PUMP IMPELLER LIFETIME IMPROVEMENT THROUGH VISUAL STUDY OF LEADING-EDGE CAVITATION

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ABSTRACT

This paper describes the successful application of cavitation flow visualization to redesign the impeller of an 8½-MW (11,400 hp) high energy double suction single stage boiler feedpump. Dual purpose of the investigation was to develop an impeller/pump case combination showing significantly less cavitation at the impeller leading-edges than the original design, and realize specific rate objectives for meeting new duty points to eliminate high pressure throttling over the discharge valve.

New impeller designs were tested in a full scale model test pump running at reduced speed, which was equipped with an acrylic viewing window for direct observation of the impeller blade cavitation. Elements of the experimental effort included determination of the effect of impeller (vane) design changes, along with the effect of suction box changes introduced at the splitter vanes.

Initially, a total of four alternative impeller designs were considered, from which the potentially best design was selected and tuned to specification. The new impeller design developed

typically had sharp elliptic leading-edges to overcome the cavitation problem.

Compared with the original design, in which impellers were replaced/repaired after less than one year of operation, the new impeller lifetime was calculated to be improved by at least a factor of eight, on the basis of cavitation bubble length. The NPSH required for the new design proved to be much better than for the original design, and, additionally the efficiency was established to be improved by one to two percent. Furthermore, a suction box splitter vane modification has led to less (cavitation) bubble activity in the eye area.

INTRODUCTION

About 10 years ago Gülich (1986, 1988, 1989a, 1989b) published the results of his work sponsored by the Electric Power Research Institute (EPRI) on the assessment of cavitation erosion in centrifugal pumps employing cavitation noise level and cavity length. Since then, numerous measurements in various installations in completely different pump types from different manufacturers have corroborated the *empirical* damage correlations of Gülich in a statistically relevant manner; e.g., Cooper, et al. (1991), Van der Westhuizen (1992), Florjancic, et al. (1993), Sloteman, et al. (1995), and Ferman, et al. (1997).

In line with these previous works, the current paper focuses on the visual study of cavitation occurring at the entrance of the impeller of an 8½-MW (11,400 hp) high energy double suction single stage boiler feedpump. The problem that is discussed concerns three (50 percent) 20 × 20 × 18 B HDR feedwater pumps suffering from premature impeller wear, due to cavitation attack on the vane leading-edges. These pumps are running in the secondary loop of a 1050-MW (1,400,000 hp) PWR nuclear power plant, and feed the plant's steam generator. Because these units were producing too much head due to block load condition changes, the customer placed an order with the pump manufacturer to rate the units for meeting new duty points, in order to eliminate the high pressure throttling over the discharge valve, and develop an impeller/pump case combination showing less cavitation at the impeller leading-edges than the original design. The work program that was set up to accomplish this was a joint effort between the customer and the pump manufacturer, in which:

- The problem was investigated,
- A test program was conducted,
- Redesign of impellers and pump case was completed, and
- Replacement impellers were commissioned (both on the test floor and in the field).

It is outlined how this particular project was handled; where attention is particularly confined to elements of the experimental effort, including determination of the effect of impeller (vane) design changes, along with the effect of suction box changes introduced at the splitter vanes.

BACKGROUND—NPSH AND CAVITATION

Cavitation is defined as the process of formation and disappearance of the vapor phase of a liquid, when it is subjected to reduced and subsequently increased pressures at constant ambient temperatures. The formation of cavities is a process analogous to boiling in a liquid, although it is the result of pressure reduction rather than heat addition. Nonetheless, the basic physical and thermodynamic processes are the same in both cases.

Clearly, from an engineering and design point of view, there are two basic questions regarding cavitation. First, one has to answer the question whether cavitation will occur or not, and secondly, if cavitation is unavoidable, the question is whether a given design can still function properly. Economic or other operational considerations often necessitate operation with some cavitation, and under these circumstances it is particularly important to understand the (deleterious) effects of cavitation.

Occurrence of Cavitation

A liquid is said to cavitate when:

- Vapor bubbles form and grow as a consequence of pressure reduction, and
- Vapor bubbles subsequently disappear or collapse due to a pressure increase.

Such bubble formation is nearly always accompanied by production of gases previously dissolved in the liquid. The phase transition resulting from the hydrodynamic pressure changes yields a two phase flow composed of a liquid and its vapor phase, which is called a cavitating flow. Obviously, a cavitating flow can imply anything from the initial formation of bubbles to large scale attached cavities (known as supercavitation). Nowadays, such cavitating flows are rather common occurrences, since designers are pushing for higher speeds for given sizes in the development of pumps (thus creating lower pressure areas).

Cavitation Inception and Three Percent Head Drop

The first appearance of cavitation is called cavitation inception. When the pressure is decreased from this inception level, the region of cavitation enlarges, eventually starting to cause noise, performance change, and possibly cavitation damage. The latter results from the fact that well beyond inception, the pressures associated with cavity collapse are high enough to cause failure of the impeller material. By the time the inlet pressure is lowered enough to cause a one to three percent drop in pump head, cavitation is usually fully established.

The above mentioned stages of cavitation are illustrated in Figure 1, in which the total pump head (H) is plotted against the net positive suction head (NPSH) for constant volume flowrate (Q) and constant angular speed (N). This abstraction, termed NPSH, is defined as the total head of the fluid at the suction nozzle above the vapor pressure of the fluid, and can be regarded as a measure for the margin against vaporization of the fluid entering the pump. From a set of test curves like Figure 1, it is possible to develop the NPSH required characteristic (say NPSH at three percent head drop) as a function of the throughflow; that is, by determining the cavitation points for three percent head drop at different (Q/N) operating points, i.e., at different specific flowrates $\Phi = Q/(\Omega R^3)$.

