Caen Hill Pumping Station

PHASE 2 SITE TEST SUMMARY REPORT

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Content

1 Introduction

This report summarises the key findings of the Phase 2 pump system test for Caen Hill Pumping Station (PS) which is owned and managed by The Canal and River Trust (the Trust).

The Phase 2 tests were undertaken on 18th November 2021 with various intervention works being undertaken since the Phase 1 testing in 2019, including:

- Reinforcement of the pump foundations
- Investigation and removal of the intermediate non-return valves
- Pump 1 bottom end overhaul including replacement of seals, bearings and sensors.
- Pump 2 general maintenance.
- Inlet and wet well general desilt maintenance

The report aims to cover the following areas:

- Derivation and analysis of the existing system and pump performance following the interventions.
- Report on current pump vibration levels
- Comparison with 2019 test results.

The site test comprised pump performance testing to establish pump duties and system curve (including rising main static head) alongside power monitoring. This incorporated recording of time-stamped data and real time measurement of pumped flow, pressure and wet well level using calibrated instrumentation and sensors in conjunction with measurement of key electrical parameters, i.e., current, voltage, power, and power factor (P.F.), using power meter voltage probes and current transducers.

2 System Description

2.1 Pumping Station

Caen Hill PS is situated near Devizes, Wiltshire, UK. Its purpose is to supply water up from Lock 22 to Lock 50 on the Kennet and Avon Canal. Constructed in 1996, it is of a dry well construction, housing two Xylem dry well submersible pumps normally operating in a duty/assist configuration.

Figure no. 1: Photos of Caen Hill PS

Table no. 1: Pump Details

Parameter	Description
Pumps	Xylem (Flygt) CT3240
No. of Pumps	2
Duty Configuration	Duty / assist
Rated Motor Output	215 kW
Impeller Diameter	535 mm
Drives	Variable speed (Mitsubishi)
VSD operation	30 s RAMP & 48.0 Hz operating frequency
Pipework	300 mm diameter
Non-Return Valves	Ball
Wet Well Level Sensor	Ultrasonic
Wet Well Level	55.5 mAOD
Pump Centre Line	54.3 mAOD

2.2 Rising Main

The rising main is approximately 3600 m in length and manufactured from Ductile Iron. There are no reports of any bursts since construction. Prior to 2021, the rising main comprised two intermediate discharge points, with in-line non-return valves (NRVs) complete with a return bypass. The NRVs were situated immediately downstream of each discharge point. The two intermediate NRV chambers were inspected and NRV doors removed by the Trust on between February and March 2021.

Table no. 2: Rising Main Details

Parameter	Description
Length	3602 m
Elevation rise	72 m
Pipe diameter	600 mm
Discharge level	127.6 mAOD
Pipe material	Ductile Iron
Removed intermediate NRV #1	1270 m from Caen Hill PS
Removed intermediate NRV #2	2410 m from Caen Hill PS

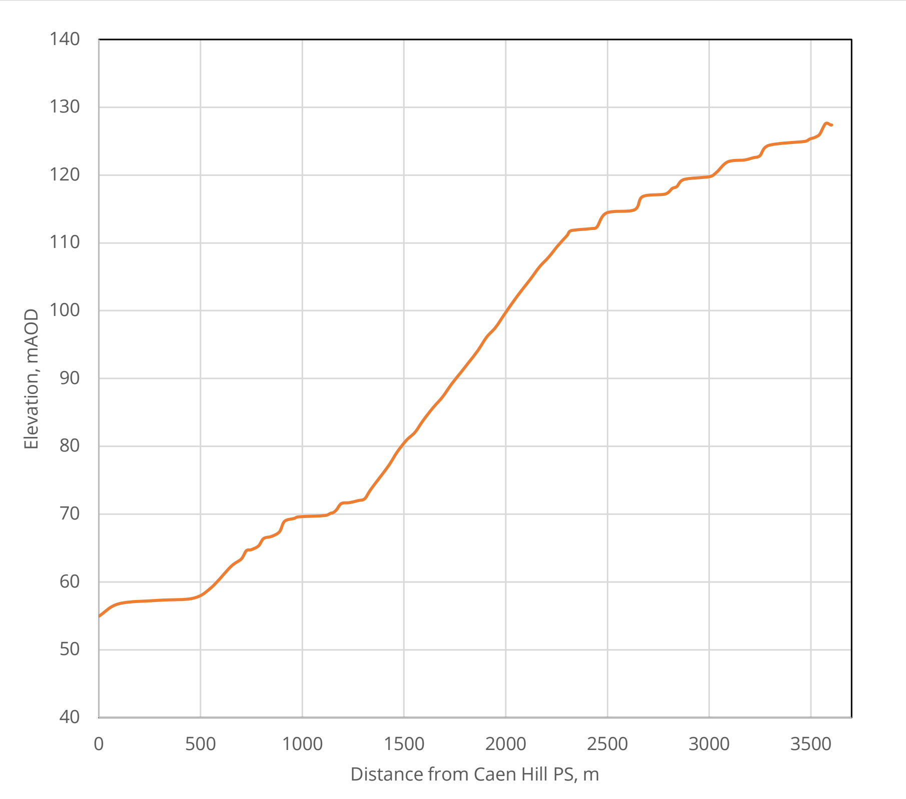


Figure no. 2: Caen Hill PS Elevation Profile

2.3 Particular Issues

The Trust has reported that the pumping station has the following issues.

- Minor impeller damage from cavitation
- Short pump bearing life
- Pipework flange leaks.

These issues are covered further within this report.

3 System Description

Site testing of pump performance was undertaken with a small/medium enterprise (SME), Samatrix, on the 18th November 2019. This was a follow up test to the original Phase 1 pump testing undertaken 11th September 2019.

The following intervention works have been undertaken since the Phase 1 testing.

- Reinforcement of the pump foundation concrete plinths to reduce pump vibration
- Investigation of the rising main and removal of the intermediate non-return valves as possible sources of high rising main head loss, and possible increased pumping energy – February / March 2021
- Pump 1 bottom end overhaul including replacement of seals, bearings, and sensors. Works were undertaken between August 2019 and December 2019.
- Pump 2 general maintenance.
- Inlet and wet well general desilting maintenance in March 2021

The following parameters were measured and logged as part of the test:

- Input power to each drive (via a portable "Fluke" power meter).
- Pumping station flow rate (via the existing installed flowmeter).
- Suction and delivery Pressures (via pressure transducers).
- Spot vibrations in RMS velocity (via magnetic vibration accelerometer).

Based upon the test results, the latest system curves have been derived for the following three operating scenarios:

- Pumps P1 and P2 operating in parallel.
- Pump P1 operating only.
- Pump P2 operating only.

The suction and delivery elevations, pipe roughness values have been based on recorded site measurements in addition to the desktop SCADA data provided.

