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Excitation mechanism of the conventional rotating cavitation in the pump with the three-bladed inducer.

Cavitation in pumps is a result of presence of the critical pressure, which is close to the evaporation pressure, occurs among others at the impeller eye. One of methods of lowering the intensity of the cavitation and even its' elimination in this zone, is application of the inducer, immediately upstream the main impeller. A wrong operation of the inducer could be however, conducted to the occurrence of the so-called rotating cavitation, frequently present in high-speed pumps. In simulations shown, there is introduced a case, when it is induced by an uneven inflow at the three-bladed inducer, conducted to presence of the elbow in the pump suction piping. Results of numerical computations were set together with real measurements, made on the test stand.

Key words: cavitation, inducer, pre-impeller, rotating cavitation, net positive suction head

Denotations

NSPHA	-	required by the suction system, net positive suction head [m]
NPSHR	-	disposable net positive suction head [m]
NPSH3	-	critical net positive suction head [m]
Q	-	pump capacity, [m^3/h]
H	-	suction head of the pump, [m]
n	-	revolutions of the pump shaft, [obr/min]
p_{skr}	-	critical suction pressure, [Pa]
p_v	-	evaporation pressure of the liquid, [Pa]
Δh_{str}	-	loss head, [m]
c_s	-	average absolute velocity of the liquid in the suction piping system, [m/s]
c	-	average absolute velocity of the liquid, [m/s]
g	-	acceleration due to gravity, [m/s^2]
ρ	-	density of the liquid, [kg/m^3]
D	-	outer diameter of the inducer,
f	-	frequency of pressure fluctuations or changes of blades stresses
f_0	-	frequency of the shaft rotation
f_n	-	nominal frequency of the shaft rotation
p_1	-	pressure at the inlet
p_2	-	pressure at the outlet
p_v	-	evaporation pressure of the liquid
U_t	-	peripheral speed of the inducer tip
v_1	-	axial velocity at the inducer eye
δ	-	angle of incidence
σ	-	number of the cavitation
ϕ	-	flow indicator

ψ_s - static pressure indicator

1. Introduction

Cavitation and suitably induced by it, cavitation erosion, is one of the most substantial cause of the hydraulic machinery damage. It is a phenomenon caused by a variable pressure field of the liquid, based on formation, increase and fading of bubbles or other closed areas containing a vapour of a given liquid, gas or a vapour-gas mixture [7]. Cavitation process is initiated in zones of the pressure close to the pressure of evaporation in the given temperature. Characteristic signs of the cavity occurrence are increase of the vibration and a noise in the initial phase of the process, until the break of the flow continuity caused by a large amount of vapour bubbles.

Increase of suction properties of the impeller may be accomplished through application of the inducer. Its' task is to increase the liquid pressure to the level, that guarantees the lack of cavitation and proper operation of the main impeller.

2. Inducers (pre-impellers)

Inducer (pre-impeller) is most often an axial perpetual screw impeller (Fig. 1), which task is to supply an energy to the liquid, in order to create a pressure surplus preventing the formation of the cavitation in the main impeller. Due to the shape of the blade and an axial flow, it is characterized by a lower pressure loss in the zone of a dynamic depression at the blade, therefore better suction properties than the centrifugal impeller.

Fig. 1 Inducer [4]

3. Suction properties of impeller pumps and anti cavitation pressure surplus

Determinant of the pump safety against the operation in the cavitation zone is the height surplus, defined as the lowest pressure value, in reference to the value of evaporation pressure in the given temperature, in which the pump may suck the liquid without the cavitation.

Because of difficulties in assumption of the pressure losses value and the local velocity c, represented by it value of the pressure, is determined empirically and denoted as *NPSH3* - the critical net positive suction head. The critical pressure p_{skr}, is considered as a point, in which the suction head of the pump is going to decrease by 3% in relation to the suction head of the pump operating with a suction pressure that guarantees an operation without cavitation (Fig. 2) [3].

$$\Delta h_{str} + \frac{c^2}{2g} = NPSH3 = \frac{p_{skr}}{\rho g} - \frac{p_v}{\rho g} + \frac{c_s^2}{2g} \tag{1}$$

