EFFECT ON THE AERODYNAMIC FORCING IN AN AXIAL TURBINE STAGE ON VARYING STATOR BLADE SHAPE

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Abstract

The aim of this paper is to describe the procedure and the outcome of a study performed in order understand the dependence of the forced response of the rotor blades in a turbine stage on a selection of different shape modifications, namely lean, sweep and a number of their combination, of the stator blades by means of computational fluid dynamics (CFD. The adopted criterion to compare the different analyzed cases is the generalized force obtained by the projection of the unsteady aerodynamic forces onto the rotor blade natural mode shapes derived by means of a finite element analysis. A mode excitability study was first conducted by putting emphasis on the lower frequencies modes widely considered as the most critical ones in real applications, in an attempt to identify and characterize their behavior with respect to the stator blade shapes. A forced response assessment exercise was then performed for the specific case under exam in order to discover the optimal configuration, intended as with the smallest maximum value of the generalized forces for each possible condition of resonance of the machine. Steady simulations were used to assess the permenance of the working conditions, as expressed by the mass flow, the pressure ratio, the loading and the flow coefficients, within reasonable limits from the baseline case for the entire set of studied variations

1 Aerodynamic Forcing in Turbomachinery

Aerodynamic forcing is an aeroelastic phenomenon which arises from the propagation of flow unsteadiness both upstream and downstream of a blade row, due to the interaction between the stationary and rotating components in turbomachinery. While the forced response of a blade is determined to be problematic, the dynamic response with respect to frequency and/or amplitude, a variation in the upstream blade shape may be considered in order to alter the frequency and amplitude of the downstream blade row response. CFD allows us to model this flow in the passage, between modified blades and blade rows and data of interest can be extracted and analyzed with respect to the unsteady pressures that contributes to the blade dynamic response and yields an indication of the integrity of the component. Knowledge of the expected excitation nature and blade Eigen frequencies will determine the position of the operating points.

To analyze the forced response to the unsteady flow conditions, the parameters of interest are identified: the sources of flow unsteadiness which can be considered as either due to gust. It is considered an integral engine order excitation in turbomachinery. The term ‘gust’ is used in turbomachine to identify periodic flow non-uniformities. Two gust components, captured in the adopted numerical model, are the potential gust due to physical obstacles in the flow, namely the stator and rotor blade rows, and vertical gusts due to wakes. Entropic gusts due to shocks were not present in this study, the reason being the employed boundary conditions not allowing for supersonic flow speeds. These two forms of gust are characterized mainly by static pressure variations and velocity gradients along with vorticity respectively. The flow deficit due to the wakes and the potential effect both lead to unsteady forces which, depending on the duration and frequency, can lead to either HCF or LCF. The forced response due to the rotor passing frequency and the natural blade frequency is best understood by the use of a Campbell diagram, a plot of frequency versus rotor speed on which known integral engine excitations (engine orders) and Eigen frequencies can be displayed to highlight areas of concern, such as resonant crossings and potential excitation frequencies. The structural features of the blades will determine the modal parameters of the structure. This study, involving parametric blade shape changes will produce changes to the modal properties of the stage and thus the vibration characteristics will be affected.

1.1 Identification of Flow Features of Prime Interest

The features of interest for the study of forced response must be identified which are the sources of flow unsteadiness. The three major flow mechanisms of interest, as listed in Table 1, are considered for the forced response case**.**

Flow feature of interest	Origin/mechanism	Associated variables
Wake interaction	Boundary layer development due to viscous effects Shedding of boundary layers at TE downstream	Total pressure Static pressure Static temperature Local Mach number
Potential Field interaction	Pressure distribution due to flow obstruction (blade) propagated upstream and downstream	Static pressure Static temperature Local Mach number
Vortex interaction	Tip vortices Horseshoe vortices Passage cross-flow	Total pressure Static pressure Static temperature Local Mach number Velocity gradients Vorticity

Table 1: Flow features of interest along with associated variable [1]

The influence of the wake and potential with respect to local Mach number is reported in literature to be such that **at** low Mach numbers the potential field power is reduced exponentially and the influence is much smaller than due to the wake, whereas for high Mach numbers (approaching sonic conditions) the potential field effect can be one order of magnitude greater than the wake effect. In order to simplify the numerical model some simplification of the geometry were used such as disregarding the tip clearance and small fillets and using a smooth surface assumption rather than applying a surface roughness, which actually does affect transition and thus the boundary layer development, yet no specifics on it were available to allow a confident employ.

2 Geometry Modification

The modification is based on re-stacking the chosen blade profile, which translates into displacing the profile sections in various directions span wise, such as to achieve leaned, swept or bowed blades while preserving the profile section. Table 2 shows the various cases considered in the study.

	Theta (degrees)	Fraction Axial Chord
Lean	4.5	0.1532
	9	0.3084
	13.5	0.4674
Sweep	3	0.1020
	6	0.2046
	9	0.3084

Table 2: Implemented parameters for the different case

Pure lean cases and combination of lean and stator blades backward sweep is considered for the study as shown in the Fig 1 & Fig 2.

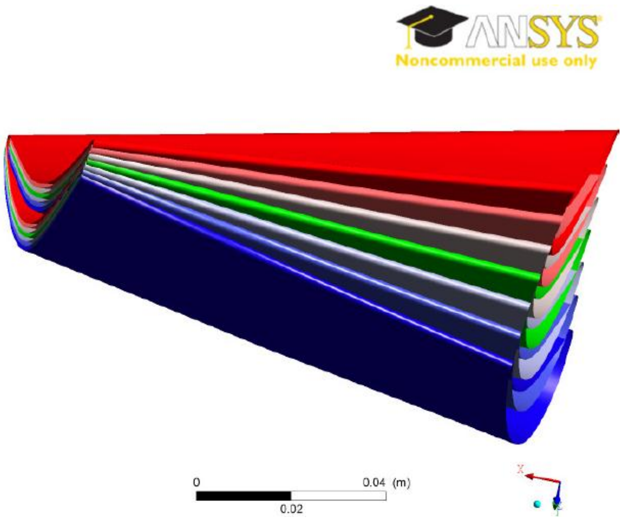

Fig. 1: Pure lean stator blades geometry: from blue to red from -13.5° to 13.5° lean angle, tip on the left hand side, and hub on the right hand side.

