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Free vibration analysis of stiffened panels with lumped mass and stiffness attachments

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ABSTRACT

Stiffened panels are basic constitutive members of ships and offshore structures, and in practice they often have different mass and stiffness attachments, which significantly influence their dynamic response. In this paper, a numerical procedure is presented for the free vibration analysis of stiffened panels with arbitrary sets of boundary conditions and carrying multiple lumped mass and stiffness attachments. It is based on the assumed mode method, where characteristic orthogonal polynomials having the properties of Timoshenko beam functions and satisfying the specified edge constraints are used as approximation functions. The Mindlin theory is applied for plate and the Timoshenko beam theory for stiffeners. The total potential and kinetic energies of the system are formulated in a convenient manner and further applied to derive an eigenvalue problem by means of Lagrange's equation of motion. Based on the developed numerical procedure, an in-house code is developed and is applied to a free vibration analysis of bare plates and stiffened panels carrying lumped masses and locally supported by pillars or springs. Comparisons of the results with those available in the literature and FEA solutions confirm the high accuracy and practical applicability of the presented procedure.

1. Introduction

The vibration analysis of stiffened panels is an important issue in almost all aspects of practical engineering. This is especially pronounced in naval architecture and ocean engineering, where stiffened panels are used as primary constitutive elements of almost all structures. In many practical applications, one can find stiffened panels carrying multiple mass elements or locally supported with concentrated supports (such as, for instance, pillars or elastic springs) which significantly change their response to dynamic loading and make vibration analysis rather complicated. In the case of masses elastically connected to the plate structure, the system exhibits even more complex behaviour than in the case of a rigidly attached mass (Avalos et al., 1993).

Vibrations of rectangular plates carrying concentrated masses have been a topic of research from the beginning of the last century, as is obvious from the paper of Gershgorin (1933) where the effect of rotary inertia was ignored, or from some later references (Stokey and Zorowski, 1959; Shah and Datta, 1969), where the mentioned rotary inertia effect is taken into account. Nicholson and Bergman (1985) studied the vibration of simply supported Mindlin plates (Mindlin et al., 1956) carrying concentrated masses which retained the effects of transverse shear and the rotary inertia of each mass. They also investigated the limit case of thin plate. Singal and Gorman (1992) presented a comparison of the analytical and experimental results for rectangular aluminium plates on rigid point supports and carrying one central mass or two masses of different weights. Their results show that rotary inertia reduces natural frequencies, while no additional modes due to the rotary inertia of masses were detected. According to Amabili et al. (2006), this is probably the consequence of relatively small rotary inertia of the masses used in calculations and experiments with respect to their weight. A Rayleigh-energy method with single-term trigonometric functions was used to study the vibration of plate with concentrated mass by Boay (1993) and comparisons with the experimental results for S-C-S-C rectangular plate were provided. Later, the same problem was investigated by multi-term trigonometric series expansion (Boay, 1995). Most of the references in this field actually deal with either plates carrying lumped masses or springs and point reinforcement, but only a few papers consider both additional inertia and stiffness attachments.

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Wu and Luo (1995) analysed the natural frequencies and corresponding mode shapes of a uniform rectangular thin plate carrying any number of point masses and translational springs by means of the so-called analytical-and-numerical-combined method (ANCM). Further, Ostachowicz et al. (2002) presented a method for the identification of the location of a concentrated mass on isotropic plates by means of a genetic algorithm search technique based on changes in natural frequencies. The location and size of the mass are determined by the minimisation of an error function, which expresses the difference between calculated and measured natural frequencies. Amabili et al. (2006) studied the effect of concentrated masses on the free vibrations of rectangular plates by considering the rotary inertia of concentrated masses and the geometric imperfections of the plate both experimentally and numerically, by using a Rayleigh–Ritz based computer code. The results showed that large rotary inertia of concentrated masses introduces additional modes, which do not appear if rotary inertia is neglected. Watkins and Barton (2010) computed normalised frequencies for a rectangular, isotropic plate resting on elastic supports using an eigensensitivity analysis, which approximates the eigenparameters in a Maclaurin series, yielding an approximate closed-form expression. Also, Watkins et al. (2010) experimentally determined the natural frequencies and mode shapes for an elastically point-supported plate with attached masses under impulsive loading. The results are compared to frequencies and to mode shapes determined from the Rayleigh–Ritz method and a finite element analysis. It was found that the Rayleigh–Ritz analysis was able to suitably capture the first three rigid body mode natural frequencies for the majority of experimental configurations, while there were intriguing differences between the experimentally obtained mode shapes and the Rayleigh–Ritz analysis mode shapes. According to Watkins et al. (2010), the selection of basis functions has a significant influence on the accuracy of results obtained from the Ritz analysis. However, both the Rayleigh–Ritz analysis and the FE model were able to effectively capture the experimentally observed high frequency plate bending modes and were in good agreement with each other, although they were slightly higher than the experimental results.

