CAVITATION STUDY IN A HIGH PRESSURE WATER INJECTION PUMP

by

Erik Torbergsen

Manager, Development Department

Sigve Gjerstad

Project Engineer, Development Department

Andreas Hosøy

Project Engineer, Development Department

Frank Mohn Flatøy AS, Oil & Gas Division

Frekhaug, Norway

and

Brian Aasland

Staff Engineer Rotating Equipment, Concept Development Department

Statoil ASA

Stavanger, Norway

Erik Torbergsen is Manager of the Development Department at Frank Mohn Flatøy AS, in Frekhaug, Norway, which produces high energy pumping systems for the oil and gas industry. The department is involved in experimental studies as well as numerical analysis of centrifugal pumps.

Dr. Torbergsen received his M.Sc. degree (Mechanical Engineering, 1990) at The Norwegian University of Science and Technology, and obtained a Doctorate in Engineering degree from the same university (1998). He joined the Frank Mohn company in 1992, and had a joint-funded research position at The Norwegian University of Science and Technology from 1994 to 1998, studying flow instability and rotordynamics of centrifugal pumps.

Sigve Gjerstad is a Project Engineer and works with general pump design and fluid mechanics in the Development Department at Frank Mohn Flatøy AS, in Frekhaug, Norway, which produces high energy pumping systems for the oil and gas industry.

Mr. Gjerstad received his B.Eng. degree (Mechanical Engineering, 1999) from the University of Newcastle, and his M.Phil. degree (Mechanical Engineering, 2000) at Cambridge University. He joined the Frank Mohn company in 2001.

Andreas Hosøy is a Project Engineer and works with general pump design and fluid mechanics in the Development Department at Frank Mohn Flatøy AS, in Frekhaug, Norway, which produces high energy pumping systems for the oil and gas industry.

Mr. Hosøy received his B.Eng. degree (Mechanical Engineering, 1997) from Bergen College of Engineering, and his M.Sc. degree (Mechanical Engineering, 1999) at The Norwegian University of Science and Technology. He joined the Frank Mohn company in 2001.

ABSTRACT

High energy pumps often operate at high speeds to achieve compact designs with few stages. These pumps need special attention on cavitation performance because the cavitation erosion rate is related to impeller speed. This paper presents a case study of cavitation problems observed in a high pressure water injection pump. To solve the problems, two different impellers were analyzed with respect to "as-built" deviations in inlet angles and leading edge shape compared to design. Experimental and numerical methods were applied to study the "as-built" and "design" versions. It is shown that "as-built" local inlet angles and shapes controlled the cavitation performance. The necessary suction pressure for cavitation-free operation is discussed by the use of a simulated (computational fluid dynamics) inception line. Visualization on a full scale test pump with different suction pressures was used to validate the numerical cavitation performance.

INTRODUCTION

Cavitation behavior of centrifugal pumps has been a major concern for all pump manufacturers. What has been known for years is that when having a typical 3 percent head drop in a centrifugal pump, which is the industrial definition of the necessary suction pressure for a pump impeller, significant cavitation will be present on the blade. The consequence of this can be cavitation erosion on the blades, vibration, and noise problems.

An early evaluation of the minimum net positive suction head (NPSH) required by a centrifugal pump for smooth and trouble-free operation was carried out by Vlaming (1981). The method, based on both an empirical and theoretical approach, is often adopted as a basis for a 40,000 hour lifetime estimation against cavitation damage. As discussed by Vlaming and others, the blade velocity is a major factor when determining the lifetime of the blade and erosion problems. Simoneau, et al. (1989), have shown that both incidence angle and velocity heavily influence the erosion rate and, on a 2-D National Advisory Committee for Aeronautics (NACA) profile, the erosion rate can vary from the eighth to eleventh power of speed. Guelich and Pace (1986) have discussed the erosion rate and conclude that for centrifugal pump impellers, the sixth power of speed is typical, based on several published experiments. This means that for high energy pumps, often operating at high speeds, the cavitation inception and bubble length is of particular interest when designing and evaluating a pump design. Several studies also discuss the erosion rate related to the cavity length and it is shown to vary approximately with the second to fourth power of the cavity length.

Altogether, this results in the need for knowledge about cavitation inception and cavity length for different capacities when designing and operating high energy pumps. A criterion often applied is zero tolerance with respect to cavitation inception in the region of continuous operation. A practical approach used, in many cases by pump manufacturers, is "in field" trimming of the blade leading edge to solve cavitation problems. The objective of this paper is to present a case study with cavitation erosion problems in a high pressure water injection pump for oil well pressure boosting, operating at high speed. The suction pressure, with respect to blade cavitation inception, is presented by the use of numerical methods. Full scale visual studies on a high pressure water injection pump have also been used to validate the numerical simulations.

The four water injection pumps described in this case are operating at the Statoil Norne oil field.