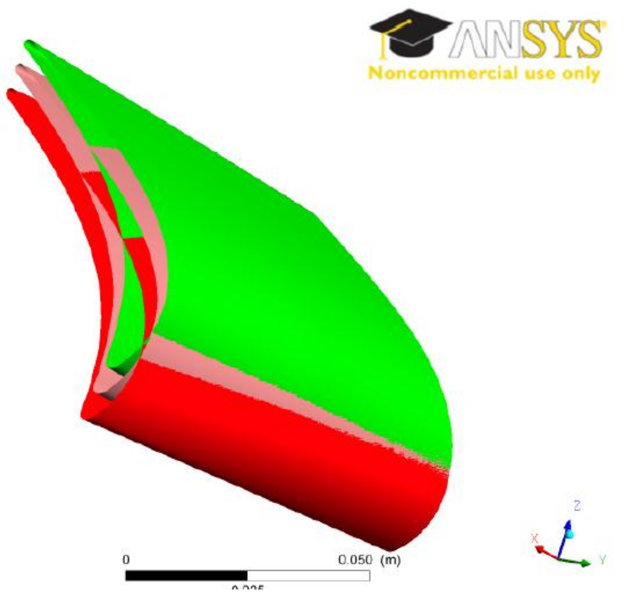

Fig. 2: Combination of -4.5° lean and backward sweep of 6 ° (red) to 3 ° (pink), hub on the right hand side, hub on the right hand side

3 Simulation Setup

The study procedure is divided in several steps in order to reach the results categorically. The analysis procedure adopted was as follows:

- Geometrical definition of stator blades
- Steady state simulation
- Transient simulations
- Modal Analysis
- Generalized Forces

Steady state simulations are a necessary starting point for transient simulations in order to check whether the working conditions were the same or at least comparable for the entire set of cases from the baseline case. Two steady simulations have been performed for all geometries, using different interface conditions between stator and rotor: the mixing plane interface (stage) to assess the operating conditions and performance followed by frozen rotor interface, providing results to be used as initial conditions for the transient runs.

Transient simulations are then required for analyses where the flow field variables are changing in the domain with time, in order to calculate the required physical quantities at every time step.

The mode shapes for the rotor blade **are** calculated by means of finite element analysis and used to produce the Campbell diagram from which the potential dangerous crossings for the rotor blade structure could be identified.

The **generalized** forces **are** calculated from the distributed unsteady forces on the blade surface obtained from the transient simulations. Fourier transformation **is** implemented to transpose into the frequency domain the forces which **are** subsequently projected onto the different mode shapes. **Normalization i**s required to allow the comparison between the different cases**.**

3.1 Mesh Generation & Boundary Conditions

A mesh is generated in ANSYS-Turbo Grid with 500000 nodes for the whole set of 5 modeled blades. Fig 3 & Fig 4 show the details of the generated mesh for the stator blades. The rotor blades meshes are created the same way [4].

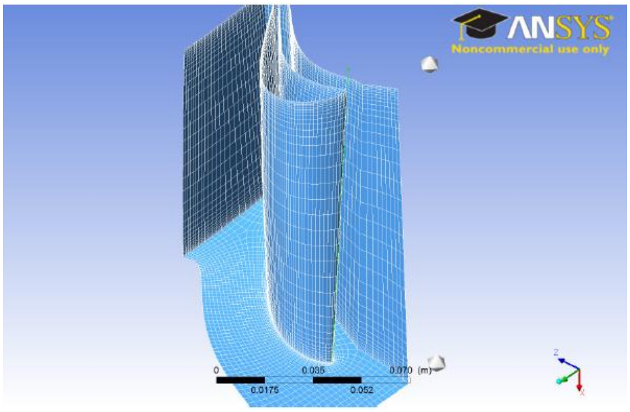

Fig 3: Span-wise distribution of nodes for the stator blade

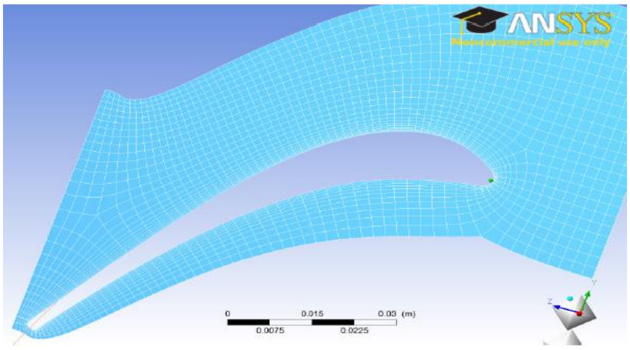

Fig 4: Generated mesh over the blade passage

3.2 Boundary Conditions

For the pre-processing part a predetermined profile boundary condition is used. The inlet pressure distribution imported from Matlab is seen to account for the boundary layer. The Turbo Chart facility also enables the user to inspect and verify that the expected profile has been adopted for the model. The adopted parameters for the flow domain are summarized in Table 3.

Analysis Parameter	Employed settings
Turbulence Model	K-Epsilon
Advection Scheme	High Resolution
Length Scale Option	Aggressive
Interface	Stage Interface

Table 3: Employed settings relative to the used model

The detailed boundary conditions with the respective values are shown at the end of the paper (Annex A)

3.3 Transient Simulations

In addition to all the settings discussed for the steady simulations setup and which apply as well to the transient one, it has to be considered that in a stage, transient simulation calculations are performed according to the rotation of the rotor blades. The rotor displacement, as expressed by the rotational speed, is discretized in time, meaning that a chosen total interval of time of the physical time simulated T is divided into intervals, the time steps, at each of which the position of the blade is varied. At each time step a number of calculations, named coefficient loop iterations, is performed in order to find the physical quantities distribution in the domain.

