ABSTRACT

The cavitation inception behavior of mixed-flow impellers has been investigated parametrically using a geometric impeller model and a three-dimensional potential flow model.

The computed cavitation inception is validated by cavitation inception measurements. Computed and measured results show a good agreement for a wide range of operating conditions.

A number of mixed-flow impellers has been developed using the same design strategy in order to investigate the influence of the number of blades and the leading-edge position (forward/backward swept) on cavitation inception.

It is found that a larger number of impeller blades results in a flatter cavitation inception curve for flow rates below the best cavitation point. Furthermore, the cavitation inception performance of the forward-swept impellers is superior to the backward-swept impeller.

NOMENCLATURE

\( N P S H \) net positive suction head [m]
\( X \) meridional coordinate [m]
\( Q \) flow rate [m³/s]
\( H \) head [m]
\( C \) constant [–]
\( n_\omega \) dimensionless specific speed \( \frac{\Omega}{\sqrt{gH}} \) [–]
\( r \) radial coordinate [m]
\( z \) axial coordinate [m]
\( x \) meridional coordinate [m]
\( s \) dimensionless meridional coordinate [–]
\( g \) acceleration of gravity \([m/s^2]\)
\( P_0 \) total inlet pressure \([N/m^2]\)
\( p_v \) vapor pressure of the fluid \([N/m^2]\)
\( v \) absolute velocity \([m/s]\)
\( f \) factor [–]

Greek symbols
\( \Omega \) angular velocity of impeller [rad/s]
\( \beta \) local blade angle [rad]
\( \delta \) angle [rad]
\( \theta \) angular coordinate [rad]
\( \rho \) density of the fluid \([kg/m^3]\)

Subscripts
\( i \) construction line index
\( l \) leading-edge
\( t \) trailing-edge
\( m \) meridional
\( opt \) best efficiency point
\( bcp \) best cavitation point

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1 INTRODUCTION

In order to minimize size, weight and cost centrifugal and mixed-flow pump impellers are required to operate at the highest possible rotational speed. The rotational speed is limited by the phenomenon of cavitation. Cavitation occurs around an impeller blade when the local static pressure falls below the vapor pressure of the liquid being pumped.

The parameter which characterizes the cavitation performance is the Net Positive Suction Head \(NPSH\), defined as

\[
NPSH = \frac{P_0 - p_v}{\rho g}
\]

where \(P_0\) is the total inlet pressure at the impeller inlet corresponding to cavitation, \(p_v\) the vapor pressure, \(g\) the acceleration of gravity and \(\rho\) the density of the fluid. When the total pressure \(P_0\) is decreased from a cavitation-free flow condition, an inlet pressure will be reached where cavitation first occurs. This corresponds to inception \(NPSH\). A further reduction of the inlet pressure can cause a reduction of the delivered head. A head reduction of 3% is referred to as \(NPSH_{3\%}\).

Many authors (Avva et al (1995), de Lange (1996) and Dreiß et al (1996)) have proposed methods for dealing with the characteristic two-face phenomenon of cavitation. However, cavitation erosion damage is most likely to occur between inception \(NPSH\) and \(NPSH_{3\%}\) (McNulty, Pearsall (1982)). It also appears that for high energy pumps cavitation inception predominantly determines the safe operating conditions (Stoffel (1995)). Thus, prediction of cavitation inception is an important issue of the impeller design process.

Previously used impeller design methods for optimization of inception \(NPSH\) are based on simple one-dimensional flow calculations along mean streamlines. The analytical aspects of these impeller inlet design methods are described in Pearsall (1973), Bunjes (1974), Noskiević (1976) and Adrizzon, Pavesi (1995). In these methods empirical coefficients are used to describe the effects of incident angle, blade thickness and non-uniformity of the inlet velocity profile. Adrizzon and Pavesi (1995) have calculated some of these coefficients by applying a quasi three-dimensional flow analysis. Cooper et al (1991) introduced a new method, known as “biased-wedge blading”, for designing impeller blades with a minimal drop in static pressure on the blade surface. This pressure drop leads to the formation of cavitation. This method uses a combination of experimental observations and computational flow analysis.

