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STRUCTURAL RELIABILITY THEORY
PAPER NO. 166

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STOCHASTIC RESPONSE OF ENERGY BALANCED MODEL FOR VORTEX-INDUCED VIBRATION

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ABSTRACT: A double oscillator model for vortex-induced oscillations of structural elements based on exact power exchange between fluid and structure, recently proposed by the authors, is extended to include the effect of the turbulent component of the wind. In non-turbulent flow vortex-induced vibrations of lightly damped structures are found on two branches, with the highest amplification on the low-frequency branch. The effect free wind turbulence is to destabilize the vibrations on the high amplification branch, thereby reducing the oscillation amplitude. The effect is most pronounced for very lightly damped structures. The character of the structural vibrations changes with increasing turbulence and damping from nearly regular harmonic oscillation to typical narrow-banded stochastic response, closely resembling observed behaviour in experiments and full-scale structures.

1 INTRODUCTION

Vortex-induced oscillations play an important role in the design of slender structures exposed to fluid flow. It is commonly observed that if the natural vortex shedding frequency, determined by the fluid flow velocity via the Strouhals relation and the eigenfrequency of the structure are close, the structure may develop oscillations that control the vortex shedding frequency, and this syncronization may lead to large oscillation amplitudes in the so-called lock-in interval. The resonance amplitude depends on the structural damping, the free turbulence, and for smooth structural shapes on the Reynolds number. It has been found experimentally e.g. by van Koten (1984) that in turbulent flow very lightly damped cylindrical structures may develop amplitudes up to around half the diameter in nearly harmonic oscillation. With increasing damping the amplitude is decreased and the response changes gradually towards nearly linear Gaussian response.

Vickery & Basu (1983) have proposed a model equation for the stochastic response during lock-in. The wind excitation is assumed in the form of a narrow-band Gaussian process, and the self-limiting amplitude is obtained by assuming that the aerodynamic damping is negative for small oscillations and increases quadratically with amplitude. This model ha been fitted to full scale observtions on chimneys by Daly (1986) and is used in simplifie form in the National Building Code of Canada (1990). An important point is the mode parameters, and their dependence on Reynolds number and turbulence. In the study b Daly (1986) only dependence on the Reynolds number was assumed. However, it appear that structures that have developed severe vortex-induced vibrations may have experience years of service without vibration problems. The incidents of severe vortex-induced vibra tions seem to be associated with atmospheric conditions with low turbulence intensit Dyrbye & Hansen (1997).

The present paper extends a double osciallator model for vortex-induced vibrations, re cently developed by the authors for non-turbulent free flow, Krenk & Nielsen (1996), t flow with free turbulence representing e.g. natural wind. The double oscillator model i based on exact energy exchange between fluid and structure, and it predicts two differen modes of oscillation in the lock-in regime leading to hysteresis effects when the wind spee passes up and down through the lock-in interval. This behaviour is also observed in ex periments e.g. by Feng (1968) and Brika & Laneville (1993). The effect of the turbulenc is to destabilize the mode with the highest amplification, thereby reducing the respons amplitude, and for higher turbulence intensity to change the self-excited harmonic respons to stochastic narrow-banded response with changing amplitude.

2 DOUBLE OSCILLATOR MODEL

Figure 1 shows cylinder of length l and diameter D suspended by linear springs in a fluid flow with mass density ρ and total fluid velocity $U_t = U + u(t)$, where U is the undisturbed mean-flow velocity and $u(t)$ is the turbulent velocity component. The cylinde can move in the transverse direction with displacement $X(t)$. The sum of the structura and added fluid mass is m_0. The structural stiffness is represented via the undampe circular eigenfrequency of the oscillator ω_0. The structural damping is modeled as a linea viscous with the damping ratio ζ_0.

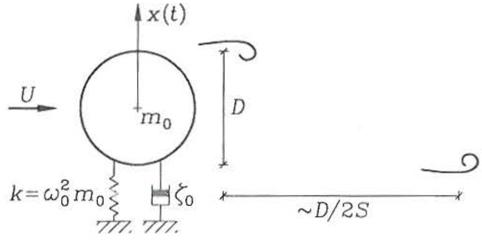

FIG. 1. Vortex-induced vibrations of cylinder in cross-flow.

