# Efficiency of pumps in parallel operation on long-distance water pipelines


**Alex van der Spek**
*Consultant, Zdoor BV, The Netherlands*

**Robert Maron*, Americo Zuzunaga**
*CiDRA Minerals Processing, Wallingford, USA*



## ABSTRACT

Mining operations in the high mountains increasingly rely on water being pumped up from sea level for their process needs. Future process plants may be able to use sea water in which case raw sea water with a higher density than desalinated water will be pumped up. Pumping large volumes of water is usually done by multiple, constant speed, high pressure, multi-stage, centrifugal pumps operating in parallel.

Such pumps operate from a common suction line and discharge into a common manifold. As a result intake and discharge pressures as well as the flow rates are correlated, which complicates the evaluation of individual pump's efficiency. Such correlations will also impact the operating point of each individual pump and will introduce dependencies between individual pump performances.

Correct treatment of such dependencies and correlations is required for predictive maintenance and efficiency improvement. In a model study of a five-pump pump station, we use simple measures of performance (head, power and efficiency) in non-dimensional form based on non-dimensional figures of merit for an individual model pump. Parallel operation of five pumps, three conforming to specifications, one with a reduced static head, and one with increased internal friction, can be modelled easily using a simple empirical model.

In order to identify underperforming pumps in parallel operation of a number of pumps, it is required to be able to compute efficiency on a single pump basis. When single pump operation in a station is rare, such single pump data can only be obtained by simultaneous measurement of discharge head and discharge flow for each pump individually.

To compliment this work it is planned to directly measure and track individual pump efficiency by installing sonar flow meters on individual pump discharges on an existing pipeline, and to report on this work in the near future.



***Robert Maron**, CiDRA Minerals Processing, 50, Barnes Park North, Wallingford, CT, USA, Managing Director, +1-203-626-3354, rmaron@cidra.com

## INTRODUCTION

The combined shortages of water and energy in arid mining regions of the world has created the challenge of pumping large volumes of water over very long distances in an energy efficient manner. In the mining region extending from central Chile into southern Peru, this has resulted in several very large water pipelines now in operation, and many more in various stages of planning or construction. The typical statistics for these pipelines are impressive: pipeline lengths of 100 km or more, vertical rise of 2,000 m, flow rates of 1,000 l/s, two to three dozen large pumps, total pumping power requirements of 20 MW, annual pumping energy costs of US$20M, and annual pump maintenance costs in the millions of US dollars.

A key to managing the energy and maintenance costs is measuring and tracking the hydraulic efficiency of individual pumps. This requires accurately measuring and tracking several parameters, one of which is flow rate. However, the pipeline system design makes this a challenge because each of the multiple pump stations has several pumps operating in parallel, and often flow meters are only installed on the main line, thus making direct individual measurement of hydraulic efficiency very difficult.

This challenge presents itself in two situations. First, during the commissioning of the pipeline, the engineering company and the client must agree that the entire pipeline system has met its design goal of pumping efficiency. If tests prove otherwise then the individual deficient component, such as a pump-motor combination, must be identified. Second, once commissioned and operating the operator would like to use a condition based maintenance program that enables extending the pump maintenance interval as long as possible. Parameters, such as vibration and bearing temperature, can indicate a need for maintenance. However, some parameters, such as hydraulic efficiency decrease due to impeller wear or increased internal hydraulic friction, can only be detected by measuring hydraulic efficiency of the pump, which requires a flow measurement.

