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Pump Cavitation Severity Evaluation Using Accelerometers and Dynamic Pressure Transducers after Installation

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ABSTRACT

This lecture demonstrates the effectiveness of using accelerometers and dynamic pressure transducers in the time domain to investigate cavitation root causes, and to quantify the cavitation severity. The technique presented allows for a minimally invasive test of the pump system. The lecture includes examples of cavitation in various pumps and services, including cases where the mean suction pressure readings indicated sufficient NPSHA (Net Positive Suction Head Available), but large pressure pulsations entering the pump suction drove instantaneous NPSHA below the NPSHR (Net Positive Suction Head Required).

INTRODUCTION

The concept of cavitation is a major consideration in the pump industry. As discussed by Brennan (1994) and Grist (1999), cavitation consists of bubble formation in the flow stream in zones where the local static pressure (in the traveling bubble frame of reference) has dropped below vapor pressure at the local temperature, followed by sudden collapse of the bubble when it progresses to a higher pressure zone downstream, as illustrated in Figure 1. The collapse can be violent, and damaging to flow path surfaces. Cavitation may be the result of insufficient suction pressure, or it may have an indirect source of operation at flow rates well away from the best efficiency point (BEP), resulting in recirculation vortices at the pump suction (and sometimes discharge) that create low pressure at their center. Other sources are possible but less common, such as rope vortices or vane tip vortex shedding. In the case of cavitation due to excessively low suction pressure over an extensive region of the primary flow path (what we will term “classical cavitation”), because the bubbles take up flow path real estate, cavitation causes degraded pump performance, in addition to the damage it can cause to flow path components such as impeller vanes. Cavitation is noticeable by ear, without instrumentation, typically causing a loud “rumbling” or grinding sound emanating from the pump that sounds like rocks in a mixer. In terms of instrumented testing, the Hydraulic Institute (HI) has an NPSH (Net Positive Suction Head) standard, ANSI/HI 9.6.1, that provides shop testing methods and criteria for contractually determining the presence of operationally-relevant cavitation based on the suction pressure at which 3% reduction in produced discharge head occurs due to the vane passage blockage by cavitation vapor bubbles.

In whatever manner the presence of cavitation is identified, there is general agreement that to quantify the extent or intensity of the cavitation, and its potential for damage in a specific pump and system, requires specialized testing and instrumentation. As vapor bubble-containing flow progresses into a higher pressure zone such that the fluid’s static pressure is above vapor pressure in the flow stream frame of reference (for vapor pressure vs. temperature of various fluids see Figure 2), the evaporated fluid bubble condenses quickly, collapsing and forming a narrow supersonic jet very similar to a shaped-charge implosion, as shown in Figure 3, along with a small but powerful “sonic boom” consisting of a roughly spherical wave. Dependent upon the extensiveness of cavitation, both TDH and efficiency of the pump are reduced due to the localized blockage of the flow stream. As a long-term consequence, cavitation bubble collapse on impeller vanes or casing walls may cause substantial erosion of the pump component surfaces, as shown in Figure 4. This occurs after a subsurface crack initiation period called “incubation”, which is material-dependent but is typically order of 40 to 100 hours of operation in strongly cavitating conditions (Figures 5 and 6). The rate at which subsequent erosion damage occurs depends on the surface material properties (Figure 7), how often bubble implosion events occur over time (these tend to be random events), and the severity of the collapse (this depends on many factors, including bubble size). When the bubbles collapse well within the flow stream rather than next to the flow path metal surface, cavitation is not harmful long-term to the pump hardware.

Over the years, the authors’ organization has developed a methodology to identify and quantify cavitation severity, as well as whether the source of the cavitation is inadequate NPSH (i.e. the NPSH available, NPSHA, is less than the NPSH required, NPSHR, plus appropriate margin depending upon the pump type and service), or rather some form of internal flow recirculation, with attendant vortices with their low pressure cores. While this is an area of on-going research, representative experimental results are presented in this lecture. To date, most of the research has been performed in the field at end-user sites, rather than in a laboratory setting. A typical field scenario is that an end-user notices “cavitation-like” noise originating from their pump. Sometimes these are new installations, and sometimes the pump has been in operation for several years but something has changed in the system. Many problematic applications involve horizontally split-case double suction between bearing single stage centrifugal pumps, primarily because such pumps are used when suction pressure is known to be a challenge. Cavitation measurement examples in several of these pump installations will be discussed, and a vertical axial flow circulator pump will also be presented. In various cases, the authors installed high frequency accelerometers and dynamic pressure transducers (in some cases, a hydrophone and microphone for comparison purposes) to determine when cavitation was occurring, and with what severity, based on structure-borne as well as fluid-borne noise. As will be shown, there is a benefit of performing cavitation testing using structure-borne noise because this can be determined using externally mounted accelerometers, which are much more convenient than installing hydrophones or dynamic pressure transducers. However, direct measurement of pressure oscillations, where this is practical, has an advantage of allowing classical cavitation (i.e. NPSHA is below NPSHR, or at least NPSHi, i.e. NPSH inception) to be distinguished from recirculation vortex cavitation, and also for determination of the incipient cavitation point, which may be helpful in certain unexpected erosion troubleshooting scenarios.

Manufacturer tests which can document the onset of significant cavitation and its associated noise typically consist of a discharge head drop test, such as the three percent head drop test (NPSH3) performed per Hydraulic Institute (HI) Standards ANSI/HI 9.6.1. Using this long-established technique on the manufacturer's test stand, data in the form of Figure 8 is obtained, and this is then used to establish published curves of NPSHR across the allowable flow range (AOR). An example of a published curve is provided in Figure 9, and how the required NPSH (based on NPSH3) as well as NPSHA change with flow is provided by example as well. Typically, keeping NPSHR below NPSHA is not sufficient to avoid cavitation, including attendant noise and possible damage. As discussed in the last several versions of the Hydraulic Institute Standards, failure of the three percent head drop test indicates that cavitation is present and severe, but unfortunately the three percent head drop test may be passed even if there is still incipient cavitation with associated substantial cavitation damage. Some degree of "margin" factor (the ratio of NPSHA to NPSHR) is required for various pump services. For this reason, HI recommends NPSH margin be maintained, such that there is a recommended ratio of NPSHA over NPSHR that is greater than 1.0. In clean lake water pumping, for example, the recommended ratio is 1.2 in most cases.

If the pump passes the three percent head drop test across the AOR with the suction pressure at the specification's minimum value, but cavitation noise is nevertheless present, then the end-user may insist on a post-installation impeller surface inspection after a significant period of time (e.g. 1000 hours) in order to prove whether or not the location of the cavitation bubble collapse, and level of bubble collapse impulse on metal surfaces, causes significant damage. This leads to contractual disputes and misunderstandings.

A reliable method to quantify harmful versus nuisance cavitation would be very useful to the industry, since NPSH testing to quantify true required NPSH margin is beyond the scope of the present Hydraulic Institute Standards. Run-disassemble-and-observe as currently applied is personnel-intensive and expensive, and requires cumbersome on-site in-service impeller erosion inspection. The process takes a prolonged period, during which fair dispersal of contract set-asides are delayed and, if a problem does exist, appropriate fixes end up being postponed.

In the field, high frequency dynamic pressure or vibration measurements, as will be described, can quantify cavitation, but no standards for interpretation exist. A typical pressure transducer used by the authors for this purpose is shown in Figure 10, and a typical single axis accelerometer that is used by the authors is shown in Figure 11. In the authors' experience, proper measurement of surface acceleration on the suction casing can provide test results that may be compared to consistent criteria, so that judgments concerning cavitation damage potential can be made. If high frequency pressure oscillations are also measured, the reason for identified damaging cavitation can be determined as well, for example so that either NPSHA or onset of internal recirculation can be modified, each of which involve a different approach.

BACKGROUND

Varga & Sebestyen et al (1969) were among the first to apply accelerometers in an attempt to quantify cavitation severity. They stated "The signs detectable by a one-component [single axis] accelerometer render qualitatively correctly the cavitation characteristics of the pump." They did not provide data plots, but the publication date suggests that the data consisted of time domain oscilloscope traces. The frequency range was reported by them to be 5 kHz to 20 kHz.

Guelich (Apr. 1989a), a long-time researcher and authority on centrifugal pump cavitation, stated: "Bubble implosion during cavitation causes the emission of pressure waves which can be measured by pressure transducers as liquid borne sound, or by accelerometers at the outside of the pump casing as solid-borne sound. The great majority of the energy density is in the range of 1kHz to 100 kHz. Accelerometers can be used on the casing surface instead of suction pressure transducers, but with some loss of sensitivity." The loss of sensitivity in the accelerometer was mentioned in the context of using a frequency spectrum rather than a raw time-based signal for the cavitation strength assessment. Guelich in his 2010 book modified his position, stating that "It is also possible to monitor solid-borne noise instead of fluid-borne noise. Solid-borne noise signals were measured on the outside of the inlet casing using an accelerometer sensor. The curves exhibit quite similar characteristics as the fluid-borne noise." Later he states "solid-borne noise measurements on the pump casing are often the only practical way to obtain information on the hydrodynamic cavitation intensity in plants." However, he warns "If the solid-borne noise is measured on the outer casing wall with an accelerometer, the [comparison with a] signal of a given fluid-borne noise depends on the casing geometry and the material." This warning was in the context of interpreting the vibration in terms of rms value of the frequency spectrum. In such a case, he says further "To avoid resonance with the transducer, the measurements were made in the range of 1 to 45 kHz." He also warns "In plants it is often difficult to determine load-dependent background noise which must be known to make a reliable assessment." Finally, "Fluid-borne noise measurements capture any bubble implosions in the pump even if these occur at a distance from the blades and do not contribute to erosion."

Guelich (April 1989a) further states that a typical individual bubble size at collapse is 1mm, and typical collapse time is 10 microseconds. This suggests an acoustic wavelength of about 15 mm in water. By the authors' calculations, the comparable wavelength in metals for the same collapse period is about 45 mm for brass or bronze, 35 to 58 mm for cast iron, and about 58 to 59 mm for various steels. Typical pump casing walls are order of 10 mm to 25 mm thick. To put this in perspective, for a 15 mm thick casing, the shock wave takes about 2.5 microseconds to pass from inside the casing wall to outside the casing wall (slightly longer for bronze). This implies very little dispersion of the bubble shock wave through the casing thickness, regardless of material. From this point forward, the sound level disperses, losing wave amplitude roughly proportional to the radius from the implosion location.

Guelich (April 1989a) further states "The results in the literature often concern measurements which have been done in a narrow frequency band. This might lead to very wrong conclusions. It is difficult to establish correlations between noise level and erosion in such cases". This is not an issue for time-based measurements instead of frequency spectrum measurements. He also points out that cavitation noise measurement must compete with many other noise sources:

1. Machine noise, generated mechanically (e.g. bearings, driver, rubbing)
2. Gas bubbles contained in the flowing fluid
3. Unsteady blade forces on the impeller ("blade pass")
4. Turbulence of incoming flow
5. Noise generated by valves, struts, or other flow system components (e.g. vortex shedding frequencies)

These other noise sources compete in total energy across the spectrum, but are of much lower energy in the brief periods of time in which cavitation pulses occur. Guelich also states that, when measuring cavitation, the following instrumentation characteristics are key:

1. Type of pressure transducer
2. Frequency response of pressure transducer
3. Usage of low pass or high pass filters
4. Type of signal processing (e.g. FFT) and form of output (e.g. rms or peak value)
5. Position of the pressure transducer versus the impeller (e.g. how close, have line-of-sight?)

These remain important for either time or frequency spectrum based approaches.

In another reference, Guelich (2010) states the velocity c of the bubble implosion at the moment of collapse as

$$c = \sqrt{\left(\frac{2}{3}\right)\left[\frac{p-pb}{rho\ g}\right]\left[(R_0/Re)^3 - 1\right]} \qquad \text{Equation 1}$$

where p is the freestream pressure, pb is the pressure in the bubble (typically vapor pressure), rho is the fluid density, Ro is the radius of the bubble at the start of implosion, and Re is the radius of the bubble at the moment of complete collapse. He surmises that the pressure at the point of collapse follows the water hammer pressure relationship:

$$P_i = rho \times a \times c \qquad \text{Equation 2}$$

Where rho = fluid density, a = fluid speed of sound, and c is the bubble implosion velocity from equation 1.

