head loss in the system, which is due to frictional losses in the pipework and valves. The dynamic head loss is a function of the pump flow rate, and can be expressed in the form:

dynamic head loss, $H_D = k.Q^2$ (m) **(equn. 2.3)**

where k is a constant and Q is pump flow rate $\left({}^{l}\!/_{s} \right)$

So, the system curve can be written in the form:

system head loss, $H_{SYS} = H_S + k.Q^2$ (m) **(equn. 2.4)**

where H_S is static head loss (m)

In the example shown in figure 2.5, it can be seen that the pump overcomes the system resistance up to point B - where the pump curve and system curve intersect. Point B, here, is known as the 'operating point'. Under these conditions the pump will deliver flow rate, C.

2.2.1.3 Flow Control

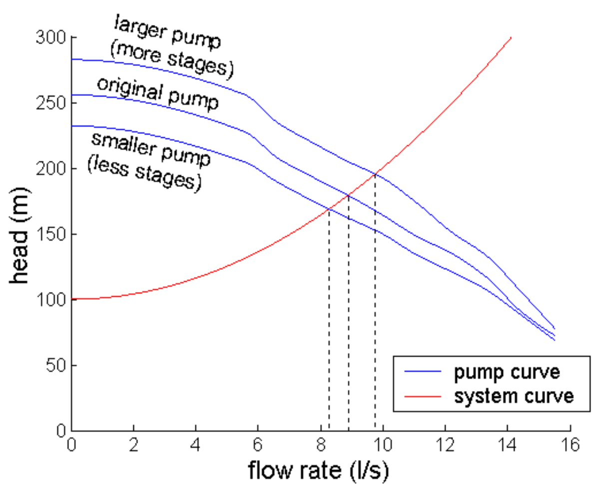
If a different flow rate is required, then it may be achieved by one of several methods. For achieving higher flow rates, possible methods which may be employed include:

- installing a larger pump
- adding impeller(s) - if the pump physically allows for this, for example, if impellers have previously been removed
- increasing rotational speed of the pump
- reducing system resistance, for example, installing pipes with less resistance

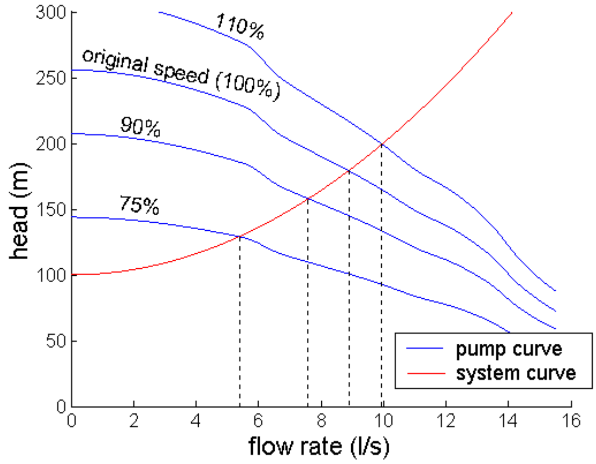
Lower flow rates may be achieved by the following methods:

- installing a smaller pump
- removing impeller(s)
- trimming impeller(s)
- decreasing rotational speed of the pump
- increasing system resistance, for example, using a throttling valve

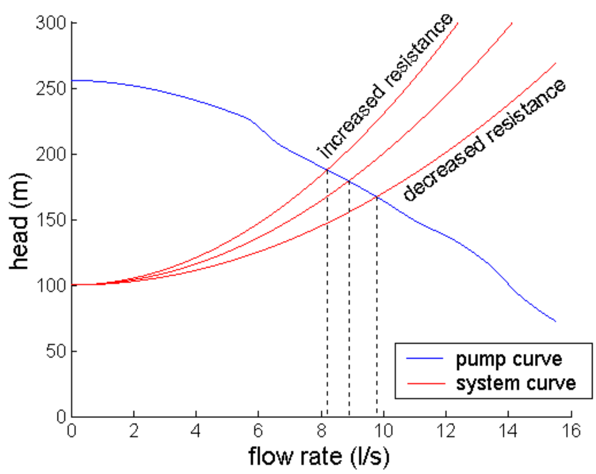
Each of these actions can be used to alter the pump flow rate achieved. The effect that each of these methods has on the pump and system curves is shown, graphically, in figure 2.6.



a. *changing pump or adding/removing stages*



b. *varying pump speed (trimming impellers has a similar effect to reducing pump speed)*



c. *varying system resistance*

Figure 2.6 *methods of altering pump flow rate*

The method that is used to vary the flow rate or head generated depends upon the particular requirements of the given pumping system. For example, installing a larger pump or a smaller pump provides only a crude new operating point, and does not allow any other setting to be achieved, other than the new operating point. Trimming impellers to reduce flow rate has more accuracy for setting the operating point, because of the varying degrees to which the impellers can be trimmed, but again this method does not allow a different operating point to be set when it is required. Changing the speed of the pump provides far more accuracy for setting the operating point. Similarly, so too does using a throttling valve. These last two methods also allow precise control over the operating point. Speed variation and throttling allow for a range of flow rates to be set, and for the set-point to be maintained, even under conditions of disturbances in the system resistance or pump characteristics. As figure 2.6c shows, throttling allows a flow rate to be set by changing the system curve. Conversely, as figure 2.6b shows, adjusting the pump speed allows a flow rate to be set by changing the pump curve.

2.2.1.4 The Affinity Laws

When concerned with speed adjustment of a centrifugal load, a group of relationships known as 'The Affinity Laws' provide useful information. These are:

$$Q \propto N$$
$$H \propto N^2$$
$$P \propto N^3$$

where Q is pump flow rate, H is head generated, P is power absorbed and N is pump speed.

The above relationships are only true when excluding overall system efficiencies, and assuming zero static system resistance.

The relationship, $\frac{H_1}{H_2} = \left(\frac{N_1}{N_2}\right)^2$, can be used to determine the pump curve for any pump speed, given the characteristic pump curve at a known speed (as figure 2.6b illustrates).

Chapter 3. Variable Speed Drives

Speed adjustment of a centrifugal pump can be achieved either mechanically or electrically.

3.1 Mechanical Variable Speed Drives

Mechanical variable speed drives, such as variable ratio belts, variable ratio chains, variable ratio friction drives and variable ratio gearboxes, can provide speed control over various centrifugal pump processes. Such mechanical drive systems require the motor, itself, to operate at full speed and, as such, are subjected to large transmission losses between motor and pump. Typically, these mechanical drives provide a transmission efficiency in the range 75% - 80%. Mechanical variable speed drives require much maintenance, such as lubrication and fine adjustment, in order to maintain the accuracy of speed settings.

A problem with many pumps, in respect to mechanical variable speed drives, is that the drives cannot easily be fitted to the system due to the self-contained construction of many motor-pump units.

3.2 Electrical Variable Speed Drives

Electrical variable speed drives function by converting the electrical signal supplied to the motor into a form which will vary the speed of rotation of the motor itself.

3.2.1 Induction Motors

The majority of centrifugal pumps are powered by induction motors. It is such pumps which will be considered hereafter. The rotational speed of an induction motor is characterised by the following equation:

rotational speed, $N = \frac{120.f}{p} \quad \left({}^{revs}\!/\!_{s}\right)$ **(equn. 3.1)**

where f is the frequency of the a.c. supply current (Hz) and p is the No. of poles

So, for any given number of poles the speed of an induction motor is proportional to the supply frequency.

3.2.2 Frequency Inverters

The only method of electrically adjusting the speed of an induction motor is to use a 'frequency inverter' - a power electronics device which converts the frequency of the supply signal to a signal of specified frequency. Figure 3.1 shows the main components of a frequency inverter.

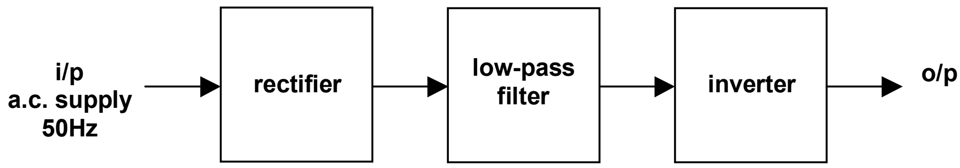


Figure 3.1 *system components of a frequency inverter*

The rectifier has the function of converting the a.c. input signal to a d.c. signal. The low-pass filter is used to filter out any a.c. components that are still present in the waveform. The lower the harmonic content of the waveform at this stage, the less harmonic distortion there will be at the output of the inverter. The inverter performs the function of converting the d.c. signal to the required frequency of a.c. signal - to output to the motor. There are three main different types of inverter section which a frequency inverter may incorporate. These are: current source inverter (CSI); variable voltage inverter (VVI); and pulse-width-modulation (PWM) inverter. The most common inverter type used with centrifugal pumps is the PWM inverter. The characteristics of a PWM inverter will, therefore, be described here. The PWM inverter performs high frequency switching (commonly between 3kHz and 12kHz) of the d.c. waveform to produce pulses of varying duration (or width), with positive voltage for half of a cycle and negative voltage for the other half of the cycle. The width of each pulse is proportional to the voltage of the output waveform. The switching frequency determines the resolution of the PWM waveform. A higher switching frequency will provide higher resolution of the PWM waveform, and so, will give a closer approximation to a pure sine wave at the inverter output, but will also decrease the efficiency of the drive due to increased heat loss in the drive's power components. Figure 3.2 illustrates an example of the waveforms present at various stages in a frequency inverter.

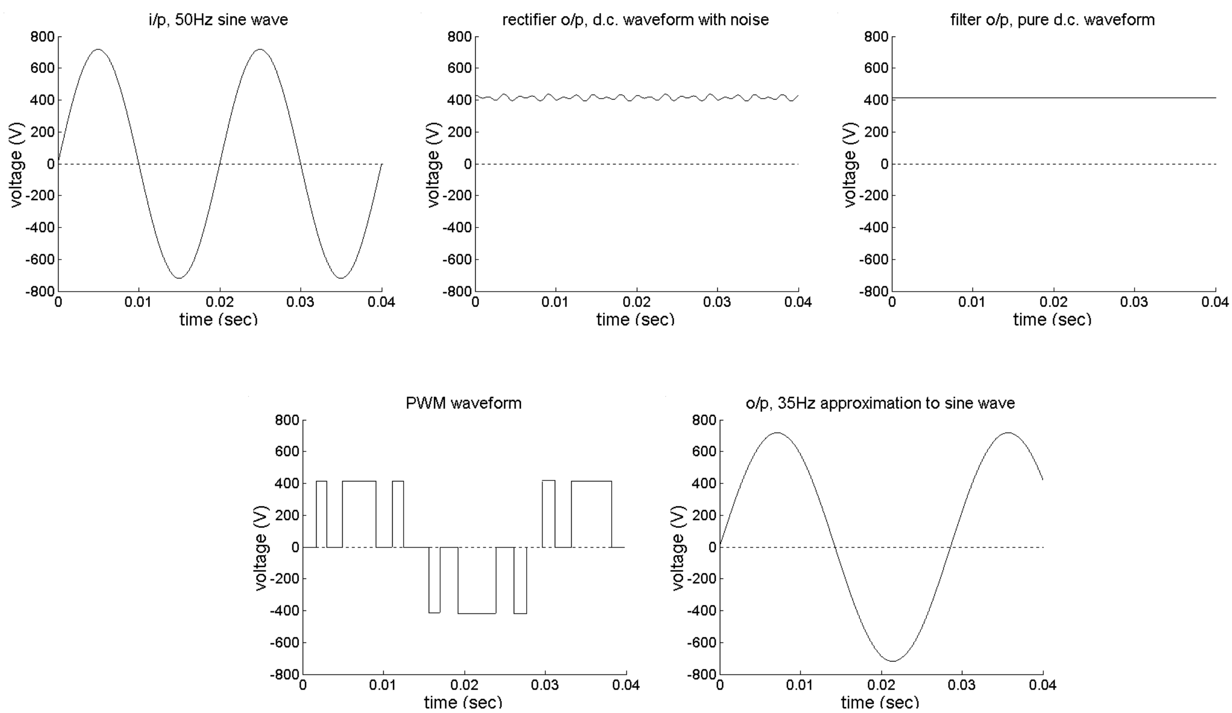


Figure 3.2 *characteristic waveforms between i/p and o/p of a frequency inverter*
(The PWM waveform is shown at a greatly reduced switching frequency to illustrate the concept.)

Since frequency inverters are the only type of electrical variable speed drive dealt with hereafter in this text, a frequency inverter is hereafter referred to, exclusively, as a variable speed drive (VSD).

3.2.3 VSD Efficiency

The typical efficiency of a VSD is approximately 95%, for loads between 30% and 100% of full load [1]. In this region the efficiency is very favourable when compared to the 75% - 80% efficiency achieved by mechanical variable speed drives. Below about 30% load the efficiency drops off sharply. For this reason it is not energy efficient to operate a VSD at the lower end of its speed range.

3.2.4 VSD Control Loops

VSD's can be used in either open-loop or closed-loop systems. In open-loop configuration motor speed can be set by specifying the output frequency of the drive. In closed-loop configuration various system parameters can be set. In water pumping applications pressure, flow rate and water level could each be used as reference set-points. A closed-loop system would incorporate sensors, which would measure the desired parameter, and return, via a transducer, the actual value of that parameter to the drive. The drive would compare this value to the reference value, and adjust the output in order to reach the reference set-point. Many VSD's facilitate proportional-integral-derivative (PID) control loops. Such control loops allow a high degree of accuracy in maintaining the reference parameter setting. The controller output can be described by an equation of the form [2]:

$$o/p = K.e + \int_T^t e.dt + C.\frac{de}{dt} \qquad \textbf{(equn. 3.2)}$$

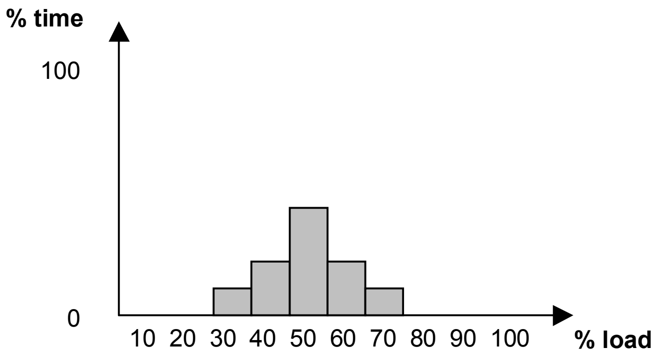
where e is the error (*ref. - feedback*), K is the proportional constant, T is the integral action time, t is time and C is the derivative constant

Each element of the PID controller performs a particular function with respect to the system's response time in maintaining the reference set-point. The proportional term gives an output that is proportional to the error signal. The integral term has the effect of reducing the offset from the desired reference value. The derivative term is used to minimize the overshoot, by reducing the oscillations. For a further explanation of PID controllers the reader is referred to Searle [2].

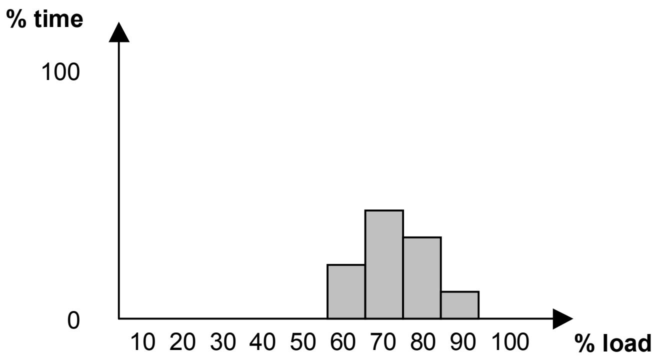
Chapter 4. Flow Control

4.1 Load Duty Cycle

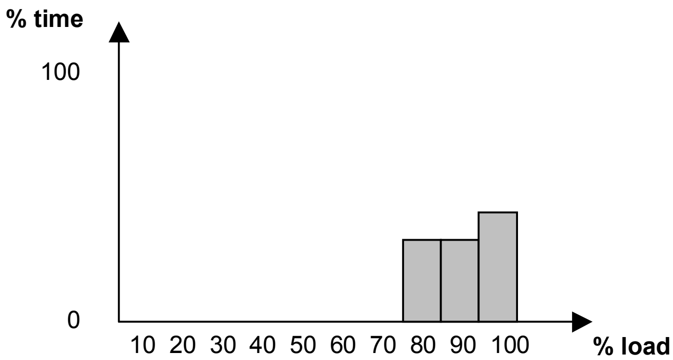
The extent of energy savings achievable through the use of variable speed control in place of throttling control is dependent upon the specific operational requirements of the pump under consideration. The load duty cycle of a pump provides essential information regarding energy savings. This describes the proportion of time for which the pump is operating, and at what proportion of full load the pump is operating as a function of percentage of time. Examples of various load duty cycles are shown in figure 4.1.



a. *large deviation from full load*



b. *medium deviation from full load*



c. *small deviation from full load*

Figure 4.1 *examples of load duty cycle*

4.2 Variable Flow Rate Control

Where these load duty cycles refer to varying degrees of flow rate, as determined by changes in demand, the pump operating points would be determined by the method of flow control used.

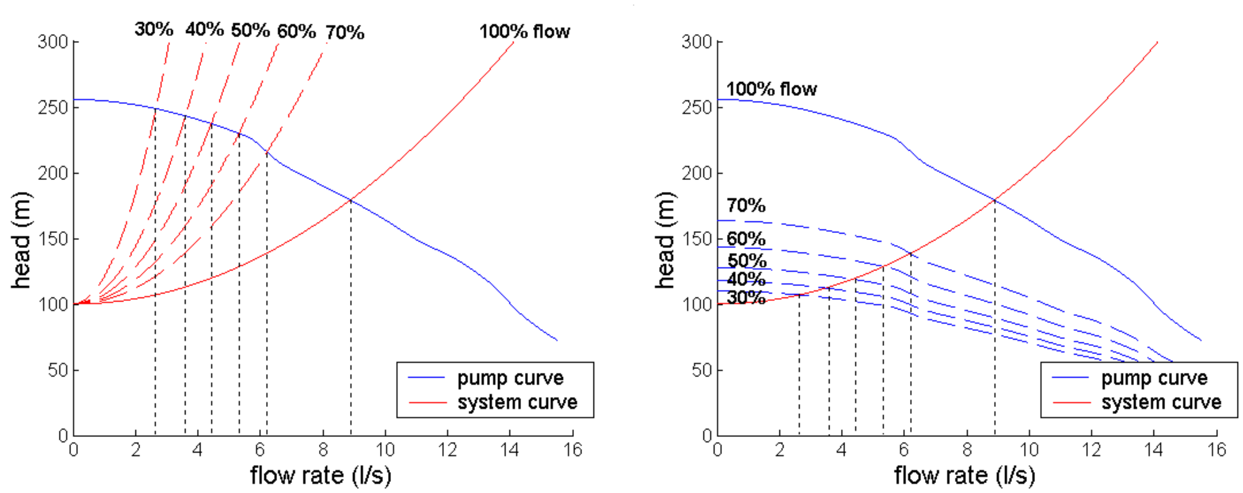
A comparison of these three load duty cycles, for such a case, is given in the example below. The example compares the two flow control methods of throttling and speed variation, in terms of power consumption, and resulting energy usage. The period of operation is taken to be one year.

In tables 4.1, 4.2 and 4.3, Q is pump flow rate, H is head generated, P is power absorbed and E is energy used. The subscripts "$_{PSV}$" and "$_{VSD}$" denote values referring to throttling control and variable speed control, respectively.

CASE A

% flow	30%	40%	50%	60%	70%
% time	10%	20%	40%	20%	10%
Q (l/s)	2.7	3.6	4.4	5.3	6.2
H_{PSV} (m)	249	244	238	230	216
H_{VSD} (m)	107	112	119	129	138
P_{PSV} (kW)	6.6	8.6	10.3	12.0	13.1
P_{VSD} (kW)	2.8	4.0	5.1	6.7	8.4
E_{PSV} (kWh)	5777	15097	35997	20951	11508
E_{VSD} (kWh)	2483	6930	17998	11751	7353

Table 4.1



a. *throttling control* ***b.*** *variable speed control*

Figure 4.2 *case A: operating points for variable flow control*

Total energy used in one year with throttling control is $89331 kWh$.

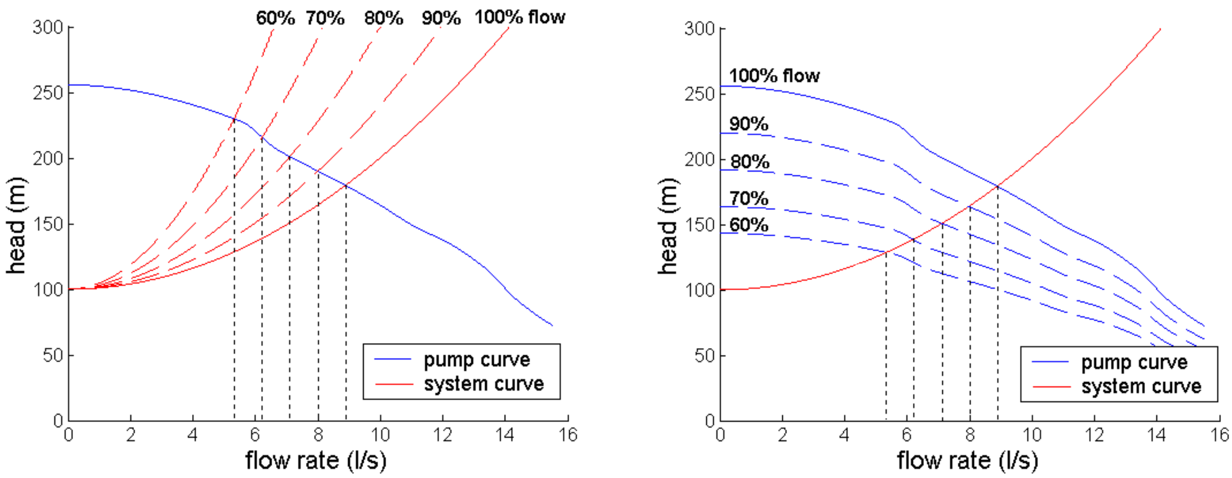
Total energy used in one year with variable speed control is $46514 kWh$.

Energy saving of 48% with variable speed control.

CASE B

% flow	60%	70%	80%	90%
% time	20%	40%	30%	10%
Q (l/s)	5.3	6.2	7.1	8
H_{PSV} (m)	230	216	201	190
H_{VSD} (m)	129	138	150	164
P_{PSV} (kW)	12.0	13.1	14.0	14.9
P_{VSD} (kW)	6.7	8.4	10.4	12.9
E_{PSV} (kWh)	20951	46034	36792	13062
E_{VSD} (kWh)	11751	29411	27456	11275

Table 4.2



a. *throttling control* ***b.*** *variable speed control*

Figure 4.3 *case B: operating points for variable flow control*

Total energy used in one year with throttling control is 116839kWh.

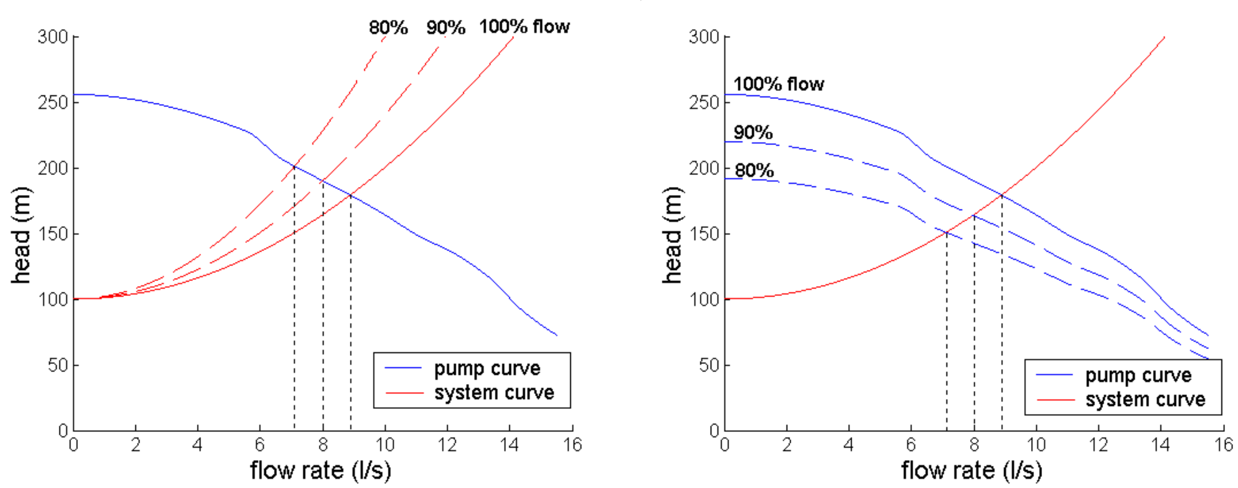
Total energy used in one year with variable speed control is 79893kWh.

Energy saving of 32% with variable speed control.

CASE C

% flow	80%	90%	100%
% time	30%	30%	40%
Q (l/s)	7.1	8	8.9
H_{PSV} (m)	201	190	179
H_{VSD} (m)	150	164	179
P_{PSV} (kW)	14.0	14.9	15.6
P_{VSD} (kW)	10.4	12.9	15.6
E_{PSV} (kWh)	36792	39187	54762
E_{VSD} (kWh)	27456	33824	54762

Table 4.3



a. *throttling control* ***b.*** *variable speed control*

Figure 4.4 *case C: operating points for variable flow control*

Total energy used in one year with throttling control is 130740kWh.

Total energy used in one year with variable speed control is 116042kWh.

Energy saving of 11% with variable speed control.

From this example it can be seen that greater energy savings can be made when there is a high concentration of operating time at greatly reduced flow rates.

4.3 Efficiency Variation with Variable Speed Control

There is, however, a limitation to this theory. In the above example, system efficiencies have not been included.

As stated in chapter 3.2.3, VSD efficiency, η_{VSD}, greatly reduces at very low speeds. Induction motor efficiency, η_m, typically remains fairly constant at around 90% for loads between 45% and 100% of full load, and significantly reduces at loads below 45% of full load [3]. So, energy savings will not be so significant at low speeds, and it will, perhaps, be less energy efficient to operate a VSD rather than operating a throttling valve in this region.

Data on pump efficiency, η_P, is normally provided with pump characteristic curves in the form of efficiency variation against flow rate for rated pump speed (see figure 2.4). This is sufficient information for determining pump efficiency in the case of throttling. However, there is a different pump efficiency curve for any given pump speed. The pump efficiency at a given pump speed and flow rate can be found through iso-efficiency lines of the form, $H = k.Q^2$. The value of k can be found by plotting the pump curve for the given pump speed (see chapter 2.2.1.4), and finding the values of H and Q at the operating point - then $k = \frac{H}{Q^2}$. Next, the intersection of $H = k.Q^2$ with the pump curve for rated speed can be found, giving a flow rate of equivalent efficiency for rated speed. Then, since $H = k.Q^2$ is an iso-efficiency line, the required efficiency value can be found by reading the efficiency curve (for rated pump speed) for this flow rate. Figure 4.5 shows efficiency curves for various pump speeds, taken by calculating several points and interpolating to give continuous curves.

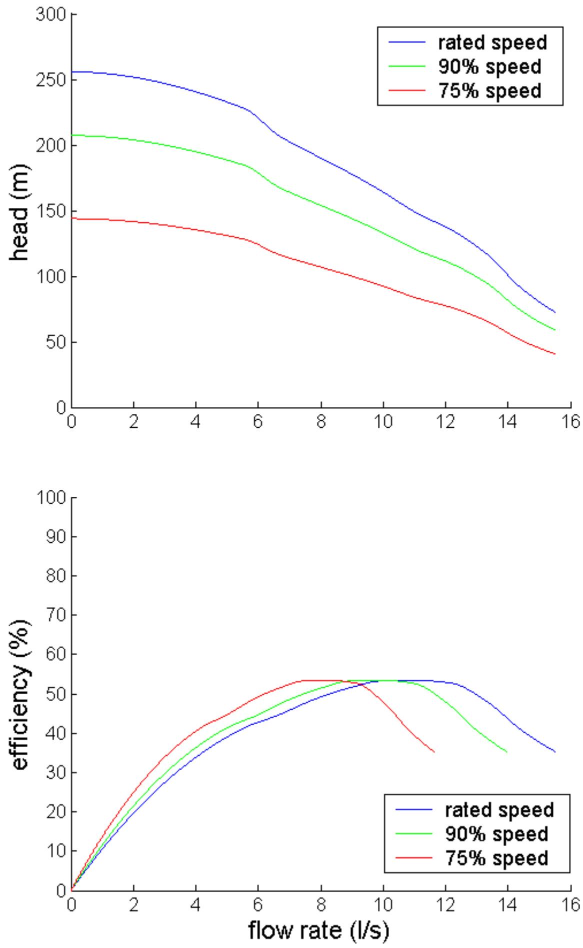


Figure 4.5 *pump efficiency curves at various pump speeds*

4.4 Effect of the Characteristics of the System Curve

When looking at the relative power savings between the two flow control methods it should be noted that the shape of the system curve contributes significantly to the result. When the system static head is zero, using variable speed control results in a power reduction which varies in direct proportion to the cube of the reduction in speed (as stated in chapter 2.2.1.4). The steepness of the system curve also affects the relative level of power saving. When using variable speed control, a steeper system curve (i.e. a higher value of k in equation 2.3) gives a greater level of relative power reduction.

4.5 Constant Flow Rate Control

So far this chapter has described systems requiring variable flow rates. Another scenario in which flow control of a pump may be employed is where a flow rate must be maintained in response to changes in system pressure requirements. A set flow rate could be maintained by throttling the pump discharge to a hold certain pressure under the conditions of a changing system pressure downstream. This form of control is described by a constant system curve and a constant pump curve. The alternative to this, using variable speed control, is to change the pump speed in response to changes in system pressure, so as to hold the desired flow rate. Figure 4.6 shows the characteristic curves of this form of variable speed control.

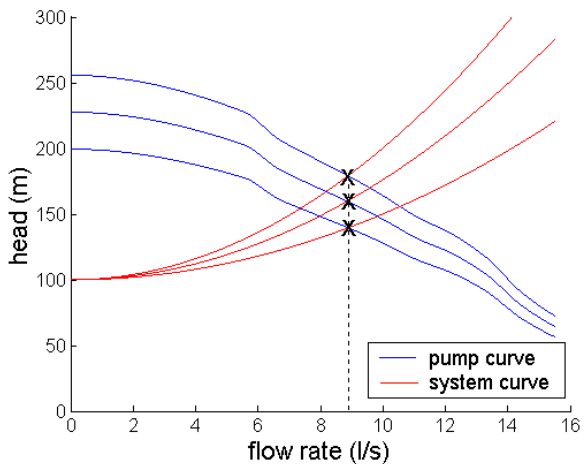


Figure 4.6 *characteristic curves for variable speed, constant flow rate control under conditions of variable system resistance*

For a system with these requirements the load duty cycles shown in figure 4.1 could describe varying degrees of system pressure (or head) requirements.

Chapter 5. Water Pumping Applications

Water pumping is used throughout a range of industries, for various purposes. This chapter looks at different applications of water pumps where VSD's have been implemented, and highlights the energy savings achieved in each system. Two cases are outlined below.

