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Novel Aerodynamic Damping Identification Method for Operating Wind Turbines

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Abstract. This contribution introduces a novel method to determine the aerodynamic damping for operating wind turbines. Previous research typically estimated the modal damping ratios in the fore-aft and side-side directions as two decoupled degrees of freedom. This can result in misleading results, as the two directions are closely and unconventionally coupled through the wind-rotor interaction. This study proposes the identification of a novel type of aerodynamic damping matrix. This matrix arises from the linearization of the aerodynamic force resultant obtained from blade element momentum theory (blade modes not included). This linearized force is then applied to a beam finite element model of the tower with a lumped mass representing the rotor-nacelle assembly. This decoupled strategy efficiently describes the physics of the system including the coupling between the fore-aft and side-side motions. The identification of the damping matrix is shown to work for simulated wind time data.

1. Introduction
Damping is a key variable in wind turbine systems as it limits vibration amplitude around resonance [1] and significantly influences fatigue life [2]. Wind turbine tower design is often governed by fatigue limit

states due to the fluctuating nature of the wind load (and wave load offshore). For accurate fatigue analyses, a dynamic model with correctly identified damping is key for wind turbine design or assessing fatigue life and future maintenance needs. Among the different sources of damping in wind turbine systems, aerodynamic damping has the largest contribution when the wind turbine is in operation. To identify the aerodynamic damping, operational modal analysis (OMA), which uses the measured vibration responses caused by ambient excitation, is the preferred method since conventional experimental modal analysis requires controlled excitation, which is very difficult to achieve due to the large size of modern wind turbines. Aerodynamic damping identification for wind turbines have been carried out by researchers such as Hansen *et al.* [3], Devriendt *et al.* [4], Dong *et al.* [5], Ozbek and Rixen [6], Koukoura *et al.* [7] These studies estimated the modal aerodynamic damping ratios in the fore-aft (FA) and side-side (SS) directions using classic or modified identification methods in time or frequency domain. However, Tcherniak *et al.* [8] and Ozbek *et al.* [9] highlighted some difficulties in using OMA in the context of wind turbines.

Previous studies typically assumed that the FA and SS motions are decoupled as the damping identification techniques were applied on measured data only for FA or SS directions. This paper proposes a new damping identification method which falls under the umbrella of OMA in the sense that it uses ambient wind excitation as an input. However, it does not aim to estimate modal parameters but rather identify on a more realistic damping model. According to the authors' recent research [10], it was found that the rotating blades introduce significant damping coupling between the FA and SS directions for an operating wind turbine and this coupling can be expressed by an aerodynamic damping matrix. In terms of aerodynamic damping, the wind turbine dynamic behaviour is quite different from that traditionally assumed by decoupled FA and SS motions. Based on this new description of the damping, this contribution presents an operational identification methodology that directly extracts the aerodynamic damping matrix and applies it to a three-blade horizontal-axis wind turbine. This paper is organized as follows. Section 2 describes the modelling strategy used in the paper and then describes the identification procedure. Section 3 gives a preliminary assessment of the identification procedure on a turbine case study using simulated wind velocity time series. Section 4 concludes the paper.

## 2. Methodology

2.1. Model description

In this study, the wind-rotor interaction is modelled separately from the tower dynamics. The two systems are then coupled through the hub resultant forces as shown in Figure 1. The tower mainly vibrates in the FA direction (x) and SS direction (y), and the vertical vibration is neglected due to its small magnitude. The model includes the linear motions represented by $\dot{x}$ and $\dot{y}$ at the tower top as well as the angular motions represented by $\dot{\theta}_x$ and $\dot{\theta}_y$ around x and y axes respectively. The inflow wind V_0 is assumed to be uniform throughout the rotor and only has a component in the FA direction. The aerodynamic forces are derived theoretically using blade element momentum (BEM) theory in steady-state conditions ignoring any blade vibration. As a result, parametric excitation due to rotor dynamics are not included. The aerodynamic forces are linearized to the sum of terms corresponding to the forces for an assumed rigid tower, plus terms proportional to the tower top linear and angular velocities, which can be expressed as

$$\mathbf{F}_{Top}(t) = \mathbf{F}_{Top}^{Rigid}(t) - \mathbf{C}_{Aero}\dot{\mathbf{u}}_{Top}(t). \tag{1}$$

