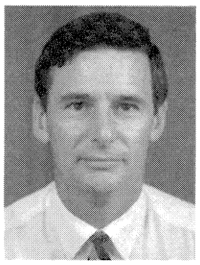
THE CONTROL OF VARIABLE SPEED PUMPS IN PARALLEL OPERATION

by
Anthony B. (Tony) Crease
Adviser, Rotating Equipment Engineering
Lagoven S.A.
Maturin, Venezuela



Anthony B. (Tony) Crease graduated in Mechanical Engineering from Imperial College, London (1966), and obtained a Master's degree in Tribology from the University of Leeds, England (1969). After two years in the English Electric Company, he spent nine years with Michael Neale and Associates as a Consulting Engineer, working on a wide variety of tribological and mechanical engineering problems for private and public industry. In 1979, he joined the Rotating Equipment Engineering group of Lagoven, one of Venezuela's three major oil companies, becoming the company's Rotating Equipment Advisor in 1985. He is involved in all aspects of rotating equipment ownership, from specification and selection, through commissioning, troubleshooting and maintenance, to performance, reliability and economic assessments, relating to over one million horsepower of major machinery, including pumps and centrifugal compressors and their gas turbine and electric motor drivers.

ABSTRACT

The features are discussed of a PLC based control scheme that was developed for a multipump pumping station to be used to supply oil field produced liquids (crude oil and associated water) from tanks through a dehydration facility to dewatered crude storage. The hydraulic resistance characteristic of this system exhibits substantial variation and the required flow is variable and covers a wide range from initial startup conditions to the eventual peak capacity requirement. The avoidance of unnecessary energy input to the product was of special importance, since it would promote the formation or stabilization of emulsions of the produced fluids, thereby complicating the dehydration process. The control scheme developed is applicable to any variable-speed, parallel pump station.

With the optimum pump number fixed in this way, pump speed is simply controlled to provide the desired overall flow. As a minimum, values of Q and N (or excitation frequency) have to be measured and input to the control scheme. The individual values of Q/N, besides providing the basis of pump number optimization, also permit protection against excessively high or low flow. Additional practical aspects are discussed.

INTRODUCTION

The potential benefits of using variable speed drives for centrifugal pumps, including flexibility and the reduction of energy consumption, have been widely presented (for example, Murphy [1]). Variable frequency drives and variable speed drives (hydraulic couplings, etc.) may be considered for a new application, and the benefits will have to be weighed against the higher initial cost and complexity.

The author's company recently adopted the use of variable frequency drives for a station using multiple horizontal centrifugal pumps to supply oil field produced liquids (crude oil and associated water) from tanks, through a dehydration facility, to dewatered crude storage.

The main factor in the decision to use variable speed in this case was the need to minimize unnecessary energy input to the pumped fluid which is particularly susceptible to the formation of stable crude/water emulsions, thereby rendering more difficult the downstream dehydration process in electrostatic precipitators. Energy cost savings and flexibility were important additional benefits of the variable speed drive.

The hydraulic resistance characteristic of the system varies considerably due to tank level and fluid viscosity changes and the required flow varies from 50 to 300 mbpd (1460 to 8460 gpm) over the project life. The range of system resistance is shown in Figure 1.

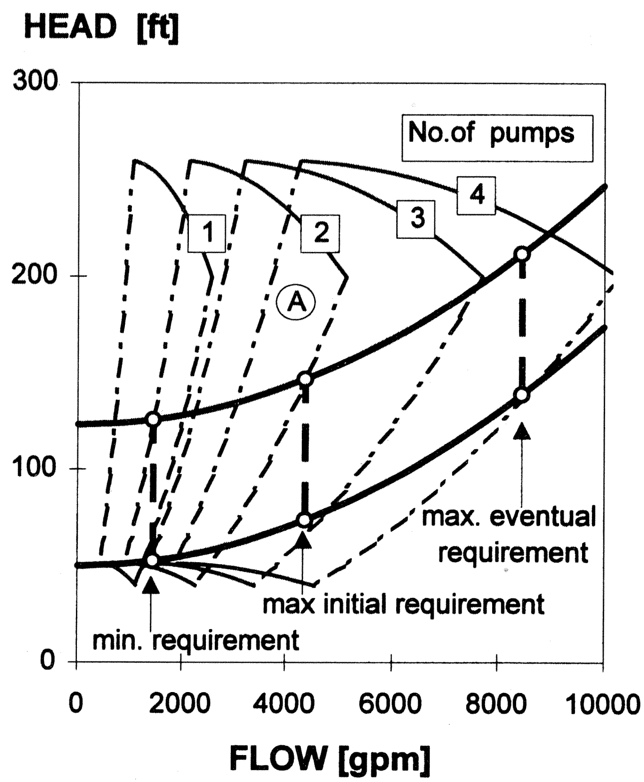


Figure 1. System Resistance Characteristic and Pump Performance Envelopes for Speeds Between 800 and 1800 RPM and Q/N Between 50 and 120 Percent of the BEP Value.

It was elected to use parallel pumps: 4 + 1 for the eventual maximum requirement and three for the initial operation. The eventual maximum requirement determined the basic head/flow capability needed from the pumps and the envelope of performance at variable speed from 45 to 100 percent of nominal for one, two, three, and four pumps is shown in Figure 1. The election of parallel pumping is in line with a published criterion [2] that recommends the use of a parallel rather than a series configuration when the fixed part of the system resistance (geodetic head) is more than half the total (fixed + dynamic) resistance.

Having selected the basic pumping scheme and pump type, attention was turned to the requirements for automatically controlling their operation, for inclusion in the project specifications for a PLC based control system to allow unattended operation if necessary of the pumping station.

The general criteria and requirements to provide optimum operation of parallel, variable speed centrifugal pumps, for application in the above mentioned project, were evaluated. The results and conclusions of the evaluation are described.

