NUMERICAL INVESTIGATION ON THE PRESSURE PERTURBATION OF A TRANSONIC AEROELASTIC COMPRESSOR RIG

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ABSTRACT

A linear aeroelastic rig was designed to serve as a validation facility for forced response research in axial compressors. This rig focuses on the study of aerodynamic damping at high-reduced frequencies when the central blade oscillates in a controlled manner. The steady and unsteady flow field of the rig has been numerically investigated previously. This paper aims to understand the pressure perturbation throughout the domain, which will give more insight into the actual transonic linear cascade rig and will improve the design of the test rig in some aspects. Such aspects are the tip gaps of the side blades that with the previously optimized upper inclined wall and tailboards are of importance in keeping flow periodicity. The design process has shown that these components do influence the unsteady pressure distribution, which might reduce the accuracy of the experiments. Thus, the aeroelastic cascade itself needs detailed investigations on the systematic accuracy losses in the unsteady testing due to the current design layout. This paper presents the impacts and mechanisms of tip gaps, upper inclined wall, and bottom tailboard on the test rig's aeroelastic performance. It provides a new validation reference for cascade test results with a perspective on the numerical studies, which can introduce a correction procedure on the existing test rig or on future cascade designs.

INTRODUCTION

Since the developed in 1947, the transonic compressor generally becomes the most significant component within the gas turbine due to its high mass flow and high-pressure ratio per stage [1]. Modern transonic compressors create new challenges. The new higher loaded and thinner blade designs have led to the compressor blade to be more prone to high cycle fatigue, which can derive in blade failure. Moreover, the high sensitivity of the transonic flow subjected to a narrow operating range increases the difficulty of their design. Thus, the prediction of forced response behaviour with the inherent high reduced frequency is of great importance in the transonic compressor design.

The forced response performance of the blade is calculated by solving the structural dynamic equations with the aerodynamic forcing and damping as inputs. One of the critical steps is to calculate the aerodynamic damping accurately. The most accessible approach is the controlled flutter method. This approach controls one or several blades vibrating as the unsteady pressure is obtained on the blade surface. Typically, this can be done in the traveling wave mode or the influence coefficient method. The aerodynamic work can be directly calculated when the traveling wave mode method is used. All blades oscillate with a specific phase shift while the unsteady pressure and the blade motion of a blade are calculated. The advantage of this method is that the blade can vibrate at different nodal diameters. In contrast, in the influence coefficient method, only one blade vibrates in a rigid body mode or with a prescribed mode shape at a specific natural frequency, and the unsteady pressure is calculated on all the blades. The blade's unsteady pressure fluctuation is influenced by the vibration of the moving blade. With the assembly of the influence coefficient matrix, the aeroelastic performance can be finally analysed. Panovsky et al studied that the coupling behaviour is mainly on the vibrating blade and the nearby two blades [2]. Theoretically, a three-blade cascade could be satisfied as long as the cascade flow is near periodic.

The linear cascade is the simplified model of the annular cascade. The annular cascade is normally recommended because the inlet and outlet flow varies along the span according to the twist angle. In the flow through a linear cascade, there is no variation in the primary flow angle at the exit of the cascade [3]. As a consequence, the secondary flow in the annular cascade is closer to the real compressor flow. This advantage of the annular cascade is not present in the current study. The test rig aims to simulate the rotor flow with a stator cascade. Therefore, the inlet flow angle needs to be determined by combining the stagger and incidence angles. Compared with the annular cascade, the linear cascade represents a more robust design. Besides, the blade profile changes significantly along the span in the modern compressor rotor. Typically, the mid-span profile is designed for high subsonic flow, while only a small tip part for transonic flow. Since the research only focuses on the transonic region, the annular cascade needs

more mass flow. From the perspective of accessibility and efficiency, the transonic linear cascade is the most promising model for validation, which its development is an ongoing effort at KTH. With the aid of CFD optimization, the test rig was successfully designed [4].

KTH Transonic Aeroelastic Compressor Rig

The blade profile is based on the first rotor of the VINK-6 compressor, designed and optimized by Lejon [5]. Figure 1 and Table 1 show the detailed information of the linear cascade. Due to the blade rotation, the inlet flow is aligned with the chord. While the radius of the VINK compressor shroud changes in the rotor section, the real pitch to chord ratio ranges between 0.718 and 0.726. A value of 0.7224 has been set for the linear cascade. The inlet pressure and temperature were adjusted according to the working conditions of the air supply.

