Effect of Twisted Vanes on Leakage Losses in Variable Geometry Radial Turbines.

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Abstract
The effects of blade twisting of the stator of a variable geometry radial turbine are presented in this paper. They have been determined using numerical simulations. As well as affecting the main flow, the variations in incidence, associated with blade-loading, had an impact on leakage vortices. The flow in the stator and the rotor and its potential to reduce losses were characterised. In the fixed row, an open stator position leading to the leakage flow passing through to the pressure side and returning to the suction side further downstream accounted for less losses. In the rotor, a configuration with a closed stator angle at the shroud gave good results, due to a greater acceleration in the meridional direction which led to more appropriate relative incidence values.

Keywords
radial turbine — stator twisting — variable vane — leakage flow

INTRODUCTION
Radial turbines have remarkable properties such as compactness and the ability to reach high pressure-ratios for acceptable efficiencies. The behaviour of radial turbines in steady conditions has been widely presented in the literature [1] [2], together with an extensive description of the flow pattern, even at off-design conditions [3]. Some design processes have also been presented to insure the good performance of the stage in its energy recovering process [4] [5]. The expansion of the flow through the turbine stage stabilizes the boundary layers, and makes separation unlikely if the guidance of the flow is appropriate. In specific applications, for which it is necessary to enlarge the operating range, a variable geometry stator can be implemented. It improves the performance in severe off-design situations. Different solutions exist, such as varying the height or the opening position of the stator blades [6], but these methods necessarily introduce secondary flows. For example, the strategy consisting of changing the stator vane angle is nowadays well known, and widely applied to automotive applications for turbocharging. However, it induces specific additional losses due to hub and/or shroud tip clearance flows. Different consequences have been observed in the literature amongst which a distortion in the free space which locally modifies the incidence of the flow on the rotor blades [7] [8] [9]. Some interactions between stator leakage flows and rotor secondary flows have also been observed. In addition, the intensity of the secondary flows, and consequently the losses, strongly depend on the opening configuration of the stator, since it alters the loading of the vane [10] [11].

An option to reduce the leakage flow of the stator blade is to modify the blade geometry. Changing the stator vane angle at the hub and/or shroud (twisted vanes) may locally reduce loading and decrease leakage flow effects at this location as a consequence. With the knowledge that equal stator reduced sections lead to unchanged performance [6], a preliminary investigation was carried out on configurations with no stator clearance to understand the effects of a distorted flow angle at the impeller inlet on the stage performance [12]. It showed that twisting the stator blades doesn’t deteriorate the flow, especially in terms of rotor inlet incidence. In addition, it was found that local flow characteristics of configurations with stator blade twisting remained much closer to the straight case whose stator opening was of the average twist angle rather than the local angle.

The aim of the present study is to identify and understand the mechanisms related to twisted geometries with stator clearance in order to comprehend their capacity to enhance performance, especially regarding leakage losses. Moreover, it is to verify twisting doesn’t induce losses at the nominal point, so it can be investigated to improve off-design performance.

This paper is organized as follows. In the first part, the test case and the numerical procedure are described. Subsequently, investigations of the flow are presented, first in the stator and then in the rotor. Some prospects are presented, before the concluding remarks.
1. TEST CASES

Table 1 gives the main specifications of the stage under investigation. It is composed of a stator and a rotor, ignoring the volute in order to reduce the calculation domain to a single blade passage. Stator clearance of 4% of the blade height is applied to the hub and shroud, and geometries with no stator gap are also considered. The extraction planes used to compute performances and cross-stream colour contours are presented in figure 1.

As mentioned previously, this research uses cases with different spanwise stator angles. Indeed, ±10° blade twisting was applied at the hub and shroud. This was done lineally so that the average stator angle remained fixed at 57° from the radial direction. An angle of 47° then corresponds to the most open and 67° to the most closed position, as presented in figure 2 (a). The straight case is also investigated as the reference configuration. References to these test cases are made of the local stator angle at the hub, at the mid-span position and at the shroud. 47_57_67 then indicates the angle is of 47° at the hub, 57° and the mid-span position and 67° at the shroud. Figure 2 (b) depicts the twisted blades in 3D for 67_57_47.

<table>
<thead>
<tr>
<th>Number of stator blades</th>
<th>13</th>
</tr>
</thead>
<tbody>
<tr>
<td>Number of rotor blades</td>
<td>9</td>
</tr>
<tr>
<td>Rotor inlet radius [mm]</td>
<td>47</td>
</tr>
<tr>
<td>Design rotational speed [krpm]</td>
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<tr>
<td>Inlet total pressure [pa]</td>
<td>171000</td>
</tr>
<tr>
<td>Inlet total temperature [K]</td>
<td>293</td>
</tr>
<tr>
<td>Expansion ratio</td>
<td>1.7</td>
</tr>
</tbody>
</table>

Table 1. Main characteristics of the test case

Figure 1. Illustration of the extraction planes used to compute performances (P) and cross-stream colour contours (C).

