Session One:

Condition Monitoring of Pumps – How to Save Three Ways

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Abstract and Introduction

Dismantling of pumps (or any major machinery) on a time-basis is rarely the lowest cost option. But, what is the alternative? Condition monitoring can reveal the extent of internal wear and can be used to help decide the optimum time for overhaul. In this workshop, we will review pump performance basics, outline the several performance monitoring technologies available for condition monitoring of pumps and how to apply them. The content is in three parts, with activities included to explain and for practice.

PART 1: PUMP PERFORMANCE TUTORIAL

1. Review of pump performance basics

The four basic quantities in centrifugal pump performance are **Total Head, Power absorbed (or, just "Power"), Efficiency** and **Flow.** Usually in works tests, Head, Flow and Power are measured and Efficiency is calculated from them. Results are plotted against flow to give curves like Fig 1. (Tolerances of a few percent are allowed given in the Standard). Manufacturing tolerances also mean that nominally identical pumps may vary slightly in performance.

Pump manufacturer catalogues, whether in printed or online form, usually show characteristics for water at 20°C for a range of impeller sizes in a given casing. To minimize clutter, Efficiency is often shown as *isoefficiency* curves or values. See the sample catalogue page later.

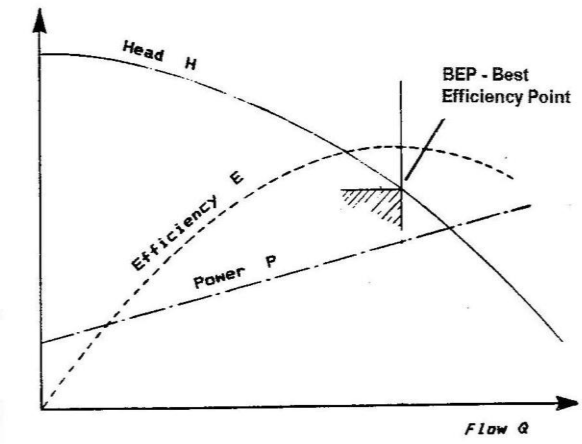


Figure 1: Pump performance basics

The basic performance equation is:

$$E = \frac{Q \rho g H}{P}$$

Where:

E = Efficiency, decimal
Q = Flow, m^3/s
ρ = Density, kg/m^3
g = gravitational constant = 9.81 m/s^2
H = Head, metres of liquid pumped (for water at normal ambient temperature, 1m = 9.8kPa)
P = Power absorbed, W
(*Specific Weight* is sometimes used in place of ρg)

The Head-Flow characteristic in volumetric terms applies for whatever liquid is pumped. The Power-Flow curve will however vary with liquid density. *An exception is viscosity effects*: the H-Q curve droops with increased viscosity, Power increases. If cold water results are to be compared with tests at much higher field temperature, efficiency will improve as the kinematic viscosity is less. Hydraulics Institute standards give the correction process.

Positive displacement pumps are usually much smaller, with the characteristic shown as Flow-Head, with Flow near linear and dropping slightly as Head is increased and internal leakage increases. Systems usually have a relief valve to prevent casing or other damage. (See Q-H-P-N curve on page 12).

A
What is the power required by a pump delivering 37.5L/s @ 22.7m, if its efficiency is 0.815?

B
What is the efficiency of a pump delivering 175 L/s at 64m, absorbing 127.7kW ?

For boiler feed pumps, H-Q characteristics are usually shown in absolute terms. *H* and *P* both vary with density. With Head in kPa, Flow in kg/s, Power in kW, Efficiency as a decimal:

$$E = \frac{QH}{\rho P}$$

C
A boiler feed pump is tested with water at 140°C, giving 146kg/s, 16300kPa total head (suction 1500, discharge 17800kPa), taking 3650kW. Correct this data to a temperature of 150°C for comparison with the datum curves.

Using the densities at the mean pressure of water in the pump:

Density at 140 degrees, 8150kPa (average pressure in pump) = 930.2 kg/m^3

Density at 150 degrees, 8150kPa = 921.2 kg/m^3

Density ratio is 0.9903, so above point becomes 144.6 kg/s, 16144 kPa, 3516 kW.

The characteristics must be corrected for any change in speed. With R as the speed ratio, Flow varies directly as R, Head as R^2, Power as R^3.

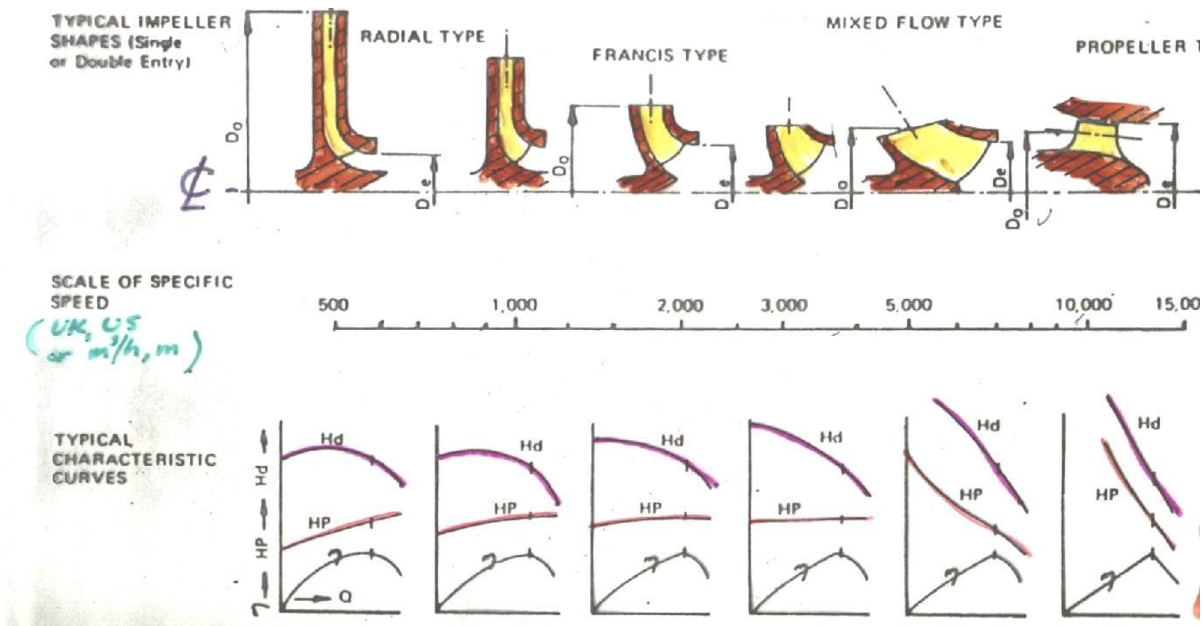
D
A test on a boiler feed pump at 5400 r/min gives 290kg/s at 11600kPa. Correct this point to compare with the datum H-Q curve at 6000 r/min.

2. Specific Speed

Calculated from the data at Best Efficiency Point, Specific Speed is a *family type number* that is useful to indicate the pump characteristics when only the nameplate data is known. Other units can be used, but using the units below gives a number close to that obtained using US units (US GPM, feet).

$$N_s = \frac{N\sqrt{Q}}{H^{0.75}}$$

Where N = rotation speed, r/min
Q = flow per first stage impeller eye, m^3/h
H = head per impeller stage, m



The maximum attainable efficiency relates
to both size and Specific Speed.

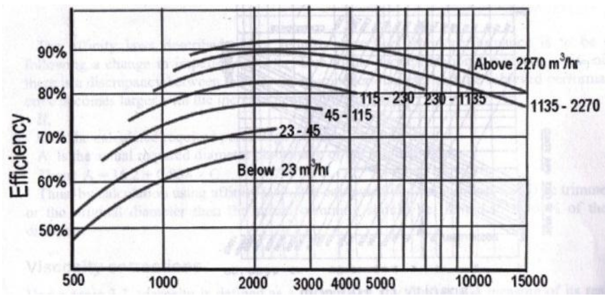


Figure 2: Specific Speed and its uses

E All you know about a single stage pump is its nameplate data: 780 m^3/h, 7m head, 980 r/min. What type of pump is it?	**F** Calculate the Specific Speed of the single-stage pump with its catalogue curves shown on page 15.

3. Pumps in parallel and series

Pumps can be arranged in *parallel*, where increased flow results with both running (or one can be kept on standby), or in *series* to obtain higher pressure (as with multi-stage pumps).

The combined characteristics are obtained:

- in parallel by adding the flow values for selected values of constant head; and
- in series, by adding the heads at constant flows. Figure 3 shows the plots for two identical pumps.

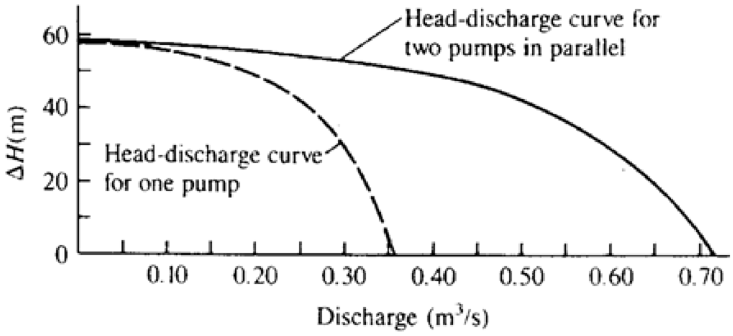
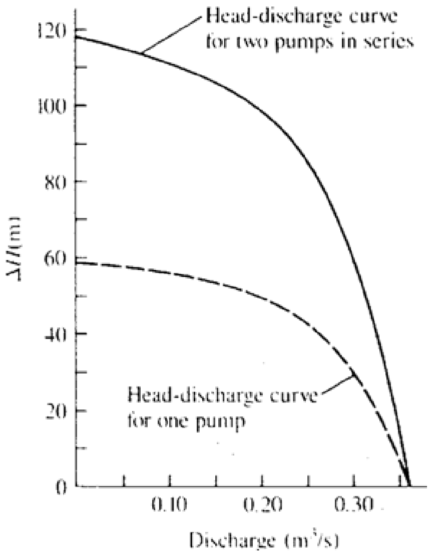


Figure 3: Pumps in parallel and series



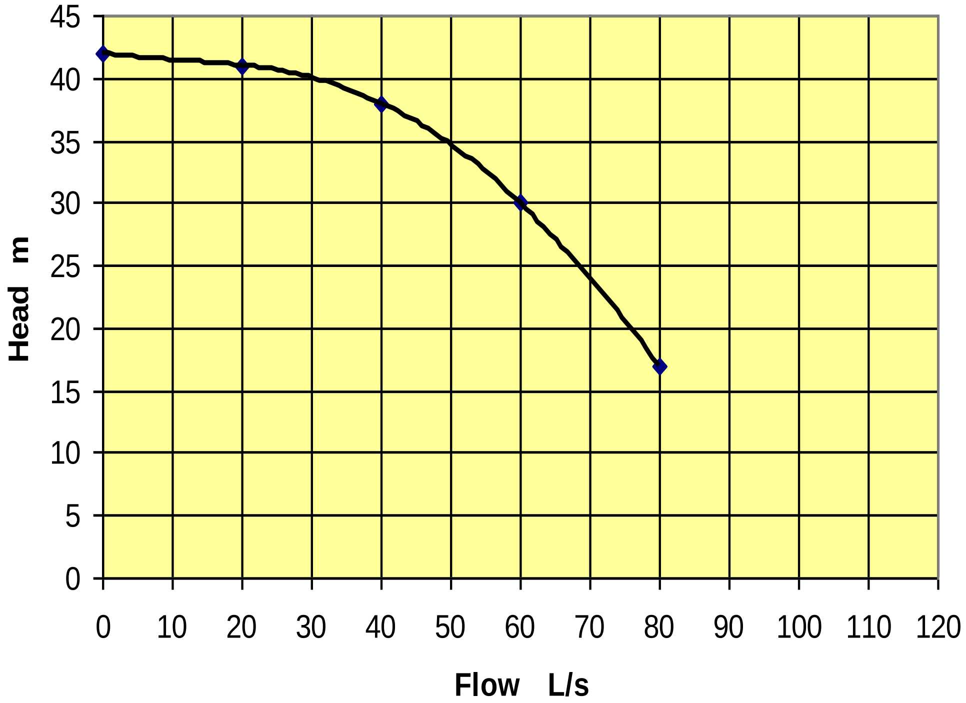
G Plot the H-Q curve for the combined performance of two of these pumps if run in parallel.	

Figure 4: Example pump H-Q curve

4. The System Curve

A pump must be considered along with the system of piping, fittings and plant items it is to serve. A system consists of one or more parts:

- *Static Head*: the difference in free liquid level between suction and discharge (measured on the plant, or from elevation drawings). Note: this can be negative, such as a pump discharging at a lower level than an overhead tank on its suction).
- *Pressure Vessel Head*: (if any) the difference between any pressure acting on the suction and discharge;
- *Friction Head*: proportional to Flow squared.

