Effects of Misalignment of Turbine Wheel Hub with Housing

Yu Wang¹, Hua Chen¹*, Chao Ma¹ and Jinhui Sun²

¹ Dalian Maritime University, 1 Linghai Road, Dalian, China 116026
* Corresponding author, E-mail: huachen204887@163.com
² Kangyue Science & Technology Company Ltd., Shandong, China

Abstract. Alignment of a turbocharger turbine rotor hub with its housing can affect aerodynamic performance of the turbine, and studies may have been done in industry to quantify this effect. However, there is no related publication in open literature and so the effect and the underline mechanism remain trade secrets. A numerical investigation into the effect and the mechanism was carried by current authors using two turbine wheels with identical flow passage but different backdisc configurations: one with a full backdisc and the other a deeply scalloped backdisc. For each configuration, five alignments of the wheel hub with housing were studied, these include a full alignment of the rotor hub and the housing and four misalignments of the two turbine components. Effects of the alignment and turbine operating condition on aerodynamic performance of the turbines were studied. The flow field of the turbines was interrogated, and the underline mechanism that produced the performance changes was investigated. The study yields some interesting results: For the full backdisc configuration, the exposure of the backdisc and the wheel recess from housing both decrease turbine efficiency. It has been found that the exposure of the backdisc in the housing outflow creates a stagnation point at the tip of the backdisc, generating some loss. The stagnation then forces the incoming flow to move away from the hub and toward the shroud side of the wheel, producing further losses within the wheel passage. The high pressure at the tip of the backdisc also reduces the flow entering the backdisc cavity between the wheel and heat shield, thus decreases the loss in the cavity. However, this loss is relatively small compared with the loss in the main flow passage. It has also found that the misalignment has greater negative effects on aerodynamic performance of the turbine with deeply scalloped backdisc, and a stronger leakage flow from the backdisc cavity into rotor passage occurs due to the opening in the backdisc by the scalloping. Turbine operating condition has an effect too: lower U/C condition shows higher impacts of misalignment than higher U/C condition. These results are reported and analysed.

Nomenclature

C Isentropic turbine spout velocity
CFD Computational Fluid Dynamics
D Rotor diameter
S Axial gap of the back cavity
L Radial gap between blade hub tip and housing
H Inlet height of scalloped backdisc turbines
K Inlet height of full backdisc turbines
U Blade tip speed
backdisc thickness at maximum radius

1. Introduction

Radial turbines have been widely applied in micro gas turbines, turbochargers, auxiliary power units and various other purposes with characteristics of low manufacturing cost, compact size, high structural strength and high efficiency under small flow rates. Radial turbines with scalloped backdisc and full backdisc are both used in industry due to their different advantages in mechanical and aerodynamic performances. Compared with scalloped radial turbines, full backdisc radial turbines bear higher thermal and mechanical stresses at hot and high speed operating conditions but their aerodynamic efficiency is higher.

There is a large amount of literature on the internal flow mechanism and structure optimization of scalloped and full backdisc radial turbines. Hiett and 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%. Kazemi et al. [2] studied three-dimensional flow in radial-inflow turbines, they recommended that any meaningful numerical simulation of radial-inflow turbines must include some sort of modelling of the tip leakage flow by comparing the model with tip clearance to the model without tip clearance. Cox et al. [3] compared a scalloped mixed-flow turbine with a conventional (full backdisc) 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. Barr and Spence [4] did a comparative numerical analysis between a normal radial rotor and a 25° back swept rotor, results show that the 25° back swept blade offers significant increases in efficiency while operating at lower than optimum velocity ratios (U/C). Chen et al. [5] deliberated eight structural optimization measures for turbine rotors to shift the peak efficiency toward a lower U/C value than 0.71, including one that uses full backdisc. Chen and Pesiridis [6] discussed the influences of pulsating inflow conditions on turbine performance through a similarity analysis, the analysis shows that Strouhal number should be scaled with the square root of turbine inlet total temperature. Rodgers [7] discussed the effects of rotor blade solidity on turbine efficiency based upon past research, and supported by analysis of a one-dimensional friction model of the rotor passage. Present authors [8] recently discussed the benefit of asymmetric scalloping of radial turbines and investigated the related flow mechanisms.

