Aerodynamic and Mechanic Analyses of An Asymmetrically Scalloped Radial Turbine

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Abstract. In order to reduce the aerodynamic efficiency loss of deeply scalloped radial turbines, asymmetrically scalloping of turbines’ backdisc has been invented for a number of years. However, the mechanism how this works and its mechanical implication have not been reported in the open literature. In this paper, taking the symmetrically scalloped turbine of a turbocharger for marine generator application as the baseline, two asymmetrically scalloped turbines were developed from it: one with scalloping bias to the pressure side, and the other to the suction side of the blades. Mechanic and aerodynamic analyses of these three turbines were carried out using ANSYS Workbench. The results indicate that the asymmetrically scalloped turbine with scalloping bias to the pressure side can reduce the backdisc-heat shield cavity leakage flow going into the blade suction surface through the scalloped backdisc, due to the influence of the high pressure near the pressure surface of the blade, thus decrease the size of vortices the leakage flow generated near the suction surface inside the blade passages and associated mixing loss. Compared to the symmetrically scalloped turbine, it increases the maximum principal stress by 0.39% and 5.2% at blade fillet and backdisc respectively, but the aerodynamic efficiency of the turbine is increased by 0.78% point at a turbine U/C value of 0.43, and the smaller the value of the U/C is, the greater the efficiency advantage will be. By contrast, the turbine with suction side bias scalloping shows the poorest aerodynamic performance, due to an increased leakage flow through its scalloped backdisc.

Nomenclature

| Symbol | Description                                    |
|--------|-----------------------------------------------|
| C      | Isentropic turbine spouting velocity           |
| D      | Rotor diameter                                 |
| S      | Axial gap of the back cavity                  |
| L      | Gap between blade hub tip and housing         |
| H      | Inlet blade height                             |
| P      | Spacing or pitch of blades                    |
| U      | Blade tip speed                                |

1. Introduction

With characteristics of compact structure, low manufacturing cost, high efficiency and high structural integrity, radial turbines have been widely applied in micro gas turbines, turbochargers, auxiliary power units and other purposes requiring small flow rates. The backdisc of the radial turbine bears
very high thermal stress and mechanical stress in working, so it is usually scalloped. The mass and the moment of inertia of the turbine wheel are both greatly reduced as material is removed from high radii and so are the stresses. However, the scalloping introduces the back-cavity leakage loss, that is, a leakage flow through the scalloped backdisc, either from the upstream high pressure gas or driven by the pressure gradient between the blade pressure and suction surfaces, which results in a great loss of aerodynamic efficiency.

Scalloped turbines have been widely studied for many years. Hiett & Johnson [1] experimentally found that the reduction of the moment of inertia of their deeply scalloped radial turbine is nearly 45% and the reduction of the peak aerodynamic efficiency is between 2% and 4%. Cox et al. [2] compared a scalloped mixed-flow turbine with a conventional mixed-flow turbine and studied their flow fields, the results indicated that at high aerodynamic loadings (low U/C values) the scalloped mixed-flow turbine allows a significant leakage flow to occur and an efficiency loss of 1.3% points was predicted. The scallop also moves the peak efficiency of the turbine to a higher U/C value. He et al. [3] investigated the back cavity sealing flow in deeply scalloped radial turbines by numerical simulation, they found that the sealing flow strengthened the casing scraping flow of the rotor backdisc in the lower radial region of the back-cavity between the backdisc and the casing, cooled the root of the backdisc and prevents hot gas ingestion. With the same disk radius ratio, the model with a radial clearance showed a better performance on cooling and sealing effectiveness than that with an axial clearance. Simonyi and Boyle [4] studied the aerodynamic loss of the back-cavity flow, and found the leakage loss in the back-cavity accounts for 50% of the total clearance loss. Kidwell et al. [5] found that turbine efficiency was sensitive to back-cavity gap: for every 1% increase in the back-cavity gap, aerodynamic efficiency of their turbines decreased by 0.17%. Takao Yokoyama et al. [6] invented a asymmetrically scalloped radial turbine, they experimentally found that this structure could effectively suppress the leakage flow in the back cavity and improve turbine efficiency.

