Inlet Swirl on Turbocharger Compressor Performance

Lei Huang, Ying Liu, Hua Chen*

National laboratory of Engine Turbocharging Technology, Tianjin, China
**corresponding author: Tel.:+86-22-5870-7069; fax: +86-22-5870-7500*
E-mail address:hchen@nlett.com

Abstract

The influence of preswirl flow generated by inlet guide vanes on the performance of a small centrifugal compressor was investigated numerically and experimentally. A number of inlet guide vanes were designed to provide constant preswirl angles and variable preswirl angle in the spanwise direction at the compressor inlet. The performance characteristics of the compressor with these different positive/negative preswirl angles were calculated by numerical simulation. The flow fields in the impeller were analyzed to reveal the mechanism of the effects of inlet swirl on the compressor performance.

Three types of inlet guide vanes were also tested. One provided positive preswirl, and another negative preswirl, while the third had twisted guide vanes with positive and negative exit blade angle at tip and root respectively. Experimental results from these three vanes and from no preswirl baseline were analysed and compared with the numerical results.

1. Introduction

The development of modern internal combustion engines requires the turbocharger centrifugal compressor to provide a broader map width and higher pressure ratio while simultaneously increasing the efficiency almost everywhere of compressor map[1]. With rising pressure ratio, the stable flow range of centrifugal compressor trends to become narrower. The surge line is located at the higher flow rates and efficiency level trends to reduce, which restricts the boost pressure rise and engine power output at high speeds. It is difficult to design a compressor to provide a broad operating range, when high pressure ratio and high efficiency are also required as part of the design specification.

The output work of a compressor is given by the Euler work equation:

$$W = m\,(U_2 C_{\theta 2} - U_1 C_{\theta 1}), \qquad (1)$$

Where m is the mass flow rate, U_1 and U_2 is the blade speed at impeller inlet and outlet respectively, $C_{\theta 1}$ and $C_{\theta 2}$ is the tangential absolute velocity at the inlet and outlet respectively. All the performance characteristics of centrifugal compressors are affected by inlet swirl $C_{\theta 1}$, which is associated with the incidence angle and the inlet Mach number, as illustrated in **Figure 1**. For example, an increase of $C_{\theta 1}$ will result in a reduction of work input to the air while simultaneously a decrease of positive incidence reducing the possibility of flow separations on the suction side of the blade. Therefore the pressure characteristic of the compressor is finally decided by the combining effects of the reductions in the incidence loss and in work input to the air. Meanwhile the surge limit and choke limit also vary too.

The inlet guide vanes (IGV) have been used in various industries to induce preswirl onto the in–coming air to adjust compressor performance. **Figure 1** shows a sketch of an IGV and inducer inlet velocity triangle. The most significant effect of inlet swirl is the variation of flow angle and velocity it brings about at impeller inlet. **Figure 1** shows that with positive preswirl (preswirl in the rotational direction of the impeller), the incidence angle and the inlet relative Mach number are both reduced. Hence it is expected that IGV may help to extend the flow range and improve the performance of turbocharger compressors operating at high incidence condition.

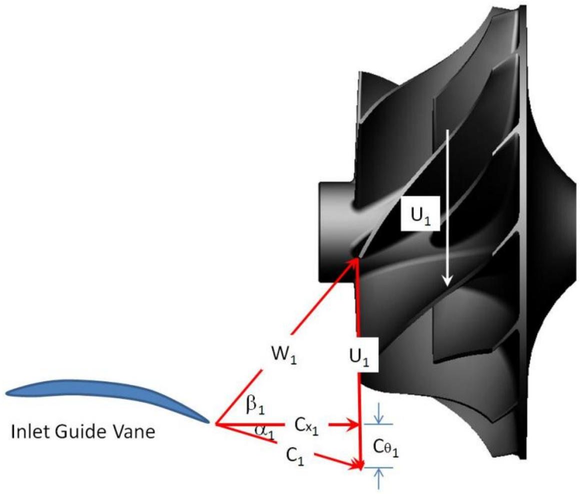


Figure 1 Inlet guide vane and inducer inlet velocity triangle

A shroud bleed slot or ported shroud has long been used to enhance the map width of turbocharger centrifugal compressors. The recirculated flow between the bleed slot and compressor inlet contains an amount of whirl which is mixed with the inflow stream and helps to optimize the incidence angle at the compressor inlet. Ref. [2] investigated the influence of operating conditions near surge and choke on the bleed slot flow field by CFD. CFD analysis revealed that the reduction of incidence due to the mixing of the recirculation flow and the main flow could be a factor for stabilising the flow within the passage and so improving the available flow range for compressor operation. The results suggested that the compressor map width could be improved by the incidence angle redistribution at the inlet to the impeller.

In [3], engine tests have been performed with different inlet swirl devices. These devices include 4 vanes with different preswirl angles of ±45°、±30°、±15°、0°, and 6 vanes with ±30°、0° preswirls. The experiment result showed that the pressure ratio was improved on the run-up line when applying the 45° positive preswirl angle while the negative swirl angle tended to improve the pressure ratio at high engine speeds, but the reduction of the efficiency offset this advantage.

Galindo et al.[4] developed a swirl generator device (SGD) specifically to improve the surge margin and compressor efficiency on a compressor with a 90º bend inlet duct. The SGD could modify the blades position to adapt to the engine operation conditions. This study concluded that the SGD improves the surge limits when the pre-whirl has the opposite direction to the compressor rotation, which is in contrast to other investigation with axial inlet flow and guide vanes. The reason proposed for this is that with the SGD the relative velocity was higher than with IGV due to the reduction of the effective section. And the incidence angle is reduced even with the negative swirl when the angle of attack decreased. The combination of them made the negative swirl having the optimal surge limit improvement.

The gain in surge margin resulting from positive preswirl is always accompanied by a drop in pressure ratio, shown by Equation (1). The theoretical computations by Yousef et al.[5] indicated that a reduction of preswirl angle from tip to hub linearly increased the work and pressure ratio up to more than 94% of the non-preswirl values and further in a parabolic relationship could restore them to the level of no preswirl when using a preswirl angle at tip up to 40°. Little in open literature however describes the effects of inlet swirl on performance of turbocharger sized compressors and the mechanism of these effects. In these small sized centrifugal compressors, the effects of viscosity or Reynolds number are important and they limit the preswirl vane count. In this paper, CFD simulations were carried out to investigate the effect of different preswirl angles on the performance of a turbocharger compressor and the underlying mechanism. The flow fields in the compressor were analyzed under near surge and choke operating conditions. The performance of the compressor with three types of inlet guide vanes were obtained experimentally and compared with that of no preswirl baseline.

