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International Journal of Naval Architecture and Ocean Engineering 8 (2016) 117–126
http://www.journals.elsevier.com/international-journal-of-naval-architecture-and-ocean-engineering/

Frequency response of rectangular plates with free-edge openings and carlings subjected to point excitation force and enforced displacement at boundaries

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Received 3 December 2014; revised 1 June 2015; accepted 16 June 2015
Available online 15 March 2016

Abstract

In this paper, a numerical procedure for the natural vibration analysis of plates with openings and carlings based on the assumed mode method is extended to assess their forced response. Firstly, natural response of plates with openings and carlings is calculated from the eigenvalue equation derived by using Lagrange's equation of motion. Secondly, the mode superposition method is applied to determine frequency response. Mindlin theory is adopted for plate modelling and the effect of openings is taken into account by subtracting their potential and kinetic energies from the corresponding plate energies. Natural and frequency response of plates with openings and carlings subjected to point excitation force and enforced acceleration at boundaries, respectively, is analysed by using developed in-house code. For the validation of the developed method and the code, extensive numerical results, related to plates with different opening shape, carlings and boundary conditions, are compared with numerical data from the relevant literature and with finite element solutions obtained by general finite element tool.

Keywords: Frequency response analysis; Plate with openings; Point excitation force; Enforced displacement; Mode superposition method

1. Introduction

Rectangular plates with openings are frequently found in engineering structures. The openings are made for many reasons, as for instance saving weight, venting, altering the natural frequencies and providing accessibility to other parts of the structures (Lee et al., 1990; Cho et al., 2013).

Literature reviews on vibration analysis of plates with openings are available in Kwak and Han (2007) and Cho et al. (2013) where different approaches are discussed. Experimentally obtained natural frequencies of rectangular plates with elliptical inner boundaries are reported by Nagaya (1981). Hegarty and Ariman (1975) investigated free vibration of rectangular plates with circular openings by a least-squares point-matching method. Application of the finite difference method to the above problem can be found in Paramasivam (1973), Aksu and Ali (1976). Different variants of Rayleigh-Ritz method are presented by Ali and Atwal (1980), Lam et al. (1989), Lee et al. (1990), Mundukur et al. (1994), Grossi et al. (1997), Laura et al. (1997), and Avalos and Laura (2003). There are also many papers where the finite element method (FEM) is applied to dynamic response analysis of plates with openings (Monahan et al., 1970; Ali and Atwal, 1980; Reddy, 1982; Chang and Chiang, 1988). Huang and Sakiyama (1999) and Sakiyama et al. (2003) treated the opening in the plate as an extremely thin part of

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Peer review under responsibility of Society of Naval Architects of Korea

http://dx.doi.org/10.1016/j.ijnaoe.2015.06.001 / pISSN : 2092-6782, eISSN : 2092-6790

the plate. In this way, a plate with an opening was turned out to be the plate with non-uniform thickness. In that study, the case of the plate with circular, semi-circular, elliptic, square, rectangular, triangular, rhombic opening were studied by applying the discrete Green function of the plate. Recently, Huang (2013) investigated effects of constraints, circular opening and in-plane loading on vibration of rectangular plates. Also, Chen et al. (2014) analysed flexural and in-plane vibration of elastically restrained thin rectangular plates with openings using Chebyshev–Lagrangian method. The above state-of-the art clearly indicates that vibration of plates with openings still represent an interesting and important topic. As explained by Cho et al. (2013), nowadays FEM is probably most important method in practical engineering, but its application still suffers from some drawbacks. A rather time-consuming model preparation is pronounced, that makes FEM appropriate only for final structure checks, when all dimensions and boundary conditions are known.

On the other hand, although the natural vibration analysis provides an insight in the dynamic behaviour of structures, it is often necessary, especially in ships and offshore units subjected to external vibration sources, to check compliance of vibration amplitudes with the prescribed criteria. Hence, this investigation is motivated with the aim to provide very simple, fast and accurate method, capable of analysing forced response of plates with openings and carlings.