THEORETICAL BACKGROUND

During the last few years, several articles and studies have described cavitation in centrifugal impellers with the use of numerical flow simulations. Dupont (2001) presents numerical simulations with the use of a simplified cavity length model and discusses the typical off-design shape of suction head at cavitation inception as a function of flow. For a typical radial impeller, the incipient visual cavitation on the blade suction side (visible side) reaches a peak at part capacity, often 30 to 70 percent of the best efficiency point (BEP), i.e., significant suction head is often required to suppress the cavitation inception at part load. Several authors relate this peak in inception to a capacity slightly higher than the critical capacity at which the swirling backflow starts in the suction flow. An excellent paper by Schiavallo (1987) discusses this phenomenon. Determination of the off-design cavitation peak is also very important when defining minimum flow operation of the pump, and, for continuously running high energy pumps, this is an essential requirement. The minimum NPSH for inception often occurs at slightly lower capacity than the BEP. This is shown experimentally by Kumaraswamy and Radha Krishna (1986). For higher flows, the $NPSH_{inception}$ increases sharply, and the incipient cavitation takes place on the pressure side of the blade leading edge.

Several field studies of cavitation in high energy pumps have been presented. Ferman, et al. (1997), present a case study with a boiler feed pump where one aim was to reduce the erosion by an improved impeller design that had smaller cavity length. Full scale flow visualization techniques were applied.

The use of numerical flow simulations (computational fluid dynamics [CFD]) has become common and important in the design of centrifugal pumps. Several publications and studies show the use of numerical predictions of complex transient flow phenomena such as rotating stall and impeller recirculation, as well as the steady-state characteristics. However, not too many papers present cavitation studies in centrifugal pumps by the use of two phase cavitation models. The impression is that there is uncertainty in using models presently available in commercial CFD-codes. There are still few engineers using cavitation models in the design of centrifugal impellers. Although CFD analyses can be inaccurate for determining NPSH outside BEP, it provides valuable information, especially when results are compared with visual information.

Hirschi, et al. (1997), present an iterative method for cavitation simulations using commercial CFD-codes, where the 3-D flow calculation is updated by including an interface shape varying with a constant pressure equal to the vapor pressure along it. The Rayleigh-Plesset equation for bubble development is applied for the initial cavity shape. Although not too much experimental data were used, the method gave fairly good results with respect to cavity length along the blade inlet and also for cavitation inception. Cavitation inception was analyzed using both minimum pressure coefficient and also a 2 percent cavitation zone length relative to the impeller outlet radius. A simplified method was developed in order to reduce the number of iterations and the initial cavity shape without updating the main flow except for longer cavities. The method is described as fast, and both incipient and cavity lengths showed values close to experimental data.

For this study it was decided to use a commercial CFD-code and analyze the cavitation inception using incompressible, single phase models and pressure profiles. A detailed analysis of the cavitation zones was considered beyond the scope of this work. The code applied in this work is widely used in turbomachinery flow analysis, and several articles regarding pump performance have been presented.

CASE DESCRIPTION

The use of high pressure injection pumps to boost pressure in oil wells is increasing. The main reason is the need to increase oil recovery from the reservoir. Some fields even need water injection to produce, i.e., when water injection is out of operation so is the oil production. The importance of water injection is therefore very high for an increasing number of installations in the North Sea. Typically, these pumps are placed on offshore platforms with limited space available. The need for ratings of several megawatts often leads to the use of high-power density pumps. One way to achieve this is to operate the pumps at high speed, typically 12 to 14,000 rpm, leading to a stiff, overhung, end suction design with few stages. These pumps run continuously at a discharge pressure of typically 200 to 300 bars (2900 to 4350 psi) and capacities of 100 to 700 m^3/h (440 to 3080 gpm). Based on the above discussion, it is quite clear that detailed knowledge about the cavitation performance is necessary. The water injection pump design consists of three main stages, where one stage is a low speed booster stage operating at 2500 to 4000 rpm, feeding the water into two high speed/high pressure stages with sufficient suction pressure. The suction pressure delivered from the booster stage discharge is typically 15 to 45 bars (218 to 653 psi), dependent on the impeller design used in the high pressure stages. The pump is integrated on a gearbox, and a typical sketch is shown in Figure 1, where the low and high speed shaft is driven by an input shaft coupled to an external electric motor (typically two-pole, 60 Hz). The critical with respect to cavitation is the first high pressure stage on the high speed shaft.

