HOW TO IMPROVE RELIABILITY IN CENTRIFUGAL PUMP SYSTEMS THROUGH THE AUTOMATIC TUNEUP OF PUMPS WITHIN THEIR BEST OPERATIONAL CONDITION

by

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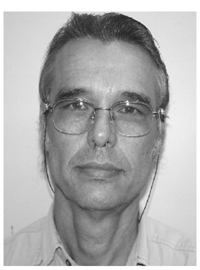

Enio Lima is with Petrobras - Petróleo Brasileiro S.A., in Duque de Caxias, RJ, Brazil. He worked with design and fabrication of petroleum equipment spare parts up until 1985, then moved to the rotating equipment maintenance area. There he dedicated most of his time to centrifugal pumps and turbomachinery failure prevention and repair. In late 2003, he started working in the reliability area, developing experiments concerning process control through centrifugal pumps speed variation, having already successfully installed this control process in 12 pumps at Duque de Caxias Refinery (REDUC), in Rio de Janeiro.

Mr. Lima received his B.S. degree (Mechanical Engineering, 1974) from the Federal University of Rio de Janeiro. He began with Petrobras in 1975, when he completed a Specialization Course in Petroleum Equipment.

ABSTRACT

Many centrifugal pump reliability problems are closely related to oversizing of pumps, which leads to highly restricted discharge valve operation, and flowrates are far below the best efficiency point (BEP), which are associated with low hydraulic efficiencies and high discharge pressures.

This paper presents the development of, and the main results achieved by, a new control technology that is able to keep the pump operating at its best condition independent of process requirements. In most cases it solves all of the above-mentioned problems, therefore improving reliability and mean time between failures. In addition, environmental conditions improve and energy is saved.

The concepts of specific energy (E_S) and life cycle cost (LCC) are discussed and, based on them, payback time of the pilot project is estimated.

Finally, some comments about this experience are made along with other important conclusions.

INTRODUCTION

Most of the process pumps, in refineries all around the world, operate substantially below their best efficiency point due to several reasons. The consequences of this low flow operation are shown in Figures 1 and 2.

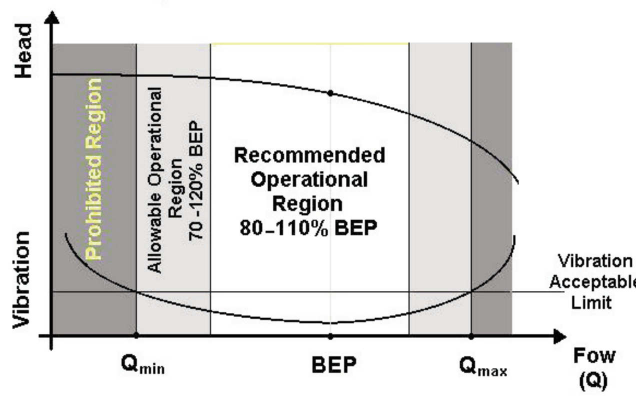

Figure 1. Pump Behavior × Operating Point.

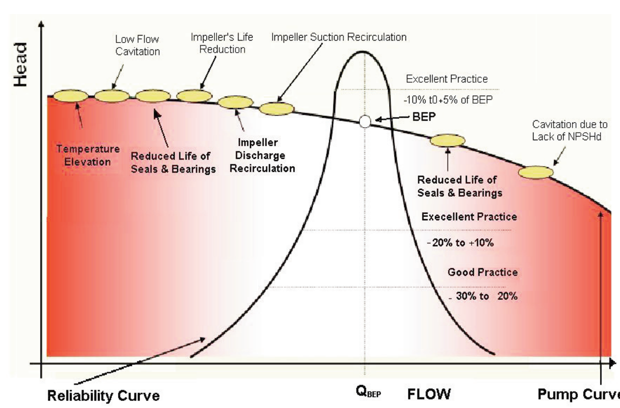

Figure 2. Consequences of BEP Deviation.

When the fluid receives all the energy generated by the pump rotor at nominal speed and does not achieve enough flow due to excessive throttling, the fluid takes random directions inside the pump volute, generating heat and producing unexpected forces, impacts, and pressure shocks, which lead to excessive wear and premature failure especially of mechanical seals and rolling elements.

Figures 1 and 2 show the main consequences of operating in regions far from the pump's best efficiency point (BEP). The large number of different overload conditions suggests that there must be another way, simpler and cheaper, to make them resistant to those aggressive conditions rather than reinforcing pump parts or sometimes trying to move process requirements to more acceptable conditions. This challenge became the authors' main goal—to design a new control system that is able to make pumps operate always near their best condition independent of process demands or human intervention.

CRITERIA FOR SELECTING PUMPS

The P-7109A pump and its spare, P-7109B, are 50 hp, 3550 rpm, back-pull-out pumps that operate in an atmospheric and vacuum distillation process unit. The pumps move kerosene from a stripper tower, T-7102B, to atmospheric tanks at pump level, all of them outside the process unit limits (refer to Figure 9).

According to the original design, using the maximum 12.5 inch impeller, the BEP is 422 gpm (95.8 m^3/h). The pump currently operates with a 10.5 inch impeller, in a range varying from 294 gpm (66.7 m^3/h, 70 percent of the BEP) to 55 gpm (12.5 m^3/h, 13 percent of the BEP). The system usually operates between 147 to 165 gpm (33 and 38 m^3/h, ≈ 37 percent of the BEP), when short aviation kerosene is produced.

This scenario itself explains the low reliability of these pumps as well as why their mean time before failure (MTBF) was less than 12 months. P-7109A/B were indeed considered two excellent pumps for testing this new control technology.

RELIABILITY PARAMETERS

Trend Display

Figures 3 and 4 show the historical vibration levels of these pumps as well as their behavior before and after installation of the new control system. It is important to note that nothing was changed in the pumps, except for their rotational speed, which became variable. Just before the test started, both pumps were scheduled to undergo maintenance due to high vibration levels and, in the case of Pump B, kerosene leakage through the mechanical seal. The first test was held on the day before the maintenance work on Pump B was to begin.