2. Compressor and numerical methods

A turbocharger compressor was studied. **Table** 1 shows the major geometry parameters of the compressor. The CFD simulations were carried out using the commercial code NUMECA. Single passages of the inlet guide vane, the impeller and the diffuser were modeled when applying inlet swirl, and no inlet guide vane passage when no preswirl was applied. The volute was not included for the computational efficiency. Periodic boundaries were applied and multi-blocked structured grids were employed. HO-topology and HC-topology were used in the guide vane and the impeller passage respectively. H-topology was used in the inlet, outlet and main flow passage domains while O-topology and C-topology were applied in the skin blocks around the guide vane and impeller blade respectively to achieve the required mesh quality. The clearance region was meshed by 17 grid points in spanwise direction. The total numbers of grid nodes were 1180645 including the inlet guide vane, and 884130 when the guide vane was left out. The grid was refined at the near-wall, leading and tailing edge regions. The minimum skew angle of grid cells was greater than 20°. The y+ values of the wall-adjacent cells were less than 2, as required for use of the one-equation S-A turbulence model. The computational grids are shown in **Figure 2**.

Table 1 Geometry parameters of the compressor

Parameter	Value
Tip diameter	122mm
Tip width	8mm
Inlet shroud diameter	90.4mm
Inlet hub diameter	27.6mm
Diffuser exit diameter	200mm
Diffuser gap	7.5mm
Blade number	8+8
Shroud clearance	0.5mm

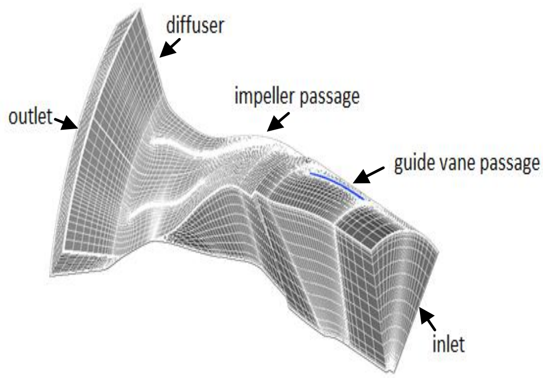


Figure 2 Computational grids of the compressor

Total atmospheric pressure and temperature values were applied at inlet boundary conditions, so was the axial inflow. The averaged static pressure over the whole outlet area was used as the outlet boundary condition near choke, and self-adaptive mass flow rate at other operating conditions. The mixing plane approach was employed at the rotor/stator interface between the guide vane passage and impeller passage, which guaranteed the conservation of mass flow, momentum and energy through the interface.

3. Inlet guide vane design

A number of inlet guide vanes were designed to provide different preswirls at the inlet of the compressor, including several types of spanwise straight vanes and one type of twisted vanes. All vanes had the same meridional shape. **Figure 3** shows the CAD model of inlet guide vanes. The length of the vane was 30mm at tip, and 10mm at hub. Vane number, vane angle distribution, distance to impeller etc were optimized by CFD. The inlet blade angles of the vanes stay zero to align with the approaching flow to avoid flow separation inside the vanes. With the variation of the outlet vane angle distribution, different straight vanes could provide relatively uniform flow but different constant preswirl angles along the span; while the preswirl angle varied from the tip to hub with the twisted vanes.

Table 2 shows the outlet blade angles of vanes at tip and hub and the preswirl angle generated at the mass flow of 0.6kg/s. **Figure 4** shows the contour of inlet swirl angle at the exit of the inlet guide vane case 3. The swirl angle is not entirely uniform due to the limited number of vanes, and the effects of trailing edge wake and casing endwall boundary layer.

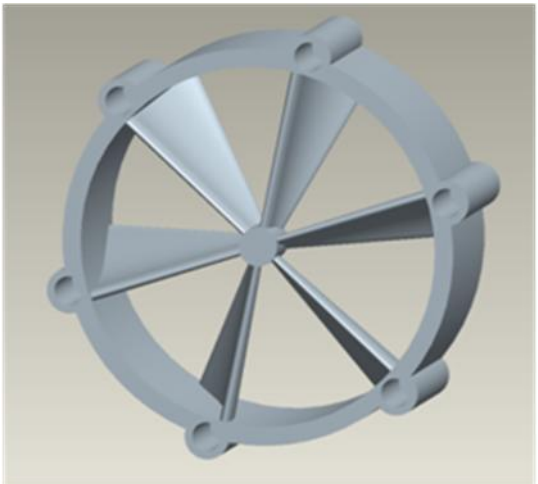


Figure 3 CAD model of inlet guide vanes

Table 2 Inlet guide vanes and effect of vane angle on preswirl

Guide vane types		Vane angle (tip/hub)	Preswirl angle
Straight vanes	Case 1	±(40°/30°)	±17°
	Case 2	±(50°/40°)	±25°
	Case 3	±(60°/50°)	±32°
Twisted vanes	Case4	+30°/-50°	+12°at tip -12°at hub

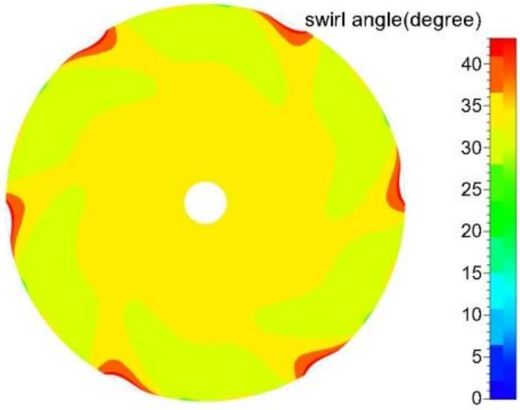


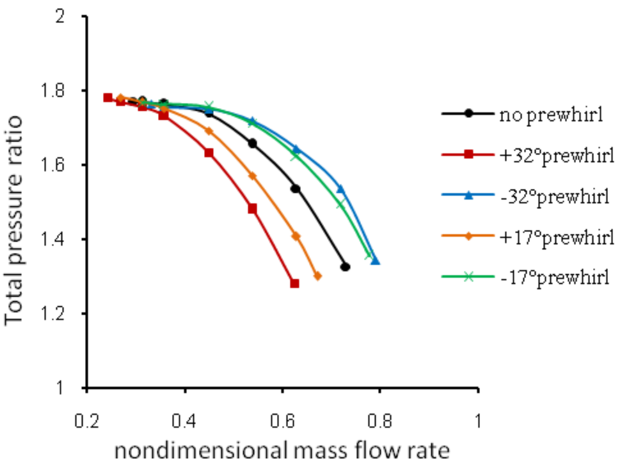
Figure 4 Swirl angle at vane exit, vane case 3, +32° mean swirl

