Verification of Employed Assumptions In Decoupled Methodology For Turbomachinery Aeroelasticity

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Abstract
This paper focuses on the validity of the used assumptions in the development of the Decoupled approaches which predict the aeroelasticity behavior of turbomachinery blades. A Decoupled approach is employed to evaluate the aerodynamic damping and the forced response of a low pressure compressor $BluM^{TM}$ induced by low engine order excitations. The validity of the assumptions like superposition principle, linearity of aerodynamic damping forces and the blade-motion independency of the aerodynamic forces are verified by employing the TWIN approach. Extensive experiments are also carried out to evaluate the overall performance of the approaches.

1 Introduction

During the last decades, several numerical approaches have been developed to predict the flutter and the aeroelastic forced response in turbomachinery. Different classifications of these approaches can be found in the literature [1, 2]. Based on the level of the fluid-structure interaction, these approaches can be categorized by the Coupled and Decoupled approaches. The main differences between these approaches rely upon the linearity of the flow and structure behavior. The linearity in behavior means that the aerodynamic forces and the vibrational level of the structure show respectively a linear variation with respect to the blade response amplitude and excitation forces.

The Coupled approaches have been developed to deal with blade-motion-dependency and the flow nonlinearities [3, 4]. These approaches do not rely on the superposition principle and linearity assumptions. In Coupled approaches, a coupling between the fluid and structure is carried out at each time step which enables the methods to compute the forced response without considering the rather restrictive superposition principle. Moreover, these methods are very expensive due to the long time to converge to a developed periodic oscillation.

The most used approaches in industry are Decoupled approaches. The numerical procedures and the applications on some of the industrial case studies are presented in the literature [2, 5, 6, 7]. These approaches are an open loop system based on the principles of aerodynamic-independency of the modes, the superposition of the aerodynamic forces and also the linearity assumption. The aerodynamic-independency of the modes means that the shape and frequency of the modes are not influenced by aerodynamic load. According to that, the modal analysis is performed without taking into account the aerodynamic loads. The superposition principle of aerodynamic forces allows evaluation of the aerodynamic damping and excitation forces through two individual CFD computations. The aerodynamic damping is obtained by performing the CFD simulation for an isolated blade-row in a clean flow (unperturbed by upstream wakes) with a prescribed forced harmonic motion. It is assumed that the aerodynamic damping forces have a linear variation with respect to the amplitude of the blade

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motion. The excitation forces are estimated using a stage computation with the assumption of the blade-motion-independency.

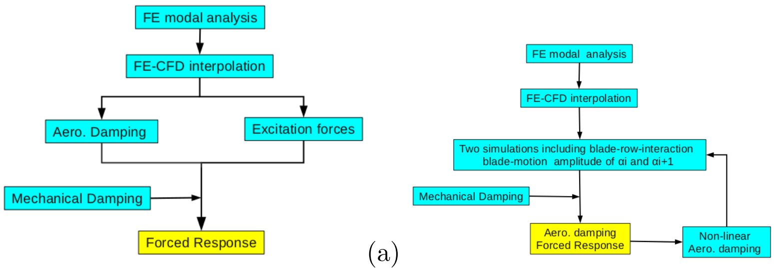
Few comparative studies have been published which support the use of Decoupled approaches for both subsonic and transonic blades. In most of these studies, the outputs of the Coupled simulations are considered as the reference solutions. Tran et al [8] observed a good agreement when different Decoupled and Coupled approaches were applied on a compressor blisk. Schmit et al [9] estimated the forced response of transonic counter rotating prop fan and they reported a full validity of superposition principle of aerodynamic forces. Moffat and He [10] made an evaluation between the use of Decoupled and Coupled forced response calculations. They show that the decoupled methodology can accurately predict the resonant vibration level of a transonic fan rotor thus, the use of coupled methods provides no gain over the Decoupled ones. Sadeghi and Liu [11] performed the flutter study for a transonic compressor blade and they verified the accuracy of the method for the high mass ratios. However, they highlighted the potential dangers of using the Decoupled methods for the low mass ratios.

The objective of this study is to investigate the overall performance of the Decoupled approach as well as the validity of the assumptions like superposition principle, linearity of aerodynamic damping forces w.r.t. the variation in amplitude of oscillations and the blade motion-independency of the aerodynamic forces. For this purpose, the Decoupled and the TWIN methodologies [12, 13] are employed to determine the forced response of a low pressure compressor $BluM^{TM}$ (monoblock bladed drum) induced by low engine order excitations. The TWIN methodology is a loosely coupled approach which is not established on the superposition principle. In this approach, the forced response is calculated through simulations in which the blade row interactions and the forced harmonic blade vibration are coexisting. On the contrary of the Decoupled approaches, the aerodynamic damping forces are deduced from two similar simulations with different amplifications of the blade motions. These properties of TWIN approach make it possible to evaluate the aforementioned assumptions which are used in the development of the Decoupled methodology. Moreover, extensive experiments are conducted to assess the overall performance of the methodologies in estimation of the forced response.