5.1 Case 1 - Water Distribution System

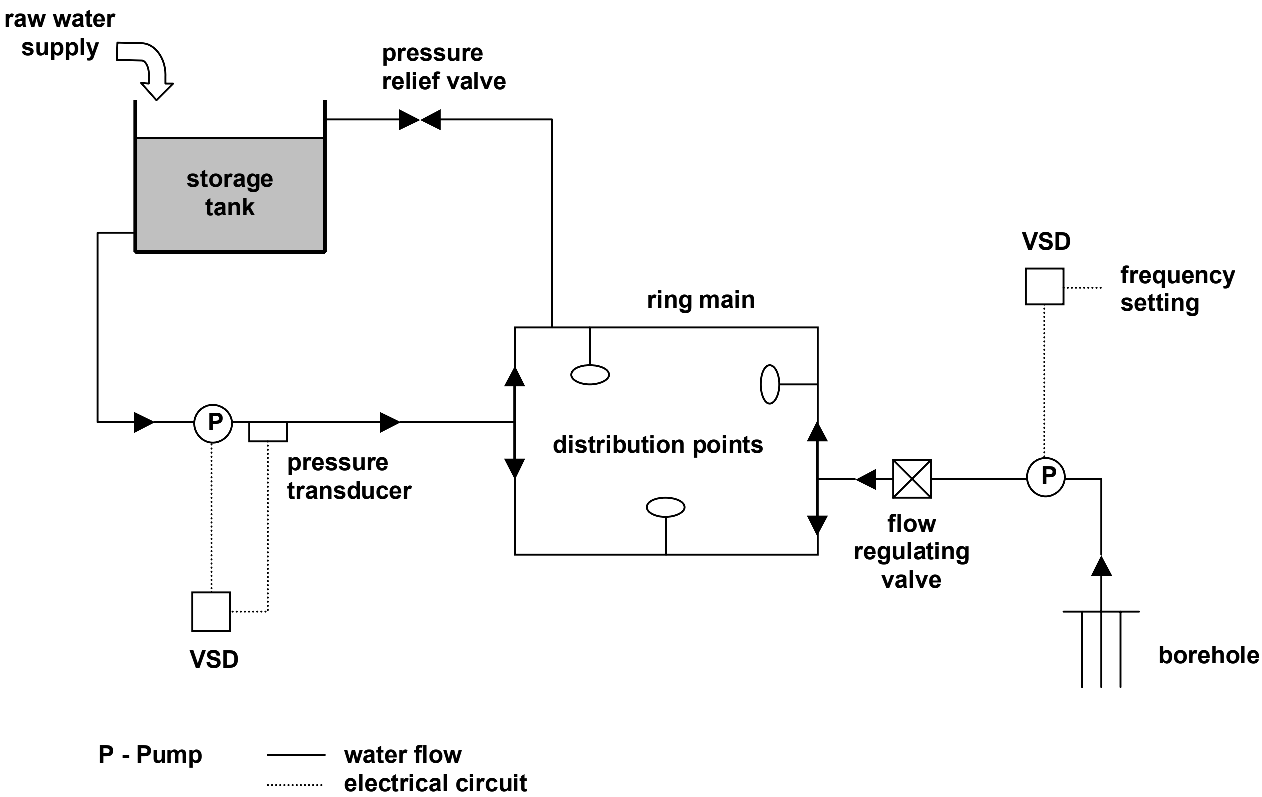


Figure 5.1 *schematic of water distribution system*

The requirements of this system are twofold:

1) the borehole water extraction rate must be kept constant, at a predetermined flow rate, according to the water company's extraction contract

2) a set pressure level in the ring main must be maintained

The original operation of the plant incorporated mechanical control devices. The borehole pump discharge was throttled by a flow regulating valve in order to maintain a constant extraction rate. The raw water pump was run at full load, and pressure in the ring main was maintained by a pressure relief valve, which diverted excess water back into the storage tank.

The plant was retrofitted with two VSD's. One was used to control the borehole pump flow rate - the flow regulating valve was fully opened - and the desired flow rate maintained by adjusting the speed of the pump. Since no pressure changes occur downstream, it was sufficient to operate the borehole VSD in open-loop control. The other VSD was fitted to the raw water pump in a closed loop configuration, with a pressure transducer providing a feedback signal, representing the measured pressure at the pump discharge line. The control loop allowed the pressure relief valve to be fully closed, and ring main pressure to be maintained by automatic adjustment of the pump speed.

This example highlights two possible uses of VSD's in water pumping applications:

1) open-loop control to set pump flow rate
2) closed-loop control to hold a pressure set-point in response to downstream pressure changes

After installation of the VSD's energy savings of $29 \cdot 7\%$ for the borehole pump, and $88 \cdot 2\%$ for the raw water pump, were recorded. It was calculated that the installation of the borehole VSD would result in a simple payback of $1 \cdot 7$ years, and that the raw water VSD would give a simple payback of 10 months [4].

5.2 Case 2 - Fresh Water Pumping System

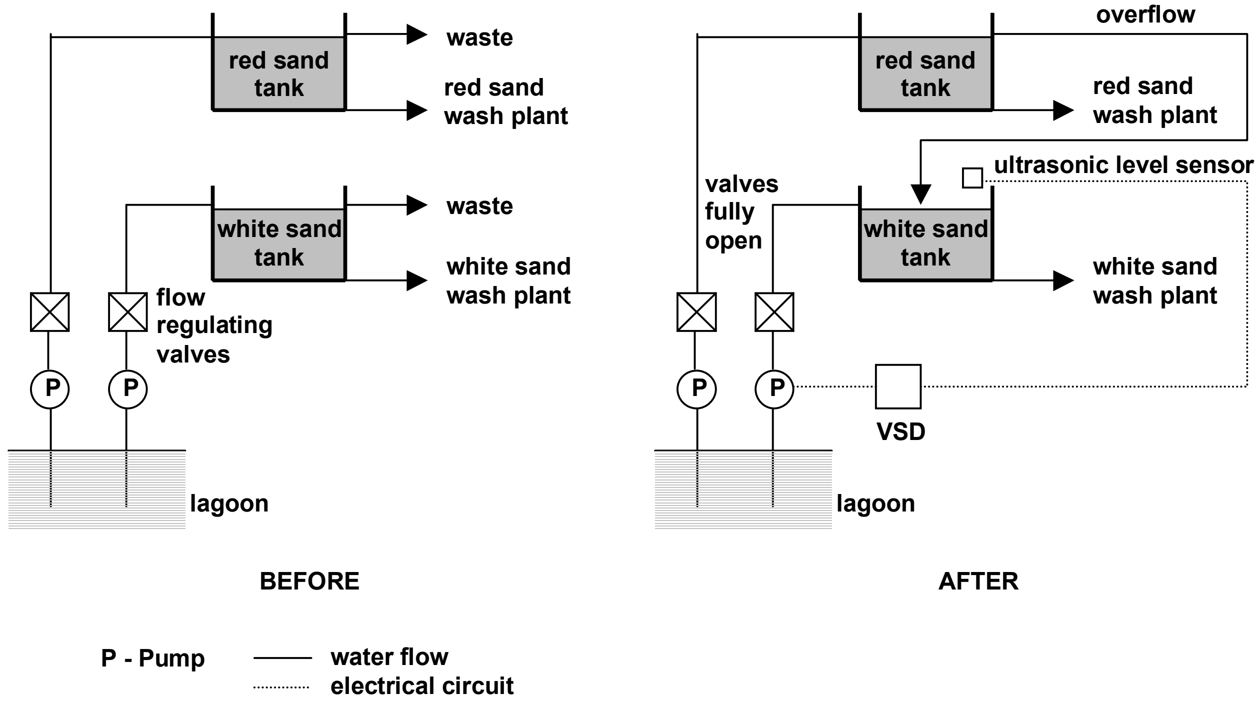


Figure 5.2 *schematic of fresh water pumping plant*

In this system water is required to be extracted from the lagoon, stored in two tanks, and supplied to the respective sand wash plants.

The original system incorporated two manually-operated pumps - one supplying the red sand storage tank, the other supplying the white sand storage tank. Flow rates were adjusted, according to demand, by flow regulating valves positioned at each pump discharge line.

The following alterations were made to the system:

- the flow regulating valves were fully opened
- the red sand tank overflow line was channelled into the white sand tank

- an ultrasonic level sensor was fitted in the white sand tank, and a VSD fitted to the white sand pump

With the new configuration no energy was wasted in throttling either pump. Any excess water in the red sand tank was channelled into the white sand tank. The ultrasonic level sensor provided a feedback signal to the VSD, which regulated the speed, and hence flow rate, of the white sand pump - to provide the required volume of water to the white sand tank.

In this system the VSD is operating in a closed-loop configuration, with liquid level as the reference parameter.

This case shows an example of the use a VSD for meeting changing flow rate requirements in response to changes in demand.

The installation of the VSD, along with the replacement of the white sand pump (with a more appropriately sized one for the new configuration) was calculated to give a simple payback of $2 \cdot 5$ years [5].

Chapter 6. Case Study: Spey Wellfields Water Pumping Station

6.1 Introduction

A scenario which has not yet been fully analysed is the use of a VSD to maintain a set flow rate in response to changes in downstream pressure. This concept was introduced in chapter 4.5. The justification for the use of a VSD in such a case is dependent upon the magnitude of the downstream pressure variations. If the pressure changes were to cause only small deviations from the desired flow rate, it is unlikely that the use of a VSD would be financially viable. The initial cost of a throttling valve is much less than that of a VSD, and the relatively small energy savings made with a VSD would be far outweighed by the relatively large cost of the VSD. So, in certain cases, throttling control would be the preferred option in financial terms.

The focus of chapter 6 is a study into the viability of using a VSD to maintain a constant flow rate in a water pumping system, in which the downstream pressure varies. The system which is examined, here, is a raw water pumping station, consisting of 36 borehole pumps, each controlled at a constant flow rate by a throttling valve. This study looks at the possibility of replacing the throttling valves with VSD's as the flow control method, with emphasis on minimising energy consumption within certain economic criteria.

6.2 Background

The pumping station under investigation, here, is a site operated by Scottish Water. It is situated on the banks of the River Spey in North East Scotland, grid reference: NJ 33 57. The site consists of 36 borehole pumps, each extracting water from flood plain alluvium. The boreholes are mainly supplied by river water, but also partly by groundwater from west of the wellfield. The boreholes are between 12m and 23m deep, and are situated at a minimum distance of 50 metres from the river's edge. The water is naturally filtered through the alluvial gravels, so only a small amount of water treatment is required after pumping. Each borehole is equipped with a submersible borehole pump. Each pump feeds into one of two collector mains. The collector mains feed into one of two transmission mains, which carry the water for 6 kilometres up a height of 115m to a storage tank at Badentinan Water Treatment Works. Figure 6.1 shows the configuration of the site.

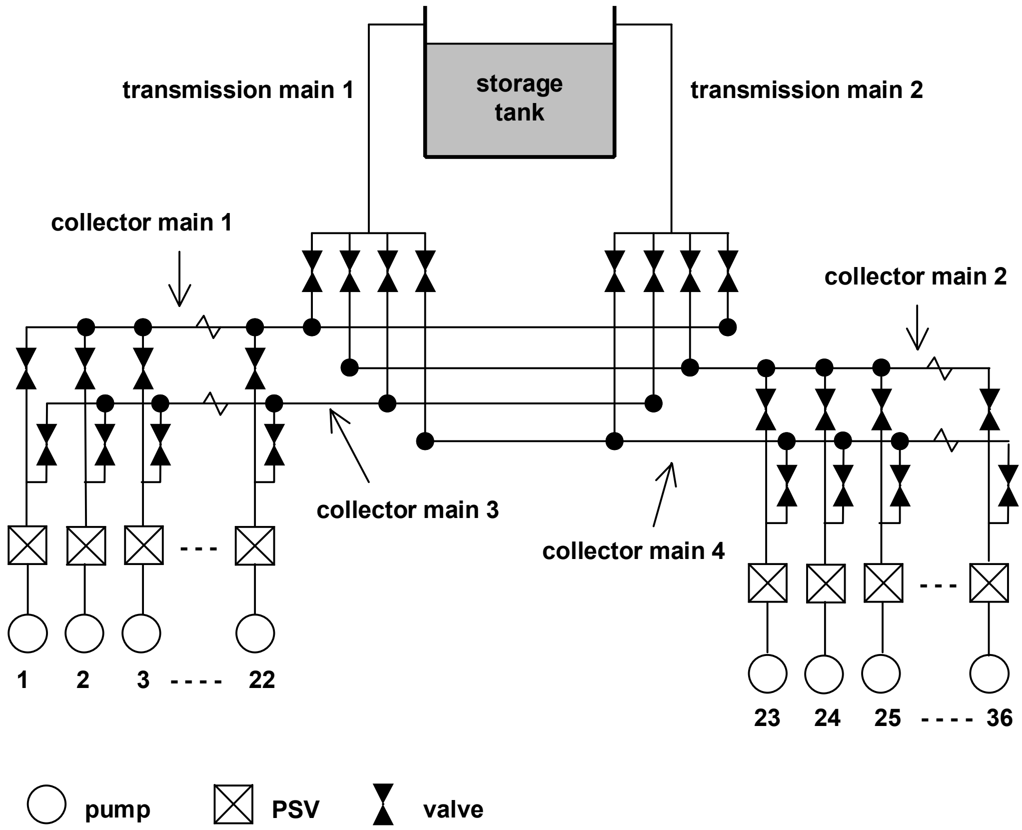


Figure 6.1 *schematic of Spey Wellfields Water Pumping Station*

6.3 Pump Models

The pumps installed at the plant are submersible borehole pumps manufactured by Grundfos. Several sizes of pump are used, with varying numbers of stages, taken from the Grundfos SP45 and SP27 ranges. The pumps are all multi-stage, ranging from 19 to 25 stages. With these pumps the motor is mounted below the pump, and is close coupled to the pump. The electricity supply is a dedicated 3-phase supply line, with each phase at $415V$, $50Hz$. The SP27 pumps are fitted with 6” Grundfos 3-phase induction motors rated at $18 \cdot 5kW$, with a full load current of $39 \cdot 5A$. The SP45 pumps are fitted with 8” Franklin 3-phase induction motors rated at $30kW$, with a full load current of $62 \cdot 0A$. All motors are connected in delta configuration. Due to the close coupling of the motor and pump, any form of mechanical speed control is practically unsuitable with these pumps.

Factory test data, providing performance characteristics, relating flow rate, head generated and wire-to-water efficiency, for each pump is reproduced in table 6.1. The values given for efficiency include motor efficiency, η_m, and pump efficiency, η_P.

	Q (l/s)	H (m)	η (%)
SP27-21			
	0.00	247.4	0.00
	1.11	240.6	13.15
	2.22	N/A	24.30
	3.89	207.6	37.91
	5.56	185.0	48.43
	6.67	160.9	51.86
	7.78	134.3	51.88
	8.89	110.6	48.46
	9.44	93.5	42.26
	10.00	78.0	37.80
SP27-23			
	0.00	247.5	0.00
	1.11	244.3	12.14
	2.22	233.8	22.91
	3.89	220.2	26.94
	5.56	201.2	47.78
	6.67	180.7	54.19
	7.78	153.0	56.78
	8.89	124.4	56.76
	9.44	110.0	52.26
	10.00	91.7	48.03
SP45-19			
	0.00	232.2	0.00
	5.56	204.9	40.46
	6.67	188.9	44.28
	7.78	174.9	44.90
	8.89	162.1	50.72
	10.00	149.9	53.01
	11.11	133.7	52.88
	12.22	120.9	51.66
	13.33	107.3	48.45
	14.44	88.1	42.17
	15.56	68.3	36.65

	Q (l/s)	H (m)	η (%)
SP45-21			
	0.00	255.8	0.00
	5.56	228.0	41.30
	6.67	207.3	44.61
	7.78	193.0	48.49
	8.89	179.3	51.40
	10.00	164.4	53.30
	11.11	147.8	53.42
	12.22	135.0	52.66
	13.33	116.9	48.05
	14.44	90.7	40.80
	15.56	72.2	35.20
SP45-24			
	0.00	282.6	0.00
	5.56	256.3	41.52
	6.67	234.2	45.08
	7.78	219.2	49.25
	8.89	205.1	52.68
	10.00	192.6	56.08
	11.11	173.3	56.24
	12.22	151.2	53.23
	13.33	133.3	49.48
	14.44	105.8	41.49
	15.56	76.8	35.42
SP45-25			
	0.00	299.0	0.00
	5.56	265.7	41.14
	6.67	242.5	45.42
	7.78	226.6	49.64
	8.89	211.8	53.01
	10.00	194.0	54.99
	11.11	173.4	54.94
	12.22	157.6	52.29
	13.33	133.9	48.46
	14.44	111.3	42.62
	15.56	52.4	41.25

Table 6.1 *pump performance characteristic data (reproduced from factory test data)*

Figure 6.2 shows plots of the data given in table 6.1, with interpolation of the data points. These curves can be used to determine pump operating ranges, and to determine the relationship between head generated, flow rate and efficiency for any point within the operating range.

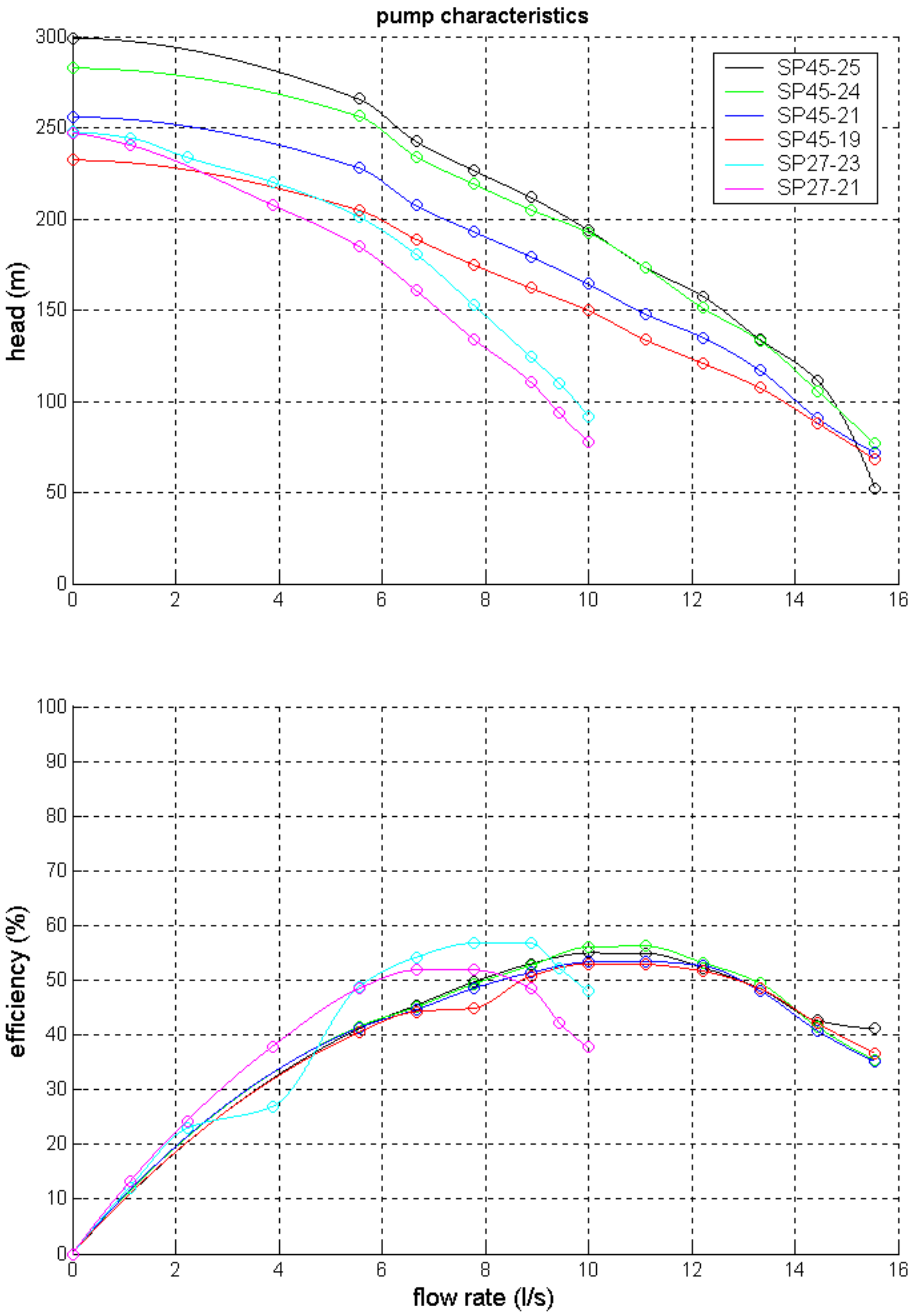


Figure 6.2 *Grundfos SP45 and SP27 pump characteristics*

6.4 Nominal Flow Rates

During the initial design stages, tests were carried out on each borehole to determine the maximum extraction rate which could be sustained whilst maintaining sufficient water levels in each borehole well. The pumps were selected, and the system designed, such that any pump operating could extract water at the required sustainable rate for that well. The maximum flow rate that each pump can sustain will be referred to as its 'nominal flow rate', $Q_{no\min al}$ (see table 6.2 for a list of nominal flow rates).

The nominal flow rate is required to be maintained in response to any changes in system resistance. The most significant fluctuation in system resistance occurs as a result of the number of pumps operating through any given collector main or transmission main. The more pumps that are operating through a main, the greater the resistance seen by any pump connected to that main.

The piping system, which each pump feeds into, consists of two collector mains for pumps on the south bank, and two collector mains for pumps on the north bank (as shown in figure 6.1). Pumps 1 - 22 (on the south bank) can be connected to either collector main 1 or collector main 2. Similarly, pumps 23 - 36 (on the north bank) can be connected to either collector main 3 or collector main 4. At the interconnection between the collector mains and transmission mains, a set of valves allow the connection of any collector main to either transmission main. So, any configuration of collector main and transmission main is possible - the only limitation being that pumps on the south bank can only be connected to collector mains 1 and 2, and pumps on the north bank can only be connected to collector mains 3 and 4. The transmission mains feed into the top of the storage tank - above the water level - so the pumps are not working against any pressure changes due to water level in the storage tank.

6.5 Water Quality Classifications

Each well provides water with different concentrations of iron and manganese, which constitute different levels of water quality. Classifications exist for various degrees of water quality. The classifications are A1, A2, B1 and B2, in order of quality - highest to lowest. Table 6.2 lists the water quality classifications for each well.

6.6 Pressure Sustaining Valves

The system was originally designed to have pressure sustaining valves (PSV's) fitted to the discharge line of each pump. The PSV's perform the function of throttling the pump discharge in order to maintain a constant head seen by each pump, which corresponds to the nominal flow rate on the respective pump curve. The amount of throttling is automatically adjusted in response to changes in downstream pressure - when pumps are switched on or switched off - to maintain a constant pressure at the pump discharge line and, hence, maintain constant flow rate.

Pumps were selected for each borehole on the basis of the ability of the pumps to meet nominal flow rates under conditions of maximum expected system resistance, so that the nominal flow rates could be achieved under any normal operating conditions. Table 6.2 lists the pump type selected for each borehole, and the pressure setting for each PSV that gives the nominal flow rate for each pump (based on the pump characteristic curves shown in figure 6.2).

pump No.	pump type	water quality	$Q_{nominal}$ (l/s)	PSV setting (m)
1	SP45-25	A1	11.75	164.5
2	SP45-25	A2	12.00	N/A
3	SP45-24	A1	12.00	155.4
4	SP45-19	A1	9.20	158.7
5	SP45-19	A1	10.00	149.9
6	SP27-21	A1	7.20	148.1
7	SP27-21	A2	7.16	149.1
8	SP27-23	A1	7.00	173.0
9	SP45-24	A1	11.30	169.7
10	SP45-25	A1	12.00	161.1
11	SP45-19	A1	9.30	157.7
12	SP45-25	A1	12.20	158.0
13	SP45-25	A1	12.10	159.6
14	SP45-21	A1	10.70	153.7
15	SP45-25	A1	12.20	158.0
16	SP45-25	A1	12.10	159.6
17	SP45-21	A2	10.69	153.9
18	SP45-25	A1	12.50	152.3
19	SP27-23	A1	7.13	169.8
20	SP27-23	A1	7.50	160.2
21	SP45-25	A1	12.20	158.0
22	SP45-19	A1	9.20	158.7
23	SP45-25	A1	12.50	152.3
24	SP45-25	A1	12.20	158.0
25	SP45-25	A1	12.10	159.6
26	SP45-25	B2	12.20	158.0
27	SP45-24	A1	11.70	161.5
28	SP45-25	A2	12.00	161.1
29	SP45-25	B2	11.90	162.5
30	SP45-21	B1	10.70	153.7
31	SP27-21	A2	7.14	149.6
32	SP27-23	A1	7.13	169.8
33	SP45-21	A1	10.71	153.6
34	SP45-21	A1	10.54	156.2
35	SP27-23	A1	7.13	169.8
36	SP27-23	N/A	7.00	N/A

Table 6.2 *pump types, water quality, nominal flow rates and PSV settings*

6.7 System Head Losses

In order to calculate the amount of the throttling employed on each pump discharge line, and the potential energy reductions resulting from variable speed operation of the pumps, the range of system resistance seen by each pump with the PSV's fully open must be analysed. This involves modelling the head losses incurred in the borehole piping, the collector mains and the transmission mains. As stated in chapter 6.4 the resistance seen by any one pump varies depending on the number of pumps operating through the same collector main or transmission main as that pump. The analysis in chapter 6.7 looks at the resistance seen by each pump when the PSV's are fully open. In the original design of the system, calculations were made describing head losses in each borehole, collector main and transmission main under certain flow conditions. The analysis describes the situation when all pumps are operating (except pumps No.2 and No.36) with flow rates and mains configuration as given in tables 6.3 and 6.4. All calculations in this chapter are made on the assumption that pump and collector main connections are as shown in table 6.3.

pump No.	collector main	Q (l/s)	H_{D_B} (m)	H_{D_C} (m)	H_{D_T} (m)	H_S (m)
1	2	9.7	2.0	1.4	11.3	125
2	2	13.0	N/A	N/A	N/A	123
3	1	11.0	1.6	1.6	12.4	122
4	1	8.8	1.0	1.5	12.4	123
5	2	7.6	1.0	0.9	11.3	123
6	2	7.1	0.7	0.9	11.3	125
7	2	6.0	0.7	0.8	11.3	124
8	2	7.0	0.7	0.7	11.3	122
9	1	12.0	2.1	1.2	12.4	124
10	1	13.0	2.6	1.1	12.4	123
11	2	6.6	0.7	1.3	11.3	123
12	2	13.0	4.6	1.3	11.3	120
13	1	13.0	4.1	1.1	12.4	121
14	1	9.9	1.4	1.1	12.4	126
15	1	12.0	2.2	1.0	12.4	125
16	1	12.0	3.8	1.0	12.4	122
17	2	9.0	1.3	0.5	11.3	127
18	2	9.6	2.0	0.4	11.3	127
19	1	4.1	0.5	0.7	12.4	127
20	1	7.2	0.8	0.6	12.4	127
21	1	10.0	1.7	0.5	12.4	126
22	2	6.4	0.8	0.1	11.3	128
23	4	13.0	2.4	0.1	11.3	125
24	4	12.0	2.5	0.2	11.3	125
25	4	11.0	3.1	0.2	11.3	128
26	3	10.0	2.3	0.2	12.4	128
27	3	7.1	1.0	0.2	12.4	125
28	3	13.0	2.0	0.3	12.4	127
29	4	9.3	1.7	1.1	11.3	131
30	4	10.0	1.9	1.1	11.3	125
31	4	2.6	0.3	0.3	11.3	126
32	3	7.5	1.1	0.3	12.4	127
33	4	6.5	1.5	0.5	11.3	126
34	3	8.3	1.7	0.4	12.4	131
35	4	6.2	0.6	0.6	11.3	128
36	3	7.3	N/A	N/A	12.4	127

Table 6.3 *original design head losses*

transmission main	Q (l/s)	H_{D_T} (m)
1	158	12.4
2	152	11.3

Table 6.4 *original design transmission main losses*

6.7.1 Static Head Losses

The static head losses for each pump are listed in table 6.3.

6.7.2 Dynamic Head Losses

When only one pump is operating through a particular collector main and transmission main, then the system resistance can be described by an equation of the form:

system resistance, $H_{SYS} = H_S + K.Q_P{}^2$ **(equn. 6.1)**

where H_S is static head, K is a constant and Q_P is the pump flow rate

The term $K.Q_P{}^2$ describes the combined dynamic losses incurred in the borehole piping, collector main and transmission main.

However, when more pumps are brought into line, the losses in the collector main and transmission main become dependent on, not only Q_P, but on the total flow rate in the collector main and in the transmission main (including Q_P).

6.7.2.1 Transmission Main Losses

Dynamic losses through the transmission mains can be described by an equation of the form:

dynamic losses in transmission main *n*, $H_{D\_T(n)} = k_{T(n)}.Q_{T(n)}{}^2$ **(equn. 6.2)**

where $k_{T(n)}$ is a constant and $Q_{T(n)}$ is the total flow rate through the transmission main

Using the values given in table 6.4, $k_{T(1)}$ and $k_{T(2)}$ can be found, as described below:

when $Q_{T(1)} = 158 \, {}^{l}\!/_{s}$, $H_{D\_T(1)} = 12.4m$

when $Q_{T(2)} = 152 \, {}^{l}\!/_{s}$, $H_{D\_T(2)} = 11.3m$

substituting into equation 6.2 gives:

$$k_{T(1)} = \frac{12 \cdot 4}{158^2}$$
$$= 0 \cdot 0004967$$

$$k_{T(2)} = \frac{11 \cdot 3}{152^2}$$
$$= 0 \cdot 0004890$$

Figure 6.3 shows plots of the dynamic losses in the transmission mains.

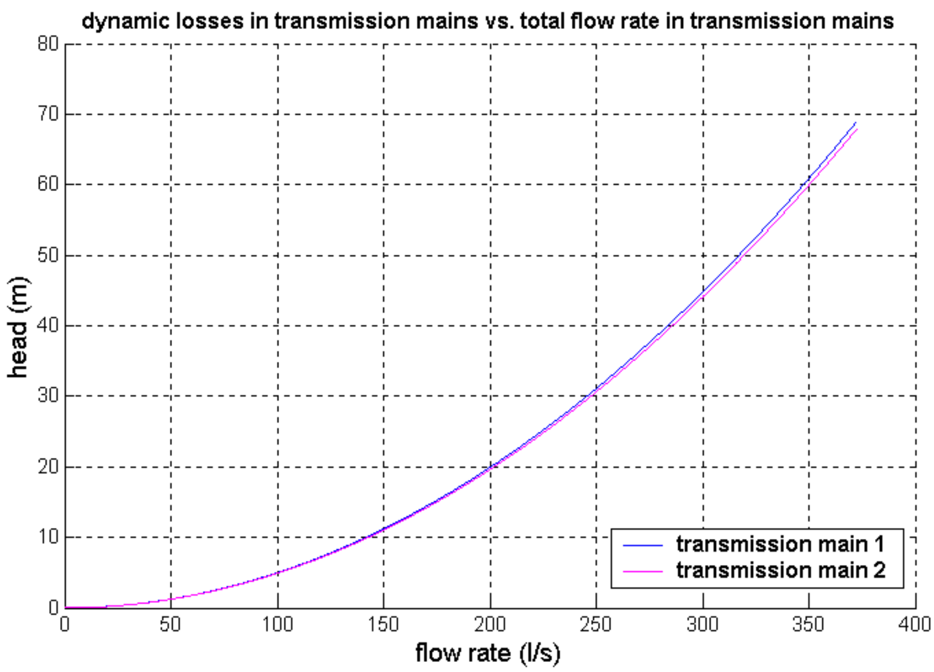


Figure 6.3 *dynamic losses in the transmission mains*

6.7.2.2 Collector Main Losses

Head losses in the collector mains, also, vary depending upon the number of pumps operating through a given collector main. Unlike the intersection of the collector mains with the transmission mains, the connections of the pump discharge lines to the collector mains do not occur at a common point. Each pump connects to a different section of the collector mains. Due to this configuration, the calculation of collector main losses as a function of total flow rate through a collector main can only be made for a given sequence of pump switching, since the output of the pumps only flow through certain sections of a collector main, and so the resistance seen by one pump depends upon the point of connection of the other pumps to the collector main, as well as the flow rates of the pumps. For the purposes of carrying out power calculations, here, collector mains head losses are assumed to be independent of the total flow rate through the collector mains, and will be assumed to vary only with pump flow rate. Thus, dynamic losses in the collector mains can be expressed in the form:

dynamic losses in a collector main, $H_{D\_C} = k_C.Q_P{}^2$ **(equn. 6.3)**

where k_C is a constant

Using the values of H_{D_C} given in table 6.3, values for k_C, found from equation 6.3, are given for each pump in table 6.5. The conditions governing these values of k_C are for 34 pumps operating, which is close to the maximum capacity. So, the values of k_C given in table 6.5 can be considered as approximately "worst case" values.

6.7.2.3 Borehole Piping Losses

Dynamic losses in the borehole piping of each pump are independent of total flow rate, since the flow from one pump does not travel through the borehole piping of any other pump. So, the losses are dependent only on the flow rate of the pump concerned. Dynamic losses in the borehole piping can be described by an equation of the form:

$$H_{D\_B} = k_B.Q_P^{\ 2} \qquad \textbf{(equn. 6.4)}$$

where k_B is a constant

Using the values of H_{D_B} given for each pump in table 6.3, values for k_B, found from equation 6.4, are given for each pump in table 6.5.

pump No.	k_B	k_C
1	0.02126	0.01488
2	N/A	N/A
3	0.01322	0.01322
4	0.01291	0.01937
5	0.01731	0.01558
6	0.01389	0.01785
7	0.01944	0.02222
8	0.01429	0.01429
9	0.01458	0.00833
10	0.01538	0.00651
11	0.01607	0.02984
12	0.02722	0.00769
13	0.02426	0.00651
14	0.01428	0.01122
15	0.01528	0.00694
16	0.02639	0.00694
17	0.01605	0.00617
18	0.02170	0.00434
19	0.02974	0.04164
20	0.01543	0.01157
21	0.01700	0.00500
22	0.01953	0.00244
23	0.01420	0.00059
24	0.01736	0.00139
25	0.02562	0.00165
26	0.02300	0.00200
27	0.01984	0.00397
28	0.01183	0.00178
29	0.01966	0.01272
30	0.01900	0.01100
31	0.04438	0.04438
32	0.01956	0.00533
33	0.03550	0.01183
34	0.02468	0.00581
35	0.01561	0.01561
36	N/A	N/A

Table 6.5 *borehole and collector main head loss coefficients*

The combined borehole piping and collector main losses for pump No.1 is plotted in figure 6.4.