FLOW CONTROL IN VARIABLE SPEED PUMPS

The main objective of the control scheme is to vary the total flow of the pumping station to match system or process requirements, without the use of control valves. This is achieved by varying the number of units in service and their speeds.

An important aspect of the control scheme is to ensure that the pump units operate within a safe continuous duty range, as determined in conjunction with the manufacturer. The envelopes in Figure 1 indicate these safe ranges for one to four pumps. In addition, as illustrated by point A in the Figure, criteria are needed to choose the best combination of pumps for a given overall duty, between two, three or more alternatives that satisfy the basic requirement of having the pumps run in the acceptable range.

DEFINITION OF THE SAFE CONTINUOUS DUTY RANGE

The continuous duty range of a variable speed pump may be considered as the range of conditions that it can accept without suffering an unacceptable adverse effect on unit reliability. This range is normally well established by the pump manufacturer for operation at nominal speed and is then simply the usable pump flow range, as determined by high and low flow cavitation or in some cases by vibration, bearing loads, etc. Guidance on cavitation limits is now available from many sources [2], and special attention to a precise evaluation of minimum flow is required for high specific speed and high energy pumps, where the flow range may be very restricted. An accurate definition of the safe continuous duty flow range at nominal speed should be obtained in consultation with the pump manufacturer as part of the evaluation of any pump for a variable speed application.

At the best efficiency point of a pump (bep), flow enters the pump with an optimum incidence angle onto the impeller vanes. At higher and lower flows the incidence angle deviates from its optimum value, leading eventually to flow separation, turbulence, flow induced vibration and cavitation. At the same time thrust and radial bearing loads, which are designed to be minimized at or near the best efficiency point, increase progressively as the percentage deviation of the flow from bep increases.

At different speeds than nominal, the flow incidence angle on to the vanes will continue to be a fundamental factor in the occurrence of flow separation, turbulence and cavitation. Since the incidence angle is directly related to the normalized flow, Q/N, it is reasonable to assume that the useable flow range will continue to be determined by the maximum and minimum limits of Q/N established for nominal speed operation. For reduced speeds, this assumption is conservative in the sense that, at a given Q/N, all forces and pressures in the pump will tend to reduce with speed.

On the contrary, at increased speed, for a given Q/N, all pressures and loads on a pump will increase. Thus, speeds significantly higher than nominal should be used with caution and only after consultation with the manufacturer.

Avoidance of Critical Speeds

It is particularly important, when applying variable speed pumps, to identify with the manufacturer, the torsional and lateral critical speeds of the equipment and to ensure that the chosen speed range of the pump excludes all lateral criticals with an adequate separation margin.

Except for special applications where the cost of a detailed engineering evaluation can be justified, it is desirable to use, for variable speed applications, pumps, and motors that run below the first lateral critical speed. This permits the units to be run at any speed up to maximum without concern. Single stage pumps of horizontal or vertical inline type, using electric motor drivers at nominal frequency or below, will generally meet this aim, according to API 610, 7th edition, paragraph 2.8.1.1. On the other hand, for multistage, vertical turbine and high speed pumps, a detailed rotordynamics analysis is necessary to locate the "wet" critical speeds as a function of clearances, and to determine rotor response at the criticals, as a basis for determining the useable speed range.

During the purchase of variable speed pumping units, steady and transient torsional analysis should be specified per paragraphs 2.8.3.4 and 2.8.3.5 of API 610. Pump sets that use direct electric motor drive are unlikely to present torsional problems that cannot be resolved by a torsional analysis and proper selection of coupling stiffness, to move criticals away from excitation frequencies.

On the other hand, geared pumping units with multiple rotors are particularly sensitive to torsional problems, due to the many critical speeds and excitation frequencies, and it will often be impossible to avoid "interferences." In such cases, it will be necessary to limit the torsional stresses which result when operating right on one or more resonances. Further information is available in the paper by Frei, et al. [3], who mention the value of damping elements (flexible energy-absorbing couplings, dampers), installed in the line shaft, as a very efficient aid in reducing torsional stresses.

It should be noted that, in the case of variable frequency drives, torsional excitation of 5.0 to 30 percent of base torque (depending on the VFD switching technology) is generated at a frequency of $6.0 \times$ base frequency [1], with lower levels of excitation at higher multiples of $6.0 \times$ base frequency. It is therefore particularly important to avoid torsional criticals at the $6.0 \times$ frequency as a minimum.

Mechanical Seals

The possible influence of reduced speed on mechanical seal operation and reliability should be reviewed. There are two main concerns. The first is that the ratio of sealing pressure to speed, which is what determines seal lubrication, may increase above the range for which the seal was designed in nominal speed operation. This can happen if the sealing pressure is maintained at reduced speeds, due to being balanced to close to suction pressure.

The second concern is that the seal temperature may rise, due to the seal flushing flow falling more rapidly than the seal heat generation. This is of particular concern when handling light hydrocarbons or other fluids with high vapor pressure.

To ensure a satisfactory seal selection and design, these aspects should be evaluated, in consultation with the pump and seal manufacturers.

Drive Motor

The final component whose behavior has to be considered for variable speed operation is the driver motor. In the case of VFDs, a derating factor must be applied when selecting the motor to allow for higher heat generation as a result of the irregular shape of the waveform supplied by the VFD.

For reduced speed operation, there is no real concern since the pump torque at any given Q/N reduces in proportion to N^2. In the case of VFDs, the maximum output torque available from the motor stays practically constant down to at least 20 percent torque [1], thus ensuring ample power at reduced speeds.

In the case of VSDs, even though the losses in the coupling increase in proportion to the slip and are typically greater than the pump power at 50 percent speed, the total power at the fixed speed motor output shaft will typically be down to 25 percent, when the coupling output speed is down to 50 percent, so that once again reduced speed operation presents no problem for the drive.