However, these quasi three-dimensional approaches fail for fully three-dimensional flow patterns occurring in the impeller inlet. Therefore a fully three-dimensional potential flow analysis is performed here to predict cavitation inception. Such an approach allows pump designers to predict the cavitation inception behavior of the impeller with more confidence.

The impeller geometry is described using the geometric impeller model presented by van Os et al (1996). With this geometric impeller model the optimization process of the impeller design is greatly simplified. Modifications to the geometry can be described in a simple and flexible manner and input files for the potential flow analysis package are prepared automatically.

This approach of a geometric model in combination with a three-dimensional potential flow analysis package is validated by comparing computed inception \(NPSH\)-values with experimental results.

In order to investigate the effect of the number of blades and the position of the leading-edge on the cavitation inception performance a number of mixed-flow impellers has been developed using the same design strategy.

2 IMPELLER GEOMETRY DESCRIPTION

The geometric impeller model is based on the Kaplan drawing method (Stepanoff (1957)) and cubic-spline functions. The description of the geometry of an impeller blade consists of two parts. The first part represents the definition of the camber surface, including leading- and trailing-edge, by means of a number of camber lines that are located on different positions between hub and shroud. By means of a cubic-spline surface through these lines the camber surface is constructed. In the second part the pressure and suction surfaces are obtained from the camber surface by means of a thickness distribution perpendicular to this surface. By using different thickness distributions for the pressure and suction side an impeller blade with an asymmetric profile can be modeled. Further details are described in van Os et al. (1996).

Figure 1 shows three camber lines on the impeller blade camber surface. The projection of a single camber line on the meridional plane, the plane \(\theta = \text{constant}\), is called the construction line. The position along the construction line is described by the meridional coordinate \(X_m\), see figure 2.
The curvature of the camber line is described by the local blade angle $\beta$, defined as (Stepanoff (1957))

$$\tan(\beta) = \frac{dx_m}{rd\theta}$$

(2)

Starting from a construction line determined by the $r$, $z$-coordinates in the meridional plane and a blade angle distribution $\beta(x_m)$, the angular coordinate $\theta$ is computed from:

$$\theta(x_m = X_m) = \theta(x_m = 0) + \int_{0}^{X_m} \frac{dx_m}{r(x_m) \tan(\beta(x_m))}$$

(3)

An example geometry of a mixed-flow impeller ($n_w = 1.6$) is shown in figure 3 and 4. The blades have a constant thickness $0.017 \times D$, with $D$ the impeller diameter, and the blade noses are rounded elliptically. Figure 3 shows the meridional plane with the intersection lines $\theta = $ constant and the plane view of the mixed-flow impeller blade.

3 THREE-DIMENSIONAL COMPUTATION OF CAVITATION INCEPTION

The inception $NPSH$ is computed by using a fully three-dimensional potential flow model. This model is based on the assumption of incompressible, irrotational and inviscid flow ($Re \gg 1$, $Ma^2 \ll 1$) within the impeller channel. This is a reasonable assumption for well-designed impellers in which no flow separation occurs.

The governing Laplace equation is solved numerically using the finite element method (FEM). This method is well suited to modeling very complex blade geometries. A detailed description of the numerical method is given in Kruyt et al. (1996).

The flow is simulated within one of the impeller blade passages. Figure 5 shows a FEM-mesh of a mixed-flow impeller channel. The domain consists of an inlet region upstream of the impeller inlet, a single blade passage, and an outlet region extending downstream. The mesh contains $39432$ nodes. A detailed description of the boundary conditions is given by van Esch et al (1995).

The three-dimensional pressure field is computed from the Bernoulli equation in the rotating frame of reference. Observation of iso-pressure surfaces indicates the occurrence and the position of cavitation inception. The $NPSH$ inception is computed from the minimum pressure.

