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Keynote address: Variability of pump system performance

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The pump is the heart of most industrial and mining plants providing the energy to separate valuable minerals from process feed. The actual effect of the pump, both in terms of production costs and overall process efficiency, is often assumed or misunderstood by engineers due to insufficient pump performance data. Pump performance can be understood and corrected, where required, through continuous performance monitoring, leading to dramatic improvements in both pump life cycle costs and system efficiencies.

The performance of every pumping application is subject to variation and the main contributing factors are as follows:

- Unavoidable deviations from pump supplier curves
- Pump wear
- Pump curve derating factors for slurry pumping
- System curve variation
- Deviation from basic principles of centrifugal pump operation.

These are discussed in detail through case studies and the author's personal experiences, proposing ways to minimize the effect of these variations, thereby reducing performance loss and preventing damage to the pump.

The paper discusses the benefits of efficient pump operation, the requirements of pump performance monitoring and its practical application to pump operation in the slurry and water environments. The cost of remedial actions is explored and compared against the savings in both power and plant reliability, resulting in a reduced life cycle cost of the pump

Introduction

Pump performance is defined as the ratio of the energy put into a pump against the work performed by the pump. While most pump operators focus on maintaining pump delivery, there is an increasing trend in industry to maximize pump performance while sustaining delivery or process requirements.

The performance of every pumping application is subject to variation. The effects of these variations are, for the most part, unavoidable, but to prevent excessive performance loss and damage to the pump, the user must understand their cause.

This paper documents these factors for slurry and water pumps.

Pump performance curves

A centrifugal pump can operate in a system only at a point where the pump performance curve intersects the system demand curve. This is the basic principle of centrifugal pump operation.

This requires the user to have a priori knowledge of the pump and system curves, in order to maintain operation within an acceptable operating envelope. The exercise is generally performed during the design phase of a pumping project, and is solely based on theoretical data. However, variations in pump performance occur throughout the life of a pump as the pump, fluid or system changes during the lifespan of the pump.

Pump water curves

Whether the user is pumping water or slurry, the manufacturer's curve will normally be based on water. Centrifugal pump curves are always provided as a family of curves, as seen in Figures 1 and 2.

The pump curve at full diameter and full speed is first generated after which the family of curves is generated, at different impeller diameters or speeds, using the affinity laws provided below.

$$\frac{Q_1}{Q_2} = \frac{N_1}{N_2} = \frac{D_1}{D_2}$$

$$\frac{H_1}{H_2} = \left(\frac{N_1}{N_2}\right)^2 = \left(\frac{D_1}{D_2}\right)^2$$

$$\frac{P_1}{P_2} = \left(\frac{N_1}{N_2}\right)^3 = \left(\frac{D_1}{D_2}\right)^3$$

Figure 1 shows an example of speed curves, where the impeller diameter remains constant while the impeller speed varies. Figure 2 shows a family of curves where speed remains constant while impeller diameter varies. (Curve taken from CH Warman pump catalogue.)

Hydraulic losses through a pump result in lost energy. At the entrance and exit of an impeller, turbulence is kept to a minimum when the flow is in the direction of the impeller vane. For this to occur, the velocity triangle, comprising the tangential and radial components, has to be balanced.

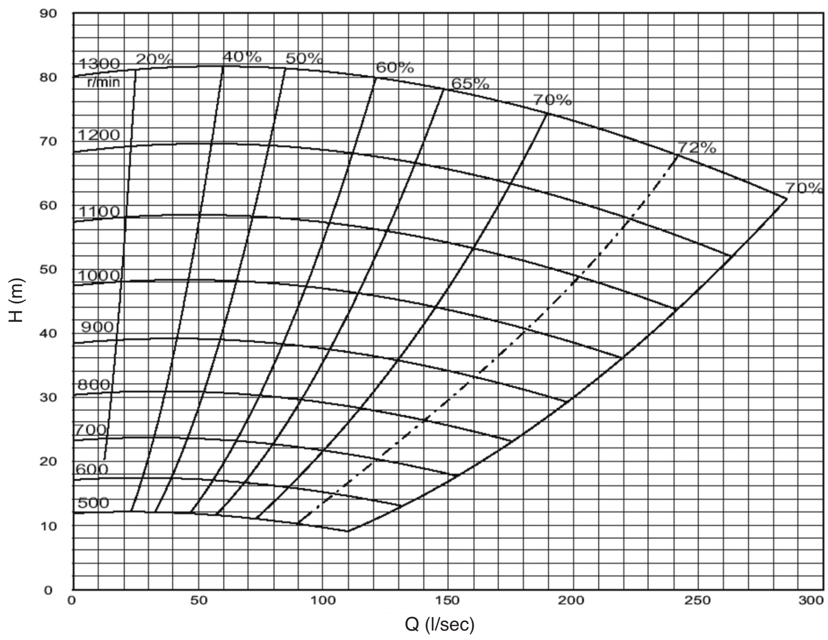


Figure 1. Family of pump performancespeed curves

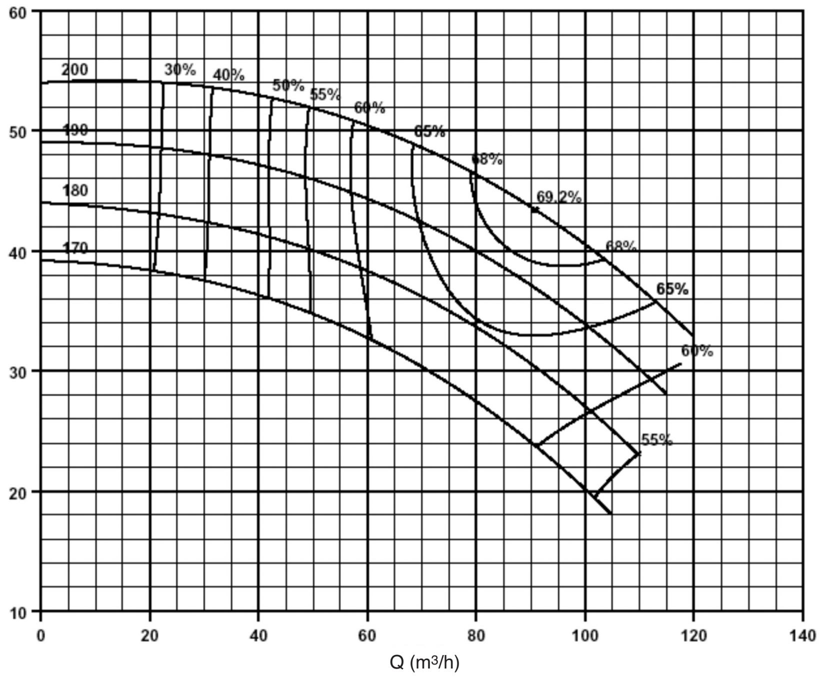


Figure 2. Family of pump performance trim curves (curve taken from Denorco pump catalogue)

The effect of impeller rotation on the fluid remains constant, and the balance is thus affected as flow rate deviates from the ideal. With flow velocity not in the direction of the impeller vane, the fluid separates from the impeller blades, increasing turbulence and energy loss.

