Technical paper presented at Ozwater 05

(Annual conference of the Australian Water Association)

Brisbane, 8-12 May, 2005

o5219

Measurement of Pump Efficiency and Flow by the Thermodynamic Method

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EXECUTIVE SUMMARY

Continuous or regular pump performance monitoring provides energy and environmental savings, improves plant and process reliability, and reduces maintenance costs. On-site measurements of pump efficiency and flow rate can be provided by the thermodynamic method, without the need for separate flow meters. Pump efficiency is determined from the differential temperature and pressure across the pump. Flow rate is derived from the efficiency and pump input power. The uncertainties in the measurements depend primarily on the accuracy of the differential temperature measurement.

Robertson Technology Pty Ltd has developed techniques for measuring the differential temperature across a pump to an accuracy of better than 1mK, with long-term stability. This technology has been utilised in both portable and installed thermodynamic pump performance monitors. The paper compares the conventional and thermodynamic methods and summaries the accuracy that can be obtained in practice with the latter, based on field experience in the water industry with pumps operating in the power range 10 kW to 5 MW. The instrumentation can also be used for turbines.

INTRODUCTION

Using the principle of conservation of energy, pump parameters are summarised by the following equation:

$$\mathbf{n.M_E.P_w = q.\rho.g.H} \quad (1)$$

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

- **n** is the hydraulic efficiency (expressed as a fraction)
- $\mathbf{M_E}$ is the motor and drive efficiency (expressed as a fraction)
- $\mathbf{P_w}$ is the electrical power to the motor (in watts)

The factor $\mathbf{n.M_E}$ is known as the 'Overall Efficiency'.

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

- **q** is flow rate, in m^3/s
- **ρ** is the fluid density, in kg/ m^3, and is a function of 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,** $\mathbf{P_w}$ **and** $\mathbf{M_E}$ are common to both 'conventional' and thermodynamic methods, with **ρ** and **g** being obtainable from reference tables.

The 'conventional' method of pump testing, commonly employed by pump manufacturers during acceptance tests, can be used when it is possible to measure the flow rate, **q,** to the requisite accuracy, using a flow meter. Then equation (1), rearranged, gives the method for obtaining pump efficiency:

$$\mathbf{n = q.\rho.g.H / (P_w.M_E)} \quad (2)$$

The accuracy of the conventional pump efficiency measurement is largely determined by the errors in the measurement of **q, H, P_W, and M_e**. The uncertainties add linearly. In practice, the accuracy of **q** is usually the limiting factor in the laboratory. On-site measurement by the conventional method requires a flow meter, but in many instances this is precluded by the layout of the pipe work. Where flow meters are fitted, accurate volumetric testing may be required for on-site calibration checks.

In the thermodynamic method, the pump efficiency, **n,** is determined from changes in enthalpy (energy per unit mass), using temperature and pressure probes. The calibration of these probes can be readily checked on-site. The uncertainty in **n** is discussed in later sections. The flow rate, **q,** is determined from equation (1), rearranged:

$$\mathbf{q = n.M_E.P_w / (\rho.g.H)} \quad (3)$$

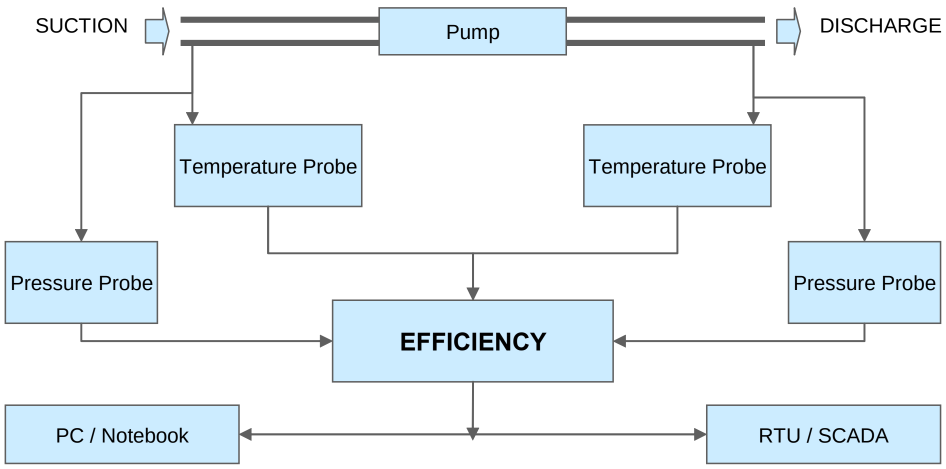
Thus flow rate can be measured thermodynamically without the need for a separate flow meter.

