Blade oscillation mechanism for aerodynamic damping measurements at high reduced frequencies

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Abstract. Accurate prediction of aerodynamic damping is essential for flutter and forced response analysis of turbomachinery components. Reaching a high level of confidence in numerical simulations requires that the models have been validated against the experiments. Even though a number of test cases have been established over the past decades, there is still a lack of suitable detailed test data that can be used for validation purposes in particular when it comes to aero damping at high reduced frequencies which is more relevant in the context of forced response analysis. A new transonic cascade test rig, currently undergoing commissioning at KTH, has been designed with the goal to provide detailed blade surface unsteady pressure data for compressor blades profiles oscillating at high reduced frequencies. The paper provides an overview of the blade actuation system employed in the test rig and presents the result of a series of bench tests characterizing the blade vibration amplitudes achieved with this actuation system.

Nomenclature

c	blade chord, mm
f	frequency, Hz
k	reduced frequency, -
M	Mach number, -
u	flow velocity, m/s
$\dot{m}$	mass flow, kg/s
γ	blade stagger angle, degrees

1 Introduction

Accurate determination of aerodynamic damping is crucial in the context of aeromechanical analysis of turbomachinery components. Although significant progress has been made over the past decades when it comes to development of the advanced simulation tools, there is still a lack of high quality, detailed measurement data for validation of these tools. One of well-established experimental approaches for aerodynamic damping investigation is to measure the aerodynamic response during controlled blade oscillation. This approach provides spatially resolved data, in particular suitable for validation purposes. Commonly the test data is acquired in simplified setups like non-rotating linear or annular cascades. The tests are performed with all the blades oscillating in a traveling wave mode and the unsteady blade surface pressure measured on one blade. Or with only one blade oscillated while the unsteady blade surface pressure is measured on the oscillating blade and a number of neighbor blades, commonly known as influence coefficient approach. Assuming small disturbances and linear superposition of the unsteady flow these two approaches correlate well for small oscillation amplitudes [1, 2, 3].

The existing cascade facilities are mainly used for controlled flutter testing, where the oscillation frequencies of interest are rather low (up to several hundred Hertz). The airfoils are oscillated as rigid bodies and the induced modes are mostly two-dimensional modes. Only in a few cases three-dimensional rigid body modes are achieved [4, 5]. The oscillation systems usually employed in these cascade rigs are either mechanical [3, 4, 6, 7] or electromagnetic [8, 9]. Mechanical systems transform rotational movement from a speed-controlled electrical motor into an oscillating bending or torsion motion of the blade. They are subject to wear and in general feature low power density. Electromagnetic systems employ one or several electromagnetic shakers to excite the oscillation of the elastically suspended blade(s). They are less sensitive to wear compared to mechanical systems and can have higher power density, but neither of these two types are capable of achieving very high oscillation frequencies.

The current work will focus on aero damping at high reduced frequencies, which is more relevant in the context of forced response analysis. The targeted reduced frequencies (calculated according to Eq.1) are in region of k~2-4, implying that in the considered transonic flow rig the blade would have to be oscillated at frequencies of ~1 kHz to 2.5 kHz.

$$k = \frac{2\pi \cdot f \cdot c}{u} \tag{1}$$

Achieving such high frequencies is simply not feasible with the above-mentioned oscillation mechanisms.

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Therefore, the choice of the oscillation mechanism has fallen upon using piezoelectric actuators that can be embedded on the surface of the blade. The idea is to excite, in a controlled manner, the natural modes of vibration present in the blade structure.

Piezo-electric elements have been previously used in a number of turbomachinery related experiments, mainly due to their ability to act both as sensors and actuators which makes them suitable for excitation purposes and active vibration control. An active control technique was described in [10], where only one pair of piezoelectric patches (actuator and sensor) were used effectively to damp several resonant modes of the test structure (a titanium cantilevered beam). Approximately 98% damping increase was achieved for the investigated 3rd bending mode of a cantilevered beam at 824Hz. The same authors later successfully demonstrated viability of using piezoelectric sensors and actuators as a part of an active control scheme to reduce resonance vibrations in composite subscale fan blades under centrifugal load [11]. The results from the conducted spin rig tests showed that active damping of 0.5% to 1.0% critical is achievable for the first bending mode over the investigated rotational speed range (up to 5000 rev/min).

