# ACCURATE MEASURMENT OF PUMP EFFICIENCY IMPROVEMENTS USING 'CUTTING-EDGE' TECHNOLOGY

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## ABSTRACT

Energy for pumping water and wastewater is a major portion of the power costs for many Councils. In the current New Zealand and world climate of increasing energy costs and climate change issues, any reduction in energy use and carbon footprint through efficiency improvements will benefit not only the Council's 'bottom line', but also its obligations as a responsible and environmentally conscious organisation.

This study utilised new technology that uses the thermodynamic method of pump performance measurement that enables measurement of the minute temperature increase of the fluid as it passes through the pump, which has a direct relationship to the energy lost to the fluid. This information, in conjunction with the pump head and input power, can calculate a pump's efficiency far more accurately and with better repeatability than any other method presently available for in-situ testing.

The objective of this study was to understand thoroughly the effect on a pump's performance of a number of standard performance enhancement techniques. The study was conducted under controlled conditions and the result of each enhancement process was measured and recorded after each stage.

This paper will discuss the testing of the GWRC Waterloo pumps, the equipment used, the method used for each pump performance enhancement and the results of the study.

## KEYWORDS

**Energy, Pump efficiency, Thermodynamic pump testing,**

## NOMENCLATURE

| | |
|---|---|
| BTO | Better Technical Options |
| CT | Current Transformer |
| GWRC | Greater Wellington Regional Council |
| VSD | Variable Speed Drive |

# 1 INTRODUCTION

The Greater Wellington Regional Council (GWRC) is responsible for the treatment and distribution of bulk drinking water to Wellington City, Upper Hutt, Lower Hutt and Porirua. In 2006, GWRC used over 21.6GWh of electricity and in 2007 over 19.6GWh, most of which was consumed by large pumps (major energy users) in the distribution network.

Early in 2007, GWRC engaged Better Technical Options Ltd (BTO Ltd) to conduct a Level 1 Energy Audit (as defined in AS/NZS3598:2000). One of the findings from the audit was that there were significant uncertainties surrounding the true efficiency and deterioration from original condition of the major energy users. Any deterioration in efficiency means that a pump will require more energy to move the same quantity of water and therefore a recommendation of the Level 1 Energy Audit was for the present efficiency of the major energy users within the GWRC water infrastructure, be accurately determined. Once the efficiency of each major energy user is accurately determined, if any efficiency enhancement work is needed, each pump can be assessed on a 'case by case' basis.

GWRC have a preventative maintenance program that includes 'pump condition monitoring' and regular inspection of major pumps. This includes the use of vibration analysis techniques and where energy metering allows, the pump efficiency is assessed from flow and energy consumption. Using existing flowmeters and power monitoring equipment to determine pump efficiency meant that the accuracy of measurement was no better than +/- 5%, which in the case of the larger pumps, related to a potential inaccuracy that could be costing $9,000 per pump per year (based on electricity cost of $0.10/kWh) .

GWRC then asked BTO Ltd to conduct a Level 3 Energy Audit, which was to carry out the recommendations of the Level 1 Energy Audit. One of the topics of the energy audit was to find a more accurate method for testing the performance of installed pumps. With a more accurate method of measuring pump performance, the benefits gained from various pump efficiency enhancements could be quantified accurately and repeatable performance measurements would provide the knowledge required to make informed energy/cost based decisions regarding the upgrade of the pumps.

Our investigations uncovered two companies that, by using new technology, had managed to adapt a technique previously restricted to test laboratory conditions, to be suitable for installed pumps. The technique referred to is the 'Thermodynamic' method of pump testing.

Late in 2007, BTO Ltd acquired the New Zealand licence from Robertson Technology Pty Ltd for using their thermodynamic technology to provide pump performance testing services using portable test equipment (P22P) for in-situ testing and for selling fixed pump performance systems (P22F) for permanent installation.

BTO Ltd used the P22P equipment to test various large pumps for GWRC and after showing that all the 630kW pumps at the Waterloo treatment and pumping station were below the original manufacturer's performance, one pump was selected as a candidate to test various performance enhancement techniques.

The chosen pump was disassembled and a mechanical health check carried out to determine the major areas for overhaul. The mechanical overhaul was completed, the pump re-assembled and then re-tested using the thermodynamic technique to determine the performance improvement from the mechanical overhaul. The next stage was the application of a propriety low friction coating to the internal surfaces of the pump, followed by another performance test using the thermodynamic technique to determine the performance improvement from the low friction coating.

This paper will describe the techniques and results of performance improvement at each stage of the process.

# 2 PUMP PERFORMANCE MEASUREMENT

Currently there are two principle methods for testing the performance of a pump. The most common approach is the traditional method that is well recognised in the industry, but can lack in accuracy and the second less know method known as the 'thermodynamic' method, which is a relatively new method and has the potential to provide much greater accuracy.

The traditional method is only known to be able to perform to AS2417-2001 Grade 1 and 2 Classes for measurement of uncertainties. The Thermodynamic method has the potential to perform to Precision Class under ISO 5198:1999 for measurement of uncertainties. These Classes and their range of 'Uncertainties' are listed below.

***Table 1 – Standards - Uncertainties of measurement***

| Permissable values of overall measurement uncertainties (+/- %) | | | |
|---|---|---|---|
| *Standard* | BS EN ISO 5198:1999 | AS2417-2001 (ISO 9906:1999) | |
| *Class* | Precision | Grade 1 | Grade 2 |
| PARAMETER | *Class A* | *Class B* | *Class C* |
| **Flow rate** | **1.5** | **2.0** | **3.5** |
| **Pump efficiency** | **2.25** | **3.2** | **6.4** |

Pump parameters are summarised by the following equation:

$\boldsymbol{\eta_p . M_E . P_W = q . \rho . g . H}$ ..........(1)

The left-hand side of equation (1) is the electrical power (joules per second) applied to the fluid, after losses in the motor drive and pump, where: -

$\boldsymbol{\eta_p}$ is the pump efficiency (expressed as a fraction)

$\boldsymbol{M_E}$ is the motor and drive efficiency (expressed as a fraction)

$\boldsymbol{P_W}$ is the electrical power to the motor (in watts)

The right-hand side of equation (1) is the energy per second imparted to the fluid, which also has the units of watts (joules per second): -

**q** is flow rate, in m3/s

**ρ** is the fluid density, in kg/ m3, a function of fluid temperature and pressure

**g** is the acceleration due to gravity, in $m/s^2$

**H** is pump total head, in m

The terms **ρ, g, H,** $\boldsymbol{P_W}$ and $\boldsymbol{M_E}$ are common to all pump test methods, with **ρ** and **g** being obtainable from reference tables.

To assess fully the energy benefit through efficiency gains achieved by the pump performance enhancement techniques used at each stage of this study, the most accurate and repeatable measurement method available is used.

## 2.1 TRADITIONAL METHOD

The traditional method of pump performance testing uses the pumps measured head, flow and input power to calculate the efficiency. The major drawback with this method is that it depends largely on the accuracy of the devices used to measure the head, flow and power input.