With VFDs, motors can be driven at speeds above nominal, subject to the mechanical limit of the motor. For such applications, the drive and motor manufacturers should be consulted to obtain a proper selection and the respective speed and power limitations.

With due attention to the various aspects mentioned, including a proper selection of seal and flushing arrangement, an ample continuous operating range can be established. In particular, direct driven electric pumps can normally run safely up to at least the nominal speed, with Q/N limits equal to those established for operation at nominal speed. Limits based on speed and Q/N can readily be incorporated in a PLC based control scheme to protect the pumps during operation.

In addition, criteria must be established for minimum acceleration rates while bringing the pumps on line. A balance must be found between disturbing the process if the pump is brought up very quickly and running a risk of damaging the pump by spending too long at very low flow if it is brought up too slowly. A stepped rampup schedule that brings the pump quickly up to at least 10 percent flow to avoid any heating problem and then more slowly up to the same operating point as the pumps already in service, was adopted in the application of this study. Soundly based guidelines for rampup schedules do not appear to be available at this time, and will have to be developed as experience is acquired.

CRITERION FOR OPTIMIZATION OF PUMP OPERATION

The useable range of Q/N for pumps other than those of high energy or high specific speed, is typically 50 to 120 percent of the bep value or even more. This means that a given overall pumping condition of a parallel pump station can generally be satisfied by various different combinations of pumps, from a small number at high speed to a larger number at lower speed, as was illustrated in Figure 1. In fact, the pumps do not even have to be identical nor run at the same speed; the only strict requirement is that all pumps should be able to make, with some margin of safety, the maximum required head under any possible operating condition when running at maximum speed and minimum allowable Q/N.

This raises the question of how to choose the pump number and speed for a given overall duty, and how to specify criteria for automatic increase and decrease of the number of running pumps.

For the application considered in the present work, it was decided to use minimum total energy output to the fluid (that is minimum pump shaft power, not minimum electrical power input) as the fundamental criterion for developing a control strategy. For applications where the main concern is minimum energy consumption by the driver rather than the pump, the fundamental criterion would be slightly different from the criterion adopted here, due to the variation of driver efficiency with power and speed. However, it could be argued that ignoring the increase of driver losses with reduction of power and speed, would be substantially offset in an economic evaluation, by the lower maintenance cost for the driver and pump. This would make the criterion of minimum pump power applicable to a wide range of applications.

In any case, the conclusions derived from the present study would often give an acceptable approximation to minimum driver power consumption and the procedure developed in this study for determining when to change pump number, could readily be modified to better reflect a criterion of minimum input power.

The criterion of minimum pump power is used, in the next section, to show how the operating points of individual pumps of different characteristics must be adjusted to satisfy the criterion. In the subsequent section it is indicated how the optimum number of pumps of nominally identical type may be determined.

CONTROL OF OPERATING POINT OF INDIVIDUAL PUMPS

If all the pump units of a parallel group are identical, and by design they normally will be, it can be shown that minimum total pump power is obtained by running all the pumps at the same operating point.

This requirement is not always immediately obvious to those involved. It is sometimes argued for example that it might be preferable to have one or more pumps operating at best efficiency and one operating less efficiently at lower flow to achieve the total required flow. It is, therefore, convenient to present the reasoning for running identical pumps at the same point.

A second subject of interest is that of where to operate pumps that are nominally identical but have somewhat different performance due to deterioration in service. Of various options for controlling the operation, such as on the basis of the same speed, or the same flow or the same value of composite parameters such as Q/N, it is interesting to try to identify a best solution.

Thirdly, although not often of practical concern, it is also of interest to consider the criterion that should be applied to the control of variable speed parallel pumps with completely different operating characteristics; and since this evaluation should provide answers to the two previous points, it has been considered here.

Optimum Operating Points of Parallel Pumps with Different Characteristics

For a specified total differential head, the performance of variable speed pumps can be described by their curves of power, P, *vs* flow, Q, which can be readily generated from the multispeed performance map, as illustrated in Figure 2.

The resulting P-Q curves for a pump of the type used in the present project (pump B) and for a second pump (pump A) of lower efficiency, lower capacity and a steeper curve, operating in parallel against a particular head, (170 ft), are shown in Figure 3, with axes interchanged to plot flow against power.

This figure illustrates how the operating points of the two different pumps must be adjusted to minimize the total power consumption for a given total flow, Qt. It can be shown that the minimum power condition, is found by operating the two pumps such that:

- the slopes at the operating points on each Q-P curve are equal, that is: $(dQ/dP)_A$ at $Q_A = (dQ/dP)_B$ at Q_B. (Note that the values of dQ/dP at any power can be derived directly from the curves of Q *vs* P and have been added to Figure 3.)
- the average flow, $(Q_A + Q_B)/2$, is equal to half the required total flow, Qt/2.

This result is logical in the sense that, if a certain total flow is required, a small change in Q_A, the flow of pump A, would have to be accompanied by an equal but opposite change in Q_B. If the change in power for pump A, determined by the slope of the Q-P curve for A, were different from the change in power for B, determined by the slope of the Q-P curve for B, the total power would change, implying that a lower total power could be found at some other combination of Q_A and Q_B capable of giving Qt. It follows that at the point of minimum total power, the slopes at the two operating points must be equal.