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A complete cavitation inception analysis takes less than one hour of CPU time on a SGI Power Challenge computer, using one processor, or less than three hours on a Pentium 90 MHz PC. The maximum internal memory use is 60 Mb.

4 VALIDATION OF THE COMPUTED CAVITATION INCEPTION

Cavitation inception measurements have been performed both visually by means of stroboscopic illumination and acoustically by means of hydrophones. Figure 6 shows computed and measured cavitation inception values for the mixed-flow impeller shown in figure 4.

Comparison of the computed and measured results shows a good agreement for 0.9 < Q/Q\text{opt} < 1.7, where Q\text{opt} is the best efficiency point. Furthermore, the predicted positions on the blade surface where cavitation inception occurs are in agreement with the visual observations of the cavitation inception measurements.

5 PARAMETRIC CAVITATION INCEPTION ANALYSIS

A previous parametric cavitation inception study by van Os et al. (1996) showed that for identical camber surfaces a larger number of blades or an increase of the blade thickness cause a reduction of the best cavitation point (Q_{bcp}). Furthermore, thickening the nose profile at the suction side (asymmetric blade profile) reduces the inception NPSH drastically for flow rates below Q_{bcp}, whereas for higher flow rates the NPSH-value is the same.

In the current parametric study a slightly different approach is followed in order to investigate the effect of the number of blades and the leading-edge position. All analyzed impellers are designed according to the same design method. Hence they are designed for the same flow rate and they are better comparable.

Impeller design strategy

The design strategy is based on a one-dimensional flow model. Hub, shroud, trailing-edge, blade angles at the trailing-edge and blade profile are identical to those of the mixed-flow impeller shown in figure 4.

The inlet blade angles are computed from:

\[
\tan(\beta_l) = f \times \frac{r_{\text{mid}}}{r_l} \times \cos \delta_l,
\]

with \(\beta_l\) the blade angle at the leading-edge, \(r_{\text{mid}} = \sqrt{r_l + r_s}\) the computed meridional velocity, \(\Omega\) the angular velocity, \(r\) the radius on the leading-edge, \(\delta\) the angle between the meridional velocity and the construction line, \(f\) a factor, and \(l\) the index for the construction line number.

The meridional velocity is computed from a meridional flow analysis that takes into account the number and the thickness of the impeller blades.

Neumann (1991) proposed a factor \(f\) of 1.05 - 1.25 in order to improve the performance of a pump for flow rates larger than the best efficiency point. Here a factor \(f = 1.05\) is used.

The variation of the local blade angles between leading- and trailing-edge has been chosen similarly for all impellers. At the leading- and trailing-edge \(\frac{\delta s}{s} = 0\) and from leading- to trailing-edge the local blade angle is obtained from:

\[
\beta_l(s) = \begin{cases} 
\frac{\beta_{bcp} - \beta_{s}}{s} - \frac{\beta_{bcp} - \beta_{s}}{2} \cos \left( \frac{s}{2} \right); & s \leq C \\
\frac{\beta_{bcp} - \beta_{s}}{s} - \frac{\beta_{bcp} - \beta_{s}}{2} \cos \left( 1 + \frac{s}{2C} \right) \frac{s}{2}; & s > C 
\end{cases}
\]

with \(\beta_{bcp}\) the blade angle at the trailing-edge, \(s\) the dimensionless meridional coordinate and \(C\) a constant defining the point of maximum gradient of the blade angle variation. For all developed impellers \(C\) lies between 0.35 - 0.65.

A side effect of this design strategy is that the camber surfaces differ slightly at the leading-edge. Due to the unchanged blade angles at the trailing-edge the blade curvature at the exit is more or less similar for all developed impellers.

Using this design strategy a number of impellers has been developed from a single reference impeller. The impeller with four blades and the same leading-edge position as the mixed-flow impeller is called the reference impeller. Figure 7 shows the meridional plane and the plane view of this reference impeller. The reference impeller differs slightly from the mixed-flow impeller of figure 4 at the leading-edge in the plane view, since the inlet blade angles differ.

