Abstract

Pumps wear at differing rates, varying with design, duty and liquid pumped. Overhaul at regular intervals can be costly and lead to unnecessary work and expense. Condition monitoring has been used for many years to detect and trend wear so that overhaul can be done when it is needed, rather than at an often arbitrary time interval. But, how far do you go? When is the optimum time to overhaul a pump when degradation has been detected? Ray explains the condition monitoring methods available, and the optimisation method he developed that has been featured in conferences, books and technical magazines around the world, and led to writing his award-winning book *Predictive maintenance of pumps using condition monitoring*. The method will be explained with an example, and a worksheet will be available, plus a spreadsheet application.

Introduction

Pumps are arguably the most common machine in power and process industry, and major consumers of energy, yet relatively little information is available on the application of predictive maintenance/condition monitoring to them. When deterioration in performance of a centrifugal pump causes a drop in plant production, overhaul is readily justified, as its cost is usually small in proportion. When the effect of deterioration is only to increase power consumption, the time to overhaul for minimum cost can be calculated from test results. Some basic condition monitoring tests for pumps are described, as is how to use these condition monitoring methods to estimate the increased power consumption caused by pump wear. This Note is based on a paper presented at conferences world-wide and also published in several technical journals.

The extent and effects of internal wear in centrifugal pumps vary with the nature of the liquid pumped, the pump type and its operating duty. Some pumps last for years, others for only months.

Overhauling of pumps on a fixed time or breakdown basis is rarely the most cost-effective policy. Use of condition monitoring ensures that pump overhauls to restore performance are performed when they are really necessary. There may of course be other factors to consider, such as field or OEM experience. However, despite the many excellent pump textbooks, there was little information available on how to apply condition-based maintenance to pumps until recently (ANSI/PI, 2000 and Beebe, 2004).

Monitoring methods should be chosen where justified that will detect each of the degradation modes which are experienced or expected:
Vibration monitoring and analysis (probably the most widely applied method of condition monitoring for rotating machines in general, and suited to detect such faults as unbalance, misalignment, looseness).

Sampling and analysis of lubricants for deterioration and wear debris (relevant for bearings/lubrication system faults).

Electrical plant tests (relevant for motor condition).

Visual inspection and Non-Destructive Testing (particularly relevant for casing wear).

Performance monitoring and analysis (relevant for pump internal condition).

For critical machines, more than one method of condition monitoring may be justified.

This Note will demonstrate use of performance analysis with some examples of condition monitoring in practice.

The Note assumes an understanding of basic pump performance characteristics and how to measure test data repeatably as for condition monitoring, repeatability is more important than absolute accuracy.

The Head-Flow Method Shows Pump Wear

The most useful condition monitoring method is by Head-Flow measurement, because as well as pump deterioration, it detects any changes in system resistance. The method can be used for all pumps where flow, or a repeatable indicator of it, can be measured.

Throttling the pump to obtain points over the full flow range is not necessary for monitoring. Some points near the normal operating duty point are sufficient to reveal the effects of wear, usually shown by the head-flow curve moving towards the zero flow axis by an amount equal to the internal leakage flow. (See "Test points-worn pump" on Figure 3).

A series of test readings at steady conditions at about 15 second intervals is sufficient, taking the average values to plot. Speed must also be measured for variable speed pumps, and the head-flow data corrected to a standard speed (Beebe, 2004).

Field tests sometimes give results slightly different to the manufacturer's works tests because site conditions for flow and pressure measurement are rarely available as required by the various Standards for pump testing. However, note again that for monitoring it is relative changes we are seeking rather than absolute accuracy.

Non-intrusive ultrasonic flowmeters are applicable in most cases. A permanent flowmeter installed as part of a pump's minimum flow protection or process measurement can be used, provided its long-term condition is considered to be constant, or it can be inspected regularly.

Where a plant has a DCS, then regular interrogation of the plant historian can be used, as shown for the multistage boiler feed pump in Figure 1 below (right). Data points are extracted each 6 months from a run of 10 days at steady load. (Constant speed pump, 4 stage, 5853 r/min, 5744kW, 171kg/s @20.4MPa)
Figure 1: Degradation of pumps shown by Head-Flow testing (left) 230kW, (right) 5744kW

Such performance information can show the extent to which a pump has deteriorated, and pumps can be prioritised for overhaul on the basis of their relative wear. But, is the overhaul of the worst pump justified economically?

The Shut-Off Head Method for PdM of Pumps

Measuring the Head at zero flow is a simple test (Beebe, 2004). It is only possible where it can be tolerated, which is not so for high energy pumps nor for pumps of high specific speed where the power at shutoff is greater than that at duty point. Some pumps have exploded when left running at zero flow!

