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ACTIVE CONTROL OF VERTICAL RISERS UNDERGOING VORTEX-INDUCED VIBRATIONS

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ABSTRACT

This paper presents a first control strategy to reduce the vortex induced vibrations (VIV). First, the system equations and an associated modal analysis are presented. This modal analysis shows that, for low damped resonant frequencies, there is a phase shift of $\pm 90^\circ$ *between a periodic external force at the riser's top, and the generated vibration along the structure. This phase shift is used in the design of the control law to reduce the VIV along the structure. For some operating conditions, simulations show that the control system attenuates the VIV and reduces the vortex shedding synchronism along the structure. Simulations are presented with two different kinds of sea current profiles. The advantages of this strategy are the small external force required to reduce VIV, and the fact that no structural change is required along the structure submerged part. But a displacement sensor near the structure bottom is needed.*

NOMENCLATURE

A State matrix.
B Input matrix.
B_m Input matrix of modal base.
C Output matrix.
E Elastic modulus.
F Hydrodynamic force.
F_{VIV} Vortex shedding force.
G Damping matrix.
I Identity matrix.
J Second moment of area.
K Stiffness matrix.
P Vortex shedding acceleration vector.
P_m Vortex shedding acceleration vector of modal base.
Q Fluid variable.
T Tension.
U Flow speed.
V Eigenvector matrix.
$\widetilde{V}$ Modified eigenvector matrix.
X Structure state vector.
W Modal state vector.
Y Transverse riser displacement vector.
$\dot{Y}$ Transverse riser speed vector.
a Constant of structure force over the fluid.
b Mode associated gain.
g Filter damping constant.
h Fluid force constant.
i Imaginary unit.
m Riser and fluid linear mass.
m_F Fluid added linear mass.
m_S Riser linear mass.
n Number of discretization points.
s Laplace variable.
t Time.
u Riser top external force.
$\hat{u}$ Laplace transform of the riser top external force.
y System output.

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$\hat{y}$ Laplace transform of the system output.
$\hat{y}^P$ Laplace transform of the output part generated by P.
z Vertical direction.
Λ Modified eigenvalue matrix.
Υ Transverse riser displacement.
α Damping constant.
γ Eigenvector real part.
ε Van der Pol's equation parameter.
ι Transfer function constant.
κ Filter gain.
λ Eigenvalue.
φ Eigenvector imaginary part.
τ Drag constant.
ω Vortex shedding natural frequency.

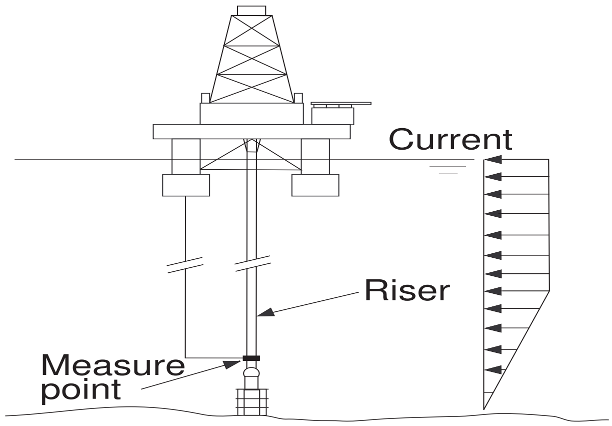

Figure 1. SYSTEM UNDER ANALYSIS

INTRODUCTION

Recently, large oil and gas reserves have been discovered in deep water far from shore. Drilling and exploration in those areas represent new technological challenges for the offshore oil industry. One of these challenges is the mechanical fatigue of extremely long structures that link the platform to the seabed and sometimes to the shore. In regions with important sea currents, a known source of mechanical fatigue is an alternating vortex shedding that periodically generates lift forces on the structure. These periodic forces generate vibrations of the structure, that are called vortex induced vibrations (VIV). The slender structures in a flow like mooring cables, catenary and vertical risers are the most affected by this phenomenon.

Instead of trying to reduce vibrations by changing the behavior of the flow around the structure, this paper proposes a system that uses an active control at the structure's top, to change the dynamic behavior of the structure. The fact that movements at the structure's top can reduce the VIV has been observed by Vandiver [1]. The present paper focuses on this aspect, and proposes a preliminary control strategy to minimize VIV of slender structures. The objective is to design a control system that reduces the VIV, while requiring the smallest external force at the riser's top (minimum energy requirement).

