USE OF SONAR FLOW MEASUREMENT FOR PERFORMANCE AND CONDITION MONITORING OF SLURRY PUMPS

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ABSTRACT

Monitoring of the performance and condition of critical slurry handling equipment in the mining industry, over extended periods of time, has been limited by the reliability of instrumentation. Techniques such as real-time pump curve monitoring, commonly applied in other industries such as power generation, have rarely been used in mining.
This paper demonstrates how an accurate, reliable, non-invasive, sonar flow measurement can be combined with existing plant measurements to produce a non-dimensional pump curve (Head vs. Flow) and efficiency curve (Efficiency vs. Flow) that can be used to monitor the performance and condition of a centrifugal slurry pump. The technique can be used with any flow measurement instrument; however an essential requirement for long-term monitoring is instrument repeatability and reliability which is difficult to obtain with invasive flow measurement instruments used in severe duty, abrasive slurry flows.
For example, in the case of variable-speed centrifugal slurry pumps, standard measurements commonly available in most plants, (e.g. pressure, flow rate, density, pump speed) can be used to obtain a non-dimensional pump curve (Non-Dimensional Head vs. Non-Dimensional Flow). This can be compared with the manufacturer's curve to monitor pump performance and wear over time. Condition-based maintenance programs are thus capable to more accurately schedule pump maintenance, thus avoiding costly unplanned shutdowns.
Additionally monitoring the efficiency of pumps allows the calculation of energy lost due to less than optimal pump operation. The energy lost could be as high as 100 MWh per month. Tracking energy loss may lead to important cost savings and may help sustain production in those cases where the power grid imposes limitations on the maximum amount of energy available.

INTRODUCTION

Monitoring performance and predicting maintenance needs of large diameter slurry pumps is a task that is approached by a variety of tools and techniques. Maintenance may be carried out either to prolong the life time of a pump or to increase the efficiency of the pump or both. Typical of life time prolonging maintenance is the dynamic balancing of impellers which reduces vibration and thereby increases the lifetime of a pump. Typical of maintenance geared to increase efficiency is the replacement of worn parts.

The preferred method for monitoring the extent of internal wear is that of head-flow, which reveals the condition of both the pump and the connected hydraulic system.[1] Head is readily measured with standard pressure gages. However the measurement of flow in severe duty slurry lines has been more problematic. In some non-lined pipes, non-invasive ultrasonic flow meters may be used. But typical severe duty lines are rubber or ceramic lined, thus making the use of these meters difficult or impossible. The recent introduction of non-invasive sonar-based flow measurement technology now makes it possible to achieve high accuracy flow measurements in severe duty pipes with virtually all types of lining, without suffering measurement degradation over time due to abrasive wear on wetted parts.[2] [3]

Since internal wear will reduce pump efficiency, it would be of interest to monitor changes in pump efficiency over extended periods of time. Pump efficiency is also affected by internal clearances between the high and low pressure sections of the pump, with large clearances leading to higher recirculation and thus lower efficiency. Typical large centrifugal slurry pumps have clearances that are adjustable to compensate for wear, thus increasing efficiency and compensating for wear of other internal components such as impeller surfaces. The competing affects of abrasive wear on non-adjustable internal parts such as impellers, and adjustable internal parts such as sealing rings, means that efficiency trends over time could be rather complex and not monotonically decreasing with abrasive wear.

One way of tracking pump efficiency is to compare over time, the hydraulic power generated by the pump and the electric power consumed by the motor driving the pump. Assuming that bearings and seals are well maintained and lubricated where necessary the wear of these might be negligible compared to the wear of wetted pump parts subjected to abrasive internal flow. If this assumption holds then the ratio of hydraulic to electric power could be a useful number to track as it should allow striking a balance between efficiency and pump wear, so as to better predict maintenance needs and avoid costly unplanned shutdowns.

This report describes a way to track pump efficiency over a fairly long (238 days) period for two large diameter centrifugal slurry pumps feeding a battery of cyclone separators. The method may be straightforward in theory but in practice it requires detailed attention to data quality.

METHODOLOGY

Pump Hydraulics

The purpose of a pump is to increase fluid pressure at a certain rate of flow. Centrifugal pumps achieve this by imparting kinetic energy to the fluid by means of a rotating impeller Subsequently by decelerating the fluid in the pump's snail house the pressure is increased. The theory of hydro kinetic machinery, including centrifugal pumps is well explained by Walshaw and Jobson.[4]

Dimensionless Numbers

In this analysis, it is desirable to use dimensionless numbers which serve to better represent various important pump metrics in terms of basic pump parameters, and permit the use of data from variable speed pumps such as those typically found in closed circuit milling operations. The six dimensionless number used are as follows.

