# MODERN CAVITATION CRITERIA FOR CENTRIFUGAL PUMPS

by

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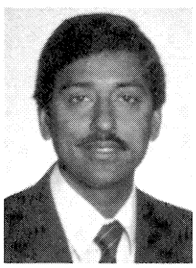

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## ABSTRACT

Historically, the NPSH requirement of centrifugal pumps has been based on head degradation. Recently, it has become quite clear that this criterion is inadequate for satisfying longlife requirements of high energy pumps. As a consequence, both the user and the manufacturer are interested in establishing more realistic criteria for determining acceptable NPSH requirements. Various criteria and empirical methods that have been proposed for establishing the longlife NPSH are examined. Even though none of the techniques investigated have achieved complete acceptance, some show great promise for eventual usage by the pump community.

## INTRODUCTION

It is well known that the performance of a rotordynamic pump deteriorates when the suction pressure is reduced below a certain value. The suction pressure is usually qualified in relation to the vapor pressure at the suction temperature and is represented by the term Net Positive Suction Head (NPSH). When the NPSH is sufficiently reduced, local pressures at the impeller inlet can fall below the vapor pressure, thus causing vapor bubbles to form. These vapor volumes can block the available passage area and cause a reduction in the pump head and efficiency. At sufficiently low NPSH values, cavitation can have advanced to such an extent that the flow area is completely blocked and the pump is unable to develop head. This performance break-off has been recognized for a long time and, therefore, specifications are established to ensure that the pump can operate satisfactorily (meaning without break-off) at the values of NPSH available at the site. Typically, the specification is stated in terms of the NPSH needed to prevent a degradation in head greater than a specified value. A value of three percent is very common.

This NPSH requirement has been found to be satisfactory in a majority of pump installations. Recently, however, the trend towards increasing the power density of pumps has shown that even when the available NPSH is well in excess of the NPSH required for a three percent head drop, significant cavitation damage can occur. Along with increases in the running speed, operation at off-design conditions has been found to exacerbate this problem. Evidently under such circumstances, NPSH specifications based only on head degradation are unsatisfactory. Both the pump manufacturer and the user are interested in establishing more realistic cavitation criteria to ensure long life.

Some of the criteria that have been proposed for different services are reviewed. They go from the extreme of no visible cavitation for very severe service to a small margin over break-off for light duties with fluid properties also considered. In some cases, cavitation criteria are specified in terms of suction specific speed. More recently, experimental work has been undertaken to establish the severity of cavitation based on short term observations. Some of these methods are reviewed. Some remarks on empirical methods for predicting NPSH requirements for long life are included.

## CRITERIA BASED ON INCEPTION

An obvious way to ensure that there will be no cavitation damage is to allow no bubble formation. Generally, this will imply a large, slow speed pump, and would not be justifiable except for severe services where long term survivability is of great importance, such as liquid sodium coolant pumps for a fast breeder reactor.

The selection of the inlet geometry of the impeller to ensure no bubble formation can be made on the basis of theoretical procedures. Since bubbles first form when the surface pressure reaches the local vapor pressure, one can predict inception NPSH if surface pressures can be accurately calculated. There are several computer based methods available for calculations of the pressure within the impeller. Generally, these methods are quasi-three-dimensional [1] and fairly cumbersome to use. Also, these quasi-three-dimensional methods generally do not provide sufficient resolution to ascertain local velocity peaks which are very important in cavitation predictions. Consequently, high resolution, two-dimensional methods are often more useful. One such method is that originally developed by Martensen [2], and later programmed for the digital computer by Jacob [3]. In this method, a set of vortex singularities is distributed on the airfoil surface. The resulting potential flow velocity field is calculated by imposing the boundary condition that there be no velocity normal to the surface. Further, the singularity distribution is made unique by utilizing the Kutta condition in which the flow leaves the trailing edge smoothly. These conditions are satisfied numerically by replacing the integral equation by a finite sum for a well-selected number of points on the airfoil surface. By solving the resulting system of simultaneous linear equations, the vortex singularity and, hence, the pressure distribution can be established.