On-site constraints often make it difficult to accurately measure pump efficiency under installed conditions by the same method that pump manufacturers traditionally use for works tests. In this technique, pump efficiency is calculated from equation (1) as follows:

$$\eta_p = q.\rho.g.H / M_E.P_W$$

This requires measurement of 3 criteria, flow, head and power. The accuracy of the pump efficiency measurement is determined by the errors in the measurement of $\mathbf{q}$, $\mathbf{H}$, $\mathbf{P_W}$, and $\mathbf{M_E}$. The accuracy of the Energy/Quantity pumped is determined by the errors in $\mathbf{q}$ and $\mathbf{P_W}$.

In practice, the flow rate (**q**) is the most difficult to determine accurately. Many pumps do not have accurate, individual flow meters, which are high cost items, especially for larger diameter pipes, and can be difficult or impossible to install, maintain, and carry out calibration checks on-site. Flow meter accuracy can be dependent on installed straight, clear pipe lengths prior to and after the measuring device, the pump's operating point and other factors, such as build-up of debris in pipes or on sensors. Often, just the total flow from the station or from each group of pumps is measured and pipe installations are often compromised in the interest of minimising civil costs. Conventional flow meters, either installed or strap-on, can have an accuracy of +/-5% or worse, and this will lead to corresponding errors in the pump efficiency and Energy/Quantity measurements. These errors are so large that the method is impracticable for accurate measurements of energy savings or for pump refurbishment or system control decisions.

For example, if each of the 3 measuring devices (flow/head/power) is 95% accurate, this would equate to almost a 9% uncertainty (i.e. the quadrate of the individual uncertainties) in the overall efficiency calculation. In turn, if a calculation determined that a pump was 87% efficient, the uncertainty would be ± 9%, meaning that the pump could actually be as low as 79% efficient. Such magnitudes of inaccuracies undermine the benefits of long term pump efficiency testing as one is not be able to reliably detect/assess pump deterioration or improvements in efficiency when work is carried out on a pump.

## 2.2 THERMODYNAMIC METHOD

The modern thermodynamic method evolved primarily from work carried out by the National Engineering Laboratory and the University of Glasgow, in the UK, in the 1960's (1), and in parallel by Austin Whillier, at the Mining Research Laboratory in South Africa. From this work, International standards have been developed (2, 3).

The thermodynamic method uses the principal that virtually all of the efficiency loss in a pump is transferred to heat and absorbed by the water/liquid it is pumping. This means that a measured difference between the input water temperature and the outlet water temperature can effectively indicate the efficiency of the pump. For example, a small difference between the inlet and outlet temperature of a pump indicates the pump is operating at a high efficiency and visa-versa.

The theoretical background to the thermodynamic method is primarily in the public domain. The performance of an instrument employing this method is largely determined by the design, accuracy and stability of the temperature probes. By this method, flow is not necessary to determine efficiency, however flow can be derived from knowing the other elements of the equation.

The flow rate (**q**) is determined from equation (1), rearranged:

$$q = \rho.g.H / \eta_p.M_E.P_W \quad \ldots\ldots\ldots(2)$$

The pump efficiency ($\eta_p$), is determined from changes in enthalpy (internal energy per unit mass), using temperature and pressure probes.

The uncertainty in $\eta_p$ is primarily due to the uncertainty in differential temperature measurements.

The thermodynamic method for determining pump efficiency relies primarily on the measurement of two parameters, (a) the differential temperature, **dT**, across the pump, and (b) the differential pressure, **dP**, across the pump.

The pump efficiency ($\eta_p$), is the ratio of two changes (in energy per unit mass), each comprising of enthalpy, kinetic energy and gravitational terms.

For pumps, $\eta_p = E_H/E_M$ , where;

$E_H$ is the hydraulic energy per unit mass of fluid, and

$E_M$ is the mechanical energy per unit mass.

In the absence of minor corrections for the kinetic energy and gravitational terms,

$E_H = dP/\rho$ and $E_M = a.dP + Cp.dT$ , where;

**Cp** is the specific heat capacity at constant pressure (change of enthalpy with temperature at constant pressure),

**a** is the isothermal coefficient (change of enthalpy with pressure at constant temperature), and

**ρ** is the fluid density, a function of fluid temperature and pressure.

***(Data for these three parameters are obtained from tables in international standards).***

The thermodynamic method determines pump efficiency to a high accuracy, since it is essentially measuring losses. For example, suppose a pump is 80% efficient and that both the conventional and thermodynamic methods had an error of 5% of the measurement quantity. Then the error in pump efficiency by the conventional technique would be 5%. However, the error by the thermodynamic method would be 1%, since the measurement error occurs in measuring the losses of 20% and 5% of 20% is 1%.

### 2.2.1 UNCERTAINTIES OF THE THERMODYNAMIC PUMP TEST METHOD

For the instrument designer, the main challenge of the thermodynamic method is the stable and accurate measurement of the differential temperature (**dT**), which will vary with total head and pump efficiency. Low head pumps give lower differential temperatures and pumps with lower efficiencies will produce higher differential temperatures. Temperatures are typically measured in millikelvin (mK), i.e. thousandths of a degree.

Table 2 shows **dT** as a function of hydraulic efficiency and head, at a water temperature of 10 °C. The signal increases slightly with water temperature.

***Table 2 - dT (mK) at 10 °C***

| **Head, m of water** | **Hydraulic efficiency, %** | | |
|---|---|---|---|
| | 70% | 80% | 90% |
| 25 m | 26 mK | 16 mK | 8 mK |
| 50 m | 53 mK | 32 mK | 16 mK |
| 100 m | 106 mK | 64 mK | 35 mK |

Table 3 shows the effect on the efficiency measurement of an uncertainty in **dT** of 1 mK.

***Table 3 - % change in hydraulic efficiency, for a 1mK variation in dT, at 10°C***

| **Head, m of water** | **Hydraulic efficiency, %** | | |
|---|---|---|---|
| | 70% | 80% | 90% |
| 25 m | 1.2 | 1.4 | 1.5 |
| 50 m | 0.6 | 0.6 | 0.8 |
| 100 m | 0.3 | 0.3 | 0.4 |

Robertson Technology Pty Ltd has developed technology for measuring the differential temperature (**dT**) across a pump to an accuracy of better than 1mK, with long-term stability over periods of years. This technology has been applied to both portable and permanently installed (fixed) thermodynamic pump performance monitors and has been utilised for performance measurements of water turbines.

## 2.3 ROBERTSON TECHNOLOGY THERMODYNAMIC TEST EQUIPMENT

For this study, the thermodynamic equipment from Robertson Technology was selected as the preferred equipment because of the excellent repeatability, less frequent re-calibration requirements and ease of use.