The alignment of a radial turbine rotor and its housing has some effects on turbine’s performance as one would expect. For various reasons including manufacturing tolerances, the rotor hub and the housing may misalign. A question rises that if a misalignment is unavoidable, what is the best configuration: should the wheel backdisc recess from or expose to the housing flow passage? Some studies on the alignment may have been done in industry, but in the open literature related issues have not been discussed, and the question remains unanswered. To answer this question, in this paper, we take a scalloped and a full backdisc radial turbine rotors as the research objects, shown in Figure 1, analyse the influence of alignment of the rotors and their housings on aerodynamic efficiency and flow fields of the turbines by numerical simulations. Rest of the paper is organised as follows: first the research objects of the turbines and different alignments of the rotors and housings are described; then the CFD method employed in the study are outlined and defined; this is followed by the results of CFD and an investigation into the reasons behind the performance differences caused by the alignments; and finally findings are summarised and conclusions are drawn.
Figure 1. Scalloped and full backdisc radial turbine rotors.

2. Research objects

Figure 2 presents the passages of the scalloped backdisc turbine (left), and the full backdisc turbine (right) both with their rotor aligned with the housing as well as the nomenclature used. It should be noted that the alignment of the rotor with its housing is subtly different between the two turbines as shown in the details of Figure 2: in the scalloped backdisc turbine, the backdisc surface is aligned with the surface of the housing flow passage; but in full backdisc turbine, the blade tip hub is aligned with the surface of the housing flow passage because of the backdisc thickness T at diameter D. Despite this small difference, in both turbines, however, the flow passages of the rotor and the housing are fully aligned at the hub side.

Figure 2. Passages of the two studied turbines, showing full alignment of rotors and housings.

The two rotors each have 13 blades with a tip diameter of 125 mm. Table 1 lists the detailed structural parameters and the two operating conditions studied. The full backdisc turbine rotor is constructed from the scalloped backdisc turbine rotor by removing the scalloping and align the rotor with the housing by changing the housing flow passage inlet height, so the difference between the inlet
height of the scalloped backdisc turbine H and the inlet height of the full backdisc turbine K is the backdisc thickness T at the maximum diameter of the rotors D.

There is a clearance between the turbine wheel shaft and the bearing housing. The clearance leakage flow 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.

Table 1. Structural parameters & operating conditions of two turbines.

| Parameters (units)                                      | Values             |
|--------------------------------------------------------|--------------------|
| Tip diameter D (mm)                                    | 125                |
| Height of back cavity S (mm)                           | 1.15               |
| Gap of hub tip L (mm)                                  | 1.25               |
| Inlet height of scalloped backdisc turbines H (mm)     | 19.55              |
| Inlet height of full backdisc turbines K (mm)          | 18.3               |
| Backdisc thickness at maximum radius T (mm)            | 1.25               |
| Polar inertia of the scalloped backdisc turbine (kg*m²)| 1613               |
| Polar inertia of the full backdisc turbine (kg*m²)     | 1765               |
| Operating condition 1                                  | 50% speed & U/C = 0.50 |
| Operating condition 2                                  | 75% speed & U/C = 0.71 |

(a) Scalloped backdisc turbine                       (b) Full backdisc turbine

Figure 3. Five backdisc configurations & alignments of the turbines.

