Effect of semi-open impeller side clearance on centrifugal pump cavitation inception

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Abstract
Cavitation in pumps is the most severe condition that centrifugal pumps can work in and is leading to a loss in their performance. Herein, the effect of semi-open centrifugal pump side clearance on the inception of pump cavitation has been investigated. The input pump pressure has been changed from 80 to 16 kPa and the pump side clearance has been changed from 1 mm to 3 mm at a rotation speed of 1500 rpm.

It has been shown that as the total input pressure decreased; the static pressure inside the impeller is reduced while the total pressure in streamwise direction has been reduced, also the pump head is constant with the reduction of the total input pressure until the cavitation is reached. Head is reduced due to cavitation inception; the head is reduced in the case of a closed impeller with a percent of 1.5% while it is reduced with a percent of 0.5% for pump side clearance of 1 mm, both are at a pressure of 20 kPa. Results also showed that the cavitation inception in the pump had been affected and delayed with the increase of the pump side clearance; the cavitation has been noticed to occur at approximate pressures of 20 kPa for side clearance of 1 mm, 18 kPa for side clearances of 2 mm and 16 kPa for 3 mm.

Keywords: Impeller Side Clearance, Centrifugal Pump Performance, semi-open impeller, cavitation inception.

1. Introduction
Cavitation is a common phenomenon in centrifugal pumps which significantly affects their performance due to the process of formation and collapse of vapor bubbles, mainly at the pump inlet. The formation of the bubbles is attributed to the drop in the static pressure at the pump inlet eye to become lower than the local vapor saturation pressure, while the sudden collapse takes place when they enter zones of higher local pressures than that of saturation pressure. This cycle of formation and collapse of bubbles causes a shock and creates small pits and erosion on the suction side of leading edge of the impeller blades. There are many factors influencing cavitation inception and development of which are undissolved air present in the liquid, flow velocity, and surface characteristics.

Many researchers oriented their studies toward the investigation of the cavitation in pumps aiming to simulate the steady and unsteady cavitating flow to shed more lights on this important phenomenon. Up to the authors’ knowledge, there is no effort has been added to investigate the effect of the pump side clearance on the centrifugal pump cavitation inception and performance. Somashekar and Purushothama (2012) studied the cavitating flow through the blade passage using ANSYS- CFX in a centrifugal pump and compared their results with experimental data. Their results showed that the inception of cavitation is taking place on the suction surface where the leading edges meet the tip. Kim et al. (2012) investigated the centrifugal pump computational domain to improve the prediction accuracy of pump cavitation. While in Brennen (2013) introduced a review paper for the recent developments in pump cavitation from the point of view of dynamics and instabilities.

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Jun et al. (2013) studied cavitation model based on Rayleigh-Plesset in steady-state and transient for centrifugal pumps and compared pump performances in case of non-cavitation and cavitation conditions. Liu et al. (2013) presented a modified k-ω model with cavitation model to model pump cavitation and their results demonstrated that the application of this model can improve the prediction accuracy. Tao et al. (2014) introduced how cavitation inception performance could be optimized in a pump by changing blade geometry. Their results showed that flow separation near leading edge was weakened and postponed to downstream with optimization. Tan et al. (2015) showed that pump inlet guide vanes have a significant influence on cavitation performance while Xie et al. (2015) showed that blade slots could enhance the pump cavitation performance. Farid et al. (2015-a) studied the effect of pump side clearance on performance for different side clearance width and different flow rates have and they showed that impeller side clearance has a great regression effect on performance. George and Muthu (2016) showed the effects of blade number, inlet and outlet pressures on pump cavitation. Rakibuzzaman et al. (2016) studied the effect of pump rotation speed on the cavitation flow while Fu et al. (2016) showed the impeller radial force fluctuations for cavitating flow in pumps.

Ji et al. (2017) investigated the cavitation in the centrifugal pump and showed the effect of the inlet diameter on the cavitation performance. Zhang et al. (2017) studied the unsteady flow in cavitating flow in a centrifugal pump in nuclear power plants and showed the vapor distribution, velocity fluctuation, and pressure fluctuation in the flow domains. Also, Zhu and Chen (2017) studied cavitation formation and evolution for unsteady pump flow in and compared their results with experiment. While in Ye et al. (2017) investigated the cavitating flows in a centrifugal pump at different flow rates and the cavity lengths under different pump inlet pressures.

