

REDUCING PUMPING POWER COSTS BY VARIABLE SPEED PUMPING

The missing link between the system curve and variable speed pump curve reveals the benefits, and potential pitfalls, of variable speed drives

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ABSTRACT

Historically, pumps have been operated by synchronous motors operating at a fixed speed to meet a design flow rate. For municipal water providers, the design flow rate might include factors such as the anticipated increase in connections, seasonal variations, and an allowance for fire fighting or other such contingencies to ensure sufficient capacity is available to meet the peak demand for the anticipated life of the pump station. For the majority of the pumps' life this design flow rate would not be required, and so traditionally the pumps run as required and then shut down.

The availability of variable speed drives (VSD) now permits pump speed to be easily varied. Generally it is not necessary to pump at the maximum rate and, by pumping at a slower rate at times of lower demand the pump power required is reduced. However, despite the reduction in pump power, the overall energy consumed is not necessarily reduced.

A synergy of equations has been developed to produce a term that quantifies the Energy Density (ED) of a system at any given flow rate. The ED is presented in a format intended to have the most relevance, that is, kilowatt hours per Megalitre (kWh/ML) pumped.

By displaying the results of the equations across a range of flow rates, it can be shown that pumping at a flow rate slower than the design flow rate of the pump through the use of a VSD often delivers significant energy savings.

Realising these benefits depends on the ability of the pump to operate effectively at the duty identified in the analysis. With the ready availability of systems that allow pumps to be run at speeds as low as 50% of the nominated pump curve speed, often there are only operational adjustments required.

Importantly, there is typically a limit to the benefit realised from reducing the flow rate, and any reduction beyond that point will result in an increase in the ED. A simple analysis is presented that will allow the ED across a range of flow rates to be displayed graphically, and permits the flow rate at which the power costs are minimised to be determined.

This paper aims to demonstrate that operating cost savings may be made by optimising the speed of pumps in a given pumping system.

INTRODUCTION

The cost of the power consumed is typically the largest proportion of the lifetime cost of any pumping package. The efficiency of each component in a pump package affects the overall efficiency, and improvements in efficiency reduce the energy consumed.

When a pump is chosen there are a number of considerations, one of the most important being that it is suitable to pump at a specific flow rate and pressure. These two characteristics are used together to form the design duty of the pump.



The pressure requirement is derived from static and dynamic components of the system. Often the static portion is essentially a given, but the dynamic portion is dependent on flow rate.

As the design duty for a municipal pump station may include allowances for anticipated future connections, seasonal fluctuations and fire fighting or similar contingencies, the duty is not actually required for most of the life of the pump. Traditionally the pump would be run at the design duty when needed and then shut down.

By reducing the speed of the pump during periods when the nominated duty is not required, thereby running the pump for longer than would otherwise be required, can actually save a significant amount of energy while delivering the same volume in an acceptable timeframe.

The concept of calculating and comparing the amount of energy required to transport a given volume is far from new. However, it seems that the potential savings available from pump packages already in place through

more intuitive control is at times being overlooked. The concept of the energy required to transport a given volume has been referred to in this document as the Energy density (ED) and the units have been chosen to be relevant to municipal water providers, that is, kilowatt hours per Mega litre (kWh/ML) pumped.

Analysing the ED across a range of flow rates for numerous pump and system combinations has shown that the decrease in pump efficiency at lower flow rates is often offset by the decrease in total pressure required, and the net result is a decrease in the ED.

Typically there is a limitation to the benefit from decreasing the flow rate, and any further reduction results in an increase in the ED.

This article utilises a synergy of well-known equations to allow the ED for a range of flow rates to be determined. This is then graphed to highlight both the trend and limitation of any potential benefit. The outputs these equations produce allow better insight into methods available to improve energy efficiency.

It is envisaged that, in time, the ED will become as much a part of assessing the suitability of a proposed pump as any of the information currently available. The outputs also allow for very quick comparison of the effect that changes planned for a system will have on the energy consumed; this includes either pump or network changes.