Robertson Technology thermodynamic performance equipment is available as either portable (P22P) or fixed (P22F) systems. Portable units are used for investigative work and regular monitoring, while the fixed installations provide on-line predictive monitoring of critical and/or large energy users. These large energy users can be centrifugal pump, blowers or hydro-turbines.

Long-term tests on temperature probes in the portable units (P22P) have shown no change in **dT** (within experimental error <0.2 mK) over a four-year period. With fixed units (P22F), a continuous condition monitoring system fitted to 7 high power pumps (1-3 MW each) has shown similar stability over the 3 years since installation.

For additional assurance of long-term stability, two temperature sensors are included in each temperature probe. The software detects any discrepancies between the two sensors, giving a warning if one of the sensors starts to drift.

Pressure probes are provided with in-built temperature sensors, to provide 0.1% accuracy (relative to full-scale) over a wide fluid temperature range. These probes have a long-term stability of typically 0.1% of full scale per year.

Electrical power to the pump motors is measured to an accuracy of 0.25%, plus errors in current and potential transformer ratios. Typical field accuracy for pump efficiency is shown in Table 4.

***Table 4 - Uncertainty in pump efficiency measurements in typical field conditions***

| **Head** | **20 m water** | **40 m water** | **70 m water** | **100 m water** |
|---|---|---|---|---|
| **Pump efficiency (%)** | Uncertainty in pump efficiency (%), at 95% confidence level | | | |
| **55** | 0.8 | 0.5 | 0.5 | 0.3 |
| **65** | 1.0 | 0.6 | 0.5 | 0.4 |
| **75** | 1.3 | 0.7 | 0.6 | 0.4 |
| **85** | 1.5 | 0.9 | 0.7 | 0.5 |

Typical <u>repeatability</u> for this method of testing is **0.2%**

Each P22P pump monitor system has two temperature probes, two pressure probes, one power meter and a software program. These are connected together using digital network technology that can utilise radio transmitters to allow the computer terminal to be situated away from the noises of a pump room. Figure 1 shows a schematic of a fixed pump monitor.

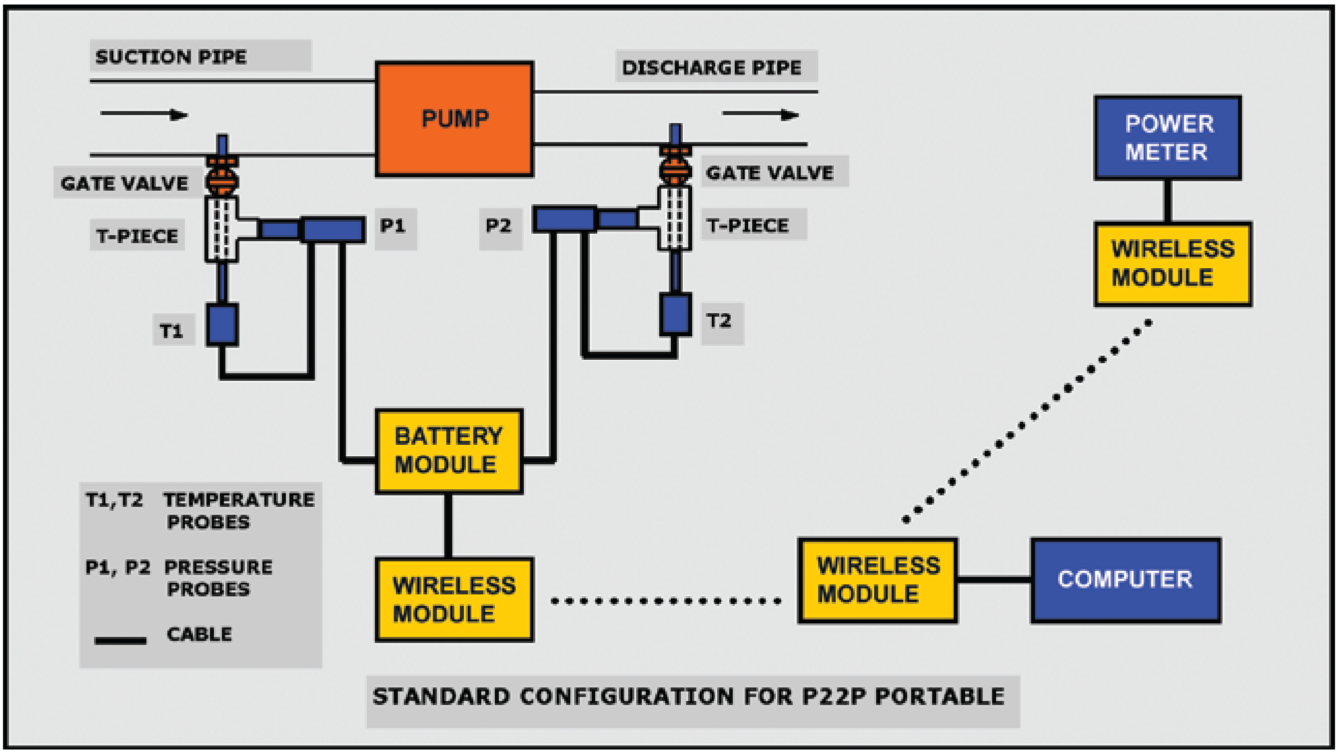



***Figure 1 - Schematic of P22 portable pump performance monitor***

# 3 INITIAL PUMP PERFORMANCE TEST

The Waterloo Treatment and Pumping Station in Lower Hutt, provides pH correction and fluoridation of borehole water from the Hutt aquifer before pumping to the distribution system using eight pumps. There are two sets of pumps that deliver treated water to the Naenae and Gracefield reservoirs, but the majority of the water is pumped into the Wellington distribution system by three 630kW pumps.

Although BTO Ltd tested all 8 pumps at the Waterloo site, a Wellington pump was chosen for this study because it is the largest and operated for greatest time, therefore would benefit most from any improvements to performance. Wellington Pump No.1 was chosen as the candidate for this trial.

Prior to each test, the calibration of each instrument was verified.

## 3.1 CANDIDATE PUMP

The Wellington pumps were installed in 1998 and have run for approximately 41,000 hours each. In that time, we can estimate that the pump has delivered 44,000,000$m^3$ of treated water at an operating head of about 93m. We estimate that the power consumption of each Wellington pump is now approximately 1.8GWh per year.

