Modelling of flow with cavitation in centrifugal pump

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Abstract. The paper concerns flow modelling in centrifugal pump with special consideration of cavitation phenomena. Cavitation occurs when local pressure drops below the saturation pressure according to the temperature of the flow. Vapour bubbles are created and then they flow through the areas with higher pressure. The bubbles collapse rapidly generating pressure wave, noise and vibration. Working under cavitation condition is very dangerous to a pump and can significantly shorten its lifetime. The investigated centrifugal pump consists of three two-flow rotors and stators working on a single shaft. The modelling process started with grid independence study. When the grid was chosen, the pump performance curve was obtained using the single phase fluid model. Next, using the results from pump performance curve calculations, the cavitation characteristic was obtained. The constant capacity was held when the pressure at the inlet was reduced. The two - phase model was used with Zwart cavitation model. The results indicate that the pump work in safe range of parameters. The analysis also provides wide range of information about the areas of vapour appearance. The most endangered regions are leading edges of rotor. When pressure at the inlet drops to about one third of pressure that calculations started from the cavitation cloud appears in whole rotor. The intense of vapour bubbles creation is greater near the shroud of the pump, rather than near the hub. As cavitation is strongly unsteady phenomena, the transient calculations were performed to check if the results are close to those obtained using the steady state type. The differences are not significant.

1. Introduction

Computational fluid dynamics provides range of possibilities to predict the flow parameters and to observe the phenomena that can occur under different work conditions. Cavitation can occur in pump flow when the conditions on suction side change (increasing hydraulic losses) or when a pump works with capacity much more higher than the capacity in Best Efficiency Point (BEP). Working under cavitation condition leads to serious damage of blades and walls. In paper the simulation of flow with cavitation in centrifugal pump is described.

The investigated pump consists of three two – flow rotors and stators. They are installed on a single shaft. The pump works on steam turbine condenser cooling water. In 2013 the modernization of pump flow system took place. Three new rotors with enlarged outflow diameter were made and installed so that the cooling water flow through the condenser increased. After the modernization the characteristics of pump changed. The numerical analysis of flow was performed to obtain pump

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performance and efficiency curve and H(NPSH) curve. To include cavitation in the analysis the cavitation model must have been defined.

2. The cavitation model

To simulate flow with cavitation it is necessary to choose appropriate mathematical model. The multiphase model is assumed to be homogenous fluid. In the analysis the cavitation model is based on Rayleigh – Plesset equation describing the growth of a vapour bubble in flow as follows:

$$R_B \frac{d^2 R_B}{dt^2} + \frac{3}{2}\left(\frac{dR_B}{dt}\right)^2 + \frac{2\sigma}{\rho_l R_B} = \frac{p_v - p}{\rho_l} \tag{1}$$

where R_B stands for the bubble radius, p_v is pressure in the bubble (assumed to be equal to the vaporization pressure), p is the pressure in liquid, σ represents surface tension between the liquid and vapor and ρ_l is the liquid density. The surface tension as well as second order terms is neglected in the used model. Therefore the equation (1) is reduced to:

$$\frac{dR_B}{dt} = \sqrt{\frac{2}{3}\frac{p_v - p}{\rho_l}} \tag{2}$$

Rate of change of bubble mass is (assuming vaporization):

$$\frac{dm_B}{dt} = 4\pi R_B^2 \rho_g \sqrt{\frac{2}{3}\frac{p_v - p}{\rho_l}} \tag{3}$$

The vapour volume fraction r_g is defined as:

$$r_g = V_B N_B = \frac{4}{3}\pi R_B^3 N_B \tag{4}$$

where N_B is bubbles per unit volume. According to Zwart model [1] the total interphase mass transfer rate per unit volume is expressed as:

$$\dot{m}_{lg} = N_B \frac{dm_B}{dt} \tag{5}$$

In case of condensation:

$$\dot{m}_{lg} = F \frac{3r_g \rho_g}{R_B}\sqrt{\frac{2}{3}\frac{|p_v - p|}{\rho_l}}\,\mathrm{sgn}(p_v - p) \tag{6}$$

In case of vaporization:

$$\dot{m}_{lg} = F \frac{3r_{nuc}(1 - r_g)\rho_g}{R_{nuc}}\sqrt{\frac{2}{3}\frac{|p_v - p|}{\rho_l}}\,\mathrm{sgn}(p_v - p) \tag{7}$$

where F is an empirical factor which is different for condensation and vaporization because they occur at different rates (condensation is usually slower than vaporization). R_B in equation (6) is replaced by nucleation site radius R_{nuc} and r_{nuc} stands for the volume fraction of the nucleation sites. It was assumed that F_{vap}=50; F_{cond}=0.01; R_{nuc}=1 μm; r_{nuc}=0.0005 [2].