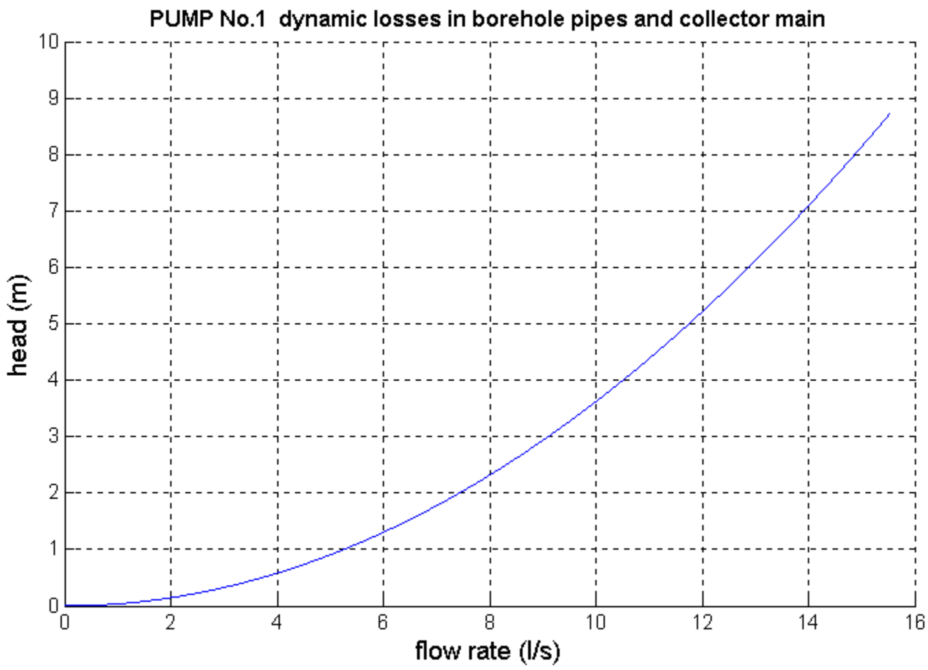


Figure 6.4 *example of dynamic losses in borehole and collector main*

6.7.3 System Curves

Since the total flow rate in a transmission main, Q_T, is the only variable determining the total head loss seen by each pump, then it is useful to find the system curves for various values of Q_T. In order to plot the system curve of any pump for a given value of Q_T, the equations describing the borehole, collector main and transmission main head losses must be expressed in terms of Q_P, for the given value of Q_T. Collector main and borehole piping losses are independent of Q_T, and are expressed in a form in terms of Q_P in equations 6.3 and 6.4, respectively. The transmission main losses are expressed in terms of Q_T in equation 6.2. To express equation 6.2 in terms of Q_P, for a particular pump, the Q_T-axis must be shifted to the left by an amount, *X*, such that (see figure 6.5):

$$X = Q_T' - Q_{nominal} \qquad \textbf{(equn. 6.5)}$$

where Q_T' is the value of Q_T for which expression must be found and $Q_{nominal}$ is the nominal flow rate of the pump

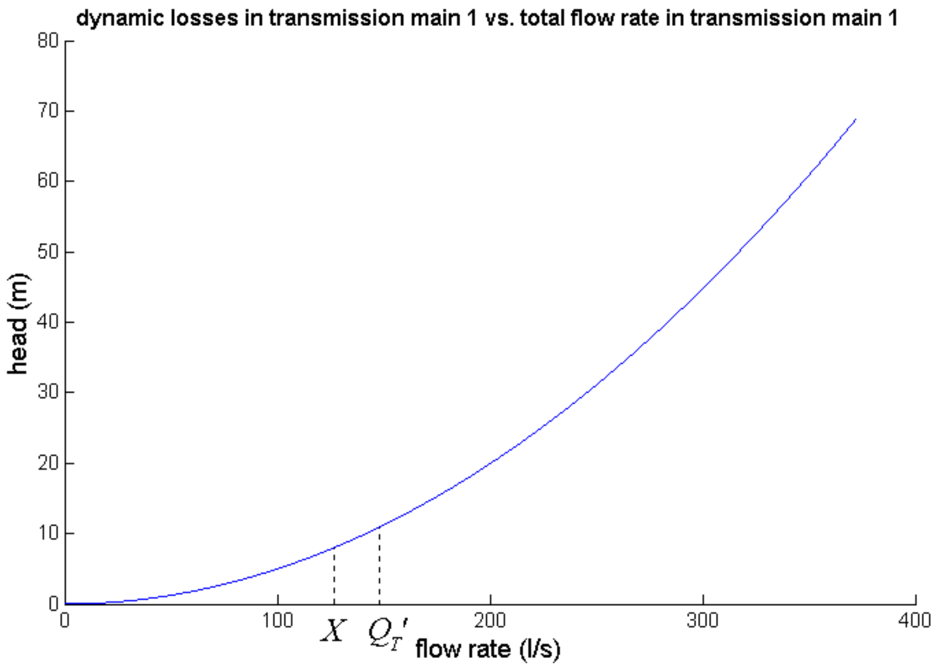


Figure 6.5 *illustration of new axis position for expressing H_{D_T} in terms of Q_P*

The Q_P - axis is related to the Q_T - axis by the expression:

$$Q_T - X = Q_P$$

$$Q_T = Q_P + X \qquad \textbf{(equn. 6.6)}$$

substituting into equation 6.2 gives,

$$H_{D\_T} = k_T.(Q_p + X)^2$$

$$= k_T.(Q_P{}^2 + 2.X.Q_P + X^2) \qquad \textbf{(equn. 6.7)}$$

So, the complete system curve can be described, for any given value of Q_T, by the following equation:

$$H_{SYS} = H_S + H_{D\_B} + H_{D\_C} + H_{D\_T}$$
$$= H_S + k_B.Q_P{}^2 + k_C.Q_P{}^2 + k_T.Q_P{}^2 + 2.k_T.X.Q_P + k_T.X^2$$
$$= (k_B + k_C + k_T).Q_P{}^2 + 2.k_T.X.Q_P + H_S + k_T.X^2 \qquad \textbf{(equn. 6.8)}$$

Figure 6.6 shows a plot of the pump curve, for pump No.1, and the system curve occurring at three significant states (when pump No.1 is connected through transmission main 1).

With only pump No.1 operating, system curve S1 would occur (with the PSV fully open). In this condition, using throttling control, the PSV would produce the largest resistance that it is required to do, in order to produce system curve S4 and force the operating point to point B (which is the only operating point that will give the nominal flow rate at rated pump speed). Using variable speed control, the pump speed would reduce and provide pump curve P2 and the pump would operate at point A.

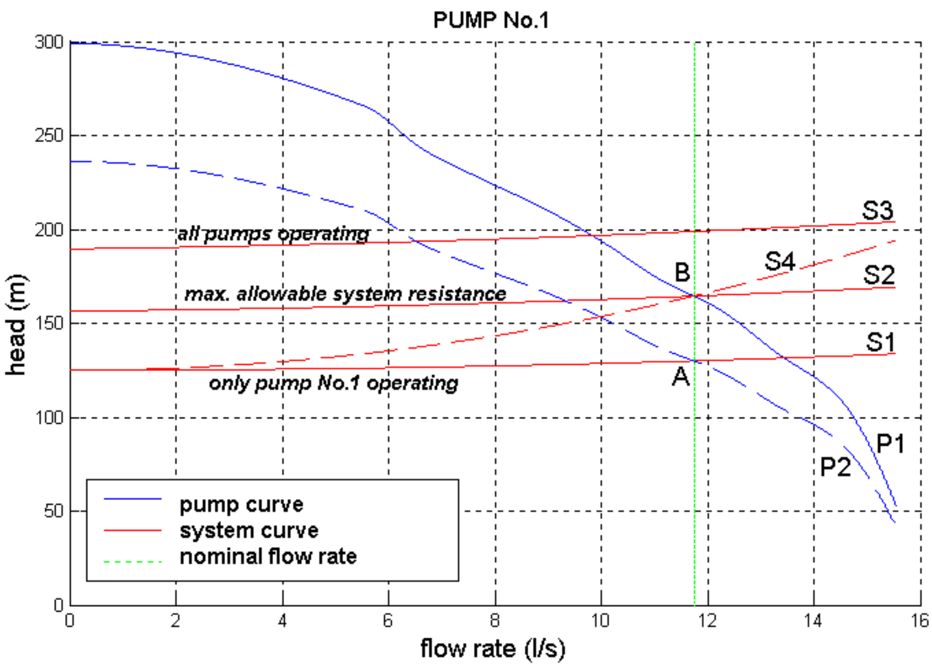


Figure 6.6 *illustration of pump operating range*

As more pumps are brought into line, system resistance increases. Line S3 represents the system curve when all pumps are operating through transmission main 1. With this level of resistance the system curve has gone beyond point B, and the pump can only be operated at flow rates less than nominal flow rate. The maximum system resistance for which nominal flow rate can be maintained is when system curve S2 occurs. With this system resistance the pump will operate at point B (nominal flow rate) with the PSV fully open and the pump running at full speed under either throttling or variable speed control. The range of system head loss for which nominal flow rate can be maintained (given that curve S1 represents the minimum system resistance) is between point A and point B. For pump No.1 the system head range at nominal flow rate is $130 \cdot 0m$ to $164 \cdot 5m$. Since the change in system resistance at nominal flow rate is dependent only upon total flow rate in the transmission main, Q_T, then the system head between point A and point B can be expressed as a function of Q_T. This relationship can be found by looking at equation 6.7. For nominal flow rate equation 6.7 is valid for $130 \cdot 0 < H_{SYS} < 164 \cdot 5$ (the range of system head between point A and point B when connected to transmission main 1). The maximum value of Q_T in this range can be found from equation 6.2:

$$164 \cdot 5 = 0 \cdot 0004967.Q_T^{\ 2}$$
$$\Rightarrow Q_T = 263 \cdot 5 \ {}^{l}\!/_{s}$$

The system resistance at point A is independent of Q_T, so when expressing system resistance at nominal flow rate as a function of Q_T, then the value of system head at point A is a constant. At nominal flow rate the minimum flow through the transmission main is the flow rate of the pump itself, i.e. the nominal flow rate. Hence, when expressing system resistance at nominal flow rate as a function of Q_T, the minimum value of Q_T for which the expression is valid is the value of the nominal flow rate of the pump itself. For pump No.1 connected through transmission main 1, the system resistance at nominal flow rate, expressed as a function of Q_T is given by:

$$H_{SYS} = 130 \cdot 0 + 0 \cdot 0004967.Q_T^{\ 2}, \text{ for } 11 \cdot 75 < Q_T < 263.5 \qquad \textbf{(equn. 6.9)}$$

Equation 6.9 is plotted in figure 6.7.

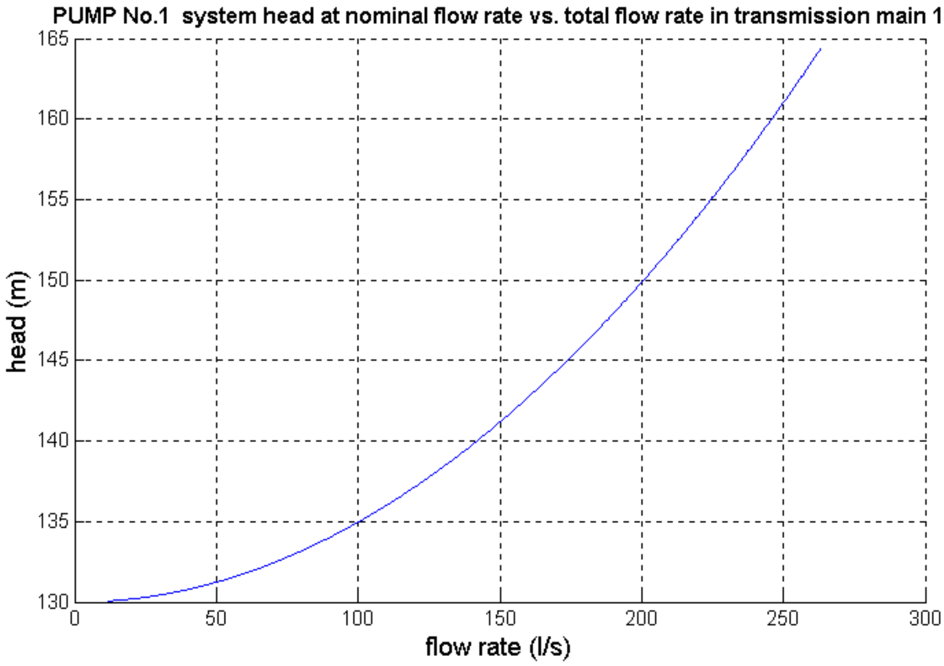


Figure 6.7 *example:* H_{SYS} *vs.* Q_T

The general form of equation 6.9 is:

$$H_{SYS} = H_{\min} + k_T . Q_T^{\ 2}, \text{ for } Q_{no\min al} < Q_T < Q_{T\_\max} \qquad \textbf{(equn. 6.10)}$$

where $H_{\min}$ is the minimum system head loss seen by a pump at nominal flow rate (defined for the condition where no other pumps are operating - for pump No. 1 this is point A on figure 6.6) and $Q_{T\_\max} = \sqrt{\dfrac{H_{\max}}{k_T}}$, where $H_{\max}$ is the maximum allowable system head loss at nominal flow rate (i.e. the head value corresponding to nominal flow rate on the pump curve for full speed - for pump No. 1 this is point B on figure 6.6)

The constants in equation 6.10 are given for each pump in table 6.6.

pump No.	$H_{\min}$ (m)	$H_{\max}$ (m)	$Q_{nominal}$ (l/s)	$Q_{T(1)\_\max}$ (l/s)	$Q_{T(2)\_\max}$ (l/s)
1	130.0	164.5	11.75	263.5	265.6
2	N/A	N/A	12.00	N/A	N/A
3	125.9	155.4	12.00	243.7	245.6
4	125.7	158.7	9.20	257.8	259.8
5	126.9	149.9	10.00	215.2	216.9
6	126.6	148.1	7.20	208.1	209.7
7	126.1	149.1	7.16	215.2	216.9
8	123.4	173.0	7.00	316.0	318.4
9	127.6	169.7	11.30	291.1	293.4
10	126.2	161.1	12.00	265.1	267.2
11	127.0	157.7	9.30	248.6	250.6
12	125.3	158.0	12.20	256.6	258.6
13	125.6	159.6	12.10	261.6	263.7
14	128.9	153.7	10.70	223.4	225.2
15	128.4	158.0	12.20	244.1	246.0
16	127.0	159.6	12.10	256.2	258.2
17	129.5	153.9	10.69	221.6	223.4
18	131.1	152.3	12.50	206.6	208.2
19	130.6	169.8	7.13	280.9	283.1
20	128.5	160.2	7.50	252.6	254.6
21	129.3	158.0	12.20	240.4	242.3
22	129.9	158.7	9.20	240.8	242.7
23	127.4	152.3	12.50	223.9	225.7
24	127.9	158.0	12.20	246.2	248.1
25	132.1	159.6	12.10	235.3	237.1
26	131.9	158.0	12.20	229.2	231.0
27	128.3	161.5	11.70	258.5	260.6
28	129.0	161.1	12.00	254.2	256.2
29	135.7	162.5	11.90	232.3	234.1
30	128.4	153.7	10.70	225.7	227.5
31	130.5	149.6	7.14	196.1	197.6
32	128.3	169.8	7.13	289.1	291.3
33	131.4	153.6	10.71	211.4	213.1
34	134.4	156.2	10.54	209.5	211.1
35	129.6	169.8	7.13	284.5	286.7
36	N/A	N/A	7.00	N/A	N/A

Table 6.6 *pump operating ranges: system head and total flow rate*

6.8 Power Consumption

The difference in power consumption between throttling control and variable speed control for each pump is examined in chapter 6.8.

6.8.1 Throttling Control

When a PSV is being used to maintain nominal flow rate, the pump operating point remains the same (point B of figure 6.6). Since the operating point does not change, then H, Q and η are constant. So, when using a PSV, the power consumption can be expressed as (as stated in equation 2.2):

$$P_{PSV} = \frac{Q_P . g . H_{SYS}}{\eta_m . \eta_P} \qquad \textbf{(equn. 6.11)}$$

P_{PSV} is a constant at nominal flow rate.

6.8.2 Variable Speed Control

When using a VSD to maintain nominal flow rate the operating point will vary (between point A and point B of figure 6.6) depending upon total flow rate, Q_T. So H_{SYS} will vary with Q_T, and η_P will vary as pump speed changes - also as a function of Q_T. Motor efficiency, η_m, is assumed to remain constant over the operating range considered here. There is a further factor which must be brought into the power calculation - the efficiency of the VSD, η_{VSD}. This will be taken to be constant, such that $\eta_{VSD} = 0 \cdot 95$. Power consumption, when using a VSD to maintain nominal flow rate, can be expressed as a function of Q_T using the equation:

$$P_{VSD} = \frac{Q_P . g . H_{SYS}(Q_T)}{\eta_m . \eta_{VSD} . \eta_P(Q_T)} \qquad \textbf{(equn. 6.12)}$$

where $H_{SYS}(Q_T)$ represents H_{SYS} expressed as a function of Q_T, and $\eta_P(Q_T)$ represents η_P expressed as a function of Q_T

The relationship between H_{SYS} and Q_T for each pump can be found from equation 6.10 and table 6.6. The relationship between η_P and Q_T for variable speed control is examined in chapter 6.8.2.1.

6.8.2.1 Pump Efficiency

Since the efficiency curve and pump curve are not described by equations, but only as plots, graphical methods must be used to find the relationship between η_P and Q_T for variable speed control. Firstly, using the method described in chapter 4.3, a plot of η_P against H_{SYS} can be found.

It should be noted that the efficiency values given in table 6.1 include motor efficiency, η_m, and pump efficiency, η_P. Within the operating range considered here, η_m would be expected to remain constant. Similarly, η_m would be expected to remain constant across the range for which η_P varies with speed variation. So, given that η_m is taken to be a constant, there is no need to separate η_m and η_P when calculating the variation of η_P with Q_T. It is sufficient to plot motor and pump efficiency ($\eta_m.\eta_P$) against Q_T under these assumptions.

A computer program was used to find various values of *k* for a sequence of curves of the form, $H = k.Q^2$, each passing through a different point between $H_{\min}$ and $H_{\max}$ at nominal flow rate. With the program, the intersection of each of these curves with the pump curve (for rated speed) was found. Then, for each point found on the pump curve, the efficiency at that flow rate was read from the efficiency curve (for rated speed). Then, these efficiency values were plotted against the initial values of H_{SYS} to show the variation of motor and pump efficiency with system head at nominal flow rate, between $H_{\min}$ and $H_{\max}$. This plot, for pump No.1, is shown in figure 6.8.

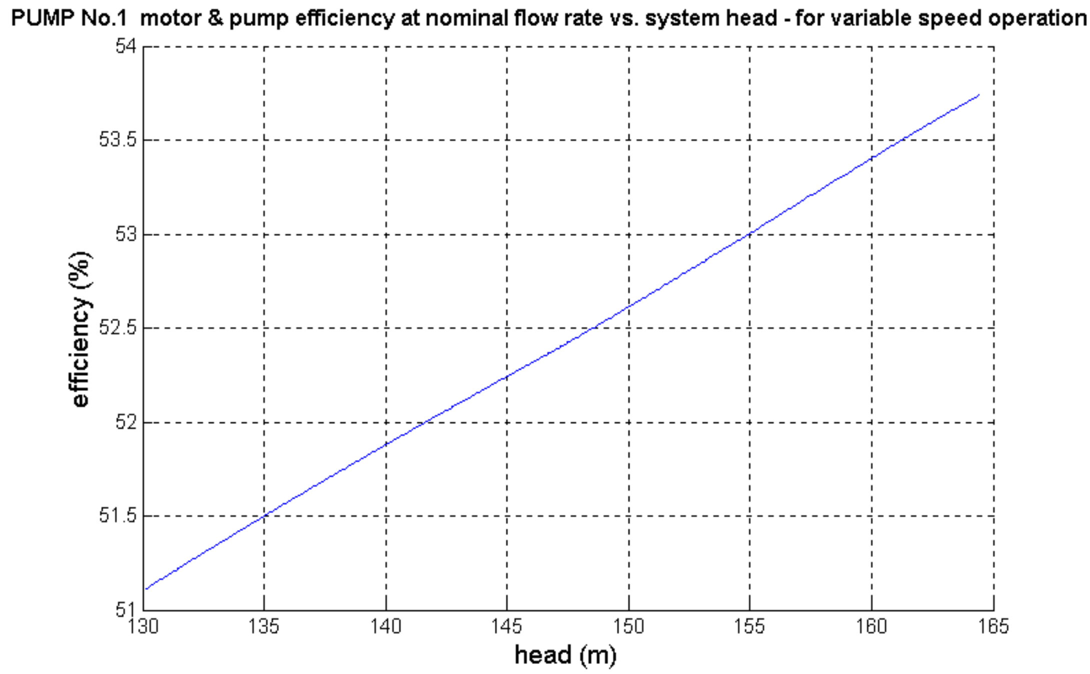


Figure 6.8 *example:* $(\eta_m.\eta_P)$ *vs.* H_{SYS}

The H-axis can be translated from H_{SYS} to Q_T by finding Q_T for various points on the curve using equation 6.9. Motor and pump efficiency $(\eta_m.\eta_P)$ can then be plotted against Q_T. Figure 6.9 shows a plot of $(\eta_m.\eta_P)$ against Q_T for pump No.1 connected through transmission main 1. Plots of $(\eta_m.\eta_P)$ against Q_T for each pump are contained in appendix A.

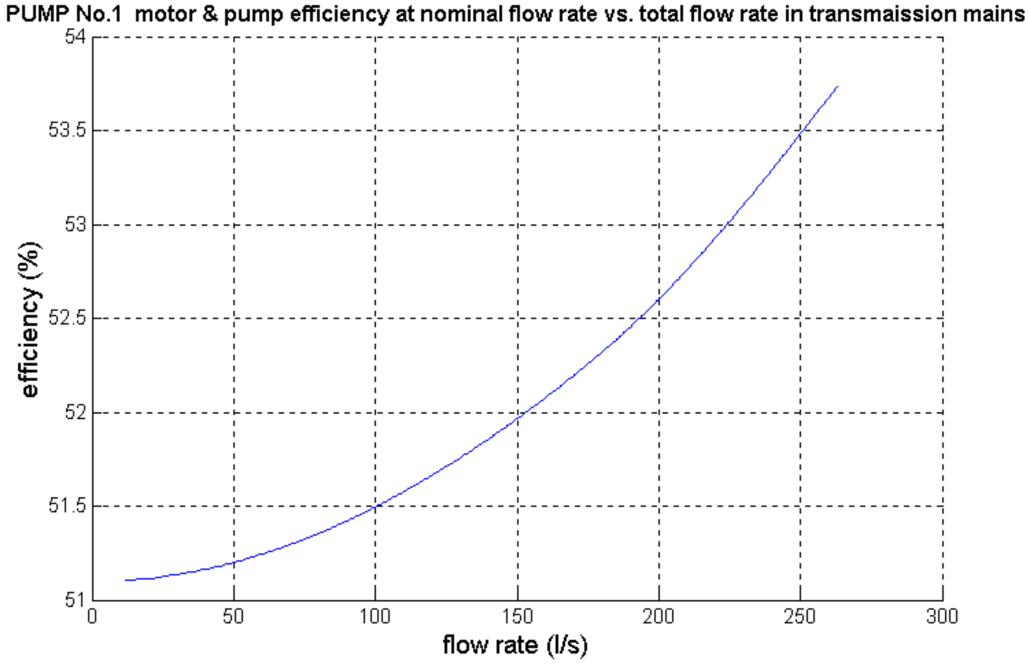


Figure 6.9 *example:* $(\eta_m.\eta_P)$ *vs.* Q_T

Figure 6.10 shows a plot of power consumption against total flow rate in transmission main 1 for pump No.1 under throttling control (PSV) and variable speed control (VSD) - plotted using equations 6.11 and 6.12.

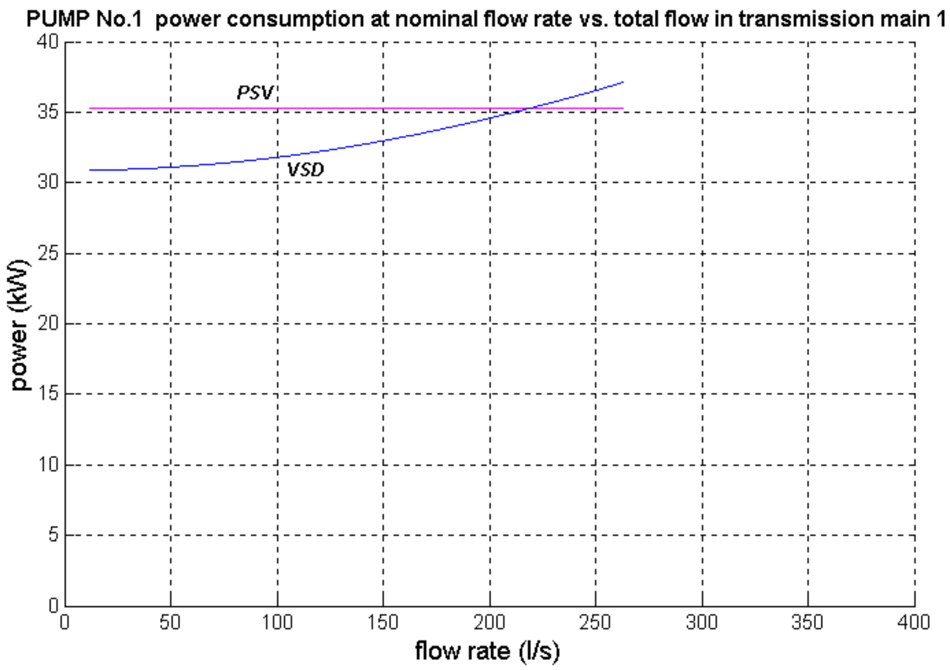


Figure 6.10 *example:* P_{PSV} *and* P_{VSD} *vs.* Q_T *for transmission main 1*

It can be seen that, at Q_{T_max}, power consumption with the VSD is greater than the power consumption with the PSV. In this condition, with throttling control the PSV would be fully open (i.e. the PSV is would not be creating any additional system resistance), and with variable speed control the pump would be operating at full speed. The additional power consumed with variable speed control is as a result of the additional efficiency factor of the VSD, η_{VSD}.

Figure 6.11 shows the power consumption for pump No.1, when connected to transmission main 2. By comparing figures 6.10 and 6.11 it can be seen that, for variable speed control, the difference in power consumption between transmission main 1 and transmission main 2 is insignificant to this degree of accuracy.

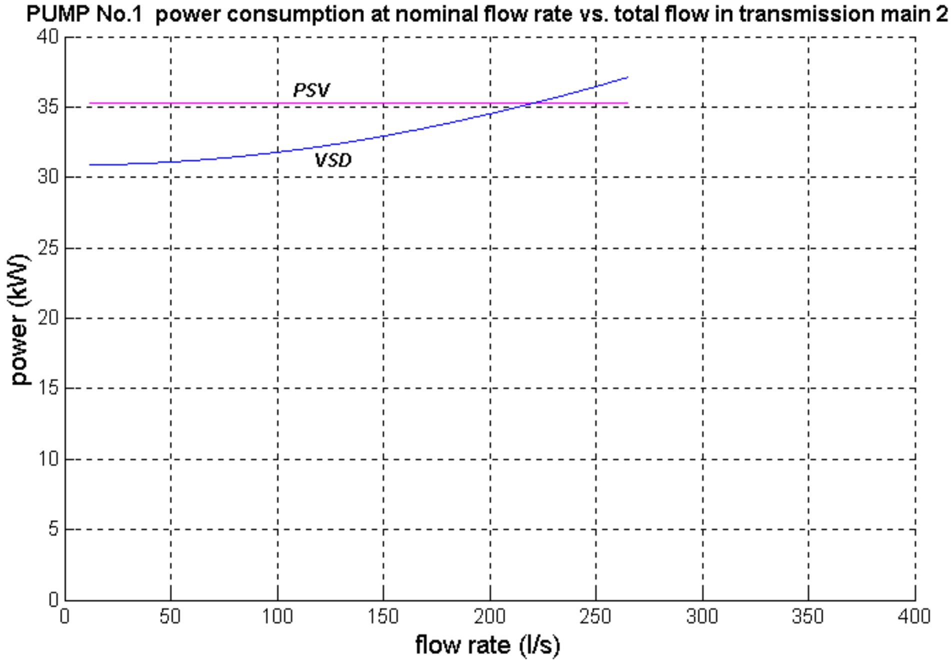


Figure 6.11 *example:* P_{PSV} *and* P_{VSD} *vs.* Q_T *for transmission main 2*

Plots of P_{PSV} and P_{VSD} against Q_T for each pump are given in appendix B. Only the plots for connection to transmission main 1 are given. The plots for connection to transmission main 2 can be assumed to be the same.

Due to differences in the pump and system characteristics, and differences in nominal flow rates for each pump, the difference in power consumption between throttling control and variable speed control, for a given system state, varies from pump to pump. So, certain pumps are likely yield greater benefits when installed with VSD's than others, depending on the load duty cycle of each pump.

6.9 Demand

An estimated demand profile, provided by Scottish Water, was used, here, to examine the energy savings which could be achieved by installing VSD's. The forecast demand data is in the form of annual average flow rates. An analysis was carried out using the forecast demand for the year 2003. The average water demand, forecast for 2003, is estimated at approximately 197 $\frac{l}{s}$. This flow rate was used for the analysis of pump operation over the year.

6.10 Pump Selection

The factors which must be taken into account when looking at the viability of installing VSD's on pumps are power consumption and water quality. For the analysis of the year 2003 the total flow rate, at 197 $\frac{l}{s}$, is approximately 53% of the maximum capacity of all the pumps. This suggests that around 19 pumps (depending on the individual flow rates) would be required to meet the demand flow rate. So, 19 pumps must be selected from the total of 36 pumps. The first selection factor to be considered is water quality. There are 27 A1 quality pumps (see table 6.2), so the first selection was made by short-listing these 27 high water quality pumps. Since the purpose of this study is to analyse the potential energy savings achievable through the use of VSD's in the present system, then the final pump selection was based on the magnitude of energy savings for each pump. The energy savings are a measure of the difference between P_{PSV} and P_{VSD} for a particular pump, in relation to the length of time for which that pump is operating over a period. Since all the pumps in this analysis are operating constantly over the year, the energy savings made on a pump are directly proportional to the power reduction in that pump.

Using the plots given in appendix B, the pumps can be listed in order of annual energy savings made with VSD's, at a given flow configuration. Initially, the power reduction was calculated on the basis of an equal flow rate through each transmission main, i.e. $98 \cdot 5 \frac{l}{s}$ in each transmission main, making up the total flow rate of 197 $\frac{l}{s}$. It is desirable to have equal flow rates in each transmission main in order to minimize head losses, since $H_{D\_T} \propto Q^2$. Table 6.7 shows the power reduction for each A1 quality pump for these flow conditions, listed in order of magnitude of power reduction.

pump No.	P_{PSV} (kW)	P_{VSD} (kW)	power reduction (kW)
8	21.5	16.3	5.2
9	33.5	28.9	4.6
4	27.7	23.4	4.3
32	21.3	17.3	4.0
35	21.3	17.4	3.9
10	35.8	32.0	3.8
11	27.7	23.9	3.8
13	36.0	32.2	3.8
1	35.3	31.8	3.5
12	36.1	32.6	3.5
16	36.0	32.5	3.5
22	27.7	24.2	3.5
27	33.7	30.4	3.3
19	20.8	17.6	3.2
24	36.1	33.1	3.0
15	36.1	33.2	2.9
3	33.9	31.1	2.8
21	36.1	33.4	2.7
20	20.9	18.2	2.7
14	30.2	27.7	2.5
25	36.0	33.5	2.5
5	27.7	25.5	2.2
23	36.3	34.2	2.1
33	30.2	28.2	2.0
34	30.3	28.4	1.9
6	20.2	18.8	1.4
18	36.3	34.9	1.4

Table 6.7 *A1 quality pumps: power consumption*

Starting from the top of table 6.7, pumps were chosen to give a total flow rate of approximately 197 l/s. The first 19 pumps in the table (down to and including pump No.20) give a total flow rate of $195 \cdot 34\, l/s$, which is sufficiently close to the estimated demand figure for this analysis. With this set of pumps operating, the power reduction for each pump must be re-calculated for the new values of flow rate in each transmission main. With the same configuration of pumps and collector mains, as described in chapter 6.7, the flow rates through each collector main are as follows:

collector main 1: $95 \cdot 53\ {}^{l}\!/\!_{s}$

collector main 2: $52 \cdot 45\ {}^{l}\!/\!_{s}$

collector main 3: $18 \cdot 83\ {}^{l}\!/\!_{s}$

collector main 4: $28 \cdot 53\ {}^{l}\!/\!_{s}$

Connecting collector mains 1 and 3 to transmission main 1, and collector mains 2 and 4 to transmission main 2 gives the most even split of the total flow. This results in transmission main flow rates as follows:

transmission main 1: $114 \cdot 4\ {}^{l}\!/\!_{s}$

transmission main 2: $81 \cdot 0\ {}^{l}\!/\!_{s}$

Table 6.8 shows the power reduction of each of these 19 pumps, re-calculated for the new transmission main flow rates.

pump No.	P_{PSV} (kW)	P_{VSD} (kW)	power reduction (kW)
8	21.5	16.1	5.4
9	33.5	29.1	4.4
11	27.7	23.6	4.1
35	21.3	17.2	4.1
4	27.7	23.7	4.0
1	35.3	31.5	3.8
12	36.1	32.3	3.8
22	27.7	23.9	3.8
32	21.3	17.5	3.8
10	35.8	32.3	3.5
13	36.0	32.6	3.4
24	36.1	32.8	3.3
16	36.0	32.9	3.1
21	36.1	33.1	3.0
19	20.8	17.8	3.0
27	33.7	30.8	2.9
15	36.1	33.5	2.6
3	33.9	31.4	2.5
20	20.9	18.4	2.5

Table 6.8 *selected pumps: power consumption*

6.11 Economics

Since the cost of a VSD rated to match the $18 \cdot 5kW$ motors will be less than the cost of a VSD rated to match the $30kW$ motors, then the SP27 and SP45 ranges must be looked at separately, when considering the economics of VSD installation.