The effect of centrifugal pump side clearance, the gap between impeller face and casing, on centrifugal pump cavitation inception is carried out using ANSYS-CFX. The 3D pump volumes; inlet, impeller and volute casing, have been introduced computationally to introduce some lights on cavitation phenomenon inside the pump and how is significant the effect of side clearance on pump performance. The study is carried out at the best efficiency point flow rate. In section two, the computational domains and governing equations are introduced. In section three, results have been illustrated and the conclusions have been addressed in section four.

2. Numerical study:

The 3D complete centrifugal pump domains; inlet, impeller and volute casing, are modeled and discretized using the CFD as described in [3 - 4]. The main aspects of the pump impeller are listed in Table 1.

| Table 1: Pump impeller dimensions [6, 17]. |
|----------------|----------|----------------|
| Blades Number (Z) | 6        | Blade discharge angle (β₂) | 20°|
| Rotational Speed (n) | 1500 rpm | Blade inlet angle (β₁) | 35°|
| Impeller Inlet Diameter (D1) | 50 mm | Blade height (b) | 15 mm|
| Impeller Outlet Diameter (D2) | 130 mm | Blade thickness (t) | 2.5 mm|

Here, the computational domains in [3 – 4] have been used and described with unstructured mesh in the current study. The modeled pump is introduced using the ANSYS-CFX as shown in Figure 1 and 2.
Fig. 1: Centrifugal pump computational domains [3 - 4].

Fig. 2: Mesh generation in ANSYS [3 - 4].

The computational domains meshes have been checked for the solution dependency as described in [3 - 4], pump head has been monitored at the pump outlet with different mesh counts and plotted in Figure 3. For this study, the unstructured mesh has been refined and tailored near the impeller blades to capture the full details of the cavitation occurrence in the pump impeller. Total of 951453 tetrahedral cells have been used from which; 288670 mesh cells used for the impeller and the inlet domain while 662783 mesh cells used for the volute. An additional 100000 mesh cells have been used for side gab domain.

Fig. 3: Mesh sensitivity analysis.

2.1 Flow Governing Equations:

The main flow governing equations, the continuity and momentum equations, for a steady state, homogeneous Newtonian and compressible mixed medium of water vapor and water liquid in the Cartesian coordinates could be written as follows [20-23]:

\[ \frac{\partial \rho m u_i}{\partial x_j} = 0 \]  

(1)
\[ \frac{\partial}{\partial x_j} (\rho_m u_i u_j) = \frac{\partial p}{\partial x_j} + \frac{\partial}{\partial x_j} (\tau_{ij}) \]  

(2)

where \( u_i \) and \( x_j \) is the instantaneous velocity and position, \( p \) is the pressure and is \( \rho_m \) is mixture density according to the volume fraction of each phase. The viscous stress tensor, \( \tau_{ij} \), is obtained as follows:

\[ \tau_{ij} = \mu_m \left( \frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i} - \frac{2}{3} \frac{\partial u_k}{\partial x_k} \right) \]  

(3)

The mixture density \( \rho_m \) and the mixture dynamic viscosity \( \mu_m \) are defined as follows [5]:

\[ \rho_m = \rho_\ell \alpha_v + \rho_v (1 - \alpha_v) \]  

(4)  

\[ \mu_m = \mu_\ell \alpha_v + \mu_v (1 - \alpha_v) \]  

(5)

where \( \rho_\ell \) is the liquid density, and \( \mu_\ell \) and \( \mu_v \) are the vapor viscosity and liquid viscosity, respectively, and \( \rho_v \) is the vapor density; \( \alpha_v \) is the vapor volume fraction. After applying Reynolds time averaged process to the main governing equations, they have the following forms:

\[ \frac{\partial \rho_m u_i}{\partial x_j} = 0 \]  

(6)  

\[ \frac{\partial}{\partial x_j} (\rho_m u_i u_j) = \frac{\partial p}{\partial x_j} + \frac{\partial}{\partial x_j} (\tau_{ij} - \rho_m (u_i u_j)) \]  