For the convenience of comparison and discussion, the two turbines with rotor-housing alignments shown in Figure 2 are chosen as baselines. By changing the housing flow passage inlet height of scalloped backdisc turbines H and the inlet height of full backdisc turbines K, while keeping the absolute position of the two turbine rotors as well as the heights of the back cavity, different alignments between the rotors and the housings were created. Figure 3 details five configurations of
the two turbines that give five different alignments. The numbers represent different inlet heights of the housing flow passage and different alignments:

- The rotor is aligned with the housing when the housing flow passage inlet height is at position 3 in the scalloped backdisc turbine and position 4 in the full backdisc turbine.
- In the scalloped backdisc turbine, alignments 1 and 2 indicate that the rotor backdisc are exposed to the incoming flow from the housing; While in alignments 4 and 5, the rotor blades are recessed into the housing/heat shield.
- In the full backdisc turbine, positions 1, 2 and 3 indicate that the rotor backdisc are exposed to the incoming flow while it is recessed into the housing/heat shield at position 5.
- It should be noted that at alignment 1, the housing/heat shield has no step and the gas can flow into the backdisc-housing/heat shield clearance cavity directly.

The inlet height and the misalignments between the rotor and housing of the two turbines are recorded in Table 2 and Table 3 respectively.

**Table 2. Inlet height H and misalignments of scalloped backdisc turbine.**

| Configurations | 1     | 2     | 3     | 4     | 5     |
|----------------|-------|-------|-------|-------|-------|
| Misalignments  | +1.15mm | +0.45mm | 0     | -0.45mm | -1.25mm |
| Inlet height H | 20.70mm | 20.00mm | 19.55mm | 19.10mm | 18.30mm |
| Misalignments/H| +5.56%  | +2.25%  | 0     | -2.36%  | -6.83%  |

**Table 3. Inlet height K and misalignments of full backdisc turbine.**

| Configurations | 1     | 2     | 3     | 4     | 5     |
|----------------|-------|-------|-------|-------|-------|
| Misalignments  | +2.40mm | +1.70mm | +1.25mm | 0mm   | -1.25mm |
| Inlet height K | 20.70mm | 20.00mm | 19.55mm | 18.30mm | 17.05mm |
| Misalignments/H| +11.6% | +8.50%  | +6.39%  | 0     | -7.33%  |

3. CFD models

In this study, the aerodynamic performance of the five different configurations of the two turbines was computed by a three-dimensional steady-state CFD simulation. Commercial software CFX™ was employed to solve Reynolds averaged Navier-Stokes equations with SST turbulence closure. The computational model of the two turbines included a turbine housing passage and a full 360° turbine rotor passage including an outlet duct. Frozen-rotor interface method was employed at the rotor-housing interface. Total temperature, total pressure and flow directions at the inlet and static pressure at the outlet of the model were imposed. The solid walls including rotor blades, casing shroud, housing 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 meshes in order to more accurately capture endwall boundary-layers.

The housing grid number is 3.1millions and the rotor passage grid number is 12.7millions, giving 15.8millions total number of the model grids. A grid independence study was carried by increasing this number by 25% but the efficiency error was found only 0.02%. The y+ values of rotor solid
surface of two turbines are shown in Figure 4, which are less than 3 and the computational model is presented in Figure 5.

![Figure 4. y+ values of two turbine rotors’ solid surface.](image)

Figure 4. y+ values of two turbine rotors’ solid surface.

![Figure 5. Computational model of turbines.](image)

Figure 5. Computational model of turbines.

4. Results and Analysis

4.1. CFD results. Table 4 presents the mass flow rate of the two turbines at the five different configurations and two U/C values. The maximum difference in the rate between the configurations is only 0.48% with the full backdisc turbine when U/C value is 0.5, and is smaller at all other cases. It confirms that the difference in inlet height H and K has little impact on the mass flow of the turbines.

