Cavitation Detection in Hydraulic Machines: A Review

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Abstract—Cavitation is the formation of vapour cavities in a liquid in the low pressure regions of hydraulic machines. The effects of cavitation are hydraulic viz. low efficiency due to flow instability and mechanical viz. surface damage, noise and vibration. Collapsing voids that implode near a metal surface cause cyclic stress and results in surface fatigue. Turbines and pumps show declined performance after few years of operation, as they get severely damaged due to erosive wear on account of cavitation. In this paper, different techniques used by several researchers for cavitation detection like measurement of pressure, vibration and sound; visual inspection as well as numerical simulations are presented which may be useful in practical conditions to minimize the effects of cavitation.

Index Terms—Cavitation, detection, pressure, vibration

I. INTRODUCTION

Whenever the pressure in any turbine part (exit) or pump part (entry) drops below the vapour pressure, the liquid boils and large number of small vapour bubbles are formed. These bubbles, mainly formed on account of low pressure, are carried by the stream to higher pressure zones where the vapour condenses and the bubbles suddenly collapse. This results in the formation of a cavity and the surrounding liquid rushes to fill it. The streams of liquid coming from all directions collide at the center of cavity giving riseto a very high local pressure [1]. Different stages of cavitation damage and pit formation are shown in Fig. 1.

II. EFFECTS OF CAVITATION

The effects of cavitation are hydraulic (low efficiency due to flow instability) and mechanical (surface damage, noise and vibration). In addition, it may also lead to surface erosion [2]. It is difficult to avoid cavitation in hydro turbines but certainly it can be reduced to an economically acceptable level [3]. The pitting on runner blades due to cavitation is shown in Fig. 2.

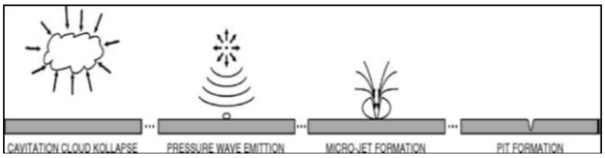

Fig. 1. Stages of pit formation due to cavitation

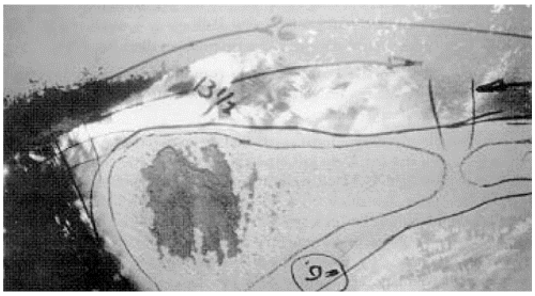

Fig. 2. Pitting at runner blades due to cavitation

There is a trend to operate the turbines in conditions far from BEP and cavitation phenomena are more prone to occur at off design operating conditions. The combination of both promotes cavitation problems. Monitoring systems for cavitation detection serve to avoid harmful situations seem to be the best solution [4].

III. THOMA CAVITATION NUMBER

Prof. D. Thoma suggested a dimensionless number, called as Thoma's cavitation factor 'σ' [5]. Typical sigma curve is shown in Fig. 3. It can be seen that as σ decreases, initially there is no effect on the efficiency. With further decrease in σ, efficiency first increases then decreases sharply. Accordingly, critical value of sigma (σ_{cr}) is defined and it is recommended to run the machines (pump/turbines) above σ_{cr} for cavitation free operation.

$$\sigma = \frac{H_b - H_s}{H} = \frac{H_{atm} - H_v - H_s}{H} \qquad (1)$$

Where, $H_b$is barometric pressure head in m of water, H_s is suction pressure at the outlet of reaction turbine, H - net head on the turbine.

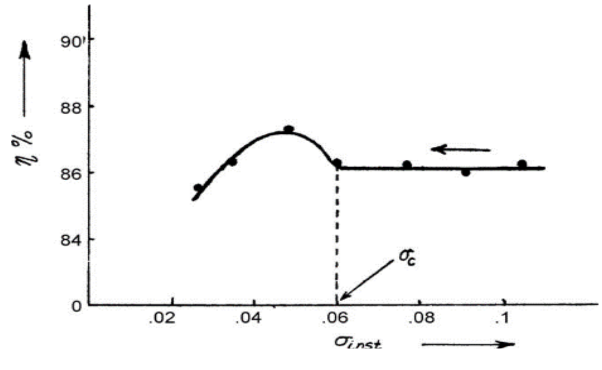

Fig. 3. Cavitation number v/s efficiency

IV. TYPES OF CAVITATION

Different types of cavitation usually found in different machines based on their operating conditions are mentioned here.

