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Effects of Material and Geometrical Parameters on the Free Vibration of Sandwich Beams

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A B S T R A C T

The aim of present study is to investigate the effect of various parameters such as length, thickness, density, shear modulus of the core and Young modulus of skins for various boundary conditions, clamped-free and simply supported, by applying the model of Khalili et al. for free vibration analysis of sandwich beams by using finite element methods. The core density is taken in consideration. The flexural vibrations of beams are analyzed by the finite element method, using the stiffness and mass matrix of beam element with three degrees of freedom per node. The three first natural frequencies are calculated by using Matlab commercial software. A comparison is established between three different lengths for each configuration. There is a good agreement between the results obtained by the present study with the results obtained by Khalili et al.

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NOMENCLATURE

L	Length of the beam	b	Width of beam
hf	Thickness for each face sheet	Ef	Young's modulus for each face sheet
hc	Thickness of the core	Gc	Shear modulus of the core
A_f	Vertical area for each face sheet	A_c	Vertical area of the core
B_j	jth constant index	D	Differential operator
i	Counter for natural numbers	j	Counter for natural numbers
K_e^{FEM}	Element stiffness matrix	l	Length of an element
K	Global stiffness matrix	M	Global mass matrix
B	Boolean matrix	K_{des} , M_{des}	Unassembled matrix
m_c	Mass per unit length of the core	m_f	Mass per unit length for each face sheet
M_e^{FEM}	Element mass matrix	t	Time
T	Kinetic energy	u_e	Element displacement vector
u (x,t)	Axial displacement	u_0, u_1	Axial displacement of the nodes
U	Strain energy	V	Volume of the element
w (x,t)	Vertical displacement	w_0, w_1	Vertical displacement of the nodes
X	Axial direction	y	Member axis
Y	Vertical direction	Z	Width direction
γ_{xy}	Shear strain	δ	First variation operator
ε_x	Normal strain	θ(x,t)	Rotation displacement
θ_0, θ_1	Rotation displacement of the nodes	ρ_c	Density of the core
ρ_f	Density for each face sheet	σ_x	Normal stress
τ_{xy}	Shear stress	ω	Natural frequency
Over head (.)	(d/dt) first derivative as function of time	Over head (..)	(d^2/dt^2) second derivative as function of time
Over head (')	(d/dx) first derivative as function of X-axis	Over head ('')	(d^2/dx^2) second derivative as function of X-axis
Over head (''')	(d^3/dx^3) third derivative as function of X-axis	Over head (" ")	(d^4/dx^4) fourth derivative as function of X-axis

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1. INTRODUCTION

Structures composed of composite materials are among the most important structures used in modern engineering and, especially, in the aerospace industry. Such lightweight structures are also being increasingly used in civil, mechanical and transportation engineering applications. The rapid increase of industrial use of these structures has necessitated the development of new analytical and numerical tools suitable for the analysis and study of mechanical behavior of such structures as defined by Sayyad and Ghugal [1]. There are many researchers who exminated the free vibration problems of sandwich beams; Sakiyama et al. [2]; Fasana and Marchesiello [3] and Banerjee [4] have studied this problem by using an analytical model. They used the same assumptions like beams in Bernoulli-Euler flexure for the bottom and top elements but with the supplementary consideration that the central element deforms especially in transverse shear. Other studies added a direct stress carring capability to the central elementas developed by He and Rao [5] and Sisemore and Darvennes [6]. The model used is simple and assumes that the top and the bottom faces of a sandwich beam deform according to the Bernoulli-Euler beam theory, whereas the core deforms only in shear as can be see in work of Khalili et al. [7] and Banerjee and Sobey [8].

Frostig and Baruch [9] presented another model for sandwich beams analysis. The free vibrations of sandwich beams were analyzed with flexible core based on higher-order theory. Several scientists used the dynamic stiffness method (DSM) to investigate the free vibration of sandwich beams as seen in literature for example in Banerjee [4], Howson and Zare [10] and Khalili et al. [7].

They presented a dynamic stiffness formulation of three-layered sandwich beam to investigate the free vibration characteristics. The theory is based on an imposed displacement field where the top and bottom layers behave like Rayleigh beams whilst the central layer behaves like a Timoshenko beam. The boundary conditions for responses and loads at both ends of the freely vibrating sandwich beam are then imposed to formulate the dynamic stiffness matrix.