Tables 6.9 and 6.10 show the energy reduction achieved by each pump over the year 2003 (assuming constant operation) and the resulting financial savings, based on electricity prices @ 1st April 2002, including Fossil Fuel Levy and Climate Change Levy charges.

SP45

pump No.	power reduction (kW)	annual energy reduction (kWh)	annual saving (£)
9	4.4	38544	1421
11	4.1	35916	1324
4	4.0	35040	1292
1	3.8	33288	1227
12	3.8	33288	1227
22	3.8	33288	1227
10	3.5	30660	1130
13	3.4	29784	1098
24	3.3	28908	1066
16	3.1	27156	1001
21	3.0	26280	969
27	2.9	25404	937
15	2.6	22776	840
3	2.5	21900	807

Table 6.9 *SP45 pumps: annual energy savings*

SP27

pump No.	power reduction (kW)	annual energy reduction (kWh)	annual saving (£)
8	5.4	47304	1744
35	4.1	35916	1324
32	3.8	33288	1227
19	3.0	26280	969
20	2.5	21900	807

Table 6.10 *SP27 pumps: annual energy savings*

The financial savings made on each pump must be weighed up against the cost of installing a VSD. Using Discounted Cash Flow Analysis a relationship between annual return on investment (financial savings made from energy reduction) and payback period can be established. Since annual savings will be approximately equal for each year of the payback period, then the annual return can be expressed in terms of the payback period thus [6]:

annual return, $R = \frac{i}{1-(1+i)^{-n}}.T$ **(equn. 6.13)**

where n is payback period (yrs), i is interest rate per annum and T is the capital cost of the VSD

This equation takes into account the assumption that the capital cost is spent within a short time, so that it need not be discounted. This will be the case with the installation of the VSD's, as these can be installed within one week.

For the required values of interest rate and payback period, the corresponding annual return can be found from equation 6.13, and compared to the annual savings in tables 6.9 and 6.10. This will provide the cut-off point for the selection of which pumps, when installed with VSD's, will provide a complete return on investment within the required payback period, and at the chosen interest rate.

Estimates for the capital cost of a VSD, including installation, are given below (due to insufficient information regarding specific prices, rough estimates have been used here, based on previous estimations - actual prices will differ for differently sized VSD's):

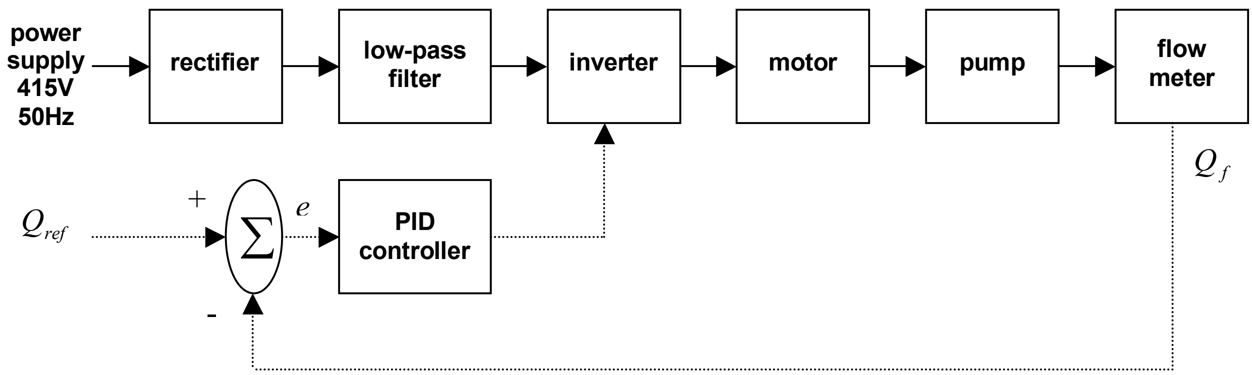
- VSD rated @ $18 \cdot 5kW$: £5000
- VSD rated @ $30kW$: £5000

The following calculations are made assuming an interest rate of 10% per annum.

Using the method described above, if all the pumps in tables 6.9 and 6.10 were fitted with VSD's, the cost could be recovered in approximately 10 years. If a shorter payback period is required, say 6 years, then by comparing the annual return for a 6 year payback against annual savings in tables 6.9 and 6.10, it can be shown that pumps 9, 11, 4, 1, 12, 22, 8, 35 and 32 would provide sufficient energy savings, if fitted with VSD's, to pay back the cost within 6 years.

6.12 Further Considerations

For this system, VSD's would be required to be installed in closed-loop configuration. A flow meter would be required in each control loop to provide a feedback signal representing the measured flow rate. A PID control algorithm would need to be developed for each control loop. This would be of the form given in equation 3.1. Mathematical modelling of the motors would be required in order to accurately specify the control algorithm coefficients. Modelling of the motors can be performed automatically by some VSD's. Figure 6.12 shows a system diagram giving the form of the required control loop.



Q_{ref} - reference set-point

Q_f - feedback value

Figure 6.12 *VSD feedback control loop*

Over a long period of operating time, the impellers of a pump become worn. This causes the pump characteristics to change. With a PSV this would result in reduced flow rates, without re-calibration of the pressure setting on the PSV. With a VSD, however, the nominal flow rate would be maintained regardless of changes in the pump characteristics, since the reference parameter is the pump flow rate - no re-calibration of the set-point is needed with VSD's.

With a PSV, any drifts above nominal flow rate might cause insufficient water levels in the well to occur, and consequently cause the pump to pull up extra gravel and block the filter. This problem would be eliminated with the use of a VSD.

Another additional advantage of VSD's is that running a pump at reduced speed reduces the wear on all drive chain components. This is of particular significance to this system, as it is based on the idea that the pumps installed with VSD's are the most frequently operated ones.

Power factors ranging between $0 \cdot 86$ and $0 \cdot 89$ have been recorded at the power supply points in the pumping station. Some VSD's are capable of correcting the power factor of the induction motors to approximately $0 \cdot 95$.

Harmonics produced by a VSD can cause a problem. Harmonics can be reflected back into the supply system. Regulations are in place that specify the permitted harmonic content of the supply. These are governed by the G5/4 regulations. External filters can be used to suppress harmonics to a certain degree.

Chapter 7. Conclusion

Various applications of variable speed control within water pumping systems were investigated. A detailed analysis of a water pumping station was carried out. For a given set of flow conditions, it was shown that, by employing variable speed control in place of throttling control, energy savings ranging from 9% to 25% could be made on each pump operated with variable speed control. An analysis of the economic viability of installing variable speed drives at the water pumping station was carried out. It was shown, for the estimated average water demand for the year 2003, that the installation of variable speed drives on all operating pumps would result in a payback period of approximately 10 years.

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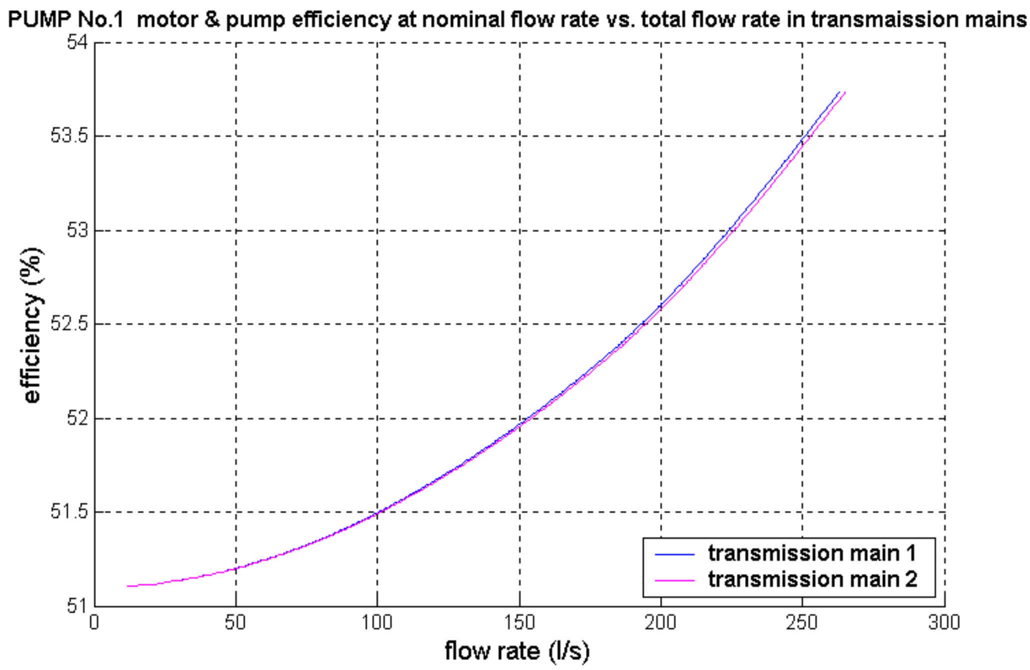
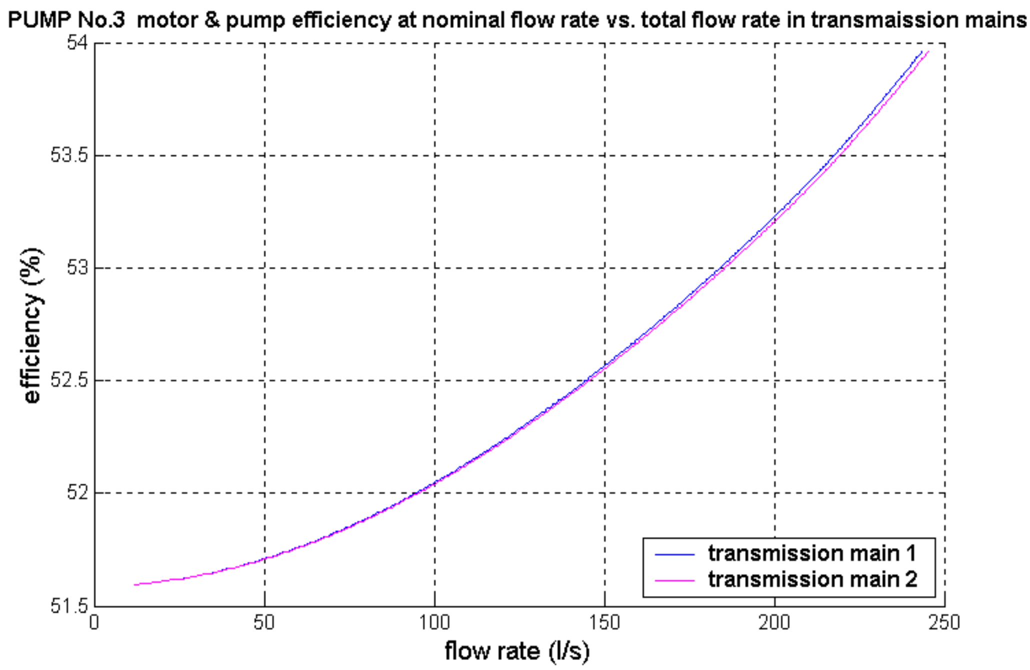
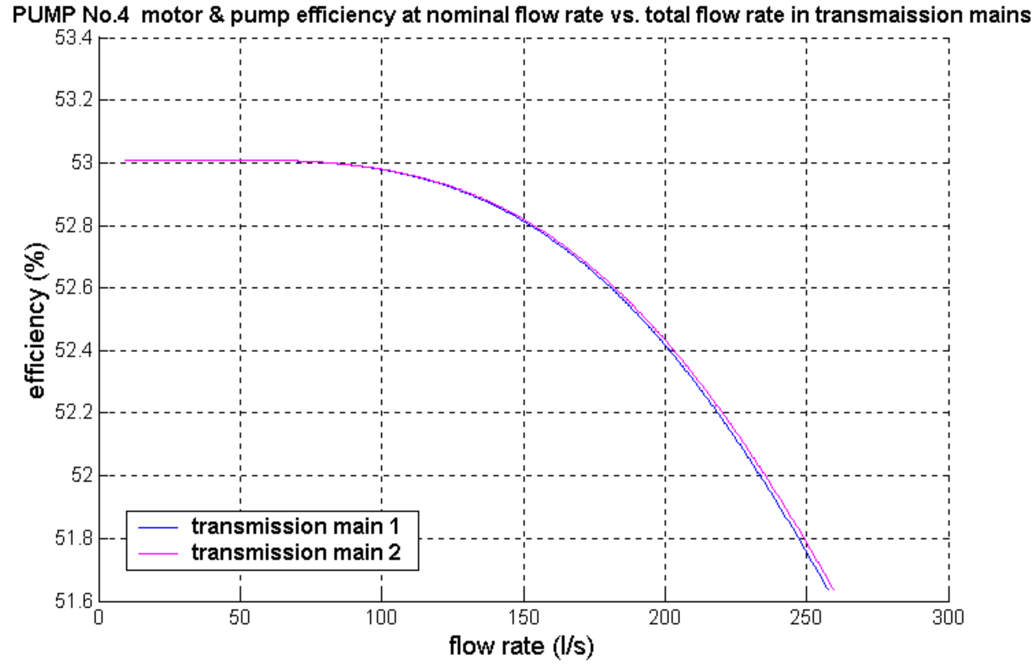
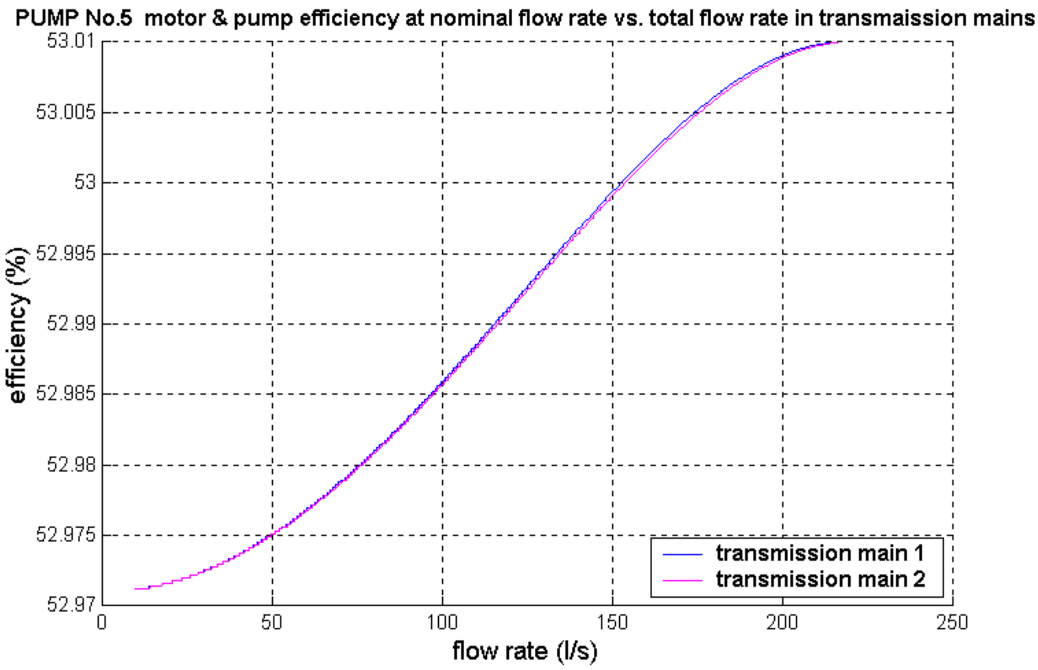
Appendix A: pump efficiency plots for variable speed control

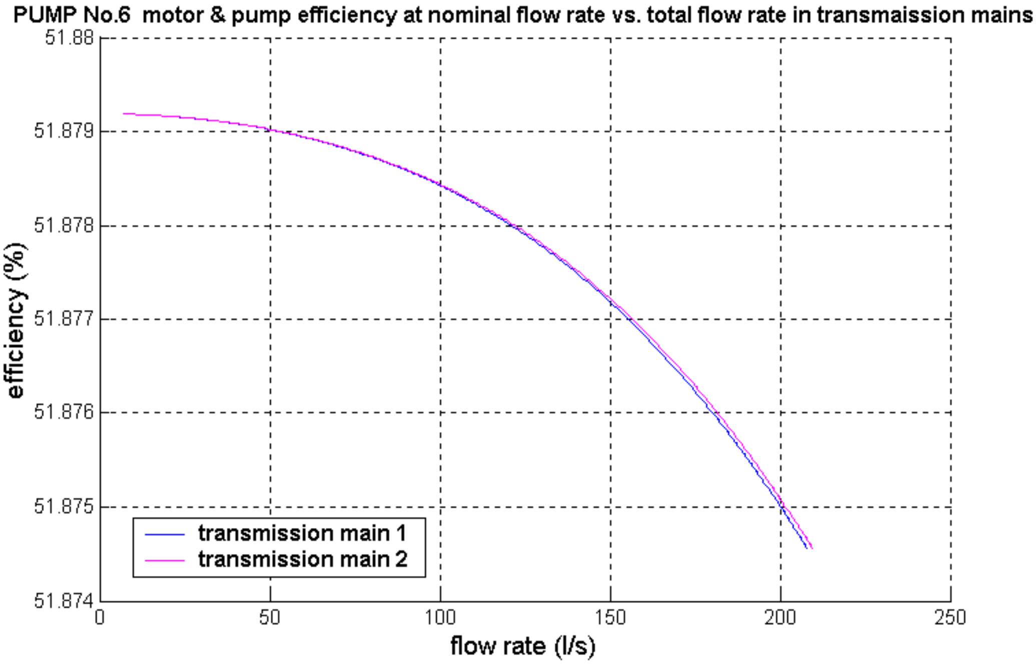
(pumps No.2 and No.36 are omitted due to unavailability of data)






PUMP No.4 motor & pump efficiency at nominal flow rate vs. total flow rate in transmaission mains
53.4
53.2
53
52.8
52.6
52.4
52.2
52
51.8
51.6
efficiency (%)
0
50
100
150
200
250
300
flow rate (l/s)
transmission main 1
transmission main 2


PUMP No.5 motor & pump efficiency at nominal flow rate vs. total flow rate in transmaission mains
53.01
53.005
53
52.995
52.99
52.985
52.98
52.975
52.97
efficiency (%)
0
50
100
150
200
250
flow rate (l/s)
transmission main 1
transmission main 2


PUMP No.6 motor & pump efficiency at nominal flow rate vs. total flow rate in transmaission mains
efficiency (%)
flow rate (l/s)
transmission main 1
transmission main 2
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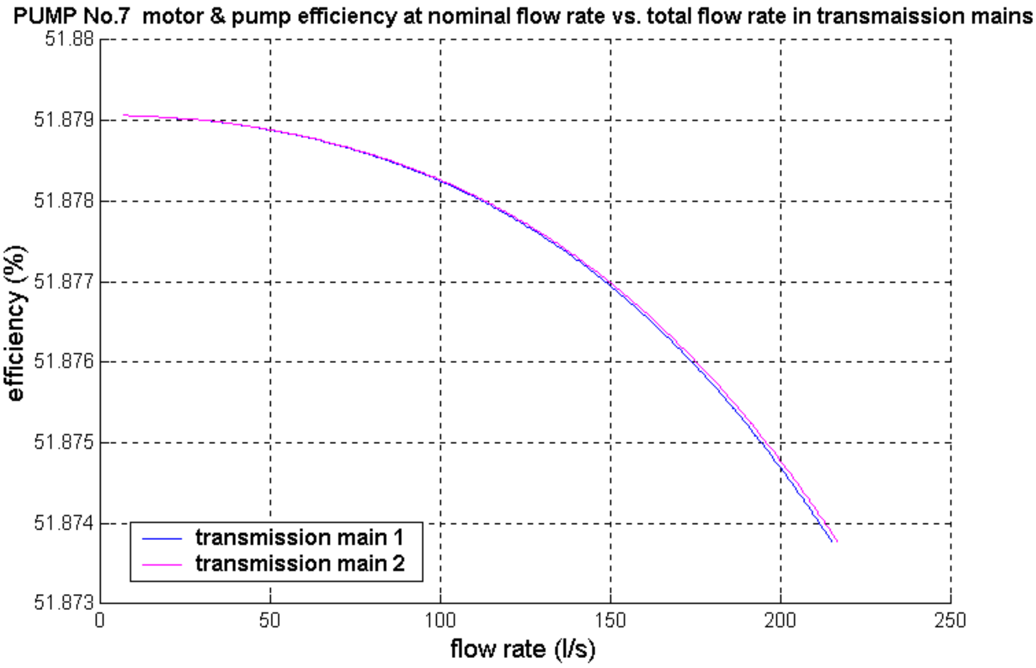
51.879
51.878
51.877
51.876
51.875
51.874
0
50
100
150
200
250


PUMP No.7 motor & pump efficiency at nominal flow rate vs. total flow rate in transmaission mains
efficiency (%)
flow rate (l/s)
transmission main 1
transmission main 2
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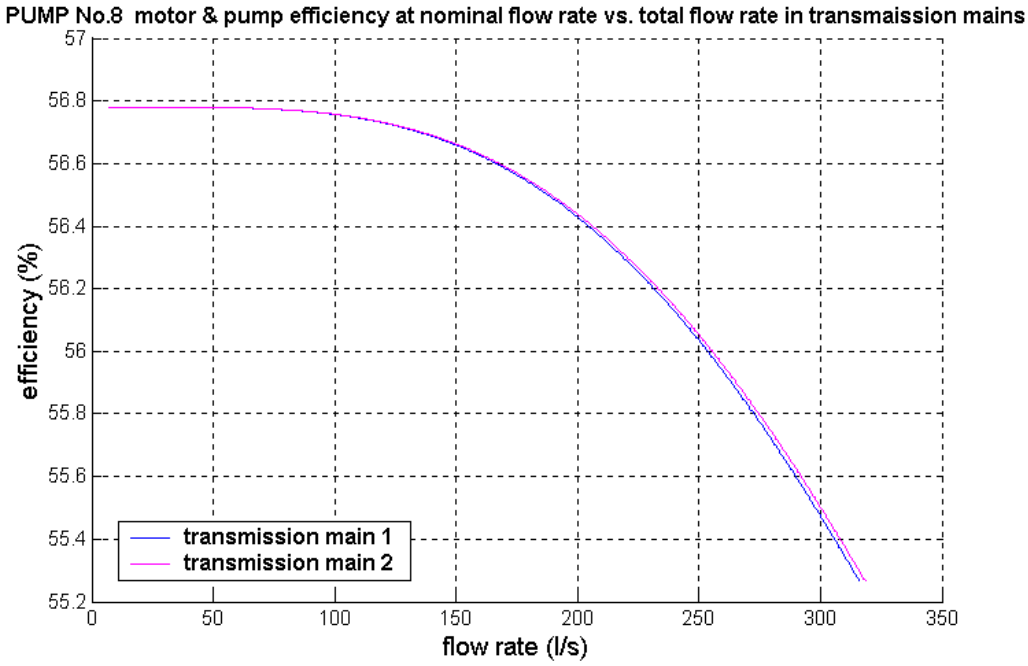
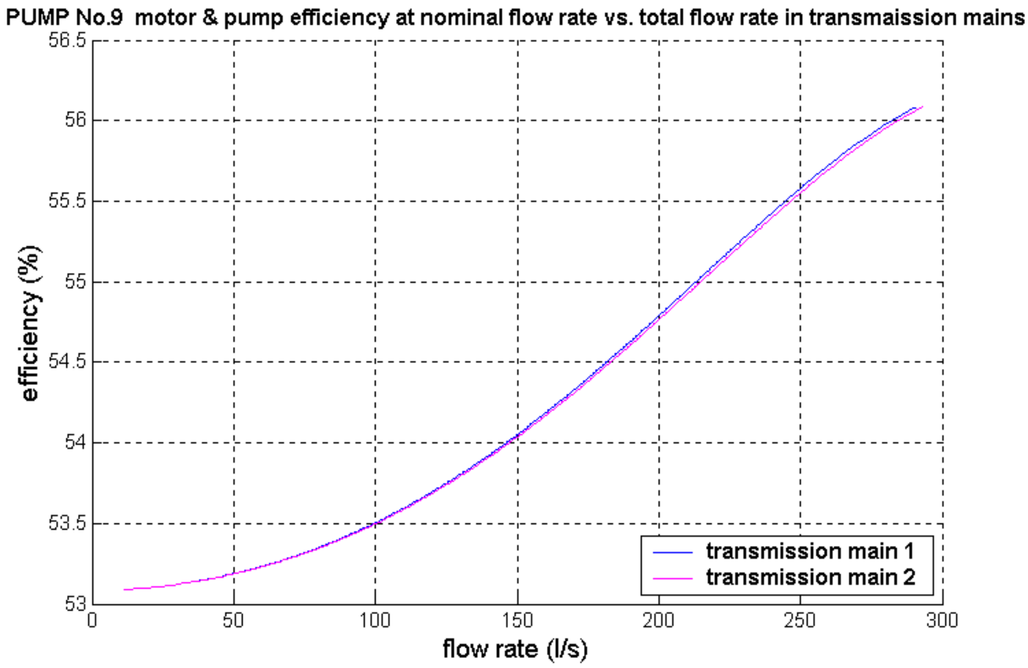
51.879
51.878
51.877
51.876
51.875
51.874
51.873
0
50
100
150
200
250


PUMP No.8 motor & pump efficiency at nominal flow rate vs. total flow rate in transmaission mains
efficiency (%)
flow rate (l/s)
transmission main 1
transmission main 2
57
56.8
56.6
56.4
56.2
56
55.8
55.6
55.4
55.2
0
50
100
150
200
250
300
350


PUMP No.9 motor & pump efficiency at nominal flow rate vs. total flow rate in transmaission mains
efficiency (%)
flow rate (l/s)
transmission main 1
transmission main 2
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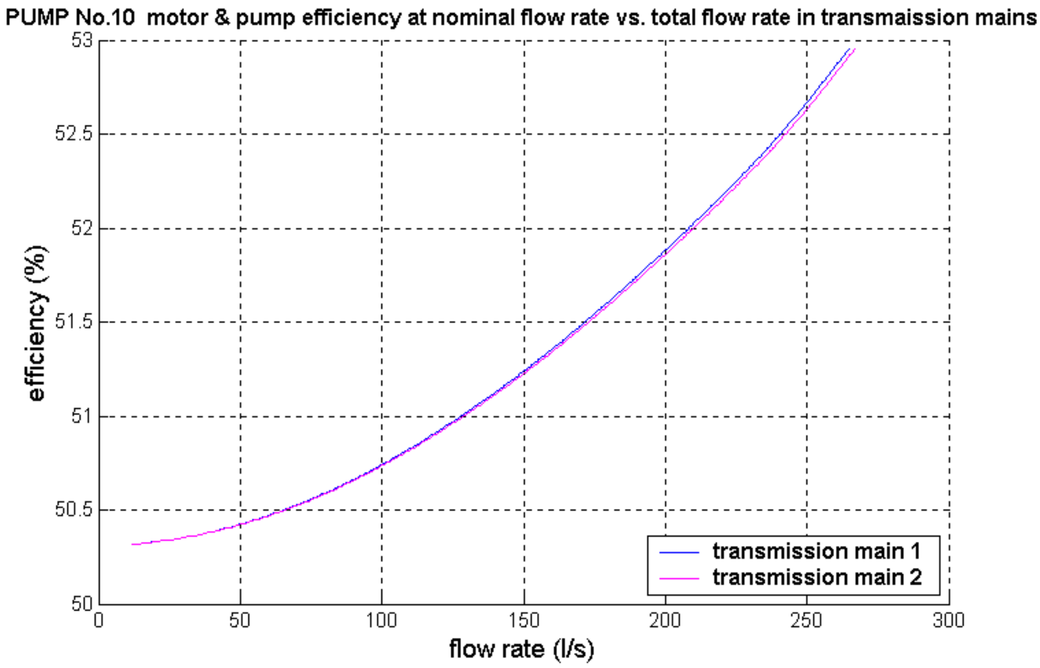
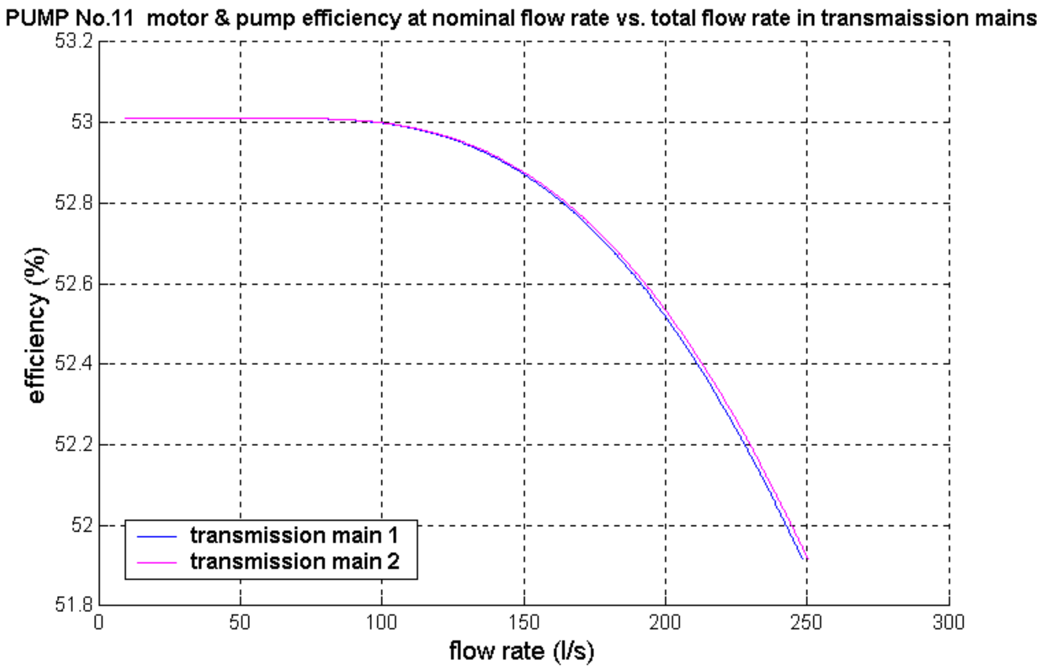
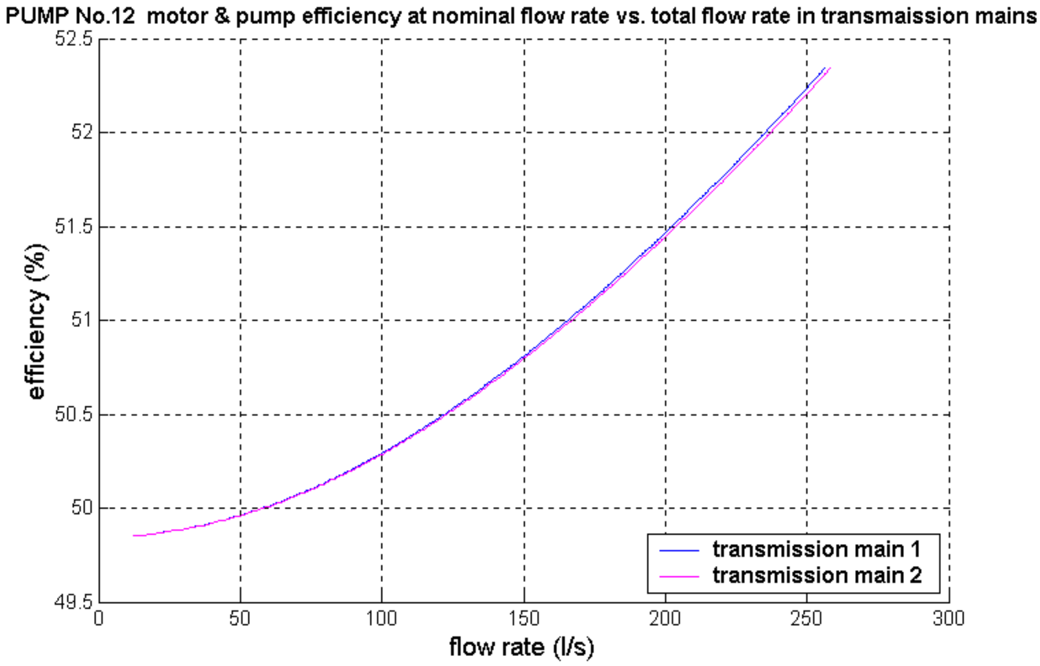
56
55.5
55
54.5
54
53.5
53
0
50
100
150
200
250
300