The other developed impellers differ from the reference impeller in the number of blades and the position of the leading-edge.
Effect of the number of blades

Figure 8 shows the computed cavitation inception for the developed mixed-flow impellers with three, four, five and six blades respectively.

The three-bladed impeller has the steepest NPSH curve for flow rates below $Q_{bcp}$. This is caused by a reduced guidance of the flow around the impeller nose in comparison with the four-, five- and six-bladed impellers. This result is similar to that presented by van Os et al. (1996) for different number of blades with identical camber surfaces.

From figure 8 it can be observed that the six-bladed impeller has the lowest $Q_{bcp}$, followed by the five-, four- and three-bladed impellers. Only the five- and six-bladed impellers have a $Q_{bcp}$ which approaches the design flow rate.

The difference between $Q_{bcp}$ and the design flow for the four- and three-bladed impellers is attributed to blockage and a phenomenon called “potential-flow vortex”, see Visser et al. (1994). The effect of this vortex on the inlet velocity near the leading-edge is reduced when the number of blades is increased. Due to this vortex, the condition of shockless flow is reached at a higher flow rate for impellers with fewer blades.

Effect of the position of the leading-edge

Three impellers with different leading-edge positions have been designed. Figure 9 (top) shows the meridional plane of these impellers. The position of the leading-edge on the hub has been kept the same, whereas this position on the shroud differs. For impeller A this position lies more upstream, while for impeller C it lies more downstream than the reference impeller referred to as impeller B.

Figure 9 (bottom) shows the plane view of the three impellers. Notice that impeller A and B are of the forward-swept type and impeller C of the backward-swept type.

Figure 10 shows the computed cavitation inception for these three impellers. The forward-swept impellers (A and B) have better cavitation inception performance than the backward-swept impeller (C). For flow rates below $Q_{bcp}$ cavitation inception occurs at half-span of the leading-edge for the forward-swept impellers. For the backward-swept impeller it occurs close to the shroud. This is in agreement with the results obtained by Bunjes (1974) and Brown (1993). They described a positive effect of the forward-swept design on cavitation inception. The forward-swept design reduced the “blade loading” at the shroud side.

The difference between the cavitation inception performance of the two forward-swept impellers is small, with impeller A showing a slightly better cavitation inception performance.

6 DISCUSSION

Three-dimensional potential flow analysis has been used for predicting cavitation inception of mixed-flow pump impellers. Computed and measured cavitation inception for the mixed-flow impeller are in good agreement for a wide range of operating
Impeller A

Impeller B

Impeller C

Figure 10: Effect of position leading-edge on cavitation inception performance.

Figure 10: Effect of position leading-edge on cavitation inception performance.

conditions. Thus the potential flow solution, which can be obtained very quickly, can be used for a parametric design of impeller inlets.

Observation of the cavitation inception behavior of the three-, four-, five- and six-bladed impellers shows that the \( NPSH \) curve is less steep for the four-, five- and six-bladed impellers compared to the three-bladed one. The lower \( Q_{bccp} \) for a larger number of impeller blades is attributed to blockage and the “potential-flow vortex”.

The cavitation inception performance of the forward-swept impellers is superior to the backward-swept impeller. This is due to a reduced “blade loading” at the shroud for the forward-swept impeller design. Therefore cavitation inception for the forward-swept impellers occurs at half-span of the leading-edge, while it occurs close to the shroud for the backward-swept impeller.

The results obtained so far suggest that the cavitation inception performance of the mixed-flow impeller can be improved by using a forward-swept impeller design, a larger number of blades and an asymmetric nose profile.

Future efforts will concentrate on further potential flow analysis for various inlet configurations in order to improve cavitation inception performance. Design parameters which will be investigated are impeller inlet diameter, inlet blade angle and the development of the local blade angle \((\frac{d}{2N_{loc}} \neq 0)\) close to the leading-edge.

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REFERENCES