3. The grid independence study

As the rotors of investigated pump work in parallel only flow through one of them was taken to the analysis. Moreover, the rotor is two – flow, so the symmetry in a plane perpendicular to the axis of rotation could be assumed. The flow was divided into 7 parts, as the rotor consists of 7 blades. Analogically, the flow in stator was split into 10 parts. Next the grid independence study was performed. The fluid was assumed to be single phase (liquid). However this assumption leaded to unphysical results near the leading edge of the rotor blade. The dropped below 0 Pa absolute pressure.

It was decided to use the two – phase homogeneous fluid model with cavitation described in chapter 2. It provided the possibility of vapour bubbles creation in the areas with pressure below the vaporization pressure. The data taken to the calculations are described in table 1.
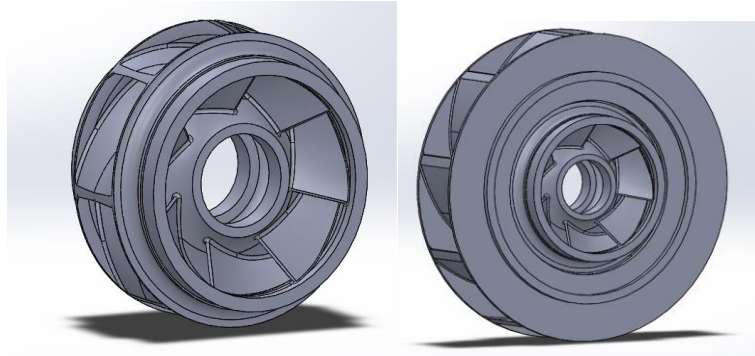

Figure 1. The two–flow rotor (left) and rotor and stator (right).

Table 1. Calculation data to perform the grid independence study.

Angular velocity	985 rev/min
Reference pressure	0 Pa
Heat transfer model	isothermal
Fluid temperature	25°C
Fluid density	997 kg m^{-3}
Vaporization pressure	3574 Pa
Turbulence model	SST
Boundary condition inlet	Static frame total pressure
Turbulence	Medium (Intensity = 5%)
Boundary condition outlet	Mass flow
Mass flow rate	23.08 kg/s

Five grids were studied. For each of them the work parameters were monitored. The pressure distribution on rotor was also compared. The pressure distribution at span 0.5 obtained for each grid is shown in figure 2. The grid with symbol S4 was chosen.

Table 2. Number of nodes for grids taken to the grid independence study.

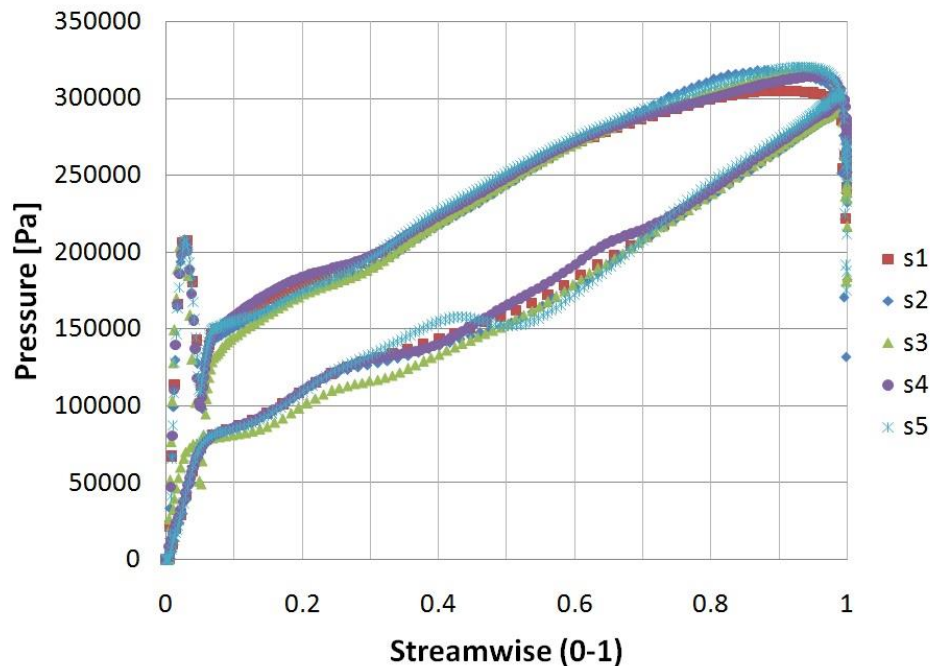| Grid symbol	Number of nodes
S1	693204
S2	967267
S3	1266659
S4	2059135
S5	3568498

