IMPROVING RELIABILITY IN A HIGH STATIC HEAD SYSTEM THROUGH VFD APPLICATION

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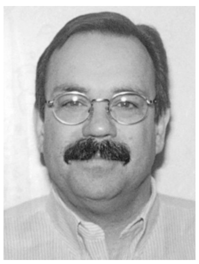
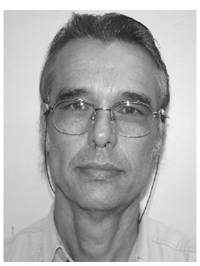


Guilherme Martins is with Petrobras, in Rio de Janeiro, Brazil. In 2008, he was invited to lead the Dynamic Equipment area of COMPERJ, the biggest petrochemical complex of Brazil, still under construction. He joined Petrobras in 1989, and began working with rotating equipment maintenance. He was transferred to the process area dealing with rotating equipment optimization, operational proceeding improvements, and operator training, moved to the Mechanical Projects area and, finally, to the Rotating Equipment Reliability area. Since late 2003 he has been developing experiments related to Process Control through Centrifugal Pump Speed Variation, with his colleague Enio Lima, at Duque de Caxias Refinery.

Mr. Martins received his M.S. degree (Mechanical Engineering, 1988) from Universidade Federal do Rio de Janeiro. He is a registered Professional Engineer in the State of Rio de Janeiro.



Enio Lima is with Petrobras, in Duque de Caxias, Brazil. He joined Petrobras in 1975, and started working with design and manufacture of petroleum equipment spare parts. He then moved to rotating equipment maintenance, dedicating most of his time to turbomachinery failure prevention and repair. In late 2003, he started developing experiments concerning process control through pump speed variation, in partnership with his colleague Guilherme Martins. Together, they have successfully installed this control system in dozens of pumps at Duque de Caxias Refinery. Since 2006 he has trained hundreds of his colleagues in this new control technology, and has been acting as a technical consultant for issues regarding the efficiency and reliability of pumping systems all over the company.

Mr. Lima received his M.S. degree (Mechanical Engineering, 1974) from Universidade Federal do Rio de Janeiro. He is a registered Professional Engineer in the State of Rio de Janeiro.

ABSTRACT

The static head content in the total head of a pumping system (static head factor—f_{HS}) plays an important role on energy and reliability potential gains attainable with the use of variable speed (VS) controls in pumping systems. In spite of this control method almost always enable important gains when compared to throttle control at constant speed, as a rule of thumb "the higher the f_{HS}, the smaller the energy and reliability gains and the higher the risk of facing an unstable pumpage (sinusoidal shape flowrate)."

The main point of this presentation is to demonstrate that even in pumping systems with f_{HS} as high as 80 percent it is still possible to save large amounts of energy and improve reliability besides an effective control of the pumpage variables.

First the present case will be described. Second, fundamental concepts of control under variable speed will be brought up in order to facilitate understanding of the applied methodology. Third, the calculation approach and graphical analysis will be shown step by step and the applied solutions will be detailed as well. Finally field records will be shown and discussed.

INTRODUCTION

B-01 A/B are two pumps, main and spare, which belong to a methyl ethanolamine (MEA) solution pumping system, each with 400 hp (298 kW), 572 psi (4,000 kPa) from an ethane treatment unit. This process is responsible for separating the CO_2 content from the feedstock and sending the treated gas directly to a process plant of a neighboring industry. Consequently, any interruption or specification loss may result in high penalties for the refinery. Therefore, reliability in this system must be high. Please refer to process flow in Figure 1.

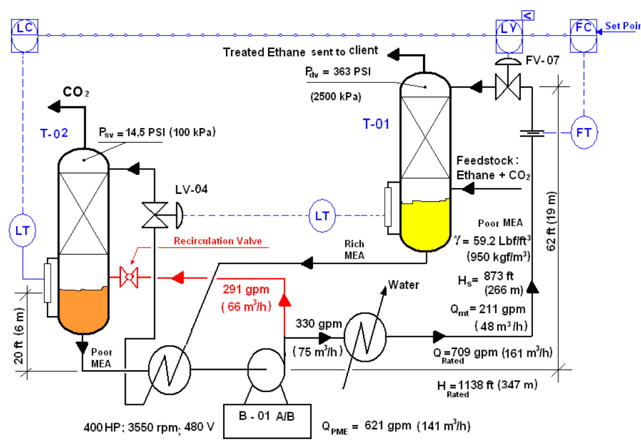


Figure 1. Simplified Process Flow Diagram of B-01 A/B system.

Each pump originally operated on its best efficiency point (BEP), 621 gpm (141 m^3/h), pumping 330 gpm (75 m^3/h) to the process and 291 gpm (66 m^3/h) to recirculation.

A drawback of this system was a manually actuated globe-valve, used to regulate the recirculating flow, back to tower T-02. The pressure drop in this valve was around 430 psi (3,000 kPa), making it vibrate strongly, with a loud noise, loosening its components and leaking to the environment. The recirculation flow was so powerful that it eroded a hole in the opposite tower sidewall, forcing the unit into an emergency situation. In August 2007 that globe valve failed definitively and, as another shutdown was impractical at that moment, it was decided to provisionally operate the unit with that valve blocked.

A few days later pump B failed and, a week later, pump A failed too, both badly damaged. The root cause of both failures was identified as lack of hydraulic fitness under eventual low flow rates. The above facts suggested that the definitive solution for this problem should include operation with the recirculating valve closed, to save energy, along with the pump hydraulically protected, to improve reliability.

The very first idea, soon discarded by the authors, was to change the manually actuated globe valve with an automatic control valve adequately designed for that purpose. Soon the authors realized that recirculation is an essentially wasteful practice and thereby opted for the use of a variable speed (VS) control. The expectation was that besides the energy savings, this control technology would increase the mean time to failure (MTTF) of the pumps and probably would protect them against flowrates as low as 211 gpm (48 m^3/h).

To implement that solution some challenges had to be overcome. The idea was to reduce pump speed and to widely open the control valve. Even the suppression of that valve was considered. The authors expected that by reducing rotation speed, a considerable differential pressure decrease could be achieved, making low flow conditions viable with the recirculation valve completely closed.

FACING THE CASE

To meet the above expectations the authors faced a huge challenge: to precisely control the operating point of the system, situated on the intersection of two curves, pump and system, almost parallel, especially on the left-hand side of the operating range of 211 to 709 gpm (48 to 161 m^3/h), as shown in Figure 2. Under this condition, even very low speed variations may provide high flow differentials.

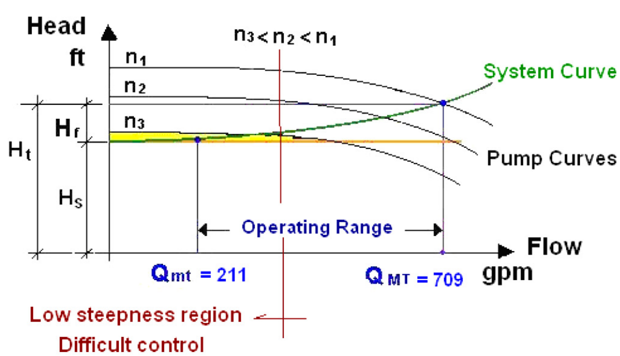


Figure 2. High f_{HS} Systems—VS Control Difficulties under Low Flowrates.

The speed interval, as a result, is also quite narrow, something around 500 rpm. These undesirable limitations are intrinsic consequences of the high static head factor (f_{HS}) of this system, around 80 percent.

There was also a second challenge: to acquire the actual system curve. The operators did not agree in stressing the system above a certain level.

NOTIONS ABOUT PROCESS CONTROL THROUGH VARIABLE SPEED

This control approach, in spite of its simplicity, is a relatively recent technology that involves many paradigm changes and some specific concepts that may be new for a part of the audience. Some of them, such as static head factor and its influence on pumping system efficiency and controllability, critical point (Pc), rotation minimum stop (RmS), destructive power (P_{WRd}) and specific energy (Es), will be brought up in order to facilitate the understanding of how these parameters were achieved and how far they impact on system reliability, controllability and energy savings.