This paper shows that with only a flow measurement on the main line it is possible to monitor hydraulic efficiency of running pumps one at a time. However, this is very burdensome operationally and, thus, is rarely done. Alternatively, running combinations of pumps to detect a poor performer is less operationally burdensome but has significantly greater detection uncertainty. For all these approaches, the variation in fluid density due to ambient temperature variation can add significant uncertainty, possibly making these approaches useless. A much better alternative is to have flow measurement on each pump discharge, thus enabling real-time monitoring of hydraulic efficiency of individual pumps while accounting for density variations. In addition to this model study, we are planning to install in the near future non-invasive sonar flow meters on the individual discharge lines of a five-pump pump station on an existing long-distance large-diameter uphill water pipeline to demonstrate the ability to track pump efficiency in real-time. The use of sonar meters eliminates concerns with using invasive meters on these high pressure lines with 1,000 psi discharge pressures, and takes advantage of their excellent long-term stability. We hope to report on this field work in the near future.

## METHODOLOGY

The simulation of a pump station with a number of pumps operating in parallel starts with a simple model of a single centrifugal pump. The model is formulated in terms of the non-dimensional head coefficient $C_Q$ and the non-dimensional flow coefficient $C_Q$. These two coefficients are respectively defined as follows:

$$C_H = \frac{gH}{\omega^2 D^2} \tag{1}$$

Where g is the gravitational acceleration (m/s$^2$), H is the head of the pump (m), w is the speed (rad/s), and D the impeller diameter (m). Since the tip speed of the impeller of the pump equals the product of the pump speed and the impeller radius, the denominator is proportional to the velocity squared of the fluid leaving the impeller. The head coefficient expresses the pump generated head in terms of the dynamic head of the fluid being pumped at the tip of the impeller.

$$C_Q = \frac{Q}{\omega D^3} \tag{2}$$

Where Q is volumetric flow rate (m$^3$/s) and the other symbols are the same as the corresponding symbols in the definition of the head coefficient. The denominator of this equation is proportional to the swept volume per revolution of the impeller. The flow coefficient, therefore, expresses the discharge relative to the impeller's swept volume.

A very simple but widely used empirical model of the performance of a centrifugal pump is as a quadratic equation of the head coefficient in terms of the flow coefficient (Walshaw, A.C. and Jobson, D. A., 1967):

$$C_H = a + bC_Q + cC_Q^{\ 2} \tag{3}$$

In this model the coefficient a expresses the generated head at zero flow, the coefficient b accounts for the increasing head with increasing rate of flow, and the coefficient c introduces frictional losses inside the pump.

The advantage of this simple model is just that, it is simple, yet it incorporates the most important effects observed in pumps. The simplicity of the model allows it to be used to describe a number of pumps running in parallel with great ease.

In order to express the generated hydraulic power of a pump, we define a power coefficient as the product of the flow coefficient and the head coefficient.

$$C_P = \frac{P}{\rho\omega^3 D^5} = \frac{gHQ}{\rho\omega^3 D^5} = C_H C_Q \tag{4}$$

By simple combination of the expression for hydraulic power and the definition of the flow coefficient and the head coefficient, it can be seen that the power scales with the cube of the pump's speed and with the fifth power of the impeller's diameter.

## Fitting and normalizing pump curve data

Pump curve data from a large diameter, fixed speed, multi-stage centrifugal pump was digitized using published performance graphs. The pump curve data, which was presented in engineering units, was converted to non-dimensional head and non-dimension flow by using equations (1) and (2) respectively. The known impeller diameter and the rated speed of the pump were used to compute the $C_H$ and $C_Q$ for each digitized point. The non-dimensional pump curve data was then fitted to the model in equation (3). Figure 1 presents the result of the digitization and normalization, the fitted model curve, and the 95% confidence interval on the fitted model data in the top left panel. The quality of the fit is good at least in the region of practical interest, i.e. away from the dead heading at zero flow.

Hydraulic power can now be computed simply by multiplying the flow coefficient and the head coefficient at each flow coefficient's value. Alternatively, we can use the fitted model equation (3), multiply on both sides with the flow coefficient, and compute the hydraulic power from the model. The two approaches should, of course, overlay, and the top right panel shows that they indeed do.