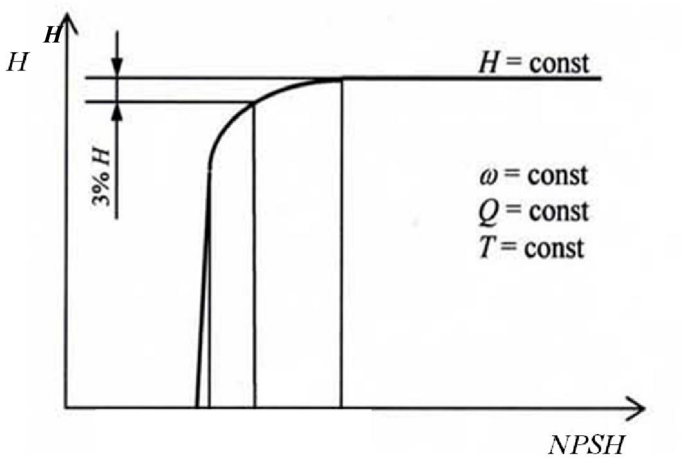

Fig. 2 Curve of the cavitation development in the impeller pump with marked 3% suction head loss [3]

No cavitation operation of the pump requires to guarantee in the suction pipe, a little bit higher pressure, which is considered as the value of NPSHR - required net positive suction head, formulated by multiplication of the NPSH3 by the coefficient k = 1.1÷1.3 dependent of the type and shape of the pump impeller (higher values respond to lover specific speed) [3].

$$NPSHR = k \cdot NPSH3 \tag{2}$$

NPSHA is the required net positive suction head of the pump suction system, defined as:

$$NPSHA = \frac{p_c}{\rho g} - \frac{p_v}{\rho g} - H_{zs} - \sum \Delta h_s \tag{3}$$

where:

p_c - total pressure at the pump inlet,

p_{v} - water vapour saturation pressure of the pumped liquid

H_{zs} - geometrical suction head loss,

$\sum \Delta h_s$ - sum of pressure head losses in the suction piping system.

Cavitation in the impeller pump does not occur, if the pressure surplus in the suction system is higher than the required level by the pump:

$$NPSHA > NPSHR \tag{4}$$

4. Rotating cavitation in inducers

Conventional rotating cavitation, in contrast to the rotating cavitation of a higher order, occurring with higher capacity values than nominal (and manifesting itself with a higher pulsation frequency), forms in zones of the passage, in which occurs a large velocity difference caused by an angle of incidence of blades and a high radius of rotation. This phenomenon is particularly dangerous due to its' unsteady character of the liquid flow and the accompanying vibration of rotating elements of the pump, which very often lead to its' damage by breaking the shaft bearing.

For the first time the rotating cavitation was observed in 1977 by T. Kamija, T. Shimura i S. Watanabe.

In 1996 it as defined that occurrence and the scope of influence of the rotating cavitation is a function of the ratio of number of cavitation (describing the pressure surplus at the pump inlet and the terminal velocity of the biggest radius at the blade tip) and the double angle of incidence of blades as it follows:

$$\frac{\sigma}{2\cdot\delta} \qquad (5)$$

On the basis of subsequent studies, as the beginning of this type of cavitation it was determined an estimate, which is 2.34 (Fig. 3)

$$\frac{\sigma}{2\cdot\delta} \approx 2{,}34 \qquad (6)$$

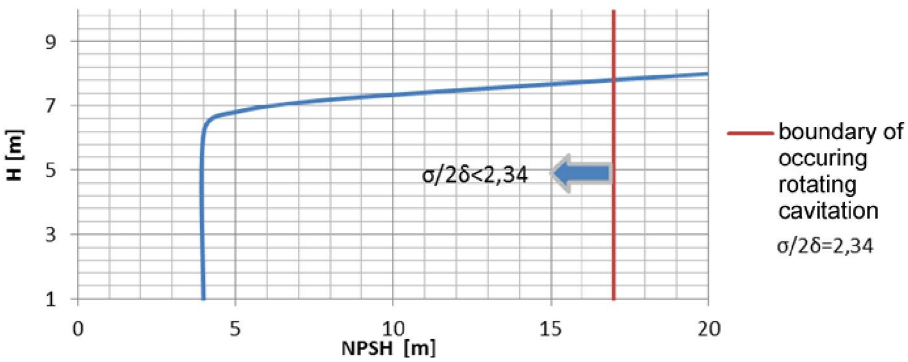

Fig. 3 Example curve of the cavitation development with marked the boundary of occurrence of the rotating cavitation

It can be found in the available literature, few recommendations for constructions preventing the rotating cavitation, inter alia:

- reduction of the inducer radious gap causes a lower number of leaks and leads to the cavitation delay.