With the discharge valve closed fully for no longer than 30 seconds or so, suction and discharge pressures are read when steady. The liquid temperature is also needed to find the density, which is used to convert the pressure readings into head values.

Wear of vane outer diameters will show readily, as the head-flow curve of a worn pump moves towards the zero flow axis. To show sealing ring wear, the pump head/flow curve needs to be relatively steep. (Note that if the pump has a rising curve, internal leakage will initially give an increase in shutoff head).

Figure 1 shows the trend in degradation of two pumps over some years. Wear amplitude is expressed at duty point flow as the percentage reduction in Total Head compared with the new datum condition. This is usually derived from Head-Flow tests near duty point, but can also be obtained using the shut-off head test where this is allowable.
The Thermodynamic Method for PdM Pumps

Another method of pump monitoring is to measure the temperature rise of the liquid through the pump. This reflects the inefficiency of the pump. As the differential temperature is very small, great care is required to measure it. Any effects of recirculation at pump inlet and outlet must be eliminated, and tests are not possible at very low flows or zero flow. The efficiency can be calculated from the measured data of inlet temperature, differential temperature and head. Comparisons if it changes with time can be made on plots of Efficiency vs Head. For high head pumps, an allowance must be made for the isentropic temperature rise which occurs as a result of pressure increase (Beebe, 2004).

Commercially available devices are widely used, especially in the water industry (Robertson, 2007). Tappings at suction and discharge are required to be two diameters away from pump flanges, for the installation of pressure/temperature probes. Tong-type detectors are placed to measure motor power. Pump efficiency is then found from the precise measurement of the head and temperature rise through the pump. From assessment of motor losses, the power absorbed by the pump is computed. From all this data, the pump flow can be found.

For condition monitoring, tests at around normal operating point are usually sufficient. The thermodynamic method would be more attractive economically if no special tapping points were required. Research at Monash University on high head pumps using special semi-conductor temperature probes on the outside surface of the piping, covered with insulation, gave usable results, provided the pump is allowed to run at steady operation conditions for 30 minutes in order for the piping temperature to stabilise (Beebe, 2002).

% Efficiency for pumps on water at up to 54°C is given by this empirical formula, which includes a correction for the isentropic temperature rise (Total Head is in kPa, temperatures in °C) (Whillier, 1972)

\[
\text{Efficiency} = 100 \left(1 - 0.003(\text{Inlet temp} - 2) + \frac{\text{Temp rise}}{\text{Total Head}} \right)
\]

Measurement of Balance Flow for PdM of Pumps

Multi-stage pumps with the impellers facing in the one direction usually have a balance disc or drum arranged such that final stage discharge pressure counteracts the axial thrust on the shaft line. Another method for condition monitoring is to measure the leakoff from the balance device [6]. The basis is that if there is increased wear in the annular space to the balance device which is evident from increased leakoff flow, then the interstage clearances are also worn. As the leakoff line is quite small compared to the pump main flow piping, a permanent flowmeter is relatively inexpensive.

For some years, overhauls have been scheduled on this basis on some boiler feed pumps. Flows are read manually, and trends plotted using a database program (Figure 2). Note that here the balance flow of 15 L/s corresponds to about 10% of the duty flow, and about 250kW of extra power. When added to the likely internal recirculation, this would mean that an even larger proportion of the power absorbed being wasted. These pumps are variable speed and other tests show that the measured flows must be corrected in direct proportion to the speed.
On a set of pumps of another design elsewhere, both head-flow and balance flow were measured for some years, but no correlation was found between the two.

On yet another pump type, of 11 stages, the head-flow performance was tested as well below the datum curve. As the pump was dismantled, measurements showed that the interstage clearances were not worn. A condition monitoring credibility crisis was averted when the balance seat area was reached and found to be severely eroded. Balance flow had obviously been very high. For the best monitoring, it is therefore considered that both head-flow and balance flow should be measured, particularly if the balance area can be separately dismantled in the field.

![Figure 2](image.png)

**Figure 2** Condition monitoring of a high energy multi-stage pump by measurement of balance device leakoff flow. (Note: flows are corrected to a standard pump speed)

### How to Calculate the Optimum Time for Overhaul

The most economic time to restore lost performance by overhaul will vary with the circumstances.

If the deterioration is constant over time, then a cash flow analysis can be done to ensure that the investment in overhaul will give the required rate of return. This is the same process as used in deciding on any investment in plant improvement.

If the deterioration rate is increasing with time, then the optimum time for overhaul will be when the accumulated cost of the increased electricity consumption equals the cost of the overhaul.

The method is now described for some of the situations which occur.

**Pump deterioration results in a reduction in plant production**

Where the cost of overhaul is insignificant in proportion to the cost of lost production, prompt overhaul is usually simply justified at a convenient “window”.

