PARAMETRIC STUDY OF LOW-SOLIDITY CASCADE DIFFUSER PERFORMANCE IN INDUSTRIAL CENTRIFUGAL COMPRESSORS USING CFD METHODS

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ABSTRACT
Using the state-of-the-art CFD methods, the influence of some important design parameters for the low-solidity cascade diffusers in industrial centrifugal compressors is investigated when 3 cases of solidity, 2 cases of the vane inlet angle and 2 cases of the radius ratio are considered. Numerical approach can be a good choice to get easily the general trend of performance details which are very difficult to measure in the test work. The compressor considered in the present study is the first stage of the industrial gear-driven integral compressor of 600 hp class. Total 12 cases of design variants are studied to suggest a guidance to aerodynamic designers which is useful in practice, because not enough information about the low-solidity cascade diffusers has been found yet.

INTRODUCTION
In the last few years, considerable progress in the low-solidity cascade vaned diffusers has been made, expecting an intermediate or better performance between the vaneless and the channel vaned diffusers. In general, they have been thought to give wider range of stable operation, while maintaining the similar level of efficiency of the conventional vaned diffusers with high solidity. Most designs are based on the extensive database of airfoils tested in linear cascades, accumulated by NACA between about 1940 and 1960 for axial-flow compressors. The final shape of the vanes is obtained by conformal transformation from the rectilinear airfoil form into the radial plane.

Since a conventional high-solidity vaned diffuser is sensitive to the incoming flow incidence, especially at a transonic condition such as in the higher level of pressure ratio, the stall-free stable operation range is usually restricted. Therefore, it is hardly used for general industrial applications requiring larger operating ranges.

Hayami(2000) performed a series of experiments about the low-solidity cascade diffusers in a transonic centrifugal compressor to broaden the operating range under the control of the geometry of impeller and/or diffuser; one was to reduce the inducer blade camber and the other to reduce the inlet passage width of diffuser. The latter failed to broaden the operating range so much, but improved the higher speed efficiency.

Tomaz Kmecl et al.(1999) investigated experimentally the influence of different geometry parameters of 20 low-solidity cascade diffusers on the stage efficiency and operating range. All diffusers were designed for industrial compressor stages with subsonic diffuser flow and had parallel side walls, diffuser vanes with simple circular arc profiles of constant thickness. The throat area of a diffuser, and consequently the number of diffuser vanes, were found the most important parameters affecting the operating range. The vaneless space extent and the diffuser vane inlet angle had a significant effect on diffuser efficiency.

Japikse D.(1996) suggested one design rule for the low-solidity cascade diffusers with NACA-65 series airfoils in modern centrifugal compressor designs which prefers an angle of attack, AOA, of approximately 6 degree or 7 degree, a low lift coefficient, and a solidity from 0.7 to 1.1 depending on the type of application. The present authors have followed the design rule from Japikse D.(1996) in several compressors, but in some cases poor surge margins and/or lower efficiency levels were found. More investigations on the design parameters are required to improve the design rule.

In the present study, 12 combinations of the design parameters of the low-solidity cascade diffusers are built to offer the basis of parametric study for an industrial centrifugal
compressor with moderate level of pressure ratio. Only a numerical approach is considered because a general trend of performance variations is of main concern in the study.

CENTRIFUGAL COMPRESSOR

The compressor considered is the first stage of the industrial gear-driven integral air compressor of 600 hp class. Its design specifications are mass flow rate of 1.6 kg/s, total-to-total pressure ratio of 2.2, and adiabatic total-to-total efficiency of 81% at the rotational speed of 35,000 rpm when the exit volute is included. Mach number at the impeller inlet tip near design point is about 0.77. In the present numerical study, however, the exit volute is not included to build a single stage calculation of the impeller and the vaned diffuser only, as shown in Fig.1. There are two vaneless diffusers; one lies between the impeller exit and the diffuser vane inlet, and the other from the diffuser vane exit to the exit boundary of the calculation as shown in Fig.1. The impeller has 16 full blades with the exit blade angle of 56 deg from tangential reference, and its outer radius is 99.5 mm. The vane profile of the present low-solidity cascade diffuser is NACA65-(4A10)06 which is one of the typical profiles.