PUMP No.10 motor & pump efficiency at nominal flow rate vs. total flow rate in transmaission mains
53
52.5
52
51.5
51
50.5
50
efficiency (%)
0
50
100
150
200
250
300
flow rate (l/s)
transmission main 1
transmission main 2



PUMP No.11 motor & pump efficiency at nominal flow rate vs. total flow rate in transmaission mains
53.2
53
52.8
52.6
52.4
52.2
52
51.8
efficiency (%)
0
50
100
150
200
250
300
flow rate (l/s)
transmission main 1
transmission main 2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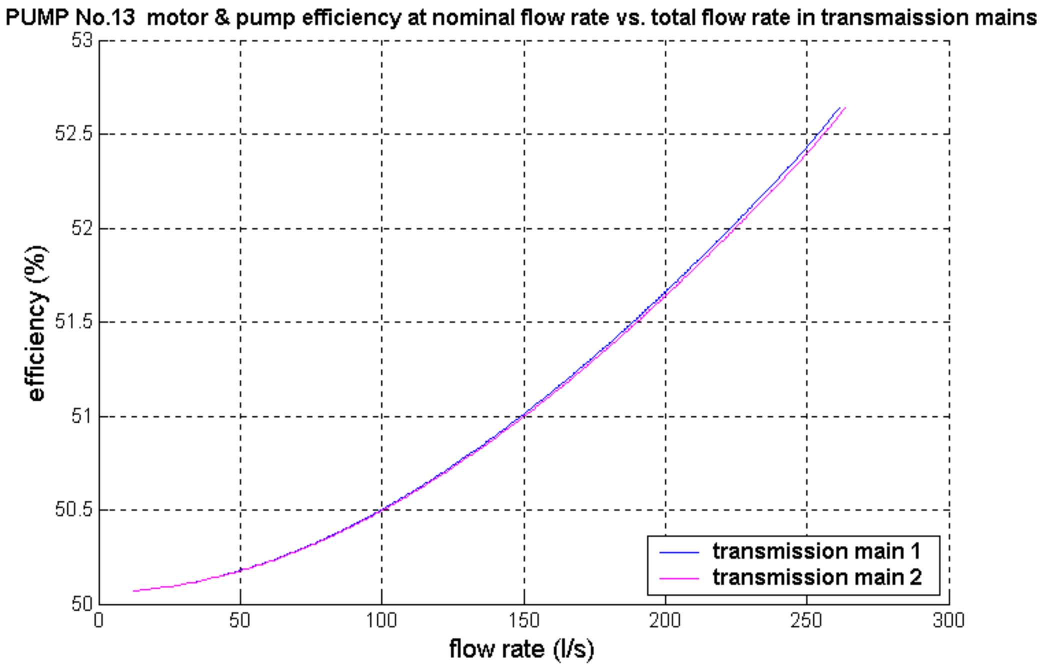
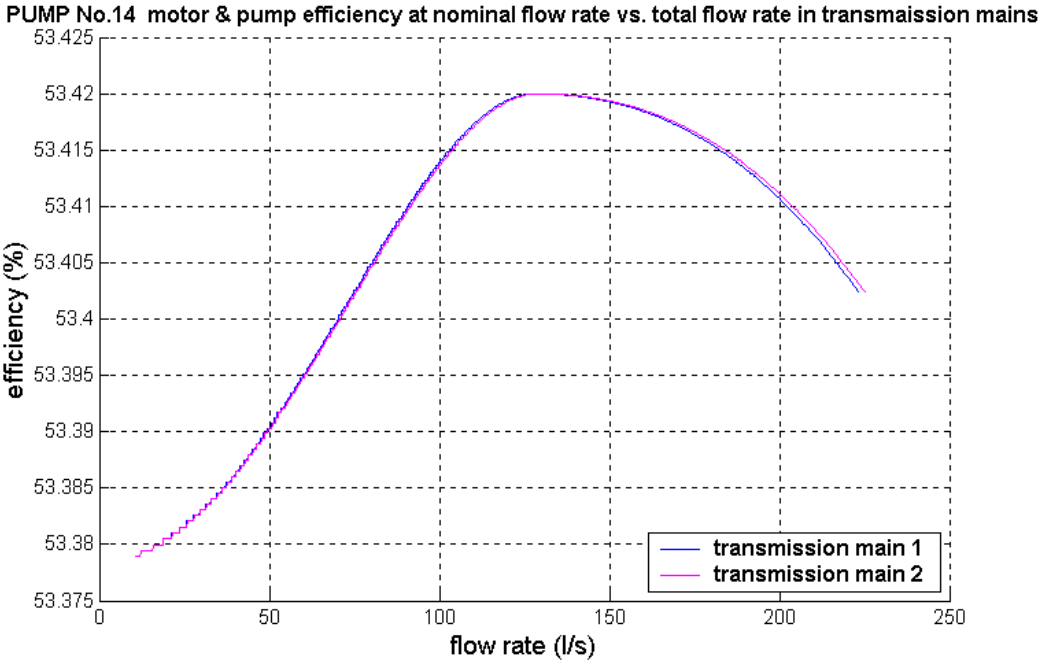
efficiency (%)
52.5
52
51.5
51
50.5
50
49.5
0
50
100
150
200
250
300
flow rate (l/s)
transmission main 1
transmission main 2


PUMP No.13 motor & pump efficiency at nominal flow rate vs. total flow rate in transmaission mains
efficiency (%)
53
52.5
52
51.5
51
50.5
50
0
50
100
150
200
250
300
flow rate (l/s)
transmission main 1
transmission main 2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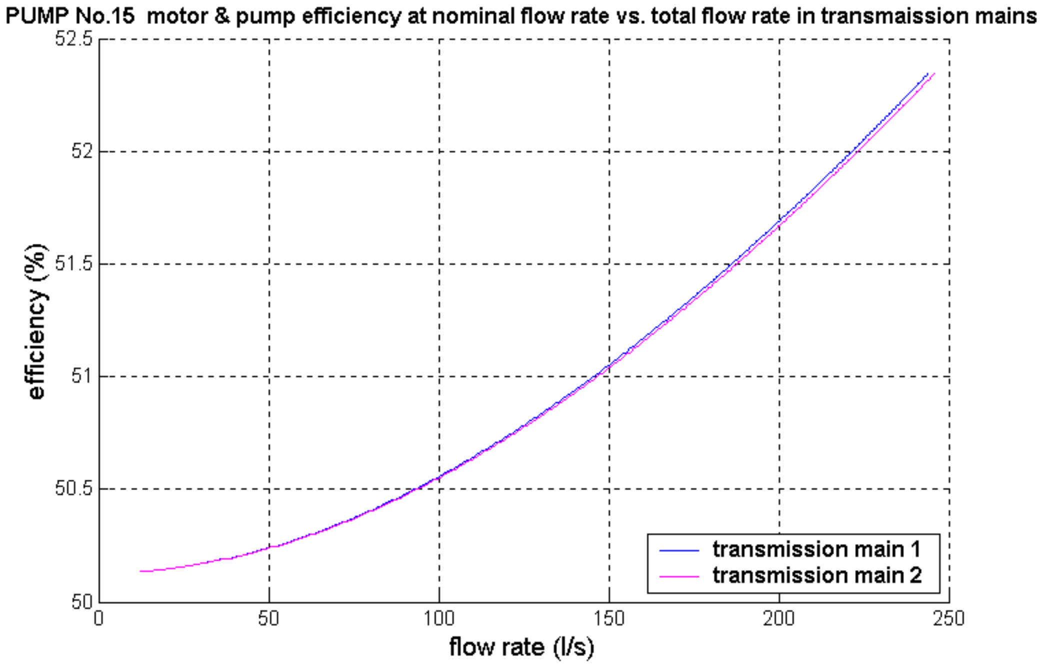
53.415
53.41
53.405
53.4
53.395
53.39
53.385
53.38
53.375
efficiency (%)
0
50
100
150
200
250
flow rate (l/s)
transmission main 1
transmission main 2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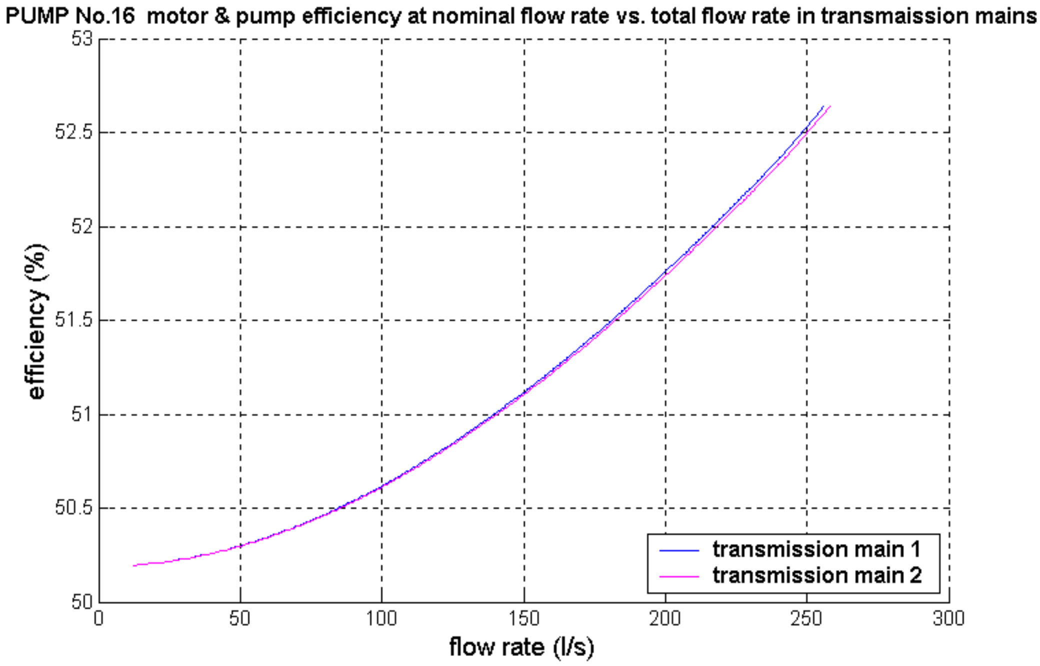
52.5
52
51.5
51
50.5
50
efficiency (%)
0
50
100
150
200
250
flow rate (l/s)
transmission main 1
transmission main 2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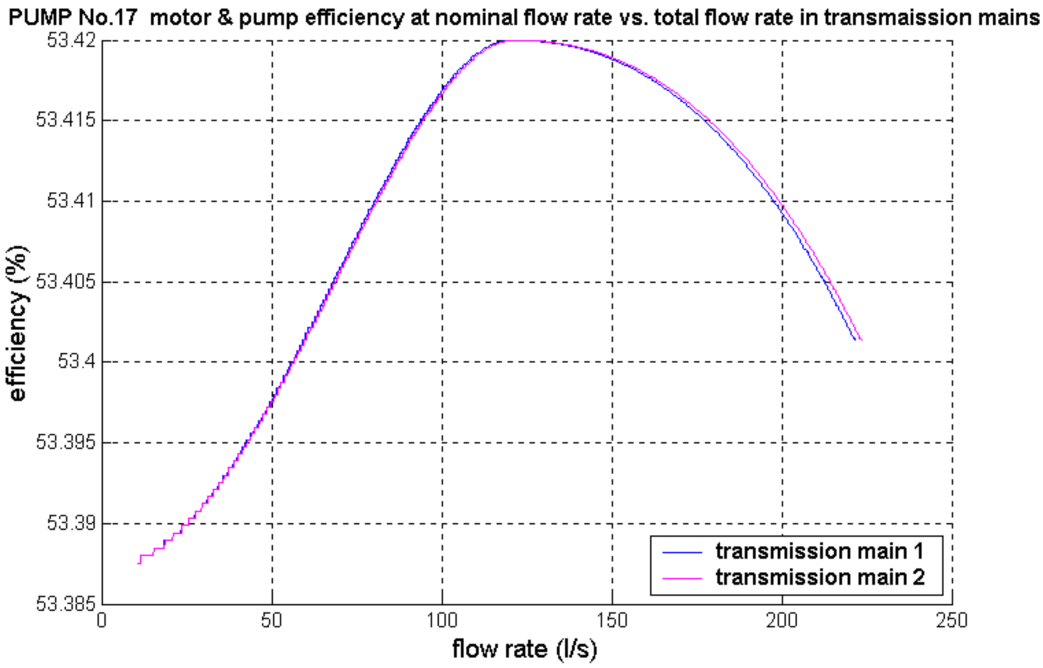
efficiency (%)
flow rate (l/s)
transmission main 1
transmission main 2
53
52.5
52
51.5
51
50.5
50
0
50
100
150
200
250
300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efficiency (%)
flow rate (l/s)
transmission main 1
transmission main 2
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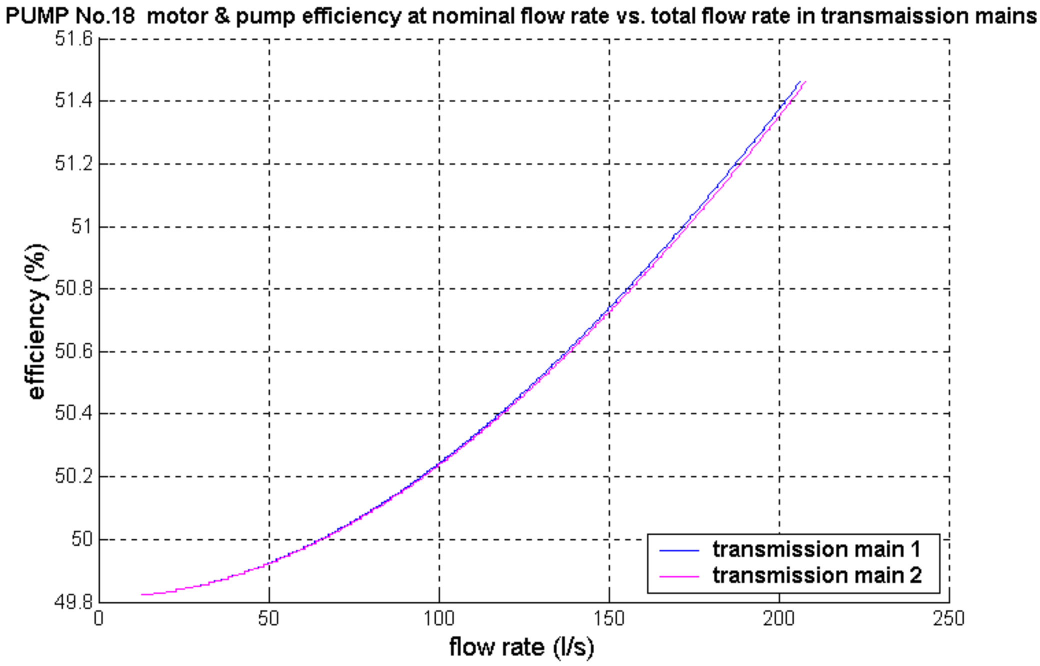
53.41
53.405
53.4
53.395
53.39
53.385
0
50
100
150
200
250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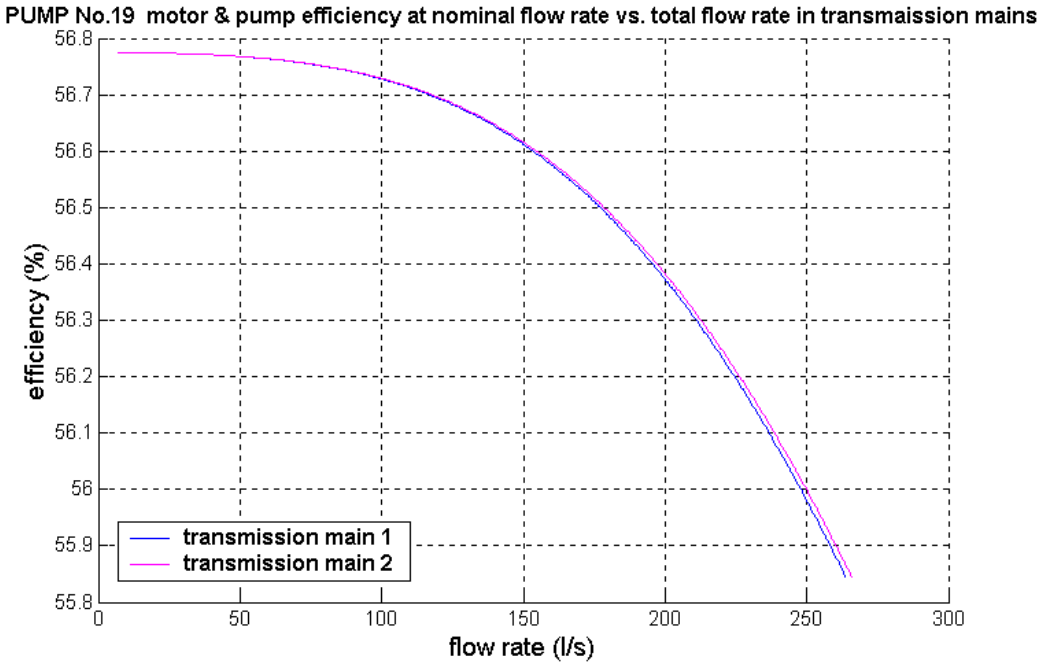
efficiency (%)
flow rate (l/s)
transmission main 1
transmission main 2
51.6
51.4
51.2
51
50.8
50.6
50.4
50.2
50
49.8
0
50
100
150
200
250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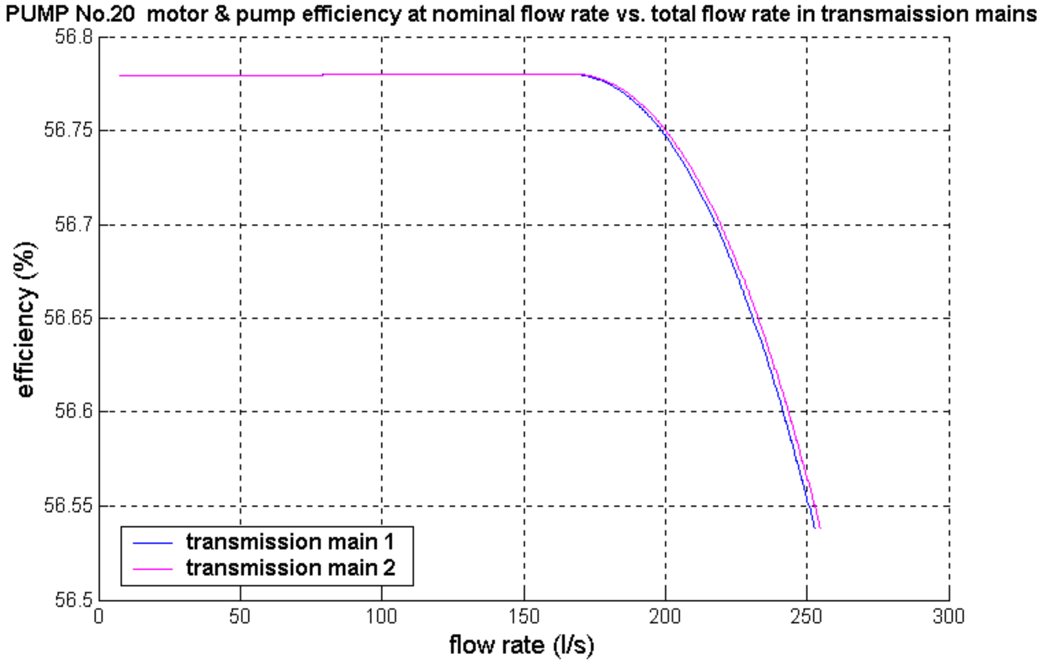
efficiency (%)
flow rate (l/s)
transmission main 1
transmission main 2
56.8
56.7
56.6
56.5
56.4
56.3
56.2
56.1
56
55.9
55.8
0
50
100
150
200
250
300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efficiency (%)
56.8
56.75
56.7
56.65
56.6
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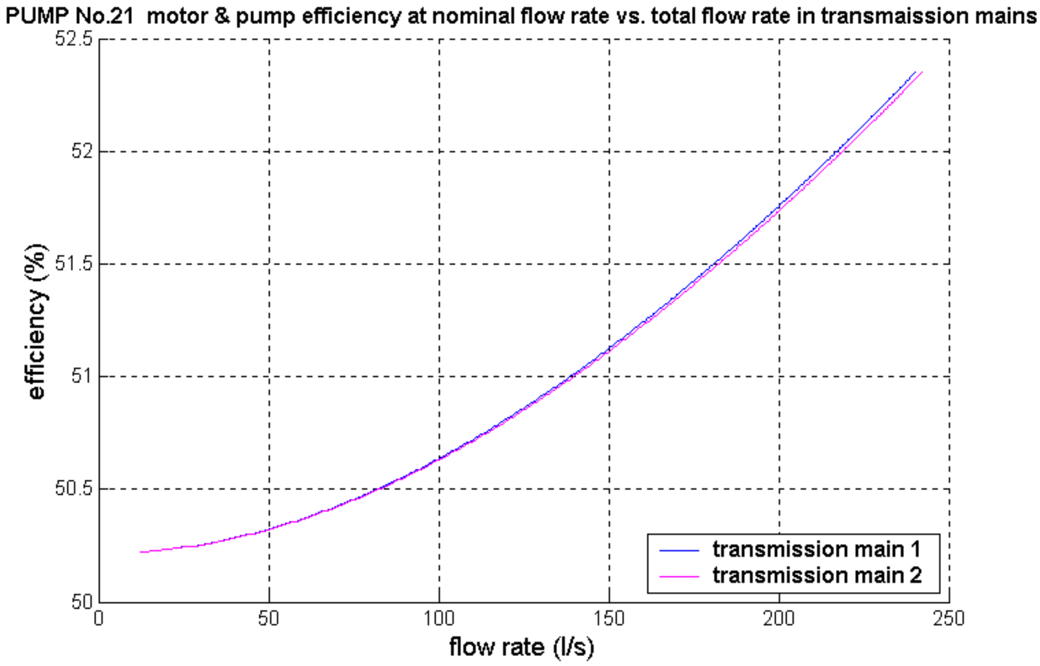
56.5
0
50
100
150
200
250
300
flow rate (l/s)
transmission main 1
transmission main 2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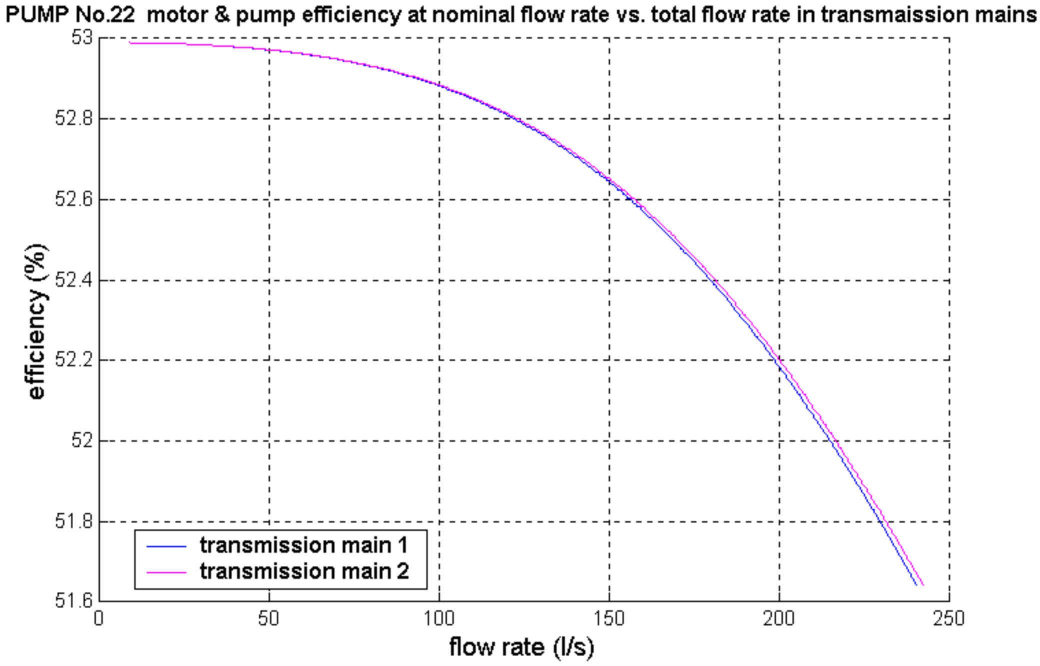
efficiency (%)
52.5
52
51.5
51
50.5
50
0
50
100
150
200
250
flow rate (l/s)
transmission main 1
transmission main 2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efficiency (%)
flow rate (l/s)
transmission main 1
transmission main 2
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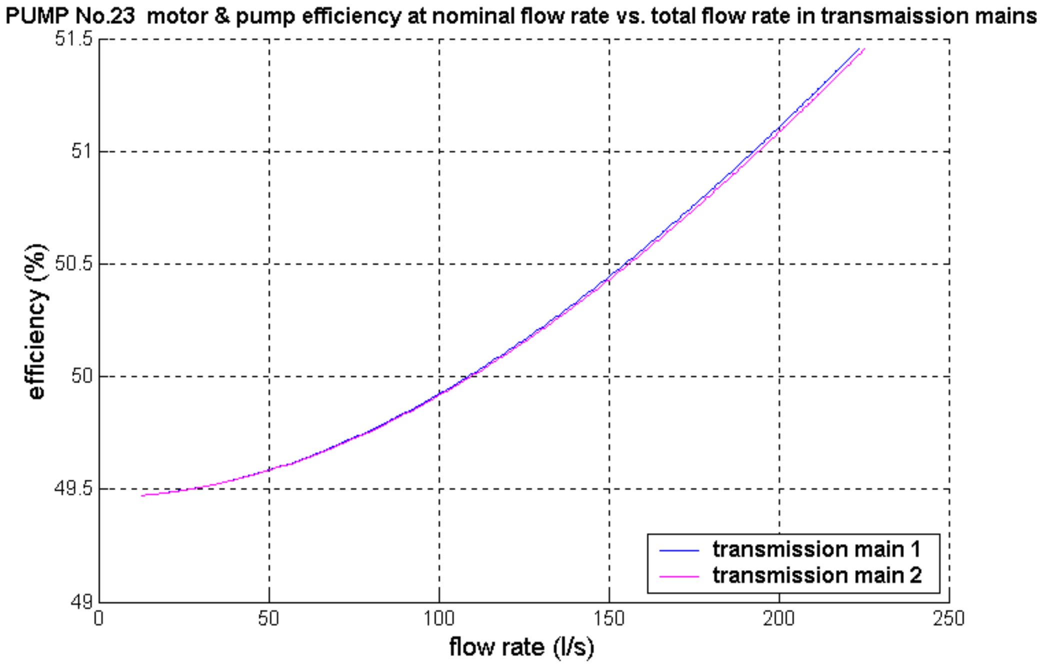
52.8
52.6
52.4
52.2
52
51.8
51.6
0
50
100
150
200
250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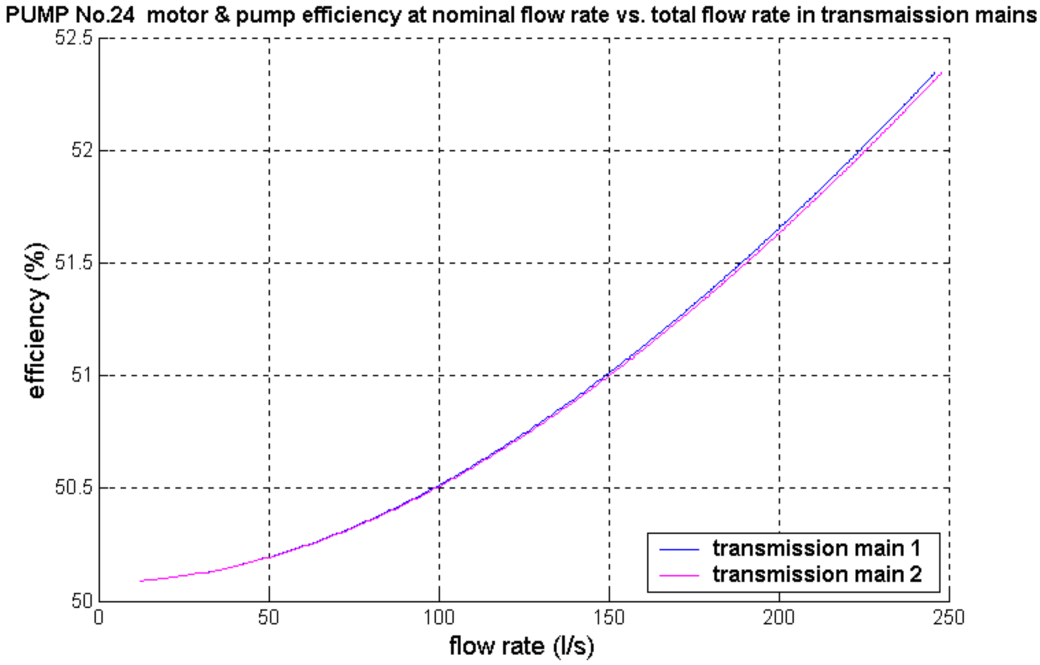
efficiency (%)
flow rate (l/s)
transmission main 1
transmission main 2
51.5
51
50.5
50
49.5
49
0
50
100
150
200
250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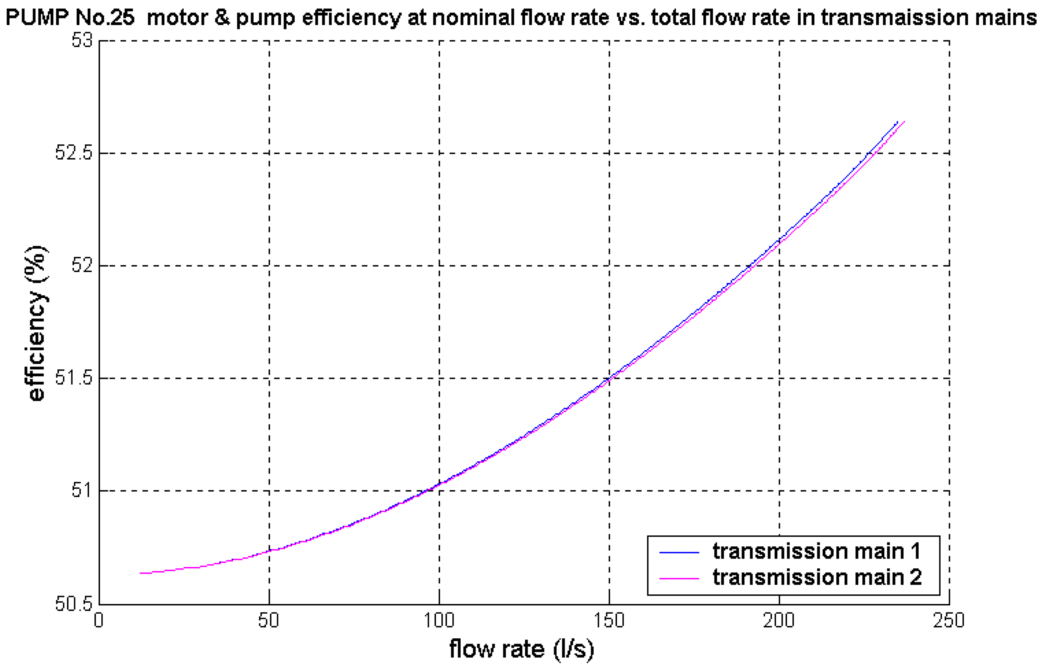
efficiency (%)
52.5
52
51.5
51
50.5
50
0
50
100
150
200
250
flow rate (l/s)
transmission main 1
transmission main 2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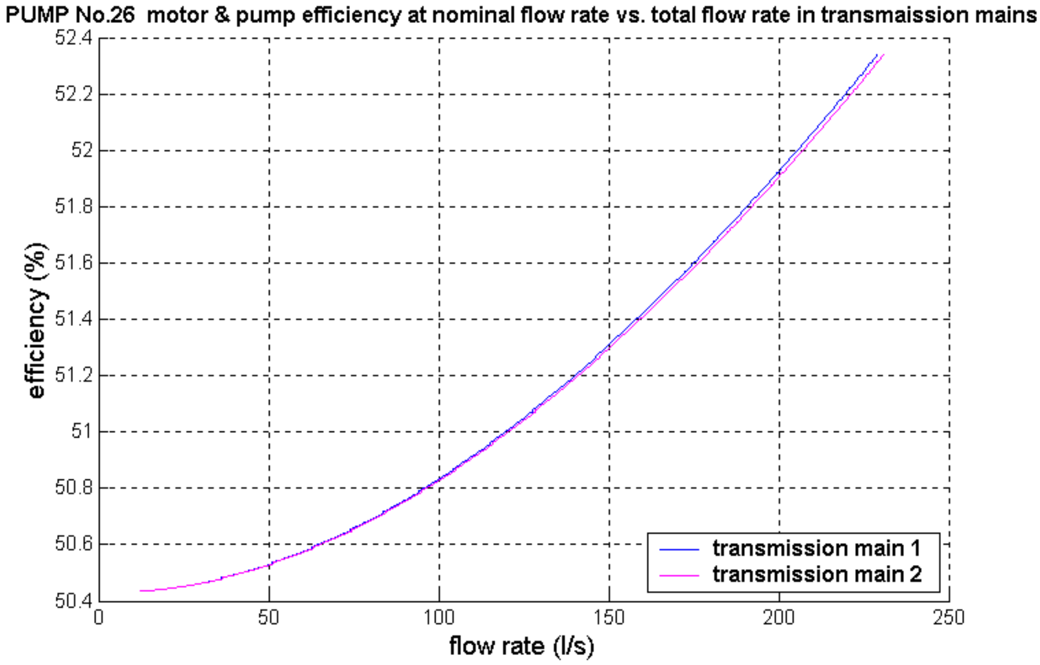
efficiency (%)
53
52.5
52
51.5
51
50.5
0
50
100
150
200
250
flow rate (l/s)
transmission main 1
transmission main 2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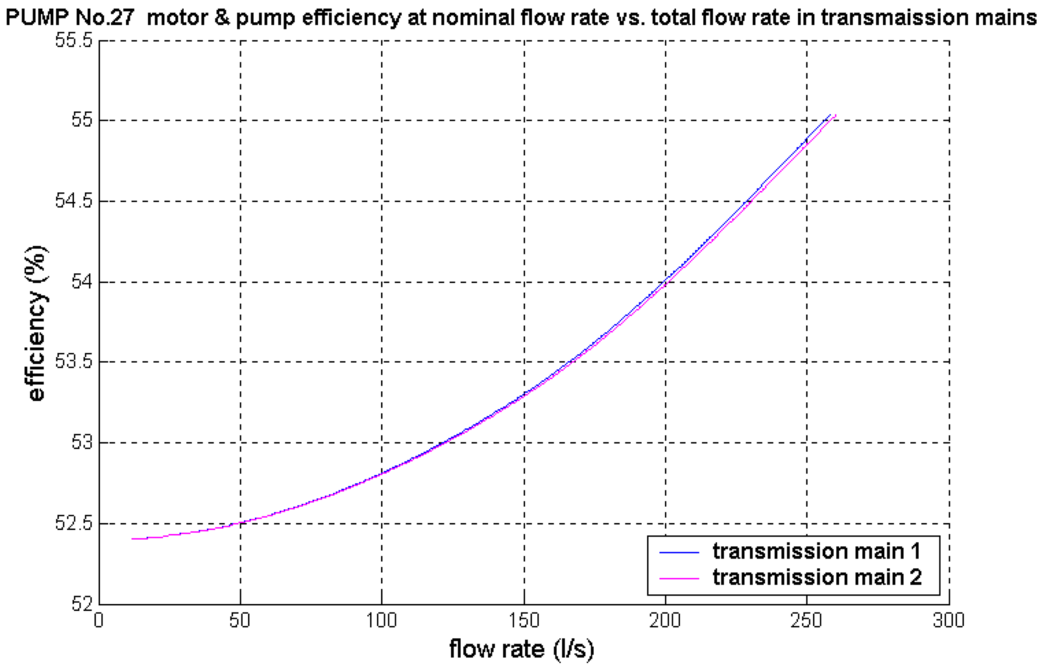
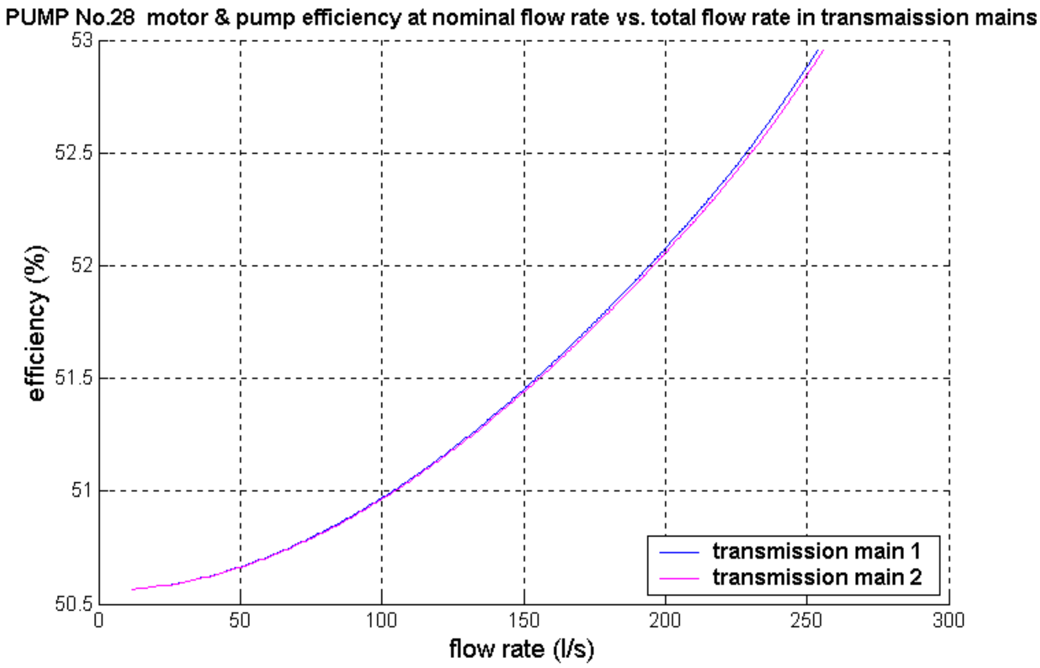
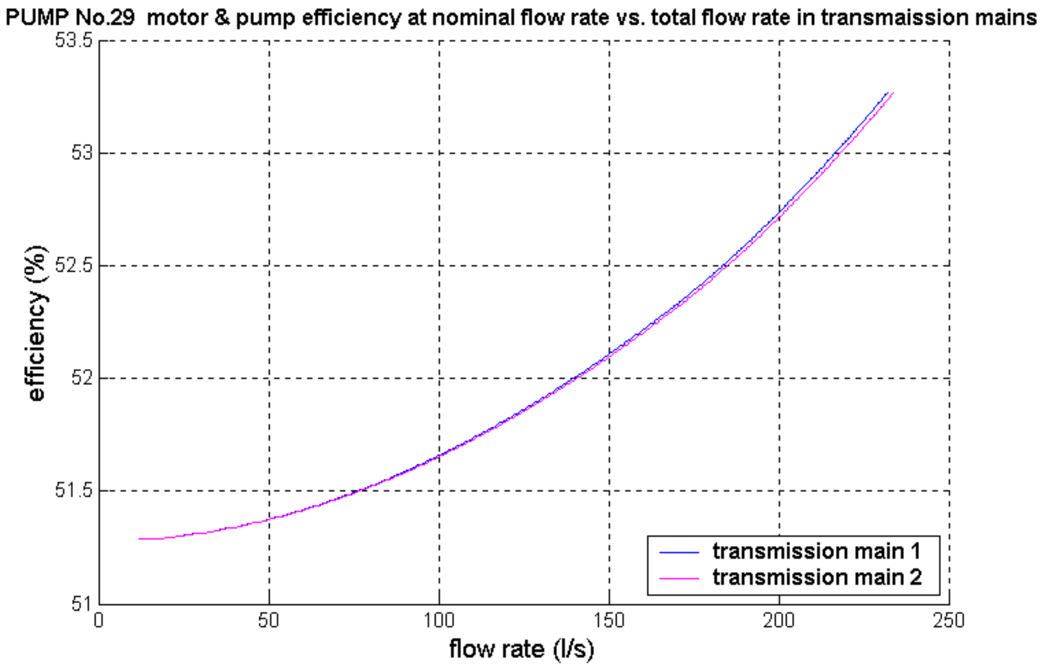
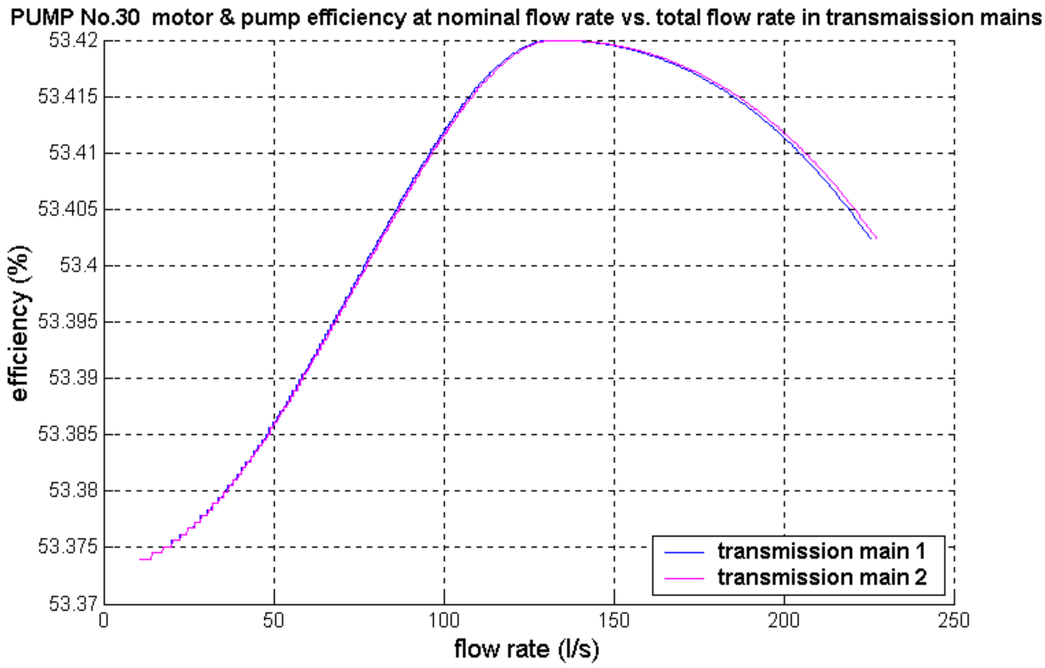
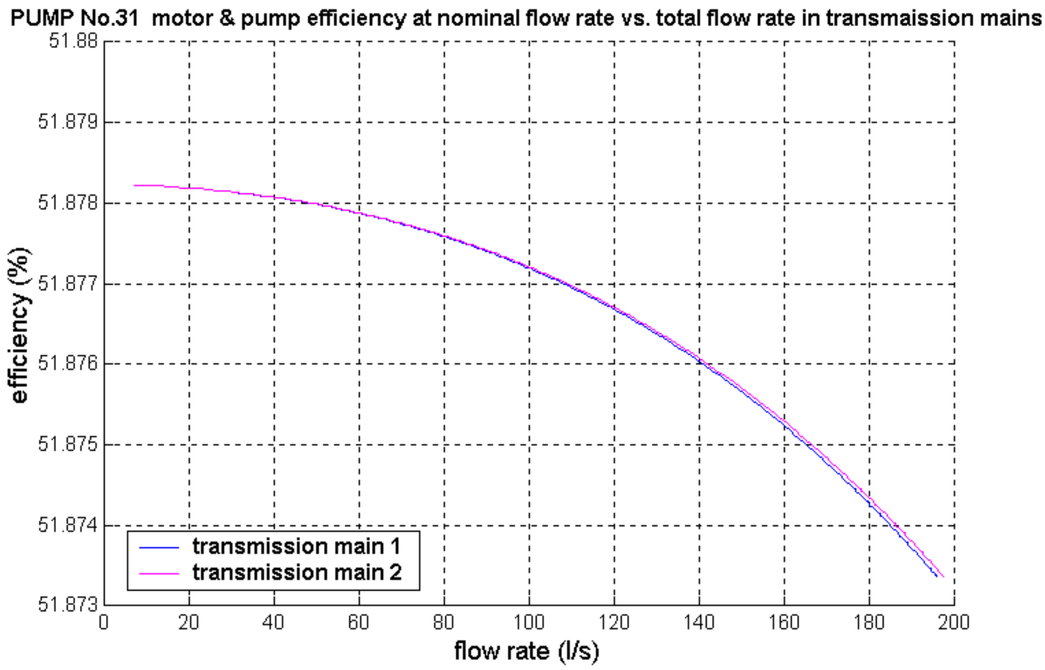
efficiency (%)
52.4
52.2
52
51.8
51.6
51.4
51.2
51
50.8
50.6
50.4
0
50
100
150
200
250
flow rate (l/s)
transmission main 1
transmission main 2