For control design, the idea is to consider the displacements of the structure points as the sum of two displacements. The first displacement accounts for the result of movements at the structure's top, the structure being described by a linear model. This model assumes small displacements (small angles). It is the linearized Euler-Bernoulli model for a constant section beam under axial traction, completed by the linear part of the hydrodynamic forces. It gives a relationship between the control u (force applied at the structure's top) and the controlled output y (displacement of a structure point located near the structure's bottom). The second displacement is due to nonlinear effects (non linear part of the hydrodynamic forces, vortex shedding). It is considered as a disturbance in the control design, even though forced displacements at the structure's top modify this disturbance, and in particular can amplify it.

The relationship between u and y that is provided by the linear partial differential equation is still too complicated for control design. We rely upon the fact that structures with low damping undergoing VIV behave like oscillators (oscillating at frequencies corresponding to the structure resonant modes). We then approximate the input-output relationship between u and y by a second order transfer function, corresponding to the most excited mode. It is derived from a modal analysis, on the spatially discretized linear partial differential equation. The control law derived from this transfer function is tested in simulations on a complete model, that couples the lift forces and the structure displacement.

The paper is organized as follows. First, we present the governing equations, the discrete model and the modal analysis, that emphasizes some specific aspects of the slender structures undergoing VIV: linear behavior of the structure for small lateral displacements, convergence of the vibration frequencies to one or some natural frequencies of the structure, synchronism and modal behavior of the structure. Then, we propose a control strategy (see [2]) to reduce the vibrations along the riser, and thus minimize the fatigue of the structure. This control tends to decrease VIV amplitude, at least in the cases under consideration. Limitations of this approach are studied in the last section. They are mainly due to the fact that the impact of amplitude reduction on vortex shedding has not been fully analyzed yet.

GOVERNING EQUATIONS

Vertical slender structures with small transverse displacements can be analyzed (see [3–5]) as a linearized Euler-Bernoulli beam with a constant section, under an axial traction plus external forces from the fluid:

$$m_S \frac{\partial^2 \Upsilon}{\partial t^2} = -EJ \frac{\partial^4 \Upsilon}{\partial z^4} + \frac{\partial}{\partial z} T(z) \frac{\partial \Upsilon}{\partial z} + F(z,t) \quad (1)$$

The hydrodynamic force $F(z,t)$ is approximated by a linear part inspired by the Morison's equation, and a non linear part represented by $F_{VIV}(z,t)$. We have

$$F(z,t) = F_{VIV}(z,t) - m_F \frac{\partial^2 \Upsilon}{\partial t^2} - \tau \frac{\partial \Upsilon}{\partial t} U(z) \quad (2)$$

Considering τ as the drag constant and m_F as the fluid added mass, noting $m = m_S + m_F$, we get

$$m \frac{\partial^2 \Upsilon}{\partial t^2} = -EJ \frac{\partial^4 \Upsilon}{\partial z^4} + \frac{\partial}{\partial z} T(z) \frac{\partial \Upsilon}{\partial z} - \tau \frac{\partial \Upsilon}{\partial t} U(z) + F_{VIV}(z,t) \quad (3)$$

The VIV present a constant amplitude when the lift force F_{VIV} compensates the drag force $-\tau \partial \Upsilon / \partial t U(z)$. The partial differential equation given by Eqn. (3) can be approximated by an ordinary differential system, using the finite difference method to discretize Υ along the $z-$axis (see [6]). Choosing n sections and concentrating the total mass of each section in its geometric center, denoting Y_j the transverse displacement for section j ($\dot{Y}_j$ denotes the transverse speed for section j), the system state X is given by $X = (Y_1, \ldots, Y_n, \dot{Y}_1, \ldots, \dot{Y}_n)^T$. The considered control u is the external force at the riser top, which, by the boundary conditions, results to a proportional displacement of the riser top. Summing $F_{VIV}(z,t)$ over z for section j and dividing by m gives an acceleration P_j. These accelerations can be grouped in the $n-$dimensional vector P. Finally, this gives

$$\dot{X} = AX + Bu + \begin{pmatrix} 0 \\ P \end{pmatrix}, \text{ with } A = \begin{pmatrix} 0 & I \\ -K & -G \end{pmatrix} \quad (4)$$

The stiffness matrix K contains the structure internal accelerations, and G denotes the damping matrix. Vector B is the acceleration over the structure associated to the riser top external force. We choose as system output y the position of the geometric center of a section, preferably close to the riser bottom. The output equation can be expressed as $y = CX$ for a given row-matrix C with only one non-zero entry.