The efficiency of the pump is a measure of how well it converts shaft power into hydraulic power. Since shaft power is also available in published performance graphs, the efficiency can be computed as well. There is, however, no simple analytical model available for shaft power. Fitting the shaft power curve is, therefore, done using a locally weighted low order polynomial fit (Cleveland, W.S. and Devlin, S.J., 1988). The result is given in the bottom right panel.

By simply dividing the hydraulic power by the shaft power, the efficiency is computed. The comparison between the digitized efficiency curve as given on published performance graphs and the model approach, where hydraulic power is computed from the pump model and shaft power from the regression model, is shown in the bottom left model.

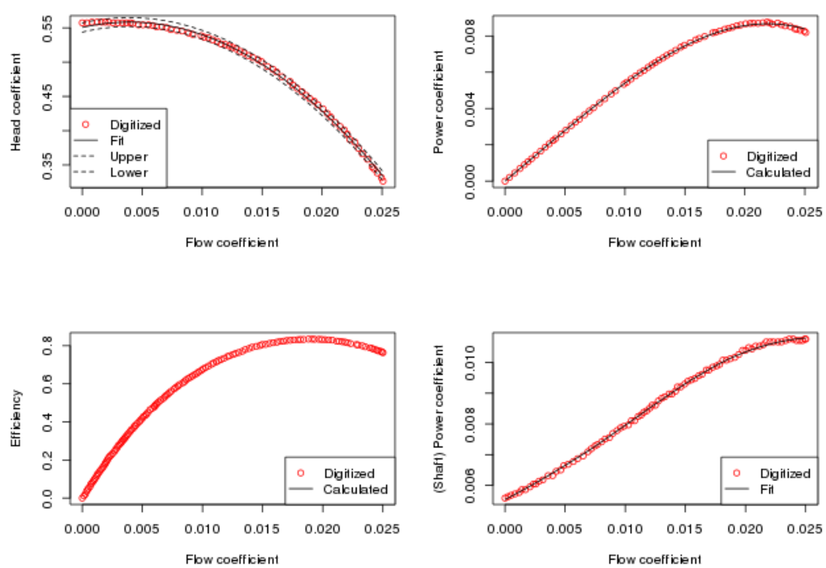


**Figure 1** Digitized pump curve with best fit model

Apparently, the simple quadratical model for pump performance in terms of a head coefficient and a flow coefficient works well and describes both hydraulic power and efficiency with great precision. It is of importance to verify this as the model will subsequently be used to describe the performance of multiple pumps operating in parallel. The quality of the fit is good, at least in the region of practical interest, i.e. away from the dead heading at zero flow.

## Modelling potential pump issues

In order to model a realistic situation, a pump station with five pumps operating in parallel is chosen where two out of five pumps each has a different issue. The pump's individual performance can be described as:

- Pump A; ideal performance as by the fitted model curve.
- Pump B; slightly better performance than pump A.
- Pump C; slightly worse performance than pump A.
- Pump D; cannot meet the zero discharge head.
- Pump E; much higher internal frictional losses.

These different pumps can now be described in model terms by their coefficients a, b and c as given in Table 1.

**Table 1** Pump model coefficients

| | A | B | C | D | E | Std Error |
|---|---|---|---|---|---|---|
| a | 0.551 | 0.552 | 0.549 | 0.539 | 0.551 | 0.001 |
| b | 4.0 | 4.2 | 3.8 | 4.0 | 4.0 | 0.2 |
| c | -503 | -496 | -511 | -503 | -581 | 8 |

The pump curves, in non-dimensional form for the five pumps A, B, C, D and E, are graphically depicted in Figure 2, top left panel. The other panels show the hydraulic power output, the pump's efficiency, and the shaft power, which is assumed to be the same for each pump.

