GPPS-CH-2020-0075

EFFECTS OF VARYING INLET GUIDE VANE PATTERN ON AERODYNAMIC MISTUNING IN TRANSONIC COMPRESSORS

Daniel Franke
Technical University of Darmstadt
Institute of Gas Turbines and
Aerospace Propulsion
Darmstadt, Germany
compressor@glr.tu-darmstadt.de

Jonas Foret
Technical University of Darmstadt
Institute of Gas Turbines and
Aerospace Propulsion
Darmstadt, Germany

Fabian Klausmann
Heinz-Peter Schiffer
Technical University of Darmstadt
Institute of Gas Turbines and
Aerospace Propulsion
Darmstadt, Germany

Thomas Giersch
Sven Schrape
Rolls-Royce Deutschland Ltd & Co KG
Dahlewitz, Germany

ABSTRACT

This paper investigates the effects of a 1.5-stage transonic axial compressor with applied aerodynamic mistuning via circumferential variation of the inlet guide vane stagger angle. The experiments were carried out at the Darmstadt Transonic Compressor at Technical University of Darmstadt. The test rig represents a modern front stage high-pressure compressor of a jet engine. Using extensive steady and unsteady instrumentation, the global compressor characteristics and performance as well as the transient behavior, focusing on aerodynamic and aeromechanic phenomena, are analyzed.

In order to suppress non-synchronous blade vibration of the BLISK rotor, triggered by modal pattern of aerodynamic disturbances, asymmetric rotor inflow conditions are applied via circumferential pattern of varying pre-swirl, given by the inlet guide vane stagger angle. The experimental study provides proof of concept and snapshot measurements regarding global performance as well as steady and unsteady aerodynamic effects. Comparing different aerodynamic mistuning pattern, the influence on performance, circumferential flow variation, synchronous vibration as well as the behavior during stall inception varies. For all configurations, the circumferential variation of the rotors operating condition is evident within the radial and circumferential aerodynamics as well as unsteady flow at the rotor tip. These findings provide similarities to effects presented for asymmetric casing geometries and related mistuning capabilities.

INTRODUCTION

Future aero engine design and optimization is driven by environmental awareness, combining higher efficiency with a reduction in exhaust and noise emissions. Besides performance, the engines' thrust-to-weight-ratio is a key factor. The compressor, as a multi-stage setup, significantly affects the overall engine weight and length, hence, resulting in requirements for more lightweight and efficient designs along with maintained power capabilities.

In order to accomplish these goals, different measures are feasible. Enabling the required overall pressure ratio with a minimum number of compressor stages, as well as combining a reduced blade count and thinner blades in a blade integrated disk (BLISK) rotor, is a fundamental design challenge.

These measures result in an increased aerodynamic stage and blade loading, and in conjunction with negligible structural damping of modern BLISK rotors lead to a severe vulnerability to blade vibration, especially in the vicinity to the stability limits.

Several studies (e.g. Baumgartner et al. (1995), Dodds and Vahdati (2014)) highlighted the problems with blade vibration in front stages. The limiting phenomena can be linked to aerodynamic disturbances and have been characterized as radial vortices, as for example presented by Yamada et al. (2013) and Pullan et al. (2015). Numerical studies of Kielb et al. (2003) and Besem et al. (2016) identify vortices as a source of non-synchronous

blade vibration (NSV). Measurements by Brandstetter et al. (2018) describe the origin of those aerodynamic disturbances, particularly radial vortices, and the interaction with blade vibration, stating that radial vortices can trigger NSV. Conducting experimental investigations, Holzinger (2017) describes coupling conditions for aerodynamic and structural propagating waves. The link between aerodynamic waves and vibration pattern of a BLISK can also be shown in numerical investigations, for example by Möller et al. (2017) as well as Möller and Schiffer (2020).

Several countermeasures to suppress NSV have already been published, including the generalized option of intended mistuning, hence features with defined non-uniformities. On the one hand, mistuning can be applied to the structure, for example by varying the blade eigenfrequencies. This has positive effects on the aeromechanical stability behavior, but also might be subjected to drawbacks with respect to forced response. There are several papers studying the mistuning effects, as for example given by Figaschewsky et al. (2017) or in the survey by Ewins (1991). On the other hand, aerodynamic asymmetries can also be considered both, within the rotor, e.g. by varying blade pitch or stagger angles, and adjacent components. Jüngst et al. (2018) and Jüngst (2019) presented non-axisymmetric rotor casing geometries enabling a novel approach of aerodynamic mistuning. Large-scale stationary asymmetric features, such as eccentric tip clearance and partial casing treatments, show the potential to reduce non-synchronous blade vibrations.

