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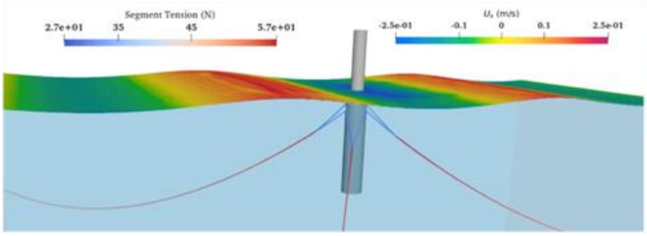

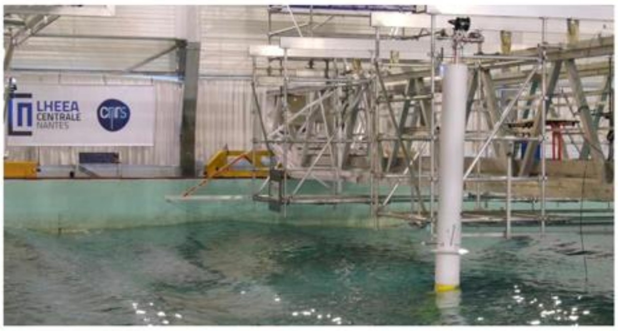

EFFET MEMOIRE APPLIQUE A L'AMORTISSEMENT STRUCTUREL DES EOLIENNES

MEMORY EFFECT APPLIED TO STRUCTURAL DAMPING OF WIND TURBINES

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Résumé

Dans la modélisation hydrodynamique des navires, des simulations en hypothèse d'écoulement potentiel sont utilisées pour fournir les efforts de houle de premier ordre, de masse ajoutée et d'amortissement de radiation. Ces deux derniers termes sont définis comme des coefficients dépendant de la fréquence. Afin d'effectuer des simulations dans le domaine temporel, la formulation de Cummins introduit une fonction de retard qui par convolution avec la vitesse du flotteur permet d'évaluer l'amortissement requis et la masse ajoutée. Pour les simulations d'éoliennes, les fabricants de turbines fournissent des informations sur l'amortissement pour différents modes propres du système. Lorsque l'on utilise la formulation classique de Rayleigh dans l'analyse dynamique structurelle, l'amortissement peut être considéré comme proportionnel à la matrice de masse et de rigidité du système et donc deux multiplicateurs proportionnels sont disponibles pour reproduire dans le domaine temporel l'amortissement modal. L'un des inconvénients de cette approche est qu'au maximum deux modes peuvent être sélectionnés pour s'adapter au niveau d'amortissement requis, mais pour les autres modes, l'amortissement ne peut pas être choisi et est imposé par la sélection des multiplicateurs.

L'utilisation d'un amortissement dépendant de la fréquence pour l'analyse structurelle dans le domaine temporel est proposée. Pour les fréquences modales sélectionnées, un niveau d'amortissement est défini et appliqué sous la forme d'une fonction triangle ou échelon autour des fréquences cibles. Le terme de convolution de la formulation de Cummins est ensuite utilisé pour appliquer l'amortissement dans le domaine temporel. Les tests numériques sont basés sur la turbine de 15 MW définie par l'IEA. La formulation de l'amortissement est d'abord vérifiée sur une seule pale, puis sur l'ensemble de l'éolienne. Il est constaté qu'il est possible d'amortir des fréquences spécifiques avec différents niveaux d'amortissement avec très peu d'influence sur les fréquences qui ne sont pas ciblées. Cette approche peut ensuite être utilisée pour un modèle d'éolienne flottante avec un amortissement spécifique sur l'éolienne qui ne ciblerait que la réponse de l'éolienne. Il est enfin

vérifié que cette formulation apporte une contribution négligeable aux charges en phase avec l'accélération (« masse ajoutée ») et que pour un essai de décroissance avec réponse monomode, le décrément logarithmique calculé converge avec le temps vers le décrément introduit par un coefficient d'amortissement direct sur la vitesse.

Summary

In hydrodynamic modelling of vessels, potential flow simulations are used to provide frequency domain loadings in terms of first-order wave forces, added mass and radiation damping. Those last two terms are defined as frequency-dependant coefficients. In order to perform time domain simulations, the Cummins formulation introduces a retardation function that is convoluted in time with the floater's velocity to produce the required damping and added mass. In simulation of wind turbine response, turbine manufacturers provide information on the damping for different eigenmodes of the system. When using classical Rayleigh formulation in structural analysis, the damping can be considered proportional to the mass and stiffness matrix of the system and therefore two proportional multipliers at most can be used to reproduce in time domain the modal damping. One of the drawbacks of this approach is that one or two modes can be selected to fit the required level of damping but for other modes or frequencies, the damping cannot be chosen and is imposed by the selection of the multipliers.

The use of a frequency dependant damping for structural analysis in time domain is tested. For selected modal frequencies of the system a level of damping is defined and applied as a triangle or step function around target frequencies. The convolution term of the Cummins formulation is then used to apply the damping in time domain. The numerical tests are based on the 15MW turbine defined by the IEA. The damping formulation is first checked on a single blade and then on the whole wind turbine. It is found that it is possible to damp specific frequencies with different level of damping with very little influence on the frequencies that are not targeted. This approach can then be used for floating wind turbine model with specific damping on the turbine that would only target the turbine response. It is finally verified that this formulation brings negligible contribution to the loads in phase with the acceleration ("added mass") and that for decay test with single mode response the logarithmic decrement converges with time towards the decrement introduced by a direct damping coefficient on the velocity.