However, in the open literature, there are few relevant studies on the internal flow fields of asymmetrically scalloped radial turbines, especially the flow fields inside the back cavity and the mechanism how the asymmetric scalloping affects turbine performance. In this paper, the mechanical and aerodynamic characteristics of a symmetrically scalloped turbine and its two asymmetrically scalloped variants are studied by numerical simulations. The aim is firstly to investigate if there is any significant difference in mechanical properties between the symmetrically and asymmetrically scalloped radial turbines that may affect the reliability of the turbines; and secondly, to present how the overall performance of the turbine may be affected by asymmetric scalloping, and how the scalloping bias to the pressure surface of the blades could be better by analyzing the internal flow fields inside the back-cavity and rotor of the three turbines.

2. Research objects

A symmetrically scalloped commercial radial turbine, from a turbocharger for marine generator application, was selected for this study and used as the baseline model. The meridional flow channel of this turbine is shown in Figure 1. The material of the turbine wheel is K418, an INCO713C equivalent, and the density is 8.0g/cm³. The wheel has 13 blades with a tip diameter of 125mm, and a maximum speed of 77,920 RPM. Table 1 records the detailed structural parameters of this turbine.

There is a clearance between the turbine wheel shaft and the bearing housing sealed by a piston ring. The leakage flow through the ring is very small compared to the main flow, so the influence of the clearance leakage is ignored in this paper, and it is considered that the wheel shaft and the bearing housing are completely sealed.
3D CAD software was used to modify the backdisc of the turbine wheel into two additional asymmetrically scalloped wheels with scalloping bias to the suction surface and the pressure surface respectively, to be compared with the baseline model. **Figure 2** depicts the three scalloped radial turbine wheels. SY presents a symmetrically scalloped wheel, SS presents a wheel with scalloping bias to the suction surface and PS presents a wheel with scalloping bias to the pressure surface.

(a) Case SY: Baseline model, the symmetrically scalloped wheel.
Figure 2. Three scalloped radial turbine wheel models.

The total width of the outermost edge of backdisc left by the scalloping of the three radial turbine wheels is 44% spacing of two blades.

Taking into consideration of the space occupied by blade root fillet near blade tip, the width of the outermost edge of backdisc on the side of the suction surface in case SS and on the side of the pressure surface in case PS is designed a third spacing of two blades.

The size and shape of the scalloped areas on the backdisc of the three turbine wheels are the same, so the three wheels have the same scalloping area, volume, mass and the polar moment of inertia. The properties of the three turbine wheels are listed in Table 2.

Table 2. The properties of three scalloped radial turbines.

| Case | Volume /mm³ | Mass /kg | Scalloping area /mm² | Polar inertia /kg*m² |
|------|-------------|----------|----------------------|---------------------|
| SY   | 225,849     | 1.807    | 64,647               | 1,712               |
| PS   | 225,848     | 1.807    | 64,647               | 1,712               |
| SS   | 225,848     | 1.807    | 64,647               | 1,712               |

3. Numerical models

3.1. Static structural models

The mechanic analyses of three radial turbine wheels were carried out by Static structural analysis of ANSYS Workbench. Figure 3 presents the mesh of radial turbine wheel SY for this purpose. The rotational speed of the three turbine wheels was set at 77,920 RPM. A uniform material temperature of 295 °K was assumed, giving a Young's modulus of 210 GPa and a Poisson's ratio of 0.26.

3.2. CFD models and verification of grid independence

3.2.1. CFD models. In this study, the aerodynamic performance of the three turbine wheels was computed by a three-dimensional steady-state CFD simulation which solved Reynolds averaged Navier-Stokes equations with SST turbulence model using Ansys CFX™. The computational model of
the three turbines included turbine housing channel, full 360° channel of turbine rotor and the domain of the backdisc-heat shield cavity (back cavity). Frozen-rotor interface method was employed at the rotor-housing interface. Total temperature (600 °C), total pressure and flow directions at the inlet and static pressure (1 bar) at the outlet of the model were imposed. The solid walls including rotor blades, casing shroud, volute wall and heat shield were all set as adiabatic, impermeable and non-slipping. Due to the complexity of the model, spatial discretization was performed using an unstructured mesh. Regions close to wall surfaces had 10 refined layers of boundary-layer type grids. The computation domain and the mesh are presented in Figure 4.