The test points for Phase 1 (yellow) and Phase 2 (green) and derived hydraulic performance curves are displayed together in Figure no. 3 to Figure no. 5

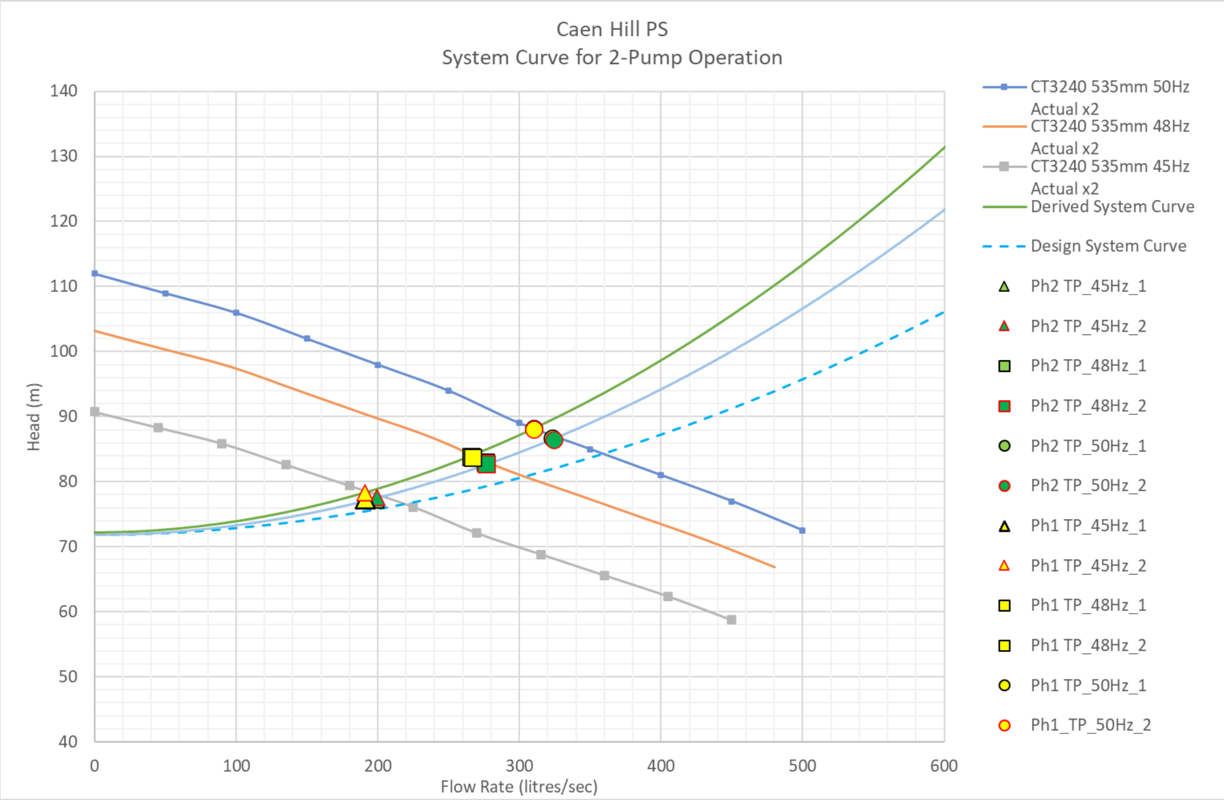


Figure no. 3: Derived System Curves for 2-Pump Operation

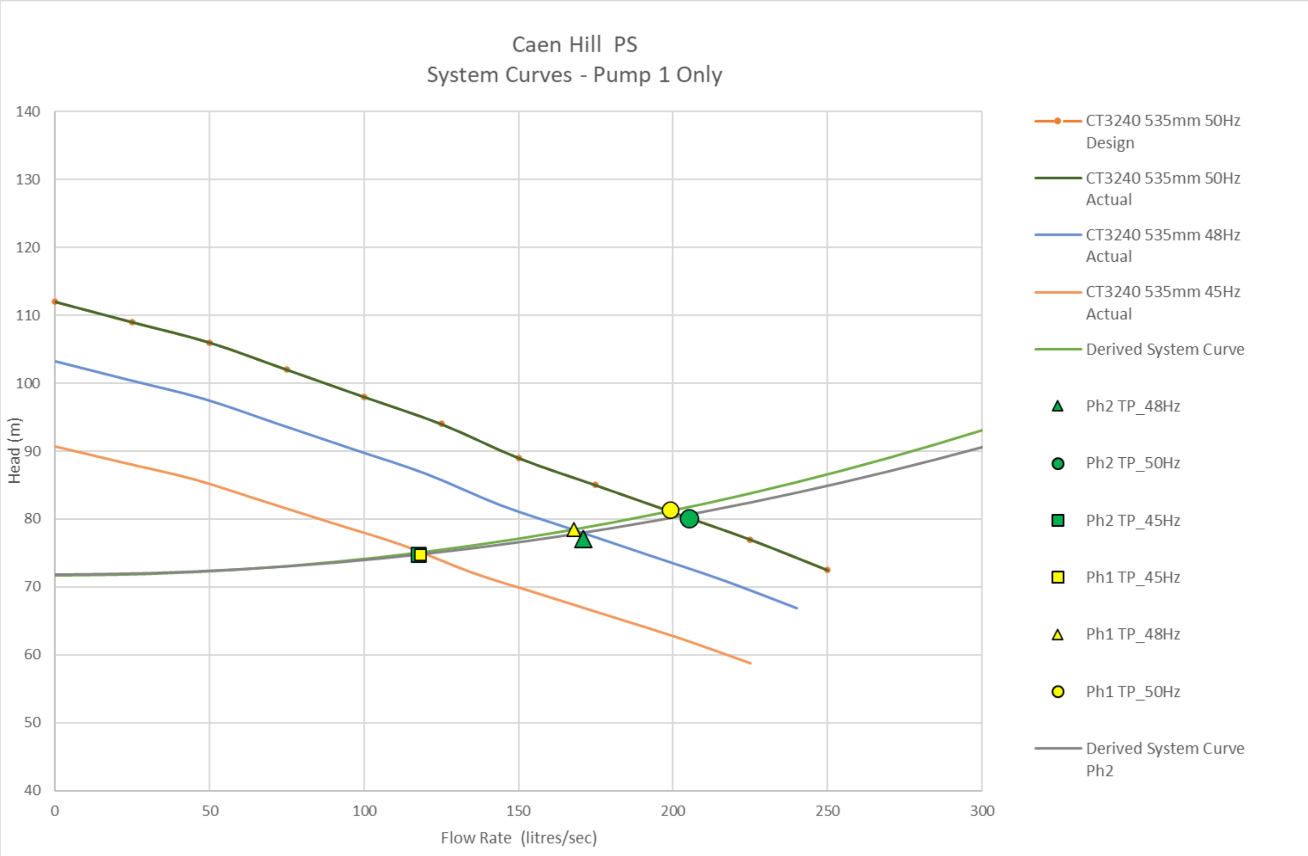


Figure no. 4: Derived System for Pump 1 Only Operation

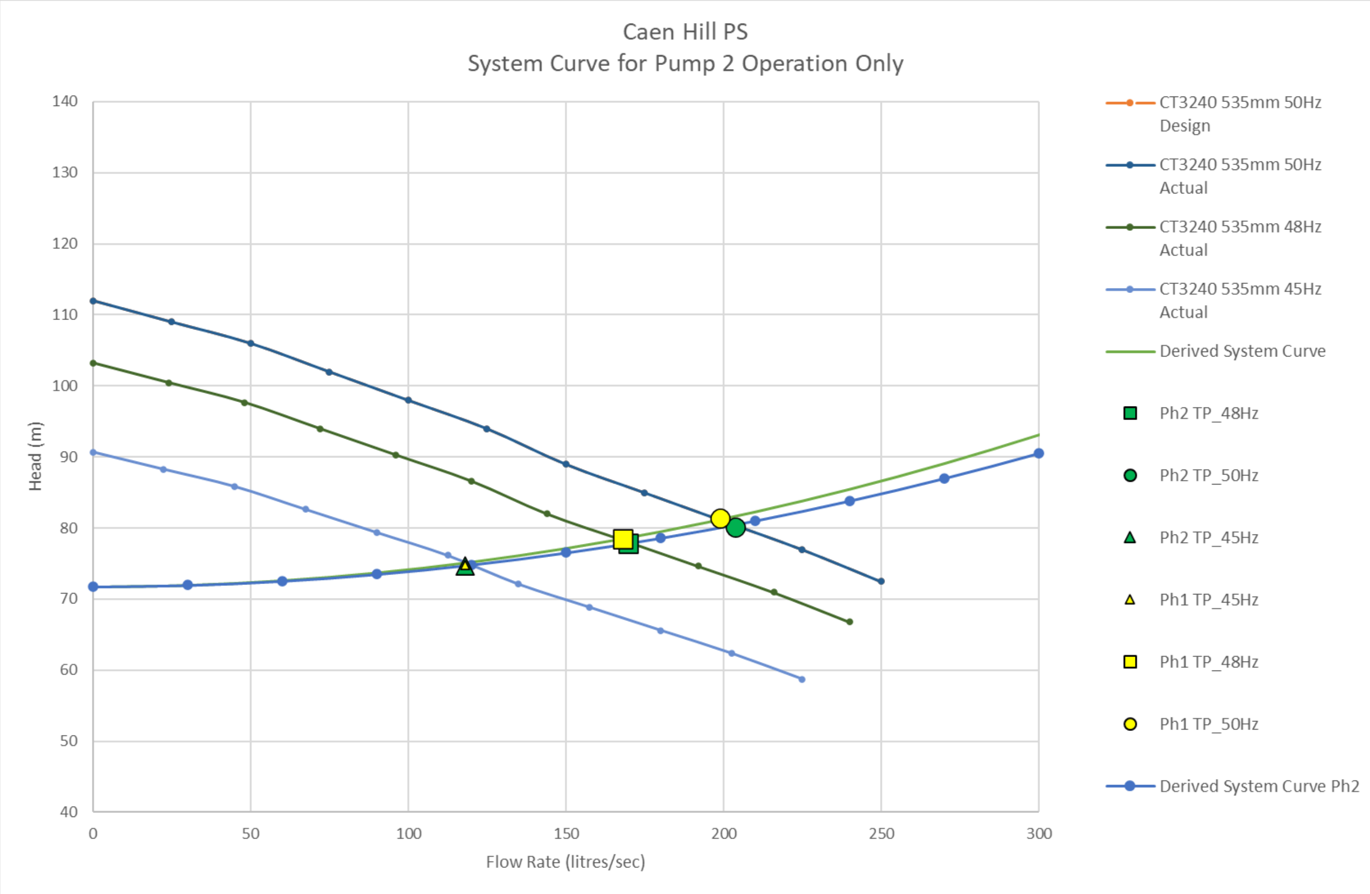


Figure no. 5: Derived System Curves for Pump 2 Only Operation

The key observations from the derived system curves are as follows:

- The site test results backed up by the SCADA data indicate a marginal improvement on rising main head losses. A reduction of approximately 2m to 3m at 300 l/s between Phase 1 and Phase 2 and removal of the intermediate NRVs was found. However, the rising main dynamic losses are still 3m to 4m higher than expected at 300l/s as indicated by the dashed “design system curve” in Figure no. 3.

- The pump curves were not adjusted from the manufacturers published performance curves in order to align with site results.

- The best efficiency point (BEP) of the installed CT3240 pump is just to the right-hand side of the pump curve extents as shown in the system curves i.e., 262 l/s at 70 m for 1-pump operation and 524 l/s at 70 m for 2-pump operation.

- Under 2-pump operation at 48Hz the operating flow rate is 55% of the flow rate at BEP. This falls outside and to the left of the typical preferred operation region (POR) of the pumps (see Figure no. 9).

4 Net Positive Suction Head (NPSH) and Submergence

Evidence of cavitation was found during the 2019 overhaul of Pump 1 as shown in Figure no. 6.

Figure no. 6: Pump 1 Impeller Cavitation

From NSPH calculations undertaken during Phase 1 and the site test results. An estimated NPSH margin of between 2 m and 10 m due to the fluctuation in measured pump inlet pressure.

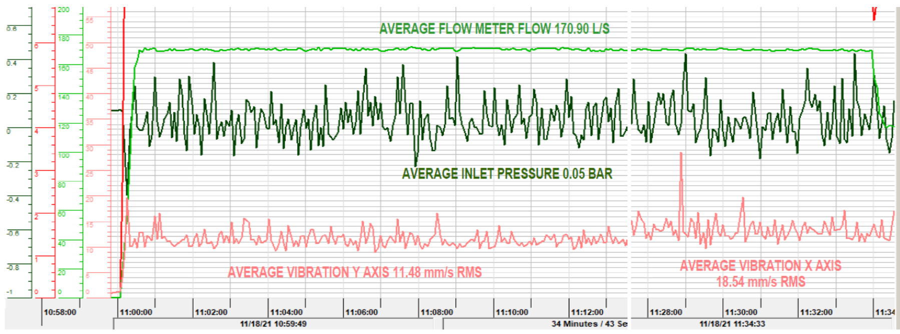


Figure no. 7: Measured Inlet Pressure (in black) for Pump 1 showing fluctuation