- increase of the blade cascade, increase of the angle of incidence (compressing edges) or slope of blades from the back side, causes a decrease of the pressure gradient in the channel part by giving the liquid an angular momentum by the part of the blade which is closer to the hub;

- maximal reduction of the angle of incidence of inducer blades, which significantly decreases the boundary of the rotating cavitation formation - impeller of such a shape has to be equipped with blades increasing the adjustable angle along the blade passage, because in the other case there would not exist a transfer of angular momentum or having two blade cascades of different angles of blades orientation, so called "tandem inducer".

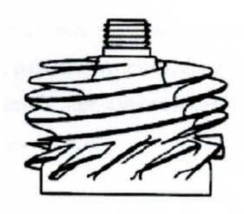

Fig. 4 Inducer of the tandem type with the inclined blade [2]

- in mixed flow pumps, in which was diagnosed the rotating cavitation operation, sometimes there are applied closed impellers to eliminate in that way the influence of flows in the gap for the impeller cavitation, even in this case there is a possibility of making the inducer in a closed form.

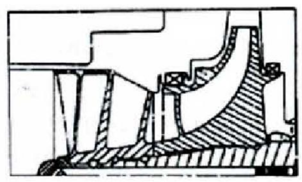

Fig. 5 Pump with the closed inducer [2]

On the basis of the publication [6], in which the authors made a classification of different type of cavitation instabilities, observed during experiments for the impeller pup, there was made a comparative analysis of them in the Table 1.

Table 1 Types of cacitations of the inducer [6]

Type of cavitation	Frequency range	Number of cavitations σ	Occurrence (efficiencies)
Cavitating surge [1]	5-10 Hz	brak danych	no data
Rotating stall [2]	0,5 - 0,7 fn	brak danych	<Qn
Conventional rotating cavitation [2]	1,1 -1,2(1,7) fn; 0,2 fn	0,04-0,06	<Qn, also Qn
Alternate blade cavitation 1	around 1fn bądź 2 fn	0,063-0,065	only for the even number of bklades
Backward rotating cavitation [3]	2 fn	0,065-0,075	<Qn
Higher order cavitation surge [3]	around 5 fn	0,058-0,068	>Qn
Higher order rotating cavitation [3]	5 fn	0,060-0,084	>Qn

1 - cavitation instabilities
3 - cavitation instabilities of the higher order
2 - may occur at the same time

More accurate description of different forms of cavitation, and their founded dependences between geometric and flow parameters, is given in [6].

5. Numerical simulations of the flow in the stage with the inducer-steady calculus

Three-dimensional computations are currently commonly applied in design of pumps. In this study, there was used a commercial software ANSYS-CFX.

Steady calculus was made for the petrochemical single-stage centrifugal pump of the following nominal parameters:

$$Q = 294[m^3/h]$$
$$n = 2974\,[rpm]$$
$$\mathrm{H} = 229\,[\mathrm{m}]$$

Computations were made for the diphase flow pattern. Transition from the liquid to the gas phase of the agent, under the influence of decreasing pressure was simulated using the Rayleigh-Plassets' equation. For simulating the turbulent flow, there was used the SST (Shear-Stress Transport) turbulence model.

The computational zone (computational mesh) was including all impeller channels, which geometry was 7x159536 nodes. In the Fig. 6. there was shown the mesh for a single channel.

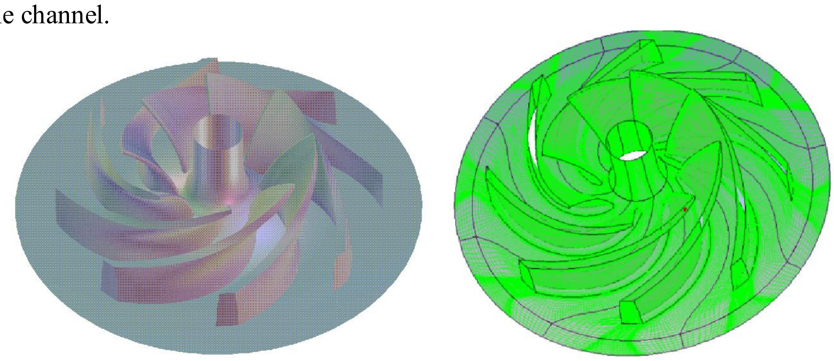

Fig. 6 Geometry of the main impeller and the computational mesh

There was analyzed a flow of the considered inducer with three blades was shown in the Fig. 7. Size of the used computational mesh was 154533 nodes.