There has been no previous maintenance work that could have improved the efficiency of this pump, so the deterioration in performance measured from the initial (benchmark) test is the result of the duty described above.

| | | **Wellington Pump No.1** | | | |
|---|---|---|---|---|---|
| **Manufacturer** | **Model No.** | **Serial No.** | **Impeller Dia** | **Capacity (ML/d)** | |
| KSB | Omega V 250-600 A 1997 | 5-L61-783 499/1 | | 30.24 MLD (350 l/s) | |
| **Liquid** | **Temp (°C)** | **S.G.** | **Total Head (m)** | **Shaft Power** | **R.P.M.** |
| Water | Ambient | 1 | 125 | | 1480 |
| **Manufacturer** | **Serial No.** | **Fixed/Variable Speed** | **Output (kW)** | **Voltage (V)** | **Current (A)** |
| Helmke - DOR 409L-4-093 | K8792 | VSD | 630 | 400/690 | 1060/612 |
| **Frequency (Hz)** | **Phases** | **R.P.M.** | **Poles** | **Motor Efficiency** | **Drive Efficiency** |
| 50 | 3 | 1480 | 4 | 94.8% (@75%) | 97% |

***Table 5 – Manufacturers data for the Wellington pump***

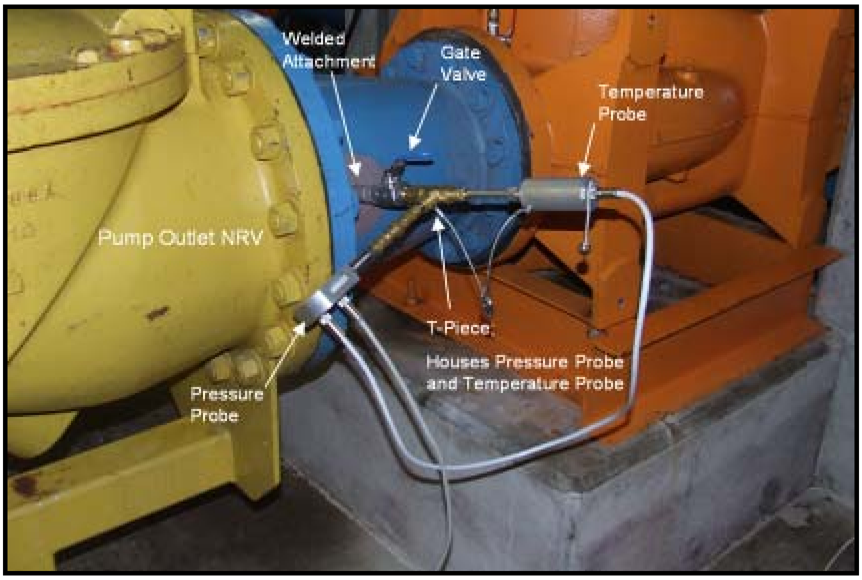
## 3.2 EQUIPMENT SET-UP

For installation of the temperature and pressure sensors, a 1/2”NPT tapping point is required on the suction and delivery side of the pump. Ideally, the tapping point should be at least 2 diameters away from the pump and before isolation and non-return valves.

BTO Ltd used the services of Aqua Engineering Ltd to install the required fittings at positions pre-determined by BTO Ltd. Using ‘hot-tap’ equipment, Aqua Engineering Ltd were able to install the fittings without shutting down the pump, which allowed operation to continue unhindered (Picture 1).

***Picture 1 – 'Hot-tapping' of pipe***

Pressure and temperature probes are inserted approximately 6cm into the flow. This distance is long enough to avoid any possible temperature stratification, but short enough to avoid vibration of the temperature probe tip (the temperature probes are so sensitive that they will detect temperature increase created by vibration at the tip), see Picture 2.




***Picture 2 – Instrumentation on delivery side of pump***

Fitting of the power monitor, normally involves the connection of voltage leads to all three phases and current CTs to two phases, but because of the cable arrangement for this drive, we were unable to connect the CTs. Fortunately, all the Wellington pumps are connected to variable speed drives (VSD), which have the facility to monitor the power input and the P22 software allows for manual input of these power values for each test.

After connecting the field instruments to the radio transmitter, the computer (located in the control room) was connected to the radio receiver, communications established and the software parameters checked ready to start the first test.

This set-up procedure remained the same for all of the tests in this study.

## 3.3 WELLINGTON PUMP 1 – BENCHMARK TEST

Initial testing showed that the original suction and discharge tapping points were too close to the pumps. Because of installation restrictions, the installed tapping points were around one-pipe diameter each side of the pumps, and ideally should be at least two pipe diameters. Two diameters is preferred in case vibration effects and/or fluid recirculation affect the temperature sensors. In an attempt to improve our test measurements, alternative suction and discharge tapping measurement points further from the pump were located.

Due to the discharge tapping point being located after the non-return valve, the measured discharge head of the pump would be slightly lower. The non-return valve losses were measured to be 0.031bar or 0.32m of head with the isolation valve fully open and the pump at 95% VSD setting (1425rpm). The Wellington pump discharge head ranged from 79m to 155m, therefore the losses associated with the non-return valve equated to 0.2-0.4% of the total discharge head. It was therefore assumed that this head loss was negligible.

Figure 2 shows graphically how the pump efficiency changed over the permissible flows. The maximum flow rate that could be obtained at the 100% VSD set point (1480rpm) was 33ML/d, which was torque limited within the VSD unit.