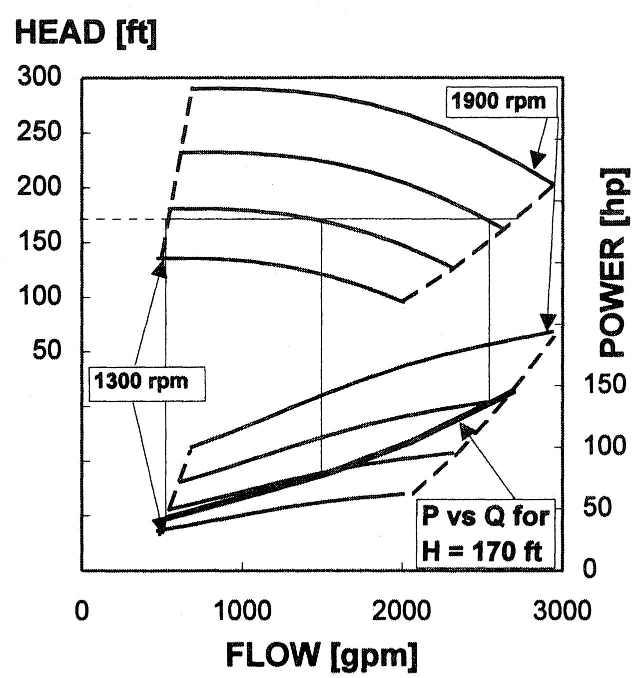


Figure 2. Pump Performance at Variable Speed and Constant Head.

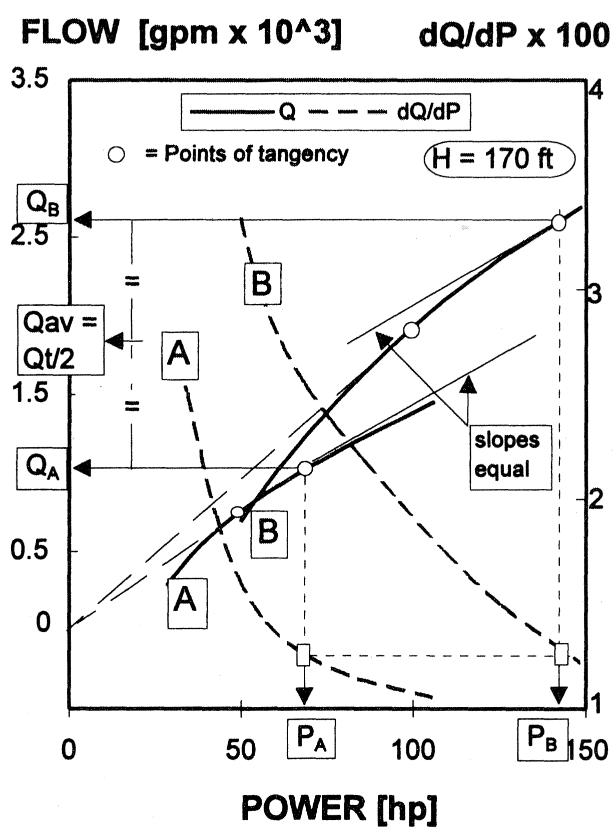


Figure 3. Performance of Different Types of Pump, A & B, in Parallel (Construction to Determine Optimum Operating Points).

Equal Marginal Efficiency As the Criterion for Optimum Operation

Another way of stating the above is to say that the optimum operating points are found where the incremental or marginal increase of flow per unit of power is the same for both pumps; or, in terms of calculus, the optimum operating points are where dQ/dP is the same for both pumps. The parameter dQ/dP could be termed a marginal efficiency factor and is related to, but different from, the normal pump efficiency, as indicated in the following:

$$\text{Pump power, } P = k \cdot H \cdot Q/\eta \qquad (1)$$

where:

k = constant for the unit system employed

η = efficiency

$$\text{Rearranging: } Q/P = \eta/k.H \qquad (2)$$

Thus, Q/P at constant head is proportional to the efficiency, as normally used, and may be termed the "normal efficiency factor." It is the flow delivered by a pump per unit of power utilized. Similarly, a marginal efficiency factor, dQ/dP, equal to the incremental flow delivered per unit increment of power utilized, may be obtained by rearranging Equation (1) as:

$$Q = P \cdot \eta \,/\, k \cdot H \qquad (3)$$

and differentiating:

$$dQ/dP = (\eta + P \cdot d\eta/dP) \,/\, k \cdot H \qquad (4)$$

Thus, dividing Equation (4) by Equation (2), we get:

$$\frac{\text{dQ/dP, the marginal efficiencyfactor}}{\text{Q/P, the normal efficiency factor}} = \frac{\text{slope of the Q-P curve at the operating point}}{\text{slope of the line from the operating point to the origin}}$$

$= 1 + \frac{d\eta/\eta}{dP/P} = 1 +$ percentage increase of efficiency per percentage increase of power.

Since the percentage change of efficiency per percentage increase of power is positive at points below bep, falls to zero at bep (where the efficiency is at its maximum value) and becomes negative for points beyond bep, it follows that the marginal efficiency is equal to the normal efficiency at bep, is higher than the normal efficiency at flows below bep and is lower than the normal efficiency at flows above bep.

This is also evident geometrically from the graph of the Q-P curve, when one observes that the slope of the tangent from the origin to the Q-P curve, which defines, at the point of tangency, the bep point, is both the normal efficiency factor (Q/P, the slope of the line from the origin to the operating point considered on the Q-P curve) and the marginal efficiency factor (dQ/dP, the slope of the Q-P curve at the operating point considered); that is the two factors are equal. At operating points below bep, the slope of the curve is greater than the slope to the origin and at operating points beyond bep the slope of the curve is less than the slope to the origin. This point is further illustrated in Figure 4, where the normal and marginal efficiency factors are shown plotted against flow.

It is interesting to note that the optimum situation, for a required total flow equal to the sum of the best efficiency flows of two different pumps, is not to have each pump running at its best efficiency point. Instead, the lower efficiency pump should be run below its bep and the higher efficiency pump above its bep.

In terms then of the marginal efficiency factor, the general criterion for minimized total pump power for pumps of any type operating in parallel with variable speed, is that all the pumps should run with the same value of the marginal efficiency factor.

