The investigation of stability enhancement within a two-stage compressor utilizing the non-asymmetric stator

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Abstract. As the compressor plays a key role in the marine-used gas turbine, its stability determines the operation condition of the power engine of the ship. The paper investigates the performance improvement of the compressor by utilizing a new-designed asymmetric stator. The results show that the flow distortion upstream is significantly constrained by the developed stator and the unsteadiness of the static pressure coefficient and Mach number at R2 inlet decrease by 74% and 48% respectively.

1. Introduction
Serving as a key component of the marine-used gas turbine, the compressor is commonly expected to have high performance and stability. However, because of the effects of complex pipe systems and non-uniform income air condition within the pipe, the compressor performance is usually threatened by the unwelcomed flow distortion and even the stall phenomenon. Up to now, several researches have paid attentions to the stability enhancement of the compressor and the casing treatments are the most commonly adopted methods. For examples, Hathaway M D [1] validates the effectiveness of the self-recirculation device within a single-stage compressor through the numerical simulation. From the results, it is noticed that the flow separation at the rear part of passage could be absorbed by the device and the separation flow is transported to the front part of the passage through the recirculation slot, which helps to compensate the flow deficit near the leading edge of blade and improve the compressor stability. Based on this classic design concept, several improved self-recirculation slot devices were invented. The works from Li J C [2], Wang W [3] and Khaleghi H [4] improved the geometry parameters of the self-recirculation slot. All of these devices transport the high pressure flow from the exit of rotor passage to the bleeding hole near the leading edge of rotor. According to the conclusions, the separation vortexes at the leading edge of rotor tip would be damaged by the bleeding flow and the goal of stability enhancement could be achieved. A special self-recirculation slot device, which connects the compressor inlet and the location of leading edge of splitter blade, was presented by Tamaki H [5] in recent years. The principle of this device lies on the absorbing of the blockage flow after the rotor shock wave. The essentials of all the inventions above are similar, which could be concluded as the following: the flow inside the self-recirculation is driven by the pressure rise produced by the rotor/impeller work. The shortage of this method is consequently brought as the rotor work ability is consumed. For the cases of
obvious flow separation at rotor exit (as described in Ref. [1]), the effect of the consumption of the rotor work may not be outstanding because of the improvement of flow condition near the rotor exit.

As the industrial compressor is always more than one stage, the transportation of the inlet distortion from the front rotor stage to the rear stage is another potential factor affecting the aerodynamic stability of the rear part of compressor. Some past researches focused on the non-uniform stator technique which improves the transportation of the flow distortion from the rotor upstream. The early studies from Gottfried DA [6] and Kaneko Y [7] altered the blade number at some specific circumferential locations of the stator, which aims to weaken the force response level of the rotors downstream. Florea RV [8] and Wartzek F [9] investigated the propagations of different boundary-flow distortions within the compressor and the consequent flow distribution at the stator exit. According to the results, the inlet flow distortion will be amplified by the transportation through the stator row. Based on the early mechanisms study, Gunn EJ [10] presented a concept of non-uniform stator with the modulation of the geometric parameters such as the chord and lean angle at part of the stator. The design is approved to reduce about 10% of the flow losses. However, Most of these designs are used for single-stage compressor and the interaction between the non-uniform stator and the stages downstream were neglected. Besides, more geometric parameters of the non-uniform stator should also be further investigated.

Considering the problems above, the paper investigates a novel layout of the compressor stator row in which the non-uniform stator blades are adopted to weaken the flow unsteadiness generated by the inlet flow distortion.

2. The concept of the design

Firstly, the concept of the new device is presented, as is shown by Fig.1. For the non-uniform stator, which is shown at the upper side, three types of stator blades are utilized and their geometric parameters are altered in terms of the following principle (the result has been optimized to some extent): (1) the circumferential range of the non-uniform blades is determined by the range of inlet flow distortion in reality ($\theta_1$), here is defined as $1.2\theta_1$; (2) the values of $l_1/l_3$ and $l_1/l_2$ are defined as 1.15 and 1.06 respectively (at the blade tip); (3) the blade thickness changes in terms of the normal distribution, from S1-1 to S1-3; (4) the stagger angle of the blade decreases from S1-1 to S1-3, which follows the normal distribution; (5) compared with S1-3, the changes of the blade lengths of S1-1 and S1-2 focus on the region upper than 55% blade height (shown by the meridional contour of the compressor).