I – Introduction

Floating wind turbines are made of a turbine, a tower, a floater and mooring lines (Figure 1). They respond to environmental loadings (wind, wave, current) as well as turbine rotation and controller in production mode. The system has rigid modes based on the floaters characteristics response in surge, sway, have roll, pitch and yaw. It has also higher frequency modes due to structural deformation of the blade, tower and floater structure.

Figure 1. Floating Wind Turbine

Focusing on the modeling of the turbine, suppliers provide damping information either for the blades or for the tower and RNA system. The damping data are provided as damping ratio or logarithmic decrement per modes. The question is then how to use these ratios in a time domain approach when the system is not solved based on a projection on its eigenmodes. In the present paper, such damping is implemented in the form of an impulse response function, constructed from the frequency-dependent damping similarly to the so-called ‘retardation function’ used in the Cummins equation to simulate the motion of floaters in time-domain. Therefore, the classical Rayleigh approach is first presented as well as the ‘retardation function’ approach as defined by Cummins. The updated damping formulation is then introduced together with some numerical comparisons between Rayleigh damping and updated method applied on a single blade or on the whole turbine. Finally, it is shown how the retardation function formulation can be linked to damping ratio per modes.

II – Rayleigh damping and retardation function

II – 1 Damping in structural analysis

The floating wind turbine is modeled by beam elements with a Timoshenko formulation [1]. The system is discretized at N nodes with six degrees of freedom for the position (three translations, three pseudo rotation vectors [2]) and the equation to solve is:

$$M\ddot{x} + C\dot{x} + Kx = f \qquad (1)$$

With x and f the position and external loading vectors for the *6xN* degrees of freedom and *M*, *C*, *K* respectively the mass, damping, and stiffness matrix. Focusing on structural Rayleigh damping [3], the matrix *C* is defined as:

$$C = \alpha M + \beta K \tag{2}$$

Which provides two parameters α and β to tune the damping.

It is also possible to solve the system by projection on its modal basis. For the i[th] mode the equation is defined as:

$$\ddot{x}_i + 2\xi_i\omega_i\dot{x}_i + \omega_i^2 x_i = \frac{f_i}{m_i} \tag{3}$$

With ω_i, ξ_i, f_i, m_i, respectively the eigenvalue, damping ratio, modal mass and modal force on mode *i*. The damping ratio is related to the coefficients in Rayleigh's formulation by:

$$\xi_i = \frac{1}{2}\left(\frac{\alpha}{\omega_i} + \beta\omega_i\right) \tag{4}$$

Therefore, it is possible to match the damping on only two of the modes. An extension of the Rayleigh damping based on Caughey series [4] is proposed by Adhikari and Phani [5] where the damping matrix is expressed as:

$$C = M \sum_{j=0}^{N} \alpha_j (M^{-1}K)^j \tag{5}$$

And the modal damping ratios as:

$$\xi_i = \frac{1}{2}\left(\frac{\alpha_1}{\omega_i} + \alpha_2\omega_i + \alpha_3\omega_i^3 + \cdots\right) \tag{6}$$

The Rayleigh formulation corresponding to the first two terms with $\alpha_1 = \alpha$ and $\alpha_2 = \beta$.In the present paper, a different approach is used by introducing the damping through a retardation function.

II – 2 Retardation function in hydrodynamic

Memory effect is a technique used to correctly account for the frequency-dependence of added mass and radiation damping in a time-domain simulation. It is accounted through the resolution of the so-called Cummin's equation of motion [6] for each of the six degrees of freedom (DOF) of a floater:

$$\left(\mathrm{M_k} + \mathrm{Ma_k}(\infty)\right)\ddot{\mathrm{X}}_\mathrm{k}(\mathrm{t}) + \int_0^\infty \mathrm{R_k}(\tau)\dot{\mathrm{X}}_\mathrm{k}(\mathrm{t}-\tau)\mathrm{d}\tau = \mathrm{F}_k^{\mathrm{ext}}(\mathrm{t}) \tag{7}$$

With M_k, the mass/inertia matrix, $Ma_k(\infty)$, the impulse added mass matrix, $R_k(t)$, the matrix of retardation functions, $X_k(t)$, the k-DOF motion, $F_k^{ext}(t)$, the external k-DOF force including viscous damping forces, mooring forces and hydrostatic restoring loads. The technique is based on the impulse response function (IRF) of a variable representative of the radiation loads. Since the non-impulsive part of the radiation transfer function verifies the Kramers-Kronig relations, the retardation function can be calculated from the radiation damping terms only. It is expressed as:

$$\mathrm{R_k}(\mathrm{t}) = \frac{2}{\pi}\int_0^\infty \mathrm{B_k}(\omega)\cos\omega\mathrm{t}\,\mathrm{d}\omega \tag{8}$$

For a regular motion pulsating at a frequency ω, the convolution term associated with the impulse inertia term restitutes the exact inertial and damping forces corresponding to $Ma_k(\omega)$ and $B_k(\omega)$, following Ogilvie's relations [7].

