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Effects of the short blade locations on the anti-cavitation performance of the splitter-bladed inducer and the pump☆

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ABSTRACT

In order to evaluate the effects of the short blade locations on the anti-cavitation performance of the splitter-bladed inducer and the pump, 5 inducers with different short blade locations are designed. Cavitation simulations and experimental tests of the pumps with these inducers are carried out. The algebraic slip mixture model in the CFX software is adopted for cavitation simulation. The results show that there is a vortex at the inlet of the inducer. Asymmetric cavitation on the inducer and on the impeller is observed. The analysis shows that the short blade locations have a minor effect on the internal flow field in the inducer and on the external performance of the pump, but have a significant effect on the anti-cavitation performance. It is suggested that the inducer should be designed appropriately. The present simulations found an optimal inducer with better anti-cavitation performance.

1. Introduction

Nowadays, centrifugal pumps are often required to run at high speed conditions. As a result, they are prone to low efficiency, cavitation, and low stability. In order to get high anti-cavitation performance of pumps, an inducer is always designed and placed upstream of the impeller to resist the cavitation. The structure of the splitter blades is an optimal design for inducers [1–3].

At present, many research works have been done on the impeller with splitter blades. Yang *et al.* [4] studied the influence of splitter blades on the anti-cavitation performance of a double suction centrifugal pump, and found that the splitter blades could improve the anti-cavitation performance of the pump. Cui *et al.* [5] calculated three-dimensional turbulent flow in a centrifugal pump with a long-mid-short blade complex impeller, and found that the back flow in the impeller has an important influence on the performance of the pump. Yuan *et al.* [6] also found that the splitter blades can reduce the pressure fluctuations. The simulation of Kergourlay *et al.* [7] indicated that the splitter blades had a positive role in improving the internal flow and hydraulic performance of a centrifugal pump. There are other researches on the effect of the impeller with splitter blades on the performance of centrifugal pumps [8–10].

The splitter blades must be designed carefully with reasonable length, number and angle. Shigemitsu *et al.* [9] studied splitter blade parameters of the low specific speed centrifugal pump impeller. They developed a design method to select the number, off-setting angle, inlet diameter, and deflection angle of splitter blades. Yang and Miao [11] investigated the effect of splitter blades' main geometry factors on the performance of pumps as turbines, including circumferential biasing degree, outlet deflection angle, outlet diameter and the number of blade. Yamada *et al.* [12] researched two types of impeller with different numbers of splitter blades. Solis *et al.* [13] reduced pressure fluctuations by adding splitter blades to the original impeller and by increasing the radial gap between the splitter impeller and the volute tongue. Golcu [14] found that the splitter blade length and blade number were important, and optimized them for a deep well pump.

Splitter blades are applied not only in pump impellers but also in other turbo-machines. The structure is proved to be beneficial for performance improvement [15–18]. Recently, the splitter-bladed inducer is used more and more wildly. Like the splitter-bladed impeller, the geometrical parameters including length, tip clearance, number and screw pitch will affect the anti-cavitation performance of the pump. Several works of simulation and experiment on centrifugal pumps with an inducer are carried out to investigate the effect of an inducer's parameters on the pump performance [1,19–23]. However, only a limited number of studies can be found concerning the effects of short blade locations on the anti-cavitation performance of the splitter-bladed inducer and the pump.

Most studies are mainly focused on the single-phase flow research and hydraulic analysis of the splitter-bladed inducer. In present work, two-phase flow is simulated. The vapor volume fraction distributions

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on the inducer and on the impeller are both analyzed. Results of simulations and experimental tests are compared correspondingly. Effects of the short blade locations on the anti-cavitation performance of the splitter-bladed inducer and the pump are disclosed.

2. Numerical Simulation of Cavitation

2.1. Pump prototype

The centrifugal pump and the splitter-bladed inducers with different short-bladed locations are investigated in this work (shown in Fig. 1). The original design parameters are: flow rate $Q_d = 4\ m^3 \cdot h^{-1}$, head $H_d = 100$ m, rotational speed $n_d = 6000\ r \cdot min^{-1}$, and specific rotation speed $n_s = 23.08$. The other main geometric dimensions are shown in Table 1.

