Design Concept with Tip Leakage Vortex Control for Centrifugal Compressor Flow Stabilization

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Abstract. Centrifugal compressor applied to turbochargers is required to operate stably in wide range from choking to surging. In our past research, authors suggested a design policy called STB (flow Stabilization by inducing Tip leakage vortex Breakdown) concept, which indicates “Operating range can be enhanced with uniform blockage region generated by tip leakage vortex breakdown”. In this study, authors tried to validate the STB concept by redesigning a given impeller into a wide range compressor. In this research, detailed analyses showed that non-uniform and unstable stall occurred at small flow rate in the conventional impeller. On the other hand, in the redesigned new impeller having high inducer loading, unsteady numerical calculation showed that the strengthened blade tip leakage flow generated a circumferential uniform blockage region, which could stabilize its internal flow. As a result of performance test of the new impeller, considering that the surging flow rate at the same shaft speed was reduced by 3% and the pressure ratio at the surging point had been improved from 2.8 to 2.9, the surging flow rate at the pressure ratio of 2.8 could be reduced by 8%. In this way, it was found that the tip leakage flow was dominant with the stall phenomenon of the centrifugal compressor, and it was also confirmed that the STB concept was one of the effective means for the operation range enhancement.

1. Introduction

Operating range enhancement of centrifugal compressor for turbochargers is strongly demanded as engine development upgrades. However, how to control internal flow structure to enhance the operating range has been hardly understood because strong secondary flow due to the centrifugal and Coriolis force causes extremely complicated flow fields in centrifugal compressor impellers especially at low flow rate condition. Therefore, phenomena near surging or stall condition have not been clarified as for the axial compressor.

Regarding to rotating stall of the axial compressor, Moore and Greitzer[1][2] showed that the flow field in the compressor became unstable at the maximum pressure rise point based on the theoretical model of the compressor, and its model was confirmed experimentally by McDougall[3], Garnier[4], Day[5], Poensgen and Gallus[6]. It was shown that disturbance wave of small scale rotating stall cells had a scale about 2-3 times of blade pitch.

The structure model of the small scale rotating stall cell was proposed experimentally by Inoue[7][8]. Its model was that tornado type vortex with a leg on its blade suction surface caused a new leading edge separation on next blade. Yamada et al.[9] confirmed by numerical analysis that the small scale rotating stall cell has the same flow structure as the flow model.
Many researches had been done for the rotating stall or the rotating instability in axial compressor for a long time, and the classification of the form of the phenomenon had been advanced comparing to the centrifugal compressor.

For the centrifugal compressor, Lenneman and Howard [10] experimentally investigated its number and the rotating speed of the rotating stall cells as a typical research on the rotating stall in impeller. And the number of stall cells had tendency to increase depending on the impeller blade number. Compressor impellers with small blade number and high blade height are often used for automotive turbocharger because of the demand of reduction in size and weight so the aspect ratio has to be small. Tomita [11] investigated an example by experimental and numerical approach that propagating phenomena like rotating stall showed unstable behavior in such centrifugal compressor having small blade number. And it was also shown that impellers which induced tip leakage vortex breakdown at high flow rate might stabilize their internal flow structure by generating circumferential uniform blockage region near blade tip at low flow rate.

Tomita also investigated whether STB (flow Stabilization by inducing Tip leakage vortex Breakdown) concept, which is modifying a given impeller to induce the tip leakage vortex breakdown to make its internal flow structure stable, could reduce the surging flow rate or not [12]. In this research, authors discussed detailed blade loading and internal flow structure obtained by CFD calculation to support their argument.