The thermodynamic method can measure 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 losses of 20% (100-80) are measured to 5% accuracy, and 5% of 20% is 1%.

THE THERMODYNAMIC METHOD

The thermodynamic method was largely developed in the 1960s in the UK (Universities of Glasgow and Strathclyde, and the National Engineering Laboratory), France (Electricite de France) and South Africa (Dr Austin Whillier, Chamber of Mines) [(1)]. This work led to ISO [(2)] and IEC [(3)] standards and a UK Code of Practice [(4)]. Instrumentation has improved with recent advances in technology, and portable units for investigative work and regular monitoring, or fixed installations for on-line predictive monitoring, are both now available.

Figure 1 Schematic of the thermodynamic method



The thermodynamic method for pump efficiency and flow rate relies primarily on the measurement of three parameters: (a) the differential temperature, **dt**, across the pump,

(b) the differential pressure, **dp**, across the pump, and (c) the electrical input power, $\mathbf{P_w}$. The theoretical background to the thermodynamic method is primarily in the public domain, being documented in ISO and other standards. The performance of an instrument employing this method is largely determined by the design, accuracy and stability of the temperature and pressure probes.

Pump (or turbine) efficiency is primarily measured by temperature and pressure probes. Flow rate is derived from efficiency and power input. For pumps with unequal pipe diameters at the tapping points, the power measurement is used to derive a relatively small velocity-dependent correction.

For pumps, the main equations and thermodynamic data that we use are those appearing in the International Standard: BS EN ISO 5198:1999 "Centrifugal, mixed flow, and axial pumps- Code for Hydraulic performance tests - Precision class". This standard is referred to for precision grade tests by Australian Standard AS2417 – 2001 "Rotodynamic pumps – Hydraulic performance acceptance tests – Grades 1 and 2".

Precision class tests were formerly referred to as Class A. Engineering class 1 and class 2 tests (former classes B and C) will be the subject of a further international standard, and are presently dealt with in ISO 2548 and ISO 3555. For turbines, the applicable standard is IEC60041:1991

The hydraulic efficiency, **n**, is the ratio of two changes in energy per unit mass, each comprising of enthalpy, kinetic energy, and gravitational terms. $\mathbf{n = E_H/E_M}$ for pumps, and $\mathbf{E_M/E_H}$ for turbines, where $\mathbf{E_H}$ is the hydraulic energy per unit mass of fluid, and $\mathbf{E_M}$ is the mechanical energy per unit mass.

$\mathbf{E_H}$ is given by $\mathbf{E_H = dp/\rho + (U_2^2 - U_1^2)/2 + g(Z_2 - Z_1)}$

$\mathbf{E_M}$ is given by $\mathbf{E_M = a.dp + c_p.dt + (U_2^2 - U_1^2)/2 + g(Z_2 - Z_1)}$

For the enthalpy terms:

$\mathbf{c_p}$ 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;

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

Data for the above three parameters are obtained from the tables in ISO 5198.

For the kinetic energy term:

$\mathbf{U_1}$ and $\mathbf{U_2}$ are the fluid velocities at the inlet and outlet measurement positions, respectively.

U = q/A, where **A** = cross-sectional area

For the gravitational term:

$\mathbf{Z_1}$ and $\mathbf{Z_2}$ are the elevation of the inlet and outlet measurement positions, respectively, relative to a datum.

Having measured the efficiency, flow rate **q** is obtained from the conventional pump performance equation

$$\mathbf{q = n.M_E.P_W / (\rho.g.H)} \qquad (3)$$

where **H** is pump total head, given by

$$\mathbf{H = dp/\rho.g + (U_2^2 - U_1^2)/(2.g) + (Z_2 - Z_1)}$$

Differential temperature

For the instrument designer, the main challenge is the stable and accurate measurement of **dt**, the differential temperature, which is the main signal. The differential temperature will vary with total head and pump efficiency. Low head pumps give lower differential temperatures. Pumps with lower efficiencies give higher differential temperatures. Temperatures are typically measured in millikelvin (mK), i.e. thousandths of a degree. It is possible to compensate for short-term drift in **dt** by swapping the probes and taking an average measurement. This is time consuming and is not an option with fixed systems for continuous monitoring. It is very desirable that the sensors measuring **dt, dp, and** $\mathbf{P_w}$ be both accurately calibrated, and have good long-term stability.