Application of piezo-electric patches for controlled blade excitation in aeroelastic investigations of a rotating compressor blisk was demonstrated in [12]. It was shown that with employed piezo-electric patches (macro fiber composites MFC) the controlled excitation was possible to a certain amplitude, if the excitation frequency was near resonance. The investigated blade mode shape was 1st bending (1F) mode at ~148Hz. The paper highlights also the limitations of the technique under engine-like operating conditions. Another study where MFC piezo electric patches were employed on rotating compressor blades was described in [13]. The tests carried out in a two-stage low speed axial-flow compressor rig showed that the forces induced by MFCs on blades of the 1st rotor were large enough to actuate the rotor blades up to a predefined vibration amplitude. The investigated blade mode shape was 1st flap at ~342 Hz (at 3000 rev/min).

The above-mentioned studies indeed demonstrate feasibility of using piezo-electric materials for controlled blade excitation, however the frequencies of the mode shapes investigated in these studies were still rather low compared to what the current work will attempt to address.

2 Transonic linear cascade rig at KTH

A new transonic compressor cascade rig has been designed and constructed at KTH (Figure 1). The rig is primarily intended to be used for aero damping measurements during blade oscillation at high reduced frequencies and for investigations of aeroelastic damping at near stall flow conditions.

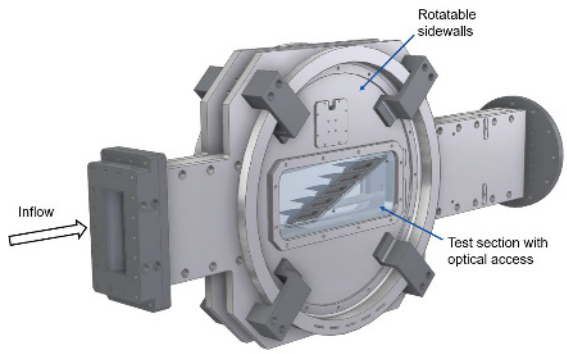


Fig. 1. Transonic compressor cascade rig

The cascade contains five compressor blade profiles. The profiles are based on the 95% span section of rotor 1 blades from a 3-stage high-speed booster developed within a Swedish national research project called VINK (Virtual Integrated Compressor Demonstrator) [14]. The blades used within the transonic linear cascade application in the KTH rig are a prismatic extrusion of above mentioned profile shown in Figure 2.

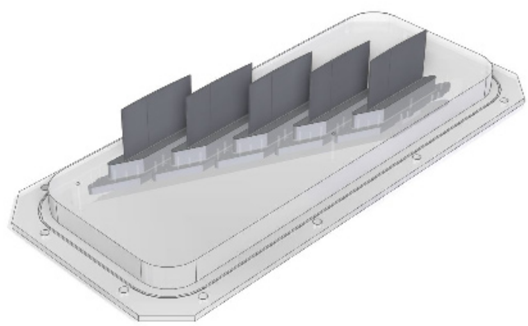


Fig. 2. Cascade blades mounted on the optical window

The rig has a fixed geometry inlet nozzle designed for achieving the targeted inlet Mach number of M=1.22. The wind tunnel will be operated at the maximum mass flow available ($\dot{m}$ ~4.8kg/s), resulting in a total pressure at the inlet of around 180kPa and at total temperature of 320 K. The operating condition in the cascade is set by controlling the pressure downstream of the cascade with help of tailboards/throttle and suction fans mounted at the far outlet of the wind tunnel piping system (the wind tunnel is operated in an open circuit mode).