The study presented in this paper applies the concept of large circumferential variations to the inflow conditions of the compressor rotor by circumferential pattern of varying inlet guide vane stagger angles. Considering different aerodynamic mistuning configurations, the experiments provide proof of concept as well as snapshots regarding global performance effects. The study includes the influence on the rotor operating conditions due to circumferential and radial flow variation, synchronous blade vibration as well as the behavior during stall inception, both aerodynamically and aeromechanically.

In general, the investigation provides proof of concept for an applicable approach to reduce non-synchronous blade vibration of modern BLISK rotors in front stage compressors, especially during transient part speed operation.

EXPERIMENTAL SETUP AND METHODOLOGY

The experimental investigations were conducted at the Darmstadt Transonic Compressor test rig, as shown in Figure 1. It is equipped with extensive steady and unsteady instrumentation, and the modular design enables easy and quick variation of the stage components.

Test Rig and Instrumentation

The compressor represents a modern 1.5-stage high pressure front stage compressor with variable inlet guide vanes (VIGV) and a BLISK rotor, designed by Rolls-Royce Deutschland. The rig is electrically driven by an 800 kW drive, enabling shaft speeds of up to 20,000 rpm. The compressor stage setup was used within several prior investigations, for example Brandstetter et al. (2018) and Jüngst (2019). The rotor has a forward swept blade design, is tip critical and suffers blade vibration at the stability limit. The available experimental investigations are supplemented by numerical studies (e.g. Möller et al. (2017)), showing a good agreement.

Figure 1 Transonic Compressor Darmstadt

To monitor the compressor operation and determine its performance, the rig is equipped with different measurement systems (see Figure 2) within several measurement sections. For example, this includes combined total pressure and total temperature rakes at the stage inlet and exit, in order to derive the compressors characteristics, as well as stator leading edge instrumentation (LE-A & LE-B) and circumferentially distributed static pressure taps.

Besides conventional performance instrumentation, the compressor is equipped with extensive unsteady instrumentation, including sensors for aerodynamic and blade vibration measurements. The blade vibrations are measured via strain gauges (SG) on the rotor blades, optimized to resolve different blade eigenmodes within the rotating frame of reference. These measurements are supplemented by a tip timing system, installed in the stationary frame of reference. The unsteady flow phenomena at the blade tip region are resolved by a variety of unsteady wall pressure transducers (WPT) in the stationary frame of reference. The 70 miniaturized sensors are flush-mounted in axial arrays as well as several circumferential positions. The axial arrays cover the whole axial blade chord length and an upstream extension, in order to resolve secondary flow features. The circumferentially distributed sensors are located at the blade leading edge, enabling spectral analysis of aerodynamics and structural vibration.

Figure 2 Instrumentation

The conducted analysis is based on comprehensive experimental investigations, including measurements at multiple constant speed lines with a variety of pre-swirl conditions, due to a VIGV angle variation. To enable transient measurements at the stability limit, the compressor can be continuously throttled to operate along a certain speed line. Additionally, varying steady state measurements at defined operating points, such as near choke (NC), peak efficiency (PE) and near stall (NS), are carried out.

Test Objectives

In order to provide the inhomogeneous inflow conditions, enabling varying aerodynamic mistuning configurations, the inlet guide vanes are set in circumferential pattern, therefore, the stagger angle of the VIGV is varied circumferentially. In prior investigations regarding aerodynamic mistuning, e.g. eccentric tip gaps or partial casing treatments as analyzed by Jüngst (2019), large scale features show sufficient effects on NSV. Hence, 180° sectors of different stagger angles with and without transition regions are used for the presented study. As shown in Figure 3, the following four different configurations are considered:

- **nom**: 360° nominal uniform VIGV setup
- **am2**: +/- 25% VIGV full range circumferential pattern without transition section
- **am1**: +/- 12.5% VIGV full range circumferential pattern without transition section
- **am1t**: +/- 12.5% VIGV full range circumferential pattern with transition section

The circumferential variation (schematically illustrated by blue-white sections) is given in percentage VIGV full operating range. The mean stagger angle, pre-swirl respectively, of the mistuning configuration match the nominal. Due to the inflow in-homogeneity and limited clocking capabilities, the instrumentation, both steady and unsteady, captures only snap-shots at distinct annulus positions or small sections (compare Figure 2).