![Figure 3. The mesh of wheel SY for static structural analysis.](image)

![Figure 4. CFD mesh of radial turbine SY.](image)
The Frozen-rotor interface method will produce different results depending on the relative position of the rotor to the housing tongue, so the same relative position of the wheel blades to the tongue was used for all the three turbines as Figure 5 shown.

![Figure 5](image)

**Figure 5.** Relative position of the rotor to the housing tongue in the three turbines.

### 3.2.2. Verification of grid independence.

Grid independence of the computational model including turbine housing channel and turbine rotor channel was first verified. Table 3 Compares the overall performance of five groups of grids of the model with different grid densities. The total number of grids ranges between 10.53 to 18.59millions.

The table shows that when the housing channel grid number exceeds 2.91millions and the rotor channel grid number exceeds 12.44millions, the turbine efficiency is less affected by the grid density of the housing and the rotor channels. For computational efficiency and accuracy purposes, the housing channel grid number and the rotor channel grid number in the subsequent models were controlled slightly higher than 2.91millions and 12.44millions respectively and the total number of the grid points was higher than 15.35millions.

The y+ values of rotor solid surface of three turbines are shown in Figure 6, which are less than 3.

![Figure 6](image)

**Figure 6.** y+ values of three turbine rotors’ solid surface.
Table 3. Verification of grid independence.

| Grid group id | Housing channel grid number (millions) | Rotor channel grid number (millions) | Turbine efficiency  |
|---------------|----------------------------------------|---------------------------------------|---------------------|
| 1             | 2.91                                   | 7.62                                  | 68.71%              |
| 2             | 5.08                                   | 7.62                                  | 68.70%              |
| 3             | 2.91                                   | 9.83                                  | 68.51%              |
| 4             | 2.91                                   | 12.44                                 | 68.43%              |
| 5             | 2.91                                   | 15.68                                 | 68.42%              |

4. Results and Analysis

4.1. Mechanic analyses

Turbine wheels bear very large mechanical stresses at the working condition of high rotational speeds. Due to their structural difference, the mechanical stresses of these three scalloped radial turbine wheels will be different. The simulation results of the maximum principal stresses of the three scalloped wheels are shown in Figure 7, and the stress details of the scalloping area are shown in Figure 8.

These figures show that the overall stress distribution of the three turbine wheels are generally similar, with high stresses mainly concentrated in the blade root fillets, and the maximum values of the first principal stress of the three turbine wheels all occurs at the filet. The stress level at the fillets and the scalloped areas is affected by the change of the location of the scalloping, there are two symmetrically distributed areas of high stress in the scalloped hub in case SY, Figure 8(a), but there is only one high stress area close to the pressure side of blades in case PS, Figure 8(b) and to the suction side of blades in case SS, Figure 8(c). The maximum values of the first principal stress of three turbine wheels are recorded in Table 4.

Figure 7. The first principal stress of three turbine wheels at 77,920 rpm.
Figure 8. The first principal stress in the scalloped region at 77,920 rpm.

Table 4. The maximum values of the first principal stress with different scallopings.

| Cases                                 | SY (MPa) | PS (MPa)    | SS (MPa)    |
|---------------------------------------|----------|-------------|-------------|
| Overall maximum values (in the trailing edge of the blades) | 773 (0)  | 776 (+0.39%) | 771 (-0.26%) |
| Maximum values in scalloped hub        | 556 (0)  | 585 (+5.22%) | 592 (+6.47%) |

Compare to case SY, the overall maximum value of the first principal stress just increases 0.39% in case PS and decreases 0.26% in case SS, so it is little affected by the circumferential location of the scalloping. However, the difference of the first principal stresses of the hubs is quite large: the maximum value increases about 5.2% in case PS and 6.5% in case SS. In these particular turbines, the values are all within the acceptable stress limit for the hub. For turbines operate at higher speed, higher temperature or more stringent life requirements, however, the increased stress in the scalloping region may cause problem.