System Head × Flow Curve

Figure 3 shows the amount of energy per unit weight necessary to move the fluid through the system. Each flowrate, indicated on the horizontal axis, is related to only one head on the vertical axis, generating this way, the curve shown in Figure 3, generically called system curve. Therefore, to deliver through the above system a flowrate of Q_M (gpm) an amount of energy or a total head of H_T (ft) is necessary.

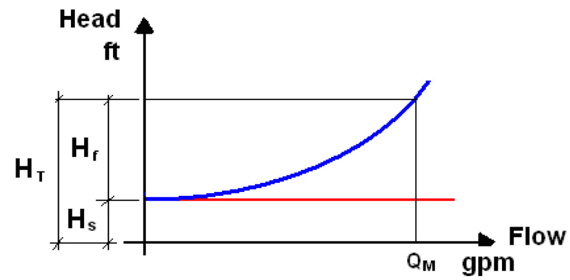


Figure 3. System Flowrate × Head Curve.

System Characteristics

The total head, H_T, generally is composed of two parts: the static head, H_S, and the friction head H_f. Thus: $H_T = H_S + H_f$ (ft or m). The first one, H_s, is the result of the differential level between delivery and supply vessels added to the differential pressure between these vessels, divided by the specific weight γ (lbf/ft^3 or N/m^3). So:

$$H_S = (Z_2 - Z_1) + (P_{dv} - P_{sv}) / \gamma \quad \text{(ft or m)} \tag{1}$$

where:

Z_2 = Elevation of the liquid entrance/surface in the delivery vessel (ft or m)
Z_1 = Elevation of the liquid surface in the supply vessel (ft or m)
P_{dv} = Delivery vessel pressure (psi or kPa)
P_{sv} = Supply vessel pressure (psi or kPa)
γ = Specific weight (lbf/ft^3 or N/m^3)

It is important to remember that the static head, H_S, in most cases, is a fixed part, intrinsic to each system, independent of flow, unlikely to be significantly changed. It is related to the amount of energy that the pump must transfer to the fluid to allow flow to start up.

The second part, H_f, represents the friction losses to be overcome in the piping, valves and fittings. Thus H_f depends on the piping roughness, flow velocity, number and type of fittings, etc., but mainly on the control valve opening.

The friction head is the variable portion of the total head of a system, over which it is possible to save energy and decrease equipment stress whenever flow or pressure demand are lower than the rated ones.

Static Head Factor (f_{HS})

Figure 4 shows several curves of different pumping systems operating at the same duty point: Q_M = 220 gpm; total head H_T = 100 ft. Depending on the considered system the static head may vary from negative values, −20 ft up to very high values, near 100 ft. As $f_{HS} = H_S/H_T$, f_{HS} in the above figure may vary from −20 percent (friction head dominated systems) up to 100 percent (static head dominated systems).

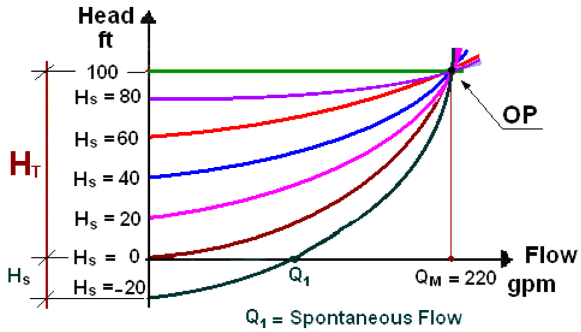


Figure 4. Systems with Different Static Heads.

In the first case, of negative f_{HS}, the suction side of the system has an intrinsic energy of pressure and elevation, able to sustain a flow (spontaneous flow) even when the pump is off. In this case the pump is necessary just to complement that flow to the value required by the process. In the second case, the pump has to run at considerably high speeds to overcome the system back pressure and enable flow to start up.

Regarding the above, one can define the parameter called "static head factor," f_{HS}, which represents the amount of static head, H_S, enclosed in the total head of the system, H_T. Thus: $f_{HS} = H_S/H_T$ or $f_{HS} = H_S/(H_f + H_S)$.

f_{HS} Influence on Potential Gains and on System Controllability

Suppose that the pump shown in Figure 5 was installed to operate at the same duty point (220 gpm; 100 ft) against any one of the plotted systems. In systems with low f_{HS}, for instance 20 percent, there is a part of 80 percent of the H_T in which economy is possible whenever flow or pressure demand is lower than the rated ones.

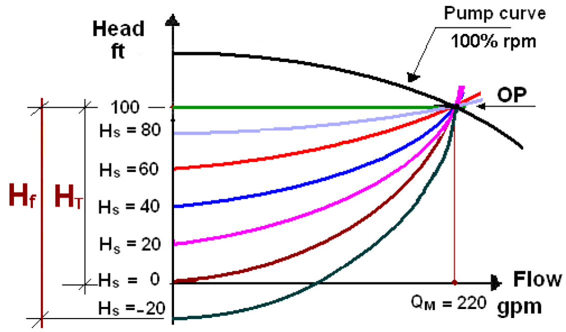


Figure 5. Pump Performance Curves—Same Operational Point.

In other words pump speed may be reduced bringing about reductions on discharge pressure, energy consumption and mainly on system deterioration, especially concerning the pumps (Figure 6). Oversized pumps are extremely common in refineries and other industrial plants all over the world. Therefore the potential gains in low f_{HS} systems are in general considerably high.

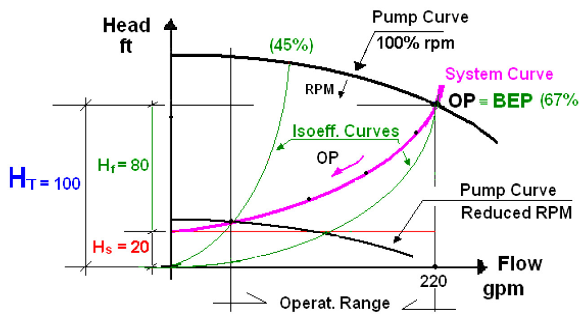


Figure 6. Potential Gains in Low f_{HS} Systems.

Suppose that the above pump is operating at its best efficiency point (BEP) (220 gpm; 100 ft). Notice that when the pump slows down, the operating point (OP) moves leftwards on the system curve (in pink), in a way almost parallel to the best isoefficiency curve (right-hand green curve). This means that, especially under medium and higher flow rates, the efficiency remains always high. Even under low flowrates, the efficiency losses are not critical. Notice in Figure 6 that, if the f_{HS} were zero, even at extremely low flowrates, the pump would still be running at its highest efficiency.

Another important point is that, within all the operating range, the angle between the tangent of both curves, pump and system, at the OP, remains adequately high, ideal for an easy and precise process control through speed variation. When this angle tends to zero, in other words, when both curves become flat, a small speed variation results in a significantly high flowrate fluctuation, making control tend to instability (sinusoidal flowrate). The concept of flow instability is better explained in the section "*Rotational Minimum Stop*," below. It is worth mentioning that a sinusoidal shape flowrate generates a sequence of accelerations and decelerations on fluid movement and, in its turn, cyclic torsional stresses in the pump shaft. This process may lead to pump shaft torsional fatigue failure.

Now consider a second system with a f_{HS} = 80 percent, like the one shown in Figure 7. Now only 20 percent of the pump's total energy, H_T, is available for any equipment or energy savings. It becomes clear that now we are dealing with a system much more difficult to control and much more limited in any type of gain. A system like this one will be analyzed further in this paper. Notice that now, when the OP moves leftwards on the system curve toward lower flowrates, both curves get quite flat and control, as mentioned before, becomes much more difficult.

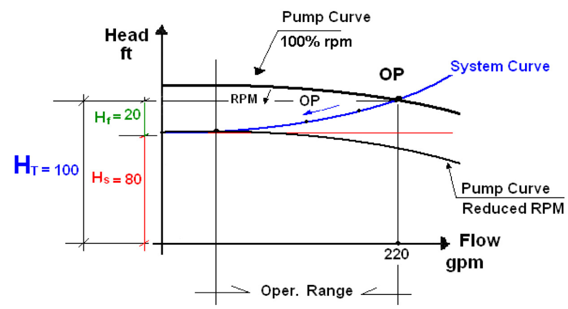


Figure 7. Potential Gains in High f_{HS} Systems.

