A CASE HISTORY—IMPROVED HYDRAULIC DESIGN LOWERS CAVITATION EROSION AND VIBRATIONS OF A WATER TRANSPORT PUMP

by

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ABSTRACT

In this case history, the successful application of well known tools for hydraulic design changes and cavitation erosion prediction is presented. A description of the tools can be found in many publications and will not be repeated here. However, the application of the tools in a difficult field situation with unfavorable hydraulic boundary conditions is outlined and the resulting successful solution is presented.

There are six pumping stations in the Rio Colorado Tijuana Aqueduct (water pipeline), which crosses northern Baja California in Mexico. The planning of the construction work did not adequately consider all the necessary hydraulic boundary conditions for the pipeline pumps; i.e., some of the suction reservoirs were built with a relatively small elevation relative to their pumping stations. Consequently, the existing available net positive suction head (NPSH) for the pumps of some stations is very low and marginal for satisfactory operation. Therefore, impellers are experiencing premature wear (repair/replacement after less than a year of operation), caused by cavitation erosion.

Construction changes to the pipeline to increase the NPSH available were cost prohibitive. Hence, to extend the life expectancy of impellers, the customer placed an order with the OEM to develop an improved hydraulic design with respect to cavitation

erosion. As nothing but the pump hydraulic could be changed, the new design would have to work under the existing, far less than optimum pipeline conditions, and compromises had to be found which would normally not be desirable for a new pipeline layout. Preliminary testing was to be done on the test bed and final testing was subject to a joint effort of the manufacturer and the user on the prototype pump installed and operating in the pipeline.

A new impeller, with slightly modified rated flow conditions to adapt to effective plant conditions, and a steeper head—a capacity characteristic to avoid runout on the curve, was developed. Pump case changes were introduced at the inlet splitters and at the volute tips.

Cavitation noise level to estimate erosion rates was measured as described by Guelich and Pace [1] and Guelich [2]. The original impeller's life with respect to cavitation was estimated. A 65 to 70 percent probability of a 16,000 hr life was determined based on data from the test bed and the field. Test results of the new configuration, both on the test bed and in the pipeline, allowed the expected impeller life to be quantified. A guarantee to reach at least double the original time span for the worst operating conditions possible, at maximum flow, i.e., single pump operation, could be given. More than three years could be guaranteed for operation at plant design conditions and the chance to reach 4.5 years was estimated to be about 80 percent. The extrapolated life has been verified through a prorata erosion rate during a field inspection after almost 3000 hrs of operation.

Vibration levels were significantly lowered due to the new hydraulic design with staggered impeller vanes. The comparison of old design and new design pump vibrations, taken during pipeline operation at the same station, indicate the clearly higher safety margin for the mechanical integrity of the upgraded machines.

Additionally, as a by product, the efficiency has been increased by about two percent, as the pump best efficiency flow had been adjusted to the effective operating flow in the pipeline.

INTRODUCTION

The pumping stations of the Rio Colorado Tijuana Aqueduct are pumping water from the Colorado River to the city of Tijuana (Figure 1). Station 0 at the river is equipped with vertical pumps and Stations 1 through 5 have four (one backup) double suction single or two stage water transport pumps from two different vendors running in parallel. General data for the units are presented in Table 1 and a cross section is given in Figure 2.

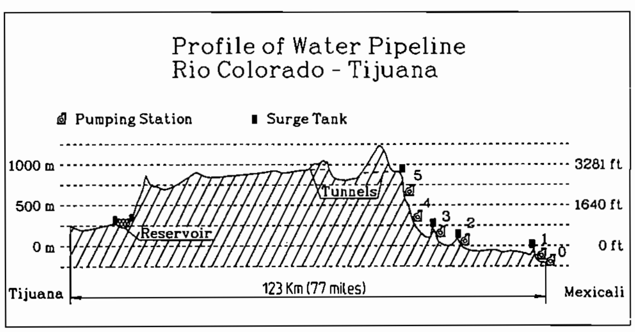

Figure 1. Profile of Water Pipeline.

Unfortunately, the available NPSH, mainly given through the differential elevation of the reservoir and the pumps, has not been distributed evenly for Stations 1 through 5. In addition, the suction piping was built quite less than optimally from a hydraulic point of view, in order to minimize construction cost. These facts leave the stations with a very marginal suction head, which consequently leads to cavitation. In stations 1 and 2, impellers have experienced the highest cavitation caused wear. This situation was unacceptable for the user and means were sought to increase impeller life.

A limited increase in the NPSH available would have been possible with improvement in the suction piping arrangement. Increasing the elevation of the suction side reservoir tank would also add to the NPSH available. However, those two constructive solutions, while easily attained during the original planning phase, were totally cost prohibitive since all the stations were already built.

Hence, in order to increase impeller life, a new hydraulic pump design for given operating conditions had to be found. As nothing but the pump hydraulic could be changed, the new design would have to work under the existing pipeline conditions. As the existing conditions are far less than optimum, compromises in

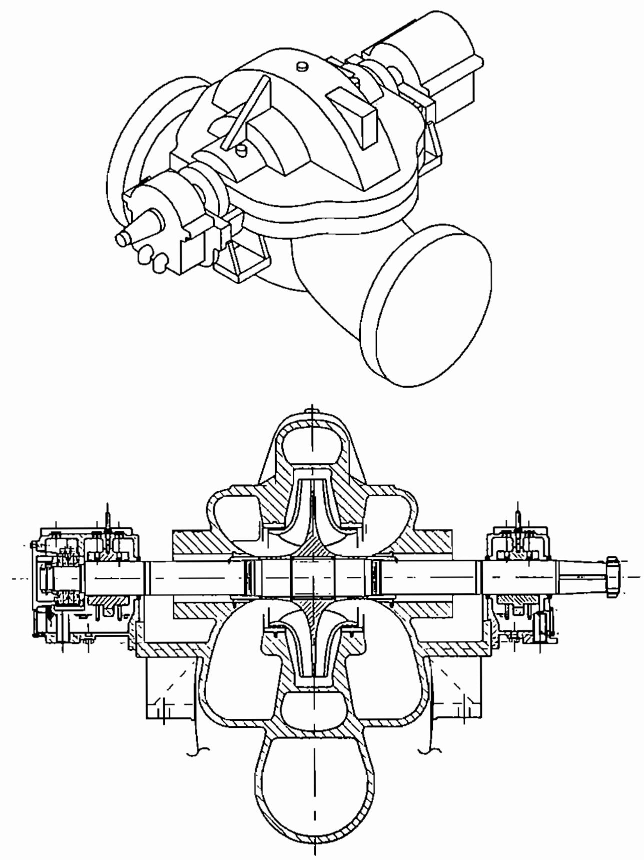

Figure 2. Cross Section of Water Transport Pump.

Table 1. Description of Pump.