The background for this study was a case with four water injection pumps at an oil field, installed on a floating production vessel (FPSO) offshore the northwest coast of Norway. After operating the pumps for one to two years, cavitation erosion was discovered on the first high pressure stage impeller and several repairs were made. In Figure 2, an example of the erosion after 4800 hours of operation is shown. The erosion pattern is typical for a centrifugal impeller cavitating at flows below best efficiency point but above onset of suction recirculation. A visual study during operation in the field was carried out using a video camera and a strobe light. The impeller inlet showed clear evidence of cavitation in the whole allowable operating region. The cavitation was present on the blade suction side. This result was surprising because the design was optimized with significant suction pressure for the first high pressure stage impeller and sufficient tolerances for the 25 Cr duplex steel impeller and to 40,000 hours' lifetime estimation. The situation was evaluated between the pump manufacturer and the end user and found to be unacceptable for long-term operation. The impeller blade design was checked and a detailed dimensional control of the actual impeller casting was carried out. The blades showed severe deviation in the inlet angle: the "as-built" local inlet angle of the cast impeller was 5 to 8 degrees higher than the design angle. The deviation was discovered on several impellers based on the same casting model, indicating a systematic deviation. An example of this is shown in Figure 3,

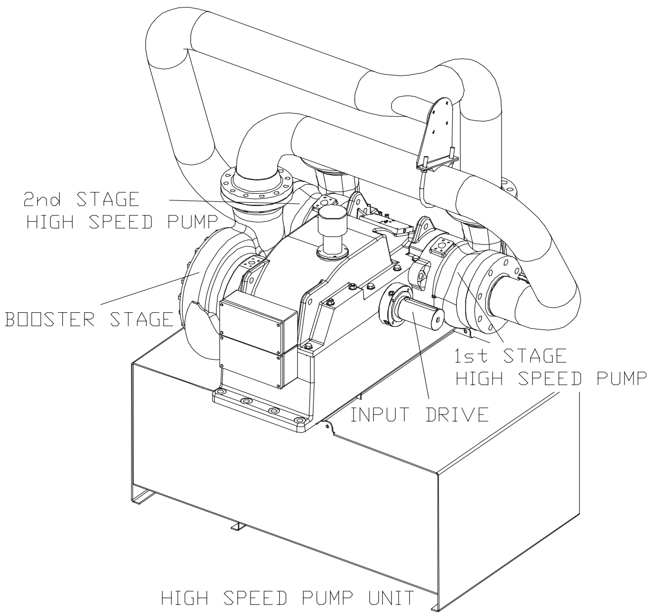

Figure 1. 3-D Drawing of a High Pressure Water Injection Pump and Integral Gearbox, Typically Rated Between 2 and 7 MW (2680 and 9380 HP) and 12,000 to 14,000 RPM.

where the inlet blade profile from the "as-built" impeller is shown together with the "design" profile. A significantly higher inlet angle can be seen on the "as-built" profile, leading to a much higher incidence angle and a blade that will be more sensitive to cavitation in the allowable operational region. The reason for the deviation was the manufacturing technique originally applied to the casting pattern by the foundry. New manufacturing techniques are now adopted for the pattern.

Figure 2. Impeller Number 1 (High Pressure Stage) with Cavitation Erosion on Blade Inlet Suction Side. Operational Time: 4800 Hours.

A detailed analysis of the cavitation performance was initiated. It was necessary to manufacture a high precision impeller (milling of blades and welding of shroud) to minimize blade geometry deviations compared with analysis. In addition, a different impeller design with slightly higher specific speed was also analyzed. This impeller satisfied the rated point of the pump operating in the field and could therefore be applied as an alternative solution for the pump. A casting version of this second impeller was also dimensionally checked in the same way as impeller number 1 and

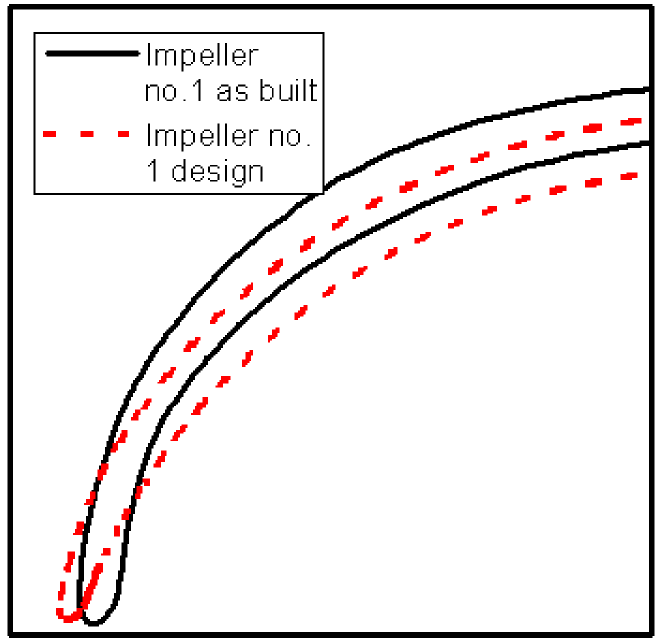

Figure 3. Impeller Number 1 "As-Built" and "Design" Inlet Profiles.

