PIEZO-ELECTRIC ACTUATION OF ROTOR BLADES IN AN AXIAL COMPRESSOR

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ABSTRACT

An actuation method for the experimental investigation of compressor rotor blade vibrations is demonstrated. Three blades of the first compressor stage in a two-stage low-speed axial-flow compressor have been equipped with piezo-electric actuators, i.e. macro fiber composites (MFC). For measuring the actuated blade vibrations, strain gages are used. The control system which has been developed within this project allows to pre-set a vibration amplitude of the blade tip. The transmission of the actuation voltage of up to 1000V into the rotor is accomplished by a slip ring. A slip ring is also used for transmitting the strain gage signals.

The tests in the rotating machine show that the forces induced by MFCs are large enough to cause sufficiently strong vibrations. The signal quality of the data transmission via slip ring is high enough to operate the control unit that allows actuating the rotor blades up to a pre-defined vibration amplitude when matching the lower eigen modes of the blades.

NOMENCLATURE

D_o	outer diameter	z_{IGV}	number of blades of IGV
h_z	blade height	z_{rot}	number of blades of rotor
m_{dot}	mass flow rate	z_{stat}	number of blades of stator
n	rotational speed	Π	total pressure coefficient

INTRODUCTION

Blade vibration and flutter have an increasing impact on the operation of aircraft compressors. This is mainly due to two factors: The demand of reducing engine weight for decreasing the fuel consumption leads to more slender blades on the one hand side and higher pressure ratios per stage on the other side. The consequences are more flexible blades with higher aerodynamic loading. This again leads to stronger vibrations. Additionally, new design and manufacturing techniques like blisk designs intensify the problem of vibration due to the decrease of mechanical damping in a blisk rotor. Hence, there is a strong interest in understanding aeroelastic phenomena in turbomachinery.

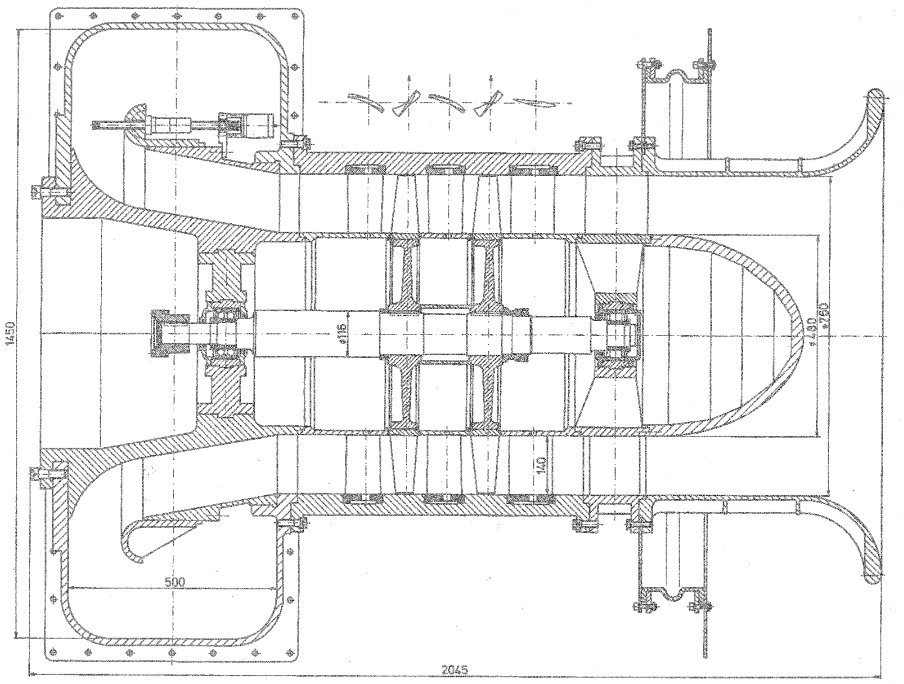
Over the years, extensive research has been carried out on the subject of rotor blade vibrations, one of the pioneers being Campbell (1924). Many recent studies have been based on computer simulations. Montgomery et al. (2005) give a good overview of the application of modern CFD tools for predicting unsteady aerodynamics and aeroelasticity. A large number of experimental investigations has been carried out in wind tunnels where the examined blades were more easily accessible than in a rotating frame of reference, e.g. Belz and Hennings (2004). Experiments investigating blades in turbomachinery or rotating test rigs have been either passive or active. For the active actuation of rotating blades mainly two techniques have been implemented. Christensen and Santos (2004 and 2006) implemented shaft-based actuation via electro-magnetic bearings. Very recent investigations by Charleux et al. (2006) use piezo-electric actuators and sensors. However, their test rig is not a real turbomachine but consists of four rotor bales rotating in a vacuum chamber. Generally, only in a small number of investigations rotor blades in real turbomachinery are actively actuated. A study very similar to the present project is provided by Kielb and Abhari (2003) investigating the aerodynamic damping in a turbine. Their paper presents an experimental measurement of forced response of a rotating blade-disk at engine representative conditions. Kielb and Abhari (2003) use engine hardware from a full scale turbine rig but not the complete turbine rig itself. Hence, the investigated turbine consists of one stage only.

