Numerical investigation on pressure fluctuations in centrifugal compressor with different inlet guide vanes pre-whirl angles

Y C Wang¹, M Shi¹, S L Cao¹ and Z H Li²

¹Institute of Fluid Machinery & Fluid Engineering, Department of Thermal Engineering, Tsinghua University, Beijing, China
²Shenyang Blower Works Group Corporation, Shenyang, Liaoning, China;
E-mail: caoshl@mail.tsinghua.edu.cn

Abstract. The pressure fluctuations in a centrifugal compressor with different inlet guide vanes (IGV) pre-whirl angles were investigated numerically, as well as the pre-stress model and static structural of blade. The natural frequency was evaluated by pre-stress model analysis. The results show that, the aero-dynamic pressure acting on blade surface is smaller than rotation pre-stress, which wouldn’t result in large deformation of blade. The natural frequencies with rotation pre-stress are slightly higher than without rotation pre-stress. The leading mechanism of pressure fluctuations for normal conditions is the rotor-stator (IGVs) interaction, while is serious flow separations for conditions that are close to surge line. A few frequency components in spectra are close to natural frequency, which possibly result in resonant vibration if amplitude is large enough, which is dangerous for compressor working, and should be avoided.

1. Introduction

The circumferential velocity at the impeller inlet directly affects the compressor performances according to the Euler momentum equation as follows:

\[ h_{th} = (C_{u2}U_{2} - C_{u1}U_{1}) \]  

(1)

On the other hand, a reduction in mass flow rate for backward impellers will results in an increase of the exit swirl, as shown in Fig. 1[1], which may reduce the energy transfer of per unit mass flow to zero, observed from equation (2).

\[ \Delta h_{th} = (U_{2}\Delta C_{u2} - U_{1}\Delta C_{u1}) = 0 \]  

(2)

This property allows the adjustment of mass flow rate with a minimum change in pressure ratio at a constant rotation speed. Therefore, the inlet guide vanes (IGVs) are used to adjust the operating conditions of the centrifugal compressor in most industrial applications, especially the positive pre-whirl (with the same direction of rotation).

For turbo machineries, the operation stability is a hotspot. Due to rotator-stator interactions and flow separations, the unsteady aerodynamics forces give rise to pressure fluctuations on blades. The different pre-whirl angles result in the mass flow rate change, so that the unsteady flow patterns in
compressor passage is varied, namely the dominant frequencies of pressure fluctuations are not a constant. From Campbell diagram shown by Dickmann[2], a number of intersections between the excitation and the natural frequencies of the impeller with the potential for resonant vibration which is dangerous for working of compressor. Hence, it is of great importance to determine the frequency of pressure fluctuation at different IGVs for safe operation, in order to avoid the resonant excitation conditions.

Figure 1. The velocity triangular for no (gray lines) and positive pre-whirl (black lines) at different flow rates: (a) impeller inlet and (b) impeller outlet

Computational fluid dynamics (CFD) techniques, as the experimental research, is indispensable at present for design and investigation of flow patterns in centrifugal compressor. A large number of studies[3] show that the unsteady flows are predicted accurately compared with experimental results, except the critical values like choke mass flow or instability onset. The validation of CFD techniques allows the use of computational results to investigate the pressure fluctuations of unsteady flow in the centrifugal compressor passage. Mangani[3] assessed the influence of various turbulence models for internal flows of centrifugal compressor with adverse pressure gradient, and found slight difference in flow field for turbulence models investigated and these didn’t affect the overall behavior for the machine. Arman Mohseni[1] designs a new tandem IGVs for centrifugal compressor and investigates their interaction with impeller by numerical simulation.

In present study, the following points will be addressed: The compressor performance map is obtained by steady computations. The steady aerodynamic force in impellers result in von-Mises stress and pre-stress under rotation are evaluated, as well as natural frequencies of blade by modal analysis. Then the transient simulations are employed to obtain pressure fluctuations and dominant frequencies on impeller surfaces at different IGVs pre-whirl angles to compare with natural frequencies.