deviations were also found. They were systematic but much less, compared to impeller number 1 (as shown in Figure 4). The inlet profile even shows a lower local inlet angle along the suction side in the "as-built" case. The two different impellers analyzed now included two versions each, identified as:

- Impeller number 1
  - "design"
  - "as-built"
- Impeller number 2
  - "design"
  - "as-built"

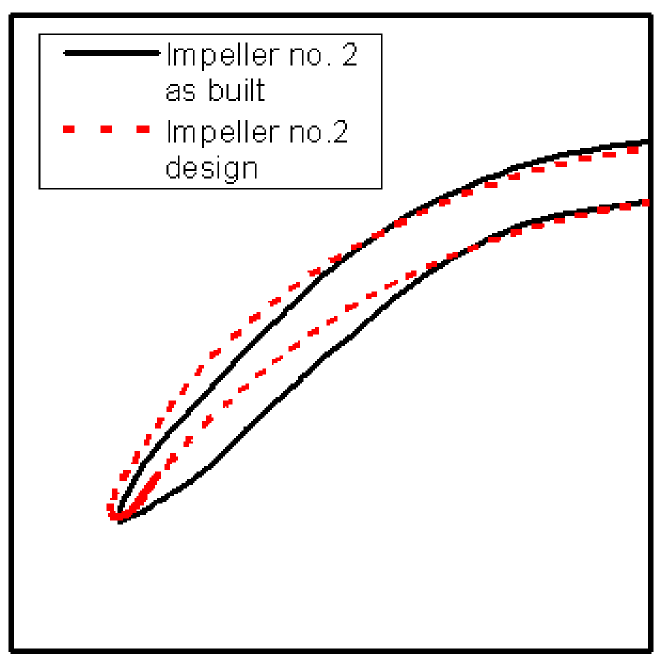

Figure 4. Impeller Number 2 "As-Built" and "Design" Inlet Profiles.

"Design" refers to very small geometrical deviations compared to the design profile, i.e., the machined high precision impeller is equal to impeller number 1 "design." The main design geometrical parameters for the two impellers are shown in Table 1.

Table 1. Main Physical Parameters for the Two Impellers Analyzed in the Study.

	Specific speed (design), Nq	Impeller diameter	Number of vanes
Impeller no. 1:	25 *(1290 US)*	224 mm *(8.8 inches)*	8
Impeller no. 2	29 *(1496 US)*	224 mm *(8.8 inches)*	8

The design version of impeller number 2 was not manufactured, but only analyzed numerically, as will be discussed below. The analysis program for the above impeller versions was included in a full scale test. The test water injection pump had the main parameters shown in Table 2.

Table 2. Full Scale Test Water Injection Pump Main Parameters.

Rated capacity	Differential head rated	Booster stage impeller speed	High pressure stage impeller speed	High pressure stage suction pressure
667 m^3/h *(2935 gpm)*	2293 mlc *(7523 feet)*	3250 rpm	13050 rpm	36-41 bara *(522-595 psi a)*

EXPERIMENTAL SETUP IN FULL SCALE PUMP

The full scale pump used in the visual study was investigated during inhouse factory testing, including standard performance instrumentation. A visualization unit for the first high pressure impeller stage was applied. The aim was to study the inlet of the first high pressure stage for different operational points and especially if cavitation were present at nominal conditions for both "design" and "as-built" versions. Due to the fixed ratio gearbox with integral booster pump, adjustment of the suction pressure to the first high pressure stage was not directly achievable. However, different restriction orifices between the booster and first high pressure stage were used to reduce the suction pressure.

The visualization unit is shown in Figure 5. A high speed digital camera was used together with a specially designed flashlight to ensure enough light. The high impeller speed and need for a detailed view of the blade made it necessary to use a camera with a shutter speed of 10 µs in conjunction with the high intensity flashlight (1000 J). An adaptor with a glass window was fitted inside the inlet pipe to enable a satisfactory view of the impeller inlet edges. The adaptor protected the borescope from the suction pressure. The borescope had a connection for the camera and light source. An external trigger was used to trigger both the camera and the flashlight simultaneously.

A compromise had to be made to minimize the adapter influence on the inlet flow and to ensure enough light to give clear and usable pictures for cavitation studies (distance to impeller inlet indicated in Figure 5). The test unit gave digital pictures with satisfactory quality for approximate evaluation of cavitation cloud sizes. It was assumed that a cavitation length down to approximately 1 percent of the impeller outlet radius, that is, a typical length of 1 mm (.04 inch), could be studied. However, the print quality of the digital pictures was limited. During the tests, several pictures were studied at each operating point to ensure repeatable results.