To achieve the aerodynamic conditions similar to the reference VINK compressor blades, pitch-to-chord ratio of 0.72 and stagger angle of γ=65° has been also maintained in the cascade. The flow into the cascade is axial, thus imposing that the blades are staggered with 65 degrees angle with respect to the inlet cross-section. Blade chord is 104.6mm and max. thickness is 2.04mm. Given the mass flow limitations, a blade span height of 70mm is achievable. The tip clearance is set to 0.7mm (1% span height).
A schematic view of the test section is shown in Fig. 3. To achieve a satisfying periodicity in the cascade, two movable tailboards are implemented, where the lower tailboard also features a smaller adjustable height throttle for control of the backpressure distribution downstream of the blade. In addition, the blades are

mounted on a turntable disc, allowing for incidence angle variability of up to ±8 degrees.

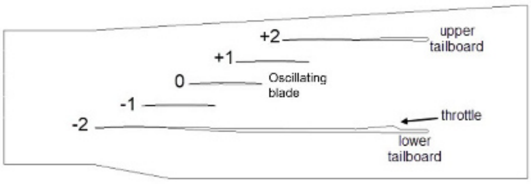


Fig. 3. Schematic view of the cascade rig test section

The test section has full optical access through sidewall windows allowing for visualization of the shock wave patterns with the help of Schlieren technique (capturing density gradients over the shockwaves). The visualization of the shock wave system in the cascade is necessary to qualitatively asses the flow periodicity in the cascade and that the targeted flow condition is being set. Detailed flow measurements upstream of the blades will be carried out with a L2F (Laser-to-Focus) system, while downstream of the cascade a three-hole pneumatic probe will be traversed.

The cascade will be operated in the influence coefficient mode i.e the central blade (blade 0) will be oscillated in different mode shapes, while the unsteady pressure data is acquired on both surfaces of the oscillating blade itself and on the adjacent blade surfaces facing toward the oscillating blade (i.e. SS of blade -1 and PS of blade +1). During these aeroelastic tests the oscillation of the blade will be monitored by means of a single-point laser vibrometers and acquired together with the unsteady pressure data on the blade surfaces.

2.1 Oscillating blade

The blade vibration in the cascade is being induced by means of two piezoelectric actuator elements. These are embedded into machined pockets on the surface of the blade as shown in Figure 4. The piezoelectric actuators are supplied with continuous voltage signal in form of a sinus wave with a frequency corresponding to the eigen mode frequencies of the blade and in that way the natural vibration mode is being excited.

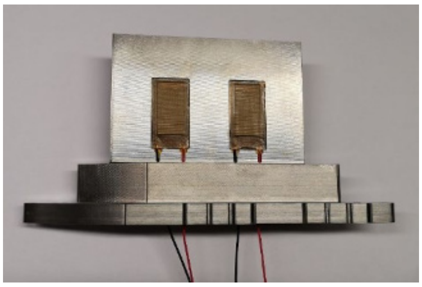


Fig. 4. Oscillating blade with embedded piezo-electric actuators

The piezo-electric actuators are Macro Fiber Composite (MFC) actuators from Smart Material, with high blocking force and good performance at high frequencies. The MFC consists of rectangular piezo ceramic rods sandwiched between layers of adhesive, electrodes and polyimide film. The electrodes are attached to the film in an interdigitated pattern, which transfers the applied voltage directly to and from the ribbon shaped rods. The actuators used on the blade will be of P1 type, meaning that under the applied voltage the active piezo element elongates, as shown in the Figure 5.

Once glued onto the surface it will transfer this elongation into a force acting on the blade. The chosen type of MFC actuator is called M-2814-P1, with active length of 28mm and active width of 14mm. It has a capacitance of 1.15nF and produces a blocking force of 195N. The overall dimensions of the piezo element are 38mmx25mm and a total thickness of only 0.35 mm, which is very important since these elements are embedded onto the blade surface and the blade maximum thickness is only 2.04mm.