On the other hand one can observe, in Figure 8, that pump efficiency no longer keeps its reasonably high values all along the operational range. In contrast, efficiency values drop to much lower values especially under low flowrate conditions and may even fall to zero (refer to the lower part of Figure 8) if the system static head line (in red) is completely crossed by the pump curve (in black).

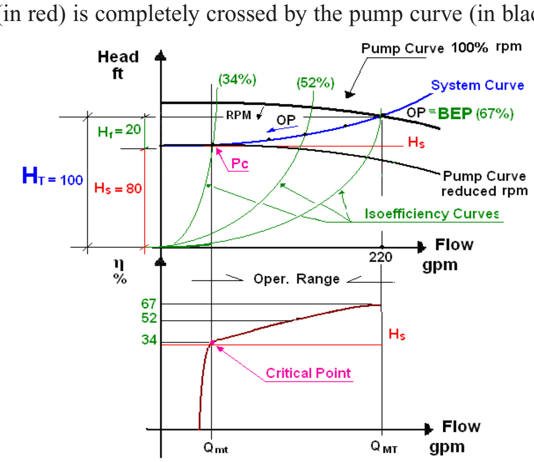


Figure 8. Pump Efficiency Evolution in High f_{HS} Systems.

Critical Point (Pc)

The above statement leads us to identify another important parameter when dealing with variable speed control in high f_{HS} applications: it is the critical point. Pc may be defined as the point that, when surpassed by the pump curve during decreasing speed, any further decrement may cause flow instability or even flow interruption. Refer to the upper part of Figure 8.

For high f_{HS} systems the head of the critical point is around 2 to 5 percent above the system static head (red horizontal line), depending on the system configuration.

Rotation Minimum Stop (RmS)

In high f_{HS} systems, rotation cannot be reduced indefinitely when searching for lower flowrates. Slowing down the pump also reduces its differential pressure and if this process is not stopped, the OP surpasses the system's static head line (red H_S horizontal line in Figure 8) and flow is momentarily interrupted. The control system, facing the flow interruption, causes rotation to rise generating a high flowrate that, in its turn, makes the control slow down the pump again. A sinusoidal flowrate is then settled.

In order to solve this problem another parameter is made necessary. It is the *rotation minimum stop*. This parameter is essential to assure reliable results in high f_{HS} VS applications. If further flow reductions are necessary they must be achieved through control valve modulation, according to Martins and Lima (2008).

Static Head Factor X Affinity Laws

Another extremely important point, regarding VS applications in high f_{HS} systems, is the calculation of a new operational condition (pressure, flow and power) when speed is changed.

The affinity laws are based on constant efficiencies along speed variations. This premise is only perfectly correct in systems where f_{HS} is nil such as, for instance, centrifugal pumps test benches or closed loop pumping systems. Actually many systems have a significant f_{HS} and the higher the f_{HS}, the bigger the error when directly applying the affinity laws on the above calculations.

Example: according to the affinity laws, flow is directly proportional to rotation speed or $Q \propto N$. In Figure 8 one can easily estimate that decreasing rotation speed by (−½), the new flowrate will be zero and not 110 gpm (220/2), as suggested by the affinity laws. This happens because the new head delivered by the pump is not sufficient to overcome the system backpressure. In these cases calculation requires some special procedures that will be discussed in this presentation.

Specific Energy (Es)

Specific energy is probably the most important *pumping system efficiency indicator* since it considers all kinds of losses, for each pumping condition. Friction losses, especially in the control valve, f_{HS}, pump efficiency, motor and converter efficiencies and finally fluid characteristics, are all taken into account.

Es is measured in hph/gallon (kWh/m^3) and represents, for each pumping condition, the energy necessary to deliver one gallon (or one m^3) through the system. One can understand that, for a certain flow rate, the lower the Es, the lower the energy required to move each gallon of fluid through the system and, therefore, less energy will be available to wear out equipment and all other components of the system. Thus Es may be considered an indicator deeply related to systems reliability, component lifespan and naturally with energy savings.

Additionally the distributed control systems (DCS) (process computer) can easily calculate the Es by dividing the power output measured in the frequency converter by the flowrate measured in the process. Associating the above information with the energy ratio, the DCS can display online, for each operational condition, in \$/gallon (\$/m^3), the actual cost for pumping each gallon. This information not only allows the operator to choose the most economical operating condition for that system but also allows the plant manager to have a permanent control of his process efficiency and energy costs.

Destructive Power (P_{WRd})

P_{WRd} is related with the amount of power transmitted to the pump shaft ($P_{WRshaft}$) and not converted into flow. In other words it represents the fraction of the power input that is mainly used to wear out the equipment and system components. So $P_{WRd} = (1-\eta/100) \times P_{WRshaft}$.

For each pumping condition, of course, there will be a specific P_{WRd}. *The lower the P_{WRd}, the lower the equipment wear and its energy consumption*.

The maximum allowable P_{WRd} level of a pump in a system may be used as a selection criterion for the pump and its control system.

RETURNING TO THE CASE

Below are some complementary data of B-01A/B:

- Rated point: Q_{Rated} = 709 gpm (161 m^3/h); H_{Rated} = 1138 ft (347 m); FV = 100 percent open ➜ from DS (data sheet)
- Normal operating point: Q_{normal} = 647 gpm (147 m^3/h; H_{normal} = 1230 ft (375 m); FV = 70 percent ➜ from DS; Pumping temperature = 199°F (93°C); rotation = 3550 rpm; γ = 59 lbf/ft^3 (950 kgf/m^3) ➜ from DS
- Impellers: $\Phi_{max.}$ = D_M = 15/15 in (380/380 mm) ➜ from DS; Φ_{rated} = D_R = 13.386/12.992 in (340/330 mm) ➜ from DS
- Best efficiency point with maximum impeller: $Q_{BEP\text{-}DM}$ = 757gpm (172 m^3/h); $H_{BEP\text{-}DM}$ = 1843 ft (562 m) ➜ from original equipment manufacturer (OEM) curves
- Minimum transient flow: Q_{mt} = 211 gpm (48 m^3/h) ➜ from DCS recordings
- Maximum transient flow: Q_{MT} = Q_{Rated} = 709 gpm (161 m^3/h); with FV = 100 percent open ➜ from DS
- Most common flowrate: Q_{MF} = 330 gpm (75 m^3/h) ➜ from DCS recordings
- Case maximum allowable working pressure = $MAWP_{(case)}$ = 725 psi (5000 MPa) ➜ from DS
- Minimum thermal flow ➜ 110 gpm (25 m^3/h) ➜ from DS

OEM performance curves, plotted in Figure 9, show that these pumps operate, main and spare, with the rated rotors, according to the brown curve. The blue curve refers to the maximum diameter rotors.

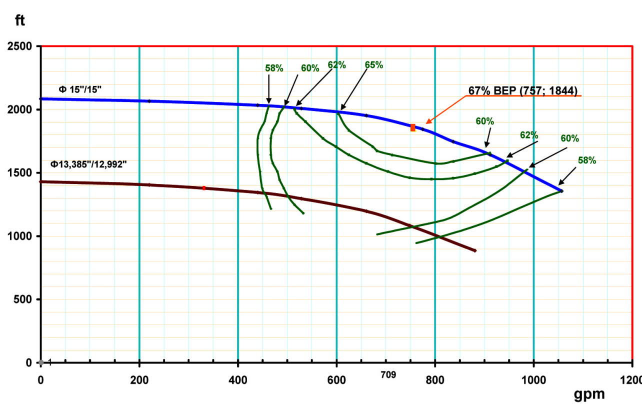


Figure 9. OEM Performance Curves.

The solution implemented to improve reliability and efficiency in this pumping system is developed below in 11 steps.