NUMERICAL ANALYSIS

The numerical modeling and simulations were carried out using a Reynolds averaged Navier-Stokes finite volume code, including a standard k-e model and logarithmic wall functions for the flow in low Reynolds number regions near a solid wall. A steady-state approach with one blade and periodic boundary conditions using uniform axial velocity field at the inlet boundary, at a sufficient

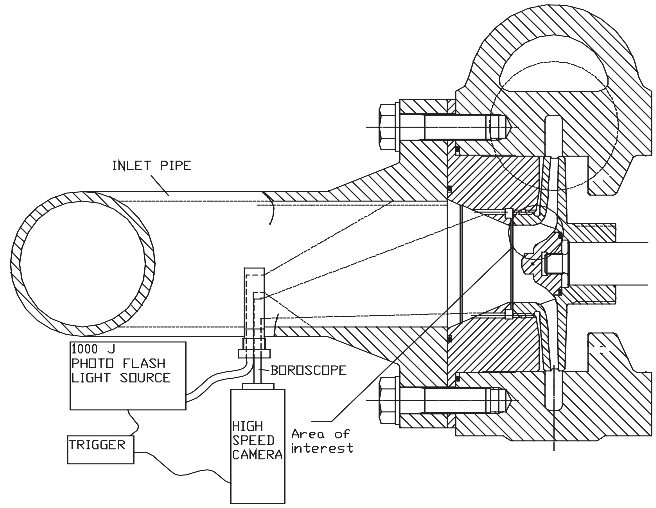

Figure 5. Sketch Showing the Visualization Unit Adjacent to the Inlet of the First High Pressure Stage Impeller.

distance from the blade leading edge, including the suction pipe. At the outlet boundary located at a 20 percent radial distance relative to the impeller radius, a constant pressure boundary was applied. Since a single phase incompressible simulation was used, only the relative pressure was of interest when studying the pressure profiles on the blade leading edge. The cavitation inception in the simulation was defined as the static inlet pressure (suction pipe pressure corresponding to suction pressure measurements in the pump) giving small pressure zones on the leading edge below vapor pressure. The zone lengths were defined as cavitation lengths (streamline direction) less than 1 percent of the impeller outlet radius, and the corresponding inlet pressure can be viewed as a simulated $NPSH_{inception}$.

Several grid sensitivity checks were carried out to minimize grid size influence. An example of one impeller model with inlet and outlet is shown in Figures 6 and 7, including a view of the inlet blade grid.

Both impellers number 1 and number 2 presented earlier, including both "design" and "as-built" versions, were simulated to study inception for several operational points.

NUMERICAL AND EXPERIMENTAL RESULTS

In Figure 8, the NPSH analysis of impeller number 1, "design" version, is shown with the flow on the horizontal axis and the inception pressure on the vertical axis. The solid line shows the results from the numerical simulation for several capacities, using the inception criteria discussed above, and defines the simulated $NPSH_{inception}$. The line is interpolated between several simulation results. Below a flow coefficient of approximately 0.18 the line corresponds to inception on the blade suction side, while above 0.18 the inception takes place on the blade pressure side. The shape of the line is in qualitative agreement with typical inception curves reported by others for centrifugal impellers. The low flow peak in $NPSH_{inception}$ is 60 to 70 percent of BEP (located at approximately 0.19). It can further be seen that at a typical suction pressure of 40 bara (580 psia), a cavitation free operation for the number 1 "design" case is expected to be from approximately $\varphi = 0.13$ to 0.20. Given the above suction pressure, this should be the region for preferred operation (simulated) when specifying a zero tolerance to cavitation. For low flow coefficients, below the peak in $NPSH_{inception}$, suction backflow and recirculation could be observed in the simulation.

From the full scale visualization tests, several points have been plotted on the same figure for the number 1 "design" version impeller. The triangular points show cavitation with corresponding