II – 3 Application of the retardation function to structural damping

In order to apply structural damping at specific frequencies, a retardation function is introduced based on the sum of elementary triangle (Figure 2) or step functions. The unitary triangle function is defined by:

$$\begin{aligned} B(\omega) &= 0 && \text{if } \omega < \omega_i - \Delta\omega_i^- \\ B(\omega) &= \frac{\omega-(\omega_i-\Delta\omega_i^-)}{\omega_i^-} && \text{if } \omega_i - \Delta\omega_i^- < \omega < \omega_i \\ B(\omega) &= \frac{(\omega_i+\Delta\omega_i^+)-\omega}{\omega_i^+} && \text{if } \omega_i < \omega < \omega_i + \Delta\omega_i^+ \\ B(\omega) &= 0 && \text{if } \omega_i + \Delta\omega_i^+ < \omega \end{aligned} \tag{9}$$

The associated retardation function is then:

$$R(t) = \frac{2}{\pi t^2}\left[\frac{\cos(\omega_i t)-\cos\left((\omega_i-\Delta\omega_i^-)t\right)}{\Delta\omega_i^-} - \frac{\cos(\omega_i t)+\cos\left((\omega_i+\Delta\omega_i^+)t\right)}{\Delta\omega_i^+}\right] \tag{10}$$

The unitary step function is defined as:

$$\begin{aligned} B(\omega) &= 0 && \text{if } \omega < \omega_i - \Delta\omega_i^- \\ B(\omega) &= 1 && \text{if } \omega_i - \Delta\omega_i^- < \omega < \omega_i \\ B(\omega) &= 0 && \text{if } \omega_i < \omega < \omega_i + \Delta\omega_i^+ \end{aligned} \tag{11}$$

With retardation function:

$$R(t) = \frac{2}{\pi}\frac{\sin\left((\omega_i+\Delta\omega_i^+)t\right)-\sin\left((\omega_i-\Delta\omega_i^-)t\right)}{\mathrm{t}} \tag{12}$$

In both cases, the function is defined for $t > 0$ and $R(0)$ is to be taken as the limit value.

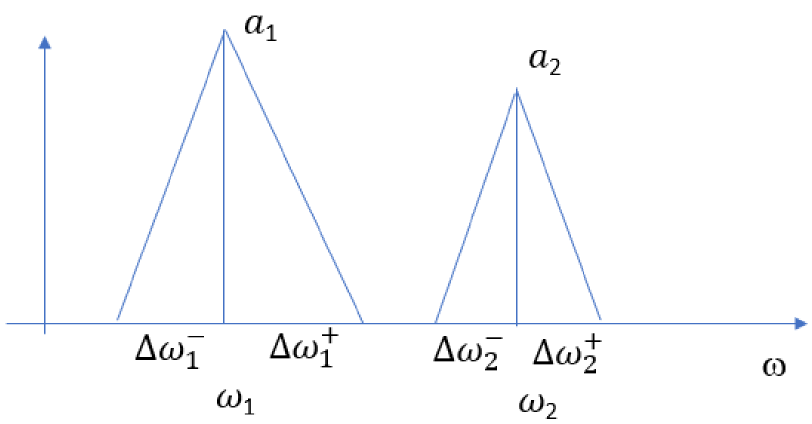

Figure 2. Example of triangle retardation function for two frequencies

As previously stated, the convolution also introduces a frequency-dependent inertial component $A(\omega)$. Nevertheless $\omega A(\omega)$ and $B(\omega)$ should verify the Kramers-Kronig relations, which predict $A \rightarrow 0$ if the area below $B(\omega)$ tends to zero. Actually, this 'added mass term' can be neglected, as will be verified later.

III – Results

III – 1 Sinusoidal excitation of a blade

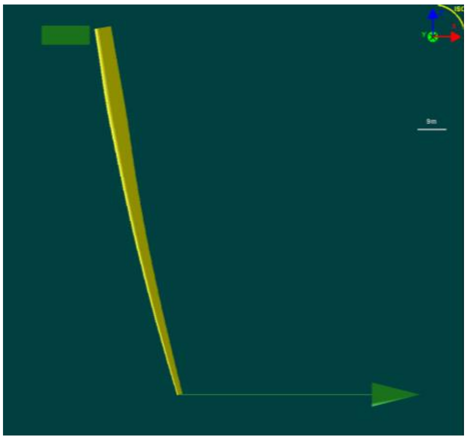

Figure 3. Blade of the 15MW IEA turbine

The blade of the 15MW IEA turbine [8] is excited at its tip by a sinusoidal force at 1Hz (Figure 4). In the direction of the force, the two closest modes are F1=0.52Hz and F2=1.48Hz. The blade is discretized with 30 elements. The dynamic simulations are performed with a Newmark algorithm in time. This test is run over 100s with the retardation function integrated over the past 30s at most. The triangle function is used, and damping is applied on the velocity of each node.

