ESTIMATE THE INCREASED

POWER CONSUMPTION CAUSED BY PUMP WEAR

Ray Beebe, School of Applied Sciences and Engineering Monash University

Centrifugal 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.

photo: Mervers

When deterioration in performance of a 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. The paper shows how to use these condition monitoring methods to estimate the increased power consumption caused by pump wear.

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 general pump textbooks, there is little information available on how to apply condition-based maintenance to pumps.

Monitoring methods should be chosen which will detect each of the degradation modes which are expected:

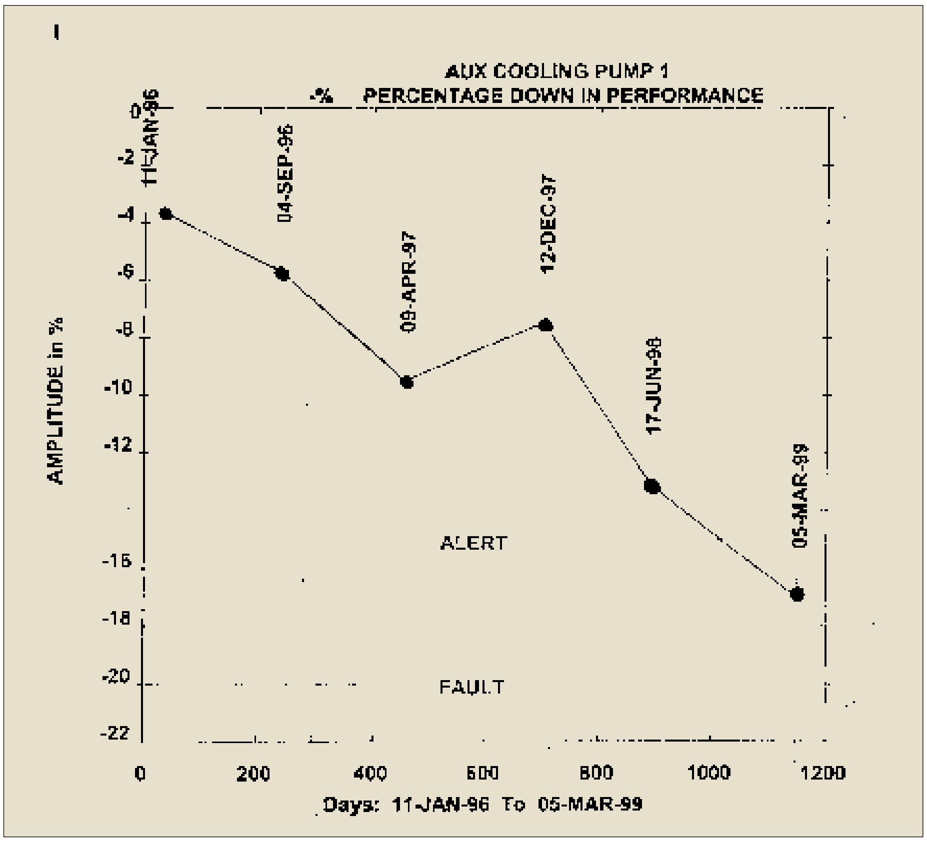


Figure 1: degradation of a 230kW pump shown by Head-Flow testing.

