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A REVIEW PAPER ON NOISE AS PARAMETER FOR CAVITATION MEASUREMENT IN CENTRIFUGAL PUMPS

ABHISHEK KUMAR SINGH[a1] AND ASHOK KUMAR[b]

[a]M. Tech. Rotating Equipment, University of Petroleum and Energy Studies, Dehradun, Uttarakhand, India
[b]Department of Mechanical Engineering, University of Petroleum and Energy Studies, Dehradun, Uttarakhand, India

ABSTRACT

Cavitation is a chief reason of disturbance in a centrifugal pump. Deficient net positive suction head available (NPSHA) is the cause of phenomenon of cavitation in centrifugal pump and it can happen at any phase of the operation. Cavitation causes numerous unwanted impacts in the pump like: (1) head capacity and efficiency of pump will drop; (2) pitting or erosion will occur in the impeller, and (3) it can cause increment in structural vibration in parts and also increment in noise level. Hence, the cavitation process must be avoided definitely. To keep cavitation from starting the initial step is to identify the start of cavitation. To recognize onset of cavitation, among different potential methods, the measurement discharged noise can be utilized as a method to gauge the onset of cavitation. This paper reviews the approach to make noise a parameter for indicating beginning of cavitation. Also this paper shows the effect of cavitation on transient condition of pump.

KEYWORDS: NPSH

When we consider the outline parameters of the pump, cavitation is one will be one of the vital point that ought to be remembered. In industry there is wide acknowledgment of the NPSH 3%-criteria for forestalling cavitation, because of this system there is detectable changes in pump discharge head [Budris and Mayleben, 1998].Since cavitation is a vaporization process as it happens because of vanishing of working liquid inside pump, the physical properties of the fluid and its vapor pressure and the flow conditions can influence the cavitation process. Therefore for cavitation we think about the consolidated impacts of liquid properties, flow conditions, and heat exchange, termed thermodynamic effect of cavitation, as utilizing this we can stop the beginning of cavitation to a vast degree.

The NPSH required change from fluid to fluid and additionally NPSH required of fluid at various temperatures can likewise be less than fluid at room temperature. It can be said that the change (diminish) in inlet pressure conditions is identified with the differing degrees of evaporative cooling related with the cavitation process. Due to the dissipation of working liquid in the pump, there is a decrement in cavity pressure and the vapor pressure of the fluid adjoining the cavity in comparison to the vapor pressure of the entire working liquid. Thus, there is a decrease in cavity pressure, which hinders the rate of further vapor formation; this enables the pump to work at lower values of NPSH.

There are distinctive engineering strategies accessible for acquiring the cavitation noise [Young, 1989 & Li, 2000]. Some of these are distinctive engineering strategies are:

a) The most well-known strategy is by finding the NPSH against a consistent speed as well as steady stream flow rate.
b) The second famous technique is picturing the method for flow of working liquid through impeller eye of the pump [Japikse et. al., 1997].
c) Another technique is by utilizing paint to do erosion testing. It is finished by covering paint on the impeller blades and shroud and after that watches it for erosion by cavitation due to apparent removal of the paint. It is used in mix with the NPSH test.
d) Static pressure measured in the stream flow or in the volute-packaging can be utilized as a strategy for checking beginning of cavitation. With this strategy the beginning of the cavitation is resolved in a roundabout way by looking at the deliberate static and the vapor pressure of the working fluid at the given flow conditions or by wavelet investigation i.e. the spectral examination of the vortex [Chu et. al., 1995] forms and pressure signal.
e) Measuring the structural vibration by putting on a transducer close to the impeller blades or near the place where cavity formation occurs [Li, 2000].
f) This strategy views the sound pressure measurement. Inspteof being basic and logical this is an infrequently utilized building strategy. The presence of cavitation can be plainly dictated by ordinary hearing; therefore,

[1]Corresponding Author

utilizing acoustic signals as an examination strategy for cavitation has been a fantasy of numerous researchers [Japikse et. al., 1997].

NOISE SOURCE OF THE PUMPING SET WITHOUT CAVITATION AT DESIGN OPERATION

A pumping set (Fig 1) comprising of a centrifugal pump 1, an electric motor 2, and a cooling fan 3 produces noise by virtue of noise generating systems and it also delivers a broadband of noise range and additionally it likewise creates discontinuous frequency. This discontinuous frequency is alluded to as "tonal noise," otherwise called rotational noise i.e. noise produced by rotating parts, and the broadband noise is alluded to as "turbulent noise," otherwise called non-rotational noise i.e. noise created by non-rotating parts of the machine.

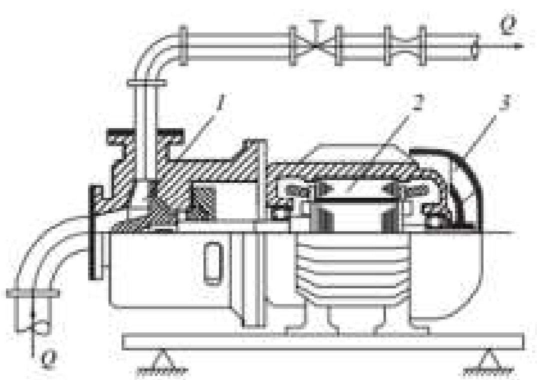

Figure 1: Pumping set

The tonal noise is mainly a result of interaction of rotor blades with unsteady flow flow and with adjacent non-moving parts.