In this study, the free vibration of sandwich beams in different configurations is studied by solving the governing partial equations of motion and using the finite element formulation (FEM), using the stiffness matrix and mass matrix of sandwich beam element with three degrees of freedom per node obtained by the model of Khalili et al. [7]. The global matrix of mass and stiffness are obtained by using assembly method. The number of elements used in this study is 250 elements.

The programming of this method was performed under the Matlab software. Parametric studies were presented to show the effect of the ratio of some parameters: ρ_c/ρ_f, h_f/h_c, G/G_c and E_f/E_{f0} for different lengths of beams on the dynamic behavior of sandwich beams for various boundary conditions: clamped-free and simply supported.

TABLE 1. Geometrical dimensions and mechanical properties of sandwich beams [7].

N^0	L (mm)	h_f (mm)	h_c (mm)	b (mm)
1	914.4	0.4572	12.7	25.4
2	711.2	0.4572	12.7	25.4
3	1218.7	0.4062	6.35	25.4

TABLE 2. Geometrical dimensions and mechanical properties of sandwich beams [7].

E_f (Nm^{-2})	G_c (N/m^{-2})	ρ_f (kgm^{-3})	ρ_c (kgm^{-3})
6.89E+10	82.68E+6	2680	32.8
6.89E+10	82.68E+6	2680	32.8
6.89E+10	68.90E+6	2687.3	119.7

2. MATERIALS

Three types of sandwiches are used in this study. These materials are extracted from literature as see in the paper of Khalili et al. [7]. For the first and the second type of sandwiches, they have the same geometrical dimensions and material properties of the beam only in length which is different as reported in Table1. For the third type of sandwiches, the geometrical dimensions and material properties are presented in Table 1 and Table 2.

3. THEORY

The theoretical formulation applied in this study is the theory presented in paper of Khalili et al. [7] to develop its model which is applied below. The model developed by Khalili et al. [7] is based on the flowing assumptions:

•The behavior is considered in the case of classical simple and linear elasticity,

•Transverse normal strains are negligible in the faces and the core. That is the thickness of the beam remains constant at the time of dynamic deformations,

•There is no slippage or delamination between the layers.

Figure 1 shows the cross sectional displacements for symmetric sandwich beam element with length l and width b, as shown in Figure 1, the thickness of each face is equal to hf and the thickness of the core is hc. In the coordinate system shown, X-axis is in the direction of the beam, Y is in the bending direction (positive upward) and Z axis is in the beam width direction. All deformations of the beam are considered to be in XY plane.

Due to the symmetry of the motion, it can be shown as seen in the paper of Mead [11] that:

$$u_t = -u_b = u(x,t) \tag{1}$$

The beam displacement field is derived by compatibility conditions for deformations:

$$X:\begin{cases} Top\ face & u - yw' & \frac{-h_f}{2} \le y \le \frac{h_f}{2} \\ Core & \left(2u + h_f w'\right)\frac{y}{h_c} & \frac{-h_c}{2} \le y \le \frac{h_c}{2} \\ Bottom\ face & -u - yw' & \frac{-h_f}{2} \le y \le \frac{h_f}{2} \end{cases} \tag{2}$$

$$Y(for\ all\ layer):\ w \tag{3}$$

$$Z(for\ all\ layer):\ 0 \tag{4}$$

According to the presented assumptions, the strain energy of sandwich beam is given by:

$$\begin{aligned} U &= \frac{1}{2}\left\{\int_{V_t} \sigma_x \varepsilon_x dV_t + \int_{V_c} \tau_{xy}\gamma_{xy} dV_c + \int_{V_b} \sigma_x \varepsilon_x dV_b\right\} \\ &= \frac{1}{2}\left\{\int_{V_t} E_f \varepsilon_x^2 dV_t + \int_{V_c} G_c \gamma_{xy}^2 dV_c + \int_{V_b} E_f \varepsilon_x^2 dV_b\right\} \end{aligned} \tag{5}$$

where E_f is the Young's modulus for each face sheet and G_c is the shear modulus of the core. By substituting the strain values and calculating the integrals in Equation (5), the strain energy for a beam element is derived as:

$$U = \frac{1}{2}\int_0^1 \left\{ \begin{aligned} & 2E_f A_f u'^2 + 2E_f I_f w'^2 \\ & + G_c A_c \left[\frac{4}{h_c} u^2 + \left(\frac{h_f + h_c}{h_c}\right)^2 w'^2 + \frac{4}{h_c}\left(\frac{h_f + h_c}{h_c}\right) u w' \right] \end{aligned} \right\} dx \tag{6}$$

where:

$$dV = dA.dx, \qquad dA = b.dy \tag{7}$$

$$\left(A_f, 0, I_f\right) = \int_{A_{t,b}} \left(1, y, y^2\right) dA_{t,b}; \quad \left(A_c\right) = \int_{A_c} dA_c \tag{8}$$

The kinetic energy for this sandwich beam element is:

$$T = \frac{1}{2}\int_V \rho\left(\dot{X}^2 + \dot{Y}^2\right) dV = \frac{1}{2}\left\{ \begin{aligned} & \int_{V_t} \rho_f \left[\left(\dot{u} - y\dot{w}'\right)^2 + \left(-\dot{u} - y\dot{w}'\right)^2 + 2\dot{w}^2\right] dv_t \\ & + \int_{V_c} \rho_c \left[\left(2\dot{u} + h_f \dot{w}'\right)^2 \frac{y^2}{h_c^2} + \dot{w}^2\right] dv_c \end{aligned} \right\} \tag{9}$$

where ρ_f is density for each face sheet and ρ_c is the density for the core. Also, by calculating the integrals in Equation (9), the kinetic energy of this sandwich beam element is derived as:

$$T = \frac{1}{2}\int_0^1 \left\{ 2m_f \dot{u}^2 + \left(2m_f + m_c\right)\dot{w}^2 + \frac{m_c}{12}\left(2\dot{u} + h_f \dot{w}'\right)^2 \right\} dx \tag{10}$$

where:

$$\left(m_f\right) = \rho_f A_f; \qquad \left(m_c\right) = \rho_c A_c \tag{11}$$

by applying the Hamilton's principle, the governing partial equations of motion and boundary conditions of sandwich beam element are derived as follows:

$$\delta \int_{t_1}^{t_2} (T - U) dt = 0 \tag{12}$$

In the above equation, δ is the first variation operator and, t_1 and t_2 define the time intervals. Substituting Equations (6) and (10) in Equation (12) yields:

$$E_f A_f u'' - \frac{2G_c A_c}{h_c^2} u - \frac{G_c A_c \left(h_f + h_c\right)}{h_c^2} w' - \left(m_f + \frac{m_c}{6}\right)\ddot{u} - \frac{m_c h_f}{12}\ddot{w}' = 0 \tag{13}$$

$$\begin{aligned} & E_f I_f w'''' - \frac{G_c A_c \left(h_f + h_c\right)^2}{2h_c^2} w'' - \frac{G_c A_c \left(h_f + h_c\right)^2}{h_c^2} u' + \\ & \left(m_f + \frac{m_c}{2}\right)\ddot{w} - \frac{m_c h_f^2}{24}\ddot{w}'' - \frac{m_c h_f}{12}\ddot{u} = 0 \end{aligned} \tag{14}$$

The results of finite element formulation (FEM) are the most appropriate estimation for the results of dynamic stiffness approach. The C_0 continuity and C_1 continuity are applied to analyze the sandwich beam shown in Figure 2:

$$\begin{cases} u = B_1 + B_2 x \\ w = B_3 + B_4 x + B_5 x^2 + B_6 x^3 \end{cases} \tag{15}$$

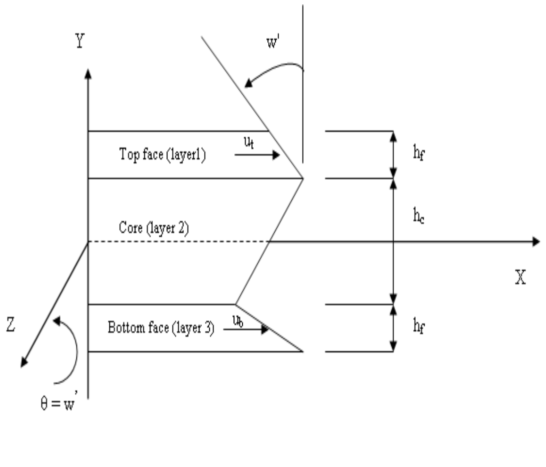

Figure 1. Coordinate system and notation for symmetric sandwich beam [7]