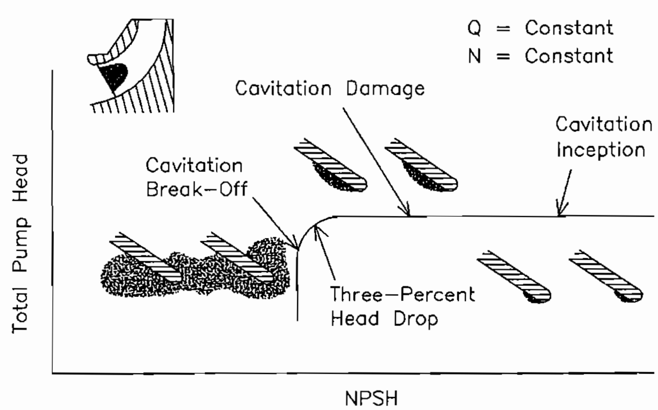

Figure 1. Cavitation Phenomena.

Cavitation Damage—Suction Specific Speed—$NPSH_{40,000}$

Cavitation Damage—Cavitation damage starts somewhere between inception and three percent head drop (Figure 1). A more accurate description is difficult to give, since many parameters influence bubble geometry and its potential for causing damage. For instance, impeller material, air content, NPSH available, vane geometry, inlet geometry, type of cavity, fluid density, and water temperature, to name a few, can be contributors or inhibitors of cavitation damage. The only certainty is that the absence of visible cavities means that cavitation damage will not be an issue. This fact is used in some conservative designs, such as liquid sodium pumps, where the NPSH available is high enough to suppress cavitation. However, in this era of intense competition, the designer pushes *suction specific speeds* (see below) to the point where suppression of visible cavitation is impossible.

Suction Specific Speed—The suction specific speed (S) determines the susceptibility to cavitation, and is defined as (for instance, Brennen, 1994):

$$S = \Omega Q^{1/2} / (\mathrm{NPSE})^{3/4} \quad (1)$$

in which:

Ω = angular speed
Q = volume flowrate through the pump
NPSE = net positive suction energy = g NPSH

with:

g = acceleration due to gravity

Like the common specific speed, $N_s = \Omega Q^{1/2}/(g\ H)^{3/4}$, the suction specific speed is a dimensionless number, and should (preferably) be computed using a consistent set of units. A typical (i.e., critical) value for the suction specific speed, using consistent units, is S_c = 3.0 (Dixon, 1978, and Table 1 (Brennan, 1994, and McNulty and Pearsall, 1979)). In traditional US evaluation, this critical value (S_c) equals about 8200. It should be recognized that this critical suction specific speed of 3.0 (8200 US) is often erroneously seen as the value at inception (S_i), while in fact it is more like the value at breakdown (S_b). So, operation below the critical value ($S_{available} < S_c$) does not necessarily imply the absence of cavitation or cavitation damage.

Table 1. Inception (S_i) and Breakdown (S_b) Suction Specific Speed for Some Typical Pumps.

Pump Type	N_S	Flow Q/Q_D	S_i	S_b	S_b/S_i
Process pump with volute and diffuser	0.31 (848 US)	0.24 1.20	0.25 (684 US) 0.8 (2188 US)	2.0 (5469 US) 2.5 (6837 US)	8.0 3.1
Double entry pump with volute	0.96 (2625 US)	1.00 1.20	0.6 (1641 US) 0.8 (2188 US)	2.1 (5743 US) 2.1 (5743 US)	3.5 2.6
Centrifugal pump with diffuser and volute	0.55 (1504 US)	0.75 1.00	0.6 (1641 US) 0.8 (2188 US)	2.41 (6590 US) 2.67 (7301 US)	4.0 3.3
Cooling water pump	1.35 (3692 US)	0.50 0.75 1.00	0.65 (1777 US) 0.6 (1641 US) 0.83 (2270 US)	3.4 (9298 US) 3.69 (10,091 US) 3.38 (9243 US)	5.2 6.2 4.1
Volute pump	1.00 (2735 US)	0.60 1.00 1.20	0.76 (1996 US) 0.83 (2270 US) 1.21 (3309 US)	1.74 (4758 US) 2.48 (6782 US) 2.47 (6754 US)	2.3 3.0 2.0

$NPSH_{40,000}$—In order to have cavitation erosion, three conditions must exist:

1. Cavitation bubbles must form in the fluid,
2. Cavitation bubbles must implode on or very near the vane surface, and
3. The cavitation intensity must exceed the cavitation resistance of the surface material.

While points one and two are relatively easy to ascertain visually, point three is rather hard to quantify. Therefore, many experimental and semiempirical studies have attempted to correlate between cavity shape and damage potential (*Cavity Length Damage Correlation*, below). Additionally, several others have applied a somewhat informed approach to predict NPSH requirements. For instance, a time honored method is the one proposed by Vlaming (1981). His NPSH required for 40,000 hour impeller life at the shockless entry point is given as:

$$\mathrm{NSPH}_{SE,40} = (k_1 C_{m1}^2 + k_2 W_1^2) / 2g \quad (2)$$

where:

k_1 = constant = 1.2
C_{m1} = upstream meridional velocity
W_1 = upstream relative velocity
k_2 = $0.28 + (U_e\ [\mathrm{m/s}]/122)^4$
= $0.28 + (U_e\ [\mathrm{ft/s}]/400)^4$

with:

U_e = peripheral velocity at impeller eye

This relation reflects a fundamental correlation, with coefficients (k_1, k_2) based on empirical data. It is believed that for reasonably good designs, adherence to NPSH values as calculated in Equation (2) would ensure an impeller life of 40,000 hours against cavitation damage.

Cavity Length Damage Correlation

A method to predict cavitation erosion that has received wide attention over the last decade is the one developed by Gülich (1986, 1988, 1989a, 1989b). It is based on the bubble or cavity length L_{cav} and can be stated as (Cooper, et al., 1991):

$$E = C \left(\frac{L_{cav}}{L_{cav,10}}\right)^n (\iota_A - \phi^2)^3\ U_e^6 \rho^3 A\ (8T_s^2)^{-1} \quad (3)$$

where:

E = erosion rate [mm/h]
C = 7.92×10^{-6} mm h^{-1} Pa^{-1} for blade suction side
= 3.96×10^{-4} mm h^{-1} Pa^{-1} for blade pressure side
L_{cav} = bubble or cavity length
$L_{cav,10}$ = reference bubble length (10 mm or 0.3937 in)
n = 2.83 for blade suction side
= 2.6 for blade pressure side
ι_A = $2g\ NPSH_A/U_e^2$
ϕ = inlet flow coefficient = C_{m1}/U_e
ρ = fluid density [kg/m^3]
A = properties factor (1 for cold water, 0.705 for 175°C/347°F boiler feedwater)
T_S = tensile strength of impeller material [Pa]

This equation is applicable to ferritic steel impellers and feedwater or drinking water services. In case one is dealing with austenitic steel impellers and/or saline applications, then the equation given needs to be corrected with a material factor and/or corrosion factor. Likewise, one has to take full account of the speed of sound in the fluid, the gas content, and the saturated vapor density if nonreference-value applications are considered; see Gülich (1986, 1988).