Type	HSB 24x24x25C (disch., suc. nozzle, imp. diameter, [in])
Design	Horizontal, Single Stage, Double Suction, Center Hung
Speed	1780 rpm
Old Imp.	D_2=22.7 in =577 mm; D_1=14 in =355.6 mm; 7 Vanes No Stagger
New Imp.	D_2=22.7 in =577 mm; D_1=14 in =355.6 mm; 7 Vanes Staggered
Old Flow	24,200 gpm or 1.53 m^3/s at bep (87.5% efficiency)
Old Head	460 ft or 140 m at bep
New Flow	20,800 gpm or 1.31 m^3/s at bep (89% efficiency)
New Head	440 ft or 134 m at bep

respect to hydraulic behavior versus cavitation erosion had to be found. Some built in operating restrictions, e.g., a steeper head-capacity characteristic resulting in no flows above 110 percent best efficiency point (BEP), would normally not be desirable for a new pipeline layout.

Being aware of those restrictions, the user placed an order with the OEM to develop an improved hydraulic design with respect to cavitation erosion. Preliminary testing was to be done on the test bed and final testing was subject to a joint effort of the manufacturer and the user on the prototype pump installed and operating in the pipeline.

NPSH and Cavitation

The net positive suction head is defined as the total head of the fluid at the suction nozzle above the vapor pressure of the fluid and is therefore a measure for the margin against vaporization of the fluid entering the pump. The required NPSH increases with flow around and above the best efficiency point due to higher local fluid velocities. Therefore, a pump running at flows higher than the BEP may need a higher suction head to maintain sufficient NPSH margin than a machine running at BEP or somewhat below.

If the suction head, and subsequently NPSH value, is too low, excessive cavitation (flashing) takes place, i.e., vapor bubbles form within the lowest static head regions of the impeller and subsequently implode in regions with a higher static head. The implosions cause extreme stress on the impeller material and therefore cavitation erosion or wear.

Causes for Problems on Site

Station 2 of the Rio Colorado Tijuana Aqueduct has the lowest NPSH available (lowest static head from tank reservoir to pump center line). The system design point was to operate three pumps in parallel, but the pipeline was often operated with only one pump running per station. While multiple pumps running in parallel will operate very close to BEP, single pump operation will lead the pump to run at a flow considerably higher than BEP, as the system resistance (head) is reduced with the lower total flow (Figure 3 (typical)). At increased pump flow, the NPSH required is higher and, therefore, single pump operation leads to the fastest cavitation erosion induced impeller wear.

A typical impeller with cavitation erosion damage after a total of 16000 hr of pump operation, 3000 hr of single and 13000 hr of dual pump operation is shown in Figure 4. The maximum erosion was measured to be approximately 8.0 mm (0.315 in) deep, i.e., the end of useful life was reached for the impeller. Cavitation noise measurements taken during this project on the old design pumps indicated that around 14200 hr operation would cause 8.0 mm deep erosion (an erosion rate of about 0.000563 mm/h, see the paragraph on impeller life calculation).

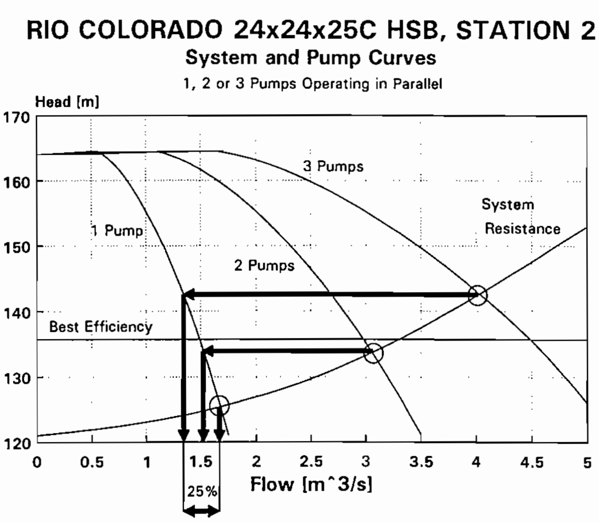

Figure 3. System and Pump Curves.

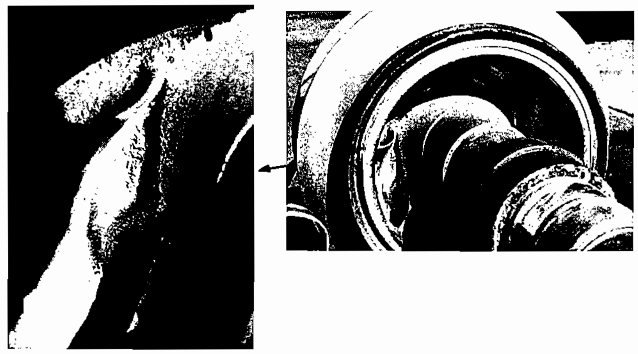

Figure 4. Typical Cavitation Erosion Damage after 16000 Hrs of Operation.

This difference between predicted and actual operating hours is not unexpected, given the mixed operating conditions, and the fact that prediction of cavitation erosion rates is based on statistical data.

The damage dimensions, i.e., the maximum depth and length, can be used to check the average erosion rate against the data base given in [5]. The similarity parameter for cavitation erosion, Θ_L, shown in Figure 5, fits well into the accumulated data of the data base. Accordingly, the erosion rate, E_R, can be calculated with the cavity length, L_{cav} (bubble length assumed to be equal to the distance from the vane tip to the location of maximum damage), and the equations given in [5]. The calculation and the agreement between measured and data base data are presented in Figure 6, and the probability distribution for this calculation is shown in Figure 7.

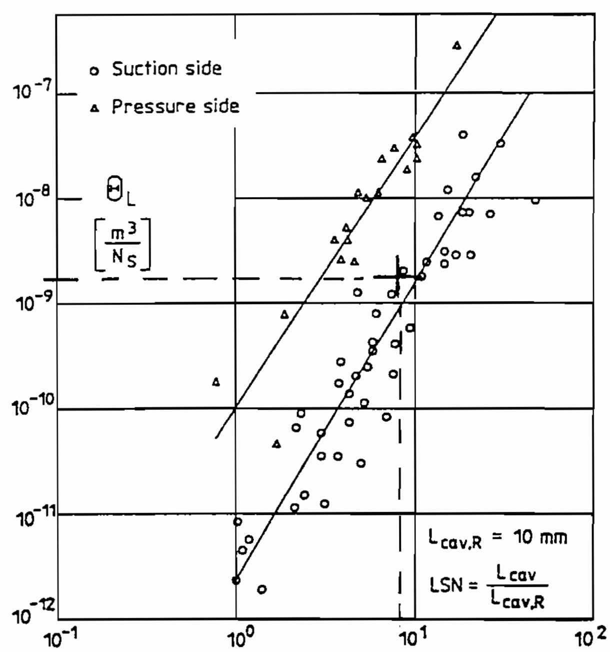

Figure 5. Cavitation Correlation Parameter for Impeller of Figure 4.