| Turbines       | Configurations | 1    | 2    | 3    | 4    | 5    |
|----------------|----------------|------|------|------|------|------|
| Scalloped      | U/C=0.50       | 0.5834 | 0.5833 | 0.5822 | 0.5819 | 0.5809 |
|                | U/C=0.71       | 0.6195 | 0.6194 | 0.6187 | 0.6186 | 0.6184 |
| Full backdisc   | U/C=0.50       | 0.5837 | 0.5849 | 0.5853 | 0.5865 | 0.5849 |
|                | U/C=0.71       | 0.6197 | 0.6199 | 0.6203 | 0.6192 | 0.6185 |
Figure 6 shows the predicted isentropic efficiency plotted against the different alignment configurations at U/C values 0.5 and 0.71 in scalloped backdisc turbine. It shows clearly that the aerodynamic performance of scalloped backdisc turbines is the best when the rotor backdisc is fully exposed (configuration 1, Figure 3), and it is the worst when the rotor is deeply recessed into the housing/heat shield (configuration 5). The aerodynamic efficiency decreases continuously with the rotor blades recessing into the housing/heat shield and the trend is consistent at both U/C values of 0.5 and 0.71. The variation is more significant when the U/C value is 0.5 than when it is 0.71: the difference in efficiency between the highest and lowest points is 0.5% at the U/C value of 0.71 while it is 1.02% at the U/C value of 0.5. The results are somewhat surprising as one expected that fully aligned configuration 3 would be the most efficient, but it is about 0.3 ~ 0.4% points lower than configuration 1 depending on the value of U/C.

![Figure 6](image1.png)

Figure 6. Predicted efficiency of scalloped backdisc turbines under two U/Cs, Config. 3 is full alignment.

![Figure 7](image2.png)

Figure 7. Predicted efficiency of full backdisc turbines under two U/Cs, Config. 4 is full alignment.
The situation is somewhat different in the full backdisc turbines, CFD results of them are presented in Figure 7. It shows that the alignment of the rotor with the housing, configuration 4, is the most efficient. Similar to the scalloped backdisc turbines, the difference between the highest and lowest efficiencies is also dependent on U/C value: it is 0.61% at the U/C value of 0.5 while it is 0.36% at the U/C value of 0.71. These differences are smaller than those observed in the scalloped turbines.

4.2. Internal flow fields analyses
To understand the reasons behind these performance differences caused by rotor-housing alignment, internal flow field of two turbines were analysed. Some 2D views of the flow field were generated, the cutting plane through the turbines and the view direction of these plots are given in Figure 8.

![Figure 8. The position of 2D flow field plots and their view direction.](image)

Figures 9 and 10 display, at the cutting plane, the streamlines and static entropy distribution of rotor passages of the scalloped backdisc turbines in the five alignments at U/C value of 0.5, respectively.
Figure 9. Relative streamlines in scalloped turbine with different alignments, U/C = 0.5.

Figure 10. Specific entropy of scalloped turbine with different alignments, U/C = 0.5.

There are some interesting points worth discussing from the CFD results of these figures:

- The incoming flow enters into rotor blade passages and separates at the step in the housing (Figure 9, Alignments 2-5). The strength of the separation increases with the height of the step. This separation reduces the flow velocity in the hub region of the rotor inducer and forces flow turning towards shroud. Majority of the gas travel downstream inside rotor passages while a small amount of the gas leaks into the back cavity.

- When there is no step, the incoming gas can leak more easily into the rotor back cavity and generate entropy in the cavity. As the step increases, the entropy in the cavity decreases. Vortices on the other hand, is generated when the step is large (Figure 9, configurations 4 and 5).
5). As a result, the entropy after the step increases with the recess (Figure 10). The divergence of the flow towards the shroud side of the rotor increases entropy in the hub region of the rotor. Because the entropy in the cavity is small compared with the loss right after the step and the loss in the rotor passage, Figures 9 and 10 thus suggest that the flow separation after the step and associated divergence of the flow toward shroud is the main flow mechanism for efficiency reduction caused by the misalignment of the housing and the rotor.

- It is important to have a smooth transition of flow from housing to rotor along the hub.

**Figure 11** shows the blade passage flow formed by the back-cavity leakage in the scalloped turbine. The amount of leakage flow is considerable in the three rotor passages under the housing tongue at Alignment 1, it decreases as the step rises but the difference is not significant.