As design parameters of the low-solidity cascade diffusers to be altered, 3 cases of AOA, 2 cases of the solidity, and 2 cases of the radius ratio of vane exit to vane inlet are considered, as shown in Table 1. For all the 12 cases, a unique impeller is used, and the same radius ratios of 1.1 and 1.573 are applied consistently for the first and the second vaneless diffuser exits, respectively, where they are measured from the impeller exit location. As defined in Fig.2, AOA is used, instead of the vane inlet angle or the stagger angle, because the design methods follow some terminology from the axial-flow compressor blades. Note that Fig.2 shows an infinite row of the axial cascades; therefore the pitch term should be measured at the vane inlet when the radial plane is considered in the present study. Fig.3 shows the front view of the compressors when AOA is 6 deg, as an example.

NUMERICAL METHOD

The state-of-the-art 3D CFD method employed in the present study is a time marching method, developed by the author(Oh J.S.,1998), which is extended to handle multiple blade row turbomachinery flows. Using the Reynolds-averaged Navier-Stokes equations, it solves the equations of motion in relative cylindrical coordinates in integral conservation form using six-sided control volumes formed by a simple H-mesh. Four stage explicit Runge-Kutta time integration scheme is used with a combined second and fourth derivative artificial dissipation with pressure gradient switching to eliminate spurious wiggles and to control shock capturing. The method utilizes a central, second-order discretization of the flux gradients and a separate time-integration scheme. The eddy viscosity is obtained using the Baldwin-Lomax mixing length model. Tip clearance is handled by gradually decreasing the thickness of the impeller blade to zero in the tip clearance gap. Total temperature and total pressure are imposed on the inlet boundary with the flow direction. Conventional wall functions are applied. Simply constant static pressure is given as fixed at the exit boundary to get a desired mass flow rate.

Table 1  Parametric work scope

<table>
<thead>
<tr>
<th>No.</th>
<th>Name</th>
<th>AOA (deg)</th>
<th>Solidity</th>
<th>Vane Radius Ratio</th>
<th>Vane Inlet Angle (deg)</th>
<th>Vane Exit Angle (deg)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>A1</td>
<td>6</td>
<td>1.0</td>
<td>1.10</td>
<td>16.71</td>
<td>37.03</td>
</tr>
<tr>
<td>2</td>
<td>A2</td>
<td>6</td>
<td>0.6</td>
<td>1.10</td>
<td>16.71</td>
<td>37.03</td>
</tr>
<tr>
<td>3</td>
<td>A3</td>
<td>3</td>
<td>1.0</td>
<td>1.10</td>
<td>16.71</td>
<td>37.03</td>
</tr>
<tr>
<td>4</td>
<td>A4</td>
<td>3</td>
<td>0.6</td>
<td>1.10</td>
<td>16.71</td>
<td>37.03</td>
</tr>
<tr>
<td>5</td>
<td>B1</td>
<td>3</td>
<td>1.0</td>
<td>1.25</td>
<td>13.72</td>
<td>34.02</td>
</tr>
<tr>
<td>6</td>
<td>B2</td>
<td>3</td>
<td>1.0</td>
<td>1.25</td>
<td>13.72</td>
<td>34.02</td>
</tr>
<tr>
<td>7</td>
<td>B3</td>
<td>3</td>
<td>0.6</td>
<td>1.25</td>
<td>13.72</td>
<td>34.02</td>
</tr>
<tr>
<td>8</td>
<td>B4</td>
<td>3</td>
<td>0.6</td>
<td>1.25</td>
<td>13.72</td>
<td>34.02</td>
</tr>
</tbody>
</table>

Fig.1  Meridional view of compressor A,B and E-type

Fig.2  Cascade terminology

(b) compressor A2
The capability of multiple blade row flows is only restricted to steady interaction between the impeller and the diffuser vane. At the interface of two different frames of reference, circumferential averaging occurs as the flow crosses the interface, assuming a complete mixing of the upstream velocity profiles. The variations in the spanwise direction are retained. The steady interaction approach is applied because the main focus of the present study is concentrated on the investigation of the meanline design parameters.