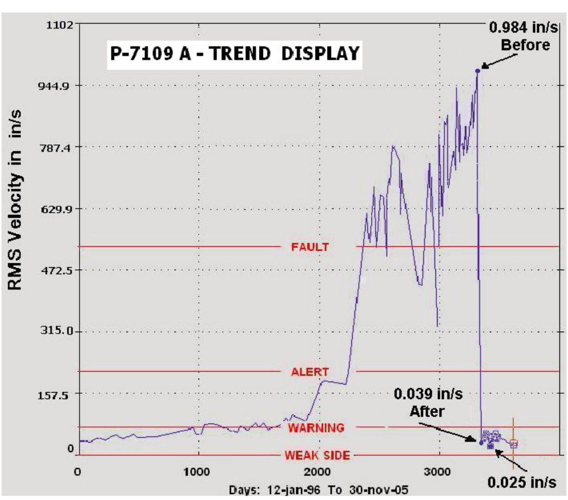

Figure 3. P-7109A Vibration Level before and after Installing the New Control System.

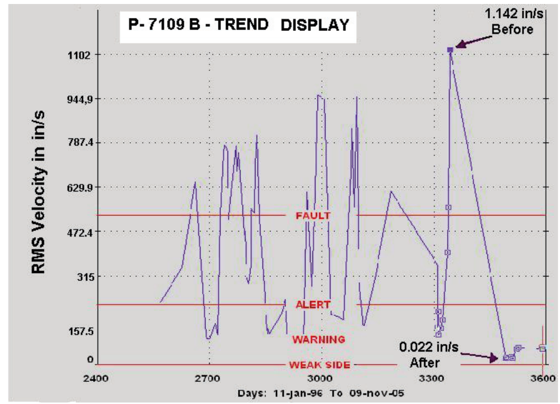

Figure 4. P-7109B Vibration Level before and after Installing the New Control System.

Comments

- The Pump A total vibration level dropped from 0.984 in/s (25 mm/s) to 0.039 in/s (0.997 mm/s) and later, to 0.025 in/s (0.63 mm/s). Pump B achieved even better results. The vibration level lowered from 1.142 in/s (29 mm/s) to 0.0022 in/s (0.57 mm/s) and the mechanical seal stopped leaking. It is important to remember that no repair had been performed on either of the pumps.
- The 0.022 in/s (0.57 mm/s) level is 4.4 times lower than the best value that had ever been measured for this pump, which was 0.098 in/s (2.5 mm/s).

Vibration Spectrum

Just before the test, Pump A spectrum showed a vibration peak of 0.875 in/s (22.23 mm/s) at 100 Hz (6000 cpm) (Figure 5). Afterwards, when operating at low speed, 37.48 Hz (2249 cpm), that peak became irrelevant. The new dominant peak became 0.016 in/s (0.4 mm/s) at a frequency of 37.48 Hz (2249 cpm) (Figure 6).

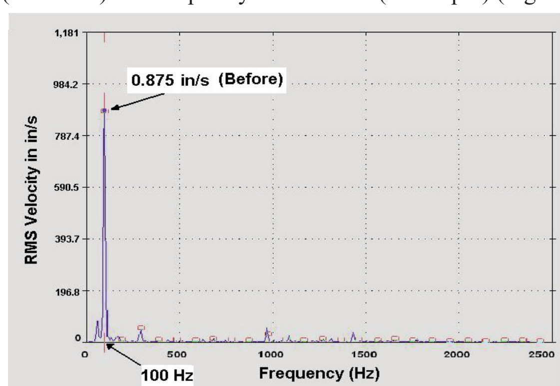

Figure 5. P-7109A—Frequency Spectrum at Nominal Speed: Dominant Vibration Peak of 0.875 in/s (22.23 mm/s), at ≈ 100 Hz (6000 cpm).

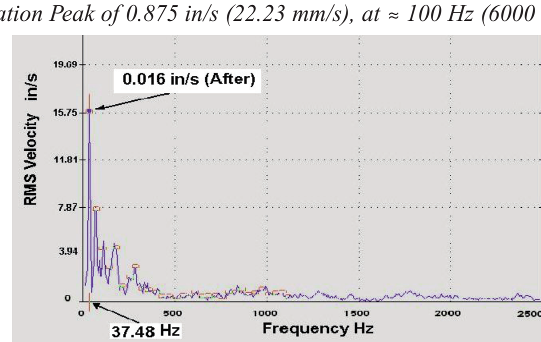

Figure 6. P-7109A—Frequency Spectrum after Dominant Peak of 0.016 in/s (0.4 mm/s, at Rotation Frequency 37.48 Hz (2249 cpm).

Figure 7 shows the vibration peak in the P-7109 B spectrum at nominal rotation. The vibration was 1.070 in/s (27.19 mm/s) at 5625 cpm. After implementing the new control system, the dominant peak changed to 0.0067 in/s (0.17 mm/s) at 2 × N or 2 × 1853 cpm, probably due to a residual misalignment (Figure 8).

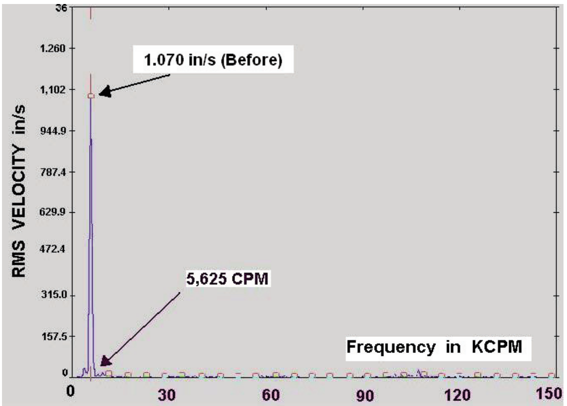

Figure 7. P-7109B—Frequency Spectrum at Nominal Speed: Dominant Vibration Peak of 1.07 in/s (27.19 mm/s) at 5625 cpm.