Beyond that, the aim of this present study is to demonstrate the piezo-electric actuation of rotor blades in two existing, complete, and rotating turbomachines and utilize it for active vibration control.

The first machine is a low-speed axial-flow compressor with IGV and two stages set up at the Institute for Turbomachinery and Fluid-Dynamics of the Leibniz University Hannover, Germany. This machine has been existing for a couple of years and has already been investigated in previous studies (e.g., Griebel and Seume (2005)). Here the first rotor stage has been equipped with piezo-electric actuators and strain gage sensors. From former investigations it is known that this machine shows only small flow induced vibrations. Hence, in this study, the vibrations detected by the strain gage sensors are mainly due to the piezo-electric actuation of the rotor blades. Therefore, this machine is very well suited for studying the major mechanisms of actuating rotor blades via piezo-electric actuators. In the present project, this first machine is used for a conceptual study and test application, the outcome of which will benefit the flutter investigations carried out in a second machine.

The second machine will be a blisk fan operated in the M2VP-wind tunnel of the DLR Institute of Propulsion Technology at Cologne. The fan will be equipped with piezo-electric actuators, strain gages, and unsteady pressure sensors. A control unit designed in the current project allows operation of the machine at a pre-set vibration amplitude and phase angle of the rotor blades. The aim is to measure the unsteady rotor blade pressure distributions induced by the vibrations.

In this paper the first machine and the results of the conceptual study are described, whereas the results of the second machine will be presented in subsequent papers.



$D_o = 760$ mm

$h_z = 140$ mm

$z_{IGV} = 20$

$z_{rot} = 30$

$z_{stat} = 20$

$n = 3000$ rev./min

$m_{dot} = 16.5$ kg/s

$\Pi = 1.035$

Figure 1: Drawing of the low-speed axial-flow machine and machine specifications

TEST FACILITY

The test facility used in the conceptual study is a closed loop facility with a low-speed axial-flow compressor with IGV and two stages. It is set up at the Institute for Turbomachinery and Fluid-Dynamics of the Leibniz University Hannover, Germany. Figure 1 shows a sketch of the investigated machine. The rotor blades have a NACA 65-(14)10 thickness distribution. The ratio of thickness to blade height is roughly 0.043. Hence, the rotor blades are comparably solid. Besides, the machine has rolling bearings and is aerodynamically very lowly loaded. Therefore, flow induced vibrations of the blades as well as the entire machine are rather minor. More information on this test facility and machine is provided in Griebel and Seume (2005).

The actuation of the rotor blades is accomplished by piezo-electric actuator foils bonded to the rotor blades' suction side. As Fig. 2a) shows, three rotor blades are equipped with actuators. So far, however, only one actuator has been used. Figure 2b) shows the pressure sides of the same three blades. Each blade is equipped with one full bridge and one single strain gage (metal foil resistive). The full bridges consist of four sensors, two of them directed in the direction of main stress and two perpendicular to it. This type of bridge was chosen to double the output signal as well as for temperature compensation. However, a full bridge needs more paths on the slip ring than a quarter bridge. For that reason, both types – full and quarter bridges – are examined in this first machine to get a direct comparison of signal quality. On the basis of these results the sensors used in Cologne have been chosen. Due to the limited number of conductor paths on the slip ring in the present machine (see Fig. 4), only four out of six strain gage signals can be transmitted out of the rotating frame of reference at a time. The sketch in Fig. 3 shows the strain gages used in the present investigation and the numbering of signals which correlates with the results presented in this paper. The positions of the strain gages on the blades have been chosen based on a numeric simulation of the eigen modes of the blades.