(7)

where \( U_i \) is the mean flow velocity in direction \( i \), and \( u' \) is the fluctuating component of the velocity. This system of equations is called Reynolds averaged Navier-Stokes Equations, RANS, and the term appeared \( \rho_m (u_i u_j) \) should be modeled. For the turbulence modeling, the Reynolds stress model has been used. The liquid-vapor mass transfer due to cavitation is governed by the vapor volume fraction transport equation [5]:

\[ \frac{\partial (\alpha_v \rho_m u_j)}{\partial x_j} = R_e + R_c \]  

(8)

where \( R_e \) and \( R_c \) are the mass transfer rates related to the evaporation and condensation in cavitation, respectively. For the cavitation modeling, the Rayleigh-Plesset equation model is used where a spherical bubble of radius, \( R(t) \) in an infinite domain of liquid whose temperature and pressure far from the bubble are \( T_\infty \) and \( p_\infty \) respectively and where \( T_\infty \) is assumed to constant [23].

The liquid density and the dynamic viscosity are constant and uniform. In the absence of mass transport across the boundary the generalized Rayleigh-Plesset equation for bubble dynamics is as follows in the absence of the surface tension and viscous terms [23]:

\[ \frac{p_B(t) - p_\infty(t)}{\rho_L} = R \frac{d^2 R}{dt^2} + \frac{2}{3} \frac{dR}{dt} \left( \frac{dR}{dt} \right)^2 + \frac{4 \mu_L}{R} \frac{dR}{dt} + \frac{2S}{\rho_L R} \]  

(9)

where \( S \) is the Surface tension, \( p_B \) is the bubble pressure and the equation is solved for \( R(t) \). The bubble contains some gas whose partial pressure is \( p_\infty \) at size, \( R_0 \), and temperature, \( T_\infty \). Then, \( p_\infty \) is formulated as follows:

\[ p_B(t) = p_v(T_B) + p_{Go} \left( \frac{T_B}{T_\infty} \right)^3 \left( \frac{R_0}{R} \right)^3 \]  

(10)

With assuming the thermal term and is equal to zero for constant liquid temperature. The Rayleigh-Plesset equation in the following general form \( s \) follows:

\[ \frac{p_B(T_B) - p_{\infty}(t)}{\rho_L} + p_{Go} \left( \frac{R_0}{R} \right)^3 = R \frac{d^2 R}{dt^2} + \frac{2}{3} \frac{dR}{dt} \left( \frac{dR}{dt} \right)^2 + \frac{4 \mu_L}{R} \frac{dR}{dt} + \frac{2S}{\rho_L R} \]  

(11)

where \( k \) is constant. The centrifugal pump head, \( H \), is calculated as follows:

\[ H = \frac{P_{out} - P_{in}}{\rho g} \]  

(12)

where \( P_{out} \) is the total pressure of volute outlet, \( P_{in} \) is the total pressure of pump inlet, \( \rho \) is density of the fluid, and \( g \) is the gravity acceleration. The required net positive suction head (NPSH) of centrifugal pump can be obtained by the following equation:

\[ \text{NPSH} = \frac{P_{in} - P_v}{\rho g} \]  

(13)

where \( P_v \) is the vapor pressure of fluid at the given temperature.

### 2.2 Boundary Conditions:

The boundary conditions for studied computational domains; inlet, impeller and volute casing, are set as shown in Figure 4. Since the flow problem is considered 3D steady, incompressible, inhomogeneous problem, the input is total
pressure and is varied from 100 to 16 kPa in order to introduce the cavitation. The output is mass flow rate and is selected to be the mass flow rate at the best efficient point, 3.5 kg/s. Fluid is water at 25°C with zero inlet vapors for the cavitation case. A MFR with frozen rotor condition rotating and non-rotating (multi-frame of reference) is applied at the interfaces between the domains.

Figure 4: Computational domain boundary conditions.

No-slip condition is applied on all wall surfaces and standard wall function has been implemented. The impeller rotational speed is 1500 rpm and turbulence model is Reynolds stress model with maximum inlet turbulence intensity of 5%. For the convergence, solution of cavitating pump is initialized with non-cavitating one. The solution residuals have been plotted as shown in Figure 5, with a maximum number of iteration of 700 while the convergence criterion is of the order of 1e-7. The numerical results for non-cavitating semi-open pump are compared to the experimental results and plotted in Figure 6 for side clearance of 1 mm at the same operating conditions of 1500 rpm and best efficiency point flow rate [10, 20].