2. Case description and numerical simulation

2.1. Centrifugal compressor stage

The meridional profile of centrifugal compressor stage is shown in fig. 2(a). The design mass flow ratio is 2.5kg/s and pressure ratio is 2.0 for centrifugal compressor stage chosen in present study, with 9 IGVs and 13 blades. The impeller rotates counter-clockwise around Z axis with rotating speed 24634r/min. The diameter of impeller is 240mm, and diameter of inlet pipe is 190mm. The pre-whirl angles of IGVs change from -40° (negative pre-whirl) to +40° (positive pre-whirl), with 10° interval. Due to large number meshes for whole passages, two IGVs passages and three impeller passages are chosen for unsteady calculation, in order to guarantee the approximately same interface area at outlet of IGVs domain and inlet of impeller domain. The computational domain for -10° IGVs pre-whirl is shown in fig. 2(b).

2.2. Numerical simulation details

Both of the grids are block structured for computational domain, and about one million hexahedron elements are generated. To monitor pressure fluctuation in impeller passage, 9 points are selected on pressure surface (PP1-PP9) and suction surface (SS1-SS9), respectively, and locations are shown in fig.3. The numerical calculations are carried out by means of Ansys CFX and k-ε turbulence model.
For the steady-state calculations the following boundary conditions are assumed: At the inlet of the computational domain the total pressure are specified. At the outlet, the static pressure are specified and varied to obtain performance curve. Both in the fixed frame and in the rotating frame the solid walls, i.e., the impeller blades, hub and shroud, the casing walls and the IGVs are modelled using a no-slip boundary condition. For unsteady computations, the same boundary conditions as for the steady state calculations are assumed.

The interfaces are connected by means of a frozen-rotor interface for steady calculations, while by means of rotor/stator interface for unsteady calculations. The frozen-rotor interface means that the relative position of the impeller and outlet/inlet region does not change through the calculations [4], but their relative position change through the calculation according to the angular velocity of the impeller for rotor/stator interface.

The time step of the unsteady calculations has been set to 0.00001*1600 seconds. The chosen time step is small enough to get the necessary time resolution. The number of iterations in each time step has been set to 10. This number of iterations is in most cases sufficient to reduce the maximum residuals by four orders of magnitude. All parameter settings, for example the time step and the number of iterations in each time step, have been retained unchanged for all test conditions. At each pre-whirl condition at first a steady-state calculation is carried out and the result is used to initialize the unsteady calculation at this test point.

![Diagram](image1.png)

**Figure 2.** (a) The centrifugal compressor stage and (b) computation domain for -10° pre-whirl of IGV

![Diagram](image2.png)

**Figure 3.** The monitor points for transient simulation: (a) on pressure surface and (b) on suction surface

### 3. Results and discussions

#### 3.1. Pressure ratio performance

Fig.4 shows the pressure ratio curves performance at different pre-whirl angles of IGVs obtained by steady computations. The conditions labeled by solid symbols are selected for transient calculation, which approximately keep the same pressure ratio. The conditions of +30° to +40° are possibly close...
to the surge line. The pressure ratios curves of 0° and without IGVs are coincide with each other, indicating that the 0° IGVs has little effect on performance of compressor. The performance curves shift upper right for the negative pre-whirl from -10° to -40°, i.e. the mass flow rate increase while the pressure ratio keeps constant. Contrastingly, the performance curves shift down left for positive pre-whirl, and the mass flow ratio decrease while the pressure ratio keeps constant.

**Figure 4.** The pressure ratio performance curves

**Figure 5.** The natural frequency modes of blade

**Figure 6.** The results of equivalent stress of one blade: (a) mesh for analysis; (b) for only rotation; (c) for only aero force; (d) for rotation and aero force. (Units for stress: Mpa)

3.2. Modal and static structural analysis

Fig.5 shows the previous six modes of natural frequencies for one blade in consideration of rotation pre-stress and without rotation pre-stress. The values of with rotation pre-stress are slightly larger than without rotation. The design and operation of compressor should consider these natural frequencies.
and guarantee that the frequencies of pressure fluctuations are far away them. The first mode frequency with rotation pre-stress is 3005 Hz. The yield strength for impeller material is of 833 Mpa, so the allowable stress is about of 400 Mpa.