3. Numerical methods

The object investigated here is a small size counter-rotating two-stage compressor, the design speed and total pressure ratio of which are 29500rpm and 1.44. The commercial CFD software FINE/Turbo was employed to finish the computational prediction work in which the S-A turbulence model was used. The calculations are based on RANS (for steady calculation) and URANS (for unsteady calculation) method. To make the prediction more accurate, the whole wheel model, adopting the grid topology of OH pattern, was used and the grid number of one passage for each row (including the IGV) reaches 253000, 316000, 253000, 316000, 253000 respectively. For the novel branch device, the grid number of one passage is 47000. As a consequence, the value of $Y^+$ for each component could be controlled less than 5. According to the research by Spalart P [14], the boundary flow characteristics inside the fluid machine could be well predicted under such $Y^+$ range when using S-A turbulence model. Figure 2 shows the detailed grid meshing of the compressor. The mix-plane interface and sliding grid techniques were used in the steady and unsteady calculation respectively. The uniform inlet boundary condition was defined as the combination of inlet total pressure of 101325 Pa, inlet total temperature of 288 K and the inlet axial flow direction. The outlet condition enforced the average static pressure as the conditions selected in the research exclude the stall limit. Besides, the convergence judgment for the steady simulation was proposed as followings: (1) the root-mean-square (RMS) residual is below $10^{-6}$, and (2) the difference between the inlet mass flow and outlet mass flow is less than 0.1%. The unsteady calculation was based on the steady results, in which the convergence was judged by the followings: the fluctuation of the selected parameter is periodic.
4. The validation of the numerical method

The comparison of the experimental and numerical results of the compressor performance at 100% speed line is shown in Fig. 3. The red dashed line in the figure represents the mass flow rate at design point. The pressure rise curves on the left side and the adiabatic efficiency on the right side indicate that the designed mass flow rate of CFD is 4% lower than that of experimental results. Meanwhile, it is noticed that the differences of the static pressure ratio and adiabatic efficiency between the CFD and experiment are less than 1.6% and 0.24% at the design point, and the trends of numerical and experimental results are similar. Thus, it is believed the CFD accuracy of the flow prediction is acceptable in this paper.

The grid independence analysis at 100% speed line is shown in Fig. 4. The results indicate that as the grid number of one passage increases from the range of 120000−170000 to the range of 250000−320000, the max decrease of static pressure ratio and the polytropic efficiency reach about 0.8% and 0.2%
respectively, which are not outstanding. When the grid number of one passage reaches the range of 250000–320000, both the static pressure ratio and the polytropic efficiency nearly remain unchanged, which indicates that the grid number of each passage chosen in the research is relatively proper.

![Figure 3 The comparison of the compressor performance](image1)

![Figure 4 The grid independence study](image2)

5. RESULTS AND DISCUSSIONS

As is shown by the distribution of relative Mach number in Fig. 5, the inlet flow distortion generates serious flow separation within the R1 passage. However, the utilization of the new-designed asymmetric stator (developed S1, the third blade row) appropriately constrains this unwelcome phenomenon. It is noticed that the distribution of the relative Mach number in each S1 passage keeps at the similar trend, especially for the developed passages corresponding to the position of inlet flow distortion of which the wake regions are significantly weakened when compared to the situation of the original S1 blade. For the design point, the distribution of the static pressure at 90% span in Fig. 6 tells more benefits of the developed S1. For examples, even under the effect of flow distortion upstream, the static pressure rise after the developed stator blade is the highest. Moreover, the performance of the developed stator under the flow condition without distortion is still acceptable and the related pressure loss is limited.
Figure 5 The distribution of the relative Mach number by using the asymmetric developed stator, at 90% span

Figure 6 The distribution of the static pressure by using different stator blades

The improvements of the flow unsteadiness inside the S1 passage by utilizing the developed stator are shown in Figs.7 and 8. The amplitude of the fluctuation of relative Mach number decreases by about 74% and it remains at the similar level with the situation without flow distortion. The amplitude of the fluctuation of static pressure coefficient decreases by about 48%, which is also similar to the situation without flow distortion.
Figure 7  The comparison of the unsteadiness of the relative Mach number between the original stator and the developed stator, at 90% span of the inlet of R2

Figure 8  The comparison of the unsteadiness of the static pressure coefficient between the original stator and the developed stator, at 90% span of the inlet of R2

The performance comparisons in Figs.9 and 10 indicate that the asymmetric stator has very limited extra performance degradation even no flow distortion exists. Detailedly, take the design point and near stall point for examples, the adiabatic efficiency of the compressor with developed stator is only 0.09% and 0.01% lower than the original design. For the comparison of total pressure ratio, almost no difference exists between both of the compressor.
Figure 9 The comparison of total pressure ratio between the compressors utilizing the original stator and asymmetric stator

Figure 10 The comparison of adiabatic efficiency between the compressors utilizing the original stator and asymmetric stator

6. CONCLUSIONS
The paper investigates the performance improvement of the compressor by utilizing a new-designed asymmetric stator. Several conclusions are shown as followings:

(1) The flow distortion upstream is significantly constrained by the developed stator, the stator wake regions are significantly weakened and static pressure rise after the developed stator blade appears to be the highest.

(2) The flow unsteadiness at the R2 inlet is improved and the static pressure coefficient and Mach number decrease by 74% and 48% respectively.

(3) Even under the flow condition without flow distortion, the developed asymmetric stator just brings very limited extra pressure loss.

ACKNOWLEDGEMENT
The authors would like to gratefully acknowledge the support of AECC Industry-University-Research-Project Fund (No.GJLZ-2020-0027).
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