The length t of the time step itself is fundamental in determining the quality of the results: the shorter it is, the closer to a continuous-in-time status the simulation is. The obvious drawback of too short a time step is the consequential heavy increase in the simulation time. As best practice in a stage transient simulation, a time step length is chosen, generally from one fortieth to one tenth, from a higher to a lower accuracy, respectively, of the smallest of the periods employed by a rotor blade to cover a pitch in the stationary frame of reference or by a stator blade to cover a pitch in the relative frame of reference.

The Frame Change/Mixing model is the most critical parameter in a transient analysis of a stator rotor stage. This model is set to Transient Rotor Stator. In this approach the transient relative motion between the components on each side of the GGI connection is simulated. It ultimately accounts for all interaction effects between components that are in relative motion to each other. The interface position is updated each time step, as the relative position of the grids on each side of the interface changes [3].

This represents the only solution to study the aerodynamic forcing in a turbine stage which relies on transient effects such as those of the wakes of upstream blade rows onto downstream ones as can be seen in Fig 5 in which the total pressure in the stationary frame of reference is depicted in order to show the propagation of the wakes from the stator to the rotor blades.

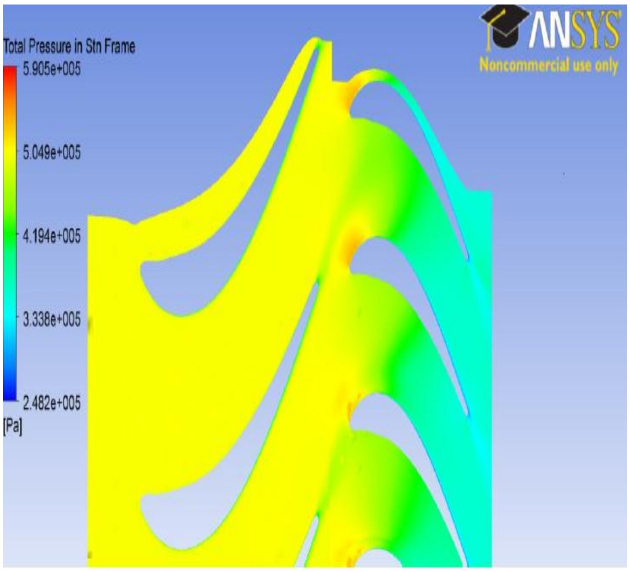

Fig 5: Total pressure in the stationary frame of reference

Furthermore, other parameters for the transient simulations such as Time Step, Max. Coefficient loops, sample time and residual target were fixed (Annex B).

4. Post Processing

Since the three force components acting on each node of the blade surface extracted from ANSYS CFX, as well as the nodes coordinates themselves, refer to the stationary coordinate systems, in order to perform the analysis described above it is necessary at each time step to rotate them back to the original position, in such a way that it becomes equivalent to having the blade fixed in the space with the forces acting on it varying with time. This has been done with a Matlab script. Fig 6 shows an example of the mesh at the initial time step, and at a generic time step, which is rotated with the devised script to perfectly match the initial one. The visible circles in different colors depend only on the angle view.

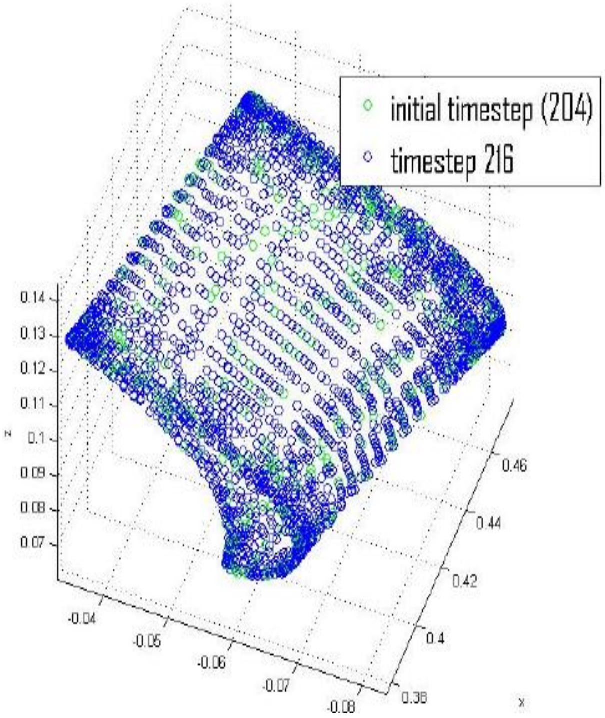

Fig 6: Meshes at the initial time step, at a generic time step and the superimposed

4.1 Modal Analysis

The rotor blade geometry was created and modified first in ICEM CFD, using original geometry files from ANSYS CFX, then imported into Solid Works to make refinements, and finally into ANSYS FEM with a resulting topology of 1 volume, 6 surfaces, 8 lines and 8 key points. Using element type Solid 45, an eight-node 3D solid isoperimetric element, and by controlling the number of divisions of the 8 lines, a hexahedral mapped mesh could be generated as shown in Figure 24. The mesh is more refined at the leading edge and trailing edge of the blade in the hope of resembling the mesh in CFD as much as possible for future mapping. The total number of elements in this mesh is 5600 (final mesh).