A. Travelling Bubbles

Bubbles usually appear around a body from micron-sized nuclei in low pressure regions of the flow as shown in Fig. 4. Travelling with the flow, they implode when they find an adverse pressure gradient. These bubbles are strongly influenced by the air content of the liquid. Nevertheless, their erosive power is considered to be relatively weak [6].

B. Attach Cavities

Cavitation can take the form of macro-cavities that develops and gets attached on a solid wall placed in the flow as shown in Fig. 5. Sheet cavitation, is characterized by thin stable cavities with smooth and transparent interfaces. At their rear part, the cavity closure presents a slight and weak pulsation due to the shedding of small cavitation vortices so that it represents a low risk of erosion. The attach cavities further disintegrate in either two forms i.e. cloud cavitation wherein small vapour bubbles are formed or in ring vortices [6].

C. Vortex Cavitation

Flow regions with concentrated vorticity can develop cavitation in their central cores due to the low pressures generated. If the tips of these vapour filled vortices are in contact with a solid surface they become potentially erosive since the final collapse of the whole cavity takes place on them. A typical example of this type of cavitation can develop if Von Karman vortex-shedding as shown inFig. 6 occurs at the trailing edge of a hydrofoil [6].

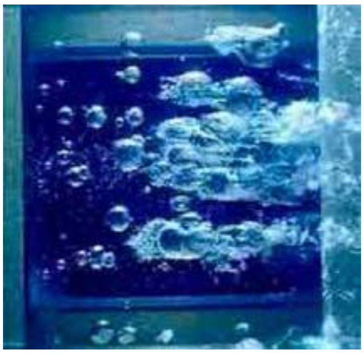

Fig. 4. Bubble cavitation

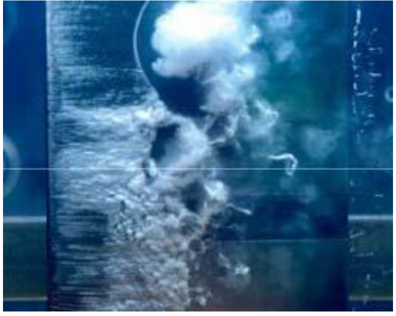

Fig. 5. Attach sheet cavitation

Fig. 6. Von-Karman vortex shedding

V. DETECTION TECHNIQUES

There are different techniques for cavitation detection in hydraulic machines viz. pressure measurement, visual inspection, vibration measurement, noise analysis, computational fluid dynamics (CFD) approach etc. These techniques are used by many researchers for the cavitation detection in hydraulic machines.

A. Pressure Measurement

The pressure measurement is a technique to determine the cavitation. When the cavity or bubbles enter the high pressure zone, they collapse and induce vibration as well as pressure pulsations. Vibration measurement may not give conclusive results due to dynamic behavior of machinery hence for detailed analysis and for verification pressure measurement technique should be adopted. Another technique is to carry out the sigma (σ) test i.e. to study the effects of variation in σ on efficiency. The σ can be varied by varying the head (H) acting on the machine.

Escaleret al.[6]carried out experiments on Francis turbine for pressure measurement with the amplitude demodulation. Fig. 7,shows the pressure pulsation against frequency for bubble type cavitation and no-cavitation flow. As it can be seen that whenever the pressure waves are generated due to cavitation, high peaks are obtained in the frequency band.

The variation in draft tube pressure at different operating conditions is shown in Fig. 8. Cavitation is detected in much lower frequency range as seen in figure. Whenever bubble type cavitation occurs the peak values are obtained compared to non-cavitation flow.

