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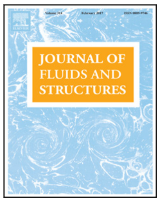

Time-domain numerical simulations of a loosely supported tube subjected to frequency-dependent fluid–elastic forces

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ABSTRACT

Flow-induced vibrations of heat-exchanger tubes are extensively studied in the nuclear industry for safety reasons. Adequate designs, such as anti-vibration bars in PWR steam generators, prevent excessive vibrations provided the tubes are well supported. Nevertheless, degraded situations where the tube/support gaps would widen, must also be considered. In such a case, the tubes become loosely supported and may exhibit vibro-impacting responses due to both turbulence and fluid–elastic coupling forces induced by the cross-flow. This paper deals with the predictive analysis of such a nonlinear situation, given the necessity of taking into account both the strong impact nonlinearity due to the gap and the linearized fluid–elastic forces. In time-domain numerical simulations, computation of flow-coupling forces defined in the frequency-domain is a delicate problem. We recently developed an approach based on a hybrid time–frequency method. In the present paper a more straightforward and effective technique, based on the convolution of a flow impulse response pre-computed from the frequency-domain coefficients, is developed. Illustrative results are presented and discussed, in connection with the previous hybrid method and with experiments. All results agree in a satisfactory manner, validating both computational methods, however the convolutional technique is faster than the hybrid method by two orders of magnitude. Finally, to highlight the subtle self-regulating frequency effect on the stabilization of such system, additional demonstrative computations are presented.

1. Introduction

Flow-induced vibrations of heat-exchanger tubes are a major concern in the nuclear industry for safety reasons and repair costs, and are thus extensively studied. Adequate designs, such as anti-vibration bars in PWR steam generators, generally prevent any problem. Nevertheless some degraded situations, although not very likely, must also be considered, as for example ill positioned or worn supports, which would widen the tube/support gaps. In such a case the tube becomes loosely supported and may exhibit vibro-impacting responses due to both turbulence and fluid–elastic coupling forces induced by the cross-flow. Predictive analysis is an essential step in component design, as well as for diagnosing anomalous behaviour. Significant efforts have been directed into the development of time-domain computational methods and tools to deal with the nonlinear dynamics of the flow-excited tube bundles — see, for instance, Axisa and Antunes (1988), Eisinger et al. (1995),

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Nomenclature

$c_d(f_r)$, $c_d(V_r)$	Dimensionless fluid–elastic damping coefficient
$c_k(f_r)$, $c_k(V_r)$	Dimensionless fluid–elastic stiffness coefficient
$C_d(f_r)$, $C_d(V_r)$	fluid–elastic damping coefficient
$C_k(f_r)$, $C_k(V_r)$	fluid–elastic stiffness coefficient
C_D	Dimensionless drag coefficient
C_L	Dimensionless lift coefficient
$\mathbf{C}_f\ (f_r)$	Flow damping matrix
$\mathbf{C}_s$	Structural damping matrix
D	Tube diameter
D_{ref}	Reference tube diameter
$\mathrm{E}_{F_{\mathrm{Cplg}}}(t)$	Energy from the fluid–elastic coupling
$\mathrm{E}_{F_{\mathrm{Impact}}}(t)$	Energy connected with the impacts at the clearance support
$\mathrm{E}_{F_{\mathrm{Turb}}}(t)$	Energy from the turbulence excitation
$\mathrm{E}_{\mathrm{Vibr}}(t)$	Vibration energy
$\mathrm{E}_{\zeta_{\mathrm{Tube}}}(t)$	Energy dissipated through modal damping
f	Frequency
f_n	Frequency of mode n
$f_r = fD/V$	Reduced frequency
f_R	Rice frequency
$F_c(t)$	Contact force
$F_{FE}^{x}(t),\ F_{FE}(t)$	Lift fluid–elastic force
$F_{FE}^{y}(t)$	Drag fluid–elastic force
$F_n^{FE}(t)$	Modal force due to the fluid–elastic coupling for mode n
$F_n^{Turb}(t)$	Modal force due to the turbulence excitation for mode n
$\mathbf{f}_{FE}\ (\mathbf{x}, \dot{\mathbf{x}})$	Motion-dependent fluid–elastic forces
$\mathbf{f}_{NL}\ (\mathbf{x}, \dot{\mathbf{x}})$	Motion-dependent contact forces at the clearance supports
$\mathbf{f}_{Turb}(t)$	Turbulence force vector
$g(\bar{t}) = \dot{\Phi}(\bar{t})$	Dimensionless time-derivative of the flow transient function
$G(\bar{s})$	Laplace transform of $g(\bar{t})$
$h_{FE}(t)$	fluid–elastic impulse response
$H_{FE}(s)$	fluid–elastic transfer function
$\hat{H}_{FE}(s)$	Experimentally identified fluid–elastic transfer function
$H(\bar{t})$	Heaviside step function
$I(\bar{t}_i)$	Normalized integral $I(\bar{t}_i) = \int_0^{\bar{t}_i} g(\bar{t})d\bar{t} / \int_0^{\infty} g(\bar{t})d\bar{t}$
k_c	Contact stiffness of the clearance support
$\mathbf{K}_f\ (f_r)$	Flow stiffness matrix
$\mathbf{K}_s$	Structural stiffness matrix
L	Tube length
L_{ref}	Reference tube length
$\mathbf{M}_s$	Structural mass matrix
$\mathbf{M}_f$	Fluid added mass matrix
P	Square bundle pitch
$\left\langle \mathrm{P}_{\mathrm{Cplg}} \right\rangle$	Time-averaged power from the fluid–elastic coupling
$\langle \mathrm{P}_{\mathrm{Turb}} \rangle$	Time-averaged power from the turbulence excitation
$\left\langle \mathrm{P}_{\zeta\mathrm{Tube}} \right\rangle$	Time-averaged power dissipated through modal damping
$\mathbf{q}(t)$, $\dot{\mathbf{q}}(t)$, $\ddot{\mathbf{q}}(t)$	Modal displacement, velocity and acceleration vectors
s	Laplace variable
$\bar{s} = sD/V$	Reduced Laplace variable
$\widetilde{S}_{ff}^{eq}(f_r)$	Turbulence excitation envelope spectrum
$\widetilde{S}_{ref}^{eq}(f_r)$	Turbulence reference envelope spectrum
t	Time
$\bar{t} = tV/D$	Dimensionless time
t_i	Time-window of flow “memory” effects
$\bar{t}_i = t_i V/D$	Dimensionless time-window of flow “memory” effects
V	Pitch (inter-tube) flow velocity