2. METHODS

2.1 Numerical methods

The structured mesh is realized using the procedure established in the preliminary study [12], with Autogrid V5 proposed by NUMECA, adapted to cases with stator clearance. Its topology is composed of H C and O. At the wall, the size of the first cell in the normal direction is equal to $10^{-3}\text{mm}$ ensuring a value of $y^+ < 1$ at the hub, shroud and blade walls. The stator mesh contains approximately $2.1 \times 10^6$ cells without and 4.0 $\times 10^6$ with clearance, while the rotor mesh is made of 3.1 $\times 10^6$ cells. The computations were performed with FineTurbo, also developed by NUMECA. The code solves the steady and unsteady Reynolds-Averaged Navier Stokes equations based on a cell centered finite volume approach. The chosen turbulence model is the one-equation Spalart-Allmaras model. Regarding the boundary conditions, the total pressure, total temperature and a flow angle of 64° are uniformly prescribed at the inlet of the domain. At the outlet, the static pressure is imposed through a simplified radial equilibrium law. All the computations are realized with a fixed expansion ratio of 1.7 and a rotational speed of 80 krpm.

This study principally uses steady simulations, for which the equations are advanced in pseudo time with an explicit four-stage Runge–Kutta scheme. The interface between the rotor and the stator is treated with the well known mixing plane approach.

The use of unsteady simulations was considered. However, since many configurations were used in this study, such CPU usage would have been excessive. Nevertheless, they were used to check the steady cases’ agreement with their results.
The phase-lagged (chorochronic) approach was chosen. This method enables a reduction of the calculation domain to a single blade passage, without losing the effects of the interaction between the wheels. In other words, the phase-lagged technique supposes that unsteady effects are only due to the movement of the wheel. Then, the flow is time-periodic in the frame of reference of the row and a phase-lag relation exists between two adjacent channels. The time integration was realized with a dual time stepping approach, meaning that at each physical time step, a steady state problem is solved in a certain pseudo time. The number of time-steps to be used to discretise one rotor blade passage was investigated using a static pressure probe just upstream of the rotor-stator interface in the middle of the channel, spanwise and pitchwise. Figure 3 shows static pressure as a function of the time period for different numbers of time-steps. Convergence for 39 time-steps, which was the number chosen for further study.

Efficiencies found using these results as well as steady computations are presented in figure 4. The steady results correspond well globally, with a relative difference between its efficiency and the averaged unsteady value of 0.82%. Furthermore, figure 5 is evidence of the local agreement of these results, showing the colour contour of cross-stream plane C (figure 1) at the rotor outlet, in steady and averaged unsteady cases. Separation is present along the hub-side half span of the pressure side, below two weaker counter rotating vortices. The clearance vortex occupies half of the passage from the shroud in the pitchwise direction, and has an impact nearly up to the suction side. These mechanisms coincide for steady or averaged unsteady results, although they are slightly weaker in the case of the latter. Then, an additional vortex can be observed towards the hub on the suction side. As a whole, the colour contours display the same topology. It can be concluded the impact of the mixing plane and averaged unsteadiness are so that steady simulations are a good basis for further analysis.

2.2 Experimental Validation

Experimental results were used to validate the mesh and the numerical model. For further detail about the setup, see references [6] and [13]. Figure 6 shows experimental and numerical results for the total-to-static efficiency at different
stator positions. Experimental data was not available for twisted configurations, and the nominal point and results corresponding to a 36° stator opening are presented. Some discrepancies may be attributed to the absence in the numerical domain of a volute, which is used for experiment, as well as different outlet pressure measurement positions. Variations are more important at the off design position, which could indicate more unsteadiness in the flow. This study is based on configurations of average stagger angle of 57°, and 57_57_57 displays consistent results.

Figure 6. Total-to-static stage efficiency as a function of expansion ratio of experimental (Exp) and numerical (Num) cases using configuration 57_57_57 with (G) and without (NOG) stator clearance.

3. RESULTS AND DISCUSSION

3.1 Global Performance

Figure 7 presents the global performances of the stage. They are computed using mass-averaged total quantities and scalar-averaged static quantities in extraction planes P1 to P4 (figure 1). Total-to-total stage and rotor efficiencies, represented in (a) and (b), show case 47_57_67 performs much better than 67_57_47. Its stage-efficiency in the presence of stator gap is less than half a point lower than in the straight case, whereas it decreases by more than a point for case 67_57_47. Most of the losses compared to the straight case are in the rotor.