2. Test Compressors

Table 1 shows specifications of the conventional centrifugal compressor for automotive turbocharger. They have open type and back swept impeller. The conventional impeller has 6 full blades and 6 splitter blades. The new compressor, which is described later part of this paper, has same blade number and exactly same meridional geometry of the impeller and vaneless diffuser. And same scroll is applied as well.

| Table 1 Specification of the Test Compressors |
|---------------------------------------------|
| Number of full blade | Conventional Compressor and New Compressor |
| Number of splitter blade | 6 |
| Impeller inlet diameter | 6 |
| Impeller outlet diameter | 40.1mm |
| Impeller outlet diameter | 51.0mm |

3. Numerical Scheme

Figure 1 shows computational grid including impeller, diffuser and scroll for this research. The number of grid is about 2,470,000 for the impeller as structured mesh, 410,000 for the diffuser and 250,000 for the scroll as unstructured mesh. Steady and Unsteady CFD were conducted with CFX ver.12 and k-ε turbulence model was applied. 178,000rpm was chosen as the research speed because the conventional compressor has an apparent pressure peak point and surging flow rate increases at that speed comparing to lower speeds.

Frozen-rotor was applied to the boundary condition between rotating part and stationary part in the case of steady CFD. In the unsteady CFD, time step per iteration was 0.0028ms, which corresponded to 20 steps per 1 full blade passing (BPF: Blade Passing Frequency).

Vortex core identification and normalized helicity Hn were applied to visualize internal flow structure. The normalized helicity corresponds to cosine value of angle between vectors of absolute vorticity and relative flow velocity and it is useful to detect vortex breakdown. Hn=+1 represents a clockwise longitudinal vortex, and Hn=−1 represents counterclockwise longitudinal vortex.
4. Conventional Compressor

4.1. Compressor Performance

Figure 2 shows the performance map of the conventional compressor obtained by measurement. The flow ratio Q* as the horizontal axis is normalized by the choking flow rate of the conventional compressor at 178,000rpm, which has a peak pressure point at Q*=0.72 and positive slope. Its operating range becomes narrow at over $\pi=2.4$ because its surging flow rate sharply increases at 178,000rpm comparing to lower speeds. The conventional compressor has a typical performance characteristic of centrifugal compressors for automotive turbocharger. The efficiency is also normalized by the maximum efficiency of the conventional compressor.

Pressure measurement was conducted to investigate unsteady behavior in the conventional compressor [12]. Although cyclic pressure fluctuation due to the blade passing was seen at the peak efficiency point (Q*=0.83) at 17,800rpm, unstable fluctuation appeared from peak pressure point (Q*=0.72) until near surging point (Q*=0.65). This fact may explain that impeller stall or tip leakage vortex fluctuation occurs at this region.
4.2. **Steady CFD Results**

Conventional compressor performance obtained by steady CFD is shown in figure 3. Air flow ratio as the horizontal axis is normalized by the choking flow rate of the conventional compressor obtained by measurement. The steady CFD was conducted at choking condition, Q*=0.95 to confirm the performance characteristic, Q*=0.84 as the peak efficiency point, Q*=0.72 as the peak pressure point and Q*=0.60 as well developed stall condition at 178,000rpm. It showed reasonable choke flow rate, efficiency and pressure ratio comparing to the measurement result. Since CFD showed that Q*=0.72 is higher than Q*=0.84 at pressure ratio, the impeller might stall at higher flow ratio in the CFD.

![Performance of the Conventional Compressor at 178,000rpm](image)

Relative Mach number distribution at the 90% span is shown in figure 4. At Q*=0.84 as a typical peak efficiency point, all of the blades had almost uniform loading and low velocity region due to leakage flow were found. It is naturally thought that no unstable phenomenon has occurred. At Q*=0.72 at the steady CFD, although two of the blades were working normally without low velocity region, a stalled region with large low velocity was identified. This non-uniformity could be occurred because of the scroll or potential unstableness of rotating stall. The stall region expanded at Q*=0.60 and was one large scale stall cell occupying almost all of the blades.

![Relative Mach number Distribution at 90% Span](image)
Figure 5 shows static pressure distribution along the blades of the conventional compressor at 90% span and 50% span. Gray lines show all blades distribution and black line shows averaged distribution. Pressure difference between pressure surface and suction surface corresponds blade loading. At design point, blade normally makes pressure difference between pressure surface and suction surface by its loading. However, when the impeller stalls, low pressure region of the suction surface at leading edge weakens and the pressure difference between pressure surface and suction surface becomes smaller. We can find shock wave around $m=0.25$ at peak efficiency point ($Q^*=0.84$). Although the blades work uniformly at the peak efficiency point ($Q^*=0.84$), blades have different loadings at the peak pressure point ($Q^*=0.72$) which means that non-stalled blades and stalled blades are existing at the same time. And it is clear that all blades stall at 90% span near surging point ($Q^*=0.60$) because pressure differences are small at all blades.