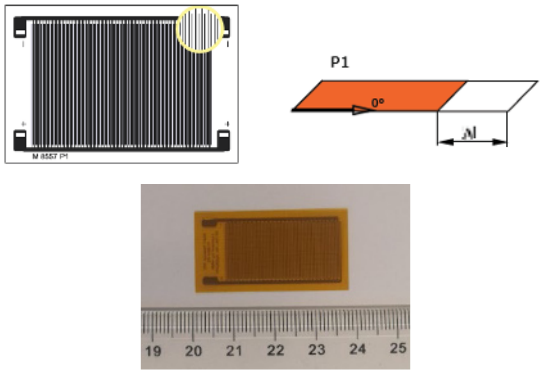


Fig. 5. Piezo-electric actuator (MFC P1 type)

An FE model has been developed to model the behavior of the blade-piezo system and to find the optimum location for the integration of the piezo patches onto the cascade blade, which results in maximum achievable vibration amplitudes for the modes of interest. The simulations carried out in ANSYS Mechanical included modal analysis to identify the modal frequencies of interest and harmonic analysis of the oscillating blade. These harmonic analysis are thought to be the best representation of the physics as the piezoelectric actuator is fed by a sinusoidal electrical signal, deforming and causing a force with harmonic sinusoidal pattern. In the setup of the harmonic analysis user can introduce different value of the damping ratio and different voltages feeding the piezo elements and by that controlling the vibration amplitude. Other parameters such as mechanical properties and piezoelectric properties are pre-set and kept constant. The results are obtained through Frequency Response analysis with frequency sweep centered in the modal frequency of interest. The electromechanical properties of the piezo actuator are defined in the FE model by defining the stiffness matrix, piezoelectric matrix and permittivity matrix.

Prior to the blade response simulations, the FE model has been validated on a 2mm thick aluminum Al-7075 plate of similar dimensions as the cascade blade. The predicted mode shapes and amplitudes of the oscillating flat plate were compared to the experimentally measured ones and a very good agreement was found [15].

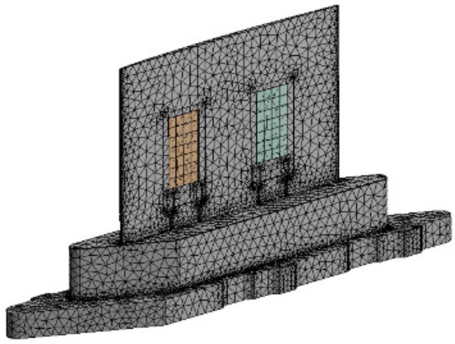


Fig. 6. FE model of the cascade blade showing optimum location of the piezo actuators

The largest uncertainty in the FE prediction of the blade vibratory response is the damping ratio, which in the performed simulations was based on the material damping values found in literature [16]. The model was not taking into account any friction damping. The amplitude was found to be directly proportional to the voltage applied to the piezo-actuators and inverse proportional to the damping value used. Even though the absolute values of the predicted response amplitudes were highly dependable on the set damping ratio values (i.e. on material damping values), the model has proven to be useful in identifying the optimum position of the piezo-actuators on the blade resulting in maximum vibration amplitudes for all modes of interest.

Since the choice of material will affect how large vibration amplitudes can be achieved, two different materials were considered when the first two blades were manufactured, namely Ti-6Al-4V and SS17-4ph (precipitation-hardened stainless steel). Both blades are then equipped with piezo-electric actuators and assessed to see which one gives the higher oscillation amplitudes, leading to appropriate material choice for all the subsequent manufacturing.

3 Experimental setup for blade vibration characterization

When excited by the piezo-electric actuators the blade will oscillate in its natural modes of vibration. Thus, it is important to characterize the blade motion by scanning the amplitude of the actual blade mode shape. To allow for detailed characterization of the blade vibration under the actuation, a bench test rig was set up as shown in Figure 7. During the bench tests, the investigated oscillating blade was fixated into a designated seat, machined in the optical window, which resembles how the blade will be mounted in the cascade rig. This is to avoid different mechanical boundary conditions compared to the ones during the aeroelastic testing in the cascade rig.