Figure 7 (c) is a graph of the stator loss coefficient, defined as

\[ \frac{P_{t1} - P_{t2}}{1/2 \rho_1 V_1^2} \]

where \( P_t \) is the averaged total pressure in the specified plane and \( \rho_1 \) is the averaged density and \( V_1 \) the averaged velocity in plane P1. As expected, it indicates stator losses are more important with clearance. Furthermore, symmetrical blade twisting has limited impact on loss levels in the stator, especially for 47_57_67.

Figure 7. Stage and rotor total-to-total efficiencies and stator loss coefficient of cases with average stagger angle of 57°, using extraction planes P1 to P4.

Taking this into account, the reasons behind the good and poor performance of 47_57_67 and 67_57_47 are analyzed in the following sections.
3.2 Impact of stator blade twisting on the flow in the stator

The effect of stator blade twisting which was presented in the reference study [12] for cases with no stator gap, is clear on the graph of the absolute flow angle $\alpha$ at the stator outlet, figure 8. The present study shows that adding stator clearance doesn’t impact the main flow, which remains a function of blade angle, while the boundary-like behaviour near the walls affects a larger region, approximately 20% of the span for all geometries.

Blade-to-blade images of entropy in figure 9 show flow separation, as well as clearance vortices. In the straight case, figures (b) and (e) display the same behaviour at heights 5% and 95 %. A symmetry is observed between the twisted configurations. Indeed, cases 47_57_67 and 67_57_47 give the same results at heights of 5% and 95% respectively (figures (a) and (d)), and vise versa (figures (c) and (f)). In accordance with previous researches [14], when the blade is locally in an open position, leakage flow travels to the pressure side before being sucked through to the suction side where the vortex can fully develop, resulting in pressure side flow separation. On the other hand, as the local angle approaches 67° in the channel. In the region of the hub, where the blade is in an open position, entropy levels are on the contrary lower than those found for the straight case. Then, as well as a later development of the leakage vortex, its formation is minimised on the pressure side by blade loading. An interesting result is that it is therefore more beneficial, despite flow separation, to have an opening of 47° at the blade tips, where the leakage flow travels to the pressure side before returning to the suction side.

**Figure 8.** Axisymmetric absolute flow angle $\alpha$ at stator outlet from hub ($h^* = 0$) to shroud ($h^* = 1$).

3.3 Impact of stator blade twisting on the flow in the rotor

Rotor inlet conditions are presented in figure 13, which shows relative incidence. As explained in the reference study [12], for cases with no stator clearance the range of relative incidence at the rotor inlet is only of a few degrees along most of the blade height, while the range of the absolute flow angle reaches 15° (figure 8). Nevertheless, it follows the same trend. Indeed, relative incidence increases with flow angle as stagger rises, and vise versa. With clearance however, a larger local stator angle no longer means an increased relative incidence.

Relative incidence is a function of rotational speed, flow angle and velocity magnitude. Rotational speed is kept constant in this study, and the flow angle abides by the expected behaviour. Relative incidence can therefore be explained by looking at the velocity magnitude at the stator outlet and rotor inlet, figure 14. Four points of interest have been identified, and are detailed in the following paragraphs. Cases without stator clearance will be considered first, followed by configurations with stator clearance.

**Velocity magnitude analysis - no stator clearance**

(i) The effect of twist on the velocity magnitude can clearly be determined for cases with no stator clearance at the stator outlet (figure 14 (a)). Positive twist increases the magnitude of velocity at the stator outlet. This angle variation leads to an increase in the meridional component of velocity, $V_\theta$, which is larger than the decrease in its radial counterpart.
Figure 9. Blade-to-blade representations of entropy [J.kg$^{-1}$.K$^{-1}$] in the stator at $h^* = h/H = 5\%$ (a, b, c) and 95\% (d, e, f). The blade loading normalised using the straight configuration, BL, is given in each case.

Figure 10. Visualisation of clearance vortices - isosurface at 5.10$^{10}$ and contour of the Q criterion.
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Figure 11. Q criterion as a function of blade height (0 at the hub, 1 at the shroud) and distance from the stator blade, in a cross-stream plane near the trailing edge shown in figure 10.

Figure 12. Axisymmetric profile of entropy as a function of blade height (hub at $h^* = 0$, shroud at $h^* = 1$) from upstream ($r = 0.039$) to downstream of the stator ($r = 0.02727$) for 47_57_67, and at the downstream position for 57_57_57.