Figure 9 records the first 20 natural frequencies of the three turbines. It can be confirmed that there are no significant differences in the natural frequencies of the turbine wheels.

4.2. Aerodynamic analyses

4.2.1. CFD results. Figure 10 details the CFD predictions of isentropic efficiency plotted against isentropic spout velocity ratio U/C for the three turbines. It is evident from the Figure that the efficiency of case PS is the highest, and that of case SS is the lowest while operating at U/C values lower than 0.7. The efficiency advantage of case PS increases as the velocity ratio decreases: when the velocity ratio is 0.68, the isentropic efficiency of case PS is only 0.11% higher than case SY and 0.28% higher than case SS, but the advantage increases to 0.78% and 0.92% respectively when the velocity ratio is 0.43. However, the difference in efficiency between case SY and case SS is not very large, the maximum difference is only 0.26%.
4.2.2. Back-cavity flow fields analyses. The observed difference in turbine efficiency is caused by the difference in backdisc structure which affects the flow fields in the back-cavity as well as in the wheels. So, it is necessary to study the flows fields in order to understand the flow mechanisms that contribute to the changes in aerodynamic performance of the turbines.

The back-cavity flow field of the baseline turbine is presented in Figure 11. The gas flows leak into the back cavity through the gap between the hub tip and the volute as well as the backdisc.
openings (the scalloped areas), and they flow counterclockwise relative to the turbine backdisc viewed from behind the backdisc. A small amount leakage flow finally enters into the blade passages through the backdisc openings away from the tongue of the volute, top-right insert, while the main leakage flow leaves the back-cavity through the openings near the tongue and the gap between the blade hub tip and the volute, right insert. The high pressure from the pressure side of the blades forces a part of the circulating gas to change their direction forming a vortex near the opening at every blade. A large, low-velocity vortex is formed near the rotating axis of the rotor in the back-cavity.

Figure 11. Back-cavity relative flow field of the baseline turbine (SY) in a plane at the middle of the back-cavity at operating condition of U/C = 0.43.

Figure 12 compares the streamlines and entropy distribution of the three turbines captured on a plane at 50% back-cavity height. The plots are taken at a velocity ratio of 0.43. There are some differences between the three cases, and some interesting comparable features:

- At the scalloped openings, especially near the tongue of the volute which locates at 6 o’clock position in the figure, the velocity of the leakage flow is very high. The average velocity of the leakage flow on this plane in case PS is significantly lower than those in case SY and case SS.
- The leakage flow forms strong leakage vortexes at the scallop openings and flows into the blade passages in case SY and case SS. However, the vortices are much smaller in case PS and the main leakage flow moves out of the back cavity directly through the gap between the blade hub tip and the volute. This point will become clearer later on.
- The location of the low-velocity vortex near the shaft center in the three turbines is basically the same but the sizes of them are different. The vortex in case PS is slightly larger than the vortex in case SY and case SS.
- The entropy is closely related to the vortices because they represent a loss. The entropy is very high near the openings where strong vortices are generated.
Figure 12. Relative streamlines (left) and entropy (right) at 50% back-cavity height, U/C = 0.43. Housing tongue is located at 6 o’clock position.
The main driving force of the back-cavity flows is the pressure gradient between the housing discharge and the back-cavity. Figure 13 shows the static pressure at the back cavity of the three turbines. It can be seen that Case PS has the highest pressure at upper back-cavity. This reduces the leakage flow to the cavity in this case compared with the other two cases, as displayed in Figure 12. The lowest pressure area corresponds to the tongue position. The pressure gradient in the circumferential direction generated by the volute drives the main back-cavity flow to this region, Figures 12.