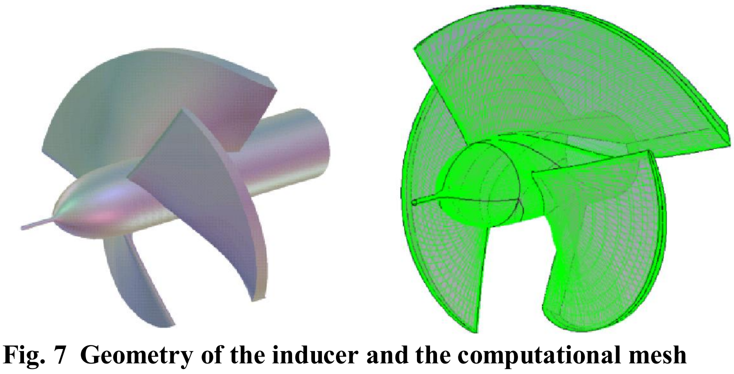

Fig. 7 Geometry of the inducer and the computational mesh

5.1. Physical model and boundary conditions

Computations of anti-cavitation characteristics were made for the real agent (mixture of propane of the volume fraction of 91% and propylene of 9%), which saturated vapour pressure is 16.2 bar for the temperature of 37°C.

In order to determine cavitation characteristic of the pump, computations were made for different values of the suction pressure of the pump with a constant capacity, till the moment of obtaining the suction head loss higher than 3% Computations were carried out for the nominal capacity Q_N and for two configurations of pump suction pipings (Fig. 8):

- A - axial-symmetic inflow,
- B - asymmetric inflow, caused by a presence of an elbow in the suction piping system, immediately upstream the pump inlet (such as it takes place in the real considered case).

  Value of the pump capacity flowing through the pup was one of the boundary conditions at the impeller outlet. Additional boundary condition was the zero pressure gradient at the inlet zone of the computational domain.

For the turbulence model, there were defined two boundary conditions in the form of turbulence intensities at the level of 5% and length of the mixing path equal to the diameter of the intake channel.

5.2. Results of computations of characteristics of the inducer pump

Graphical illustration of computations results was shown in the Fig. 8.

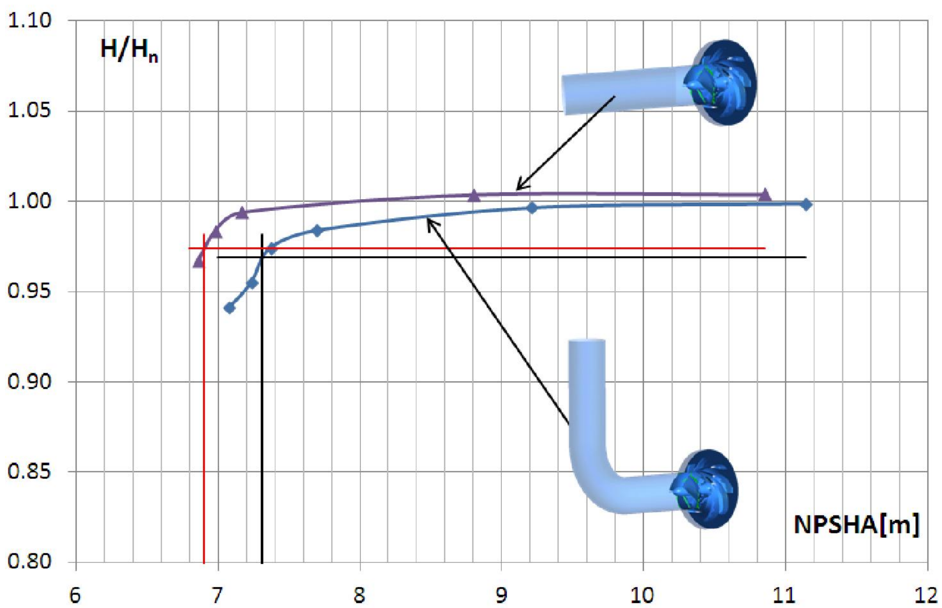

Fig. 8 Characteristics H(NPSH) for the principal impeller cooperating with the inducer for the efficiency equal to 0,9Q_n in the configuration with the suction piping system elbow (A) and without (B).

