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Development of eco-friendly and lightweight insulation panels for offshore plant

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Abstract

Recently, regulations pertaining to the noise and vibration environment of offshore plants have been strengthened. For example, the NORSOK standards have been applied, which are very strict regulations that are comparable to those applied to passenger ships. Furthermore, the use of porous materials, such as those used in most of the current insulating panels, has been forbidden. Therefore, honeycomb-backed Micro-Perforated Plates (MPPs) are now regarded as next-generation absorber materials. This paper reports the results of parametric studies that were performed using numerical methods to determine the effect of the thickness on the performance of a honeycomb panel and the effect of the perforation ratio on the MPP performance. The numerical results were verified through experiments. Finally, we propose a combined honeycomb/MPP panel where the MPP is placed between upper and lower honeycomb panels and one end surface is also replaced with an MPP.

Keywords: Eco-friendly; Insulation panel; Honeycomb; Micro perforated plate (MPP); Intensity method

1. Introduction

Vibration and noise regulations for offshore plants have been strengthened in recent years, and environmental concerns are also now a consideration. The NORSOK (2004) (Norsk Sokkels: Norwegian shelf) and United Kingdom Health & Safety Executive (UK-HSE) standards are now being applied, and these are very strict regulations that require ratings comparable to those for passenger ships. In addition, the use of porous materials as acoustic absorbers is now being avoided because of health concerns. In this respect, honeycomb panels and Micro-Perforated Plates (MPPs) are considered to be good alternative.

Honeycomb panels have different stiffness moduli in the planes perpendicular and parallel to the direction of the cells, and these can be characterized as orthotropic with nine independent stiffness constants. In addition, honeycomb panels have good mechanical properties and are widely used in various industries because of their high strength-to-weight ratios. In particular, the panels have the advantage of a light weight because they consist of about 97% air and 3% honeycomb core.

Dym and Lang (1974) were the first to attempt a theoretical prediction of the sound transmission loss (STL or TL) of laminated panels with an infinite size. Since then, Moore and Lyon (1991) proposed using this numerical method for laminated panels with isotropic and orthotropic cores, and the absorption and insulation performances were estimated by assuming that the core of the honeycomb panel was orthotropic.

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Maa (1975) was the first to propose MPPs, and these have now been theoretically established. They have since been used as sound-absorbing materials for various applications (Fuchs et al., 2001; Zha et al., 2002; Wu, 1997). Asdrubali and Pispola (2007) investigated the sound absorption of multiple MPPs for noise barriers. Toyoda and Takahashi (2005,2008) and Toyoda et al. (2007) conducted an analysis that considered the transmission and sound absorption of a combination of MPPs, a solid plate, and honeycombs.

In this study, honeycomb panels and MPPs were used to increase the insulation performance without the need for a porous material, and this composition is now being applied to shipbuilding.

2. Theory of honeycomb insulation panels

Fig. 1 shows an analytical model with an infinite xy plane. Here, S.S. is a steel sheet, and H/C is a honeycomb.

The honeycomb can have different stiffness moduli in the planes perpendicular and parallel to the direction of the cells, and it can be characterized as orthotropic with nine independent stiffness constants. The following equation expresses the relationships among the stress, strain, and stiffness constants (Lin et al., 2007).

$$\begin{Bmatrix} \sigma_x \\ \sigma_y \\ \sigma_z \\ \tau_{yz} \\ \tau_{xz} \\ \tau_{xy} \end{Bmatrix} = \begin{bmatrix} E_{11} & E_{12} & E_{13} & 0 & 0 & 0 \\ E_{12} & E_{22} & E_{23} & 0 & 0 & 0 \\ E_{13} & E_{23} & E_{33} & 0 & 0 & 0 \\ 0 & 0 & 0 & E_{44} & 0 & 0 \\ 0 & 0 & 0 & 0 & E_{55} & 0 \\ 0 & 0 & 0 & 0 & 0 & E_{66} \end{bmatrix} \begin{Bmatrix} \varepsilon_x \\ \varepsilon_y \\ \varepsilon_z \\ \gamma_{yz} \\ \gamma_{xz} \\ \gamma_{xy} \end{Bmatrix} \tag{1}$$