![Figure 11. Rotor passage flow formed by the back-cavity leakage flow in scalloped turbine, U/C = 0.5. Housing tongue location indicated by the arrow.](image)

Although significant, the changes in the leakage flow rate in the back cavity still affect the back-cavity loss which is further shown in **Figure 12**. The figure clearly shows that the back-cavity leakage loss is reduced as the step rises. All the information combined, one can conclude that the effect of rotor passage loss on efficiency is more important than that of back-cavity leakage loss.

**Figures 13** and **14** present the streamlines and static entropy distribution in the rotor passages of the *full backdisc* turbines, **Figure 15** shows the passage flow formed by the back-cavity leakage flow in the same turbines. The plots were taken at a velocity ratio of 0.5.
Figure 12. Specific entropy at 50% back-cavity height, $U/C = 0.5$ and scalloped turbine.
The situation in the full backdisc turbines is similar to that in scalloped backdisc turbines but there are also some differences in the details.

- When the rotor backdisc is exposed to the incoming flow, (Figure 12, Configurations 1 to 3), the flow hits the tip of the backdisc and stagnates. The high pressure generated then forces the flow away from the hub, increasing the flow velocity at higher spans. The stagnation and the movement of flow away from the hub generate entropy near leading edge of the hub as shown in Figure 13. In configuration 1, housing exit flow is able to get into the backdisc cavity freely,
generating the highest entropy in the cavity. In configuration 4, the alignment of the rotor backdisc with the housing passage enables a smooth transfer of the flow in the hub region from the housing into the rotor, reducing the loss along hub region. When the rotor backdisc is recessed into the housing, Configuration 5, a step above the backdisc is created in the hub flow passage. The sudden expansion of the flow over this step causes a significant velocity loss of the gas flow and incorrect incidence to the rotor, generating high entropy near the hub tip and leading to further losses in the rotor passage.

- The static entropy inside the back cavity in the full backdisc turbines is, however, higher than that in the scalloped turbines because of the different backdisc structures and it seems to decrease with the recess. In the scalloped turbines, the flow going to the back cavity from upstream is affected by the adjacent blades whose rotation reduces the total pressure and total temperature of the gas into the cavity, thus decreases the entropy in the cavity. However, the major turbine loss is the rotor passage loss, and by maintaining a better hub flow, the full backdisc turbines all achieved better efficiency than the best scalloped turbines at U/C value of 0.5.

![Figure 15. Rotor passage flow formed by the back-cavity leakage flow in full backdisc turbine, U/C = 0.5. Housing tongue location indicated by the arrow.](image-url)

Figure 15 shows that the leakage flow decreases with the recess just like in the scalloped turbines. However, the reduction in the full backdisc turbines is more significant than that in scalloped turbines, as now the leakage gas can only flow in and out of the back cavity through the gap between rotor hub and the housing, while in the scalloped turbines scalloping creates additional passages for the leakage gas. Comparing Figure 11 and Figure 15 also indicates that the amount of leakage flow from the back cavity in the full backdisc turbines is much smaller than that in the scalloped backdisc turbines, this has no doubt contributed to the higher efficiency of the full backdisc turbines (Figures 6 and 7).
In full backdisc turbines, the hub tip and the housing are close together with a gap (L) between them of 1.25 mm only, so the loss near the housing step (Cases 2 to 5) is small and the influence of rotor passage loss and back-cavity leakage loss on turbine efficiency dominates.

5. Conclusions

A turbine with both scalloped backdisc and full backdisc was numerically investigated for the effects of rotor-housing/heat shield alignment on turbine performance. Five different alignments, including extruded backdisc, fully aligned backdisc and recessed backdisc, were created for the both full and scalloped backdiscs by modifying the housing/heat shield. CFD was performed with all the ten configurations at two U/C operating conditions. Following conclusions can be drawn:

Turbine efficiency is the highest when there is no step in the flow passage between the housing and the rotor in hub region. In the scalloped backdisc turbines, this means that the entire backdisc is exposed in the incoming flow from the housing; and in the full backdisc turbines, this implies a full alignment of the rotor and the housing flow passages. The alignment has a less effect on turbine mass flow.