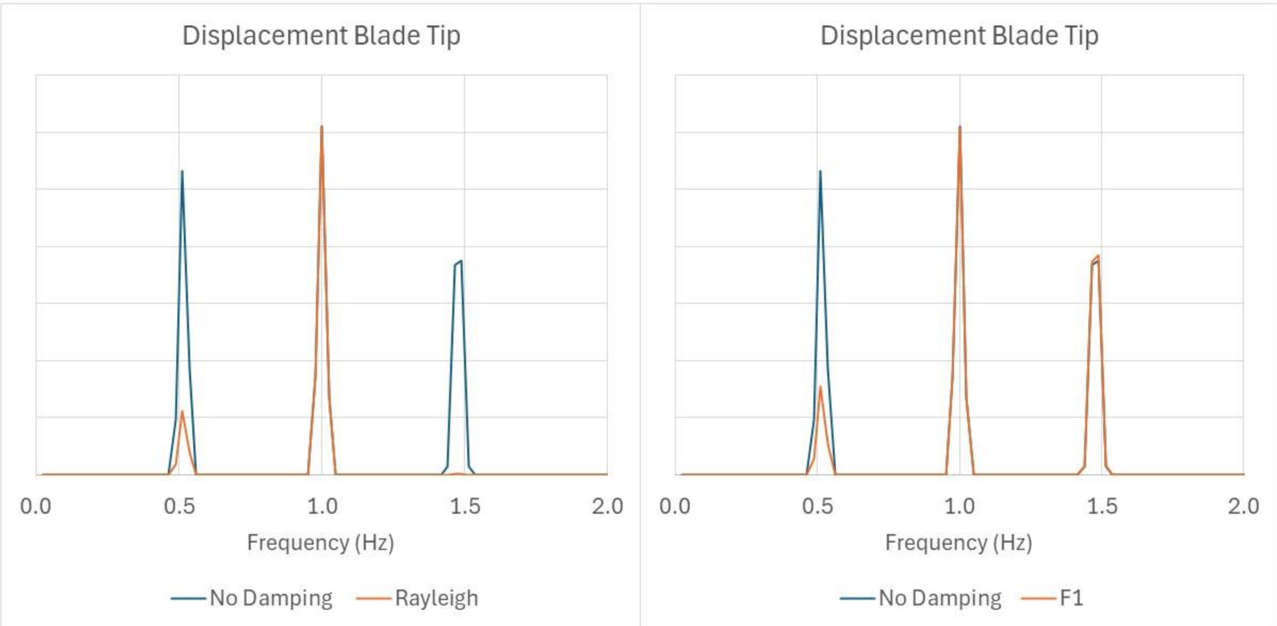

Figure 4. Comparison of Rayleigh damping (left) and retardation damping (right)

The response spectrum of the solution without damping is compared with the Rayleigh damping on the stiffness only (i.e. α =0) and the retardation damping with the first modal frequency F1 is targeted with a frequency range of Δf=+/-0.02Hz. For both damping methods, the direct response to the excitation (response at 1Hz) is unchanged. As expected, the Rayleigh damping on stiffness damps both modal response at F1 and F2 and since damping on stiffness is used the higher frequency is the most damped. With the retardation function only the targeted frequency F1 is damped, the energy response at F2 is very similar to the undamped response.

III – 2 Sinusoidal excitation of the turbine

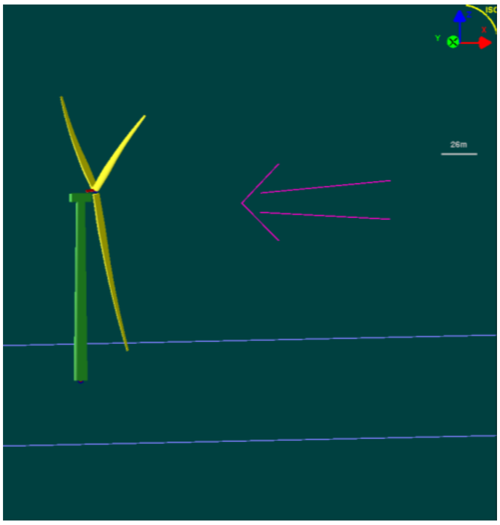

Figure 5. Blade of the 15MW IEA turbine

In this section, the whole 15MW turbine is modelled including the tower (Figure 5). The blades are in parked mode and a turbulent wind field is applied aligned with the turbine with an average hub height velocity of 30m/s. As in the previous section, simulations are performed with and without damping over 500s. In Figure 6, the response on 3 modes (F1=0.28s, F2=0.62Hz, F3=1.67Hz) is observed in the wind direction at bottom blade root. the damping is applied through the retardation function on the first mode (F1). As observed with the single blade, the response energy is reduced for the targeted mode while the other modes have similar response as the case without damping.

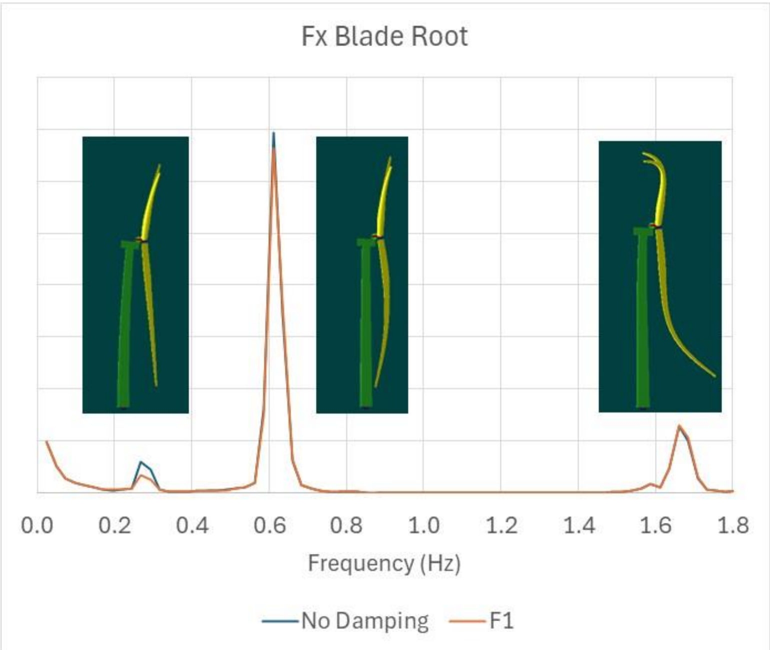

Figure 6. Blade of the 15MW IEA turbine

The retardation functions are shown in Figure 7. In the top graphs, the function used to dampen mode 1, then mode 2 and finally both modes 1 and 2 indicates that the function are closed to 0 in about 5s with the triangle function. For mode 1 damping, increasing the frequency range leads to a slower convergence to 0. Similarly, the triangle function is more efficient to converge to 0 than the step function. Faster convergence to 0 decreases the time for integration of the memory effect and improves the simulation time. Therefore, triangle functions are used for all results below. For the frequency range, other considerations might come into play (precision on the modal frequency).