It should be noted that, while the figures identified will be seemingly exact, the outputs should be viewed as an approximation of the potential savings available. The methods used to determine the results are an aid for estimating behaviour only.

The utilisation of a process control program such as a Supervisory Control and Data Acquisition (SCADA) package will allow the actual ED to be observed. By trending an algorithm of the instantaneous power consumed in kilowatts (kW) divided by the flow rate in Mega litres per hour (ML/h), the ED in kWh/ML can

be displayed in real time. Then, changes to the rate of pumping on energy consumption can be trended. This will allow the energy consumption of the entire pump package to be reviewed, whereas this document focuses only on the theoretical energy the pump consumes.

METHODOLOGY

To determine the ED across a range of flow rates rather than for a given duty there are a number of steps required. At first they appear laborious, but once the equations have been developed and correctly entered into a spreadsheet package such as Microsoft Excel, the outputs can be manipulated reasonably quickly.

Care must be taken to ensure the results are relevant, the equations in the raw form do not account for an array of other factors (e.g. time of use power tariffs and the manufacturer's recommended range of application for the pump) that need to be considered when assessing the suitability of a pump to perform a specific duty.



For this reason it is necessary to observe and review all of the results of the various equations to ensure the proposed benefits can be realised.

The equations need the inputs to be in specific formats. For centrifugal pumps the pump curve can be expressed as:

$$H_1 = aQ_1^2 + bQ_1 + c$$

where H_1 represents the pump head developed at full speed at a flow rate of Q_1 .

This equation does not need to represent the entire pump curve, but the closer it represents the region of the pump curve being analysed the better the results will be. The curve can be expressed as a higher degree polynomial if desired / required, but the equations derived will vary from those developed in this paper.

Then, the power curve is also required. For ease of calculation the form used for the following equation assumes a linear relationship between pump power (P_1) and flow (Q_1). However, as has been discussed previously, this can be expressed as a higher order polynomial to improve the accuracy of the output, but the equations derived will again be slightly different:

$$P_1 = dQ_1 + e$$

where P_1 is the power required to develop a flow Q_1 from by the pump at full design speed.

Next, a slight rearrangement of the Pump Affinity Laws for centrifugal pumps,

$$\frac{Q_1}{Q_2} = \frac{n_1}{n_2}$$

$$Q_1 = \frac{Q_2 n_1}{n_2}$$

where Q is the flow developed by a pump at rotational speed n .

Let n_1 (the design full speed) = 100% = 1

$$Q_1 = \frac{Q_2}{n_2}$$

Likewise:

$$H_1 = \frac{H_2}{n_2^2}$$

$$P_2 = P_1 n_2^3$$

The next step is to re-write the pump curve with the new definitions:

$$\frac{H_2}{n_2^2} = a \left(\frac{Q_2}{n_2} \right)^2 + b \frac{Q_2}{n_2} + c$$

$$H_2 = aQ_2^2 + bQ_2 n_2 + cn_2^2$$

$$0 = \langle c \rangle n_2^2 + \langle bQ_2 \rangle n_2 + \langle aQ_2^2 - H_2 \rangle$$

This equation can now be solved to give n_2 , the speed at which the pump needs to be driven, as a proportion of the maximum speed n_1 , to meet any given duty point (i.e. a nominated flow rate Q_2 at the required pressure H_2) for the system curve being analysed. The standard form for finding the roots of a quadratic equation where $y = 0$ is:

$$x = \frac{-b \pm \sqrt{b^2 - 4ac}}{2a}$$

Which; in this example, becomes:

$$n_2 = \frac{-(bQ_2) \pm \sqrt{(bQ_2)^2 - 4c(aQ_2^2 - H_2)}}{2c}$$

(Equation 1)





Experience has shown that the '+' version of the calculation returns the relevant answer, but it needs to be checked.

Once the speed is determined, this can be used to determine the pump power consumed at any flow rate:

$$P_2 = \left(d \frac{Q_2}{n_2} + e \right) n_2^3$$

(Equation 2)

Now that both flow rate and pump power is available, the ED can be calculated

$$\text{EnergyDensity} = \frac{P_2}{Q_2}$$

(Equation 3)

A similar but simpler set of equations can be derived for positive displacement pumps as, in this class of pump, the flow rate is proportional only to motor speed and does not vary significantly with the resulting system head.