Substituting Equation (15) into Equations (13) and (14) and using the finite element formulation, gives the followings :

$$0 = \left(K_e^{FEM} - \omega^2 M_e^{FEM}\right) \quad (16)$$

The element stiffness matrix Ke^{FEM} and the element mass matrix Me^{FEM} which are 6×6 symmetric matrices can be obtained.

$$K = B^T K_{des} B$$
$$M = B^T M_{des} B \quad (17)$$

The squared natural frequencies are calculated from the eigen-values of (KM^{-1}) where K and M, the global matrix of stiffness and mass [12-17] are obtained by using assembly method. Where:

- B is the Boolean matrix.
- K_{des} and M_{des} are unassembled matrix, they contain only elementary matrix of mass and stiffness.

$$K_{des} = \begin{bmatrix} \left[K_e^1\right] & 0 \\ 0 & \ddots\left[K_e^N\right] \end{bmatrix} \quad (18)$$

$$M_{des} = \begin{bmatrix} \left[M_e^1\right] & 0 \\ 0 & \ddots\left[M_e^N\right] \end{bmatrix} \quad (19)$$

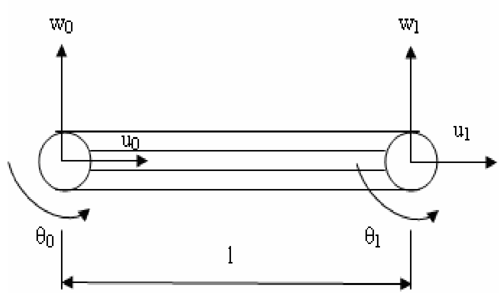

Figure 2. Symmetric sandwich beam element and displacements notations [7]

TABLE 3. Third lowest natural frequencies for a configuration C-F of sandwich beam

Methods	Frequency (Hz)		
	ω_1	ω_2	ω_3
Present study (FEM-250)	33.63	199.73	511.34
Khalili et al. [7] (FEM-250)	33.74	198.79	511.40
Khalili et al. [7] (FEM-500)	33.74	198.78	511.37
Ahmed [12]	33.97	200.5	517
Ahmed [18]	32.79	193.5	499
Sakiyama et al. [2]	33.14	195.96	503.43

TABLE 4. Third lowest natural frequencies for a configuration S-S of sandwich beam

Methods	Frequency (Hz)		
	ω_1	ω_2	ω_3
Present study (FEM-250)	57.50	225.82	464.45
Khalili et al. [7] (FEM-250)	57.12	219.42	464.58
Khalili et al. [7] (FEM-500)	57.12	219.42	464.57
Ahmed [12]	57.5	-	467
Ahmed [18]	55.5	-	451
Sakiyama et al. [2]	56.15	215.82	457.22

4. RESULTS AND DISCUSSION

In this section, a parametric study is developed for different configurations. In our case, we have two configurations: clamped-free (C-F) and simply supported (S-S). For each configuration, we presented the effect of the ratio of some parameters: ρ_c/ρ_f, h_f/h_c, G/G_c and E_f/E_{f0} for different lengths of beams (L1= 711.2 mm, L2= 914.4 mm and L3= 1218.7 mm) on the dynamic behavior of sandwich beams.

The first step in this study is to validate the efficiency of the programme in Matlab software and the results are reported in Table 3 and Table 4 for the two configurations (C-F, S-S):

4. 1. First Configuration: Clamped-Free(C-F)

4. 1. 1. ρ_c/ρ_f Figure 3 shows a decrease in the first natural frequency when the ratio of ρ_c/ρ_f increases for the three lengths studied. We observe also a slight decrease in frequency for L2 compared to L1 (4.16%) but for L3 there is a higher increase in frequency (18.43%). These curves show the effect of length on the first natural frequency of sandwich beam.

4. 1. 2. h_f/h_c As shown in Figure 4, increasing the ratio of h_f/h_c, increases the first natural frequency of sandwich beam for the three lengths studied. We observe also a slight decrease in frequency for L2 than L1 (4.22%) but for L3 there is a higher decrease in frequency (40.11%). These curves show the effect of length on the first natural frequency of sandwich beam.