Table.1 Comparison of VINK-6 and the test profile		
Parameters	VINK6	Test profile
Chord length(c/mm)	104.6	
Maximum thickness(t/mm)	2.03	
Height length(s/mm)	70	
Stagger angle(deg)	64.78	
Pitch chord ratio	0.7179-0.7259	0.7224
Inlet Mach number	1.2	
Inlet static pressure(p_{in} /kPa)	39.8	74.94
Inlet temperature(T_{in} /K)	267.8	238.3

Figure 1. VINK-6 profile

Figure 2 provides an overview of the cascade test rig.

Figure 2. Schematic view of KTH test rig

This test rig is a cascade wind tunnel with a one-outlet shape test section and five transonic compressor blades, as shown in Figure 2(a). The blades and tailboards are installed on the rotating disk to adjust for the incidence variations of the flow. To achieve a near periodic flow, there are four important components: the upper wall (red part), the bottom tailboard, the upper tailboard (blue parts), and the throttle (green parts). In the two-dimensional design, the difference of the blade loading is only 5% when the three middle blades are considered [4]. In the experiment, the unsteady pressure is measured from pressures holes on the suction and pressure surface of the middle three blades while the central blade is vibrating in a controlled manner, driven by piezoelectric actuators that trigger the eigenmodes of the blade. Arbitrary control voltages drive the actuators without significantly altering the intrinsic dynamic properties of the underlying structures [6]. Two MFC piezoelectric actuators are directly attached to the pressure

surface of blade0 [7], as shown in Figure 2(b). The third, fourth, and fifth mode shapes can be achieved when the vibrating frequency is up to 1000~3000 Hz (reduced frequency $\omega^* > 2$).

One of the key aspects regarding blade motion is the tip gap. The tip gap will affect not only the steady performance but also the aeroelastic behaviour [8],[9]. The tip leakage flow significantly influences the steady and aeroelastic behaviours in compressors [10]-[12]. This complex phenomenon needs an experimental facility to provide a standard configuration for numerical studies. The enlargement of tip gaps in the transonic wind tunnels cascade might cause other system errors. It becomes even more necessary in the KTH aeroelastic rig with only five blades because the change of the blade loading will affect the periodicity as a consequence.

Figure 2(c) shows the shockwaves inside the test rig. It should be noticeable that the 1st shockwave is generated close to the leading edge of blade+2. This shockwave affects the inlet flow and interacts with other blades with its reflected shock waves. Previous studies indicate that the 1st shock wave can be well controlled by both the bottom bypass shape and the backpressure. The distribution of mass flow between the bottom bypass passage and the central passage is also affected by it. Hence, the flow field near blade+2 largely determines the performance of the whole cascade. A large tip gap of blade+2 will inevitably affect the backpressure in the bottom bypass, affecting the periodicity as the consequence. One possible solution is to use smaller tip gaps on the side blades (blade±2) while using large tip gaps on the middle blades (blade±1,0). However, this solution means the middle three blades are different from the two side blades concerning the tip regions, which might also cause other errors.

The inclined upper wall (red part) is used to prevent choking, as shown in Figure 2(a). Due to the shockwave reflection on the upper wall, the flow near the upper wall is complicated. Tailboards (blue parts) are essential in the test rig to keep the periodicity. The angle of the tailboards is a sensitive factor, and it should follow the exit flow angle [13], [14]. The last component is the throttle used to generate a non-uniform back pressure to adjust the periodicity. The reflection between blade0 and the later walls will cause pressure perturbations, which affect the aeroelastic performance [15]. The controlled vibration of blade0 will cause the leading edge motion, which might induce high unsteadiness due to the movement of the shockwave and its reflections. If the unwanted unsteadiness effects compromise the aeroelastic performance and these are difficult to eliminate, it is necessary to clarify whether these effects contribute to the overall unsteadiness linearly or not. If these are linear, it will be possible to correct the test results during data post-processing. Otherwise, it will be too complex to deal with the unsteadiness the only comes from the vibration of blade0 necessary to calculate the aerodynamic damping.

This paper shows a numerical simulation of the flow throughout the test rig when subjected to a controlled blade oscillation at a high-reduced frequency. Firstly, it provides a benchmark for the initial idea of having a real transient flow in the new test rig. Secondly, the tip gap effect is analysed, providing the reference of the determination of the side tip gap. Lastly, the roles of the sidewalls and tailboard concerning the pressure perturbation will be discussed. The understanding of the systematic accuracy losses will be beneficial in future unsteady testing.