The measured pump inlet pressures were seen to fluctuate significantly, more noticeably in Pump 1, which could indicate excessive flow turbulence / instability.

Key aspects of the inlet pipe arrangement against model design guidance[1,2], are shown in Table 3.

Table 3 – Existing inlet pipe comparison to model design guidance

Parameter	Caen Hill PS Existing	Recommended Range	
	48Hz Operation	Prosser / CIRIA	ANSI/HI-9.8
Inlet pipe/bellmouth diameters (mm/mm)	250/370	-	-
Inlet bellmouth velocity (m/s)	1.28 to 1.59	1.3	0.6 to 2.7 (recommended = 1.7)
Distance between inlet bellmouth and floor (m)	0.6 (est.)	0.19	0.11 to 0.19
Inlet bellmouth submergence (m)	1.4 (est.)	0.56	0.95 to 1.08

The key deviance from “recommended” practice is the height of the inlet bellmouths above the wet well floor, presumably due to silt concerns.

Figure 8 – Wet Well Intake during 2019 Desilting Activities

This could result in the inlet pipework being susceptible to adverse hydraulic conditions such as vorticity and potentially air entrainment or local low-pressure zones that could cause cavitation.

Siltation can impact on flow presentation to the inlet pipes. The well was desilted in February 2022 and, although a possibility, this is not considered the most likely root cause issue.

1 MJ Prosser, *The hydraulic design of pumps sumps and intakes*, BHRA/CIRIA 1977
2 ANSI/HI-9.8, *Pump Intake Design*, Hydraulic Institute 1998, 2008.2016, 2020

The pump operating point, being left of the preferred operating region, could lead to local flow recirculation at the pump entry. This is a known cause of cavitation.

Given the calculated suction head available, possible reasons for the presence of cavitation marks as found in the pump impeller include:

- Inlet pipework design causing hydraulic instability, and poor presentation leading to turbulent low pressure or air entrainment;
- Internal or entry recirculation of flows at the pump due to operation left of preferred operating region (POR).