Fig. 6 plot the static structural analysis results of one blade, in which (a) is mesh for calculation, and the connection surface of blade and hub is set as fixed support; (b) is equivalent stress for only rotation; (c) is the equivalent stress only for aero pressure distribution by computed; (d) is the results of jointly function of aero force and rotation. It’s clear that, compared to stress generated by high speed rotation, the stress generated by aero pressure load is far smaller, which could be ignored. The stress concentrations appear on region of blade outlet edge and hub connection, which conclusion is agreement with theory analysis. The largest stress generated by aero pressure load concentrates on the largest curvature region and inlet edge region, indicating the high pressure gradient between pressure and suction surface of blade due to sharp change of flow.

![Figure 6](image1.png)

**Figure 6.** The spectra of pressure fluctuations of monitor points: (a) on pressure surface at condition C; (b) on suction surface at condition C; (c) on pressure surface at condition B; (d) on suction surface at condition A;

3.3. Spectra of pressure fluctuations
Fig. 7 plots the spectra of pressure fluctuations at IGVs -20° pre-whirl at conditions A, B, C that labeled in fig. 4. From fig. 7 (a) and (b), it can be seen that, on pressure surface and suction surface, the frequency components of all monitor points are the same, and the largest amplitude appear in inlet region of suction surface. The amplitudes of pressure fluctuations on suction surface are larger than on pressure surface, implying the pressure fluctuations on pressure surface are weaker. As the pressure ratio decreasing, i.e. the flow rate increase, the pressure fluctuations become stronger. The primary frequencies for all monitor points at IGVs angle -20° is about 3600 Hz, which is same as the rotor-stator (IGVs) interaction frequency elevated by equation (3).
\[ f_i = n z_i / 60 \]  \hspace{1cm} (3)

Where \( n \) is rotating speed (\( r/min \)), \( z_i \) is blade number of IGVs. Therefore, if there is no flow separation, the dominant mechanism of pressure fluctuations is rotor-stator interaction, which is reasonable to explain the largest amplitude appearing in inlet region. In addition, a frequency component about 2800 Hz is observed in spectra (a) and (c), which is close to the first mode of natural frequency, and possibly result in the resonant vibration.

![Figure 8](image8.png) \hspace{0.5cm} Figure 8. The spectra of SS5 for different pre-whirl angles

![Figure 9](image9.png) \hspace{0.5cm} Figure 9. The streamlines of 50% impeller surface

Fig.8 plots the spectra of pressure fluctuations on monitor point SS5 for different pre-whirl angles and without IGVs. It’s obvious that the pressure fluctuations of negative pre-whirl are much stronger than positive pre-whirl due to mass flow increase. Because of non-linear rotor-stator interaction, several frequency components, such as \(1/3 f_i\), \(2/3 f_i\), and \(2 f_i\), are observed, in which \(2 f_i\) approximately equal to the second natural frequency. The spectra of +30° and +40° conditions are different as other conditions, which show disorderly and complicatedly. The reason is that these two conditions are close to surge line and serious flow separations occur, as shown in figure 9. Fig.9 shows the streamlines of 50% impeller height, from which the serious flow separation is observed on suction surface.

4. Conclusions

(1) The equivalent stress of blade generated by aero pressure load is far smaller than by high speed rotation. But a few frequency components in spectra are close to natural frequency, which possibly result in resonant vibration if amplitude is large enough.

(2) The natural frequencies with rotation pre-stress are slightly higher than without rotation pre-stress.

(3) The dominant mechanism of pressure fluctuations is rotor-stator (IGVs) interaction for normal conditions, while is flow separation for conditions in vicinity of surge line due to serious flow separations.

References

[1] Mohseni A, Goldhahn E, Den Braembussche R A V and Seume J R 2012 J.Turbomachinery 134(2) 021006.1-021006.8

[2] Dickmann H P, Secall Wimmel T, Szwedowicz J, Filsinger D. and Roduner C H 2006 J.Turbomachinery 128(3) 455-465

[3] Mangani L, Casartelli E and Sebastiano Mauri 2012 J. Turbomachinery 134(3) 061033.1-061033.10

[4] Majidi K 2005 Journal of turbo-machinery 127(2) 363-71