As mentioned above, the actuation voltage of the piezo patches as well as the strain gage signals are transmitted into and out of the rotating system through a slip ring, respectively. The slip ring mounted on the engine shaft behind the two compressor stages is shown in Fig. 4.

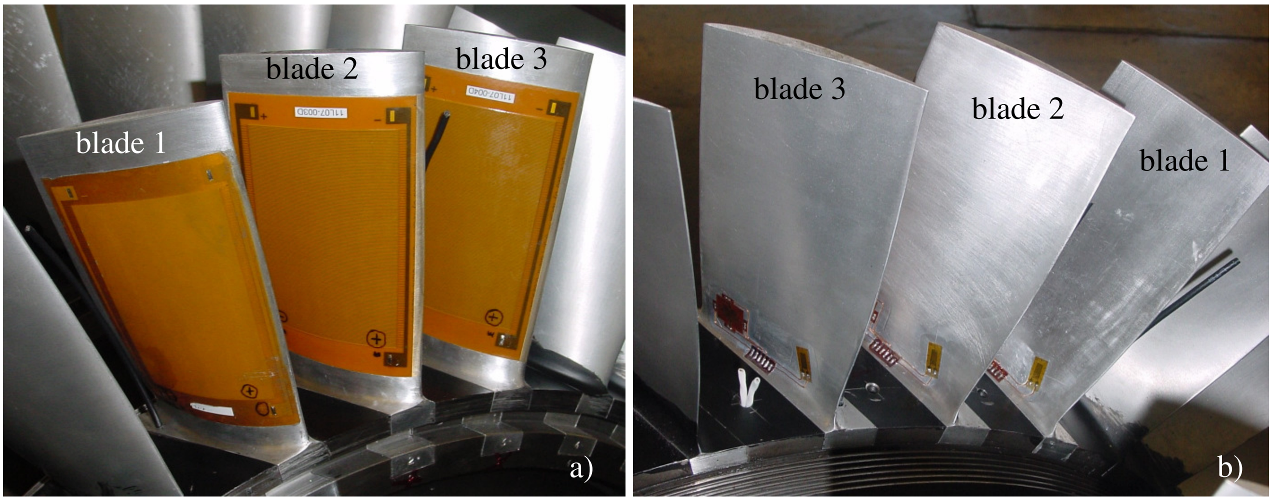


Figure 2: Piezo-electric actuators at the blade suction side (a) and the strain gages at the blade pressure side (b), none of them wired yet

Abbreviations in Fig. 3:

sg 1f:	strain gage on blade 1, full bridge
sg 2f:	strain gage on blade 2, full bridge
sg 2q:	strain gage on blade 2, quarter bridge
sg 3q:	strain gage on blade 3, quarter bridge