Erosion Rate Calculation Based on Bubble Length [5]
(Assumption: Bubble Length = Damage Length)

OL	Similarity Parameter for Cavitation Erosion
Emax	Maximum Erosion Depth
ER	Erosion Rate
Lcav	Cavity Length
Rm	Tensile Strength
po	Suction Pressure
psat	Saturation or Vapor Pressure of Liquid

For Water at Ambient Temp. in an Open Loop, Suction Side:

OL = (ER * Rm^2) / (po - psat)^3	(equ. 2-13 in [5])
ER = 2.3E-12 * (Lcav / 10)^2.83 * (po - psat)^3 / Rm^2	(equ. 2-14 in [5])

with

Emax (depth)	8	[mm]	0.315	[in]	
Oper.Hours	16000	[h]			(by user)
ER (meas.)	1.39E-07	[mm/s]	5.47E-09	[in/s]	
Lcav	85	[mm]	3.35	[in]	
Rm	560E+06	[N/m^2]	80000	[psi]	(average value)
po	2.94E+05	[N/m^2]	42.6	[psi]	
psat	1.70E+03	[N/m^2]	0.25	[psi]	

OL (graph.)	9.82E-10	[m^3/Ns]	(from Database)		(fig. 2-7 in [5])
OL (calc.)	1.74E-09	[m^3/Ns]	(from Meas. Res.)		(equ. 2-13 in [5])
ER (calc.)	7.82E-08	[mm/s]	3.08E-09	[in/s]	(equ. 2-14 in [5])

Measured Values relative to Averages of Database:

ER-Ratio	1.78	[-]	(Meas./Dat.base)
OL-Ratio	1.78	[-]	(Meas./Dat.base)

Probability for survival	W	31	[%]
	W = f(ER-Ratio), see figure 4d		

Figure 6. Comparison of Measured Corrosion Erosion with Database for Old Design.

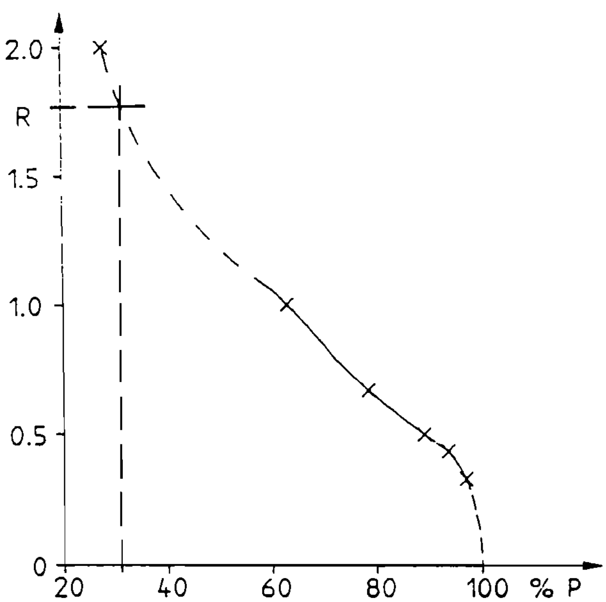

Figure 7. Probability Function for Impeller Life Prediction from Cavity Length.

From the above comparison of calculated and effective operating hours, it can be concluded that single pump operation of the old design hydraulic is worse than dual pump operation. This effect has been explained. The estimated erosion rate from the damage dimensions shows that the average erosion rate for the primarily dual pump operating hours is indeed lower than the estimate for single pump operation based on cavitation noise measurement.

While the comparison of measured damage with predictions based on cavitation noise was possible only after the test series, the comparison based on the cavity length (damage length) at the beginning indicated the classical cavitation erosion phenomenon was present.

DESIGN CHANGES

A new impeller design, less susceptible to damage, was developed at the OEM's headquarters in Switzerland, using state of the art methods. Flow conditions were slightly modified, and the design head was reduced to prevent unnecessarily high flows when operating a single pump against the system curve. Changes are described only globally, as the main focus here is not the hydraulic design but the successful field application. The main changes made to the impeller are indicated in Figure 8:

- The vane surface area at the inlet was increased with the vanes extended further towards the impeller eye and hence, the vane load was reduced.
- The vane inlet angle was reduced by an average of 3.5 degrees to reduce incidence.
- The vane leading edge was elliptically profiled to be less sensitive to variations in flow.
- The splitter between the two impeller halves was extended to the full impeller diameter to accommodate vane stagger of the two sides, reducing vane pass frequency excitation.
- The discharge side of the impeller was designed to produce a steeper curve slope by narrowing the discharge width by four percent and reducing the discharge vane angle by one degree. The steeper curve slope in conjunction with a reduction in impeller diameter limits the one pump operating flow to a reduced value.
- The impeller material was upgraded to increase cavitation erosion resistance.
- The impeller was cast using ceramic core techniques for better tolerances and improved surface finish as compared to normal sand castings.

As an additional improvement, pump case changes were introduced at inlet splitters, Figure 9, and at the volute tips, Figure 10:

- Inlet splitters were cut back and profiled to avoid vortex shedding into the impeller eye.
- Volute lips were profiled.

Again, it has to be noted that some of those hydraulic changes do not necessarily represent the best solution for a new pipeline layout, but are needed to achieve reasonable impeller operating life time under the existing adversary boundary conditions.

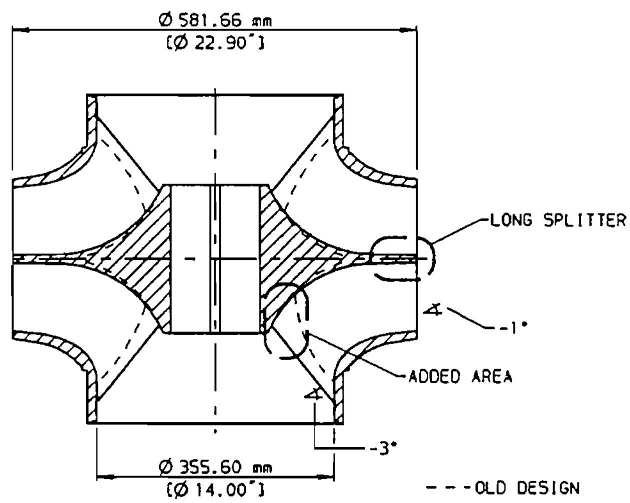

Figure 8. Impeller Design Changes.

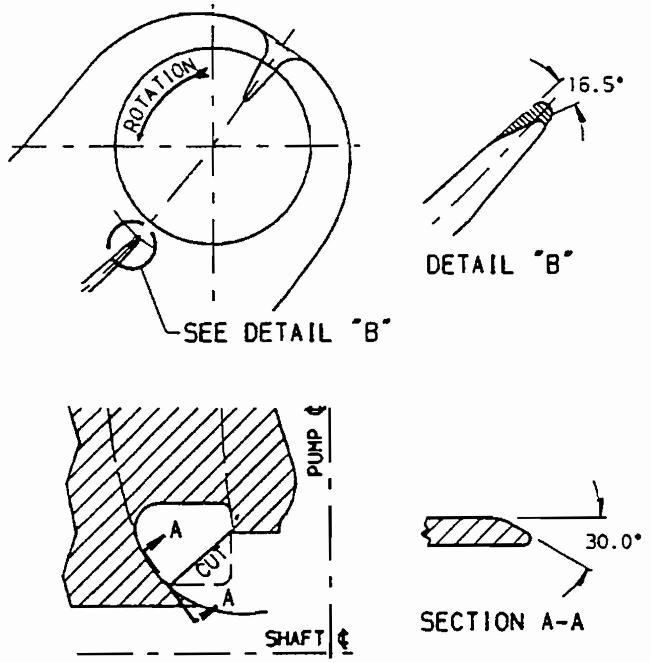

Figure 9. Pump Case Design Changes, Inlet Splitters.