![Figure 5 Static Pressure Distribution along the Blades](Conventional Compressor, Steady CFD)
Mach number distribution near blade tip and identified vortex core colored by normalized helicity in one of the blades at Q*=0.84 is shown in figure 6. The compressor in the figure was rotating clockwise as viewed from the upstream, and the longitudinal tip leakage vortex rolls counterclockwise. Once the tip leakage vortex breakdown occurs, reverse flow generated by adverse pressure gradient makes longitudinal vortex rolling clockwise. If the vortex core shape cannot be maintained, spiral type vortex breakdown occurs. It is found that the tip leakage vortex was not broken in spite of the interaction with shock wave generated near blade suction side in the conventional compressor.

![Mach number distribution](image)

**Figure 6** Internal Flow Structure in the Conventional Compressor at Peak Efficiency Point (Q*=0.84, Steady CFD)

4.3. Unsteady CFD Results

Unsteady CFD was conducted to research unsteady flow behavior at Q*=0.84, 0.72 and 0.60. Following figures show instantaneous result after three rotation calculation. Pressure fluctuation near the impeller leading edge was discussed by Tomita[12]. Stable cyclic pressure fluctuation can be found at Q*=0.84. At Q*=0.72, although the BPF is still dominant, the shape and amplitude were fluctuating like the measurement result. At Q*=0.60, low frequency phenomenon occurred and the cyclic fluctuation couldn’t be seen any more.

Figure 7 shows limiting streamline on suction surface, vortex cores colored by normalized helicity and regions of an axial flow velocity Vz=0m/s in an axial cross section of the impeller leading edge at an instantaneous flow field. From the limiting streamline, it is possible to visualize secondary flow caused by separation on the suction surface or blockage and to easily detect changes in the flow structure. When the tip leakage vortex collapsed or hit next blade, the meridional velocity decreases. At Q*=0.84, the tip leakage vortex were generated but no reverse flow could be found. On the other hand, reverse flow near the blade tip was generated at the half of the blades at Q*=0.72. And tip leakage vortex with positive normalized helicity could be seen there. The reverse regions consisted of the broken tip leakage vortex and reverse flow coming from the secondary flow on the suction surface. The other half of the blades worked normally as Q*=0.84 without reverse flow. The reverse flow region expanded near the surging point at Q*=0.60. And the reverse flow kept its non-uniformity, where the blade 1 stalled but the blade 4 worked normally shown in the figure.

Vortex cores colored by normalized helicity and limiting streamline of the blade 1 and 4 of the figure 7(c) are shown in figure 8. The tip leakage vortex breakdown occurred at the blade 1, and strong secondary flow existed around the blade 1. On the other hand, it was found that the blade 4 worked normally with weak secondary flow. Although this kind of vortex structure appeared when a rotating stall occurred, its behavior was strongly unstable like a compressor shown by Tomita[11].
5. NEW COMPRESSOR

5.1. Impeller Design

As shown in the previous chapter, it was found that a circumferential non-uniform blockage region was generated by tip leakage vortex breakdown or stall cells in the conventional impeller. In our past research, it was suggested that impellers which induced tip leakage vortex breakdown at relatively high flow rate might stabilize internal flow by generating circumferential uniform blockage region near blade tip at low flow rate. In this study, authors investigated whether modifying a given impeller to induce the tip leakage vortex breakdown can reduce the surging flow rate or not.

Authors designed a new impeller which had higher loading at the inducer part to strengthen the leakage flow and the secondary flow to generate a circumferential uniform blockage to stabilize the flow field. As shown in Table 1, the main specification such as the number of blades and the meridional geometry such as the blade inlet diameter, the outlet diameter, the blade height were maintained. And the impeller was also designed to increase not only near the inducer loading but also overall loading. Same vaneless diffuser and scroll were used for the new compressor as well.
5.2. Numerical Results

Figure 9 shows steady CFD result at the peak efficiency point ($Q^*=0.84$) at 178,000rpm. Bubble type tip leakage vortex breakdown interacted with shock wave could be seen at all of the blades.