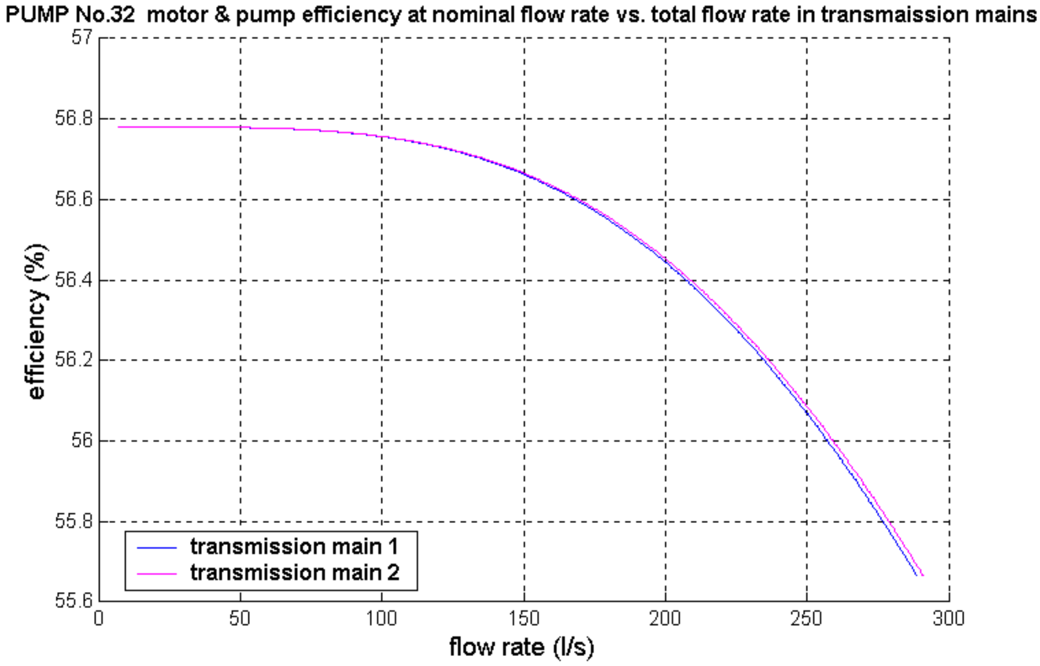

PUMP No.27 motor & pump efficiency at nominal flow rate vs. total flow rate in transmaission mains
efficiency (%)
55.5
55
54.5
54
53.5
53
52.5
52
0
50
100
150
200
250
300
flow rate (l/s)
transmission main 1
transmission main 2

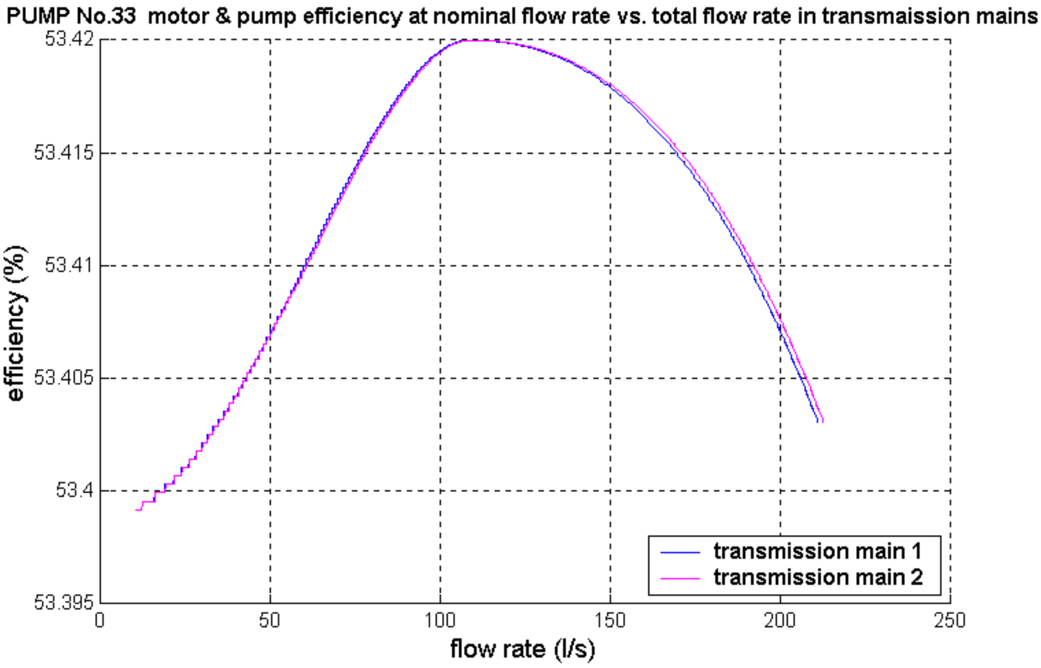

PUMP No.28 motor & pump efficiency at nominal flow rate vs. total flow rate in transmaission mains
efficiency (%)
flow rate (l/s)
transmission main 1
transmission main 2
53
52.5
52
51.5
51
50.5
0
50
100
150
200
250
300


PUMP No.29 motor & pump efficiency at nominal flow rate vs. total flow rate in transmaission mains
efficiency (%)
flow rate (l/s)
transmission main 1
transmission main 2
53.5
53
52.5
52
51.5
51
0
50
100
150
200
250


PUMP No.30 motor & pump efficiency at nominal flow rate vs. total flow rate in transmaission mains
53.42
53.415
53.41
53.405
53.4
53.395
53.39
53.385
53.38
53.375
53.37
efficiency (%)
0
50
100
150
200
250
flow rate (l/s)
transmission main 1
transmission main 2


PUMP No.31 motor & pump efficiency at nominal flow rate vs. total flow rate in transmaission mains
51.88
51.879
51.878
51.877
51.876
51.875
51.874
51.873
efficiency (%)
0
20
40
60
80
100
120
140
160
180
200
flow rate (l/s)
transmission main 1
transmission main 2


PUMP No.32 motor & pump efficiency at nominal flow rate vs. total flow rate in transmaission mains
efficiency (%)
flow rate (l/s)
57
56.8
56.6
56.4
56.2
56
55.8
55.6
0
50
100
150
200
250
300
transmission main 1
transmission main 2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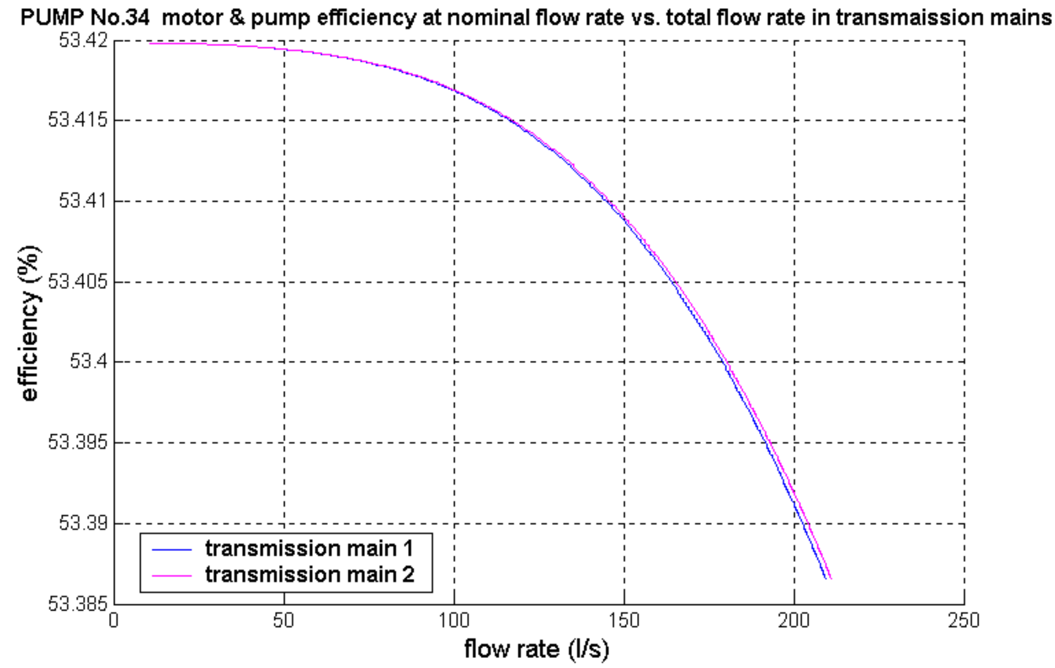
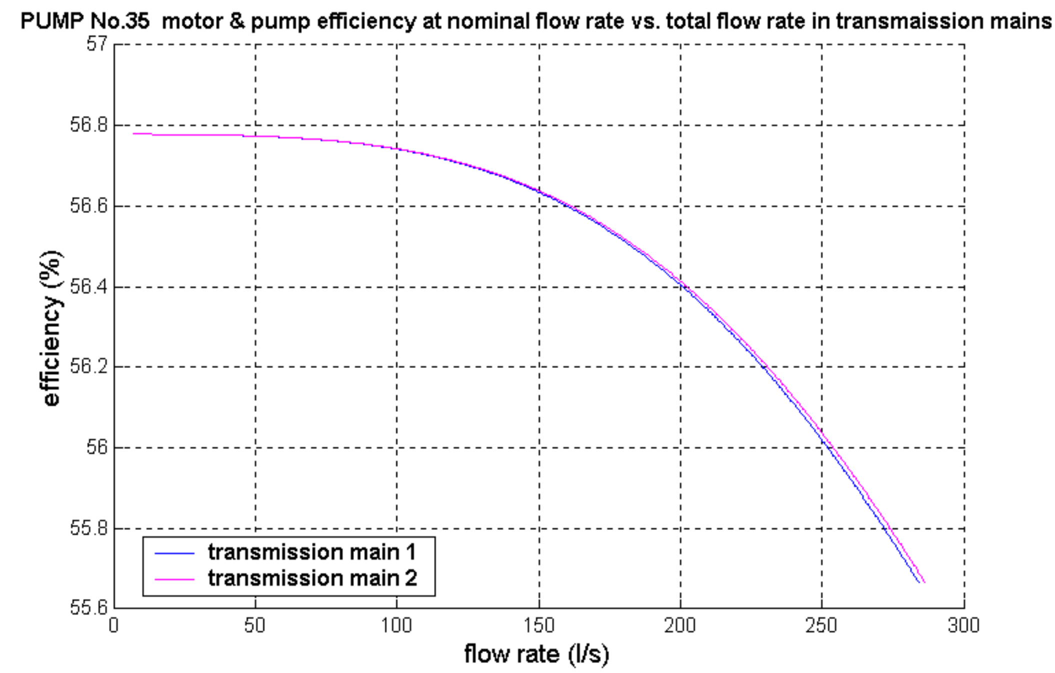
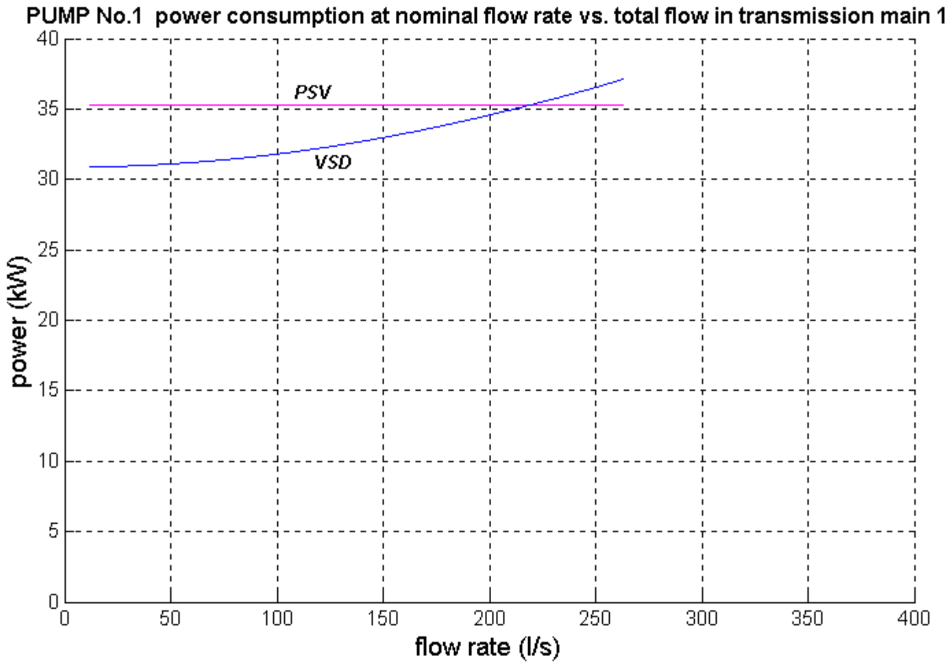
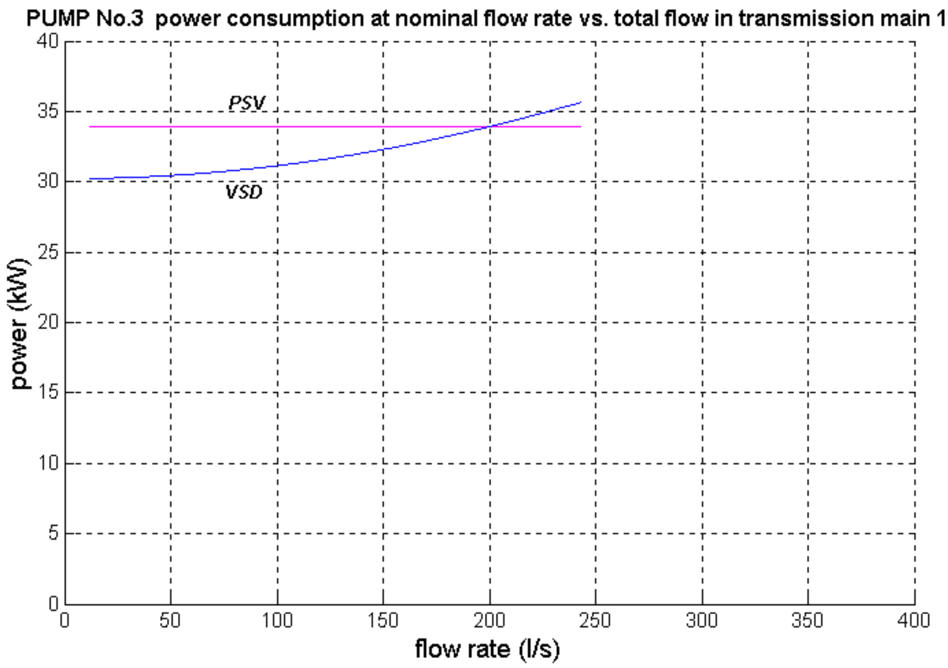
efficiency (%)
flow rate (l/s)
53.42
53.415
53.41
53.405
53.4
53.395
0
50
100
150
200
250
transmission main 1
transmission main 2

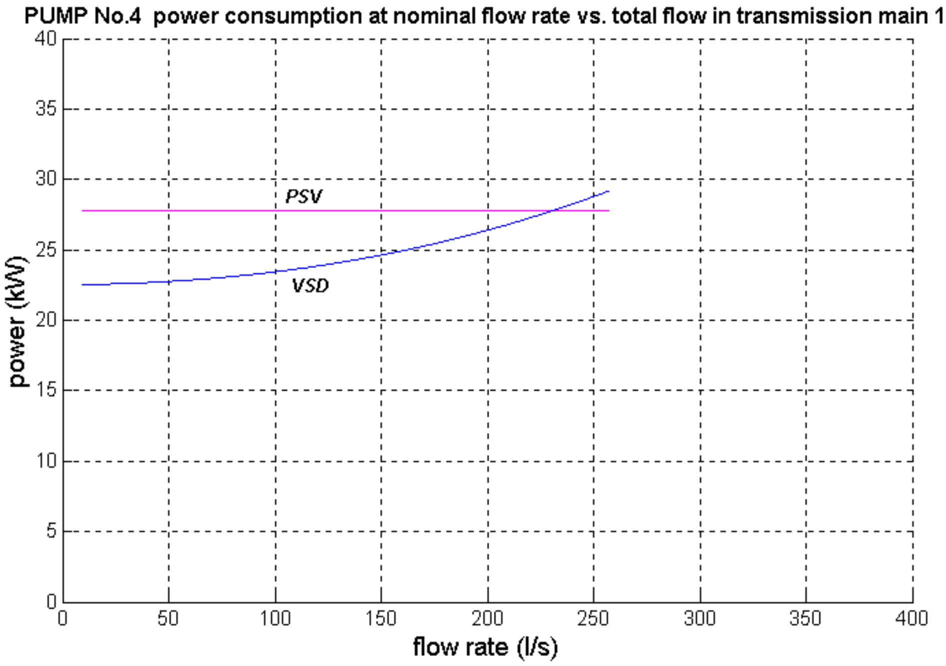

PUMP No.34 motor & pump efficiency at nominal flow rate vs. total flow rate in transmaission mains
efficiency (%)
flow rate (l/s)
transmission main 1
transmission main 2
53.42
53.415
53.41
53.405
53.4
53.395
53.39
53.385
0
50
100
150
200
250

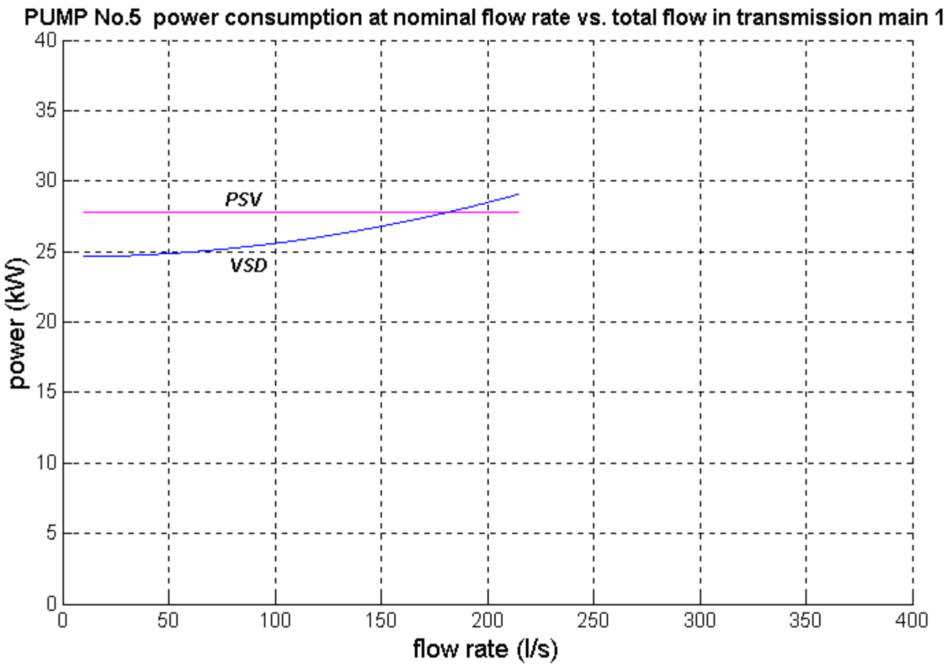

PUMP No.35 motor & pump efficiency at nominal flow rate vs. total flow rate in transmaission mains
efficiency (%)
flow rate (l/s)
transmission main 1
transmission main 2
57
56.8
56.6
56.4
56.2
56
55.8
55.6
0
50
100
150
200
250
300

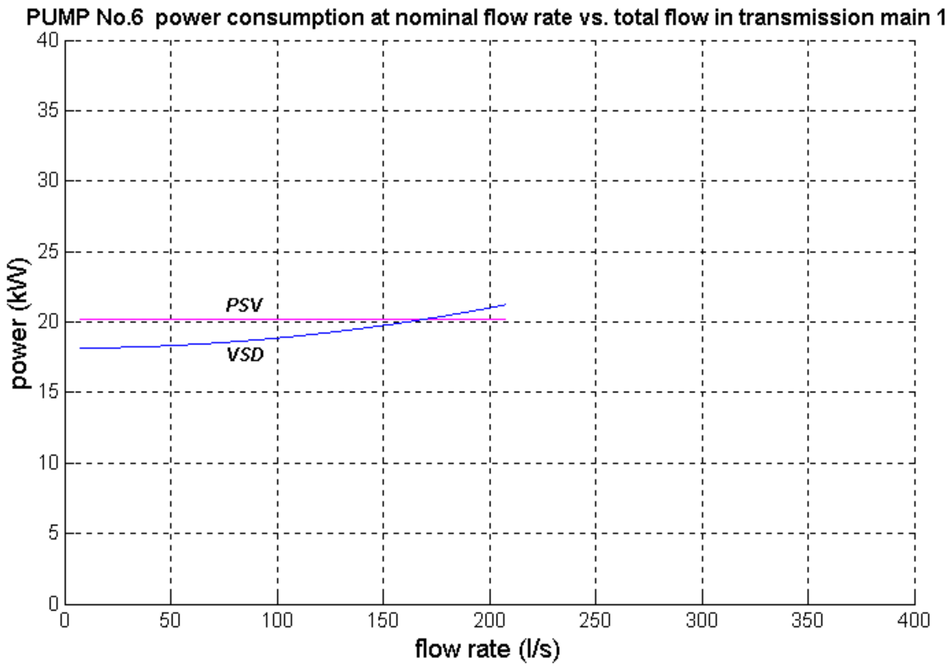
Appendix B: plots of power consumption against total flow rate in transmission main 1 - for individual pumps (pumps No.2 and No.36 are omitted due to unavailability of data)