Figure 3 Aerodynamic Mistuning Configurations

RESULTS AND DISCUSSION

Based on the comprehensive measurements, global trends can be derived, showing effects on performance and steady aerodynamics, as well as the behavior at the stability limit. From the aeromechanical point of view, both the non-synchronous and synchronous blade vibrations are affected.

Global Trends

In the nominal stage setup, the investigated BLISK rotor suffers 1T-NSV at the vicinity of its stability limit, as for example shown by Holzinger (2017) and Möller et al. (2017).

The applied aerodynamic mistuning due to an asymmetric rotor inlet flow field by VIGV pattern provides capabilities to reduce non-synchronous blade vibration. The normalized 1T blade vibration amplitudes during stall inception, measured by different strain gauges on several blades, for the nominal and mistuning cases are shown in Figure 4.

Figure 4 Normalized Blade Vibration Amplitudes during Stall Inception

The advantages of the VIGV mistuning can be depicted for both, part speed and design speed operating conditions. Compared to the nominal case, all VIGV mistuning configurations reduce the blade vibration amplitudes to below 30%.

Nevertheless, it does not necessarily lead to increased aerodynamic operating ranges. In general, the stability limits are reached at similar mass flows, even though some speed lines show slightly increased or reduced mass flows at the stall boundary. For some part speed characteristics with am2 VIGV, the compressor can be throttled even further during operation in stall without suffering increased blade vibration. Those operating conditions indicate a partial annulus rotating stall.

These findings indicate the capabilities to suppress the fluid-structure-coupling at the stability limit.

Since the aerodynamic boundary conditions of the rotor vary with respect to the nominal case, the aeroelastic advantages are accompanied by impacts on steady aerodynamics and performance.

Steady Aerodynamics

In order to determine the influence of VIGV mistuning on the compressor stage performance, steady state measurements have been performed to derive the characteristics, as shown in Figure 5. Besides the nominal case as well as am2 and am1, the speed lines for nominal VIGV with the corresponding maximum stagger angle variations due to the mistuning are considered for design speed operating conditions. The annulus-averaged performance of the compressor is affected by the VIGV mistuning, even though no quantitative analysis is possible due to missing full-circumferential clocking of the varying inflow and instrumentation. Qualitatively, the speed lines with aerodynamic mistuning are shifted to reduced pressure ratios as well as mass flows, depending on the degree of mistuning. The overall operating range, as indicated by the separated operating points of stall onset, is only slightly affected. For detailed analysis of the effects on efficiency as well as the circumferential distribution of operating points and the corresponding characteristics, full-circumferential performance traverses are necessary within future work. Based on that, theoretical explanation attempts, for example the parallel compressor model, described by Pearson and Mc Kenzie (1959), or global comparison to other non-uniform features will be enabled.

The characteristics indicate an influence of the aerodynamic mistuning on the rotor performance, hence steady aerodynamics.

Figure 5 Compressor Characteristics (qualitatively, no full-circumferential resolving performance traverses)

In order to get a first impression, the rotor exit flow field is analyzed, using the stator leading edge instrumentation, located in both VIGV mistuning sections (LE-A & LE-B, Figure 2). Figure 6 shows the differences in the radial total pressure rise distribution of the rotor with respect to the nominal case at near stall operating conditions. For both mistuning configurations, a variation can clearly be perceived, both radially and circumferentially. The trend is similar for am1 and am2, with increased effects due to higher degrees of mistuning. For am2, the circumferential variation in the tip section is more than 5% and the radial variation up to 8%. This indicates varying radial loading and potential mass flow redistribution.

Figure 6 Varying Radial Total Pressure Rise at Rotor Exit

In order to determine the origin of the above-mentioned trends, the circumferential variations of the VIGV mistuning can be characterized by the static pressure distribution at the rotor inlet. Figure 7 visualizes the circumferential static pressure distribution at the casing with respect to the dynamic inlet pressure for the mistuning configurations at design speed and exemplary at one part speed operating point.

The two transition regions of the aerodynamic mistuning are clearly visible, even though they differ in shape and intensity. At the transition of opened to closed VIGV, a steep pressure drop occurs with varying intensity (up to 10%). For am2, the circumferential variation is even up to 15%. The mistuning configuration with “smoothed” transition (am1t) shows a significant effect on the circumferential variation as well as the shape at the transitions. In comparison to design speed, the part speed operating conditions with adjusted VIGV schedule result in degraded circumferential variation and transitions.

