SENSITIVITY OF THE AERODYNAMICS DAMPING COEFFICIENT PREDICTION TO THE TURBULENCE MODELLING CONJUGATED WITH THE VIBRATION MODE SHAPE

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ABSTRACT
The flutter corresponds to an aerodynamic loading of the structure which amplifies the natural blade vibration. In this paper, a modern design of a high pressure compressor is investigated using a time-linearized RANS solver on 2D blade to blade channel. Two operating points at part speed have been selected, the first with only small supersonic pockets and the second with the interblade channel blocked. Two vibration modes are investigated, the first torsion mode (with a nodal diameter at 2) and the first flexion mode (with a nodal diameter at 2, 4 and 6). Two different two equations turbulence models, $k-l$ and $k-\omega$ have been used to resolve the steady state. The unsteady resolution is based on the previous steady state field. Turbulent variables are calculated over time based on a $k-\omega$ turbulence model. It was found that for some mode shapes, but not for all, the work exchange between the flow and the blade presents a large disparity depending on the turbulence model used primarily in the steady calculation. This paper proposes a parametric study in terms of rotor velocities, nodal diameters and vibration mode shapes to determine which flow phenomena are sensitive to the turbulence modelling. Main results point to the effect of the shockwave motion, and its interaction with the boundary layer and its separation.

KEYWORDS
FLUTTER, HIGH PRESSURE COMPRESSOR, L-URANS, TURBULENCE
NOMENCLATURE

$N_b$ the number of compressor blade
$ND$ the nodal diameter
$P_s$ the static pressure
$V$ the velocity vector
$S$ the surface vector
$S_B$ the total blade surface
$U$ the maximal vibrating kinetic energy
$W$ the work extracted
$x_b$ the domain boundary
$\delta x$ the displacement
$\rho$ the blade material density
$\sigma$ the interblade phase angle
$\omega$ the vibration pulsation
$\Omega$ the fluid-structure contact interface

INTRODUCTION

Flutter instability can lead to the failure of compressor blades. The flutter corresponds to an aerodynamic loading of the structure which amplifies the natural blade vibration. To avoid delays in new development, low fidelity unsteady numerical methods need to be used early in the design stage to detect flutter situations. In this framework, the turbulence is modelled to perform a large set of calculations in a realistic computation time. Classic approach is to use one of the two equations turbulence models. In the best case, it is expected that all physical values calculated, including the aerodynamics damping coefficient, are not very sensitive to the chosen turbulence model.

In this paper a high pressure compressor is analysed with linearised RANS calculations on 2D blade to blade channel. The steady state turbulence variables are estimated with two classical different turbulence model, $k-l$ and $k-\omega$. The configuration database is composed of two operating conditions (the first is in majority subsonic and the other has a choked interblade channel) and two vibration modes (first torsion and flexion modes). For the flexion mode, three nodal diameters have been investigated (2,4 and 6).

This paper is not advocacy for one or the other turbulence model. The relevant fact is that for some configurations the flutter prediction is independent of the turbulence model chosen. In some other configurations, the global damping coefficient is relatively robust to the turbulence model, but a more complete analysis shows that these cases have large disparities which self-compensate. Finally, in the worst cases, the global damping coefficient shows opposite stability tendencies.

The damping coefficient is linked to the pressure fluctuation generated by the flow dynamics. The proposed analysis tracks the effects of the turbulence modelling on the flow dynamics and their effects on the work exchange.
NUMERICAL METHODS

Steady solver

ONERA elsA CFD software is used for all the steady state calculations. A Jameson convective flux (Jameson et al., 1981) with the second and fourth order dissipation coefficient is used with a backward Euler time integration scheme. The term source of the turbulent transport equations is computed from vorticity rather than from the strains tensor to avoid turbulent overproduction near the leading edge and across the shock wave. The geometry has been extracted from the 3D-structured mesh. The final 2D-mesh consists in around 20 000 points. The grid-convergence has been checked with a 2D-mesh with 80 000 points.

The position and the pattern of shockwaves are conserved between the 2D calculation and the 3D solution at the same height. To make this possible, the inlet azimuthal velocity and the outlet static pressure are adjusted. Other boundary conditions, total pressure, total energy and turbulence variables have the same values as 3D simulation.

Two models of turbulence with two equations are compared in this article for the steady state calculation, with the length scale of turbulence ($k$-l turbulence model of Smith, 1990) and with the specific rate of dissipation ($k$-$\omega$ turbulence model of Kok, 1999).