Figure 2. The pressure distribution on the rotor blade (span = 0.5).

4. Pump performance curve

To predict new operating point after the modernization of pump flow system it was vital to obtain pump performance curve H(Q). The series of calculations were made. The flow rate was changed from 5000 to 9000 m^3/h with step 500 m^3/h. The intersection of pump performance curve and system curve takes place at the flow rate about 7600 m^3/h. The efficiency curve is relative flat; in range of flow rate from 6500 to 9000 m^3/h it varies from 89.5 to 92.7 %. The highest efficiency occurs when flow rate is equal to 8000 m^3/h.
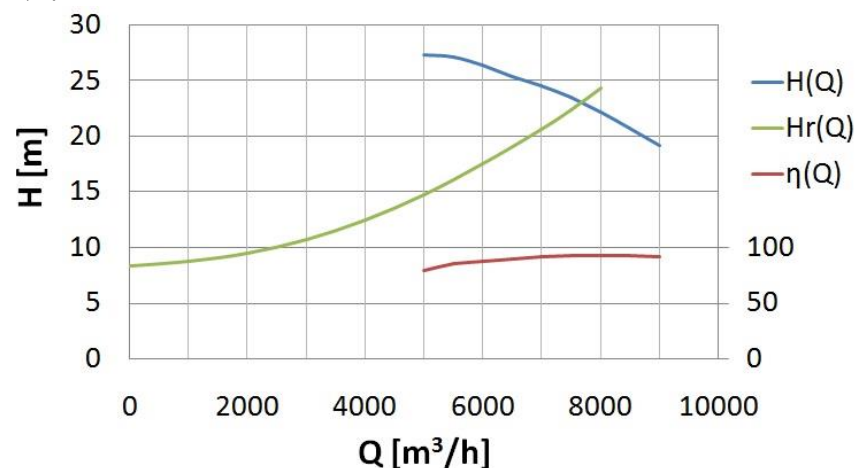

Figure 3. The pump performance curve.

5. Modelling cavitation in the flow

The suction conditions are crucial to avoid cavitation occurrence in a pump. Net positive suction head (NPSH) is a value in hydraulic circuit that indicates the difference between pressure at the inlet (p_s) and vaporization pressure (p_v) [3].

$$NPSH = \frac{p_s - p_v}{\gamma} + \frac{c_s^2}{2g} \tag{8}$$

Where γ stands for specific gravity and c_s is the velocity at the inlet. To preserve pump from working in cavitation condition the NPSH value must be greater than the Required NPSH (NPSHR), usually specified by the manufacturer. NPSHR is defined on the basis of measurements. One of the symptoms of cavitation is a change in pump characteristics. It means that if a pump works with constant flow rate and cavitation occurs, the head of pressure drops significantly. To determine the critical value of NPSH corresponding to the initial cavitation, the H(NPSH) curve is plotted.