As a measure of a pump's efficiency the head coefficient C_H is often used which a dimensionless number is defined as:

$$C_H = \frac{\Delta p}{\rho(\omega D)^2} \tag{1}$$

In this equation Δp is the pressure differential created, ρ the density of the fluid, ω the circle frequency of the impeller and D the diameter of the impeller. Under ideal conditions the fluid at the tip of the impeller leaves the impeller at the tangential speed of the impeller which is equal to the product of the circle frequency and the impeller radius. Thus the denominator in equation (1) measures the dynamic head of the fluid leaving the impeller whereas the numerator measures the static head. The ratio of the two is a measure of how well the pump converts the impeller's kinetic energy into pressure.

Ideal conditions only apply when the fluid enters the pump without axial or tangential velocity components and when the fluid leaves the impeller without radial or axial velocity components. To account for non ideal conditions it is useful and instructive to relate the head coefficient to a dimensionless value of the rate of flow sustained:

$$C_Q = \frac{Q}{\omega D^3} \tag{2}$$

Where Q is the volumetric rate of flow. The denominator in equation (2) is proportional to the volume swept by the impeller per unit time. Therefore the flow coefficient C_Q relates the flowrate through the pump to the pump size and speed.

The hydraulic power generated by the pump is simply the product of the rate of flow and the pressure differential created. By comparing this to the power consumed by the (electric) motor driving the pump shaft an overall efficiency may be defined as:

$$\eta = \frac{\Delta p Q}{VI \cos\varphi} \tag{3}$$

This efficiency definition assumes that the motor is a one phase motor with a power factor equal to the cosine of the phase shift between the current I and the voltage V.

Using the same equation for hydraulic power a dimensionless power coefficient may be defined that is independent of the pump's speed:

$$C_P = \frac{C_Q}{\sqrt{C_H}} = \frac{P}{\Delta p D^2 \sqrt{\frac{\Delta p}{\rho}}} \qquad (4).$$

A dimensionless speed coefficient is derived from the head coefficient as:

$$C_S = \frac{1}{\sqrt{C_H}} = \frac{\omega D}{\sqrt{\frac{\Delta p}{\rho}}} \qquad (5).$$

Clearly this is related to the impeller's tangential speed and the fluid velocity in the pump outlet.

All dimensionless quantities so far defined are related in some manner to the size of the pump expressed in terms of the impeller diameter D. A specific speed N_S may be defined as:

$$N_S = \frac{C_H{}^3}{C_Q{}^2} = \frac{\Delta p^3}{\rho^3 \omega^4 Q^2} \qquad (6),$$

Which is now a number that for geometrically similar pumps will be the same and will be constant at the maximum efficiency point.

The Best Efficiency Point (BEP)

The maximum efficiency of a pump may be calculated by curve fitting actual pump data. The model equation used for the head coefficient to flow coefficient is a quadratic:

$$C_H = a + bC_Q - cC_Q^2 \qquad (7).$$

In equation (7) a, b and c are positive coefficients. This quadratic model is widely used in pump hydro-kinetic theory and expresses both increase in head with increasing flow and the effects of friction. As a result of equation (7) the equation for the efficiency will be a cubic, the maximum of which can be found analytically. Now, corresponding quantities for the head, speed, power coefficients, and specific speed, all at the BEP can be derived easily.

Pump Data and Consistency

Data was obtained from a plant data historian, as shown in Table 1, and is commonly available in a modern concentrator plant.

Table 1. Pump data items

Item	Unit	Mnemonic	Range
Flow rate of pump	gpm	f	0 – 20000
Pressure of battery	psi	p	0 – 15
Density of slurry (by weight)	%	d	-20 – 100
Speed of pump motor	rpm	s	200 – 400
Amperage of pump motor	A	a	200 – 1500
Level in sump tank	%	l	-1 – 101
Number of cyclones open to flow		c	1 – 10

Consistency of data must be addressed, to overcome issues with missing data, unrealistic values, instrumentation and communication failures. This resulted in the use of automated data filtering which reduced the volume of data by as much as 20%. After proper filtering of the data, the pair wise correlations are obtained as displayed in Figure 1. Note the high correlation between pump current (a) and flow (f).