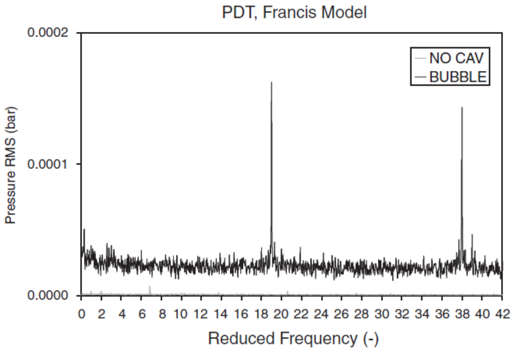

Fig. 7. Peak pressure values measured

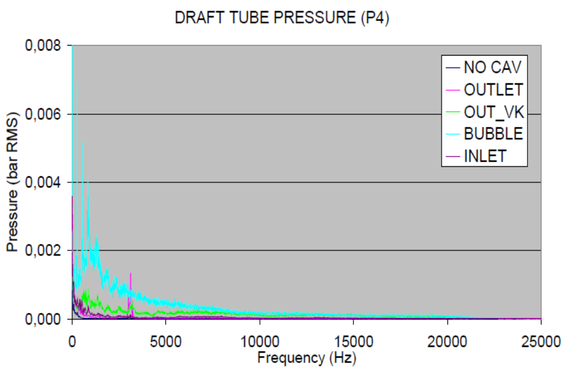

Fig. 8. Draft tube pressure at different condition

Schiavello and Visser [7] carried out experimental investigations on cavitation on pumps and discussed various net positive suction head required criteria, net positive suction head available margins and impeller life expectancy and plotted the same. They also determined various models to deal with cavitation. Fig 9 shows various cavitation modes. NPSH curve is shown which is responsible for significant erosion damage throughout the whole range of operation. A curve for NPSHR at 3 percent head drop is shown which describes key cavitation aspects, effects and location.

B. Visual Inspection

Many investigators have provided transparent window in different parts of the machine to visualize the cavitation. A small transparent glass window can be fitted near the drfat tube entry for flow visualization. The use of high speed frame capturing cameras as well as stroboscopic light is useful in such studies. Some researchers also suggest the use of glass draft tube to visualize the draft tube swirl phenomena as well as the travelling bubble type cavitation.

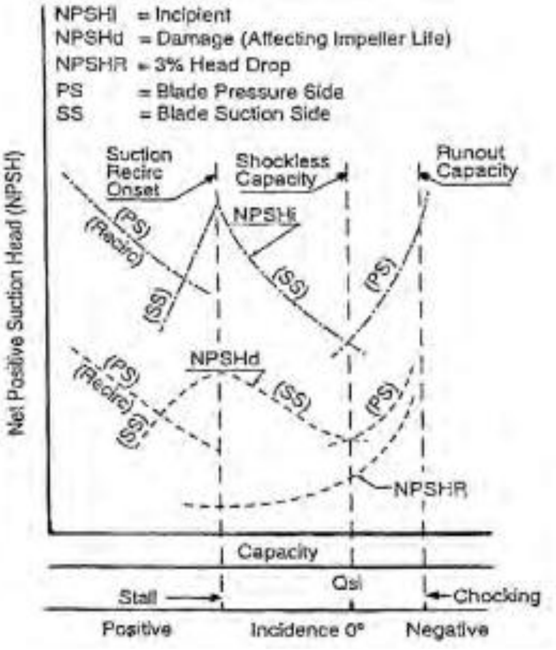

Fig. 9. Curves defining cavitation

Grekula and Bark [8] studied the cavitation in Kaplan turbine with high speed filming, video filming and visual observations with stroboscopic light. Mainly two types of cavitation in Kaplan turbine was found by them viz. travelling bubble and attached cavities. The attach cavities further shredded into either cloud bubble cavitation or vortices cavitation. The main reason of shredding was the Re-Entrant jet as shown in Fig 10 and its penetrating length. If the penetrating length was small then it would result in vortex shedding and if the penetrating length of Re-entrant jet was large it would result in cloud cavitation. Generally the attach type cavities was found at the blade leading edge as shown in Fig 11. This was due to mismatched angle of attack causing a flow separation close to leading edge. Travelling bubble and sheet type cavitation was found by them at blade root as shown in Fig 12. This was mainly due to reduction in pressure caused due to shape of body because of fillet as well as mixing of two boundary layers which causes flow separation and resulted in to low pressure areas. At the blade tip the cavitating vortices were formed mainly due to tip clearance flow and scraping of the boundary layer as shown in Fig 13.