The value of NPSH required for bubble inception can be determined from the pressure distribution by assuming that bubbles form as soon as the local pressure reaches the vapor pressure. Assuming that the relative total pressure remains the same between the pump suction and the local cavitating zone, the result is:

$$\frac{1}{\rho}\, p_\ell + \frac{W_\ell^2 - U_\ell^2}{2g} = \frac{1}{\rho}\, p_1 + \frac{C_{m_1}^2}{2g}$$

where

- $p$ = static pressure
- $W$ = relative velocity
- $U$ = blade peripheral velocity
- $C_m$ = meridional velocity
- $\rho$ = fluid density
- $g$ = acceleration due to gravity

and subscript:

- $1$ = pump suction
- $\ell$ = local, on the blade surface

Putting $p_\ell$ = vapor pressure, $p_v$, the result is

$$\frac{1}{\rho}(p_1 - p_v) + \frac{C_{m_1}^2}{2g} = \text{NPSH} = \frac{W_\ell^2 - U_\ell^2}{2g}$$

With no prewhirl upstream,

$$U_\ell^2 = W_1^2 - C_{m_1}^2$$

$$2g(\text{NPSH}) = W_1^2 \left\{ \left(\frac{W_\ell}{W_1}\right)^2 - 1 \right\} + C_{m_1}^2$$

The ratio $W_1/W_1$ is determined by the Jacob computer program for any given inlet flow angle.

The accuracy of the predictions can be judged by comparing results with actual inception test data on a single airfoil (NACA $66_1$-012). The profile and aerodynamic data are given by Abbott and von Doenhoff [4] and the cavitation data are given by Kermeen [5]. The comparison between test data and the Jacob computer program plotted in terms of the cavitation number K is shown in Figure 1, where K is defined as:

$$K = 2\,\frac{p_1 - p_v}{\rho V_1^2}$$

where

- $p_v$ = vapor pressure
- $V_1$ = upstream velocity

It can be seen that the agreement is quite satisfactory over a range of angle of attacks.

A comparison of the measured inception NPSH versus predicted values as a function of flowrate for a mixed-flow pump impeller is shown in Figure 2. As this is an open impeller, cavitation could be observed on both the pressure and suction sides of the vanes. The meridional profile of the impeller is also shown in Figure 2, along with pertinent dimensions. The comparison between theory and test data is very good for the NPSH levels, although the point of shockless entry occurs slightly to the left of the predictions.

It should be pointed out that the Jacob program is applicable only for two-dimensional flow. The impeller shown in Figure 2 has a shroud stream surface that is close to being cylindrical in the inlet portion. Since cavitation inception occurs only in the inlet region in centrifugal pumps, the Jacob program has wide

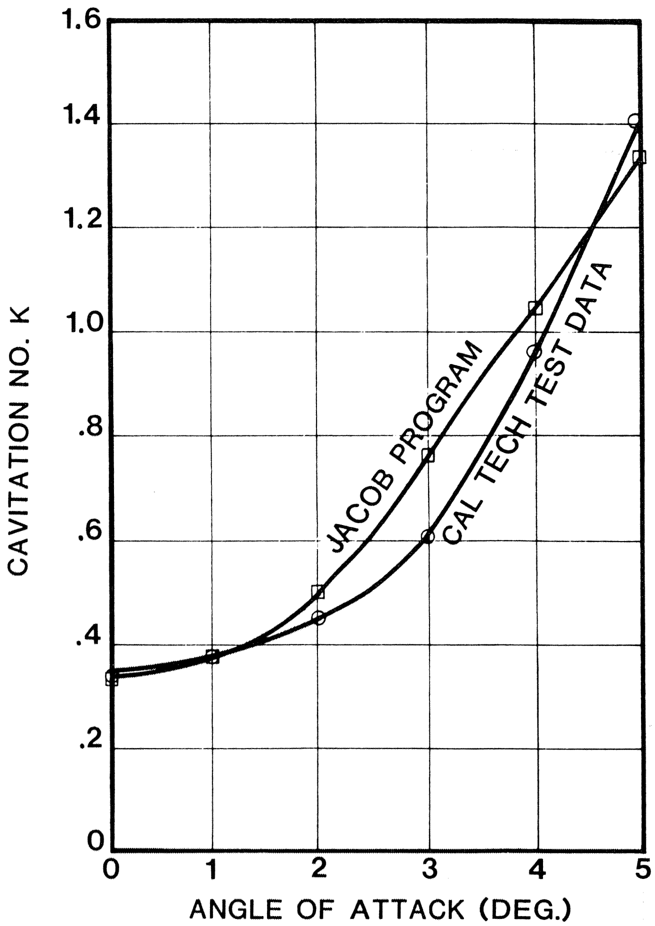


*Figure 1. Comparison between Measured and Predicted Cavitation Inception Numbers on NACA $66_1$-012 Airfoil (Test Data from [5]).*