The penalty of the misalignment of the rotor and the housing is severe with the scalloped backdisc. The maximum efficiency reduction is predicted to be 1.0% at U/C value of 0.5 and 0.50% at U/C value of 0.71. The step in the housing/heat shield reduces the leakage flow to the back cavity, but can cause flow to separate behind the step and generate vortices, which produces high aerodynamic loss at the inlet hub region of the rotor and forces the flow in the region to migrate towards shroud further augments the loss in the hub passage of the rotor. The losses increase with the size of the step.

The leakage flow from the back cavity into rotor passages is significant in the passages under the housing tongue and it mixes with the main flow in the passages generating losses. The leakage is not greatly affected by the misalignment. The loss inside the cavity itself is lower than that in the full backdisc cases because of the pre-expansion of the gases when they first coming to the cavity from upstream.

In the full backdisc turbines, the difference in turbine efficiency is not as significant as in the scalloped backdisc turbines: the maximum difference in turbine efficiency between the five alignments is 0.60% at U/C value of 0.5 and 0.36% at U/C value of 0.71. The influence of the misalignment on turbine flow field is attenuated due to a weaker leakage flow. Without the scalloping, the flow can only squeeze through the small gap at the rotor tip into the back-cavity and from the cavity into the rotor. When the backdisc is exposed to the incoming flow, the strong flow stagnation at the tip of the backdisc generates some loss and contaminates main passage flow. Alignment of the backdisc with the housing passage eliminates the stagnation, thus reduces losses. When the backdisc is recessed from the housing, similar to the effect of a step in the scalloped backdisc turbines, there is a velocity loss happening after the step near the hub region of the rotor.

Acknowledgement

The turbines studied were provided by Kangyue Science & Technology Co. Ltd., the authors are grateful for the permission to publish this work granted by the company.

References

[1] G F Hiet and I H Johnson 1963 Experiments concerning the aerodynamic performance in inward radial flow turbines. Proceedings of the Institution of Mechanical Engineers 178(31(ii)) 28-42.

[2] M Zangeneh-Kazemi, W N Dawes and W R Hawthorne 1988 Three dimensional flow in radial-
inflow turbines. *The American Society of Mechanical Engineers 345E. 47St., New York, N.Y. 10017, 88-GT-103.

[3] Graham Cox, Jason Wu and Ben Finnigan 2007 A study on the flow around the scallops of a mixed-flow turbine and its effect on efficiency. *ASME Turbo Expo 2007: Power for Land, Sea and Air, May 14-17, 2007, Montreal, Canada*, GT2007-27330.

[4] Liam Barr and Stephen W T Spence 2008 Improved performance of a radial turbine through the implementation of back swept blading. *Proceeding of ASME Turbo Expo 2008: Power for Land, Sea and Air, June 9-13, 2008, Berlin, Germany*, GT2008-50064.

[5] Hua Chen et al. 2019 Turbocharger turbine rotor design for low U/C values. *Proceeding of ASME Turbo Expo 2019: Turbomachinery Technical Conference and Exposition, June 17-21, 2019, Phoenix, Arizona, USA*, GT2019-90070.

[6] Hua Chen and Apostolos Pesiridis 2016 Turbine performance under pulsating inflow - a similarity analysis. *Proceeding of ASME Turbo Expo 2016: Turbomachinery Technical Conference and Exposition, June 13-17, 2016, Seoul, South Korea*, GT2016-56612.

[7] C Rodgers 2000 Radial turbines - blade number and reaction effects. *Proceedings of ASME TURBOEXPO 2000 May 8-11, 2000, Munich Germany*, 2000-GT-456.

[8] Y Wang, H Chen, C Ma and J H Sun 2019 Aerodynamic and mechanic analyses of an asymmetrically scalloped radial turbine, Submitted to *ISROMAC 18, 18TH International Symposium on Transport Phenomena and Dynamics of Rotating Machinery, April 19 – 23, 2020, Honolulu, Hawaii, USA.*