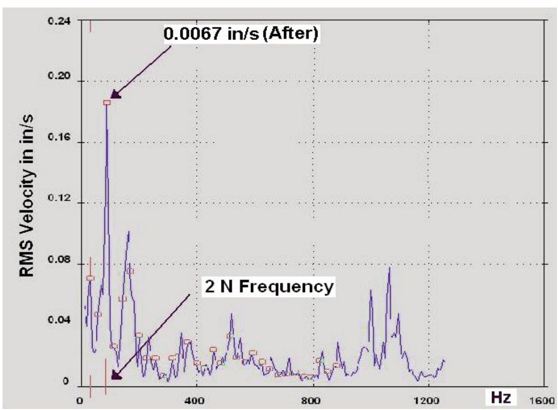

Figure 8. P-7109B—Frequency Spectrum after Dominant Peak of 0.0067 in/s (0.17 mm/s), at a frequency of 2 × N.

THE NEW CONTROL SYSTEM

The U-1710 atmospheric and vacuum distillation unit, is controlled by a distributed control system (DCS), a software application that controls the whole process. Control valve FV-023, which regulates the discharge flow of P-7109A/B, receives its set points from the control system (Figure 9).

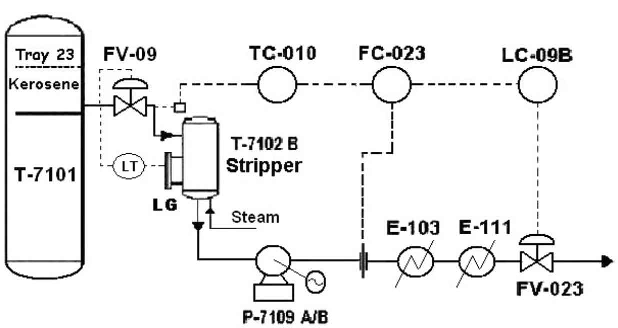

Figure 9. Simplified Flow Chart of P-7109 A/B System.

In Figure 10, the SC-004/005 and ZC-002 blocks represent the modifications introduced in the control system network. Two frequency converters, SC-004/005, were installed in the local substation to control the rotational speed of each pump motor. Both frequency converters receive their setpoint from the DCS block ZC-002.

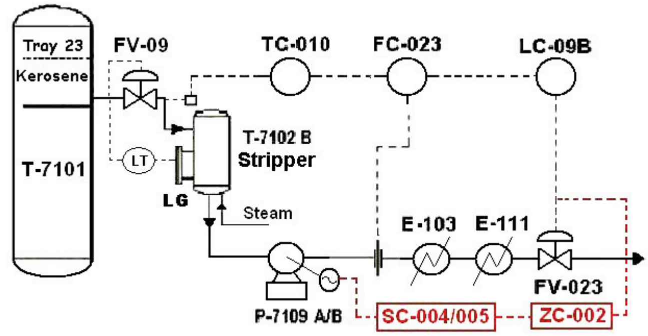

Figure 10. Simplified Flow Chart of P-7109 A/B System after Modification.

Level controller LC-09B sends the setpoint for FV-023 to the position controller ZC-002. For a certain opening range, it sends a zero signal to the speed controller in operation, so that the output is a constant frequency. This interval, that the authors' called *converter immobility range*, corresponds to the shaded area in Figure 11. When the valve operates within this range, for instance between 45 to 55 percent, motor speed will remain constant and flow will be controlled exclusively by control valve FV-023.

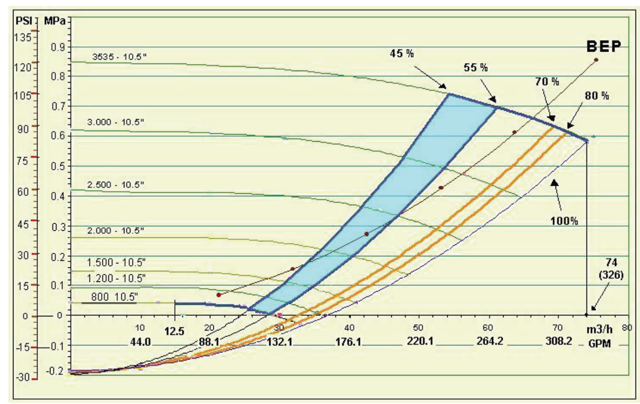

Figure 11. Converter Immobility Range Tuned-Up Around the BEP.

When the process conditions require FV-023 to open beyond the 55 percent limit, ZC-002 sends a positive signal to the converter, which in response increases the output frequency and, consequently, the pump rotational speed and flow. The flow increases until it exceeds the process requirements, then LC-09B sends a signal to close the control valve FV-023 until the valve returns to its immobility range, between 45 and 55 percent (Figure 11).

When the process requires a smaller flow and control valve FV-023 receives a signal to close below the lower limit of 45 percent, ZC-002 sends a negative signal to the converter, which in turn reduces the output frequency and, consequently, the pump rotational speed until the valve returns to its immobility range between 45 and 55 percent.

The distributed control system has to be programmed with a *minimum frequency limit*. Once this is attained, the flowrate decrease is controlled only by FV-023. In Figure 11 it corresponds to the bold dark line of the 800 rpm curve, leftward extension of the low side of the shaded area, which represents the immobility range.

The distributed control system has to be programmed with a *maximum frequency limit*. As above, once this threshold is attained, the flowrate increase starts to be controlled only by FV-023. This upper limit is not necessarily equal to the nominal rotation. In Figure 11 it corresponds to the bold line of the 3535 rpm curve, rightward extension of the upper side of the shaded area, which represents the immobility range.