A sinusoidal input voltage signal to the piezo actuators is supplied by using a two-channel signal generator and a high-voltage amplifier. The amplifier used is HVA 1500/50-2, a two-channel amplifier from Physical Instruments, capable of delivering an output voltage in the range of -500 V to +1500 V at an output current up to 50 mA DC. Each amplifier provides a large signal bandwidth of DC to 10 kHz depending on the load capacitance and has a gain of 200 V/V. Input voltage signals are separately controlled to each of the two piezo-electric elements allowing both in-phase and out-of-phase signal feeding. Out-of-phase operation is in particular useful when torsion modes are to be excited (due to a symmetric placement of the piezo elements from the mid-chord of the blade), resulting in augmented blade amplitudes for those modes. The maximum voltage input to the MFC piezo elements used here is ±500V.

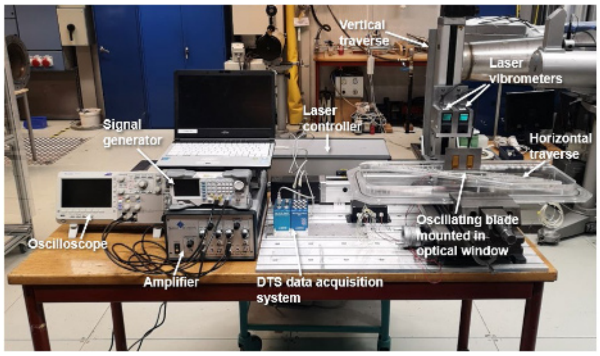


Fig. 7. Bench rig setup for blade vibration measurements

Blade vibration amplitude is measured by traversing two single-point laser vibrometers LK-G152 high-speed laser displacement sensors from Keyence. The laser sensor has a maximum sampling frequency of 50kHz. The linearity of the sensor is ±0.05% of F.S. (F.S.=±40mm) and given repeatability is 0.5µm. The traverse measurement shown in Figure 8 consists of 22x13 measurement points covering in principle the entire suction side surface of the blade.

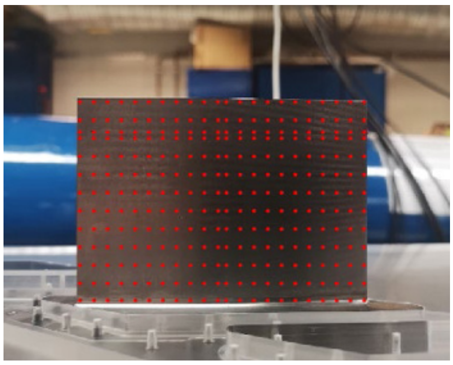


Fig. 8. Laser traverse measurement grid

The laser signal is acquired using a high-speed data acquisition system SLICE PRO SIM from DTS. The system features 18 channels with 16bit A/D conversion for each channel and a maximum sampling rate for all 18 channels simultaneously of 500ksps/ch. The present tests were performed at a sampling rate of 20ksps. The vibration amplitude is determined by ensemble averaging the laser signal data (typically 200 periods) with respect to the oscillation period.

4 Results and discussion

As mentioned previously, the targeted reduced frequencies are in region of k~2-4. This implies that in the considered transonic flow regime the blade would have to be oscillated at frequencies of 1 kHz to 2.5 kHz. The results of the modal analysis carried out using the FE model indicate that this frequency region corresponds to modes 3 to 6, as displayed in Figure 9.