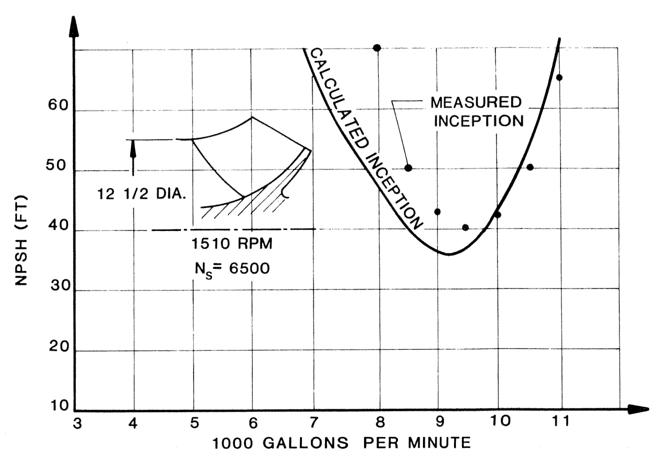


*Figure 2. Comparison between Measured and Predicted NPSH Values for Inception on Mixed Flow Pump Impeller.*

applicability. However, for low specific speed impellers, computations will have to be carried out using methods which allow for a radial shift of the streamlines and varying stream filament thickness.

As has been mentioned previously, the criterion of no bubble formation will lead to large, low speed pumps. These may be economically feasible only for very severe or unique applications. However, it is still important to determine the inception NPSH values, as these can be used to ascertain the allowable margin. This will be discussed further under RADIOACTIVE METHODS.

## CRITERIA BASED ON PAINT REMOVAL

In the vast majority of commercial pump installations, it would not be necessary to absolutely eliminate cavitation. A certain volume of collapsing bubbles can be tolerated. A number of methods are being pursued to determine the degree of acceptable cavitation. A method that has long been investigated is the removal of a soft coating from the impeller passages. Most of the published work on this subject have come to the conclusion that this method is a good indicator of where cavitation damage, if it occurs, is likely to take place. Conclusions as to the rate of damage cannot be drawn from paint removal [6, 7].

Experiments on paint removal have been conducted with a number of paints and coating materials. The material generally preferred is either Roller System Ink S-1 (Black) (a stencil ink manufactured by Diagraph Bradley Industries) or Magic Marker Red Ink. It is believed that if the coating is not removed during a cavitation test of 15 to 30 min duration, the impeller is unlikely to undergo significant damage during actual operation. Unfortunately, paint removal cannot be correlated to actual damage.

## CRITERION BASED ON CAVITY LENGTH

Recently, suggestions have been made that damage rate can be correlated with the extent of the bubble cavity on the visible or "non-working" surface of the vane under cavitating conditions. A physical interpretation for such a relationship to exist may be expressed as follows:
The intensity of damage can be expected to be proportional to the pressure at which bubbles collapse on the surface. It has been shown [8] that

$$p_i \propto p_o \left( \frac{R_o}{R_c} \right)^2$$

where

$p_o$ = initial bubble pressure
$p_i$ = impact pressure
$R_o$ = initial bubble radius
$R_c$ = final collapse radius of bubble

The bubble size just before collapse ($R_o$) depends upon the time available for the bubble to grow. The growth time is proportional to the cavity length and inversely proportional to the translational velocity of the bubble. Thus, $R_o \propto \ell / V_o$ where $\ell$ is cavity length, and $V_o$ is approximately equal to the local fluid velocity. From this it can be shown that the intensity of erosion, I, is proportional to the square of the cavity length. Thiruvengadam [9] has developed non-dimensional expressions for the intensity of spherical bubble collapse as well as jet impact cases. His expression for the jet impact case is:

$$D \propto \frac{1}{S} \frac{1}{2} \rho V_o^3 \frac{\ell}{d} K \sqrt{\Delta K} \exp \left[ \frac{-2.67}{W \Delta K} \right]$$

where

D = damage rate
S = erosion strength of material
d = average nuclei size
$\Delta K$ = difference between the operating cavitation number and inception cavitation number
W = Weber number expressed as

$$W = \frac{1}{2\gamma} \rho V_o^2 d$$

$\gamma$ = surface tension

From the above expression, it can be seen that for a given liquid and operation at a given cavitation number, the damage rate depends on the cavity length and fluid velocity. Correlation of the measured bubble length with observed damage in the field has been made [10]. The data show that the product of the cavity length times the peripheral vane tip velocity at the impeller eye is a good indicator of the number of hours the pump can operate without appreciable damage. It has been suggested by several authorities that a cavity length of 16 mm is an acceptable limit for eye velocities of about 200 ft/sec.