Figure 13. Pressure at 50% back-cavity height, U/C = 0.43. Housing tongue at 6 o’clock position. In Case PS, the pressure at upper back cavity is higher and more uniform than the other two cases.

Figure 14 shows the inducer surface pressure of the three turbines. The exposed blade loading is in clockwise direction in Figures 11 and 12, which is opposite to the relative flow direction at the upper back-cavity, this creates the vortices at the openings and produces losses. The figure also shows that because blade suction surface is less disposed by the scalloping in Case PS, the pressure in the hub opening in this case is more uniform than the other two cases. This transfers to a more uniform pressure at the upper back-cavity for Case PS as shown in Figure 13. As the pressure on the suction surface of blades is generally lower than the pressure in the upper back-cavity, some gases in the back-
cavity flow out of the cavity through the openings and enter blade passages as depicted in the inserts of Figure 11. The amount of this flow depends on the pressure at the opening, so Case PS will have smaller, and Case SS larger amounts of the leakage flow than Case SY. Under the cross-passage pressure gradient of blade loading, this leakage to the blade passages always moves to the blade suction surface. When U/C decreases, the pressure gradient increases, and the difference in the leakage flow will be more noticeable between the three scallopings.

4.2.3. Rotor blade passage flow structure analyses. In scalloped radial turbines, the back-cavity leakage flow will mix with the mainstream flow and the tip clearance flow, and form a strong passage vortex near the suction surface in blade passages. Figure 15 depicts a portion of the passage vortices formed by the back-cavity leakage flow for the three turbines. The figure shows that the back-cavity leakage flows form a vortex in each blade passage having been sucked toward blade suction surface by the passage pressure gradient and stopped by the blade in tangential direction. In Case SS, the strong leakage flow leads to strong vortices with high swirl velocity. In the baseline, Case SY, a large amount of the leakage still flows into the blade passages through the scalloped backdisc openings, and only a small amount of the leakage flow comes out of the cavity and reenters blade passages around the hub tip. However, in Case PS, only a small portion of the leakage flow enters the blade passages through the openings, thus the formation of the passage vortices is greatly inhibited. As blade loading increases when U/C reduces, Case PS becomes more efficient while Case SS less so than Case SY with the reduction of U/C.

![Figure 15. Comparison of the passage vortices formed by the back-cavity leakage flow, U/C = 0.43.](image)

5. Conclusions

Three symmetrically and asymmetrically scalloped radial turbines were compared from the mechanic and aerodynamic aspects respectively.

The results of mechanical analysis by FEA show that the stress distributions of the three turbines at the maximum speed are generally the similar. The maximum value of the first principal stress, which occurs at blade trailing edge fillet, of the asymmetrically scalloped turbine with scalloping bias to the blade pressure surface is only 0.39% higher than that of the symmetrically scalloped radial turbine. The maximum first principal stress in the scalloping area with the pressure-surface bias scalloping is about 5.2% higher than that of the symmetrically scalloped baseline, but the stress is within the acceptable level. For the turbine with scalloping bias to blade suction surface, the two values are 0.26% smaller and 6.47% higher respectively.

By running CFD simulations of the three turbines across a range of U/C values, it is found that at low U/Cs, the turbine with scalloping bias to the pressure surface has better efficiencies than the
baseline, the advantage increases as the value of U/C decreases and it reaches 0.78% at a velocity ratio value of 0.43. The turbine with scalloping bias to the suction surface has slightly poorer efficiency than the baseline.

The variation of the turbine wheels’ backdisc structure results in the different flow patterns inside the back-cavity and interactions between the back-cavity flow and the main passage flow. The higher pressure near the pressure surface of the blade passages can inhibit the leakage flow in the turbine back-cavity and prevent the flow from leaking into the blade passages. It weakens the vortices near the scalloped openings of the backdisc in the back-cavity, and allows the unscalloped backdisc regions to cut off the strong connection between the leakage flow in back-cavity and the flow in blade passages thus reducing the mixing loss inside blade passages. Therefore, the radial turbine with scalloping bias to the pressure side has the best aerodynamic efficiency.

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