In the following, first, the analytical formulations of the forced response to a synchronized excitation force are derived and the employed methodologies are elaborated. Next, the test cases are introduced and the experimental and numerical setups are illustrated. Finally, the obtained results are presented and the validity of the assumptions are analyzed. At the end, a summary of the work is presented and some conclusions are offered.

2 METHODOLOGY TO CALCULATE FORCED RESPONSE

In Decoupled approaches, the fluid and structured domains remain uncoupled. It means that the unsteady fluid flow does not affect the modal behavior of the structure. In other words, the Decoupled approach splits an inherently coupled non-linear phenomenon into two separate (non-)linear and uncoupled analyses. In this research, the employed Decoupled approach calculates the forced response in four steps [6, 7]. First, a modal analysis is obtained using Finite Element (FE) calculations. In the second step, the aerodynamic damping is estimated by performing the CFD simulation for a single rotor blade with the prescribed harmonic forced motion. The third step is the calculation of the excitation forces which are predicted by a stage computation. At the end, by solving the equations of motion (in their modal form), the forced response is estimated. In this approach, it is assumed that the upstream stator row is tuned in such a way that all rotor blades are forced to oscillate with the same frequency and inter-blade phase angle (IBPA). Figure 1 (a) presents the procedure to



(a) (b)

Figure 1: Procedure to calculate the forced response. (a) Decoupled approach, (b) TWIN approach.

calculate the aerodynamic damping and forced response in Decoupled approach.

The estimation of the forced response in TWIN approach is similar to the Decoupled approach but the excitation forces and the aerodynamic damping are deduced from two (stage) computations in which the blade row interaction and the blade motion are present. Therefore the incoming flow is not uniform but includes the upstream stator wakes. Figure 1 (b) illustrates the procedure to calculate the aerodynamic damping and forced response in TWIN approach. As it is illustrated in the figure, for the cases where the blade row interactions and also the aerodynamic damping froces are not linear, more iterations can be performed by updating the amplitude and initial phase of the blade motion with the results of the previously computed forced response.

3 FORCED RESPONSE FORMULATION

In this section, the analytical formulations to calculate the forced response in terms of the estimated parameters using Decoupled and TWIN approach are derived and presented.

3.1 Decoupled approach

The equilibrium of the forces of a structure characterized by a mass M, a damping C and a stiffness K exposed to external forces F can be written as

$$M\ddot{u} + C\dot{u} + Ku = F(t) \tag{1}$$

Let's assume an harmonic motion of the form

$$u(t) = \alpha\phi_k \exp^{i\omega_k t} \tag{2}$$

where ϕ_k is the complex mode of interest, ω_k is the related frequency of the mode k. The amplification factor α is a complex number $\alpha(a,\theta) = a\exp^{i\theta}$ characterizing by the amplitude "a" and the dephasing "θ" of the deformation with respect to the k mode.

Substituting Eq.(2) into Eq.(1) yields

$$\alpha\left(K\phi_k - \omega_k^2 M\phi_k + i\omega_k C\phi_k\right)\exp^{i\omega_k t} = F(t) \tag{3}$$

By multiplying the Equation (3) by $\overline{\phi_k}^T$:

$$\alpha\left(\overline{K}_k - \omega_k^2\overline{M}_k + i\omega_k\overline{C}_k\right)\exp^{i\omega_k t} = \overline{\phi_k}^T F(t) \tag{4}$$

where the terms $\overline{K}_k$, $\overline{M}_k$ and $\overline{C}_k$ are generalized matrices defined by $\overline{X}_k \stackrel{\text{def}}{=} \overline{\phi_k}^T X\phi_k$.

The right hand side of the Equation (1) is the unsteady force $F(t)$ acting on the blade. The application of an harmonic motion on the blade will lead to an harmonic force characterized by the frequency ω_k. Therefore, the force can be expressed by

$$F(t) = \sum_{n=1}^{\infty} f_{k_n}(\alpha)\exp^{in\omega_{k_n} t} \tag{5}$$

where f_{k_n} are Fourier coefficients provided by the FFT signal decomposition of the forces. From a combination of Eq.(5) and Eq.(4) and by applying the identification limited to the first harmonic, one derives

$$\alpha\left(\overline{K}_k - \omega_k^2\overline{M}_k + i\omega_k\overline{C}_k\right) = \overline{\phi_k}^T f_k(\alpha) \stackrel{def}{=} GAF(\alpha) \tag{6}$$

$$= GAF_{forcing} + \alpha\frac{GAF_{aero}(\alpha_{aero})}{\alpha_{aero}} \tag{7}$$

The term on the right hand side of Eq.(6) is called the **G**eneralized **A**erodynamic **F**orces (GAF) and is assumed to be linear wrt α in the decoupled approach. It should be pointed out that the terms $\overline{\phi_k}^T F(t)$ in Eq.(4) is the temporal GAF and for further convinient it is shown by tGAF.