Measurement of the bubble length can easily be made in a flow visualization rig where the impeller eye can be viewed through a plexiglass cover. A typical rig setup is shown in Figure 3 and a cross-sectional view of the arrangement is shown in Figure 4. It can be seen that the rig is set up to simulate the inlet configuration of between-bearing pumps, in that it has a shaft through the eye and a top suction arrangement. It is important for a meaningful flow visualization test that no free air bubbles enter the impeller. To measure cavity length, it is convenient to make grid marks on the vane surface. The measured cavity length contours for various flows and NPSH values on the model of a boiler feed pump impeller are given in Figure 5. The eye diameter of the impeller was 6.5 in and cavity length measurements were made at 1800 rpm. From the data in Figure 5, one can see that the shockless entry flow is about 1200 gpm and recirculation onset is at about 600 gpm.

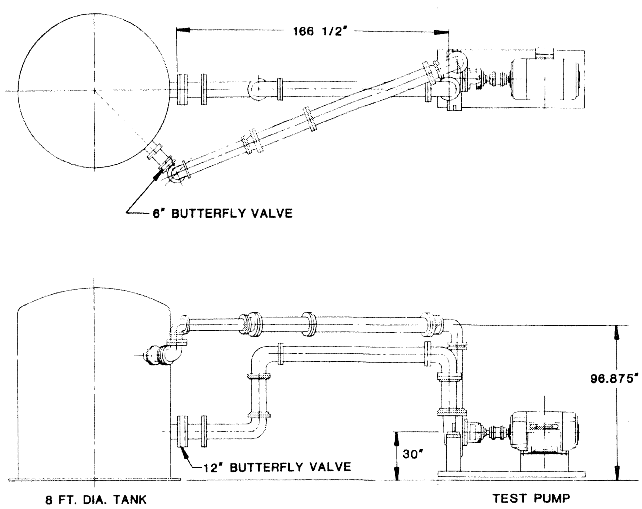


*Figure 3. Flow Visualization Loop for Single Suction Impeller.*

It must be pointed out that even though the cavity length method is easy to adopt, there are some significant shortcomings. Unless extensive field data have been acquired, it is not possible to determine the permissible cavity length. Further, as collapse may occur away from the vane surface, large cavity

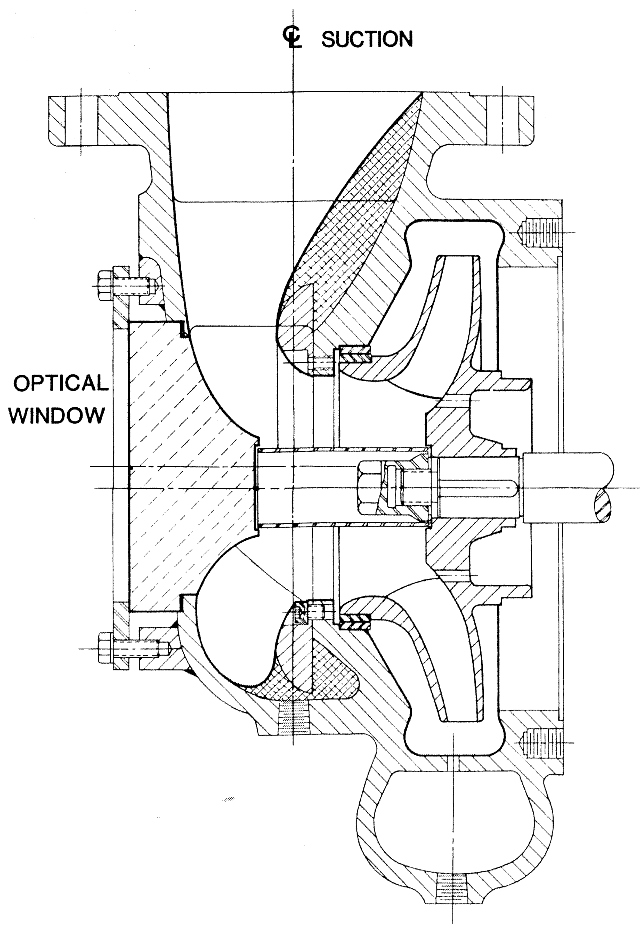


*Figure 4. Cross-sectional View of Model Impeller for Flow Visualization. Eye Diameter 6.5 In; 1800 RPM.*