where

$$\begin{Bmatrix} E_{11} & E_{12} & E_{13} & 0 & 0 & 0 \\ E_{12} & E_{22} & E_{23} & 0 & 0 & 0 \\ E_{13} & E_{23} & E_{33} & 0 & 0 & 0 \\ 0 & 0 & 0 & E_{44} & 0 & 0 \\ 0 & 0 & 0 & 0 & E_{55} & 0 \\ 0 & 0 & 0 & 0 & 0 & E_{66} \end{Bmatrix} = \begin{Bmatrix} 1/E_x & -\nu_{yx}/E_y & -\nu_{zx}/E_z & 0 & 0 & 0 \\ -\nu_{xy}/E_y & 1/E_y & -\nu_{zy}/E_z & 0 & 0 & 0 \\ -\nu_{xz}/E_y & -\nu_{yz}/E_y & 1/E_z & 0 & 0 & 0 \\ 0 & 0 & 0 & 1/E_{yz} & 0 & 0 \\ 0 & 0 & 0 & 0 & 1/E_{xz} & 0 \\ 0 & 0 & 0 & 0 & 0 & 1/E_{xy} \end{Bmatrix}^{-1} \tag{2}$$

where σ_x, σ_y, and σ_z are the normal stresses in the x, y, and z directions and τ_{yz}, τ_{xz}, and τ_{xy} are the shear stresses, respectively. E_x, E_y, and E_z are the Young's modulus values in the x, y, and z directions, and E_{xy}, E_{xz}, and E_{yz} are shear modulus values, respectively. υ is Poisson's ratio. The values used for Young's modulus and Poisson's ratio in this paper were taken from a reference Gibson and Ashby (1997).

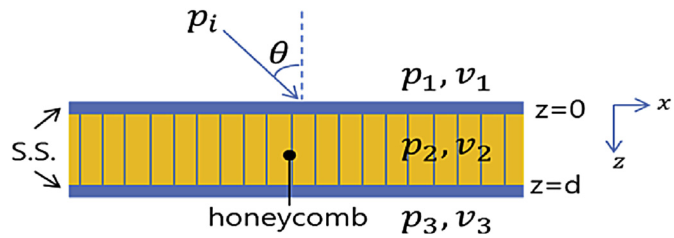

Fig. 1. S.S. + H/C + S.S.

The strains are defined using u, v, and w, which are the displacements in the x, y, and z directions, respectively. These displacements were introduced by Ford et al. (1967).

$$\varepsilon_x = \frac{\partial u}{\partial x}, \quad \varepsilon_y = \frac{\partial v}{\partial y}, \quad \varepsilon_z = \frac{\partial w}{\partial z},$$

$$\gamma_{xy} = \frac{\partial u}{\partial y} + \frac{\partial v}{\partial x}, \quad \gamma_{xz} = \frac{\partial u}{\partial z} + \frac{\partial w}{\partial x}, \quad \gamma_{yz} = \frac{\partial v}{\partial z} + \frac{\partial w}{\partial y} \tag{3}$$

where ε_x, ε_y, and ε_z are the strains in the x, y, and z directions, respectively, and γ_{xy}, γ_{xz}, and γ_{yz} are the shear strain.

The stored elastic potential energy density, W, for a given strain field is

$$2W = E_{11}\varepsilon_x^2 + 2E_{12}\varepsilon_x\varepsilon_y + 2E_{13}\varepsilon_x\varepsilon_z + E_{22}\varepsilon_y^2 + 2E_{23}\varepsilon_y\varepsilon_z + E_{33}\varepsilon_z^2 + E_{44}\gamma_{xy}^2 + E_{55}\gamma_{xz}^2 + E_{66}\gamma_{xy}^2 \tag{4}$$

The total elastic potential energy (*P.E.*) stored in a body of finite volume is computed as the integral of the potential energy density over the volume of the body, and the kinetic energies (*K.E.*) are similarly defined in terms of the volume integrals. Each equation is as follows:

$$P.E. = \int_{Vol} W \ dVol \tag{5}$$

$$K.E. = \frac{1}{2} \int_{Vol} \rho\left(\dot{u}^2 + \dot{v}^2 + \dot{w}^2\right) dVol \tag{6}$$