Beside the above reviewed references to rectangular plates, circular and elliptical plates with attached masses have also been the subject of investigation (Laura et al., 1984; Bambill et al., 2004; Maiz et al., 2009). The vibration analysis of circular plates carrying a concentrated mass reported by Bambill et al. (2004) was motivated by the practical need to place a centrifugal pump rigidly attached to the thin, circular cover plate of a water tank in a medium-sized ocean vessel. Namely, due to a lack of space, it was necessary to place the system off-centre of the circular configuration and to calculate the fundamental frequency of the coupled system, where the Rayleigh–Ritz method was applied. In the case of stiffened panels, it was rather difficult to find a paper dealing with the vibration analysis of such structures carrying lumped mass attachments or having additional point supports.

During the last few decades, the finite element method (FEM) has become the most powerful tool for strength and vibration analysis, widely used in practical engineering. It is applied to very complex structures, and is also used to determine the static and dynamic response of different simple beam-like and plate-like structures. Various finite elements have even been developed and incorporated in general FE software. However, among other drawbacks, such software still requires rather lengthy model preparation, and modifications of the models when different topologies are being investigated are time consuming. Therefore, the application of simplified energy-based solutions, such as the one proposed in this paper, may have some advantages in the initial design stage.

In this paper, a simple numerical procedure is presented for the free vibration analysis of stiffened panels carrying multiple lumped mass attachments and/or having point reinforcements arbitrarily placed within the plate area. The procedure is based on the assumed mode method, which was recently applied by the authors to both the dry and wet vibration analysis of simpler plate structures (Cho et al., 2015a,b,c). The effect of lumped attachments is expressed by adding their potential and kinetic energies to the corresponding panel energies, and the ordinary procedure by utilising Lagrange's equation is applied to calculate the natural frequencies and their corresponding mode shapes of the structures. The procedure is validated through several numerical examples dealing with the free vibration of both bare plates and stiffened panels with different sets of boundary conditions and lumped attachments. The results are compared with numerical solutions from open literature, and the FEA solutions obtained by the general FE tool, where the high accuracy and practical applicability of the method are confirmed.

2. Mathematical formulation

The dynamic response of stiffened panels with attachments is obtained by the assumed mode method which is actually an energy method (Kim et al., 2012; Cho et al., 2015a,b,c). Hence, a calculation of the total system potential and kinetic energies is required, which are further used in Lagrange's equation to formulate an eigenvalue problem. A stiffened panel of length a and width b with an arbitrary number of point masses (arbitrarily placed within the panel area) and having spring or pillar supports is considered, Fig. 1.

The plate model used here is based on the Mindlin thick first-order shear deformation plate theory which operates with three potential functions, i.e. plate deflection w, and angles of cross-section rotation about the x and y axes, ψ_x and ψ_y, respectively (Mindlin et al., 1956). All combinations of classical (simply supported, clamped, free) and non-classical (elastically supported in translation and rotation) boundary conditions are considered, Fig. 2. The equations of motion yield:

$$\frac{\rho h^3}{12}\frac{\partial^2\psi_x}{\partial t^2} - D\left(\frac{\partial^2\psi_x}{\partial x^2} + \frac{1}{2}(1-\nu)\frac{\partial^2\psi_x}{\partial y^2} + \frac{1}{2}(1+\nu)\frac{\partial^2\psi_y}{\partial x\partial y}\right) - kGh\left(\frac{\partial w}{\partial x} - \psi_x\right) = 0, \quad (1)$$

$$\frac{\rho h^3}{12}\frac{\partial^2\psi_y}{\partial t^2} - D\left(\frac{\partial^2\psi_y}{\partial y^2} + \frac{1}{2}(1-\nu)\frac{\partial^2\psi_y}{\partial x^2} + \frac{1}{2}(1+\nu)\frac{\partial^2\psi_x}{\partial x\partial y}\right) - kGh\left(\frac{\partial w}{\partial y} - \psi_y\right) = 0, \quad (2)$$

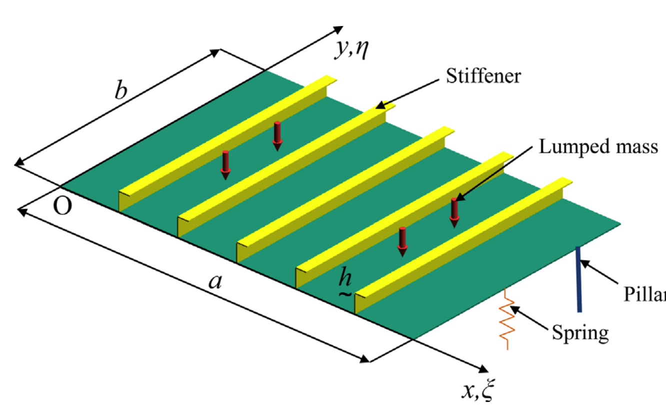

Fig. 1. Schematic presentation of a stiffened panel with attachments.