The varying static pressure distribution at the rotor inlet casing indicates a circumferential variation of axial flow velocities, hence mass flow as well as changed

annulus averaged flow through the compressor. Similar results of circumferentially varying axial velocities due to steady 180 degree inlet distortions have been reported by Hynes and Greitzer (1987), even though the transition between the sectors seem to be magnified by the VIGV mistuning.

The circumferential variation affects the incidence of the rotor, especially at the tip, thus results in varying loading conditions of the BLISK. In order to get a better understanding on the resulting effects, the unsteady aerodynamics at the rotor tip are analyzed.

Figure 7 Circumferential Static Pressure Variation at the Rotor Inlet Casing

Unsteady Aerodynamics

The extensive unsteady instrumentation of the rotor casing segment, using time-resolved pressure transducers, provides the unsteady pressure field at different circumferential casing positions. Figure 8 shows the ensemble averaged flow field at the rotor tip for the nominal case and am2 as well as different operating points along the design speed line.

For the nominal case, the expected varying loading as well as the detaching shock and its interaction with the tip gap leakage flow is visible while throttling towards the stability limit. At near stall, the shock (compare dotted line) is detached by about 40% blade pitch, as already presented by Brandstetter et al. (2016) for the same rotor.

For the mistuning cases, the flow field varies for the two circumferential sections (given by the casing position of the WPT arrays). At all operating points along the speed line, the blade tip loading is higher in the vicinity of the transition sector from closed to opened VIGV (264° WPT position). Furthermore, the circumferential variation of the shock position is about 20% blade pitch. This is in good agreement with the findings by Hah (1998), stating a similar shock oscillation for inlet distortions, and thus underline the similarities. For the other mistuning configurations, the trends are the same, however minimized.

Figure 8 Circumferential Variation of Unsteady Pressure Field at Rotor Blade Tip

Around the annulus, some parts of the compressor are operating in a flow regime that can be referred to as stable, while others are unsteady and thus potentially affect the stability, both aerodynamically and aeromechanically.

Figure 9 shows the unsteady pressure fields for am2 at operating conditions close to the stability limit (same WPT array positions as in Figure 8). Additionally, the deviation from the periodic pressure field is derived by subtracting the local ensemble average based on prior revolutions. Hence, the unsteadiness of the flow or aerodynamic disturbances can be detected and characterized with respect to the dynamic pressure at stage inlet, as shown by Brandstetter et al. (2018).

Figure 9 Unsteady Flow Field at Rotor Blade Tip for am2 at Near Stall Operating Conditions

Figure 10 WPT Spectra at Different Casing Positions for am2

As already mentioned, the distinct difference in flow field and thus the corresponding blade loading can be depicted from the unsteady pressure field. For the WPT measurement position close to the transition between closed and opened VIGV, small low pressure spots can be located, indicating aerodynamic disturbances such as radial vortices. This finding is supported by the highly disturbed pressure deviation plot, with high passage to passage variations. In contrast, only slight variations occur at the other WPT measurement location. Even though large disturbances exist in some parts of the annulus, the global operation remains stable. These findings are supported by smaller WPT arrays at intermediate casing positions with respect to those shown in Figure 9 (not shown for brevity).

In order to determine the various operating conditions in the entire annulus, spectra of the throttling process from steady near stall to rotating stall (indicated by revolution 0) are shown in Figure 10. The spectra of eight circumferentially distributed sensors at the blade leading edge are plotted for am2. For a wide annulus sector, no

disturbances or even distinct peaks, indicating explicit aerodynamic wave pattern or blade vibration, are visible prior to rotating stall. Compared to the nominal case (see Figure 11), which is suffering a distinct fluid-structure-coupling with an aerodynamic wave pattern of 13 cells interacting with a 1T-ND+8 BLISK mode, the VIGV mistuning seems to suppress the underlying excitation, thus the blade vibration.

Within the transition sector between closed and opened VIGV, broadband scattered disturbances, in literature referred to as “rotating instabilities”, occur and vanish again. These disturbances are spread between EO10 and EO19 and are already visible 1000 revolutions prior to stall, with increasing intensity during stall inception. These findings provide strong similarities to the investigations of the aeroelastic effects due to asymmetric tip clearance, as shown by Jüngst et al. (2018), although some differences are noticeable. The eccentric tip gap also causes various circumferential operating conditions, with low pre-stall disturbances and no fluid-structure interaction for some parts of the annulus.