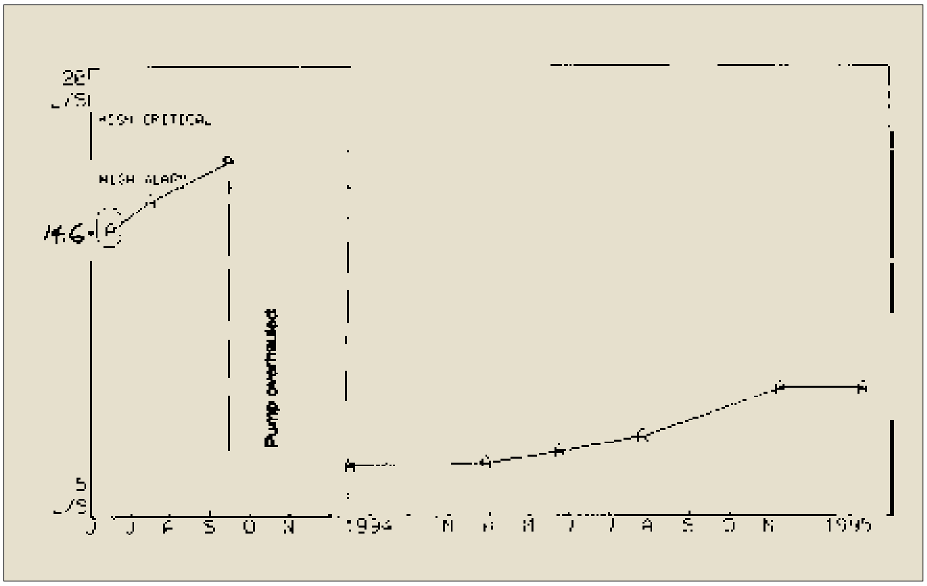


Figure 2: condition monitoring of a high energy multi-stage pump by measurement of balance device leakoff ow. (Note: ows are corrected to a standard pump speed).

- Vibration monitoring and analysis, probably the most widely applied method of condition monitoring for rotating machines in general, and suited to detect such things as unbalance (maybe due to impeller failure), misalignment, looseness, soft foot, resonance.
- 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, which is most relevant for pump internal condition, and because there is little published, is the focus of this paper.

For critical machines, more than one method of condition monitoring may be justified. This paper will demonstrate use of performance analysis with some examples of condition monitoring in practice. The paper assumes an understanding of basic pump performance characteristics and how to measure test data repeatably.

The head flow method

The most useful condition monitoring method is by Head-Flow measurement, because as well as pump deterioration, it detects any changes in system resistance, provided any throttle control valve is in the same opening position. 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. 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 using the usual affinity laws and as given later in this paper.

Field tests sometimes give results that differ from 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, for monitoring it is relative changes we are seeking rather than absolute accuracy.

Unless a plant’s permanent instrumentation is verified as being in calibration, test instrumentation will be needed.

Pressures can be read with test-grade bourdon tube gauges, or electronic types, one at least (CRYSTAL™) of which has a greater accuracy than most deadweight testers. Non-intrusive ultrasonic and similar clamp-on 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.

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

Figure 1 shows the trend in degradation of a 230kW water pump over three years to be closely linear. Wear amplitude is expressed here as the percentage reduction in Total Head at a selected datum flow compared with the new datum condition. (The non-consistent 12 Dec 97 point should have led to a re-test).

This is usually derived from Head-Flow tests near duty point, but can also be obtained using the shut-off head test (see next section) where this is allowable.

The shut-off head method

Measuring the Head at zero flow is a simple test. It is only possible where it can be tolerated, which is not so for high energy pumps: some pumps have exploded when left running at zero flow! It is not allowable for pumps of high specific speed where the power at shutoff is greater than that at duty point and may exceed the motor rating.

With the outlet valve closed fully for no longer than 30 seconds or so, inlet and outlet pressures are read once steady. The liquid temperature prior to any significant heating up 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).

The thermodynamic method

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.

Commercially available devices are widely used, especially in the water industry. Tappings for the installation of pressure/temperature probes are required at inlet and outlet to be two diameters away from pump flanges. 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 was conducted by Monash University on high head pumps using special semiconductor temperature probes on the outside surface of the piping, covered with insulation. Usable results were obtained, provided the pump is allowed to run at steady operation conditions for 30 minutes in order for the piping temperature to stabilise

% 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). (1psi = 6.895kPa):