4. CFD results of straight vanes

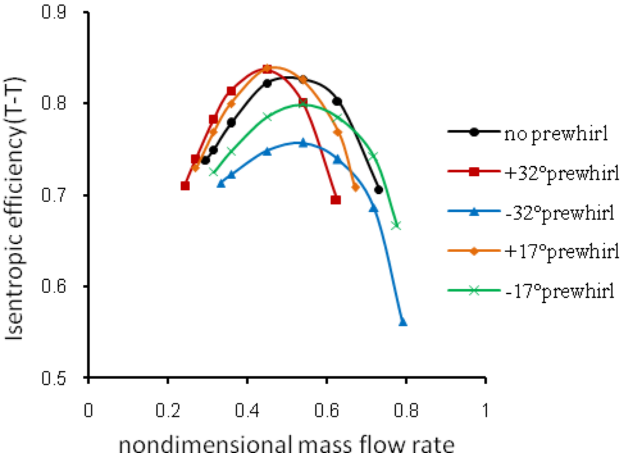
4.1. Overall performance

The straight inlet guide vane case 1 and case 3 were chosen for the numerical analysis with the compressor stage. The pressure ratio and efficiency of the compressor with and without inlet swirl against mass flow rate at two rotation speeds are shown in **Figure 5**. The mass flow rate values are non-dimensionalised. Implementing the inlet swirl device causes a considerable change in the compressor characteristics. The two positive preswirls shift the characteristic curves to the lower values of mass flow rate and improve the peak efficiency of the compressor slightly. Increasing positive preswirl angle results in lower 'surge' or numerically unstable flow (here the last numerically stable point from CFD was regarded as the 'surge' point.), and choke flow, and more reduction in pressure ratio at high mass flows. At the rotational speed of 50000rpm, +32° preswirl leads to a lower 'surge' flow from 0.30 with no inlet preswirl to 0.24, while choke flow is reduced from 0.73 to 0.63. These implies that the positive preswirl is useful for the extension of surge limit but has negative effect on choke limit.

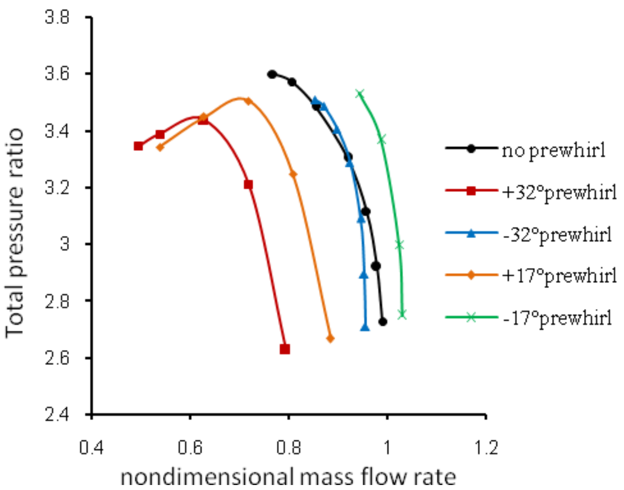
The CFD results from the negative preswirls show the opposite effects on the compressor performance compared to the positive preswirls. The negative preswirls cause a significant reduction in compressor efficiency and the reduction increases with the negative preswirl angle. At the rotational speed of 50000rpm, the 'surge' flow is increased to 0.33 and the choke flow increases by 8.2% when applying -32° preswirl. The pressure ratio is almost the same between the -32° preswirl and -17° preswirl. At the higher rotational speed of 80000rpm, the choke flow of -32° preswirl is even lower than that of no preswirl, while the -17° preswirl still shows an improvement of 3.3%, indicating that there is a limitation of raising the negative preswirl to improve the compressor choke performance. Note also the 'surge' flow of -32° preswirl is lower than that of -17° one.



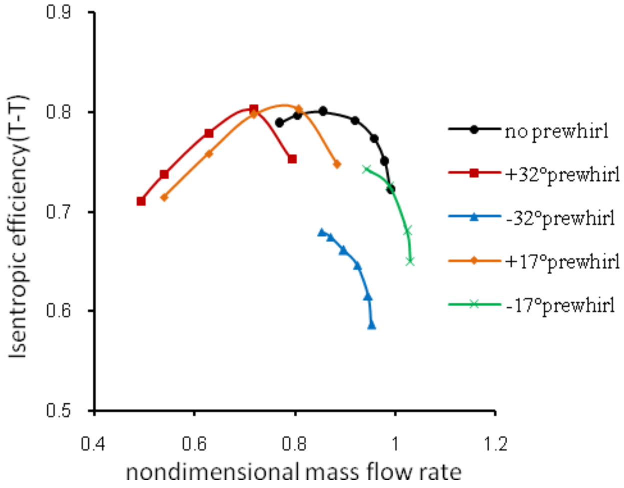
(a) Pressure characteristic at 50,000rpm



(b) Efficiency characteristic at 50,000rpm



(c) Pressure characteristic at 80,000rpm



(d) Efficiency characteristic at 80,000rpm

Figure 5 Performance characteristics of the compressor with and without inlet swirl

4.2. Flow field analysis at surge end

The flow field analysis was carried out to figure out the mechanism of the effects of inlet swirl on the compressor performance. Investigation was performed at the non dimensional mass flow rate of 0.36 at 50,000rpm. **Figures 6** and **7** show the pitchwise averaged impeller inlet relative flow angle and relative Mach number at the spanwise direction respectively. The positive preswirl increases the relative flow angle at the hub significantly. Due to the higher blade tangential velocity and the smaller solidity of the inlet guide vanes at the shroud (See **Figures 3** and **4**), the inlet relative flow angle changes little near the shroud with inlet swirl except for the negative preswirl which displays a sharp peak. The peak corresponds to a flow reversal at impeller inlet as shown in **Figure 8**. **Figure 7** shows that the negative preswirl has higher mean relative Mach number up to 65~70% span of the inlet compared to the positive preswirl and no preswirl, which can be explained through the analysis of the velocity triangles at the compressor inlet. The relative Mach number of the negative preswirl peaks at 55% span and then drops until at about 90% span where it starts to increase due to the relative motion of the impeller and casing. Similar trends of the relative Mach number distribution are found with no preswirl and positive preswirl. However, the location of the peak relative Mach number moves to higher span and the distribution becomes more uniform with the increase of the preswirl angle from negative to positive. Hence the positive preswirl has higher relative Mach number at shroud, which cannot be revealed by a simple analysis of the inlet velocity triangles.

