THE INFLUENCE OF THE SUCTION VORTEX OVER THE NPSH AVAILABLE OF CENTRIFUGAL PUMPS

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Lucrarea de față tratează problema mixtă a aspirației de aer în pompele centrifuge şi a influenței acestui fenomen asupra NPSH-ului disponibil. Studiul se bazează pe ipoteza că pătrunderea vârtejurilor cu antrenare de aer din camerele de aspirație în rotoarele maşinilor reduce secțiunea de curgere şi modifică triunghiurile reale de viteze la intrare adăugând o componentă corespunzătoare grosimii siajului. Concluziile prezentului studiu propun completarea formulei de calcul al NPSH-ului disponibil cu coeficienți care țin seama de fenomenele de mai sus fiind prezentat un studiu de caz bazat pe o cercetare anterioara. Se evidențiază modificări semnificative între valoarea calculată cu formula clasică şi cea amendată, de până la 14%. Cercetările ulterioare vor lua în considerare cazul vehiculării de lichide având temperaturi ridicate ceea ce creşte riscul cavitațional.

This paper approach the mixed problem of air absorbed in centrifugal pumps and of its influence over the NPSH available. The study is based on suction section decrease due to vortices from the suction chamber of centrifugal pumps. In Hydraulic machines, this action modify speeds triangles and addition a component specific to the streamlines thickness. Conclusions of this study propose to complete the NPSH available relation by coefficients taking account of above phenomenon. We make also a study case based on a previous research, to demonstrate that the difference between classical relation and our proposed is by 14%.. Other research will made account by fluid rise temperature that increase the cavitations danger

Keywords: vortex in suction chamber, cavitations, centrifugal pumps, NPSH.

1. Introduction

This paper proposes to study the influence of absorbed air on the available NPSH of centrifugal pumps.

In present there are known different methods to reduce the intensity of the vortex flow in suction chambers of centrifugal pumps [1][2][3][4][5]. These methods have 2 objectives:

- to reduce the absorbed air due to the vortex to penetrate inside the pump (as short pipes or with big diameters and complicated installation).

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- to increase the working range of instalation at minimal level. Below this level the pumps could be failed.

Due to reduced flow in tanks and suction basin of centrifugal pumps, it is justified and opportune to find solutions and to design a device for different geometries of the suction chambers, to increase the working domain of the installation of minimal level, with best efficiency.

To avoid destructives effects of the vortices on the pumps working and on the suction basins, we will correlate the working parameters of machine (H,Q) with the static level in suction chamber (H_{st}).

2. Vortex in suction chamber of centrifugal pumps

In suction chambers of centrifugal pumps we pursue to obtain design indication (solutions) to result the minimal level for a steady working of the pumps. Unsteady work changes the parameters and fail the pumps.

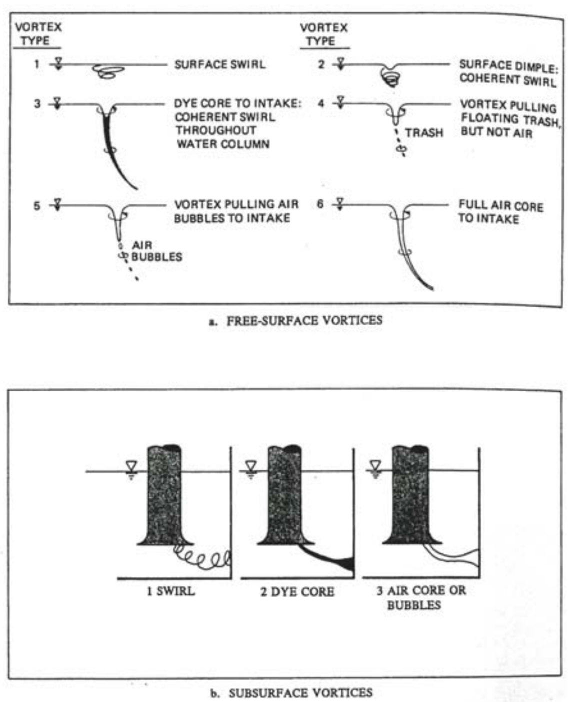

Fig. 1 Vortex strength in suction bell [6]

The tangential velocity component may be measured by a vortimeter, by electronic or manual countering of rotation over a given time. Prerotation can also be determined from velocity transverses obtained with a two directional Pitot tube.