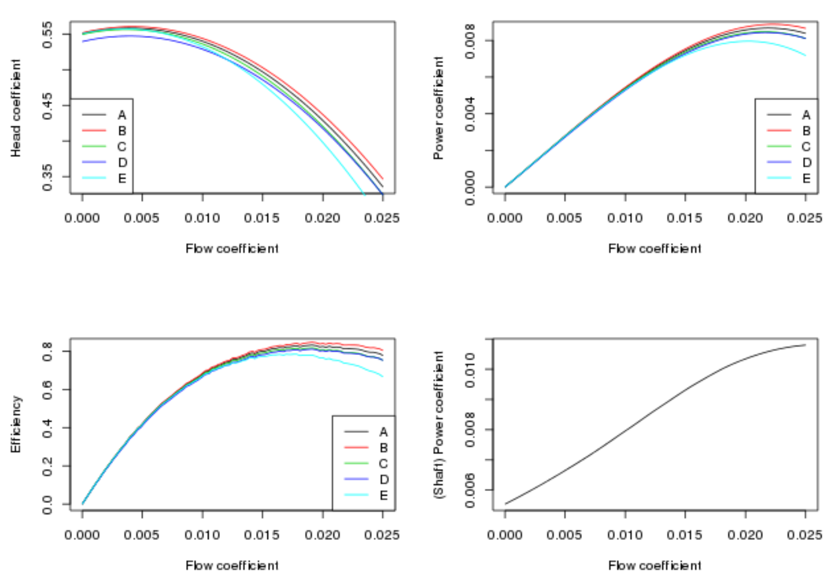


**Figure 2** Five pumps individually: three normal, two with issues

As can be seen the degradation of performance in pump D is hard to see at higher rates of flow; whereas, the E pump, which suffers from increased frictional losses, is clearly performing poorer than the other pumps.

Note that only hydraulic issues are being addressed in this model. Mechanical issues with the pump, such as increased frictional losses in the shaft's bearings, are not taken into account. It is assumed that such additional effects remain negligible or can be effectively cured by simple in-service maintenance.

## Operating pumps in parallel

In a five-pump pump station the operation of pumps in parallel is governed by the addition of the flow (coefficients) at the same head (coefficient). Since not all five pumps are operating all the time and fewer than five pumps can operate in different combinations, it becomes difficult to observe the effect of misbehaviour of a single pump. This difficulty is shown in Figure 3. The black curve in this graph shows the pump curve of the five pumps in parallel where two of the five have a serious issue. The red curve, by comparison, shows the pump performance of the five pumps in parallel when all five show ideal performance as pump A does. Clearly, there is an issue as the black (observed) pump curve is below the ideal performance.

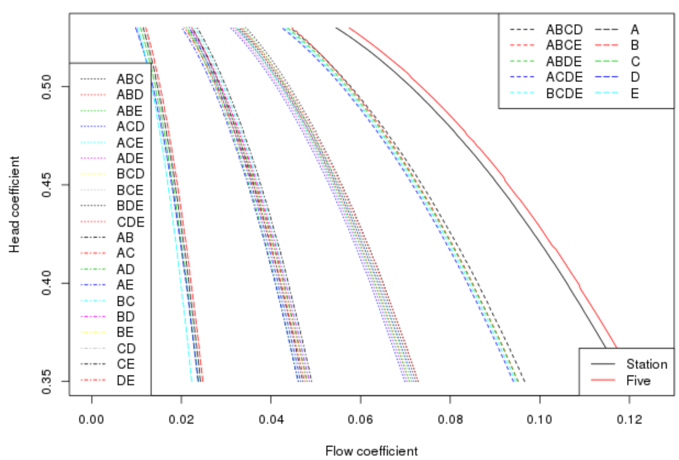


**Figure 3** Parallel pump operation

Would it be possible to find the mal performing pump(s) by selectively switching off one pump and operating just four pumps in various combinations? This is a strategy that a maintenance engineer might follow in the absence of individual pump discharge flow measurement data.