MODAL ANALYSIS

The VIV frequencies belong to the set of the structure's natural frequencies [7]. So, structures with low damping undergoing VIV, have a behavior close to the behavior of the resonant modes corresponding to the VIV frequencies. One way to better visualize this phenomenon is to use a change of coordinates to express the system in its so-called modal base (for details, see [6, 8]).

Let us first study the eigendecomposition of A. We will use this decomposition to propose a new change of coordinates. Consider the eigenvector matrix V. It contains two kinds of columns (eigenvectors): one, corresponding to negative real eigenvalues, with real entries, and the other, pairwise, corresponding to complex conjugate eigenvalues λ_j and $\bar{\lambda}_j$, and containing complex conjugates values. The latter pairs of columns can be replaced by two columns, the first with the real part γ, and the other with the imaginary part φ. This leads to a new change of coordinates matrix $\widetilde{V}$. In these coordinates, the dynamics by Eqn. (4) with $\Lambda = \widetilde{V}^{-1} A \widetilde{V}$, $W = \widetilde{V}^{-1} X$, $B_M = \widetilde{V}^{-1} B$ and $P_M = \widetilde{V}^{-1}(0,P)^T$, writes

$$\dot{W} = \Lambda W + B_M u + P_M \quad (5)$$

where Λ (only with real entries) has the form

$$\Lambda = \begin{pmatrix} \lambda_1 & & & & \\ & \ddots & & & 0 \\ & & Re(\lambda_j) & Im(\lambda_j) & \\ & & -Im(\lambda_j) & Re(\lambda_j) & \\ & 0 & & & \ddots \\ & & & & & \lambda_{2n} \end{pmatrix} \quad (6)$$

and $(\lambda_j, \bar{\lambda}_j)$ is a pair of complex conjugate eigenvalues with negative real parts. The dynamics of the pair of elements $(W_j, W_{j+1})^T$ is given by

$$\begin{aligned} \begin{pmatrix} \dot{W}_j \\ \dot{W}_{j+1} \end{pmatrix} &= \begin{pmatrix} Re(\lambda_j) & Im(\lambda_j) \\ -Im(\lambda_j) & Re(\lambda_j) \end{pmatrix} \begin{pmatrix} W_j \\ W_{j+1} \end{pmatrix} \\ &+ \begin{pmatrix} B_{M_j} \\ B_{M_{j+1}} \end{pmatrix} u(t) + \begin{pmatrix} P_{M_j} \\ P_{M_{j+1}} \end{pmatrix} \end{aligned} \quad (7)$$

The output equation is transformed into $y = C_M W$, with $C_M = CV$. It is worth recalling that the meaning of input u and output y are not modified by the change of coordinates. In particular, y can still be viewed as a position. This is also true in the sequel, when model reduction is considered. The modal base shows that, for vibrations at a frequency corresponding to a low damped mode, the amplitudes associated to the excited mode are much larger than the amplitudes of the other modes. In this case, the structure dynamics can be approximated by the dynamics of the corresponding excited mode. Figure 2 compares the Bode diagram of a 200th order system with the Bode diagram of a second order model that only represents one system mode. This is

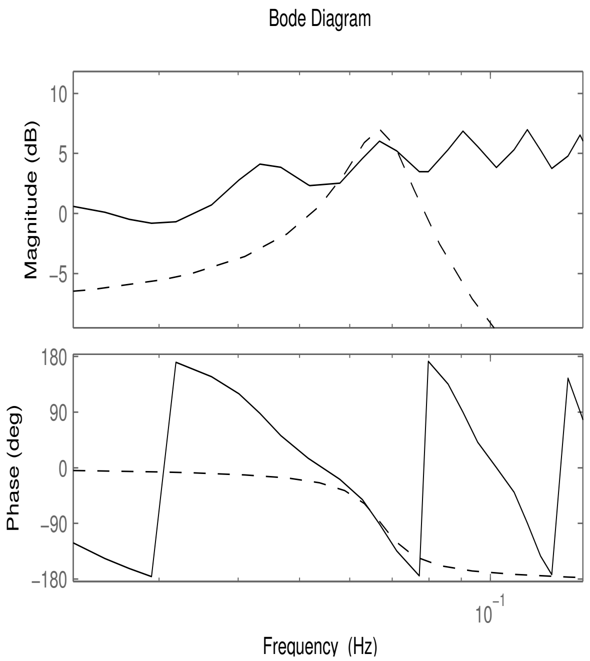

Figure 2. BODE DIAGRAMS OF THE STRUCTURE (CONTINUOUS LINE) AND THE SINGLE MODE MODEL

a key aspect to understand the behavior of a structure undergoing VIV, because VIV have large amplitudes only when they are associated to these frequencies.