The oscillating fluid in the near wake of the cylinder is modelled phenomenologically as a single degree of freedom non-linear oscillator of the van der Pol type, where the sign of the damping term, which serves as the power supply to the integrated system, is controlled by the mechanical energy of the fluid as proposed by Hartlen & Currie (1970). Several proposals have been made for the form of coupling between the solid an fluid oscillator, e.g. Iwan & Blevins (1974). Recently the authors proposed a model in which the coupling is prescribed such that the energy exchange between the two oscillators is balanced at all times, Krenk & Nielsen (1996). The governing equations for non-turbulent flow were derived from dimensional analysis in the form

$$m_0[\,\ddot{X} + 2\zeta_0\omega_0\dot{X} + \omega_0^2 X\,] \;=\; \tfrac{1}{2}\rho U^2 Dl\frac{\dot{W}}{U}\gamma \tag{1}$$

$$m_s[\,\ddot{W} - 2\zeta_s\omega_s(1 - \frac{W^2 + \dot{W}^2/\omega_s^2}{w_0^2})\dot{W} + \omega_s^2 W\,] \;=\; -\tfrac{1}{2}\rho U^2 Dl\frac{\dot{X}}{U}\gamma \tag{2}$$

γ is a non-dimensional coupling parameter, and the quantity $\gamma\dot{W}/U$ may be considered as a time-dependent lift-coefficient. The velocities $\dot{X}$ and $\dot{W}$ are normalized with respect to the wind velocity U. m_s is a generalized mass proportional to the fluid mass density ρ and the volume of the cylinder. The proportionality factor can be absorbed in W and γ, so the equivalent mass can be set to $m_s = \rho D^2 l$. The natural frequency of the cylinder is defined as the circular Strouhals frequency $\omega_s = 2\pi SU/D$, where $S \simeq 0.2$ is the Strouhals number, indicating the circular shedding frequency on a fixed cylinder in laminar flow. In this case the exact solution to the fluid oscillator equation is $W(t) = w_0\sin(\omega_s t + \psi)$, with the arbitrary phase ψ. Hence w_0 is the amplitude of the fluid oscillations on a fixed cylinder in laminar flow. The power supplied to the mechanical oscillator becomes $\frac{1}{2}\rho U_t^2 Dl\dot{W}\dot{X}\gamma/U$, which exactly balance the power extracted from the fluid oscillator. Previous double oscillator models have not met this power flow condition.

2.1 Equations for turbulent flow

The effect of the turbulence is to modify the right-hand side of the equations by introduction of the instantaneous wind pressure $\frac{1}{2}\rho U_t^2$ and by evaluating the Strouhals frequency in the fluid oscillator stiffness term by use of the instantaneous wind velocity U_t, whereby the fluid 'stiffness' takes the form $\omega_s^2(1 + u(t)/U))^2$. The fluid damping term was devised to supply a typical rate of energy and is not known in sufficient detail to warrant more detail. The equations including turbulence then are

$$m_0[\,\ddot{X} + 2\zeta_0\omega_0\dot{X} + \omega_0^2 X\,] \;=\; \tfrac{1}{2}\rho U^2 Dl\,(1 + \frac{u(t)}{U})^2\frac{\dot{W}}{U}\gamma \tag{3}$$

$$m_s[\,\ddot{W} - 2\zeta_s\omega_s(1 - \frac{W^2 + \dot{W}^2/\omega_s^2}{w_0^2})\dot{W} + \omega_s^2(1 + \frac{u(t)}{U})^2 W\,] \;=\; -\tfrac{1}{2}\rho U^2 Dl\,(1 + \frac{u(t)}{U})^2\frac{\dot{X}}{U}\gamma \tag{4}$$

The following non-dimensional variables are introduced for the structural and fluid displacement

$$Y = \frac{X}{D} \qquad , \qquad V = \frac{W}{w_0} \tag{5}$$

The equations (3)-(4) can then be written

$$\ddot{Y} + 2\zeta_0\omega_0\dot{Y} + \omega_0^2 Y = \mu_s c\,\omega_s(1 + R(t))\dot{V} \tag{6}$$

$$\ddot{V} - 2\zeta_s\omega_s[\,1 - V^2 - (\dot{V}/\omega_s)^2\,]\dot{V} + \omega_s^2(1 + R(t))V = -v_0^{-2}c\,\omega_s(1 + R(t))\dot{Y} \tag{7}$$

with the following non-dimensional quantities

$$\begin{aligned} v_0 &= \frac{w_0}{D} & \mu_s &= \frac{m_s}{m_0} = \frac{\rho D^2 l}{m_0} \\ c &= \frac{w_0}{D}\frac{\gamma}{4\pi S} & R(t) &= 2\frac{u(t)}{U} \end{aligned} \tag{8}$$

μ_s is the mass ratio, and c is a rescaled coupling coefficient proportional to the amplitude of the lift coefficient. In (6)-(7) the turbulence intensity is assumed sufficiently small to justify the omission of quadratic terms in $u(t)$, and the turbulence is therefore represented by the non-dimensional stochastic process $R(t)$.