On the basis of carried out computations and characteristics shown (Fig. 8), one can notice that the anti-cavitation characteristics of the pump are getting worse, that is the value of NPSH3, for the asymmetrical inflow case, caused by a presence of the elbow in the suction piping system.

6. Computation of the cavitation number for the considered inducer

Input data for computations was assumed on the basis of results of the pump measurements with an inducer:

$$Q = 262\frac{m^3}{h}$$

$$p_s = p_1 = 16.7bar$$

$$p_2 = 29bar$$

$$p_v = 16.2bar$$

$$\rho = 485\frac{kg}{m^3}$$

$$D = 0.126m$$

$$f_n = 49.75Hz$$

$$U_t = \pi D f_n = 19.7\frac{m}{s}$$

Cavitation number:

$$\sigma = (p_s - p_v)/(\rho U_t^2/2) = 0{,}53$$

Flow indicator:

$$\phi = \frac{Q}{nD^3} = 0{,}297$$

Indicator of the static pressure:

$$\psi_s = (p_2 - p_1)/(\rho U_t^2/2) = 6.24$$

Verification of the initiation condition of the rotating cavitation:

$$\frac{\sigma}{2\cdot\delta} \approx 2{,}34$$

Angle of incidence on the basis of numerical computations was assumed, equal to: $\delta = 6.8^\circ = 0.118682rad$.

$$\boxed{\frac{\sigma}{2\cdot\delta} = \frac{0.53}{2\cdot 0.118682} = 2.24 < 2.34}$$

It results from these computations that the analyzed inducer operates in the zone of rotating cavitation for assumed operation parameters.

7. Numerical computations of the unsteady flow in the pump level with the inducer

Computations were made for the nominal capacity $0,9Q_N$ equal to 262 m^3/h and the rotational speed 2985 rpm, that is the parameters such as during measurements on the pump stand. Temperature of the liquid - mixture of propane in the volume fraction of 91% and the propylene of 9%, it was assumed a constant temperature T= 37°C, for which the saturated vapour pressure is equal to 16,2 bar.

For unsteady computations it was assumed:

- value of the single timestep: 1,6807E-04 s (so that the average value of the Curant number was around 1.0),
- number of all time steps: 1008,
- a single timestep responds to the rotation of the inducer at about 2,66° by the rotational speed of 2985 rpm,
- total number of rotations, which the inducer made during computations was equal to 7.4 rotations.

Carried out computations allowed to obtain time runs (Fig. 9) for the following values:

- suction head,
- inlet pressure (before the elbow),
- pressures at inlets before the inducer,
- pressures after the inducer (before the main centrifugal impeller eye),
- pressures after the main centrifugal impeller,

Knowledge of timesteps of these values was used to carry out the FTT, which enabeld to inentify harmonics responsible for the conventional rotating cavitation - Table 1.