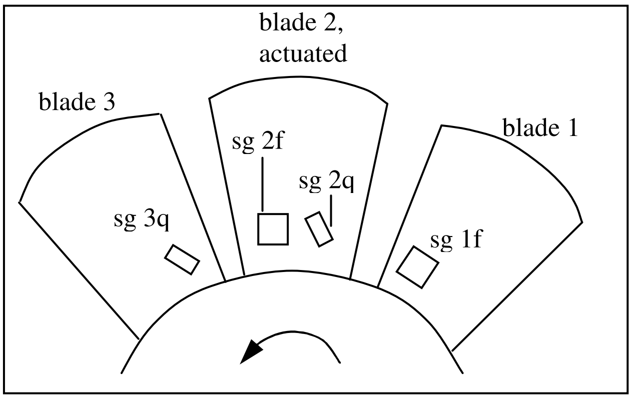


Figure 3: Sketch of used strain gages and numbering of signals

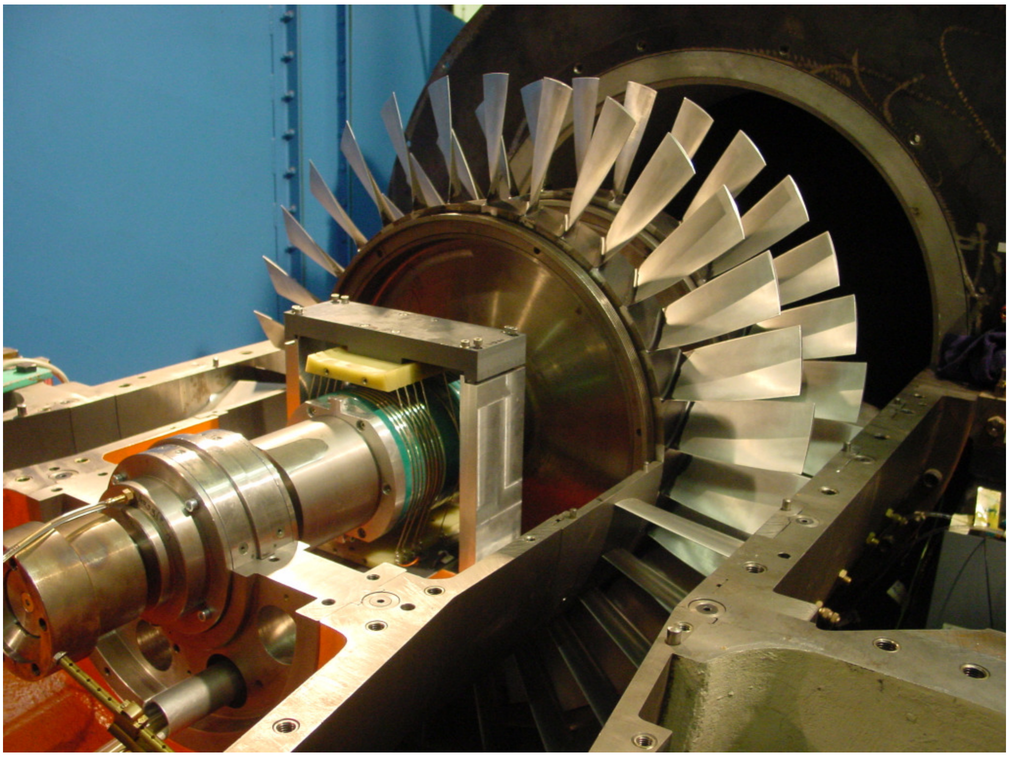


Figure 4:

Test rig with slip ring, perspective from the pressure side against the flow direction