METHODOLOGY

The whole test rig and grids are shown in Figure 3. The mesh in the test section domain was made with ICEM. The element number is between 4M and 5M in most cases. The first layer thickness is of 0.002mm, with a maximum y^+ of 2.5. It includes two sections of the nozzle and a test section with five blades and two tailboards, as shown in Figure 3(a). Double O blocks mesh surrounding the blade was applied to provide a smooth transition on the grids' size. The blade tip is dealt with the C blocks, aiming to focus on the tip leakage flow, as shown in Figure 3b. The commercial solver, ANSYS CFX, was used for the RANS steady and URANS unsteady simulations. SST $k-\omega$ turbulence model was applied with its capacity to predict the separation flow with a certain accuracy. The inlet total pressure is set to 195kPa according to the air supply chamber.

Figure 3. Flow domain and the mesh around the blade

This simulation focuses on the fifth mode shape that is considered a mode with a high reduced frequency. The fifth mode shape was investigated with ANSYS APDL, as shown in Figure 4. The mode's frequency is 2033.4Hz and the reduced frequency is 3.44, which was calculated with Equation 1.

$$k = \frac{2\pi fc}{V_{ref}} \quad (1)$$

Where, f means the blade vibration frequency, c is the chord length. V_{ref} represents a reference velocity, which corresponds to the inlet velocity of the test rig.

Figure 4. Blade displacement of the fifth mode shape

Figure 5. Mesh sensitivity analysis

A mesh sensitivity study was conducted for validation, as shown in Figure 5. The test rig with 1% tip gap (about 4.5M elements) was selected as the reference. To eliminate the effect from y^+ , the first layer thickness on the blades is kept the same with a distance of 0.002 mm. The steady aerodynamic force was used for the study since it takes into consideration the difference in blade loadings that is the main variable of interest. The steady force comes to converge with the increase in element numbers. The difference between the 4.5M case and the 10M case is of only 1.22%. The comparison of the variant torque on blade0 with two oscillating cycles is presented in Figure 6. The prediction of the torque is important because blade0 is the only vibrating blade to induce unsteadiness. Relative to the amplitude of the torque variation, the peak-to-peak difference between the case with 160 time-steps and 40 time-steps is 2.6%. So, 40 time steps do not compromise the accuracy of the unsteady performance prediction for this case.

RESULTS AND DISCUSSION

Tip gap

Table 2 presents five cases with different tip gaps in the numerical simulations. The tip gap changes from 0.5% span length to 2% in Cases 1-3. Narrowed tip gaps of 0.5% span are used in Case 4 and Case 5 for the side blades. Because large tip gaps change the blade loading, the outlet pressure for Case 3 is changed from 132.8kPa to 132kPa in order to achieve periodicity. In Case 5, the outlet pressure drops from 132.8kPa to 132.5kPa.

Table.2 The simulation cases with tip gaps

	Middle three blades (blade-1,blade0,blade+1)	Side blades (blade-2, blade+2)	Outlet pressure (kPa)
Case 1	0.5%	0.5%	132.8
Case 2	1%	1%	132.8
Case 3	2%	2%	132
Case 4	1%	0.5%	132.8
Case 5	2%	0.5%	132.5

Figure 6 (a) shows the Mach number contour at 50% span. The shockwave becomes stronger and moves downstream from Case 1 to Case 2 and Case 3. The isentropic Mach number around the blade's span at mid-span is presented in Figure 6(b). The increase of the tip gap induces a shorter blade and a different blade loading. When the tip gap increases from 0.5% to 1%, the blade loading remains periodic. When the tip gap further increases to 2%, blade0 and blade-1 are still kept periodic while the periodicity of blade+1 worsens. The use of smaller tip gaps on the side blades benefits the periodicity, as in Case 4 and Case 5.

Figure 6. Comparison of the Mach number distribution at 50% span

Figure 7 shows the Mach number contour and the isentropic Mach number around the blade at the 95% span. The flow at 95% span is not as periodic as the mid-span. The low-speed region indicates the tip leakage vortex's location and size in Figure 7(a). The size of leakage vortices in Case 2 and Case 3 are larger than in Case-1 while the angle of the low-speed region increases when compared to Case 1. Figure 7(a) shows that the blade+1 loses its periodicity with 2% tip gap in Case 3. The implementation of 0.5% tip gap for the side blades, seems to assist the blade loading to become more periodic but not too much, presented in Case 5. The results confirm that larger tip gaps worsen the periodicity of the cascade while smaller tip gaps of a side blade can improve the periodicity of blade+1. However, the effect is also limited.

Figure 7. Comparison of the Mach number distribution at 95% span

Figure 8 presents the streamline of the tip region, which indicates the structure of tip leakage vortices. The streamline shows a concentrated vortex that starts from the front part of the blade. The origin of the vortex in Case 1 is earlier than in Case 2 and 3. It indicates that the leakage flow is easier to swirl with a smaller tip gap. However, the tip leakage vortex expands both in pitch-wise and span-wise with a larger tip gap, shown in Case 3. Furthermore, the centre of the tip leakage vortex is also far away from the suction surface in Case 3 and Case 5.