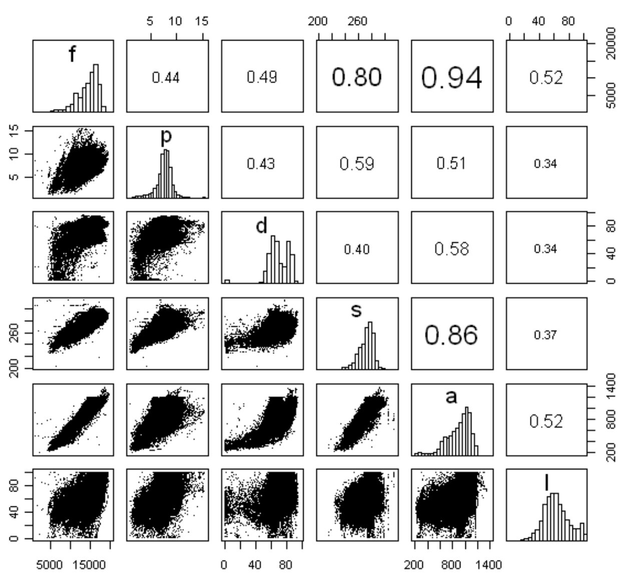

Figure 1. Pair wise data correlations.

Pump Performance Analysis

In order to assess the cyclone feed pump performance quantitatively by using the dependence of the efficiency of the pump on the flow coefficient it is necessary to first calculate the differential pressure over the pump. A schematic piping layout is shown below. The cyclones that are fed from the battery are not shown; only one cyclone inlet pipe (from a total of 10) are drawn.

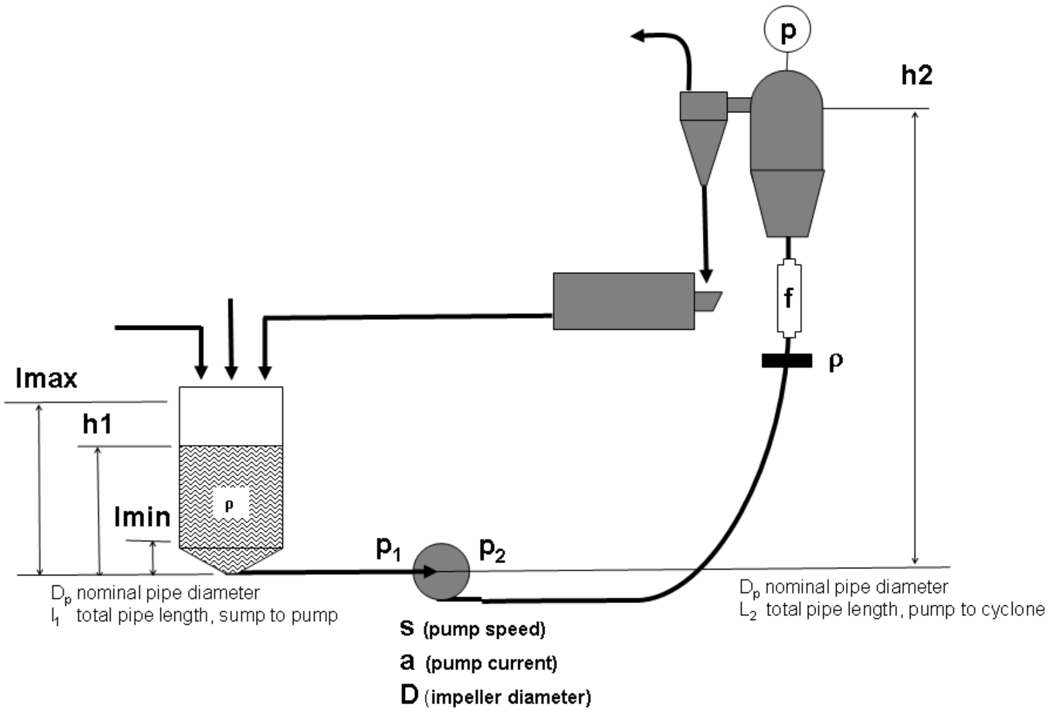

Figure 2. Pump, sump, battery piping layout.