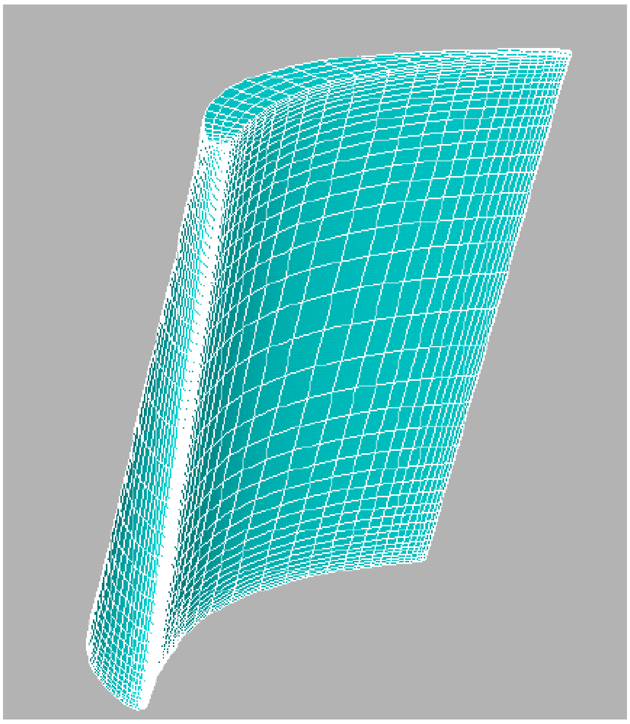

Fig 7: Final mesh for FEM modal analysis of rotor blades

Eigen-frequencies are calculated at different rotational speed (Annex C). The Campbell diagram (Fig 8) is hence obtained to spot the dangerous crossings. Only the first 8 modal solutions are extracted. It can be seen that the Eigen-frequencies of the modes display a small increasing trend with an increasing rotational speed, due to the stiffening effect which arises with the centrifugal loads. In this analysis the working range is defined as -60% to 15% of the working point rotational speed (4100 rpm), shown as blue vertical lines. Also to consider the forcing for the present case the 54 and 108 engine order lines were drawn, where 54 is the number of stator blades in the stage.

For calculating the generalized forces and identifying mode excitability, the following results were exported from ANSYS FEM to ANSYS CFD: list of nodal coordinates and nodal displacements in the x, y and z directions (rotational speed at 4100 rpm).

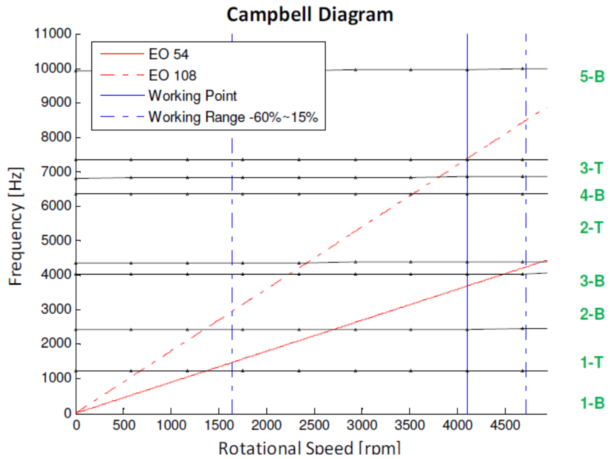

Fig 8: Campbell Diagram for the rotor blade

The nodal displacements of the 8 modes, named according to the common practice of identifying the dominant deformation, as in bending B, torsion T or edgewise E and the order of the mode as it appears from lower to higher natural frequencies and related to the neutral point/lines recognized in the shapes, are displayed in Annex D. It can be shown that only bending and torsion modes are found up to the first 8 considered modes, since it can be expected that edge-wise modes often have much higher frequencies. Afterwards, a mapping of the nodal displacements from the FEM environment to the CFD domain was performed using Matlab scripts (Fig 9)

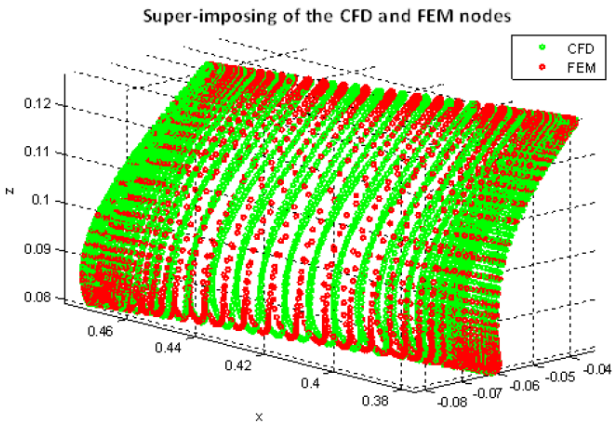

Fig 9: Superimposed CFD & FEM nodes

4.2 Generalized Forces

The parameter employed to assess the forced response phenomenon in the present work is the *generalized force*, a concept which derives from the Principle of Virtual Works and the idea of generalized coordinates, and from the result of structural dynamics according to which any vibratory motion of a structure can be described in terms of the its mode shapes (modal analysis). In brief, given a mode shape of the structure, the rotor blade in this case, the generalized force expresses the energy transferred from the aerodynamic forces to the structure vibrating in that mode. The higher this value, the more energy is transferred and the higher the amplitude of the resulting oscillations. A comparison among the generalized forces of different modes allows therefore identifying the most critical ones with respect to the others.

For the present study, the calculation of the generalized forces was performed by extracting from the CFD results the *three components* of the unsteady aerodynamic force acting on each node and projecting them onto the mode shapes of the rotor blade, obtained by a modal analysis. The mathematical tool used to handle a force which is a function of time is the *Fourier transform*, which allows the transformation of a function $f(t)$, whose domain is time t, into a complex function called the frequency domain representation $F(w)$, where w is the frequency. The principle which allows analyses in the frequency domain is the fact that it holds that $f(t)$ is equivalent to the sum of oscillatory functions, one for each value of w, with the frequency being w itself and amplitude and phase being the magnitude and phase of $F(w)$, respectively. For a function periodic in time as the aerodynamic forcing in a turbomachine, the values of w are discrete and all multiples of the so called base frequency w_0 which is equal to $2\pi/T$, where T is the period of the function.