The empirical correlation presented in Equation (3) enables assessment of cavitation erosion impact and lifetime expectancy. Typically, Gülich's method states that a depth penetration of 75 percent of the blade thickness, t, constitutes the end of the useful life of the impeller in question; that is, after 0.75 t/E hours.

Although the damage rate, as indicated above, is affected by a number of factors, it can be argued that a usable correlation can be deducted with cavity length as the primary independent variable; i.e.,:

$$E \propto L_{cav}^n \quad (4)$$

This particular relation provides a very reasonable basis to project a change in impeller life when the impeller (vane) geometry is modified, while all other factors remain (practically) unchanged. Typically, the problem discussed here has been tackled using this simplified correlation of Gülich's formula. Another successful example of this approach can be found in Ferman, et al. (1997).

THE FIELD PROBLEM

Installed in the secondary loop of a 1050-MW (1,400,000 hp) PWR nuclear power plant are three 50 percent (i.e., two running and one backup) fixed speed feedwater pump trains that feed the system's steam generator (Figure 2). Each pump train comprises a booster pump and a feedwater pump. These feedwater pumps produced too much head due to a change in plant load conditions. They also showed premature short term impeller wear caused by cavitation erosion on the suction side surface of the vanes at impeller inlet, although the NPSH available was far over the $NPSH_{3\%}$, and also well above Vlaming's (1981) NPSH required for 40,000 hour impeller life. In fact, impellers were replaced/repaired after less than one year of operation. Therefore, to tackle this problem of short term cavitation damage, a cavitation study was included in the rate program that was contracted for meeting new duty points to eliminate high pressure throttling over the discharge valves.

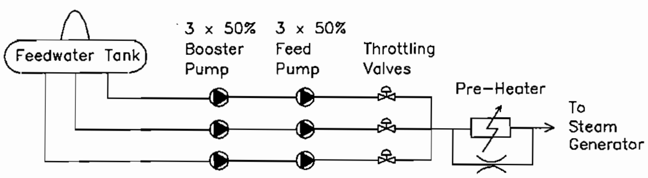

Figure 2. Feedwater System Layout.

Originally, the feedpumps were rated at 110 percent block load to develop a total head of 760 m (2493 ft) at 3553 m^3/h (15,644 gpm)—feedwater temperature, 148°C/298°F. Because this condition was no longer considered as a plant load condition, it was decided to refurbish the design of the feedpumps to operate optimally at 100 percent block load, producing minimally 736 m head (2415 ft) at 3230 m^3/h (14,222 gpm). As such, the feedpumps would be down rated to 7.6 MW (10,200 hp), yielding a more than 6 percent savings in driving power (originally, the feedpumps produced 785 m/2575 ft head at 100 percent block load). The original head performance characteristic of the feedpumps, including the original and new design points, is shown in Figure 3. Further particulars are (at 110 percent):

- Impeller speed - 5300 rpm,
- Impeller eye peripheral velocity - 75.6 m/s (248 ft/sec),
- Specific speed (total flow, double entry impeller) - 0.687 (1879 US),
- Three percent suction specific speed - 4.5 (12,300 US),
- Suction specific speed available - 1.8 (4930 US),
- Specific head rise or head coefficient - $\Psi = gH/(\Omega R)^2 = 0.50$ [−], and
- Specific volume flowrate or discharge flow coefficient - $\Phi = Q/(\Omega R^3) = 0.17$ [−].

These values qualify the feedpumps as so called high energy machines.

Besides new default operation requirements, there was a further requirement to meet a transient condition, in which only one out of three pumps is feeding the steam generator for 30 seconds. This ended up in specifying a runout condition of 4390 m^3/h (19,329 gpm) with a minimum required head of 593 m (1946 ft).

Although an important part of the project was to reduce the feedpumps' total head, it was the requirement of providing improved cavitation characteristics that drove the project.

PROJECT DEVELOPMENT

Test Approach

At the preparatory stage of the project, the final solution was specified to last at least 60,000 hours of operation. Taking into account that the original impellers were replaced after one year of operation (that is, after about 8,000 hours), it readily followed from Equation (4)—taking n = 2.83—that the cavity length on the new design had to be a little less than half the cavity length on the original design. As such, the (suction side) cavitation erosion rate would diminish by a factor greater than $2^{2.83} \approx 7.1$; thus, giving the desired impeller lifetime increase.

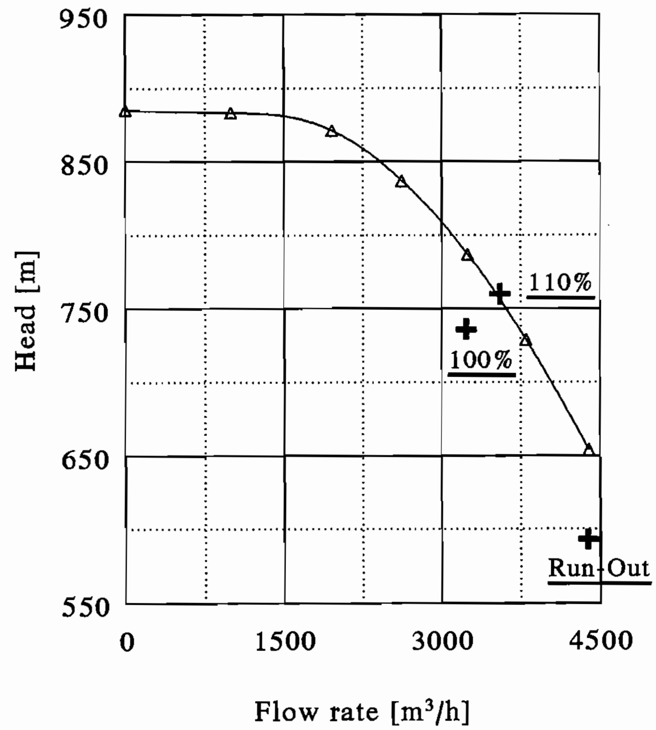

Figure 3. Original Feedpump Head Characteristic with Old (110%) and New (100%) Design Points, and New Runout Condition.