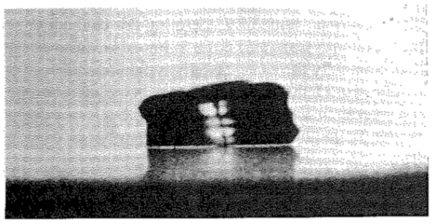

Fig. 10. Re-entrant Jet

Fig. 11. Attach cavity at leading edge

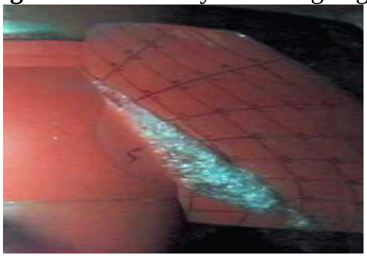

Fig. 12. Cavitation at blade root

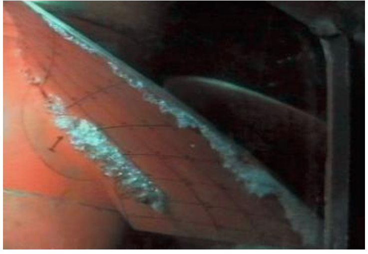

Fig. 13. Cavitation at blade tip

Avellan [9] carried out experiments on centrifugal pumps and Francis turbine with an observation window to visualize the cavitation. It was found that in case of pumps the cavity development depends on the discharge coefficient according to the relative flow velocity and incident angle at impeller inlet. At rated discharge travelling bubble cavitation occurred on suction side of blades while minimum pressure was observed at impeller throat. For a lower discharge value, leading edge cavitation occurred as shown in Fig 14. For a lower value of σ, cavitation vortices were found runner inlet. Draft tube swirl observed in Francis turbine using a glass draft tube as shown in Fig 15.

Siroket al.[10] presented cavitational structures quantification with aid of computer-aided visualization method on the model of the Kaplan turbine as shown in Fig 16.The machine at selected integral turbine parameters, topological cavitational structures on the draft side of rotor blade was analyzed by them. High speed filming along with the stroboscopic light was used to observe the cavitation phenomena in Kaplan turbine model.

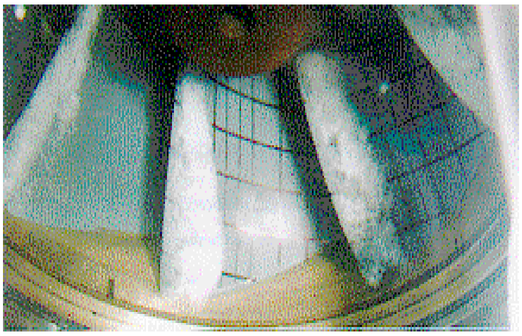

Fig. 14. Leading edge cavitation at inlet of pump

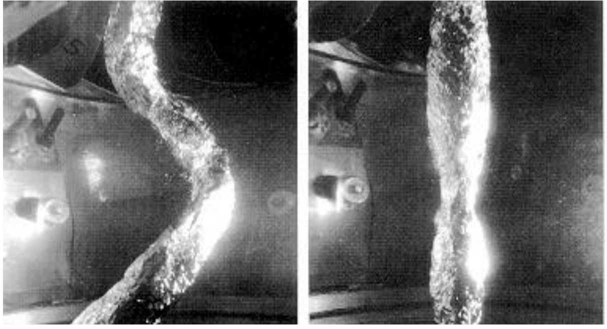

Fig. 15. Draft tube swirl

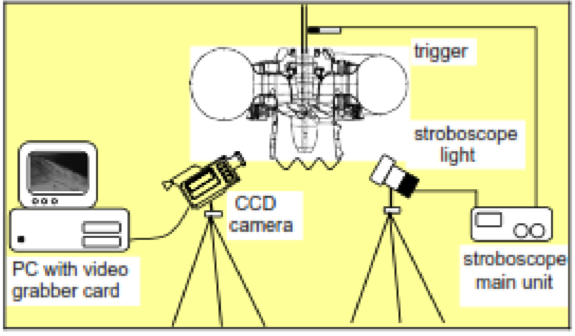

Fig. 16. Computer aided visualization

Fig. 17. Cavitation erosion of a runner

Patel [11] carried out experiments on Pump running in Turbine mode. A glass tube was installed at the entry of the draft tube in order to visualize the cavitation. Mainly the bubble type of cavitation and the swirl were found as shown in Fig 18 and in Fig 19.