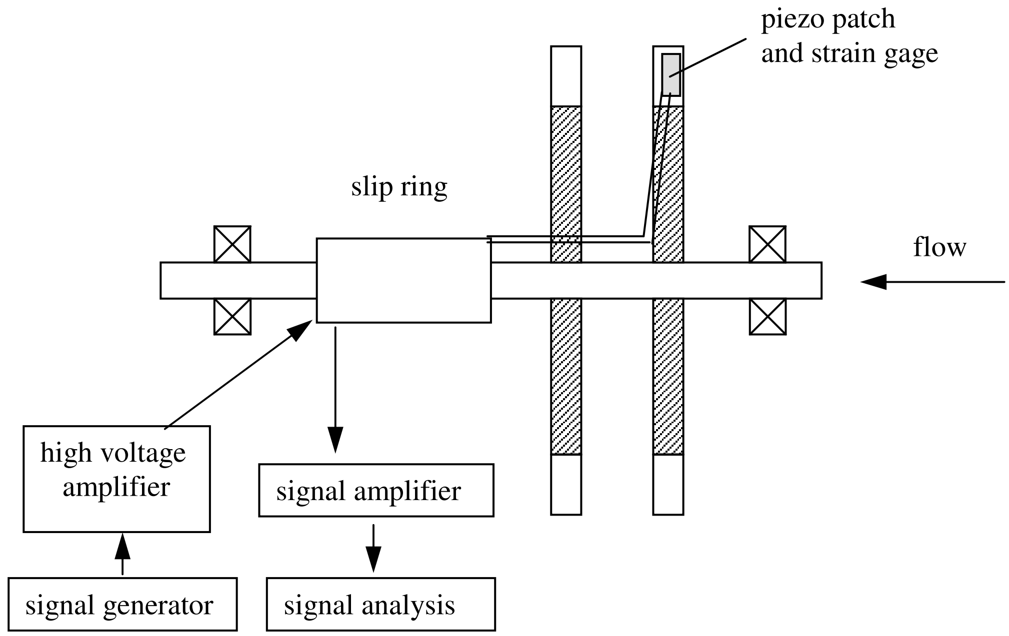


Figure 5: Sketch of measuring chain

Due to the limited space on the engine shaft in the already existing machine, this was the choice for transmitting actuation voltage and strain gage signals. The last paragraph of this paper (Future Investigations) describes the implementation in Cologne.

For generating the supply voltage, first a conventional signal generator is used. By means of this generator a sinusoidal signal oscillating between 0 and 10V is generated (stepped sine). In a second step the sinusoidal signal is generated by the control unit which has been developed within the scope of this project. In a subsequent high voltage amplifier this signal is amplified by factor 100. Hence, the piezo-electric actuators are charged by a voltage oscillating between approximately 0 and 1000V. The frequency of this high voltage signal can be adjusted manually. Figure 5 shows a sketch of the measuring chain. The wires are well shielded in the stationary and the rotating system to minimize the effects of electrical stray.

The signals of the strain gages are amplified and gathered by a PCI card (containing an anti-aliasing filter) and analyzed via LabView and Matlab codes. For the analysis, the relative amplitudes and the phase shift between actuating voltage and strain gage signals are calculated, and Fast Fourier Transformations (including Hanning windowing) are run.

RESULTS AND DISCUSSION

Results

Before investigating the rotating machine, experimental bench tests were done with a single rotor blade clamped in a vise. Amongst others, these tests showed that gluing a piezo patch to the blade surface did not change the blade eigen modes significantly. Moreover, during these bench tests a strain gage which was not located on the vibrating blade and which was not strained, therefore, was held next to the actuated piezo patch. This strain gage did not show an increased noise level or electrical stray. Therefore, it was assumed that the signal obtained from the strain gage on the blade is due to the mechanical vibration of the blade and not due to electrical stray.

The attempt to actuate the first torsion eigen mode was not successful using this type of piezo actuator and hence was not pursued any further.

In the rotating machine the signals obtained from the two full bridges (sg_1f and sg_2f) are very clear containing only little electrical stray. The signals from the two quarter bridges (sg_2q and sg_3q) however are superposed by a stray signal of about 4.6 kHz the origin of which needs more investigation. Due to this fact all figures shown in this paper are restricted to the two signals obtained from the full bridges (sg_1f and sg_2f).

The above mentioned control system was bench marked in the test rig in Hannover. The actual implementation of the control system will take place in the test rig in Cologne. Hence, the results presented here only show the un-controlled system with the middle rotor blade (blade 2) actuated by a preset amplitude of the supply voltage. For all figures shown here the actuation voltage was a sinusoidal signal oscillating roughly between 0 and 1000V. Due to the fact that the amplification factor of the high voltage amplifier is a function of the frequency, most of the results presented here are normalized by the actuation voltage. The relative amplitudes stand for

(amplitude of strain gage signal[V])/(amplitude of actuation voltage[V])

and the phase shift denotes

(phase angle of actuation voltage)-(phase angle of strain gage signal).

The absolute value of the strain gage signals are depending on a number of factors such as the exact position and orientation of the strain gage on the blade and the amplification factor of the signal amplifier. Therefore, the absolute strain gage signals are not sufficient for judging the response level of the blade. For that purpose a calibration becomes necessary to get a correlation between actuation voltage and displacement of the blade. This calibration has not been done yet but is planned in future investigations.