Since the researchers had to battle with a very short project time of nine months from start to finish (including commissioning in the field), it was decided to:

- Preselect/design four possible *hydraulic* solutions to the cavitation problem,
- Benchmark these possible solutions, and
- Select and fine-tune the most promising one.

Although several hard cavitation resistant materials exist to extend the life of the impeller in question, the use of a metallurgy other than the original impeller material (CA6NM) was only briefly considered, since it was believed that the root cause of the problem (i.e., development of cavitation vapor bubbles) could be thoroughly weakened or even eliminated by proper hydraulics. Moreover, the presence of vapor in a high energy pump is highly undesirable, and should therefore be avoided if possible.

Experimental Verification

Because of the lack of suited calculation techniques, the investigation required a flow visualization testing capability to determine the cavity length on the impeller blades. A precondition of the visualization was the geometric similarity of the test floor model to the field unit. To determine the model test conditions from field data, the following relationships were used.

- Volume flowrate:

$$Q_m = Q_f \, f^3 N_m / N_f \tag{5}$$

- Net positive suction head:

$$NPSH_m = NPSH_f\, f^2\, (N_m / N_f)^2 \tag{6}$$

in which:

f = scale factor [−]
N = rotational speed [rpm]

and where the subscripts denote model (m) and field unit (f).

Furthermore, two additional requirements had to be fulfilled in order to conduct successful flow visualization; namely, good visual access, and good imaging possibilities. Full visual access is almost a necessity for illumination and visualization, since in order to disclose the shape and form of the cavitation area, it is often necessary to vary the direction of the incident light. A small access limits this considerably.

Test Arrangement

For verifying both the hydraulic performance of the new designs, and carrying out the visual study of the impeller inlet flow, a full scale test pump (Figure 4) was constructed, with an acrylic viewing window for direct observation of the impeller blade cavitation. This test pump was installed in a test loop as shown in Figure 5.

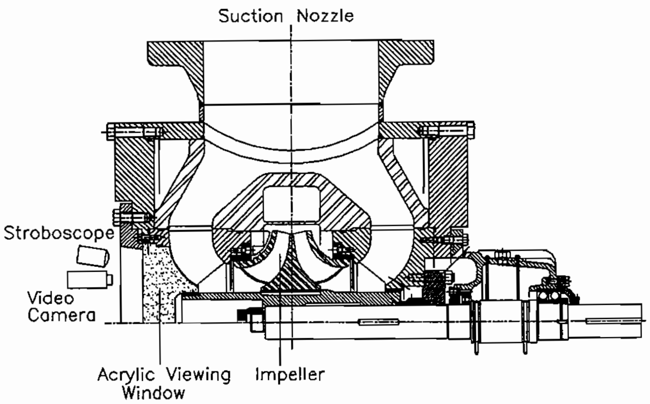

Figure 4. Flow Visualization Test Pump.

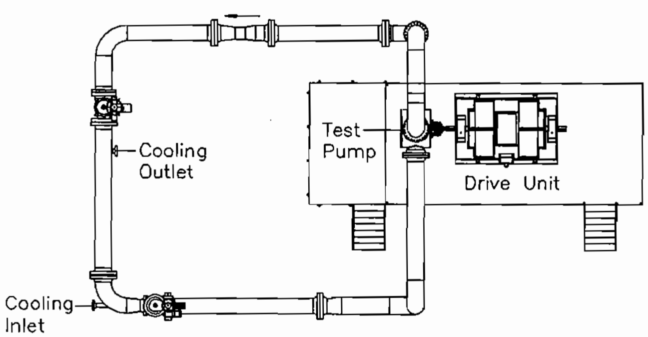

Figure 5. Flow Visualization Test Loop.

To enable proper visual access, the impeller was placed in an overhung position, whereas the field unit comprised a between bearing design. The pump casing of the rig was cast directly from drawings of the original equipment, but had a reduced wall thickness, since it only had to withstand approximately one third of the field pressures, because the rig was operated at reduced speed (i.e., 2990 versus 5300 rpm). Special attention was given to the design and shape of the window to assure that the rig was hydraulically identical to the field units. The window itself was composed of two thick layers of acrylic plate (glued together), and had to withstand (maximally) 10 bar/145 psi suction pressure. The test fluid was plain (cold) tap water.

To image the leading-edge cavities, a synchronized high frequency stroboscope was employed, while permanent records were obtained through conventional videographic and photographic registration, along with plain witnessing techniques using sketches. To quantify the length of the cavitation bubbles, the visible side or suction side of the impeller vane surfaces was furnished with marker stripes at increments of 10 mm (0.3937 in), parallel to the leading-edge.

Prior to the start of the experiments, the water in the test circuit was deaerated by running a vacuum pump at 76 mm Hg (3.0 in) for at least two hours. During the tests, the dissolved air level (i.e., oxygen contents) was monitored to ensure a less than 10 ppm test condition, in line with the field situation.

Cavitation Test Results

Investigated in the test rig were, at 100 percent, 85 percent, and 70 percent of the rated throughflow (Table 2), the inlet flows inside the eye of:

- A multiple repaired field impeller,
- An original casting (as delivered in the past as original design), and
- The four possible solutions to the cavitation problem (Table 3).

This included NPSH testing, Q-H performance testing, and observation and measurement of cavitation bubbles at the impeller inlet vane surfaces by means of stroboscopic imaging. For example, Figure 6 shows a typical cavitation bubble that was encountered at the leading-edges of the original design.

Table 2. Model to Field Unit Cavitation Visualization Test Conditions.