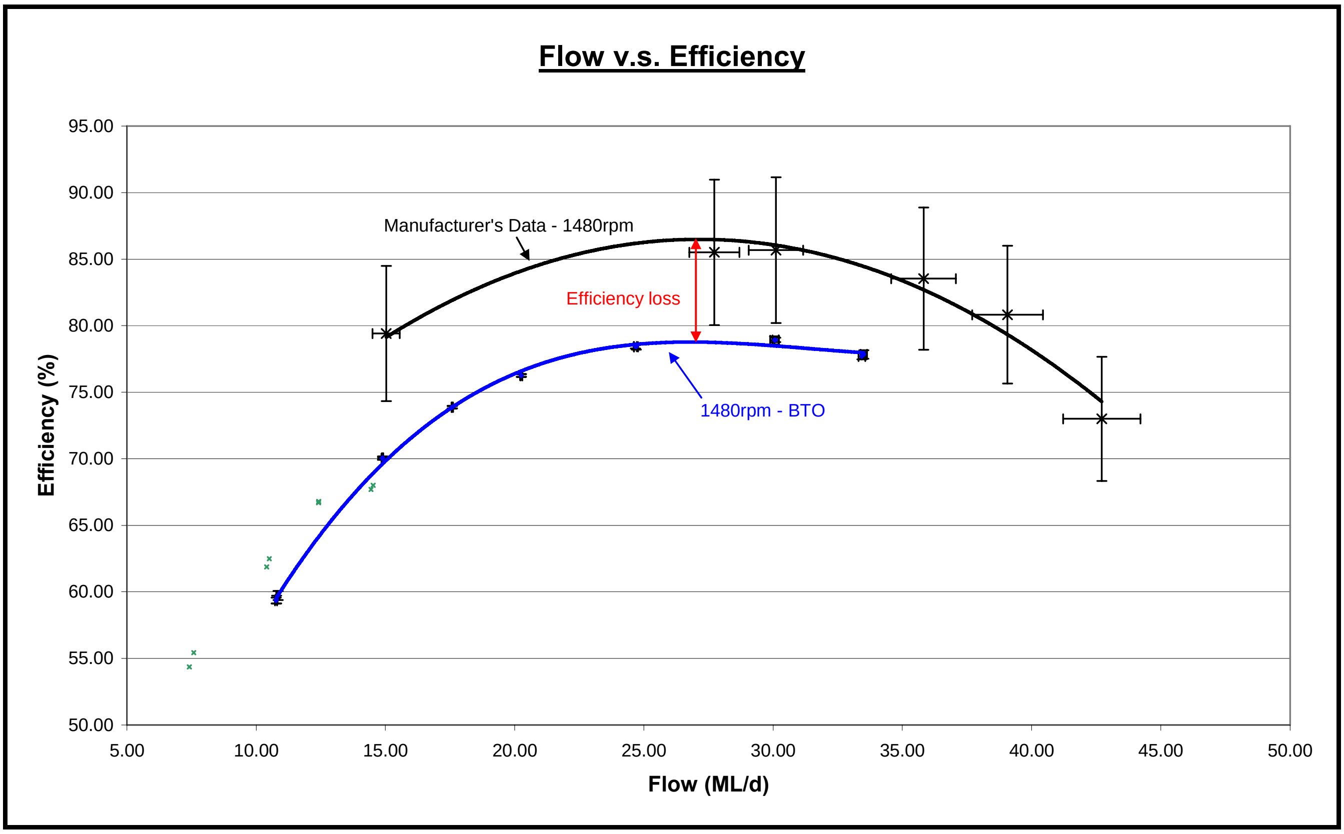



***Figure 2 - Wellington Pump 1 – Flow vs. Efficiency for various VSD set points***

The manufacturer's test was conducted by KSB in 1997 to 'class C' accuracy, which would indicate that the flow rate and pump efficiency is calculated to an accuracy of within +/-3.5% and +/-6.4% respectively (see error bars in Figure 2). The uncertainty in the original data is much greater than the uncertainty of the data obtained from the P22 test.

Figure 2 clearly shows the deterioration in pump efficiency since Wellington Pump No.1 was installed in 1997. A measure of the deterioration in pump performance can be estimated from the difference between the BEP of the manufacturer's data (86.5%) and the BEP of the thermodynamic testing at the same speed (79.1%). Therefore the estimated deterioration in performance of the pump in the first 10 years of service is **7.4%,** which is equivalent to an additional cost for power of about **$12,900/year** (based on power cost of $0.1/kWh and assuming even duty rotation).

# 4 PUMP PERFORMANCE ENHANCEMENT METHOD 1

For a pump that has run an estimated 41,000 hours, it was suspected that the wear rings might need to be replaced and so the first method of pump performance enhancement to be tested was a mechanical overhaul.

After removing the Wellington pump No.1 from service, the pump case was split for visual inspection.

## 4.1 VISUAL INSPECTION

After removing the Wellington pump No.1 from service, the pump case was split for visual inspection.

The epoxy surface treatment applied by the manufacturer to the internal surfaces of the casing had deteriorated over time to leave an intermittent covering. Picture 3 shows the state of the internal surfaces.

***Picture 3 - Surface condition of pump casing after 41,000 hours service***

***Picture 4 - Discharge vane tip of pump impellor***

***Picture 5 - Suction vane tip of pump impellor***

Further inspection revealed that significant cavitation damage had occurred to the volute wall just after the cutwater, which is probably caused by increased water speed due to a pressure differential that is sufficient to induce cavitation as it slows again on the other side of the cutwater.

***Picture 6 - Cavitation damage to the volute wall***

## 4.2 DETAILED INSPECTION

The pump casing and pump rotor were shipped to a local engineering company for detailed inspection and report. The inspection report consisted of tolerance checks, fatigue cracking detection, mechanical seal health check and others, but the most relevant to this study was the pump ‘wear ring’ tolerance.

The tolerance between the wear ring and the rotor was measured as 1.65mm to 1.7mm clearance, compared to the manufacturer’s recommendation of 0.9mm. So, over the operating life of 41,000 hours, the wear ring had lost 0.75/0.8 mm.

## 4.3 PUMP OVERHAUL PROCEDURE

From the detailed inspection report, GWRC engaged the same engineering company to carry out a full overhaul of the pump that included shaft balancing, reconditioning of mechanical seal etc, but the most relevant work to this study was the replacement of the pump wear rings. New bronze wear rings were manufactured to the manufacturers recommended tolerances of 0.45mm each side (0.9mm total).

The cost of the work was;

- Independent inspection - $2,300
- Pump overhaul work - $10,500
- Remove and re-install pump (internal labour resource) - $700

Of all the overhaul work, it is expected that the new ‘wear rings’ will make the major contribution to any improvement in pump performance. The estimated cost for replacing the wear rings only (including pump disassembly and reassembly was $6,000.

## 4.4 PUMP PERFORMANCE TEST AFTER MECHANICAL OVERHAUL

The re-assembled pump was returned to Waterloo PS and installed. The pump performance test was repeated, using Robertson Technology P22 equipment using the same test points and equipment as the first ‘benchmark’ test to give a repeatability of results to within 0.2%.