For typical fluid properties and service conditions in a centrifugal pump, this indicates that Pi can easily exceed 1000 bars (about 15000 psi). This is roughly the corrosion-assisted fatigue strength of carbon steel in oxygenated water. At 3000 bars, Pi would exceed the notched (i.e. imperfect surface) fatigue strength of nearly all pump impeller materials. This is consistent with industry experience that cavitation as it occurs in centrifugal pumps will lead to gradual surface fatigue and subsequent erosion of engineering metals, with some metals being much more resistant than others, but all being susceptible at worst case realistic cavitation intensities.

The bubble formation which enables later collapse, and controls its intensity, is determined by NPSH. NPSH as discussed above is defined as the pump suction local total pressure minus the vapor pressure at the prevailing fluid temperature, with the quantity divided by density times the acceleration of gravity in order to convert the pressure units into units of head. Converting total pressure into static pressure p_s plus dynamic pressure based on freestream velocity Vs, NPSH may be redefined as:

$$\text{NPSH} = \left[\frac{p_s - pv}{rho\ g}\right] + \left(\frac{{V_s}^2}{2\ g}\right) \qquad \text{Equation 3}$$

A variety of additional excellent investigators have focused on the cavitation assessment issue over the years. For example, Gopalakrishnan (1985) observed pressure signals in the time domain with roughly 50 kHz frequency response, with line-of-sight to the impeller by the pressure transducer. He defined a Signal Above Threshold (SAT) to qualitatively assess cavitation severity. Schiavello et al (1988) used high speed photography on a full scale representatively operating feed pump, operating on the test stand, with specially marked impellers and Plexiglas observation ports. This work was among the first to document the occurrence and importance of incipient cavitation, occurring at pressures well above the 3% head drop NPSHR. In a similar vein but less applied, Naka et al (1995) used very high speed photography on hydrofoils exposed to cavitation, to assess cavitation bubble formation and collapse and attendant “violent vibration”.

Koivula et al (2000) used accelerometer signals for cavitation detection and cavitation severity determination. They did not look at pump hardware, but rather signals from a robust wall in which a cavitating orifice was installed. Time domain peak G levels were observed and were the basis for the prediction of the force occurring at the bubble implosion point. G levels of about 50 G’s were registered when cavitation was strong, with sampling rates of 200 kHz and low pass filter set at 100 kHz, with a high frequency response B&K 4384 accelerometer. Escaler et al (2001) used a vortex generator to induce cavitation bubble formation and collapse next to test coupons. A Dytran 5800SL high frequency accelerometer was used, with a 500 kHz sampling frequency. Transfer functions, including resonant behavior, were determined by impact tests, to obtain a relationship between accelerometer reading and force on the surface of the test coupon in the zone of bubble implosion. Cavitation pit size for a given number of implosions was plotted versus inferred forces, and this plot showed that at a given bubble implosion force (120 N in their rig) erosion ceased, apparently because the implosion peak pressure fell below the material fatigue endurance limit.

Kiesbauer et al (2006) developed a cavitation severity assessment method based on using pressure transducers to acquire acoustic events in the time domain, rather than in a frequency spectrum. They state “The signals caused by cavitation have a special shape. Large amplitudes that trail off fairly slowly [i.e. by ringing down over hundreds of microseconds according to their data] arise when cavitation events (implosions) occur.” Such events, in valves as well as pumps, are very distinctive in the time domain, and nearly always rise far above (in brief amplitude spikes) any other noise in the pump or system. Conversely, in the frequency domain, these brief spikes simply cause a mild rise across the entire frequency spectrum, out to order of 100 kHz, with some resonance zones rising more, but still a modest amount that is pump-specific and instrumentation-specific, and therefore is difficult to interpret.

Sloteman (2008) performed cavitation testing on pumps in the laboratory using six probes: 1) inlet pipe pressure transducer, 2) suction bay pressure transducer, 3) suction bay accelerometer, 4) bearing housing accelerometer, 5) front bearing piezoelectric load cell, and 6) airborne noise microphone in front of the suction bay casing wall. He evaluated the data both in terms of the frequency spectrum, per the widely applied EPRI method (Guelich, Nov. 1989b), as well as plotted raw signals in the time domain, with all data acquisition performed at 70 kHz The easiest to interpret signals appeared to be in the time domain. The pressure transducers 1 and 2 presented modest response, the accelerometer on the suction bay outer wall showed the strongest response, the bearing housing accelerometer showed response but with similar sensitivity to the pressure transducers, the load cell showed negligible response (presumably an F=ma issue), and the airborne noise showed response only to the strongest pulses, and significantly muted the peak value of the ringdown. Sloteman’s results resemble those of the authors’ experience.

There is an important lesson that the authors gathered from the previous research literature. Cavitation spikes are very clear and have a unique "signature" shape when plotted in terms of oscillation versus time. In this form, either high frequency pressure or casing acceleration are clear indicators of when cavitation of significant strength is occurring, and when it is not. With proper calibration and interpretation, researchers have been able to define a consistent correlation between the height of the surface acceleration vs. time spike amplitude, versus cavitation damage rate. Furthermore, when observed in the high time resolution (i.e. high frequency response) time domain, the amplitude of other sources of noise, e.g. from flow phenomena or mechanical forces, are generally not competitive with cavitation noise, thus avoiding the lack of clarity in this regard when cavitation signal response is evaluated in the frequency domain, which has been a chronic problem with frequency spectrum techniques such as the EPRI technique (Guelich Nov. 1989), with its erosion rate estimation method summarized in Fig.12. In this technique (one of the best frequency spectrum based techniques), the cavitation energy is distributed over a very broad frequency range, instead of being focused in very high and easily distinguishable spikes over very limited periods of time for each bubble collapse. This can lead to poor quantitative prediction of erosion rate, as illustrated in Figure 13 (RCCE, Japan, 2008).

METHODS OF DETERMINATION OF CAVITATION USING ACCELEROMETERS AND PRESSURE TRANSDUCERS

General Considerations

Cavitation bubbles form in a flow when the local static pressure (typically presented in the cavitation context as psi absolute, not psi gage) is less than the vapor pressure of the fluid. These bubbles then rapidly collapse some distance downstream when the local pressure once again exceeds this vapor pressure. In a centrifugal pump impeller, if classical cavitation occurs it would typically initiate with bubble formation near the leading edge of the vane (the lowest static absolute pressure location), with the bubbles then traveling downstream with the through-flow of the impeller, such that they later collapse near or on the suction side of the vane (visible part of the vane, as seen when staring down the "eye" of the impeller). If the collapse occurs on a solid metal surface, such as the impeller vane surface, a high-intensity stress wave is created that propagates through the metal as a structure-borne noise pulse. This initially unidirectional wave conserves most of its energy as it propagates initially by compression through the thickness. Following this, it loses energy at a rate roughly proportional to radius, by either compression or shear throughout the pump casing. There are also waves that propagate through the fluid and reflect from opposing walls, but these are much lower in energy than structural waves from bubbles that collapse on or very near the surface. Bubbles that collapse on the rotating impeller vanes, hub, or shroud, in the authors' experience, appear to give line-of-sight casing acceleration levels similar to bubble collapse directly on the casing wall. This process is not understood, and suggests "wave-guiding" of the cavitation acoustic wave from the impeller to the casing, perhaps by the thin annual cavity of the wear ring. The issue deserves further study.

The structure-borne vibration acceleration spikes have very high amplitudes, order of 100 to 1000 G's if the cavitation bubble collapse is on or close to either the casing or impeller metal surfaces. The high G level exists for a very brief moment (order of 10 to 50 microseconds), such that gross structural motion, or extended areas of damage, from a single bubble collapse do not occur. However, the very high local acceleration at the rapidly moving wave front can cause fatigue crack initiation, and later propagation near the wave front, which eventually results in fatigue cracking and material removal in the form of cavitation erosion damage. This has the appearance of pitting, and results in reduced thickness of critical functional locations such as impeller vanes.

If the collapse of the cavitation bubbles occurs in the flow stream some distance away from a solid surface, then the event will produce a lesser shock wave and cause audible sound, but will not result in material damage. In this case the peak acceleration amplitude has been observed by the authors to be considerably lower, typically order of 5 to 25 G's, which still may be considered high, but is for a very brief period of time, and by calculation does not lead to sufficient stresses to cause fatigue (i.e. stress oscillations remain below the fatigue endurance limit, and stress intensity stays below the fracture mechanics crack propagation threshold stress intensity for typical pre-existing flaws, Kth). Cavitation damage to a pump is due to metal fatigue from the repeated collapse of cavitation bubbles on the metal surface. Damage is dependent upon several key factors which include the peak stress resulting from each bubble collapse, the rate at which the collapse occurs at a particular location, and the resistance of the material to fatigue damage. In general, as pointed out by Guelich (2010), the material removal rate will be highest at the highest flow rates, everything else being equal. This compounds the fact that, as flow rate increases, typically there is greater loss in the suction line and therefore a lesser value of NPSHA, while the local static pressure reduction at impeller vanes, proportional to velocity squared, reduces NPSHR as flow increases for a given impeller. Also at these higher flow rate, whether the bubbles collapse near or far from metal surfaces, there remains the adverse performance effect of cavitation bubbles before their collapse, in that the formation of the vapor bubbles acts as an obstruction to the flow through a pump impeller and results in a decrease in the pump head and flow. This latter factor can be quantified by the NPSHR3 test of ANSI/HI 9.6.1.

In general, as per the literature discussed in the Background above, those attempting to quantify cavitation based on vibration or pressure pulsation have done so using changes in the amplitude vs. frequency spectrum. Cavitation events are random in nature and except in special circumstances (e.g. cavitation surge, discussed later) do not repeat periodically with time, so they produce in general a minor rise in the vibration spectrum, but not distinct narrowband peaks. Viewing the cavitation events in the time domain with a high frequency data acquisition system (>25 kHz, and preferably >100 kHz) allows for the capture of the distinct cavitation events. In the experience of the authors, the potential for pump flow path damage from the cavitation is best assessed based on the instantaneous peak values of these time-based measurements.

The existence of cavitation is detectable by the human ear as airborne sound. As pointed out by Guelich (2010), the audible noise is mostly in the range of 1 kHz to 10 kHz, and is actually from the oscillations of the bubble cloud, not the actual bubble collapse; human hearing can detect sound from about 20 Hz to approximately 20 kHz. However, it is the authors' experience that microphone measurements of this airborne sound are not the best indication of the ability of cavitation to damage a pump. The measurement of the sudden acceleration of the structure which results from the collapse of the cavitation bubbles has proven to be a better indication of the damaging effect of the cavitation, better in fact than the less convenient direct measurement of pressure near the impeller suction, whether this pressure is obtained by piezoelectric pressure transducer or line-of-sight high frequency hydrophone. This is due to the fact that the material erosion, if any, is occurring on the structure, and the accelerometer measurement is directly on the structure.

As the structure vibrates from a damaging cavitation pulse, it also causes increased audible (and ultrasonic) airborne noise. However, measuring the structural vibration directly avoids transmission losses which exist in the detection of cavitation with airborne sound measurements (e.g. via a microphone). A quantitative damage potential can be assessed using the surface acceleration that is not possible measuring sound intensity or sound pressure level at some distance from the casing walls. In the authors' experience, instantaneous peak structural acceleration levels of 15 g's or less indicate that the cavitation will not produce damage to the pump, and acceleration levels of 100 g's or more indicate the presence of damaging cavitation. The range between 15 to 100 g's may not clearly indicate whether the cavitation is damaging, and this depends upon material cavitation resistance, casing wall thickness, and how close the measurement location is to the bubble collapse zone. In the author's experience in the field, the acceleration peaks measured on the suction casing wall are either well below 15 G's, and in those cases no cavitation damage is found in practice, or well above 100 G's, and in those cases it is accompanied audible noise and noticeable cavitation damage after about 1000 hours. In instances in the authors' experience, audible cavitation noise is present at a suction head at or below NPSH3, but if there is no significant rate (spikes per second) of acceleration peaks above 15 G's, then inspection after 1000 hours has exhibited negligible cavitation erosion damage.