Figure 5 shows how these are combined to give the system curve.

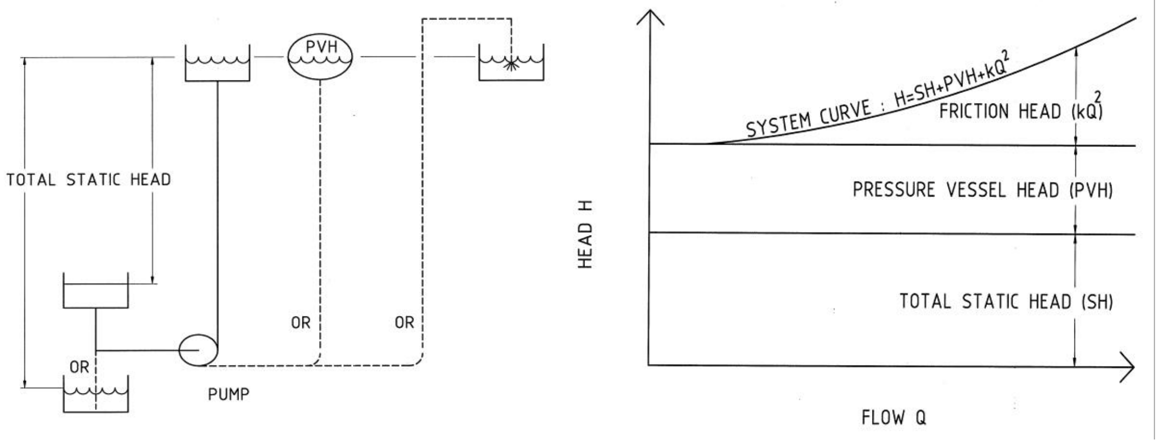


Figure 5: Static Head and the system curve

Note that the static head and/or pressure vessel head may not be constant with flow.

To obtain the combined curve for *systems* in parallel and series, the same rules apply as for pumps. Free software such as *epanet* is available for complex networks.

5. Operating or duty point

Where the system curve and the pump (or combined pumps) curves intersect gives the operating point (Figure 6).

Designers aim to have this within range of the Best Efficiency Point. However, pumps wear and systems can change (erosion, corrosion, buildup in pipes, etc.)

Note that operating two “identical” pumps in parallel does not give twice the flow to a system. The extra flow that does result reduces as friction losses increase. SKETCH a typical steeper system curve on the Figure 7:

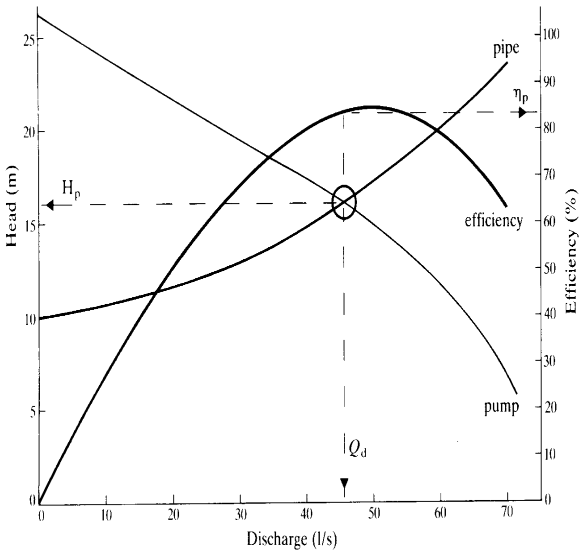


Figure 6: Operating point: at intersection of Pump H-Q curve and system curve

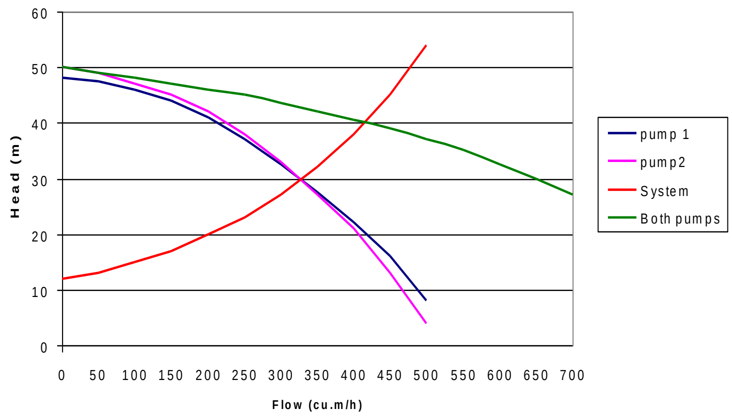
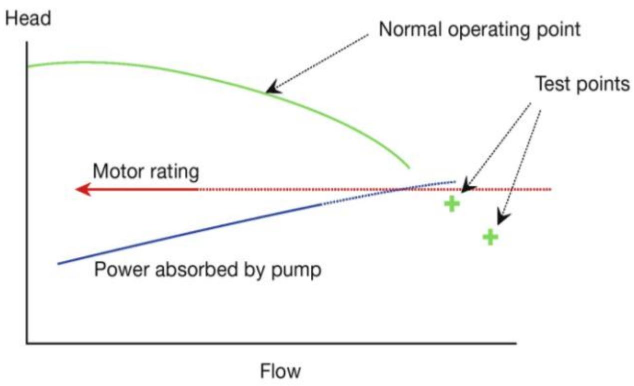


Figure 7: Why operating two identical pumps in parallel does not double the flow to a system

H
A multi-stage pump supplies water through an agitator nozzle in an ash pit. It kept tripping on high motor current, so was removed and overhauled. On return, it still tripped.

A test was run using pressure gauges and flow meter. The test points are shown.

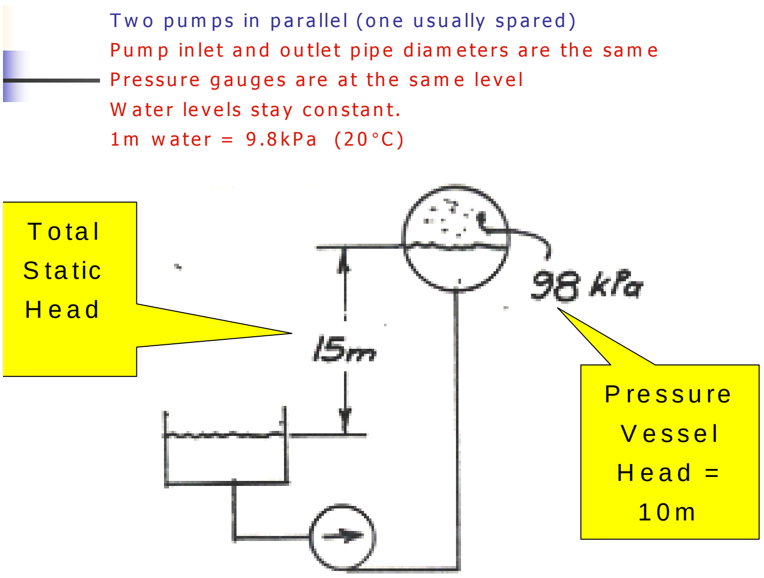
Can you explain what has happened?



I
When new, running unthrottled, suction head is +2m, discharge head 20m. On the diagram for the pump in Figure 4, plot the operating point for one pump in service.

Plot the operating point when both pumps are run in parallel and a flow of 88L/s is measured.

Plot the zero flow point, then draw the system curve through the three points (obtained by test).



6. Cavitation and NPSH

If the pressure of the liquid at the entry impeller drops below its vapour pressure, vapour bubbles form – i.e. localised boiling of the liquid occurs. The vapour bubbles implode and can severely damage metal surfaces. So, *cavitation* must be avoided.

- **NPSH-R**: Net Positive Suction Head *Required*: energy required by a pump at its inlet to prevent **Cavitation**. It is a function of *pump design* and shown on catalogue curves, as NPSH-R vs Flow, or as in the Appendix, noted on the H-Q curve

- **NPSH-A:** Net Positive Suction Head *Available:* function of *system design* on suction side of pump. If this exceeds what the pump requires, with a margin, then cavitation should not occur.

With all items expressed as Head of liquid pumped, NPSH-A is calculated from:

> Atmospheric pressure + Static Lift (or - Static Suction Head, if liquid level is above pump centreline) – Vapour Pressure – Friction losses in suction piping.

(Vapour Pressure is usually small: see table for water on last page).

<table>
<tr>
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J
Calculate the NPSH-A for the system shown at flow of **60L/s**. Flow velocity varies through the system: V = Q/A.

$H_L = K\, v^2/2g$

K values for fittings:

Foot valve: 15

45° bend: 0.22

Reducer: 0.2

Pipe friction loss is 2m/100m

Is this OK for the pump with catalogue curves later?
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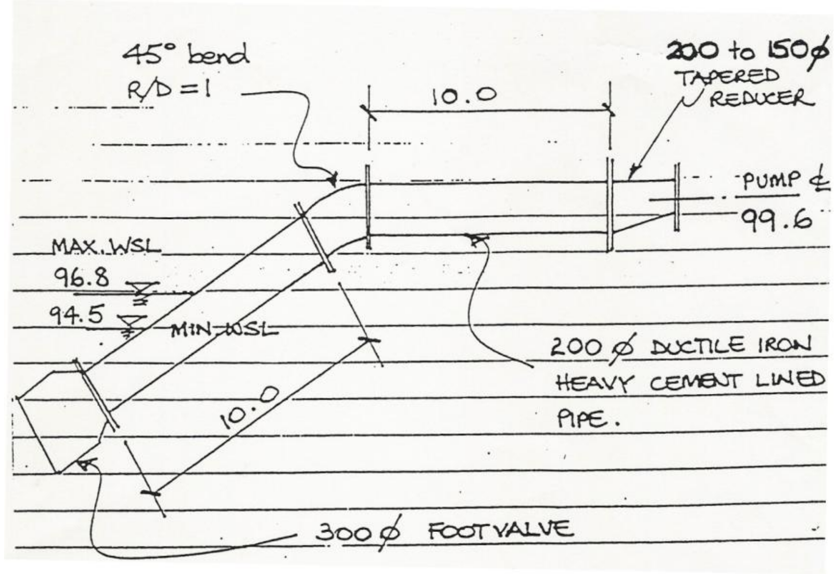

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The *Suction Specific Speed* is also used: the same formula as before applies, but with Q in m^3/s, and NPSH-R in place of H. Hydraulics Institute advise that over 175 the probability of repeat installation failures increases exponentially. Above 180, sustained flow should not be below 85% of BEP.

7. Effect of wear inside a pump

Some liquid from the discharge of an impeller recirculates to its suction through the sealing (wearing) rings. Increased wear moves the Head-Flow curve towards the zero flow axis as shown in Figure 9. Experience shows that the rate of wear is close to linear (Beebe, 2004).

Some of the power taken by the pump is wasted, so efficiency reduces.

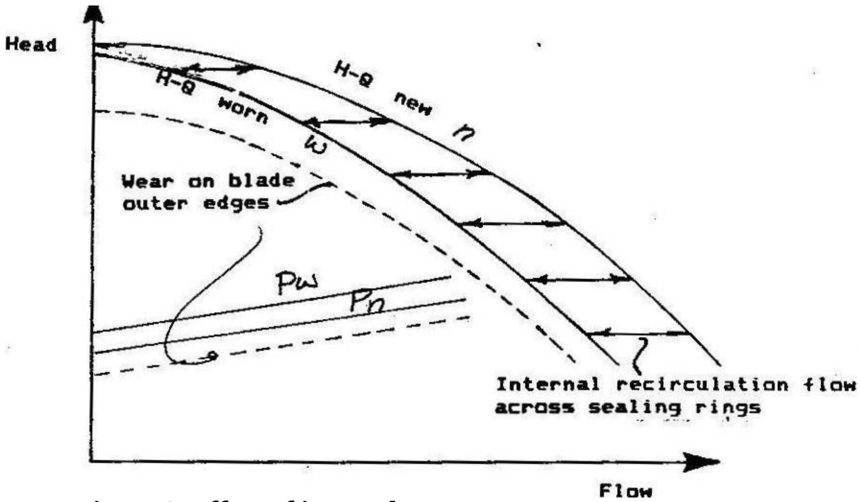


Figure 9: effect of internal wear

Impeller wear on the outer diameter on the vanes results in a smaller impeller.
Casings can also erode, and thickness can be measured from the outside using NDT.

8. Modifying a pump to change performance

To improve performance, impeller vanes can be *underfiled*, thus varying the exit angle. The volute can be chipped to remove metal at the throat. Diameter can be reduced by machining the vanes. A different impeller of the same diameter but wider, or with more vanes, is another possibility. OEMS can advise.