This reduction considers that the structure is synchronized with the VIV. Its points either have a displacement in phase (0°) or in opposite phase (180°), so that the phase shift between the displacements of two points is close to 0° or 180°. Note that the punctual force applied to the structure top is an exception, since it locally deforms the structure and changes the behavior of this small part of the structure. This local deformation can not be represented by a single mode system.

Expressing the transfer function between the top external force $\hat{u}(s)$ and the displacement $\hat{y}(s)$ of another point of the structure, we get (s denotes the Laplace variable and $\hat{x}(s)$ denotes the Laplace transform of $x(t)$)

$$\frac{\hat{y}(s)}{\hat{u}(s)} = \frac{b_j(\iota_j s + 1)}{s^2 - (\lambda_j + \overline{\lambda}_j)s + |\lambda_j|^2} \tag{8}$$

In the specific case of structures undergoing VIV, the damping along the structure is small and has soft variations along z. That implies $\iota_j \ll 1$ (see [9]). The interesting point of the modal reduction is that for a vibration with a frequency equal to one natural frequency ($s = i|\lambda_j|$), the displacements are orthogonal (phase $= \pm 90°$) to the force phase that generates them. We have:

$$\frac{\hat{y}(i|\lambda_j|)}{\hat{u}(i|\lambda_j|)} = \frac{b_j(\iota_j i|\lambda_j| + 1)}{2Re(\lambda)|\lambda_j|} i \tag{9}$$

In other words, considering a force with the same frequency as a natural mode, for all the points in phase with the force application point ($b_j > 0$), the phase between the displacements of these points and the force is $-90°$; for the other points ($b_j < 0$), the phase is 90°. This property allows to determine the required phase relationship between a measured vibration and the top riser external force, in order to generate a vibration in opposite phase (anti-resonance effect). Consider a vibration associated to a mode for which the structure presents, at the measure point, a phase of $-90°$ (resp. 90°) between the top external force and the displacement at the measured point ($b_j > 0$, resp. $b_j < 0$): the force should have a phase shift of $-90°$ (resp. 90°) with respect to the measured signal, as that gives a generated vibration with a phase shift of $-180°$ (resp. 180°) with respect to the measured vibration ($(-90°) + (-90°) = -180°$).

CONTROL STRATEGY

The objective is to attenuate the vibration associated to the most excited frequency, using a control based on this frequency. This choice for the control design benefits from the fact that VIV frequencies are low damped and that the vortex shedding along the structure amplifies small displacements at this frequency. These two points help the control to propagate along the structure, and reduce the control amplitudes required to modify the whole structure behavior.

The idea is to generate a vibration in opposite phase ($\pm 180°$) to the VIV, in order to neutralize it. The control strategy uses some system information: VIV have large amplitudes just in natural frequencies [10], and for these frequencies, the modal analysis shows that structure displacements are orthogonal ($\pm 90°$) to the forces that generate them. In other words, using Eqn. (9) and $\hat{y}^P(s)$ as the Laplace transform of the vibration induced by P, we can write in the Laplace domain

$$\hat{y}(s) = \frac{b_j(\iota_j s + 1)}{s^2 - (\lambda_j + \overline{\lambda}_j)s + |\lambda_j|^2} \hat{u}(s) + \hat{y}^P(s) \tag{10}$$

Considering the system behavior at the natural frequency ($s = i|\lambda_j|$) and an external force at the riser top of the form $\hat{u}(i|\lambda_j|) = -i\kappa\hat{y}(i|\lambda_j|)$, Eqn. (10) becomes

$$\hat{y}(i|\lambda_j|) = \frac{-b_j \kappa \hat{y}(i|\lambda_j|)}{2Re(\lambda_j)|\lambda_j|} + \hat{y}^P(i|\lambda_j|) \tag{11}$$

Rewriting Eqn. (11), we obtain a relationship between the effect of vortex shedding and the structure vibration:

$$\hat{y}(i|\lambda_j|) = \hat{y}^P(i|\lambda_j|)(1 - \frac{b_j\kappa}{2Re(\lambda_j)|\lambda_j|})^{-1} \tag{12}$$

According to Eqn. (12), as $b_j\kappa > 0$, the influence of an unmodified disturbance $\hat{y}^P$ on $\hat{y}$ is reduced at the resonance frequency. However, it must be noticed that:

(i) According to simulations not reported in this paper, the gain κ must be kept sufficiently small. Otherwise, movements at the structure's top become too large, regarding the assumptions made in the control design: the structure behavior cannot be reduced to the behavior of a simple oscillator anymore. For instance, for large κ, the structure can suffer from amplified vibrations, due to a lock in with vortex shedding, at another natural frequency.
(ii) It is well known, that disturbance $\hat{y}^P$ increases when the amplitude of $\hat{y}$ is reduced, i.e. when control is applied. That such increases can always be compensated by a suitable choice of κ is an open question. Furthermore, the global stability of the controlled system remains to be proved. However, at least for the cases under consideration in this paper (and indeed for all the cases that we have studied so far), this phenomenon is not critical. Yet, as it can be seen, for example in Fig. 3 and Fig. 4, the control can slightly increase the mean amplitude of the vibrations for some points of the structure.