Total grid number used in the present study is 150 x 29 x 29 (streamwise x circumferential x spanwise direction) where 70 and 80 are the maximum streamwise numbers in the impeller and the diffuser domains, respectively. Tip clearance at the impeller exit is set to 4% of the exit axial width, and no clearance is provided in the diffuser domain.
To investigate the behavior of the meanline design parameters using three-dimensional CFD results, a mass-averaging concept is introduced to produce one-dimensional representative values at each concerned cross section. All the variables three-dimensionally calculated on each concerned cross section, such as pressure, density, temperature and each velocity component, are directly mass-averaged to get one-dimensional representative values. Then, the mass-averaged flow angles or the mass-averaged pressure losses, etc. are obtained.

RESULTS AND DISCUSSION

Fig. 4 shows the distributions of the impeller total-to-total pressure ratio and adiabatic efficiency calculated using the mass-averaged pressures and temperatures at the impeller inlet and exit. The overall CFD calculations are executed from deep choke conditions toward minimum flow rates by raising the static pressure values at the second vaneless diffuser exit boundary, as shown in Fig. 1. It is hardly possible to find out flow rates at real surge or stall with any time marching numerical methods, because of their exit boundary condition requirements for subsonic exit flow. The present minimum flow rates are found when the very next solutions begin to diverge, and therefore they should not be real surge or stall flow rates.

Because a unique impeller is used in all the calculations of 12 cases, theoretically unique distributions of the pressure ratio and the adiabatic efficiency have to be generated. But there are some differences in the convergence tolerances imposed by the authors, especially in the lower flow rates.

Fig. 5 Overall total-to-total pressure and adiabatic total-to-total efficiency variations
As an example, total temperature ratio at the impeller exit at a constant flow rate should be the same in all cases, but small changes are inevitable within about 1.5% deviation. In addition, the same number of grids are simply applied in all cases, allowing small variations of grid density; A4, B4 and E4 cases with coarser grids deviate relatively more as shown.

Fig. 5 shows the distributions of overall total-to-total pressure ratio and adiabatic efficiency calculated using the mass-averaged pressures and temperatures at both very ends in Fig.1. At first sight, it is found that AOA has an influence on the operating range or the surge margin at constant speeds by noting the pressure rise slopes. As AOA increases from 3 deg to 6 deg and further to 9 deg (in the order of Compressor B-A-E), the compressor operating ranges are found decreasing by noting the calculated flow fields at the minimum flow rates, as shown in Fig. 6, while the level of pressure ratio and maximum efficiency falls. In Fig. 5, the curves of Compressor B1, B2 and B4 seem to give narrow operating ranges, but their minimum flow rates are never stall points, as found in Fig. 6. No further solutions are converged due to the inherent limitations, as said above, the time marching methods have. In addition, the slopes of their pressure rise are higher, and therefore their real minimum flow rates must be much lower than those in Fig. 5. Maximum efficiency is found in Compressor B4, while minimum efficiency in Compressor E4. Compressor B1 and B3 are expected to meet all the design requirements about pressure ratio, efficiency and surge margin. The effect of the variation of solidity and radius ratio on the compressor efficiency is found higher in Compressor E.