Data Recording Methodology
The data reported in this lecture originated from tests which searched for evidence of cavitation by monitoring vibration/structure-borne noise with high frequency response accelerometers rigidly mounted to the external wall of the suction casing, and fluid-borne noise obtained from either hydrophones or high frequency response pressure transducers. Measurements were taken at various operating conditions, and in the case of the accelerometers, on various locations on the pump suction casing. Likewise, when cavitation was suspected to be from discharge recirculation, acceleration as well as fluid-borne noise measured by high frequency pressure transducers was obtained at the discharge volute. Single axis accelerometers (Figure 11) with temporary rigidly-behaving attachments were used to obtain the structural acceleration readings in the direction perpendicular to the casing surface, and occasionally triaxial accelerometers were used to obtain comparison signals in the two perpendicular directions as well. The frequency range of the accelerometers, limited by their lowest natural frequency dependent upon the attachment to the casing and the casing wall stiffness, varied depending upon specific tests, from 25 kHz to about 50 kHz. Pressure transducers (Figure 10) possessed a frequency range limited by their lowest natural frequency of about 70 kHz, and had relatively low high frequency cross-sensitivity to acceleration.

A multi-channel B&K FFT signal analyzer was used to translate probe signals into vibration or pressure pulsation readings. The anti-alias filtered frequency range of this analyzer was over 25 kHz, and test data was taken in the time domain up to about 60 kHz. Some data was taken using National Instruments hardware with 50 kHz anti-alias filtering, and time domain signal recording of 100 kHz. Guelich (2010) and others recommend (and the authors concur) that cavitation data measurements ideally should be taken over a frequency range of 1 k Hz to 150 kHz, although most of the cavitation energy can be expected to be contained in the 10 kHz to 60 kHz range. The authors' intent was to interpret the resulting signals in the time domain, as a high frequency "oscilloscope trace" of amplitude vs. time, providing the raw cavitation pulse and its "ring-down" reverberation. The probes and analyzers used were adequate for this purpose. If the frequency domain was to be used instead to assess and interpret the cavitation, as has been the approach for many decades by most researchers and field investigators, probes with much higher frequency range would have been required to avoid probe resonance, since such resonances would artificially amplify the frequency response near any resonances within the spectrum of the cavitation energy. This has been problematic for the industry, since probes with significantly higher frequency response are much more expensive in the least, and in the case of accelerometers may not be able to be practically implemented because of the compliance of the pump casing wall they are attached to.

The ideal location to place the high frequency accelerometers is on the suction volute casing in the axial direction opposite, and in line with, the impeller eye. This is illustrated by Figure 1, which indicates that the static pressure decreases as the measurement approaches the impeller eye, such that classical cavitation due to inadequate NPSH would be expected on the suction side of the vane, near the impeller eye. Whatever the absolute pressure at the pump suction pipe reads prior to the impeller, the local static pressure is less within the impeller eye prior to rapid head development within the vane passages. If classical (non-recirculation) cavitation is to occur, it is in this region, with subsequent bubble collapse with a few centimeters of the leading edge.

If the negative portion of the dynamic pressure at the impeller eye causes the suction pressure to drop below the vapor pressure, as shown in Figure 1, cavitation will occur. The rapid volume increase as the liquid flashes to vapor typically prevents additional pressure drop, which in any event would not be expected to drop below pure vacuum, 0 psia (although Brennan 2010 illustrates how sub-zero pressure- i.e. tension- can develop in very pure liquids). Therefore, any negative spikes from the dynamic pressure transducer response in the time domain will generally be "clipped" at a local pressure that relates to the vapor pressure being present at the cavitation zone. Figure 14 shows a double suction pump impeller subjected to the early stages of cavitation damage on the visible side of the vane inlets. Suction bay wall acceleration spikes, frequently above 100 G's, are evident in Figure 15. The vapor-pressure-boundary "clipping" for the same pump, over the same time period, is illustrated in Figure 16. Conversely, as also demonstrated in Figure 16, there is no reason for the positive spikes to be clipped. The clipping of the negative pressure oscillations such that they do not drop below the vapor pressure has been found to be a clear sign that there is cavitation in a pump or valve.

The time waveform from the accelerometer measuring structural vibration on the suction part of the pump casing, shown for "Pump A" in Figure 15, was taken on the outboard side of the casing "suction bay" wall, with line-of-sight to the impeller, using a small accelerometer which had a frequency response in excess of the data acquisition frequency. As can be seen in this plot, there is a peak acceleration level of more than 200 g's, which in the authors' experience typically indicates rapid erosion damage taking place, and which in fact was true in this pump.

Note that this pump was previously tested at the OEM's plant, on a test stand consistent with Hydraulic Institute standards. The authors do not know if the site conditions matched the laboratory conditions, such as degree of entrained and/ or dissolved gas content. At the plant site, the authors monitored vibration and structure-borne noise (accelerometers), and fluid-borne (dynamic pressure transducer) noise at various operating conditions. The testing was performed because the installed pump made audible cavitation noise when running at a flow at or above its design point. The authors' measurements indicated relatively large suction pressure oscillations (up to 12 psi peak was observed) and accelerations (the maximum was in excess of 200 G's peak as shown in Figure 15). The high-pressure oscillations (Figure 16) and G levels consisted primarily of high frequency pulsations, in a pulse time scale consistent with >30 kHz. The reason for the cavitation in this pump, which possessed average suction head above its NPSH3, was the 12 psi suction pressure oscillation, which instantaneously reduced suction head well below the NPSHR in an oscillating manner, and in fact reduced suction head well below the vapor pressure of the room temperature water being pumped.

When suction and/ or discharge recirculation is present, typically when the pump operates well away from BEP, elevated acceleration levels are often also present, indicating that cavitation is occurring within the core of some of the recirculation vortices. In this instance, the negative spikes from suction dynamic pressure transducers will not necessary be clipped off because the freestream static suction pressure away from the vortex tip is typically not actually falling below the vapor pressure. Recirculation vortex cavitation is not a "classical cavitation" issue, but the cavitation bubble formation within the recirculation vortices can still cause significant damage. Typically, such recirculation vortices form on the suction side of the vanes, but before the bubbles reach a high enough pressure to collapse, each vortex axis rotates and points its core tip at the pressure side of the neighboring vane, damaging the vane pressure surface or "back-side" as seen from the front of the impeller eye. The practical steps that cure such recirculation cavitation damage, e.g. altered impeller incidence angle, correctional guide vane pre-swirl in front of the impeller, overt siphoning of stalled flow zones, or an increase in flow rate (e.g. with a minimum flow recirculation valve) are typically different than the steps to solve classical cavitation, such as increased NPSHA, the use of an inducer as explained by Japikse (1997), or a redesigned impeller with reduced NPSHR. The latter may actually exacerbate suction recirculation potential due to higher suction specific speed, the details of which are beyond the scope of this lecture.

Test Examples
In addition to Pump A discussed above, examples are shown for Pumps B through F in Figures 17 through 31. In each case, the end-user complained of a noisy pump during operation. The data the authors collected showed random high frequency spikes. Spikes in the time domain, such as those observed, are typical of what the authors have found to be present in a strongly cavitating pump. This evidence was present to a similar degree on all of the pumps measured. Along with acceleration and dynamic pressure time traces, pictures of the damaged impellers are shown in most cases as well. The pumps shown as "Pump B", "Pump E, and "Pump F" are double suction pumps from various manufacturers, and each showed signs of damage due to a combination of apparent suction and/ or discharge recirculation, as well as classical cavitation. Often, discharge recirculation is indicated by high discharge pressure dynamic oscillations. The pump shown as "Pump C" is an axial flow solids handling circulator pump. The Pump D is a horizontal split case double suction pump oriented 90 degrees from horizontal to accommodate a motor on top for space saving purposes.

Figures	Pump Type	Pump Specifications	Application
14-16	Double Suction Horizontal Split Case "A"	12" suction, 10" Discharge/ 200 HP/ 6100 GPM/ 175 Ft Head / 1785 RPM	Water Treatment (clean)
17-20	Double Suction Horizontal Split Case "B"	10" & 12" Discharge/350 & 500 HP/ 2000 & 5000 GPM/ "B"375 Ft Head/ 1785 & 1770 RPM	Water Treatment (clean)
21-23	Axial Flow "C"	24" discharge /200 HP/ 18307 GPM/ 22 Ft Head/ 698 RPM	Wastewater Treatment (influent sewer)
24-25	Double Suction Horizontal Split Case "D" (Vertical Axis)	12" Discharge/ 700 HP/ 9000 GPM/ 200 FT HD/ 1800 RPM	Water Treatment (clean)
26	Double Suction Horizontal Split Case "E"	24" Discharge/ 2000 HP	Water Treatment (clean)
27- 31	Double Suction Horizontal Split Case "F"	30" Discharge, 2500 HP	Water Treatment (clean)

"Case History" details on these examples are given in the figure captions.

Some additional discussion will be useful in the case of Pump E as shown in Figure 26, however. It was designed by a reputable manufacturer in Japan using good hydraulic procedures. Nevertheless, the pump was cavitating strongly. In a zoomed-in time plot, on average there were about ten instances in which the peak acceleration exceeded 100 g's in a one second period (10 times per second is considered at the moderate-to-severe boundary, based on the authors' experience). This poor cavitation behavior was in spite of previously being tested in a well-regarded laboratory in France, where the pump was reported to have made modest noise at off-design points, but was also reported to have passed the EPRI-based cavitation criteria, as defined based on the EPRI procedure (Figure 12). As discussed previously, the EPRI procedure, while widely applied and included in API-610 as an informative section, may not be as accurate as desired (Figure 13). In part this is because the EPRI report mentions that "the conditions in the application must be equivalent to those for the tests used to establish the correlation" (see page S-5), and this is difficult to achieve in most cases in actual installations in the field, in terms of water air content, even-ness of inlet flow velocity, and inlet swirl control.

The flow entering the pump was calculated (by others) through computational fluid dynamics (CFD) to have excessive swirl. It is very likely that this swirl was a factor in the cavitation of this pump. Swirl in the flow entering the pump results in a mismatch of the inlet flow angle and the impeller inlet vane tip angle which can cause unexpected classical cavitation or early onset of recirculation cavitation. The CFD analysis indicated that the suction piping configuration would induce swirl near the impeller inlet at a level which would exceed the level recommended by the Hydraulic Institute pump inlet standard. Thus, the cavitation may have been entirely the result of mismatch between the incident flow angle and the impeller vane angle.

The data acquired during the test included airborne sound measurements, waterborne sound measurements, dynamic pressure measurements, and structural vibration measurements. This test provides a good example of the usefulness of the authors' time-based technique vs frequency-based techniques. As cavitation is a random, transient event, and typically does not repeat at a regular frequency (even when groups of cavitation events repeat themselves at lower frequency, as in cavitation surge), the authors have found that transforming this data into the frequency domain (via Fast Fourier Transform or FFT) does not adequately represent the very high amplitude instantaneous spikes which occur during cavitation. These spikes result from the rapid implosion of the cavitation bubbles. Traditionally, the frequency domain data has been used as an indication of cavitation by comparing the "floor" of the spectrum to one in which there is no cavitation. Cavitation causes a general rise in the broadband "noise floor" in the spectrum, and somewhat enhanced response from certain high frequency resonances, but in the authors' opinion it is difficult to quantitatively evaluate whether the detected cavitation is damaging, using such a method. The EPRI method discussed earlier makes an admirable attempt, apparently with a degree of success when applied by experts, under pristine flow conditions. In general, using frequency to predict cavitation occurrence and severity in a plant setting requires baseline data during for which the pump is operating without cavitation, for comparison. This may not be practical. Therefore, the authors have found it more useful to view the data in the time domain with a high speed data acquisition system (50 kHz or higher in the unfiltered time domain).

There was no opportunity for the authors to gather time-based surface acceleration data in the research lab in France, but at the installation site the time-based acceleration data provided unambiguous evidence of damaging cavitation. In spite of the pump's NPSHR being well below the nominal (without swirl) NPSHA, with adequate margin, the high G levels of the test results predicted damage to the long term reliability of the pump. Strong cavitation in the area of the impeller eye was evident by sharp, high amplitude suction pressure spikes, clipped on the negative side, and suction casing wall acceleration spikes at random intervals, at well over 100 G's, and as high as 400 G's. Upon opening the casing and inspecting the impeller, the problem was identified as classical cavitation, clearly evidenced by the severe rate of erosion on the suction-side of all impeller vanes, slightly behind each vane's leading edge. Cavitation damage depth measurements covered a consistent one inch by 1½ inch area on each 316L stainless steel vane, with a depth of material removal of between 0.18 and 0.22 inches at the worst location in each scar, after only 1000 hours of service.