Optimal energy efficient flow is thus achieved at the flow rate where this separation and increased turbulence does not occur or is kept to a minimum. This point is described as the best efficiency point or BEP, and the flow rate at that point is described as Q_{BEP}.

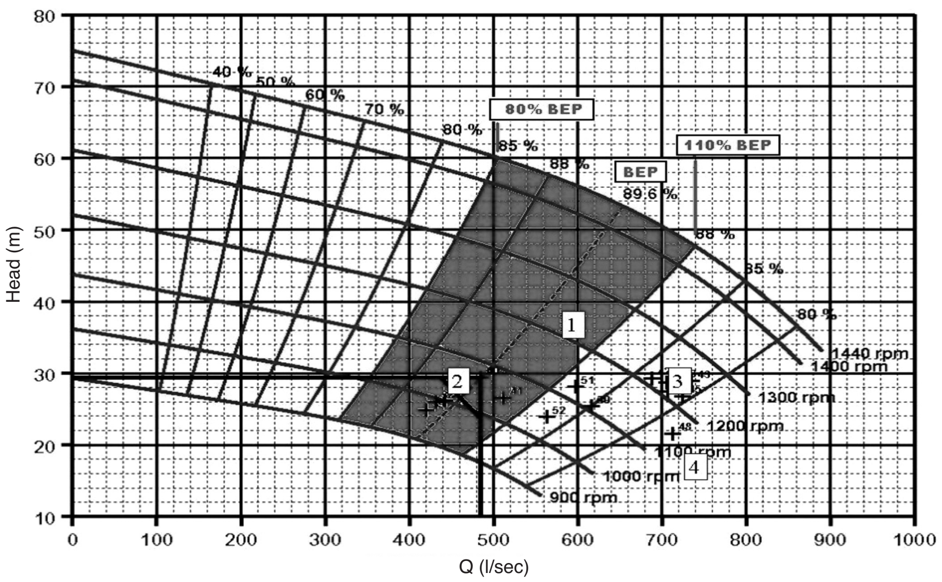
Centrifugal pumps are required to operate within a confined envelope, typically where flow is more than 80% and less than 110% of Q_{BEP}. (See Figure 3.)

Factors contributing to pump performance variation

Unavoidable deviations from expected pump curves

Pump users will encounter variations from pump supplier curves when performing their own pump tests on site. These deviations can usually be explained by:

- Pump manufacturing standards allow for varying accuracy during casting. Up to 8% deviation from the standard performance curve can be allowed.
- As discussed above, variation of pump speed will cause a variation in the pump water curve.



1 Pump Best Efficiency Point (BEP)

2 Ideal pump duty

3 Pump operating to the right of BEP

4 Pump cavitating (very far right of BEP)

Figure 3. The optimal pumping envelope

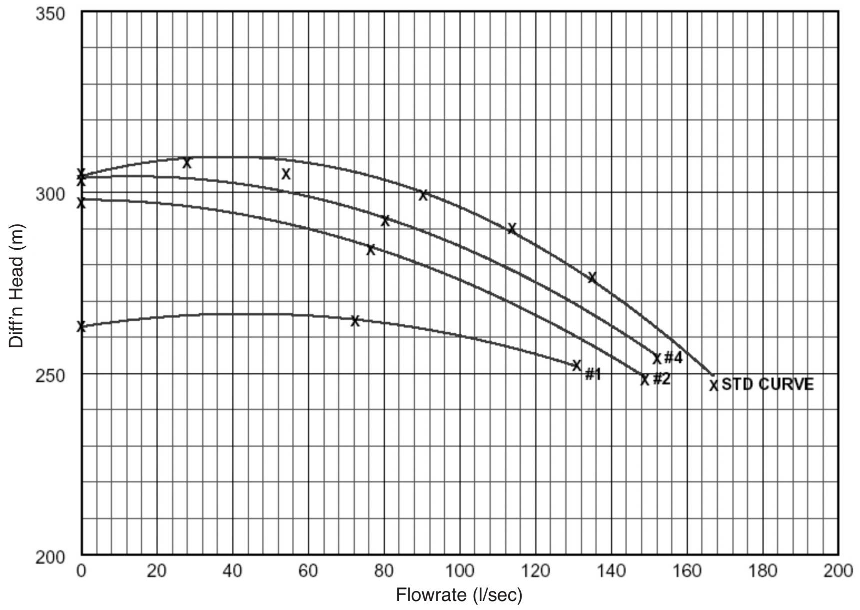


Figure 4. Comparison between new and worn pump curves (curve taken from TAS PumpMonitor report)

– Belt driven pumps always incur slip between pulleys and belts. This slip varies over the life of the pump or as pump load changes.
– Motors are rewound or replaced, resulting in variation of motor speed by up to 5%.

Pump wear

The most dramatic variation in pump performance comes as a pump wears and is able to produce less flow and head at the same speed. Thus a pump that is ideal for a specific duty when new may not be ideal once wear has occurred.

Figure 4 shows the results of an *in situ* pump test performed on three high-lift water pumps, identical except for their state of wear. One pump, Pump 4, is relatively new while another, Pump 1, is in a severe state of wear.

The test indicates that drop-off in performance due to wear can be approximated by using a smaller sized impeller. Accuracy of this method deteriorates as wear increases.

Case Study 1: mismatched pumps in parallel

The high-lift pumps shown in Figure 5 are run in parallel. When two new, identical pumps operate in parallel into a common discharge pipe, the pump flows are summed. This results in the combined pump curve as shown in Figure 5.