4. 1. 3. G/G_c The variation of the first natural frequency is shown in Figure 5; increasing the ratio of G/Gc, increases the first natural frequency of sandwich beam for the three lengths studied. We observe also a same value in frequency with L2 and L3 for the ratio equal to 0.01 and a slight decrease in frequency for L2 than L1(3.71%) for the ratio equal to 0.20 but for L3 there is an increase in frequency (13.10%) for the ratio equal to 0.20. These curves show the effect of length on the first natural frequency of sandwich beam.

4. 1. 4. E_f/E_{f0} Figure 6 shows a decrease in the first natural frequency when the ratio of E_f/E_{f0} decreases for the length L1 studied. We observe also a decrease in frequency (9.75%). This curve shows the decrease in natural frequency which can be an important tool to detect the deterioration of the sandwich material.

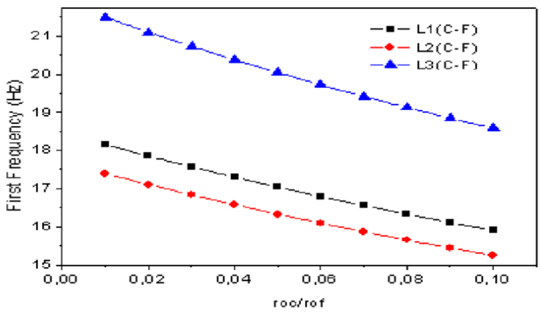

Figure 3. Evolution of the first natural frequency with C-F configuration with the variation of the ratio ρ_c/ρ_f for different lengths.

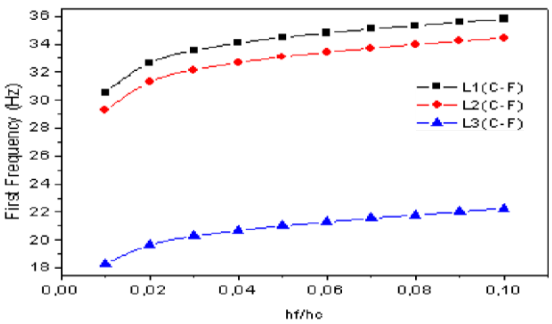

Figure 4. Evolution of the first natural frequency with C-F configuration with the variation of the ratio h_f/h_c for different lengths.

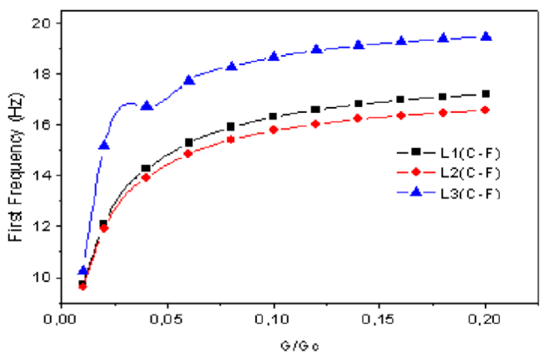

Figure 5. Evolution of the first natural frequency with C-F configuration with the variation of the ratio G/G_c for different lengths.

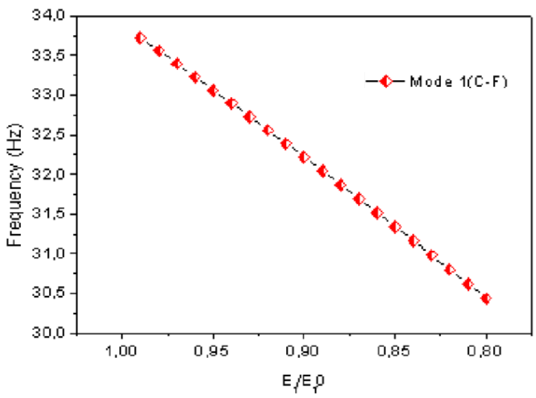

Figure 6. Evolution of the first natural frequency with C-F configuration with the variation of the ratio E_f/E_{f0} for length L1.