Hydraulic Fitness Analysis

Q_{BEP-DR} is defined as the best efficiency point flowrate with rated impeller. API 610 considers a pump correctly fitted in its system, with a rated impeller, when the flow range delivered to the system is between 0.7 × (Q_{BEP-DR}) and 1.1 × (Q_{BEP-DR}). Thus it is necessary to know Q_{BEP-DR} in order to calculate the ratio $Q/(Q_{BEP-DR})$.

To calculate Q_{BEP-DR}, it is necessary to apply the affinity laws. This application supposes constant efficiency through all the speed range. So the BEP for the rated impeller and for the maximum impeller must be on the same isoefficiency curve. Thus we first need to plot in Figure 9 the best isoefficiency curve. In the intersection of that curve with the pump curve for the rated diameter (D_R), one can determine the correct pump efficiency at that point.

To plot the best isoefficiency curve at least two points are necessary, since it is a quadratic function of flow (parabola) with nil derivate at the origin. The curve equation is of the type $H = K \times Q^2$, where K is a constant. The first chosen point is the origin itself (0; 0). The second point may be the BEP for the maximum diameter impeller. From the complementary data: Q_{BEP-DM} = 757 gpm (172 m^3/h); H_{BEP-DM} = 1843 ft (562 m). So, using USC units: $H_{BEP-DM} = K \times (Q_{BEP-DM})^2 \rightarrow K = 1843/(757)^2 = 0.0032$. Thereby the equation is:

$$H = 0.0032\ Q^2. \qquad (2)$$

Using SI units: $K = 562/172^2 \rightarrow K = 0.0019$. The equation is:

$$H = 0.019\ Q^2. \qquad (3)$$

Now we have to plot this curve in Figure 9, above, by choosing five or more flowrates and calculating the respective heads. These data are shown on Table 1. Plotting the Table 1 data in Figure 9 one obtains the graphic shown in Figure 10.

Table 1. Best Isoefficiency Equation and Curve.

ISOEF. 67 percent		ISOEF. 67 percent	
$K_{67percent}$ = 0.0032		$K_{67percent}$ = 0.019	
H = 0.0032 x Q^2		H = 0.019 x Q^2	
Q	H	Q	H
gpm	ft	m^3/h	M
0	0	0	0
88	26	20	8
220	154	50	47
308	305	70	93
440	620	100	189
572	1050	130	320
660	1394	150	425
709	1608	161	490
757	1844	172	562

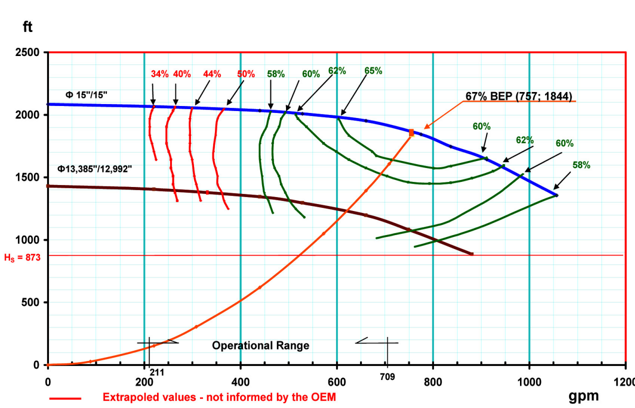


Figure 10. Best Isoefficiency Curve.

From Figure 10 one can easily see that Q_{BEP-DR} = 621 gpm (141 m^3/h). Entering this value in Equation (2): $H_{BEP-DR} = 0.0032 \times 621^2 = 1234$ ft or $H_{BEP-DR} = 0.019 \times 141^2 = 378$ m. Q_{mt}/Q_{BEP-DR} = 211/ 621 = 48/141 = 0.34 ➜ 34 percent.

As 34 percent is much lower than 70 percent one can conclude that the calculated value is far below API standard recommendation (for a reliable operation with the recirculation valve completely closed). On the other hand recirculation is an essentially wasteful practice that should be, whenever possible, avoided, especially in cases like this one, where the valve operates under extreme conditions and plays a vital role in system reliability.

Calculating the System Static Head

Using data from Figure 1 in Equation (1): $H_S = P + Z$ = [(2500 100)kPa × 10000 /(100 × 950 Kgf/m^3)] + (19 – 6) m; H_S = (253 + 13) m; H_S = 266 m or H_S = 873 ft.

Calculating the System Curve
for the FV at 100 Percent Opening

Due to difficulties in making significant changes in operational conditions, these data were based on the OEM curves, datasheet and some DCS readings.

Selecting the Points (At Least Two Points Are Needed)

- Rated Point: Q_{Rated} = 709 gpm (161 m^3/h); H_{Rated} = 1138 ft (347 m)
- Shut off: Q(0) = 0; H(0) = 873 ft (266 m) ➜ H_S of the system

Calculating the Constant
of the Equation and Curve Points

$H_R = H_S + K \times (Q_R)^2 \rightarrow K = (H_R - H_S)/(Q_R)^2 = (1138 - 873)/709^2$ = 0.00053 or in SI units, $\rightarrow K = (347 - 266)/(161)^2 \rightarrow K = 0.0031$.

Plotting data of Table 2, Figure 11 may be generated. The operational range was also plotted. It is important to notice that the efficiencies plotted in red were initially estimated by the authors. Afterwards, with the frequency converters in operation, based on the recorded values, they were recalculated to the plotted values.

Table 2. Equation and System Curve.

FV = 100 percent		FV = 100 percent	
K_{Rated} = 0.0031		K_{Rated} = 0.00053	
H = 266 + 0.0031 x Q^2 Equation (3)		H = 873 + 0.00053 x Q^2 Equation (3)	
Q	H	Q	H
m3/h	m	gpm	ft
0	266	0	873
20	267	88	877
50	274	220	898
70	281	308	923
100	297	440	974
150	336	660	1102
161	346	709	1136

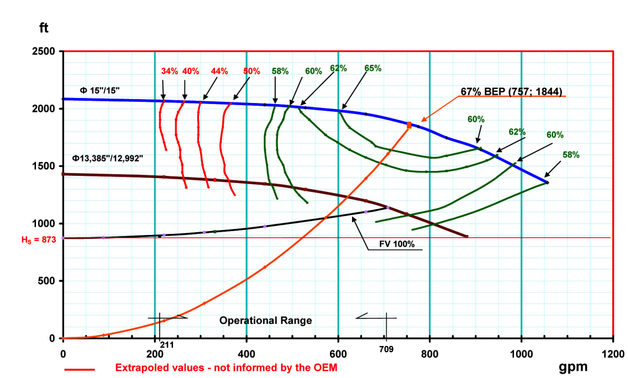


Figure 11. System Curve with FV 100 Percent Open.

Calculating f_{HS} of the System

$f_{HS} = H_S/(H_S + H_f) = H_S/H_T = 873/1{,}138 = 0.77$ or, in SI units: $f_{HS} = 266/347 = 0.77$ ➜ $f_{HS} = 77$ percent ➜ very high—almost 80 percent!

As shown previously in the section "*f_{HS} Influence on Potential Gains and on System Controllability*," f_{HS} plays an important role in VS controlled systems when selecting and dimensioning control parameters. Especially in cases as above, where f_{HS} has a significant value, largely above 30 percent, high limitations are expected, regarding to speed range, controllability and efficiency. $f_{HS} \approx 30$ percent is the highest estimated value for VS control applications totally free from limitations.

Plotting Other Isoefficiency Curves

As mentioned before, the affinity laws cannot be directly applied in high f_{HS} systems since pump efficiency varies substantially even within narrow speed ratios. Therefore all speed variations must be taken along isoefficiency curves in order to guarantee reliable results when calculating new conditions whenever pump speed is changed. Thus, to have some isoefficiency curves previously plotted in the graphic is usually very helpful.