CONVERTER IMMOBILITY RANGE SELECTION PROCEDURE

One will observe, in Figure 12, that the most frequent flowrange in the process is between 88 to 185 gpm (20 to 42 m^3/h). This feature led the authors to select the immobility range of the converter between 45 and 55 percent.

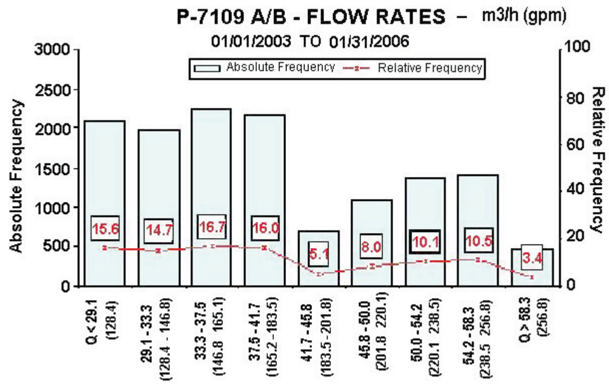

Figure 12. Flowrates Curve of the System along a Period.

Note in Figure 11 that the best isoefficiency curve, which begins in BEP, crosses diagonally the converter immobility range. In fact the approach outlined above will enable the pumps to operate around their best isoefficiency curve most of the time.

OVERHEATING PROBLEMS AT LOW SPEEDS

Tests performed between December 27 and 28 demonstrated that, during Rio de Janeiro's hottest summer days, at a reduced speed of 1300 rpm, the motor temperature *decreased* consistently with the rotational speed, as can be seen in Table 1. The authors' intention was to compare the test pump motor temperature, running at reduced speed, with other 50 hp pump motors running at nominal speed.

Table 1. Temperature Records at Low Speeds × Nominal Speed.

MP-7109B - MOTOR TEMPERATURE TEST AT LOW SPEED				
		12/27/2005	12/27/2005	12/28/2005
Ambient Temperature > 37°C (99°F)		Morning	Afternoon	Morning
Flow	m^3/h (gpm)	37.5 (165.1)	35.5 (156.4)	37.4 (164.6)
Speed	rpm	2.152	1.315	2.152
FV-023	%	46	68	47
Power Input	Kw (hp)	9.8 (13.1)	2.7 (3.6)	9.8 (13.1)
Motor Temperature	°C (°F)	46 (114.8)	43 (109.4)	40 (104)
Pump Vibration	mm/s (in/s)	0.64 (0.025)	0.41 (0.016)	0.65 (0.026)
OTHER 50 HP MOTORS AT NOMINAL SPEED				
Ambient Temperature > 37°C (99°F)		All Readings on 12/27/2005 morning		
Tag	MP-7118A	MP-7125A	MP-7111A	MP-7130A
°C (°F)	54 (129)	60 (140)	86 (186.8)	88 (190.4)

Comments

- Field tests confirmed that down to 1300 rpm, the motor temperature decreases with the rotation speed. When comparing temperature readings obtained in the morning of 12/27/05 at 2152 rpm, and in the afternoon, at 1315 rpm, one will observe that the second reading, 109.4°F (43°C), is lower than the first one, 114.8°F (46°C).

- The input power is a cubic function of rotation. Indeed, in another test, held at 800 rpm, the authors recorded temperatures as low as 102.2°F (39°C).

- It is important to observe that when lowering the speed while opening the control valve and keeping a reasonably constant flow, the power required by the pump decreases dramatically, along with the power available to wear out the equipment (destructive power). One can see that when comparing data obtained 12/27/2005 in the morning and in the afternoon, input power decreases from ≈13.14 hp (9.8 kW) to ≈3.62 hp (2.7 kW). That is an afternoon-to-morning ratio of less than one third. Simultaneously the overall vibration level decreases from 0.025 in/s (0.64 mm/s) to 0.016 in/s (0.41 mm/s) signaling a lower *destructive power*.

- *Destructive power may be defined as the part of the energy, absorbed by the pump from its driver, that is not converted into flow*.

- The above data encouraged the authors to further lower the pump rotational speed.

FINAL BATTERY OF FIELD TESTS

Based on the results in Table 1, the authors decided to hold another battery of field tests. This time they decided to move the immobility range up to 70 to 80 percent and the electronic stop for the minimum rotational speed down to 800 rpm. The data in Tables 2 and 3, as well as the comments below, deserve special attention. Tables 2 and 3 are presented in two versions: one in US current units and the other in metric units.

Table 2. Tests with Converter Immobility Range between 70 Percent and 80 Percent.

P-7109 - EFFICIENCY EVALUATION UNDER ECONOMICAL SPEED						
CONVERTER IMOBILITY RANGE: 70 to 80 % - FV-023						
PARAMETER	UNIT	REMARKS	RECORDS & CALCULATIONS			
Q_{LS}	gpm	on DCS	73.6	146.8	73.4	155.9
R_{LS}	rpm	on DCS	1200	800	800	1121
FV-023	%	on DCS	22	69	31	74
PI_{LS}	hp	on converter	2.41	1.07	0.94	2.41
PV_{LS}	in/s$\times 10^3$	on vibrometer	13.8	10.2	6.3	10.2
Pd_{LS}	psi	on pump	49.1	39.8	44.8	41.2
Ps_{LS}	psi	on pump	30.6	32.7	32.7	32.7
ΔP	psi	$Pd_{LS} - Ps_{LS}$	18.5	7.1	12.1	8.5
ΔP	ft	$Pd_{LS} - Ps_{LS}$	65.6	25.3	45.6	30.2
TPI_{LS}	hp	Q.H.δ / 3960	0.77	0.59	0.50	0.76
η_{LS}	%	TPI_{LS}/PI_{LS}	32.5	56.2	53.8	31.9
DPI_{LS}	hp	$(1 - \eta_{LS}/100) \times PI_{LS}$	1.62	0.47	0.43	1.65
Q_{LS} = Flow at Low Speed						
R_{LS} = Low Speed Rotation						
DCS = Distributed Control System						
PI_{LS} = Power Input at Low Speed						
PV_{LS} = Pump Vibration at Low Speed						
Pd_{LS} = Discharge Pressure at Low Speed						
Ps_{LS} = Suction Pressure at Low Speed						
ΔP = Differencial Pressure Between Discharge and Suction						
TPI_{LS} = Theoretical Power Input at Low Speed						
η_{LS} = Total Efficiency at Low Speed						
δ = Specific Gravity						
DPI_{LS} = Destructive Power at Low Speed						