Figure 11 SG and WPT spectra of throttling process for nom at design speed, Jüngst (2019)

But for a large angle of the annulus, the aerodynamic disturbances excite blade vibration and even synchronize with 1T or even higher eigenmodes, depending on the local tip gap and the resulting aerodynamics.

The positive effects of mistuning features on non-synchronous blade vibration at the stability limit might most likely be accompanied with drawbacks in forced response vibrations.

Synchronous Vibration

The measurements show a distinct influence on the synchronous vibration of the rotor due to forced excitation, noticeable at all fundamental engine order crossings within the operating range. Figure 12 shows the maximum normalized blade vibration amplitudes for the first three fundamental eigenmodes for the nominal case and am2 with different VIGV schedules. The increased response is visible for all present modes, but particularly for M1, representing the first flap. Here, a strong dependency of the VIGV schedule can be emphasized, with up to seven times the amplitudes, whereas higher modes are less affected.

In general, the synchronous vibration amplitudes depend on the mistuning configuration, each forcing a different inlet flow pattern, as shown in Figure 7, on the rotor. Smoothed transition sections enable a reduction of resonance amplitude (not shown for brevity).

Figure 12 Normalized Blade Vibration Amplitudes for Forced Response Excitation

CONCLUSIONS

The effects of aerodynamic mistuning due to inlet guide vane stagger angle pattern on a transonic axial compressor rotor are investigated. The experiments are performed at the 1.5-stage Darmstadt Transonic Compressor test rig, using extensive steady and unsteady instrumentation. As common for modern BLISK rotors in compressor front stages, the nominal case suffers a distinct fluid-structure-coupling of an aerodynamic wave pattern interacting with a 1T BLISK mode, leading to non-synchronous blade vibration at the vicinity of the stability limit. To influence or even suppress these corresponding vibrational issues, mistuning features are state-of-the-art, mostly focusing on structural mistuning.

The presented study provides proof of concept measurements with respect to the comprehensive possibilities of aerodynamic mistuning. Therefore, different non-uniform VIGV configurations are tested, showing the ability to suppress the fluid-structure-coupling of modern BLISK rotors for part speed operation and design conditions. Due to the varying inlet conditions, distinct aerodynamic effects, such as circumferential mass flow redistribution, occur. Thus, the rotor encounters varying operating and loading conditions, preventing the formation of aerodynamic modal waves, which are able to excite and couple aeromechanically with the rotor. These benefits in stability behavior are accompanied by drawbacks in global aerodynamic performance as well as forced response excitation.

In future research, the detailed effects on compressor performance as well as unsteady aerodynamics have to be considered, for example by full annulus resolving traverses and probe measurements. Based on that, general correlations and design strategies could be derived and compared to basic models, such as the parallel compressor model. For potential engine application, the multi-stage effects should be considered as well.

ACKNOWLEDGMENTS

The authors gratefully acknowledge the financial support and thank Rolls-Royce Deutschland Ltd & Co KG for the permission to publish this paper. The support and helpful discussions with the TU Darmstadt compressor team, especially Maximilian Jüngst, is gratefully acknowledged. The authors would also like to thank for the continuous support of the compressor aerodynamic and aeroelasticity group of Rolls-Royce Deutschland, in particular Bernd Becker and Frank Heinichen.

NOMENCLATURE

Θ	circumferential angle difference
Δp	pressure difference
1T	first torsional blade eigenmode
am	aerodynamic mistuning configuration
BLISK	blade integrated disk
EO	engine order

LE	leading edge
M1, M2, M3	blade eigenmode 1, 2, 3
N100	design speed
NC	near choke operating conditions
ND	nodal diameter
nom	nominal compressor stage setup
NS	near stall operating conditions
NSV	non-synchronous blade vibration
$p_{dyn,in}$	dynamic pressure at stage inlet
PE	peak efficiency operating conditions
p_s	static pressure
SG	strain gauge
VIGV	variable inlet guide vanes
WPT	wall pressure transducer