5. Results & Discussion

In order to look into the steady features and the effects of leaning/sweeping of the different cases, the loadings of the rotor at tip, hub and mid-span are analyzed. First the pure lean cases were plotted in and then the combined lean and sweep cases. Only the curves defining the borders of the area within which all the curves were located are plotted. it is observed that for both plots at the 50% span all cases converge and there is not a significant difference along the stream-wise direction of the blade. For the pure lean cases at 10% span, there is some deviation of the loading near the leading edge of the blade at both pressure side and suction side, and all the curves of the 7 cases (including the baseline case) are to some extent linearly distributed versus leaning angles [5].

The maximum loading corresponds to a forward 13.5 lean case, and the minimum loading corresponds to a backward 13.5 lean case. All the other case curves lie in between (which were not shown here for clarity of the graph). On the other hand, for the pure lean case at 90% span the plot shows an opposite trend, which is backward lean producing overloading effect of the blade and forward lean resulting in a discharging effect near the leading edge.

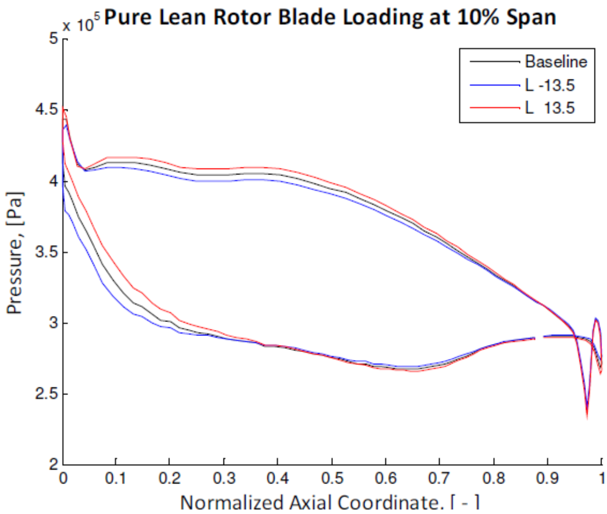

Fig 10. Blade loading at 10% span for lean cases

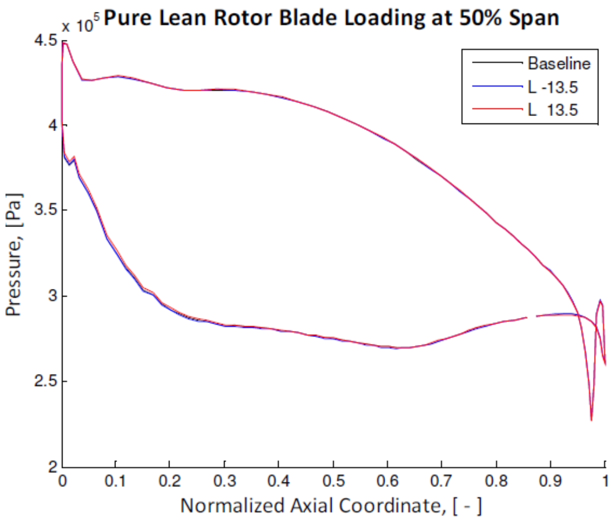

Fig 11. Blade loading at 50% span for lean cases

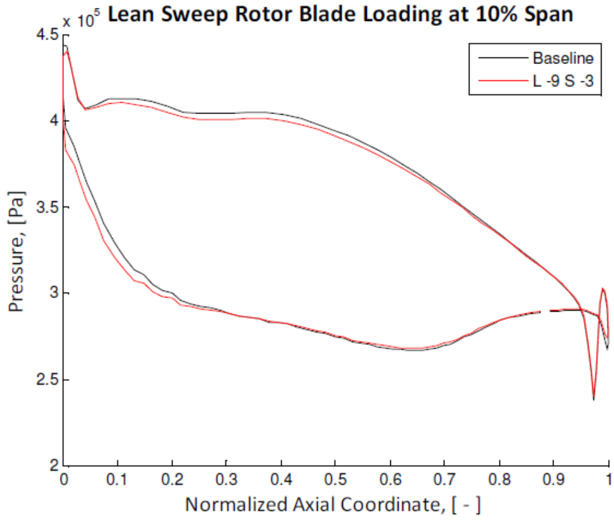

Fig 13. Blade loading at 10% span for lean sweep cases

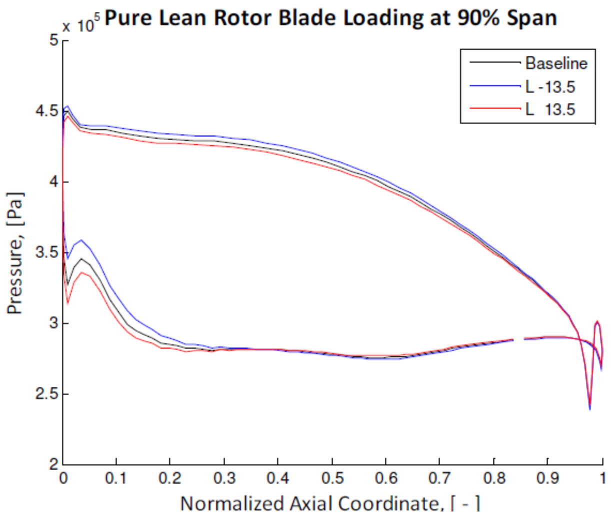

Fig 12. Blade loading at 50% span for lean cases

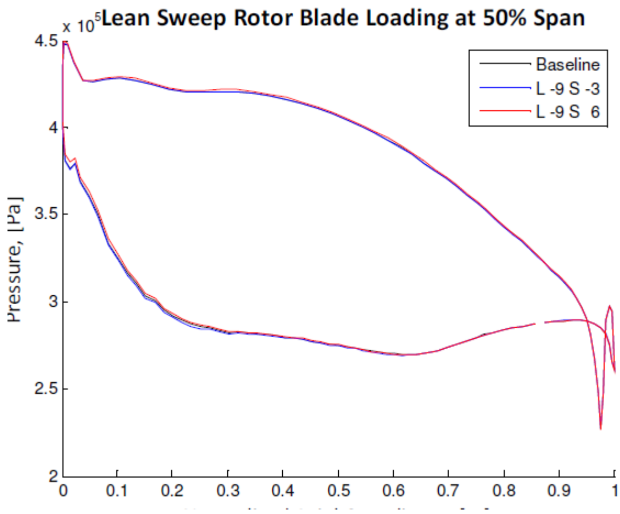

Fig 14. Blade loading at 50% span for lean sweep cases

For lean and sweep combination cases, the deviation of the loading is relatively small compared to pure lean cases because sweep modification in itself does not produce significant flow variations. It is also observed that deviation occurs at the leading edge while converging near the trailing edge. At 10% span the case with the most backward lean has the lowest loading while baseline has the highest, and at 90% the opposite is seen to be the case. This again supports the finding that the flow variations are more affected by modifications of the lean rather than sweep.

5.1 Study Case Forced Response Analysis