9. Field testing of pumps

Standards apply for tests in the works, and can be used as a guide for field testing. The required conditions are often not available in the field, but for repeatability, expedient methods can be used.

Head is measured with pressure gauges or transducers at pump suction and discharge flanges, where tapping holes are usual. For water at low pressures, a simple manometer made of clear precision plastic tubing open to the atmosphere works well.

Head is referred to the pump centreline. If the suction and discharge tappings and instruments are not at the same level (based on centre of the scale for gauges), allowance must be made for liquid heads in the tapping lines. If the instrument is below the tapping, then it will read too high, and the *static liquid leg* must be subtracted from the reading, and vice versa.

Pump suction pressures are often below atmospheric, and the instrument connecting piping is probably filled with vapour (air which un-dissolves from the liquid) rather than liquid. The static leg of vapour is negligible. If in doubt, use clear plastic piping, or position the instrument at pump centreline. Tapping lines should be bled to ensure they are filled with liquid or vapour, as the case may be. To check transducer sensitivity and contents of the instrument line, lower the transducer a metre and check that the reading changes by this amount.

Use of quick-connect couplings speeds connection of test gauges or transducers where the liquid is not hazardous.

Sensors should be calibrated using a deadweight tester before and after the test. A curve of calibration correction against reading can be drawn to interpolate. Calibration at several points spanning the range expected on test is more useful than over the full range of the instrument. If excessive hysteresis is observed, then the instrument should not be used.

Tappings in the side of the pipe measure the *line* or *static* pressure, and if the pipe bore diameters there differ greatly the *velocity head must be added* to obtain the total head. Often operating pressures are much higher in proportion and velocity head can t be neglected. For condition monitoring, repeatability is essential and provided the total head is always obtained in the same way, velocity head can be neglected.

Velocity Head, in metres, is calculated from: $\dfrac{[Velocity]^2}{2g}$

The Total Head is the difference between the total heads at pump discharge and suction.

Field tests may not give the same values as that measured at works tests where Standard conditions require pressure measurements made 2 pipe diameters away from the pump flanges (to avoid any recirculation effects) and four tappings around the pipe, each with its own isolating valve, and connected by a piezometric ring. On a 22 m^3/h pump where both types of pressure tappings were used, the head was 10% higher and efficiency 5% higher using flange tappings. (Yedidiah, 1996). In critical cases such as when disputing guarantee performance, the special tappings would be needed.

When investigating the matching of a pump to a system, any such effects would affect both pump performance and system equally, and this refinement can be ignored. For condition monitoring, this effect would be constant with time, and can also be ignored.

Speed must be measured for variable speed pumps with a tachometer, and data corrected to a datum speed using the affinity laws. Speed may vary significantly on a large constant speed pump if voltage changes.

Flow (or *Capacity, Discharge*) can be measured repeatably for condition monitoring on most pumps with a portable ultrasonic or other non-intrusive flowmeter, which clamps on the outside of a straight section of suction or discharge pipe. These meters give the average velocity of flow, so any bore deposition must be allowed for in determining the pipe bore diameter.

Where available, the flow elements of permanently installed flowmeters can be used. Care must be taken when connecting in parallel with permanent instruments, as a local pressure drop may trigger pump an alarm or minimum flow protection, if fitted.

In flanged pipes, an orifice plate may be possible, installed at a joint. Tappings are welded at distances from the orifice plate of one pipe diameter upstream, half a diameter downstream (called D and D/2).

Repeatable results may be obtained for condition monitoring even without the full lengths of straight pipe required by flow standards such as ISO5617-1:1997. The units can be volumetric (m^3/h, or L/s) or as mass flow (kg/s). The basic equation for orifice plates and other differential pressure producing devices is:

$$Flow\ (volumetric) = kd^2\sqrt{\frac{Differential\ pressure}{Density}}$$

The grouped constant k includes any conversion constants, etc. This can be simplified if the temperature of the liquid does not vary much from ambient, to:

$$Flow\ (volumetric) = k_2\sqrt{Differential\ pressure}$$

Included in the grouped constants above is the *discharge coefficient.* At low flows, this may no longer be constant, as it increases below a certain Reynolds Number.

If the system includes a tank, timing of the depth (volume) change as it fills or empties can be used. Tanks may have a chamber for level sensors to give a settled level unaffected by surges of the liquid. An alternative, also useful for closed tanks, is a manometric level gauge.

Complex pumps may have separate sealing flows into and/or from their shaft glands that bypass the pump flow measuring point. Usually these are very small in comparison and variations from design may be neglected for condition monitoring. For critical pumps, separate gland sealing flow metering may be justified, particularly if the glands can be repaired without dismantling the complete pump.

Each case must be examined to ensure that any minimum flow leakoff valve leakage or axial thrust balance device flows are properly allowed for in measuring flow *through* the pump.

Withdrawable pitot tubes are another device which can be used on lower pressures. These are installed when required through a gate valve in the side of the pipe and sealed by a suitable gland fitting. Note that the pressure inside the pipe acts to force the tube out, so a restraining system is essential. The Annubar™ multi-point device is one type, with high accuracy.

A single point double-tip tube can also be made and used (Beebe, 1995). If the flow is not expected to change significantly, a traverse across the pipe at the Standard positions can relate the centre velocity to the average. Future tests can then be made with centre measurements only. The flow can be found by multiplying the average velocity by the pipe cross-sectional area, using consistent units.

If required for motor drives, *power* is best measured with a test kWh meter, or the two-wattmeter method, which can however be expensive to arrange. Current and potential transformers must be identified and their characteristics known.

For 3-phase motors, power usage of a motor in Watts = $\sqrt{3} \times Volts \times Amps \times \cos\phi$ For single phase, power is Watts = $Volts \times Amps \times \cos\phi$

To obtain the **Power absorbed by the pump**, the motor efficiency and the efficiency of any gearbox or fluid coupling (from works test data) must be taken into account.

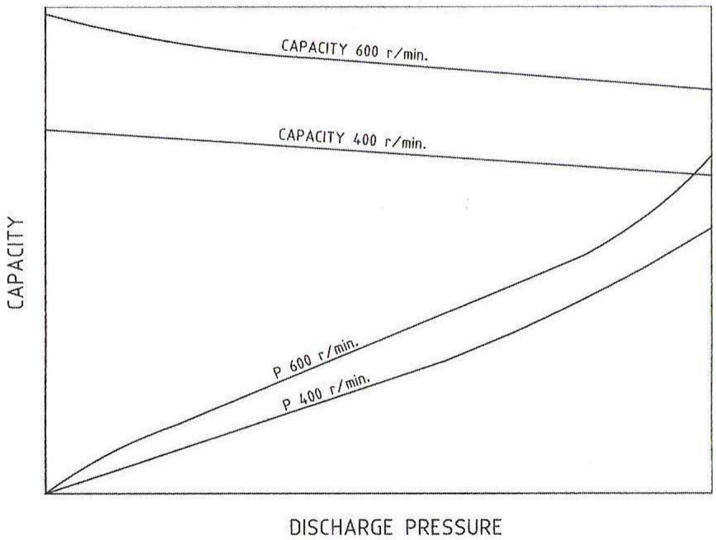
For routine condition monitoring, it is worth seeing if a repeatable indication can be given from a panel ammeter, and using system volts. This is however likely to be crude and of limited use.

K	Return to the plot of Figure 4. After some service, a test on Pump #1 gives flow of 54 L/s @ a total head of 29m. **What has happened?**
L	Later with both pumps in new condition in the scenario of Figure 4, operators report that both pumps must be run to obtain the required flow. Your test gives 60 L/s @ 40m. Plot this point and decide whether a pump needs overhaul.

10. Other factors when investigating poor performance

Consider these as well as the previous issues:

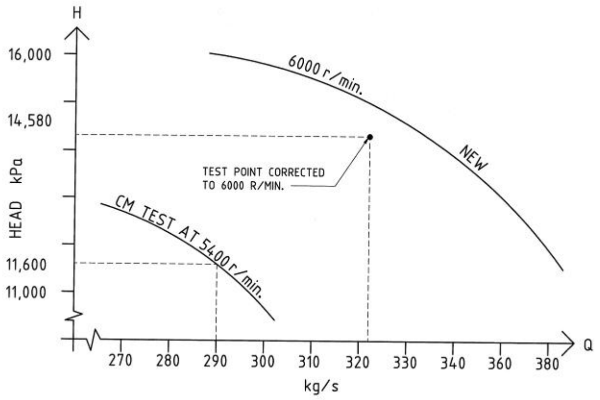
- Pump running backwards (dc motors) – still pumps but very poorly.
- Double entry impeller installed wrong way round – still pumps but poorly (vanes become forward curved).
- Passing isolating valves such that one pump of two in parallel sends water in reverse through the standby one, rotating it in reverse.
- Poorly matched "identical" pumps to a system that deadheads one pump when run in parallel.



Performance characteristics of positive displacement pumps are shown differently to those for centrifugal pumps. The Flow (i.e. Capacity) decreases as discharge pressure increases – internal recirculation or tip leakage increases at the higher pressure.

Figure 9: characteristics of positive displacement pumps

ANSWERS to activities in Part 1

A	10.2kW
B	86.3%
C	This is self-contained
D	Speed ratio is 6000/5400 = 1.111 Mass flow corrected is 290 × 1.111 = 322 kg/s. Head (pressure really) corrected is 11600 × $(1.111)^2$ = 14320kPa 
E	Specific Speed is 6360, so it is a Mixed-Flow pump.
F	Specific Speed is 2681, typical for a radial flow pump (remember to use Q in m^3/h here)
G	We shall discuss how the system curve is obtained = Static Head + PVH +kQ^2
H	The pump H-Q points are consistent with extrapolation of the curve, so the pump condition is fine. The system has reduced in resistance, and the pump is drawing more power than the motor rating, so the motor trips. The nozzle at the end of the pipe had worn off.
I	We will discuss this in the session
J	NPSH-A calculates at 4.08m. At 60L/s, the pump needs 2.3m (see pump curve) , so there is adequate margin.
K	What do you decide from the test point? We will discuss this in the session.
L	Same as for K

11. Let’s test a pump

The AJAX pump (details in the catalogue curves in Figure 11) was supplied with an impeller of 258mm diameter. It has just returned from overhaul at a reliable workshop. A new impeller and wearing rings were installed.

BUT, the pump does not supply the required flow! You are to run a H-Q test, unthrottled, and compare the test point with the only data available - the catalogue curve.

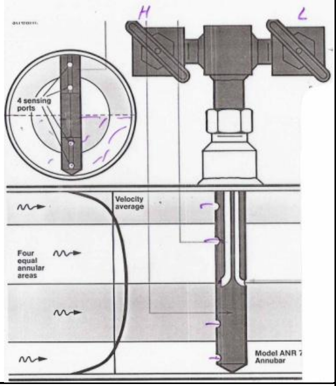
Also plot the static head and sketch the system curve. What do you conclude?
The test sheet is given below.

FLOW Q

"Eagle Eye" meter supplied with Annubar

Reading	% × 0.6
%	L/s

Measured with "Annubar" pitot tube in pipe.



(Flow equation given by OEM from Lab calibration)

Test pressures are measured at pump flanges.

Bore diameters at flanges: Suction: 200 mm, Discharge: 150mm.

Suction pressure gauge is at pump centreline level

Discharge gauge is 0.17m above centreline

(Difference in water levels, suction to discharge, is 12.5m)

Head at suction								
Suction pressure reading	Calibration correction (test sheet)	Suction pressure corrected	Static Suction Head (calc'd)	Static leg (measured)	*Corrected Static Suction Head*	Velocity at Suction V_s (Calc'd)	*Velocity Head at Suction $V_s^2/2g$ (calc'd)*	**Total Suction Head** (includes velocity head)
Test gauge	-0.2		kPa ÷ 9.8	0		Q/A_s		**A**
kPa	kPa	kPa	m	m	m	m/s	m	m
				0				
				0				

Head at discharge									
Discharge pressure reading	Calibration correction (test sheet)	Discharge pressure corrected	Static Discharge Head (calc'd)	Static leg (measured) (gauge above pump C/L)	*Corrected Static Discharge Head*	Velocity at discharge V_d	*Velocity head - discharge $V_d^2/2g$ (calc'd)*	**Total Discharge Head**	**TOTAL HEAD H**
	0		kPa ÷ 9.8	+0.17m				**B**	**B – A**
kPa	kPa	kPa	m	m	m	m/s		m	m
	0			*+0.17m*					
	0			*+0.17m*					

(Calibration error on this instrument is nil).