Performing this equation across the range of flows would be tedious, but a spreadsheet package such as Microsoft Excel removes a degree of the tedium and gives quite illustrative results.

Additionally, there is a range of information that can be derived for the three equations above. An example of a typical output is shown in Table 1,

An additional column has been included in Table 1, Pump Efficiency (Eff_2). This is the theoretical minimum power divided by the actual power. ($\text{Eff}_2 = Q_2 H_2 g / P_2$), where g is the gravitational constant. This has been included to assist identifying when the system curve has exceeded the minimum flow for the pump. For example, if, for a given pump, the minimum recommended flow correlated with 70% efficiency at 100% speed, the recommended minimum flow would typically occur at 70% efficiency for all speeds, but this would need to be confirmed for each pump being analysed. Once confirmed, it can be quickly determined that, for the example analysed in Table.1, the minimum speed at which the pump should be run before the flow would be less than that recommended by the manufacturer would be approximately 200 L/s at 66.4% of the full speed.

TABLE 1: An example of typical results

System Curve		Equation 1	Equation 2		Equation 3
Q ₂ (L/s)	H ₂ (m)	η ₂ (%)	P ₂ (kW)	Eff ₂ (%)	PD (kWh/ML)
0.0	60.0	58.3	113.0	0.0	N/A
50.0	60.6	58.8	120.8	24.6	671.4
100.0	62.5	60.4	135.8	45.2	377.2
150.0	65.6	63.0	158.8	60.8	294.1
200.0	70.0	66.4	191.3	71.8	265.7
250.0	75.6	70.6	234.8	79.0	260.8
300.0	82.5	75.4	291.0	83.4	269.4
350.0	90.6	80.7	361.8	86.0	287.1
400.0	100.0	86.4	449.3	87.3	312.0
450.0	110.6	92.4	555.4	87.9	342.8
500.0	122.5	98.7	682.4	88.1	379.1
510.0	125.0	100.0	710.4	88.0	387.0

It can also be observed that there would be no benefit for this pump to run below 250 L/s, as the energy density becomes greater below that flow rate.

The results also demonstrate that, for this example, the energy required per unit volume increases from 250 L/s up to 510 L/s, the nominated BEP for this pump, but, for the system analysed, it is the least energy efficient rate of pumping above 250 L/s.

TYPICAL RESULTS

To illustrate the potential benefits of understanding the ED of a system, an analysis of three typical systems has been compiled. For this analysis a fictional pump curve has been developed that has a BEP of 88% at the duty 510 L/s@125m.

This pump would typically have been selected by the designer as the system curve passes through the BEP at the design duty for the foreseen lifetime of the new pump package.

It is not a requirement for this analysis for the system curve to pass through the BEP of the pump, but it has been done to illustrate that greater energy efficiency can be obtained if the pump can be slowed to meet the non-peak demand system requirements. Three fictional scenarios will be quickly analysed to illustrate the

limitation and variation between the benefits available. The three fictional scenarios are:

- a) System 1:** No Static Head, High Dynamic Head (Static Head 0m, design duty 510L/s@125m total pump Head)
- b) System 2:** Moderate Static Head, Moderate Dynamic Head (Static Head 60m, design duty 510L/s@125m total pump Head)
- c) System 3:** High Static Head, Low Dynamic Head (Static Head 100m, design duty 510L/s@125m total pump Head)

System 1 – No Static Head, High Dynamic Head

It can be clearly seen in Figure B that for system 1, as the flow rate is decreased, less energy per ML is consumed.

Therefore, if today's demand is forecast to be 20 ML, despite the BEP nominating 510 L/s (43.2 ML/d) as the best pump efficiency, if instead the pump is run at 310 L/s then the power to pump the 20 ML is 63% less than if the pump had been run at its nominated duty.

Additionally, at the flow rate of 310 L/s, the 20 ML will be pumped in 18 hours, allowing significant additional pumping time should the forecast have been incorrect.