Load	Flow [m³/h]		NPSH available [m]	
	Model	Field Unit	Model	Field Unit
100%	1820	3230	70	220
85%	1547	2745	72.3	227
70%	1274	2260	76.4	240

Table 3. Impellers Investigated.

Field Impeller	Provided by customer
Base-Line Impeller	Original casting; identical as delivered in the past (5-vanes)
Option 1	Modified base-line impeller; thicker vanes (5), adjusted vane angle
Option 2A	Adapted from existing (20 × 20 × 18 A HDR) impeller (5-vanes, smaller eye)
Option 2B	New 5-vane design
Option 3	New 7-vane design

After intensive testing, including various fine-tuning actions, a final solution was determined, yielding not only the definition of the new impeller design, but also a modification of the suction box (i.e., repositioning of a splitter vane). This will be discussed below. Here, attention will be confined to the case of 100 percent flow. The tests at 85 and 70 percent throughflow were conducted to see whether there would be excessive cavitation compared with the situation of 100 percent flow. At the 85 and 70 percent part load conditions, it was found that the cavitation bubbles were roughly twice the size of the bubbles encountered at 100 percent flow. As such, running at part load may cause (comparatively) severe cavitation damage. The power plant, however, rarely runs at part load.

Impeller Selection—Table 3 lists the impellers that were tested in the model pump. The visual study of these impellers initially

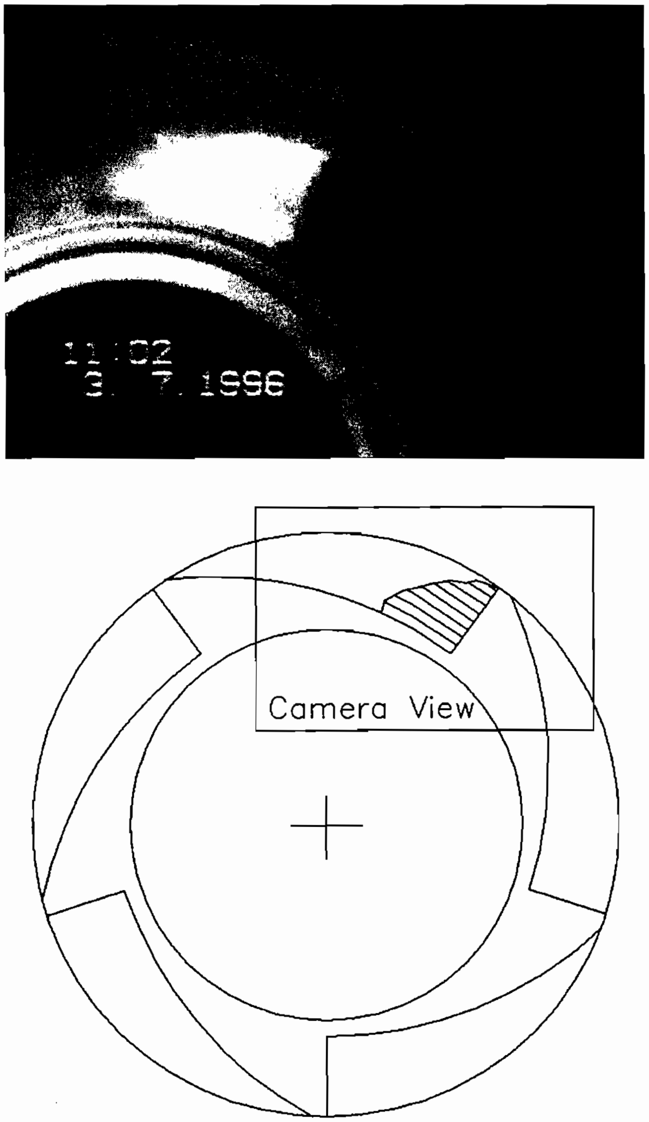

Figure 6. Typical Cavitation Bubble on Original Impeller Leading-Edge Surface at 100 Percent of the Rated Throughflow and Duty NPSH Available.

indicated that option 2A provided the best possibility to reduce the cavitation bubble length. However, this impeller showed a very unfavorable NPSH behavior, despite the favorable bubble behavior. To improve the NPSH, it was therefore decided to open the impeller eye (twice) by cutting back the vanes. This improved the NPSH behavior a little bit, but seriously worsened the bubble behavior. Consequently, option 2A was no longer considered as a solution. Instead, the attention was focused on option 2B, which showed to be the second best in potential to solve the cavitation problem. The other options showed a cavity behavior similar to the baseline impeller and the field impeller, and were therefore ruled out for further investigation.

Although option 2B already had a very good NPSH behavior, its cavitation bubble behavior still had to be improved. To accomplish this, the vane leading-edges were given an elliptical shape at the suction side, since it was argued that the flow stalled on this particular surface. This assumption was corroborated by the tests that followed. After several fine-tuning actions, the vanes finally got the elliptical shape shown in Figure 7. This way, the cavitation occurring at the vane tips was minimized to a level that fulfilled the lifetime requirement (below).

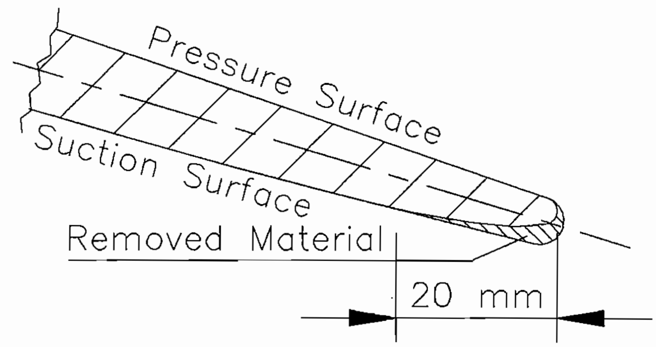

Figure 7. Elliptically Shaped Leading-Edge Contour (Vane Suction Surface).

Having experimentally determined a satisfying solution on the basis of option 2B, four look-a-like impellers were cast for the field units (one for each feedpump, plus a spare). In order to get the correct vane shape on these four new field impellers, a set of vane templates were manufactured from the final rework option. Thereupon, the four new impellers were commissioned in the test rig to verify the improved cavitation behavior.