Commented [BK1]: Here we sound as if we are categorically identifying the sizing as *causing* hydraulic instability, in Section 8.2 we use the word *may* and I wonder if these should be more aligned. The potential reson not is that this line is referring to sizing (ie dia) whereas 8.2 is referring to height bellmouth above floor.

Commented [JN2R1]: Agree – the word sizing was a legacy and now rmeoved

5 Energy Analysis

5.1 Specific Energy

At the pump audit visit by Samatrix, a temporary "Fluke" power meter was connected at the individual pump start compartment to record power into the pump VSD. From the measured power, flow rate, and pressure undertaken at the Samatrix audit visit, an analysis of pumping efficiency and energy has been undertaken.

Table no. 4 summarises the measured VSD input power, efficiency, and derived specific energy findings.

Table no. 4: VSD Input power, Efficiency and Specific Energy

Pump Configuration	Measured Flow Rate (l/s)	VSD Freq. (Hz)	Measured Power (kW)	String Efficiency (%)	Specific Energy (kWh/1000 m³) – PHASE 2	Specific Energy (kWh/1000 m³) – PHASE 1	Specific Energy (kWh/1000 m³) – % Difference
Pump 1 Only	205.1	50	217	74	294	312.7	-6%
	170.9	**48**	**181**	**71**	**294.2**	**310.0**	**-5.1%**
	117.7	45	134	64	316.4	323.7	-2.3%
Pump 2 Only	203.7	50	219	73	298.7	296.4	+0.8%
	169.4	**48**	**184**	**70**	**301.3**	**296.0**	**+1.8%**
	118	45	136	64	320.2	313.1	+2.3%
Both Pumps (Power Measured at Pump 1)	323.4	50	196	70	338.1**	352.2**	-4.0%
	276.2	**48**	**166**	**67**	**336.6****	**347.6****	**-3.2%**
	198.9	45	126	60	352.9**	360.7**	-2.2%
Both Pumps (Power Measured at Pump 2)	324.1	50	198	70	338.1**	352.2**	-4.0%
	276.8	**48**	**169**	**67**	**336.6****	**347.6****	**-3.2%**
	199.4	45	127	60	352.9**	360.7**	-2.2%
	**String Efficiency is overall "wire to water" efficiency including the VSD **Averaged from both Pump 1 and Pump 2 individually measured power readings and flow rates*						

The normal running frequency of each pump at Caen Hill PS is 48 Hz. Operating at 48 Hz results in a lower specific energy and therefore energy cost than running 45 Hz on a VSD, these results can be seen in Table no. 4. It can also be seen that 2-pump operation results in a higher specific energy than 1-pump operation and a lower overall operating efficiency.

Pump 1 is operating more efficiently since its overhaul, and Pump 2 has marginally deteriorated in efficiency since 2019. Overall specific energy on 2-pump operation has improved by approximately 3% at 48Hz speed.

This is as expected as the individual pump duty points under parallel operation are further to the left of its performance curve and further away from its best efficiency point which is exacerbated with a reduction in drive frequency.

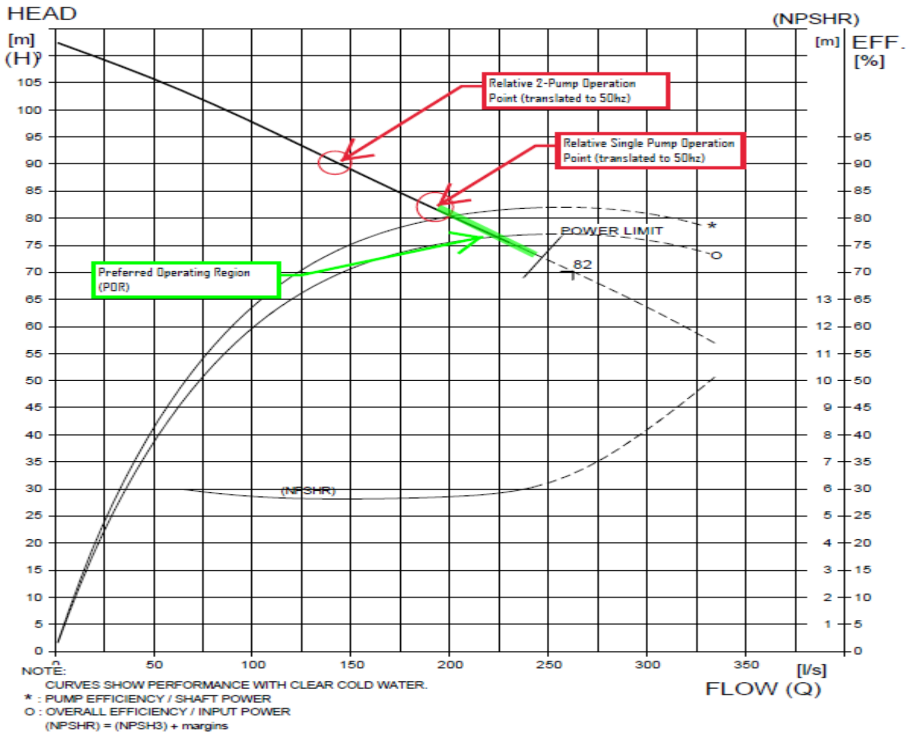


Figure no. 9: Manufacturer's Pump Curve and Relative Operating Points at Caen Hill

5.2 Annual Energy

Based on 2018 operations, an estimate of annual energy consumption and savings has been derived.

Pumping Configuration	Utilisation split (by time)	Estimated Volume (m3)	Specific Energy Phase 1 (kWh/1000m3)	Specific Energy Phase 2 (kWh/1000m3)	Specific Energy Difference (kWh/1000m3)	Energy Difference (kWh)
Pump 1 only	24%	1006266	310	294.2	-15.8	-15899
Pump 2 only	24%	1006266	296	301.3	+5.3	+5333
Both Pumps	52%	2147062	347.6	336.6	-11.0	-23618
Total	100%	4159154				-34183

6 Vibration Issues

6.1 Pump Vibration

Both pumps 1 and 2 have poor reliability and it is understood that the typical bearing life for both pumps generally is short at between 2,000-hours to 3,000-hours, against a design life expectation of 50,000-hours. This is indicative of excessive vibration.

Figure no. 9 shows the manufacturer's pump curve. From the system curve the relative operating points for Caen Hill PS for single and dual pump operations have been derived.

The pump curve has a preferred operating region (POR), defined by being 70 % to 120 % of flow rate at the best efficiency point (BEP), where vibration should be lowest. This is highlighted in green in and shows that single pump operation is within the POR but falls to the left of POR under 2-pump operation, which may be an indicator of higher vibration.

Since the 2019 Phase testing, the pump foundation plinths have been repaired and reinforced with steel straps and plates, though other concrete support/restraints have failed.