REFERENCES

- Baumgartner, M., Kameier, F. and Hourmouziadis, J., (1995). Non-engine order blade vibration in a high pressure compressor. In *Proceedings of 12th International Symposium on Airbreathing Engines*.
- Besem F. M., Kamrass, J. D., Thomas, J. P., Tang, D. and Kielb, R. E., (2016). Vortex-induced vibration and frequency lock-in of an airfoil at high angles of attack. In *Journal of Fluids Engineering* 2016, volume 138, 011204.
- Brandstetter, C., Wartzek, F., Werner, J., Schiffer, H.-P. and Heinichen, F., (2016). Unsteady Measurements of Periodic Effects in a Transonic Compressor with Casing Treatments. *Journal of Turbomachinery*, Vol. 138, 051007.
- Brandstetter, C., Jüngst, M. and Schiffer, H.-P. (2018). Measurements of radial vortices, spill forward and vortex breakdown in a transonic compressor. *Journal of Turbomachinery*, 2018. doi:10.1115/1.4039053.
- Figaschewsky, F., Kühnborn, A., Beirow, B., Nipkau, J., Giersch, T. and Power, B., (2017). Design and analysis of an intentional mistuning experiment reducing flutter susceptibility and minimizing forced response of a jet engine fan. In *Proceedings of ASME Turbo Expo 2017: Turbomachinery Technical Conference and Exposition*.
- Dodds, J. and Vahdati, M., (2014). Rotating stall observations in a high speed compressor: Part 1 – experimental study. In *Proceedings of ASME Turbo Expo 2014: Turbine Technical Conference and Exposition*.
- Ewins, D.J., (1991). The effects of blade mistuning on vibration response – a survey. *IFTOMM 4th International Conference on Rotordynamics*.
- Franke, D., Jüngst, M., Möller, D., Schiffer, H.-P. and Giersch, T. (2020). Influence of pre-swirl, rotor speed and blade count on aeroelastic coupling mechanisms during stall inception of a transonic compressor. In *Proceedings of the ASME Turbo Expo 2020: Turbomachinery Technical Conference and Exposition*.
- Hah C., (1998). Effects of Inlet Distortion on the Flow Field in a Transonic Compressor Rotor, *Journal of Turbomachinery*. 1998, 120(2): 233-246.
- Holzinger, F. (2017). *Coupling of tip leakage flow and blade vibration in transonic compressors – Mechanism and countermeasures*. PhD thesis, Technical University of Darmstadt.
- Hynes, T. P. and Greitzer, E. M., (1987). A Method for Assessing Effects of Circumferential Flow Distortion on Compressor Stability. *Journal of Turbomachinery* 109, pp. 371-379.
- Jüngst, M., Franke, D., Schiffer, H.-P. and Giersch, T., (2018). Aeroelastic effects in a transonic compressor with nonaxisymmetric tip clearance. In *Proceedings of GPPS Forum 18 Global Power and Propulsion Society Montreal*.
- Jüngst, M., (2019). *The transonic compressor with non-uniform tip clearance: Effects on aerodynamics and aeroelasticity*. PhD thesis, Technical University of Darmstadt.
- Kielb, R. E., Barter, J. W., Thomas, J. P. and Hall, K. C., (2003). Blade excitation by aerodynamic instabilities: A compressor blade study. In *Proceedings of the ASME Turbo Expo 2003: Power of Land, Sea and Air*
- Möller, D., Jüngst, M., Holzinger, F., Brandstetter, C., Schiffer, H.-P. and Leichtfuss, S., (2017). Mechanism of nonsynchronous blade vibration in a transonic compressor rig. *Journal of Turbomachinery* 2017, vol. 139, pages 011002-1-10, doi: 10.1115/1.4034029.
- Möller, D. and Schiffer, H.-P., (2020). On the mechanism of spike stall inception and near stall non-synchronous vibration in an axial compressor. In *Proceedings of ASME Expo 2020: Turbomachinery Technical Conference and Exposition*.
- H. Pearson and A. B. McKenzie, (1959). Wakes in axial compressors. *Journal of the Royal Aeronautical Society*.
- Pullan G., Young, A. M., Day, I. J., Greitzer, E. M. and Spakovszky, Z. S., (2015). Origins and structure of spike-type rotating stall. *Journal of Turbomachinery* 2015, vol. 137, 051007, doi: 10.1115/1.4028494.
- Yamada, K., Kikuta, H., Iwakiri, K.-I., Furukawa, M. and Gunjishima, S., (2013). An explanation for flow features of spike-type stall inception in an axial compressor rotor. *Journal of Turbomachinery* 2013, vol. 135