The starting point for the forced response study of a turbomachine is the Campbell diagram, which depicts the natural frequencies of the structure, in this case the rotor blade, as function of the rotor rotational speed, together with the engine order excitation lines. Resonance occurs where a crossing between an excitation line and a natural frequency of the structure exist inside the operating range of the machine. The Campbell diagram for the present study case, with the first eight eigen-frequencies of the rotor blade structure. The engine order lines of interest are 54 and 108, corresponding to one time and twice, respectively, the number of the upstream stator blades, which coincide as well to the first and second harmonic content of the aerodynamic forcing obtained by the Fourier transform. By observing the generalized forces of the crossing in the baseline reported in Fig 15, it is clear how the crossings of 1st and 2nd mode with the first harmonic and of the 7th mode with the second harmonic are the most critical, as they show a higher value with respect to the others. In particular, crossing of mode 7 with the engine order 104 is almost exactly on the operating line, which makes it a most dangerous one.

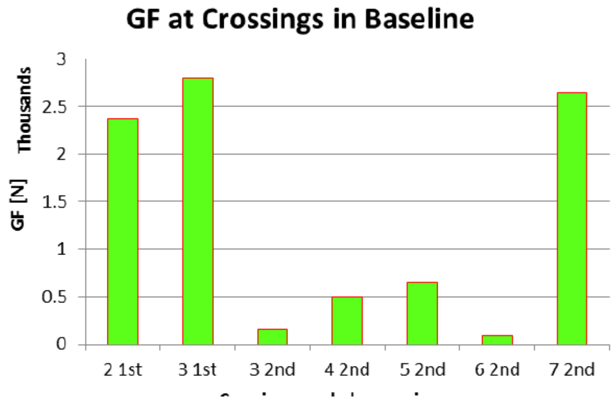

Fig 15. Generalized forces on the rotor blades for the crossings between natural and excitation frequencies in the base-line case

From Fig 16, it can be deduced, based on the requirement for the crossing with the highest generalized force magnitude for a configuration to be the lowest with respect to the others, that the case with a positive lean angle of 4.5° represents the optimal one. The choice is strongly affected by the crossing of mode 3 and 1st harmonic for the negative lean angles , as the magnitude of the corresponding generalized force grows up to about double its value in the baseline, whereas towards positive angles it is the crossing of mode 1 being decisive, as it shows a positive gradient in that direction.

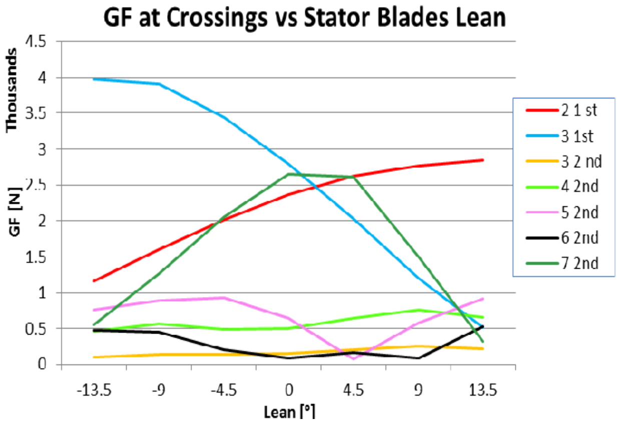

Fig 16. Generalized forces on the rotor blades for the crossings between natural and excitation frequencies as function of the stator blades lean

The generalized forces at crossings are reported for the cases where stator blades sweep variations have been combined with two negative leans variations, according to the study plan: Figure 17 and 18 show the results for combinations with a lean angle of -4.5° and -9°, respectively. It can be noted how the trend is the same for both values of the lean onto which the sweep is imposed

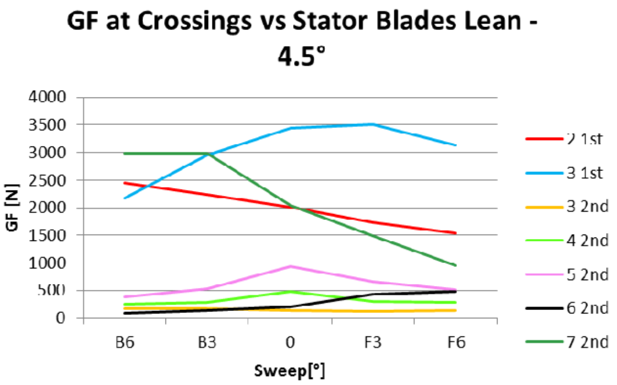

Fig 17. Generalized forces on the rotor blades for the crossings between natural and excitation frequencies for -4.5° stator blades lean angle as function of sweep