According to [10], $\mathbf{C}_{Aero}$ is

$$\mathbf{C}_{Aero} = \begin{bmatrix} c_{xx} & 0 & c_{x\theta_x} & 0 \\ 0 & c_{yy} & 0 & c_{y\theta_y} \\ c_{\theta_x x} & 0 & c_{\theta_x \theta_x} & 0 \\ 0 & c_{\theta_y y} & 0 & c_{\theta_y \theta_y} \end{bmatrix}, \tag{2}$$

where the off-diagonal terms representing the damping coupling terms are not symmetric and have analytical expressions derived from BEM. The terms in $\mathbf{C}_{Aero}$ are defined in Appendix 1. The aerodynamic damping matrix and $\mathbf{F}_{Top}^{Rigid}(t)$ are calculated in MATLAB for a particular parameter set of mean wind speed, rotor rotation speed and blade pitch angles and $\mathbf{F}_{Top}(t)$ as defined by Equation (1). This force is then applied at the top of a finite element (FE) model (also written in MATLAB) of the tower modelled as a cantilever made of 11 Euler-Bernoulli beams elements as shown schematically in Figure 1.
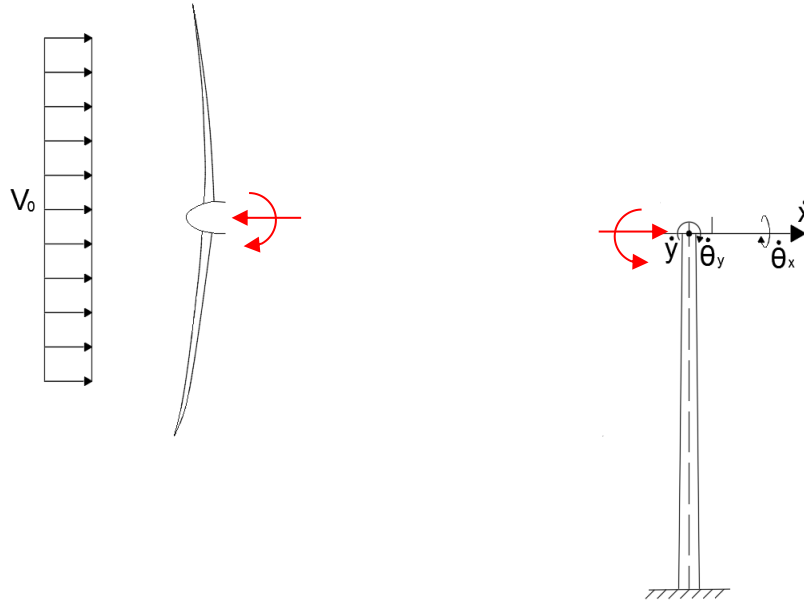
(a) Stage 1: rotor aerodynamic interaction.   (b) Stage 2: tower dynamics.

Figure 1. Schematic diagrams illustrating the two-stage modelling strategy.