The reference level for (hydrostatic) pressure is the center line of the pump. All vertical distances will be measured from this level. The pressure of the cyclone battery is measured at the top of the battery by a pressure gauge measuring relative to atmosphere. The sump level in the tank produces a positive intake pressure also dependent on the barometric pressure. Hence the difference is not dependent on the barometric pressure. Sonar based flow meters in the riser to the battery measure the pump's discharge flow rate.

Pump Differential Pressure

The losses in bends are not taken into account because the radii of the bends are very large. The acceleration losses are dependent on the rate of diameter increase from the pipe line to the battery which is small. The hydrostatic losses, however are by far the largest; an estimate of the other terms shows that at maximum pump rate the losses due to acceleration effects, sharp bends or sump tank outflow are each less than 3% of the hydrostatic loss. Note that density has no impact on this estimate because all losses scale linearly with density. Due to lack of precise measurements, various values are estimated as shown in Table 2.

Table 2. Hydraulic, pump and motor quantities

Quantity	Symbol	Value	Unit
Pump impeller diameter	D	54	inch
Carrier fluid density	ϱ_f	1000	kg/m^3
Solids density	ϱ_s	2750	kg/m^3
Carrier fluid viscosity	υ_f	1	mPa.s

Vertical height to top battery	h_2	50	ft
Vertical height to pump inlet	h_1	5 +10*l	ft
Gravitational acceleration	g	9.81	m/s^2
Flow line diameter	D_p	24	inch
Absolute flow line roughness	ε	0.4	mm
Motor voltage	V	460	V
Motor power factor	cosφ	0.7	
Downstream piping length	l_2	50	ft
Upstream piping length	l_1	30	ft
Level in sump tank	l	0 – 100	%

Using these hydraulic quantities (converted to SI units where necessary) we can calculate the density of the slurry being pumped as:

$$\rho = \frac{\rho_f}{1-\phi\left(1-\frac{\rho_f}{\rho_s}\right)} \tag{8}$$

The solids volume fraction follows from the calculated slurry density by

$$\varphi = \frac{\rho-\rho_f}{\rho_s-\rho_f} \tag{9}$$

Which in turn gives the viscosity of the slurry, according to the Einstein formula

$$\nu = \nu_f\left(1+\frac{5}{2}\varphi\right) \qquad (10).$$

The pump's inlet pressure follows as:

$$p_1 = \rho g h_1 - f\,\frac{1}{2}\rho V^2\,\frac{\pi D_p}{\frac{1}{4}\pi {D_p}^2}\,l_1 \tag{11}$$

The pump's outlet pressure equals:

$$p_2 = p + \rho g h_2 + f\,\frac{1}{2}\rho V^2\,\frac{\pi D_p}{\frac{1}{4}\pi {D_p}^2}\,l_2 \tag{12}$$

In both of the above, the Fanning friction factor f can be calculated from the Reynolds number Re by using the Churchill equation[5] which covers both rough and smooth pipes over the entire range of Re numbers including the transition region from laminar to turbulent flow.

RESULTS AND DISCUSSION

Sample Results

After calculating the inlet and outlet pressure of the pump the differential pressure is easily found. This enables the calculation of the six non dimensional quantities introduced previously. The head coefficient and efficiency are plotted as a function of the flow coefficient. The power coefficient and specific speed are plotted versus the speed coefficient. This gives a four plot presenting different views of pump performance each which is own merit.

1. The C_H, C_Q curve. This shows the increase/decrease of head of the pump with increasing flow and is modelled as an inverted parabola with the top a zero or a low discharge.
2. The η, C_Q curve. This curve defines the Best Efficiency Point (BEP), as the maximum.
3. The C_P, C_S curve. The power coefficient is a linear function of the speed coefficient.
4. The N_S, C_S curve. The specific speed, whilst independent of the pump's size is inversely proportional to the fourth power of the speed.

The data set (almost a year) is broken up in periods of one day (1440 minutes). Data from two days is cross plotted and fitted as per the formulas given previously. The BEP is determined and plotted as a single point in all four performance graphs. The fitted, empirical lines as well as the 95% prediction intervals are also plotted. The data points are grey value coded by the number of cyclones open to flow.

This cross plotting is repeated for each pair of days, moving forward in time by day. The determined BEP can thus be seen as an average over 2 days of data. Note that the 'days' as used here need not be consecutive calendar days because the data set had to be filtered heavily to remove non representative and bad data.