The control system is made of two physical parts: an actuator to impose horizontal forces at the riser's top, and two sensors, one close to the riser top, and the other close to the riser bottom. The system starts in a VIV detection mode. In this mode, the system measures the vibrations at the two points, detects their amplitudes and the phase shift between the two. Once the vibration crosses a specified threshold, a control law is generated, considering the main frequency of the VIV and the phase shift between the displacement at the bottom measure point and the force application point (this part identifies if b_j is positive or negative). This control law is designed with three features:

(i) A bandpass effect, that makes the control system only react to the dynamics of the main frequency, excited by the vortex shedding. This condition avoids the control system to generate vibrations in other frequencies.
(ii) A phase shifter, to generate a force at the same frequency, but in opposite phase to the VIV.
(iii) A stationary gain κ, between the measured vibration and the force to be applied.

This can be implemented through a linear control law with transfer function:

$$\frac{\hat{u}(s)}{\hat{y}(s)} = \frac{2\kappa\alpha s}{s^2 + 2\alpha s + |\lambda_j|^2} \times \frac{\pm(|\lambda_j| - s)}{|\lambda_j| + s} \tag{13}$$

Notice that, as the vibrations are perpendicular to the current, the reference position at rest always corresponds to $y = 0$. That is why no reference appears in Eqn. (13). This transfer function combines two terms in series. The first term is a second order bandpass. It filters the signal, in order to keep just a narrow frequency band around the excited natural frequency. It also defines the gain between the measured signal and the top riser external force. The second term is a phase shifter, that gives $\pm 90°$ depending on the sign of b_j, identified during the VIV detection mode.

The width of the frequency band is based on the precision of the main frequency estimation. In the transfer function represented at Eqn. (13), the bandwidth is related to the damping constant α. Replacing $s = |\lambda_j|$ in Eqn. (13), it is readily verified that this transfer function satisfies the condition $\hat{u}(i|\lambda_j|) = -i\kappa\hat{y}(i|\lambda_j|)$.

WAKE MODEL

The model presented in Eqn. (3) used to design the controller is not rich enough to suitably simulate the whole system behavior. This is why we introduce a wake model only used for simulations.

Several models have been designed to describe the forces associated to a relative movement between a solid and a fluid. A common fact for all these models is a nonlinear damping ratio that always increases with the speed of the relative movement. This nonlinear effect causes a kinetic energy concentration around some frequencies, because the displacements associated to one frequency increase the damping ratios for the other frequencies. In other words, the fact that one frequency is excited helps to reduce the lift force for the other frequencies.

Our interest is to use a wake model that is complicated enough to represent the main aspects of VIV (frequency concentration of the movement, phase lock between the structure and the vortex shedding at the natural frequencies of the structure), but kept simple enough for numerical simulations to be tractable. The model chosen to represent the nonlinear fluid forces is the wake oscillator model described by Facchinetti [11], and compared to direct numerical simulations of Navier-Stokes equations in [12]. It can be written as:

$$\begin{cases} F_{VIV} = hU(z)^2 Q \\ \dfrac{\partial^2 Q}{\partial t^2} = -\varepsilon U(z)(Q^2 - 1)\dfrac{\partial Q}{\partial t} - (\omega U(z))^2 Q + a\dfrac{\partial^2 \Upsilon}{\partial t^2} \end{cases} \tag{14}$$

Table 1. STRUCTURE DIMENSIONS AND CONSTANTS.

