Cavitation performance and flow characteristic in a centrifugal pump with inlet guide vanes

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Abstract. The influence of prewhirl regulation by inlet guide vanes (IGVs) on cavitation performance and flow characteristic in a centrifugal pump is investigated. At the impeller inlet, the streamlines are regulated by the IGVs, and the axial velocity distribution is also influenced by the IGVs. Due to the total pressure loss on the IGVs, the cavitation performance of the centrifugal pump degrades. The cavitation area in impeller with IGVs is larger than one without IGVs. The specify values of total pressure loss between the suction pipe inlet and impeller inlet for three cavitation conditions show that the IGVs will generate additional pressure loss, which is related to the IGVs angles and cavitation conditions.

1. Introduction
The prewhirl regulation of IGVs is a widely used approach in industrial centrifugal compressors. It can induce a controlled prewhirl at the impeller inlet so as to regulate the pressure ratio and mass flow at the constant rotational speed. Due to the advantage of convenient installation and practical effect, the IGVs for centrifugal compressors have been sufficiently investigated for decades. The research found in the literature on IGVs for centrifugal compressors mainly fall into four categories: (1) the effect of IGVs on compressors' energy performances (pressure ratio, efficiency) [1-12]; (2) the effect of IGVs design method and structure parameter on the regulation results [2-5, 11]; (3) the unsteady interaction between the stable IGVs and the rotating impeller [6-12]; (4) the common failure modes of IGVs [13]. The focus of investigation on IGVs’ effect on compressor and fan is the pressure and efficiency. For example, Fukutomi [1] experimentally investigated the effect of IGVs on a cross flow fan, and the results showed that the pressure and efficient of the fan with IGVs is higher than that without IGVs. Tan [11] investigated the effects of IGVs on the centrifugal compressor with radial and axial inlets. The results showed that the IGVs system had different performances in the radial and axial inlet models.

Though the IGVs system is a stable component of the compressor, there are still several geometric parameters of IGVs that influence the regulation results. The spherical section passage was used to eliminate the blade hub and tip clearance and the tandem vanes was presented to produce comparable swirl with significantly lower loss than flat plate vanes [2]. The significant features and effects of different settings of the original, s-cambered, and tandem inlet guide vanes on a centrifugal compressor were investigated [4]. The results showed that the tandem inlet guide vanes had superior
aerodynamic performance and extended the operating range of the compressor at both positive and negative inlet swirls. Ferro [5] designed an IGVs system by using a quasi-three-dimensional method, combining the streamline curvature method for the solution of the meridional flow and a panel method for the blade-to-blade flow.

Despite the great improvement of pressure ratio and efficiency by IGVs, the operation instability of compressors resulting from IGVs should be taken into consideration. Oro [9, 10] developed an experimental openloop facility to investigate the dynamic and periodic interaction between both fixed and rotating blade rows in a single stage fan with IGVs. The wake transport through the rotor passage was influenced by the mass flow rate and the geometrical characteristics, such as the blade camber and the stagger angle. Zhou [12] used a constant-temperature hot-wire anemometer to investigate the unsteady interaction of IGVs, impeller, and diffuser in a centrifugal compressor. The unsteadiness caused by IGVs in the diffuser with negative prewhirl angle was larger than that for positive prewhirl angle.

For the centrifugal pump, the circumferential velocity at the impeller inlet directly affects the pump’s efficiency and head according to the Euler equation. The main working principle of prewhirl regulation on centrifugal pump is to change the IGVs prewhirl angle to induce the circumferential velocity, thus regulating the operating condition. According to the investigation by Tan [14-16], the three dimensional IGVs can greatly improve the flow pattern at impeller inlet and widen the pump high-efficiency operation region. However, due to the friction loss on IGVs’ surface the pump cavitation performance will be influenced inevitably.

Much research has been performed toward understanding the cavitation in centrifugal pumps by both means of experiment and numerical simulation. The Particle Image Velocimetry (PIV) was used to investigate the cavitation flow for the case of 3% head drop in a centrifugal pump operating at over load conditions by Bachert et al. [17]. Ding et al. [18] adopted the full cavitation model to simulate the cavitating flow in a centrifugal pump.

2. Problem statement

2.1. Parameters of centrifugal pump

The main parameters of the centrifugal pump are as follows: design flow rate \(Q_d=340 \text{ m}^3/\text{h}\), head \(H=30 \text{ m}\), rotation speed \(n=1450 \text{ r/min}\). Figure 1 shows the tested centrifugal pump and test rig.

![Figure 1. Centrifugal pump and test rig.](image)

2.2. Numerical method

The fluid in the cavitation flow field is considered a homogeneous and compressible mixed medium of
The vapourization rate $\dot{m}_{\text{vap}}$ and condensation rate $\dot{m}_{\text{con}}$ in cavitation model [16] based on the mass transport are as follows:

$$\dot{m}_{\text{vap}} = C_{\text{vap}} \frac{3\alpha_{\text{nu}} (1 - \alpha_{v}) \rho_{v}}{R_{b}} \sqrt{\frac{2}{3} \max (p_{v} - p_{c}, 0) \rho_{l}}$$

$$\dot{m}_{\text{con}} = C_{\text{con}} \frac{3\alpha_{v} \rho_{v}}{R_{b}} \sqrt{\frac{2}{3} \max (p - p_{v}, 0) \rho_{l}}$$

where $C_{\text{vap}}$ and $C_{\text{con}}$ are the empirical calibration coefficients for vapourization and condensation rates, respectively; $R_{b}$ is the bubble radius; $p_{v}$ is the vapour pressure; $\alpha_{\text{nu}}$ is the nucleation site volume fraction, $\alpha_{v}$ is the vapor volume fraction.