EXAMPLE of head calculation (shown for suction):

Say the flow is **70** L/s**.** With the diameter of 200mm at the suction measuring point, pipe area is

$$\pi d^2/4 \;=\; \pi\; 0.2^2/4 \;=\; 0.03 \text{ m}^2.$$

Therefore velocity = Q/A_s [units: m^3/s and m^2]= (70 ÷ 1000)/0.03 = **2.33** m/s.

Test suction pressure reads -19.8 kPa. Corrected pressure is (–19.8 – 0.2) = – 20.00kPa;

Static leg here is Nil, as the suction gauge centre is at pump centreline level, so Static Suction Head =

$$-20.00 \div 9.8 = -2.04\text{m}$$

But this pressure does not include that due to flow, so need to add on Velocity Head = $V^2/2g$ [g = 9.8]
= $2.33^2/(2 \times 9.8)$ = 0.277m, so

$$\text{Total Suction Head} = -2.04 + 0.277 = -1.763\text{m}$$

Note: Velocity head and static leg are negligible for high pressure pumps.

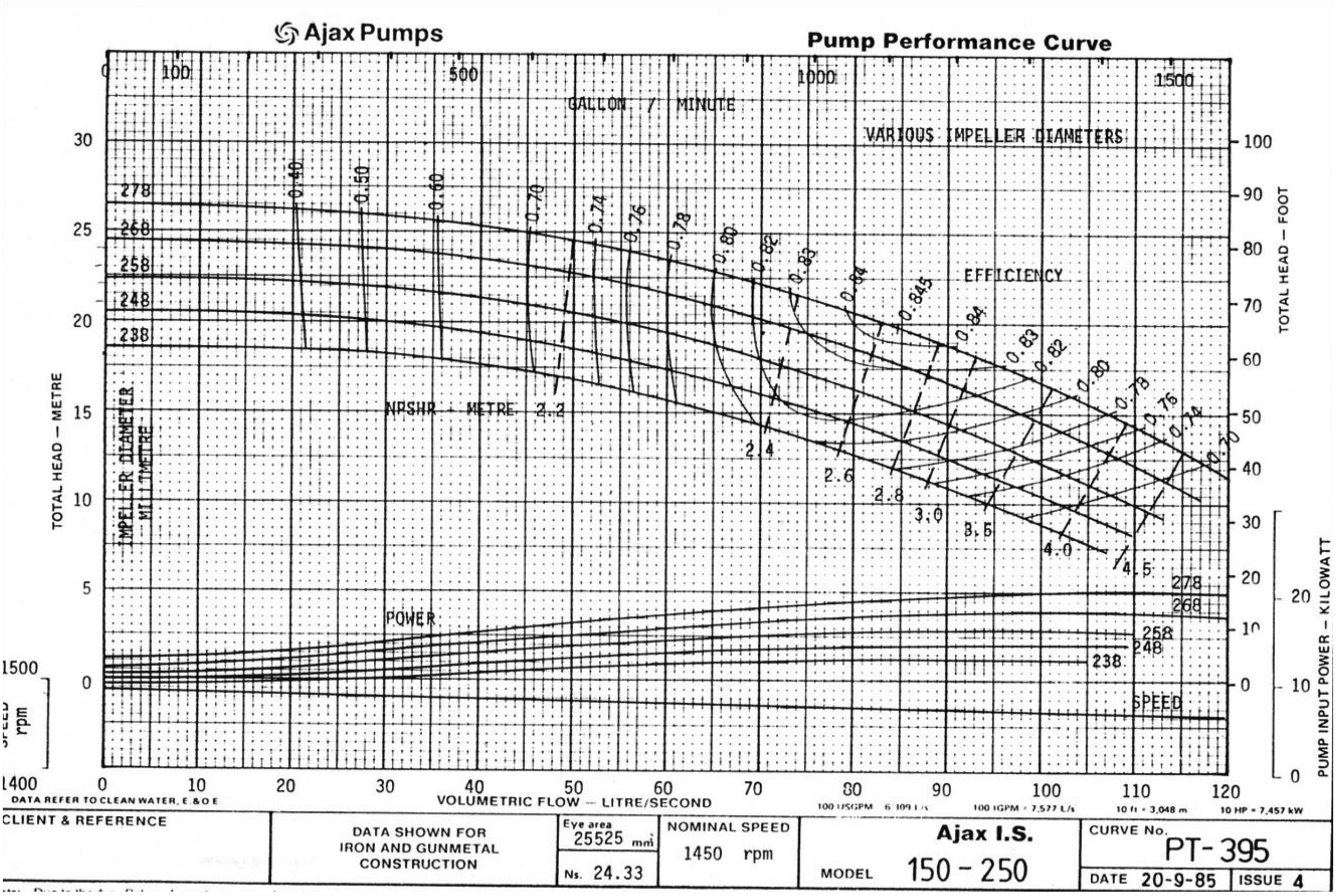


Figure 11: Catalogue curves for test pump, showing for a range of impeller diameters

Your calculations and conclusion:

REFERENCES and resources (just some) for Part 1

Europump.org – several excellent guides. Hydraulics Institute pumps.org

Pump Industry Association – Australian Pump Technical Handbook

Beebe (2004) *Predicting maintenance of pumps using condition monitoring* Elsevier UK

AS2417-2001 *Rotodynamic pumps – Hydraulic performance acceptance tests- Grades1 and 2*

ISO 5617:1997 Measurement *of fluid flow by means of orifice plates, nozzles and venturi tubes inserted in circular cross-section conduits running full*

Yedidiah, S (1996) *Centrifugal Pump Users Guidebook* Chapman & Hall (and others by Palgrave, Karassik, Stepanoff, Shiels, Bacchus, Mackay, Bloch, etc.)

World Pumps magazine	www.worldpumps.com
Hydraulics Institute (see Tip Sheets, etc.)	www.Pumps.org www.PumpsLearning.org
Pump System Improvement Modelling Tool (PSIM)	www.PumpSystemsMatter.org
Pumps & Systems magazine	http://pump-zone.com/
Lots of free stuff	http://www.pump-flo.com/
Same again- and FAQs, examples	http://lightmypump.com/
US Dept of Energy: Pumping System Assessment Tool (PSAT) (free)	http://www1.eere.energy.gov/industry/bestpractices/ (search under "Pumps")
Texas A&M University – Turbo Labs	http://turbolab.tamu.edu/resources/default.aspx
ISO 9906-1999 - Pump testing is the same as AS2417-2001	ISO13709: 2003 (API 610)

PART 2: CONDITION MONITORING OF PUMPS

1. When to overhaul a pump?

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, until recently there was little information available on how to apply condition-based maintenance to pumps (ANSI/HI, 2000 and Beebe, 2004).

Monitoring methods should be chosen where justified that will detect *each of the degradation modes* which are experienced or expected. For critical machines, more than one method of condition monitoring may be justified.

1.1 Vibration monitoring and analysis, probably the most widely applied method of condition monitoring for rotating machines in general, is suited to detect such faults as bearing wear, unbalance, misalignment, looseness. Standards for assessing severity of vibration have been developed over many years from wide experience and are a useful guide.

Vibration severity zone boundaries to ISO10816-3:1998 Measurements on bearing, maximum of H, V directions, machine at steady state operation. Displacement values also given, and differ between all four Groups.				
Velocity mm/s rms 10-1000Hz (For speeds <600r/min, 2-1000Hz).	**Group 1**: Large machines 300kW to 50MW, electrical machines shaft height 315mm+ **Group 3:** Pumps with multi-vane impeller, separate driver, 15kW+, at rated flow		**Group 2**: Medium machines 15kW to 300kW, electrical machines shaft height 160mm to 315mm **Group 4:** Pumps with multi-vane impeller, integrated driver, 15kW+, at rated flow	
Supports:	Rigid	Flexible	Rigid	Flexible
1.4			A/B	
2.3	A/B			A/B
2.8			B/C	
3.5		A/B		B/C
4.5	B/C		C/D	
7.1	C/D	B/C	D+	C/D
11.0	D+	C/D		D+
above		D+		

Systems are available for regular routine or continuous monitoring to analyse vibration signals and detect changes in the spectrum that point to developing faults. Special ultrasonic methods can be used for rolling element bearings.

1.2 Sampling and analysis of lubricants is widely used. Lubricant quality checks have extended oil change intervals. Analysing the presence and developing extent of tiny metal particles that point to wear is well-established, with analysis on site or by external services.

1.3 Electrical plant tests apply for monitoring condition pf motor insulation.

1.4 Visual inspection using aids such as boroscopes can check inside pump casings and **Non-Destructive Testing** using ultrasonic thickness testing can monitor wear of casings

1.5 Performance monitoring and analysis is relevant for pump internal condition, and here we will demonstrate its use with some examples of condition monitoring in practice.
This Part 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.

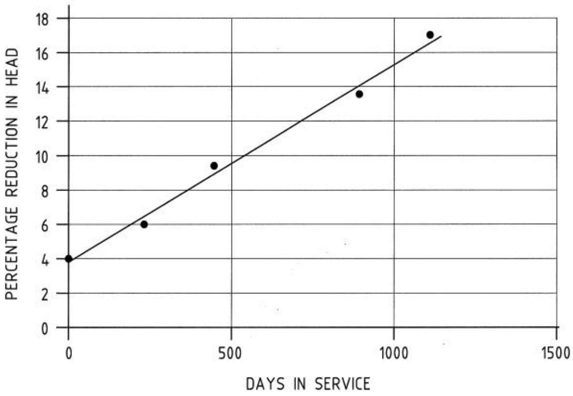
2. The Head-Flow method shows pump wear

The most useful condition monitoring method by performance analysis 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, as in Figure 1.

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 affinity laws (Beebe, 2004)

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 the figure 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)



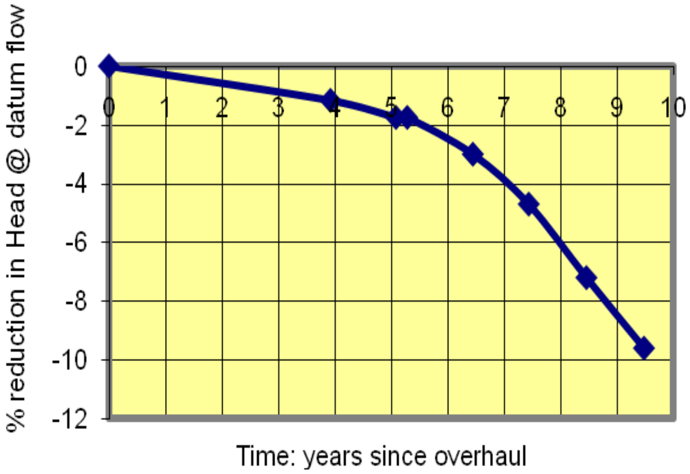


Figure 12: Degradation of pumps shown by Head-Flow testing (left) 230kW, (right) 5744kW

This figure 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.

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? A method for helping this decision is given later.

3. The shut-off head method for CM 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 from built-up pressure 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. 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 results in Figure 12 were obtained on a 19kW wastewater pump where flow could not be measured. Overhaul restored its performance.

Figure 12: Trend of pump wear from Shut-off Head tests

4. Condition monitoring by indirect eficiency measurement (Thermometric)

Hydraulic inefficiency in a pump converts into heat, so if the liquid temperature rise is closely measured, the efficiency can be found. For all but small pumps, mechanical losses are negligible by comparison.

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. From assessment of motor losses, the power absorbed by the pump is computed. From all this data, the pump flow can be found.

% Efficiency for pumps on water at up to 54°C is also given by this empirical formula, which includes a correction for the isentropic temperature rise which occurs as a result of pressure increase, not from inefficiency. (Total Head is in kPa, temperatures in °C):

$$\frac{100}{[1 - 0.003\ (Inlet\ \ temp\ \ - 2) + 4160\ \dfrac{Temp\ \ rise}{Total\ \ Head}]}$$

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).

5. Condition monitoring by measuring the balance valve leakoff flow

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, as shown in Figure 13.

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.

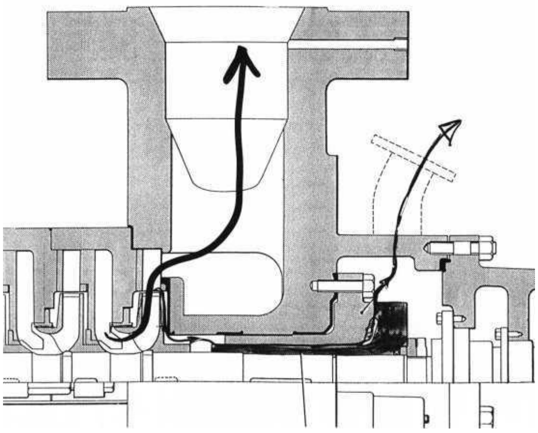


Figure 13: cross-section at discharge end of multi-stage pump showing balance valve

The trend of wear ands the effect of overhaul on a boiler feed pump is shown in Figure 14. Flows are read manually, and trends plotted using a database program. Note that here the balance flow of 15 L/s when worn corresponds to about 10% of the duty flow, and about 250kW of extra power. When added to the likely internal recirculation, this would mean 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.