Suction Box Modification—Testing the various impellers showed that there were comparatively large cavities on the blades in the top region of the eye area and small cavities in the bottom region. In addition to the adjustment of the vane leading-edges, it was therefore decided to investigate the effect of modifying the suction box of the pump; that is, examine the possibility of making the inlet flow more uniform over the impeller eye area, so that cavities would become more equally sized across the eye area, and, hence, overall impact would diminish. Having tested several configurations, it was finally found that the best result was obtained if the second splitter was repositioned by a 45 degree angle as indicated in Figure 8. This way, occurrence of large bubbles was reduced to a relatively small (30 percent) portion of the circumferential area at the top of the suction bay. Without this modification, the region of large cavities was considerably larger (30 versus 50 percent).

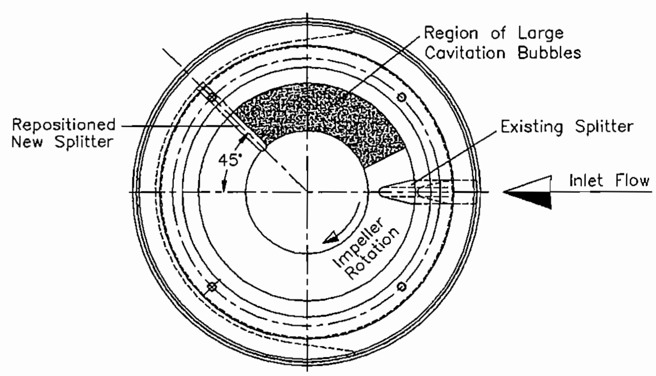

Figure 8. Region of Comparatively Large Cavitation Bubbles in Suction Bay/Impeller Eye Area (Modified Twin Splitter Configuration).

Impeller Lifetime Improvement—The visual testing of the final design showed that it was not completely free from cavitation bubbles at duty NPSH available (70 m/230 ft, $\iota_A = 0.24$), but cavitation bubble length had decreased considerably. A graphical survey of the cavity lengths that were encountered on the original and new field impeller designs is given in Figures 9 and 10. They show the minimum and maximum values of the cavity lengths, occurring at bottom and top region respectively.

With reference to Figures 9 and 10, it is seen that compared with the original design, the cavity length at duty NPSH (100 percent

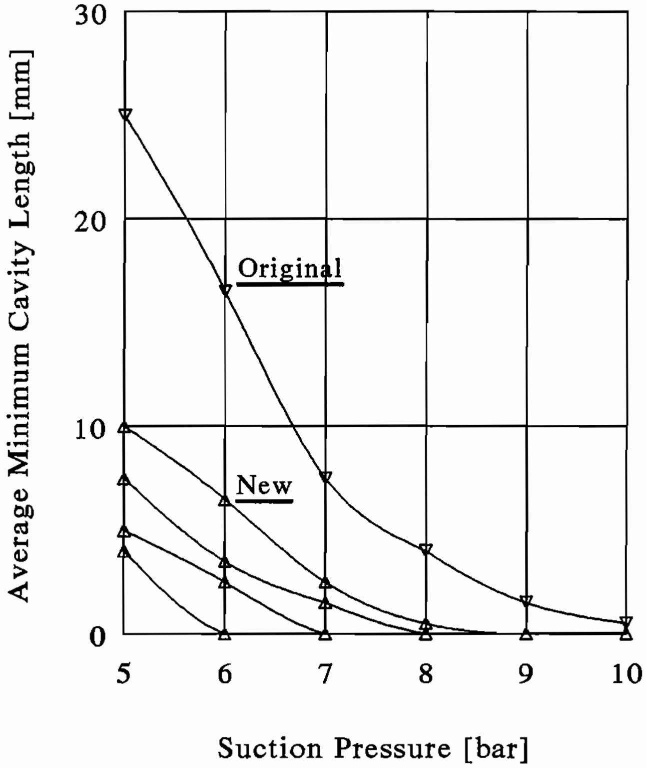

Figure 9. Average Minimum Cavity Length Measured at 100 Percent Flow.

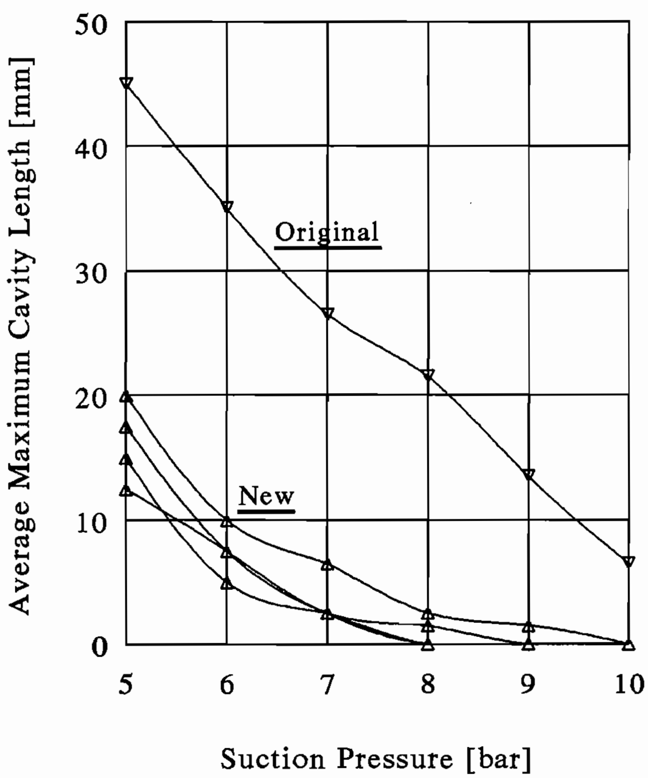

Figure 10. Average Maximum Cavity Length Measured at 100 Percent Flow.

flow) has been reduced by a factor smaller than 7/15, and, hence, impeller lifetime can be expected to be improved by (at least) a factor eight, employing the theory of Gülich. It should be recognized that, for this calculation, a conservative value of 15 mm (0.59 in) for the original cavity length has been used, while the average length equals around 26 mm (1.02 in). Considering further that with the modification of the splitter vane there is a reduced span of time of comparatively severe bubble impact, one may actually expect a lifetime well over 100,000 hours (for operation at the rated condition for all time, i.e., at base load).