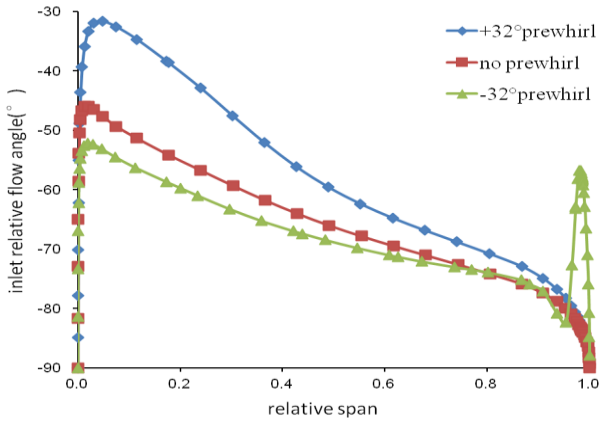


Figure 6 Impeller inlet relative flow angle distributions at 50,000rpm and mass flow rate 0.36

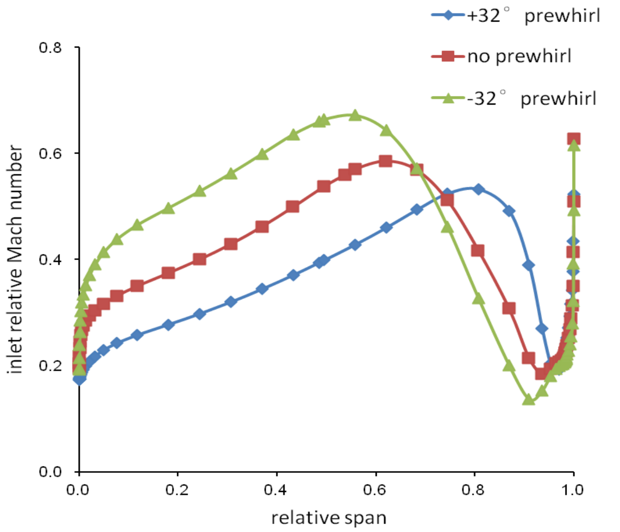
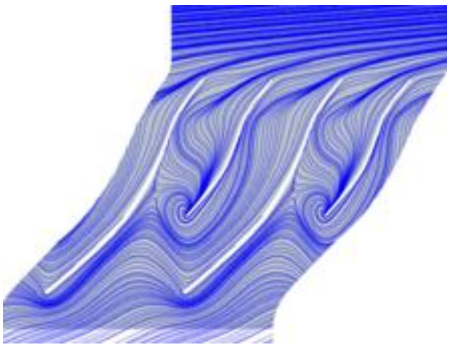
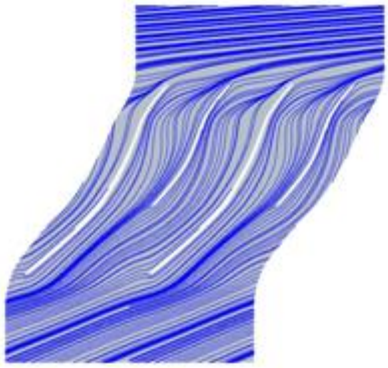


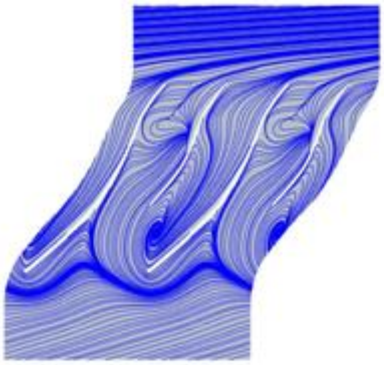
Figure 7 Impeller inlet relative Mach number distributions at 50,000rpm and mass flow rate 0.36



(a) no preswirl



(b) +32° preswirl



(c) -32° preswirl

Figure 8 Streamlines at 90% span, 50,000rpm and mass flow rate of 0.36. Showing some inlet flow reversal with no preswirl, no inlet flow reversal with positive preswirl and severe inlet flow reversal with negative preswirl

Figure 9 shows the full blade pressure loading at 90% span and non-dimensional mass flow rate of 0.36 with three different inlet swirls. The positive preswirl has the highest blade loading at the leading edge while that of negative preswirl is the lowest. This is consistent with the results in **Figure 7** and also provides a reason for the better surge end performance of the positive preswirl. **Figure 10** shows pitchwise averaged impeller exit absolute flow angle distribution. At the hub, the positive preswirl produces the highest exit flow angle, meaning that the corresponding exit relative flow velocity is the lowest. This is at least because of the lowest inlet relative flow velocity produced by the positive preswirl (**Figure 7**). It seems that the positive preswirl provides a more uniform distribution of the flow angle at impeller exit, which will improve the flow condition in the downstream diffuser.

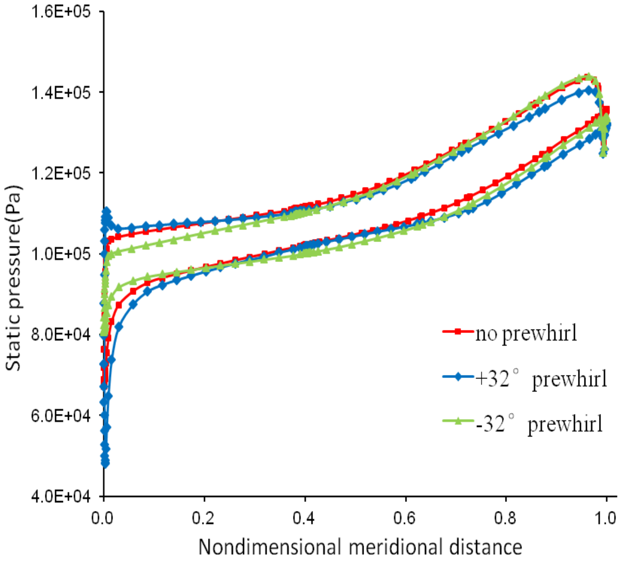


Figure 9 Blade loading at 90% span, 50,000rpm and mass flow rate of 0.36

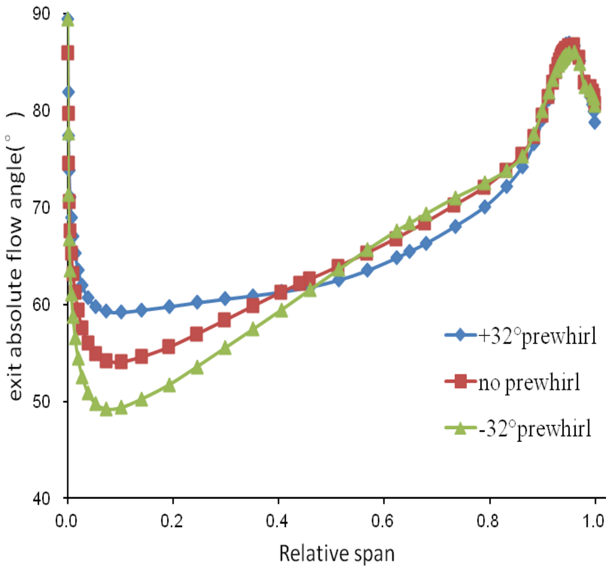
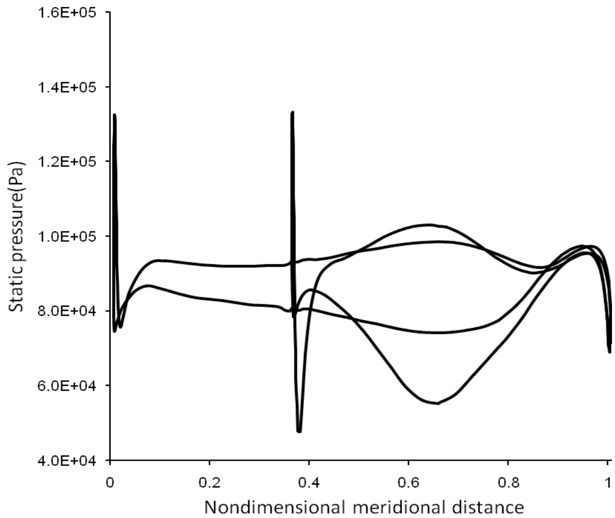