So far, the tests have been done with a non-rotating machine (n = 0 rev/min) and at two rotational speeds of n = 2000 rev/min and about 3000 rev/min, respectively, in order to distinguish between the influence of electrical stray fields, mechanical unbalance, and the actual vibration frequencies caused by the actuation. Figure 6 shows the amplitudes of the two full bridges relative to the actuation voltage of the piezo-electric actuator measured for 25 different frequencies. The phase angle of the signal obtained from sg_2f is related to the phase angle of the actuating voltage, since the piezo actuator and strain gage sg_2f are located on the same rotor blade. The phase angle of the signal obtained from sg_1f on blade 1 is related to the phase angle of sg_2f on blade 2 to get the direct information of the blade motion of blades 1 and 2 relative to each other.

n = 0 rev/min

The amplitudes of the actuated blade 2 show a clear peak at 312 Hz (1^{st} flap eigen frequency, 1F) whereas the propagation of vibrations towards blade 1 seems to be low. The two curves of phase shift show the expected jump around the eigen frequency.

n = 2000 rev/min

The vibration amplitudes measured at n = 2000 rev/min show that the 1F eigen frequency of blade 1 and 2 moves to 330 Hz (see Fig. 7). Moreover, the maximum amplitude of blade 2 increases from about 0.22×10^{-2} at n = 0 rev/min to roughly 0.3×10^{-2} for n = 2000 rev/min.

The fact that apart from the eigen frequency blade 1 seems to show stronger vibrations than blade 2 is due to a stronger interfering signal in the voltage obtained from sg_1q. This misleading impression will be avoided by using a low pass filter for future investigations.

about 3000 rev/min

In Fig. 8 the same trends are shown for about 3000 rev/min. The peak in the relative amplitude of sg_2f moves to 342 Hz and a little more than 0.4×10^{-2}. Here again, the somewhat higher relative amplitude of sg_1f at frequencies except the eigen frequency is due to a stronger interfering signal and has no physical relevance.

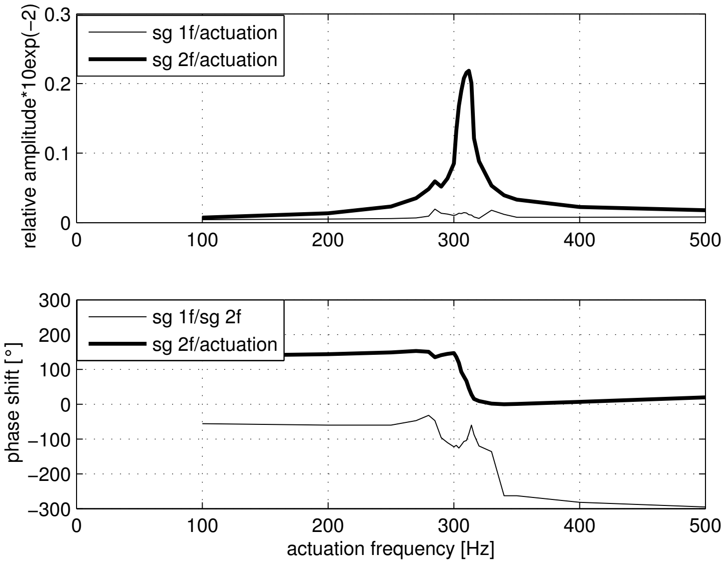


Figure 6: Amplitudes and phase shift in non-rotating machine, n = 0 rev/min

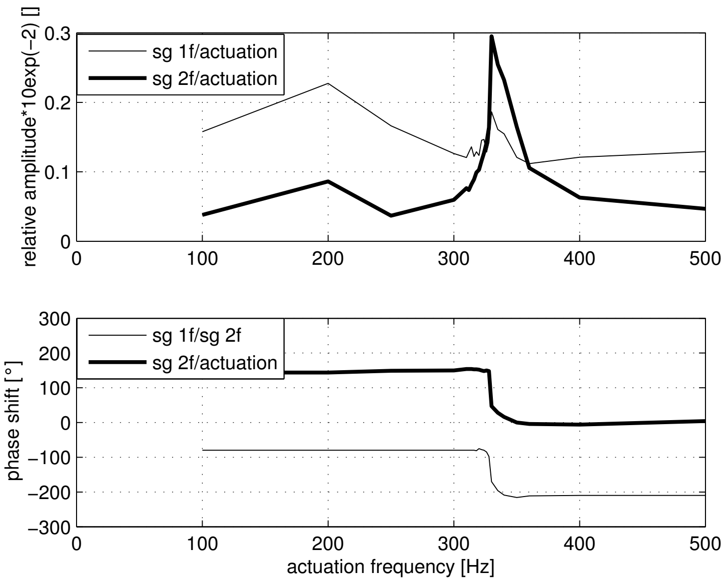


Figure 7: Amplitudes and phase shift at reduced rotational speed, n = 2000 rev/min