Figure 8. Comparison of the tip leakage vortices

The unsteady pressure coefficient C_p is introduced to present the unsteadiness on the blade loading. It is defined in Equation 2, where p represents the wall pressure, and A means the maximum amplitude of the oscillation.

$$C_p = \frac{pc}{(p_{t,in} - p_{s,in}) A} \quad (2)$$

Figure 9 presents the unsteady pressure coefficient distribution at 50% span and 95% span with a 0.5% tip gap, where the red points represent the pressure side, and the blue points show the suction side. The mode shape at 50% span is similar to a torsion mode. The torsion motion of the blade0 directly affects the shockwave fluctuation near the leading edge and in the blade passage. The first peak of the unsteady pressure amplitude on the suction and pressure side of the blade0 is due to the shockwave motion. Furthermore, the torsion mode affects the flow area of the blade passage, thus introduce high unsteadiness on the rear part of the blade0, the rear part of the suction surface of blade+1, and the front part of the pressure side of the blade-1. Due to the subsonic inlet flow at 95% span, the reason for the unsteadiness has to do with the flow area variation. The pressure unsteadiness is smaller compared with the mid-span transonic flow. It should be noticed that the vibration of the blade0 also affects the unsteadiness of the blade-1 suction surface. It might be due to the existence of the tip leakage flow.

Figure 9. Unsteady response of the middle three blades

Figure 10 shows the comparison of the unsteady pressure response at the mid-span with different tip gaps.

Figure 10. Comparison of the unsteady pressure coefficient distribution at 50% span

The first comparison (cases 1, 2, and 3) focuses on the tip gap influence. A higher tip gap leads to higher unsteadiness at the mid-span. The unsteady pressure increases when the tip gap increases from 0.5% to 1% span length. The location with the most growth is on the rear pressure surface of the blade0 and blade+1. The shape of the peaks remains almost the same which means the

shockwave structure is similar. When the tip gap increases to 2% span, the pressure ratio of the cascade cannot withstand in a near periodic manner, so the outlet pressure has to be changed 0.8kPa, which causes a smaller blade loading. The steady and unsteady pressure of the 2% tip gap case is lower than the 0.5% and 1% tip gaps. Nevertheless, the unsteady pressure amplitude increases in the shockwave region, which means the shockwave motion is affected. The second comparison (cases 1, 2, and 4) focuses on the side tip effects. The unsteady amplitude with the side blade tip case of 0.5% has in overall similar properties than the one with a 1% tip gap. The most affected region is the rear part of the pressure side of the blade0 and blade+1. Since the non-uniform tip gap keeps high periodicity, the use of a narrow tip gap on the side blades is recommended.

Figure 11 presents the comparison of the unsteady pressure response at the 95% span of different tip gaps. The first comparison (cases 1, 2, and 3) clearly shows that the higher tip gap decreases the pressure unsteadiness significantly. It means that the tip leakage flow can reduce the aerodynamic damping in this case. The effect is not linear to tip gaps because the 2% tip gap causes different unsteady pressure distribution. The second group (cases 1, 2, and 4) indicates that the non-uniform tip gap is almost the same as the uniform 1% tip gap. Thus, the use of a narrow tip gap on the side blades does not affect the aeroelastic performance of the tip part.

Figure 11. Comparison of the unsteady pressure coefficient distribution at 95% span

To quantify the influence of different cases on the unsteadiness. The force in the Y direction (e.g. perpendicular to the chord line) is integrated at 50%, 65%, 75%, 85%, 90% and 95% span. The unsteady force in the Y direction is the amplitude of the harmonic force after a Fourier transformation, which represents the unsteadiness at a certain span. Figure 12 shows the unsteady aerodynamic forces on blade-1 and blade+1.

Figure 12. Comparison of the unsteady forces in the Y direction

The comparison between Case 1 and Case 3 indicates that large tip gaps reduce the unsteady forces both blade-1, shown in Figure 12(a), and blade+1 in Figure 12(b), especially near the tip region. The enlargement of the tip gap also changes the blade loading distribution. Thus, the unsteady forces might increase slightly, such as at the mid-span of blade-1 and 65% span of blade+1. Case 3 and Case 5 are considerably similar, which means the change of the tip gap of side blades (blade $\pm$ 2) does not affect the unsteady performance. The change of the side blades' tip gaps (blade $\pm$ 2) is recommended, so smaller tip gaps can be used because they benefit the periodicity without affecting the unsteady performance fundamentally.