C. Vibration Measurement

The methods to detect cavitation in real machines are based on the measurement and the analysis of the induced signals. Detection is not an easy task because depending on the turbine design and the operating condition; the type of cavitation, its behavior and its location are different. So, this affects the nature of the excitation and determines the transmission path followed up to the sensor. Furthermore, the measured signals can be contaminated by noise coming from other excitation sources of hydrodynamic, mechanical or electromagnetic origin. Therefore, the selection of the most adequate sensor and measuring position on the machine is of relevant importance to improve the detection.

Fig. 18. Travelling bubble type cavitation

Fig. 19. Vortex rope cavitation

Escaleret al. [4] carried out experiments and vibration analysis and suggested to measure the structure and fluid borne noise, high frequency content amplitude demodulation and low frequency content. The amplitude of a given frequency band could be compared for various operating conditions by computing the auto-power spectrum of time signals. Cavitation erosion of runner is shown in Fig.18 and vibration peaks with amplitude demodulation is shown in Fig. 19. The high frequency content may not give conclusive results because other phenomena can also cause vibration of machinery hence to carry out amplitude demodulation of high frequency content was suggested by them using the HILBERT Transform as follows.

$$\mathrm{Hi}\{x(t)\} = \frac{1}{\pi}\int_{-\infty}^{\infty} x(\tau)\frac{1}{t-\tau}\,d\tau \quad (2)$$

$$\tilde{x}(t) = \mathrm{Hi}\{x(t)\}$$

Then

$$\dot{x}(t) = x(t) + j\tilde{x}(t)$$

Escaleret al. [6] carried out experiments on Francis turbine model for vibration analysis using three accelerometers at different position. One on shaft and two on guide bearings at 90 degrees apart. Various acceleration values within the given bandwidth of frequency for different guide vane openings was measured and concluded that as the turbine run away from its BEP, higher peaks were obtained nearer to 6 kHz frequency as shown in Fig20 and Fig 21.

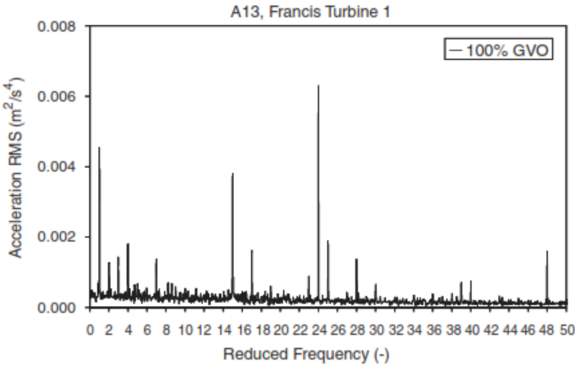

Fig. 19. Vibrations peak with amplitude demodulation

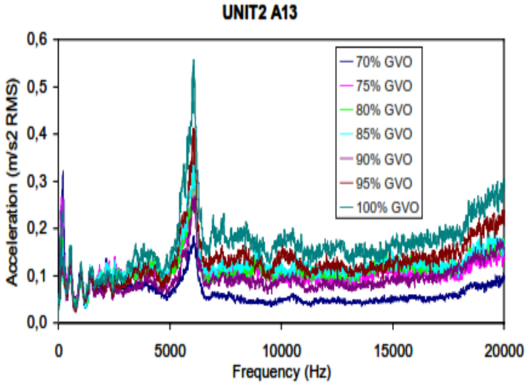

Fig. 20. Vibration at bearing for different GVO

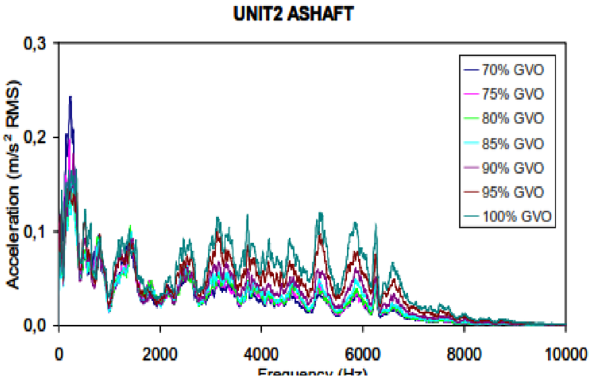

Fig. 21. Vibration at shaft for different GVO

D. Noise Measurement

In higher frequency ranges the vibration measurement becomes difficult and results obtained are not satisfactory hence the use of acoustic emission sensors serves to extend this analysis to upper frequencies that the accelerometers cannot reach [4].