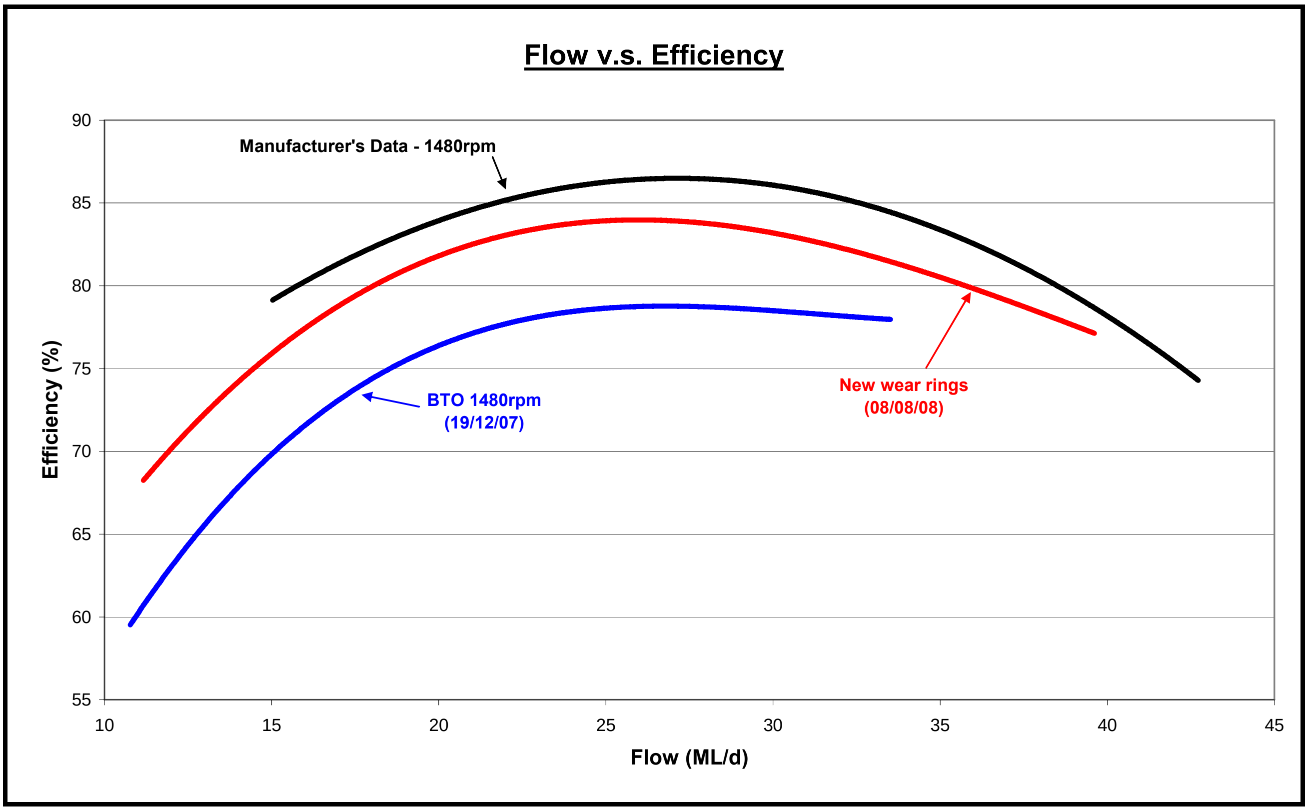



*Figure 3 - Wellington Pump 1 – Flow vs. Efficiency – Performance after Method No.1*

Figure 3 shows graphically how the pump characteristics have improved. Not only has the efficiency increased, but he curve has moved further to the right, which shows that there is more flow from the pump at the same speed.

The estimated efficiency improvement is about **5.3%** from the pump overhaul, which can be attributed in main to the installation of new wear rings. This improvement in efficiency is estimated to save electricity costs of about **$8,300/year** (assuming even duty rotation). Although this does not meet the manufacturer's original performance curve, it does fall well within the measuring tolerance of the original data.

### 4.5 PAYBACK PERIOD OF MECHANICAL OVERHAUL

The total cost, including internal labour, for the mechanical overhaul is in the region of $13,500. This cost does not reflect the true cost of the work relating to efficiency improvements, because while the pump was disassembled other maintenance work was carried out. A cost estimate for the efficiency related work only, including removal, re-instatement and internal labour would be in the region of $6000.

The following payback calculations are based on the cost estimate of $6000 for the efficiency related work only and two operational scenarios.

Because the refurbished pump is one of three and it is the only one to receive an overhaul, there are two ways to assess the payback period. The first is to run the refurbished pump continually as it is the most efficient and use the 'less-efficient' parallel pumps as assist and standby pumps. The second method is to rotate the duty of all the pumps equally.

By running the refurbished pump as the duty pump continually, the power saving would be about **$15,900**, which equates to a payback period of about **5 months**.

By rotating the duty of the refurbished pump with the other two, the power saving would be about **$8,300/year**, which equates to a payback period of about **9 months**.

## 5 PUMP PERFORMANCE ENHANCEMENT METHOD 2

The second method of pump performance enhancement tested was the application of a proprietary low friction coating. The 'Supermetalglide 1341' product from Belzona was chosen as the main surface coating, which claims "Performance gains range from 7% on new pumps to 40% on those in service".

### 5.1 APPLICATION OF LOW FRICTION COATING

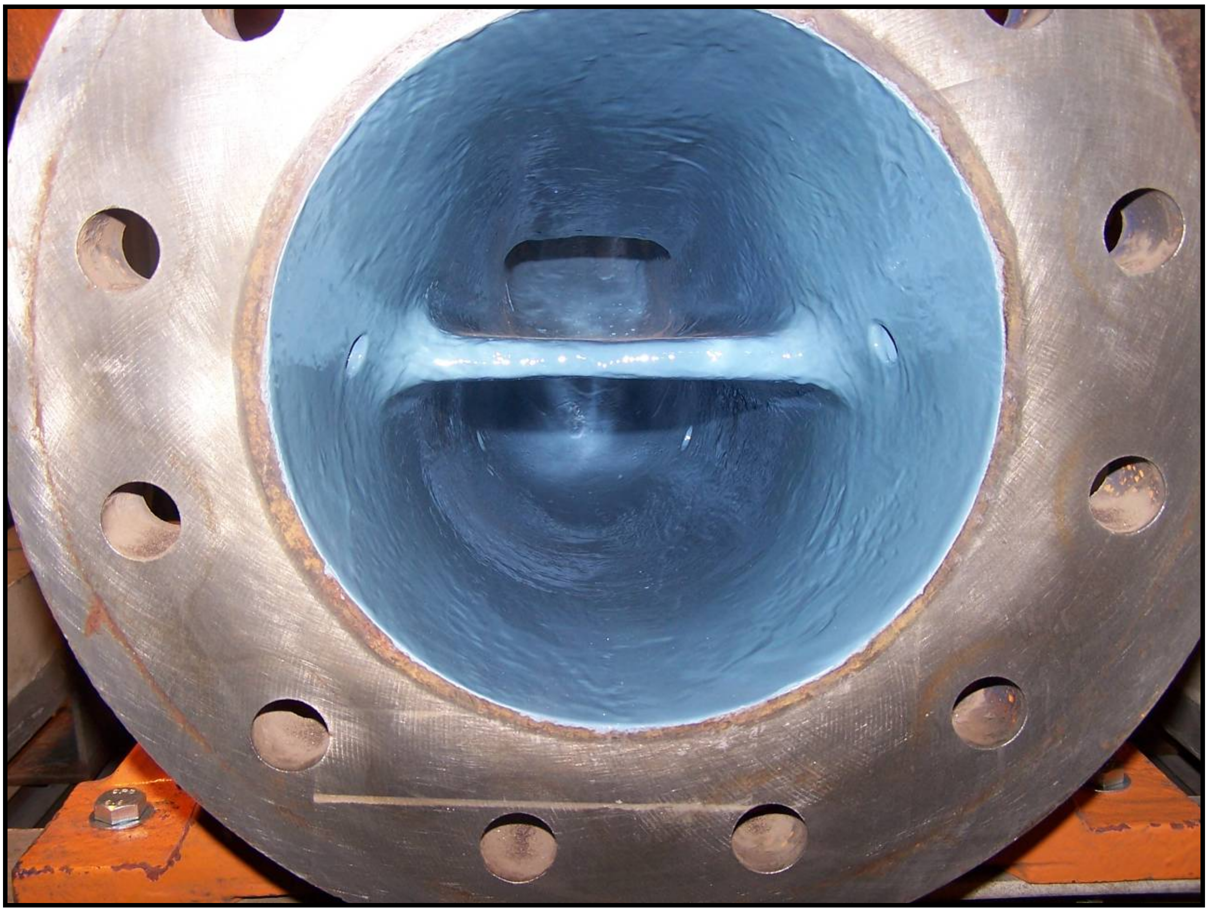
Wellington pump No.1 was removed from service and delivered to Aqua Engineering Ltd for preparation and application of the Belzona products. The application process consisted of the following steps;