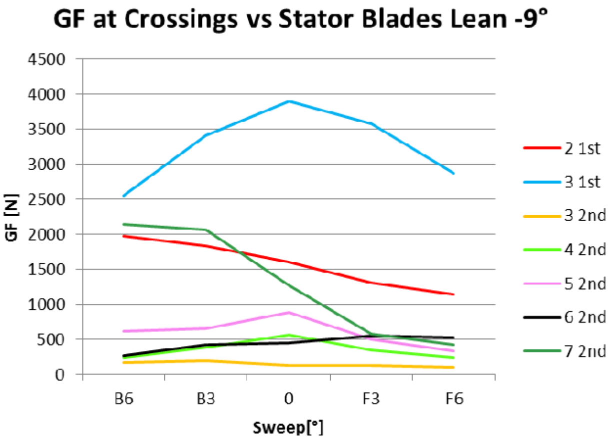

Fig 18. Generalized forces on the rotor blades for the crossings between natural and excitation frequencies for -9° stator blades lean angle as function of sweep.

The last meaningful comparison with respect to a forced response analysis is among all the studied cases, in order to determine which one is the most recommendable, again as the one in which the highest generalized force has the lowest magnitude among all of them. From Figure 19 and Table 4, where the results are plotted and reported, it is clear how from the baseline, actually only three cases show a smaller amplitude for the highest point, which are L4.5, L9 and L-9 B6, and it is the latter which can be considered the best case, although the highest amplitude for the crossing of mode 3 with the 1st harmonic is only 9% less than the same value in the baseline, a modest value when compared to the reduction achievable by the single modes in some configurations.

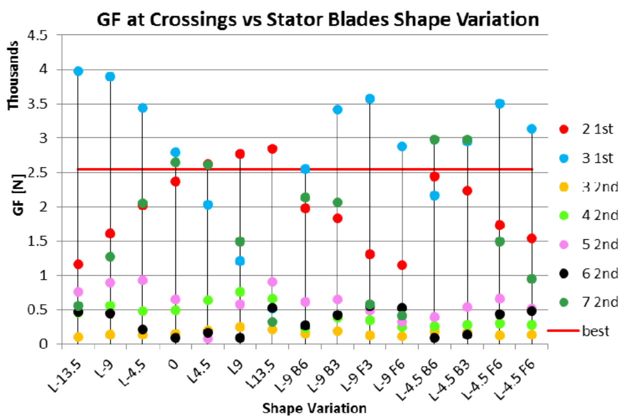

Fig 19. Generalized forces on the rotor blades for the crossings between natural and excitation frequencies as function of all the analyzed stator blades geometries

Case	Max GF [N]	Cross Max
L-13.5	3977.778	3 1st
L-9	3902.721	3 1st
L-4.5	3444.878	3 1st
0	2797.69	3 1st
L4.5	2621.169	2 1st
L9	2771.578	2 1st
L13.5	2843.798	2 1st
L-9 B6	2546.276	3 1st
L-9 B3	3414.591	3 1st
L-9 F3	3571.565	3 1st
L-9 F6	2877.04	3 1st
L-4.5 B6	2985.946	7 2nd
L-4.5 B3	2984.548	7 2nd
L-4.5 F6	3510.633	3 1st
L-4.5 F6	3135.683	3 1st

Table 4. Max generalized force for each case and crossing which attains it; highlighted are the baseline and the cases with a smaller value than the baseline

6. Conclusion

The present work shows how computational fluid dynamics can be a powerful tool for studying the aerodynamic forcing on the blades in a turbomachine, with the aim of assessing the suitability of the different known and newly devised techniques for reducing the forced response magnitude and thus increasing the life expectancy of highly stressed components and relaxing the structural requirements in the design phase. In particular, the dependence of the excitability of the natural modes of the rotor blades, obtained by means of finite element modal analysis on the 3D shape variations of the stator blades in the AETR turbine stage, has been investigated by studying the variation with the lean and sweep of the stator blades of the normalized generalized force for the modes observed to be the most critical ones in real applications, i.e. those with lower natural frequencies, and for the first two harmonics in the forcing signals, shown to be the most significant ones in the case under exam. A more pragmatic forced response analysis has also been conducted by taking into account the actual occurrence of resonance between the excitation forces and the natural modes of the considered structure, evaluable from the occurrence of crossings of the respective curves in the machine operating range on the Campbell diagram.

From the results obtained it has been understood that:

- the operating conditions in the several analyzed cases, although not by a significant amount, differ from the base line ones in terms of loading coefficient versus flow coefficient trend, as their corresponding points wander away from the baseline operating line in a perpendicular fashion instead of moving along it; a similar trend is seen for the total pressure ratio and mass flow

- the detected general trend for the generalized force of the examined modes, the first, second and third bending and the first torsion, is that the magnitude relative to the first harmonic content is always from 2 to 5 times greater than the second harmonic one with the exception being the first bending mode, for which the two of them are of the same order of magnitude; the first harmonic is generally strongly affected by leaning in contrast with the second content whose dependence on stator blades' lean is weaker;

- the changes in the generalized force magnitude can be either generally monotonic or show different gradients for different ranges of the lean angle, with no correlation found with the mode type

- sweeping of the stator blades produce changes in the generalized forces magnitude whose magnitude is generally lower than those determined by leaning for the same angle variation

- in terms of machine forced response analysis, only three cases have shown to lead to smaller values for the highest generalized force at resonance obtained by correcting the rotational speeds to take into account where the corresponding crossing are actually localized on the Campbell diagram, with the best configuration being a combination of mild, as defined, negative lean and backward sweep of the stator blades, with an achieved reduction of about 9% from the baseline case.