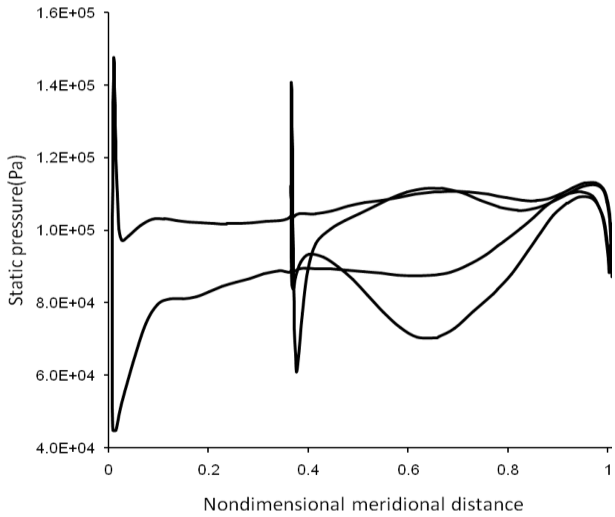
Figure 10 Impeller exit absolute flow angle distributions at 50,000rpm and mass flow rate of 0.36

4.3. Flow field analysis at choke end

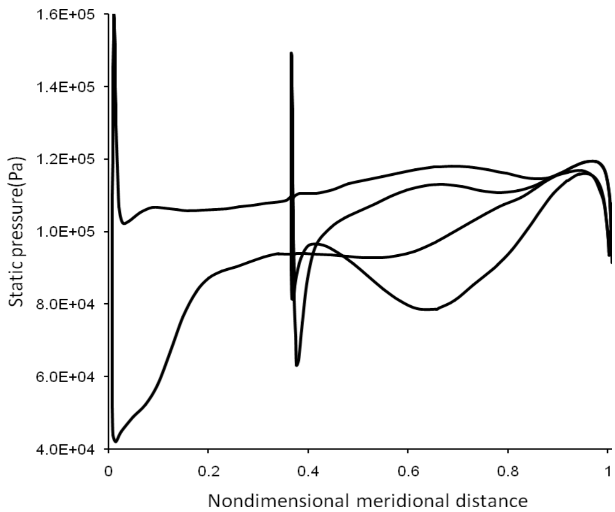
At a higher non-dimensional mass flow rate of 0.72, flow field was investigated to reveal the mechanism of the effects of negative inlet swirl on the compressor performance. **Figure 11** shows the static pressure loading at 90% span. The full blade is under a small negative incidence with no preswirl as shown in **Figure 11(a)**, while the splitter blade is under a considerable negative incidence. The largest pressure loading of full blade and splitter blade locates both at about 65% chord length, where low pressure regions appear on the blade suction surfaces. Two supersonic regions are formed on the blade suction surfaces, especially on the splitter blade, as shown in **Figure 12(a)**.



(a) No preswirl

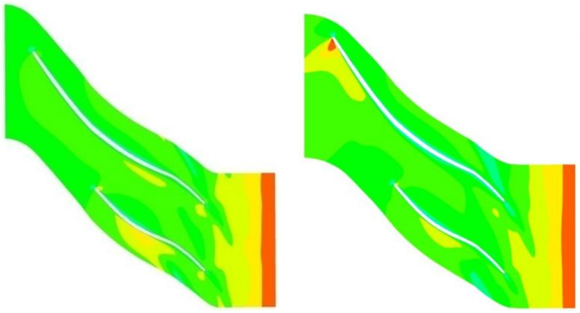


(b) -17° preswirl

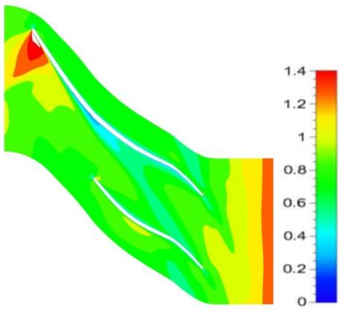


(c) -32° preswirl

Figure 11 Blade loading at 90% span, 50,000rpm and mass flow rate of 0.72



(a) No preswirl (b) -17° preswirl

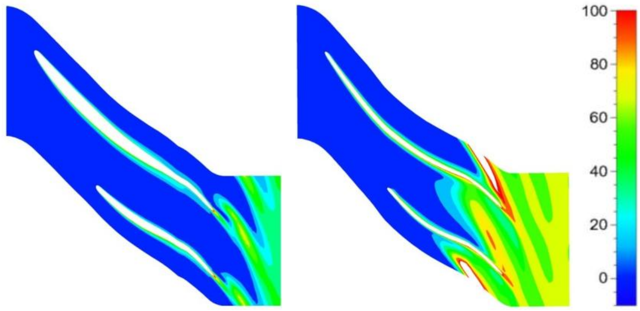


(c) -32° preswirl

Figure 12 Relative Mach number contour at 90% span, 50,000rpm and mass flow rate of 0.72

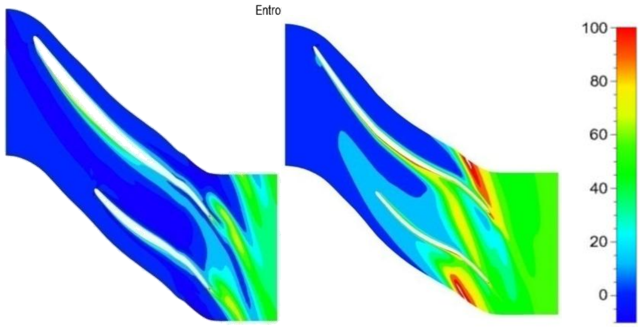
When the negative preswirls are imposed, the incidence to the full blade near the shroud turns positive, and the inducer blade loading is improved. This is shown in **Figure 11** (**b** and **c**). Splitter incidence is also slightly improved and splitter loading shifted forward. The relative Mach number is increased too at the inducer suction surface of the full blade as shown in **Figure 12** (**b** and **c**) due to the large positive incidence. A supersonic flow region is formed near the leading edge followed by a shockwave and a thickened boundary layer caused by shockwave-boundary layer interaction. While the compressor pressure ratio is increased by more Euler work, the benefit of earlier diffusion in the inducer is lost to the higher shockwave losses, so the compressor efficiency improves little at -17° preswirl, and reduces at the larger negative preswirl.