Figure 5: Convergence criteria for the solution (residuals and point monitored “head”)

Using assumption of no volumetric leakage, 10% difference between the CFD head and the experimental head is accepted and the performance; head and efficiency, showed a good agreement with that from numerical simulation.

3. Pump Cavitation Inception Results:
The effects of pump side clearance on pump performance in cavitating flow have been carried out by changing input pump pressure from 80 to 16 kPa and pump side clearance from 1 mm to 3 mm.

3.1 Varying pump input pressure:
Pump total input pressure has been changed from 80 to 16 kPa at pump rotational speed of 1500 rpm for a semi-open pump side clearance of 1 mm to introduce its effect on the pump performance with cavitation. The results are represented in contour forms in a cutting plane at a span of 0.5 of the distance between hub to shroud. Static pressure contours, as shown in Figure 7, and total pressure contours in a streamwise direction, as shown in Figure 8, have been
plotted for the different total input pressures. The velocity vectors, as shown in Figure 9 for the different total input pressures. All results clipped to the same ranges. It has been shown that as the total input pressure decreased; the static pressures inside the impeller are reduced and are equal in all impeller passages while the total pressure in the streamwise direction has been reduced. There are no effect has been noticed on velocity vectors with the decrease of the total input pressure.

Fig. 6: Experimental vs CFD for side clearance $e=1\text{mm}$.
Fig. 7: Static pressure contours for different total input pressure at side clearance of 1 mm.

Pt = 20 kPa  
Pt = 18 kPa  
Pt = 16 kPa

Fig. 8: Total pressure contours in streamwise contours at pump side clearance of 1 mm.

Pt = 20 kPa  
Pt = 18 kPa  
Pt = 16 kPa

Pt = 80 kPa  
Pt = 70 kPa  
Pt = 60 kPa

Pt = 50 kPa  
Pt = 40 kPa  
Pt = 30 kPa
To study the effect pressure on pump performance with side clearance in case of cavitation, a direct comparison has been plotted between static pressure for both closed-impeller and semi-open-impeller with side clearance of 1mm as shown in Figure 10. It has been noticed that the static pressure is higher in the case of a closed pump from that of the semi-open with side clearance in cavitation due to the ability of pressure to be redistributed across impeller side clearance.

In Figure 11, pump head variations and in Figure 12, NPSH variations have been plotted with total input pressure for both cases of closed-impeller and semi-open-impeller with side clearance of 1 mm. In Figure 13, a direct comparison between the vapor volume fraction of 10% for both cases of closed-impeller and semi-open-impeller side clearance of 1 mm has been plotted to show the effect of the side clearance on the vapor formation due to cavitation existence. It has been noticed that the cavitation is higher in the closed pump rather than that in the semi-open-impeller with side clearance. This may be interpreted as in the semi-open-impeller with side clearance and due to the impeller clearance, the flow can redistribute from the pressure side to the suction side through this gap and hence the pressure on the pressure side is decreased little and the pressure on the suction side is increased. This may lead to that cavitation in
leading edge of suction side is less in semi-open-impeller when compared to closed-impeller.

Fig. 10: Static pressure contours for different total input pressure for a closed and semi-open pump with side clearance of 1 mm.
Fig. 11: Pump head for closed and semi-open impellers with side clearance of 1mm at different total input pressure.

Fig. 12: NSPH for closed and semi-open impeller with side clearance of 1mm.

It has been noticed also that the head is constant with the reducing of the total input pressure until the cavitation is reached, the head is decreased. At this point the head is starting to decrease at approximate pressure of 20 kPa for the closed-impeller and with a percent reduction in the head of 1.5%. While for the semi-open-impeller with side clearance of 1mm, the head is noticed to decrease at approximate pressure of 20 kPa also but with a percent reduction in the head of 0.5%.