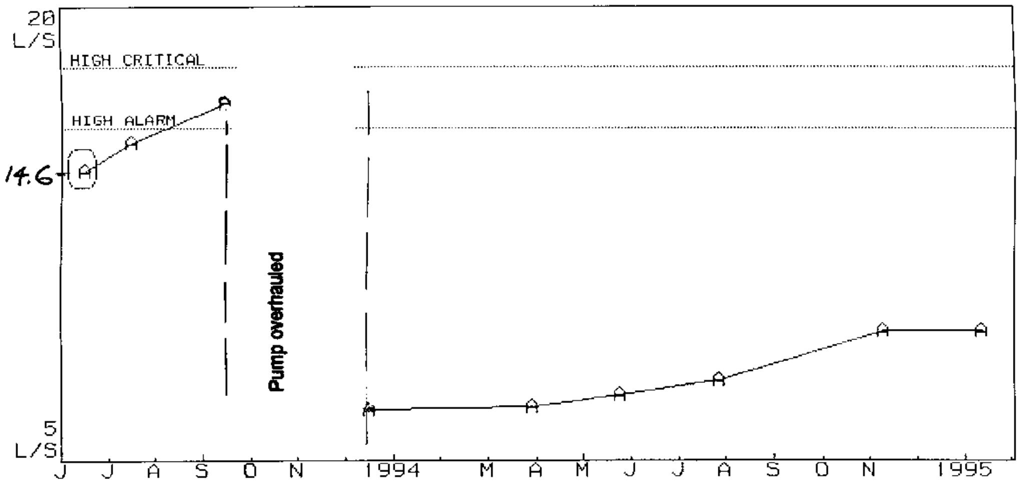


Figure 14: Condition monitoring of a high energy multi-stage pump by measurement of balance device leakoff flow. (Note: flows are corrected to a standard pump speed)

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.

6. 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.

6.1 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".

6.2 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.

6.3 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 15 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.

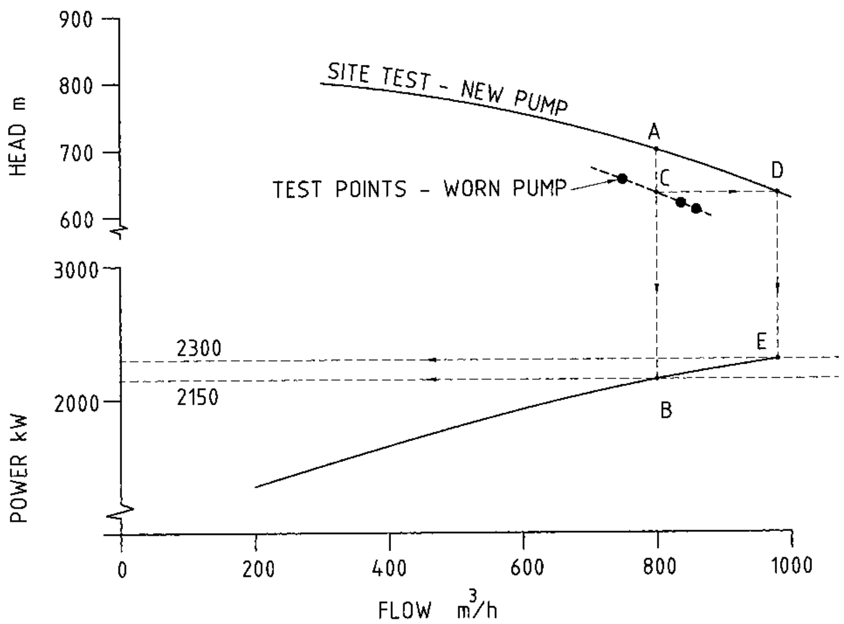


Figure 15: Head-flow-power characteristics of new pump, and head-flow points from worn pump.

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).

6.4 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) from $T = \sqrt{\frac{2 \times O}{C}}$

where: O = cost of overhaul

C = cost rate of deterioration

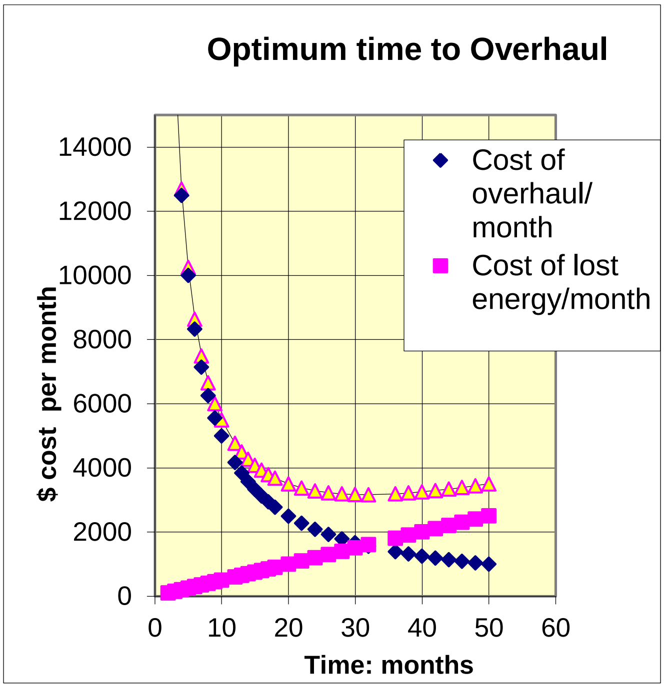
...... giving here T = 27.2 months, but it is better to calculate and plot the average total cost/month values for a range of times. Seen clearly will be the cost impact of doing the repairs at some other time, such as at a scheduled plant shutdown.

6.5 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:	the sum of these two figures =	$3578/month.

Repeat this calculation for several months, perhaps using a spreadsheet, and look for the minimum total cost, which is at 27.2 months, as shown in **Figure 15**. When plotted as cost/month against time, the resulting curves will show the cost per month of overhaul decreasing 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 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 × 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! Other formulae apply for rates of change which are not linear (Haynes & Fitzgerald, 1986).

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.*
- *If a pump varies in its duty, then the energy usage would be corrected in proportion.*
- *The cost of electricity to be used here may vary with the power supplier's tariff structure. The cost may be less in stepped blocks with higher consumption levels for the plant.*

6.6 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 (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 16 following 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.

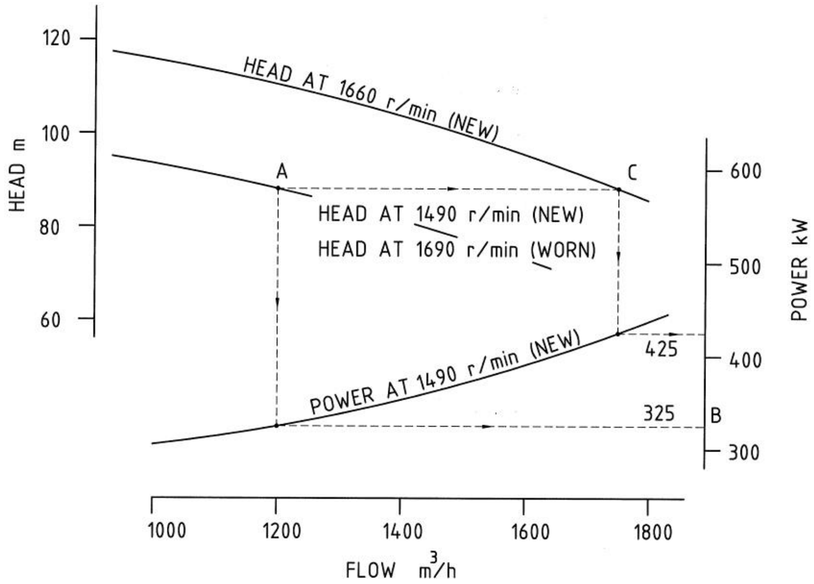


Figure 16: Head-flow-power characteristics of new variable-speed pump, and head-flow points from worn pump.

7. 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.

OPTIMISATION OF TIME FOR PUMP OVERHAUL: EXERCISE

A boiler feed pump has Head-Power-Flow site test curves as in Figure 17. It runs at constant speed and its output controlled by a regulating valve.

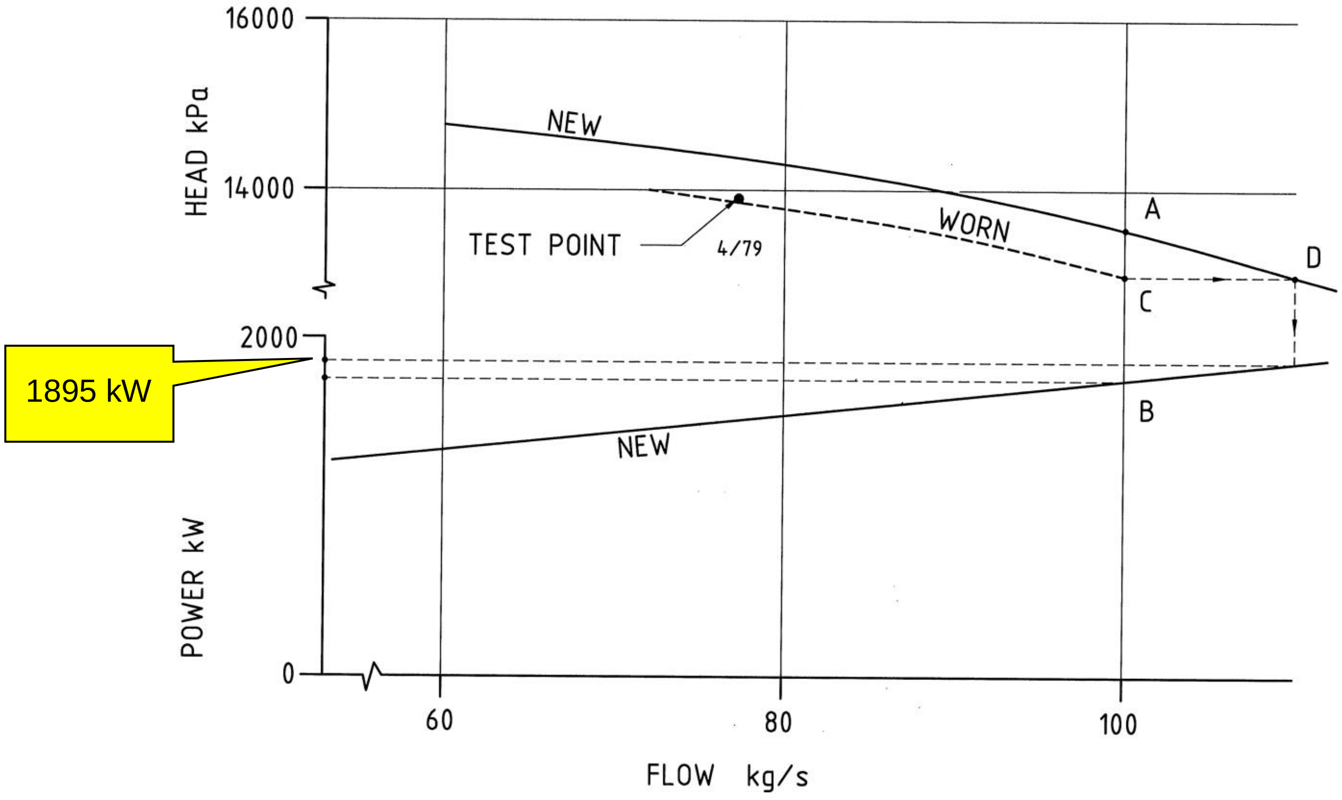
Duty flow is 100 kg/s, at **A** in **NEW** condition. Power required at this duty is read off the Power-Flow curve as 1785kW at **B**. With a motor of efficiency 95%, power consumed is 1880kW with the normal hot water.

After 15 months of service, a Head-Flow condition monitoring tests show that internal wear has occurred: **TEST POINT 4/79**.

Extrapolate the "**WORN**” curve to **C** to find the effect at normal duty. (Production is not lost yet because the system resistance curve lowers when the regulating valve is opened further, and pump output flow is the same).

From previous experience, an overhaul costs $50 000. At this time, the cost of extra used-in-station electricity is 2c/kWh (as extra internal consumption is lost sales), and the pump runs for 90% of the time on average.

- ***WHEN is the best time to overhaul to minimise total cost?***



- Estimate the increased power required in the worn condition: 1895 kW (Scale off figure)

- Calculate the extra electricity consumption: 1895 – 1785 = 110kW ÷ motor efficiency (95%)
  = kW.