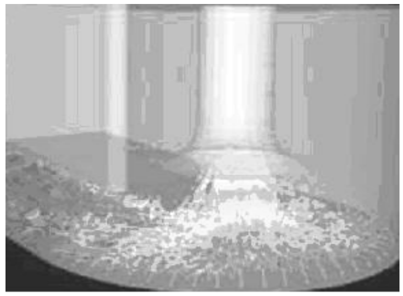

Fig.2 Flow with vortex in suction chamber of pump

3. Corrective measures

Problems in pump sumps such as severe vortex, intense swirl or uneven flow distribution at the suction bell reduced or eliminated by corrective methods. On can try several suitable methods in the model. These are incorporated in the geometry of the pump and include same devices to dissipate the vortices energy. Introduction of horizontal grids below the water surface, changing of wall and floor clearances, improvements in approach channel configuration, and changes in lengths and spacing of piers are some of the common techniques for reducing vortex energy. One of these methods of eliminating vortices problems in suction chamber of centrifugal pumps is presented in figures 3.

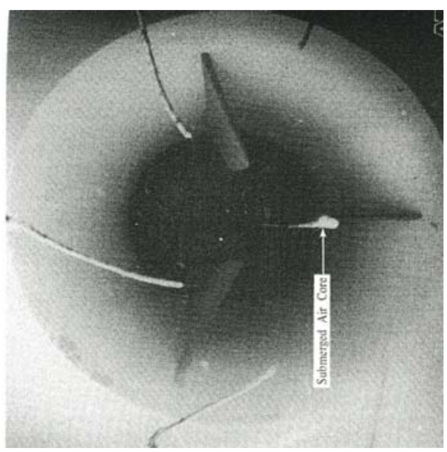

Fig.3 Vortex in suction bell and a solution with radial blade to dissipate its energy

Installing a curtain wall immediately upstream of the pump can also control surface vortices. Installing splinter vanes or floor cones under the bell usually eliminates submerged vortices.

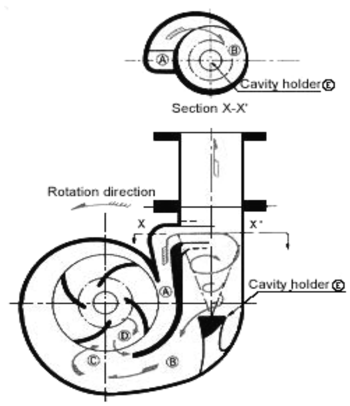

Fig.4 Pump case to dissipate vortices energy, solution by TOKOTA PUMPS

To optimize the pumps working domain at the BEP (best efficient point) is an important objective of pumps manufactures, by adequate geometries. For example, TOKOTA Pumps [7] give a new solution to reduce pumps cavitation and to improve available NPSH. A special case has destroys bubbles (by separating the air from the water and by ousted it), as shows figure 4. Other solutions propose inducers in front of the impellers.

Vortex intensity is mainly dependent of some important parameters: the speed and the flow rate discharge, the static level, and the pressure on the liquid surface from suction chamber, by physico-chemical characteristics of the fluid.

The reduced pressure in the vortex core causes fluctuating load on the pump impeller along with associated vibration and noise, increased possibility of cavitation, higher inlet losses and decreased pump efficiency, when the pump core pressure is sufficiently low to release dissolved air or other gases from the fluid.

4. Cavitation in centrifugal pump

If the pressure of the liquid at any place inside the pump falls below the saturated vapour pressure at the prevailing temperature, small vapour bubbles begin to form and the dissolved gases are evolved. The flowing liquid catches up the vapour bubbles and swept into a region of higher pressure, where they condense violently (time of collapse smaller than 0,003 seconds).

The air absorbed from the vortices arrives in the centrifugal pump and increase the cavitation process. Vortices and prerotation will influence pump performances because the flow approaching the impeller already has a rotational flow field that may oppose or add to the impeller rotation, depending on direction. The design of the pump blades (shape and angle) usually assume no prerotation . The prerotation implies flow separation along one side of the impeller blades. Prerotation could be quantified, determined by detailed velocities measurements.