Sidewalls

Figure 13 shows the steady and unsteady results of an infinite cascade. The infinite cascade domain was built by using 11 blades, the middle blade (Blade0) has been oscillated with the fifth mode shape, and the unsteady pressure propagation on blade +5 and -5 is considered negligible, for this reason is considered as an infinite cascade. Since the inlet and outlet pressure is 188kPa and 115kPa respectively, the shockwave position is different from the one in the test rig, shown with the blade loading in Figure 13(b). Figure 13(c) compares the unsteady pressure amplitude distribution of the blade0 between the infinite cascade flow and test rig flow. The unsteadiness on the front part of the blade0 is different. The test rig needs further pressure and geometric adjustment to simulate a similar shockwave performance. The second difference is the rear part of the pressure surface. It is obvious that the unsteady pressure amplitude is much higher in the test rig flow than in the infinite cascade flow. The phenomenon needs to be analysed since it causes high systematic accuracy loss in the testing.

Figure 13. Comparison of the infinite cascade flow and the rig flow

Figure 14 compares the unsteady pressure distribution at mid-span between the infinite cascade and the test rig cascade flow. Two maximum oscillation amplitudes are set as 1mm and 0.5mm, which represents approximately 0.5% and 1% of the chord length. It should be noticed that the unsteady pressure range in the first case is ± 10 kPa in Figure 14(b) and in the second case is ± 5 kPa in Figure 14(c), half as the first case.

Figure 14. Unsteady pressure distribution at 50% span during an oscillating cycle with different amplitudes

Compared to Figure 14(a) with Figure 14(b), the most crucial difference is that the unsteady pressure reflection on the rear part, which explains the high unsteady pressure coefficient on blade0. The highest unsteadiness occurs between blade0 and the bottom tailboard in the test section, while the unsteadiness in the infinite cascade is not as high, though the trend remains indicating localized unsteady pressure at the trailing edge as well. It highlights the requirement of an absorbing material on the bottom tailboard or a correction method on the unsteady testing data. The unsteadiness near the leading edge due to the vibration of blade0 is similar in the infinite cascade and the test rig cascade. Although the transonic flow near the upper wall is complex due to several shockwaves, the unsteadiness is still at a considerably low level, which was not expected. It proves that the interaction with the sidewalls does affect the unsteady performance but will not cause systematic errors in the unsteady testing. Figure 14(c) shows that the unsteady pressure distribution for the different amplitudes is qualitatively the same.

Figure 15 shows the unsteady pressure distribution at 95% span. The high unsteadiness close to the leading edge exists at the 50% span due to the shockwave motion, but disappears at 95% span. It is due to tip inlet flow that reduces the Mach number to 0.8. The highest unsteadiness is still located near blade0 and close to the bottom tailboard. Simultaneously, the unsteadiness between the suction side of blade0 and the pressure side of blade-1 increases, which is due to the tip leakage flow. The unsteady pressure distribution is similar in the two amplitudes cases, as shown in Figure 15(b).

Figure 15. Unsteady pressure distribution at 95% span during an oscillating cycle with different amplitudes

Table 3 summarizes the amplitude of the unsteady forces in one oscillating cycle. While the blade vibrates with 1mm amplitude, the unsteady force is almost double as the 0.5mm case. It proves that the sidewall effect exists, but mainly on the bottom tailboard, and that the sidewall effect on the aeroelastic performance behaves linearly.

Table.3 Comparisons of the unsteady force in Y direction

Unsteady force in Y Direction (/N)	Blade-1	Blade0	Blade+1
Amp.0.5mm	6.0936	9.3688	2.7682
Amp. 1mm	12.3807	18.2827	5.6984

CONCLUSIONS

In this paper, numerical simulations are applied to investigate the effects on the unsteady performance due to the components inside the cascade wind tunnel. The conclusions can be summarized as the following:

1. In this case, the tip gap has a positive effect on reducing the unsteady pressure variations due to the oscillation of blade0. However, the phase needs to be taken into account to have a better understanding. Since the tip leakage flow of blade+2 worsens the periodicity of the transonic cascade wind tunnel, it is recommended to minimize the tip gap for the side blades. When smaller tip gaps are used on the side blades (Case 4 and Case 5), the steady performance becomes periodic, and the unsteady performance remains similar.
2. The unsteady pressure reflections on the tailboard have a significant impact. The bottom tailboard causes more unsteady reflections than the upper wall and the upper tailboard. This bottom tailboard contributes to the largest unsteady difference when compared to the infinite cascade. Nevertheless, the effect on unsteadiness due to this pressure perturbation is linear, which can be corrected.

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