The switching off of one pump leaves five combinations of four pumps operating. This is the first fan curve to the left of the black and red curves in Figure 3. Likewise, switching off 2 pumps results in ten possible pump combinations in the subsequent fan curve to the left. Switching off 3 pumps also results in 10 possible combinations, given as another fan curve left. Finally, the five pumps operating individually are presented by the left most fan curve.

Obviously, it is very hard to discern anything from such narrowly spaced fan curves. Unfortunately, zooming in to each individual fan curve as in Figure 4 does not really help.

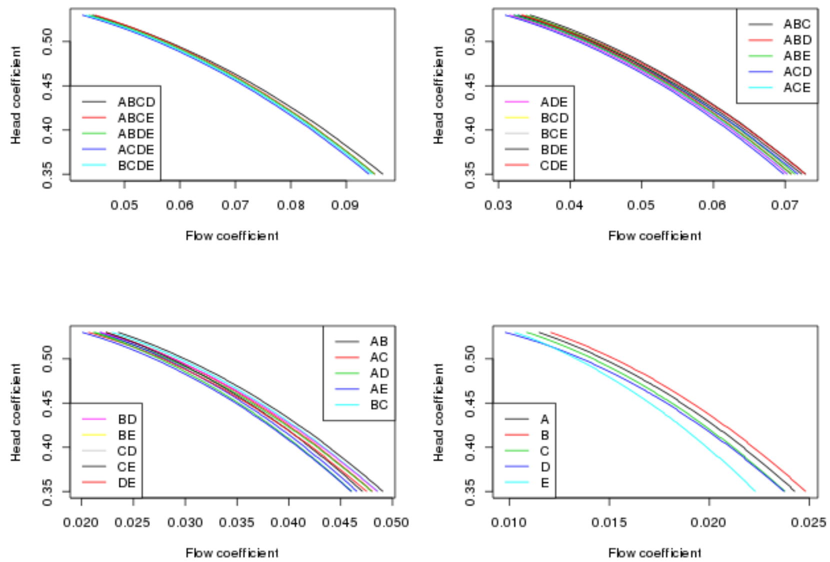


**Figure 4** Parallel pump operation (zoomed)

The top left panel of Figure 4 shows the combined pump curve for four pumps operating in parallel for each of the five possible combinations. The black curve in this panel is the curve for that particular combination where the E pump, the pump with highest internal frictional losses, is missing. It is, however, also that curve where the B pump, the pump which operates better than the published performance data, is in the combination. Therefore, concluding that the E pump is underperforming is a conclusion one can only draw when it is assumed that all other pumps are performing normally and not any one performs better than anticipated.

In comparison, with one pump only operating there is no doubt that the E pump is suspect and requires attention. Incidentally, whereas it is difficult or rare to see a five pump station operating with only one pump in service, the same information could have been obtained by individual discharge pressure and flow measurement data.

## *Parallel hydraulic power*

Since hydraulic power is the product of the discharge flow rate and the pump generated head, the effect of an underperforming pump should be easier to see on a plot of the hydraulic power. This is so as both a defect in the generation of head, for instance due to increased clearances, as well as a defect in the capacity of the pump, as may be caused by increased internal friction, reduce the pump's generated hydraulic power.