The swirl decays along the pipe as result of walls friction, turbulence should be located near the impeller.

Places attacked by cavitation in a centrifugal pump are the impeller blades and walls of the casing bounding the liquid flowing through the pump.

Similary studies about unsteady flow with vortex at centrifugal pumps intake and impeller has recently effectuated by K. Kamemoto [5]. He has established in which manner the absorbed air and cavitations bubbles reach through the impeller. Figure 5 shows the flow with absorbed air, for a flow range of approximate 60% from the nominal [5].

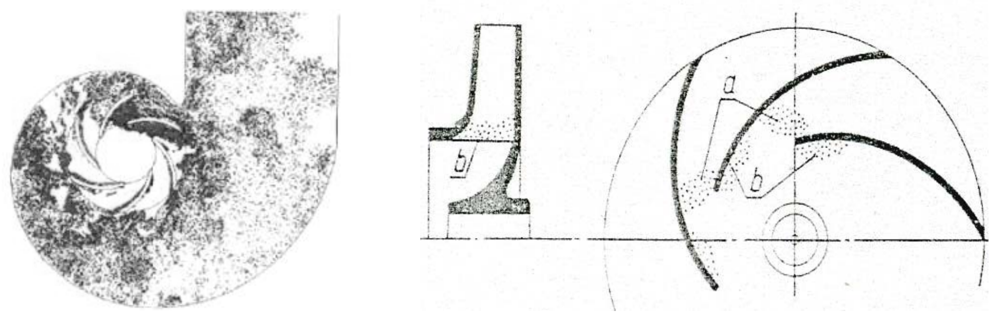

Fig. 5 Absorbed air in the impeller of the pump and pitting due to cavitation in a pump impeller

When cavitation is fully developed, this cause vibrations that are transmitted through the whole pump, even to the foundation. The vibration of the walls of the metal parts attacked by the bursting and rapidly condensing bubbles causes the vibrations of the pump. The noise is produced by the collapse of the bubbles in region of high pressure.

Breaking-off or cut-off of characteristic curves occurs in various ways, depending of specific speed of the pump. At the cut-off point, cavitation is fully developed. At different suction heads, characteristic curves are sown in figure 6. a- D_2=255 mm; b- D_2=165 mm, n=2250 rpm , (1-H_s=0,8 m, 2-H_s=5m, 3-H_s=7m).

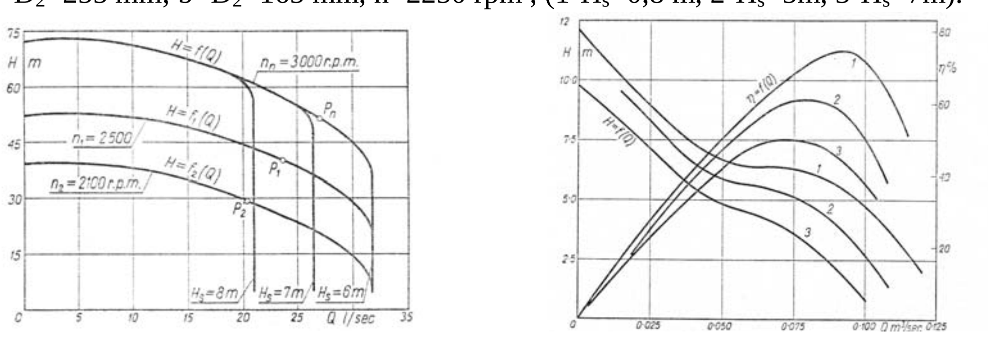

Fig.6 a, b Characteristic curve at different speed and different Hs (suction head)

5. Influence of vortex over the NPSH available of centrifugal pump

The absorbed air will reduce the fluid rate flow coming into pumps. In laboratory of Pumps, fans, blowers and compressors, at Hydraulic and Hydraulics machines Department, we make a research study, proposing a method with a video cam that wills control the film speed. We will make a 3D plainer to determinate the air volume in mass of liquid.

In this research we propose an original method to establish the air-absorbed volume due to the vortices from the suction chamber of centrifugal pump, by photographing the suction nozzle from the flowing axis. Due to different way in which the light pass through the water and the air, the vortices will have a different brightness.