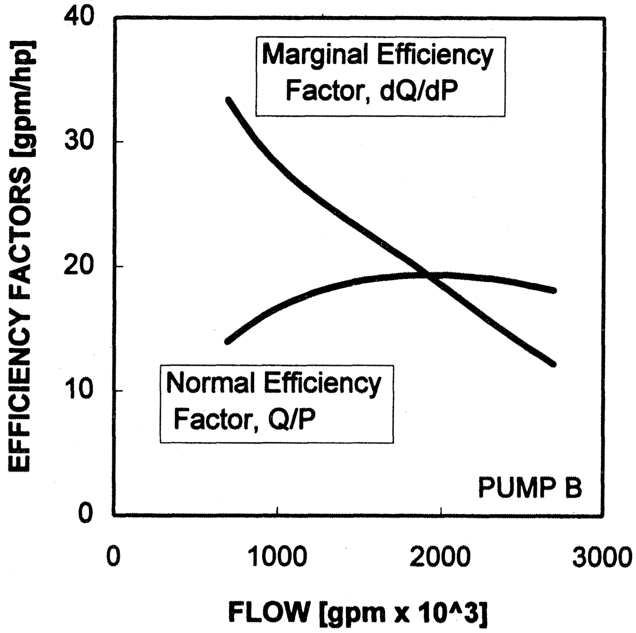


Figure 4. Normal and Marginal Efficiency Factors vs *Flow.*

Determination Of Optimum Operating Points Of Pumps With Different Characteristics

In general, the optimum combination for a given pumping requirement, defined by the values of Qt and head H, could be determined by developing for each pump and head H, the curves of the marginal efficiency factor, dQ/dP, as shown in Figure 3. Then, for each value of dQ/dP, the corresponding total flow could be obtained from ($Q_A + Q_B$), and the speeds required of pumps A and B would be deduced from the values of Q_A and Q_B respectively, together with the known value of head, H. Thus, by repeating the exercise for various values of dQ/dP and H, a matrix of required speeds is obtained that could, at least theoretically, be used to provide optimum control of the two different parallel pumps.

Operation of Identical Pumps in Parallel

Turning now to the common situation of pumps that are identical for practical purposes, it follows, from the general conclusion that the optimum combination of operating points for two pumps is obtained where the values of the marginal efficiency factor, dQ/dP, for each pump are equal, that the two pumps must be run at the same point (same speed, flow, etc.) since the Q *vs* P and hence dQ/dP *vs* P curves, for identical pumps must be identical.

It follows that, contrary to what is often assumed, it is not convenient to operate a group of identical, parallel pumps with all but one at best efficiency and the remaining one controlled to match the total flow to the level desired.

Effect of Performance Deterioration on the Method of Operating Point Control

The question of the best way to control nominally identical pumps, of which the performance of one has been affected by deterioration in service, is complicated by the variety of deterioration mechanisms and the different effects these may have on the pump performance curves [4]. A comprehensive evaluation of the possible effects and their implications for optimum pump control was outside the scope of the present study. Nevertheless, some guidance was required on a specific question, namely: what parameter of the running pumps should be controlled to be the same for all of them.

With all pumps identical, the selection is immaterial, since any parameter would serve and the easiest to measure could be adopted. In the face of possible differences of performance due to deterioration, the question can be reformulated as: what parameter, of those that could be determined with reasonable precision by simple measurements, would, when made equal for both pumps, provide the closest approximation to the desired situation of equal dQ/dP for both pumps. The curves of dQ/dP, for a new and deteriorated pump, are shown in Figure 5, plotted now against Q instead of P. To obtain the curve of the deteriorated pump, constant reductions by 10 percent of bep values are assumed for head and efficiency over the whole flow range. A similar exercise was also carried out for the case of 10 percent reduction of head with seven percent reduction of efficiency, and gave very similar results. Based on the range of deterioration mechanisms discussed by Yedidiah [4], the above two cases give a reasonable representation of the range of possibilities.

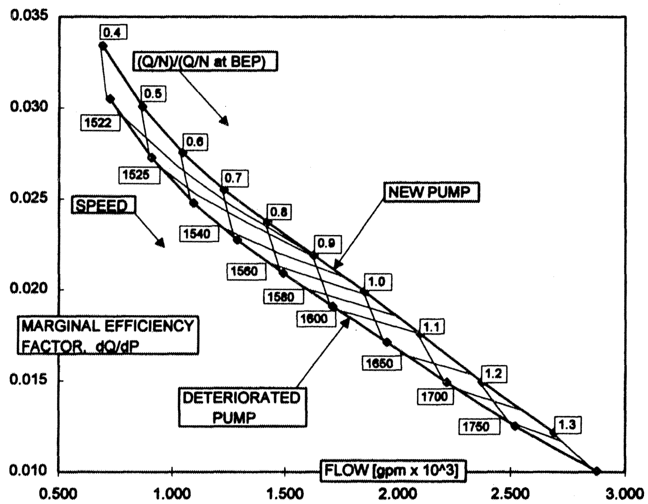


Figure 5. Effect of Deterioration on the Marginal Efficiency Factor vs *Flow Curve of a Pump.*

The curves are marked up to show points of equal Q/N and speed on the two curves. The results show that control of the pumps to equal Q/N is not a good solution since it results in a relatively large difference of dQ/dP between the two pumps. Control to equal speed or flow would give closer and similar, though not complete agreement. Control to equal speed is generally preferable due to the ease and relative accuracy with which speed can be estimated from the excitation frequency in a VFD.