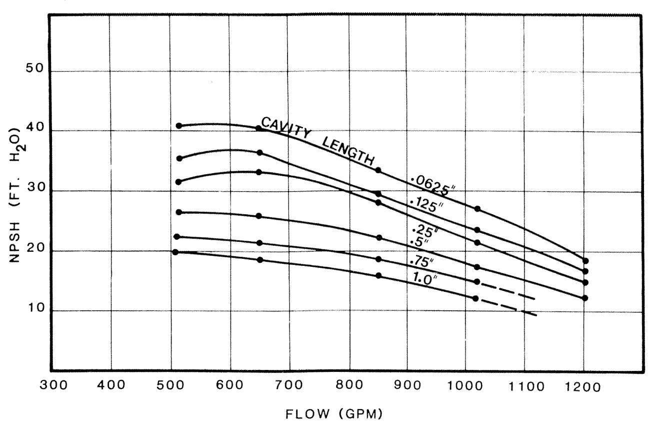


*Figure 5. Contours of Constant Cavity Length on Impeller of Figure 6.*

lengths would in some instances be acceptable. For example, super-cavitating propellers with the cavity length exceeding the blade chord provide acceptable service life. Finally, since "back-side" or working surface cavitation cannot generally be visually observed, conclusions which are too optimistic may be drawn.

It was stated in the beginning of this section that the physical basis for the cavity length idea is that the collapse pressure increases with the cavity length. This assumes that the bubbles travel along the full length of the liquid-to-vapor interface and that they collapse at the cavity closure point. There are several circumstances under which this assumption may not be true:

1) There are no bubbles along the interface.

2) The cavity closure may not be well defined and fresh bubbles may be generated there (and not upstream), as a result of the vorticity due to the break-up of the cavity and the sudden diffusion of the liquid as it flows past the cavity closure point.

3) Cross flows sweep the trailing portion of the cavity and transfer the bubbles generated (as in 2) into the interior of the passage, where collapse may be harmless. *(The author is indebted to Dr. Paul Cooper of Ingersoll-Rand Research, Incorporated, for these observations).*

For these reasons, it is not clear that the damage rate must necessarily be related to the cavity length. Indeed, the variety of two-phase flows that may exist in a pump makes it unlikely that the cavity length alone is a true determinant of the damage rate. Therefore, it is important that other erosion prediction techniques be considered. In the following sections, new methods that may be useful on short-term testing and empirical methods of prediction will be described.

## ACOUSTIC CRITERIA

The collapse of cavitating bubbles generates noise which can be picked up with casing mounted transducers. The acoustic emission spans a very large frequency band from about 10,000 Hz to 120,000 Hz. Typical noise signatures are shown in Figure 6 for a 5 in diameter inducer pump running at 3600 rpm. The transducer was mounted on the suction pipe at about one diameter upstream of the inducer. The top portion of Figure 6 shows the head degradation with NPSH, while the bottom portion shows the variation of noise filtered into different frequency bands. It is interesting to notice that the high frequency noise shows a gradual increase as NPSH is decreased, culminating in a sharp increase at an NPSH value of 11 ft. The low frequency noise variation is not as distinct. Several attempts have been made to correlate the high frequency noise signal to actual erosion rates.