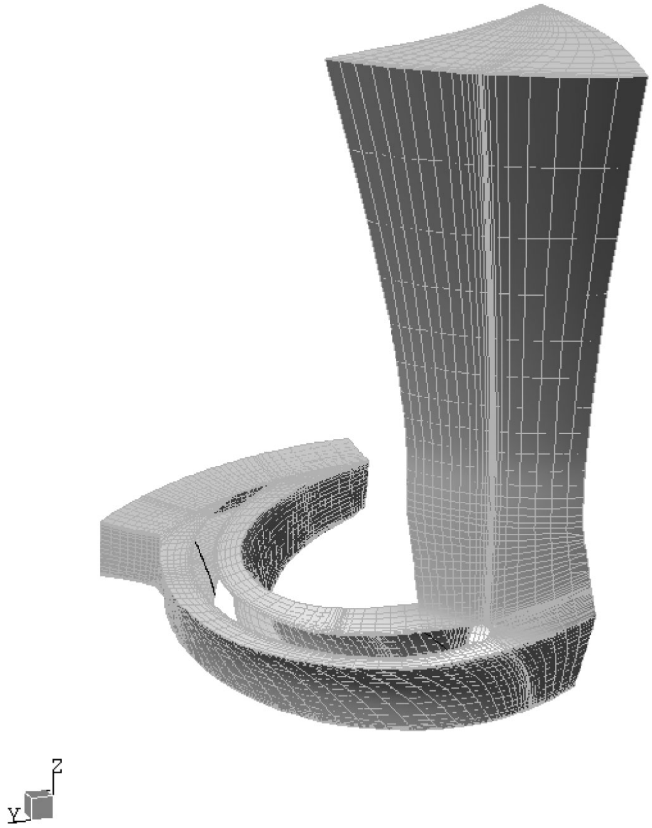

Figure 6. Example of Impeller Grid Model Showing Inlet and Outlet Boundaries and One Single Blade with Periodic Boundary Conditions.

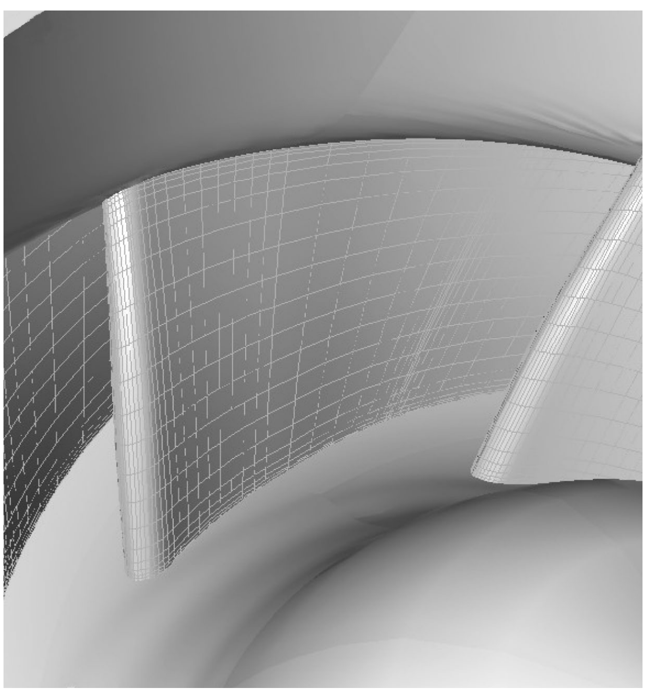

Figure 7. Example of Impeller Grid Model Blade Leading Edge with Grid (Right, Multiple Blades Shown for Viewing Purposes).

cavity lengths determined from the visualization (length in streamline direction relative to impeller outlet radius). Few points were available, but it can be seen that below the theoretical

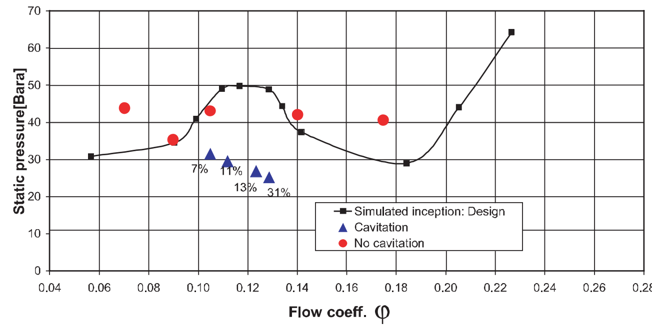

Figure 8. Impeller Number 1 "Design" Version. Simulated Inception Line and Experimental Points (Visualization).

inception line, significant cavity lengths are present in the low flow region. The circled points, representing cavitation-free operation, lie close to and above the inception line. However, at low flow, one point without cavitation can be seen located close to but below the line.

In Figure 9, the same analysis is presented for impeller number 1 "as-built" version. The simulated inception line has changed significantly. The off-design peak in $NPSH_{inception}$ is dramatically increased and covers a wide range of capacities. The minimum $NPSH_{inception}$ point is moved from $\varphi = 0.18$ to approximately 0.27. However, this result is not surprising, recalling the significant deviations in the inlet angle shown in Figures 3 and 4. This curve also explains the field case erosion problem with this impeller, operating with a suction pressure of 40 bara (580 psia) (preferred region in the field case was from $\varphi = 0.13$ to 0.20). The triangular points from visualization show significant cavitation in the operating region with up to 20 percent cavity lengths, in correspondence with the observed erosion marks (as shown in Figure 2). However, one point at low flow indicates cavitation also outside the simulated inception line.

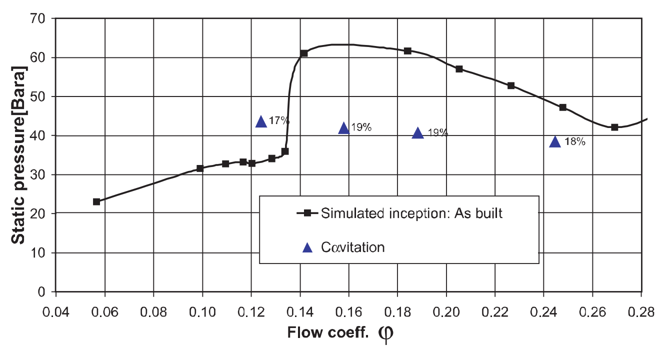

Figure 9. Impeller Number 1 "As-Built" Version. Simulated Inception Line and Experimental Points (Visualization).