CONCLUSIONS

Data presented in the figures provided were in the form of vibration versus time rather than vibration versus frequency spectra, because in the authors' opinion time domain plotting of cavitation spike peaks, in either terms of either surface acceleration or suction bay pressure, is the most sensitive and quantitative method of detecting damaging cavitation. It is important to implement this at high sampling rate (>50 kHz, or preferably more than 100 kHz). In such oscilloscope-like signal vs. time plots, individual bubble collapses during cavitation are evidenced by a characteristics sharp spike and rapid "ring-down", in which the spike rises for a very brief time to well above the otherwise typical acceleration oscillation amplitude from other hydraulic or mechanical phenomena. Classical cavitation can be distinguished versus recirculation or vortex cavitation by non-axisymmetric dynamic pressure spikes in the suction, where negative spikes are clipped-off when pressure oscillation extremes in the suction attempt to fall below the vapor pressure of the fluid being pumped. In the authors' experience, damaging cavitation occurs when acceleration spikes occur with an amplitude of greater than about 100 G's, when measured by the described method, although the threshold value is expected to be dependent upon casing wall dimensions, and material erosion resistance. For acceleration spikes below 15 g's, the cavitation damage is considered mild for any pump or material, likely occurring in the flow stream with sufficient separation from the vane or casing walls to prevent damage. The most cavitation resistant materials such as 316L stainless steel or nickel aluminum bronze will have their fatigue limit surpassed at somewhat more than 100 G's spike level. Impellers manufactured with cast iron on the other hand are more prone to accelerated erosion, and damage would be expected to initiate at levels perhaps as low as 50 G's. The rate of occurrence of acceleration spikes also affects the rate of damage of the cavitation.

RECOMMENDATIONS FOR FUTURE RESEARCH

The data provided in this lecture provides specific cases of casing wall acceleration levels associated with damaging cavitation. A threshold value of G level is proposed based on experience, but there is insufficient data to permit quantitative calculations to predict the erosion rate. Different materials will have different threshold G levels and erosion rates at a given G level. Further research can develop experimental correlations of measured dynamic pressure and/ or acceleration vs. damage. Comparison of such data to CFD simulations of bubble collapse dynamics and energy transfer to vane and casing walls should also lead to useful simulations of the bubble collapse loads vs. material fatigue strength.

NOMENCLATURE

FFT = Fast Fourier Transform
g, G = Gravitational Constant (L/s^2)

For cavitation formula nomenclature see Fig. 9

FIGURES

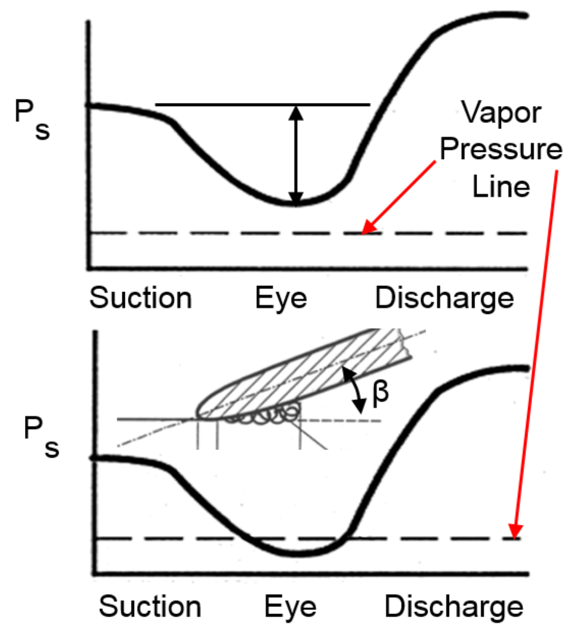

Figure 1. Plot of the pump suction pressure as it enters the suction nozzle, through the impeller eye, and out into the vane-to-vane passage way. The lowest pressure in the system is at the impeller eye just downstream from the vane leading edge. The upper picture shows the no-cavitation situation, and the lower picture shows the cavitating situation. β defines angle between shockless flow coincident with vane meridian and the vane leading edge rotational trajectory. The vane moves right-to-left. (Derived from Guelich, 2010).

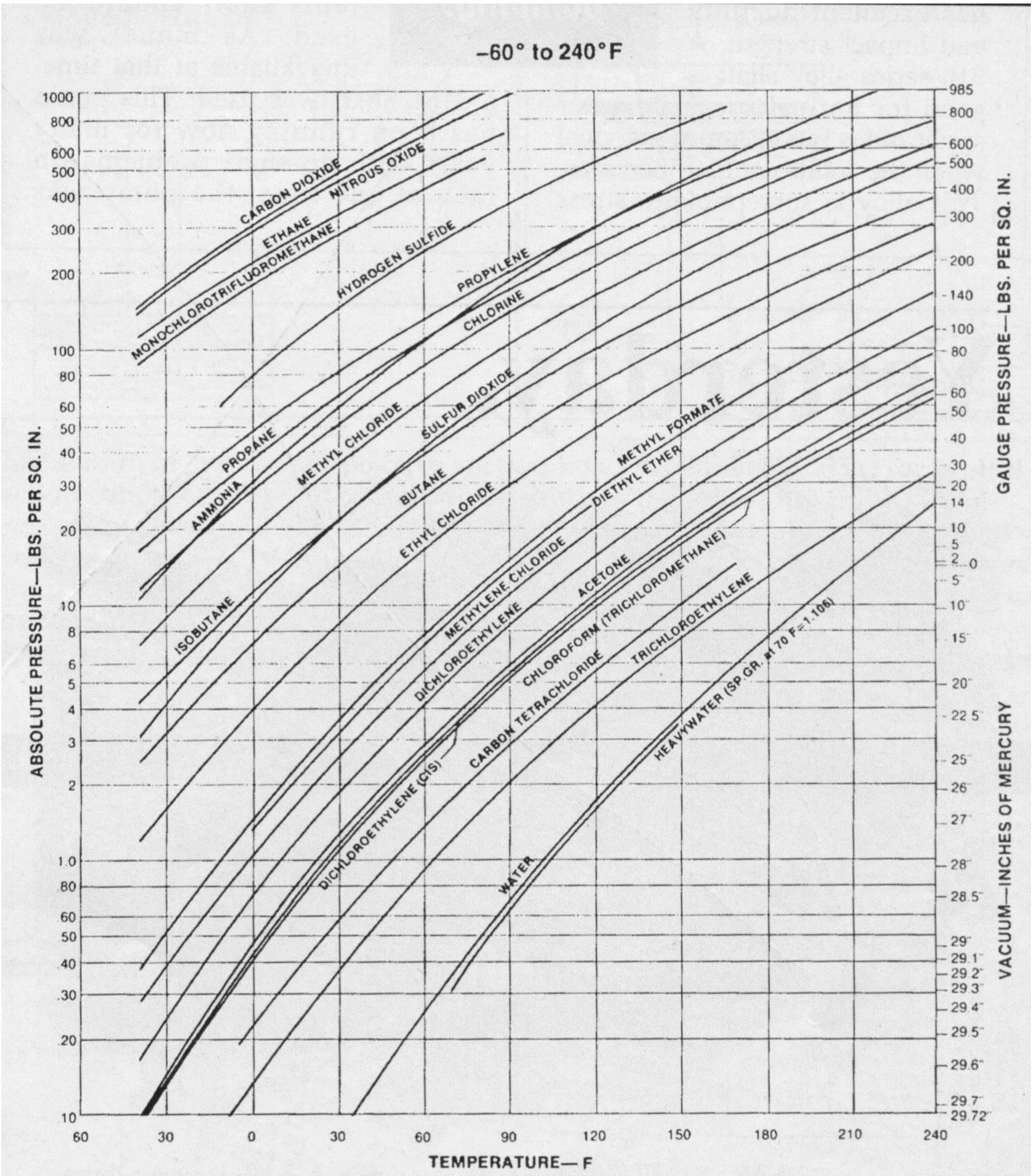

Figure 2. Vapor pressure for various liquids as a function of temperature (Lahr, 1993)

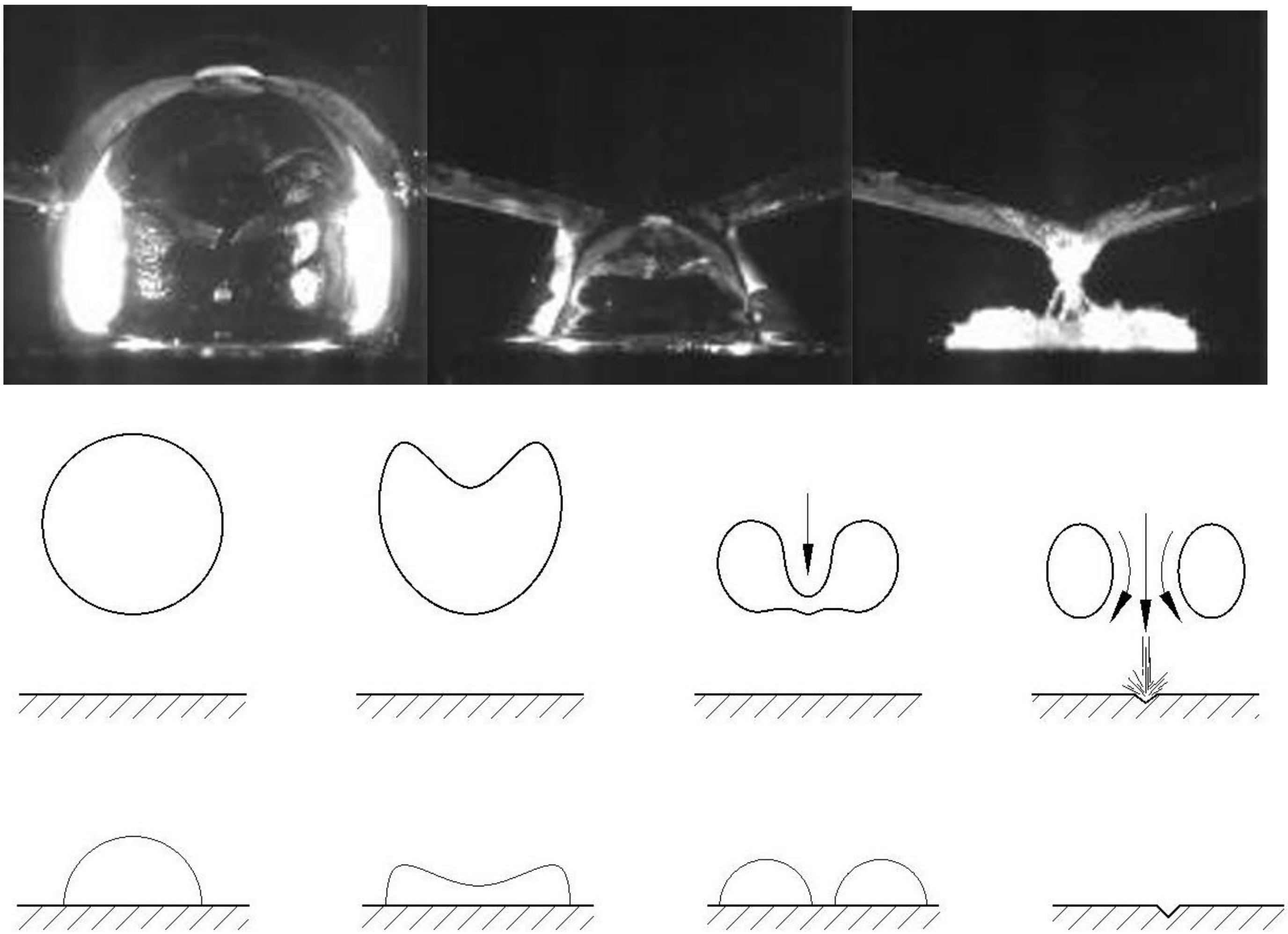

Figure 3. Implosion of a bubble at or near a wall.

a) Upper photograph, high speed video (24000 frames/sec), obtained by Laboratory for Hydraulic Machinery, Ecole Polytechnique Fédérale, Lausanne, Switzerland.
b) Middle picture shows stages of bubble collapse at a short distance from the wall. Collapse occurs away from wall if bubble is not at wall when static pressure increases above vapor pressure of the liquid at the local temperature.
c) Lower picture shows similar stages of bubble collapse if bubble reaches a zone of pressure above vapor pressure at the wall. The jet still forms within the collapsing bubble annulus, but over a typically shorter distance. Either (b) or (c) can produce strong impulse force on the wall, which if strong enough will lead to a stress wave of amplitude higher than the material fatigue endurance limit. After sufficient impacts, cracking initiates and then propagates until material is removed (eroded) from the surface, typically one very small particle at a time.
d) If the bubble collapse is far enough away from the surface, the jet has insufficient impact force to cause material damage, and the cavitation then becomes relevant only with regard to degree of performance degradation, or nuisance “grinding” noise.