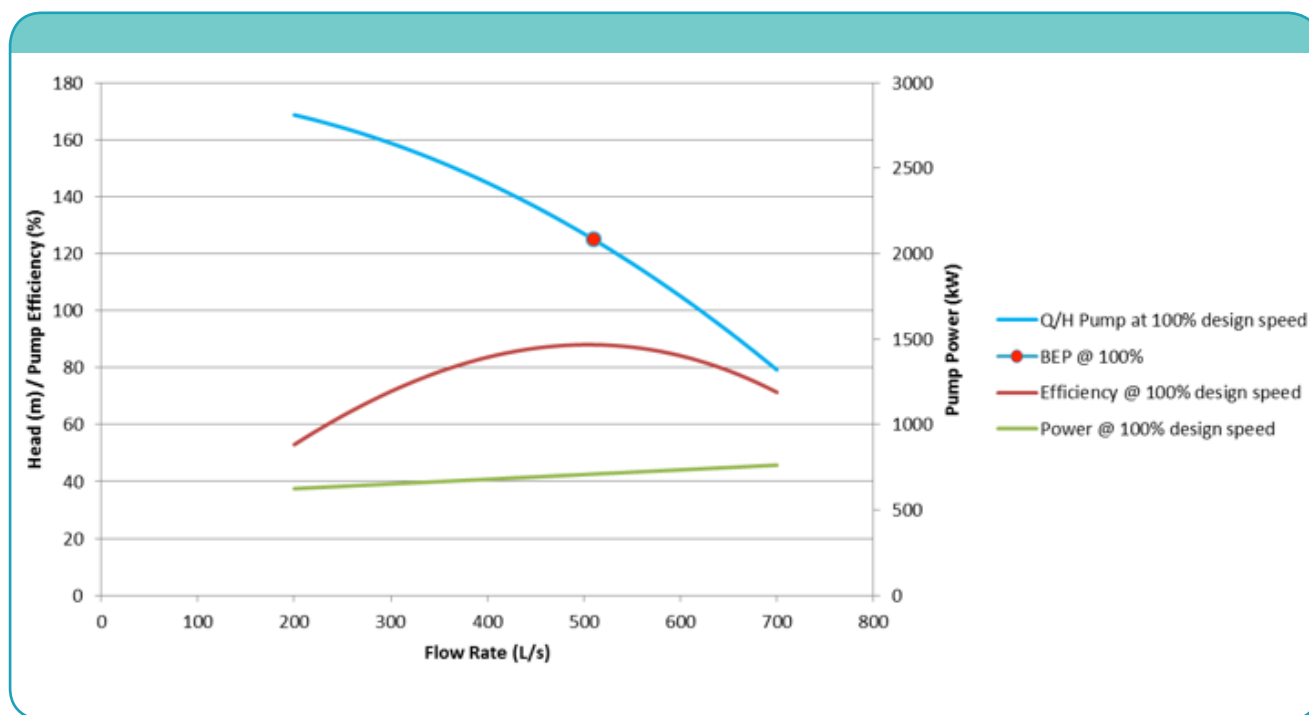


Figure A: Operating characteristics of pump used for analysis

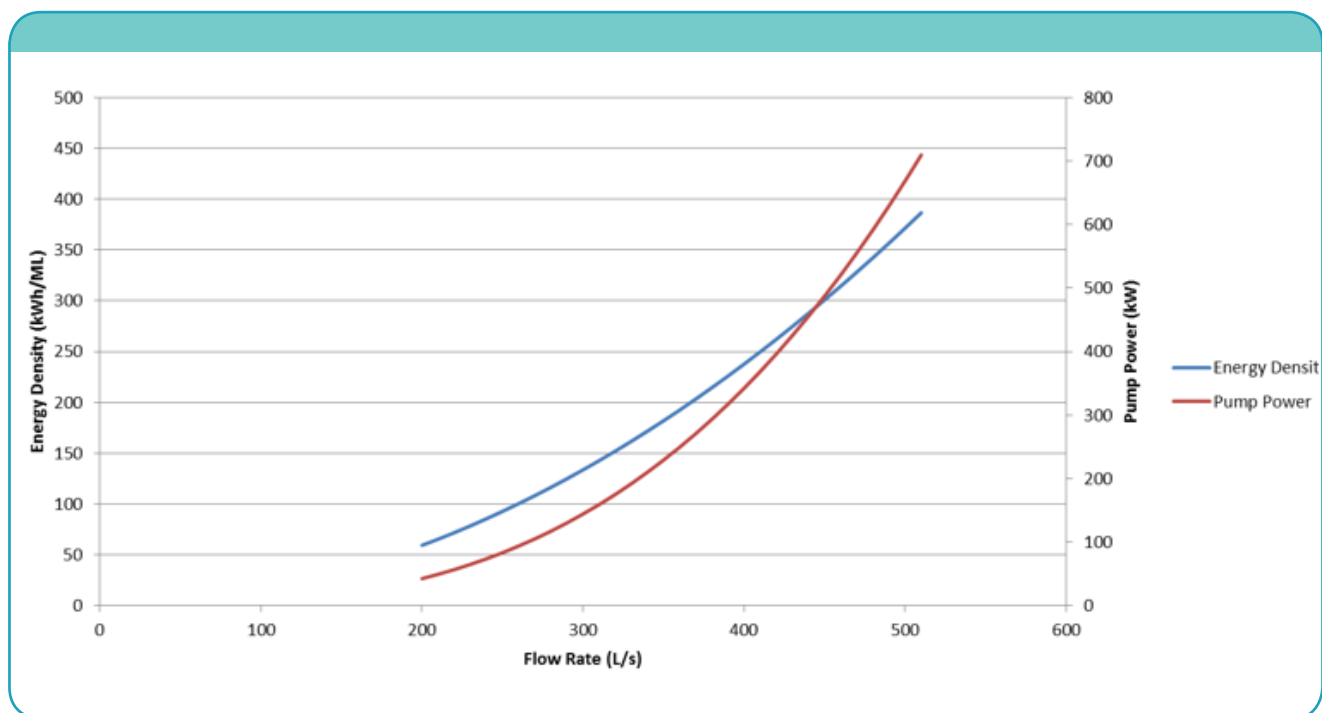


Figure B: System 1 - No Static Head, High Dynamic Head

Additionally, to achieve this saving the pump needs to run at 61% of its rated speed, or 30.5 Hz for a 50 Hz package. This degree of turn down is widely available in standard pump packages.

Furthermore, if continuous pumping is acceptable, and the pump slowed further to 231.5 L/s, while it would take the entire 24-hour period to pump the required 20 ML, the pump energy consumed to do so would be 79.7 kWh/ML. Compared to the nominated design duty, where pump energy consumed is 386.8 kWh/ML, this is a saving of 79.4%. To achieve this however the pump needs to run at 45% of its rated speed, which may be beyond the capability of standard pump packages.

While this scenario offers the best savings, appreciable

savings are also available for systems with moderate static head.

System 2 – Moderate Static Head, Moderate Dynamic Head

For system 2 the results are shown in Figure C, the ED has a minimum value at 220 L/s. At this flow rate the energy consumed per ML pumped is 32.4% less than if the pump had been run at its nominated design duty. This is despite this point correlating 75.1% pump efficiency, a seemingly poor figure, but well within the application range.

For the aforementioned scenario of pumping at 310 L/s, the power saving is in the order of 29.7% to deliver the 20 ML in 18 hours.