- Shot-blast pump casing to remove residue of original surface coating,
- Dry casing over 24 hours to remove any residual moisture,
- Fill and dress areas of sever cavitation with Belzona 1111 Super Metal
- Surface Coating. Apply 2 x coats Belzona 1341 Supermetalglide

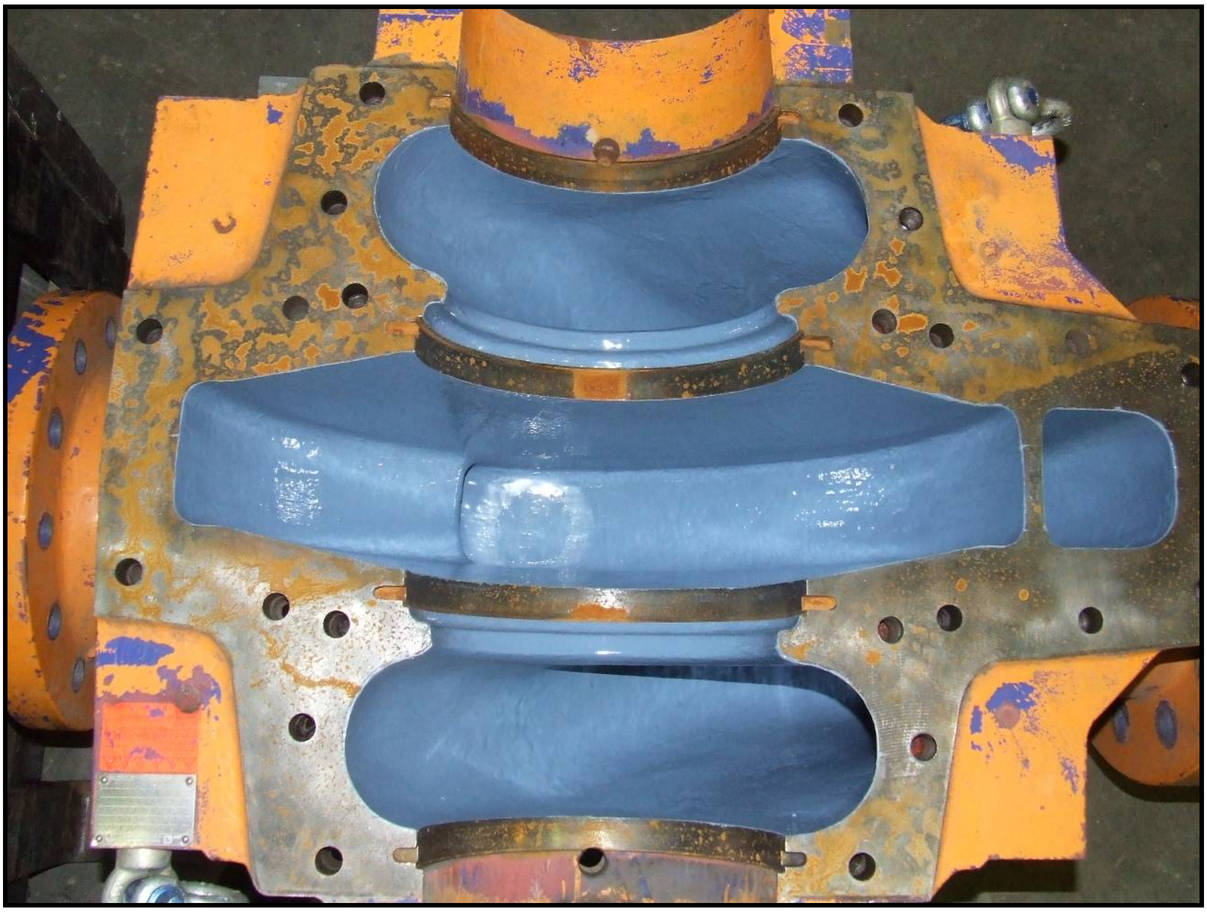
The following are a sample of photographs taken of the pump outlet and 'cutwater' during the preparation and coating process.

***Picture 7 - Cavitation damage to the original surface coating before shot-blasting***

***Picture 8 - Original surface coating removed by shot-blasting***

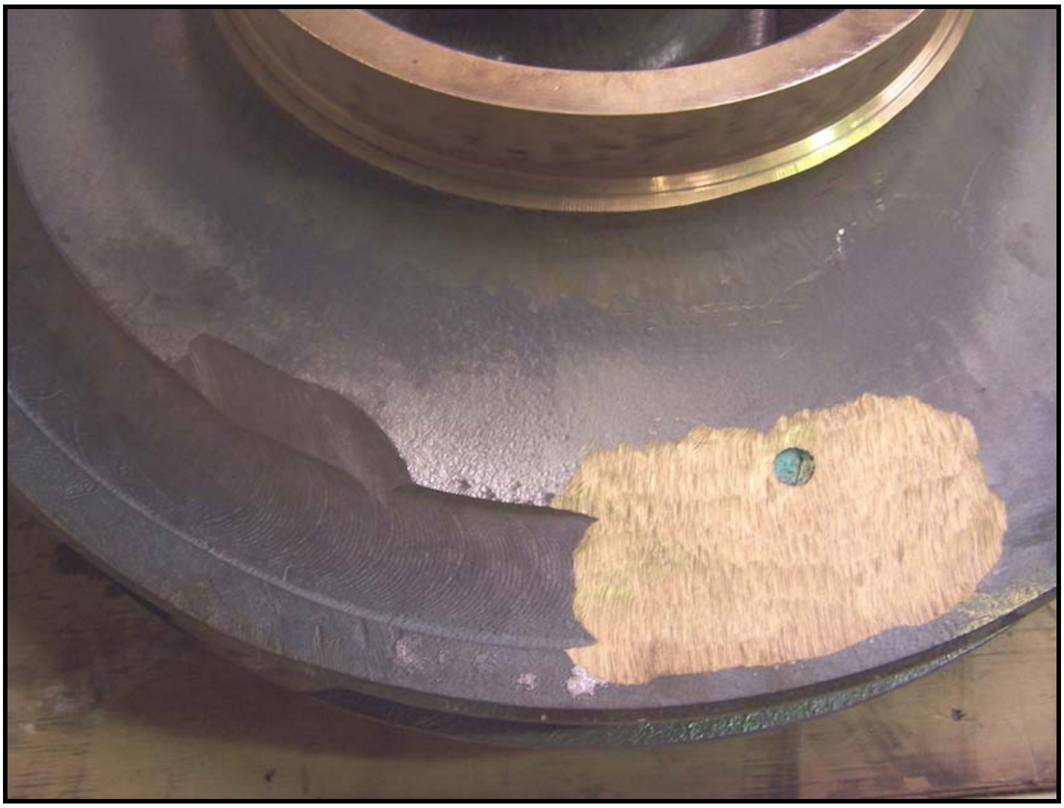


*Picture 9 – Final surface finish – Pump outlet volute*



*Picture 10 – Final surface finish – Pump casing*

In addition to the low friction coating applied to the pump casing, the pump rotor also received a surface treatment. The original casting of the rotor had various imperfections that were considered detrimental to pump performance. Picture 11 shows an example of a casting imperfection.