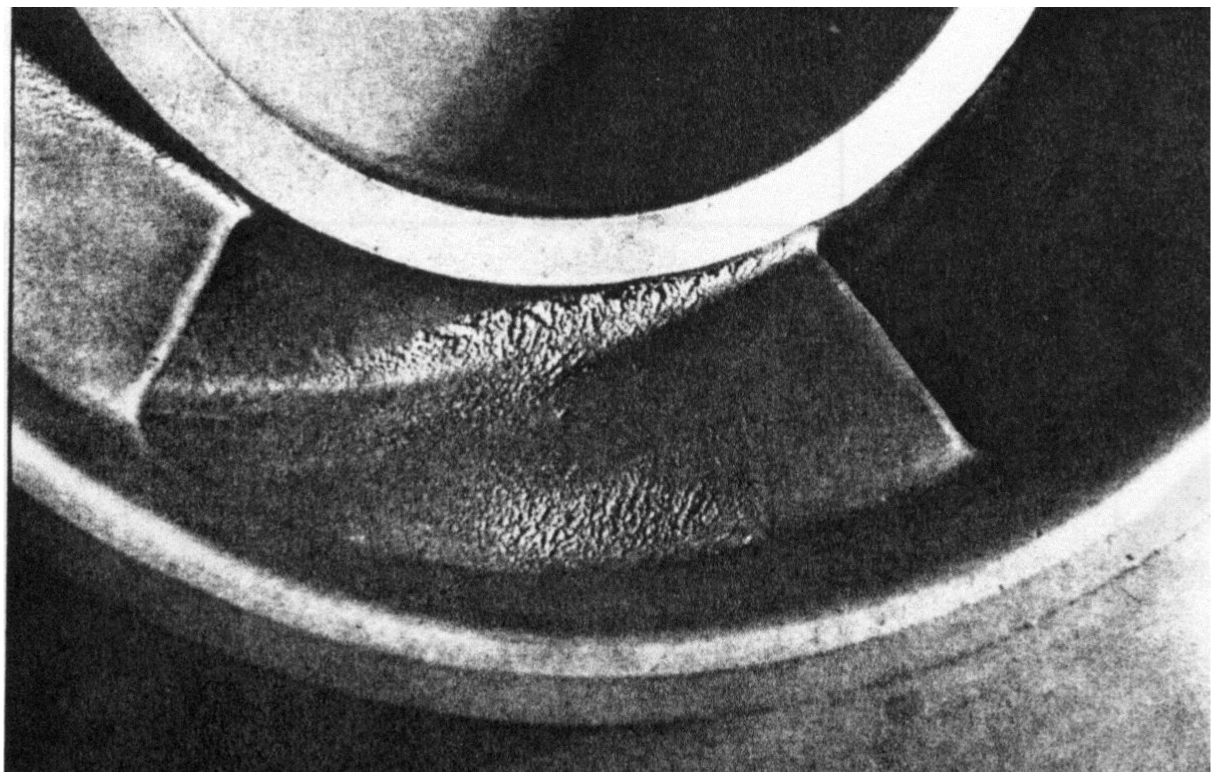

Figure 4. Classical cavitation erosion on the vane of a boiler feed pump (Sloteman, 1995)

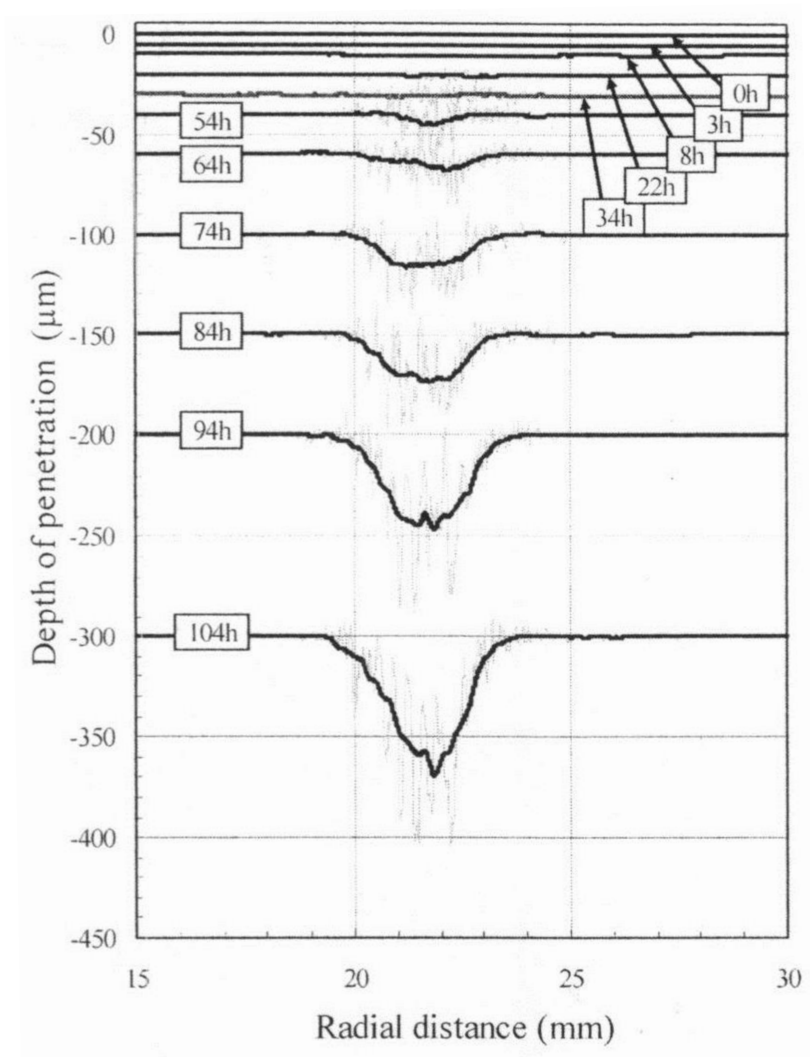

Figure 5. Influence of exposure time on erosion profiles of flat plate specimen. The light gray lines are the raw data with radial step size of 10 microns. The dark black primary lines are the 100 point moving average, taken over a 1mm total length. Note that once the roughly 50 hour incubation period was passed, the erosion rate picked up quickly. Data is from Franc (2009).

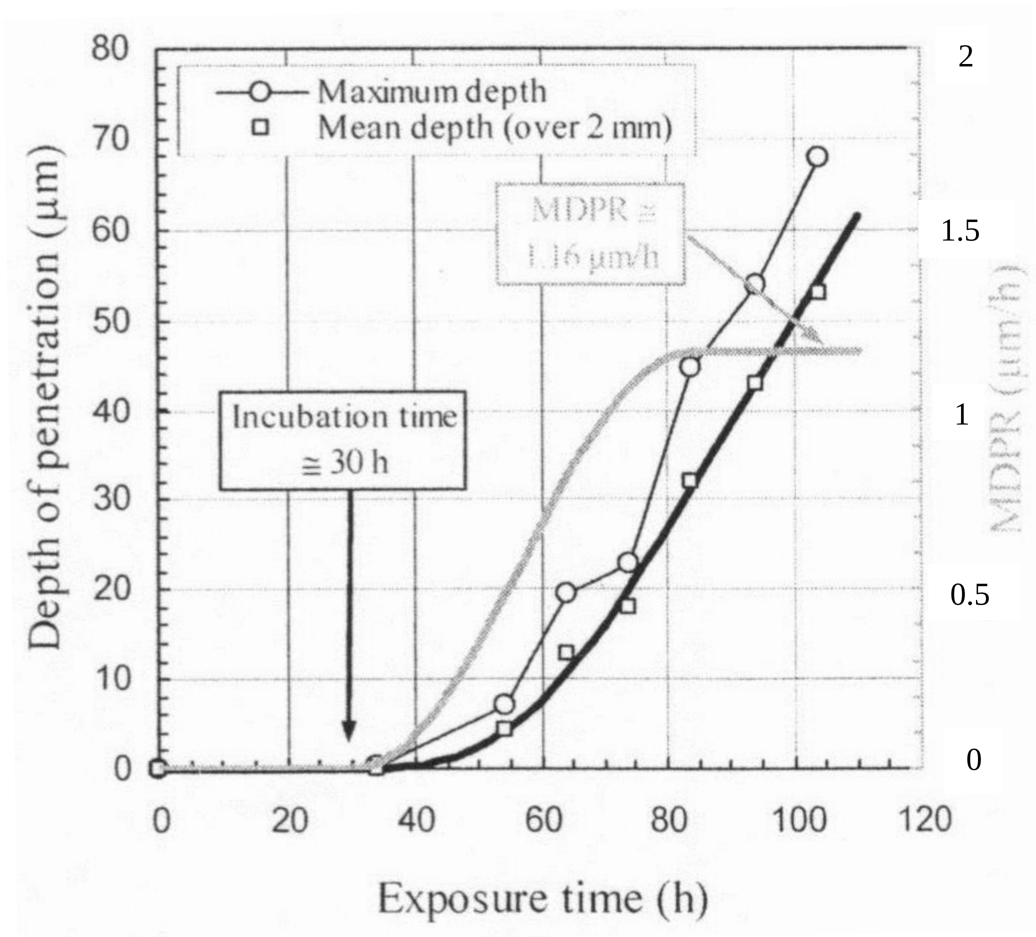

Figure 6. For the test coupon of Figure 5, the mean depth of penetration (black, left scale) and mean depth of penetration rate (gray line, right scale, MDPR in microns per hour, i.e. the slope of the penetration vs. time), both as a function of exposure time. Data is from Franc (2009).

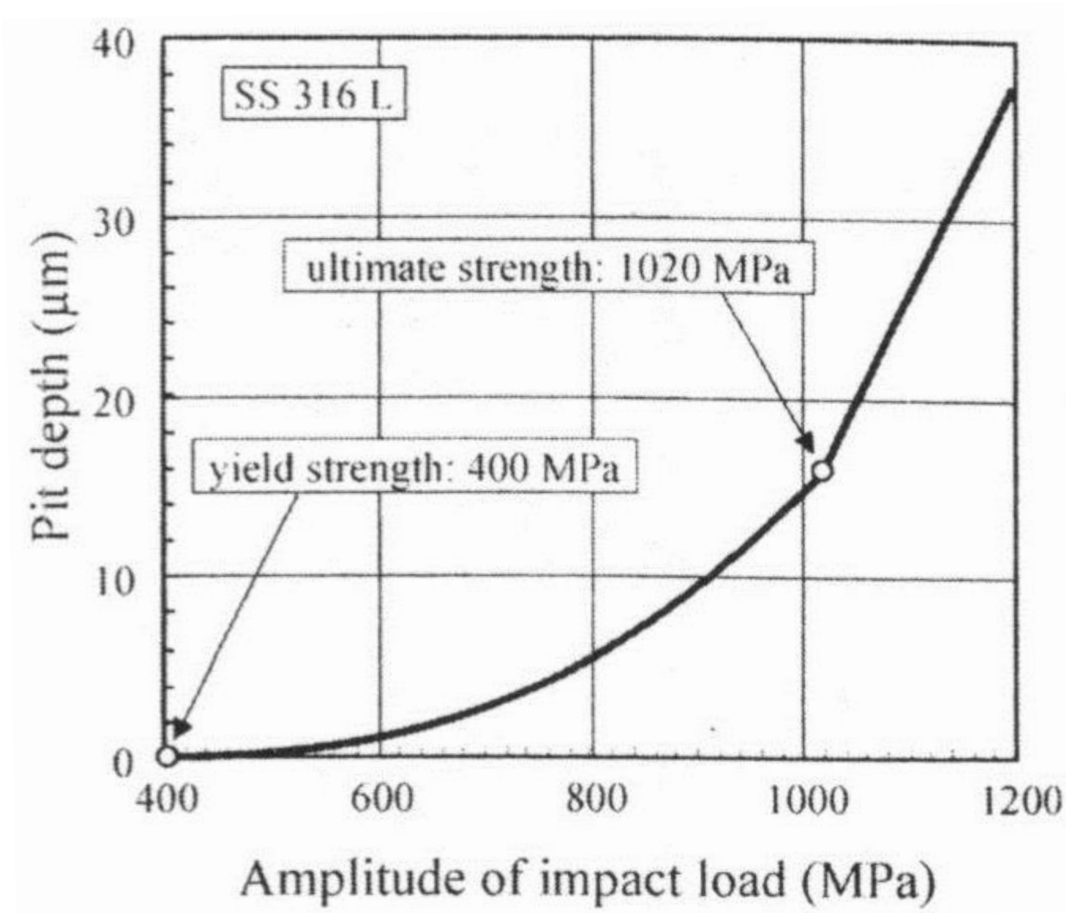

Figure 7. For the test coupon of Figure 5, use of accelerometer data to determine amplitude of bubble implosion impact pressure, and then relating this to pit depth at the conclusion of a 100 hour test. Data is from Franc (2009). The specimen was cold worked 316L stainless steel. Note that in 100 hours, there was negligible erosion if the implosion pressure (as calculated from the accelerometer data using their technique) was less than the material bulk yield strength. 316L is a superior material with respect to cavitation erosion resistance. The Table below compares it qualitatively to other materials used for impellers in various industrial, marine, and military applications.