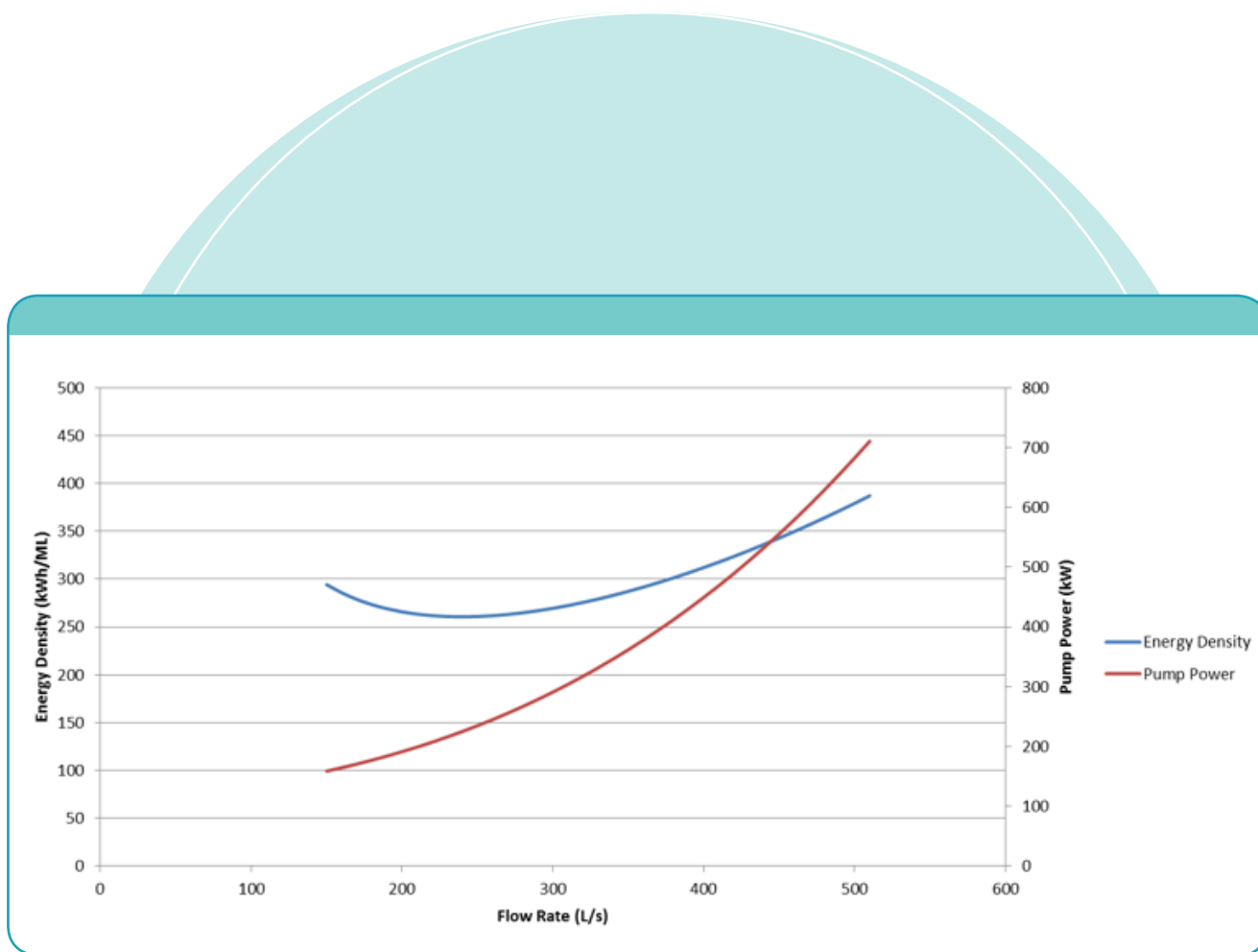


Figure C: System 2 – Moderate Static Head, Moderate Dynamic Head

System 3 – High Static Head, Low Dynamic Head

For system 3 the results are shown in Figure D., the ED has a minimum value at 380 L/s, at which point the ED is 6.1% less than the design duty.

This is to be expected for this scenario as the dynamic head losses are very small compared to the static lift. Therefore, despite less instantaneous pump power being consumed, there would be no benefit to pumping at 310 L/s.

CONCLUSION

The systems analysed offer a useful insight into the limitations of the benefits available from VSD's. It also demonstrates that despite the pump power being less

at lower flow rates, the energy required to pump a given volume does not always follow the same trend.

Therefore, taking the time to review the ED across a range of flow rates may result in considerably lower overall power consumption, and can at times be achieved with minor operational changes.