The equation of motion for the wind turbine system can be originally written as

$$\mathbf{M}\ddot{\mathbf{u}}(t) + \mathbf{C}_{Struc}\dot{\mathbf{u}}(t) + \mathbf{K}\mathbf{u}(t) = \mathbf{F}(t), \tag{3}$$

where $\mathbf{M}$, $\mathbf{C}_{Struc}$ and $\mathbf{K}$ are the mass, structural damping and stiffness matrices respectively. $\mathbf{C}_{Struc}$, representing standard structural damping is assumed to be a Rayleigh damping matrix. $\mathbf{M}$, $\mathbf{C}_{Struc}$ and $\mathbf{K}$ are assumed to be known either through modelling or standard OMA on a parked turbine. $\mathbf{F}(t)$ is the external aerodynamic force vector which includes $\mathbf{F}_{Top}(t)$ at the relevant degrees of freedom, and $\mathbf{u}(t)$ is the generalised displacement vector. In this study, the aerodynamic force on the tower itself is not considered, so $\mathbf{F}(t)$ is only non-zero at the tower top degrees of freedom. After implementing the two-stage modelling strategy, the equation of motion for the wind turbine system can be changed to

$$\mathbf{M}\ddot{\mathbf{u}}(t) + \mathbf{C}\dot{\mathbf{u}}(t) + \mathbf{K}\mathbf{u}(t) = \mathbf{F}^{Rigid}(t), \tag{4}$$

where $\mathbf{C}$ is the addition of the damping matrices $\mathbf{C}_{Struc}$ and $\mathbf{C}_{Aero}$ (adding the terms of $\mathbf{C}_{Aero}$ at the relevant degrees of freedom in $\mathbf{C}_{Struct}$). In Equations (3) and (4), $\mathbf{F}(t)$ or $\mathbf{F}^{Rigid}(t)$ represent the

complete aerodynamic force vector applied to the flexible tower or the rigid tower, whose components corresponding to the tower top are expressed as $\mathbf{F}_{Top}(t)$ and $\mathbf{F}_{Top}^{Rigid}(t)$ in Equation (1). The model expressed by Equation (4) is referred to as the "full model", to distinguish it from the simplified two-degree of freedom (2-DOF) model which will be described in Section 2.3.

Table 1. Basic properties of the NREL 5MW reference onshore wind turbine.

Rotor Diameter, R	126m
Hub Height from MSL	87.6m
Tower Diameter, D	3.87-6.00m
Tower Thickness, t	19-27mm
Lumped Mass at Top	3.5×10^5 kg
Rated Wind Speed	12.1m/s
Natural Frequency	0.34 Hz

2.2. Model validation
This two-stage modelling strategy was validated against the fully-coupled aero-elastic package FAST provided by NREL [11] for inflow wind field with a range of constant and uniform wind speeds. Calculations were carried out using the NREL 5MW reference onshore wind turbine with main characteristics given in Table 1. Figure 2 shows time histories of the FA and SS tower top displacements following an initial displacement imposed in the FA direction. The responses generated by the full model agree well with the responses from FAST simulation.

(a)   (b)
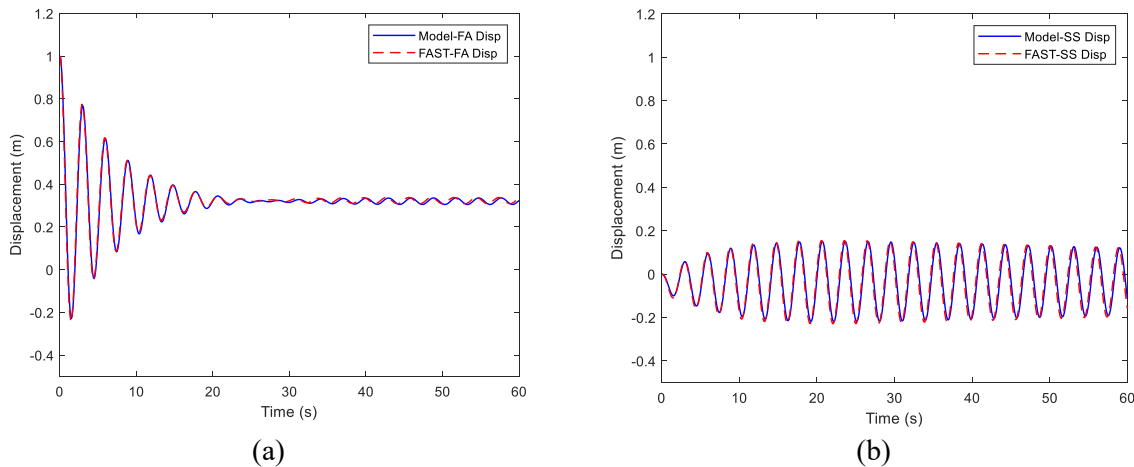
Figure 2. Comparison of the FA (a) and SS (b) vibration displacements between the full model and FAST simulation with initial displacement of the tower top in the FA direction, wind speed 10m/s.