An excellent example of the analysis is shown in Figure 3. The black dots in the lower panel plots are right on top of the fitted lines showing good consistency as the dots are calculated from the BEP not from the data fit in those panels.

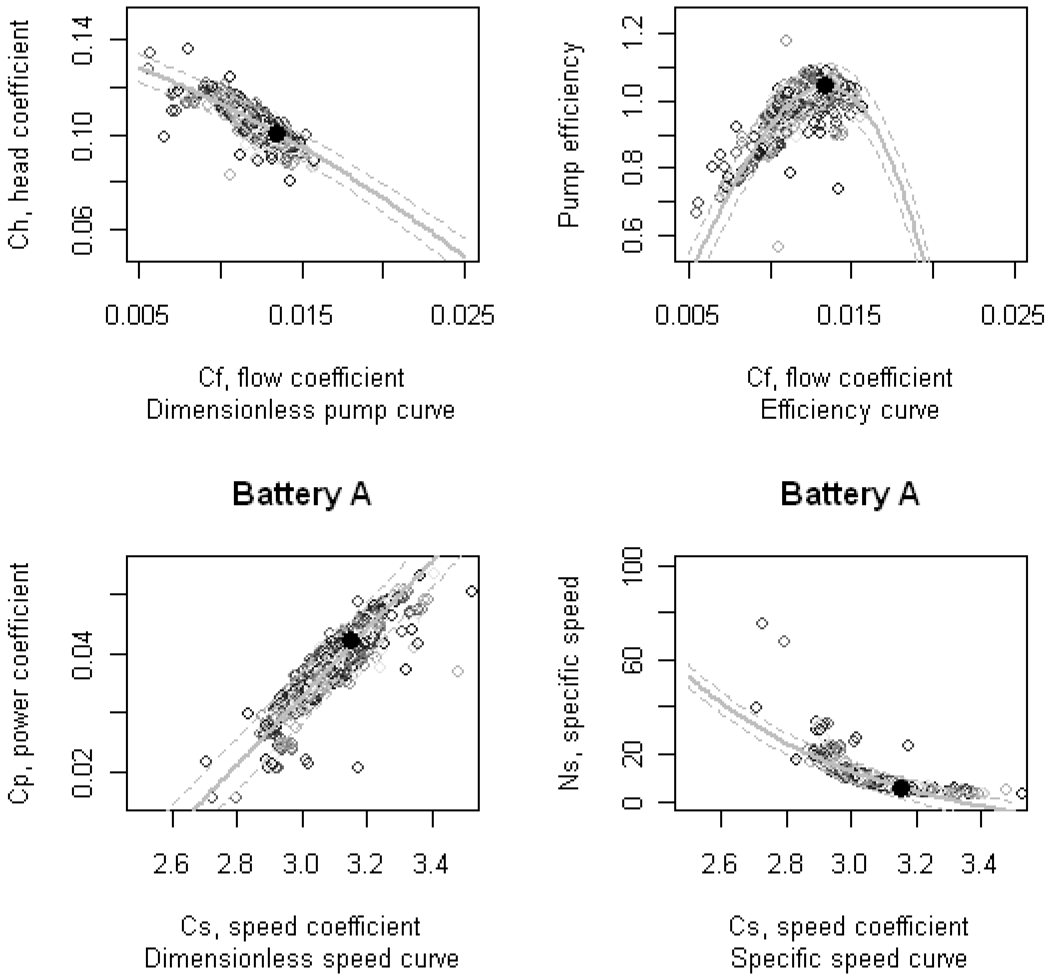

Figure 3. **Four plot of hydraulic dimensionless numbers.**

Furthermore the C_P – C_S data does indeed follow a straight line as it should according to the dimensional numbers introduced. Similarly the specific speed N_S, with the exception of some outliers, trends as the inverse of the fourth power of the speed coefficient.

Final Results

The main result of course is the value of the BEP. If we plot the BEP value versus the actual calendar day for both pumps we see that for Battery A this results in a downward trend in BEP whereas for Battery B no clear trend is discernible. Figure 4 below shows both trend plots next to each other. Because of the fact that certain parameters had to be estimated e.g. the power factor of the motor was estimated to be 0.7, it is possible that the efficiency is slightly higher than one. Calculations for both batteries were done with the same set of parameters. Therefore the efficiency differences between pumps are correct and so is the efficiency trend over time.