5.1. Simulation to obtain H(NPSH) curve
The H(NPSH) curve was plotted at the flow rate equal to 8000 m³/h. The value of NPSH was gradually decreased by lowering the inlet pressure. The Head of pressure was monitored. The data used to calculations are the same as specified in table 1. The H(NPSH) curve is shown in figure 4.
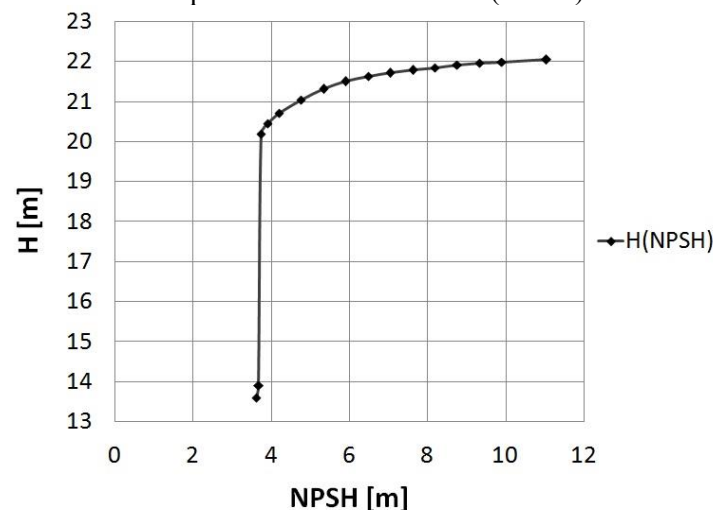

Figure 4. The plotted H(NPSH) curve.

The critical value of NPSH is defined as NPSH for which the Head drops to 97% of its initial value. The Required NPSH is greater than critical NPSH, usually about 10-30% [4]. Initial value of NPSH in the described case was 11,03 m and critical NPSH is about 5,35 m. It indicates that the pump works in a safe range of parameters. Lowering the inlet pressure causes Head to decline, but the changes are smooth. The rapid drop of Head occurs when inlet pressure is decreased to about one third of initial pressure.

5.2. Areas of vapor appearance in rotor
The lowest pressure in a rotor occurs near the leading edge of the blade. In this area the vapour bubbles may appear even during the work with NPSH significantly higher than NPSHR. In figure 5. the area of vapour bubbles appearance is compared for four values of inlet pressure. The cloud of vapour bubbles firstly occurs near the leading edge of the rotor blade and then gradually enlarges as the inlet pressure is decreased. The cloud was located only in the rotor. In the stator there was no vapour in the whole range of inlet pressure. When inlet pressure reaches the value of one third of the initial inlet pressure the cavitation cloud covers whole area beyond rotor blades. However the vapour distribution through the blade span is not uniform. In figure 6 the area of vapour occurrence at lowest inlet pressure is compared for different values of blade span. Most of the cavitation cloud gathers near the shroud of the rotor (span greater than 0.5). This distribution tallies with available data on cavitation

damage in centrifugal pump [5, 6]. The most endangered place is the blade surface near the shroud. The results show that the used model correctly predict this phenomenon in rotor.
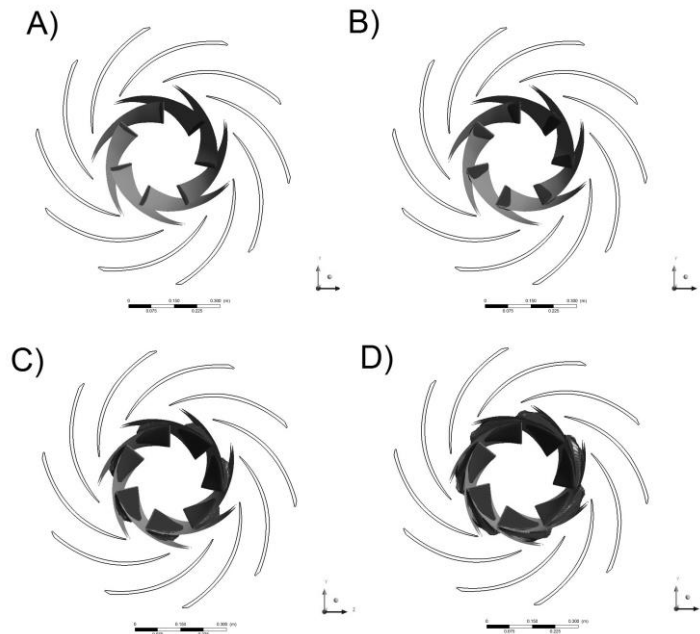