Resistance to Cavitation	**Metals**
Group I (Best)	Stellite, Cobalt Alloys, 300 Series SS, Super Duplex SS, Inconel, Titanium
Group II (Good)	Ni-Cu-Al (K-500), Ni-Cu Alloy 400, Ni-Al Bronze
Group III (Fair)	Cu-Ni 70/30, Mn Bronze, G & M Bronzes
Group IV (Inferior)	Carbon & Low Alloy (e.g. 4140) Steel, Cast Irons, Aluminum

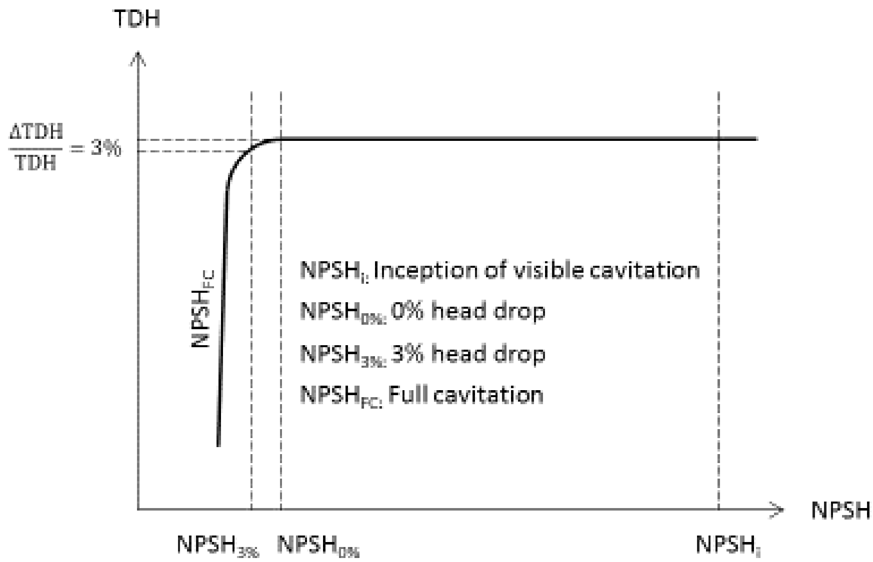

Figure 8. Typical "data-smoothed" NPSHR 3% curve exhibiting discharge head as suction head is reduced to the left. When the head developed drops by 3% on a steady state averaged basis, the Net Positive Suction Head at which this occurs is designated the NPSH3. Note that the head at which cavitation initiates (NPSHi, "incipient" cavitation) is typically at a much higher suction head than NPSH3.

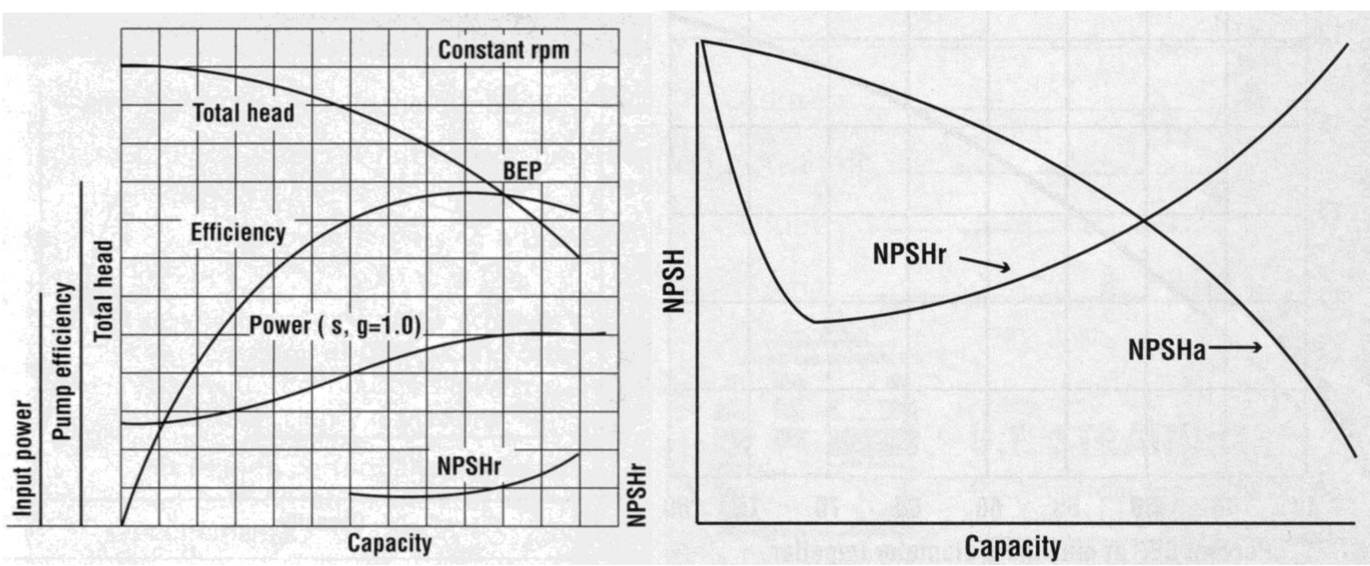

Figure 9. The typical relationship of actual NPSHR 3% head drop and NPSHA to flow capacity in a centrifugal pump, as related to the manufacturer's head vs, capacity and NPSHR published curves. The NPSHR rise on the left side is due to suction recirculation. Originally presented in Dec. 1996 Chemical Processing magazine by the Hydraulic Institute.

Figure 10. PCB Piezotronics Model 111A26 General Purpose ICP Pressure Transducer
+/- 500 psi range, 400 kHz frequency response

Figure 11. PCB Piezotronics Model M353B17 High Frequency, Quartz Shear ICP Accelerometer
+/- 500 G's range, 30 kHz to 3 dB roll-off point, useful (but distorted) output to 100 kHz

Blade surface coefficient　　Pressure　　Gas content

$$ER_p = C_L\left(\frac{L_{cav}}{L_{cav\ ref}}\right)^x \frac{(p_0 - p_v)^3 F_{cor}}{R_m^2 F_{mat}}\left(\frac{\alpha_{ref}}{\alpha}\right)^{0.36} \qquad (1)$$

Cavitation length　　Material properties

where,

ER_p	:	Predicting erosion rate at maximum damage point	(m/s)
C_L	:	Constant of suction side $= 2.3\times10^{-12}$	(m^3/N/s= m/s/Pa)
		Constant of pressure side $= 1.1\times10^{-10}$	(m^3/N/s= m/s/Pa)
L_{cav}	:	Cavitation length measured from leading edge (Distance from leading edge of blade to end of damaged area is defined as cavitation length here.)	(m)
$L_{cav\ ref}$	:	Reference value of cavitation length L_{cav}(0.01 m)	(m)
x	:	Index of suction side = 2.83 / Index of pressure side = 2.6	
p_0	:	Static pressure at impeller inlet	(Pa)
p_v	:	Saturated vapor pressure of water	(Pa)
R_m	:	Tensile strength of material (impeller)	(Pa)
F_{cor}	:	Corrosion factor which corrosion exerts on erosion (Table 1)	
F_{mat}	:	Material factor which material exerts on erosion (Table 1)	
α	:	Gas content of water	(ppm)
α_{ref}	:	Saturated gas content of water	(ppm)

Table 1 Factor of Eq. (1)

Material	F_{mat}	F_{cor}	
		Feedwater, Drinking water	Seawater
Ferritic steels	1.0	1.0	1.5
Austenite steels	1.7	1.0	1.3

Figure 12. A description of the EPRI-GS-6398 report concerning prediction of cavitation severity. Excerpted from: The Research Committee on Cavitation Erosion, Turbomachinery Society of Japan, April 2008, “Comments on NPSH40K and Flow Visualization”, ISO/TC115/SC3 JWG for ISO 13709 (API-610), New Orleans LA. This was designated “informative” (suggested) rather than “normative” (mandatory) in ISO 13709, with regard to prediction of 40000 hour minimum impeller life relative to cavitation damage. The reason for the informative designation was that the life prediction accuracy was found low.

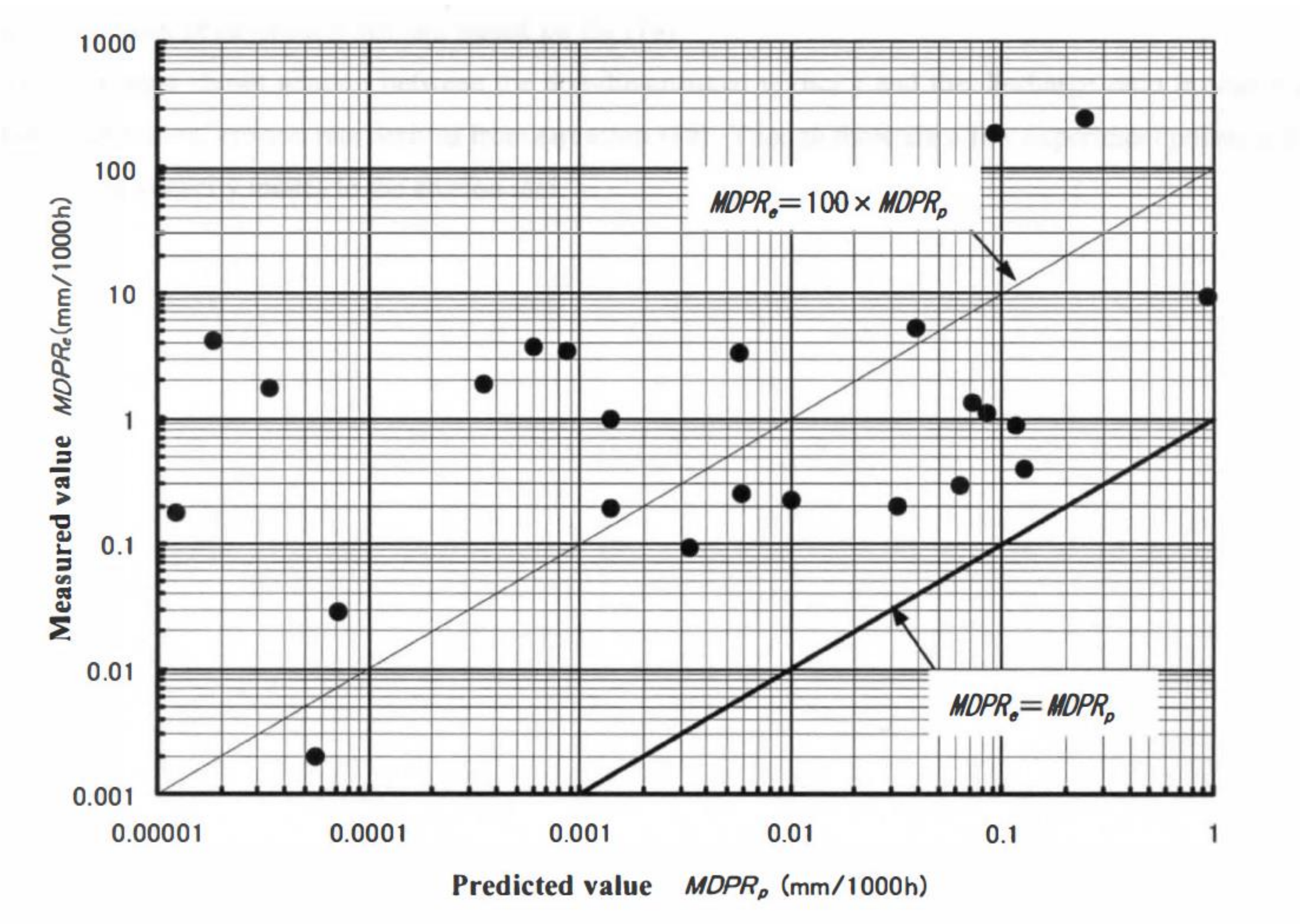

Figure 13. Correlation Plot of Accuracy of Measured vs. Predicted Values of Cavitation Damage (Maximum Erosion Depth of Penetration Rate of Damaged Region "MDPR", Based on EPRI Equation (Source: Turbomachinery Society of Japan, Guideline for Prediction and Evaluation of Cavitation in Pumps, TSJ G 001, Jan. 20, 2003). Note that the correlation is much less accurate than desired, and that the measured value is typically much higher than the predicted value.