P-7109 - EFFICIENCY EVALUATION UNDER ECONOMICAL SPEED						
CONVERTER IMOBILITY RANGE: 70 to 80% - FV- 023						
PARAMETER	UNIT	REMARKS	RECORDS & CALCULATIONS			
Q_{LS}	m^3/h	on DCS	16.7	33.3	16.7	35.4
R_{LS}	rpm	on DCS	1.200	800	800	1.121
FV-023	%	on DCS	22	69	31	74
PI_{LS}	Kw	on converter	1.8	0.8	0.7	1.8
PV_{LS}	mm/s	on vibrometer	0.35	0.26	0.16	0.26
Pd_{LS}	Mpa	on pump	0.34	0.27	0.31	0.28
Ps_{LS}	Mpa	on pump	0.21	0.22	0.22	0.22
ΔP	Mpa	$Pd_{LS} - Ps_{LS}$	0.13	0.05	0.09	0.06
ΔP	m	$Pd_{LS} - Ps_{LS}$	20	7.7	13.9	9.2
TPI_{LS}	Kw	Q.H.δ / 3960	0.57	0.44	0.37	0.57
η_{LS}	%	TPI_{LS}/PI_{LS}	32.5	56.2	52.9	31.7
DPI_{LS}	Kw	$(1 - \eta_{LS}/100) \times PI_{LS}$	1.21	0.35	0.32	1.23

Table 3. Comparative Values under Nominal Speed.

P-7109 - EFFICIENCY EVALUATION AT NOMINAL SPEED						
PARAMETER	UNIT	REMARKS	RECORDS & CALCULATIONS			
Q_{NS}	gpm	on DCS	69.1	146.6	69.1	155.9
ΔP_{NS}	psi	OEM Data	14.3	14.3	14.3	14.3
ΔP_{NS}	ft	Pd_{NS} - Ps_{NS}	6.26	2.43	4.33	2.89
η_{NS}	%	OEM Data	25	44	25	45
PI_{NS}	hp	OEM Data	29.4	33.3	29.4	34.5
DPI_{NS}	hp	$(1 - \eta_{NS}/100) \times PI_{NS}$	22.03	18.6	22.03	16.9
Q_{NS} = Flow at Nominal Speed						
DCS = Distributed Control System						
PI_{NS} = Power Input at Nominal Speed						
ΔP_{NS} = Pump Differencial Pressure						
TPI_{NS} = Theoretical Power Input at Nominal Speed						
η_{NS} = Total Efficiency at Nominal Speed						
DPI_{Ns} = Destructive Power at Nominal Speed						
P-7109 - EFFICIENCY EVALUATION AT NOMINAL SPEED						
PARAMETER	UNIT	REMARKS	RECORDS & CALCULATIONS			
Q_{NS}	m^3/h	on DCS	15.7	33.3	15.7	35.4
ΔP_{NS}	Mpa	OEM Data	0.10	0.10	0.10	0.10
ΔP_{NS}	m	Pd - Ps	1.91	0.74	1.32	0.88
η_{NS}	%	OEM Data	25	44	25	45
PI_{NS}	Kw	OEM Data	21.91	24.82	21.91	25.74
DPI_{NS}	Kw	$(1 - \eta_{NS}/100) \times PI_{NS}$	16.43	13.9	16.43	12.6

- At 800 rpm and 69 percent valve opening, pump efficiency reached its largest recorded value during the tests, 56.18 percent. According to the original equipment manufacturer (OEM), the maximum efficiency that may be reached with an impeller diameter of 10.5 inches, at nominal speed, is 58.5 percent.

- At 800 rpm the control valve position has a large influence on the flowrate and a minimal impact on the corresponding power input. When FV-023 opened from 31 to 69 percent, the flowrate increased from 73.4 to 146.8 gpm (16.7 to 33.3 m^3/h, +100 percent) and the required pump power input increased from 0.94 to 1.07 hp (0.7 to 0.8 kW, +14.3 percent). It became clear that the pressure and height potential energies, which earlier caused the equipment to wear out, now are being used to push the fluid through the system. It should be remembered that the suction pressure is 32.7 psi (0.22 MPa) caused by the pressure in the tower (suction side) and the differential level between the tower and the tank (discharge side).

- When speed starts to rise, the corresponding power input rises much more steeply since it is a cubic function of rotation.

- At constant speed, one of the worst operating conditions for these pumps was 73.4 gpm (16.7 m^3/h). The vibration level reached 1.14 in/s (29 mm/s) with a discharge pressure of 188.5 psi (1.3 MPa) and power input of 29.38 hp (21.91 kW). With variable speed, at 800 rpm the pump vibration level dropped to 0.006 in/s (0.16 mm/s), discharge pressure dropped to 44.8 psi (0.31 MPa), and power input dropped to 0.94 hp (0.7 kW). The *destructive power* dropped from 22.03 to 0.43 hp (16.43 to 0.32 kW). A reduction in all of the above-mentioned parameters is a strong indicator of increased reliability and MTBF for the whole pumping system and especially for the pump itself. Also note that under these conditions the suction pressure of 33.4 psi (0.23 MPa) and the available differential height are being used entirely for fluid displacement.

  The authors had never before seen a pump vibrating at such a low level as 0.006 in/s (0.16 mm/s). Most remarkable is that this result was achieved under conditions that earlier had resulted in low reliability and MTBF for this pump.