The forced response α_{fr} is thus obtained by

$$\boxed{\alpha_{fr} = \frac{GAF_{forcing}}{[\overline{K}_k - \omega_k^2\overline{M}_k + i\omega_k\overline{C}_k] - \frac{GAF_{aero}}{\alpha_{aero}}}} \tag{8}$$

By comparison of the denominator terms of Eq.(8), one can derive a physical interpretation of the $-GAF_{aero}/\alpha_{aero}$ term, i.e.

$$-\omega_{shift}^2\overline{M}_k + i\omega_k\overline{C}_{aero,k} = -\frac{GAF_{aero}}{\alpha_{aero}} \tag{9}$$

The frequency shift induced by the added mass [7] and the aerodynamical damping are identified by the real and imaginary parts of GAF

$$-\omega_{shift}^2\overline{M}_k = -\Re\left(\frac{GAF_{aero}}{\alpha_{aero}}\right) \tag{10}$$

$$\overline{C}_{aero,k} = -\frac{1}{\omega_k}\Im\left(\frac{GAF_{aero}}{\alpha_{aero}}\right) \tag{11}$$

The non-dimensional aerodynamical damping in this study is defined as

$$\delta = \frac{\omega_k\overline{C}_{aero,k}}{2\overline{K}_k} = -\frac{1}{2\overline{K}_k}\Im\left(\frac{GAF_{aero}}{\alpha_{aero}}\right) \tag{12}$$

where $\overline{K}_k$ is the generalized stiffness of the structure.

Finally, the forced response α_{fr} can be written by substituting Eq.(9) into Eq.(8)

$$\alpha_{fr} = \frac{GAF_{forcing}}{[\overline{K}_k - (\omega_k^2 + \omega_{shift}^2)\overline{M}_k + i\omega_k(\overline{C}_k + \overline{C}_{aero,k})]} \tag{13}$$

3.2 TWIN approach

In TWIN approach, the excitation forces are superposed to the damping forces acting on the surface of blades. These damping forces are induced by a prescribed harmonic motion. This means that the computed GAF is a combination of both the excitation and the aerodynamic damping forces.

$$GAF_{TWIN}(\alpha) = GAF_{forcing} + GAF_{aero}(\alpha) \tag{14}$$

Thus, Eq.(4) for TWIN simulation is:

$$\alpha\left[\overline{K}_k - \omega_k^2\overline{M}_k + i\omega_k\overline{C}_k\right] = GAF_{TWIN}(\alpha) \tag{15}$$

The forced response α can be written as the solution of the equation

$$\mathcal{F}(\alpha) = 0 \quad \text{where} \quad \mathcal{F}(\alpha) \stackrel{\text{def}}{=} GAF_{TWIN}(\alpha) - \alpha\left[\overline{K}_k - \omega_k^2\overline{M}_k + i\omega_k\overline{C}_k\right]. \tag{16}$$

By applying the Taylor expansion and taking into account only the first order terms, a linear system is obtained

$$\frac{\delta \mathcal{F}}{\delta \alpha}|_n(\alpha^{n+1} - \alpha^n) = -\mathcal{F}(\alpha^n) \tag{17}$$

The computation of the Jacobian can be a very difficult and expensive task because of the non-linear behaviour of the Navier-Stokes equations. In practice, the Jacobian is computed approximately using a finite difference formula like:

$$\frac{\delta \mathcal{F}}{\delta \alpha}|_n = \frac{\mathcal{F}(\alpha^n + \epsilon) - \mathcal{F}(\alpha^n)}{\epsilon} \tag{18}$$

where ϵ is a small complex value. Hence

$$\frac{\mathcal{F}(\alpha^n + \epsilon) - \mathcal{F}(\alpha^n)}{\epsilon}(\alpha^{n+1} - \alpha^n) = -\mathcal{F}(\alpha^n) \tag{19}$$

The solution of this 1D complex equation is given by

$$\alpha^{n+1} - \alpha^n = \frac{GAF_{TWIN}(\alpha^n) - \left[\overline{K}_k - \omega_k^2 \overline{M}_k + i\omega_k \overline{C}_k\right] \alpha^n}{\left[\overline{K}_k - \omega_k^2 \overline{M}_k + i\omega_k \overline{C}_k\right] - \frac{GAF_{TWIN}(\alpha^n + \epsilon) - GAF_{TWIN}(\alpha^n)}{\epsilon}} \tag{20}$$

Here, the non-dimensional aerodynamical damping is calculated by

$$\delta = -\frac{1}{2\overline{K}_k} \Im \left(\frac{GAF_{TWIN}(\alpha^{n+1}) - GAF_{TWIN}(\alpha^n)}{\epsilon} \right) \tag{21}$$

For the case where $\alpha^0 = 0$ and $\epsilon = \alpha_{aero}$, the first Newton iteration of Eq.(20) becomes:

$$\boxed{\alpha_{fr} = \frac{GAF_{TWIN}(0)}{\left[\overline{K}_k - \omega_k^2 \overline{M}_k + i\omega_k \overline{C}_k\right] - \frac{GAF_{TWIN}(\alpha_{aero}) - GAF_{TWIN}(0)}{\alpha_{aero}}}} \tag{22}$$

For the case where the variation of the aerodynamic damping with respect to the amplitude of the blade movement α is highly non-linear, a Newtonian iterative method is proposed. In the method, the value of α^n is updated with the calculated forced response α_{fr} following Eq.(20) and computations are repeated until a converged solution is obtained.