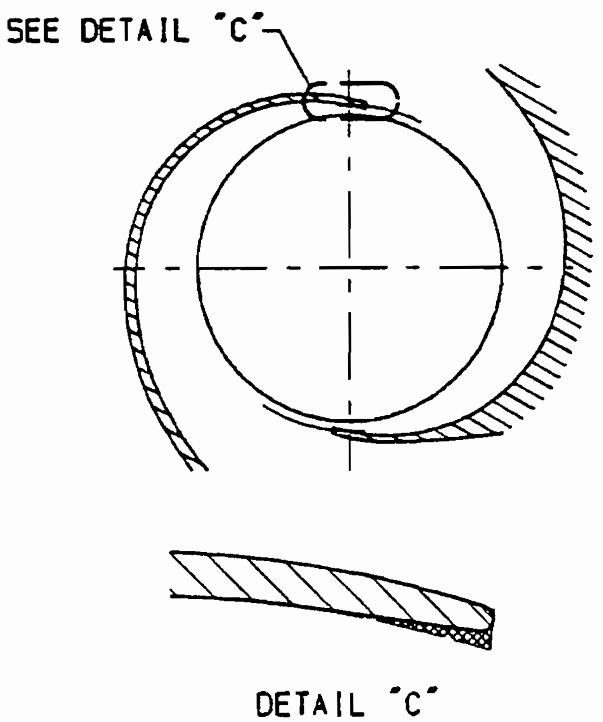

Figure 10. Pump Case Design Changes, Volute Tips.

TESTING

Testing Methods Available

Required NPSH values are a measure for the potential of a pump to be cavitating and for possible head loss caused by the cavitation. It is indicated by most manufacturers for one to three percent head loss, as this operational condition can be measured relatively easily. However, NPSH values per se, do not directly define cavitation erosion wear and damage. Literature indicates that cavitation starts long before the pump discharge head deteriorates [3, 4]. A typical example is shown in Figure 11, where, for a constant flow, the pump head is plotted against the (normalized) NPSH or suction head. It can be seen that, below a certain suction head, the pump head starts to drop, but the cavitation bubbles start to appear much earlier. In the case of Rio Colorado-Tijuana, the available static tank levels do not allow operation with enough NPSH margin to totally preclude cavitation damage.

Cavitation Bubble Distribution and Weight Loss per Unit Time as a Function of Cavitation Coefficient at Constant Speed

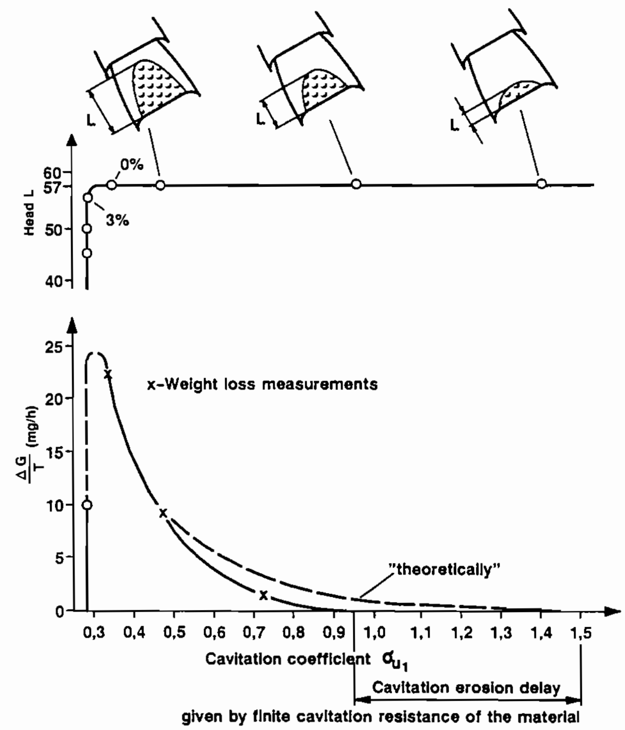

Figure 11. Cavitation Bubbles Vs NPSH.

Hence, while measuring the NPSH is relatively easy and shows where the pump performance starts to deteriorate, it is not a definitive indicator of where cavitation starts and how severe cavitation damage is, i.e., what the erosion rate is. NPSH values and corresponding cavitation related head drops are *global* information, describing pump behavior, but are not a direct measure of the *local* cavitation erosion and wear.

Two methods of obtaining reliable results and prediction of cavitation wear have been developed and are well documented in literature [1, 2]. The observation of bubble extension on the impeller through a plexiglas window in the casing, normally done for model testing, may be most straightforward, however, it was not feasible for the pump to be investigated. The measurement of cavitation noise with a piezoelectric pressure probe for the acoustic pressure waves in water needed only a minor modification, i.e., a tap in the suction chamber, of the pump casing. Cavitation noise is at very high frequency and is measured in the frequency range from clearly above vane pass frequency up to 180kHz, while the maximum noise is expected to be around 80 to 100Khz.

With the noise produced by the imploding cavitation bubbles, and the data base on cavitation noise established by Sulzer under EPRI contract 1884-10 [5], estimated wear over time (mm-vane-thickness per hour) can be calculated. This method has been verified on numerous model tests and on some full scale tests [6]. Obviously, the cavitation is not the only source of noise, and it must be ensured that the background noise level is not too high. With excessive background noise, the level of cavitation noise cannot be evaluated. Alternatively, no air should be entrained in the fluid as this would dampen the transmission of the cavitation noise at the impeller blades through the water to the probe in the casing.

Cavitation noise measurement will typically give a nearly constant background level at very high NPSH levels, long before head loss occurs. The noise then starts to increase below a certain NPSH level, still without head loss. The cavitation noise then peaks *around* the NPSH level for one percent head loss and decreases for lower NPSH levels. The decrease takes place due to the dampening effect of the cavitation (vapor) bubbles on noise transmission from the location of implosion to the location of measurement.

Additional Parameters

The (dissolved) gas content of the fluid, an important parameter for cavitation erosion prediction, did not need to be determined for the investigated case, as the system in the plant is open (tank surface) as is the system on the test bed. In an open system, the fluid is assumed to be saturated with gas.

Pressure pulsations at frequencies lower than cavitation noise result from bubble oscillations and from impeller vanes passing the volute tips. These pulsations in the range of 1.6 to 2,200 Hz were measured in the suction and the discharge chamber of the pump, in addition to the cavitation noise.

Testing Configurations

On the test bed, the old design and new design pumps were measured. Suppression tests were done for varying flows. Hence, NPSH characteristics for both configurations can be compared and the influence of the changes evaluated.

The cavitation noise measurement on the test bed was difficult. The suppression tank of the test loop contained only 10,000 gal, compared to the rated 20,000gpm flow of the pipeline pump. At BEP of the test pump, the equivalent of one tank content was passed through in less than half a minute and air probably was sucked into the flow. The background noise of the loop was high, and the cavitation noise measured did not appear as a peak. Only increased damping of the noise signal with increasing entrained air content and larger volumes of cavitation bubbles was found.