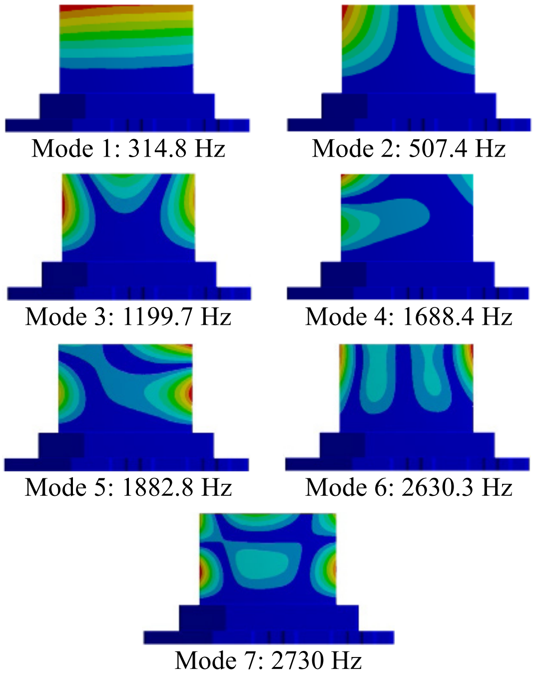
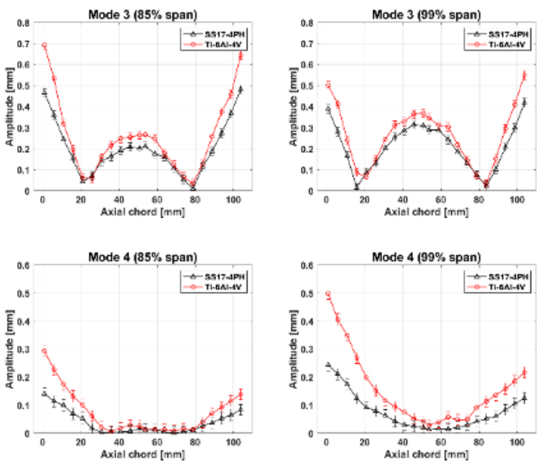


Fig. 9. FE predicted mode shapes of the cascade blade (Ti-6Al-4V)

The first step in the tests was to identify which of the two materials (Ti-6Al-4V, SS17-4PH) provides higher blade vibration amplitudes. The amplitude comparisons were made at 85% and 98% span sections and for modes 3 to 6. The measured amplitudes at these two span sections are plotted in Figure 10.



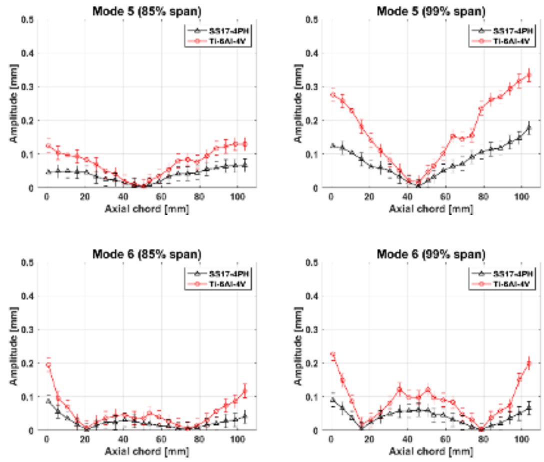


Fig. 10. Comparison of measured vibration amplitudes for Titanium and Stainless Steel blade

The titanium alloy blade gives higher amplitudes for the same voltage applied to the actuators for all of the modes of interest. This difference is in particular obvious at higher frequencies. Since it is important that sufficient amplitudes can be achieved even at these high frequencies, it is decided that the oscillating blades for the rig tests will be manufactured in titanium alloy.

Based on this, further detailed mapping of the vibration mode shape and amplitudes was done for the titanium blade and the results of these measurements are presented in Figures 11-17. The voltage fed to the piezo actuators during these measurements was maximum voltage achievable i.e. ±500 V, except for the 1st bending mode, where ±300V was already sufficient to achieve amplitudes higher than 1mm.

