Performance improvement of a centrifugal compressor stage by using different vaned diffusers

Y C Zhang*, X Z Kong, F Li, W Sun, and Q G Chen

College of Mechanical and Electronic Engineering
Shandong University of Science and Technology, Qingdao, 266590, P. R. China

E-mail: zhangyongchaobgx@126.com

Abstract. The vaned diffuser (VLD) is usually adopted in the traditional design of the multi-stage centrifugal compressor because of the stage’s match problem. The drawback of the stage with vaneless diffusers is low efficiency. In order to increase the efficiency and at the same time, induce no significant decline in the operating range of the stage, three different types of vaned diffusers are designed and numerically investigated: the traditional vaned diffuser (TVD), the low-solidity cascade diffuser (LSD) and the partial-height vane diffuser (PVD). These three types of vaned diffusers have different influences on the performance of the centrifugal compressor. In the present investigation, the first part investigates the performance of a centrifugal compressor stage with three different vaned diffusers. The second part studies the influences of the height and the position of partial height vanes on the stage performance, and discusses the matching problem between the PVD and the downstream return channel. The stage investigated in this paper includes the impeller, the diffuser, the bend and the return channel. In the process of numerical investigation, the flow is assumed to be steady, and this process includes calculation and simulation. The calculation of 3-D turbulent flow in the stage uses the commercial CFD code NUMECA together with the Spalart-Allmaras turbulence model. The simulation of the computational region includes the impeller passages, the diffuser passages and return channel passages. The structure and surrounding region are assumed to have a perfect cyclic symmetry, so the single channel model and periodic boundary condition are applied at the middle of the passage, that is to reduce the calculation region to only one region. The investigation showed that the low-solidity cascade diffuser would be a better choice as a middle course for the first stage of the multistage centrifugal compressor. Besides, the influences of the height and the position of partial height vanes on the stage performance are intensively investigated and concluded at the design point, the isentropic efficiency and the static pressure ratio of the stage are improved with the increasing of the partial vane’s height, and that installing the half-height vanes on the shroud side the stage would obtain a more uniform diffuser outflow and a better aerodynamic performance.

1. Introduction
In the traditional design, multistage centrifugal compressors are used to be equipped with vaneless diffuser at the first stage for its wide operating range. However it is well-known that the efficiency of the stage with the vaneless diffuser is lower than that of the stage with the vaned diffuser. So as to increase the efficiency of a centrifugal compressor stage and at the same time to keep the operating range not obviously reduced, the low-solidity cascade diffuser and the half guide vane diffuser (or named as the partial-height vane diffuser) are usually applied.

The LSD is first proposed by Senoo at 1978 in the Japanese patent application disclosure[1]. Later, Kaneki and Ohashi’s experimental investigations revealed that the efficiency of the stage with the LSD increased 4% and 2% at the design flow rate and the surge flow rate respectively, comparing with
that of the VLD stage\textsuperscript{[2]}. Senoo et al. demonstrated that the LSD had an equivalent operational range with the VLD but provided considerably better pressure recovery in the study of a low-speed centrifugal blower\textsuperscript{[3]}. Hayami et al. applied a LSD of 0.69 solidity to a transonic centrifugal compressor got the similar conclusions and indicated that the LSD demonstrated a good pressure recovery over a wide range of flow angles even the inflow Mach number to the cascade is over unity\textsuperscript{[4]}.

The PVD is first adopted for a high specific speed centrifugal compressor by Yoshinaga et al. at the year of 1987\textsuperscript{[5]}. They installed the half guide vanes on the shroud side wall of the diffuser and found that the pressure recovery of this diffuser is considerably improved by the half guide vanes and that the optimum height of the vanes is a little less than one half of the diffuser width. While, Sikari fixed the partial guide vanes to the diffuser hub of a low specific speed centrifugal compressor and obtained a good result when the vane height is 0.4 times the impeller exit width\textsuperscript{[6]}. Issac et al., however argued that the optimum PVD vane height is 0.3 times the diffuser width, and the effect of the partial height vanes’ position on the hub or shroud is negligible, but when the partial vanes are staggered placed on both the hub and the shroud the performance of the compressor is substantially improved\textsuperscript{[7]}.

It is easily recognized that when the PVD is adopted in the multistage centrifugal compressor, the incidence angle to the return channel blade will be changed with the changing of the PVD vane height, so the matching problem between the PVD and the downstream return channel should also be investigated when evaluating the performance of PVD.

In the present investigation, the first part mainly investigates the performance of a centrifugal compressor stage with three different vaned diffusers. And the second part studies the influences of the height and the position of partial height vanes on the stage performance, and discusses the matching problem between the PVD and the downstream return channel.

2. Centrifugal compressor stage
The centrifugal compressor stage in study is the first stage of an industrial thirteen-stage centrifugal compressor, of which the working substance is CO\textsubscript{2}. And the original stage is equipped with the vaneless diffuser. In the present paper three type vaned diffusers are designed, so the stage investigated here consisted of the impeller, the vaned diffuser, the bend duct and the return channel, as shown in figure 1. When \( h/b = 1 \), it meant that the diffuser is the TVD or LSD, while \( 0 < h/b < 1 \) denoted that the diffuser of PVD.