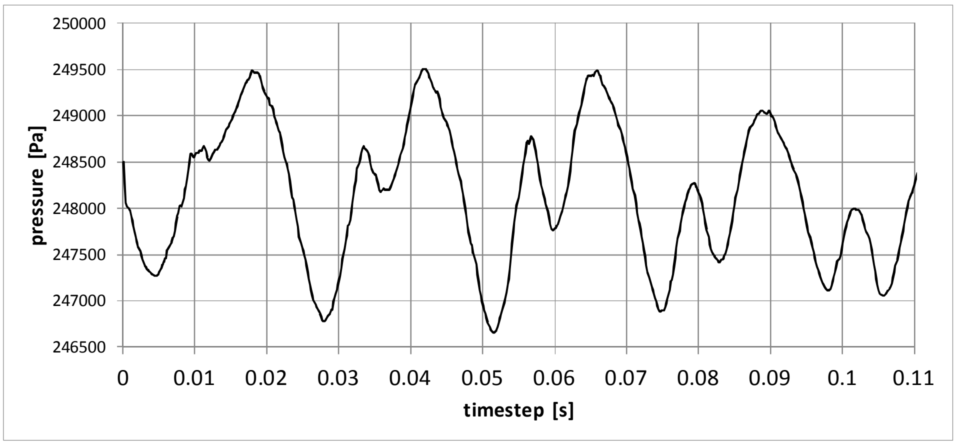

Fig. 9 Time run of pressure fluctuations after the inducer

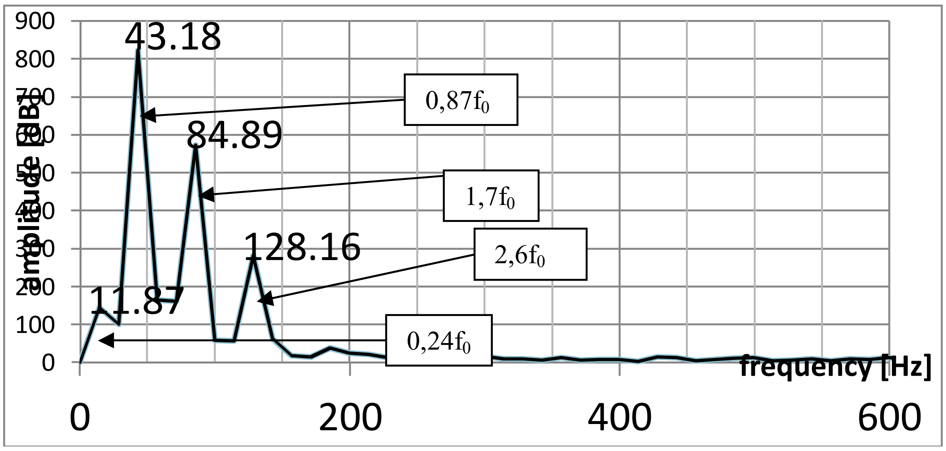

Fig. 10 Time run of pressure fluctuations after the inducer - FFT analysis

On the basis of harmonic analysis of time runs of monitored values (among all inlet pressure, pressure after the inducer), it was found that dominant are harmonic components in values of: 0,2÷0,3f_0; 0,8 f_0; 1,7f_0; 2,6 f_0. Components 0,2f_0 i 1,7f_0 are most likely a result of the rotating cavitation presence in the studied inducer. It is confirmed by studies on the pump stand (Fig. 11), in which these components (1,6f_0) are also present.

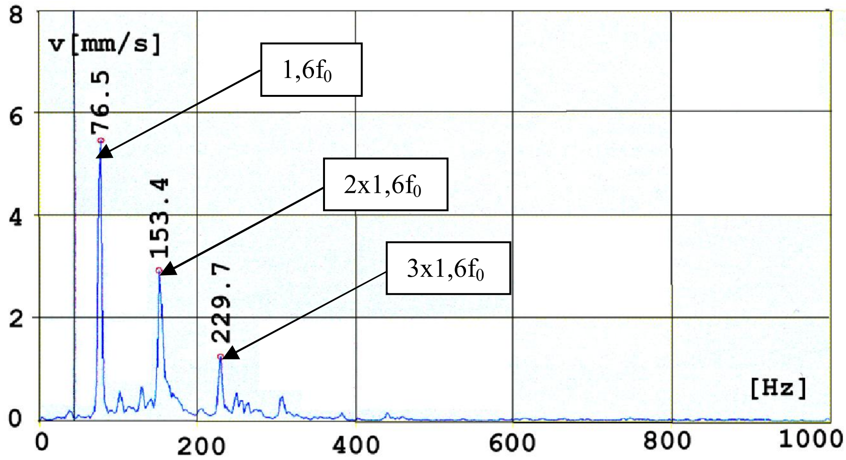

Fig. 11 Spectrum of vibrations measured at the pup hub in the perpendicular direction to the rotation axis (f_0=49,75 Hz=2985 orb/min).

8. Conclusions and summary

Conventional rotating cavitation is formed in zones of the blade passage at the impeller eye, while as there occurs a huge velocity difference, caused by the angle of incidence of blades and the huge rotation radius. This phenomenon is particularly dangerous due to the unsteady character and the accompanied vibration of rotating elements of the pump.

On the basis of carried out steady computations and characteristics shown (Fig. 8) it was noticed that anti-cavitation properties are getting worse - there occurs the increase of NPSH3, in the case of the asymmetrical inflow caused by a presence of the elbow in the suction piping system. This is caused by increase of the intensity of occurrence of vapour-gas bubbles in the blade passage of the pump impeller, causing pressure pulsations.

On the basis of the monitored time runs (among all the inlet pressure, pressure after the inducer), it was stated that harmonic components are dominating in values of: $0{,}2\div0{,}3f_0$; $0{,}8\ f_0$; $1{,}7f_0$; $2{,}7f_0$. Component $0{,}2f_0$ and $1{,}7f_0$ are a symptom of rotating cavitation in the studied inducer. Result of unsteady numerical computations are close to the real, obtained on the pump stand.

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