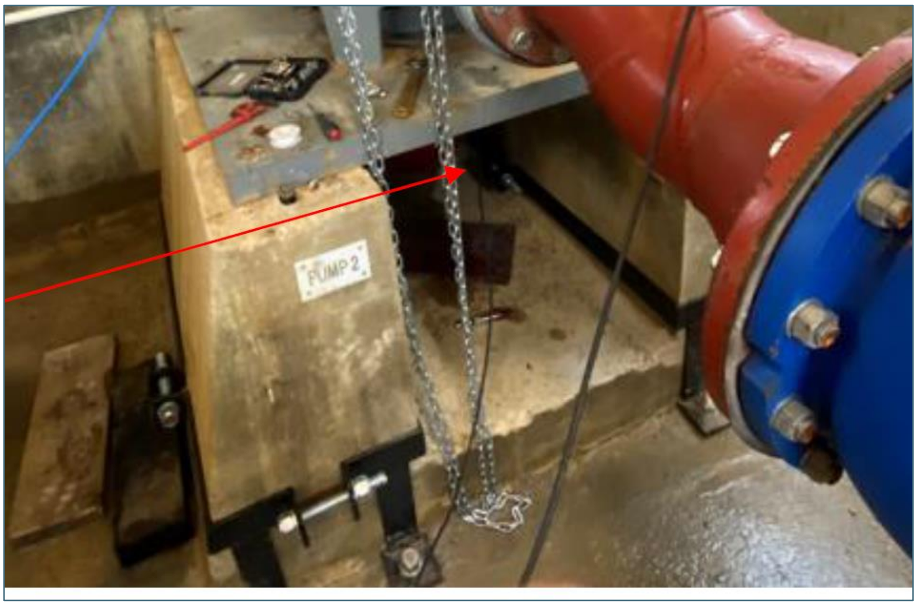


Figure no. 10: Steel Reinforcement to Concrete Pump Plinth

6.2 On-Site Vibration Measurement

Vibration measurements were taken using magnetic accelerometer transducers at the pumping station and positioned in accordance with the relevant guidance BS ISO 10816-7:2009 - Mechanical vibration - Evaluation of machine vibration by measurements on non-rotating parts - Part 7: Rotodynamic pumps for industrial applications, including measurements on rotating shafts.

The measurement was taken via the temporary in-situ placement of a magnetic transducer on the X and Y axes at the drive bearing end (just above volute casing), and at the corner of the mounting plate.

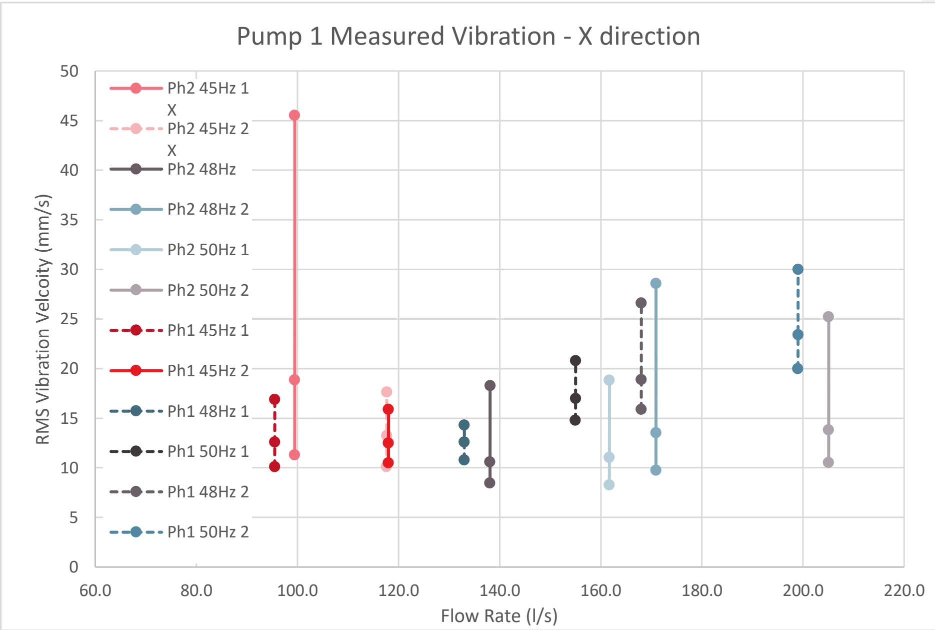


Figure no. 11: Pump 1 Vibration (X - Direction) Recorded in Sep- 2019 (Ph1) and Nov-2022 (Ph2)

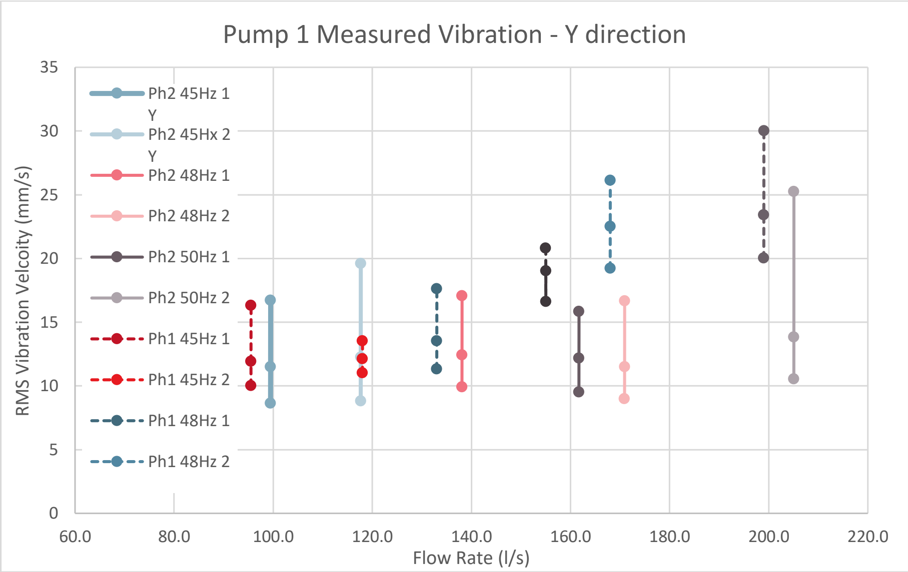


Figure no. 12: Pump 1 Vibration (Y - Direction) Recorded in Sep-2019 (Ph1) and Nov-2022 (Ph2)

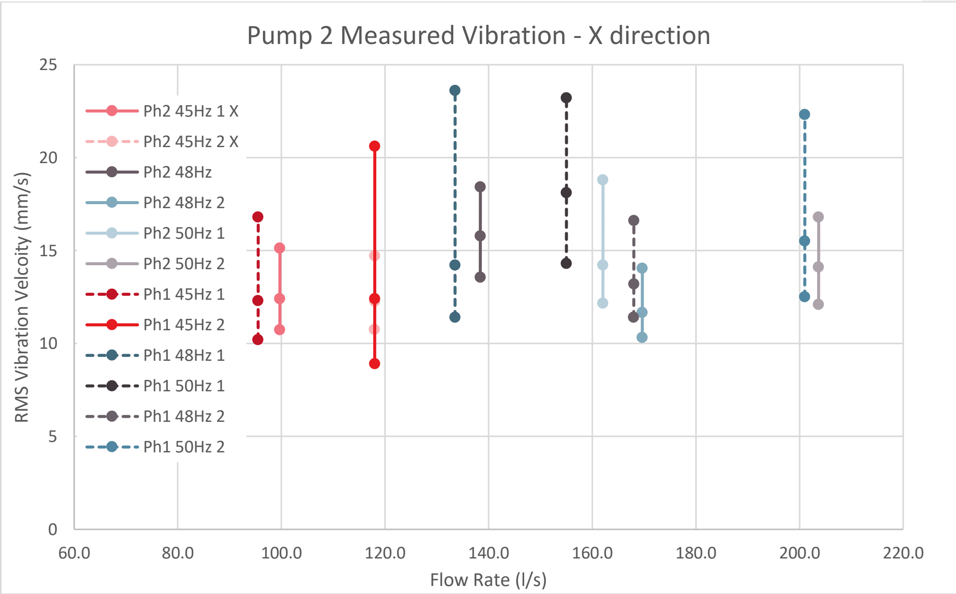


Figure no. 13: Pump 2 Vibration (X - Direction) Recorded in Sep-2019 (Ph1) and Nov- 2022 (Ph2)