Following the same methodology shown in the previous section, "*Hydraulic Fitness Analysis*," for the maximum isoefficiency curve, 65, 62, 60, 58, 50, 44, 40 and 34 percent isoefficiency curves are calculated and plotted in Figure 12. Their equations are:

- 65 percent ➜ $H = 0.0054 \times Q^2$ (USC units) or $H = 0.032 \times Q^2$ (SI units)
- 58 percent ➜ $H = 0.0094 \times Q^2$ (USC units) or $H = 0.057 \times Q^2$ (SI units)
- 50 percent ➜ $H = 0.0154 \times Q^2$ (USC units) or $H = 0.091 \times Q^2$ (SI units)
- 44 percent ➜ $H = 0.023 \times Q^2$ (USC units) or $H = 0.1354 \times Q^2$ (SI units)
- 40 percent ➜ $H = 0.0293 \times Q^2$ (USC units) or $H = 0.1728 \times Q^2$ (SI units)
- 34 percent ➜ $H = 0.0429 \times Q^2$ (USC units) or $H = 0.25123 \times Q^2$ (SI units)

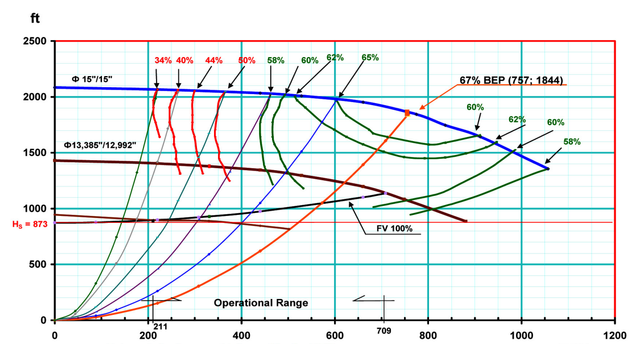


Figure 12. Isoefficiency Curves.

Determination of the Critical Point

As said, Pc may be defined as the point of the system curve where any further pump speed reduction may lead pumpage into instability.

If Pc is outside the operational range, the control valve (CV) may be suppressed. In this case the control will be fully done through speed variation.

If Pc is within the operational range, the CV can no longer be removed. It will be necessary then to associate CV, DCS and VFD.

In high f_{HS} systems, the critical point head uses to be around 2 to 5 percent above the system's static head (H_S). To check that, one has to calculate the head at the minimum transient flow, Q_{mt}. It should be higher than $1.02 \times H_S$.

Supposing $Q_{mt} = Qc = 211$ gpm (48 m^3/h) and entering this value in Equation (3), system curve: $Hc = 873 + 0.00053 \times (211)^2 = 896$ ft, or in SI units: $Hc = 266 + 0.0031 \times (48)^2 = 273$ m. Thus: Pc = (211 gpm; 896 ft) or Pc = (48 m^3/h; 273 m). $Hc/H_S = 896/873 = 273/266 = 1.026$ ➜ $Hc = 1.026 \times H_S$ or $Hc > 1.02\ H_S$ ➜ OK! The control valve may be suppressed since (2 percent < 2.6 percent < 5 percent).

Suppressing the control valve means to keep it fixed at 100 percent open or replace it by an orifice plate that generates the equivalent friction loss. This small loss is necessary to preserve the system curve slope and process controllability.

Determination of the Rotation Minimum Stop

All isoefficiency curves start at the origin and end at the pump D_M (maximum diameter impeller) performance curve, where efficiencies measured by the OEM are indicated. Notice in Figure 12 that, interpoling an isoefficiency curve that passes on the Pc (48 m^3/h; 273 m), one can estimate that it would touch the D_M performance curve in a value between 44 and 50 percent, something round 46 percent. This 46 percent isoefficiency curve crosses the rated impeller (D_R) curve at 260 gpm (59 m^3/h), and will be used to calculate the RmS by applying the affinity laws: 260 gpm/3550 rpm = 211 gpm/RmS; 59 m^3/h/3550 = 48 m^3/h/RmS ➜ RmS = (211/260) × 3550 or RmS = (48/59) × 3550 ➜ RmS = 2888 rpm.

Determination of the Rotation Maximum Stop

If the impellers are kept the same, there are no restrictions for the top speed. In principle it may be the nominal speed, 3550 rpm.

Plotting the RmS Curve in Six Points

- Point 1: Intersection of D_R 3550 rpm curve with vertical axis ($Q_1 = 0$) ➜ ➜ The point Q = 0 and H(0) = 1430 ft (436 m). Applying the affinity laws within the speed ratio: $H_1 = 1430 \times (2888/3550)^2 = 945$ ft or $H_1 = 436 \times (2888/3550)^2 = 288$ m
- Point 2: The critical point belongs to the RmS curve so: PC ➜ (211 gpm; 892 ft) or (48 m^3/h; 273 m)
- Point 3: Intersection of 58 percent isoefficiency curve [K = 0.0094 (USC units) or K = 0.0057 (SI units, refer to previous section, "*Plotting Other Isoefficiency Curves*") with D_R 3550 rpm curve] resulted in the following point: Q = 374 gpm; $H = 0.0094 \times 374^2$ => H = 1317 ft or Q = 85 m^3/h; $H = 0.057 \times 85^2 = 405$ m.

  Applying the affinity laws along the 58 percent isoefficiency curve: $H_3 = 1317\ \text{ft} \times (2888/3550)^2 = 879$ ft; $Q_3 = 374$ gpm × (2888/3550) = 304 gpm or $Q_3 = 85 \times (2888/3550) = 69$ m^3/h; $H_3 = 405 \times (2888/3550)^2 = 268$ m.
- Point 4: Intersection of the 62 percent isoefficiency curve with D_R 3550 rpm curve resulted in the following point: Q = 440 gpm; H = 1345 ft (from Figure 12) or Q = 100 m^3/h; H = 410 m (from Figure 12).

  Applying the affinity laws along the 62 percent isoefficiency curve: $Q_4 = 440$ gpm × (2888/3550) = 357 gpm; $H_4 = 1345\ \text{ft} \times (2888/3550)^2 = 890$ ft or $Q_4 = 100 \times (2888/3550) = 81 m^3/h$; $H_4 = 410 \times (2888/3550)^2 = 271$ m.
- Point 5: Intersection of the 65 percent isoefficiency curve (refer to previous section, "*Plotting Other Isoefficiency Curves*") with D_R 3550 rpm curve resulted in the following point: Q = 493 gpm; $H = 0.0054 \times 493^2 = 1315$ ft or Q = 112 m^3/h; $H = 0.032 \times 112^2 = 401$ m.

  Applying the affinity laws along the 65 percent isoefficiency curve: $Q_5 = 493 \times (2888/3550) = 400$ gpm; $H_5 = 1315 \times (2888/3550)^2 = 870$ ft or $Q_5 = 112 \times (2888/3550) = 91$ m^3/h; $H_5 = 401 \times (2888/3550)^2 = 265$ m.
- Point 6: Intersection of 67 percent isoefficiency curve (K = 0.0032; refer to previous section, "*Hydraulic Fitness Analysis*") with D_R, 3550 rpm curve resulted in the following point: Q = 621 gpm; $H = 0.0032 \times 621^2 = 1234$ ft or Q = 141 m^3/h; $H = 0.019 \times 141^2 = 376$ m.

Applying the affinity laws along the 67 percent isoefficiency curve: Q_6 = 621 gpm × (2888/3550) = 505 gpm; H_6 = 1234 ft × $(2888/3550)^2$ = 817 ft or Q_6 = 141 × (2888/3550) = 114 m^3/h; H_6 = 376 × $(2888/3550)^2$ = 249 m.

Theoretical Parameterization

The parameterization below was based on the theoretical graphic of Table 3 and Figure 13 and was installed in January, 2009, during plant operation.

Table 3. RmS Curve.

RmS Curve (SI units)		RmS Curve (USC units)	
Q	H	Q	H
m3/h	M	gpm	Ft
0	288	0	945
48	273	211	892
69	268	304	879
81	271	357	890
91	265	400	870
114	249	505	817

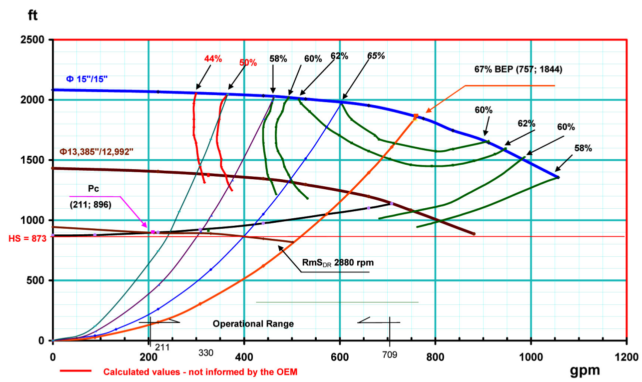


Figure 13. Critical Point and RmS Curve.