We make a theoretical and experimental research simulating the flow in suctions chamber and intake of pump, also in different hydraulics of impellers; the experimental stand has the possibility to modify the speed and also the flow rate of centrifugal pump, to modify water level and suction nozzle position in suction basin.

5.1 NPSH theoretical approach

Losses caused by vortex disipative devices such as screens, cone, walls, etc. may add up to a value so great that the required NPSH of a pump is not satisfied. Losses lead to insufficient NPSH. Inlet losses in a preliminary design wherein air core vortices and a hight degree of swirl were presented were 20 % higher than in a revised design with no strong vortices , in similar pipe entrance geometry and flows. It is not possible to calculate inler losses reliable, thatfore ussualy they are obtained from model studies. With the experiments result values for inlet losses and the NPSH available should be checked by recalculation.

The study of dependence between cavitation factor and available NPSH of the pumps will be completed by our research.

With Thoma's cavitation factor [6] can give an approximative relation as exemple:

$$\sigma = \frac{NPSH}{H} = \frac{2g \cdot NPSH}{u_2^2} \tag{1}$$

We will try to complete this relation and to define the concordance between energetic parameters of pumps, fluid characteristics, impeller geometry, so:

$$\eta, \Delta H, P = f(Q, d_2, \Omega, \rho, \upsilon, NPSH, p_v, n_s) \tag{2}$$

On demonstrate from Bernoulli equation that:

$$\frac{p_{in} - p_v}{\rho g} \equiv NPSH \Rightarrow p_{in} = \rho g NPSH + p_v \tag{3}$$

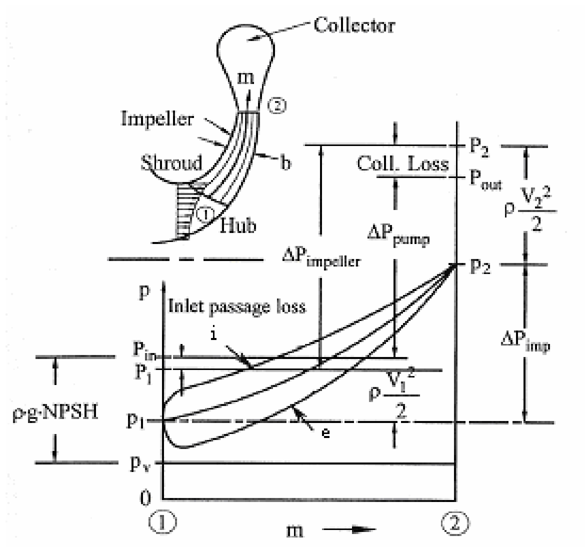

Fig.7 Internal pressure in impeller [6]

$$p - p_v = \rho g NPSH + \frac{1}{2}\rho(u^2 - w^2) - \sum h_i - \rho g(z_2 - z_1) \tag{4}$$

Cavitation parameter is also related to u_2 *and* v_2, and for small and medium pumps can be writes as [6]:

$$NPSH_{3\%} = \left(\frac{v_{1,sh}}{v_2}\right)^n \frac{v_2^2}{2g} + k_2 \frac{w^2{}_{1,sh}}{2g} = k_1 + k_2 \frac{w^2{}_{1,sh}}{2g} \tag{5}$$

$$\sigma = (k_1 + k_2)\left(\frac{v_2}{u_2}\right)^2 + k_2 \Rightarrow \sigma_{3\%} = \frac{NPSH_{3\%}}{\frac{u_2^2}{2g}} \tag{6}$$

With orthogonal inlet, typical breakdown of NPSH give the coefficients:

$$k_1 = 1{,}69 \qquad k_2 = 0{,}102 \tag{7}$$

The value $\frac{v_2}{u_2}$ at BEP or design point rarely exceeds 0,3, regardless of how much NPSH is available.

Because tangential and absolute velocities to outlet depend of the impeller geometry, with small values for β_1 and small number of blades, result a good NPSH available feature.