Escaleret al. [6] carried out experiments on Francis turbine model for acoustic emission measurement. Fig. 22 shows the measured values of Acoustic Emission v/s the frequency ranging from 0 kHz to 20 kHz for different Gate Valve Openings. The values increased with the increase in GVO with the exception of sudden decrease at 90% which accelerometer could not plot were concluded by them.

Patel [11] carried out acoustic emission analysis on pump as turbine (PAT) running at different speeds to detect the cavitation. It was found that the statistical value of equivalent noise level fluctuated over different time periods. The device was set at ‘C’ level to measure the noise at draft tube entry and values were recorded at 10 seconds for 20 times. Following equation was used to found equivalent noise level.

$$L_{eq} = 10\log_{10}\frac{1}{n}\sum_{i=1}^{i=n}10^{(L_i/10)} \quad (3)$$

The graph was plotted of equivalent noise level and cavitation number. It was found that at critical cavitation number where the cavitation occurred largely the sound level found was satisfying as shown in Fig 23.

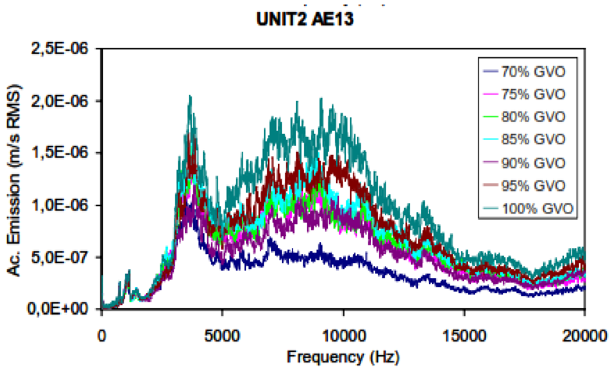

Fig. 22. Acoustic emission values

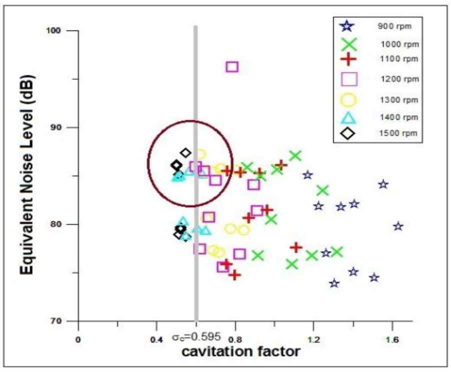

Fig. 23. Equivalent noise level v/s cavitation number

E. CFD Analysis

Sedlaret al.[12] described a new model design to provide an efficient picture of the water cavitation erosion potential using numerical modeling of the turbulent cavitating flow. They described the dynamic behavior of the cavitation bubbles resulting from the rapidly changing static pressure inside the hydraulic machine by the solution of the generalized version of the full Rayleigh-Plesset equation obtained from the 3D Reynolds-averaged Navier-Stokes equations. The model of the cavitation erosion potential was based on the estimation of the energy dissipated by the collapses of the cavitating bubbles. All the energy dissipated during the bubble collapse was used to form the shock wave propagating from the bubble center. A part of the shock wave energy emitted towards the solid surface represented the erosion potential.

Nohmiet al.[13] carried out cavitation analysis of centrifugal pump using CFD. They used two different CFD codes for same. First; a compressible air-vapor-liquid two phase model, widely known as TE model and secondly the Constant enthalpy vaporization model, known as CEV model. Both the CFD codes gave results nearer to the measured value at BEP flow rate as shown in Fig. 13. Either approaches showed steep head drop at cavitation breakdown even though the computation by TE model was unstable. By using the CEV model the head drop was gradual but there was a slight increase in head. Analysis concluded that at higher flow rates both models needed modification to predict better results.