For impeller number 2, a similar analysis was carried out, but only the "as-built" version was studied experimentally. In Figure 10, the "as-built" version is shown with the simulated inception line and even more visualization points. It can be seen that a significant reduction in $NPSH_{inception}$ is present compared to the impeller number 1 "as-built" version. With a typical suction pressure of 40 bara (580 psia) for impeller number 2 "as-built" version, numerically a cavitation-free operation can be expected in the complete region up to approximately $\varphi = 0.23$. Studying the experimental visualization points, several cavitation-free operation capacities are found in the region of 28 to 40 bara (406 to 580 psia). This means that according to both simulations and visualizations, a cavitation-free operation is expected for this version when having the above-mentioned suction pressures and flow coefficients.

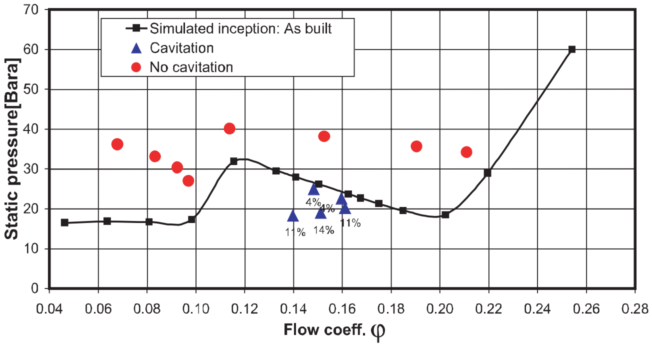

Figure 10. Impeller Number 2 "As-Built" Version. Simulated Inception Line and Experimental Points (Visualization).

For pressures below the simulated inception line and between $\varphi = 0.14$ and 0.16, cavitation with cavity lengths from 4 percent to 14 percent is present, the lowest lengths lying close to the simulated inception line. In Figure 11, the simulated inception line is shown for impeller number 2 "design" version. No visualization data for this were available, but the shape is quite similar to the "as-built" version with the minimum point at the same capacity. However, the curve shows between 10 to 20 bars (145 to 290 psi) higher $NPSH_{inception}$, decreasing the cavitation-free operational region and shifting it to higher flow with 40 bara (580 psia) suction pressure.

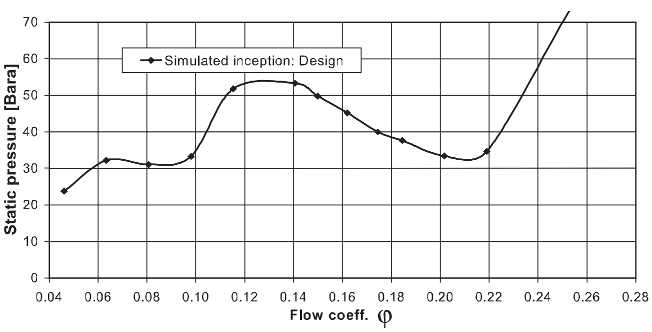

Figure 11. Impeller Number 2 "Design" Version.

DISCUSSION

The significant difference between the number 1 and number 2 "as-built" impeller, with respect to cavitation performance, was one of the clearest conclusions from this study. The overall performance including efficiency did not show such a big difference, although the BEP for the number 1 "as-built" version was at slightly higher capacities. The field experience and visual observations with the "as-built" version of impeller number 1 was in good agreement with the simulated inception line with respect to presence of cavitation. The inlet profile deviation, as shown in Figure 3, was the clear reason for the cavitation erosion problem. However, the main shape of the inception line was difficult to verify because of the few observation points. The number 1 "design" version shows an inception line corresponding to a 40,000 hour lifetime estimation carried out during the original design work. The cavitation-free operational region with typically 40 bara (580 psia) suction pressure is in accordance with the recommended preferred operation region. The visualization points also suggest a cavitation-free region for this version. Consequently, this means that if a high precision manufactured impeller had been delivered and operated in the field, no cavitation erosion would have been expected in the preferred operating range. However, sufficient field data for this impeller are not yet available to get the complete long-term validation.

On the other hand, for impeller number 2 the deviations in the inlet profile for the "as-built" version improved the cavitation performance compared with the number 1 impeller. The leading edge shape as shown in Figure 4, can explain the differences. It can be seen that the suction side tends to decrease the inlet angle while the pressure side is close to the design angle, resulting in the "as-built" leading edge tolerating flows over a wider range without significant pressure drop on the leading edge.