Figures 13 ~15 further show the entropy contours at 50% and 90% spans. With no preswirl, **Figure 13**, at 90% span, two high entropy regions appear just after the supersonic regions and extend to the impeller exit, indicating the influence of the overtip leakage caused by the large blade loading at these locations. The -17° preswirl reduces aft blade loading at 90% span and so the entropy at this span, **Figure 14**. But the entropy is increased at 50% span by a thickened boundary layer. When -32° preswirl is applied, an inducer boundary layer separation can be observed at the both 50% and 90% span, **Figures 15**.



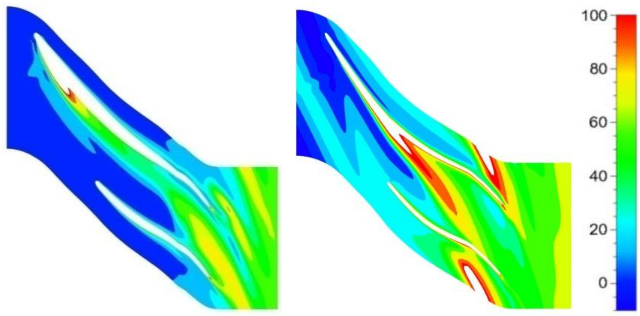
(a) 50% span (b) 90% span

Figure 13 Entropy contour with no preswirl at 50,000rpm and mass flow rate of 0.72



(a) 50% span (b) 90% span

Figure 14 Entropy contour with -17° preswirl at 50,000rpm and mass flow rate of 0.72

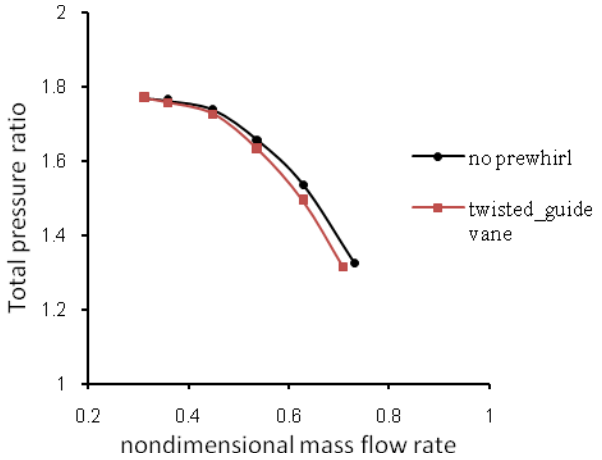


(a) 50% span (b) 90% span

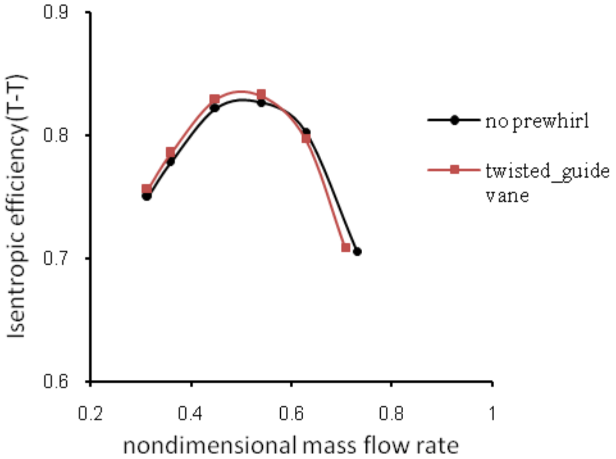
Figure 15 Entropy contour with -32° preswirl at 50,000rpm and mass flow rate of 0.72

5. CFD results of twisted vanes

Figure 16 shows the pressure ratio and efficiency characteristics of the compressor with the twisted vanes and with no inlet swirl at the rotational speed of 50000rpm. The low ends of the characteristic curves were at the same non-dimensional mass flow rate of 0.31. The twisted vanes result in a lower choke flow and some reduction in pressure ratio at high mass flows. Compressor stability at low flows increases with the vanes and the peak efficiency is improved slightly, just as with the positive preswirls. **Figure 17** shows pitchwise averaged impeller inlet absolute flow angle distribution. The preswirl angle increased from -12° at root to 12° at tip and the positive preswirl is imposed at the compressor inlet above 30% span.



(a) Pressure characteristic at 50,000rpm



(b) Efficiency characteristic at 50,000rpm

Figure 16 Performance characteristics of the compressor with twisted guide vanes

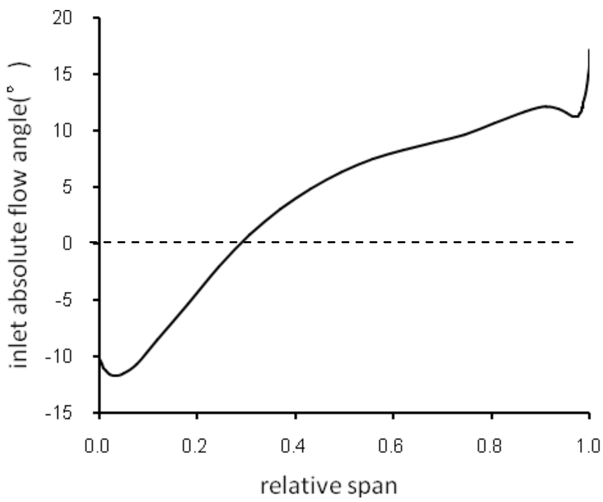
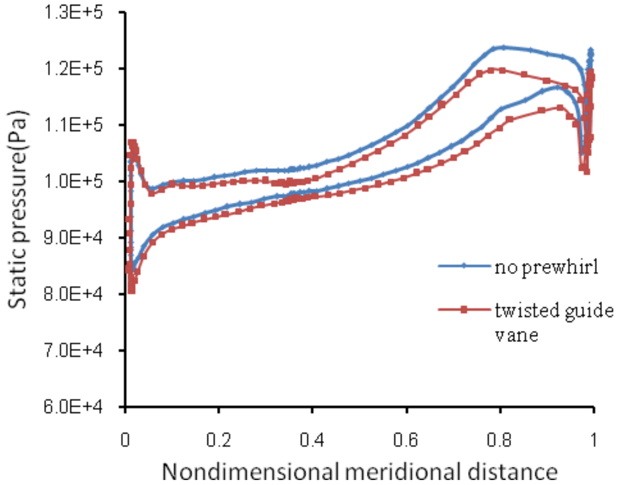
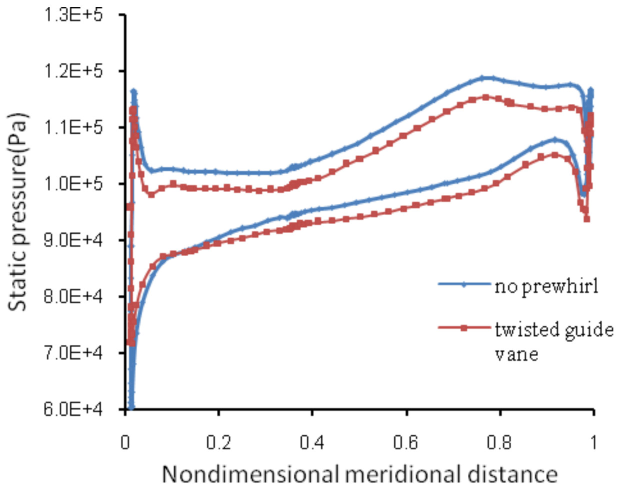