The quantity of flow loss when operating pumps in parallel depends on the gradient of the system curve. The steeper the system curve the greater the flow sacrifice. The sacrifice is shown below as c, i.e. $b + c = a$. High-lift pumps

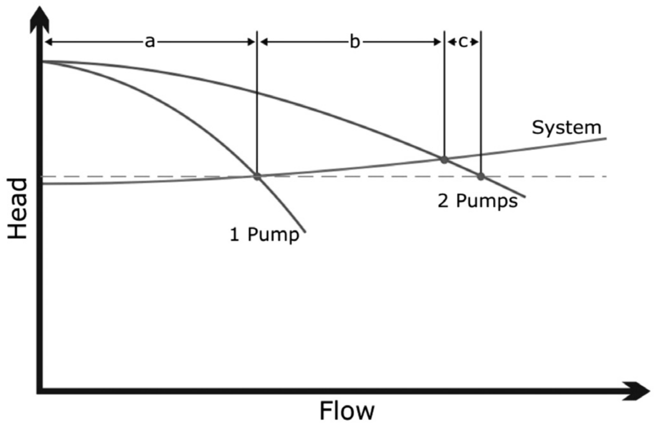


Figure 5. Pumping in parallel

generally have shallow system curves, but this cannot always be taken for granted, especially in older pump stations.

This effect is well known and always catered for in pump system design. The effect of running identical pumps, in different states of wear, in parallel is, however, seldom taken into account.

The additional losses incurred are explained in Figure 6. The older, weaker, pump is pushed back along its curve by the higher pressure achieved by the stronger pump. Generally, this means that the weaker pump will sacrifice even more efficiency. Again, the steeper the system curve the more the weaker pump will sacrifice.

The total sacrifice is again shown as *c*, i.e. $b + c = a$.

Case Study 2: overpumping/high recirculation

Fixed speed pumps in applications where a specific flow rate is required, have to be oversized to achieve the requirement over the life of the pump. In other words, the pump has to be selected to produce the required flow when worn.

The pump curve is not known for any condition except new, and pump selections are thus based on the as-new pump curves with estimates for the effect of wear on flow rate as wear sets in.

If the pump remains in service for too long, then the flow requirement may not be met for the remaining life of the pump.

Figure 7 shows the system requirement, shown as a red line, for a pump, to deliver 600m^3/hr. A new pump would thus need to deliver substantially more than this to meet the requirements once wear has set in. In this case the initial flow requirement is around 900 m^3/h.

The energy cost of such overpumping must be weighed up against the cost of a variable speed drive to eliminate overpumping.

The pump monitored drew 150 kW on average. An estimate of annual pumping cost is thus:

150 kW x 7200 h/year = 1 080 MWhrs/year
1 080 000 x 15*c* = R162 000

- Overpumping by 20% (720 m^3/hr) requires 20% more energy at a cost of R32 400 and 30kW
- Overpumping by 40% (840 m^3/h) requires 40% more energy at a cost of R64 800 and 60kW
- Overpumping by 60% (960 m^3/h) requires 60% more energy at a cost of R97 200 and 90 kW.

In conclusion, a business case exists to fit a VSD in this application.

Accurate calculation of pump curve derating factors for slurry pumping

A pump imparts its energy on the fluid being pumped. When acting on slurry, the water is required to drag the solids along, resulting in a loss of energy. This slip between the water and solids is known as derating and the result is that when a pump at a given speed operates on slurry instead of water, the head decreases while the power consumption increases.

This means that at design stage pumps are sized to provide the required head and flow. Derating factors are then applied to determine the correct operating speed of the pump. Derating factors, used at this point, are calculated or assumed from previous experience.

Two ratios, head ratio, HR, and efficiency Rrtio, ER, can be used to describe the pump's derating.

$$\text{Head Ratio} = \frac{\text{Total head developed on slurry}}{\text{Total head developed on water}}$$

$$\text{Efficiency Ratio} = \frac{\text{Pump efficiency on slurry}}{\text{Pump efficiency on water}}$$

If calculated, various methods exist to determine pump derating from theoretical data. Most methods will require the following inputs:

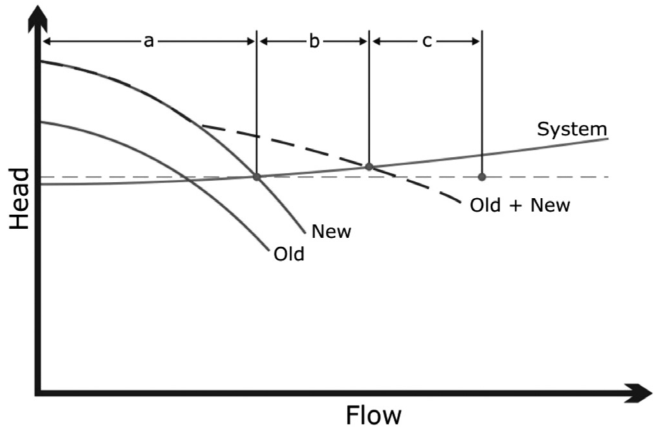


Figure 6. Running two identical pump in different states of wear in parallel

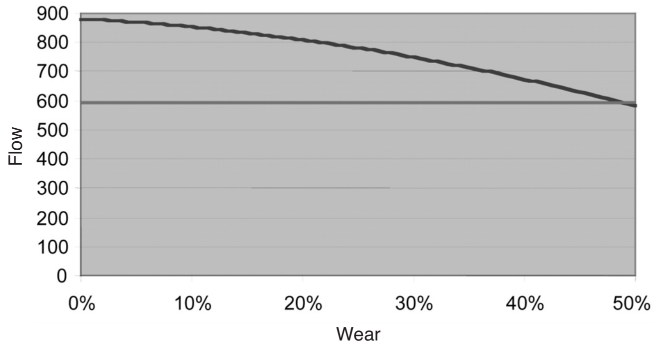


Figure 7. Actual pump delivery vs. % of full wear

- Density of solids
- Particle size distribution
- Average particle size, D_{50}
- Concentration of solids in slurry, CV
- Impeller diameter.

Even if the above data were available during the design phase, there is no guarantee that the method used for calculating HR and ER is correct. Once a pump is selected, based on the available data, any or many of the variables change over time, resulting in significant deviation from the estimates of efficiency and head ratios.

It should be noted that all methods of predicting pump derating are for Newtonian slurries. This means that as a fluid moves, the variation in shear stress exerted between different layers within the fluid in constant. When this is not the case, fluids are said to behave as non-Newtonian fluids. There is no predictive method for determining pump curve derating factors for non-Newtonian fluids.

Case Study 3: determining pump derating through pump testing

A correct medium pump at Anglo Coal's Klein Kopje Colliery was monitored. The fixed speed pump is required to supply 600 m^3/h to a header box, which supplies three cyclones.