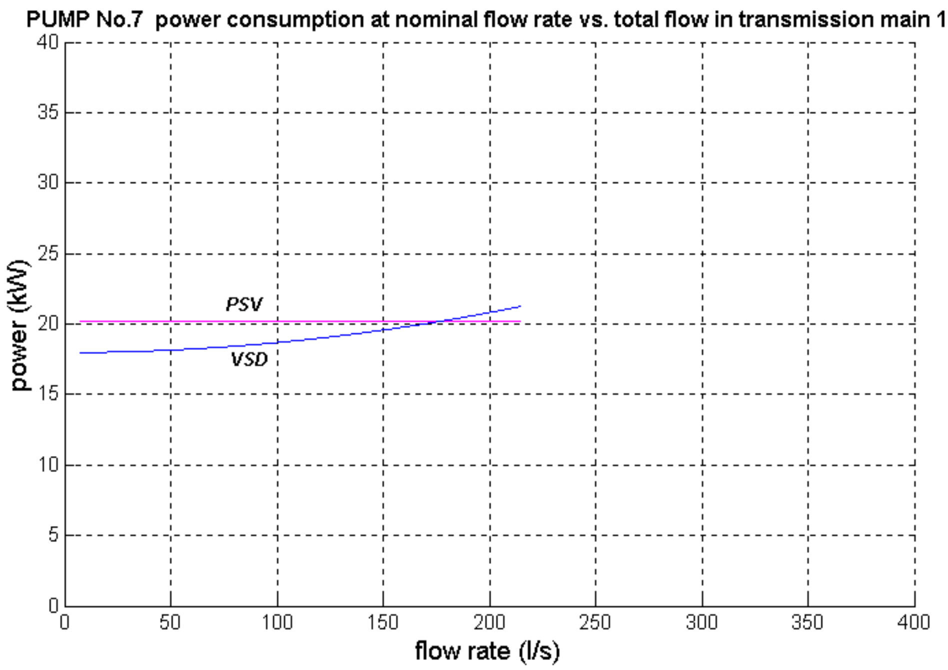


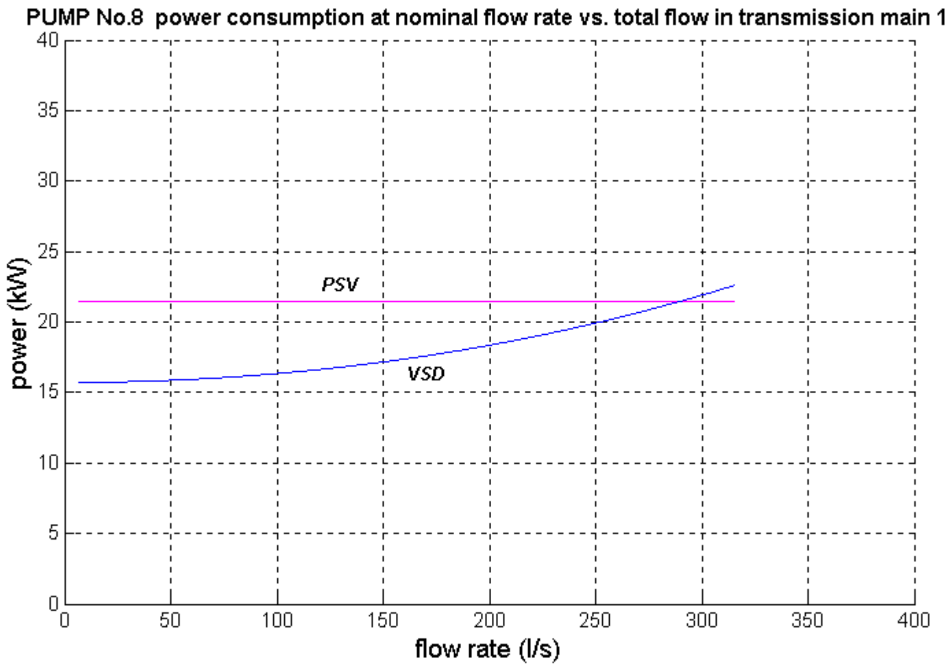


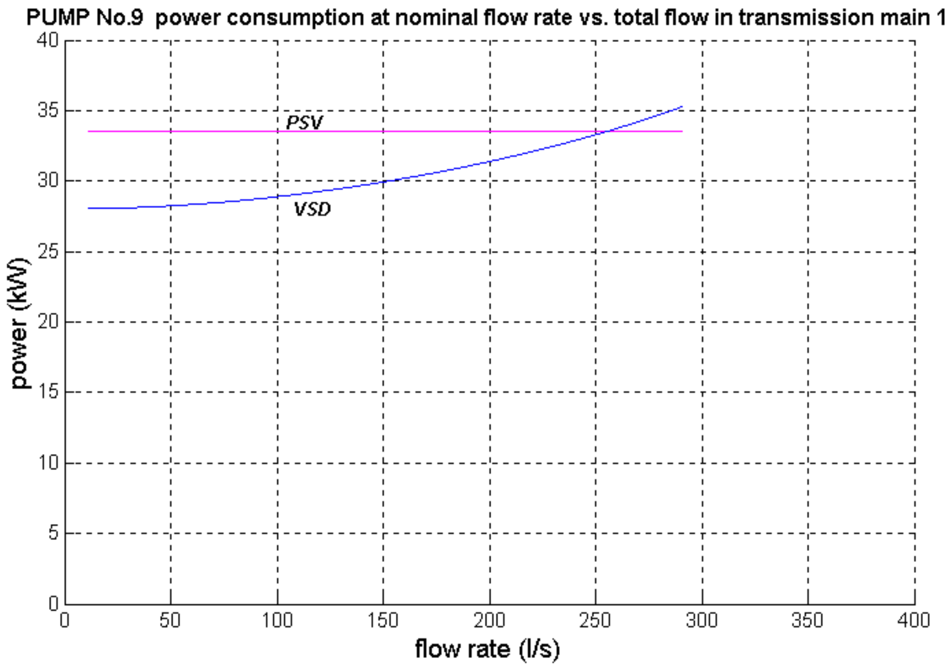

PUMP No.4 power consumption at nominal flow rate vs. total flow in transmission main 1
PSV
VSD
power (kW)
flow rate (l/s)
0
5
10
15
20
25
30
35
40
0
50
100
150
200
250
300
350
400

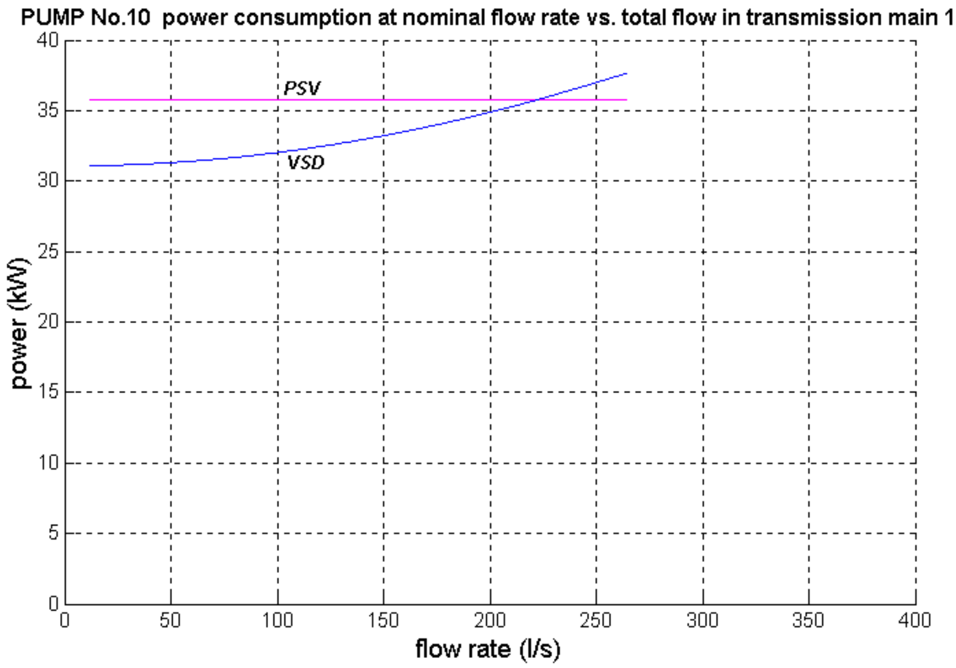

PUMP No.5 power consumption at nominal flow rate vs. total flow in transmission main 1
PSV
VSD
power (kW)
flow rate (l/s)
0
5
10
15
20
25
30
35
40
0
50
100
150
200
250
300
350
400

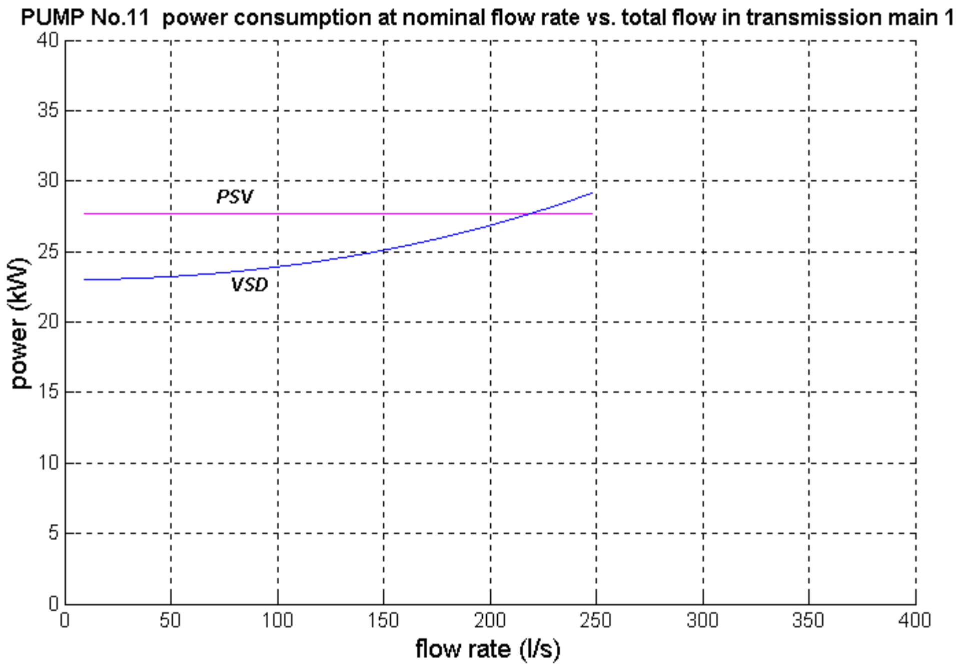

PUMP No.6 power consumption at nominal flow rate vs. total flow in transmission main 1
power (kW)
flow rate (l/s)
PSV
VSD
0
5
10
15
20
25
30
35
40
0
50
100
150
200
250
300
350
400


PUMP No.7 power consumption at nominal flow rate vs. total flow in transmission main 1
power (kW)
flow rate (l/s)
PSV
VSD
0
5
10
15
20
25
30
35
40
0
50
100
150
200
250
300
350
400

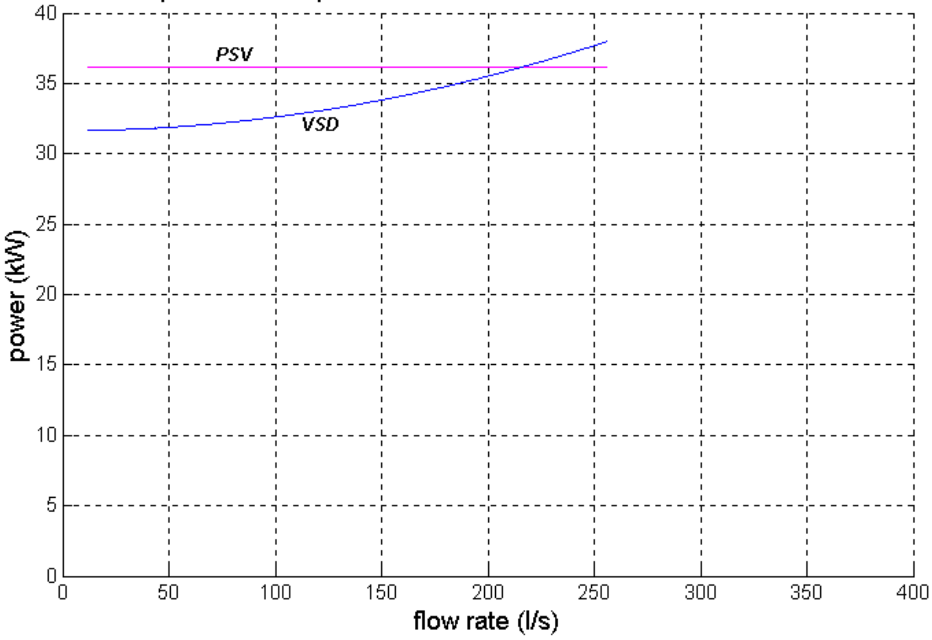

PUMP No.8 power consumption at nominal flow rate vs. total flow in transmission main 1
PSV
VSD
power (kW)
flow rate (l/s)
0
5
10
15
20
25
30
35
40
0
50
100
150
200
250
300
350
400


PUMP No.9 power consumption at nominal flow rate vs. total flow in transmission main 1
PSV
VSD
power (kW)
flow rate (l/s)
0
5
10
15
20
25
30
35
40
0
50
100
150
200
250
300
350
400

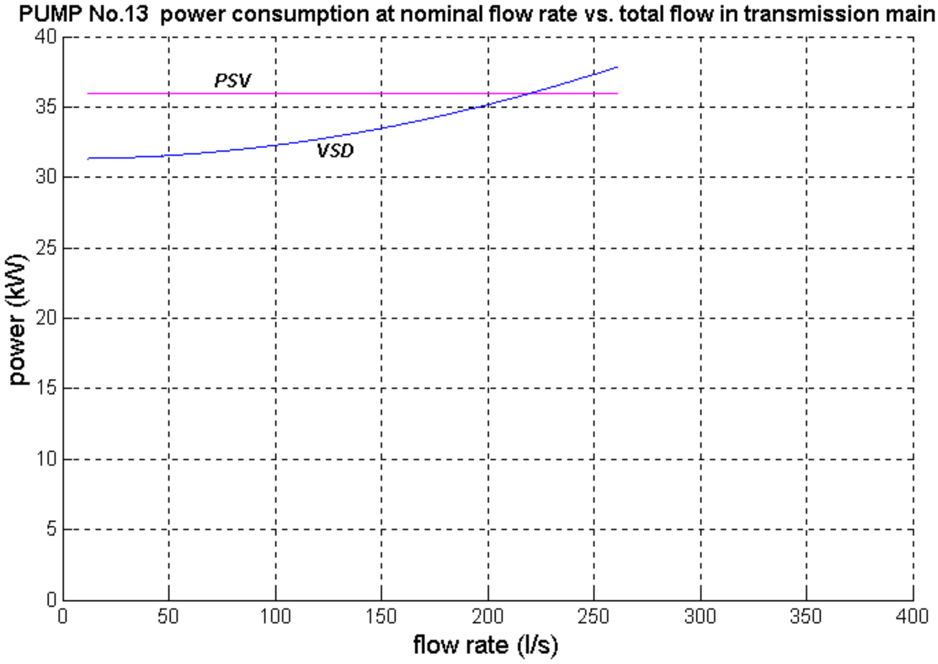

PUMP No.10 power consumption at nominal flow rate vs. total flow in transmission main 1
PSV
VSD
power (kW)
flow rate (l/s)
0
5
10
15
20
25
30
35
40
0
50
100
150
200
250
300
350
400

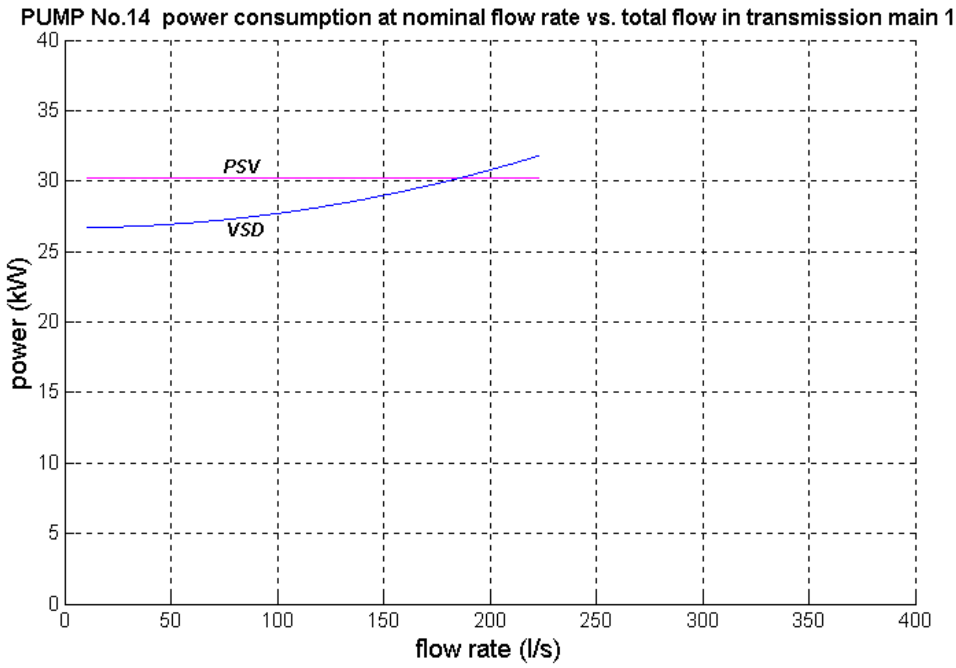

PUMP No.11 power consumption at nominal flow rate vs. total flow in transmission main 1
PSV
VSD
power (kW)
flow rate (l/s)
0
5
10
15
20
25
30
35
40
0
50
100
150
200
250
300
350
400

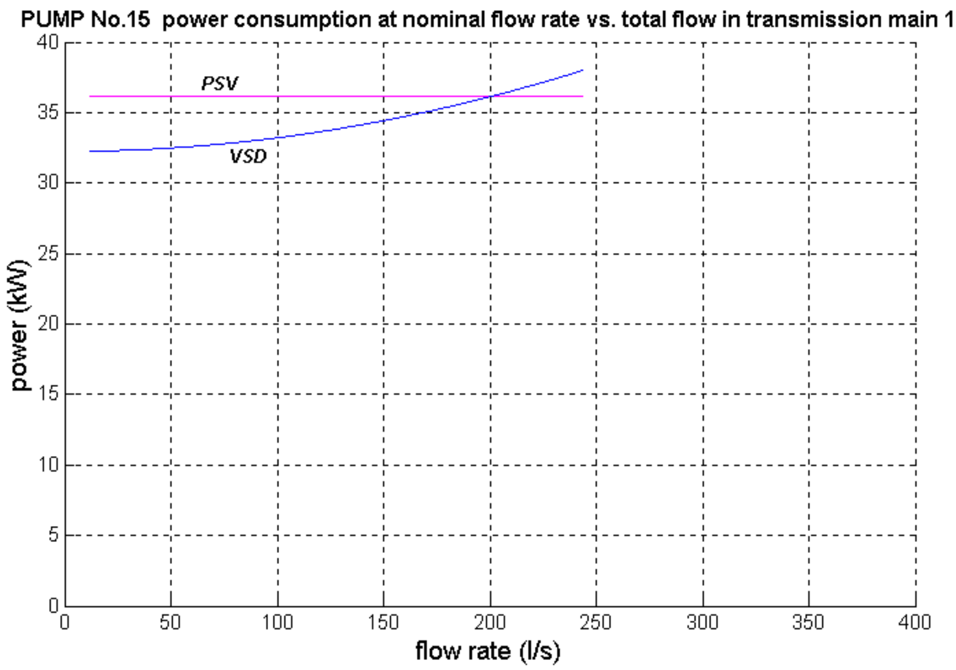
PUMP No.12 power consumption at nominal flow rate vs. total flow in transmission main 1

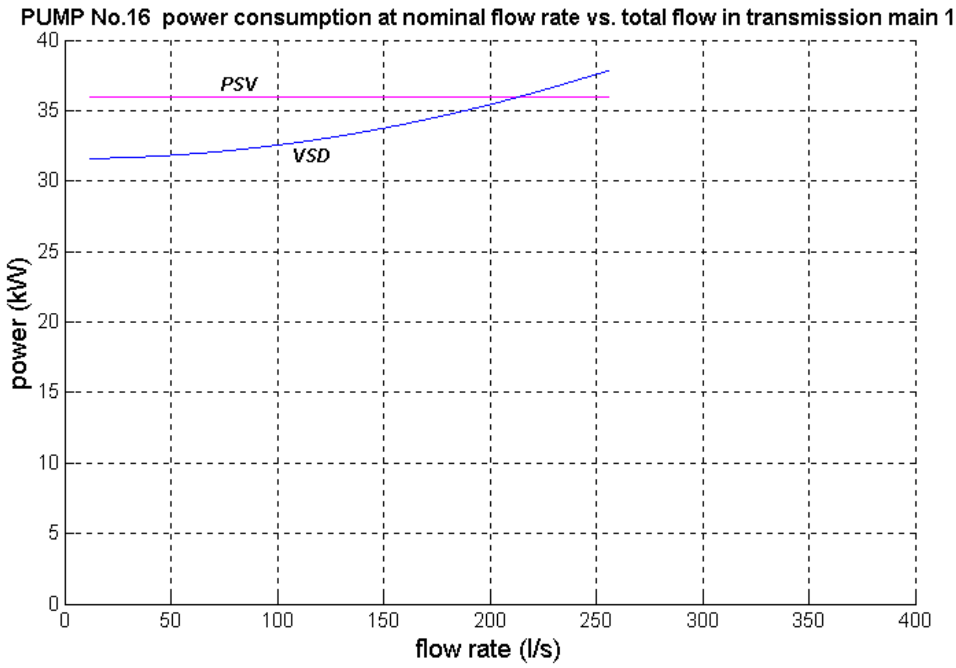


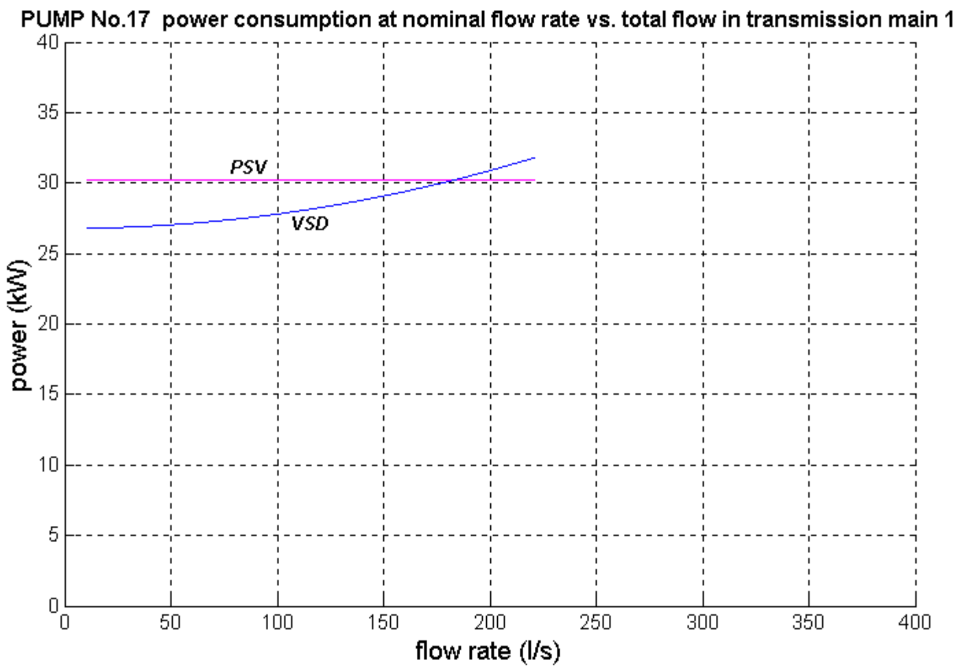
PUMP No.13 power consumption at nominal flow rate vs. total flow in transmission main 1

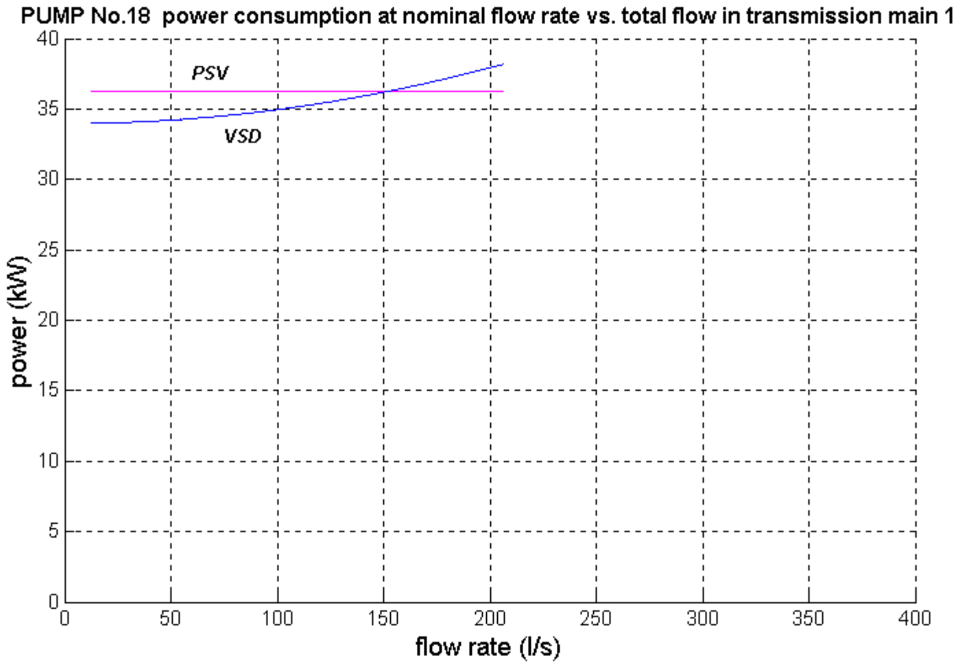


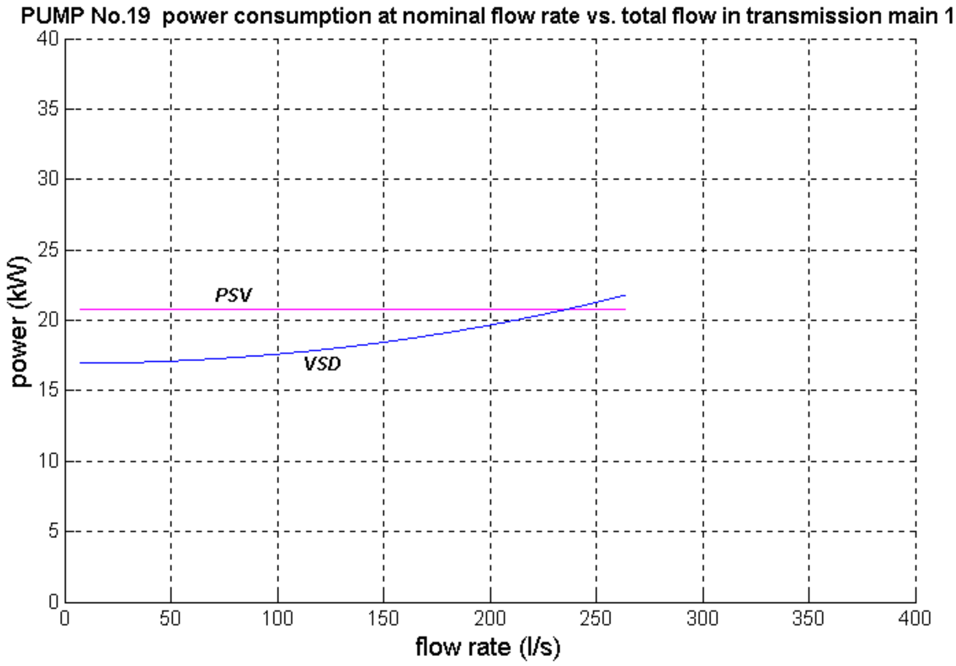

PUMP No.14 power consumption at nominal flow rate vs. total flow in transmission main 1
PSV
VSD
power (kW)
flow rate (l/s)
0
5
10
15
20
25
30
35
40
0
50
100
150
200
250
300
350
400

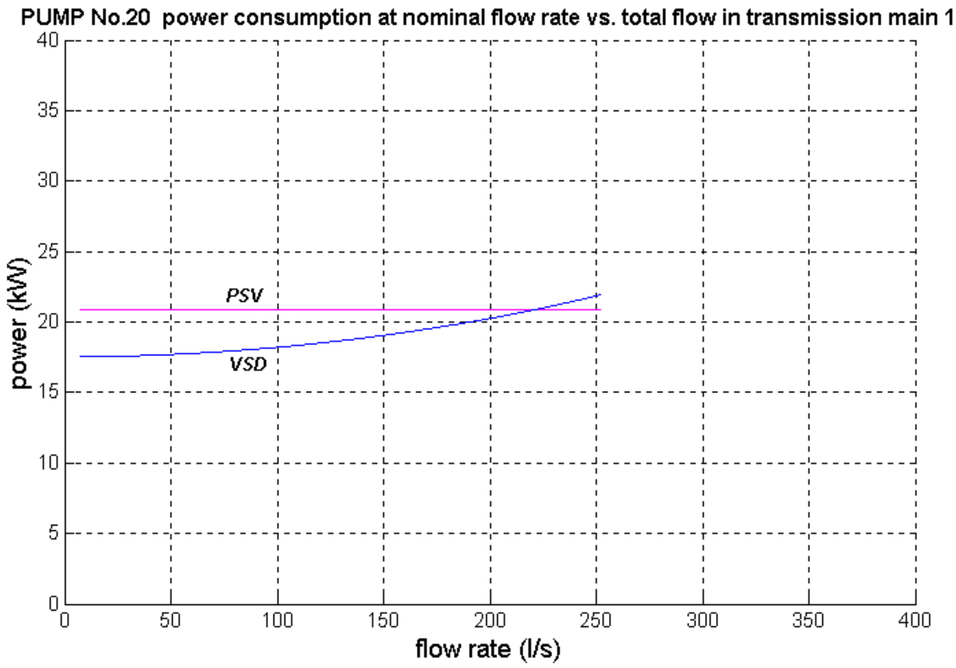

PUMP No.15 power consumption at nominal flow rate vs. total flow in transmission main 1
PSV
VSD
power (kW)
flow rate (l/s)
0
5
10
15
20
25
30
35
40
0
50
100
150
200
250
300
350
400

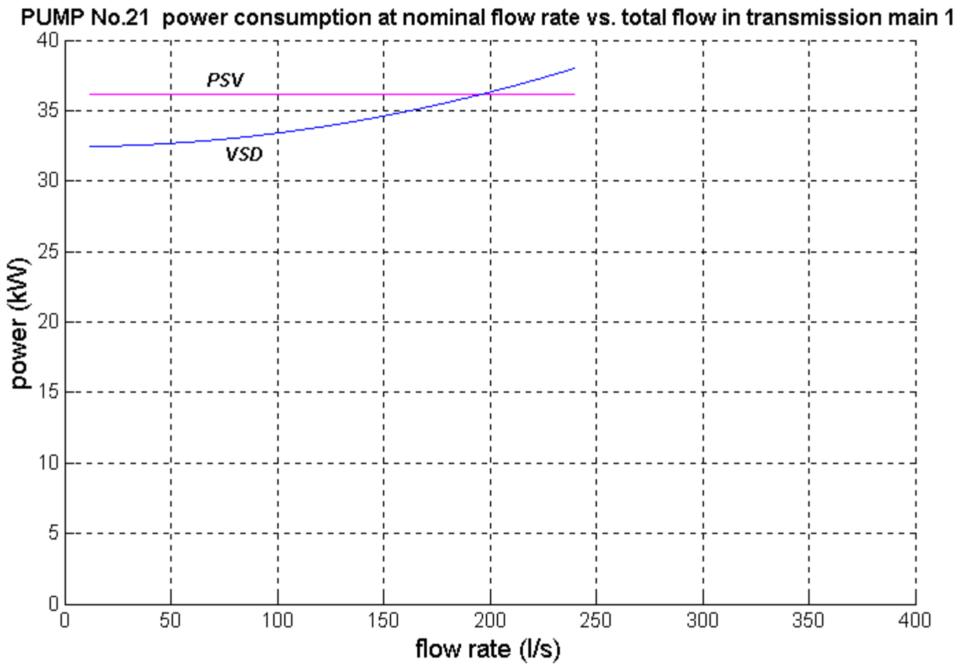

PUMP No.16 power consumption at nominal flow rate vs. total flow in transmission main 1
PSV
VSD
power (kW)
flow rate (l/s)
0
5
10
15
20
25
30
35
40
0
50
100
150
200
250
300
350
400