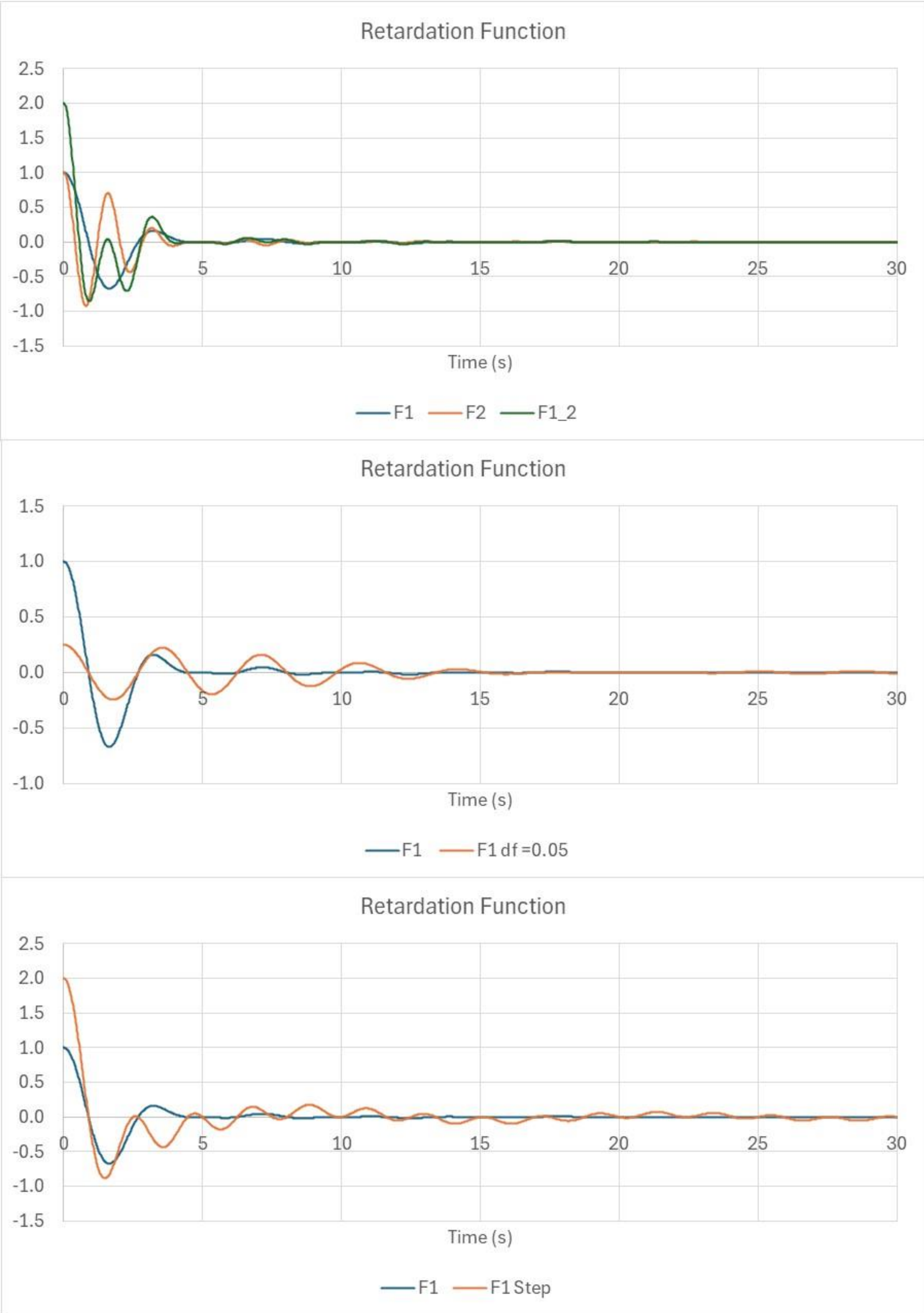

Figure 7. Retardation function for damping on frequencies F1, F2, F1 and F2 (top); with s range of +/-0.02Hz versus 0.05Hz for F1 (middle); with a triangle versus step function for F1 (bottom)

Coming back to the global turbine model, the damping formulation is checked by targeting either one mode or two modes. In Figure 8, the response spectrum of the force in the wind direction is plotted for cases with damping on F1 only, F2 only, F1 and F2, F1 and F3. The main observation is that any damping frequency can be targeted and damping of one frequency has only a very small effect on other frequencies. Therefore, damping amplitude can be defined independently per frequencies.