For fixed systems, the specification for **dt**, to obtain an efficiency measurement to an accuracy of 1% for the low head pumps, is that **dt** should be accurate to within 1 mK for a period of 1 year. It is reasonable to assume that initially, calibration checks on **dt** and **dp** will be carried out at 6 months and 12 months, and thereafter at yearly intervals.

Table 1 Changes in efficiency for 1mK variation in differential temperature, at different pressures

dp	dt	Change in dt (mK)	Hydraulic efficiency (n)	% change in n
10 bar (102.3m)	100 mK	0	0.746	
		1	0.744	0.3
5 bar (51.1 m)	50 mK	0	0.746	
		1	0.741	0.6
2 Bar (20.5 m)	20 mK	0	0.746	
		1	0.735	1.5

Table 1 illustrates the changes in efficiency due to a variation in 1 mK in the measured differential temperature across a water pump operating in normal ambient temperatures, for a range of differential pressures. In our experience, an accuracy of better than 1 mK is generally achieved. ISO 5198 (section 11.6) specifies an accuracy of 1 to 20 mK, according to the pump total head and the temperature of the water.

Long-term tests on the temperature probes have typically shown no change in **dt** within experimental error (0.25 mK) over a two-year period. We are not restricted to calibrating two probes together for **dt**, and indeed for some applications (e.g. multiple probes for very large pipes, or testing pumps in series) we have calibrated more. For additional assurance of long-term stability on fixed units, we include two temperature sensors in each temperature probe. Any discrepancies between the two sensors are detected by the software and are indicative of drift of one of the sensors.

Differential pressure

Standard pressure sensors are available from many manufacturers. Typically the accuracy may be 1 to 2% over the temperature range 0 to 40ºC. Most of the inaccuracy is due to temperature effects. IS0 5198 (section 11.6) specifies better than 0.1% accuracy for precision manometers for the thermodynamic method. To achieve this, we have designed pressure probes with in-built temperature sensors, to provide 0.1% accuracy over a wide fluid temperature range. Table 2 illustrates the % change in hydraulic efficiency for 1% pressure changes. It can be seen, for this typical situation, that a 1% uncertainty in differential pressure leads to an uncertainty in efficiency of 0.3%. The ISO specification of 0.1% accuracy will typically result in an uncertainty in the measured efficiency of 0.03%.

Table 2 Changes in efficiency for 1% variation in differential pressure, at different pressures

dp	dt	Change in dp (%)	Hydraulic efficiency (n)	% change in n
10 bar (102.3m)	100 mK	0	0.746	
		1	0.748	0.3
5 bar (51.1 m)	50 mK	0	0.746	
		1	0.748	0.3
2 Bar (20.5 m)	20 mK	0	0.746	
		1	0.748	0.3

Flow rate

The uncertainty in the flow rate, calculated from equation (3), will be somewhat higher than that for the hydraulic efficiency measurement, due to uncertainties in the following parameters:
(a) motor efficiency (typically 0.01 to 1%, depending on the availability of manufacturer's data),
(b) electrical power measurement (typically 0.25%),
(c) the accuracy of the current transformers and, for high voltage pumps (> 600 V), potential transformers (typically 0.5%).

Tapping points

The following remarks apply to 'cold-fluid' applications (0 – 60 ºC). Other arrangements may need to be made for higher temperature fluids, for safety reasons.

One tapping point is required on each side of the pump. Tee-pieces allow connection of both the temperature and pressure probes to the same tapping point (see Figure 2).

Figure 2 Temperature and pressure probe fitted to a tapping point

15 mm (½ inch BSPT / ISO) tapping points are required on the inlet and outlet of each pump to be tested, fitted with gate or ball valves to allow insertion of 9.53 mm diameter temperature probes or thermowells. Thinner temperature probes, 6 mm in diameter, have been developed for use with thermowells. They can also be used with 3/8 inch BSPT / ISO

Tee-pieces and tapping points, which are sometimes already fitted to the pipe work. The 9.53 mm and 6 mm diameter probes can be inserted when the pump is running, up to a maximum pressure of about 15 bar (153 m of water) or 30 bar (306 m of water) respectively. Probes can be used at higher pressures if they are inserted with the pump not running.

Pressure probes can be connected via a ring manifold on separate tappings, if required, to check for a uniform static pressure distribution. However, note from Table 2 that the requirement for a ring manifold is not so great for the thermodynamic method as it is with the conventional method, as a 1% error in the pressure measurement will lead to a smaller error, 0.3%, in the pump efficiency measurement.