Pumps with "Flat" Curves

The exception to the previous point would be with pumps having curves which are very flat at the lower end of the reliable operating range, as was the case for the pumps selected for the application of this paper. In this case, control to equal speed could well result in unacceptably large differences of flow between the new and deteriorated pumps, as may be clearly seen at low flow in Figure 5. For pumps of this type, control to equal flow is preferable. As an arbitrary criterion of "flat," the following is suggested:

Determine for the pump, the slope, s, of the head – flow curve at the minimum continuous flow point from:

$$s = -(dH/dQ)/(H_{bep}/Q_{bep}) \qquad (5)$$

If s is less than 0.25, the curve should be considered flat and pumps should be controlled on the basis of flow rather than speed.

In practical terms the criterion signifies that a given percent drop in head at a speed would result in a percent change in flow four times higher, for example a 3.0 percent head deficiency would result in a 12 percent drop in flow.

Control to equal values of flow requires that each individual pump flow must be measured. As a side benefit, knowing both the individual flows and speeds allows the relative deterioration between the running pumps to be continuously monitored, and in the event the difference between the worst and best of the running pumps becomes greater than some specified criterion, an alarm can be incorporated in the control system to alert to this situation.

It is recommended that in cases of doubt for any specific application, curves of marginal efficiency function, dQ/dP, be generated as a basis for determining which parameter to use for control. Theoretically, a better approximation to maximum overall efficiency could be obtained by controlling to equal values of a parameter of the type QN^x where x could be found from the curves of dQ/dP as the value which results in a best approximation to equal dQ/dP, and appears to be around 2.0 or 3.0. In practice this would seem to be an unnecessary refinement.

DEFINITION OF OPTIMUM NUMBER OF IDENTICAL PUMPS FOR A GIVEN OVERALL DUTY

Applying the criterion of minimum pump power, the optimum number of identical pumps running in parallel at the same condition is the number that gives lowest total power Pt for a given overall flow Qt and resulting head H. In other words, the optimum number is that which gives highest overall efficiency, k·H·Qt/Pt (where k is the appropriate constant for the unit system adopted).

Now individual pump efficiency is the same as overall pumping efficiency since:

Individual pump efficiency = k·H·Q/P = k·H·n·Q/n.P = k·H·Qt/Pt = overall pump efficiency. So the optimum pump number is also that which results in highest individual pump efficiency.

The question then is how to determine the individual pump efficiency for any given overall duty (Qt and H) and for different numbers of pumps.

For constant speed service, the efficiency is simply the well known function of flow. For variable speed, assuming the affinity laws apply, efficiency can be expressed as an approximately unique (that is, speed-independent) relationship against Q/N.

For the particular number of pumps in service at any moment, it would be an easy matter to determine the Q/N by direct measurement of Q and N. For more or less pumps in service at the same overall duty (Qt and H), the individual pump flow could also be readily determined from the (measurable) total flow and the pump number. However, to obtain the required value of Q/N, the expected speed with more or less pumps would still be required and it is not immediately obvious how this may be estimated.

Instead, the efficiency and corresponding values of Q/N for different pump numbers and overall duty, may be deduced directly from curves of efficiency *vs* H/Qt^2, for each pump number. The reasoning is as follows:

For an individual pump, assuming the affinity laws apply, the efficiency and normalized head, h ($=H/N^2$), are functions of Q/N, as shown in Figure 6.

Now: $$H = h \cdot N^2 \quad (6)$$

and : $$Q = q \cdot N \quad (7)$$

From (7): $$Q^2 = q^2 \cdot N^2 \quad (8)$$

From (6)/(8): $$H/Q^2 = h/q^2 \quad (9)$$

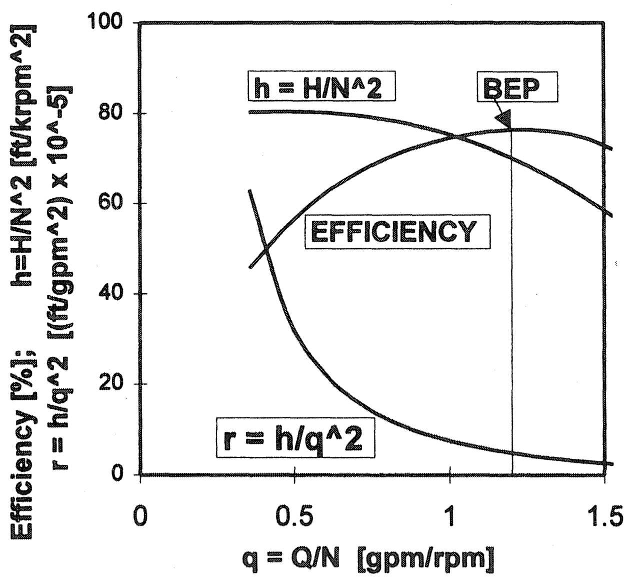


Figure 6. Pump Performance Expressed in Terms of Normalized (Speed-Independent) Parameters.

Using the symbol, r, for h/q^2, note that since h is a unique function of q, then r must also be a unique function of q. And since the efficiency is a function of q, it can also be presented as a function of r, with the advantage that r only depends on the overall duty and the number of pumps. That is: $r = H/Q^2 = H/(Qt^2/n^2) = n^2 \cdot H/Qt^2$. This signifies that it must be possible to present the efficiency as a function of H/Q^2 and the number of pumps n, which will provide a convenient way of selecting the optimum pump number for a given duty.

The various functions of q of interest can be determined in the following way, using the subject pumps discussed herein as an example. From a table of several values of head and efficiency *vs* Q for a given speed, the corresponding values can be obtained of h and efficiency *vs* q. These values can then be fitted by polynomials to provide smooth continuous functions of q. Using these functions, a table such as Table 1 can be developed using a simple spreadsheet program.