Measurement of balance flow

Multi-stage pumps with the impellers facing in the one direction usually have a balance disk or drum arranged such that final stage outlet pressure counteracts the axial thrust on the shaft line. This design gives another method for condition monitoring, by measuring the leakoff flow from the balance device. 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 the four electric-driven boiler feed pumps on a 500MW unit. These pumps are variable speed and other tests were run to find that the measured flows must be corrected in direct proportion to the speed. The flows are read manually from an Annubar ™ installed in each leakoff line and panel instruments. Figure 2 shows trends plotted using a database program also used for vibration trending. Note that here a "worn" balance flow of 15 L/s corresponds to about 10% of the duty flow, and absorbs about 250kW of extra power. When added to the effect of the likely matching internal recirculation, this would mean that an even larger proportion of the power absorbed being wasted. In the power plant, this is not entirely lost, as the heat is recovered in the cycle, but it is created at an overall efficiency of about 38%, and still represents maybe 500kW per pump that is not output from the plant to be sold. Unfortunately, the author learned of this case after the overhaul, and found that new wearing rings were installed without the internal clearances being measured. Co-relation of as-found condition with the predicted condition is important in gaining experience and credibility, and the opportunity here was missed. On a set of pumps of another design elsewhere, both head-flow and balance flow were measured for some years, but no correlation in predicting condition was found between the two. On yet another pump type, of 11 stages, condition monitoring tests revealed that the head-flow performance was well below the datum curve. But, as the pump was dismantled, measurements showed that the interstage clearances were not worn! Later, the balance seat area was reached and severe erosion of

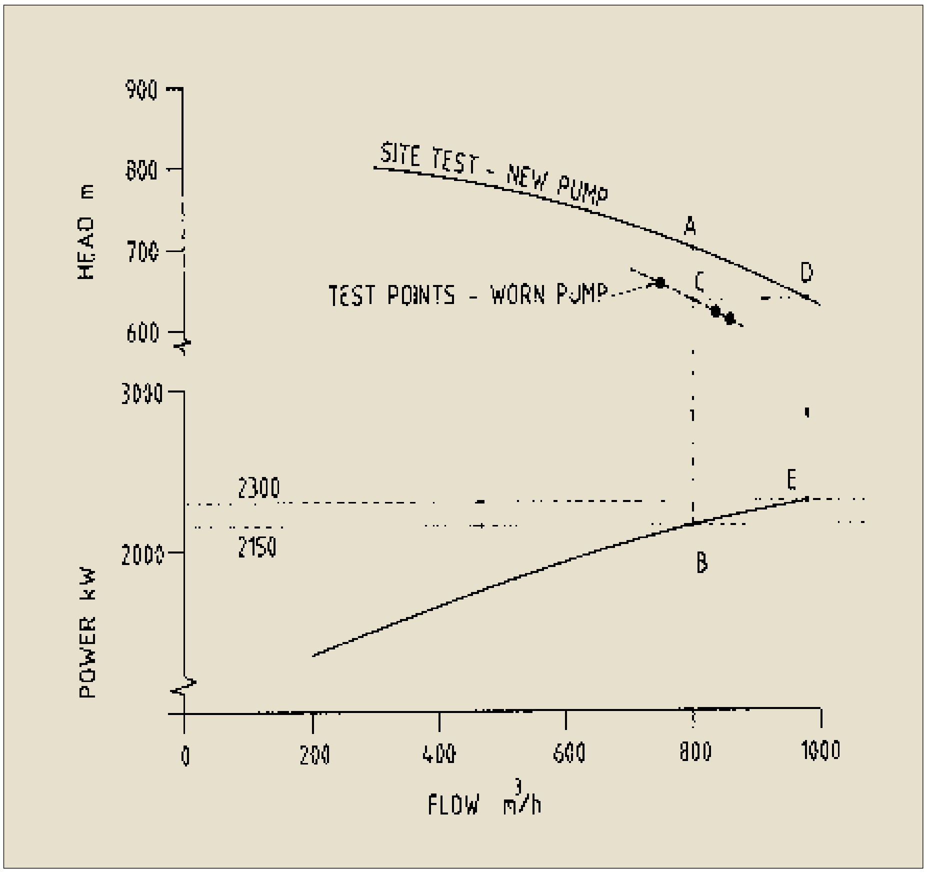


Figure 3: head- ow-power characteristics of new pump, and head- ow points from worn pump.