After negotiating with the operational staff and based on the theoretical values of Figure 13, the group decided for the following parameters:

- Speed range between 2920 and 3400 rpm. The RmS was preventively adjusted at a value a little higher than the theoretical one (2888 rpm), since it was not viable, at that occasion, to test the behavior of the system at the minimum flow.
- The RMS was adjusted at 3400 rpm. The group understood that the process would never require flowrates above the 621 gpm (141 m^3/h) produced by the pump at this rotation speed. The idea was to preserve the piece of equipment from higher loads.
- Control valve fixed at 100 percent open through an electronic stop
- Q_{mt} ≡ Pc ≡ (211 gpm; 896 ft) or (48 m^3/h; 273 m); Q_{MT} ≈ 616 gpm (140 m^3/h); H_{MT} = 1078 ft (328 m) (limited by RMS = 3400 rpm)
- Two operating modes: full automatic and semiautomatic. The first mode keeps the FV fixed in 100 percent and regulates flow through speed variation. Under this mode the FV electronic signal, 4 to 20 mA, is shifted to the frequency converter. In the semiautomatic mode rotation speed can be manually adjusted in any value between 2920 and 3400 rpm while flow is regulated by the FV, between 55 percent and 100 percent, according to the adjusted flowrate set point.
- Power output between 117 and 309 hp (88 and 230 kW)
- Power output at the most used flowrate (330 gpm or 75 m^3/h): 145 hp (108 kW)

Ultimate Parameterization Based on Values Recorded in the DCS

Just before the first turnaround of the process unit, in June 2009, the authors had their first chance to stress the theoretical parameterization installed in that system. The following values were then recorded:

- Speed range between 2920 and 3400 rpm
- Control valve fixed at 100 percent (full automatic mode)
- Q_{mt} ≡ Pc (211 gpm; 896 ft) or (48 m^3/h; 273 m); Q_{MT} = 630 gpm (143 m^3/h); H_{MT} = 945 ft (288 m), (limited by RMS at 3400 rpm)
- Power output between 118 and 282 hp (88 and 210 kW)
- Power output at the most used flowrate (330 gpm or 75 m^3/h): 147 hp (110 kW)

Just before the process shut down the authors turned the control to semiautomatic mode and, with the pump at the RmS, 2920 rpm, they reduced the flowrate set point to a very low threshold. FV-07 closed to 55 percent and the flowrate stabilized at 110 gpm (25 m^3/h). This condition was kept one hour. No disturbances in the flow were noticed. Some values collected during this phase are displayed in Table 4.

Table 4. Recorded Values During Shutdown Phase.

US CUSTOMARY UNITS

Q	H	Rotation	η	$V_{vibr.}$	$P_{WR.\ conv.}$	$P_{WR.destr.}$	Es
gpm	ft	rpm	%	mil/s	HP	HP	10^3 x HPh/gal
0	874						
211	896	2925	38.6	34.6	118	69	9.303
330	919	3025	49.4	33.5	147	71	7.442
440	968	3170	58.8	35.4	174	68	6.597
630	1066	3400	57.3	39.8	282	114	7.452
709	1116	3550	56.6		335	138	7.879

SI UNITS

Q	H	Rotation	η	$V_{vibr.}$	$P_{WR.conv.}$	$P_{WR.destr}$	Es
m^3/h	m	rpm	%	mm/s	kW	kW	kWh/m^3
0	266		0				
48	268	2925	38.4	0.88	88	52	1.840
75	275	3025	51.0	0.85	110	53	1.467
100	283	3170	58.8	0.9	130	51	1.300
143	288	3400	55.4	1.01	210	85	1.469
161	303	3550	56.6	1.40	250	103	1.553

The values registered in the yellow pattern correspond to field recordings or on DCS. The white pattern recordings correspond to calculations based on the yellow pattern values. The orange pattern values correspond to extrapolations estimated to compose curve extremes.

The New Control System Evaluation and Comparison with Throttle Control

Based on Table 4 values, the authors plotted three graphics comparing power input, destructive power and specific energy under throttling control at full speed with variable speed control with the control valve fixed at 100 percent opening.

Power Input

In Figure 14 one can notice that power input under VS control, within the operational range, will always be lower than *full-speed-throttling-control*, even under high flowrates, due to loss reductions in the control valve, now fully open. At 330 gpm (75 m^3/h) the power input at nominal speed is around 225 hp (168 kW) against 147 hp (110 kW) in variable speed. This shows a difference of 78 hp. If one considers that earlier the pump delivered

621 gpm (141 m^3/h), regardless of process demand, and now only 330 gpm (75 m^3/h) are being delivered with the pump rotating at reduced speed and the control valve completely open, the energy saving becomes even higher. That difference is 306 hp (full speed, 621 gpm or 141 m^3/h) to 147 hp (variable speed, 330 gpm or 75 m^3/h), which corresponds to 159 hp, a reduction of almost 50 percent.

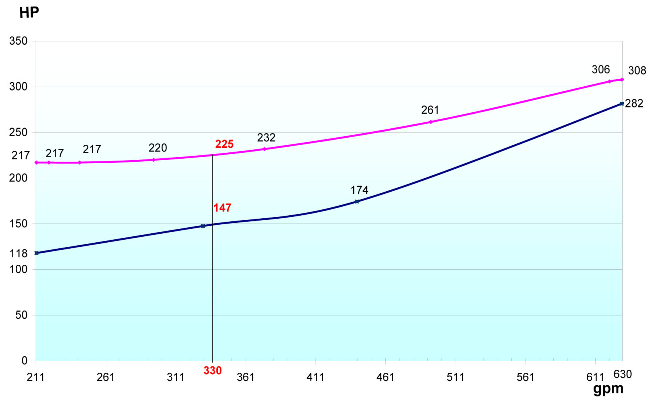


Figure 14. Shaft Power × Flowrate under Full Speed and Reduced Speed.

Destructive Power under
Full Speed × Variable Speed

Analyzing Figure 15, one can notice that at the most common operational condition, 330 gpm (75 m^3/h), destructive power is limited to 71 hp (53 kW), 59 percent of the BEP destructive power of 120 hp (90 kW), calculated for the maximum efficiency flow of 621 gpm (141 m^3/h). Comparing the two control systems at 330 gpm (75 m^3/h), with the recirculating valve blocked, destructive power is approximately 40 percent lower [1 – (71/120)] when operating under VS control.

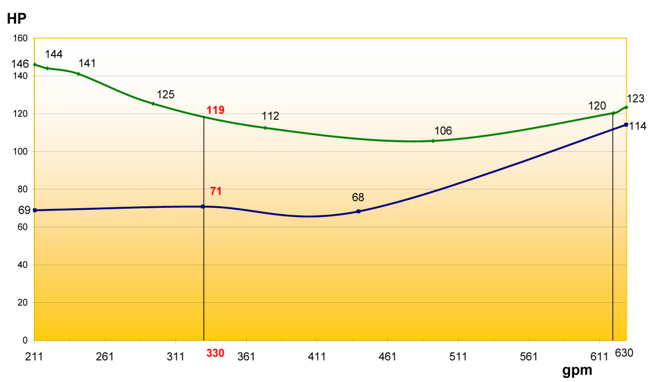


Figure 15. Destructive Power under Full Speed × Variable Speed.

These numbers point to a higher reliability and a higher equipment life span, besides a lower energy demand. *It becomes clear that destructive power may be used as a reliable criterion for selecting the pump and its control system*.