Blade loading comparison with static pressure distribution along the blade is shown in Figure 10. It is found that the new compressor impeller has higher loading near leading edge both 90% span and 50% span. It is the design concept increasing blade loading near leading edge to make tip leakage vortex and secondary flow large. Although aft part loading was also increased because the blade angle was tuned to maintain its pressure ratio, the STB concept was on the front part. Figure 11 shows static pressure distribution of the new compressor impeller. It is successfully achieved to create uniform blade loading not only at peak efficiency point and peak pressure point but also near surging condition.
Unsteady CFD was performed on the newly designed compressor as well as the conventional one. Pressure fluctuation near impeller leading edge was discussed in the previous research [12]. Unsteady phenomena was not detected in the new compressor at least at $Q^* = 0.72$ in spite of the fact that unstable phenomena happened in the conventional compressor. At $Q^* = 0.60$, low frequency disturbance occurred and the cyclic fluctuation couldn’t be seen as well as the conventional compressor.

Same as figure 7, figure 12 shows limiting streamline on suction surface, vortex cores colored by normalized helicity and regions of an axial flow velocity $V_z = 0 \text{m/s}$ in an axial cross section of the impeller leading edge at an instantaneous flow field. No reverse flow appeared at $Q^* = 0.84$. Reverse flow caused by the tip leakage vortex breakdown and secondary flow near the blade tip could be found at all of the blades at $Q^* = 0.72$ but their scale and condition of the limiting streamline were almost uniform. Although the reverse flow region expanded at $Q^* = 0.60$.

All of the differences on the phenomena and performances were caused by the new impeller because the same vaneless diffuser and volute were applied in this study. From the fact that impeller
pitch-wise distribution was uniformed by the new impeller, it is possible that the STB concept also has a tendency to stabilize the diffuser part as well.

![Diagram](image)

(a) Peak Efficiency Point (Q*=0.84) (b) Peak Pressure Point (Q*=0.72) (c) Near Surging Point (Q*=0.60)

Figure 12  Instantaneous Internal Flow Structure (New Compressor, Unsteady CFD)

5.3. Performance

Figure 13 shows the pressure ratio and the efficiency ratio obtained by a performance test. Air flow rate is normalized with the choking flow rate of the conventional compressor obtained by measurement. As a result of the performance test of the new impeller, considering that the surging flow rate at the same shaft speed is reduced by 3% and the pressure ratio at the surging point has been improved from 2.8 to 2.9, the surging flow rate at the pressure ratio of 2.8 can be reduced by 8%. In this way, although efficiency ratio was deteriorated by 0.03pt, it was found that the tip leakage flow was dominant with the stall phenomenon of the centrifugal compressor.

![Graph](image)

(a) Total Pressure Ratio (b) Efficiency Ratio

Figure 13  Performance Comparison at 178,000rpm (Measurement)
6. Conclusion

In this study, authors tried to validate the STB concept indicating “Operating range can be enhanced with uniform blockage region generated by tip leakage vortex breakdown” by modifying a given impeller into a wide range compressor.

Unsteady CFD revealed that circumferential uniform blockage region generated by weak tip leakage vortex breakdown was seen in the new compressor, which increased its inducer loading, and the blockage region gradually developed from the peak efficiency condition to near surging condition. It is thought that the uniform blockage region can keep the internal flow stable in the new compressor, in spite of the fact that non-uniform complex phenomena happened in the conventional compressor.

As a result of the performance test of the new impeller, the surging flow rate at the pressure ratio of 2.8 can be reduced by 8%. In this way, it was found that the tip leakage flow is dominant with the stall phenomenon of the centrifugal compressor, and it is confirmed that the STB concept is one of the effective means for the operation range enhancement. Since authors have not done any optimization on the loading design, research to find the best tuning to minimize the efficiency loss can be done with CFD as a future work.

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