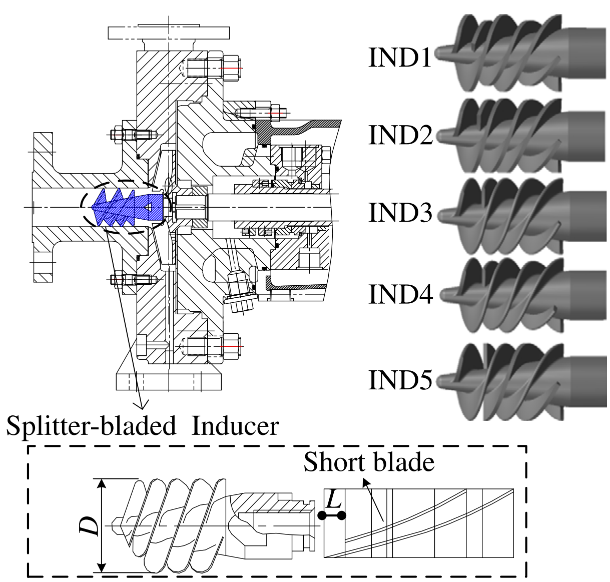

Fig. 1. Pump and five splitter-bladed inducers with different short blade locations.

Table 1
Main geometric dimensions of the pump with splitter-bladed inducer

Parameter	Value
Suction diameter D_i/mm	40
Discharge diameter D_o/mm	40
Blade number of inducer	4 (two long and two short)
Blade number of impeller	8
Impeller inlet diameter D_1/mm	28
Impeller outlet diameter D_2/mm	118
Thickness of the impeller blades at the tip W_1/mm	4
Thickness of the impeller blades at the root W_2/mm	5.5
Thickness of the inducer blades W_3/mm	1
Helical pitch S_i/mm	$S_i = 97.8\ \tan\beta_i\ (15.5° < \beta_i < 28°)$

In Fig. 1, the parameter D is the diameter of the inducer, and L is the distance from the tip of the short blade to the tip of the long blade. To observe the effects of the short blade locations on the anti-cavitation performance of the inducer and the pump, five inducers are designed. They will be denoted IND1–IND5, respectively hereafter. The short blade location is configured in Table 2.

Table 2
Locations of the short blades

	IND1	IND2	IND3	IND4	IND5
L/D	0.15	0.3	0.45	0.6	0.75

2.2. Computational domain and grid

Fig. 2 shows the three-dimensional computational domain and grid. The clearance of the blade's tip and the pump is not considered in the present simulation. The inlet pipe and the volute outlet are extended properly to reduce the effect of boundary conditions on the internal flow. The commercial code GAMBIT is used to generate the meshes. Tetrahedral meshes are chosen in the inducer and the impeller domains, while hexahedral meshes are chosen in the inlet pipe and volute domains. The grid sizes of the pump with the various inducers are listed in Table 3. In order to simulate more accurately, mesh independence is analyzed on the case of IND1. The result is given in Table 4. From Table 4, at the case where the mesh interval size is less than 0.5 mm, the head is relatively stable. Thus the mesh interval size is chosen as 0.5 mm. 'EquiAngle Skew' and 'EquiSize Skew' of all grids are less than 0.85, therefore, the grid quality is satisfactory.

2.3. Mixture model

In order to explore the cavitation mechanism in the splitter-bladed inducer and the impeller, cavitation flow is numerically calculated. During simulation, a physical model is based on the assumption that the mixture of water and vapor in a cavitating flow is a homogeneous fluid. The Reynolds average N–S approach is used for turbulent flow in this work. A mixture model is adopted, and the number of the phases is set as two. The two phases are considered as water and vapor [24, 25]. As the inlet pressure and outlet pressure are higher than the saturation pressure, the vapor volume fraction is assumed to be zero at the inlet and at the outlet of the pump. The liquid phase is water under the standard condition. Equations of continuity and of momentum conservation are

$$\frac{\partial \rho}{\partial t} + \frac{\partial}{\partial x_j}\left(\rho u_j\right) = 0 \quad (1)$$

$$\frac{\partial}{\partial t}(\rho u_i) + \frac{\partial}{\partial x_j}\left(\rho u_i u_j\right) = -\frac{\partial P}{\partial x_i} + \frac{\partial}{\partial x_j}\left(\mu \frac{\partial u_i}{\partial x_j}\right) \quad (2)$$

with

$$\rho = \alpha_w \rho_w + \alpha_v \rho_v. \quad (3)$$

Volume fraction equation for the vapor phase is

$$\frac{\partial}{\partial t}(\alpha_v \rho_v) + \frac{\partial}{\partial x_i}(\alpha_v \rho_v u) = -\frac{\partial}{\partial x_i}\left(\alpha_v \rho_v u_{dr,v}\right). \quad (4)$$

The above equations are formulated in terms of the mass-averaged mixture velocity u and drift velocity of the vapor phase $u_{dr,v}$, which are defined as follows, respectively:

$$u = \frac{\alpha_w \rho_w u_w + \alpha_v \rho_v u_v}{\rho} \quad (5)$$

$$u_{dr,v} = \frac{(\rho - \rho_v) d_v^2}{18 \mu_c f} \cdot [g - (u_v \cdot \nabla u_v)] - \frac{1}{\rho} \sum_{i=1}^{n} a_i \rho_i u_{wi}. \quad (6)$$