The computation fluid dynamics code CFX 13.0 is employed in the present calculation by combining the RNG $k-\varepsilon$ turbulence model and mass transport cavitation model. The boundary conditions are defined as: the total pressure at pump inlet, and the mass flow at pump outlet, no slip wall at walls. Figure 2 shows the computational domain of centrifugal pump and mesh of impeller. The flow rate $Q=315 \text{ m}^3/\text{h}$ is chosen in the present study. The good agreement between the experimental and numerical results validates the reliability of the numerical model and method [16].

3. Result and discussion

Figure 3 shows the streamlines in impeller for the centrifugal pump without IGVs and with IGVs at $12^\circ$, $24^\circ$, $-12^\circ$, $-24^\circ$. When the IGVs are set to different angles, the streamlines in suction pipe and impeller will be regulated accordingly. As shown in Fig.3, the streamlines at the impeller inlet are different for various IGVs angles, but the difference is not as obvious as that in suction pipe.

Figures 4 shows the axial velocity at impeller inlet of the centrifugal pump without IGVs and with at $12^\circ$, $24^\circ$, $-12^\circ$, $-24^\circ$. For both positive and negative prewhirl regulations, the axial velocity contours at impeller inlet are more even than that without IGVs. Correspondingly the high velocity region shrinks as the IGVs angle increases or decreases. Though the IGVs are circumferentially symmetrical in the suction pipe, the axial velocity distribution is still dissymmetrical at impeller inlet. The reason is that the interaction between the impeller and volute induces the uneven velocity in impeller and then influences the flow pattern at impeller inlet. In several regions close to the impeller shroud, the axial velocities are negative due to the reverse flow at partial load condition of $Q=315 \text{ m}^3/\text{h}$. 

Figure 2. Computational domain of centrifugal pump and mesh of impeller.
Figure 3. Streamlines in impeller: (a) no IGVs; (b) IGVs 12°; (c) IGVs 24°; (d) IGVs −12°; and (e) IGVs −24°.

Figure 4. Axial velocity at impeller inlet: (a) no IGVs; (b) IGVs 12°; (c) IGVs 24°; (d) IGVs −12°; and (e) IGVs −24°.

Figures 5 and 6 show the cavitation structure in impeller without IGVs and with IGVs -24° for the cavitation conditions of NPSHA=4m, 3m and 2m, where the NPSHA is the difference between the total energy and vapourization energy for the unit weight of the fluid at pump inlet. With the pressure decrease at pump inlet, the cavitation develops in the impeller gradually. In comparison with the cavitation in impeller without IGVs in Fig.5, the cavitation in impeller with IGVs -24° in Fig.6 is larger, because the pressure loss increases when the IGVs are fixed in the suction pipe. Therefore, the IGVs will degrade the cavitation performance in centrifugal pump, but the influence on the cavitation structure is very tiny as shown in Figs.5 and 6.
Figure 5. Cavitation structure in impeller without IGVs: (a) NPSHA=4 m; (b) NPSHA=3 m; and (c) NPSHA=2 m.

Figure 6. Cavitation structure in impeller with IGVs -24°: (a) NPSHA=4 m; (b) NPSHA=3 m; and (c) NPSHA=2 m.

To further specify the influence of IGVs on cavitation performance of the centrifugal pump, Table 1 lists the total pressure loss between the suction pipe inlet and impeller inlet. For NPSHA=4 m, the total pressure loss is 319 pa for the pump without IGVs, it increase to 2699 pa and 1199 pa when the pump installs the IGVs at 24° and -24° respectively. For NPSHA=3 m and 2 m, the difference of pressure loss for pump without and with IGVs is also very obvious. The positive prewhirl regulation has greater effect on the pump cavitation performance than the negative prewhirl regulation. The cavitation condition will also influence the pressure loss. For example, the total pressure loss for NPSHA=2 m is much larger than that for NPSHA=4 m at IGVs -24°.

| pump          | total pressure loss (pa) for NPSHA=4 m | total pressure loss (pa) for NPSHA=3 m | total pressure loss (pa) for NPSHA=2 m |
|---------------|----------------------------------------|----------------------------------------|----------------------------------------|
| No IGVs       | 319                                    | 315                                    | 326                                    |
| IGVs 24°      | 2699                                   | 2759                                   | 2729                                   |
| IGVs -24°     | 1199                                   | 1242                                   | 1731                                   |

4. Conclusions  
As an effect regulation approach of operation condition for centrifugal pumps, the IGVs can greatly widen the high-efficiency operation region. However, the pressure loss on IGVs should not be neglected for centrifugal pumps. Based on the previous experiment, the numerical research is conducted to investigate the cavitation performance and flow characteristic in a centrifugal pump.
without and with IGVs. The IGVs in suction pipe will generate positive or negative prewhirl on the streamlines at the impeller inlet and then influence the flow pattern in impeller. The distribution of axial velocity becomes more even after the prewhirl regulation of IGVs. Due to the pressure loss on IGVs, the cavitation performance of the centrifugal pump degrades. The cavitation area on blade surface expands when the centrifugal pump installs the IGVs. The pressure loss induced by IGVs is influenced by the IGVs angles and cavitation conditions.

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