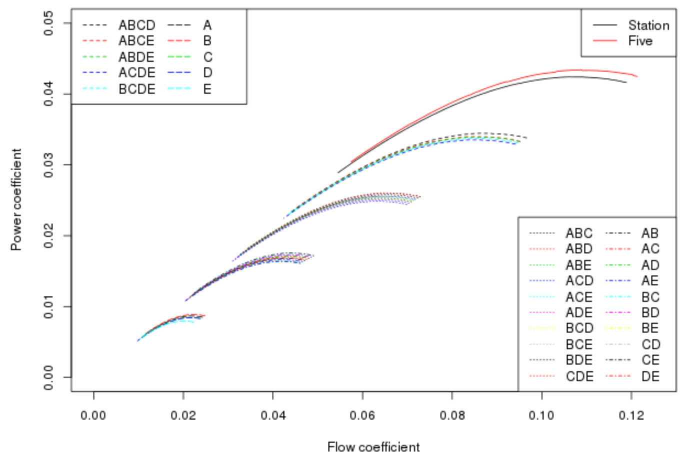


**Figure 5** Combined hydraulic power

Combined power curves for a five-pump station operating with either all five pumps or any combination of four, three, two or one pump only are given in Figure 5. The black curve in this graph shows the station's generated hydraulic power for all five pumps in contrast to the red curve, which is the curve for five identical pumps as per the published specifications. Clearly then, a difference exists, and the one pump which is slightly better than the published specification cannot make up for the shortfall in performance of the three others. The fan curves at lower throughput are difficult to interpret because of the scale of the plot. Zooming in to each individual fan curve in a separate multiplot as in Figure 6 gives a clearer picture.

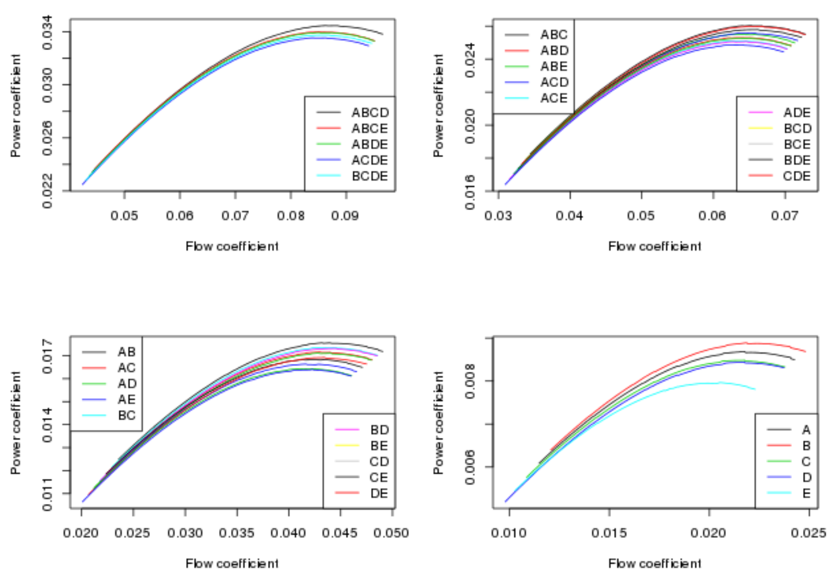


**Figure 6** Combined hydraulic power (zoomed)

Indeed, as expected in the top left panel, the black curve, representing the case where pump E is missing from the pumps in service, stands out a little more than in the pump curves itself. Likewise however, the bottom right panel singularly identifies pump E as the underperforming pump and pump B as the best pump, better even than pump A, which operates "by the book."

Note that the chosen representation shows the differences in pump performance by the changes in curvature. Had there been an additional and unwanted loss due to bearing friction being higher than called for, this would not have changed the shape of any curve. In this case only the position of the curves would have shifted down as the pumps are fixed speed pumps.

### *Parallel efficiency*

Parallel efficiency is the efficiency obtained by a number of pumps operating in parallel. It can be computed by dividing the sum of the hydraulic power of each pump by the sum of the shaft power required at the given discharge flow rate and head.

Since efficiency is equal to hydraulic power normalized by shaft power, it is anticipated that any differences in pump performance can be seen still more clearly on parallel efficiency data than on combined hydraulic power output data.