While this effect does not impair the global $NPSH_R$ measurement, it does distort the absolute levels of cavitation noise. Therefore, the absolute cavitation noise levels measured on the test bed cannot be used for erosion prediction. Yet, assuming comparable levels of entrained air and noise damping for identical setups, the test bed noise levels allow for a relative comparison of the old and the new hydraulic design in respect of cavitation erosion.

A detailed description of cavitation noise measurement instrumentation and setup are described by Guelich [2, 5].

Vibration measurements on the machine were taken on the bearing housings. The test bed support stiffnesses were much lower than on site and, therefore, vibration readings of individual configurations on the test bed are elevated compared to those achieved at site. Because of this, vibration readings of the original pumps and the modified pump are compared only for the units as installed in the pipeline.

Cavitation noise measurement was also done on site with the same prototype pump. Here, the pump runs in the environment for which it was designed, and no problems with cavitation noise measurement were encountered. The limiting factor on site was the small possible variation of suction pressure. The suction head and hence the NPSH available on site is given by the geodetic head, i.e., the water level within the reservoir tank plus the elevation of the tank compared to the pump. The static head can be varied by the tank level from about 26ft (8.0m) down to some 6.5ft (2.0m), resulting in a relatively small variation of actual NPSH available. Effectively measured NPSH in the plant include velocity head and piping losses and vary from 94 ft to 74ft (28.5 m to 22.5m) only. Cavitation noise measurement on the remaining three pumps (one spare) could not be measured, as their casings had not been tapped for the instrumentation.

Test Results

Hydraulic Data: The pump head-capacity performance is shown in Figure 12 and the suppression test results are shown in Figure 13 for one and three pump operating conditions with head,—$NPSH_A$ curves.

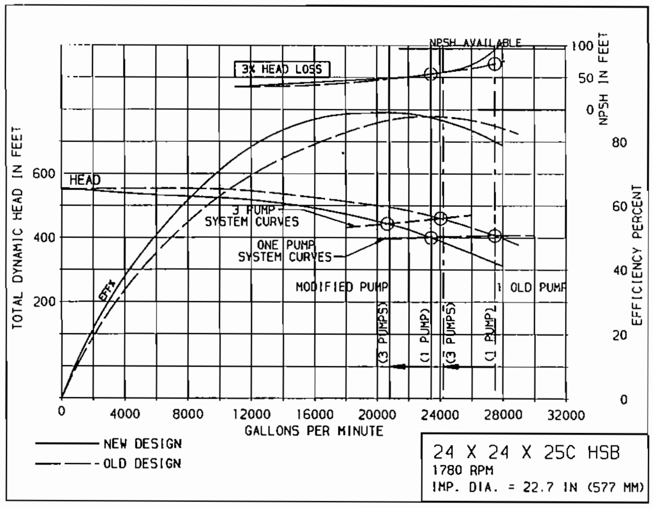

Figure 12. Hydraulic Performance Old-New.

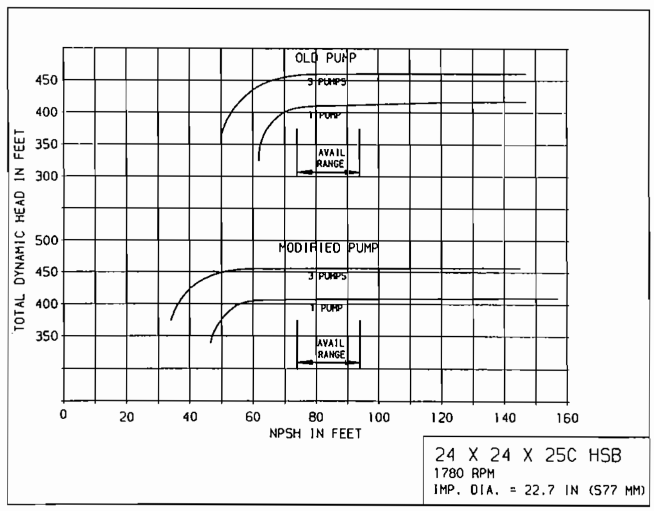

Figure 13. Suppression Test Old-New.

A comparison of the two tests shows the distinct change in curve slope and BEP of the new hydraulic design. The NPSH three percent characteristics of the two designs are similar near the individual rated flows, however, the NPSH three percent of the new design is increased at higher capacities due to the optimization of the new impeller's suction vanes with respect to cavitation for the lower flows.

A test bed comparison of the performance of the old impeller in the unmodified case, and the new impeller with the modified case clearly indicates the improvements made through design changes. Plotting the system curves on the pump characteristic (Figure 12) shows the limitation in the maximum operating flow, and the accompanying reduction in NPSH required. The increased performance curve slope reduces the difference in the flow per pump between one- and three-pump operation. Originally, the difference was more than 17 percent per pump while the new difference has been reduced to 14 percent. The maximum single pump flowrate is lowered from 117 percent to 112 percent of BHP with the design change.

The margin of $NPSH_A$ against $NPSH_R$ is indicated in Figure 13 for the two designs under the limitations given in the pipeline. It can be seen that for single pump operation the margin is smaller than for three pumps running in parallel. In addition, the increase in NPSH margin with the new design under both types of operation becomes obvious in Figure 13.

Operating conditions for the prototype pump on site could not be varied over as wide a range as on the test bed. It was decided that the readings were to be taken for the test pump running at maximum flow, where the most severe cavitation conditions exist, and at 100 percent flow as a reference. The tank reservoir was to be varied from the maximum to the minimum levels permitted for maximum flow, in order to vary the $NPSH_A$ to its most extreme values. The static suction head could be varied from 14 m to 20 m (46 ft to 66 ft), which leads to a possible variation of $NPSH_A$ from 24m to 30 m (74 ftto 94 ft), only.

The pump capacity was measured *indirectly* by three means:

a) From the pressure differential across the pump with the known pump characteristic, the capacity could be determined. This is assumed to be the most accurate method.

b) The common discharge header was tapped and the velocity profile of the flow was measured with a Pitot tube. The velocity profile was integrated to yield the total quantity. The error of this method is small, and the flow measured compares very well with method a) (Table 2 for numerical values).

c) Current, voltage, and power factor were measured on the three phase motor. The average power multiplied by the power factor and by the efficiency defined the power input to the pump. This was used on the pump characteristic curve to estimate the capacity. The relatively flat BHP curve and the procedure of calculating the power input to the pump includes some errors, and the flow estimated is the least reliable.

The maximum flow occurred with the maximum suction head at the maximum tank level. This flow was found to be about 103 percent of BEP and decreased with lower tank levels. As maximum flow and BEP flow were almost identical, only one point was measured exactly at BEP flow.

Flow and head measurement at Pump Station 2 indicates that the system curve data originally obtained from the user is slightly low. The measured one-pump flow at Station 2 was 1.412 m^3/s (22,380 gpm) vs the expected value of 1.476 m^3/s (23,400 gpm), and targeted maximum flow of 1.527 m^3/s (24,200 gpm). Increased impeller diameters could increase capacities, if the user desires such a change. This can be done without endangering the current impeller life guarantees, as long as the maximum flowrate per pump is restricted by throttling to flows below 1.527 m^3/s.