Type	Value	Unit
Height	10^3	m
External Diameter	.2	m
Internal Diameter	.18	m
Elastic Modulus	2.1×10^{11}	Pa
Steel Density	7.8×10^3	kg/m^3
Water Density	10^3	kg/m^3

This model proposes a phenomenological model to describe the wake dynamics, using a Van der Pol equation, instead of using Navier-Stokes equations by direct simulation. The term ω in the second equation represents the natural frequency of the vortex shedding. Replacing Eqn. (14) in Eqn. (3), the complete system model is

$$\begin{cases} m\dfrac{\partial^2 \Upsilon}{\partial t^2} = -EJ\dfrac{\partial^4 \Upsilon}{\partial z^4} + \dfrac{\partial}{\partial z}T(z)\dfrac{\partial \Upsilon}{\partial z} - \tau U(z)\dfrac{\partial \Upsilon}{\partial t} + hU(z)^2 Q \\ \dfrac{\partial^2 Q}{\partial t^2} = -\varepsilon U(z)(Q^2-1)\dfrac{\partial Q}{\partial t} - (\omega U(z))^2 q + a\dfrac{\partial^2 \Upsilon}{\partial t^2} \end{cases} \tag{15}$$

This model suitably represents the fact that, when the vortex shedding frequency is close to a natural frequency of the structure, the amplitude of the riser vibration is largely increased (this can be shown and will be the topic of a forthcoming paper).

NUMERICAL SIMULATIONS

Equation (15) is discretized the same way than Eqn. (3). The simulated structure is a steel vertical riser held by rotary joints at both ends. The tension at the riser bottom is taken equal to zero. Geometric dimension and other constants used to simulate the system are presented in Tab. 1. Two different cases are simulated. The first case represents the structure totally immersed, with a constant flow along the structure. The considered flow speed is 0.1m/s. The second case represents a marine current that is constant with respect to time, but linearly increases from the sea bed ($z = 0$) to the surface ($U(z) = 0.05 + 0.00005z$). We choose two different profiles of sea current that excite the same mode. The interest of this choice is to show that the structure cross displacement has a form mainly defined by the structure and its most excited mode, and that the sea current profile is important to define the amplitude envelope.

Note that the structure behavior is much more complex than the one used for the control design. However simulations show that the vibration control suitably reduces the vibration envelope.

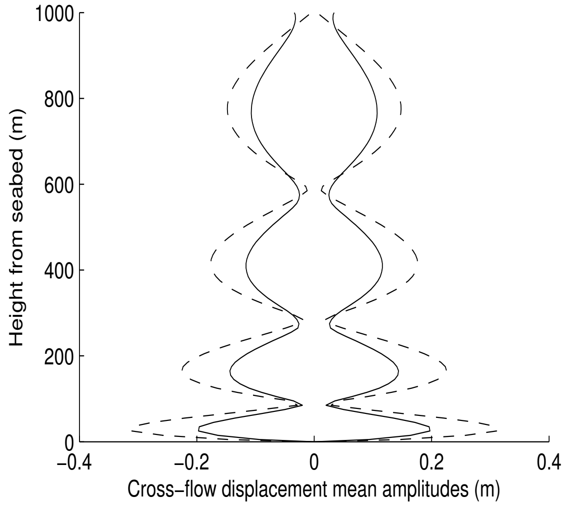

Figure 3. CROSS-FLOW DISPLACEMENT FOR A CONSTANT SEA CURRENT: OPEN LOOP(DASHED LINE) CLOSED LOOP (CONTINUOUS LINE)

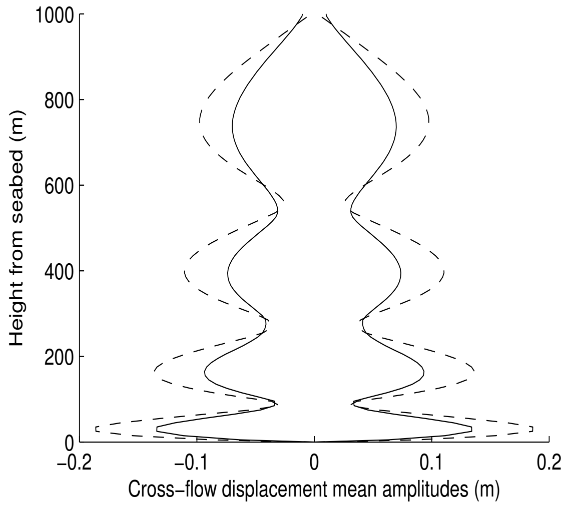

Figure 4. CROSS-FLOW DISPLACEMENT FOR A VARIABLE SEA CURRENT: OPEN LOOP(DASHED LINE) CLOSED LOOP (CONTINUOUS LINE)

The structure envelope is close to the modal envelope defined by the associated eigenvectors $(V_j, V_{j+1})^T$. In figures 3 to 10, the vibrations are compared considering the following four aspects: cross-flow displacements, speeds, highest punctual displacement and its Fourier transform. The interest is to compare the vibration intensity and to observe the impact of the feedback control along the structure and on the other structure modes.