Field Verification—Subsequent to the shop testing, the impellers have been commissioned in the field units (November 1996), where they are accumulating operational hours. After nearly one year of operation (October 1997) the first unit has been opened and the impeller has been inspected for possible cavitation damage.

It was found that one vane had some slight cavitation erosion damage at the visible (suction) side. The other (nine) vanes showed no cavitation erosion damage. (Recall that the original design showed considerable cavitation erosion damage on all vanes after one year of operation.) The damage that was found on the eroded vane had a depth less than 0.5 mm (0.02 in). A small leading-edge surface irregularity on the vane in question probably caused some cavitation attack resulting in the erosion damage found. The other unit(s) was also scheduled to be checked for possible cavitation damage in October/November 1997, but results were not yet available while preparing this paper.

Performance Testing

Figures 11, 12, and 13 show the results of the shop performance testing. In Figure 11, the head characteristics of the four new field impellers are shown, including the design and runout points, along with the original feedpump characteristic. Figure 12 shows the $NPSH_{3\%}$ requirement, and Figure 13 gives the new efficiency relative to the original feed-pump efficiency.

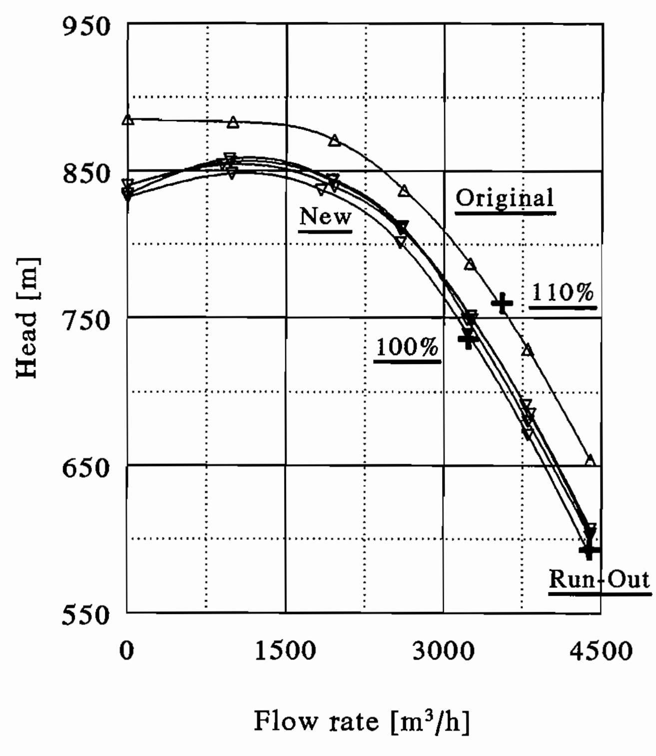

Figure 11. Test Floor Head Characteristic of Original Design and New Design (4×).

From the NPSH characteristic, it is seen that the NPSH requirement for the new design is much better, that is to say lower, than for the original design; especially at part load. Furthermore, it is seen that the efficiency has improved by one to two percent at design point.

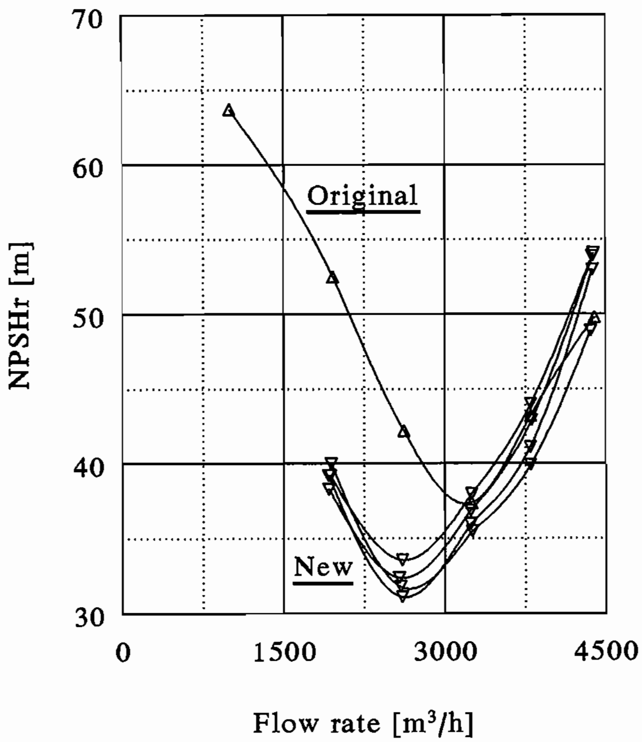

Figure 12. Test Floor NPSH Characteristic of Original Design and New Design (4×).

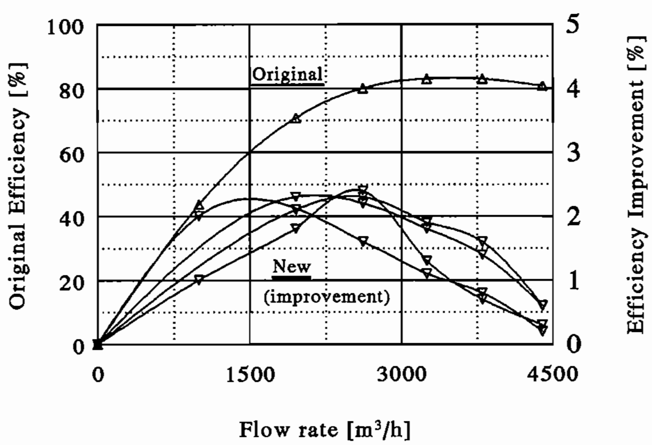

Figure 13. Test Floor Efficiency of the New Design (4×) Relative to the Original Design.