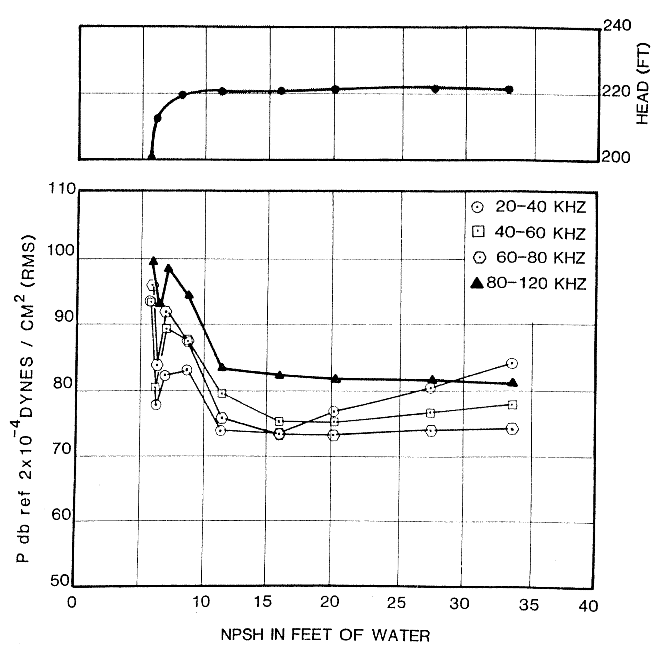


*Figure 6. Variation of Acoustic Emission with NPSH Measured Using a Pressure Transducer Mounted on Suction Pipe. Inducer diameter 5 in; 3600 rpm; 600 gpm.*

A 1968 paper by Pearsall and McNulty [11] showed that at least qualitatively there is a relationship between the erosion rate and the amount of noise, as shown in Figure 7. It can be seen that as velocity increases, both erosion rate and cavitation noise increase monotonically. Tests were conducted in a tube erosion rig at a constant cavitation number. Recent work, however, has indicated that this measurement is perhaps misleading, because noise is emitted whenever a cavitation bubble collapses. It is conceivable that a bubble could collapse in the free stream far from the vane surface and not cause damage. It would, however, generate noise. As a consequence, the noise measurement method for erosion rate assessment is in doubt.

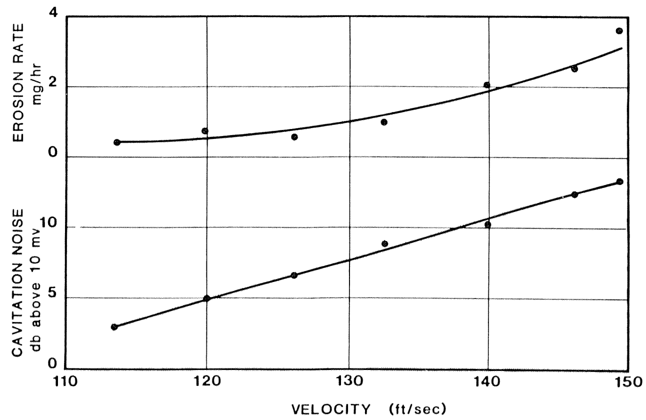


*Figure 7. Correlation between Cavitation Noise and Erosion Rate at K = 0.54 in a Constricted Tube Erosion Rig [11].*

Other attempts have been made to correlate the measured noise intensity with cavitation damage. An approach used by Hammitt [12] was to distinguish between bubble collapses having a large amount of energy and those having a small amount. This results in a bubble collapse spectrum (the number of pulses versus the energy per pulse). Recognizing that there is a threshold energy level below which no damage will take place, it is possible to integrate this pulse count spectrum curve and obtain an acoustic energy level that would cause damage. This method was tried on an ultrasound vibratory test facility. Comparing the acoustic energy measurements against direct weight-loss measurements, it was found that the spectrum area curve generally followed the same trends as the weight-loss curve. Hammitt went further to define an erosion efficiency that is obtained by dividing the spectrum area by the weight loss. It was expected that this efficiency would remain a constant, but it was found to vary by about a factor of five. It is believed that the method can be extended further and if successful, this method would become a very reliable indicator of the cavitation damage rate occurring on a prototype machine.

Recent advances in instrumentation techniques have made it possible to detect high frequency pulses that are characteristic of material damage and possible to discriminate between collapses that cause damage and those that do not. The principle to this concept is that when a material undergoes damage by repeated impacts, a distinct noise is emitted in a very high frequency range of the order of 100 kHz. It has been suggested that the amplitude of this noise may be directly related to the rate of damage.

Acoustic flow analyzers developed by some manufacturers measure not only the average root mean square (RMS) energy in the high frequency range, but also a new quantity that is a measure of the energy contained in the intense acoustic bursts generated when a material undergoes damage. In order to delete small spikes, the acoustic energy of the intense bursts is measured only above a certain threshold level. The threshold level itself is considered to be a function of the RMS amplitude. The new measure is called SAT, meaning Spike Above Threshold. The measurement system consists of a sensor, which can be mounted on the surface of the test housing, and an electronic data reduction package.