2.2 *Harmonic response for non-turbulent flow*

In the absence of turbulence, i.e. for $u(t) \equiv 0$, the response can be represented by the harmonic approximation

$$y = A\sin(\omega t) \qquad , \qquad v = B\sin(\omega t) \tag{9}$$

The frequecy of oscillation ω is determined by the Strouhals frequency ω_s. The inverse relation giving ω_s as function of ω is, Krenk & Nielsen (1996),

$$(\frac{\omega_s}{\omega_0})^2 = \left(\frac{\omega}{\omega_0}\right)^2 \frac{\left(1 - (\frac{\omega}{\omega_0})^2\right)^2 + 4\,\zeta_0^2\,(\frac{\omega}{\omega_0})^2}{\left(1 - (\frac{\omega}{\omega_0})^2\right)^2 + 4\zeta_0^2(\frac{\omega}{\omega_0})^2 - \mu_s\left(\frac{c}{v_0}\right)^2(\frac{\omega}{\omega_0})^2\left(1 - (\frac{\omega}{\omega_0})^2\right)} \tag{10}$$

The amplitudes can then be found as

$$B^2 = \frac{4\omega_s^2}{\omega_s^2 + 3\omega^2}\left(1 - \frac{\zeta_0\omega_0}{\zeta_s\omega_s}\frac{\omega_s^2 - \omega^2}{\omega_0^2 - \omega^2}\right) \tag{11}$$

$$A = \frac{\mu_s\,c\,\dfrac{\omega_s}{\omega_0}\,\dfrac{\omega}{\omega_0}\,B}{\left[\left(1 - (\frac{\omega}{\omega_0})^2\right)^2 + 4\,\zeta_0^2\,(\frac{\omega}{\omega_0})^2\right]^{1/2}} \tag{12}$$

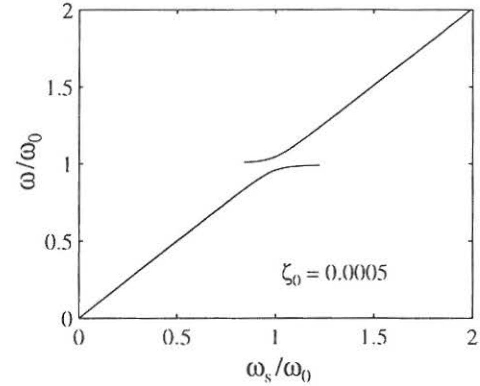

FIG. 2. Response frequency branches.

Representative values of the model parameters are $c \simeq 0.10$ and $\zeta_s \simeq 0.010$. Figure 2 shows typical frequency branches for a very lightly damped system with $\mu_s = 0.05$, $\zeta_0 = 0.0005$, and $v_0 = 0.26$. The corresponding amplitude curves are shown in Fig. 3.

The figures show two solutions with distinctive resonance when the Strouhals frequency is close to the natural structural freqency. The solutions are limited by the condition $B^2 \geq 0$. A stability analysis carried out in Krenk & Nielsen (1996) shows that the low frequency branch is only stable for frequencies lower than an upper limit approximately equal to the peak amplitude frequency. Similarly the high frequency branch is only stable for frequencies above a frequency corresponding approximately to the peak amplitude of this branch.

3 THE TURBULENCE PROCESS

The non-dimensional turbulence process $R(t)$ is here used to represent the rapidly fluctuating part of the natural turbulence. This is the part of the turbulence that leads to a modified stationary rsponse, while low frequency components may lead to transients when entering and leaving lock-in intervals of limited length. The turbulence process $R(t)$ is introduced in the form of an Ornstein-Uhlenbeck process with the stationary covariance function

$$\kappa_{RR}(\tau) \; = \; \sigma_R^2 \exp\left(-\frac{|\tau|}{\tau_c}\right) \tag{13}$$

where σ_R is the standard deviation and τ_c the correlation time scale. The turbulence intensity is defined by $I_u = \sigma_u/U = \frac{1}{2}\sigma_R$.