Time-linearised URANS solver

The Linearised RANS (LRANS) solver Turb’Lin is used to compute the harmonic flow around the steady state. This solver has been previously validated on transonic separated flows (Philit et al., 2012, Rendu et al. 2015). The solution is obtained in the frequency domain by solving the linear system. Spatial discretisation relies on Jameson et al.(1981) centred scheme with linearised pressure sensor.

Only the $k$-$\omega$ turbulence model of Wilcox (1988) is implemented in the linearised solver. The turbulence model is fully linearised (no frozen turbulence assumption here, Duquesne et al., 2018). Physically, that corresponds to consider that the turbulence reacts without delay to the flow fluctuations. Change of turbulent variables between steady and unsteady solvers is performed keeping the turbulent kinetic energy and the turbulent viscosity.

Aeroelasticity

The complex amplitudes of displacement $\tilde{\delta}x$ and velocity $\tilde{V}$ are imposed at each node of the blade mesh to model the blades oscillation. The steady position of the blade is chosen as the phase origin. This yields

$$\Re(\tilde{\delta}x) = 0 ; \quad \Im(\tilde{V}) = 0$$

(1)

The local work $W$ extracted by the flow from the structure is written according to the convention of Verdon, 1987.

$$W = \int_{0}^{T} \left[ -\tilde{P}s(x, t) \ast \tilde{S}(x, t) \right] \ast \tilde{V}(x, t) dt$$

(2)

where $\tilde{P}s$ is the instantaneous static pressure, $\tilde{S}$ the vector associated to the instantaneous surface, oriented towards the structure.

The damping coefficient $\zeta$ is then obtained by the integral along the blade surface of the extracted work normalised by the maximal vibrating kinetic energy of the blade $U$

$$\zeta = \frac{1}{4\pi} \frac{\int_{\Omega} W d\Omega}{U}$$

(3)
where $\Omega$ is the fluid-structure contact interface and $U$ the maximal vibrating kinetic energy of the blade. $\zeta > 0$ denotes a stable configuration; $\zeta < 0$ denotes a flutter case.

**STUDIED CONFIGURATION**

**Mode shapes**

The high pressure compressor geometry has been designed by Safran Aircraft Engines to be representative of the state of art. Due to confidentiality concerns, all the figures have a modified aspect ratio. The mode shape is a result of the mechanical finite element analysis (FEA) software SAMCEF. Two vibration modes are selected, the first torsion mode and the first flexion mode. A 2D sketch of the compressor blade positions at two instants during the vibration cycle is plotted in fig. 1. At 50% height, the blade first torsion mode corresponds mostly to a horizontal translation and the blade first flexion mode to a rotation around the trailing edge. As the reader can notice, both mechanical modes are not pure translation or rotation. Modes are superposition of a translation and a rotation with a phase shift. The classification of torsion or flexion mode refers to the main deformation of the 3D blade.

![Blade motion for the first torsion mode (left) and first flexion mode (right).](image)

Figure 1: Blade motion for the first torsion mode (left) and first flexion mode (right). Green and red lines are the blade position at t=0 and t=0.5 T. Vibration amplitude, interblade distant and airfoil aspect ratios are modified for presentation and confidentiality purposes.

The motion of adjacent blades can present a phase shift called InterBlade Phase Angle (IBPA or $\sigma$), while the frequency and the mode shape remain identical between adjacent blades. The IBPA is by convention positive when the deformation wave propagates in the same direction as the rotor speed and negative otherwise. The IBPA can be expressed in function of the nodal diameter ($ND$) and the number of blades ($N_b$) by: $\sigma = 2\pi ND/N_b$. For the torsion mode only the case with $2ND$ is investigated but for the flexion mode three nodal diameters are selected ($2, 4$ and $6 ND$).

**Global steady flow pattern**

Two operating points are selected for this paper. The first one, named OPA, is at partial speed (90 Nn). As presented on the left side in fig. 2, the flow at 50% height is subsonic almost everywhere. Only two small supersonic pockets are present on each side of the leading edge inside the interblade channel. A small flow separation zone is also present on the suction side near the trailing edge (represented in fig. 2 between points S and R).
The second operating point, named OPB, is at higher, but not at nominal speed (98 Nn), near the choke line. The steady state of OPB includes two large supersonic pockets (right side in fig. 2). The upstream supersonic region begins near the leading edge on the suction side and represents approximately a quarter of the chord. This supersonic zone extends up to the front of the leading edge of the adjacent blade in the azimuthal direction. The other supersonic zone is downstream the previous one. It chokes the interblade channel and exhibits a strong shockwave. On the suction side, the reached Mach number is strong enough to lead to the separation of the boundary layer downstream of the shockwave. The flow separation zone extends up to the trailing edge and its length represents more than one third of the chord.