Typical measurements obtained on the previously mentioned inducer pump are shown in Figure 8. Both the RMS and SAT variations are shown as functions of NPSH at a constant flow and speed. It can be seen that at about 30 ft NPSH, RMS begins to increase rapidly. However, only at about 6 ft NPSH does the SAT increase. The suggestion is that even though considerable cavitation activity starts at 30 ft NPSH (as evidenced by RMS), damage begins to occur only at 6 ft NPSH.

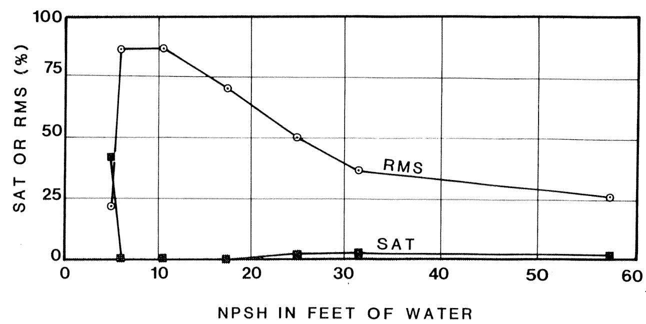


*Figure 8. Variation of RMS and SAT Intensity with NPSH on Inducer Pump of Figure 6 at 400 GPM.*

The cavitation behavior of the inducer pump is plotted in Figure 9. Inception was detected both acoustically and visually. The acoustic inception was based on the NPSH at which the noise signal in the 80-120 kHz band increased rapidly. The SAT curve was plotted by referring to the NPSH at which the SAT value reached ten percent. This number was arbitrarily assumed to represent a damaging NPSH value. The best efficiency point (BEP) of the pump is at 600 gpm. Both visual and acoustic inception NPSH values are close to each other down to about 400 gpm, where appearance of inlet recirculation greatly changed the acoustic characteristics. At BEP the margin between inception and breakoff is small, and NPSH based on SAT criterion suggests that the pump should be operated above the acoustic inception point!

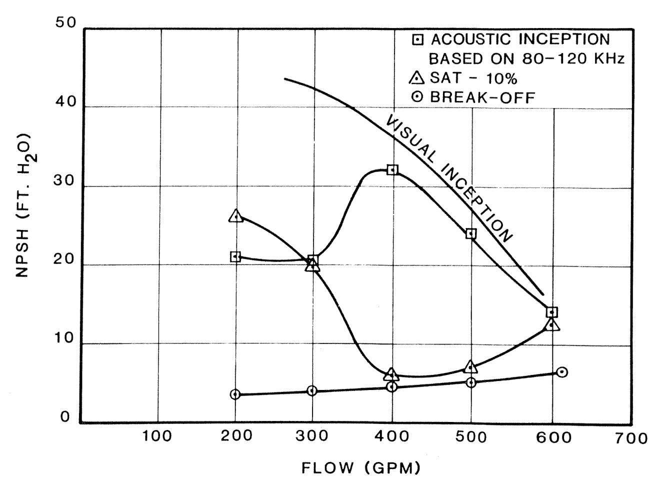


*Figure 9. Cavitation Behavior of Inducer Pump.*

It is clear that the SAT technique offers great promise as a short term method for estimating longlife NPSH. Bloch also utilized this technique and promising results were reported [13]. Unfortunately, not enough correlative data between SAT and damage rates has been acquired to suggest use of this method for pump acceptance tests.

## RADIOACTIVE METHODS

Radioactivity has been used to measure the rate of removal of material by cavitation. In a method described by King [14], a thin Cadmium overlay was applied on a specimen undergoing erosion. A beta-emitting radioactive source, Strontium 90, was used to irradiate the overlay, which scattered the incident particles. A counter was used to measure the intensity of the back-scattered particles. This intensity was proportional to the remaining Cadmium overlay. This method revealed local erosion rates; however, it has not been extended to rotating systems.

Currently, a new radioactive emission method is being investigated [15]. This method utilizes safe radiation levels, and can be applied to any metallic centrifugal pump impeller material. The particle beam from an accelerator impinges on the surface of the part of interest and causes a nuclear transmutation of a few of the atoms of the material. The resulting radionuclides, being unstable, decay by emitting gamma rays and other forms of radiation. The depth to which the surface is activated depends upon the material, the beam species used, energy of the beam, etc. The activation depth determines the sensitivity of the measurement. The beam chosen has an appropriate half-life for the emissions of the decaying radionuclides. When the surface wears due to any erosive action, the quantity of emitted gamma rays decreases. The resulting gamma ray activity must be calibrated beforehand by using a calibration sample. This method is being used at present to detect the wear rates of engine bearings and other mechanical components. No applications for cavitation damage detection have been reported.