4 TEST CASE

The test case for the evaluation of the performance of TWIN approach and also Decoupled approach is a low pressure compressor $BluM^{TM}$ stimulated by low engine order excitations. The related Campbell diagram shows crossings between the first bending mode (1F) and the 10 and 20 engine orders at the speeds ω_1 and ω_2. The engine speed ω_1 is the operating point. Also from the ZZENF diagram corresponding to the engine speed ω_1, a crossing at 10 nodal diameters (1F-10ϕ) is observed.

The stage compressor includes 100 upstream stator and 76 rotor blades, thus, the dominant excitation is 100N. To generate the low engine excitation 10N, one stator blade out of 10 is replaced by a different (thicker) one in order to promote the excitation of the mode 1F-10ϕ. The configuration for the excitation force corresponding to 10N is presented in Figure 2(a). For further convenience, here, it is called $1S9s|1r$ which stands for 1 thick stator blade, 9 similar stator blades and one rotor blade. In a similar way, to generate the excitation corresponding to the 20N excitation, one stator blade out of 5 is replaced. Figure 2(b) shows the configuration $1S4s|1r$ for 20N excitation.

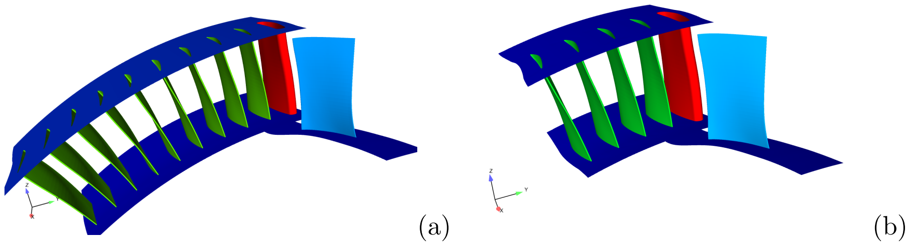


Figure 2: Configurations (a) $1S9s|1r$ corresponds to the 10N excitation. One stator blade out of 10 is different (in red). (b) $1S4s|1r$ to promote the 20N excitation. One blade out of 5 stator blades is different (in red).

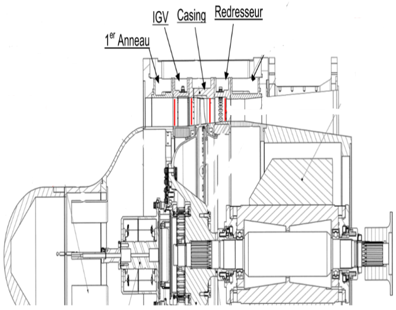


Figure 3: Schematic of the experimental setup including 1.5 stages compressor.

5 EXPERIMENTAL AND NUMERICAL SETUPS

The experimental and numerical simulations are conducted to investigate the forced response of a Low Pressure compressor $BluM^{TM}$ designed by Techspace Aero. In the following the experimental and numerical setups are illustrated.

5.1 Experimental setup

The experiments are carried out on a $\frac{1}{2}$ scale aerodynamic test bench. The test rig is depicted in Figure 3. The facility is a 1.5 stage compressor with the combination of stator - rotor - stator. The uniformity of the upstream pressure and yaw angle distributions was checked with a 5-hole pneumatic probe in both pitchwise and spanwise directions. The characteristics of the upstream boundary layer developing along the end walls were measured by means of a miniature total pressure probe. The absolute uncertainty on the measured pressure is 0.1%. Tip timing technique and the strain gauges measurements are used for blade response analysis. For the tip-timing, all blades are considered while a few blades are equipped with strain gauges. The accuracy of the measured blade vibration amplitude during forced response is ±4%.

5.2 Numerical setups

As it was described in the previous section, the forced response calculation is performed in several steps which requires different numerical setups, see Figure 1. In the first step, the modal analysis is obtained from a Finite Element (FE) calculation. With the assumption of the cyclic symmetry of the structure, only one sector of $BluM^{TM}$ is taken into account for the modal analysis. The FE calculations are performed using the Samcef code. In particular, the DYNAM solver is used after an initial ASEF(static) computation that enables us to account for centrifugal pre-stress. Moreover, the DYNAM analysis is performed by considering specific families of nodal diameter modes.

In order to interpolate the modal shape from FE mesh to the CFD mesh the MpCCI library is used (Ref.[14]). Figure 4 shows the real part of normalized displacement of the first bending mode (1F) on CFD mesh. It should be pointed out that the sector of $BluM^{TM}$ is not presented in this figure.

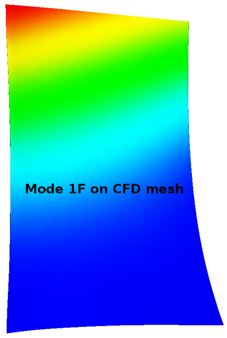


Figure 4: The real part of normalized displacement of the first bending mode (1F) on CFD mesh.