2.3. Simplification to a 2-DOF model
In wind turbines, the FA and SS responses are dominated by the first FA and SS bending modes so the behaviour of the system can be efficiently described by only considering these two modal coordinates. Applying modal decomposition to Equations (3) and (4) using the first two bending mode shapes (FA: $\boldsymbol{\phi}_x$ and SS: $\boldsymbol{\phi}_y$), the two equations of motion for the first bending modes can be written as:

$$\bar{m}_x \ddot{\alpha}_x(t) + 2\bar{\zeta}_x \sqrt{\bar{m}_x \bar{k}_x}\, \dot{\alpha}_x(t) + \bar{k}_x \alpha_x(t) = \boldsymbol{\phi}_x^T \mathbf{F}_x(t),$$
$$\bar{m}_y \ddot{\alpha}_y(t) + 2\bar{\zeta}_y \sqrt{\bar{m}_y \bar{k}_y}\, \dot{\alpha}_y(t) + \bar{k}_y \alpha_y(t) = \boldsymbol{\phi}_y^T \mathbf{F}_y(t),$$

$$(5)$$

where $\bar{m}_x$, $\bar{m}_y$, $\bar{k}_x$ and $\bar{k}_y$ are the modal masses and stiffnesses for the first FA/SS mode respectively, $\bar{\zeta}_x$ and $\bar{\zeta}_y$ are the structural modal damping ratios, and $\alpha_x(t)$ and $\alpha_y(t)$ are the modal coordinates for the FA and SS modes respectively. $\mathbf{F}_x(t)$ and $\mathbf{F}_y(t)$ are the linearized aerodynamic forces which also include tower top velocity terms. These terms can be combined with the structural damping ratios into a 2x2 damping matrix

$$\bar{\mathbf{C}} = \begin{bmatrix} \bar{c}_{11} & \bar{c}_{12} \\ \bar{c}_{21} & \bar{c}_{22} \end{bmatrix}. \tag{6}$$

The above modal decomposition converts the full model to a simpler 2-DOF model, the coefficients of which will be identified in the following procedure outlined next.

2.4. Damping identification
Standard OMA implicitly assumes proportional damping and therefore calculates modal damping ratios. The damping identification method introduced by [12] was used instead as it does not assume that the damping matrix is symmetrical. It operates in the frequency domain with transfer functions and is outlined in Appendix 2. In practice, wind time series and structural responses could be obtained from actual measurements but in this paper, they were simulated. Wind time series were generated by inverse Fourier transform of a Kaimal Spectrum, using customised MATLAB code or TurbSim in FAST [13]. The turbulent wind time series can be converted into aerodynamic forces on a rigid turbine using steady BEM code. The 2-DOF model can generate the responses in FA and SS directions from these forces. Given the inputs (aerodynamic forces) and the outputs (the responses), the frequency response functions (FRFs) of the 2-DOF model can be estimated from the cross spectral density (CSD) matrices of the inputs and the outputs. The method calculates an auxiliary, frequency dependent damping matrix from the FRF matrix of the 2-DOF system. Damping coefficients are then determined by selecting an appropriate frequency range and averaging the results from the auxiliary damping matrix. The identification procedure is described schematically in Figure 3.
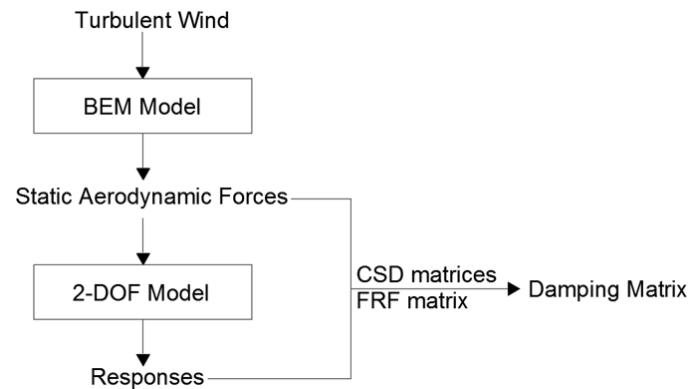