σ - cavitation parameter; $NPSH$ - net positive suction head,

$NPSH_{3\%}$ - net positive suction head to prevent loss>3%,

u_1, v_1, w_1 - tangential , absolute and relative velocities impeller inlet,

u_2, v_2, w_2 - tangential , absolute and relative velocities impeller outlet,

$v_{1,sh}$, $w_{1,sh}$ - absolute and relative shroud velocities,

$\sum h_i$ - pressure losses in impeller plus in inlet passage,

D_2 – outlet diameter of impeller; z_1, z_2 - elevation coordinate; ρ, ν, p_v - fluid characteristics;

β_1, β_2 - inlet and outlet angles of the blades; Ω - angular speed of impeller.

5.2 Case study

With the stand and equipment of our Pumps, fans, blowers and compressors laboratory, stand composes from suction chambers, centrifugal pumps with variable speed at nominal parameters

Q = 54 m^3/h, H= 28 m at n = 2900 rpm, and

Q = 30 m^3/h, H= 8 m at n = 1550 rpm,

We obtained the characteristic curves (fig.8).

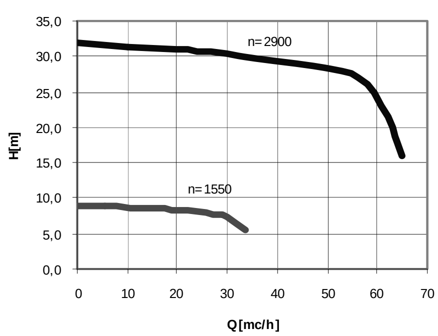

Fig.8 Specific curves with pump from stand in PVSC laboratory - UPB

Taking account by the experimental research [4] concerning air absorbed to intake of the pump, the maximum values for area occurred by air is of 10 % and flow air absorbed in pump is $Q_{asp} = 40{,}27\ cm^3/s$. – 1550 rot/min (fig. 9).

Fig. 9 Sections with air absorbed intake the pump [4], PVSC laboratory - UPB

We established the effect over the NPSH available of pump and we represented this feature in figure 11. The absorbed air rise the NPSH feature with 0,03 m for this size of pump, and the effect of strong vortex was simulate by increasing the intake losses with 20 %, influencing with 0,05 m the NPSH. The percentage of losses is tacking into account the modification of the real velocity triangle at the rotor inlet due to air vortex. One consider the air inlet influence on velocity (pct.1 - fig.7) acting near the suction pipe walls [4]. The shape of the real velocity at inlet is presented in fig.10. The NPSH feature was modified with 14%.

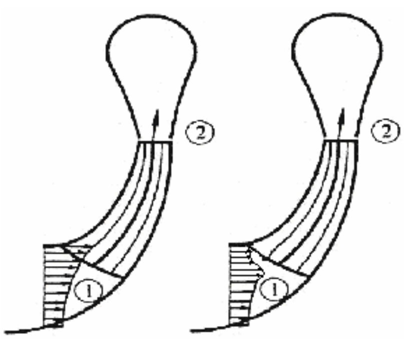

Fig.10 Air inlet velocity influence

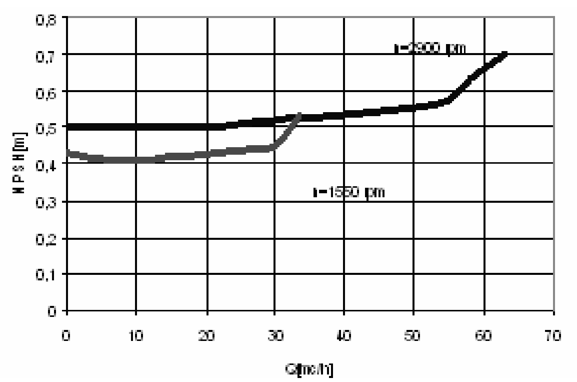

Fig.11 NPSH available feature, influenced by vortices.

6. Conclusion

In our research we propose an original method to establish the air-absorbed volume due to the vortices from the suction chamber of centrifugal pump, by photographing the suction nozzle from axial flowing.

We will continued the theoretical and experimental research simulating the flow in suctions chamber and to intake of pump (with Fluent and CFD application), also in different hydraulics of impellers (with own applications and CFTurbo); the experimental stand has the possibility to modify the speed and also the flow rate of centrifugal pump, to modify water level and suction nozzle position in suction basin. We like to complete the NPSH relation and to define the concordance between energetic parameters of pumps, fluid characteristics, and impeller geometry. Research study will be continued in next papers.

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