Figures 8 and 9 show a pump selection performed for the application.

- Figure 8 shows a selection using a relative density of 1.6 and the head and efficiency ratios typically used for this type of slurry, 0.88.
  - Flow = 860 m^3/h
  - Head = 25.69m
  - Efficiency = 62.21%
- Figure 9 shows a selection using a relative density of 1.8 and the typical derating factors for the slurry at this density, 0.84.
  - Flow = 800 m^3/h
  - Head = 25.35m
  - Efficiency = 59.14%

The solid used for correct medium cyclone applications in South African washing plants is magnetite, a very fine iron oxide. After a period of performance monitoring, it was believed that the slurry might be behaving as a non-Newtonian fluid, and more specifically as a type of non-Newtonian fluid called a Bingham fluid. Bingham fluids display different shear stress characteristics at low flow, where additional energy is required to induce fluid slip.

For this reason, a pump test was performed on an overhauled pump to ascertain the pump's water and slurry curves for the specific application. Test results are shown in Figure 10.

The supplier's theoretical pump curve for a pump speed of 822 rpm is labelled 'Speed=822', while the curve generated during the test on water is labelled 'SG=1'. The difference between these two curves falls within the required specification.

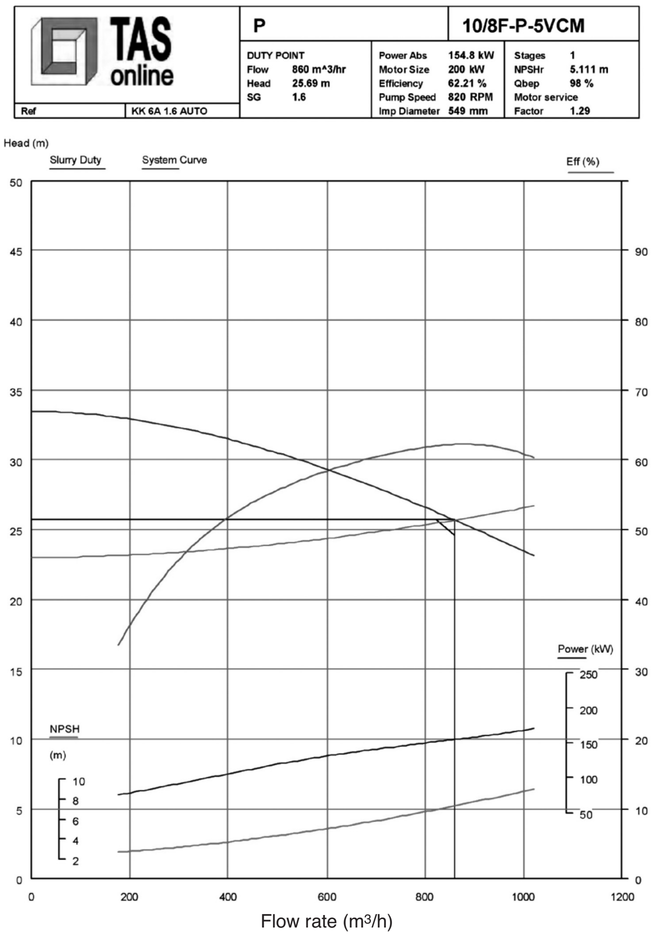


Figure 8. Pump selection using a relative density of 1.6 and head ratio of 0.88

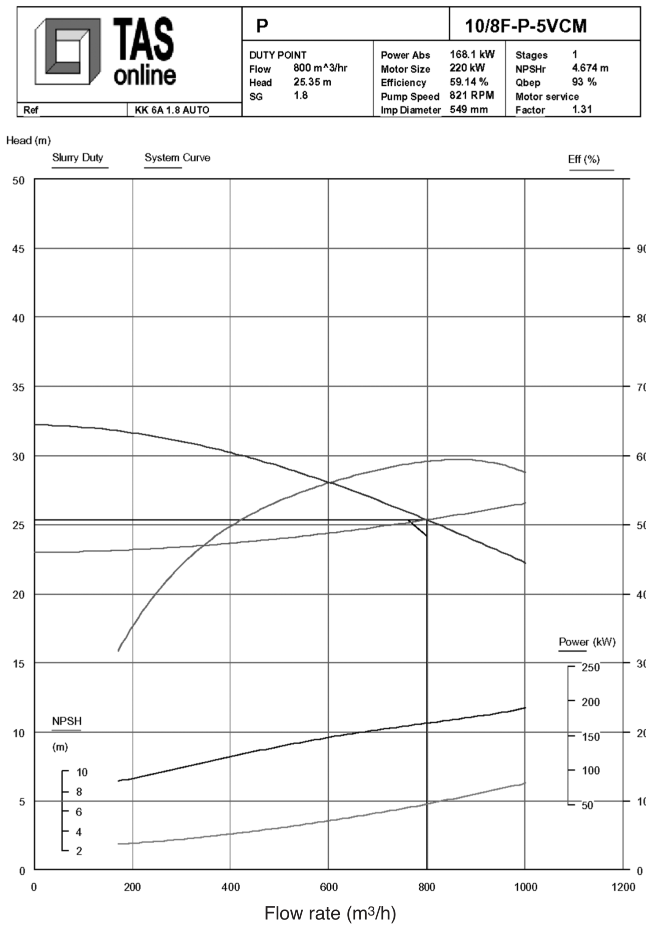


Figure 9. Pump selection using a relative density of 1.8 and head ratio of 0.84

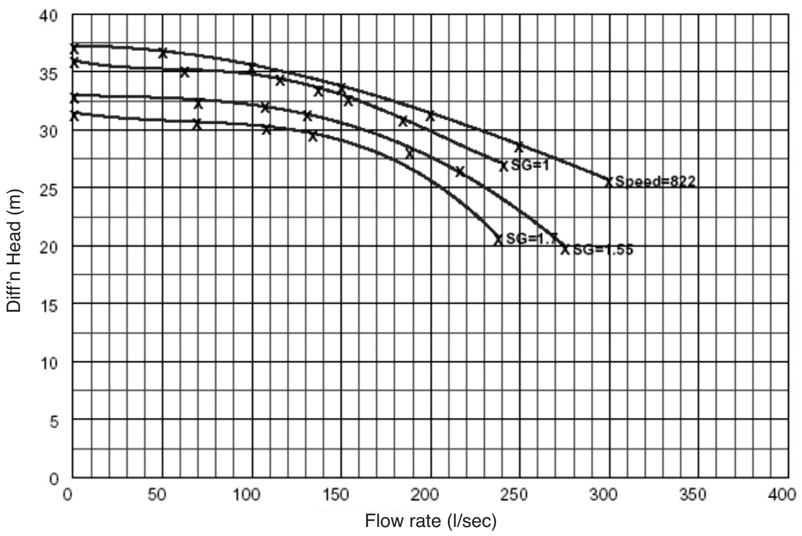


Figure 10. Determining head and efficiency ratios through testing

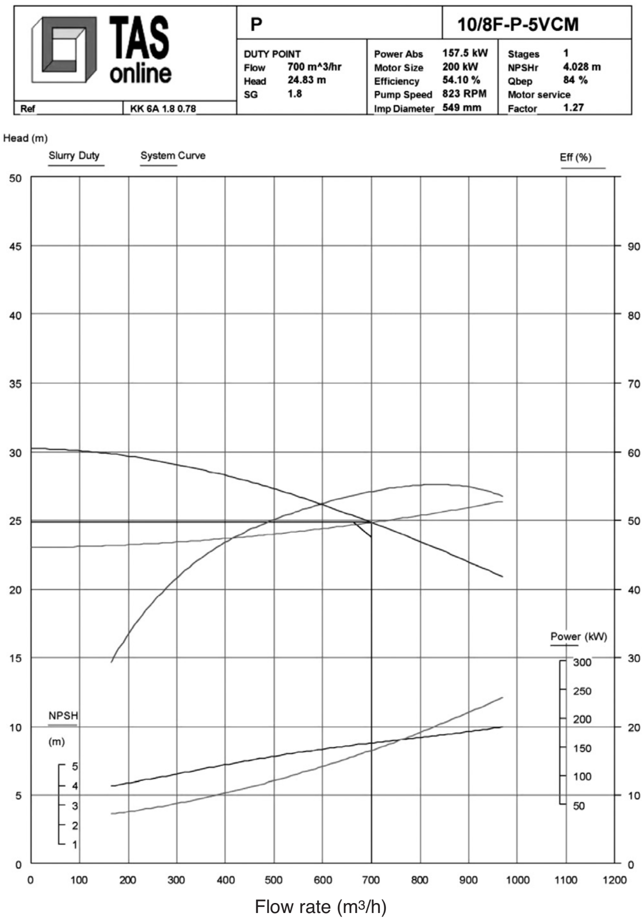


Figure 11. Pump selection using a relative density of 1.6 and pump test head ratio of 0.78

The slurry curves for SGs of 1.55 and 1.7 can be seen below the water curves. Results from this test allow the user to determine more accurately HR and ER.

The slurry was thus not behaving as a non-Newtonian fluid, but derating factors were higher than previous values used.

- When SG = 1.55, HR = 0.8
- When SG = 1.7, HR = 0.7.

From this we can estimate the HR for SGs of 1.6 and 1.8, which are 0.78 and 0.63 respectively. These empirical derating factors can now be used to determine actual pump performance. Figures 11 and 12 show the resultant selection using the test results.