Figure 3. Schematic for the aerodynamic damping matrix identification procedure.

3. Results
The identification procedure developed was tested using the FE model established in MATLAB. The onshore 5MW NREL reference wind turbine from FAST provided the tower dynamic properties and the airfoil properties for the FE model. Given a particular mean wind speed, the terms in the 2x2 damping matrix can be calculated. These calculated values represent reference analytical values. To simulate a more realistic incoming wind flow, turbulent wind time series with same mean speed were used as input for the 2-DOF system to generate the system responses. The aerodynamic damping matrix was determined for every frequency point employing the identification procedure outlined above, using the

generated responses and the estimated aerodynamic forces applied to the rigid tower. The estimation of damping coefficients is eventually made by averaging the frequency-dependent damping coefficient estimates.

As an example, a mean wind speed equal to 20m/s is chosen to test the proposed damping identification procedure. With a predefined 1% structural damping ratio [1] for the first bending modes, the structural modal damping matrix is

$$1 \times 10^4 \times \begin{bmatrix} 1.7265 & 0 \\ 0 & 1.7265 \end{bmatrix} (N \cdot s/m),$$

and the reference aerodynamic damping matrix is

$$1 \times 10^5 \times \begin{bmatrix} 1.0805 & 0.2128 \\ 0.4138 & 0.1123 \end{bmatrix} (N \cdot s/m).$$

The estimation of the FRFs was obtained by averaging 5 simulations of 1-hour duration. If the number of simulations or the simulation duration increases, the estimation of FRFs is more accurate. After obtaining the FRFs for the 2-DOF model, the damping matrix can be estimated following the method by Chen *et al.*'s method [12]. The estimated frequency-dependent damping matrix is shown in Figure 4, fluctuating around the analytical solution. Averaging the frequency-dependent matrix from 0.01Hz to 0.3 Hz, a final estimation of the aerodynamic damping matrix after subtracting the structural modal damping matrix can be obtained, which is

$$1 \times 10^5 \times \begin{bmatrix} 1.0708 & 0.2318 \\ 0.4093 & 0.1208 \end{bmatrix} (N \cdot s/m).$$