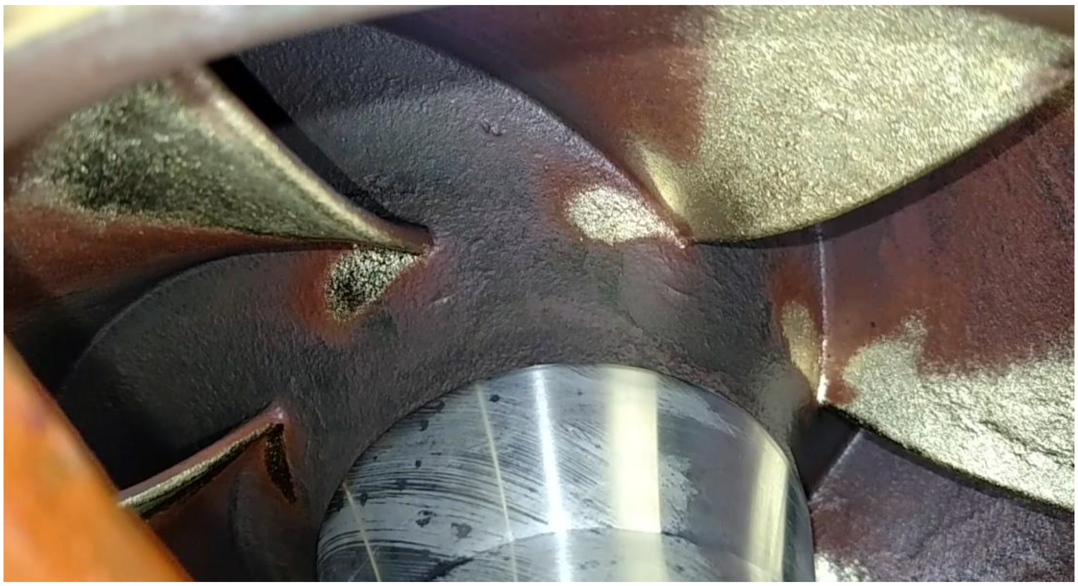

Figure 14. Pump A impeller picture. Note the polished surface of the low pressure side of the vanes which extends onto the hub.

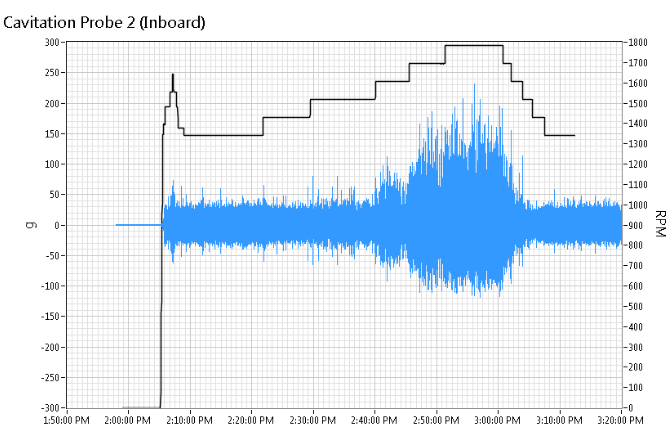

Figure 15. Pump A high frequency accelerometer time domain response. Note the g levels exceeded 200 g's as the pump went higher in speed, when NPSHR increased and NPSHA reduced.

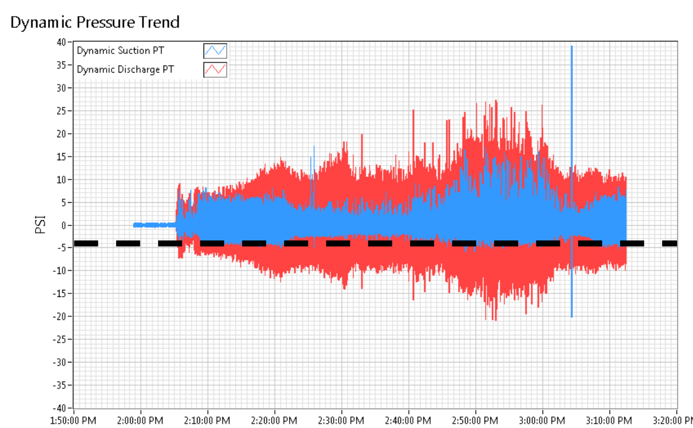

Figure 16. Pump A dynamic pressure transducer time domain response. Note the negative psig spikes on the suction transducer get clipped at just below -5 psi (except for the one anomaly near the end of the test). This shows the pump was experiencing cavitation and the pressure was dropping below the vapor pressure.

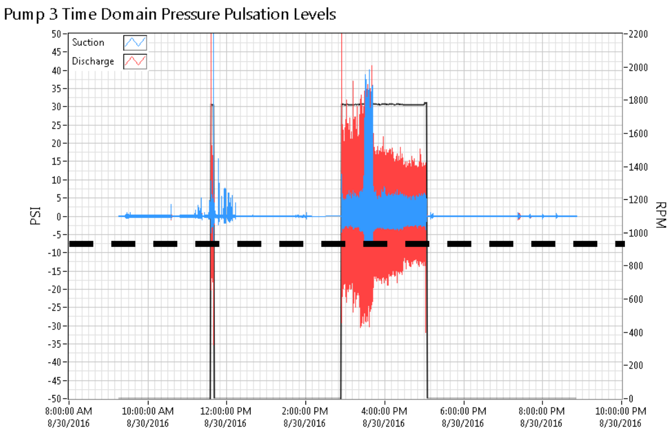

Figure 17. Pump B dynamic pressure response. Negative spikes clipped off because the suction pressure level fell below the vapor pressure of the water, indicating cavitation.

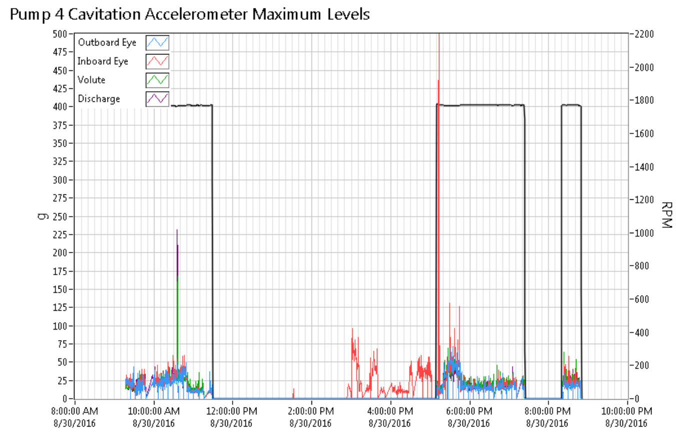

Figure 18. Pump B casing high frequency accelerometer response

Figure 19. Pump B impeller exhibits cavitation damage from discharge recirculation (yellow).

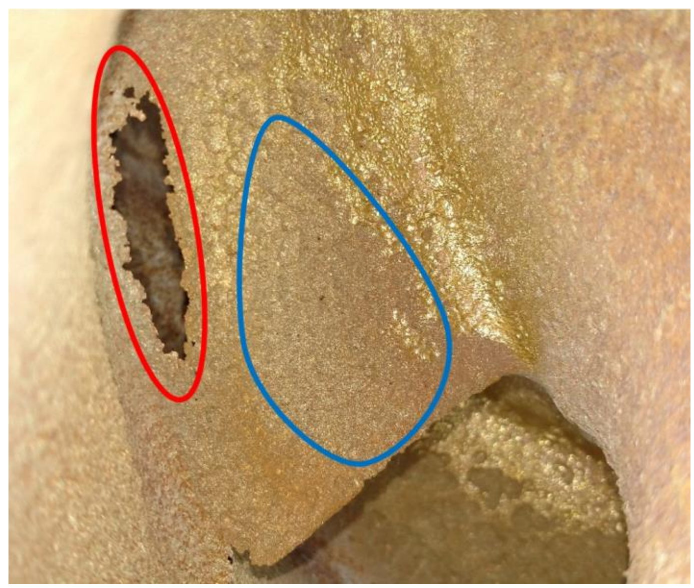

Figure 20. Pump B The red shows damage of suction recirculation. The telltale that the damage is mainly due to recirculation are the holes and pitting on the pressure side of the vane and/ or shroud. Note that suction and discharge recirculation often are observed together. There is also some typical suction-side cavitation phenomena damage. This is shown by the discoloration in the area circled in blue.

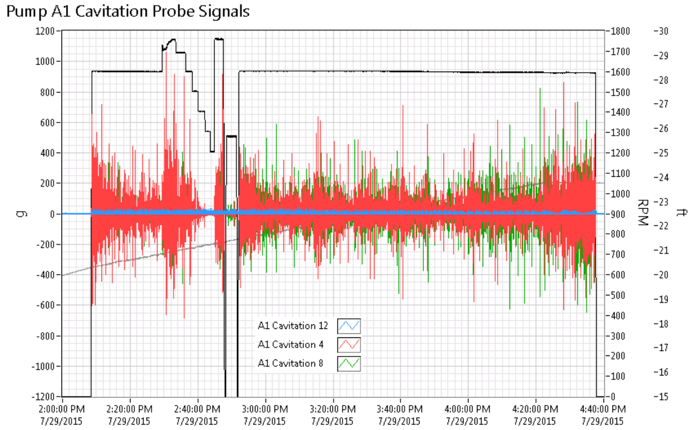

Figure 21. Pump C (axial flow circulator pump) accelerometer g level over time during operation. As can be seen, the g levels exceeded what the authors consider damaging levels, and are well above 100. The accelerometer was placed on the pump suction piping near the impeller, radial with respect to the impeller eye due to pump type.

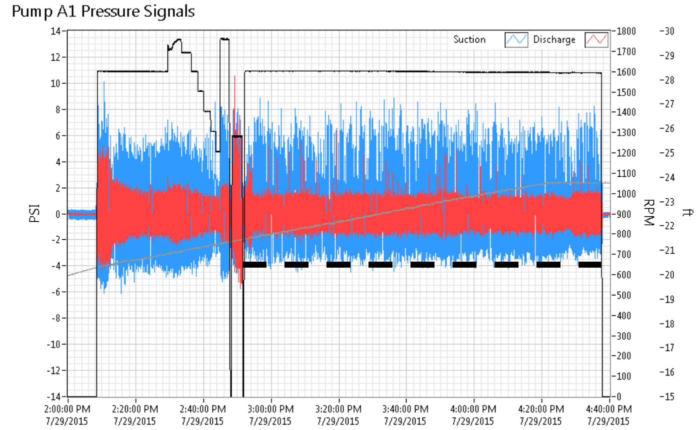

Figure 22. Pump C time waveform from dynamic pressure transducers indicated that localized cavitation was present at the suction end of the pump (+8/-4 psig). The negative spikes (clipped at -4 psig) were clipped off, which is indicative that instantaneous suction pressure was below the vapor pressure. The discharge end did not show large pressure pulsations (±2 psi).