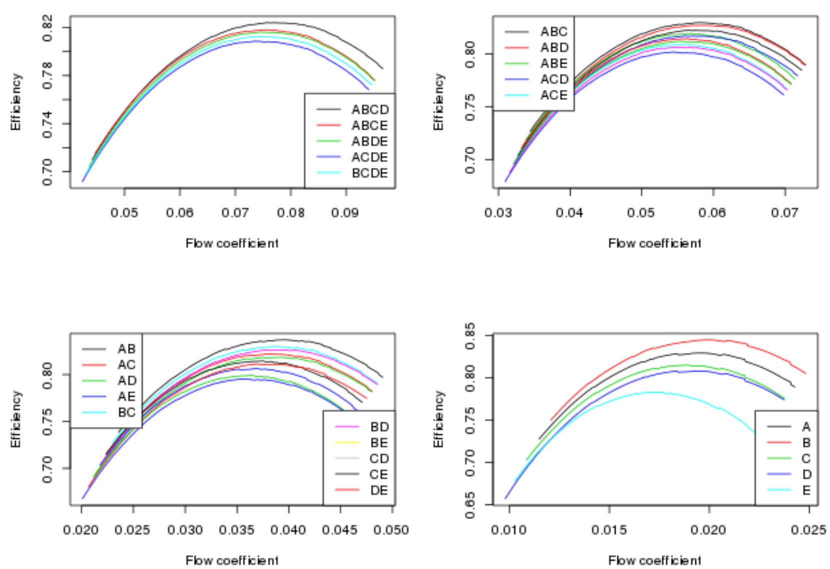


**Figure 7** Parallel efficiency (zoomed)

Each subplot is now one of the four possibilities of subsequently switching off more pumps out of the five pumps in parallel. Only on the individual level, however, does it become clear that the E pump has a serious issue. For the four pumps operating, as shown in the top left panel, the pump combination with best performance can be identified but not the individual pump causing underperformance of the station as a whole.

Plotting all of the available efficiency data on one common scale of efficiency shows how much larger the differences are in the individual pump's efficiency as opposed the parallel efficiency of each pump. By the normalization to common shaft power, the fan curves are now no longer stacking up as was the case for the hydraulic power curves. Instead, the fac curves partly overlay and cross each other.

Whereas in the case of individual pumps the difference in efficiency between the best and the worst pumps is about 8% absolute, the difference in efficiency between that combination of pumps which operates best and that combination which operates worst for a four-pump combination is only 2% absolute. The difference between best and worst increases with fewer pumps operating, but so does the number of possible combinations and, therefore, the time of testing, not to mention the complexity of the analysis.

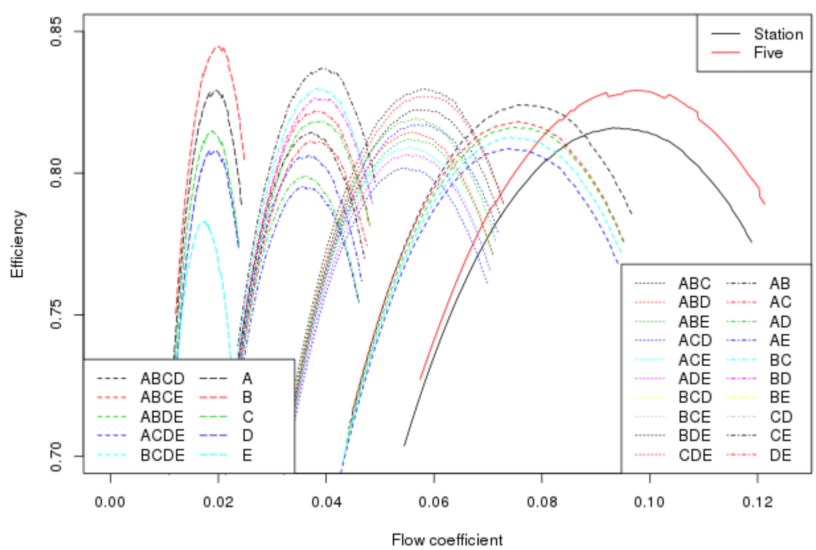


**Figure 8** Parallel efficiency (zoomed)

It is noteworthy that the best efficiency obtained for single pumps is by the pump with slightly better performance than the published performance data. The combined five pump best performance, however, is based on five pumps which conform exactly to the published performance data.