The estimated aerodynamic matrix is very close to the predefined one above and it was found that the maximum difference is for C_{12} and less than 10%. Identified values for the four damping coefficients in the 2x2 damping matrix for mean wind speeds from 8m/s to 20m/s（2m/s interval）are plotted against the analytical values, as shown in Figure 5. It can be seen that the identified values are close to the analytical values.
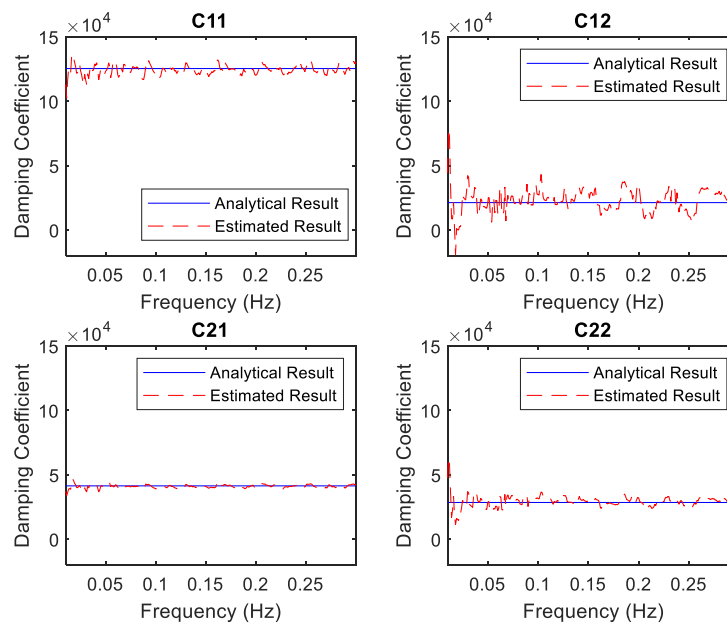
Figure 4. Comparison between analytical (blue solid) and identified results (red dashed) for the frequency dependent aerodynamic damping coefficients. (Unit: N·s/m).
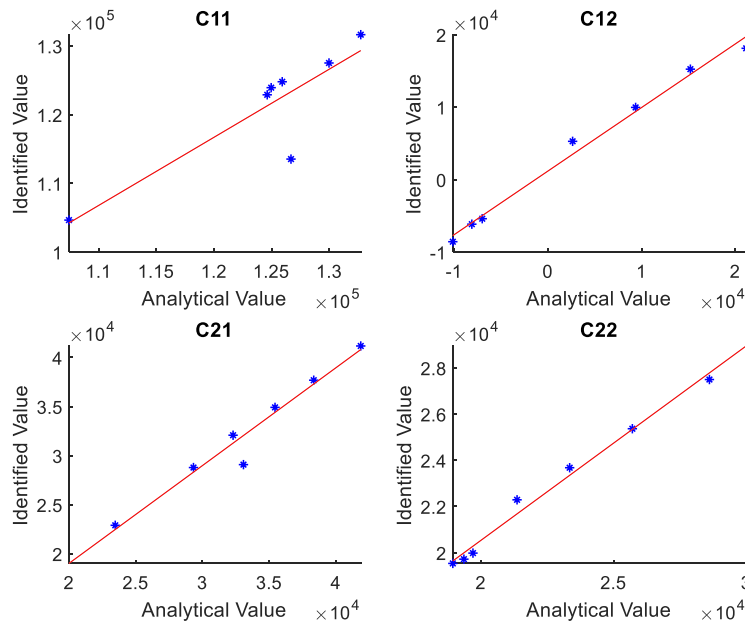

Figure 5. Analytical values against identified values after averaging for mean wind speeds from 8m/s to 20m/s in 2m/s steps (Unit: N·s/m).

4. Conclusion

This contribution presented the development of a new methodology to identify the aerodynamic damping in wind turbines in operation by using vibration responses caused by ambient excitation. The identification obtains the aerodynamic damping matrix rather than conventionally-used damping ratios. The coupling between FA and SS motions is included by the developed aerodynamic damping matrix and therefore the proposed identification method provides better information to characterise the aerodynamic damping in wind turbines, which is key for robust fatigue and reliability analyses. The methodology was devised so that it can be used in a practical situation. It requires wind time series, the dynamics response of the tower and modal properties of wind turbines in parked condition.