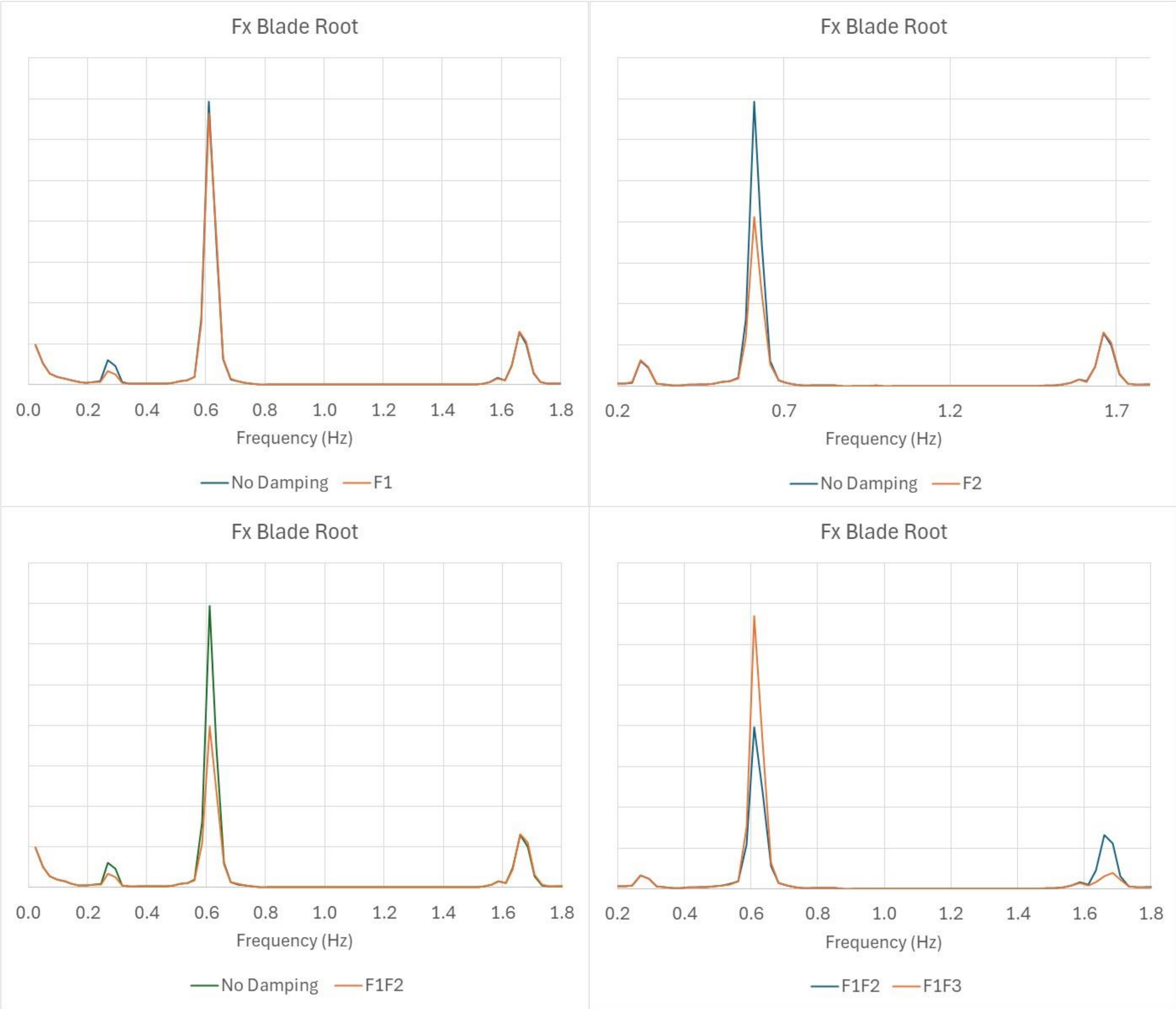

Figure 8. Response spectrum of bottom blade root force in the wind direction – damping on F1 only (top left) – damping on F2 only (top right) – damping on F1 and F2 (bottom left) – damping on F1 and F2 versus damping on F1 and F3 (bottom right)

A final test is performed with the turbine in production on a floater. The turbine is in a turbulent wind with an average hub height velocity of 12m/s. The wave peak period is at 12s. The tower modes in the x-direction (turbine out-of-plane) and y-direction (turbine in-plane) are respectively at 0.37Hz and 0.33Hz. Acceleration spectra in both directions are shown in Figure 9 with damping on one of the modes or both modes. Again, even for close modes, it is possible to pinpoint the frequency associated to a damping ratio. Indeed, the frequency at 0.37He can be damped without damping the frequency at 0.33Hz.

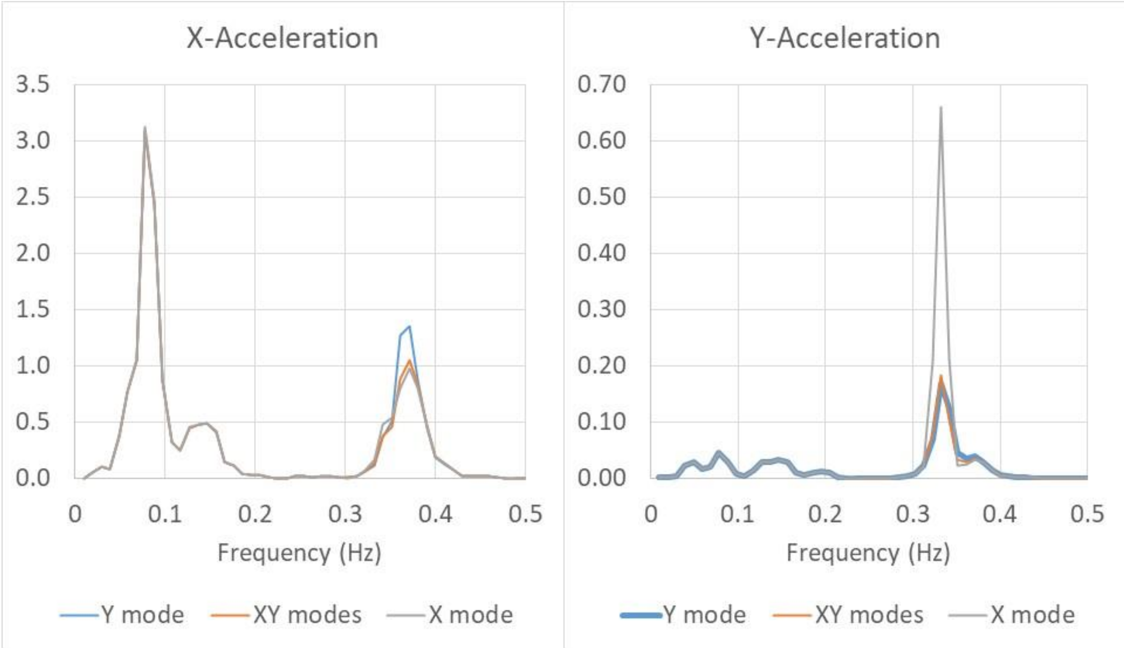

Figure 9. Response of acceleration at Nacelle, turbine in production in a 12m/s turbulent wind and 12s peak wave period – damping on in-plane/out-of-plane modes