Table 1. Data Development for the Curves of Efficiency and Q/N vs *H/Qt^2.*

n. No. of pumps	q/q.BEP	q=Q/N	Efficiency	h=H/N^2	H/Q^2 = (H/N^2)/(Q/N)^	H/Qt^2 = (H/Q^2)/n^2
		[gpm/rpm]	%	[ft/pm^2]	[ft/gpm^2]	(ft/gpm^2)*E-06
1	0.4	0.476	54.93	8.04E-05	3.5E4-04	354.24
1	0.5	0.596	61.93	8.02E-05	2.26E-04	226.06
1	etc.	....	....	....	....	....
1	1.2	1.429	74.71	6.16E-05	3.02E-05	30.18
1	1.3	1.548	71.39	5.62E-05	2.35E-05	23.46
2	0.4	0.476	54.93	8.04E-05	3.54E-04	88.56
2	0.5	0.596	61.93	8.02E-05	2.26E-04	56.52
2	etc.	....	....	....	....	....
2	1.2	1.429	74.71	6.16E-05	3.02E-05	7.54
2	1.3	1.548	71.39	5.62E-05	2.35E-05	5.86
etc.						

The values of efficiency and q can then be plotted against H/Qt^2 for different numbers of pumps using the relevant columns of the table, as shown in Figure 7. The intersection of the curves of efficiency *vs* H/Qt^2 indicate the points where a change of pump number is required to maximize pumping efficiency.

In practice it is desirable to maintain a pump combination in service until the addition or removal of one pump would result in an improvement of the efficiency by a small amount (chosen to

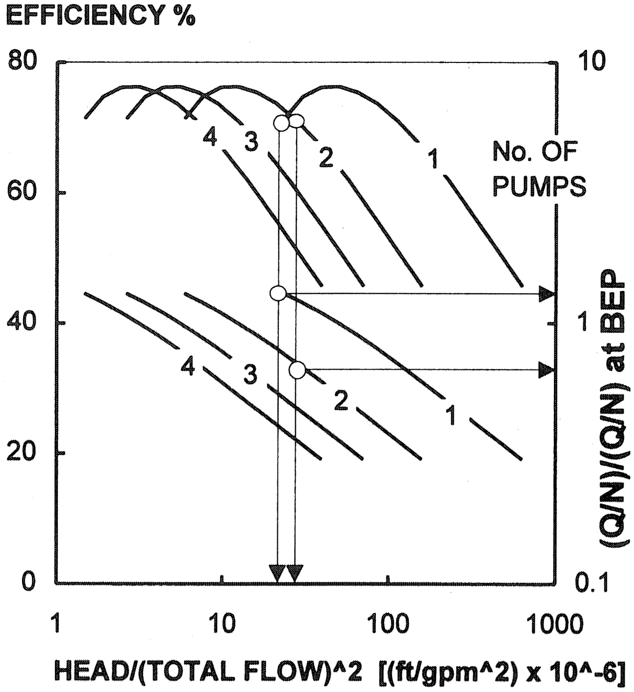


Figure 7. Determination of the Optimum Number of Pumps for Any Operating Condition (H & Qt) as a Function of the Terms H/Qt^2 and Q/N (Showing Change Points Between 1 and 2 Pumps).

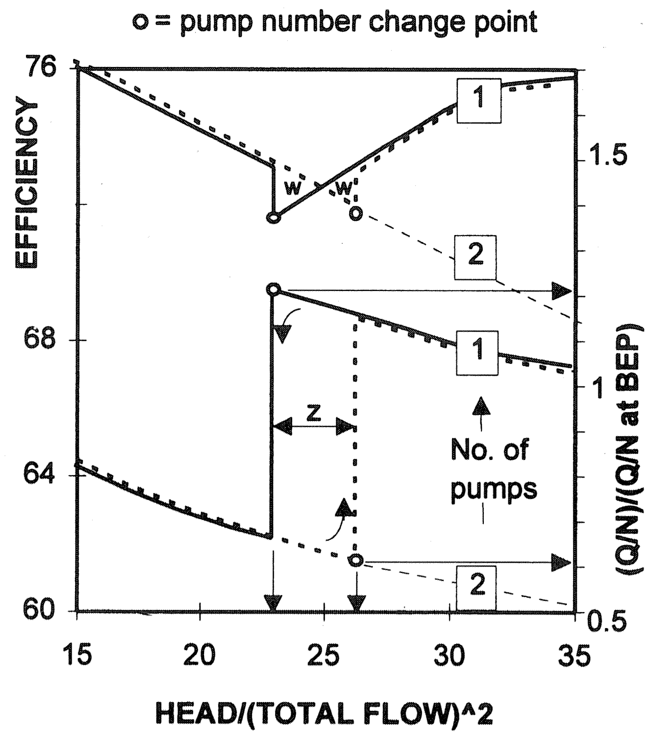


Figure 8. Zoom of the Optimum Change Point Between 1 and 2 Pumps.

avoid unnecessary cycling of the pump number). A zoom of the region of changeover between 1 and 2 pumps is shown in Figure 8. The solid line shows the situation during flow increase, when the pump number has to be increased, and the dotted line represents the situation during flow decrease, when the pump number has to be reduced. Using a suitable value for the separation, z, between the change points, to obtain the desired hysteresis effect, the specific values of H/Qt^2 and q (=Q/N) can be chosen and incorporated into a PLC based control algorithm. The control values which were determined for the application of this paper are shown in Table 2.

Table 2. Parameter Values at the Optimum Change Points.

CHANGE POINT:	CORRESPONDING VALUES FOR THE RUNNING PLUMPS OF:		
	H/Qt^2	(Q/N)/(Q/N at BEP)	Q/N
	[ft/gpm^2]		[gpm/rpm]
From 1 pump to 2	23.0E-6	1.25	1.49
From 2 pumps to 3	7.0E-6	1.22	1.45
From 3 pumps to 4	3.4E-6	1.2	1.43
From 4 pumps to 3	4.4E-6	0.85	1.01
From 3 pumps to 2	8.5E-6	0.8	0.95
From 2 pumps to 1	27.0E-6	0.7	0.83

It should be noted that when the control system has to adjust flow to control another process variable such as level, it should be designed and tuned to minimize the amplitude of flow fluctuations associated with control of the process variable. In this way the separation, z, between the pump number change points during flow increase and flow decrease, can be reduced, and the potential efficiency loss near the change point (areas w, for waste, in Figure 8) can be minimized.