The excitation forces $GAF_{forcing}$ in Decoupled approach and all aerodynamic forces in TWIN approach $GAF_{TWIN}(\alpha)$ are estimated through a stage chorochronic computation [15]. Figure 2 presents the computational domains for the test cases of 1F-10N-10ϕ and 1F-20N-20ϕ. The AutoGrid 8.9-2 software of Numeca International is used to generate the grid. Based upon the experience, roughly 1% of the blade span is considered as the tip gap. The meshes $1S9s|1r$ and $1S4s|1r$ include 23 and 13.2 million grid points, respectively. The first grid cell size next to the wall is 1×10^{-6} m corresponding to a y^+ value about one. The unsteady Reynolds-Averaged Navier-Stokes equations are solved as the governing equations. The simulations are carried out using the elsA solver which is a cell-centered, finite volume, multi-block structured solver developed at ONERA [16, 17]. The Smith k -L turbulence model [18] is implemented to add the turbulence effects. At the inlet boundary, the total pressure, total temperature, flow direction and turbulence variables are prescribed according to measured values. The boundary layer profile is obtained from experiments. At the outlet, a valve type boundary condition is imposed which tunes the static pressure to obtain a target mass flow. The Jameson scheme is used for space discretization. For the time advancing the backward Euler scheme - which is a first order implicit scheme - is used.

The aerodynamic damping forces of the decoupled approach GAF_{aero} are calculated on a single row domain including the same rotor passage. The mesh contains 2.4 million grid points. The inlet and outlet boundary conditions are extracted from the stage (stator-rotor) computation which is performed to estimate the excitation forces. The chorochronic boundary condition is applied at both side boundaries. The surface displacement (harmonic blade motion) is imposed using a mesh deformation technique. The same technique is also used in the TWIN computation. The amplification factor of blade motion corresponding to the modal movement of 1F is $\alpha_1 = (0.40, 0)$, where the phase is in radian and the amplitude is normalized.

6 RESULTS AND DISCUSSION

As it was mentioned in the section 1, the objective of this study is to assess the overall performance of the Decoupled approaches as well as the validity of the assumptions these approaches are based on. In this section, the different steps of forced response calculations, i.e. excitation forces and aerodynamic damping, are illustrated through the test cases and the overall performance of the methodology is evaluated by comparing the predicted forced responses with the measured values. At the end, the validity of the assumptions like superposition principle, linearity of aerodynamic damping forces and also blade-motion independency of aerodynamic forces are analyzed and discussed.

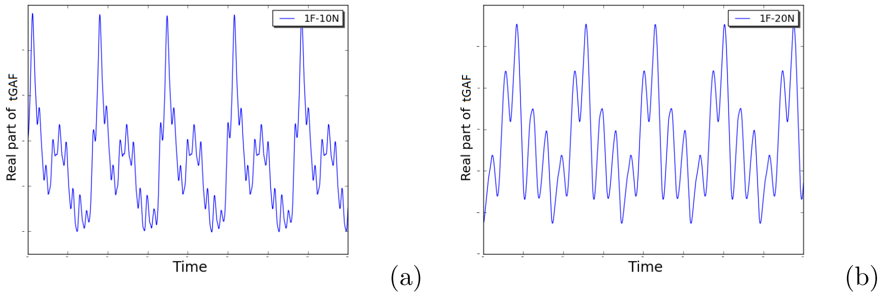


Figure 5: Time evolution of $\mathcal{R}e.tGAF_{TWIN}(\alpha 0)$; (a) the case 1F-10N case, (b) the case 1F-20N.

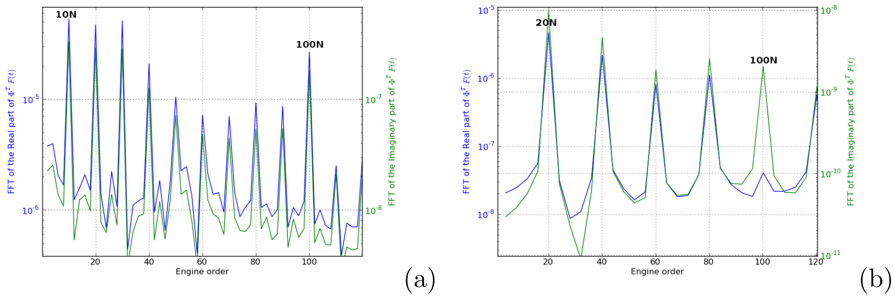


Figure 6: The calculated FFT of $tGAF_{forcing}$ signal over 5 periods of thick blade passing in frequency domain. (a) the case 1F-10N, (b) the case 1F-20N.