Consequently, it could say that the reduction of pump head due to cavitation in the case of the semi-open-impeller is lower than that in the case of closed-impeller but one has to take into consideration that the head in case of semi-open-impeller is already lower than that for closed-impeller. It has been noticed also that head is constant with the reducing in NSPH until the cavitation is reached at a value of 1.68, the head is started to decrease for both cases.
It has been noticed that small cavities of water vapor can be clearly seen on the suction side attached to impeller blades’ leading edge cavitation evolution. With decreasing the total input pressure, it is observed that this vapor cavity is increased in the case of closed-impeller rather than that in semi-open-impeller. In addition, it is noticeable that the share of vapor fraction is not equal in all impeller passages in both cases, closed and semi-open, due to the existence of the volute and its effect on the impeller characteristics and generally these water vapor cavities are responsible for reduction in pump head.

3.2 Varying pump side clearance:
In this section, different centrifugal pump side clearances have been applied to study their effects on the pump cavitation inception. The pump side clearance is varying from 1 to 3 mm with different pump total input pressure from 20 to 16 kPa and at the same rotational speed of 1500 rpm. In Figure 14, static pressures contours have been plotted for the different side clearances while in Figure 15, the velocity vectors have been plotted and all results have been clipped to the same range.
It has been noticed that the static pressure is decreased with the increase of the semi-open-impeller side clearances and this may lead to delay in cavitation inception in the semi-open pump rather than that in closed-impeller. In Figure 16, pump head variations with total input pressures have been plotted for different impeller side clearance for sake of comparison, the closed pump has been also plotted. It has been noticed that the pump head is constant with the reducing of the total input pressure until the cavitation is reached for different side clearances.

The cavitation inception has been noticed for different impeller side clearance as follows; for side clearance of 1 mm cavitation is reached at an approximate pressure of 20 kPa as the closed-impeller, for side clearance of 2 mm cavitation is reached at an approximate pressure of 18 kPa and for side clearance of 3 mm cavitation is reached at an approximate pressure of 18 kPa.

Fig. 14: Static pressure contours for different total input pressures and different impeller side clearances.
Fig. 15: Velocity vectors for different total input pressures and different impeller side clearances.

The cavitation is noticed to be delayed with the increase of the impeller side clearance. This could be illustrated from the plotting of the vapor volume fraction of 10% for different impeller side clearance as shown in Figure 17. It could notice that the effect of the side clearance on the cavitation inception and hence the vapor formation is to delay the mechanism of the cavitation generation.
Fig. 16: Pump head for closed and for different impeller side clearances.
4. **Conclusions:**

The effect of impeller side clearances in centrifugal pumps on the cavitation inception are carried out using ANSYS-CFX for different total input pressure from 80 to 16 kPa and different side clearance from 1 to 3 mm at the same pump rotation and flow rate.

It has been noticed the following:

- Static pressure is higher in the case of a closed pump than that of the semi-open with side clearance in cavitation due to the ability of pressure to be redistributed across impeller side clearance.
- The cavitation is higher in the closed pump rather than that in the semi-open-impeller with side clearance. This may be interpreted as in the semi-open-impeller with side clearance and due to the impeller clearance, the flow can redistribute from the pressure side to the suction side through this gap and hence the pressure on the pressure side is decreased little and the pressure on the suction side is increased.
- Head is starting to decrease at approximate pressure of 20 kPa for the closed-impeller and with a percent reduction in the head of 1.5%. While for the semi-open-impeller with side clearance of 1mm, the head is noticed to decrease at approximate pressure of 20 kPa also but with a percent reduction in the head of an approximate of 0.5 %.
- Vapor cavities be clearly seen on the suction side attached to impeller blades’ leading edge cavitation evolution. With decreasing the total input pressure, it is observed that this vapor cavity is increased in the case of closed-impeller rather than that in semi-open-impeller. In addition, it is noticeable that the share of vapor fraction is not equal in all impeller passages in both cases, closed and semi-open.
- The cavitation inception has been noticed for different impeller side clearance as follows; for side clearance of 1 mm cavitation is reached at an approximate pressure of 20 kPa as the closed-impeller, for side clearance of 2 mm cavitation is reached at an approximate pressure of 18 kPa and for side clearance of 3 mm cavitation is reached at an approximate pressure of 18 kPa.
- The cavitation is delayed with the increase of the impeller side clearance and the effect of the side clearance on the cavitation inception and hence the vapor formation is to delay the mechanism of the cavitation generation.

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