Figure 23. Pump C, used pump impellers (4000 hours of service). In both cases, large amounts of material were removed from the impeller due to the cavitation measured by the authors

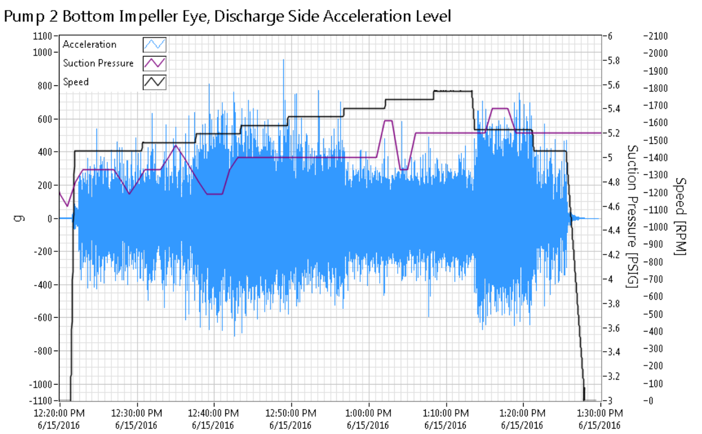

Figure 24. Pump D accelerometer g level over time during operation. As can been, the g levels exceeded what the authors consider damaging levels, and are well above 100. The accelerometer was placed on the pump volute casing near the impeller eye.

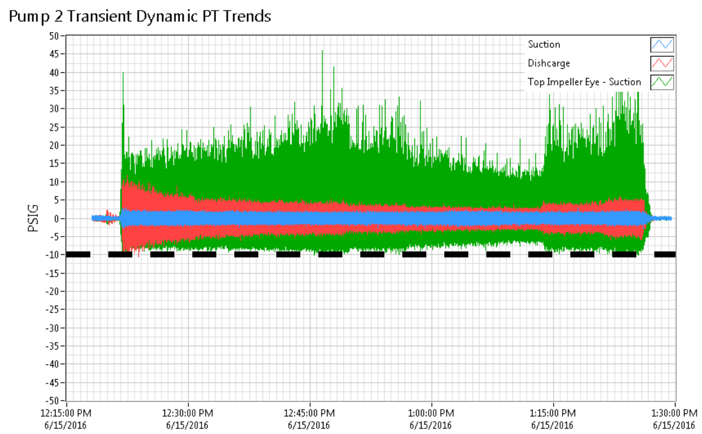

Figure 25. Pump D dynamic pressure time domain signal. The negative spikes at the "Top Impeller Eye" location were clipped off because the suction pressure fell below the vapor pressure of the water. This is a reliable indicator of cavitation phenomena.

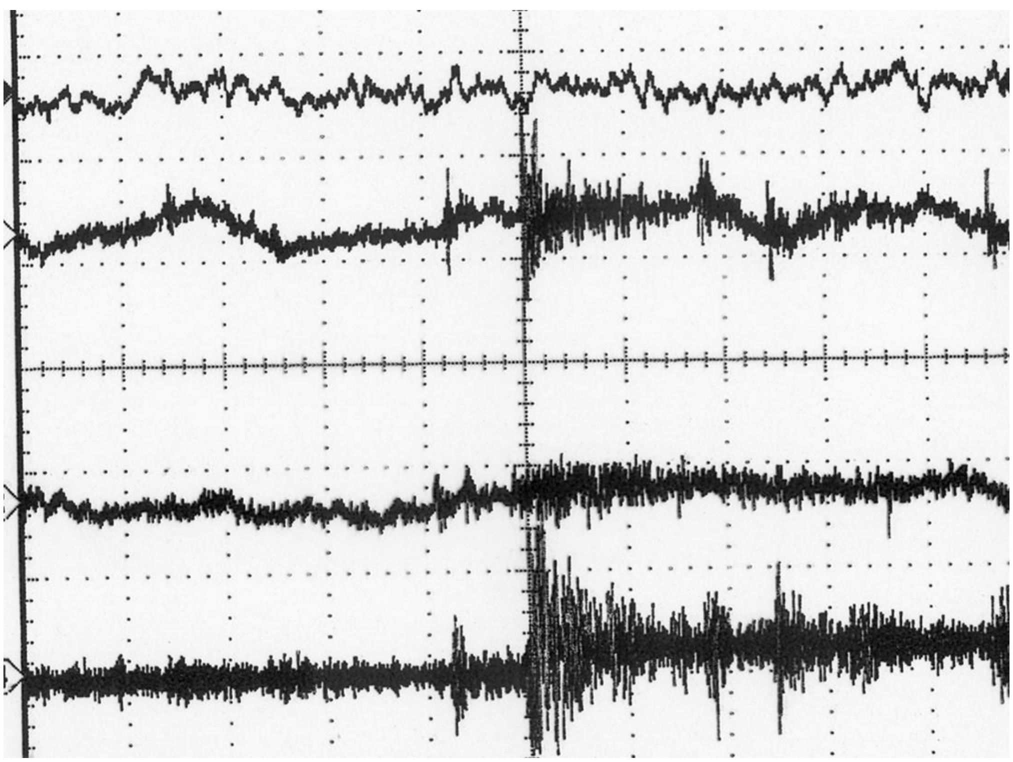

Figure 26. Pump E: Double suction pump instrumented in each bay with multiple sensors types. 316L stainless steel impeller eroded 0.2 inch deep in 1000 hours of operation, in service in a water plant.

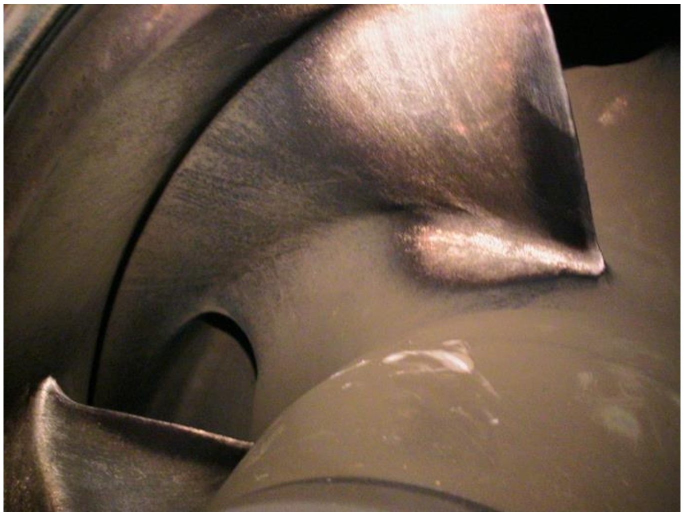

Figure 27. Pump F impeller, view is looking into the outboard impeller suction. Note the polished surface of the low pressure side of the vanes which extends onto the hub.

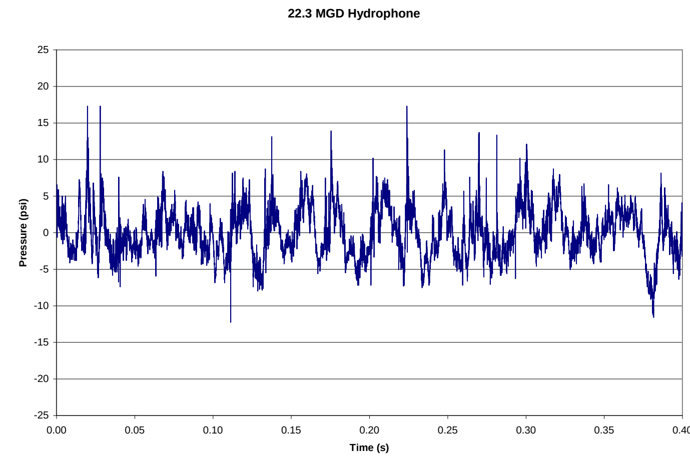

Figure 28. Pump F hydrophone measurement in the inboard suction chamber when operating with a full open discharge valve. The hydrophone was located approximately 8 to 12 inches in front of the inlet of the impeller, with line-of-sight. This time waveform shows sudden impulses which are related to the cavitation superimposed on low frequency oscillations.

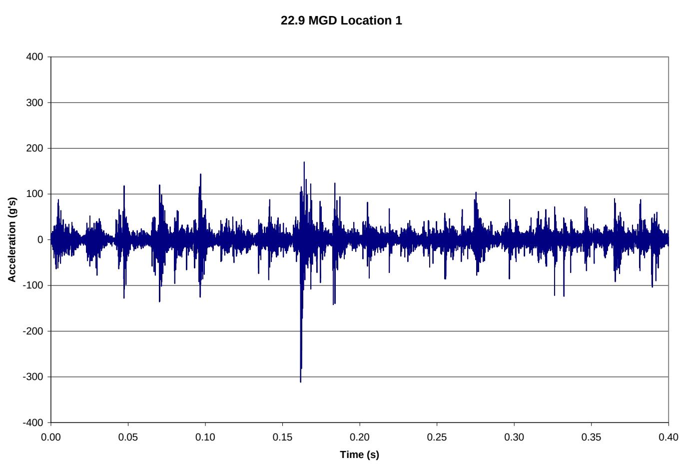

Figure 29. Pump F Structural vibration trace (amplitude in g's) taken on the suction of the casing while operating with full open discharge valve. Note that in this 0.40 second sampling time, the peak amplitude exceeded 100 g's for nine events, and exceeded 300 g's once. Note how much more evident the cavitation spikes are than they are in the hydrophone data.

Figure 30. Pump F also exhibited cavitation damage on the wear ring exit (above), and the volute tongue (below). Accelerometer measurements on the casing indicated local acceleration peaks in these areas. The tongue erosion likely occurred due to discharge recirculation, since this pump often ran at 60% BEP flow.

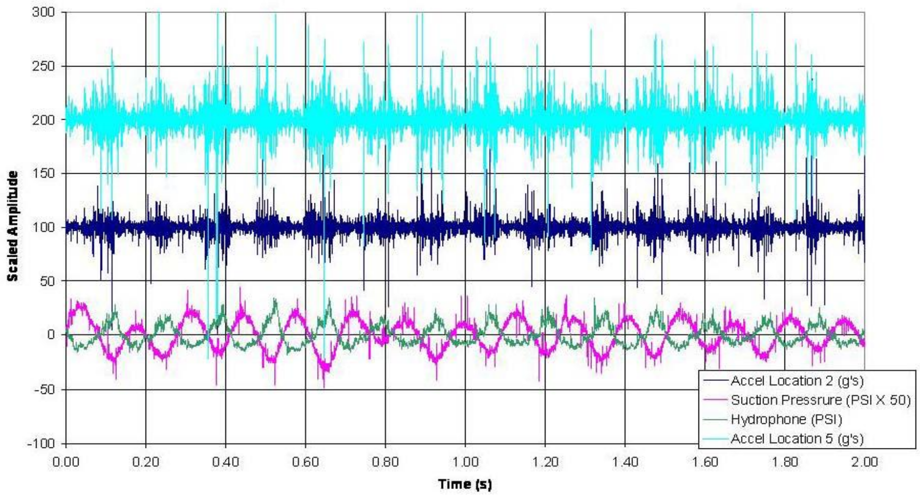

Figure 31. Pump F, exhibiting a phenomenon known as "cavitation surge". Accelerometer readings at two locations on the suction bay are compared to output of a suction pressure piezoelectric transducer, and a line-of-sight hydrophone. Cavitation surge involves the cyclic occurrence of classical cavitation the vane passages, generally at the rate of 5 Hz to 20 Hz. The passages fill with cavitation bubbles, restricting further flow, which then reduces dynamic pressure drop. In response, static pressure in each vane passage increases above the vapor pressure, eliminating the cavitation and permitting full flow to develop again. The velocity associated with the full flow reduces the suction pressure below the vapor pressure in the vane passage, and cavitation returns, blocking the passage. This cycle continues with an audible chugging noise. The authors have encountered this unusual phenomenon primarily in double suction pumps.

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