Tapping points should ideally be about two pipe diameters from the pump flanges, and fitted into straight inlet and outlet measuring sections of equal diameter. The reasons for this are as follows. If the inlet tapping is closer to the pump, water heated by the impeller may re-circulate back to the inlet measuring position, particularly at flow rates well below the best efficiency flow rate for the pump, and lead to erroneous temperature measurements. We have however made accurate measurements on efficient pumps at full flow with tapping points on the pump flanges, so there can be exceptions. On the pump outlet, the distance of two pipe diameters allows the turbulent flow from the impeller to thoroughly mix, creating a uniform temperature over the pipe cross-section at the measurement point.

Where it is not possible to have equal pipe diameters, the tappings can be fitted in straight lengths of pipe of unequal diameters. The water velocity is then not the same at the inlet and outlet measurement positions, and corrections for two velocity dependent effects, velocity head and viscous heating, are then made in software. Viscous heating is the temperature rise in the temperature probe due to the kinetic energy dissipated in the stationary probe by the moving fluid. The correction terms are proportional to the square of the velocity, so given a choice, it is better to fit tapping points in larger diameter pipes, with lower water velocity, to minimise any uncertainty in the correction.

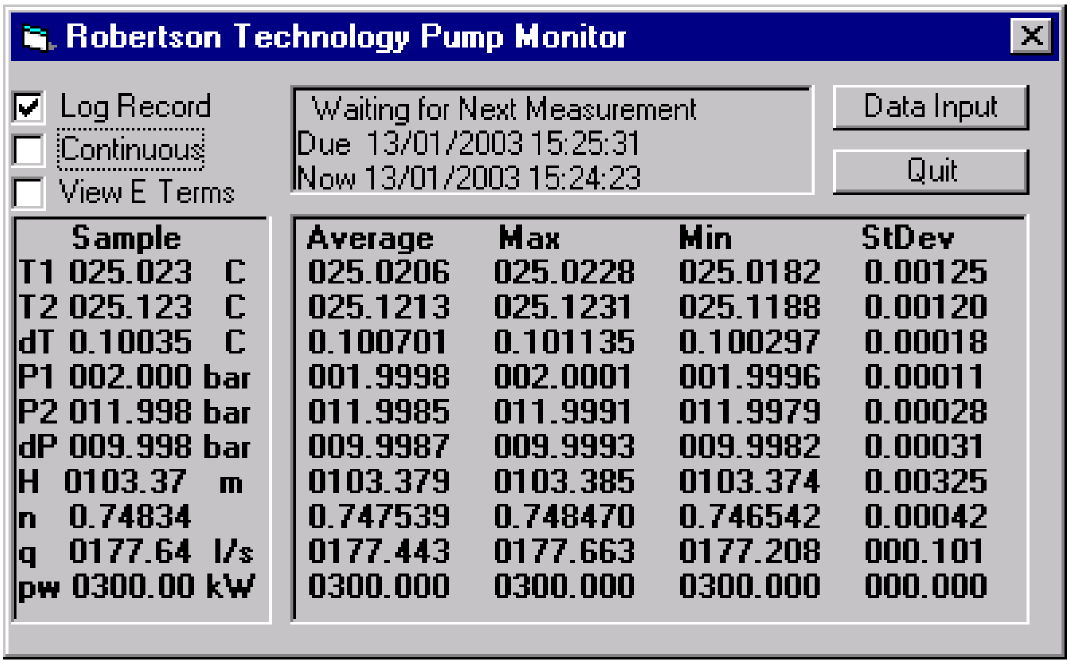
If tappings have to be in curved sections, they are best placed on the side of any bend, to reduce uncertainty in the pressure measurement. Generally, tappings should not be positioned at the top or bottom of the cross-sections of horizontal pipes, to avoid air or debris respectively.

Data display and derived information

Data is displayed on and logged by the control computer (a notebook computer is generally used with the portable unit). Averages, maxima, minima, and standard deviations are automatically calculated. This enables the operator to see if the pump operating point is sufficiently stable. In Figure 3:

T1, T2 and **dT** are the inlet, outlet and differential temperatures respectively
P1, P2, and **dP** are the inlet, outlet and differential pressures respectively, in bar
H is total pump head, in m of water
n is the hydraulic efficiency, **q** is flow rate in l/s, and **pw** is the electrical power supplied to the pump, in kW.