Figure 17 Impeller inlet absolute flow angle distribution at mass flow rate 0.54, twisted guide vanes

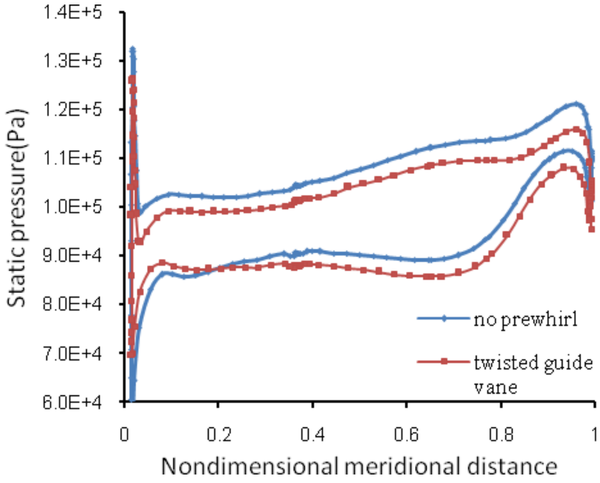
Figure 18 shows the static pressure loading of main blade at different spans at the mass flow rate of 0.63 near the choke end. At 10% span, the inducer blade loading is improved with the twisted guide vanes as shown in **Figure 18(a)**. It is because the negative preswirls are imposed at the hub, as shown in **Figure 17**. At 50% and 90% span, the twisted vanes result in the reduction in inducer blade loading when the full blade is under the positive preswirls at high spans. Increasing positive preswirl angle leads to more reduction in blade loading. The improvement of inducer blade loading at the hub and smaller reduction at mid span lower the reduction of compressor pressure ratio at high mass flows. And the compressor surge margin also benefits from the positive preswirl at the blade tip when the twisted vanes are used.



(a) 10% span



(b) 50% span



(c) 90% span

Figure 18 Blade loading of main blade at 50,000rpm and mass flow rate of 0.63

6. Experimental results

Three fixed geometry inlet guide vanes were manufactured and tested. One such vanes is with blade angle of 60° at the shroud and 50° at the hub to produce a mean inlet preswirl of +32°, another has the same magnitude but opposite blade angle sign to the first to produce a mean inlet preswirl of -32°, and the third has twisted guide vanes of 30° at the shroud and -50° at the hub. **Figure 19** shows the three inlet guide vane prototypes. These devices were mounted directly onto the compressor housing inlet and tested together with the same impeller and housing, the same compressor without any inlet guide vanes was also tested as the baseline.

Figure 19 Photos of the inlet guide vane prototypes (from left to right: positive preswirl vane, twisted guide vane and negative preswirl vane)

6.1. Test results of the positive preswirl vane

Figure 20 shows that the test results of +32° preswirl and no preswirl. Test results confirm the findings of the CFD. The positive preswirl causes a considerable change in the compressor characteristic: The compressor surge line is shifted to smaller flow rates especially at high speeds, and compressor peak efficiency is increased. The compressor efficiency at the surge end is improved too, although all these benefits are at a price of reduced compressor performance at the choke end.

6.2. Test results of the negative preswirl vane

Figure 21 shows the performance characteristics of -32° preswirl and no preswirl. The negative preswirl slightly increases the choke flow at 50,000rpm and reduces the flow at 80,000rpm, it also has little effect on the compressor pressure ratio. These are consistent with the CFD investigation. The test also shows the large reduction of compressor efficiency as suggested by the CFD. However, the test results does not show any significant effect on compressor surge flow by the negative inlet preswirl and this contradicts with the CFD finding. Further investigation into this disagreement is necessary.

6.3. Test results of the twisted guide vane

Due to the exit blade angle distribution of the twisted guide vane, positive preswirl and negative preswirl are applied at the tip and the hub of the impeller inlet respectively. **Figure 22** shows the tested result of the compressor with the twisted guide vanes and no inlet swirl. The map of the compressor is shifted to the left with the twisted guide vanes compared to that of no preswirl. Although the improvement of compressor surge at high speeds is not as large as that of +32° preswirl, the reductions of compressor pressure ratio and choke flow are much smaller than the positive preswirl. Compressor peak efficiency and compressor efficiency at bottom-left corner of the map are improved too. The result indicates that surge margin is mainly affected by the preswirl at the shroud and the negative preswirl at hub can help improve the pressure ratio and choke flow.