- Another remarkable point is that both pumps are still running reliably 24 months after both had been scheduled for maintenance, which was not carried out at all. In addition, they still are the smoothest running pumps in the refinery.

NOISE LEVEL READINGS

Unfortunately it is quite difficult to accurately measure the noise level reductions on pumps and motors running at low speed due to the high noise levels in the surrounding area. As a result, the authors decided to take comparative readings between P-7109 and other 50 hp pumps in the same plant. The corresponding data are plotted in Table 4. Two readings were made for each electric motor. The first reading was about 1 inch (≈ 25 mm) from the fan inlet and the second one at about 1 yard (≈ 914 mm) from the fan inlet.

Table 4. Decibel Meter Readings.

Equipment	rpm	Noise Level (dB)	
		At 1 in from the Fan Inlet	At 1 Yard form the Fan Inlet
P – 7111 A	3550	109.1	96.6
P – 7111 B	3550	109.3	97.5
P – 7118 A	3550	100.6	94.5
P – 7125 B	3550	***112.6***	103
P – 7130 A	3550	111.3	98
P – 7109 A	1121	***92.6***	92.9
	800	*91*	91.2

Comments

- The noise level of MP-7109 A, at 91 dB, was lower than all other 50 hp motors.

- For all data recorded, the noise level got lower when the decibelmeter was distanced from the motor fans, except for MP-7109, when an opposite behavior was noted. Conclusion: under those recording conditions, the noise level inside MP-7109 was lower than outside.

SPECIFIC ENERGY

Specific energy (E_s) is the energy per unit volume consumed by a pump, when operating in a certain system, with a certain fluid at a certain condition, needed to displace 264.1 gal (1 m^3) of that fluid. In other words, E_s may be expressed by the following units:

$$E_s = kWh / m^3 \left(E_s = hph / gal\right) \tag{1}$$

where:
E_s = Specific energy
kW (hp) = Power input at the driving train
h = Time in hours
m^3 (gal) = Pumped volume

Specific Energy Applications

The main E_s applications are:

- Pumping system efficiency indicator at a certain operating point.

- Pumping system efficiency indicator at an average condition, along a period of time. The lower the E_s, the higher the system's efficiency.

- Element for life cycle cost (LCC) calculation as described in the next section.

- Reliability and MTBF indicator for a pumping system. The lower the E_s, the higher the reliability and the MTBF *of the system*.

For the time being, attention is going to be concentrated on the third application that is life cycle cost calculation element.

$$E_s = \left(Power \times Time\right) / Volume = Power / Flowrate \tag{2}$$

That means that E_s is a function of the system flowrate and, consequently, it will be necessary to calculate it for every pumping condition, especially when speed variation is considered.

E_s and LCC Calculations

Life cycle cost is a parameter used to calculate the total cost of a pumping system through its expected life when operating at design conditions. Generally it includes equipment acquisition costs, installation costs, maintenance and spare parts costs, and finally operation costs, where energy is almost always the main component.

To calculate the pumping energy cost, the E_s must first be calculated for a representative number of operating conditions depicted on the system demand curve (Figure 12). The E_s can be easily calculated for each condition by dividing the power input at the frequency converter by the corresponding flowrate.

The amount of energy required to pump the total volume of fluid at each operating condition can be calculated by multiplying the calculated E_s values by the total volume of fluid pumped throughout a year for each respective condition. Adding these portions one can finally calculate the total energy required to pump all of the process production throughout a year. By multiplying the total energy required by the energy rate, the total energy cost for the corresponding control solution can easily be calculated. An example of the above-mentioned calculations can be found in Table 5. Thus, the difference between them, in this case 174.8 − 49.62 = 125.6 MWh (168,430 hph) represents the energy saved by replacing the constant speed control by this variable speed control technology. That means a reduction of over 71 percent.

Table 5. E_s Calculations for Different Operating Points and the Total Pumping Energy Required under Constant Speed Control. (In US Current and in Metric Units.)

P-7109 Specif Energy and Energy Consumption Calculation at Nominal Speed									
Flow Range	% year	hs/ year	Qconsid.	Dif. Press.	Head	η	m³/ year	Es	Mwh/ year
m³/h	%	hs	m³/h	Mpa	m	%	m³	Kwh/ m³	Mwh
Q<29.1	15.6	1364	22.5	0.93	143	35	30.688	0.74	22.6
29.1 - 33.3	14.7	1283	31	0.92	142	42	39.784	0.61	24.1
33.3 - 37.5	16.7	1463	35.4	0.90	138	45	51.787	0.55	28.7
37.5 - 41.7	16.0	1402	39	0.89	137	48	54.662	0.51	28.1
41.7 - 45.8	5.1	448	44	0.86	132	51	19.696	0.47	9.2
45.8 - 50.0	8.0	699	48	0.84	129	53	33.554	0.44	14.7
50.0 - 54.2	10.1	888	52	0.83	128	54	46.190	0.43	19.7
54.2 - 58.3	10.5	916	56	0.82	126	56	51.313	0.41	20.8
Q>58.3	3.4	297	60	0.80	123	57	17.818	0.39	6.9
		Total hs	Pump H x Q Curve @ 3550 rpm without strainer losses SpGr = 0.65				Total m³	Aver. Kwh/m³	Total Mwh
		8.760					345.492	0.506	174.8