Specific Energy at Full Speed × Variable Speed

Notice, in Figure 16, that at the full speed condition (blue curve), the more throttled the CV, the higher the energy required to deliver each gallon of fluid through the system. In other words a higher specific energy becomes necessary. On the other hand, under VS, the control valve is kept 100 percent open and the energy requirement for delivering each gallon will always be lower than before (pink curve). In a first moment the efficiency losses due to speed decrement are overcome by the reductions on friction losses due to lower flow velocity. Thereby the Es demand decreases. At a certain moment, around 100 m^3/h, the efficiency losses overcome the friction losses, reversing the above described tendency.

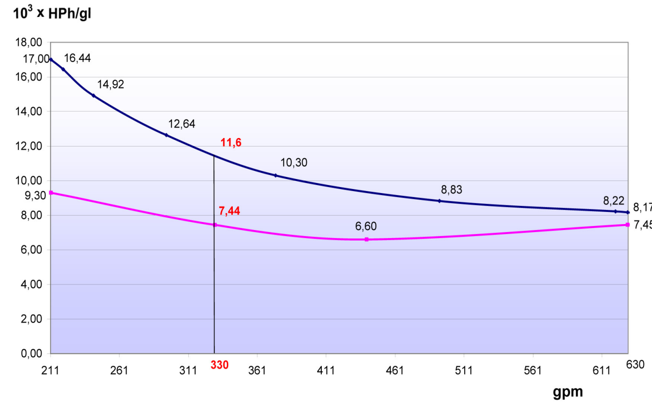


Figure 16. Specific Energy under Full Speed × Variable Speed.

Another interesting point one can notice in Figure 16 is that under VS control, at the most common condition, 330 gpm (75 m^3/h), in spite of a much lower pump efficiency, the energy required to deliver each gallon (m^3), 1.47 kWh/m^3, is lower than the one required on the pump's BEP at nominal speed, 1.67 kWh/m^3. This fact gives evidence that the BEP might not be the best operational condition for the pump nor the most economical one.

Also it becomes clear that, for a proper selection of the pump and its control methodology, the system characteristics must be taken into account. These characteristics are: system curve, operational range and demand curve (or load profile).

Still at the most common flowrate, comparing the two control systems operating with the recirculating valve blocked, the specific energy required under variable speed (1.47 kWh/m^3) is 35 percent lower than under nominal speed (2.29 kWh/m^3).

One should keep in mind that, for a given pumping system and a given flowrate, the lower the energy input, the lower the energy available to wear out the pump. So, specific energy is also a strong pump lifespan indicator, a system reliability indicator and a *pumping system efficiency* indicator as well. Hence, in VS controlled systems, the E_S display in the DCS screen is strongly desirable.

Temperature Tests

Case Temperature

The lower temperatures recorded after 15 minutes under low flowrates, 146°F (63.3°C) (Figure 17), are probably related with the more intensive heat exchange in the two exchangers situated in the suction line (observe in Figure 1). No significant temperature raise was detected inside the pump.

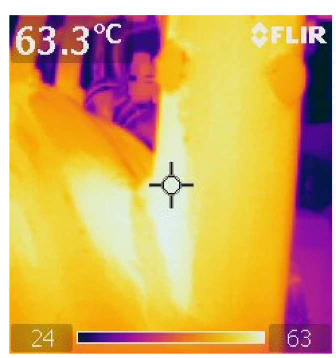


Figure 17. Thermography of the Pump Case at 2970 rpm, 220 gpm (50 m^3/h).

At high flows after 15 minutes, as can be observed from Figure 18, the temperature reached 152°F (67°C) at the discharge end.

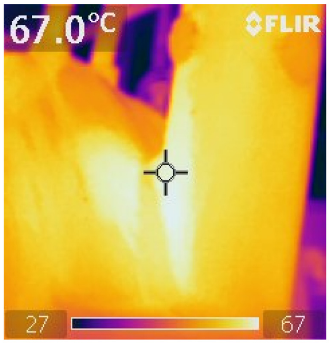


Figure 18. Thermography of the Pump Case at 3380 rpm, 621 gpm (141 m^3/h).

Bearing Temperature

Comparing records of Figure 19 and Figure 20, one may notice that, at low flowrates, the temperature decrease observed in the bearings after 15 minutes operation, may be related with the destructive power reduction. Indeed in Figure 15 one can observe that Destructive power dropped from 120 hp, at 621 gpm (141 m^3/h), to 69 hp, at 220 gpm (50 m^3/h), a 43 percent reduction.

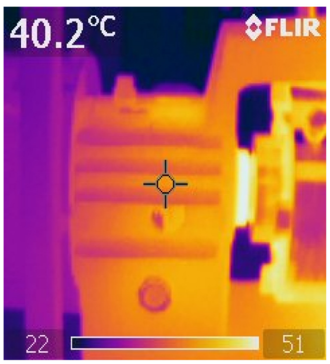


Figure 19. Combined Bearing at 2970 rpm, 220 gpm (50 m^3/h).

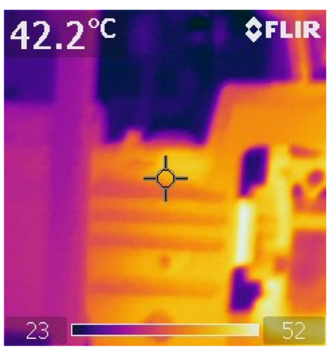


Figure 20. Combined Bearing at 3380 rpm, 621 gpm (141 m^3/h).

Dynamic Behavior

These pumps have a very close diametral clearance between the interstage diaphragm bushing and the shaft, 0.006 to 0.010 in (0.15 to 0.25 mm). This feature provides an exceptional rigidity to the rotor assembly. Actually it runs very smoothly all along the flow range. At 330 gpm (75 m^3/h), 3025 rpm, the highest recorded value was around 31.5 mil/s (0.8 mm/s). When operating at full speed, values like 55 mil/s (1.4 mm/s) were recorded, as shown in Figure 21.

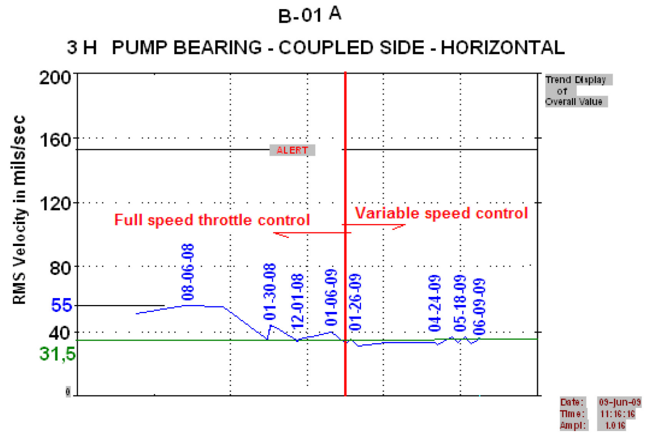


Figure 21. Vibration Trend Display Graphic.

Once again the effect of the lower destructive power under variable speed becomes clear. It is important to notice that, in this pump, the records taken at 3H position are historically the highest ones. All other values are lower.

Efficiency Evaluation Around the Most Common Operational Condition

Comparing the two control methods it becomes clear that, under variable speed, around the most common condition, not only the head developed by the pump is lower but its efficiency is also higher. One can notice in Figure 22 that, under full-speed-throttle-control, the efficiency is 47 percent, while under variable speed it is around 53 percent.

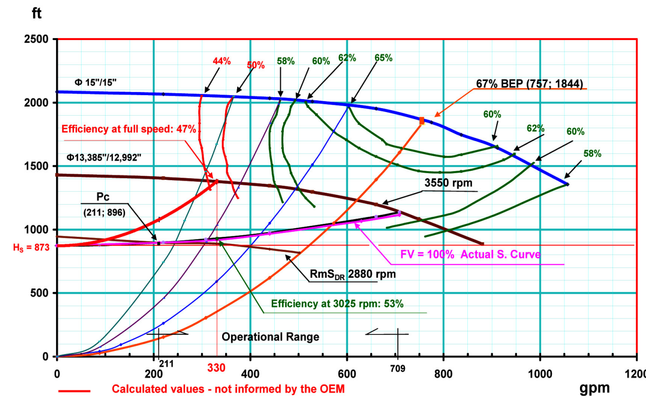


Figure 22. Efficiencies at 330 gpm (75 m^3/h) at 3550 rpm and at 2920 rpm.