Design AOA determines the inlet vane angle of the diffuser and also the diffuser vane setting angle which are one of the important control parameters in the surge margin control. Changing the diffuser vane setting angle is a well-known tool to shift the compressor characteristics towards lower or higher flow rates. Generally when the angle is reduced, the diffuser characteristics is shifted towards lower flow rates, resulting in a

Fig. 6  Relative/absolute Mach contours in the blade-to-blade section at midspan at the minimum flow rates
change of choke flow rates in case of a high-solidity diffuser. But, in the present study, all the maximum flow rates are determined by the impeller throat area, because no throats are present in the low-solidity cascade diffusers. Larger AOA means the vanes experience larger incidence, when the incidence is defined as an angle, vane inlet angle minus inlet flow angle, as shown in Fig.2, and when all the angles are measured from tangential reference. Fig.7 shows the incidence distributions in the diffuser vanes for all the cases. It is found that the diffuser vane stall begins to be predicted when the vane incidence rises approximately from –5 deg. Design incidence is found around –10 deg when Compressor B1 and B3 are considered.

Fig.8 shows the deviation distributions in the diffuser vanes for all the cases. It is found that the diffuser vane stall begins to be predicted when the vane incidence rises approximately from –5 deg. Design incidence is found around –10 deg when Compressor B1 and B3 are considered.

Fig.9 represents the distributions of the static pressure recovery coefficient, a non-dimensional static pressure rise divided by upstream dynamic pressure, in the diffuser vanes. Compressor A3 and E3 show a poor recovery, while Compressor A2 shows the best recovery. The results show an advantage of a long diffuser vane with respect to maximum static pressure recovery of the diffuser vanes in Compressor A2, A4, B2, B4, E2 and E4. Higher pressure recovery comes from the higher area ratio between the vane throat and the vane exit. However, when the total length of the diffuser is considered including the downstream vaneless diffuser, higher pressure surface. Therefore, the deviation has a positive value in the lower flow rates, and a negative value in the deep choke. But, it has a maximum value between the two end flow rates. Compressors with lower solidity and lower radius ratio give higher level of deviation, as seen in Compressor A3, B3 and E3, because of higher blade-to-blade loadings due to smaller number of vanes.

Fig.6 Continued
Compressor efficiency is not always predicted for the long vane cases, as shown in Fig.5.

Fig.10 shows the distributions of the total pressure loss coefficient, a non-dimensional total pressure loss divided by upstream dynamic pressure, in the diffuser vanes. Compressor B3 and A2 give lower level of the total pressure loss, while Compressor B1, E3 and E4 give higher level. Herrig et al(1951) suggested the variations of the design AOA, at minimum loss, for the change of the cascade solidity for different airfoil profiles through various cascade tests, as shown in Fig.11. According to the figure, for the present airfoil profile, the suggested design AOA is about 5 deg at the solidity of 0.6, and about 7 deg at the solidity of 1.0. Therefore, Compressor A1, A2, A3 and A4, which have AOA of 6 deg, show relatively low level of the total pressure loss in the diffuser vanes. Compressor B1 and B2, which have AOA of 3 deg and the solidity of 1.0, show relatively higher level of the total pressure loss, because the design AOA is about 7 deg in Fig.11 which is far from 3 deg. Compressor E3 and E4, which have AOA of 9 deg and the solidity of 0.6, show relatively higher level of the total pressure loss, because the design AOA is about 5 deg in Fig.11 which is far from 9 deg.

SUMMARY

In the investigation of the effect of some design parameter variations on the performance of the low-solidity cascade diffuser in a centrifugal compressor, the following summary is drawn.
(a) The angle of attack has an influence on the operating range of the compressor. Its design range is found between 3 deg and 6 deg if the operating range is of the first concern.

(b) It is found that the diffuser vane stall begins to be predicted when the vane incidence rises approximately from −5 deg. Design incidence is found around −10 deg.

(c) A long diffuser vane has an advantage with respect to maximum static pressure recovery of the diffuser vanes, due to higher area ratio in the passage.

(d) The level of the total pressure loss in the diffuser vanes with respect to the solidity is found similar to test data supplied in the work by Herrig L.J. et al (1951).

REFERENCES


Fig. 11 Design angles of attack for NACA65-series