## EMPIRICAL METHODS

From the foregoing, it would be obvious that none of the existing test techniques are technically completely acceptable. As a consequence, empirical calculation methods are being proposed in the literature. A method which has received wide attention is the one proposed by Vlaming [16]. The NPSH required for 40,000 hour impeller life at the shockless entry point is given as

$$NPSH_{S.E.} = K_1 \frac{C_{m_1}^2}{2g} + K_2 \frac{W_1^2}{2g}$$

where

$K_1$ = constant $\cong 1.2$
$C_{m_1}$ = upstream meridional velocity
$W_1$ = upstream relative velocity
$K_2 = 0.28 + \left(\frac{U_1}{400}\right)^{4.0}$
$U$ = peripheral velocity at eye

For flows at other than the shockless entry point, the variation of NPSH with flow is given by Vlaming [16] in a graphical form. The method is based on empirical data. It is believed that for reasonably good designs, adherence to the NPSH value as calculated above would ensure an impeller life of 40,000 hours against cavitation damage.

Alternate methods based on the ratio of NPSH required for long life to NPSH required at head degradation have also been proposed. This ratio, sometimes referred to as the "R-factor," has been expressed in terms of meaningful physical parameters by Cooper and Antunes [17]. In this representation, which was derived from fundamental physical considerations of the damage process, the R-factor depends upon the pressure coefficient within the blading, details of flow geometry, material properties, etc, and strongly on the impeller inlet blade tip velocity. The value of R can be as low as 1.0 or as high as 20, depending upon materials, tip speeds, flow configuration, etc.

The R-factor method cannot at present be commonly used, because coefficients in the final expression must be quantified via model testing of actual pump geometries. An empirical data-based procedure for estimating longlife NPSH using the R-factor has been published recently [18]. Since the ordinates in the various figures are not specified, the method cannot be applied.

Until the approaches mentioned above can be refined and fully quantified, empirical predictions will continue to be employed. A method used by the present author is based on defining the long life NPSH as a ratio with respect to inception NPSH. This factor can be denoted as $R_i$. Evidently, when $R_i$ is greater than 1, there can be no cavitation damage as bubbles will not be formed. Since inception NPSH obeys the hydraulic affinity laws quite closely, knowledge of $R_i$ can lead to NPSH predictions for different speeds, etc. Of course $R_i$ will be a function of material properties, flow geometry, blading design, and impeller inlet blade tip velocity, $U_1$. The relationship between $R_i$ and $U_1$ can be written as:

$$R_i = \frac{K_1}{\pi} \arctan\left(\frac{U_1 - K_2}{K_3}\right)$$

where

$K_1 \cong 2.0$ to $2.2$
$K_2 \cong 10$ to $40$
$K_3 \cong 15$ to $25$

and the inverse tangent expression stated in radians. The variations in the factors, $K_1$, $K_2$, and $K_3$, allow for different materials and fluid properties. The above expression is valid for shockless entry flow. For other flowrates, larger margins may be necessary. The margins can be estimated from Vlaming's curve [16].

## CONCLUSION

Several criteria that have been proposed for establishing the NPSH level required by centrifugal pumps for damage-free operation have been examined. For applications where cavitation damage would cause severe liability, a conservative criterion of no cavitation bubble is adopted. In less severe cases, bubble cavity length is allowed as a function of the impeller inlet blade tip velocity. In other instances, lack of removal of a suitable soft coating on the impeller surface is taken to indicate low probability of eventual metal removal. New acoustic and radioactive techniques are also being studied as possible candidates for short term damage evaluation. Empirical methods for calculating long life NPSH are also described.

It is clear that experimental techniques have yet not been developed to the point that they can be used as acceptance criteria during pump witness testing. Similarly, neither have published empirical procedures received wide acceptance. However, the need for suitable criteria is increasingly being recognized worldwide, and a great deal of active research is being carried out. Therefore, rapid progress may be expected in the next few years and criteria acceptable to users and manufacturers alike will soon emerge.

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