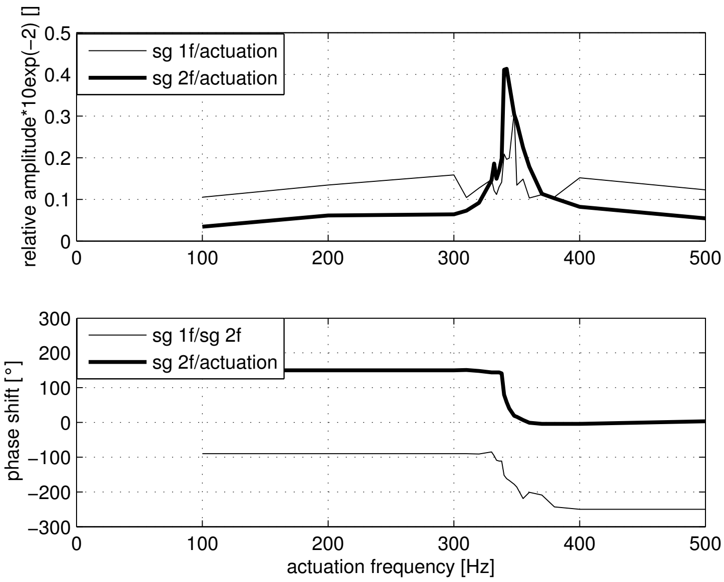


Figure 8: Amplitudes and phase shift at about 3000 rev/min

As an example for the signal quality, Fig. 9 shows the time signals of sg_1f and sg_2f at the blade eigen frequency (1F) of the actuated blade 2. The FFTs of the two signals (see Fig. 10) show minor peaks at about 48 Hz (and multiples) which result from the rotating frequency. The major peak of both signals can be seen at 342 Hz (1F).