At the designed operating conditions, there are vortices displayed in the flow which are made because of viscous friction of the working liquid and furthermore because of the pressure distinction between the suction and release side of the blade, the blade communicates with vortices present in axial and radial clearances and the noise created from this is one of the major contributor in creating turbulent noise.

At the point when the pump works at high flow rates there is increment in the quantity of vortices while the blade is kept completely occupied. Along these lines, extra water driven noise is created and this noise increments with the expansion in high flow rate of the pump. Though At partial flow rates, there is angular rotation of the fluid in the interior running clearances which causes the increment of destabilizing powers likewise noise made by stalls also increases the noise produced.

NOISE SOURCE IN THE PUMPING SET WITH CAVITATION

As cavitation happens, cavitation noise is created which is the additional noise produced because of cavitation. Hydro dynamically prompted cavitation noise is of two sorts. One type is connected with the development of stalls which is because of recirculation of local flow at low flow rate. These stalls have high speed at their center this causes gigantic lowering down of static pressure around there. The other sort of noise is made due to suction head deficiency in the pump. This causes general mass flow and pressure oscillation all through the pump and is known as surge.

The required NPSH value is the required head over the fluid vapor pressure where the static pressure is minimum e.g., at the suction nozzle. The NPSH can be expressed by the following equation (NPSH = σH). Where σ is the Thomas cavitation coefficient, a measure of the resistance to cavitation. The lower the value of the coefficient the more it is plausible that the cavitation will happen. σ is portrayed by the quotient of the flow speed ‘w’ and static pressure: $\sigma = (p_{st} - p_v)/0.5\rho w^2$ where p_{st} is static pressure, p_v is vapor pressure of the pumping liquid at the local temperature, ρ is the density of the liquid

TEST PROCESS

For this test the pump is made to work at different pressure. The pump is made to take liquid from a closed vessel in which there is a fluctuated air pressure which is finished by changing the pneumatic force over the fluid level. To estimate cavitation onset, the aggregate head delivered is estimated at a consistent speed and steady flow rate, yet at a shifting NPSHA (net positive suction head available) conditions (Fig 2). In the meantime, measurement of the spectra and total emitted level of noise is taken from a distance of 0.5m via a microphone which is kept perpendicular to the motor-pump axis.

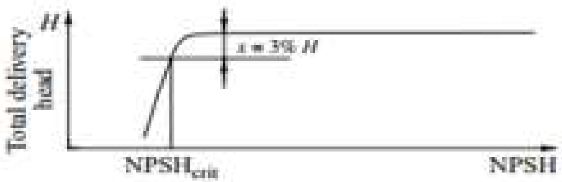

Figure 2: Determination of NPSHA required

NOISE ANALYSIS OF THE PUMPING SET WITH CAVITATION

Amid the estimation it was discovered that, at the design point, noise level of the of the entire pumping set is about almost 13 dB (A) higher in contrast with that electric motor without fan, and noise produced by electric motor and cooling fan is almost 4 dB (A) lower than total noise [Chudina, 2003]. This implies in the noise produced by pump set most of the noise is made by the disturbances in the pump.

After the above test it was discovered that the difference between the total noise level before cavitation happens (Lp) and after the cavitation has completely created (Lp(NPSHcrit)) are around 3 dB(A).

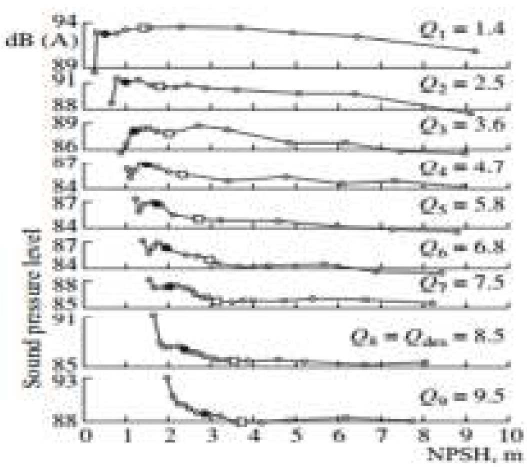

Figure 3: NPSH and noise characteristics at constant flow rate

The larger the cavitation in pump the lower is NPSH. It can be seen that noise peaks of the BPF and sounds at higher flow rates is not affected by cavitation (Fig 3). Be that as it may, it affects lower flow rates noise peaks, particularly the first harmonic, where large amount of stalls is present. The first BPF crest diminishes when the cavitation occurs, whereas the other peaks somewhat increases. This is clarified by the way that the impact of rotating stalls reductions where NPSH available diminishes and the amount of cavitation bubbles rises. Then again, noise due to cavitation increments through the advancement of cavitation with the goal that it covers the area around the higher harmonics of the BPF (Fig 4).