In addition all signals that go to the PLC for control purposes should have a damping function applied to them either at the signal source or electronically in the PLC, to avoid undesired fluctuations of speed and start/stop of pumps.

Alternative Parameters for Defining the Pump Number Change Points

Definition of the change points in the control algorithm can be on the basis of H/Qt^2, H/Q^2, Q/N, or (Qt/n)/N, depending on what parameters are to be measured and fed to the PLC. The first two can conveniently be transformed to a ratio of differential pressures that not only increases the accuracy of their determination via measurement and division in the PLC, but also makes the parameter independent of the specific gravity of the fluid. This is a distinct advantage in applications where the gravity may vary significantly, as is the case in the application of this paper due to substantial variations in the water content of the crude/water mixture.

The transformation of the H/Q^2 terms into a ratio of differential pressures is achieved in the following way:

$$H = (p_d - p_s)/\rho g \tag{10}$$

and, using the equation for the orifice employed for the flow measurement:

$$Q^2 = C^2 \cdot A^2 \cdot 2g.\Delta p/\rho \tag{11}$$

where C = orifice flow coefficient

ρ = density of fluid

and Δp= differential static pressure across the orifice.

Thus, dividing Equation (5) by Equation (6), we obtain:

$$H/Q^2 = r = K \cdot (p_d - p_s)\,\Delta p \tag{12}$$

where K is a constant.

If, as discussed in the previous section, individual pump flows are to be measured to enable controlling of individual pumps on the basis of flow rather than speed, $(p_d - p_s)/\Delta p$ or Q/N could be used as the parameter that determines changes of pump number in the control algorithm. The former would be preferred for applications with variable gravity.

If the pumps have a steep curve, the control of individual pump operating points could be done on the basis of speed equalization and change points could be derived from the total flow, using $(p_d - p_s)/\Delta p_t$ or (Qt/n)/N. Once again the former would be preferable for variable gravity situations.

CONCLUSIONS

• For best overall efficiency, identical pumps running in parallel should all be operated at the same condition; it is not convenient to run some pumps at best efficiency with others removed from the bep.

• The control of nominally identical, variable speed, parallel pumps should include the following features:

- (Protection of a reliable operating range using maximum (and if necessary minimum) allowable speeds and maximum and minimum values of Q/N. Control, alarm and shutdown values should be incorporated for each limit.
- If the pumps have a flat curve ($s < 0.25$), control all running pumps to equal individual flow. If the pumps have a steep curve ($s > 0.25$), control to equal speed.
- Control the start and stop of pumps on the basis of predetermined values of the parameters q = Q/N, or $(p_d - p_s)/\Delta p$, and include some measure of "hysteresis" in the values to avoid cycling on and off of pumps when the required duty is close to a change point. Use the procedure discussed herein for determining the change values. Use $(p_d - p_s)/($p as the controlling parameter, if the fluid gravity is subject to appreciable variation. If not, either parameter may be used.
- If it is chosen to measure individual pump flows (or Δp's) as well as speeds, monitor the differences between the individual values of Q/N (or $\sqrt{\Delta p}/N$) for the running pumps and incorporate an alarm to signal unusually low Q/N relative to the pump with highest Q/N, in order to detect and warn of excessive pump deterioration.
- If it is required to operate pumps with different characteristics in parallel, a suitable control algorithm for this purpose could be developed using the methodology indicated herein, which is based on minimization of the energy input to the pumped fluid and translates into a criterion of running the pumps at equal values of a "marginal" efficiency factor, the slope at the operating point of the constant head, variable speed, Q-P curve for each pump. An additional basic requirement to allow nonidentical pumps to be run satisfactorily in parallel is that each, at its maximum speed and minimum continuous Q/N value, should generate a head at least equal to the maximum possible head which could arise in any possible operating condition of the system.

NOMENCLATURE

Q,P	Flow, power for an individual pump
Qt, Pt	Total flow and power for a number of pumps in parallel
N	Pump speed
dQ/dP	Slope of the curve of Q *vs* P for a pump at constant head and variable speed
H	Differential head of a pump or a group of pumps in parallel
η	Efficiency of a pump
n	Number of running pumps in parallel
h	H/N^2
q	Q/N
r	h/q^2
p_d, p_s	Discharge, suction pressures of a pump
ρ	Fluid density
g	Acceleration due to gravity
Δp, Δp_t	Differential static pressure across an orifice for measuring individual, total flow
s	slope of head-flow curve at minimum flow $= -(dH/dQ)/(H_{BEP}/Q_{BEP})$

REFERENCES

1. Murphy, S.P., "Application of Variable Speed Electric Motors for Pumps," *Proceedings of the Tenth International Pump Users Symposium,* Turbomachinery Laboratory, Texas A&M University, College Station, Texas (1993).
2. Sulzer Centrifugal Pump Handbook, 3rd. Edition, Sulzer Brothers Limited, Winterthur, Switzerland (1987).
3. Frei, A., Grgic, A., Heil, W. and Luzi, A., "Design of Pump Shaft Trains Having Variable-Speed Electric Motors," *Proceedings of the Third International Pump Symposium,* Turbomachinery Laboratory, Texas A&M University, College Station, Texas (1986).
4. Yedidiah, S., *Centrifugal Pump Problems–Causes and Cures,* pp. 4-12, Tulsa, Oklahoma: Petroleum Publishing Company (1980).