- Calculate the current extra cost of electricity: × × × 720 = $ /month.
  (Use average month @720h).

- Calculate the *average cost rate of deterioration:* $ ÷ gives $ /month/month.

- Calculate the optimum time for overhaul from $T = \sqrt{\frac{2\ O}{C}}$

where: O = cost of overhaul =

C = cost rate of deterioration =

...... giving here T = months

Suppose the pump is not in a utility power station, but in another process plant that buys its electricity, but in addition to the supply charge, has a maximum demand charge. What would be the effect of a demand charge of say, $14/kW/month?

REFERENCES for Part 2

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

PIA: *Australian Pump Technical Handbook* (2009)

Beebe, R S (2004) *Predictive maintenance of pumps using condition monitoring* Elsevier, London

Beebe, R S (2002): *Thermometric testing of high energy pumps using pipe surface measurements* 3rd ACSIM (Asia-Pacific Conference on Systems Integrity and Maintenance), Cairns, Australia (2002)

Haynes, C J and Fitzgerald, M A (1986): *Scheduling Power Plant Maintenance Using Performance Data* ASME Paper 86-JPGC-Pwr-63

Karassik, I J et al (Eds) (2001) *Pump Handbook* McGraw-Hill

Robertson, M et al (2007) *Continuous Pump Performance Monitoring and Scheduling* IMechE Symposium - Energy Savings in Pumps and Pumping, London

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

Whillier, A (1972): *Site testing of high-lift pumps in the South African mining industry* IMechE paper C155/72 Conference on Site testing of Pumps London (1972) pp209-217 (I attended this while working in the UK 1971-73)

(The optimisation method has been presented at several conferences and also published in various forms in several books and technical magazines around the world)

Answer to optimisation example

The power required for this duty in the worn condition is scaled off the figure by the projection from the duty flow of 100kg/s to the “worn” curve to find 13107 kPa head, then across to the "Site test-new" curve, then down to the power-flow curve, to find 1895 kW.

The extra electricity consumption is therefore 1895 - 1785 = 110kW ÷ motor efficiency (95%) = **116kW**.

Our test shows that the *rate* of increasing cost/month has reached 116 × 0.02 × 0.9 × 720 = $1503/month. (A month has 720h on average).

As the time now is 15 months, $1503 ÷ 15 gives the *average cost rate of deterioration* as $100/month/month.

The optimum time for overhaul calculated from $T = \sqrt{\frac{2O}{C}}$

gives T = 32 months

PART 3: ANOTHER ENERGY SAVING OPPORTUNITY

1. “Tune” a pump to save energy?

In selecting a pump for a given duty, the designer assumes frictional conditions, and allows margins on head and flow to be sure that the pump will meet design requirements. But, when the plant is built and put into service, this may mean that the pump is oversized. As an example, the results of a survey in Finland by Jantunen (2000) gave the average overall pumping efficiency as less than 40%. Pumps consume about 10% of the total electricity in Finland, so the wastage is very high. The US Department of Energy estimates that 5% of all power is consumed by pumps and that a 20% reduction is possible (HI, 1997). A similar situation can be expected in other countries.

This Part shows how a smaller impeller or one of reduced diameter can give big savings in power used. The method can also be used to investigate a suspected smaller impeller in the case of a new pump or an overhauled one not performing as expected. All plant systems with benign liquids where the wear rate is slow should be examined to see if this oversized impeller situation occurs.

There are other efficiency improvement measures:

- Modern software facilitates speedy checks of the system design, to evaluate the effect of changes such as diffuser entry-exit sections at pipe-tank connections.
- Variable-speed drives may be retro-fitted
- Adding a smaller pump in parallel, for use where the required flow is sometimes less than the full flow required from a pump.
- If there are several pumps in parallel of different sizes, power may be able to be saved by running differing combinations of pumps to suit varying duty flows.

Condition monitoring by performance analysis can also provide the information to determine when overhaul of a pump to restore worn clearances or other degradation is justified on an energy cost basis (Beebe 2004). Other condition monitoring methods could also be used as appropriate to detect other modes of wear. For example, vibration analysis for looseness, misalignment, or bearing condition; and non-destructive testing by ultrasonic thickness measurement for casing wear.

Here we look at two examples of different sizes and types of pump, and show how performance analysis of even a medium size pump can give large savings if a smaller diameter impeller is sufficient for the required duty. Impellers can be machined down for radial-flow and to a limited extent with mixed-flow pumps. Reducing the diameter of star vane impellers of side channel pumps is not possible (SIHI, 1988).

A workbook exercise concludes, giving an opportunity to practice the method explained.

Changing speed where this is possible can dramatically reduce power consumption and how to estimate this is included.

2. Impeller trimming: Medium size pump - 132kW

2.1 Design selection. A plant system handling clean water requires a flow of **615m³/h**. The system has a static head of 37m, and adding the calculated frictional resistance gives a total head at this flow of **55m.** With the usual design margins added, a constant speed pump with a 375mm diameter impeller was selected to meet this duty: see Head-Power-Flow curves "375ϕ" on Figure 18. This pump has a Specific Speed of 1920 (US units).

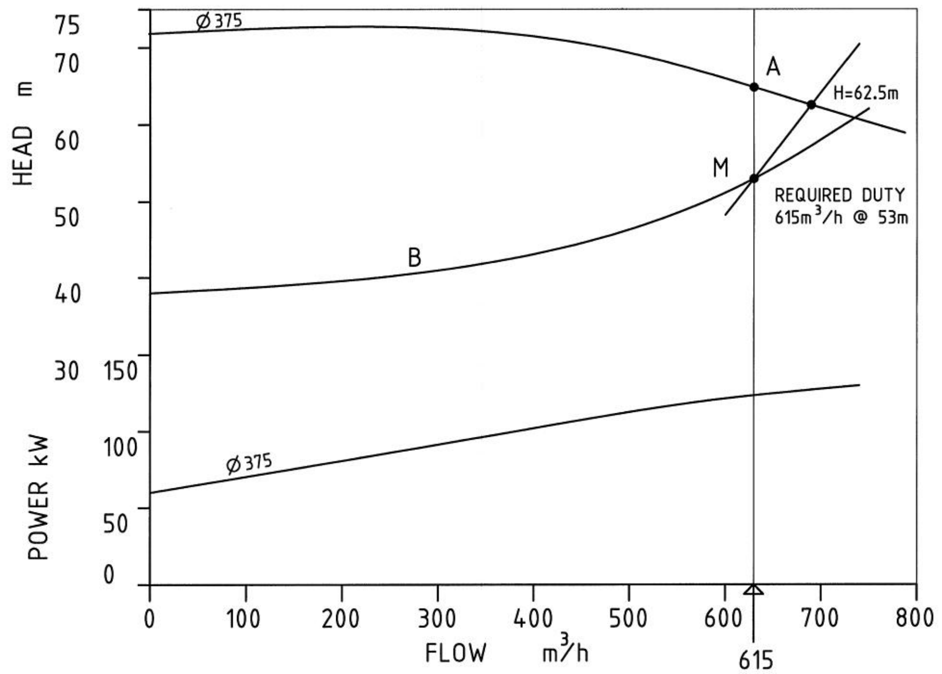


Figure 18: System curve and original pump characteristics

2.2 Operation in service. When the pump is in service, its discharge must be throttled to keep the flow down to the required 615m³/h. A head-flow test shows that the operating point is **A**, at a head of 64m. From the Power-Flow curve in the above figure, the pump *absorbs* 123kW.

The *actual* system curve was found on test on site by running the pump *unthrottled*, measuring the suction and discharge pressures and the flow. For such tests velocity head is allowed for if the suction and discharge pipes are of different diameters at the points of pressure measurement, and if the gauges are not at the same level, readings are corrected to allow for static leg (Beebe, 2004).

The system static head of 37m - the zero flow point – is found from site measurements or plant elevation drawings. The operating point, unthrottled, is found to be 725 m³/h @ 60m, so the actual system curve **B** shows that system resistance is below the design expectation. The required operating point - the total system resistance for the desired flow - is therefore **615 m³/h@ 53m.** We shall designate these points as H_1 and Q_1.

2.3 How to calculate the reduced impeller diameter. Pump makers typically use a range of impellers in any one casing size. If a geometrically similar smaller impeller is used, the performance characteristics would follow the affinity laws given in pump textbooks. For economy in manufacture, a "standard" impeller can be turned down to get reduced head-flow performance. As geometric similarity is not maintained, other rules are needed to predict performance. (On most radial pumps (i.e. those of lower specific speed), diameter can be reduced by about 10% without affecting efficiency much. Note that cuts of 5% or more may affect the Net Positive Suction Head Required by the pump to avoid cavitation.

Manufacturer's catalogues give curves for smaller diameter impellers. If this data is not available, as with engineered pumps, the approximate method here can be used. Experts differ on how the cut should be calculated, but this method from Stepanoff (1957) and Karassik et al (2008 and earlier editions) should be close enough to *estimate* likely power savings. Another, but slower, approach is given in the Hydraulics Institute training publications (HI, 1997).

Designating **R** is the diameter ratio, Table 1 gives the relationships: (For pumps with blade outer edges not parallel to the axis, the calculation should use the mean diameter).

Impeller details	Flow varies as	Head varies as	Power varies as
The usual situation: the ***impeller width will alter*** with the cut (i.e. the shrouds are not parallel).	$\mathbf{R}$	$\mathbf{R}^2$	$\mathbf{R}^3$
If the ***impeller width and exit angle will not be altered*** by the cut (vanes will still overlap):	$\mathbf{R}^2$	$\mathbf{R}^2$	$\mathbf{R}^4$

Table 1 Relationships with cut-down impellers

In this case, the first set of relationships applies.

- Choose an arbitrary flow above the desired flow to obtain a point on the parabola that is defined by the affinity laws: at say, Q_2 = 720m³/h. (The parabola starts at the origin). Calculate the Head corresponding to this flow from:

$$H_2 = H_1 \times \left(\frac{Q_2}{Q_1}\right)^2 = 53 \times \left(\frac{720}{615}\right)^2 = 72.6m$$

- Plot this new point, and draw a line to the desired duty point (Figure 1). This part of the parabola is essentially a straight line, but the process can be repeated if desired to get another point and plot the very slight curve.

- Read off the he intersection of this line with the 375ϕ pump curve in Figure 1: H_3 = 62.5m

- Calculate the new diameter:

$$D_2 = D_1 \times \sqrt{\frac{H_1}{H_3}} = \sqrt{\frac{53}{62.5}} = 345 \ mm$$

As these affinity laws are not exact, a cut to this diameter would probably give a greater reduction in head and flow than required. A correction originating with Stepanoff (1957) is updated in Karassik (2008). For pumps of Specific Speed up to 2500, the *actual diameter for machining* is found from:

[(Calculated diameter, as decimal fraction of original) × 0.857 + 0.143].

In this example, the machining diameter required is therefore:

(345 ÷ 375) × 0.857 + 0.143 = 0.931, or **349*mm***

Tests comparisons in our laboratory of a 20kW pump with two impellers: one 258mm diameter, and another trimmed to 238mm diameter, showed that the closest prediction of performance was given by the formula in Appendix B (normative) of AS2417-2001 *Rotodynamic pumps – Hydraulic performance acceptance tests Grades 1 and 2* [same as ISO 9906:1999(E)].

With D_1 = the mean diameter of the suction eye, D_r = the reduced diameter, and D_t = the initial diameter

$$R = \sqrt{\frac{D_r^2 - D_1^2}{D_t^2 - D_1^2}}$$

Then calculate pairs of new points and plot: $Q_r = R \times Q_t$

$H_r = R^2 \times H_t$

2.4 Power savings with a cutdown impeller. The manufacturer's catalogue should be consulted to find the performance data with a smaller impeller of this size.

If the catalogue is not available, points from the 375ϕ Head-Flow curves could be corrected using the above relationships, *but calculated using the uncorrected, i.e. initially calculated 345mm diameter*, and plotted if desired. For Flow and Head, R = 345 ÷ 375 = 0.92. Therefore, R = 0.92 to correct Flow, and R^2 = 0.8464 is the correction factor for Head.

Power-Flow points are however *calculated using the true 349mm diameter*. R for this situation is 0.931, therefore R^3 = 0.8061. Table 2 gives original and corrected data for a selection of points:

Flow *Q*		Head – *H*		Power - *P*	
375ϕ impeller	*349ϕ impeller*	375ϕ impeller	*349ϕ impeller*	375ϕ impeller	*349ϕ impeller*
Correction factor: 0.92		Correction factor: 0.8464		Correction factor: 0.8061	
400m³/h	*368 m³/h*	68.7m	*58.1*	106kW	*85.4*
500	*460*	67.2	*56.8*	116	*93.5*
600	*552*	64.5	*54.6*	124	*100*
700	*644*	61.2	*51.7*	130	*104.7*
668 ←	***615***			128.5 →	***103.5***

Table 2 Cut-down impeller- some original and corrected data

When plotted on the first figure in this Part, to a suitably larger scale, the power absorbed at 615m³/h can be read off as 103.5kW. An alternative way is to calculate the power directly, by starting with the required flow of 615m³/h. Dividing by 0.92 gives the flow at the original diameter: 668m³/h. From the original curve, the power at 668 m³/h is read off as 128.5kW. Multiplying by 0.8061 gives 103.5kW, as before.