The influence on the single modes considered alone was found to be much greater than the final result obtained for the particular machine under exam where the highest generalized force at resonance in the optimal configuration was 9% less than the same value in the baseline. In fact, no general rule for any turbine stage can be derived from the results of the study hereby reported. Conversely, its usefulness is in providing a general trend for the most usual critical modes despite the fact that they may have different natural frequencies or might be excited by forces generated in different settings, and an example of establishing a systematic procedure for a parametric study of the forcing in a turbomachine.

7. References

[1] Contran, A., October 2010, *Parametric Study of the Effect of Blade Shape on the Performance of Turbomachinery Cascades, Part II B: Forced Response Behavior of a Compressor Stage*, Master's Thesis, Politecnico di Torino, Italy

[2] Vogt, D.M., "Experimental Investigation of Three - Dimensional Mechanisms in Low - Pressure Turbine Flutter", PhD thesis, Royal Institute of Technology, Sweden, ISBN 91 - 7178 - 034 - 3, Stockholm, 2005

[3] ANSYS CFX Help Documentation

[4]http://www.cfdonline.com/Wiki/Best_practice_guidelines_for_turbomachinery_CFD

[5] Clark. J. P., *Design strategies to mitigate unsteady forcing*, Von Karman Institute for Fluid Dynamics Lecture Series 2008

Annex A

Boundary Conditions	
Total pressure at the inlet p_{01}	Blasius BL (500k/492k[Pa])
Static pressure at the outlet p_2	300 518 Pa
Total temperature at the inlet T_0	1073 K
Axial velocity component at inlet u_{ax}	121.6 m/s
Radial velocity component at inlet u_{rad}	0 m/s
Circumferential velocity component at inlet u_θ	82.6 m/s
Boundary layer thickness	hub 5% span, casing 10% span
Turbulence values	length 2 mm, intensity 2%
Rotational Speed	4100 rpm

Table 5: Boundary Conditions for base case

Annex B

Parameter	Setting
Time Step t_s	$T_r/16 = 1.1292\times10^{-5}$ s $(=T_s/24)$
Max Coefficient Loops	3
Total Time	$8\times T_s = 2.1680\times10^{-3}$ s
Mixing Model	*Transient Rotor/Stator*
Sample Time	$2\times t_s = 2.2584\times10^{-5}$ s, *last 2 periods*
Output Variables	*Pressure, Density, Velocity, Total Pressure in Stn Frame, Total Enthalpy in Stn Frame*
Residual Target	RMS 10^{-5}

Table 6: Transient Simulation Setup

Annex C

Ω [rpm]	f_1 [Hz]	f_2	f_3	f_4	f_5	f_6	f_7	f_8
0	1229	2436	4036	4369	6351	6818	7345	9915
585	1230	2436	4037	4369	6351	6819	7345	9916
1170	1231	2437	4037	4370	6352	6821	7346	9919
1755	1232	2438	4039	4371	6353	6824	7347	9924
2340	1234	2439	4040	4373	6355	6829	7348	9932
2925	1237	2440	4043	4376	6357	6835	7350	9941
3510	1240	2442	4045	4379	6359	6842	7353	9953
4100	1244	2444	4049	4383	6362	6851	7356	9967
4680	1249	2447	4052	4388	6366	6861	7359	9983
5265	1254	2449	4056	4393	6370	6872	7363	10001
Den.	**1-B**	**1-T**	**2-B**	**3-B**	**2-T**	**4-B**	**3-T**	**5-B**

Table 7: Eigen- Frequencies at different operating speeds

Annex D

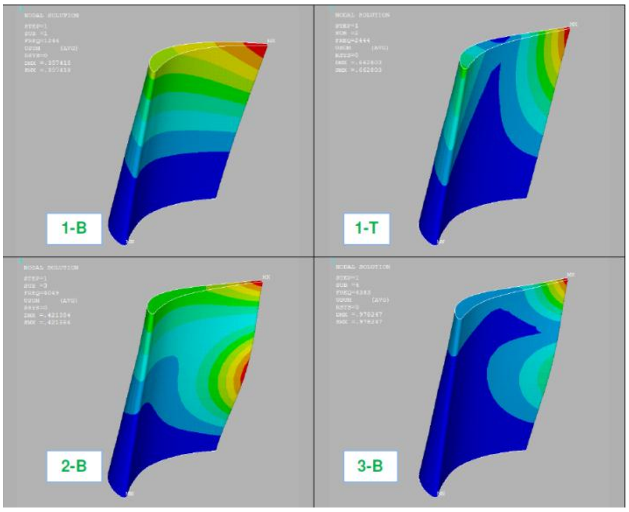

Fig 20. Nodal displacement contour plot of the 8 extracted mode shapes with denomination (first 4 modes)

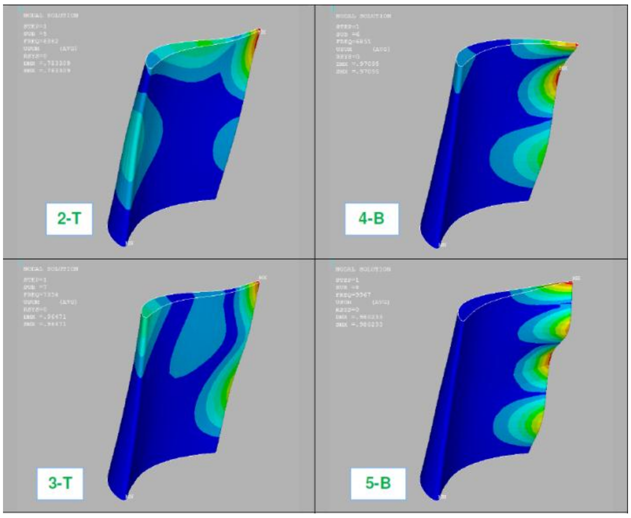

Fig 21. Nodal displacement contour plot of the 8 extracted mode shapes with denomination (last 4 modes)