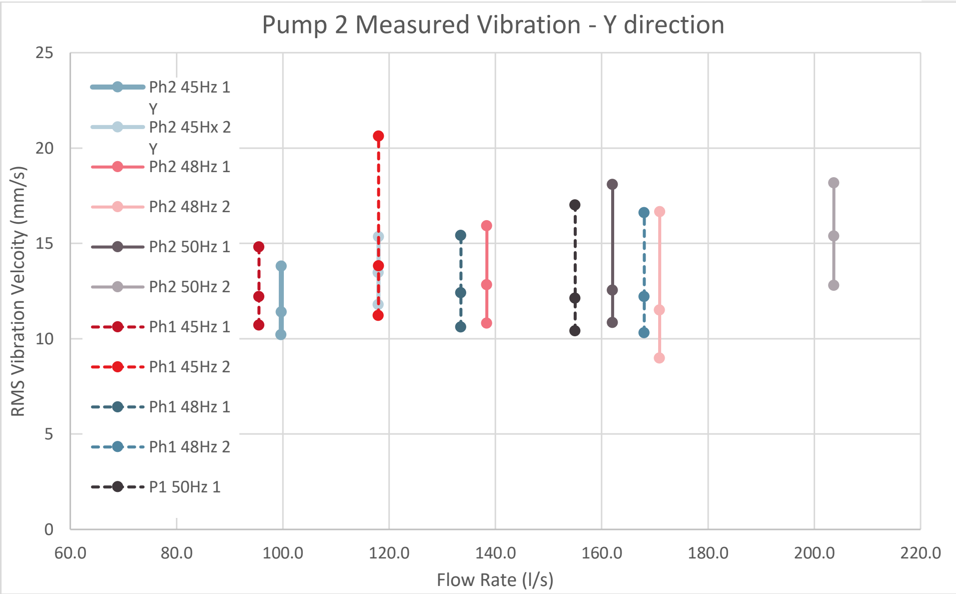


Figure no. 14: Pump 2 Vibration (Y - Direction) Recorded in Sep-2019 (Ph1) and Nov- 2022 (Ph2)

6.3 Discussion

BS ISO 10816 Part 7 defines the requirements for the measurement and evaluation of rotodynamic pumps. This suggests that risk of damage occurs at vibration velocities of above 9.5 mm/s RMS. However, it should be noted that submersible pumps are excluded from the scope of this standard.

Under Water Industry Mechanical and Electrical Specification (WIMES) 1.03 (dry well submersible pumps), a vibration limit of 11.2 mm/s RMS would apply to the 2-channel impeller pump. However, the manufacturer would need to advise their recommended limits.

As can be seen from Figure no. 11 to Figure no. 14, although there is a slight improvement since the Phase 1 testing , the peak vibration from both pumps significantly exceed the 11.2 mm/s RMS value under all tested operation scenarios. Therefore, we conclude that, despite the plinth improvements, vibration remains a concern and a probable cause of premature pump failure.

A dedicated test by a vibration specialist would help to understand the root causes from Fast Fourier Transform (FFT) spectrum analysis and motion analysis. This would identify the vibration frequencies and if the vibration cause is related to foundation, pump structure, imbalance, or hydraulics. The results should then inform on the required solution focus, e.g., foundation, flow presentation, supporting.

Some examples of specialist case studies on similar pump installations are listed below:

- Case Study | Fixing Resonance in a Non-Clog Pump Motor (mechsol.com)
- Case Study | Resolving a Sewage Pump Problem via Specialized Testing and Analysis (mechsol.com)

7 Additional Observations

7.1 Pipework

From the site visit, it was noted that pipe flange gasket failures have occurred at Caen Hill PS. The following items were also observed:

1. There is visible pipe movement following a pump hard stop event e.g., power cut/emergency stop under pressure surge conditions. The pump concrete piers already have steel bracing. At least one of the concrete thrust blocks / pipe restraints for the manifold have failed since previous visits, notably the bend from pump number 1. This is now no longer supported in the vertical at the bend, nor restrained from movement in the horizontal, other than via the bolted flanged joints to the individual pump pipework and the downstream manifold. Under a hard-stop this section of pipework was seen to move substantially, and regular lateral movement may cause failure of the pipework/flange joints.
2. Each pump delivery branch has ball check valves. These typically have a poor dynamic response.
3. The non-return valves are located close to the pump delivery (i.e. not achieving 3 or 5 diameters typically recommended by manufacturers) but this is largely dictated by the available space.
4. The delivery manifold knife gate valve seal is leaking.

7.2 Pressure Transients

During Phase 1, a basic hydraulic transient model was constructed in VariSim to estimate the surge pressures that could arise under normal operation. Several scenario simulations were then studied using the model, based on the current set up (48 Hz).

The findings, together with observed pipework movement following pump hard stop, suggest high pressure transients may be present

Table no. 5: Estimated Peak Surge Pressures at PS Manifold

Transient Event	Calculated Peak Pressure (bar.g)	Measured Peak Pressure (bar.g)
Uncontrolled 2-Pump Trip/Power failure	17.5	
Uncontrolled 1-Pump Trip (1-pump running)	17.4	15.8
Uncontrolled 1-Pump Trip (2-pumps running)	21.8	
Controlled Pump Stopping (30s ramp)	11.5	

8 Conclusions

8.1 Existing Pump Hydraulics Performance

The installed pumps have been selected appropriately by Xylem. Although they do not operate within the POR they are the best Xylem standard option available for a duty/assist configuration.

Both pumps are operating very close to their design hydraulic performance.

The inlet pressures monitored during the Phase 2 testing fluctuate which suggests less than ideal hydraulic conditions at the pump inlet and potentially contributing to cavitation damage. Operating left of the POR may also be a contributing factor to the cavitation wear of the impeller.

8.2 Pumping Station Pump Inlet Pipework

The inlet suction bellmouths are located higher above the floor than recommended by established pump intake design literature including *ANSI/HI-9.8 Pump Intake Design*. This may be contributing to hydraulic instability issues and cavitation.

Adopting a simple modification such as lowering the bellmouth and adding a suction cone (see adjacent figure) or divider plate may address this issue

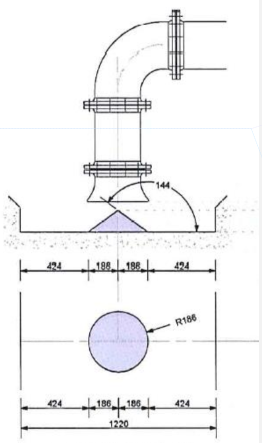


Figure 15 – Example of suction cone to improve inlet hydraulic stability

Commented [BK3]: We mention bellmouth / suction pipe diameters also being a contributary factor to this. Is there anything we can do to improve that, eg taper d/s of bend, then upsize bend dia and bellmouth dia? If not, then also probably worth stating this and explaining why - ie perhaps considered would have neglible impact vs the length thru the wall and bolted to the pump inlet.

Commented [JN4R3]: The sizing is a red herring and removed not that the comparison has been made. The scope of the changes as I see would be to lower the bellmouths and add a suction cone.