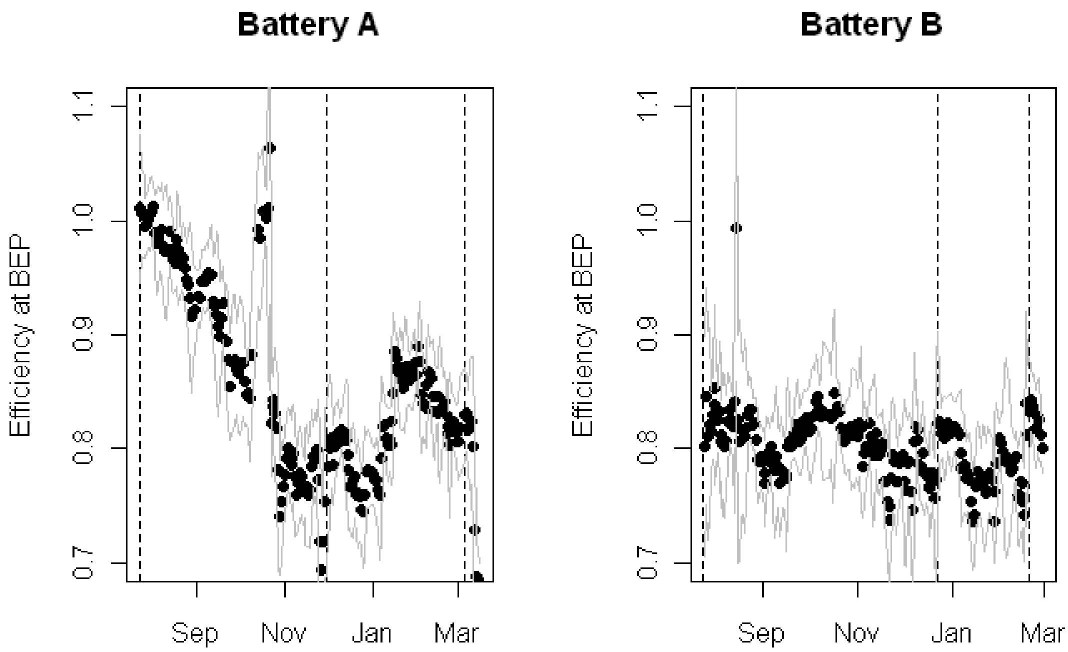

Figure 4. Best Efficiency Point (BEP) vs. time.
Vertical dashed lines mark the days on which pump maintenance was carried out. The mid period line indicates an impeller replacement only, the two outer period lines represent a complete overhaul. The grey lines depict the evolution of the 95% prediction interval on the value of the BEP. Data regarding the timing of pump clearance adjustments was not available. The unusual spike in the data for Battery A in October is believed to be associated with the motor current sensor since that anomaly appeared only in only one of the four plots which contained that measurement.

Energy losses

Pumps may well be the most common type of industrial machinery after motors. It has been estimated that pumps use about 25% of the world’s total motor driven electricity[6]. This equates to 6.5% of global electricity production. Inefficient operation of pumps leads to losses in energy which may have been prevented. There are at least three distinct reasons to monitor energy usage in mining operations.

1. *The cost of energy*. Monitoring energy use and maximizing efficiency of energy use may bring about substantial cost savings.

2. *Power grid limitations*. In remote areas the power grid may be limited in its capacity to deliver power. In such cases the mill production may be limited by power limits. Optimizing energy use may thus lead to increased production.
3. *Peak shaving*. During rush hour times it is not uncommon that utilities charge a premium price for any power usage above a contract maximum. This has led to 'peak shaving' where machinery is temporarily shut down in order to stay under contractual maxima. Knowing which devices are most energy inefficient will help decide which pumps to shut down in order to minimize the impact on production.

The chart below shows the cumulative energy lost because of the Battery A and B pumps running below a certain attainable maximum (chosen here to equal 1). In an ideal case, where maintenance is able to restore efficiency to maximum every day or every week the cumulative energy lost would have been insignificant. Note that the white stripes in the plots correspond to periods where no data on pump performance was available.