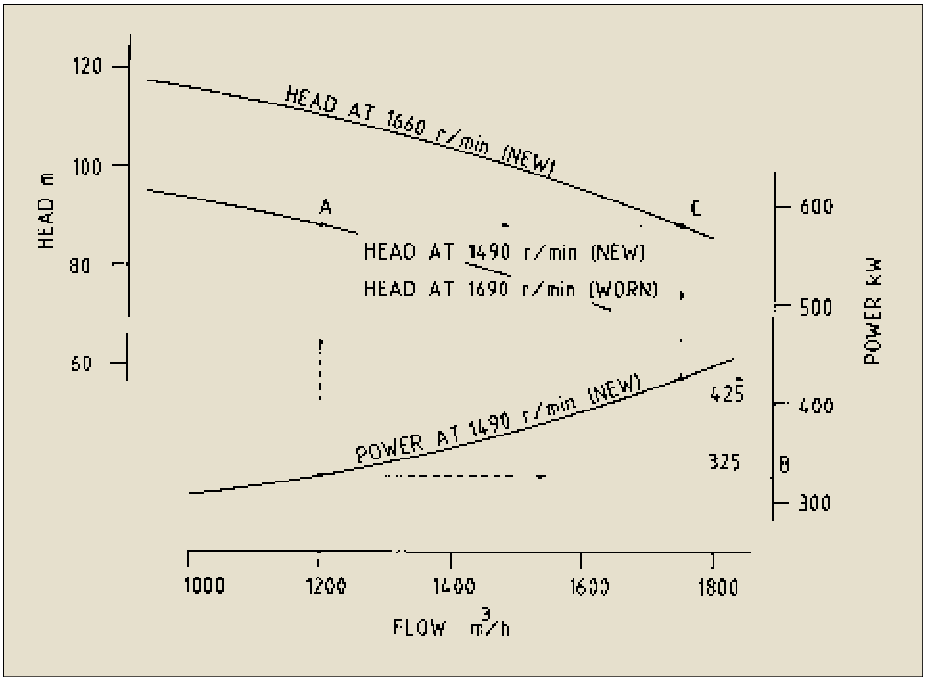


Figure 4: head- ow-power characteristics of new variable-speed pump, and head- ow curve from worn pump.

the pump casing, showing that water had passed between the seat and the casing. A condition monitoring credibility crisis was averted! 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.

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 close to 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, when flow is relatively constant over a period, deterioration in the pump will result in it 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

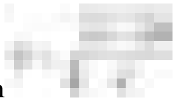
Here, the internal wear does not cause any loss in production from the plant, as the control valve opens more fully to ensure that pump output is maintained at the required rate. Eventually, as wear progresses, pump output may be insufficient to avoid loss of production, or the power taken will exceed 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 m^3/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. 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 intersecting the Power-Flow curve for new condition at constant flow: E. Follow the arrowed line in Figure 3. The basic assumption here is 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). The flow difference C – D represents the internal leakage. If the pump was motor-driven, the actual power could of course 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 m3/h to the test curve at 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. Using this method, a number of pumps of varying wear conditions could be prioritised for maintenance, based on their increased power consumption and their relative costs of overhaul, i.e. the cost/benefits. However, a further calculation enables

Finding the optimum time for overhaul

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. Based on experience as reported in Figure 1, this is assumed to be linear with time. The optimum time for overhaul can be calculated from 
where:
O = cost of overhaul
C = cost rate of deterioration
giving here T = 27.2 months.

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:

- The average cost of overhaul is now:
  \$50 000 ÷ 22
  \$2273/month
- The average cost of extra energy is now:
  \$135 × ½ × 22
  \$1485/month
- The total average cost/month is now:
  \$375/month.

Repeat this calculation for several months, perhaps using the spreadsheet mentioned. 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, but such refinement is rarely needed). The minimum total cost, as before, is at 27.2 months. Usually the total cost curve is fairly flat for ± 20% or so. If the overhaul was delayed until, say, 30 months, then the accumulated cost of lost energy would have reached \$135 × ½ × 30² = \$60 750. At 27.2 months, the cost is \$135 × ½ × 27.2² = \$49939. The cost of delaying overhaul is thus the difference, \$10811. 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! Note that 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 even a reduction, in power with increased flow

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 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 (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. The Head at the duty flow - point A – is projected across to meet the head-flow curve @ 1660 r/min (new condition). (Line C in Figure 4). 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 its power consumption cannot be measured). The same calculations as before are followed to find the time for overhaul for minimum total cost.

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

Some methods for condition monitoring of pump have been described, with the Head-Flow test at duty point method recommended. 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 adapted to any item of plant where deterioration results in loss of efficiency and energy consumption can be measured or estimated. <<