Figure 5. Areas of vapour bubbles appearance in the pump rotor. A) Inlet pressure 110 kPa B) Inlet pressure 55 kPa C) Inlet pressure 40 kPa D) Inlet pressure 39 kPa.
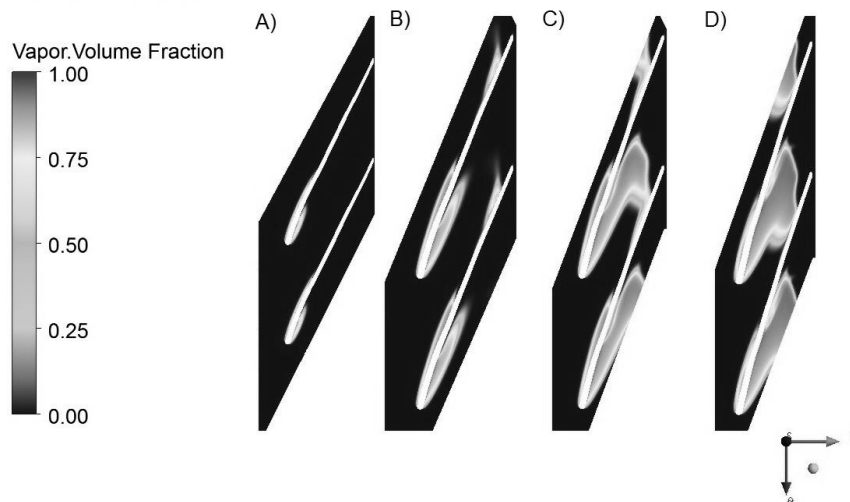

Figure 6. Areas of vapour bubbles appearance in the pump rotor. A) Span 0.2 B) Span 0.5 C) Span 0.7 D) Span 0.9.

5.3. Transient calculations

Cavitation is strongly unsteady phenomena. The process of vaporization and condensation of the bubbles proceed in very turbulent way. The results shown above were obtained with the assumptions of steady state type of calculations. It definitely shortened the time of calculations. The next step was to recalculate a few of the cases above and check if the results from transient calculations are similar to those obtained with the assumption of steady state. The chosen points to recalculations was the points characterized by the inlet pressure: 110 kPa (100% initial inlet pressure), 55 kPa (50% initial pressure), 40 kPa (36% initial pressure) and 39 kPa (35% initial pressure). The results are shown below in figure 7. The values of pressure in both rotor and stator are changing periodically and as a consequence the values of H are changing as well. The values of H shown below are calculated as the average in one period. The differences of H values are specified in table 3.
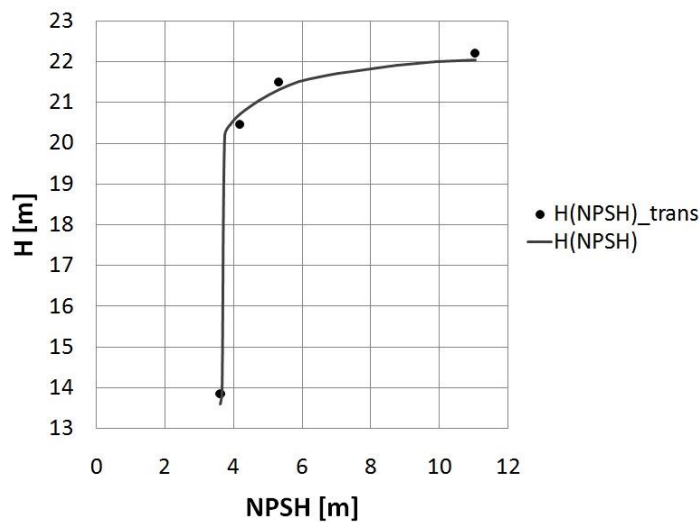
Figure 7. Transient results compared to steady state results.

Table 3. H values for different type of calculations.

NPSH, m	H steady state, m	H transient, m	Relative difference, %
11.03	22.05	22.20	0.67
5.33	21.32	21.50	0.85
4.19	20.73	20.45	1.33
3.62	13.60	13.86	1.90

After analysing differences between transient and steady state results it was affirmed that characteristic H(NPSH) on the basis of transient results would be very close to the curve obtained after steady state calculations. The differences are below 2%. In this case to predict cavitation occurrence in a pump it is sufficient to use steady state type of calculations.