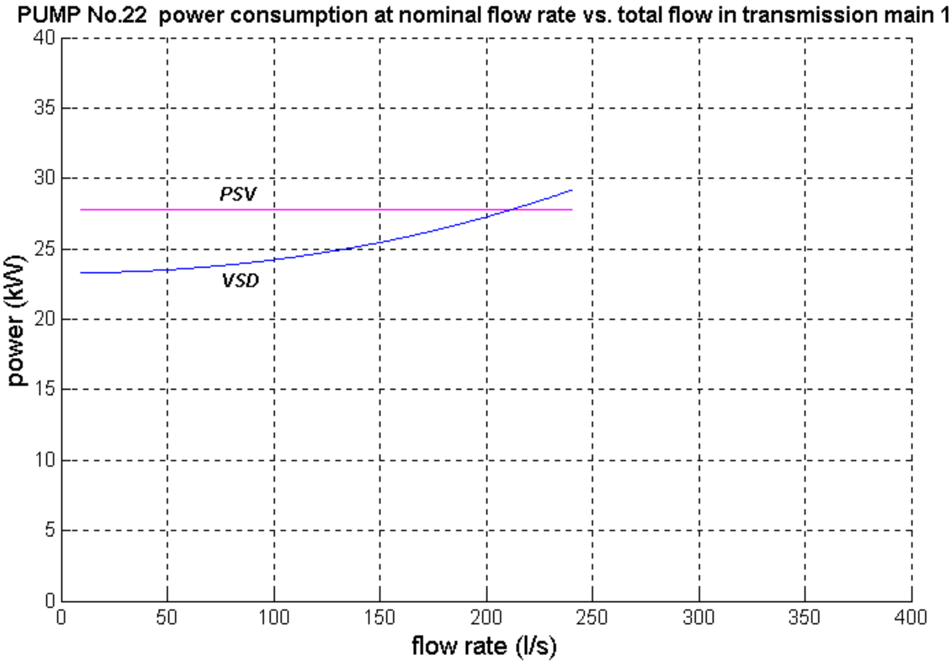

PUMP No.17 power consumption at nominal flow rate vs. total flow in transmission main 1
PSV
VSD
power (kW)
flow rate (l/s)
0
5
10
15
20
25
30
35
40
0
50
100
150
200
250
300
350
400

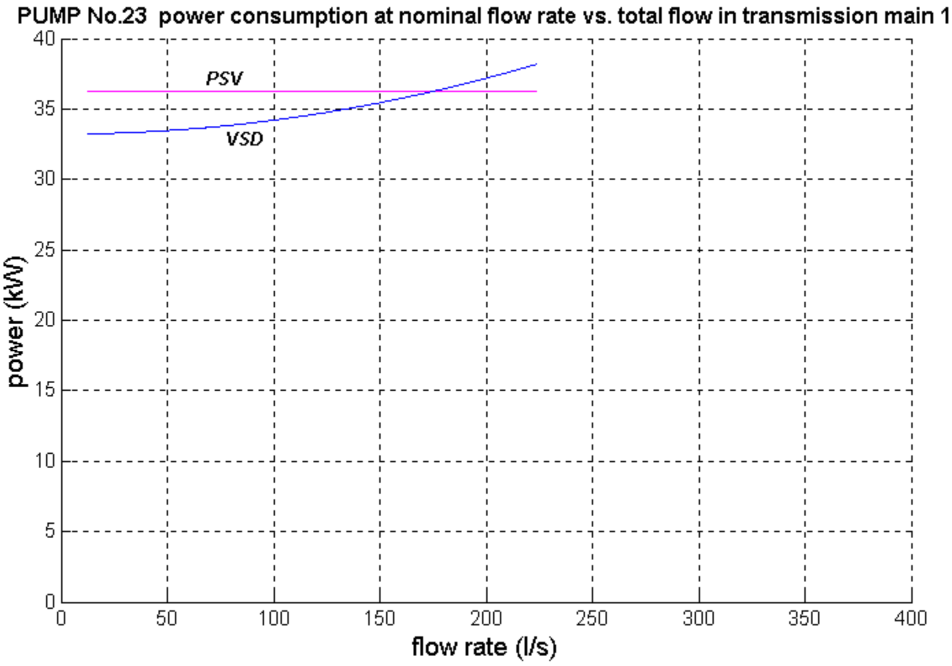

PUMP No.18 power consumption at nominal flow rate vs. total flow in transmission main 1
PSV
VSD
power (kW)
flow rate (l/s)
0
5
10
15
20
25
30
35
40
0
50
100
150
200
250
300
350
400

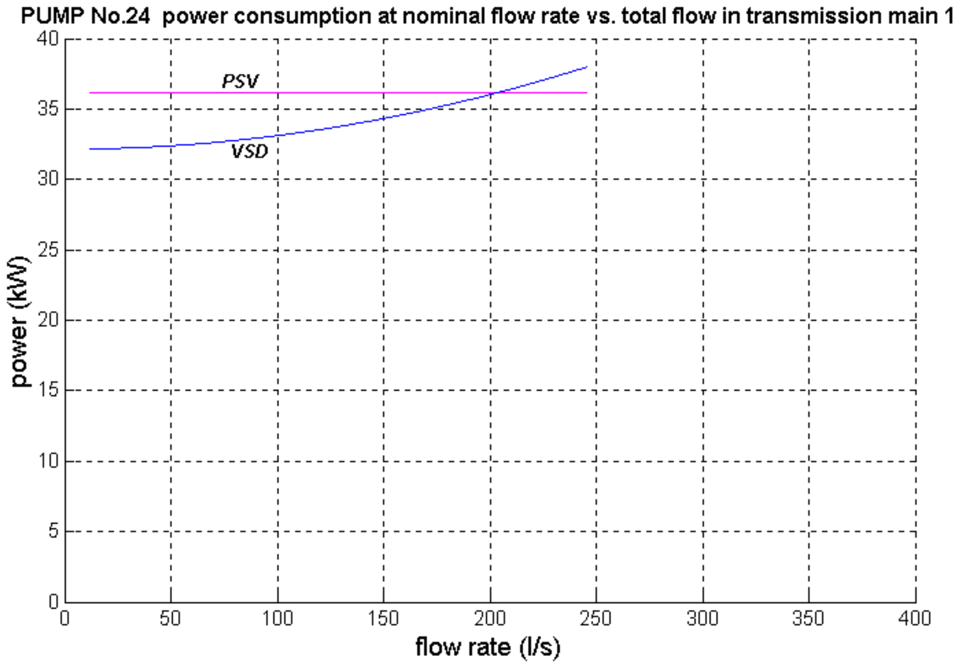

PUMP No.19 power consumption at nominal flow rate vs. total flow in transmission main 1
PSV
VSD
power (kW)
flow rate (l/s)
0
5
10
15
20
25
30
35
40
0
50
100
150
200
250
300
350
400

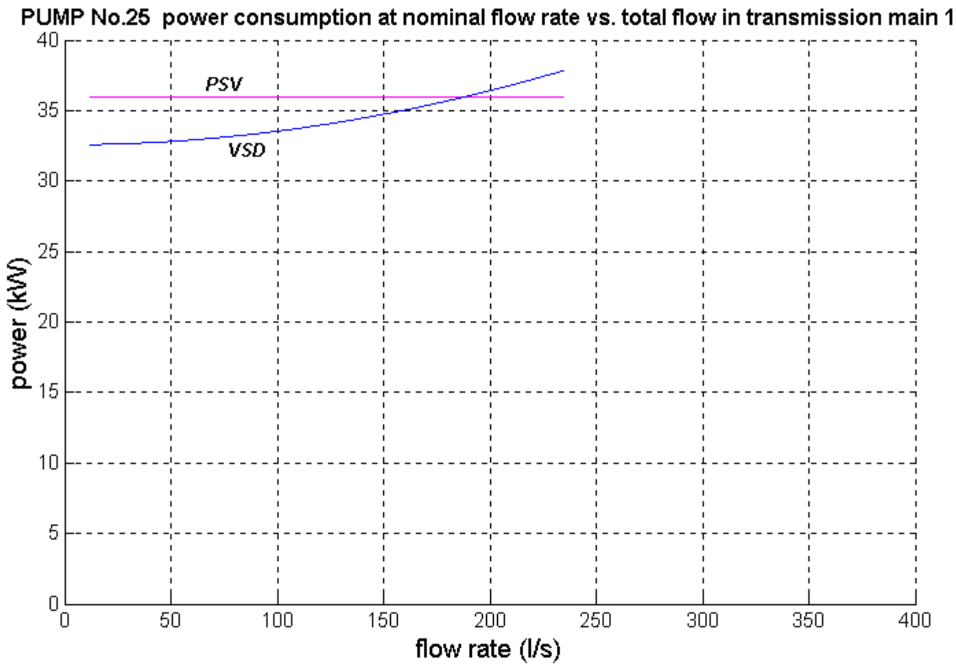

PUMP No.20 power consumption at nominal flow rate vs. total flow in transmission main 1
power (kW)
flow rate (l/s)
PSV
VSD
0
5
10
15
20
25
30
35
40
0
50
100
150
200
250
300
350
400

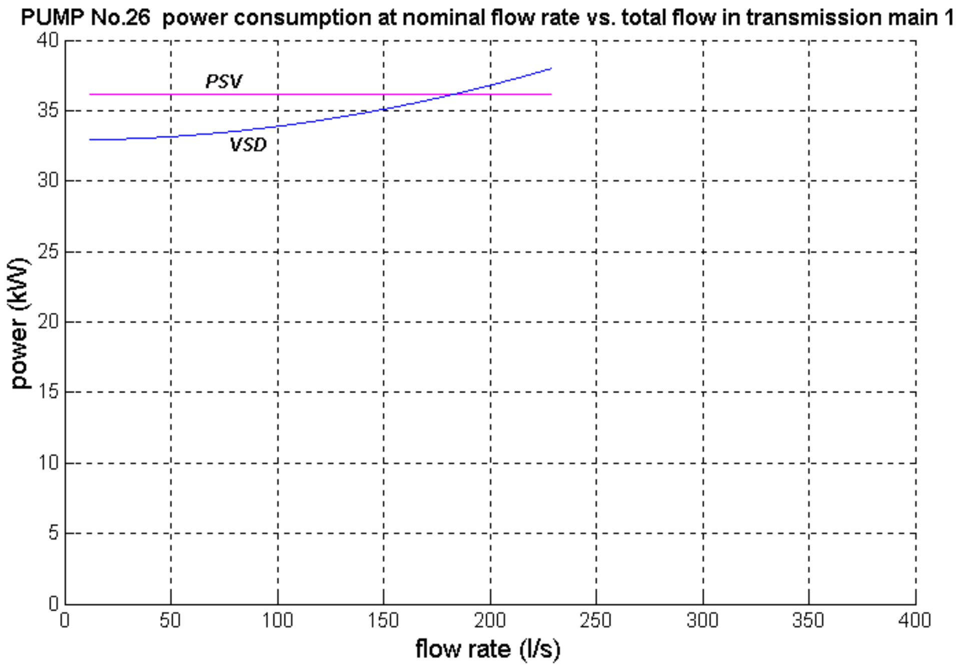

PUMP No.21 power consumption at nominal flow rate vs. total flow in transmission main 1
power (kW)
flow rate (l/s)
PSV
VSD
0
5
10
15
20
25
30
35
40
0
50
100
150
200
250
300
350
400

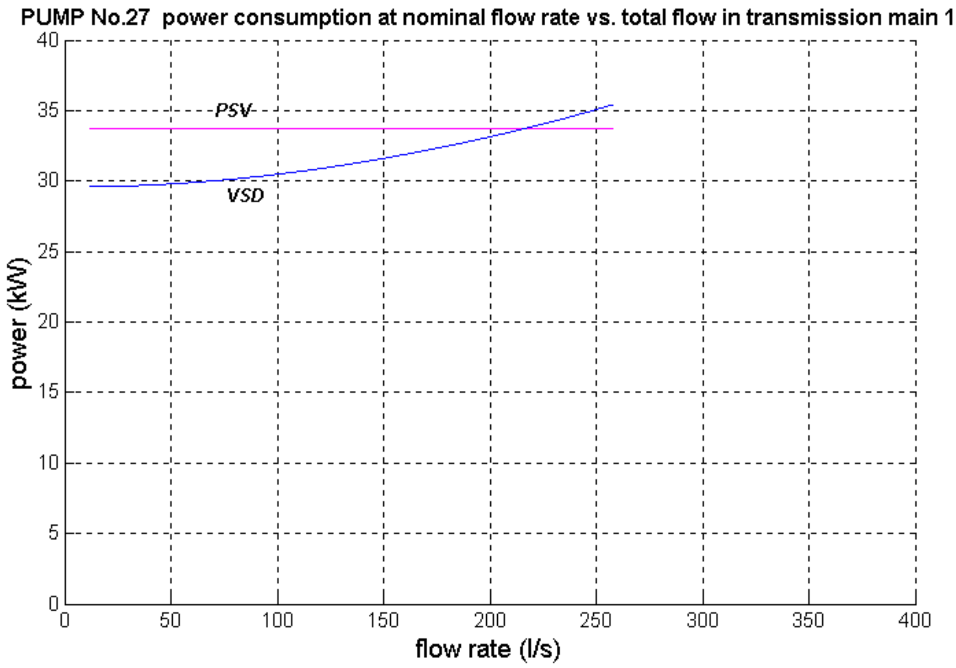

PUMP No.22 power consumption at nominal flow rate vs. total flow in transmission main 1
PSV
VSD
power (kW)
flow rate (l/s)
0
5
10
15
20
25
30
35
40
0
50
100
150
200
250
300
350
400

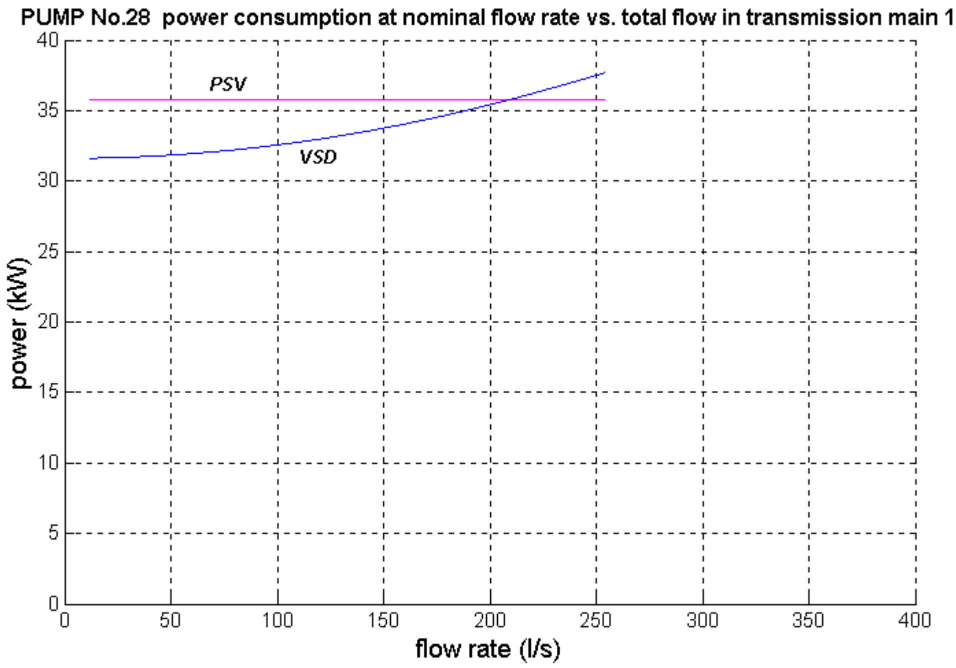

PUMP No.23 power consumption at nominal flow rate vs. total flow in transmission main 1
PSV
VSD
power (kW)
flow rate (l/s)
0
5
10
15
20
25
30
35
40
0
50
100
150
200
250
300
350
400

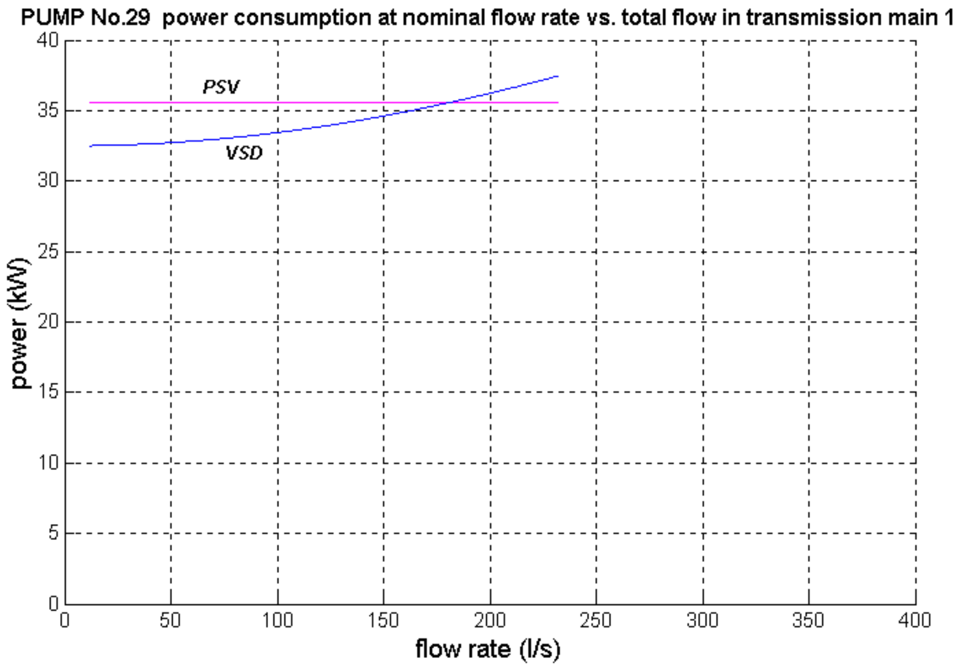

PUMP No.24 power consumption at nominal flow rate vs. total flow in transmission main 1
PSV
VSD
power (kW)
flow rate (l/s)
0
5
10
15
20
25
30
35
40
0
50
100
150
200
250
300
350
400

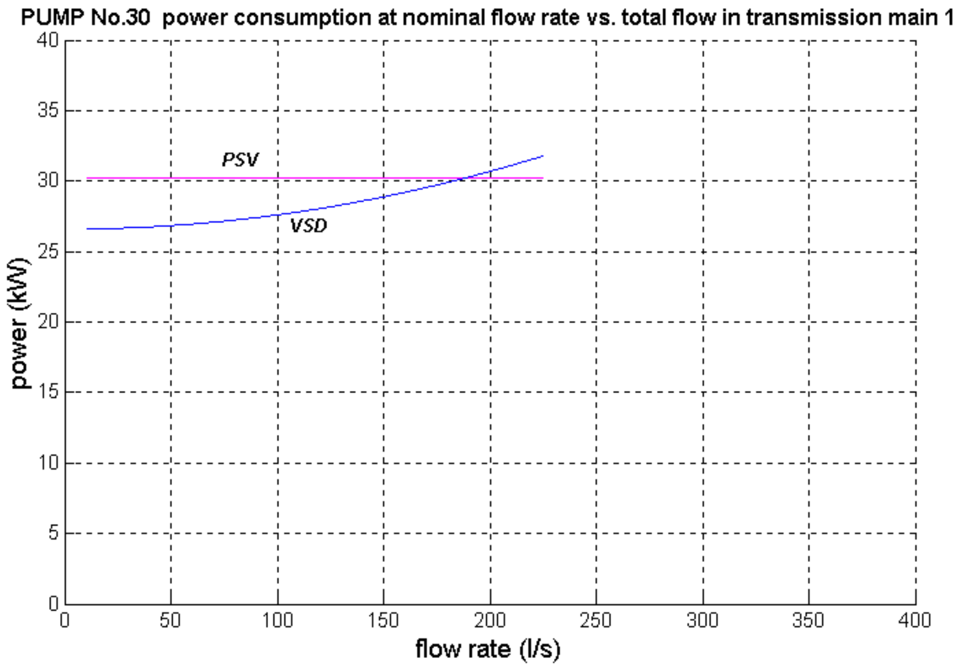

PUMP No.25 power consumption at nominal flow rate vs. total flow in transmission main 1
PSV
VSD
power (kW)
flow rate (l/s)
0
5
10
15
20
25
30
35
40
0
50
100
150
200
250
300
350
400

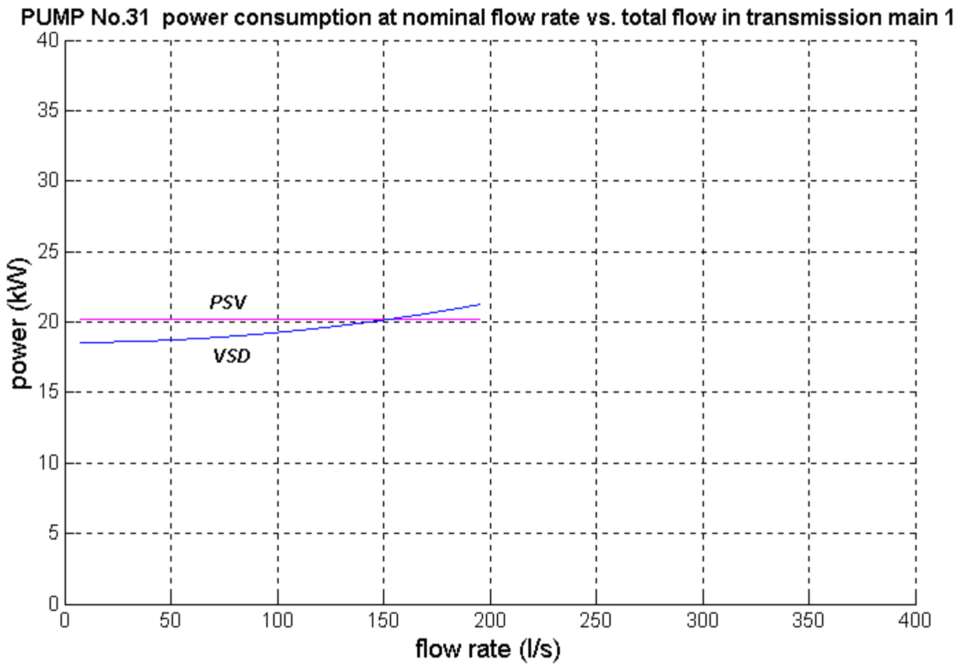

PUMP No.26 power consumption at nominal flow rate vs. total flow in transmission main 1
PSV
VSD
power (kW)
flow rate (l/s)
0
5
10
15
20
25
30
35
40
0
50
100
150
200
250
300
350
400

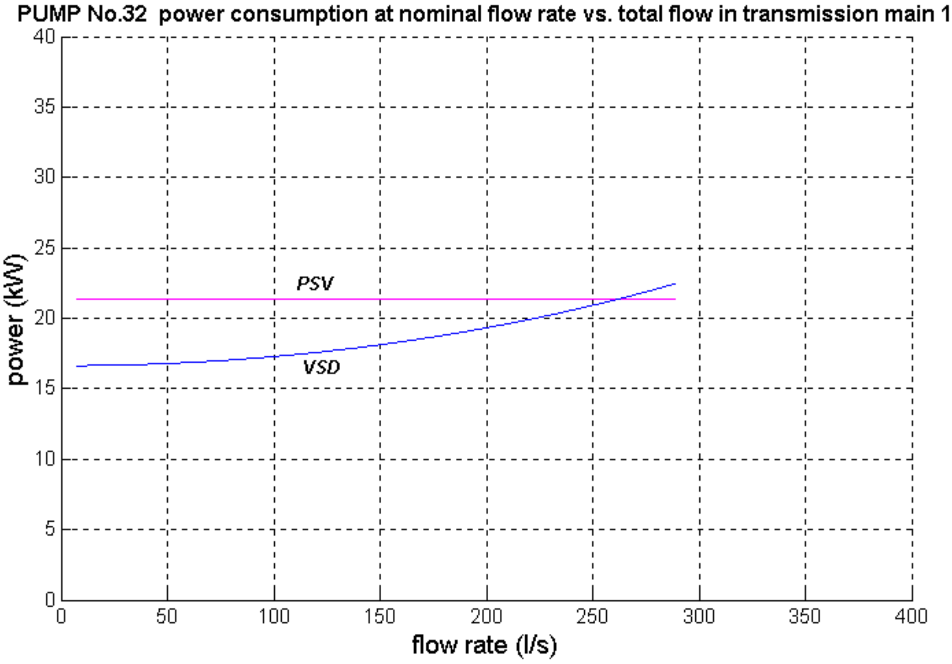

PUMP No.27 power consumption at nominal flow rate vs. total flow in transmission main 1
PSV
VSD
power (kW)
flow rate (l/s)
0
5
10
15
20
25
30
35
40
0
50
100
150
200
250
300
350
400

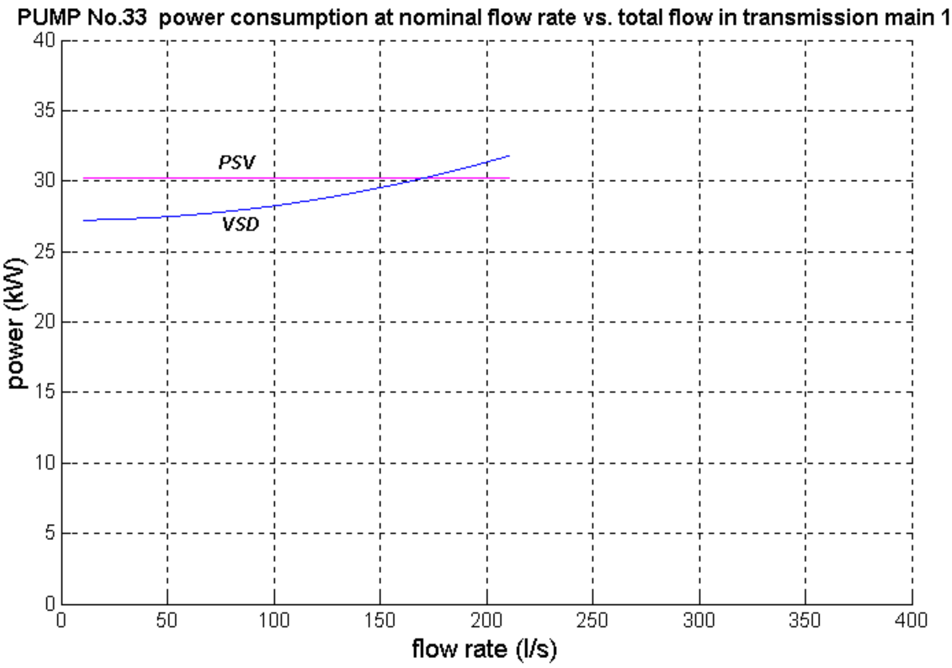

PUMP No.28 power consumption at nominal flow rate vs. total flow in transmission main 1
PSV
VSD
power (kW)
flow rate (l/s)
0
5
10
15
20
25
30
35
40
0
50
100
150
200
250
300
350
400

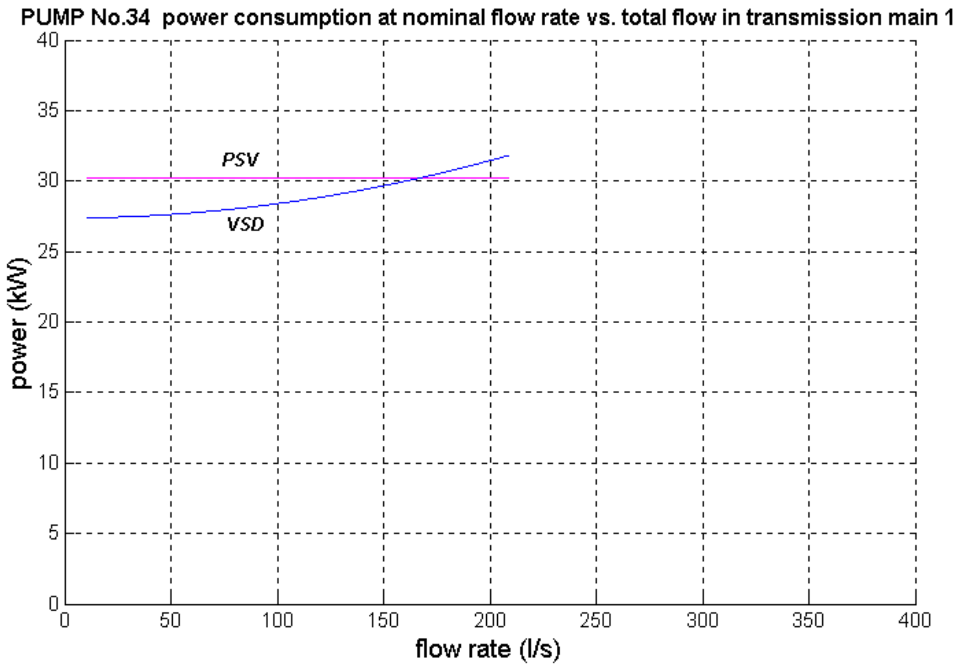

PUMP No.29 power consumption at nominal flow rate vs. total flow in transmission main 1
PSV
VSD
power (kW)
flow rate (l/s)
0
5
10
15
20
25
30
35
40
0
50
100
150
200
250
300
350
400

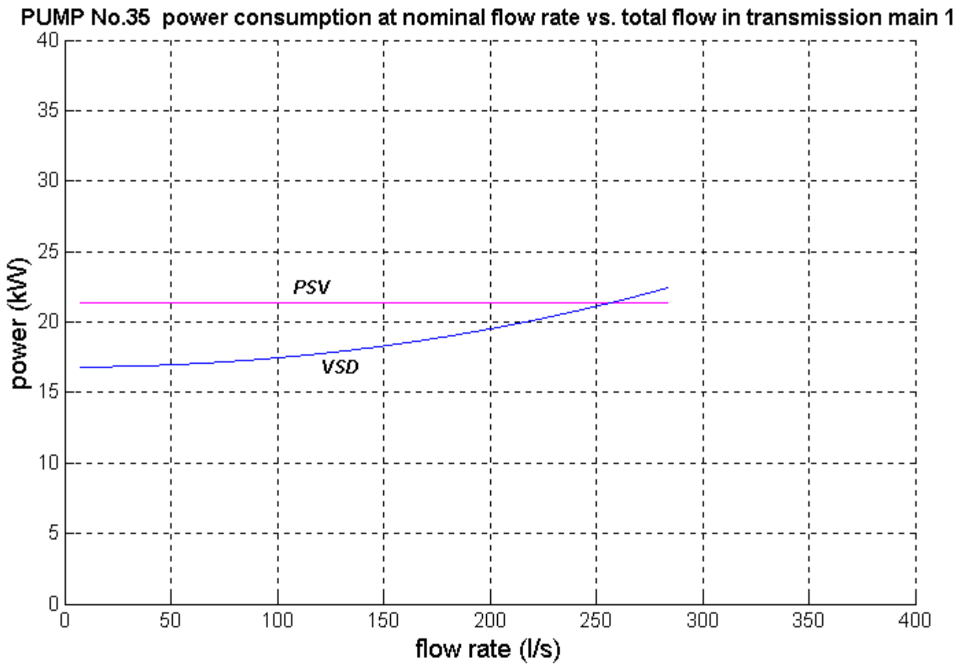

PUMP No.30 power consumption at nominal flow rate vs. total flow in transmission main 1
PSV
VSD
power (kW)
flow rate (l/s)
0
5
10
15
20
25
30
35
40
0
50
100
150
200
250
300
350
400


PUMP No.31 power consumption at nominal flow rate vs. total flow in transmission main 1
PSV
VSD
power (kW)
flow rate (l/s)
0
5
10
15
20
25
30
35
40
0
50
100
150
200
250
300
350
400


PUMP No.32 power consumption at nominal flow rate vs. total flow in transmission main 1
power (kW)
flow rate (l/s)
PSV
VSD
0
5
10
15
20
25
30
35
40
0
50
100
150
200
250
300
350
400


PUMP No.33 power consumption at nominal flow rate vs. total flow in transmission main 1
power (kW)
flow rate (l/s)
PSV
VSD
0
5
10
15
20
25
30
35
40
0
50
100
150
200
250
300
350
400


PUMP No.34 power consumption at nominal flow rate vs. total flow in transmission main 1
PSV
VSD
power (kW)
flow rate (l/s)
0
5
10
15
20
25
30
35
40
0
50
100
150
200
250
300
350
400


PUMP No.35 power consumption at nominal flow rate vs. total flow in transmission main 1
PSV
VSD
power (kW)
flow rate (l/s)
0
5
10
15
20
25
30
35
40
0
50
100
150
200
250
300
350
400