The response analysis is based on Monte Carlo simulation. The broad-banded turbulence

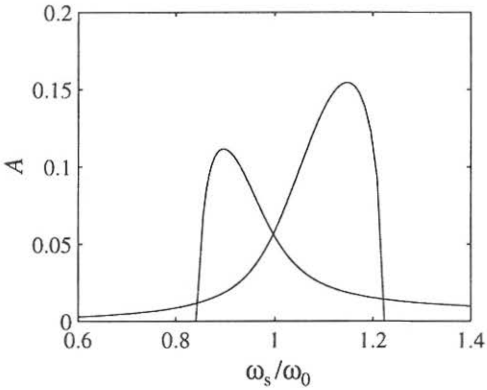

FIG. 3. Harmonic amplitude versus frequency.

process $R(t)$ is simulated from the Itô-differential equation

$$dR(t) \;=\; -\frac{1}{\tau_c}\, R(t)\, dt \;+\; \sqrt{\frac{2}{\tau_c}}\, \sigma_R \, dW(t) \tag{14}$$

where $dW(t)$ is the increment of a unit intensity Wiener process.

The reference time is the 'Strouhals period' $T_s = 2\pi/\omega_s$, representing a typical oscillation period. The correlation time τ_c is chosen to be 'short' relative to T_s, $\tau_c = T_s/50$. The simulation interval Δt of the process $dW(t)$ was $\Delta t = T_s/250$, corresponding to $\Delta t = \frac{1}{5}\tau_c$, and the simulated points connected by a broken line as suggested by Clough & Penzien (1975). The response was obtained by introducing the state space vector $[Y(t), \dot{Y}(t), V(t), \dot{V}(t), R(t)]$ and integrating the differential equations (6), (7) and (14) with a 4th order Runge-Kutta scheme using the time-step Δt. By this integration scheme the response to rapid fluctutions are represented with good accuracy. The probability density $f_Y(y)$ and the standard deviation σ_Y are obtained from ergodic sampling over a time interval of length $10^5 T_s$, following an initial interval of length $3 \cdot 10^3 T_s$ to allow for decay of transients due to initial conditions. The probability density function was sampled with 301 classes of equal length covering the interval $[-4\sigma_Y, 4\sigma_Y]$ for an estimated value of σ_Y.

4 RESPONSE CHARACTERISTICS

In the absense of turbulence two near-harmonic solutions exist in the frequency interval between the peaks of the amplitude curves in Fig. 3. The effect of turbulence is illustrated in Figs. 4 and 5, showing the simulated probability density function $f_Y(y)$ at mean wind speeds corresponding to the Strouhals frequencies $\omega_s/\omega_0 = 0.9, 0.95, 1.0, 1.05, 1.1, 1.15$ for turbulene intensities $\sigma_u/U = 0.02$ and 0.10, respecively. At low turbulence intensity the

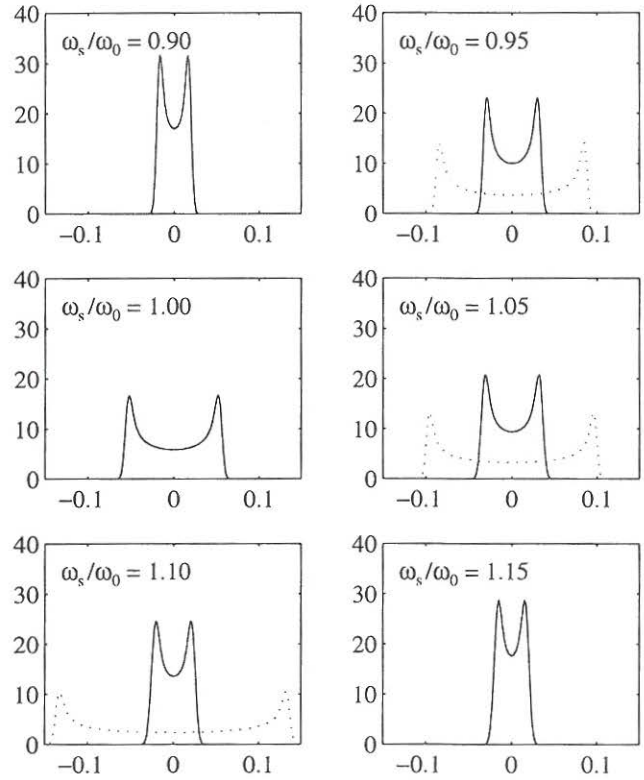

FIG. 4. Probability densities, $\zeta_0 = 0.0005$, $\sigma_u/U = 0.02$.