Figure 3   Typical data display



Single readings can be taken continuously at preset intervals. Alternatively, measurement sets can be taken at preset intervals. Each measurement set may consist of 25 readings (for example), acquired at preset intervals (e.g. every 3 seconds). Averages and standard deviations are automatically calculated and logged. The time interval between measurement sets can be varied over a wide range, but may typically be 15 minutes for a continuous monitoring system.

Figure 4 is an example of the data format in the Record log for each pump. The Record log is in a csv (comma separated variable) format, and is imported into other programs for data analysis, graph plotting, and pump scheduling.

Figure 4   Example of data log

Record No	Time	Timer	Date	T1	T2	dT	P1	P2	dP	H	n	q	pw	Eh	Em	dEm	Ex
1	22:51:14	82274.11	28/01/03	25.0000	25.1000	0.1000	0.000	9.672	9.672	100.0	0.741	7.1	10	982.7	1325.5	0	0
2	22:51:17	82277.11	28/01/03	24.9973	25.0978	0.1005	0.000	9.672	9.672	100.0	0.740	7.0	10	982.8	1327.6	0	0
3	22:51:20	82280.11	28/01/03	24.9974	25.0973	0.0999	0.000	9.672	9.672	100.0	0.742	7.1	10	982.7	1325.2	0	0
4	22:51:23	82283.12	28/01/03	24.9970	25.0969	0.0999	0.000	9.673	9.673	100.0	0.742	7.1	10	982.8	1325.2	0	0
5	22:51:26	82286.12	28/01/03	24.9970	25.0972	0.1002	0.000	9.672	9.672	100.0	0.741	7.0	10	982.7	1326.2	0	0

ACCURACY OF ON-SITE MEASUREMENTS

Pumps in the energy range 10 kW to 5 MW have been tested to date, with internal pipe diameters ranging from 0.08 to 1.2 m.

With field tests it is not possible to guarantee accuracy beforehand, due to unknown system considerations, for example, pipe work configurations, velocity distributions, faulty valves, and pump control problems. However, lightweight portable temperature and pressure calibrators are available, for on-site verification of probe accuracy. Differential temperature can be checked on-site to an accuracy of about 1mK, and pressures to 0.05%.

Following the test, an estimate of uncertainties can obtained from a Case Study assessment. Several corrections may need to be taken into account, including:

(a) Heat energy from motor cooling water, if cooling water is taken from the high-pressure side of the pump and returned in between the measurement points. This may typically be 2% of the heat energy dissipated by the pump.

(b) Heat lost to surroundings, if the pump is at a much higher temperature than ambient. This correction is applied for boiler feed pumps, and is not generally required for cold-water applications.

(c) Heat dissipated in bearings, if not water-cooled, or if the cooling water is not returned to the pump. This correction may be required for smaller pumps.

(d) Transit time correction, if the temperature of the inlet water is varying more than about 10 mK per minute. This requires an estimation of the volume of water between the measurement points. It is not often required for fresh or raw water pumps, but may be needed for sewerage or boiler feed pumps.

The main criterion for accurate pump efficiency measurements is how representative the inlet temperature measurement is of the average temperature of the water entering the pump. ISO 5198 recommends temperature probe insertion of 1/7 of the pipe diameter, for single probe measurements. However, an insertion depth of 50 mm or more is generally sufficient, and is recommended in reference (3), which applies to Class 1 and 2 tests. At high water velocities, the temperature probes may vibrate when inserted deeper, due to vortex shedding, and vibration may cause excessive heating of the probe tip. Longer probes have been developed to allow verification of temperature distributions, or averaging of measurements at different insertion depths, also in accordance with ISO 5198. We have identified a requirement for investigating the profile of the inlet temperature in the following situations: (a) where the inlet measurement point is unavoidably close to the pump's inlet flange, and (b) for wet well pumps.

The outlet temperature is generally measured to high accuracy, as the water is mixed well after passing through the pump. However, if the outlet measurement point is too close to the pump flange, high standard deviations in the outlet temperature will be seen.

Unstable or unusual readings can be used as a diagnosis tool. The standard deviation of a set of measurements can often provide a good indication of the suitability of the tapping points. If the inlet measurement is too close to the pump flange, re-circulation may even be noticeable at full flow.

Above 20 m head, uncertainties will generally be about 1% for pump efficiency, and 2% for flow rate measurement, when tapping points are optimal. Repeatability can be an order of magnitude better than this, allowing rapid detection of pump degradation by continuous condition monitoring. At higher water velocities (> 5 m/s), the uncertainty in the viscous heating correction may be the limiting factor for accurate differential temperature measurements. Further work is in progress to determine this correction more accurately.

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