In this spectrum investigation two ordinary conditions were looked at (low flow rate condition and high flow rate condition). The information gathered from the instruments at the two regular conditions investigated and the time domain signal (flow induced noise) is changed into power spectrum diagram by Fourier transform [Yuan et. al., 2012].

These discrete frequency noise are chiefly because of the rotor irregularity and the association amongst impellers and tongue. The noise band at low flow rate is for the most part centered on the area 0– 40 times shaft frequency. Moreover the broadband noise at low flow rate and configuration condition is frail amid the district 40– 100 times shaft recurrence. However, there are numerous peak points the broadband noise is solid amid 40– 100 times shaft frequency under high flow rate condition. Under the high flow rate condition, the blade wake expanded. What's more, as the pump flow rate builds, the local static pressure lessens further and the volume of cavitation bubble increments, in this manner advancing cavitation. At this point the broadband noise under the given conditions are analyzed and discovered that the broadband noise under low flow rate condition is the lower and the high flow rate condition is higher.

After examination it was found that the peak at the discrete frequency of 147 Hz could well be used to recognize the cavitation method in the pump [Alfayez and Mba, 2005].

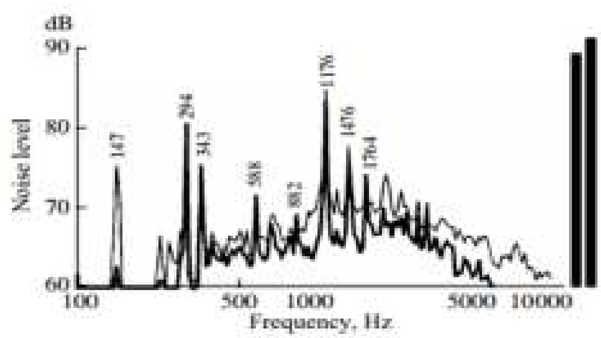

Figure 4: Noise spectra before the inception of cavitation (thick line) and noise spectra after cavitation fully developed (thin line)

The difference in noise level of the peak at 147 Hz are sufficiently enough to utilize the noise signal to recognize the beginning of the cavitation process in the pump [Tsujimoto et. al., 1997]; for utilizing this frequency for estimating vibration it doesn't make a difference if flow rate is known or not.

TRANSIENT CHARACTERSTICS OF THE CAVITATING PUMP

Influenced by heavy cavitation in the beginning, the hydrodynamic efficiency of pump clearly diminished compared to operation without cavitation. Amid the quick increase of rotational speed, instantaneous head H, all of a sudden decreases and start fluctuating after critical speed is surpassed by rotational speed. Before critical speed is attained, the quasi-steady supposition head H_s is lower compared to instantaneous head H. Be that as it may, the instantaneous head H is less than quasi-steady supposition head H_s in the wake of reaching the critical speed [Tanaka and Tsukamoto, 1999].

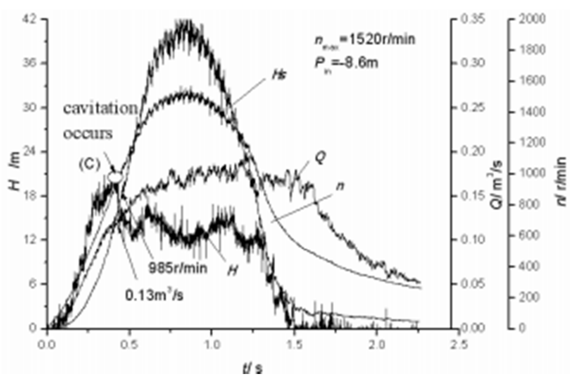

Figure 5: Hydrodynamic performance during the transient operation under cavitation condition

The delayed cavitation might be due to impulse pressure of pump startup [Tsukamoto and Ohashi, 1982]. During rise of rotational speed, pump flow carries potential flow without partition. In this manner, it prompts a more prominent incentive for head H than semi state supposition head H_s, and furthermore prompts the postponement of cavitation initiation [Wu et. al., 2010].

CONCLUSION

The noise emitted by pump relies upon flow rate and load on the pump as the higher the flow rate the more is noise and on the instability in the pump. Stalls, surge, and cavitation can be some of the causes of instability in pump. Instability caused by cavitation can result in vibration, cavitation noise, material erosion, pitting hence bringing down performance of pump. Cavitation is caused by deficiency of sufficient NPSH value. Therefore, the NPSH available must be kept up to prevent corrosion.

The difference in noise levels is sufficiently high, hence signal of frequency 147 Hz is used to decide NPSH and control operating conditions of pump.

In the beginning time of the non-cavitation pump, acceleration profoundly affects centrifugal pump transient qualities. The head 'H' makes a beeline for winding up noticeably bigger than the supposition head H_s, and the head fast rise stage is notably influenced by the rate of acceleration. Be that as it may, after the cavitation happens in transient condition, the peak head right off the bat clearly declined then started to fluctuate.

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