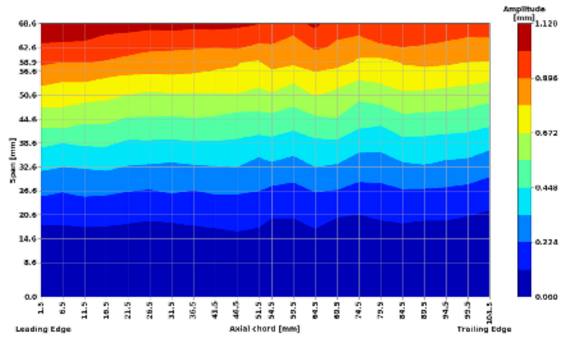


Fig. 11. Measured amplitudes for Mode 1 @ 285Hz

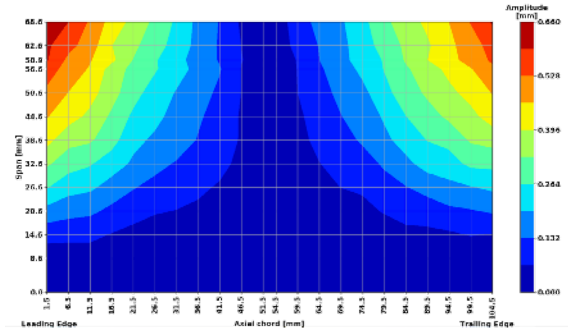


Fig. 12. Measured amplitudes for Mode 2 @ 499Hz

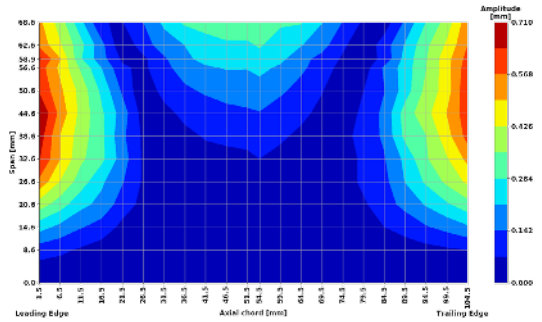


Fig. 13. Measured amplitudes for Mode 3 @ 1230Hz

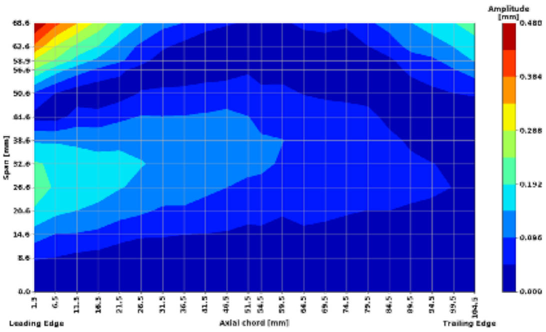


Fig. 14. Measured amplitudes for Mode 4 @ 1627Hz

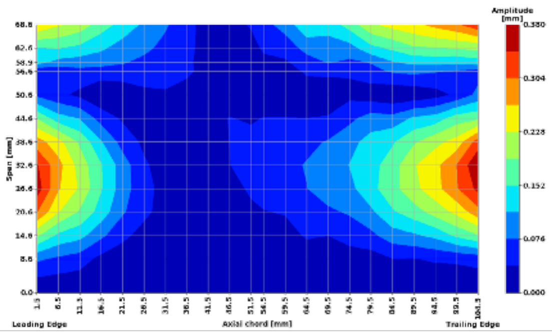


Fig. 15. Measured amplitudes for Mode 5 @ 1884Hz

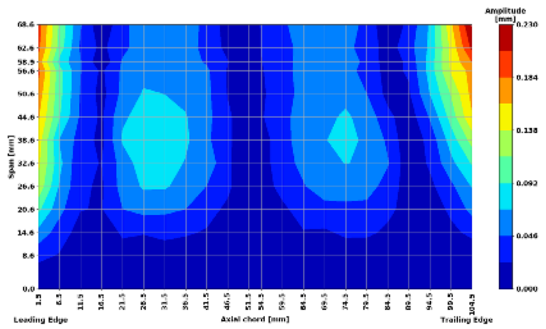


Fig. 16. Measured amplitudes for Mode 6 @ 2517Hz

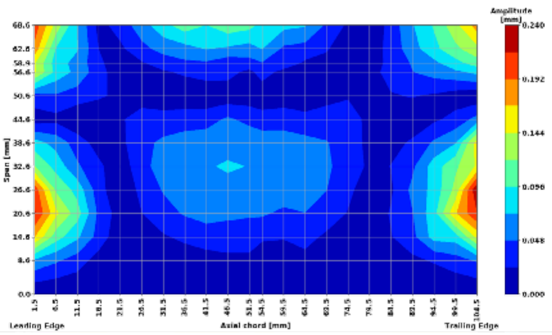


Fig. 17. Measured amplitudes for Mode 7 @ 2692Hz

It can be observed that qualitatively there is a good agreement between the predicted mode shapes shown in Figure 9 and the ones measured during the blade excitation. There is a slight difference in shape of the 5th mode where the experimental results indicate somewhat different and more pronounced deflection in the leading edge region.

Another reflection made is that the predicted mode shape frequencies do not match the frequencies where the maximum vibration amplitudes were obtained in the experiments. The starting point in all of the conducted amplitude measurements was to set the input voltage signal frequency to the predicted mode shape frequency and make a fine step sweep around this value until the resonance point occurs and maximum amplitude is achieved. For all modes the resonance occurred at lower frequency than the predicted mode shape frequency, except for mode 3 where opposite was observed. The difference in these two frequencies can be as high as 113Hz as seen for mode 6.

Table 1. Measured max. amplitudes for modes 1-7.

Mode	f [Hz]	k [-]	Max. amplitude [mm]
1	285	0.49	1.12
2	499	0.85	0.66
3	1230	2.1	0.71
4	1627	2.78	0.48
5	1884	3.22	0.38
6	2517	4.31	0.23
7	2692	4.61	0.24

It is considered that the achieved max. vibration amplitudes will be sufficient for the aeroelastic tests in the cascade rig. The amplitude decreases with increased frequency: max amplitude for mode 3 was 0.71mm while for mode 7 max amplitude was measured to 0.24mm. This can be explained with that the higher modes in general are more difficult to excite as well as that blocking force of the piezo actuators drops down with an increase in frequency.