III – 3 Relation to damping ratio

As shown in the previous sections, it is possible to target specific frequencies in the response spectrum and adjust the damping level. A remaining question is the amount of damping to introduce and therefore the link between the amplitude applied on the unitary functions and the Rayleigh coefficients or the modal damping ratio. Indeed, damping ratio (or logarithmic decrement) is a quantity which can be obtained from measurement of the system's response. The damping ratio is linked to the logarithmic decrement δ by:

$$\xi = \frac{\delta}{\sqrt{4\pi^2+\delta^2}} \text{ or } \xi = \frac{\delta}{2\pi},\ \delta \ll 1 \qquad (13)$$

A simple beam model is used, the beam is clamped at one end and free at the other. It is excited on the first second of the simulation by an imposed displacement at the clamped end and then the motion is stopped generating a decay test at the free end. The first mode of the beam is at 3.55Hz. As this is an excitation on a single mode, Rayleigh damping is defined with the stiffness term only with α =0 and b=βK. In Figure 10, decay tests are compared with Rayleigh damping and the triangle function with an amplitude a equal to the Rayleigh damping b. It indicates that the damping ratio are very similar with the two methods. Then for the retardation method, the bottom graph of Figure 10 indicates that the damping is linearly dependent on the amplitude but almost independent on the frequency range.

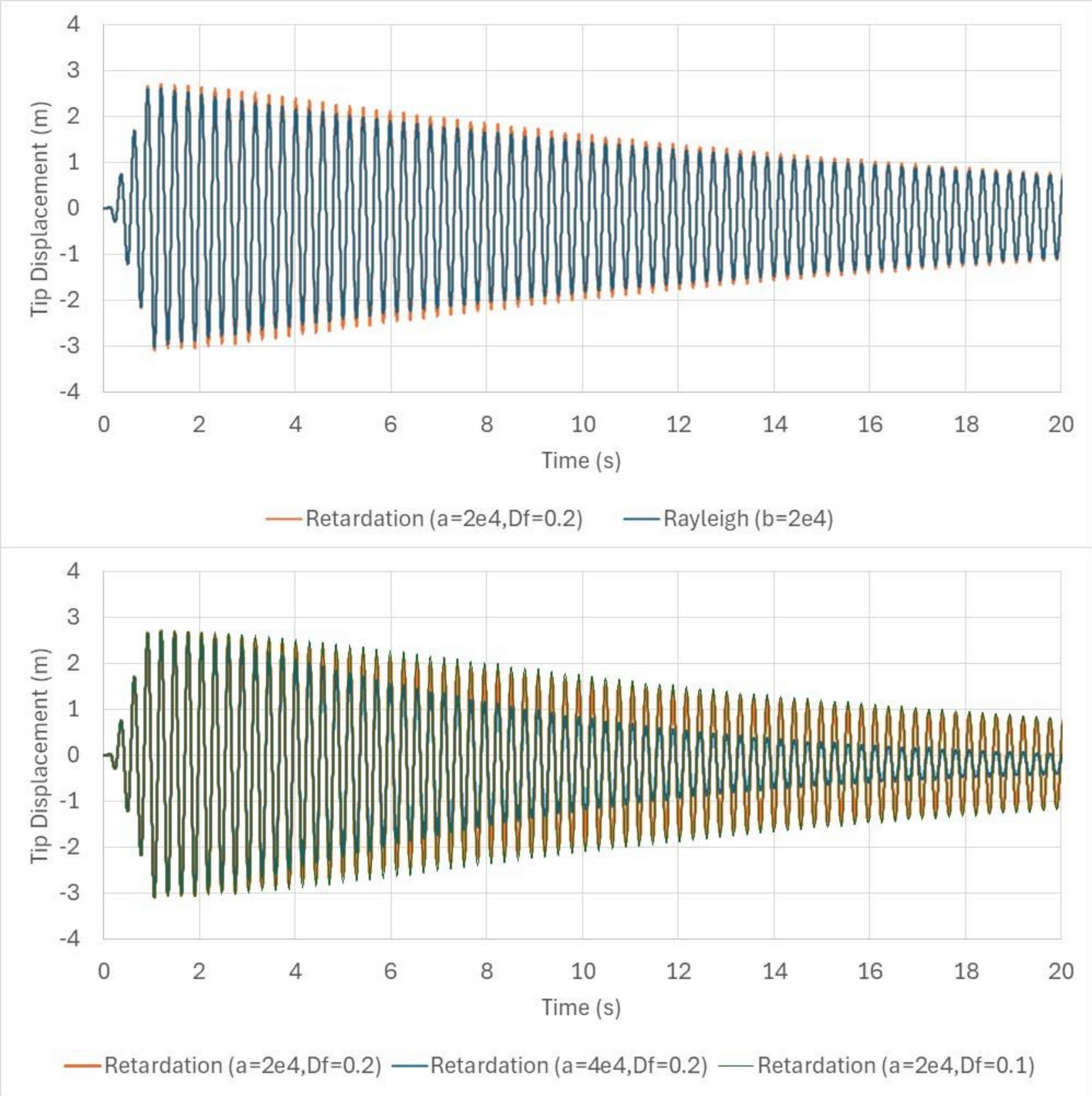

Figure 10. Decay test; Comparison Rayleigh damping and Retardation method (top) – Influence of amplitude and frequency range on retardation method (bottom)