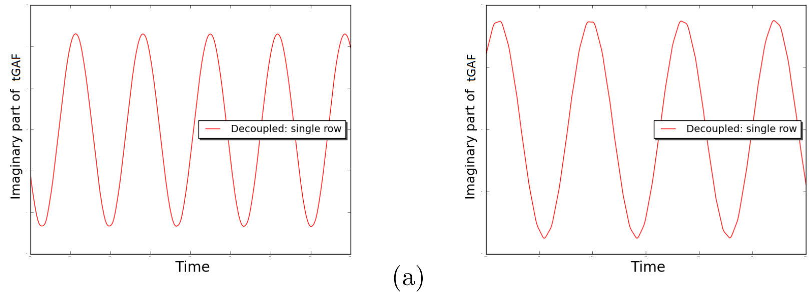
6.1 Excitation forces

The excitation forces in Decoupled and TWIN approaches are calculated using a stage configuration. For excitation corresponding to 10 and 20 engine orders, the configurations of $1S9s|1r$ and $1S4s|1r$ are used, respectively. Here, the used amplification factors are $\alpha_0 = (0, 0)$ and $\alpha_1 = (0.40, 0)$ where the phase is in radian and the amplitude is normalized. Therefore, the used simulation to calculate the excitations for Decoupled approach is the same as TWIN(α_0) which means $GAF_{forcing} = GAF_{TWIN}(\alpha_0)$.

The unsteady RANS computations start from the converged solutions of RANS simulations as the initial flow field. The computations are carried out until they reach a periodic behavior. After stabilization of the periodic behavior of flow, the unsteady pressure acting on the rotor blade is measured and converted to the GAF format (see Eq. (6)). The excitation forces are the combination of the different upcoming wakes induced by the regular and thick blades. Figures 5 (a) and (b) present the time evolution of real part of the temporal GAF for the cases 1F-10N and 1F-20N, respectively. It can be seen that the induced forces by the passing wakes of the similar blades are affected by the thick blade so that the forces are changed from blade to blade. Nevertheless, a periodic behavior with the frequency of the thick blade passing is observed for both cases. By applying the Fast Fourier Transform (FFT), the temporal GAF signal is converted to the frequency domain. Figure 6 presents the amplitude of GAF versus the frequency for the cases 1F-10N and 1F-20N. The general aerodynamic forces are calculated over the last 5 periods. The part of the GAF which shows the impact of the thicker blade (10N and 20N) and the part of all stator blades (100N) are marked in the figures. It should be pointed out that for forced response calculation, here, the $GAF_{forcing}$ is computed using the recorded pressure signal over the last period of the thick blade passing. In the frequency domain, it is tailored so that only the contribution of the first harmonic mode is taken into account and the other frequencies are ignored as they do not respond to the mechanical mode of interest.

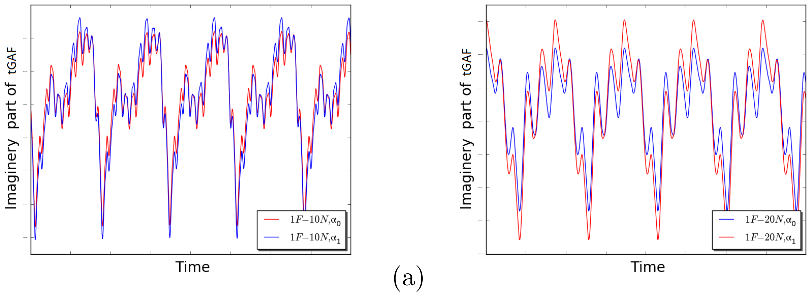
6.2 Aerodynamic damping

Aerodynamic damping not only is an input parameter for forced response calculation but also is a key parameter to predict the flutter behavior. The aero-



(a) (b)

Figure 7: Time evolution of $\mathcal{I}m.tGAF_{aero}(\alpha_1)$ obtained from a single row calculation in Decoupled approach; (a) the case 1F-10N, (b) the case 1F-20N.

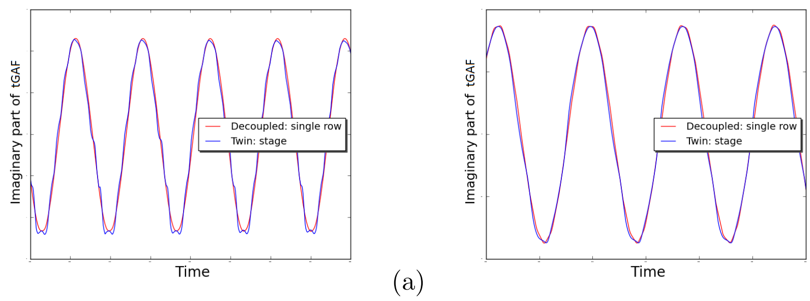


(a) (b)

Figure 8: Evolution of Imaginary part of temporal $GAF_{Twin(\alpha_0)}$ and $GAF_{Twin(\alpha_1)}$ (a) the case 1F-10N, (b) the case 1F-20N.

dynamic damping is related to the fluid unsteadiness generated by the blade dynamic motion. In the Decoupled approach, the aerodynamic damping is calculated through a single rotor blade simulation with a forced harmonic motion corresponding to the mode 1F with the prescribed amplification factor α_1. This blade motion generates unsteady pressure which can either excite the blade vibration or damp it. Figure 7 presents the time evolution of the temporal GAF for both test cases obtained from the single row configuration.