In terms of practical usage of the piezo-electric actuators as devices for inducing blade vibration following was observed during the tests:

- Strong bounding between the piezo element and blade surface is essential for transfer of the deformation of the piezo element to a force acting on the blade. On the other side, this implies that once glued properly onto the blade surface, the elements can only be removed by applying a raw mechanical force.
- The performance of the piezo-actuators is maintained even after a longer period of use. The actuators employed during the tests have 15+ hours of operation at various frequencies and still perform like when they were entirely new.
- Long-term operation at frequencies above 2.5kHz leads to certain heat development from the element itself (in particular if it sits on a blade made in material with a very low thermal conductivity like titanium alloys). This should be monitored carefully, since the excess heat of 90 °C could be

detrimental for the performance and health of the piezo element.

5 Conclusions

An approach for using embedded thin piezo-electric actuators as a device for controlled blade oscillation has been presented. The technique showed promising results for the investigated blade profiles and significant blade vibration amplitudes were achieved even at high frequencies (f >2kHz), which would in general be out of reach for other traditional devices (mechanical or electromagnetic) used in the existing aeroelastic test rigs.

The amplitude measurements presented in this paper have been performed in a room ambient, and in the next steps the performance of the actuators also needs to be assessed in the transonic flow conditions. At a later stage, the mode shape measurements of the oscillating blades instrumented with Kulite transducers will be performed in a vacuum chamber to assess the effects of the induced blade oscillation on pressure transducers' diaphragm and sensor readings

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