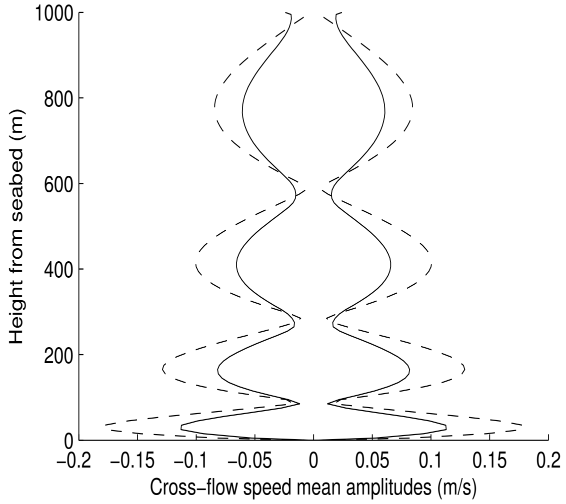

Figure 5. CROSS-FLOW SPEED FOR A CONSTANT SEA CURRENT: OPEN LOOP(DASHED LINE) CLOSED LOOP (CONTINUOUS LINE)

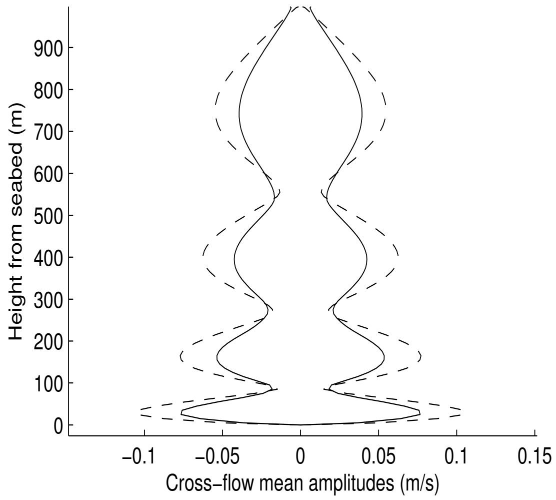

Figure 6. CROSS-FLOW SPEED FOR A VARIABLE SEA CURRENT: OPEN LOOP(DASHED LINE) CLOSED LOOP (CONTINUOUS LINE)

The cross-flow displacements presented in Fig. 3 and Fig. 4 present a displacement reduction of 35% at the critical points. It is interesting to observe a small amplitude amplification in the minimum displacement regions. This phenomenon happens because of the non linear damping. A vibration reduction in the main frequency increases the lift forces, and consequently increases the vibrations associated to other modes.

The cross-flow speeds presented in Fig. 5 and Fig. 6 show an average speed decrease of almost 35%, the same rate as for the

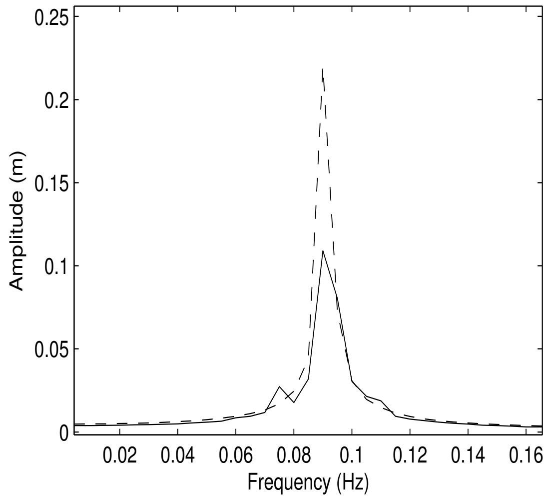

Figure 7. CROSS-FLOW HIGHEST VIBRATION FOURIER TRANSFORM FOR A CONSTANT SEA CURRENT: OPEN LOOP(DASHED LINE) CLOSED LOOP (CONTINUOUS LINE)

displacement reduction. This point shows that the structure under active control keeps the same relation between displacement and speed, this happening because the structure continues to oscillate with the same open loop natural main frequencies.

Figure 7 and Fig. 8 show an important reduction of the vibration associated to the main frequency. They also show a slight amplification of other modes, because the lift force increases when the structure mean speed is reduced. These other frequencies remain of minor importance for the structure fatigue, according to their small amplitudes and to the fact that the displacement peaks do not coincide.

Figure 9 shows a single mode open loop behavior due to the uniform current profile, which becomes multi-modal in closed loop, thus showing the lock in attenuation at the structure natural frequency closest to the vortex shedding natural frequency. Fig. 10 shows this tendency even for open loop multi-modal behavior.