By recalling the Eq.(14), the superposition principle of forces can be verified by comparing the $GAF_{aero}(\alpha_1)$ with the reconstructed signals from TWIN simulations, i.e. $GAF_{TWIN}(\alpha_1) - GAF_{TWIN}(\alpha_0)$. Figure 8 presents the time history of temporal $GAF_{TWIN}(\alpha^1)$ and $GAF_{TWIN}(\alpha^0)$. The differences between two signals in Figure 8 is due to the blade vibration. Figure 9 presents the calculated $tGAF_{TWIN}(\alpha_1) - tGAF_{TWIN}(\alpha_0)$. For better comparison, temporal GAF_{aero} which is plotted in Figure 7 is also shown in the Figure. It should be noticed that the steady forces corresponding to the temporal $GAF_{aero}(\alpha_0)$ has been reduced from the signal, i.e. $tGAF_{aero}(\alpha_1)$-$tGAF_{aero}(\alpha_0)$. It can be seen that the signals are similar which means the superposition principle is valid for these cases. However, the reconstructed signals ($tGAF_{TWIN}(\alpha_1) - tGAF_{TWIN}(\alpha_0)$) are not as smooth as the one obtained from the single row calculation ($tGAF_{aero}$) and the effects of each passing wake can be observed. Table 1 presents the calculated non-dimensionalized aerodynamic damping (δ) based on Eq. (12) and Eq. (21) using Decoupled and TWIN approaches.

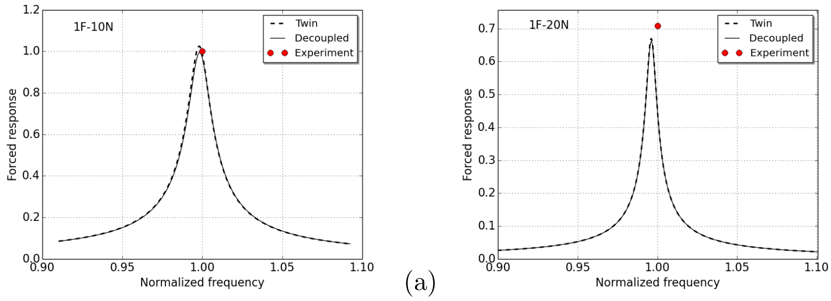


(a) (b)

Figure 9: Verification of the superposition principle based on Eq.(14). Imaginary part of $tGAF_{aero}(\alpha_1) - tGAF_{aero}(\alpha_0)$ and $tGAF_{TWIN}(\alpha_1) - tGAF_{TWIN}(\alpha_0)$; (a) the case 1F-10N, (b) the case 1F-20N.

Table 1: *Computed non-dimensionalized aerodynamic damping using Decoupled and TWIN approaches. The values are normalized by the computed aero. damping of the case 1F-10N using Decoupled approach.*

Approach	1F-10N	1F-20N
Decoupled	1.0	0.48
TWIN	0.97	0.47



(a) (b)

Figure 10: Estimated and measured forced response; (a) the case 1F-10N, (b) the case 1F-20N.

6.3 Forced response

The forced response quantity might be the best parameter to assess the overall performance of a methodology since the effects of all assumptions are accumulated in this quantity. In this section, the forced responses using both approaches are calculated and compared with the measurements.

As it is illustrated in Figure 1, the last input parameter for forced response calculation is the Rayleigh mechanical damping. This parameter is obtained through the experimental modal analysis in laboratory conditions at 20°C and 0 rotational speed. Although the mechanical damping is not measured at the operating condition, it is assumed that variation with respect to the engine speed is negligible.

The maximum forced response does not always take place at the given frequency (predicted resonance frequency from modal analysis) while in industries the estimation of the maximum values is more appreciated. Therefore, the forced response is extrapolated around the resonance frequency by keeping all parameters constant in Eq. (8) and Eq. (22) and solving those equations for a range of frequencies. Figure 10 presents the variation of the forced response amplitude versus frequency. The frequencies are normalized with the eigenfrequency of the mode (1F) at each nodal diameter. It can be observed that the maximum predicted forced responses take place at the frequencies which are slightly different from resonance frequency. This shift in response frequency is due to the "added mass", i.e. the real part of the terms $\frac{GAF_{aero}}{\alpha_{aero}}$ and $\frac{GAF_{TWIN}(\alpha_1)-GAF_{TWIN}(\alpha_0)}{\alpha_1}$ presented in Eq. (8) and Eq. (22), respectively [10].

Table 2 presents the calculated forced response at the peak frequency for the test cases 1F-10N and 1F-20N and also corresponding measured values. It can be seen that the estimated values following the Decoupled and TWIN approaches are very close. In comparison with the experiments, both approaches predict the forced response within 7% of the measured values which is an excellent performance.