Cavitation and Pressure Pulsation Data: The normalized cavitation noise levels (CNL) where no background noise is subtracted are presented in Figure 14. The pressure pulsations measured in the suction chamber and in the discharge nozzle are displayed in Figure 15 and 16.

Table 2. Hydraulic Characteristics.

Measured Values (Speed Corrected)					
Point #	Speed [rpm]	Head [ft]	Flow [gpm]	Power [bhp]	Eff. [%]
1	1760	407	22018	2622	86
2	1760	397	22043	2664	83
3	1760	404	22043	2626	87
4	1760	408	22043	2605	88
5	1760	412	22055	2594	88
Deviations of Field Values From Test Bed Values					
Point #	Speed [rpm]	Head [%]	Flow [%]	Power [%]	Eff. [%]
1	1760	0	-0.4	1.4	-2.5
2	1760	0	-2.9	2.4	-5.5
3	1760	0	-0.7	1.4	-1.2
4	1760	0	-0.0	1.0	-0.3
5	1760	0	0.7	0.9	-0.5

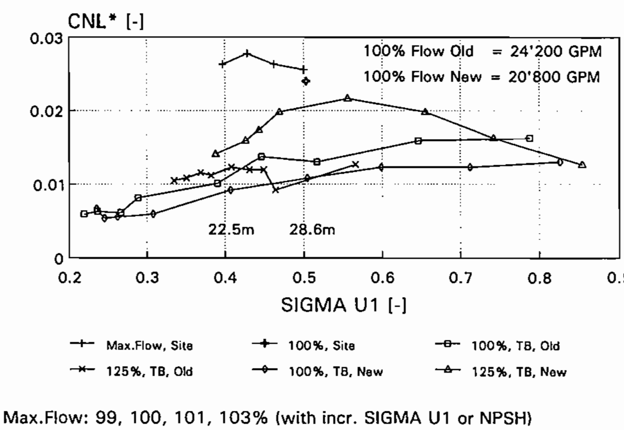

Figure 14. Cavitation Noise Level (including background).

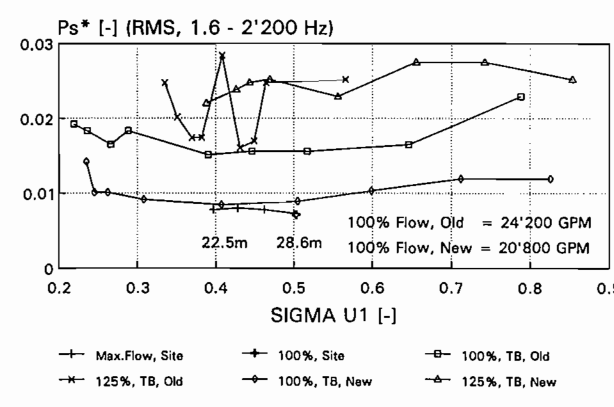

Figure 15. Suction Pressure Pulsation.

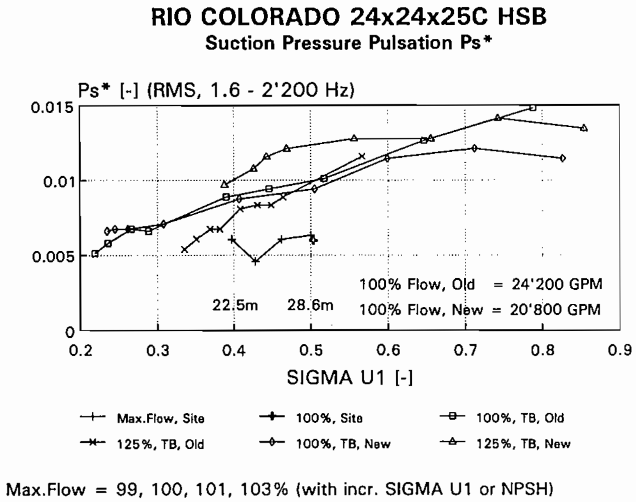

Figure 16. Discharge Pressure Pulsation.

As discussed earlier, the cavitation noise level data measured in the test loop were rendered unreliable for *absolute* calculations due to excessive air entrainment from the reservoir. As the nominal maximum content of the biggest reservoir available was turned over twice per minute, the chance of including air bubbles in the water and for high background noise was given. However, the *relative* magnitudes of cavitation noise measured provide a clear indication that, as intended, the new hydraulic design provides an improvement of at least 20 to 30 percent at design flow. At a flow of 125 percent of BEP, no improvement was achieved, as the new hydraulic was optimized for flows close to 100 percent (the shockless flow for the old design was above BEP and for the new design close to BEP). However, higher flows had to be precluded by the new design anyway, as the margin of $NPSH_A$ available to $NPSH_R$ in this range was too small (Figure 12) to allow the operation of any hydraulic design without fast cavitation erosion.

The relative change in cavitation noise levels for the new and old design found on the test bed was used in conjunction with the cavitation noise measurement on the new design installed in the plant in order to estimate the effective noise level of the old design. The erosion rate of the old design was estimated with this absolute cavitation noise level in accordance to Guelich [5], and was compared to the cavitation erosion rate as found on the old impeller shown in Figure 4. As mentioned earlier, this comparison proved to be quite satisfactory (16,912 estimated operating hours *vs* 16,000 actual operating hours).

The new hydraulic design provided generally lower suction and discharge pressure pulsations. The suction pressure pulsations for nominal flow at the sigma (normalized $NPSH_A$) values expected in the field (around sigma 0.45) differed only slightly for test bed results. The suction pulsations were lower for the pump in the pipeline, indicating superior conditions upstream of the pump.

The discharge pressure pulsations clearly indicate the advantageous influence of the case work and the impeller vane is staggering. The new design exhibits a strong reduction in discharge pressure pulsations at BEP, and the test bed measurement was repeatable in the field. At flows considerably above design conditions, no improvement can be attained, as the flow pattern is already otherwise distorted.

Cavitation noise and pressure pulsation data were taken in the same way as on the test bed, though in the limited range available. Results were found to be in the same order of magnitude as compared to test bed data. Numerical data are given in Table 3.

The suction and discharge pressure pulsations measured on site were, to some extent, lower than measured on the test bed. The cavitation noise levels however, were distinctly higher at site—a further indication that the air entrainment on the test bed damped the cavitation noise and that the background noise on the test stand was dominating. The cavitation noise levels obtained at site were sufficient for performing an impeller life calculation.

Table 3. Cavitation Noise and Pressure Pulsation Measurement.

Nondimensional				
Point #	Sigul [-]	Cnl* [-]	Puls* S [-]	Puls* D [-]
1	0.503	0.024	0.006	0.007
2	0.500	0.026	0.006	0.007
3	0.462	0.026	0.006	0.008
4	0.428	0.028	0.005	0.008
5	0.397	0.026	0.006	0.008
Dimensional				
Point #		Cnl [bar]	Puls S [bar]	Puls D [bar]
1		0.132	0.096	0.105
2		0.142	0.094	0.109
3		0.146	0.090	0.116
4		0.154	0.068	0.119
5		0.146	0.090	0.116

Vibration Data: As the test bed set up was only a temporary arrangement with relatively low support stiffnesses, only measurement from the pumps running in the pipeline are presented.