## RESULTS AND DISCUSSION

Simple measures of performance for a set of pumps operating in parallel were derived from non-dimensional figures of merit, the flow coefficient and the head coefficient, for a single pump. The computation of these performance measures is facilitated by a simple analytical but empirical model of a pump. This model is able to capture pump performance degradation of several kinds, including those of loss of static head as a result of increasing clearances, as well as those of loss of throughput due to increased internal friction.

For a model five-pump pump station the performance measures (head, power and efficiency) were computed for all possible combinations of pumps in and out of service. It is apparent that efficiency shows clearest those pump combinations with best and worst performance; the identification of the pump that is underperforming and in need of maintenance is only possible at the single pump level.

Note that the analysis presented here pertains to pumps operating in parallel and discharging into a common pipeline. The discharge pressure of the pumps must, therefore, be equal. The rate of flow per pump, however, may vary considerably with the pump's ability to discharge at the common pressure.

For pumps operating in series, the converse holds. Here the rate of (mass) flow is equal for each pump whereas the discharge pressures are different. In the case of pumps in series, the minimum required information is discharge pressure and shaft power. In the case of pumps in parallel the minimum required is rate of flow per pump and shaft power.

Whenever (mechanical) shaft power is not available an equivalent analysis may be made based upon motor power (current draw and voltage). The disadvantage of this is that it is no longer possible to split the pump's hydro-kinetic efficiency from the motor's efficiency.

## CONCLUSION

In order to identify underperforming pumps in parallel operation of a number of pumps, it is required to be able to compute efficiency on a single pump basis. When single pump operation in a station is rare, such single pump data can only be obtained by simultaneous measurement of discharge head and discharge flow for each pump individually.


## ACKNOWLEDGEMENTS

Susan McCullough’s help during the preparation of this article was indispensable.


## NOMENCLATURE

| | |
|---|---|
| CH | Head coefficient |
| CQ | Flow coefficient |
| CP | Power coefficient |
| a | Pump model coefficient, zero discharge head |
| b | Pump model coefficient |
| c | Pump model coefficient, internal frictional losses |
| D | Pump impeller diameter |
| g | Gravitational acceleration |
| H | Pump head |
| Q | Flow rate |
| P | Power |
| $\omega$ | Pump speed |
| $\varrho$ | Fluid density |

## REFERENCES

Albraik, A., Althobiani, F., Gu, F. and Ball, A. (2012) 'Diagnosis of Centrifugal Pump Faults Using Vibration Methods', *25th International Congress on Condition Monitoring and Diagnostic Engineering*, Journal of Physics: Conference Series 364, IOP Publishing Ltd.

Cleveland, W.S. and Devlin, S.J. (1988) 'Locally-Weighted Regression: An Approach to Regression Analysis by Local Fitting', *Journal of the American Statistical Association,*83 (403): pp. 596–610.

Harihara, P.P. and Parlos, A.G. (2012) 'Fault Diagnosis of Centrifugal Pumps Using Motor Electrical Signals', *Centrifugal Pumps*, Dr. Dimitris Papantonis (Ed.), InTech, Croatia.

McKee, K.K., Forbes, G., Mazhar, I., Entwistle, R. and Howard, I. (2011) 'A review of major centrifugal pump failure modes with application to the water supply and sewerage industries', Asset Management Council, ICOMS Asset Management Conference, Gold Coast, QLD, Australia, 16 May.

R Core Team (2012) *R: A language and environment for statistical computing*. R Foundation for Statistical Computing, Vienna, Austria. ISBN 3-900051-07-0, URL http://www.R-project.org/.

Yedidiah, S. (1996) *Centrifugal Pump User's Guidebook: Problems and Solutions*, Chapman & Hall, New York.

Walshaw, A.C. and Jobson, D. A. (1967) *Mechanics of Fluids*, Longmans, London, pp. 431-483.