Field Verification—In order to verify the hydraulic performance in the field, a field performance test was conducted. To that end, the plant startup unit was temporarily equipped with a data acquisition system to collect performance data at four different flowrates during plant startup. These flowrates were between 40 and 70 percent of the (100 percent) duty flow. At duty flow, a fifth point was measured after the plant was in full operation (i.e., two units running). The volume flowrate through the field unit was measured by a venturi, while the pump head was measured using pressure taps in the suction and discharge of the pump. The power input was obtained by a Watt meter.

Figure 14 shows the results of the field performance test. It shows a slightly higher head (roughly 1.3 percent) at the duty point. At the lower flowrates, the curve rises to 5.5 percent above the shop test characteristic. This is the result of a higher efficiency due to higher speed, higher fluid temperature, and a better casting quality of the discharge volute (waterways) in the field. (The shop test arrangement was a prototype casting having a relatively rough surface finish.)

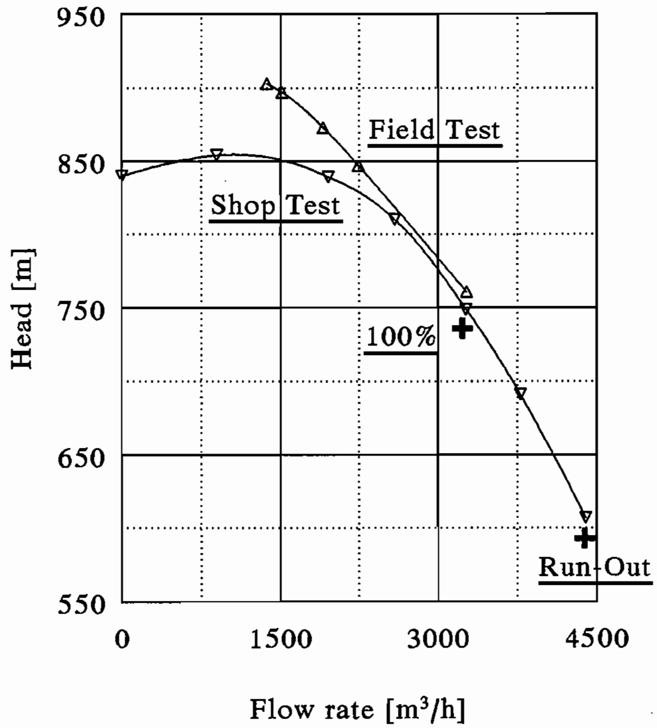

Figure 14. New Field Performance Head Characteristic Versus Shop Performance Head Characteristic.

Vibrations—Lastly, without discussing it in detail, it is mentioned that after the refurbishment, the overall vibration level of the feedpump went down significantly, and that RMS pressure pulsation levels were considerably reduced in the main suction piping (i.e., 60 percent reduction at duty flow). Furthermore, the maximum pressure pulsation levels were far below the normal design practice criterion of six percent of the nominal pressure; i.e., 0.44 and 0.22 percent at suction and discharge at duty flow.

CONCLUSIONS

In a joint effort of the user and the pump manufacturer, the design of an 8½-MW (11,400 hp) high energy double entry feedpump impeller and its inlet casing have been investigated in a full scale model test arrangement. It has been established that the newly developed impeller would suffer significantly less from cavitation and cavitation erosion than the original design, which was replaced/repaired because of cavitation attack after one year of pump operation.

The new impeller was designed to meet a cavity length criterion of "less than half the cavity length on the original design," which is to give a cavitation resistance lifetime improvement of at least seven years or more. This was accomplished by correctly matching the vane leading-edge angles to the incoming flow, and, to a lesser extend, through repositioning of a suction box inlet splitter; so that the inlet flow would become more uniformly distributed across the eye area, and, hence, less flow distortions (cf bubble activities) would emerge.

In the test arrangement, the impellers were compared by means of visualization of the cavitating flow in the inlet regions. This comparative approach corroborated the improvement in cavity length that was to be expected from the design changes. Resorting to another kind of special hard impeller material with better resistance to cavitation erosion was not necessary, because cavities were properly reduced by the hydraulic design changes.

Apart from the cavitation problem, specific performance objectives to eliminate high pressure throttling over the discharge valve were to be achieved with the new design as well. The head rise required was properly met, and the NPSH requirement was established to be much better than with the original design. Furthermore, the efficiency had improved by one to two percent.

The new impellers have been commissioned in the (three) field units, including the modification of the suction box splitter, and will be inspected for potential cavitation damage after one year of operation. In the field, the refurbished plant pumps were tested at different flowrates to verify satisfactory system performance.

NOMENCLATURE

A	=	constant = 0.705 (for boiler feedwater)
C	=	constant = 7.92×10^{-6} mm h^{-1} Pa^{-1} for blade suction side, 3.96×10^{-4} mm h^{-1} Pa^{-1} for blade pressure side
C_{m1}	=	meridional inlet velocity
E	=	erosion rate [mm/h]
f	=	the scale factor
g	=	acceleration due to gravity
H	=	pump head
k_1	=	constant = 1.2
k_2	=	$0.28 + (U_e\ [m/s]/122)^4$
	=	$0.28 + (U_e\ [ft/s]/400)^4$
L_{cav}	=	bubble or cavity length
$L_{cav,0}$	=	reference bubble length (10 mm)
n	=	constant = 2.83 for blade suction side, 2.6 for blade pressure side
N	=	rotational speed [rpm]
N_s	=	specific speed = $\Omega Q^{1/2}/(g\ H)^{3/4}$
NPSH	=	net positive suction head
NPSE	=	net positive suction energy = g NPSH
Q	=	volume flowrate
Q_D	=	design volume flowrate
R	=	impeller (discharge) radius
S	=	suction specific speed = $\Omega Q^{1/2}/(NPSE)^{3/4}$
T_S	=	tensile strength [Pa]
U_e	=	peripheral velocity at impeller eye [m/s]
W_1	=	relative inlet velocity
ρ	=	fluid density (kg/m^3)
ι_A	=	$2g\ NPSH/U_e^2$
Φ	=	specific flowrate = $Q/(\Omega R^3)$
φ	=	inlet flow coefficient = C_{m1}/U_e
Ψ	=	specific head rise = $gH/(\Omega R)^2$
Ω	=	angular speed

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