Effect of turbulence modelling on steady results

For the steady state results $k-l$ or $k-\omega$, the previous global description is similar independently of the turbulence model. The goal of these simulations is to predict the flutter triggering, not the compressor efficiency, therefore the position of shockwaves need to be preserved. For OPA, the same boundary conditions have been applied for both turbulence models. For OPB, to obtain a similar shockwaves position, the outlet static pressure needs to be adjusted. Because of this adjustment, the isentropic Mach number along the blade downstream part is slightly different.

As a consequence of the turbulence model selection and the eventual modification of outlet boundary condition, the steady flow presents some localised differences. For OPA, the small supersonic pockets are modified. As presented to the left of fig. 2, the supersonic pocket on the suction side is larger with $k-l$ than with $k-\omega$ turbulence model. This effect is reversed on the pressure side, supporting a small shift of the flow incidence. For OPB, the flow upstream the shockwave is very similar (Mach number gradient, shockwave position...), but the flow separation zone downstream is different between turbulence models. As shown at the right of fig. 2, the thickness inside the inter-blade channel and the downstream extension of the flow separation are larger when the $k-\omega$ turbulence model is used.

Figure 2: Mach number from low (blue) to high (red) for OPA (left) and OPB (right) configurations with $k-\omega$ turbulence model. Black line represents the sonic line (Ma=1), S and R are for separation and reattachment points. Enlargements present the differences between $k-l$ (in blue) and $k-\omega$ (in green) results.
Effect of the steady turbulence modelling on the damping coefficient prediction

It is important to notice that only the steady state flow use two different turbulence models. This steady state is an input of the linearized calculation. The time evolution of turbulent variables is calculated only with a $k-\omega$ turbulence model. Fig. 3 presents the different values of the damping coefficient for all configurations. The damping coefficient prediction for the first torsion mode (1T) is very similar with both turbulence models for the two operating point. For the first flexion mode (1F), at OPA, the calculation based on the $k-l$ steady turbulence model predicts a lower damping coefficient with a constant gap for all NDs, and all configuration remain stable. For the flexion mode (1F) at the higher speed operating point (OPB), the calculation based on the steady state with the $k-l$ predicts again a lower damping coefficient but this time the difference increases with the ND. The coefficient can become negative for example at 6ND with the $k-l$ turbulence model (it remains always positive for the $k-\omega$ turbulence model). From these observations, the next section presents a detailed analysis of the damping coefficient from the case with the smallest difference (OPA, 1F6ND) up to the worst case (OPB, 1F6ND).

Figure 3: Damping coefficients for the different configurations versus the rotation speed. Steady flow obtained with $k-l$ (in blue) and $k-\omega$ (in green), OPA:90 Nn, OPB:98 Nn

Case with a torsion motion

As previously noticed, the damping coefficient for the first torsion mode (1T) is closed for both turbulence models. The next step is to check if it is not due to compensatory effects. As shown, on the right of fig. 4, the global trend of the work exchange along the chord is coherent for OPB (when the interblade channel is chocked). The major difference corresponds to the second half of the suction side, where the $k-l$ case exchanges more work from the blade to the flow (stabilising behaviour). This zone corresponds with the flow separation predict by the steady state flow. The analysis of pressure fluctuation amplitude and phase (not presented) reveals that it is due to different amplitudes with a similar phase. It is reasonable to associate this difference to the difference of the flow separation averaged size.

For the operating point at the lowest speed (OPA), the main differences are near the leading edge (see fig. 4 at the left): a bump of negative work on the suction side and a bump and a localised peak on the pressure side both. These zones correspond to the two supersonic pock-ets presented fig. 2 at the left. The analysis of pressure fluctuations near the leading edge (not presented) indicates different amplitude with a similar phase. The difference of flow incidence angle induces different supersonic pockets size that react with different amplitudes to the blade vibration motion. Despite this, the differences between the two turbulence models are rela-
tively small and by a lucky strike here, the differences are self-compensated and the damping coefficients are the same.