LIMITATIONS

The limitations of this control law are due to the following assumptions:

(i) Small-angle approximation (linear structure behavior).
(ii) Smooth current profile to insure that ι_j in Eqn. (9) is small enough for the 90° phase shift assumption to hold true.
(iii) Vibration mainly concentrated on a single resonant mode.
(iv) Uncoupling of the control reaction with the perturbation amplitude (the lift force increases when the vibration is

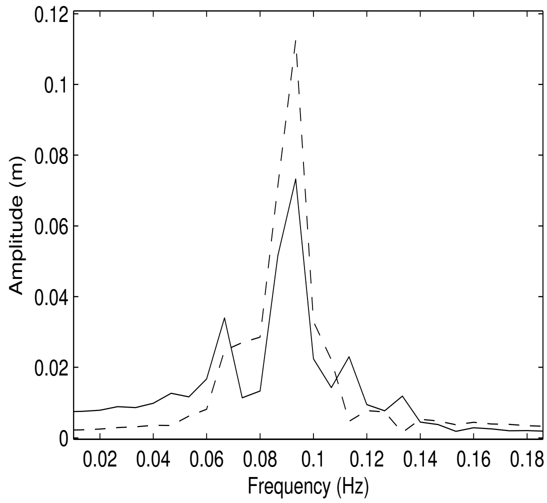

Figure 8. CROSS-FLOW HIGHEST VIBRATION FOURIER TRANSFORM FOR A VARIABLE SEA CURRENT: OPEN LOOP(DASHED LINE) CLOSED LOOP (CONTINUOUS LINE)

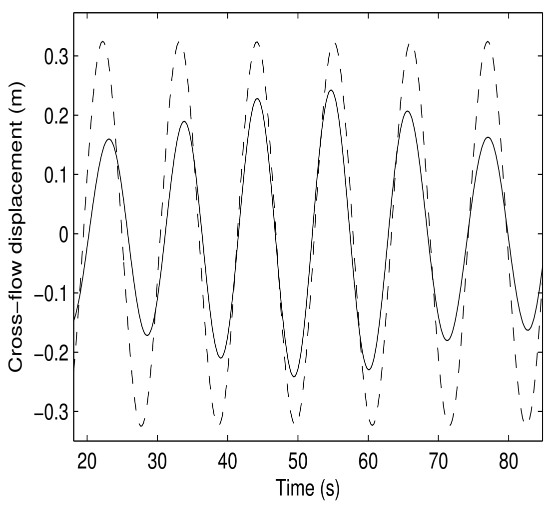

Figure 9. CROSS-FLOW HIGHEST DISPLACEMENT FOR A CONSTANT SEA CURRENT: OPEN LOOP(DASHED LINE) CLOSED LOOP (CONTINUOUS LINE)

reduced).

(v) The wake model used was compared and validated with experimental data just in the cases of structures without an active control.

Further investigations are needed to test if this theoretic limitations really affect the results of our control approach.

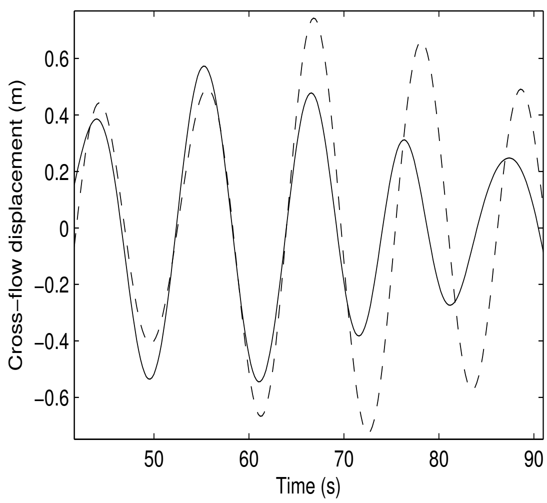

Figure 10. CROSS-FLOW HIGHEST DISPLACEMENT FOR A VARIABLE SEA CURRENT: OPEN LOOP(DASHED LINE) CLOSED LOOP (CONTINUOUS LINE)

CONCLUSION

We present preliminary results for an active control VIV reduction with small top forces and associated displacements, using two sensors to on-line identify the main vibration frequency and amplitude, thus eliminating the need to estimate which structure mode is excited.

In the considered simulation conditions, displacement and speed reduction, of 35% along all the structure, are obtained with a riser top displacement of about 10% of the maximum punctual displacement.

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