P-7109 Specif Energy and Energy Consumption Calculation at Nominal Speed									
Flow Range	% year	hs/ year	Qconsid.	Dif. Press.	Head	η	gal / year	Es	hph/ year
gpm	%	hs	gpm	psi	ft	%	gal x 10⁶	(hph/ gal) x 10⁻³	hph x 10³
Q<128	15.6	1364	99	135	469	35	8.10	3.74	30.3
128 - 147	14.7	1283	136.4	133	464	42	10.50	3.08	32.3
147 - 165	16.7	1464	155.9	131	454	45	13.68	2.81	38.5
165 - 183	16.0	1402	171.7	129	449	48	14.44	2.61	37.6
183 - 202	5.1	448	193.7	125	434	51	5.20	3.37	12.3
202 - 220	8.0	699	211.3	122	424	53	8.86	2.23	19.7
220 - 238	10.1	888	228.9	120	419	54	12.20	2.16	26.4
238 - 257	10.5	916	246.6	119	414	56	13.56	2.06	27.9
Q>257	3.4	297	264.1	116	404	57	4.71	1.97	9.3
		Total hs	Pump H x Q Curve @ 3550 rpm without strainer losses SpGr = 0.65				Total gal x10⁶	Aver. HPh/GPM	Total hph
		8.760					91.26	2.568	234.3

Comments

- The system demand curve (Figure 12) informs the amount of hours in a year that the system will operate under a number of pumping conditions. Thus, one can calculate the total volume pumped under each condition throughout a year.

- In Table 5 one can see that under constant speed, when the flowrate decreases, E_s consistently increases, especially under very low flowrates. (Table 5 calculations are based on OEM data applied to the system demand curve.)

- In Table 6 one can see that, under variable speed, when the flowrate decreases, E_s also decreases. At 73.4 gpm (16.7 m^3/h), the slightly increased value of 0.043 kWh/m^3 (0.219×13^{-3} hph/gal) makes sense on the control valve position, which will have moved from 69 percent to 31 percent. (Table 6 calculations are based on field recorded data.

Table 6. E_s Calculations for Different Operating Points and the Total Pumping Energy Required under Variable Speed Control. (In US Current and in Metric Units.)

P-7109 A/B Specific Energy and Energy Consumption Calculation at Low Speed										
% year	h / year	Qconsid.	Speed	Ctrl. Valve	Dif. Press.	Head	η	m³ / year	Es	Mwh / year
	h	m³ / h	rpm	% open.	Mpa	m	%	m³	Kwh / m³	Mwh
15.6	1.364	16.7	800	31	0.085	1.3	55	22.778	0.043	0.98
14.7	1.283	33.3	800	69	0.05	0.8	56	42.735	0.025	1.05
16.7	1.463	35.4	1.121	74	0.06	0.9	32	51.787	0.052	2.70
16.0	1.402	39.1	1.499	80	0.038	0.6	33	54.803	0.032	1.75
5.1	448	45.2	1.999	70	0.152	2.3	35	20.233	0.120	2.43
8.0	699	46.4	1.998	80	0.133	2.0	37	32.436	0.100	3.23
10.1	888	50.2	2.214	72	0.31	4.8	38	44.591	0.225	10.05
10.5	916	56.5	3.000	60	0.455	7.0	36	51.771	0.350	18.12
3.4	297	66.8	3.208	76	0.42	6.5	25	19.837	0.469	9.30
								Total Volume	Aver. Kwh / m³	Total Mwh
								340.971	0.146	49.62
								Total Save	71.23%	

P-7109 A/B Specific Energy and Energy Consumption Calculation at Low Speed										
% year	h / year	Qconsid.	Speed	Ctrl. Valve	Dif. Press.	Head	η	gal / year	Es	hph / year
	h	gpm	rpm	% open.	psi	ft	%	gal x 10⁶	(hph/gal) x 10⁻³	hph x 10³
15.6	1.364	73.6	800	31	12.33	4	55	6.02	0.219	1.32
14.7	1.283	147	800	69	7,252	3	56	11.32	0.125	1.42
16.7	1.463	156	1.121	74	8,702	3	32	13.69	0.265	3.63
16.0	1.402	172	1.499	80	5,511	2	33	14.46	0.162	2.34
5.1	448	199	1.999	70	22.05	8	35	5.35	0.610	3.26
8.0	699	204	1.998	80	19.29	7	37	8.56	0.505	4.32
10.1	888	221	2.214	72	44.96	16	38	11.78	1.144	13.47
10.5	916	249	3,000	60	65.99	23	36	13.69	1.777	24.32
3.4	297	294	3.208	76	60.92	21	25	5.24	2.381	12.47
								Total Volume gal x 10⁶	Aver. (hph/gal) x 10⁻³	Total hph x 10³
								90.11	0.739	66.55
								Total Save	71.23%	

- Table 5 shows that the lowest E_S consumed under nominal speed is 0.39 kWh/m^3 (1.97×10^{-3} hph/gal), which corresponds to the flowrate of 264 gpm (60 m^3/h), rarely reached by the process (Figure 12). On the other hand, in Table 6, under variable speed, the smallest E_s, reached at 800 rpm, is 0.025 kWh/m^3 (0.125×10^{-3} hph/gal), 15 times lower and in a flow region widely practiced by the process. It is important to note that, at the same considered flowrate (of 147 gpm or 33.3 m^3/h), the E_s demanded under nominal speed is approximately 0.58 MWh/m^3, that means, 23 times higher. The low E_s of 0.025 kWh/m^3 (0.125×10^{-3} hph/gal) became possible thanks to the control valve opening at low speed, allowing the system to make use of its potential energy (height and pressure) to save most of the energy required for moving the fluid. Note that the system itself, due to that potential energy, is able to support a gravitational flow. The pump does only complement that flowrate up to the volume required by the process. It should be remembered that the suction pressure is 32.7 psi (0.22 MPa).

- In Table 5 one can notice that the efficiencies at nominal speed increase consistently with the flowrates since the operating point gets closer to the BEP. On the other hand, in Table 6, under variable speed, the efficiencies start high at low flowrates and should keep the same high level along all the flow range. Nevertheless these efficiencies decrease when flowrate and rotation speed get higher. An improper strainer installed in the suction line of both pumps, P-7109 A and B, may explain this fact. In other words, without the strainer, efficiency would keep high all along the flow range.