The blade number of the TVD is 20, which is counted adopting the rule of optimal equivalent divergence angle. The cascade solidity of the TVD is 1.428, and the definition of the corresponding parameters is shown in figure 2. For the convenience of subsequent experiments, the LSD is obtained by getting rid of every other blade of the TVD. So its blade number is 10, and its solidity is 0.714. And the PVD is produced by resetting a partial height of the vane from the TVD. Therefore, all the three type vaned diffusers have the same blade profile and setting angles. The blade profile of the vaned diffuser is from Xi et al\textsuperscript{[8]}, which is quite suitable for the high subsonic flow. The inlet blade angle of the diffuser is chosen as 22 deg to match the outflow from the impeller at the design point. In addition, the design flow rate the absolute Mach number of flow at the diffuser inlet is about 0.8, according to the recommend value, the diameter ratio of the vane diffuser at the inlet section \( D_3/D_2 \) is 1.147 and that at the exit section \( D_4/D_2 \) is 1.4.

![Figure 1. Centrifugal compressor stage](image1)

![Figure 2. Characteristic parameters of the vaned diffuser](image2)

All the three type vaned diffusers share the same redesigned return channel. The original return channel had the tandem cascade, but the redesigned one possessed only a single row of blades, which is designed to match the stage with the TWD best. The return channel blades are produced adopting the
“banana” vane profile and a single-circular-arc camber line, and to match the outflow from the bent duct their inlet setting angle is fixed at 34 deg. The main technical data of the stage is shown in Table 1.

| Table 1. Main technical data of the stage |
|------------------------------------------|
| Evolving fluid                          | CO$_2$ |
| Design rotation speed $n$ [rpm]         | 6900   |
| Design mass flow rate $Q_m$ [kg/s]      | 14.64  |
| Design pressure ratio with VLD          | 1.685  |
| Inlet total pressure $p_{r,i}$ [Pa]     | 136000 |
| Inlet total temperature $T_{r,i}$ [K]   | 318.15 |

Impeller
- Blade number: 9+9
- Peripheral Mach number $M_s$: 0.977

Diffuser
- Inlet setting angle $\alpha_{3,i}$ [deg.]: 22
- Outlet setting angle $\alpha_{4,i}$ [deg.]: 36
- Vane number: TVD 20

Return channel
- Inlet setting angle $\alpha_{5,i}$ [deg.]: 34
- Outlet setting angle $\alpha_{6,i}$ [deg.]: 96
- Vane number: 17

3. Numerical method
In this paper, the flow is assumed to be steady. The 3-D turbulent flow in the stage is calculated by using the commercial CFD code NUMECA together with the Spalart-Allmaras turbulence model. In the present simulation, one of the impeller passages, one of the diffuser passages and one of the return channel passages are selected as the computational domain and the periodic boundary condition is applied at the middle of the passage.

The calculation is performed with a second-order central difference scheme, second and fourth order artificial dissipation terms, and the W-cycle multigrid technique. Besides, a four-order Runge-Kutta scheme together with local time stepping and implicit residual smoothing techniques is adopted for convergence acceleration. It is noteworthy that to obtain a more accurate simulation of the flow in the compressor stage, the perfect gas model with variable gas properties is introduced into the CFD code$^{[9]}$.

For the investigated stage, the variation of the pressure, with the change of the measuring point’s position in the passage and with that of the stage’s operating condition, lied in the range [0.1MPa, 0.2MPa]. And within this pressure range, the effects of the pressure to the CO$_2$’s thermodynamic parameters such as $c_p$, $\gamma$, $\kappa$ and $\mu$ are negligible. However, the influences from the temperature variation to the four thermodynamic properties above are distinct. According to the operating conditions of the present stage the data list of the CO$_2$’s four thermodynamic properties over the temperature range [200K, 1000K] is added into the CFD code, respectively.

4. Results and discussions
4.1. Comparison of the three vaned diffusers
Figure 3 shows the predicted performance of the stage with different diffusers. In this figure, the PVD performance curves are obtained through the stage with the 0.5b-height vane diffuser, and of which all the half-height vanes are fixed on the hub side of the diffuser wall. In Figure 3 it can be seen that, using the TVD the stage has the highest isentropic efficiency at the design flow rate ($Q_m, \text{des}$), while utilizing the VLD the stage has the widest stable operating range.

Comparing with the original stage with the VLD, the stage with the TVD gains an isentropic efficiency benefit of 6.8% and a static pressure ratio improvement of 4.7% at the design point. But a distinct shrink of the stable operating range occurs; the mass flux decreased by 6% of $Q_m, \text{des}$ at the choke point, and the small flow point near the stall is increased by 20% of $Q_m, \text{des}$.
As a compromised choice, the stage with the LSD shows good qualities. Comparing with the original stage, it gains a 5.4% isentropic efficiency improvement at the design point, at the same time it kept the maximum flow rate with only 9% of $Q_{m_{des}}$ increase at the stall point. So its operational range is greatly widened, comparing with that of the stage with the TVD. Besides, using LSD the stage can achieve a quite higher static pressure ratio in a quite wider stable operating range.