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Appendix 1

The terms in aerodynamic damping matrix in Equation (2) are defined by the following expressions according to [10]:

$$\mathbf{C}_{Aero} = \begin{bmatrix} c_{xx} & 0 & c_{x\theta_x} & 0 \\ 0 & c_{yy} & 0 & c_{y\theta_y} \\ c_{\theta_x x} & 0 & c_{\theta_x \theta_x} & 0 \\ 0 & c_{\theta_y y} & 0 & c_{\theta_y \theta_y} \end{bmatrix}$$

$$= \begin{bmatrix} N_b \int_0^R \frac{\partial(dT)}{\partial V_0} & 0 & -N_b \int_0^R r\frac{\partial(dT)}{\partial V_r} & 0 \\ 0 & -\frac{N_b}{2}\int_0^R \frac{\partial(dS)}{\partial V_r} & 0 & -\frac{N_b}{2}\int_0^R r\frac{\partial(dS)}{\partial V_0} \\ N_b \int_0^R r\frac{\partial(dS)}{\partial V_0} & 0 & -N_b \int_0^R r^2\frac{\partial(dS)}{\partial V_r} & 0 \\ 0 & \frac{N_b}{2}\int_0^R r\frac{\partial(dT)}{\partial V_r} & 0 & \frac{N_b}{2}\int_0^R r^2\frac{\partial(dT)}{\partial V_0} \end{bmatrix},$$

where dT and dS are the steady-state forces in normal and tangential directions respectively applied to one blade element at distance r from the hub. V_0 is an steady inflow wind velocity in the FA direction, V_r is the tangential speed of the blade element at distance r which is caused by the rotor rotation, N_b is the number of blades, R is the radius of the blade.

Appendix 2

Outline of Chen *et al.*'s method [12]
Given a dynamic system with mass, stiffness and damping matrices **M**, **K** and **C** excited by an external force **f**, the equation of motion is

$$\mathbf{M}\ddot{\mathbf{x}} + \mathbf{C}\dot{\mathbf{x}} + \mathbf{K}\mathbf{x} = \mathbf{f}.$$

Rewrite this equation of motion in frequency domain:

$$(-\omega^2\mathbf{M} + i\omega\mathbf{C} + \mathbf{K})\mathbf{X}(\omega) = \mathbf{F}(\omega).$$

And the frequency response function (FRF) matrix $\mathbf{H}(\omega)$ is defined as:

$$\mathbf{H}(\omega) = (-\omega^2\mathbf{M} + i\omega\mathbf{C} + \mathbf{K})^{-1}.$$

The "normal" FRF $\mathbf{H}^{\mathbf{N}}(\omega)$ is defined with the undamped system:

$$\mathbf{H}^{\mathbf{N}}(\omega) = [\mathbf{K} - \omega^2\mathbf{M}]^{-1}$$

With the "normal" FRF, the frequency domain equation of motion can be written as

$$[\mathbf{H}^{\mathbf{N}}(\omega)]^{-1}\mathbf{X}(\omega) + i\omega\mathbf{C}\mathbf{X}(\omega) = \mathbf{F}(\omega),$$

or

$$\mathbf{X}(\omega) + i\mathbf{G}(\omega)\mathbf{X}(\omega) = \mathbf{H}^N(\omega)\mathbf{F}(\omega),$$

where

$$\mathbf{G}(\omega) = \omega\mathbf{H}^N(\omega)\mathbf{C}.$$

Therefore, the relationship between the measured FRF $\mathbf{H}(\omega)$ and the "normal" FRF $\mathbf{H}^N(\omega)$ is

$$\mathbf{H}^N(\omega) = [\mathbf{I} + i\mathbf{G}(\omega)]\mathbf{H}(\omega),$$

where **I** is an identity matrix. Since $\mathbf{H}^N(\omega)$ and $\mathbf{G}(\omega)$ are real matrices, the imaginary part of the RHS in the above equation is zero, giving

$$\mathbf{G}(\omega) = -\mathrm{im}\big(\mathbf{H}(\omega)\big)\big[\mathrm{Re}\big(\mathbf{H}(\omega)\big)\big]^{-1}.$$

Finally, the damping matrix at any given frequency can be expressed by

$$\mathbf{C} = \frac{1}{\omega}[\mathbf{H}^{\mathbf{N}}(\omega)]^{-1}\mathbf{G}(\omega).$$

It should be noted that this damping identification method needs the prior knowledge of the stiffness and mass matrices.