The bearing housing vibration signatures of the modified and one original pump on site are shown in Figures 17 and 18, respectively, for the inboard horizontal and vertical and the outboard horizontal, vertical, and axial direction. The influence indicated in Figure 19 is of decreasing $NPSH_A$ on the inboard bearing housing vibrations in vertical direction for the modified pump. The bottom curve on the graph is at maximum $NPSH_A$ and the top curve is at minimum $NPSH_A$.

Generally, it can be noted that the vibration level of the pumps are low (note the fine scale). In the vertical direction, a slight increase of vane pass frequency vibrations can be found for decreasing $NPSH_A$ (Figure 19). Vibrations of 1 × rpm and 2 × rpm vibrations indicate small residual unbalance and a very minor misalignment. The staggered vane configuration of the new impeller (Figure 17) leads to a small two × vane pass frequency. The overall vibrational behavior of the modified pump on site is

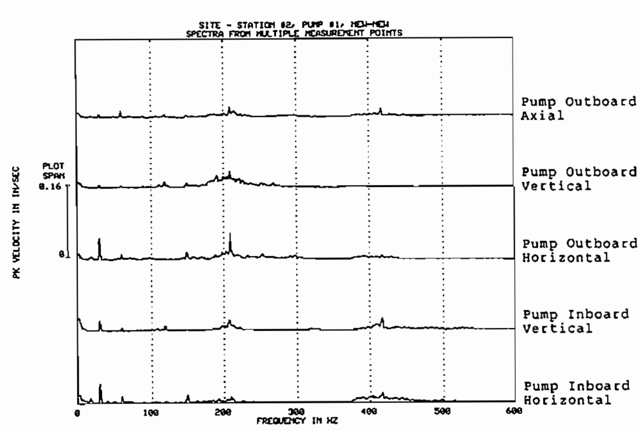

Figure 17. Pump Vibrations on Site, New Design.

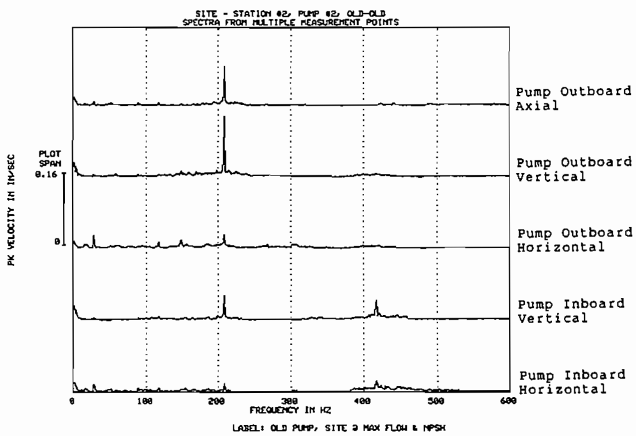

Figure 18. Pump Vibrations on Site, Old Design.

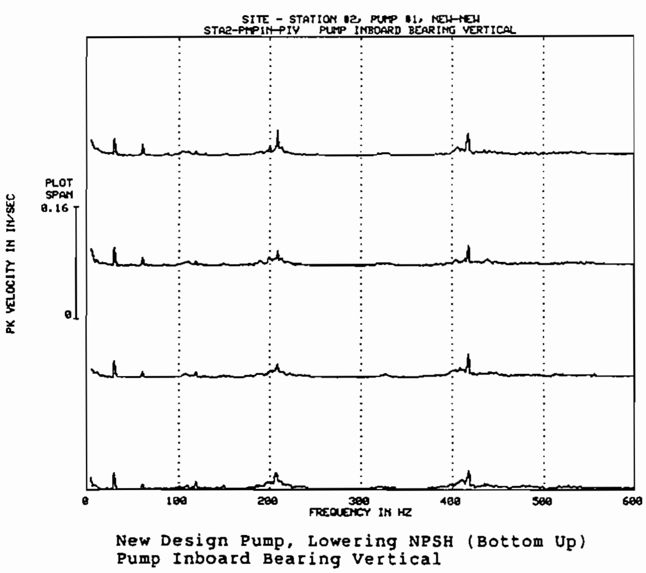

Figure 19. Pump Vibrations on Site, Varying $NPSH_A$.

excellent, and it is apparently the smoothest running pump in the plant.

The overall vibrational behavior of the original pumps (Figure 18) at maximum $NPSH_A$ is acceptable, though peaks are generally about twice as high as for the modified pump. Vane pass frequency vibrations are predominant while unbalance and misalignment vibrations are as low as can be expected.

IMPELLER LIFE EVALUATION

New Impeller Design

The detailed formulas and data for the life evaluation are presented by Guelich and Pace [1] and Guelich [2], and the probability function for impeller life prediction from cavitation noise level is given in Figure 20. The end of life conditions are reached when 75 percent of the vane thickness are eroded. The calculated impeller erosion rates shown in Figure 21 are based on the most reliable measurements taken in the field. The calculations allowed the manufacturer to *guarantee* a lifetime of 25,000 hr for three pump operation.

Considering the increased flow per pump for single pump operation at a lower NPSH margin ($NPSH_A/NPSH_R$) the lifetime could be *guaranteed* to exceed 13,000 hr for one pump operation.

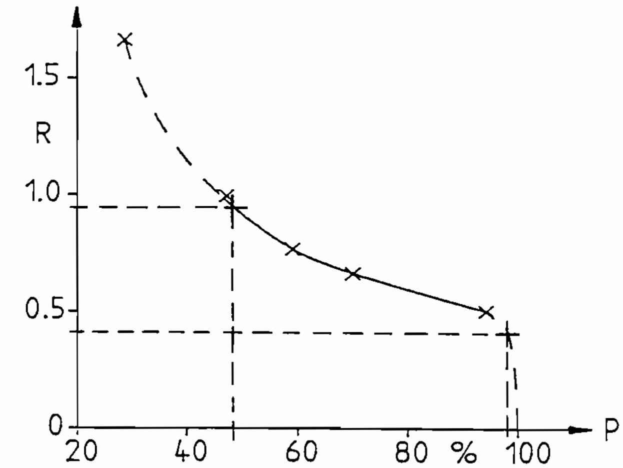

Figure 20. Probability Function for Impeller Life Prediction from Cavitation Noise Level.