For SG = 1.6,

- HR = 0.78 (not 0.88)
- Flow = 700 m^3/h

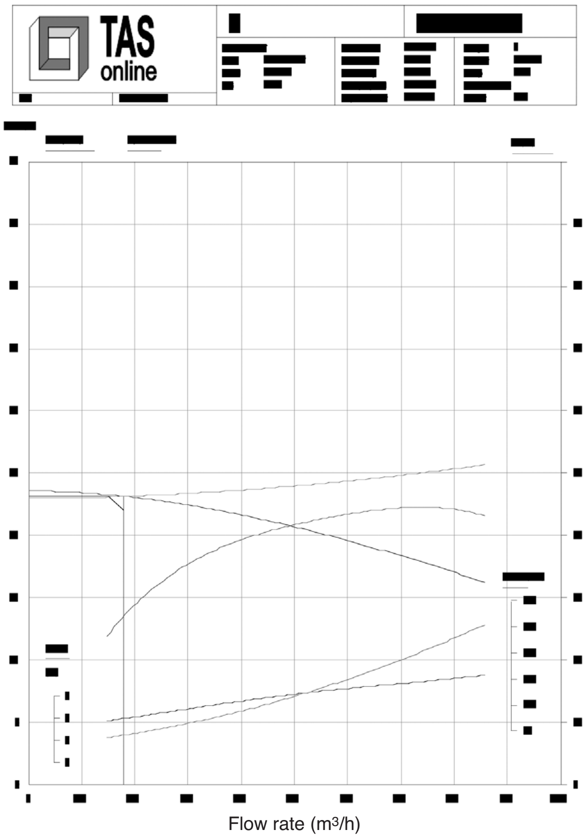


Figure 12. Pump election using a relative density of 1.8 and pump test head ratio of 0.63

- Head = 24.83 m
- Efficiency = 54.10%

For SG = 1.8

- HR = 0.63 (not 0.84)
- Flow = 180 m^3/h
- Head = 23.15 m
- Efficiency = 27.21%.

This indicates that the pump performance for a new pump in this application is above the required 600 m^3/h only when the density is kept around 1.6. This flow reduces as the slurry density increases to unacceptable levels.

In addition, flow will decrease as pump wear sets in, resulting in delivery dipping below 600 m^3/h. In addition HR and ER would change once again if the magnetite wears.

System curve variation

As the system curve changes, the operating point moves along the pump curve. System curves typically change for the following reasons:

- Suction pressure changes due to sump levels or variation in slurry density
- Piping wear occurs, reducing friction losses
- Pipe exit conditions change
- Density changes.

Water pumps

Figure 13 shows the pump curve and operating points for a high-lift, multistage pump. Each cross represents data taken once per hour over a day.

The level of the dam, from which this pump draws, does not vary by more than 20 m, and the effect of this change on suction pressure is thus minimal.

The more substantial variation seen in Figure 13 is from operating the pump with a second pump in parallel, while the dam level changes. Variations can be seen as the pump operating point (crosses) moves along the pump curve, resulting in variations in pump flow.

Case Study 4: system curve variation in variable speed driven water pumps

A water supply pump has a varying demand through a 24-hour cycle. This change in demand moves the system curve up and down.

The change in pump performance seen in Figure 14 below indicates pump performance outside of the pump's acceptable operating envelope. Each cross represents a duty that has been monitored during a 24-hour cycle. A variable speed drive is used to maintain the downstream pressure. Due to this simple control, it is evident that the pump operates outside of its optimum envelope, and in some cases on the extreme right-hand side of the graph, the pump will start experiencing severe cavitation.