A simplified energy-based method for the natural vibration analysis of Mindlin plates with openings and arbitrary boundary conditions is presented by Cho et al. (2013). The procedure is based on the assumed mode method (Chung et al., 1993; Kim et al., 2012; Cho et al., 2014, 2015), and it can be applied to plates with multiple free-edge openings, arbitrarily placed within the plate area. As indicated above, this paper extends application of the developed procedure to the forced vibration analysis of such structures under point forced excitation and boundary displacement load, by using the mode superposition method. Furthermore, in the mathematical model the effect of carling is introduced, as a novelty. Namely, in ships and offshore structures, in the vicinity of openings and below heavy equipment carlings are fitted to prevent local deformation, and at the same time they influence the structure dynamic properties.

Since the natural vibration analysis is a prerogative for the forced response calculation by using the mode superposition method, in the next section the assumed mode method is briefly outlined. In the third section, forced response assessment is described and it is followed by extensive illustrative examples considering plates with different openings, carlings and boundary conditions. Comparisons with other available methods confirmed high accuracy of the presented procedure.

2. Natural vibration of plates with openings and carlings – assumed mode method

In this mathematical model the Mindlin thick plate theory which takes shear influence and rotary inertia into account (Mindlin et al., 1956) is adopted. It operates with three general displacements, i.e. plate deflection w, and angles of cross-section rotation about the x and y axes, ψ_x and ψ_y, respectively. In the formulation of the eigenvalue problem, Lagrange's equation is applied. In this sense, it is necessary to express system potential energy, V, and kinetic energy, T, respectively, in a convenient manner (Kim et al., 2012; Cho et al., 2013, 2014, 2015). One can write for a plate with opening and carlings:

$$V = V_p + V_c - V_o, \tag{1}$$

$$T = T_p + T_c - T_o. \tag{2}$$

where V_p is the plate potential energy, V_c represents the potential energy of carlings and V_o is the potential energy of openings. Similarly, T_p is the plate kinetic energy and T_c and T_o are the kinetic energies of the carlings and openings, respectively.

The assumed mode method is applied to analyse the natural flexural vibrations. Lateral displacement and rotational angles can be expressed by superposing the products of the orthogonal polynomials:

$$\begin{aligned} w(\xi,\eta,t) &= \sum_{m=1}^{M}\sum_{n=1}^{N} a_{mn}(t)X_m(\xi)Y_n(\eta), \quad \psi_\xi(\xi,\eta,t) \\ &= \sum_{m=1}^{M}\sum_{n=1}^{N} b_{mn}(t)\Theta_m(\xi)Y_n(\eta), \quad \psi_\eta(\xi,\eta,t) \\ &= \sum_{m=1}^{M}\sum_{n=1}^{N} c_{mn}(t)X_m(\xi)\Phi_n(\eta) \end{aligned} \tag{3}$$

where $\xi = x/a$ and $\eta = y/b$. $X_m(\xi)$, $Y_n(\eta)$, $\Theta_m(\xi)$ and $\Phi_n(\eta)$ are the orthogonal polynomials to represent deflections and rotation angles satisfying the specified elastic edge constraints with respect to ξ and η, respectively. Furthermore, $a_{mn}(t)$, $b_{mn}(t)$ and $c_{mn}(t)$ are the influence coefficients of orthogonal polynomials. Also, M and N are the number of orthogonal polynomials used for an approximate solution in the ξ and η directions, respectively.

Lagrange's equation of motion reads:

$$\frac{\mathrm{d}}{\mathrm{d}t}\left(\frac{\partial T}{\partial \dot{q}_i}\right) - \frac{\partial T}{\partial q_i} + \frac{\partial V}{\partial q_i} = 0, \tag{4}$$

where

$$\{q(t)\} = \{a_{11}\ldots a_{MN}\, b_{11}\ldots b_{MN}\, c_{11}\ldots c_{MN}\}^T. \tag{5}$$

If Eqs. (1) and (2) are substituted into (4) the discrete matrix equation with $3 \times M \times N$ degrees of freedom is obtained as the following equation:

$$[M]\left\{\frac{\partial^2 q(t)}{\partial t^2}\right\} + [K]\{q(t)\} = 0, \tag{6}$$

where $[M]$ and $[K]$ are the mass and the stiffness matrix, respectively. Their constitution using characteristic orthogonal polynomials is described in detail by Kim et al. (2012).