Impeller Life Calculation (New Hydraulic) [1], [2]
Plant: Rio Colorado - Tijuana, BC, Mexico
Pump Type 24x24x25C HSB

Impeller	Blade Thickness	12.7	[mm]	Vanes	7	
	Eye Diameter	0.3556	[m]	u1	33.3	[m/s]
Impeller Material	CA6NM	Martensitic				
	Rm	8.37E+08	[N/m^2]	Fmat	1	[-]
Water	Density	1000	[kg/m^3]			
	Chem. Comp.:			Fcor	1	[-]
Flow	BEP	1.31	[m^3/s]			
Speed		1787	[RPM]			

Noise Level	CNL*	0.0268	[-]	CNL	14835	[N/m^2]
	CNL = (CNL*) * Density * u1^2 /2					

Erosion Rate	ER	1.56E-04	[mm/hr]	4.34E-08	[mm/s]
	ER = 9.4 E9 * Fcor / Fmat * (CNL / Rm)^2.9				

Calculated Impeller-Life	IL calc	60946	[hr]
	IL calc = 0.75 * Blade_Thickness / ER		
Required Impeller-Life	IL req	25000	[hr]
	(for guarantee)		
Life Ratio	RL	0.41	[-]
	RL = IL req / IL calc		
Probability for survival	W	95	[%]
	W = f(RL), see figure16c		

Figure 21. New Impeller Life Calculation [1, 2].

This will be achieved using the new impeller design, when operating at an NPSH of 25 m (83 ft) or more.

Old Impeller Design

The old impeller exhibited 20 to 30 percent higher cavitation noise than the new impeller. The material was austenitic which, according to literature, sets the material factor (F_{Mat}, Figure 22) to calculate the erosion rate at 1.7 instead of 1.0 for CA6NM. The original 316 stainless steel also has a lower tensile strength (Figure 6) than the new material.

Together, all these elements increase the predicted erosion rate by a factor of about 3.6 to 0.000563 mm/h for the old design impeller as compared to the erosion rate for the new impeller, see Figure 22 (at BEP flow).

INSPECTION OF PROTOTYPE IMPELLER ON SITE

After almost 3000 hrs of operation, the impeller of the pump with the new hydraulic design was inspected on site. The measurement of the material erosion was supposed to allow for a calculation of the remaining life of the impeller and to check the guarantee given by the manufacturer.

Impeller Life Calculation (Old Hydraulic) [1], [2]
Plant: Rio Colorado - Tijuana, BC, Mexico
Pump Type 24x24x25C HSB

Impeller	Blade Thickness	12.7	[mm]	Vanes	7	
	Eye Diameter	0.3556	[m]	u1	33.3	[m/s]
Impeller Material	316 SS	Austenitic				
	Rm	5.60E+08	[N/m^2]	Fmat	1.7	[-]
Water	Density	1000	[kg/m^3]			
	Chem. Comp.:			Fcor	1	[-]
Flow	BEP	1.53	[m^3/s]			
Speed		1787	[RPM]			
Noise Level	CNL* estimated*)	0.0335	[-]	CNL	18543	[N/m^2]
	CNL =(CNL*) * Density * u1^2 /2			*) 125% of New Hydraulic		
Erosion Rate	ER	5.63E-04	[mm/hr]	1.56E-07	[mm/s]	
	ER = 9.4 E9 * Fcor / Fmat * (CNL / Rm)^2.9					
Calculated Impeller-Life	IL calc	16912	[hr]			
	IL calc = 0.75 * Blade_Thickness / ER					
Required Impeller-Life	IL req	16000	[hr]			
	(for guarantee)					
Life Ratio	RL	0.95	[-]			
	RL = IL req / IL calc					
Probability for survival	W	35	[%]			
	W = f(RL), see figure16c					

Figure 22. Old Impeller Life Estimate [1, 2].

The data logging by the user indicated the durations of operating conditions as indicated in Table 4. With this information, the used percentage of the calculated available life time can be estimated. Accordingly, the maximum allowable wear rate to be found acceptable during inspection is established.

Table 4. Pro Rate Live Evaluation.

Operating Units	Operating Hours	Guaranteed Hours	Percentage Used	Predicted Erosion [mils]	Predicted Erosion [mm] *)
1 Pump	985	13000	7.6%	28	0.72
2 Pumps	1955	18000	10.9%	41	1.03
3 Pumps	0	25000	0%	0	0.00
Total	2940	N/A	18.5%	69	1.75

*) Vane thickness is 12.7 mm (0.5 in), End of Live is reached when 75 percent of the vane thickness is eroded

The inspected prototype impeller is shown in Figure 23. As the picture suggests, no metal loss has occurred. Measurement of the vane thickness with an ultrasonic probe revealed variations well below 10 mils, i.e., clearly within casting tolerances, only.

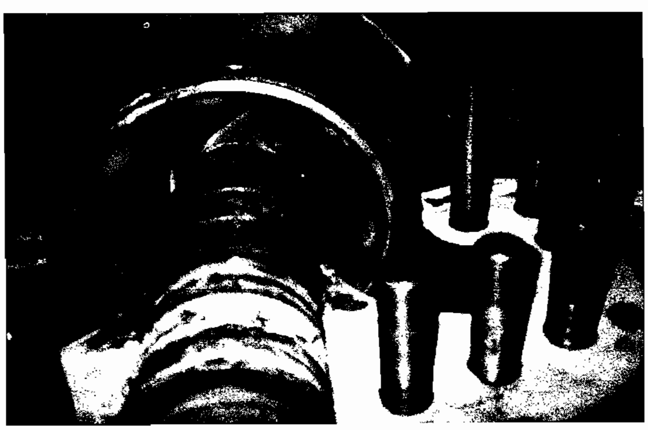

Figure 23. Inspected Prototype Impeller after 2940 Hrs of Mixed One and Two Pump Operation.

However, the area close to the vane inlet appears to be polished, indicating the region where some cavitation still persists. As has been pointed out earlier, the existing boundary condition, i.e., the available NPSH in the station, is too small to prevent cavitation entirely. But the new design clearly lowers the intensity of the cavitation and eliminates premature impeller wear and erosion. The results found during inspection were evidently better than the predicted results and, most certainly, the guaranteed life time of the impeller will be exceeded.

CONCLUSIONS

A case of inadequate construction planning in a pipeline pumping station has led to quite unfavorable boundary conditions for the pumps with respect to available suction head, i.e., to marginal $NPSH_A$. As construction changes were absolutely cost prohibitive, hydraulic design changes were the only means to improve cavitation erosion resistance and, hence, the life time of the impellers. However, the existing installation required compromises in respect of operating ranges versus cavitation erosion.

In a joint effort with the user of the Rio Colorado Tijuana Aqueduct, the OEM has developed a new hydraulic design to increase and guarantee life expectancy of the impellers installed in the pumps of the pipeline. The original and the new design were tested on the manufacturer's test facilities, and the new design pump was finally tested installed in Station 2 of the aqueduct.

The testing techniques applied allowed measurement of the relevant parameters to determine cavitation erosion. Comparison of test results on the test bed and data taken on site confirmed the superiority of the new design and the guaranteed life expectancy.

Single pump operation results in a shorter life of the impeller based on the higher flow of the unit. The hours of operation will be thoroughly monitored and logged by the user and will determine the remaining hours of guaranteed impeller life during operation. The new impeller was inspected by the manufacturer after 2940 hrs of mixed operation (calculated equivalent to 10 to 11 percent loss of vane thickness). Based on the inspection data, which revealed no material loss or roughening/pitting at all, the predicted life time proved to be substantially higher than the calculated and guaranteed values.

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