Table 2: *Normalized value of the maximum forced response.*

Approach	Amplification	1F-10N	1F-20N
Measurements	–	1	0.71
Decoupled	α_1	0.99	0.66
TWIN	α_0, α_1	1.02	0.67

Amplication factor $\alpha = (a, \theta)$	α_1	α_2	α_3	α_4
Amplitude (a)	0.4	0.4	0.4	0.4
phase (θ)	0	$\frac{\pi}{2}$	π	$\frac{3\pi}{2}$

Table 3: *Different amplification factors. The phase is in radian and the amplitude is normalized.*

Regarding the accuracy of the method, almost the same range of accuracy, i.e. 15%, is reported when a Decoupled approach is used to determine the forced response of a turbine blade [5]. It should be pointed out that in the current study, the mechanical damping is measured and is taken into account in the forced response calculation while in the other studies a zero value is assumed. This might be the reason to observe a slightly better accuracy in this study.

6.4 Verification of the employed assumptions in the Decoupled Methodology

The superposition principle was verified in Section 6.3 where the variations in the aerodynamic forces caused by blade-motion were extracted from the stage simulations. It was shown the reconstructed signal is very similar to the ones obtained from single blade passage simulations.

The linearity assumption of aerodynamic damping forces can be investigated by performing simulations using different amplitudes of blade motion. The estimated forced response α_{fr} by TWIN approach using α_0 and α_1 for the case $1F-10N-10\phi$ is $\alpha_{fr1} = (1.025, 0.948\pi)$. As it was illustrated in section 2, for the case that the aerodynamic damping forces are not linear, the amplification factor should be updated $\alpha = \alpha_{fr1} = (1.025, 0.948\pi)$ and another stage simulation should be carried out. The TWIN calculations using α_1 and α_{fr1} reveals that $\alpha_{fr2} = \alpha_{fr1}$ which means for the selected test case, the variation of the aerodynamic damping forces for the range of blade movement is linear. Similar behavior for the case $1F-20N-20\phi$ is also observed.

The TWIN approach is based on the stage computation coexistent with the forced harmonic blade motion. To perform this kind of simulations the initial phase of the blade motion with respect to the upstream blade azimuthal position should be given. Therefore, the assumption of the blade-motion independency of aerodynamic forces can be verified by studing the impact of the initial phase. Here, the impact of the initial phase on the aerodynamic damping and the forced response is investigated through the case study of $1F-10N-10\phi$. For this purpose, the TWIN calculations are performed for various pairs of amplification factors which have the same amplitude 0.4 but different phases ranging from 0 to $\frac{3\pi}{2}$. Table 3 presents the amplification factors.

The non-dimensional aerodynamic damping and the forced response are calculated based on the combination of α_0 and the amplification factors presented in Table 3. The calculated values are shown in Table 4. For better comparison the values are presented with three digits accuracy. It can be observed that the current test case is not sensitive to the initial phase and the calculated non-dimensional aerodynamic damping and the maximum forced response using different combinations of α respectively result to the same values of ≈ 0.97 and $(1.02, 0.95\pi)$. This shows the interaction of the upcoming wakes and the vibrating blade is linear or in other words, the blade-motion independency of aerodynamic forces is valid for these cases.

7 SUMMARY AND CONCLUSIONS

In this research, the performance of a Decoupled approach to predict the forced response of a low pressure compressor $BluM^{TM}$ excited by low engine order excitations is investigated. The experiments are conducted to assess the over-

Table 4: *Calculated non-dimensional aerodynamic damping and forced response using different combination of α.*

Amplification factors for TWIN calculations	α_0, α_1	α_0, α_2	α_0, α_3	α_0, α_4
Non-dimensionalized Aerodynamic damping	0.969	0.969	0.966	0.964
Max. forced response $\alpha_{fr} = (a, \theta)$	a = 1.024 $\theta = 0.948\pi$	a = 1.023 $\theta = 0.948\pi$	a= 1.027 $\theta = 0.948\pi$	a= 1.029 $\theta = 0.948\pi$

all performance of the methodology. The TWIN approach is also employed to verify the validity of the used assumptions in the Decoupled approach. In this study the assumptions of superposition principle, linearity of aerodynamic damping forces and the blade-motion independency of the aerodynamic forces are investigated.

The time evolution of general aerodynamic forces acting on the surface of vibrating blade is used to calculate the aerodynamic damping in the Decoupled approach. A similar signal is deduced from two simulations of TWIN approach. The comparison of these signals reveals that the superposition principle is valid for the selected case studies. Calculated aerodynamic damping for a range of amplification factor of the blade movement shows a linear variation of aerodynamic damping forces with respect to the amplitude of blade vibrations. Moreover, it is shown that the initial phase of the blade motion with respect to the upstream blade azimuthal position in TWIN approach has a negligible effect on the calculated aerodynamic damping and the forced response. This validates the assumption of the blade-motion independency of aerodynamic forces.

The essential requirement to apply the Decoupled approach is that the induced blade motion should be small enough around the point at which the linearization is performed. In the current research, the test study is the first bending mode of a subsonic compressor blade where the Decoupled approach shows an excellent performance to estimate the forced response with a maximum difference of 7% between the numerical predictions and measurements. The reasons for that may be a large value for bending stifness. In the case of more flexible blades, such as large compressor blades, fans or propellers, the resulting motion might go beyond the linearity domain of the Decoupled approach application. In that case, the iterative TWIN approach might be a good candidate to tackle this new challenge but the assessement of the method is well beyond the scope of the present paper.

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