This demonstrates the danger of using a variable speed drive in an application where the inadequate control feedback to the VSD allows the pump to operate outside its allowable operating conditions

Slurry pumps

Figure 15 shows the variation of operating points for the correct medium pump at Klein Kopje Colliery. Variation in a system curve in a slurry application is far more pronounced from changes in suction pressure, density and friction losses brought about by changes in slurry density.

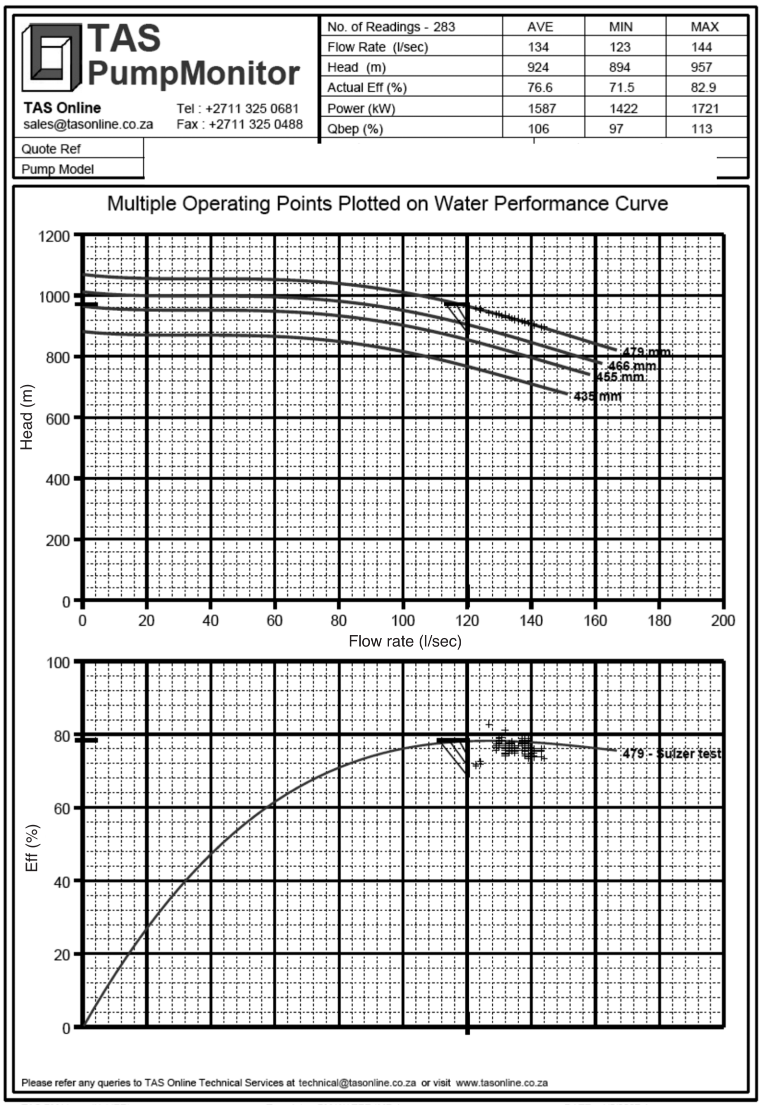


Figure 13. Pump operating point in water application moves along curve

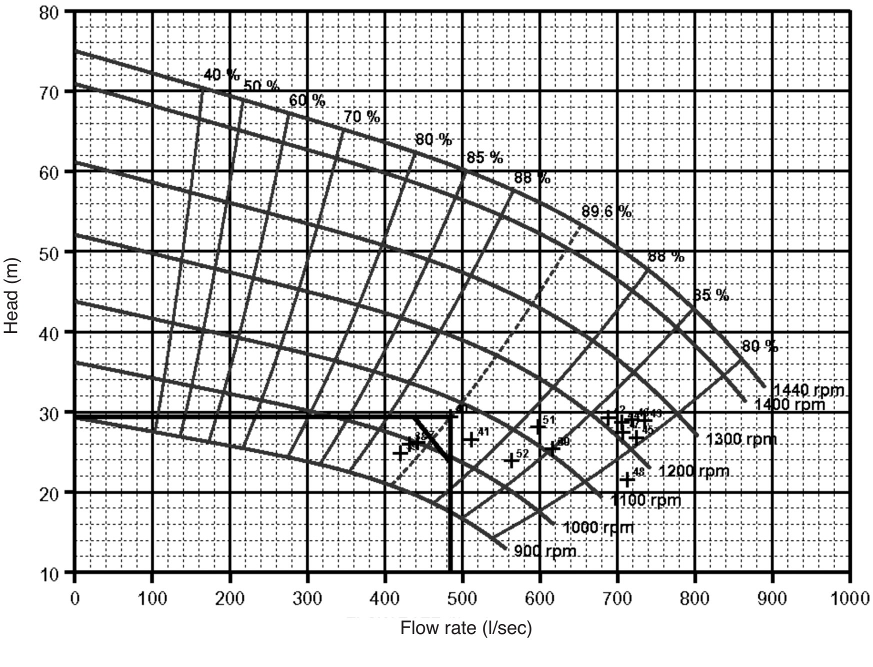


Figure 14. View current pump duty superimposed on standard performance curve

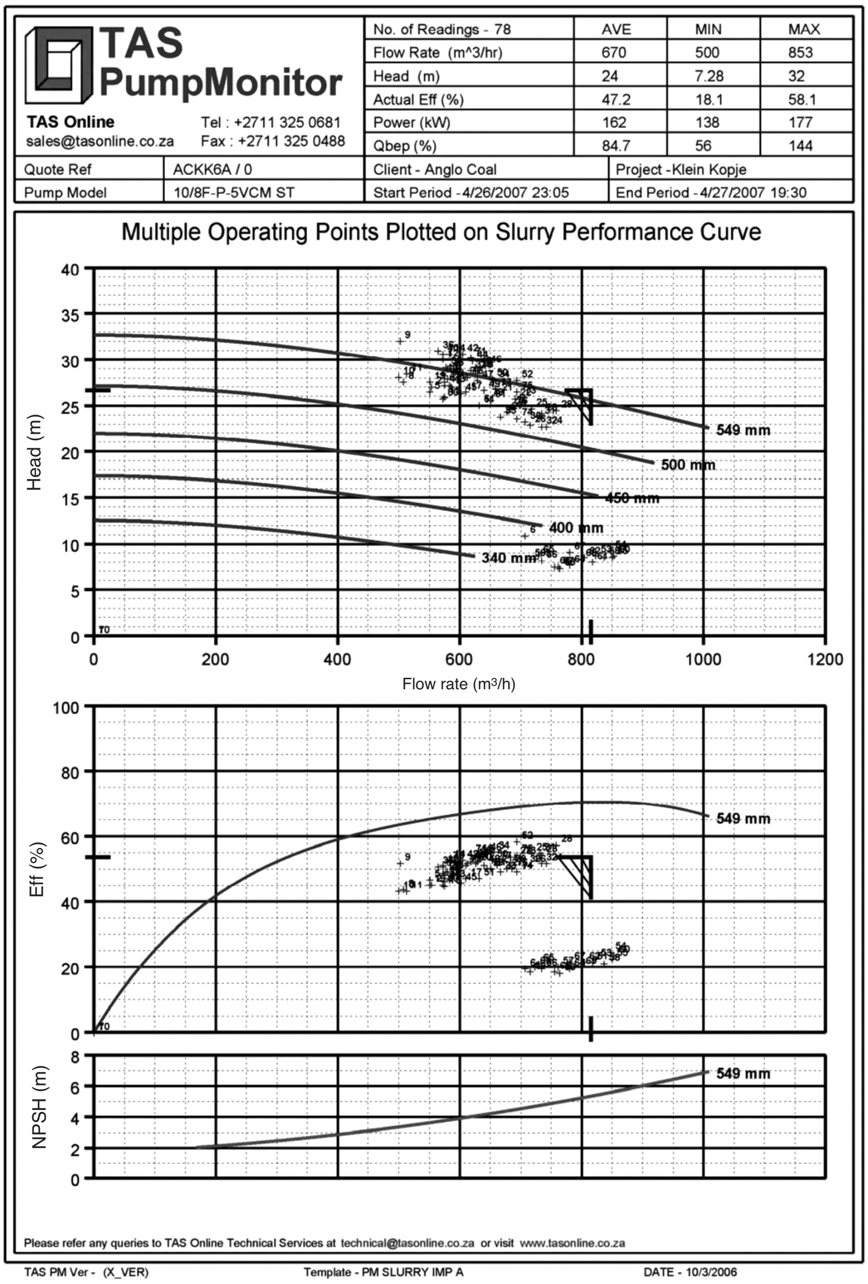


Figure 15. Pump operating point in slurry application moves due to variations in density

Case Study 5: system curve variation in variable speed driven slurry pumps

Cyclone feed pumps are variable speed driven to maintain correct supply during changes in sump level, density and pump wear. Changes in sump level and density change the system curve.

Monitoring pump performance will reveal whether the control philosophy applied effectively compensates for the variation in the system curve.