For a simple sinusoidal motion, the displacement, velocity and damping force can be expressed as

$$x = x_o \sin \omega t,\ \dot{x} = x_o\, \omega \cos \omega t,\ F_d = \ b x_o\ \omega \cos \omega t \tag{14}$$

For the retardation damping with triangle function and considering $\Delta\omega = \Delta\omega^{+} = \Delta\omega^{-}$, the force is then:

$$F_d = \frac{2 a x_o \omega}{\pi \Delta\omega} [I_1\, cos\, \omega t + I_2\, sin\, \omega t] \tag{15}$$

With

$$I_1 = \int_0^t \frac{2\, cos^2\, \omega\tau - cos\, \omega\tau\ cos(\omega - \Delta\omega)\tau - cos\, \omega\tau cos(\omega + \Delta\omega)\tau}{\tau^2} d\tau \tag{16}$$

$$I_2 = \int_0^t \frac{2\, sin\, \omega\tau cos\, \omega\tau - sin\, \omega\tau\ cos(\omega - \Delta\omega)\tau - sin\, \omega\tau cos(\omega + \Delta\omega)\tau}{\tau^2} d\tau \tag{17}$$

The first integral is in phase with velocity while the second integral is in phase with acceleration. Focusing on I_1:

$$\lim_{t\to 0} I_1 = \Delta\omega^2\, t \tag{18}$$

And using [9]:

$$\lim_{t\to +\infty} \int^{t} \frac{\cos(cx)\cos(dx)}{x^2} dx = -\frac{\pi}{4}(|c-d| + |c+d|) \tag{19}$$

Then

$$\lim_{t\to +\infty} I_1 = \frac{\pi\Delta\omega}{2} \tag{20}$$

Comparing with the expression for the Rayleigh damping, the limits are then:

$$b \xrightarrow{t\to 0} \frac{2a\Delta\omega}{\pi} t, \quad b \xrightarrow{t\to +\infty} a \tag{21}$$

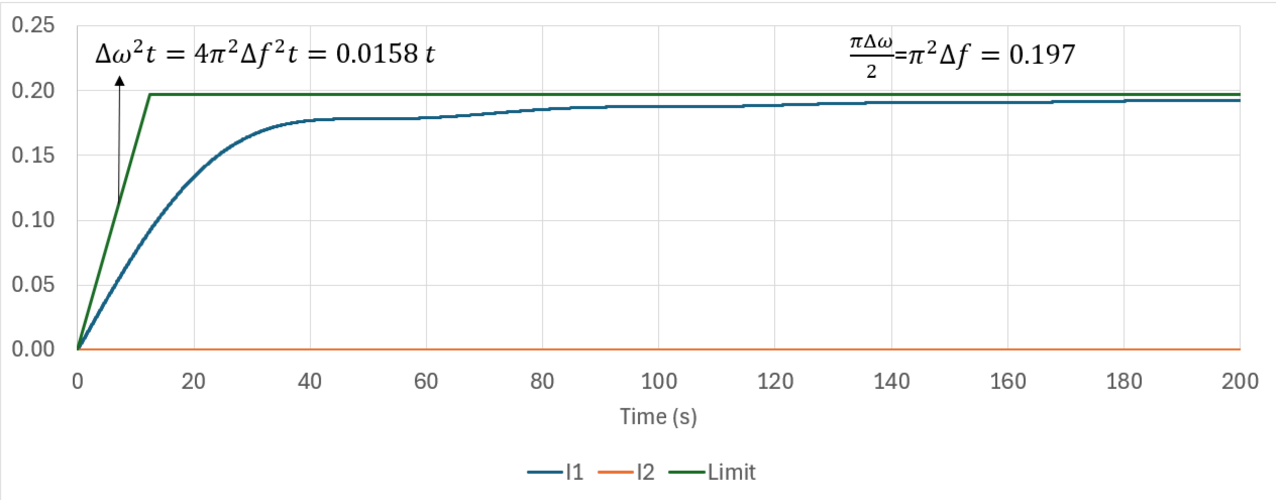

Figure 11. Function I_1 and I_2 together with I_1 limits

As presented in Figure 11, the damping force in phase with acceleration is negligible compared to the force in phase with the velocity. The amplitude defined on the unitary triangle function tends towards βK, therefore the retardation function for the i$^{\text{th}}$ mode defined by circular frequency ω_i and damping ratio ξ_i can be defined as:

$$\mathrm{R_i(t)} = \frac{2}{\pi} \int_0^{\infty} \frac{2\xi_i}{\omega_i} K\mathrm{B}(\omega) \cos\omega t \, d\omega \tag{22}$$

IV – Conclusion

In modeling turbine response to environmental loadings, specific damping ratios are provided by turbine suppliers either for individual blades of for the whole turbine and tower system. In time domain simulation (without projection on the modal basis), the classical Rayleigh damping formulation is too limited to introduce these ratios on several modes without impacting the global system response. A formulation based on a retardation function (an impulse response function) is used that is able to target individual frequencies with a specific damping amplitude and with minimal cross influence between frequencies response. Such formulation can then be used to model the complete floating wind turbine.

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