- Due to the above, one can conclude that the energy consumption calculated in Table 5 is underestimated since the improper strainer at the pump suction was not taken into account.

• In addition, in Table 6, the specific energy demands are overestimated since the improper strainers will soon be removed.

• Based on the above comments, one can conclude that the energy reduction achieved by variable speed control, when compared with control under nominal speed, is higher than 71.23 percent.

• Beside the drastic reduction in energy consumption, one can observe a correspondent reduction in the destructive power, which certainly will impact on a higher reliability and higher MTBFs.

The main conclusion is that: *"The best operational condition for a pump is not necessarily its best efficiency region. It will be on the region where the pumping system requires the lowest E_S and consequently the lowest destructive power input."*

INVESTMENT PAYBACK PERIOD CALCULATION

Electric Power

Considering the conservative annual energy gain of 71.23 percent or 125.6 MWh (168,4230 hph) and the Brazilian rate of US$ 40.00/MWh (US$ 0.03/hph), one may calculate the annual electric energy savings of: AEES = 125.8 MWh × US$ 40.00/MWh = US$ 5007.00 (annual electric energy savings). It is important to note that if both strainers are removed from the suction lines of the pumps the energy savings will certainly increase.

Savings with Maintenance and Spare Parts

Based on historical data up to 2003, the average cost of an overall maintenance in a P-7109 is around $6000.00. Considering an average of two maintenance events a year ($6000.00 × 2 = $12,000.00) for both pumps, and the new MTBF of approximately 10 years, the annual cost reduction would be around 90 percent, which means: MAR = 0.9 × $12,000.00 = $10,800.00 (maintenance average reduction).

Total Annual Savings

The total annual savings would be: TAS = $5007.00 + $10,800.00 = $15,807.00 (total annual savings).

Installation Cost and Payback Time

The total installation cost of the variable speed control system was $19,230.00 for both pumps. Thus, the payback time may be estimated as: PbT = IC/TAS = $19,230.00/$15,807.00 = 1.21 years (payback time).

In spite of the results above, the main reasons for the variable speed control installation on these pumps were not related to energy savings but primarily to reliability factors, secondly to environmental factors (due to leakage and fugitive emission reductions), and finally due to MTBF improvements. However, except for MTBF, evaluating these factors is much more complex.

It is important to remember that in some applications it may be necessary to upgrade the electric motor insulation class or even to certify the electric motor to operate with the frequency converter. Anyway, these additional costs generally do not affect the decision of adopting variable speed control.

A PARADIGM SHIFT ON VARIABLE SPEED PUMPING TECHNOLOGY

The state-of-the-art in variable frequency device (VFD) application is on low static head systems, where variable speed drives have been used successfully for the last 20 years. Nevertheless, it has been impossible to operate them in negative static head systems, like P-7109 A/B. In these systems there can be a flow (called *gravitational flow*) independent of pump rotation (Figure 11).

By combining control valve, VFD, and a process computer, the new technology allows one to make use of the system potential pressure and height energy that produce a gravitational flow, which is complemented with the pump's flow, in order to meet process requirements. Thus, *negative static head systems became the most attractive application for VFDs*.

Another important application for the new control technology is in systems with midrange static heads, up to 75 percent of the total head. Earlier, in those systems, when the total head delivered by the pump was decreased, via rpm reduction, to values close to the system's static head, the pumping conditions got unstable, leading to resonant flows, erratic flows, or sometimes to no flow at all. The combination of the VFD with the control valve and the process computer allows one to limit speed reduction, increase the system curve inclination, and avoid such unstable conditions.

Finally, when the system static head gets high, generally over 75 percent of total head, the benefits generated by variable speed rotation become much more limited, generally leading to unattractive investments.

CONCLUSIONS

• The above variable speed control technology, combining control valve, VFD, and process computer is applicable to most industrial processes involving centrifugal pumps, since it enables the application of VFDs to negative static head systems and medium range static head systems.

• On the other hand, operational transient problems, faced by centrifugal pumps when process units are starting or under emergency conditions, will be automatically avoided by the new control system. Certainly the high failure rate, typical of these situations, will not be a problem anymore.

• Besides the reliability and MTBF improvements, important environmental gains are also produced by this new technology. Operating under reduced speed, at a fraction of normal discharge pressure and with extremely reduced vibration levels, seal leakages and even fugitive emissions are largely reduced. Considerable noise reductions are also expected. These characteristics, in spite of being difficult to evaluate, must be taken into account when deciding about adopting such a variable speed control technology.

• Another important factor is simplicity. To install such a system, all one needs is some available space in the local substation and a distributed process control system (process computer). The installation can also be done along with process operation and the connection with the process computer, in most cases, can be made without process interruption.

• Finally, another factor to consider is the low initial cost of frequency converters, especially for low voltages like 480 volts. Also the installation costs are likewise low in most cases.

NOMENCLATURE

Converter immobility range—Corresponds to the control valve operating range in which the VFD delivers a constant frequency output. Within this range the flowrate control is maintained exclusively by the control valve.

VFD—Variable frequency device, generally a frequency converter

Destructive power—May be defined as the part of the energy absorbed by the pump and not converted into flow. Destructive power = $(1 - \eta_{pump})$ × power input at pump shaft.

Negative static head system—A system where the pump suction pressure is higher than its discharge pressure. In these cases the pump is used to complement the flowrate required by the process.

Specific energy (E_s)—The amount of energy necessary to move one cubic meter or one gallon of a certain product through a pumping system. In a given system each operating condition will correspond to a typical E_s. Different control technologies will also generate different E_s for the same flowrate. The lower the E_s, the higher the system efficiency.

Life cycle cost—The total cost of a pumping system considering all costs involved like purchasing costs, installation costs, operating costs, energy costs, etc., during a given length of time.

System demand curve—A graphical record of all operational conditions encountered by a pumping system during one or more years. It is an important tool for life cycle cost calculation.

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