As a conclusion, for the torsion mode the damping coefficient prediction seems to be relatively robust to the turbulence model, the difference noticed can be explained by the different size prediction of steady flow features and it is linked to the pressure fluctuation modulus, not to its phase.

Figure 4: Exchanged work along the axial chord for the first torsion mode with 2ND at OPA (left) and OPB (right). The trailing edge is at X/Xc=1 and -1, the leading edge at X/Xc=0, the pressure side for X/Xc<0 and the suction side for X/Xc>0.

Case with a flexion motion

For flows with small supersonic pockets at OPA

Figure 5: Left: exchanged work along the chord for the first flexion mode with 6ND at OPA. The trailing edge is at X/Xc=1 and -1, the leading edge at X/Xc=0, the pressure side for X/Xc<0 and the suction side for X/Xc>0. Middle and right: pressure fluctuation phase respectively for \(k-l\) and \(k-\omega\) results.
For the flexion mode (1F), the damping coefficient has been calculated for different nodal diameters, 2, 4 and 6\(ND\). The damping coefficients for the operating point at lower velocity (OPA) present a relatively constant gap between the \(k-l\) and the \(k-\omega\) cases (see fig. 3). As presented at left in fig. 5, for 6\(ND\), the zones of high work exchange are located at the same positions as for the torsion mode, but the work exchange direction is opposite. The disparities between the turbulence models have the same order of magnitude as for the torsion mode. Major disparities are located at the place of supersonic pockets and the first half of the blade. Differently to the torsion mode, with a flexion motion the disparities do not compensate themselves and the resulting damping coefficient is lower for the \(k-l\) case.

These work exchange disparities correspond to different levels of pressure fluctuation modulus (as the torsion mode). For example, at 6\(ND\) the pressure side supersonic pocket induces larger pressure fluctuations and the pocket on the suction side induces smaller pressure fluctuations for the \(k-l\) case (not presented). In addition, zones of disparities correspond with areas where phase shift can be observed (different to the torsion mode). For example, at 6\(ND\) on the pressure side, as presented on the pressure fluctuation phase mapping in fig. 5, in the zone marked with A symbol. Regressive pressure waves (waves propagating in the inverse direction of the flow convection) are predicted by the \(k-l\) case instead of a pressure fluctuation phase relatively constant for the \(k-\omega\) case.

The work exchange disparity, and the corresponding pressure fluctuation amplitude and phase, is modulated by the \(ND\) evolution. The larger difference between the \(k-l\) and \(k-\omega\) cases remains on the pressure side. But when the \(ND\) increases, the disparity on the pressure side tends to decrease and the disparity of the suction side tends to increase. That contributes to keep the global damping coefficient relatively constant for the different \(ND\).

The flexion mode at OPA points to an effect of the pressure fluctuation phase to explain the disparity between the two turbulence models. The effect of the phase shift here is not important and the prediction remains robust for the detection of the flutter triggering.

For the flow with choked interblade channel at OPB

For a flexion mode on OPB, the prediction on the damping coefficient can present important disparities (see fig. 3 right). As presented in the 4\(ND\) case in fig. 6, the analysis of the work exchange along the chord reveals important disparities at the position of the traversing shockwave. In particular on the pressure side, the work exchange presents a larger and higher peak of negative value at the traversing shockwave position (X/Xc=-0.15) for the \(k-l\) case. This difference corresponds to simultaneously, a more intense pressure fluctuation amplitude of the shockwave, and a phase shift of the pressure wave in the inter-blade channel. The phase fields at the right on fig. 6 show an important difference in the phase pattern between both cases in the interblade channel (zone marked with B symbol). For the \(k-l\) case, the phase pattern presents regular iso-phase of progressive pressure waves (waves propagating in the direction of flow convection). That is completely different to the \(k-\omega\) case where the phase pattern reveals a complex superposition of pressure waves. Progressive pressure waves are dominant on the suction side and regressive waves on the pressure side. The pressure fluctuations in \(k-l\) are not only more intense but reach the shockwave at a different time in the vibration cycle of the blade. This effect seems to amplify the work exchange from the fluid to the blade at the shockwave.