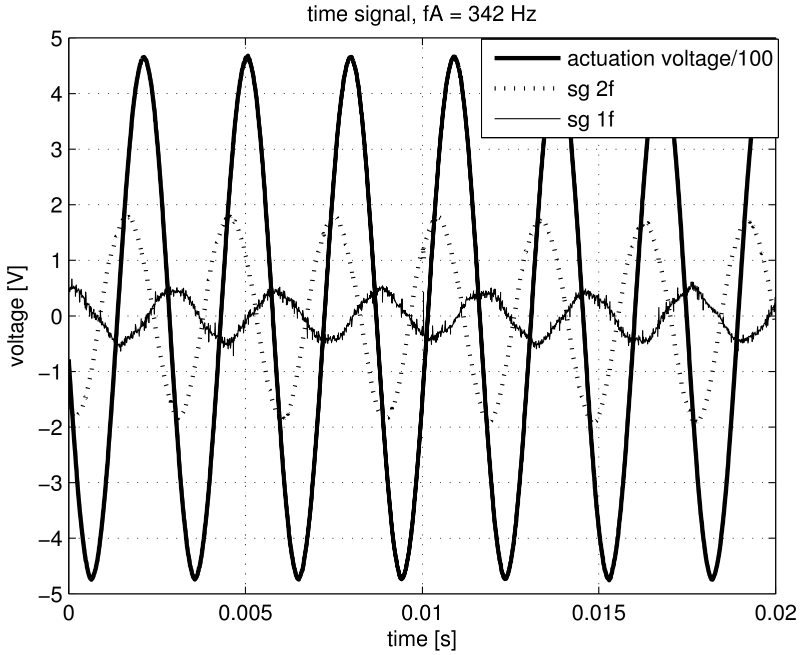


Figure 9: Time signals at about 3000 rev/min

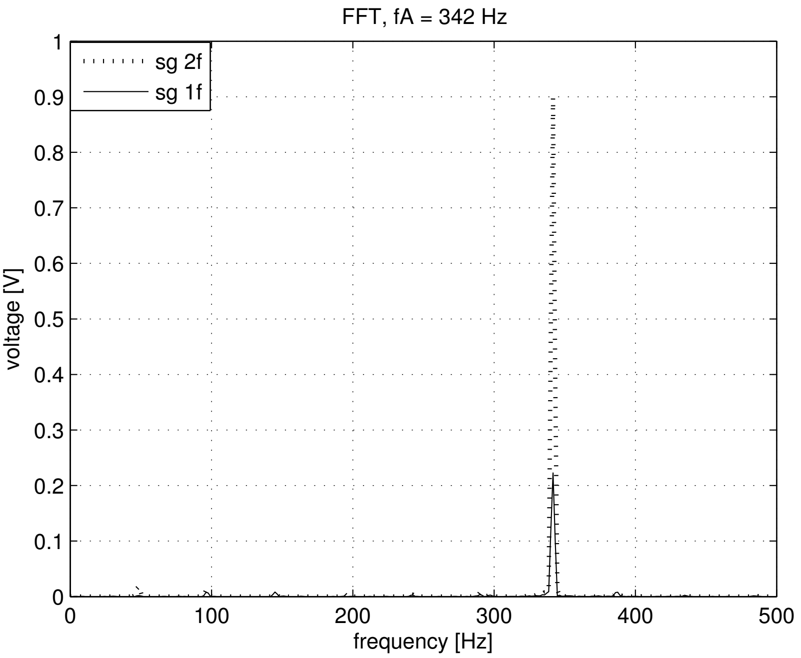


Figure 10: FFTs at about 3000 rev/min

Discussion

Comparing the three test series shows that the frequency of the first flap eigen mode moves from 312 Hz (for non-rotating machine) to 330 Hz (n = 2000 rev/min) and 342 Hz (about 3000 rev/min) while the maximum amplitude is increasing also. The experimental bench tests described before have shown that eigen frequency and amplitude highly depend on the clamping condition of the blades. Hence, it is assumed that the shift in eigen frequency and the increase in amplitude are caused by a changing clamping condition in the machine when increasing the rotational speed as previously reported e.g. by Kielb and Abhari (2003) in their turbine study.

For all three test series the blade response shows multiple response peaks which indicate mistuning of the blade and disk assembly due to small geometric differences between the individual blades.

CONCLUSIONS

The described investigations show that it is possible to actuate compressor rotor blades by piezo-electric actuators in a real machine. The actuation is strong enough to obtain strain gages signals of good quality. First tests using the control unit show that the vibration amplitude can be controlled. Based on this proof of principle implementation, a number of future investigations in the described machine in Hannover as well as in the compressor rig in Cologne are planned.

FUTURE INVESTIGATIONS

The tests in the Hannover rig include the following:

(1) The tests described in this paper will be re-run using a low-pass filter, in order to obtain four signals of good quality giving information about the physical phase shift between the three rotor blades. Furthermore, the unsteady wall pressure right behind the first rotor stage will be measured. This will give a first hint about the aerodynamic pressure fluctuations caused by the blade vibrations. To get a relation between actuation signal in volts and physical vibration amplitude in µm, a laser vibrometer will be used for calibration. Finally, it is planned to actuate all three blades – in phase and out of phase.

(2) The actuation and control system will be used for actuating five blades of a transonic fan blisk operated in the M2VP-wind tunnel of the DLR-Institute of Propulsion Technology. In order to obtain the aerodynamic damping under operating conditions, the five blades are equipped with piezo-electric actuators, strain gages, and unsteady pressure sensors. The five blades will be controlled independently creating a so-called travelling wave mode, i.e. constant amplitudes and a given phase shift between the blades (interblade phase angle). The vibration induced unsteady pressure distribution and the blade vibration will be measured. Comparing this data will yield the aerodynamic damping.

To avoid capacitive cross-talk at the slip ring (which possibly is an issue in the Hannover test rig), in Cologne there will be two separate transmitting systems located at the two opposite ends of the rotor shaft.

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