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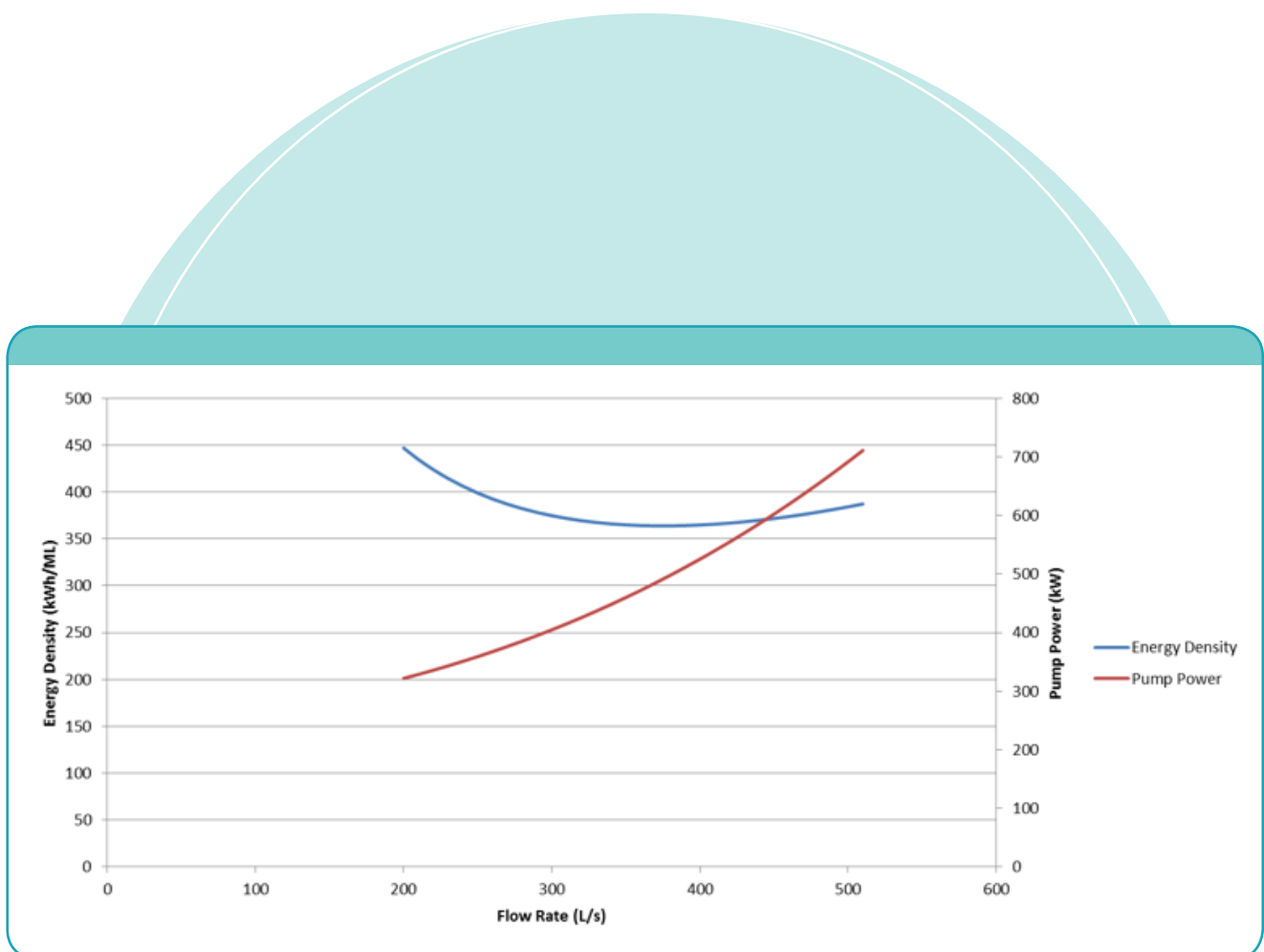


Figure D: System 3 – High Static Head, Low Dynamic Head