Figure 16 shows the monitoring of a VSD driven pump with requirements as described above. The oversized pump cannot be controlled due to the flatness of the pump curve at the far left, resulting in the sump filling and emptying in short periods as pump flow varies by 50% of full flow.

Once again we have seen an application where a VSD is unable to control the pump operation, in this case due to the fact that the pump is oversized for the required duty

Deviation from the basic principle of centrifugal pump operation

As discussed, a pump operating in a system can operate only at a point where the pump curve intersects the system curve. This is the basic principle of centrifugal pump operation.

When pump performance monitoring indicates that this principle no longer holds true then further investigation is required. This occurred at the Klein Kopje site where dramatic variations in performance were recorded over shorts periods, as can be seen in the performance plots below. The pump is no longer operating on its system and pump curves.

Further investigation was thus indicated and the root causes of the flow fluctuation were found to be cavitation when low tank levels were experienced and air entrainment due to a central fluid return to the tank. Both conditions are easily remedied to ensure stable pump performance.

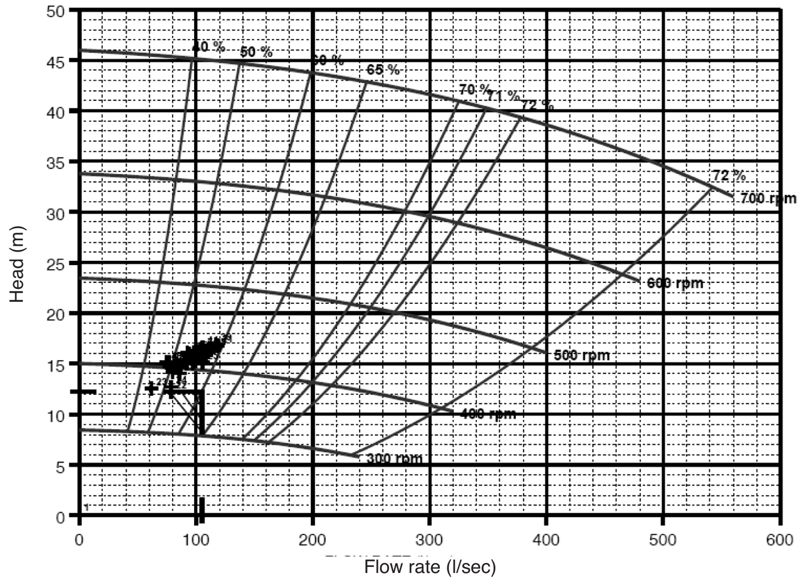
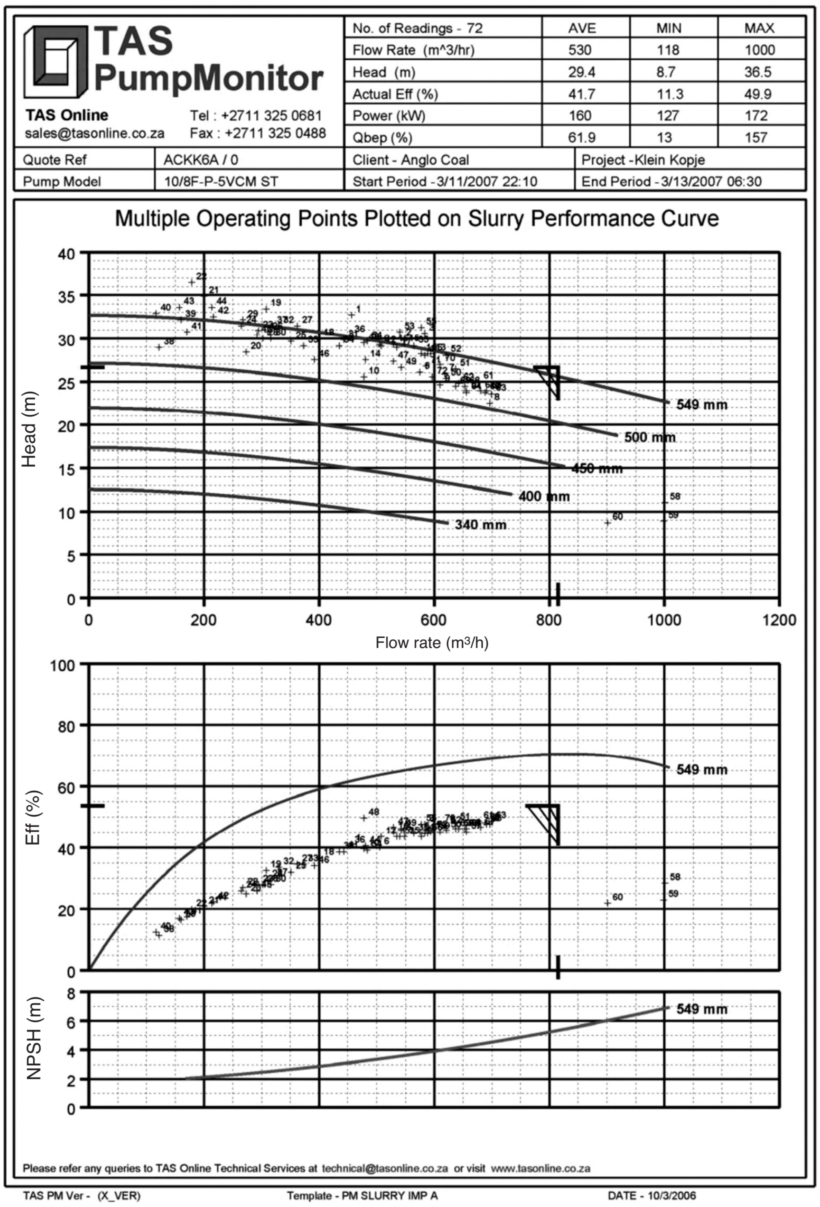