In addition, the efficiency peak point of the stage with the PVD is shifted to large flow rate side. This shortcoming may be overcome by improving the matching between the PVD and the return channel, which will be discussed in the next section.

4.2. The effects of the vane height for the stage with PVD

Figure 4 shows the predicted performance of the stage with different PVDs, of which the vane height respectively is 0.2$b$, 0.4$b$, 0.6$b$ and 0.8$b$, and all the partial vanes are installed on the hub side of the diffuser wall. It can be seen from figure 4 that, at the design point, the isentropic efficiency and the static pressure ratio increased with the height increasing of the PVD vanes, though the increment is tiny.

Figure 5 and figure 6 show the distribution of the pitch-averaged absolute flow angle along the spanwise at the exit of the vaned diffuser and the inlet of the return channel, respectively. In figure 5 it can be seen that, at the exit of the diffuser all the pitch-averaged absolute flow angles dropped behind in the vicinity of the shroud. And there is a noticeable turning point respectively for the curve of $h/b=0.4$, $h/b=0.6$ and $h/b=0.8$, from where the flow angles diminished rapidly, even turning into negative values at last. And the turning point for each of the three curves just located at the corresponding position of the partial vane’s tips, which means that in the passage near the shroud (where the partial height vanes do not exist), the absolute flow angles are much smaller than that of the flow in the region with the partial vanes.
Figure 6 shows the flow angle distribution of the outflow from the bend duct. Due to the influences of the bend and the upstream flow, the flow angle is uneven entering the return channel blades. And in figure 6 it can be seen that the higher the partial vane is, the closer the averaged flow angles’ mean value of the PVD is to that of the TVD. So this is the reason for that the isentropic efficiency and the static pressure ratio increased with the height increasing of the PVD vanes at the design point.

4.3. The effects of the vane position for the stages with PVD

Figure 7 shows the predicted performance of the stage with two different PVDs. Both of the two PVDs’ vane height is 0.5b, but the position of the partial vanes are different, one row of the partial vanes is installed on the hub side named as h_0.5 in figure 7, and the other row is fixed on the shroud side named as s_0.5 in the figure.

In figure 7 it can be seen that when the half height vanes are installed on the shroud side the stage’s predicted performance is better than that of the stage with half height vanes fixed on the hub side.

Fig.8 shows the distribution of the pitch-averaged absolute flow angle along the spanwise at the exit of the vaned diffuser. It could be clearly seen that the flow angles near the shroud increased greatly due to the effect of the shroud side installed half vanes, and that at the exit of the vaned diffuser, the pitch-averaged flow angle’s distribution of the shroud half-height vane diffuser, is more uniform than that of the hub half-height vane diffuser along the spanwise.

The numerical analysis showed that, for the stage with half-height vane diffuser, installing the half vanes on the shroud side would obtain a more uniform outflow at the diffuser exit and a better aerodynamic performance than doing that on the hub side.

5. Conclusions

In this paper, the performance of the first stage of a centrifugal compressor with different diffusers is numerically investigated. The conclusions can be drawn as the following:

(1) Using the traditional vaned diffuser (TVD) the stage has the highest isentropic efficiency at the design flow rate, while utilizing the vaneless diffuser (VLD) the stage has the widest stable operating range. The stage with the low-solidity cascade diffuser (LSD) has not only a quite higher efficiency, but also a rather wider stable operating range.

(2) The stage with the PVDs nearly has the same stable operating range with the LSD stage, but it’s efficiency does not show a obvious superiority to the stage with vaneless diffuser at the design point. The problem of lower efficiency may be solved through better match between the PVDs and the return channel. At the design point, the isentropic efficiency of the stage with partial height vane diffusers is increased with the height increasing of the partial vanes.

(3) For the stage with half-height vane diffuser, installing the half-height vanes on the shroud side, the outflow from the diffuser is more uniform and the performance of the stage is also better.

Nomenclatures

\[ D = \text{diameter} \]
\[ n = \text{rotational speed} \]
\( R = \) gas constant
\( T = \) static temperature
\( M = \) Mach number
\( Q_m = \) mass flow rate
\( b = \) width of the diffuser
\( c = \) chord length
\( c_p = \) specific heat at constant pressure
\( p = \) pressure
\( s = \) cascade solidity, \( c/t \)
\( t = \) pitch at the diameter of \( D_3 \)
\( h = \) height of the vane
\( \alpha_{\text{in}} = \) setting angle, from circumferential direction, \( i=2, 3, 4, 5, 6 \)
\( \gamma = \) specific heat ratio
\( \kappa = \) heat conductivity
\( \mu = \) dynamic viscosity

Subscripts

des = design point value
in = inlet
t = total value
u = peripheral velocity
2 = impeller exit
3, 4 = diffuser inlet, diffuser exit, respectively
5, 6 = return channel inlet, return channel outlet, respectively

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