The reduction in power drawn by the motor can be found by subtracting that required with the smaller impeller from the previous value, and dividing by the motor efficiency (typically 0.97). The cost of power must of course be known. In this example, if a power cost of 14c/kWh applies.

With a 24h/day pumping operation, this saves:

$$\left[\frac{123 - 103.5}{0.97}\right] \times 0.14 \times 8760 = \$24\ 640 \quad per \quad year$$

2.5 Action to take? Large estimated savings certainly justify action, but before proceeding to machine an impeller, the pump manufacturer should be consulted to verify and advise.

If it is decided to machine the impeller, it would be safer to do this in steps and retest after each stage. (Machined-off impeller vane ends cannot be readily stuck back on, although this method can also be used with centrifugal fans, where extensions are feasible).

If a particular pump is known to wear rapidly, or the frictional resistance in the system it supplies increases due to internal deposition, then the smallest impeller calculated this way may not be appropriate. Some performance margin may be justified. Another possibility would be to order another impeller for the present actual duty, and keep the original size one in store just in case. An alternative narrower impeller design may be available. In this example, the cost of am extra impeller would be recovered in a few weeks.

The reduced power required may enable a smaller motor to be used. Further energy savings could result if this runs nearer its rated output. Even though motor efficiencies may be similar, less overall losses in kW will occur with a smaller motor required to give the same output at the same motor efficiency (http://www.productiveenergy.com/resources/resources.asp). A high efficiency motor would give further energy savings.

Incidentally, the **effects of changes in speed** follow the same affinity laws as when the impeller alters with a diameter cut. The same method is used as above, but using *speed* in place of *diameter* in the steps given. The correction to impeller diameter for machining does not of course apply in this case. It is possible to change both diameter and speed together to obtain a desired performance characteristic.

3. Impeller trimming: large size pump – 1320kW

3.1 Design selection. In this case, there are two power station units, each with two cooling water pumps designed to operate in parallel (2 ×50% capacity). Each pump was selected to supply a total flow of 4.675 m³/s at a design head of 24.5m. The combined flow must not be exceeded or unacceptable erosion of condenser tubes is likely. The pumps are single-stage, with double entry impellers of 986mm outside diameter. The impeller shrouds are not parallel. Motor efficiency is 95.3%.

3.2 Operation in service. Site tests revealed that the system resistance was only 21.6m, much less than design, such that the flow must be restricted by partly closing the outlet isolating valves. Portion of the original Head-Flow and Power-Flow curves for one pump are shown in Figure 19. The duty points are shown as “Design” and “Actual”.

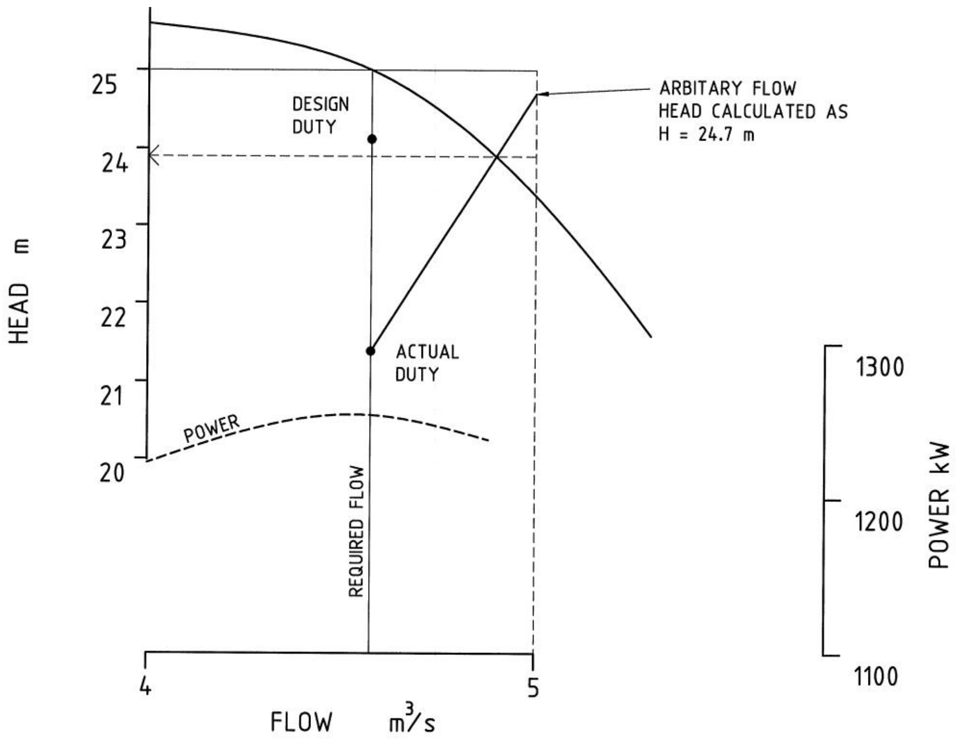


Figure 19: Cooling Water Pump – original performance characteristics and system points (Not to scale)

3.2 Calculation of reduced impeller diameter. Following the same process as in the first example, an arbitrary flow is selected as Q = 5m³/s, and the corresponding Head calculated:

$$H_2 = H_1 \times \left(\frac{Q_2}{Q_1}\right)^2 = 2.1 \cdot 6 \times \left(\frac{5 \cdot 0}{4 \cdot 68}\right)^2 = 24.7\,m$$

When plotted and joined to the desired duty point, the Head is read off the intersection of this line with the original pump curve: H_3 = 23.8m. The new diameter is therefore:

$$D_2 = D_1 \times \sqrt{\frac{21 \cdot 6}{23 \cdot 8}} = 939\ mm$$

The correction used before for machining applies for pumps of Specific Speed up to 2500. This pump has a Specific Speed of 4400, outside the range for which a correction is given in the later texts, so the correction does not apply.

As this is an engineered pump, manufacturers catalogue data is not likely to be available, and some points from the existing Head-Flow curves could be corrected using the above relationships, based on the first calculated diameter (i.e. 939mm) and plotted if desired.

3.4 Power savings with cutdown impeller. From the Power-Flow curve when plotted, the power saved for the required duty flow with a cut-down impeller was estimated as in the first example:

$$\left[\frac{1253 - 1097}{0.953}\right] = 164\ kW\ \ saved\ \ per\ \ pump$$

With four such pumps operating 90% of the year, this totals 5170MWh, equivalent to $206 900 at an average selling price of 4c/kWh.

The impact would of course be proportionally greater with a higher selling price, or if the pumps were in an industrial plant where power was bought in. Such estimated savings in energy cost and in matching greenhouse emissions certainly justify action, and the advice of the pump manufacturer should be sought. *Proceeding to machine such a large impeller without reference is not recommended.*

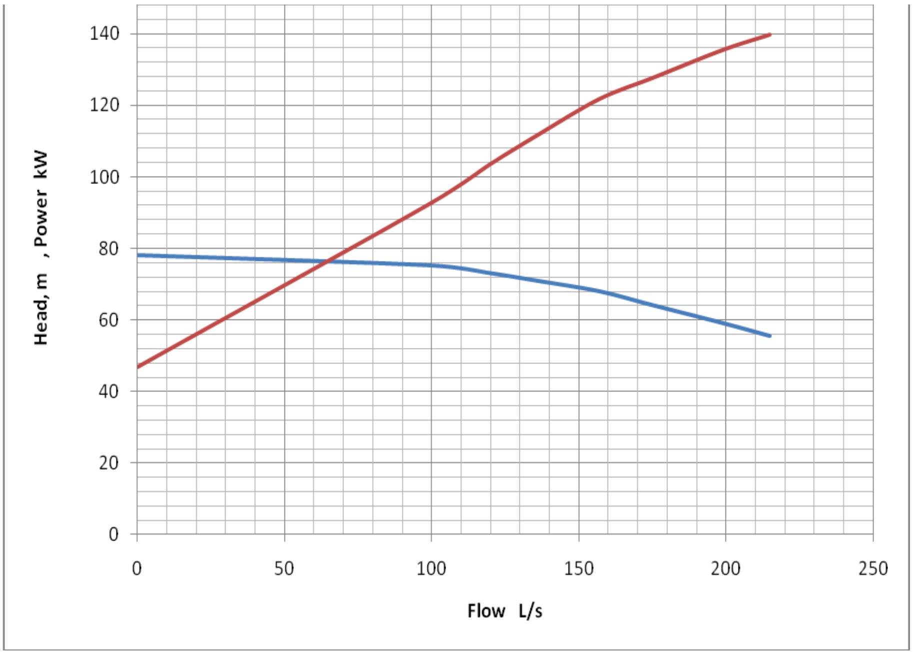
4. Workshop Exercise - 150kW pump

Here is another pump in this situation for you to try. Characteristics are shown below, with origin zero-zero. A later figure is an enlarged version around the area of our interest.

Exercise example (rising curve is Power)

Pump design duty point (draw it on graph): 175L/s @ 64m.

The double-entry type impeller diameter is 449mm. Operating speed is 1460 r/min. Motor 150kW, with efficiency 93.9% @ 100% load and 94.5% from 50 – 75% load.



CM tests with the pump unthrottled show that the system curve is right on the design, with a Total Static Head of 45m. However, in service, the duty flow is less than design, and this pump has to be throttled to maintain a desired 165 L/s. The pump operates on a continuous basis, for 95% of the year.

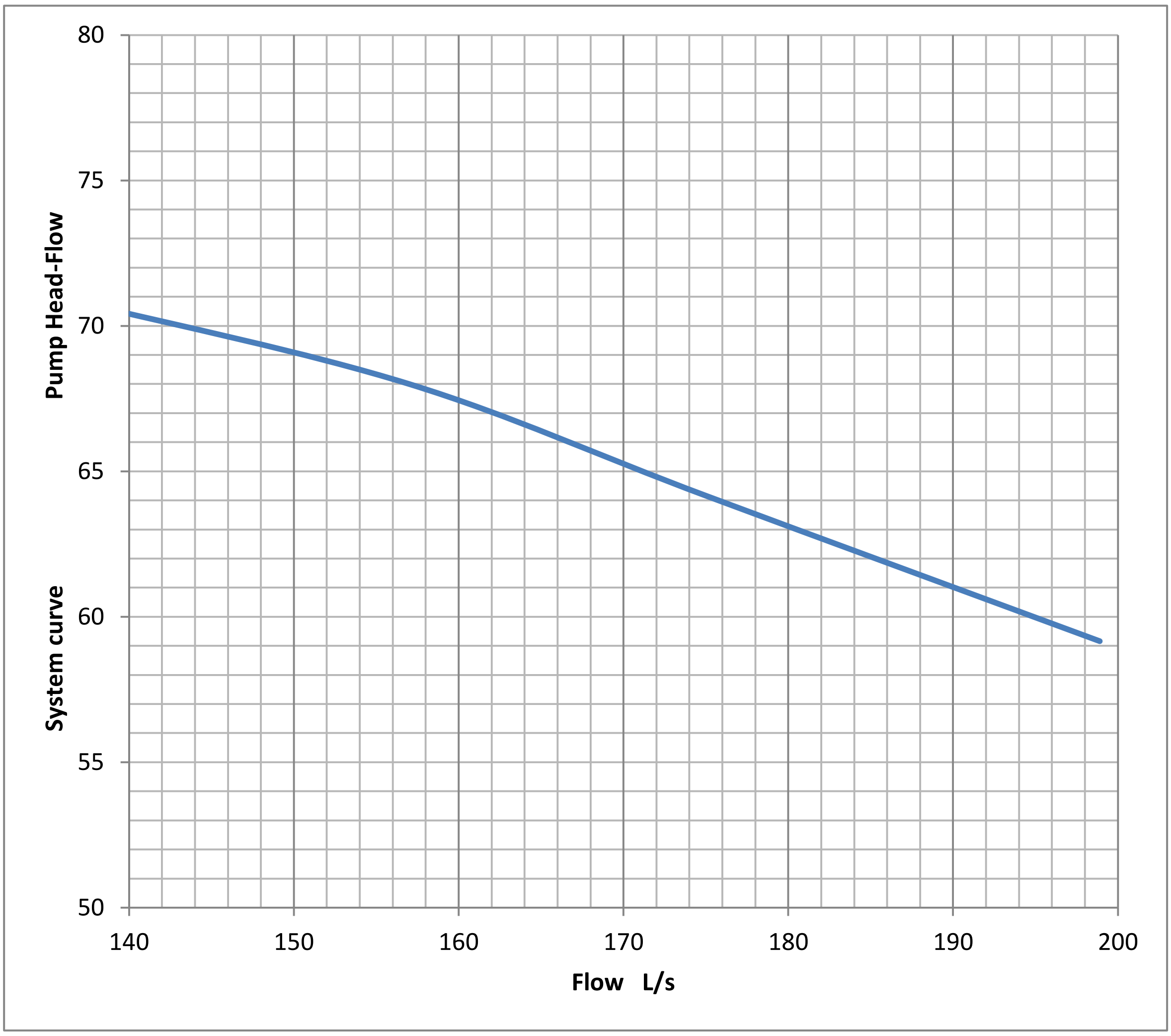
Power costs 12c/kWh, and there is a maximum demand charge of $14/kW.