**Pump which runs intermittently to meet a demand**

In a pumping installation such as topping up a water supply tank or pumping out, deterioration will result in the pump taking more time to do its duty. The
extra service time required therefore results in increased power consumption which can be related to the cost of overhaul.

**Pump deterioration does not affect plant production, at least initially: constant speed, throttle valve controlled pump**

The internal wear does not cause any loss in production from the plant, as the control valve opens more fully to ensure that pump output is maintained. Eventually, as wear progresses, pump output may be insufficient to avoid loss of production, or the power taken will exceed the motor rating.

Figure 3 shows the Head-Power-Flow site test characteristics of such a pump. Its output is controlled using a throttle control valve. The duty flow is 800 m³/h, and the duty point in the new condition is A. The power absorbed by the pump is read off the Power-Flow curve as 2150kW: B. The power-flow curve should ideally be found on site, but the works tests information may have to suffice.

After some service, the "Test points - worn pump" plotted indicate that internal wear has occurred. When worn to this extent, the operating point moves to C, as the system resistance curve lowers when the throttle valve is opened further.

![Figure 3](image)

**Figure 3**  Head-flow-power characteristics of new pump, and head-flow points from worn pump.

The increased power required in the worn condition can be estimated by extending from the Head-Flow curve at constant head from the operating point to D, and then dropping to intersect the Power-Flow curve for new condition at constant flow: E. Follow the arrowed line in Figure 3. This assumes that the original curve still represents the flow through the impellers, of which less is leaving the pump to the system due to internal wear. (If the pump was motor-driven, the actual power may be able to be measured on test at extra expense).

In our example, the power required for this duty in the worn condition is shown in Figure 3 by the projection from the duty flow of 800 3/h to the test curve to
find 640m head, then across to the "Site test - new pump" curve, then down to
the power curve, to find 2300 kW.
The extra electricity consumption is therefore 2300 - 2150 = 150kW ÷ motor
efficiency (here it is 90%), to obtain 167kW.
If the sealing clearances are known, by previous experience of correlation with
measured performance, or if the pump is opened up already, the extra power
consumed likely to be saved by overhaul can be estimated (Stepanoff, 1957, and
HI).

**Finding the optimum time for overhaul from Head-Flow data**

For this example, the test points were obtained following 24 months of service
since the pump was known to be in new condition; an overhaul would cost $50
000; electricity costs 10c/kWh; and the pump is in service for 27% of the time
on average.

Our test shows that the rate of increasing cost/month has reached 167 × 0.10 ×
0.27 × 720 = $3240/month (taking an average month as 720h).

As the time now is 24 months, $3240 ÷ 24 gives the average cost rate of
deterioration as $135/month/month.

The optimum time for overhaul can be calculated (Haynes and Fitzgerald, 1986)

\[ T = \sqrt{\frac{2O}{C}} \]

where:  
O = cost of overhaul 
C = cost rate of deterioration

...... giving here  \( T = 27.2 \) months, but it is better to calculate and plot the average total cost/month values
for a range of times. Seen clearly will be the cost impact of doing the repairs at
some other time, such as at a scheduled plant shutdown.

**How to calculate the total average cost per month, month by month**

For example, take the time as 22 months:

<table>
<thead>
<tr>
<th>The average cost of overhaul is now:</th>
<th>$50 000 ÷ 22</th>
<th>$2273/month</th>
</tr>
</thead>
<tbody>
<tr>
<td>The average cost of extra energy is now:</td>
<td>$135 × \frac{1}{2} × 22</td>
<td>$1485/month</td>
</tr>
<tr>
<td>The total average cost/month is now:</td>
<td>the sum of these two figures =</td>
<td>$3578/month.</td>
</tr>
</tbody>
</table>
Repeat this calculation for several months, perhaps using a spreadsheet, and look for the minimum total cost, which is at 27.2 months. If plotted as cost/month against time, the resulting curves will show the cost per month of overhaul dropping with time, with the cost of lost energy increasing with time.

(The time value of money could also be taken into account if required). Usually the total cost curve is fairly flat for ± 20% or so. The special spreadsheet application shown in Figure 4 (adjacent) is free from the author.

If the overhaul was delayed until, say, 30 months, then the accumulated cost of lost energy would have reached $135 \times \frac{1}{2} \times 30^2 = \$60 \,750$. At 27.2 months, the cost is $135 \times \frac{1}{2} \times 27.2^2 = \$499\,39$. The cost of delaying overhaul is thus the difference, $\$10\,811$.

Note that this calculation is only correct if the wear progresses at a uniformly increasing rate with time, but as Figure 1 shows, this is not unusual. Information may not be available to make any other assumption, but decision makers have to start somewhere! Other formulae apply for rates of change which are not linear (Haynes & Fitzgerald, 1986).