response retains a nearly harmonic character and around the resonance frequecy two solutions exist, as in the non-turbulent case. In the imulation study the two solutions are obtained by using the initial conditions $\dot{Y}(0) = V(0) = \dot{V}(0) = 0$ together with either $Y(0) = 1$ or 0. Segments of the corresponding time histories are shown in Figs. 6a and 6b for $\omega_s/\omega_0 = 1.05$. The mean amplitudes correspond closely to the deterministic values predicted by Fig. 3. The resonant solution remains most regular. This confirms the experiental observaion of Goswami et al. (1993), who found only little effect of turbulence of this low intensity. When the turbulence intenity is increased the upper branch is destabilized and the response changes character from slightly perturbed harmonic to narrow-banded stochastic response containing also very small amplitudes as shown in the probability density functions $f_Y(y)$ in Fig. 5 and the time history show in Fig. 6c.

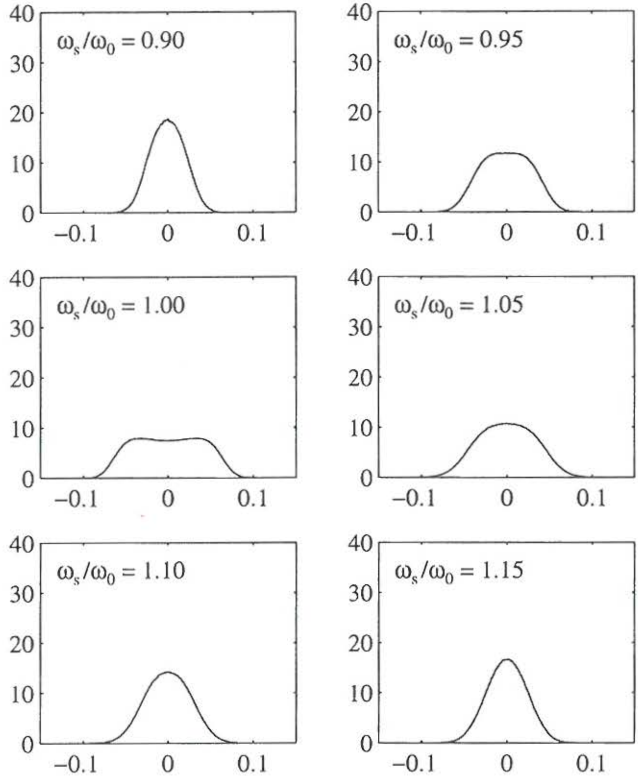

FIG. 5. Probability densities, $\zeta_0 = 0.0005$, $\sigma_u/U = 0.10$.

5 CONCLUSIONS

A recently developed energy balanced double oscillator model has been used to investigate the effect of turbulence. Two effects were identified: turbulence tends to decrease the response by destabilising the most resonant branch, and with increasing turbulence intensity the response changes to typical stochastic narrow-banded with varying amplitudes.

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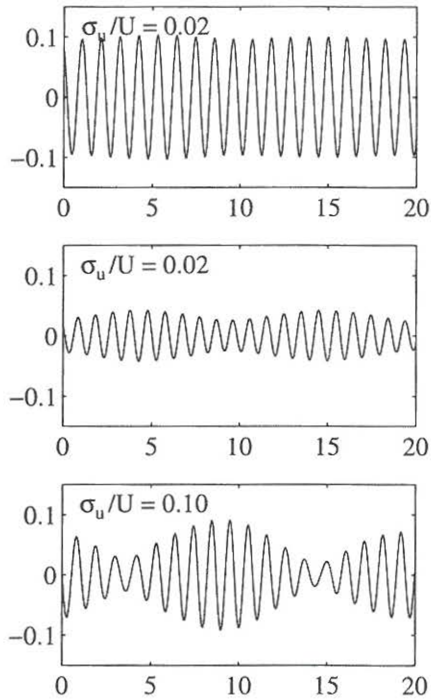

FIG. 6. Time histories for $\zeta_0 = 0.0005$.

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