Numerical simulation of flow in an impeller at severe off-design conditions can introduce uncertainties. Part of the reason for this lies in the inaccuracy of CFD analysis at severe off-design conditions. The flow is more unsteady and a convergence of a steady-state simulation can be difficult. This can be the reason for some observed points lying on the wrong side of the inception line at off-design. However, the inception line is based on interpolation between several simulated points, and more simulated points need to be included. An additional uncertainty factor is that the simulated flow has been compensated for wear ring leakage to arrive at the true flow through the impeller (i.e., there is uncertainty in evaluating the exact leakage flow).

Generally, since the use of $NPSH_{3\%}$ often is the adopted approach to estimate cavitation limits, it is in some cases a need to evaluate suction pressure, especially for high energy pumps. Project personnel often tend to use $NPSH_{3\%}$ as the cavitation limit. This approach can lead to cavitation problems in the field. To achieve better validation, more attention to cavitation erosion and lifetime estimation should be based on a close cooperation between the end user and pump supplier. The need for such attention increases as the power increases.

CONCLUSIONS

A continuously running, high pressure water injection pump needs special attention on cavitation performance. Despite the limited amount of experimental data, the simulated cavitation inception line seems to capture the main trends satisfactorily, based on visualization tests on a full scale test pump and field observations. The typical shape of the inception line is also in qualitative agreement with typical characteristics published by others. The simulation of inception has been adopted as a design rule for achieving cavitation-free operation of the first high pressure stage impeller on high pressure water injection pumps. The second interesting point from the study is that the local deviations in the inlet profile, often observed in castings, can have a significant impact on cavitation performance. Although the deviation observed in the number 1 case is extreme and too high for leading edge trim, it is shown that local inlet angle deviations often control the cavitation performance. The shape of the leading edge is important, as shown for impeller number 2, and leading edge trim can often be used to optimize the local angles. When designing for zero tolerance, the deviations observed in the casting blade profile mean that carrying out numerical simulations on a design version could lead to significant errors in the local pressure profiles and thereby cavitation performance, even though the overall performance of the impeller is satisfactory. This topic needs further attention, with both numerical and experimental studies.

The main objective in this exercise was to reduce cavitation problems. This means having suction pressures with sufficient margins to achieve at least 40,000 hours' lifetime or, as discussed, zero-tolerance to cavitation inception. Generally, using numerically generated inception curves can be time and resource consuming when designing high energy pumps. Additional simple rules and criteria are therefore needed. This can be developed and evaluated by a closer cooperation between the end user and pump manufacturer.

NOMENCLATURE

A = Area
N_q = Specific speed of high pressure impeller in ISO units

Q = Flow rate
U = Impeller peripheral speed
BEP = Best efficiency point
NPSH = Net positive suction head
φ = Flow coefficient ($\varphi = Q/A\ U$)

Subscript

1 = Impeller eye (inlet)
inception = Cavitation inception condition

REFERENCES

Dupont, P., 2001, "Numerical Prediction of Cavitation in Pumps," *Proceedings of the Eighteenth International Pump Users Symposium*, Turbomachinery Laboratory, Texas A&M University, College Station, Texas, pp. 59-65.

Ferman, R. S., Cugal, M., Gopalakrishnan, S., and Fuller, P. D. K., 1997, "Boiler Feed Pump Re-Rate for Increased Head and Reduced Cavitation," ASME FEDSM97-3337.

Guelich, J. F. and Pace, S. E., 1986, "Quantitative Prediction of Cavitation Erosion in Centrifugal Pumps," IAHR Symposium.

Hirschi, R., Dupont, P., Avellan, F., Favre, J. N., Guelich, J. F., and Handloser, W., 1997, "Centrifugal Pump Performance Drop Due to Leading Edge Cavitation: Numerical Predictions Compared with Model Test," ASME FEDSM-3342.

Kumaraswamy, S. and Radha Krishna, H. C., 1986, "Cavitation Studies of Centrifugal Pumps," *Journal of Irrigation and Power*, pp. 99-116.

Schiavello, B., 1988, "The Impact of Fluid Dynamic Design on Minimum Flow," from Panel Session on Specifying Minimum Flow, *Proceedings of the Fifth International Pump Users Symposium*, Turbomachinery Laboratory, Texas A&M University, College Station, Texas, pp. 186-192.

Simoneau, R., Avellam, F., and Kuhn de Chizelle, Y., 1989, "On Line Measurement of Cavitation Erosion Rate on a 2D NACA Profile," *Proceedings of the International Symposium on Cavitation Noise and Erosion in Fluid Systems*, ASME-FED, pp. 95-102.

Vlaming, D. J., 1981, "A Method for Estimating the Net Positive Suction Head Required by Centrifugal Pumps," ASME Paper Number 81-WA/FE-32.