The previous example at 4\(ND\) has been selected because of the evident disparity in the pressure fluctuation phase pattern, but the effect on phase and amplitude is modulated by the \(ND\). In the extreme case, at 6\(ND\), the \(k-l\) and \(k-\omega\) cases present a phase pattern like the \(k-\omega\) case.
Figure 6: **Left:** exchanged work along the chord for the first flexion mode with 4ND at OPB. The trailing edge is at X/Xc=1 and -1, the leading edge at X/Xc=0, the pressure side for X/Xc<0 and the suction side for X/Xc>0. **Middle and right:** pressure fluctuation phase respectively for \( k-l \) and \( k-\omega \) results.

To investigate the flow dynamics, the flow field is reconstructed in the time domain based on linearized results during one vibrating cycle. Fig. 7 presents the main flow features at three instants of interest for \( ND = 4 \). The first instant selected at \( t = 0.2T \), corresponds to the smallest separation zone and the last instant at \( t = 0.76T \), corresponds to the largest for the \( k-\omega \) case. The intermediate instant corresponds to the minimal flow separation size for the \( k-l \) case.

The flow separation zone size fluctuation is larger for the \( k-l \) case (the minimal size of the flow separation is smaller and the maximal size is larger). The breathing motion of the flow separation zone is different in terms of magnitude. In addition, the extreme size of the flow separation zone is not reached at the same time in the two simulations. The breathing motion is also shifted in time between both cases. As a consequence, the pressure waves travelling in the vicinity of the flow separation zone differ strongly. For example, in fig. 7 in the last presented instant with arrows, only regressive pressure waves are emitted by the flow separation zone up to the shock wave for the \( k-l \) case, but for the \( k-\omega \) case the pressure waves travel in both directions downstream and upstream. The flow separation differences in terms of magnitude and timing induce a different pressure waves emission and propagation, that explains the difference in the phase patterns observed in fig. 6.

The position of the shockwave seems to be relatively similar between both cases. The flow
separation zone is an emitter of pressure waves. These pressure waves can travel upstream up to the shockwave. The shockwave represents a receptor which induces a work exchange with the blade. This exchange is strongly affected by the timing and the magnitude of pressure waves.

Figure 7: Time capture for OPB 1F4/ND, at t= 0.2, 0.36 and 0.76 T with $k-l$ (upper) and $k-\omega$ (lower) calculations. Blue, black and red lines represent respectively iso-pressure lines, $Ma=1$ iso-line and $Vx=0$ iso-line.

CONCLUSIONS

In this paper the sensitivity of the triggering of flutter to the turbulence modelling is analysed with a Linearised RANS (LRANS) calculation on 2D blade to blade channels. The turbulence modelling of the steady state flow has been done with two different turbulence models, the $k-l$ and the $k-\omega$. The investigated geometry is a modern design of a high pressure compressor, two operating points and two vibration modes (first torsion and flexion modes) have been analysed. For the investigated cases, the turbulence model has a limited effect on the damping coefficient for the torsion mode. The difference in the work exchange is linked to different levels of pressure fluctuation modulus corresponding to different flow features size, like supersonic pockets.

For the flexion mode, the flutter prediction is impacted by the pressure fluctuation modulus and the phase. This phase shift in the interblade channel seems to be sensitive to the type of flow features: shockwave, flow separation ... For the operating point with a flow separation zone, the size of this bubble differs depending on the turbulence model. The difference of the flow separation in the steady state also conducts to a different flow separation zone dynamic in terms of phase and amplitude. These disparities induce different pressure fluctuations and so different aerodynamic loading of the structure. The dynamics of the flow separation differs from a nodal diameter to another and in the worst case the damping coefficient sign can be opposite depending on the turbulence model selected.

An important conclusion of the paper results is that the turbulence model choosen can have, or not, an effect on the flutter triggering. Because of that, the trap is to check different turbulence models on one configuration and to consider the result valid for all vibration modes or operating...
conditions. Another subtlety is to restrict the analysis only to the damping coefficient which can be the result of compensating effects. An analysis of the flutter triggering should include analysis of the work exchange along the chord and the mapping of the pressure fluctuation modulus and phase. These tools allow to detect the compensation effects and the work exchange sign modification.

Analysis is focused on the link between the flutter triggering and the flow phenomenon dynamics for different turbulence models. The next step is to determine how the turbulence models modify the flow phenomenon dynamics. In the light of presented result, the comportment of turbulent variables in the vicinity of the wall is certainly interesting to investigate.

REFERENCES
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