***Picture 11 – Casting imperfection in pump rotor***

Rough external areas of the rotor were removed with a hand grinder and then polished to a smooth finish. In comparison the internal rotor surfaces were reasonably smooth so no work was done in this area.

***Picture 12 – Casting imperfection in pump rotor***

## 5.2 PUMP PERFORMANCE TEST AFTER LOW FRICTION COATING

The re-assembled pump was returned to Waterloo PS and installed. The pump performance test was repeated, using Robertson Technology P22 equipment under the same conditions as the first and the second tests.

Figure 4 shows graphically how the pump characteristics have improved. Not only has the efficiency increased, but again the curve has moved further to the right, which shows that there is more flow from the pump at the same speed.

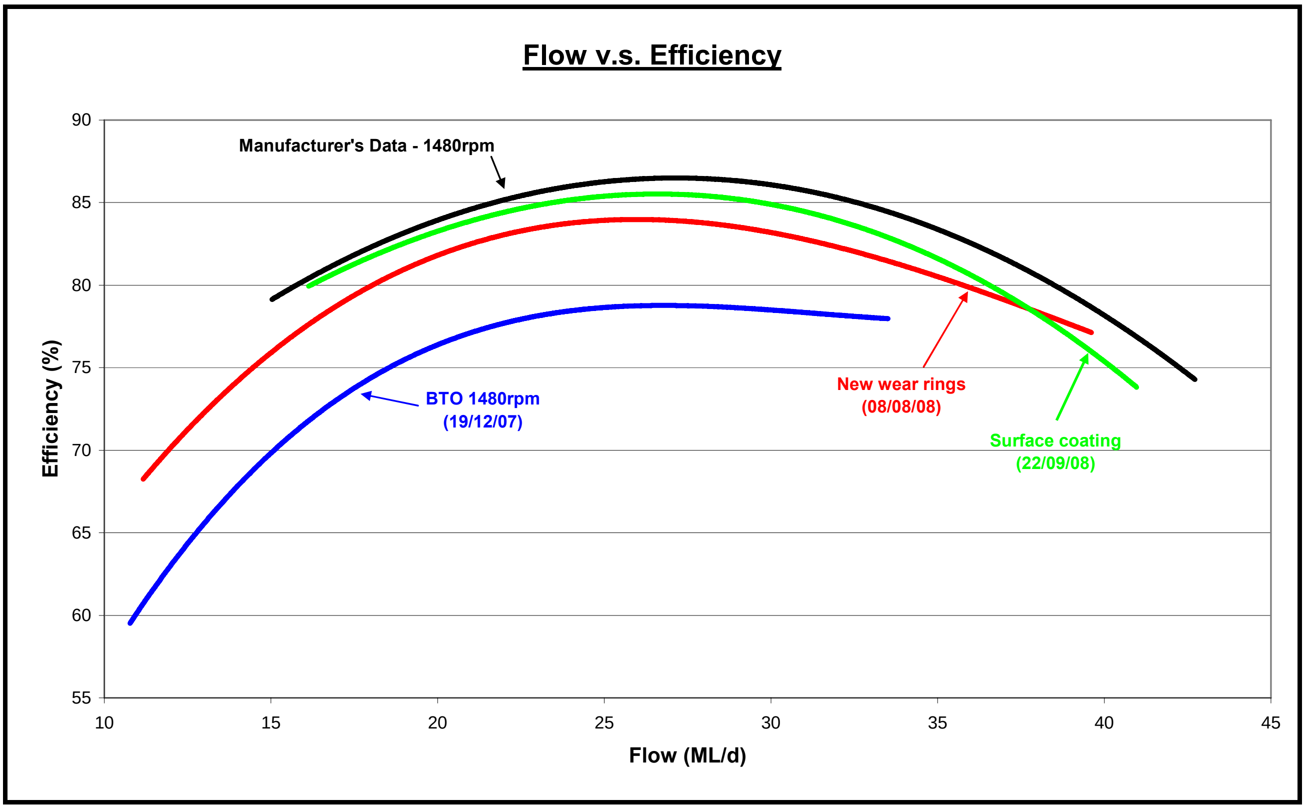



*Figure 4 - Wellington Pump 1 – Flow vs. Efficiency – Performance after Method No.2*

Figure 4 shows the efficiency improvement is about **1.6%** from the application of the low friction coating and rotor enhancement, which is equivalent to a saving in electricity costs of about **$2,900/year** (assuming even duty rotation). Although this does not meet the manufacturer's original performance curve, it does fall well within the measuring tolerance of the original data. The new curve is also much more representative of the original curve.

## 5.3 PAYBACK PERIOD OF LOW FRICTION COATING

The total cost for the application of a low friction coating, including removal, re-instatement and internal labour would be in the region of $8,500. This value will be used to calculate the approximate payback period.

As before, the refurbished pump is one of three and the only one to receive the low friction coating. Therefore, there are two ways to assess the payback period. The first is to run the refurbished pump continually as it is the most efficient and use the 'less-efficient' parallel pumps as assist and standby pumps. The second method is to rotate the duty of all the pumps equally.

By running the refurbished pump as the duty pump continually, the power saving would be about **$5,600**, which equates to a payback period of about **18 months**.

By rotating the duty of the refurbished pump with the other two, the power saving would be about **$2,900/year**, which equates to a payback period of about **35 months**.

# 6 CONCLUSIONS

Our tests found that by replacing the worn pump wear rings with new wear rings, the amount of water re-circulating in the pump is reduced and the efficiency improved by 5.3%, which is within the measurement tolerance of the original tests. For a 630kW pump the payback for a mechanical overhaul is estimated to be 9 months based on normal duty rotation and 5 months, based on dedicated duty.

Furthermore, the low friction coating improved the efficiency of the pump by a further 1.6%. This method has an estimated payback of 35, months based on normal duty rotation and 18 months, based on dedicated duty.

Applied together, these pump performance enhancement techniques improved the efficiency of the pump by nearly 7%, which could provide an estimated annual saving of over $11,000 on equal duty rotation.

It is well known that the efficiency of a pump will deteriorate over time due to normal ‘wear and tear’, but this wear rate will be dependant on the conditions that the pumps operates within. In the current New Zealand and world climate of increasing energy costs and climate change, continual efficiency monitoring or regular efficiency testing offers owners of large energy consuming equipment the ability to be able to determine the optimum time for overhaul or application of performance enhancement techniques.

With the thermodynamic measuring method, an accurate and reliable method is now available for use on site, thus further reducing the cost of pump removal for physical inspections etc.

## REFERENCES

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