- Find the diameter the impeller could be machined to, and
- Estimate the power savings likely with the smaller impeller.

Start by calculating the system curve and plot some points to find the **required** duty operating point.

System head H = Total Static Head + k Q^2 =

Flow Q	150	160	170	180
Head calculated				



Pump Head-Flow curve for plotting the system curve and other points

- Choose an arbitrary flow above the desired flow, to obtain a point on the parabola which is defined by the affinity laws: say, $Q_2 =$

Calculate the corresponding Head $H_2 = H_1 \times \left(\frac{Q_2}{Q_1}\right)^2 = \qquad \times \left(\frac{\qquad}{\qquad}\right)^2 = \qquad$ m

- Plot this new point and join it to the desired duty point in Figure 4. This part of the parabola is essentially a straight line, but the process can be repeated if desired to get another point and plot the very slight curve.

- Read off the intersection of this line with the original pump curve $H_3 =$ m (Figure 4).

- Calculate the new diameter from: $D_2 = D_1 \times \sqrt{\frac{H_1}{H_3}} = \qquad \times \sqrt{\frac{\qquad}{\qquad}} =$ mm

- Apply machining correction. First compute the Specific Speed from: $N \frac{\sqrt{Q}}{H^{0.75}}$ using r/min, m³/h, m (Remember that Q is flow per impeller eye):

- If Specific Speed is below 2500, calculate the *actual diameter for machining* from:

[(Calculated diameter, as decimal fraction of original) × *0.857 + 0.143]* × *Original diameter*

Here we get: [(÷ 449) × 0.857 + 0.143] × 449 = mm

No catalogue information is available, so points from the 449ϕ Head-Flow curves could be corrected using the above relationships, and plotted if desired, *but calculating with mm diameter,*

For Flow and Head, R = ÷ 449 = Therefore:

R = to correct the Flow;

R^2 = is the correction factor for Head;

R^3 to correct Power (using the smaller diameter initially computed) =

Calculate and enter original and corrected data for a selection of points:

Flow *Q*		Head – *H*		Power – *P*	
449ϕ impeller	*ϕ impeller*	449ϕ impeller	*ϕ impeller*	449ϕ impeller	*ϕ impeller*
Correction factor: 0.		Correction factor: 0.		Correction factor: 0.	
L/s	*L/s*	m		kW	
←				→	

- Calculate the reduction in power drawn by the motor by subtracting that required with the smaller impeller from the previous value, dividing by the motor efficiency as given at this load.

- Calculate the saving with the 95% time pumping operation. In this example, a power cost of 12ckWh applies:

$$\left[\frac{\quad - \quad}{\quad}\right] \times \quad \times \quad = \$ \quad \textit{per year}$$

- Calculate the potential savings in demand charge:
- What do you conclude?

4. Estimating the speed required for a new duty

A somewhat similar method applies when a change in duty occurs for a given pump and speed can be changed to suit. The new speed and relative change in power can be found:

- Using the new operating point data, calculate the constant to the affinity parabola $H_1 = kQ^2$.
- Choose a flow about 10% below the initial operating point.
- Calculate the corresponding H value and plot the point.
- Join the points and read off the intersection with the pump curve to find H_1
- Calculate the new speed from Initial Speed × $\sqrt{(H_2/H_1)}$
- Change in power required is proportional to $(N_2/N_1)^3$

REFERENCES and resources for Part 3

Bachus, L and Custodio, A (2003) *Know and understand centrifugal pumps* Elsevier, UK

Beebe, R (2004) *Predictive maintenance of pumps using condition monitoring* Elsevier, UK]

Bleu, JL and Lobach, J (1998) *Continuous monitoring of sealless pumps – the next step* Proceedings of 15th International Pump Users Symposium, Texas A&M University

Bloch, HP and Geitner, FK (1994a) *Machinery failure analysis and troubleshooting* (2nd Ed) Vol 2 of Practical Machinery Management for Process Plants, Gulf

Bloch, HP and Geitner, FK (1994b) *An introduction to machinery reliability assessment* (2nd Ed) Gulf

Europump (2006) *System efficiency – a guide for energy efficient pumping systems*]

Pumping Systems Matter and HI *Optimising pumping systems*

http://www.pumpsystemsmatter.org/content_detail.aspx?id=332]

Hydraulics Institute (1997) *Energy reduction in pumps and pumping systems* (Training package) USA

Jantunen, Errki et al (1998) *Flexible expert system for the diagnosis of centrifugal pumps* Proceedings of COMADEM 98 11th International Conference on Condition Monitoring and Diagnostic Engineering Management pp433-442

Karassik, I J et al (2008) *Pump handbook* McGraw-Hill

Netzel, JP and Sabini, EP (2002) *Toward reduced pump operating costs, Part 3: Minimizing the effects of wear and optimising pump efficiency* Proceedings of 19th International Pump Users Symposium, Texas A&M University

Palgrave, Ron (2003) *Troubleshooting centrifugal pumps and their systems* Elsevier, UK

PSAT (2000) Pump System Assessment Tool: free download http://www.oit.doe.gov/bestpractices/steam/psat.html

Stepanoff, A J (1957) *Centrifugal and axial flow pumps* Wiley]

SIHI (1988) *Basic principles for the design of centrifugal pump installations* SIHI-Halberg

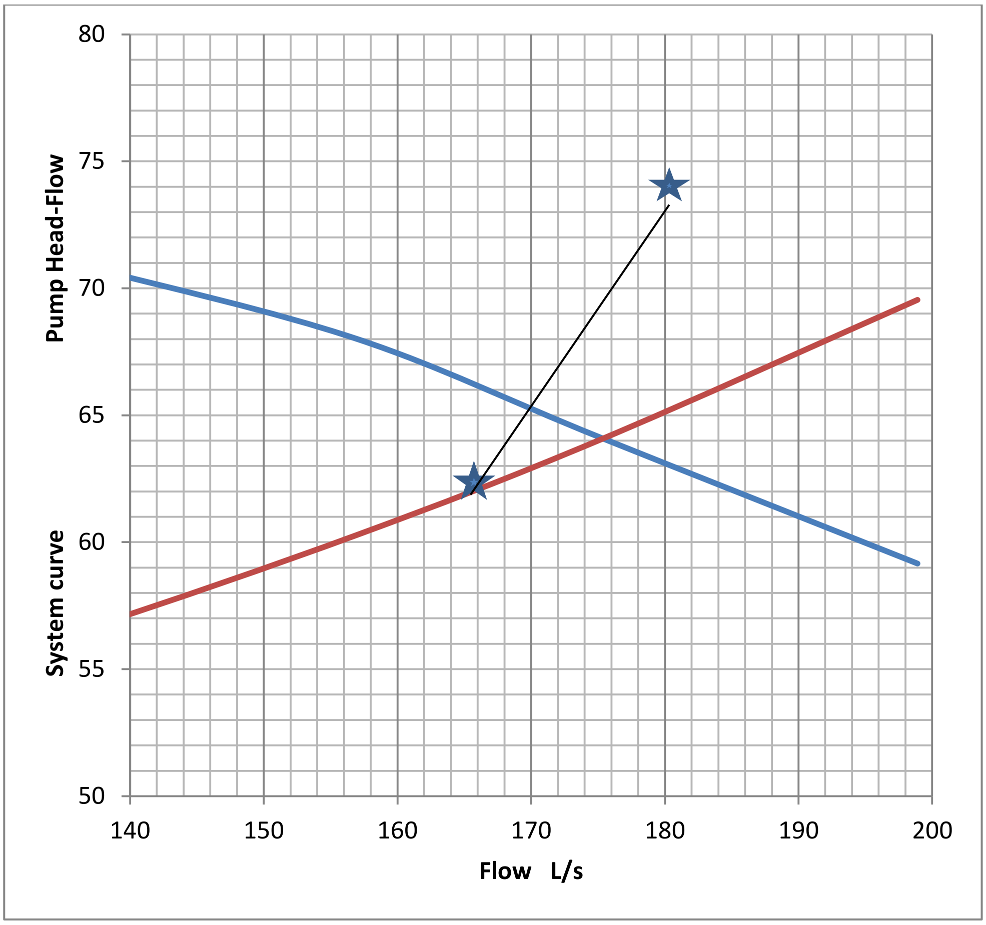
ANSWER to Workshop Exercise:

- Calculate the system curve and plot some points on the portion of the total curve below to find the required duty operating point.

System head H = Total Static Head + k Q^2 = 45 + k Q^2, using Q = 175 L/s gives k = 6.204×10^{-4}

Q	150	160	170	180
H calculated	*58.9*	*60.9*	*62.9*	*65.1*

Plot the required duty point flow (165L/s) on the system curve and read off the value of H_1 *as 62m*



- Choose an arbitrary flow above the desired flow, to obtain a point on the parabola which is defined by the affinity laws: say, Q_2 = *180L/s*

Calculate the corresponding Head $H_2 = H_1 \times \left(\frac{Q_2}{Q_1}\right)^2 = 62 \times \left(\frac{180}{165}\right)^2 = 73.8m$

Plot this new point and join it to the desired duty point in Figure 5. This part of the parabola is essentially a straight line, but the process can be repeated if desired to get another point and plot the very slight curve.

- Read off the intersection of this line with the original pump curve to get H_3 = *65 m*

- Calculate the new diameter from: $D_2 = D_1 \times \sqrt{\frac{H_1}{H_3}} = 449 \times \sqrt{\frac{62}{65}} = 438.5\,mm$

- Apply machining correction. First compute the Specific Speed from: $N\frac{\sqrt{Q}}{H^{0.75}}$ using r/min, m³/h, m (Remember that Q is flow per impeller eye): *1145*

- As this is below 2500, calculate the *actual diameter for machining* from:

[(Calculated diameter, as decimal fraction of original) × *0.857 + 0.143]* × *Original diameter*

Here we get: [(*438.5* ÷ 449) × 0.857 + 0.143] × 449 = **440**mm

No catalogue information is available, so points from the 449ϕ Head-Flow curves could be corrected using the above relationships, and plotted if desired, *but calculating with* 440 *mm diameter*,

For Flow and Head, R = 440 ÷ 449 = 0.980 Therefore:

R = *0.980* to correct the Flow;
R^2 = *0.960* is the correction factor for Head;
R^3 to correct Power (using the smaller diameter initially computed) = *0.931*

Calculate and enter original and corrected data for a selection of points:

Flow *Q*		Head – *H*		Power – *P*	
449ϕ impeller	*440mm ϕ impeller*	449ϕ impeller	*440mm ϕ impeller*	449ϕ impeller	*440mm ϕ impeller*
Correction factor: *0.980*		Correction factor: *0.960*		Correction factor: *0.931*	
L/s	*L/s*	m		kW	
140	*137.2*	71	*68.2*	114	*106.1*
150	*147*	68	*65.3*	118	*109.9*
160	*156.8*	67.5	*64.8*	123	*114.5*
168.4 ←	***165***			126.0 →	***117.3***

- Calculate the reduction in power drawn by the motor by subtracting that required with the smaller impeller from the previous value, dividing by the motor efficiency as given at this load.

- Calculate the saving with the 95% time pumping operation. In this example, a power cost of 12c/kWh applies:

$$\left[\frac{126 - 117.3}{0.94}\right] \times 0.12 \times 0.95 \times 8760 = \$9242 \ \textit{per year}$$

- Calculate the potential savings in demand charge:

7 × 14 × 12 = $1237 per year

Total cost of extra energy per year with oversized impeller = $10479

Water properties at normal atmospheric pressure

Temp °C	0	5	10	15	20	25	30	35	40
Density kg/m^3	999.8	1000	999.7	999.2	998.3	997.1	995.7	994.0	992.3
Vapour pressure m	0.06	0.09	0.12	0.17	0.25	0.33	0.44	0.58	0.76