To study pressure changes in time the monitor points were added in both rotor (point A) and stator (point B). The point in the rotor was located near the trailing edge. The point in the stator was far from the area of vapour appearance. In the figures 8 and 9 the values of pressure in both points are plotted in condition of flow without and with cavitation.
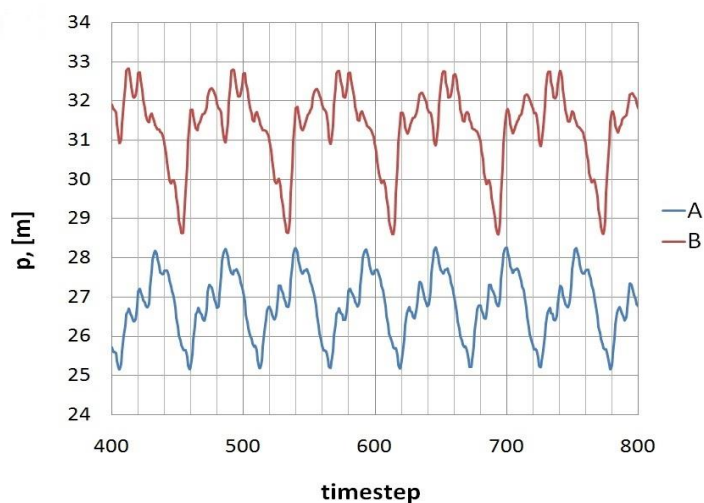

Figure 8. The changes of pressure values in point A and B in condition of non – cavitating flow.
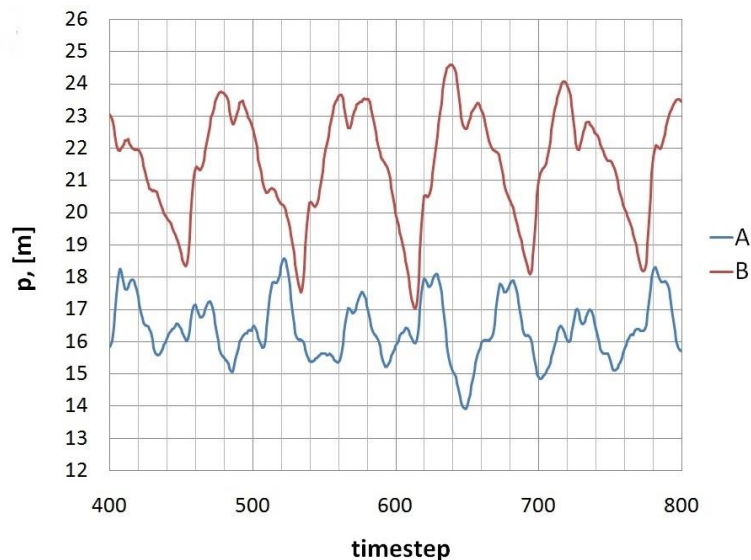
Figure 9. The changes of pressure values in point A and B in condition of cavitating flow.

Frequency of changes in both points was related to the rotational velocity and geometry of the stage. In case of point A this frequency was about 170 Hz, in case of point B – about 115 Hz. However, in case of point A during the flow with cavitation the additional frequency was observed. This frequency was equal to 96.5 Hz and most probably resulted from cavitation model that was used during the calculations.

6. Conclusions

The performed analysis of the flow in a centrifugal pump enabled to predict new working conditions after the modernization of pump flow system and define pump performance curve. For modelling characteristic of pump the single phase fluid model was sufficient, but it gave unphysical results of pressure in the areas where bubbles of vapour should appear. To study cavitation phenomena it was necessary to use two-phase fluid model with appropriate model of cavitation. In the analysis the H(NPSH) curve was also plotted. It indicated that there is no risk of cavitation during work with specified flow rate, because operating NPSH is almost two times greater than critical NPSH corresponding to the initial cavitation. In the case of developed cavitation the location of vapor cloud in the rotor is described. Most of the cloud gathered near the shroud of the rotor. Transient type of calculations was also performed. The results do not vary significantly from the results attained by steady state type of calculations.

7. Literature

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