(ii) In addition, an asymmetry caused by rotor effects on $V_\theta$ is observed between the twisted configurations. It is stronger in case 47_57_67, whose stator angle is more closed towards
This explains the significant difference between the velocity magnitudes for most of the blade height at the rotor inlet (figure 14 (b)).

(iii) At this position, the flow is further accelerated up to 20% span from the shroud because of the impact of rotor clearance on the radial component of velocity. A symmetry between the flow of the two twisted configurations has been observed in section 3.2. Nonetheless, rotor curvature and clearance effects acting in the vaneless space accelerate the flow towards the shroud, and cause rotor inlet conditions vary significantly and asymmetrically. They tend towards the distribution of the straight case 57_57_57 without stator clearance at the rotor inlet. Then, the velocity magnitude increases linearly with blade height until the acceleration is sharpened in the vicinity of the shroud, because of rotor clearance. Twist induces a similar velocity distribution for 47_57_67, and it is accentuated by the rotor. On the other hand, the opposite twisting, 67_57_47, leads to an unnatural increase in velocity with height at the stator outlet. The flow is then forced to accelerate in the radial direction, and the main flow decelerates because of mass flow rate conservation. This explains the significant difference between the velocity magnitude of this case and the straight case at the rotor inlet.

Velocity magnitude analysis - with stator clearance
(iv) Adding stator clearance decreases the velocity magnitude. In addition, it is further reduced up to 20% from the endwalls because of clearance vortices, especially the higher the local stator angle. The shape of the profile is different for cases with a local angle of 47° because of separation. Clearance in conjunction with vaneless flow acceleration dampens the differences between the three geometries towards the hub at the rotor inlet, while they are accentuated at the shroud by rotor clearance. These effects, superposed to the impact of twist, explain the behaviour of the relative incidence.

The impact of different inlet conditions is illustrated in figure 16. Firstly, the vortex which develops due to rotor leakage flow is greatly diminished for 47_57_67, even compared to the straight configuration, while it induces the most losses in case 67_57_47. In addition, passage vortices of great intensity compared to the other configurations are present in this case. This also appears in the plot of rotor outlet swirl, presented in figure 17, whose value shows distortions in the hub-side half-span, whereas it increases uniformly in the other cases. Again, this is in accordance with global performance. It is also worth noting exit swirl exhibits the same values for the three cases on 40% of the span from the shroud, as the rotor leakage flow is dominant on twist effects.

4. PROSPECTS

This study has shown the flow isn’t deteriorated by the use of a stator blade positively twisted at the shroud at the nominal stator angle. This means blade twisting may be used in off design conditions, and requires further investigation at other average stagger angles. Figure 18 presents total-to-total efficiencies of cases with different stagger angles, with and without stator clearance. Straight stator blades, and geometries positively twisted at the shroud such as 47_57_67 are used.

As described in section 3.1, the difference between values corresponding to an average stagger of 57° can be neglected. This is also the case for cases with an average stagger angle of 67°. With a more open stator angle of 47°, the results are promising, with 1 point increase in efficiency with stator
Figure 16. Colour contours of helicity on plane C, near the rotor outlet. Positive values indicate clockwise motion of the vortices, and the opposite is true for negative values. The white areas are out of the range which was chosen to be able to visualise more mechanism of the field.

Figure 17. Axisymmetric swirl at rotor exit from hub ($h^* = 0$) to shroud ($h^* = 1$).

clearance, and 3 points without. This is particularly interesting as losses are significant at this opening. Stator blade twisting at open average stagger angles could prove an effective way of reducing losses, and calls for particular attention.

As well as the average stator angle, other geometries and conditions could lead to improved performance, especially the twist angle and its spanwise distribution, as well as the rotational speed.

Figure 18. Total-to-total efficiency of cases whose stagger angles are of 47, 57 and 67°. Straight configurations and those twisted positively at the shroud, such as 47_57_67, are presented with and without stator clearance.
5. CONCLUSION

Characterisation of the flow in the stator showed local blade loading determines the position of clearance vortex formation. Higher blade-loading induces more upstream clearance vortex development. In addition, symmetrically twisted geometries result in symmetrical flow in the stator.

This symmetry is lost at the rotor inlet, notably because of rotor curvature and clearance effects in the vaneless space. Stator twist distribution must be adapted to these mechanisms to yield to satisfactory relative incidence, which explains 67_57_47’s poor performance and of why 47_57_67 doesn’t deteriorate the flow.

These findings could be amplified by other stator geometries and in other conditions to lead to improved performance. Ultimately, stator blade twisting is a tool to increase the efficiency of regions of the operating range that are of interest for a specific application.

REFERENCES