8.3 Pump Control

The specific energy analysis and review of the daily pumped volume, indicate the pump control could potentially be optimised to provide energy savings. The review of control would be subject to the hydraulic modelling review being undertaken by University of Liege and agreement from the Trust on potential and future requirements.

8.4 Vibration

Despite the recent installation of bracing to the pumping station pump plinths, concerningly high levels of vibration were measured on both pumps. Although, both pumps show slightly lower levels of vibration than measured during Phase 1 site testing. This excessive vibration is a probable cause of the short bearing and seal life being experienced by both pumps.

Resolution can be difficult under such circumstances, but further improvements to resolve the situation should be based upon the results of a specialist vibration audit including FFT spectrum analysis and motion analysis.

8.5 Pumping Station Delivery Pipework

There are transient pressure surges following hard stops that cause pipework movement and may be contributing factor to gasket failures. Pipework improvements to resolve gasket failure issues by means of additional thrust supports/anchors should be undertaken as a priority.

Consideration to a new pipework layout including changing the NRVs, increasing inlet pipe diameters, and new delivery isolation valve could help address ongoing issues.

Stop/start ramp rates could also be increased to “cushion” the return flow and lower the resulting surge pressures.

8.6 Rising Main

The removal of the intermediate NRVs has partly reduced the dynamic head losses. From our calculations, an unknown excess head loss of 3m to 4m at 300l/s remains within the rising main. It is unknown what is causing this higher head loss but further investigation may be subject to the law of diminishing return.

8.7 Energy Saving and Remaining Potential

The improvements from the rising main by removing the intermediate non-return valves and refurbishment to Pump 1 are estimated to achieve a 2% saving. This equates to approximately 34,000kWh based on 2018 operation volumes.

There is further potential to reduce overall energy usage at this site, although the priorities should initially focus on resolving resilience issues, such as vibration.

Several aspects could be explored in more earnest, such as further exploration of the rising main losses, changing impeller diameter, and the potential change to more efficient drives. Further improvements may follow from the University of Liege modelling review to optimise the pumping regime.

Table no. 8: Potential energy savings by option/action (based on 2018 flows and 4.6M m^3 total volume)

Option/Action	% Saving Over Existing	kWh per Annum
IE3 Motors	0.2	3,000
Fixed Speed Drives (+IE3)*	5	75,000
Larger impeller/motor Duty/Standby Configuration (+ IE3)	10	150,000
Improving Rising Main Head loss**	4	60,000
** Subject to vibration issues being resolved*		

9 Recommendations

1. Undertake additional anchoring of the pump station outlet pipework to stop movement under surge conditions.
2. It is recommended that the vibration issues are subject to a specialist vibration analysis. Resolving this issue would have the potential benefit of increasing the bearing life of the pumps, leading to increased reliability and efficiency.
3. The site test data suggests there is a higher dynamic loss than expected in the rising main of up to 4 metres. Further investigation is recommended if this situation worsens.
4. Consult with University of Liege with respect to modelling recommendations and finalise the levels and flow rates required to maintain the system in operation before finalising the pump selection, duty configuration, and control.
5. Continuing existing pump operation at 48Hz achieves the best specific energy and vibration balance.
6. Assuming the flow rates are not changing, consider new IE3 motor Flygt CT3420 pumps on replacement and retain existing as boxed spare units.
7. Implement inlet and delivery pipework and anchorage improvements for the purposes of achieving:
    - improved inlet hydraulic conditioning to the pumps
    - resistance to movement during surge conditions
    - address existing valve leakage
    - a better separation of pump and NRV

Appendix A

Guide to System Curves and Pump Performance Curves

The System Curve

Consider a pump system (Figure no. 16) where water is required to be conveyed from Point A to Point B at a Flow Rate of Q_D.

As the elevation of the water surface at the delivery Point B is higher than at A, it cannot flow under gravity, so pumps are required to lift the water. The elevation difference that the pumps are required to overcome is known as the static head, H_S, where H_S = Surface Elevation @ B – Surface Elevation at A.

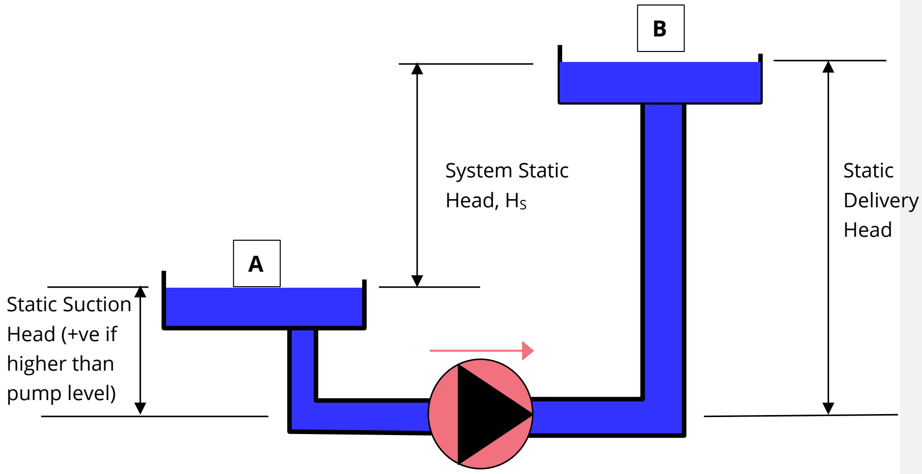
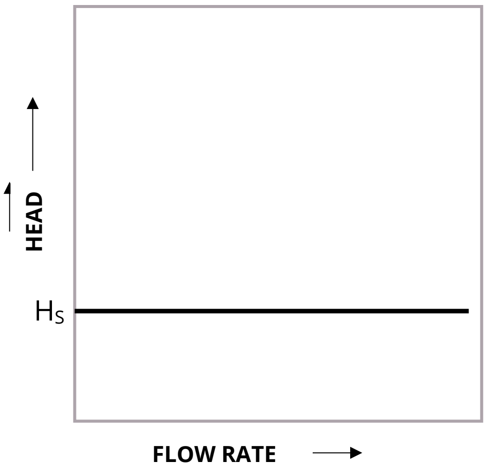


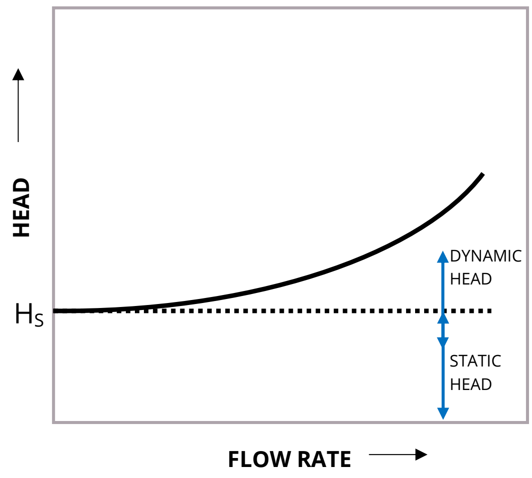
Figure no. 16: Pump System Representation

The calculated static head can be represented on a chart with head on the y-axis and flow rate on the x-axis, as follows:

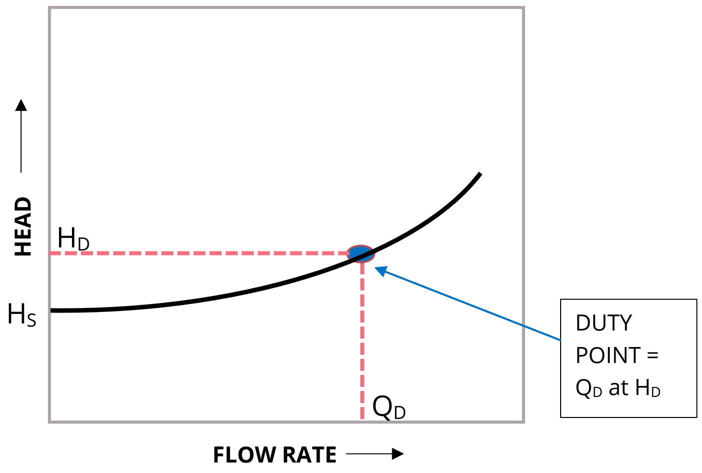


With an increasing flow rate, the flow resistance of the pipe and pipe fittings increases due to friction. So to achieve a higher flow rate, more pressure (or head) is then needed to be generated by the pump. The head losses due to friction increase proportionally to the square of flow velocity and is referred as "Dynamic Head". The Total Head for a given flow rate is the sum of Static Head and Dynamic Head.