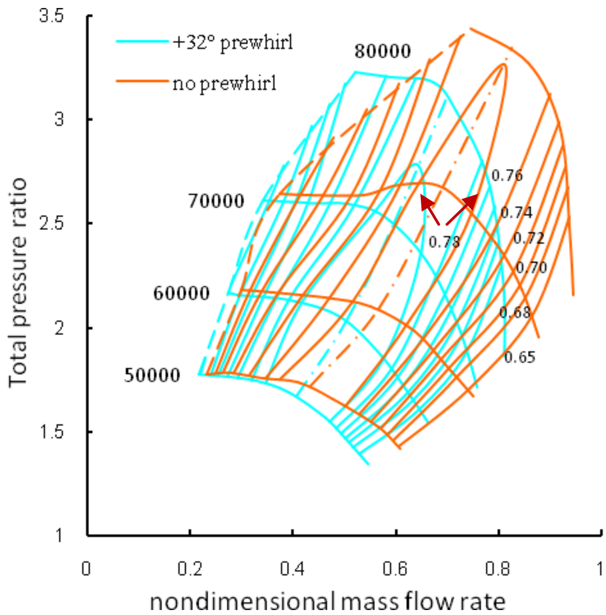
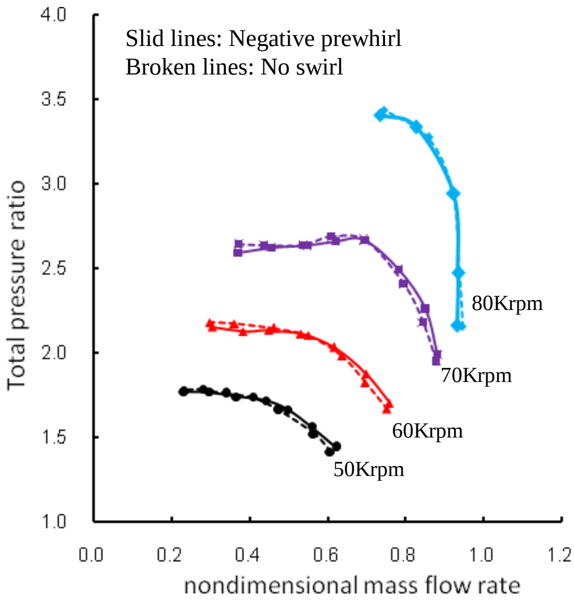
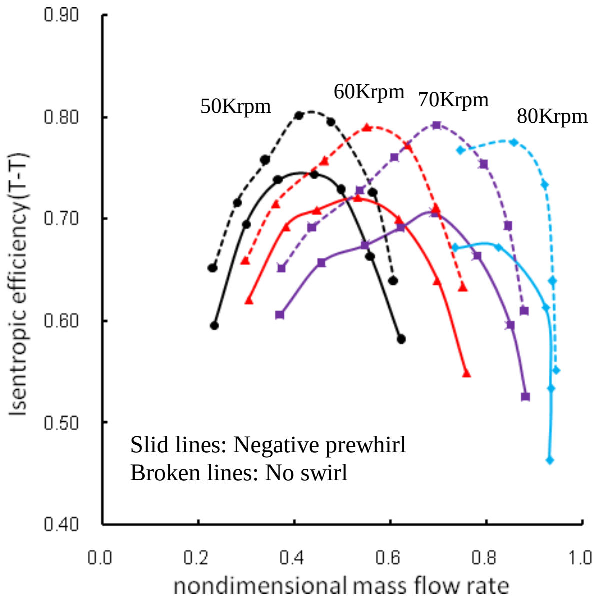


Figure 20 Performance characteristics of +32° preswirl and no preswirl



(a) Pressure charateristic



(b) Efficiency characteristic

Figure 21 Performance characteristics of -32° preswirl and no preswirl

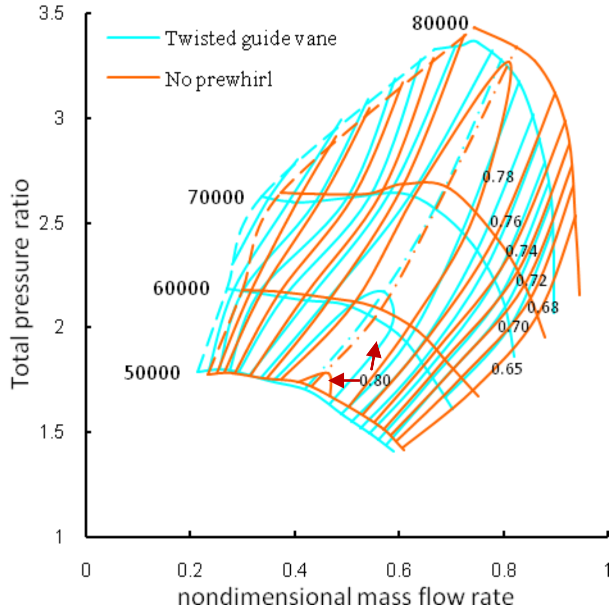


Figure 22 Performance characteristics of twisted guide vane and no preswirl.

7. Conclusions

CFD simulation and experiment were carried out to investigate the effects of inlet swirl on the performance of a turbocharger compressor and the mechanism of these effects. Several types of inlet guide vanes were designed to generate spanwise uniform preswirl flow and one twisted vane to provide variable preswirl flow at the impeller leading edge. ±17°, ±32° preswirl vanes and the twisted vane were simulated with a single impeller passage to obtain the compressor performance, and results compared with those of no preswirl.

CFD results revealed that the positive preswirls improved the surge margin and peak efficiency of the compressor but caused a reduction of pressure ratio at chock end. Near the numerical instability point, Impeller inlet flow became more uniform spanwise, and blade loading increased near the shroud and reduced at the hub. Impeller exit flow also became more uniform suggesting smaller losses in the downstream diffuser.

In contrast, the larger negative preswirl led to efficiency drop almost at the whole flow range. At high flow rates, the pressure loading was shifted to the leading edge of the full blade. The large positive incidence generated a strong shockwave-boundary layer interaction, causing the flow to separate from inducer suction surface and higher losses.

With twisted vane, the preswirl angle increases from negative at the hub to positive at the tip. The negative preswirl improves the inducer blade loading at the hub while the positive preswirl has opposite effect at higher spans. The variable of preswirl angle at the compressor inlet could reduce the negative effect to compressor pressure ratio when a uniform positive preswirl is applied.

The compressor with ±32° preswirl was tested and its performance compared to that of no preswirl. Results revealed that the positive preswirl improved the stability and efficiency of compressor at surge end with a reduction of chock limit and compressor pressure ratio. The negative preswirl led to a large reduction in compressor efficiency as suggested by CFD. It had little effects on the pressure characteristic and stability of the compressor, indicating that the stability mechanism could not be found by the current CFD simulation using a single rotor passage.

Additional test was performed on a twisted guide vane with positive exit blade angle at the tip and negative exit blade angle at the hub. With it the surge margin and peak efficiency of compressor were both improved and the pressure ratio recovered from that of the positive preswirl. The result suggests that positive preswirl at the shroud and negative preswirl at the hub seem to be the best preswirl strategy.

Acknowledgements

This work is supported by Science Foundation for National Laboratories of China, Grant No. 9140C330104120C33119.

References

[1] S. Sivagnanasundaram, S. Spence, J. Early, An impact of various shroud bleed slot configurations and cavity vanes on compressor map width and the inducer flow field, ASME, GT2011-45629, 2011.

[2] S. Sivagnanasundaram, S. Spence, J. Early, Investigation of compressor map width enhancement and the inducer flow field using a shroud bleed slot, ASME, GT2010-22154, 2010.

[3] N. Fraser, T. Fleischer, J. Thornton, Development of a Fully Variable Compressor Map Enhancer for Automotive Application, SAE Paper 2007-01-1558, 2007.

[4] J. Galindo, et al., Potential of flow pre-whirl at the compressor inlet of automotive engine turbochargers to enlarge surge margin and overcome packaging limitations, International Journal of Heat and Fluid Flow, 28(2007) 374-387, 2007.

[5] S. H. Yousef, A. M. S. Saleh, Effect of preswirl on the performance of centrifugal compressors, International Journal of Rotating Machinery, 8(6): 397–401, 2002.