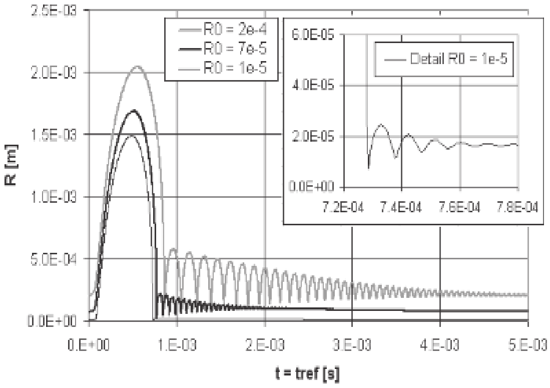

Fig. 24. Bubble collapsing

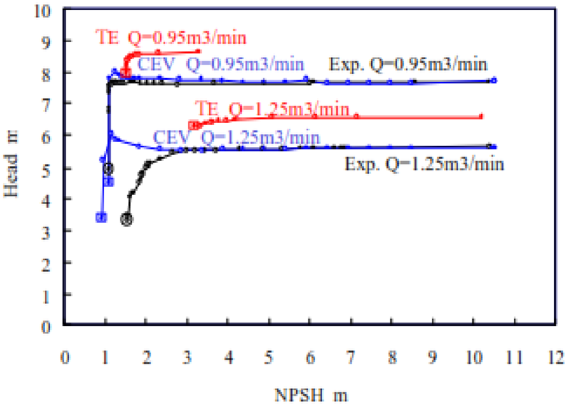

Fig. 25. Different models in CFD

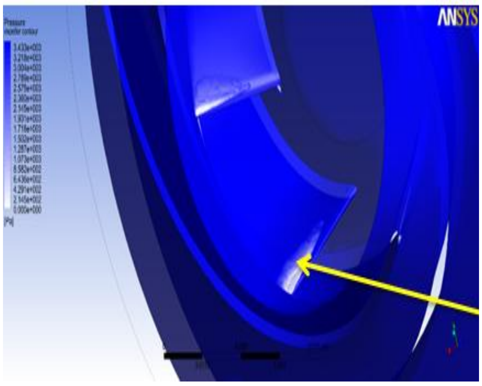

Fig. 26. Trailing Edge cavitation detection

Iosif and Sarbu [14] presented an explicit numerical model based on finite element method and dual reciprocity method. The use of the model to transform the 3D fluid flow to a simple 2D problem with ideal incompressible fluid was suggested by them. The axisymmetric potential motion using FEM method was solved and found the pressure and velocity distribution along the stream lines. The fluid motion was solved around the radial-axial profile cascades using the DRM and found the values of stream function ψ and normal derivatives of this function. The results were analyzed for a reversible hydraulic machine and found different discharge values which were used further to determine the cavitation characteristics and sensitivity curves..

Chauhan [15] used CFD as a numerical simulation tool to carry out cavitation analysis of pump running in turbine mode using Ansys-CFX software. To consider the cavitation effects multiphase homogeneous model was used. The study revealed that the trailing edges of the blades were more prone to cavitation, as shown in Fig. 26 and it resulted in somewhat decrease in efficiency.

VI. CONCLUSION

Cavitation being an unavoidable phenomenon, its effect can surely be reduced and can be predicted if proper methods are applied. Different techniques have been applied by many researchers for cavitation detection in hydraulic machines viz. pressure measurement, visual inspection, vibration measurement, noise analysis, CFD approach etc.With the help of vibration and acoustic analysis the detection can be made effectively but it is a time consuming and expensive process. Vibration measurement technique is best suited for detection compared to different techniques. CFD based cavitation analysis could be a cost effective solution for extensive analysis but it requires thorough understanding of numerical techniques and applications.

In Centrifugal Pumps, the cavitation is mainly found near the impeller eye and is subjected to leading edge cavitation of attached type. Mainly the NPSH criteria govern the cavitation in case of pump along with the discharge coefficient values.

In Francis turbines, flow coefficient governs the draft tube swirl cavitation. Leading edge cavitation does not mainly depend upon the Thoma number but depends upon the energy coefficient values. Travelling bubble occurs at design values of energy coefficient.

In Kaplan turbine, the main reason for attach cavity shedding, which is the main cause for blade erosion, is the re-entrant jet. Other than attach cavity on blades, the attach sheets are found near the blade root and hub. In case of PAT, when the machine is run at point away from its BEP the draft tube swirl as well as bubble cavitation are found at the exit.

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