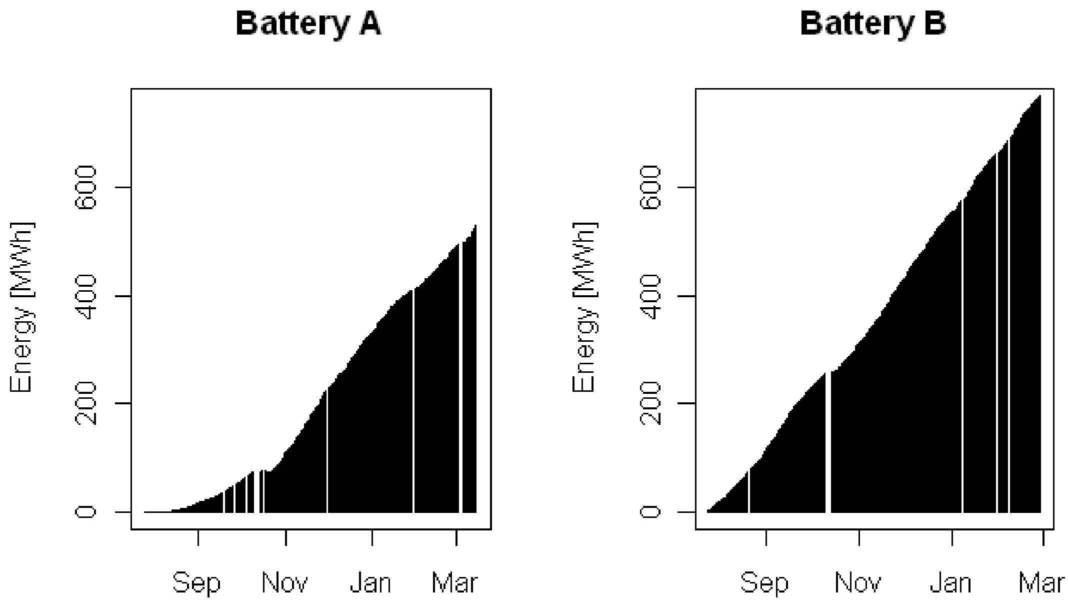

Figure 5. Energy losses.

The energy lost because of less than optimal pump efficiency was calculated by using the calculated and the daily mean power draw by the pump's motor. Accumulating the lost energy by day then results in substantial losses of energy as high as 100 MWh per month.

CONCLUSIONS

Long periods of pump historical data are sufficiently consistent to use in a statistical analysis geared towards inferring pump performance changes. The data can be filtered reliably to remove outliers. Where high pair wise correlations are expected they indeed occur such as between motor current and flow rate.
It is possible to estimate the generated pressure differential over the pump by back calculation from a downstream measured pressure (battery pressure) and an upstream level measured in the tank from which the pump draws fluid (sump tank).
Having a direct measurement of pump differential pressure will increase the reliability and accuracy. Presently losses in pressure due to friction are estimated, whilst pressure losses due to bends and pipe diameter changes are neglected.

The model equations for the head and efficiency do in most cases describe the pump performance adequately. When the range of flow rates covered within an analysis period is too small it may be necessary to change the curve fitting procedure.

The non-dimensional coefficients remove the effect of varying pump speed thereby allowing the analysis to proceed regardless of the pump speed. Non-overlaying data can thus only be due to measurement data that is no longer in line with past data, possibly because of gauge repair or recalibration.

In rare cases the non-dimensional analysis shows non-overlaying features in the efficiency curve. Since the non overlay occurs only in the efficiency curve the suspected gauge is the current clamp used to monitor motor current as this is the unique measurement in the efficiency.

The final result for Battery A shows a very clear downward trend in pump efficiency from a maintenance period to the next. From thereon the efficiency varies but no clear trend with time is discernible. Mid second period however the efficiency does increase by a significant amount. This could be due to pump clearance re-adjustment. Thereafter the efficiency trends down again.

The final result for Battery B does not show a downward trend in efficiency at all. A slight increase in efficiency is observed though right after the maintenance period.

Based upon the efficiency and average daily power draw of the pump's motors the energy lost by less than optimum pump performance can be calculated. Tracking the lost energy may help saving cost and may help maintaining or increasing throughput.

To improve interpretation and usefulness of this analysis procedure for pump monitoring, more detailed maintenance data is required, such as the timing and extent of clearance adjustments, and status of associated instruments.

The analysis techniques applied here rely on reliable plant instrumentation whose performance does not change with time. To date, the flow measurement has been most subject to degradation with time due to the use of invasive meters subject to abrasive slurry flows. The use of a sonar-based non-invasive flow meter significantly improves the weakest link in the instrumentation chain, and appears to make this type of long term performance more feasible.

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