Figure 16. VSD driven oversized pump


TAS PumpMonitor

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No. of Readings - 72	AVE	MIN	MAX
Flow Rate (m^3/hr)	530	118	1000
Head (m)	29.4	8.7	36.5
Actual Eff (%)	41.7	11.3	49.9
Power (kW)	160	127	172
Qbep (%)	61.9	13	157

Quote Ref	ACKK6A / 0	Client - Anglo Coal	Project -Klein Kopje
Pump Model	10/8F-P-5VCM ST	Start Period -3/11/2007 22:10	End Period -3/13/2007 06:30

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Figure 17. Dramatic variation in pump performance due to cavitation and air entrainment

Figure 18 highlights the pump operating points when the pump is placed on recycle mode. Correct medium recycling is required when the plant if off-line to prevent the medium from settling to ensure smooth start up thereafter. In this mode the slurry is not pumped to height of the cyclone but returns straight to the tank.

The operating points highlighted below indicate that there is insufficient head against which the pump should operate. The operating points should be further to the right, but the very high flow rates initiate severe cavitation in the pump, reducing flow substantially.

Conclusion

There are many factors that result in variation in pump performance. Without monitoring pump performance data, throughout the life of a pump, it is impossible to understand the cause and effect of variations to determine appropriate corrective action.

The case studies discussed in this paper are a result of monitoring pump performance continuously to ensure complete visibility of pump performance.

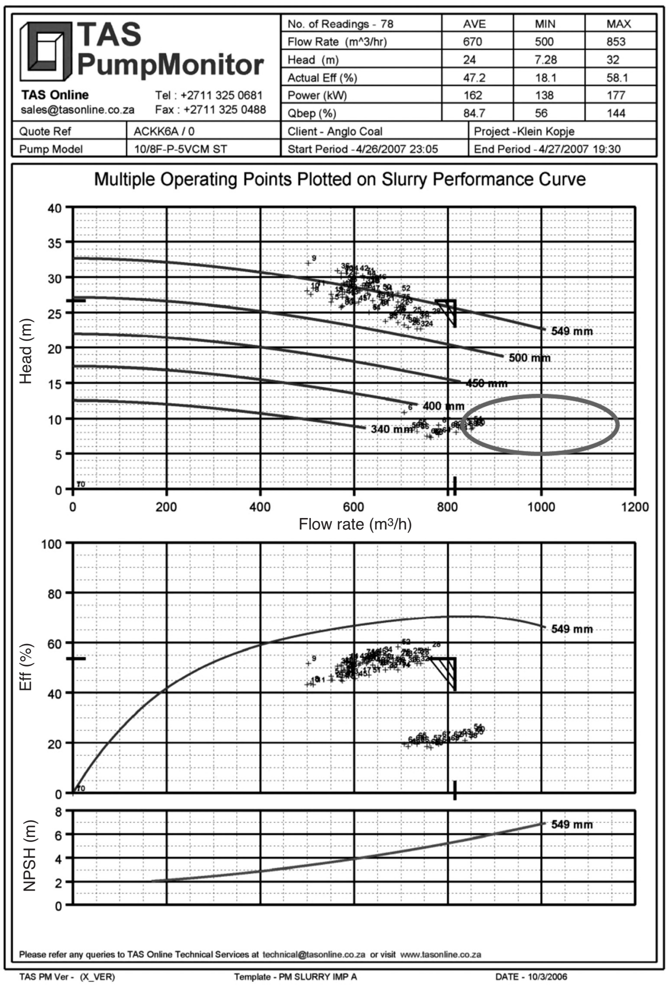

Figure 18. Severe cavitation altering pump operating point

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Harry's expertise has been developed over more than 20 years in the pump industry. In 1991 he founded Technical Application Software (TAS) in order to produce software for the pump manufacturing industry. These products, Testbed, Graftec and Aquatec, are in use with major pump manufacturers worldwide.

In 2000 Harry turned his attention to the user side of the pump industry and TAS PumpMonitor was developed to provide users of industrial pumps with continuous visibility into the operating performance of their pumps in near real time. He also leads the TAS Online team, which undertakes pumping system audits and provides recommendations on improving system efficiency and stability and reducing maintenance costs.

Harry is past Chairman of the South African Institution of Mechanical Engineering, Central Branch Region, which includes the industrial heartland of South Africa. In this capacity he founded the International Pump User Conference, which is now approaching its third event under his direction. He has had a number of papers and news articles published in leading international and South African pump journals.