Using established equations and loss coefficients the head unique to the pipe system can be calculated at various flow rates and its curve plotted as shown below. This is known as the "SYSTEM CURVE" or "SYSTEM CHARACTERISTIC".

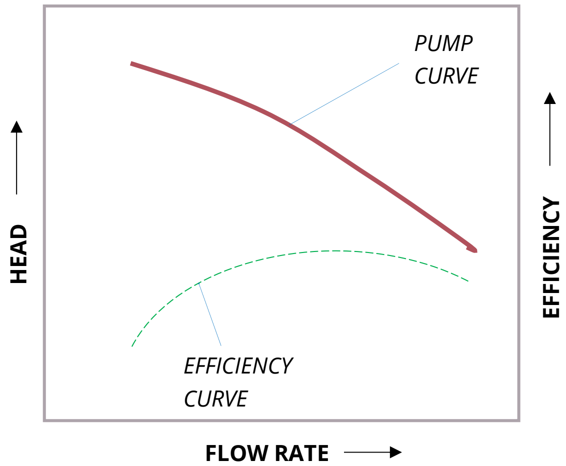


To select a pump for the desired flow rate, Q_D, the intersection point at H_D, is translated from the system curve. This is known as the desired "Duty Point".

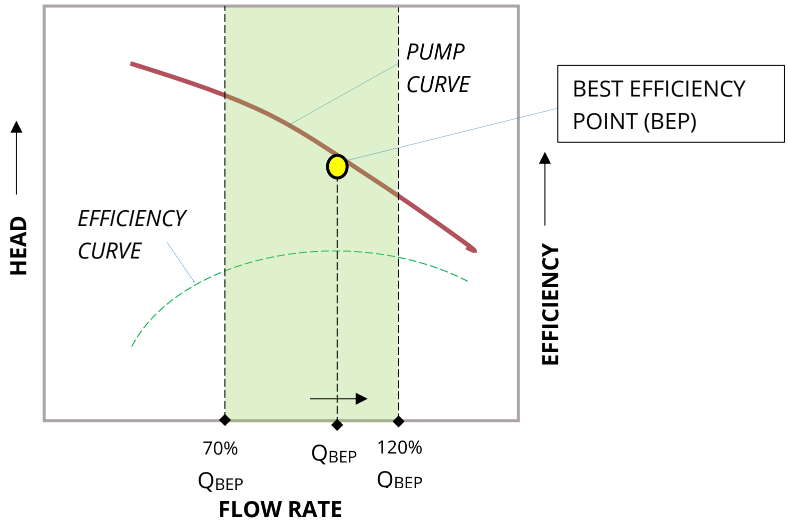


Pump Performance Curves

Centrifugal pumps have a flow-head characteristic known as the “PUMP PERFORMANCE CURVE” or “PUMP CURVE”. This shows the flow rate that can be generated by the pump for a given head. The pump curve will typically fall as flow rate increases. The pump efficiency will typically vary with flow rate, initially rising to a peak and then falling away at higher flow rates. These can be represented by curves aginst flow rate as shown below.

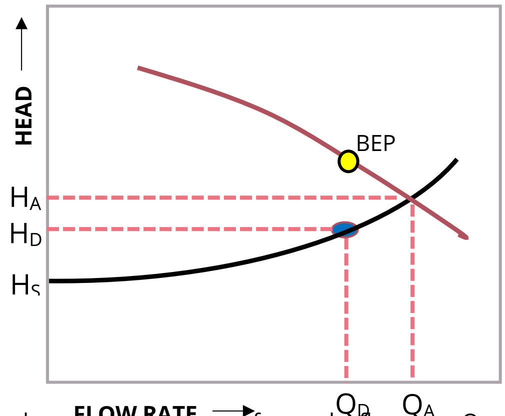


The point on the pump curve where efficiency is highest is known as the “BEST EFFICIENCY POINT”, abbreviated to “BEP”. The flow rate at this point is abbreviated to Q_{BEP}. Well selected pumps generally perform at flow rates within 70%< Q_{BEP} < 120%, in what is known as the “PREFERRED OPERATING REGION” as highlighted below.



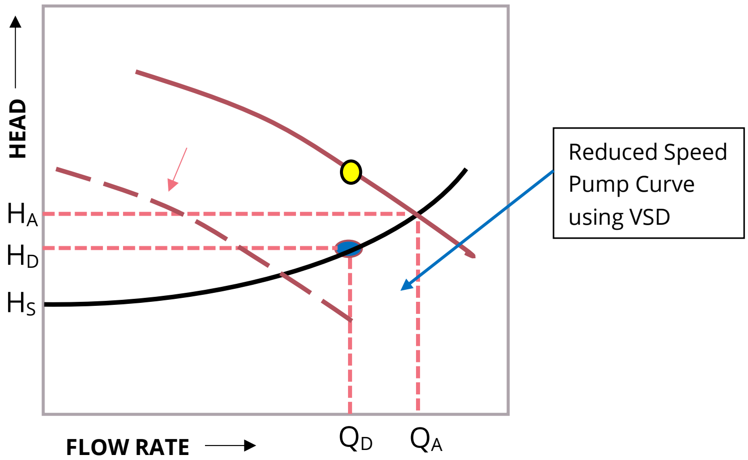
Combining System and Pump Performance Curves

The performance curve for a pump can be overlaid on the system curve. The expected flow rate for a particular pump is indicated where the pump curve intersects the system curve.



In the above exampl ass forward a flow rate, Q_A at a head of H_A (Actual Duty Point) which is higher than the desired duty flow rate, Q_D and higher than (to the right) of its best efficiency flow rate, Q_{BEP}.

Pump Manufacturers can provide a pump curve reflective of a particular model of pump "as new". Sites tests can provide an actual pump performance curve through varying pump speed or adjusting valves and measuring flow rate and pressure.



Using variable speed drives it is possible to change the pump speed by changing the frequency of input power to the pump motor. This varies its performance curve meaning that it can meet a desired flow rate. In this example, the speed is reduced to meet the desired duty flow rate, Q_D.

Appendix B

Samatrix Site Test Report

Interreg
North-West Europe
Green WIN

Canal & River Trust
Making life better by water

LIÈGE université

Ministerie van Infrastructuur en Waterstaat

Waterways Ireland
Uiscebhealaí Éireann Watterweys Airlann

vnf
Voies navigables de France

VLAAMSE LAND MAATSCHAPPIJ