Note:

Some relatively small pumps may never justify overhaul on savings in energy use alone, but may be justified on reduced plant production rate.

The method does not apply to pumps of high specific speed that show little change, or reduction, in power with increased flow.

If a pump varies in its duty, then the energy usage would be corrected in proportion.

**Pump deterioration does not affect production, at least initially: variable speed controlled pump**

For a pump where the speed is varied to meet its desired duty, the effect of wear on power required is much more dramatic than for the case of a constant speed throttle controlled pump. This is because the power usage increases in proportion to the speed ratio cubed.

Unless the pump output is limited by the pump reaching its maximum speed, or by its driver reaching its highest allowable power output, then no production will be lost. However, power consumed will increase more dramatically for a given wear state than for a constant speed pump.
To estimate the power required in the worn state, the Head-Flow curve must be drawn for the current higher speed in the new condition. Select a Head-Flow point on the original new condition curve, and correct it to the higher speed: multiply the Flow by the speed ratio, multiply the Head by the \((\text{speed ratio})^2\). Repeat this for some other points at flows above duty flow to draw the new condition Head-Flow curve.

Follow the same method and calculations as before to find the time for overhaul for minimum total cost. The operating point is projected from the worn curve to the new curve at the same speed as the worn curve. Figure 4 shows the performance of a variable speed pump. When new, operation at 1490 r/min meets the desired duty flow, at operating point A, requiring 325 kW power: point B.

After some time in service, internal leakage has increased such that the pump must run at 1660 r/min to meet the required duty - still point A.

To estimate the power required now, the Head-Flow curve must be drawn for the higher speed in the new condition. Several Head-Flow points are selected and corrected to the higher speed: multiply each Flow by the speed ratio, and multiply each matching Head by the speed ratio squared. This will result in the Head-Flow curve @ 1660 r/min in the new condition.

Project across from the Head at the duty flow - point A –to meet the head-flow curve @ 1660 r/min (new condition). (Line C in Figure 5). Projection downwards at constant flow leads to the increased power required at 425 kW. The extra power is 31% more! (This pump is driven by a steam turbine, so power consumption cannot be measured).

The same calculations as before are followed to find the time for overhaul for minimum total cost.

![Figure 5](image-url)  
**Figure 5** Head-flow-power characteristics of new variable-speed pump, and head-flow points from worn pump.
**Optimisation Using Shut-Off Head Test Results**

The shut-off head test information can also be used to estimate power used in the worn state, and do the optimisation calculations as explained in the above section.

Head-Power-Flow characteristics in the "new" state are needed as before, and the operating point must be known. Note the power required at operating point as before.

Make an overlay trace of the Head-Flow curve in the new condition. Place it over the "new" curve and move to the left horizontally until the curve cuts the Head axis at the value of shut-off head obtained on the test. The trace is now in the position of the "worn" Head-Flow curve which is being experienced. Exactly the same process can be followed as explained above.

**Conclusion**

A method of determining the optimum time for overhaul of a pump based on energy savings has been given. It is hoped that this valuable tool will help asset managers and engineers in their role of managing assets to provide capacity for production, and to improve energy efficiency and minimise greenhouse impact. This optimisation approach can also be applied to any item of plant where deterioration results in loss of efficiency and energy consumption can be measured or estimated.
References

ANSI/HI 9.6.5-2000  American National Standard for Centrifugal and Vertical Pumps for Condition Monitoring


Stepanoff, A J:  Centrifugal And Axial Flow Pumps Wiley (1957), and Figure 1-77A of the PDF figures on www.pumps.org


Method presented and published -
Conferences:

First Australian Congress on Applied Mechanics, 1996 Melbourne


Prognosis of residual life of machinery and structures:  Proceedings of 52nd Conference of Society for Machinery Failure Prevention Technology  Virginia Beach, March 1998


Proceedings Maintenance and Reliability Conference (MARCON99), Tennessee 1999

11th Process and Power Plant Reliability Conference, Houston, Nov 2002

International Pump User’s Conference, SALMechE, Johannesburg, June 2005

3rd Rotating machinery conference, Marcusevans, Melbourne February 2008

Euromaintenance2008, Brussels, April 2008

Publications

Maintenance Journal (1992) (Australia)

PUMPING TECHNOLOGIES, April/June 2000 (Australia)

INTERNATIONAL WATER & IRRIGATION Vol 21 (2) 2001 (Israel)

HYDROCARBON PROCESSING, April 2003 (USA)
Beebe, RS (2004) Predicting maintenance of pumps using condition monitoring
Elsevier (UK)

WORLD PUMPS May 2004 (UK)

Fairmont /CRC Press (USA)

CHEManager 9/2008 (Europe)