Actually the efficiency at VS is higher than that at nominal speed along most of the operational range. Even when the efficiency at nominal speed is slightly higher, like for instance at the BEP, 330 gpm, the specific energy and destructive power under VS are both lower, as one can observe in Figures 15 and 16. In other words, the BEP may not be the smoothest condition for the pump.

SAVING AND IMPROVEMENT PERSPECTIVES FOR VS OPERATION

According to Europump and Hydraulic Institute (2004), "wear in bearings and on rubbing surfaces reduces as the seventh power of the speed, hence major benefits can be realized by running a pump more slowly."

Supposing the process will operate during the whole year at 330 gpm (75 m^3/h), or 3025 rpm, and supposing that under the original condition of 621 gpm (141 m^3/h), 3550 rpm the MTTF would be four years, one can estimate: new expected MTTF = 4 years × $(3550/3025)^7$ ➜ NEMTTF = 3.06 × 4 years; NEMTTF > 12 years. That means that besides the lower energy demand, less energy is driven to wear out all the system components, like PIs, FVs, FTs, heat exchangers and, in particular, the pumps.

Among all the achieved advantages, reliability is certainly the most desired in a short term point of view. It is important to remember that the emergency shutdown in mid 2007, when both pumps failed simultaneously, resulted in a production loss of almost $5 million, not considering the commercial penalties. In a long term point of view, environmental issues would certainly be the most important benefit of the new control method.

Nevertheless reliability related production losses in such a new plant like this one may be questionable and hence they were not considered. Other expected gains follow.

Maintenance Costs Reduction

Maintenance costs occurred between August 2005, when the plant started up, until August 2007, when both pumps failed.

- B-01 A—08/21/2007—$49,109.00
B-01 B—08/28/2007—$85,657.00
Total cost: $129,766.00 (in two years)
Annual cost = $129,766.00/2 years
Maintenance cost/year = $64,883.00

Supposing that under variable speed the pump wear will be reduced to one-third (item 6), one could expect a maintenance cost reduction of two-thirds of the maintenance annual cost, which corresponds to: MCR = 2 × $64,883.00/3; MCR = $43,255.00/year.

Energy Savings

Unfortunately a reliable system demand curve, able to forecast all the operational conditions for the next campaign is not available. Nevertheless it is well known that this unit operates most of the time requiring 330 gpm (75 m^3/h) of MEA solution. Supposing this flowrate will remain constant all along the year, one can estimate: Power at BEP @ nominal speed = 308 hp = 230 kW (original condition); Power at 330 gpm @ 3025 rpm = 147 hp = 110 kW (most common condition); P_{WR} = 161 hp = 120 kW.

Along one year ➜ ES = 120 kW × 8,760 hs/year ➜ ES = 1,051 MWh/year (1,409 hph/year). Supposing $76.00/MWh ➜ ES = 1,051 MWh × $76.00/Mwh; ES ≈ $79,900.00/year.

Expected Annual Savings with Maintenance and Energy

EASME = MCR + ES = $43,255.00 + $79,900.00; EASME = $123,155.00/year (only considering maintenance and energy).

Overall Installation Cost and Estimated Payback Time

Two frequency converters 480 V ➜ $73,000.00; Installation materials and manpower ➜ $190,500.00; Parameterization and final adjustments ➜ $9,500.00; Overall installation costs (OIC) ➜ $273,000.00.

Estimated Payback Time

EPT = OIC/EASME = 273,000.00/123,155.00 = 2.2 year ≈ 27 months; EPT ≈ 27 months after new control system start-up.

Note: If the 2007 production loss (neglecting the commercial penalties) were considered, the payback time of this investment would drop to less than 21 days!

Carbon Dioxide Emission Reductions

Probably in a few years this gain will be considered the most important one. Considering that this refinery has an energetic matrix where 60 percent of the total consumed energy is based on fossil fuels, one can work with the ratio of 590 kg CO_2/MWh generated. Thus it is possible to calculate: CDER = 1,051 MWh/year × 590 kg CO_2 /MWh; CDER = 620 ton CO_2/year.

FINAL COMMENTS

This pumping system has a peculiar characteristic: it normally works with very slow flow variations. On the other hand modern converters may vary their frequency with very high precision as 0.1 rpm. These two features certainly contributed to make feasible variable speed control, with the flow control valve fully open, even in such a combination where system and pump present very flat curves, almost parallel, along a significant part of the operating range.

If rapid flowrate responses were required, the control exclusively through speed variation would become very difficult or maybe impossible. In this case the solution would come through the association of the control valve with the DCS and the frequency converter, by applying the technology shown by the authors (Martins and Lima, 2008). In both situations the gains attained are unquestionable when compared with full-speed-throttle-control.

Another remarkable point is that the control valve will not be removed. During the campaign the system will operate fully automatically. Nevertheless, if during the start-up lower flowrates are required, down to 110 gpm (25 m^3/h), the control system may be switched to semiautomatic mode and the new flowrate set point adjusted in the DCS. The flow CV will then close down to 55 percent, producing the above condition at 2920 rpm. The DCS is programmed not to allow valve openings below 55 percent nor rotation speeds lower than 2920 rpm.

CONCLUSIONS

The use of the above technology:

- Proved that VS control may be feasible and advantageous in many high f_{H0} applications. Each case analysis will certainly take longer and demand a bigger effort but, in many cases, like the one above, it may be worth it
- Enabled the definitive closure of the manually actuated recirculation valve along with the hydraulic protection of the pump. That was the main drawback of the old control system, specially under low flowrates.
- Permitted the system to operate adequately, with the control valve fully open, within all the operating range. This fact allowed the suppression of a considerable amount of dynamic loss in the system and permitted reduction of pump speed, pump wear, to energy savings and reliability improvement. The energy required before to wear out the equipment now is being saved.
- Also enabled the system to require, for each operating condition, a lower amount of energy (hph or kWh), to deliver each gallon (m^3) of fluid. In other words, in each duty point less energy remains free to wear out the pump and system components, especially at very low flowrates as 48 m^3/h.
- Enabled an important improvement in system reliability, due to the lower destructive power levels, especially around the most common condition, 330 gpm (75 m^3/h). These lower levels can be attributed, as seen in Figure 22, to lower rotation speeds, lower differential pressures and higher efficiencies.
- Enabled an important improvement on MTTF expectation, from four years to 12 years.
- Enabled a reduction in energy consumption of 1,051 MWh/year (1,409 hph/year) or $79,900.00/year.
- Last but not least, enabled the reduction of CO_2 emissions about 620 tons CO_2/year. As the climate change issues escalate in importance, this feature may soon become the most important consideration.

NOMENCLATURE

BEP = Best efficiency point
CV = Control valve
Es = Specific energy
f_{HS} = Static head factor $f_{HS} = H_S/H_T = H_S/(H_S + H_f)$
FV = Flow control valve
H_f = Friction head, due to fluid velocity
H_S = Static head, due to differential pressure and elevation
H_T = Total head, $H_T = H_S + H_f$
MEA = Methyl ethanolamine
MTTF = Mean time to failure
OP = Operating point = duty point
Pc = Is the point that, when surpassed by the pump curve during decreasing speed, any further decrement may cause flow instability or even flow interruption. Refer to upper part of Figure 8. If the Pc is situated within the operational range the control exclusively through speed variation will be impossible.
P_{WRd} = Destructive power, $P_{WRd} = (1- \eta /100) \times P_{WRshaft}$
$P_{WRshaft}$ = Pump shaft power imput
Q_{BEP} = Flowrate at the best efficiency point
Q_{mt} = Minimum flowrate at an operational transient or process upset. An example of operational transient is during the start-up of a petroleum distillation unit when the feedstock is still under heating process.
Q_{MT} = Maximum continuous flowrate at an operational transient. For a correct design $Q_{MT} \leq Q_{Rated}$. An example of a maximum flow transient is when a process drum is overfilled and the CV opens 100 percent order to normalize its level.
RmS = Rotation minimum electronic stop
RMS = Rotation maximum electronic stop
VS = Variable speed

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