4. 2. Second Configuration: Simply Supported (S-S)

4. 2. 1. ρ_c/ρ_f Figure 7 shows a decrease in the first natural frequency when the ratio of ρ_c/ρ_f increases for the three lengths studied. We observe also a slight increase in frequency for L2 than L1(0.65%) but for L3 there is a higher decrease in frequency (28.47%) for the ratio equal to 0.01 but for the ratio 0.1 this value becomes 40.90%. These curves show the effect of length on the first natural frequency of sandwich beam.

4. 2. 2. h_f/h_c As shown in Figure 8, increasing the ratio of h_f/h_c, increases the first natural frequency of sandwich beam for the three lengths studied. We observe also a slight increase in frequency for L2 than L1(0.58%) but for L3 there is a higher decrease in frequency (28.82%) than L1 except for the ratio 0.02 (43.84%). These curves show the effect of length on the first natural frequency of sandwich beam.

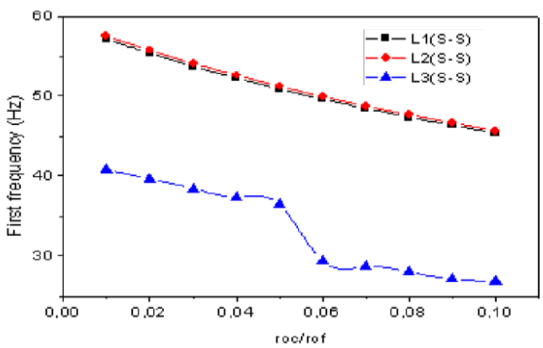

Figure 7. Evolution of the first natural frequency with S-S configuration with the variation of the ratio ρ_c/ρ_f for different lengths.

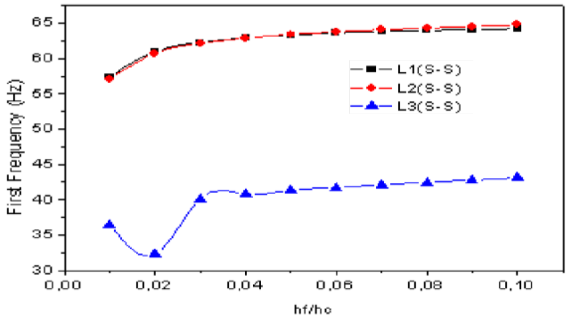

Figure 8. Evolution of the first natural frequency with S-S configuration with the variation of the ratio h_f/h_c for different lengths.

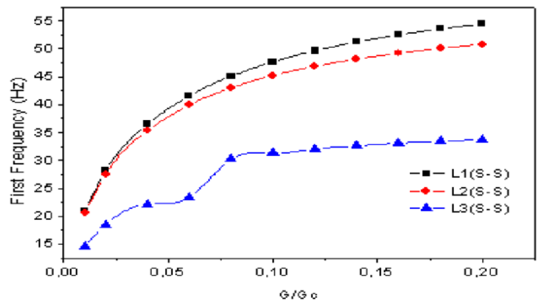

Figure 9. Evolution of the first natural frequency with S-S configuration with the variation of the ratio G/G_c for different lengths.

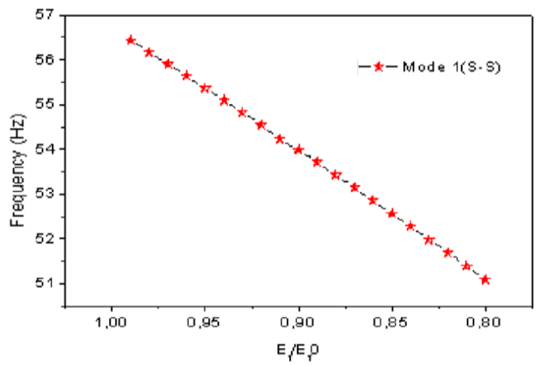

Figure 10. Evolution of the first natural frequency with S-S configuration with the variation of the ratio E_f/E_{f0} for length L1.

4. 2. 3. G/G_c The variation of the first natural frequency is shown in Figure 9; increasing the ratio of G/G_c, increases the first natural frequency of sandwich beam for the three lengths studied. We observe also a same values in frequency with L2 and L3 for the ratio equal to 0.01 and a slight decrease in frequency for L2 than L1(6.77%) for the ratio equal to 0.20 but for L3 there is a decrease in frequency (30.74%) for the ratio equal to 0.20. These curves show the effect of length on the first natural frequency of sandwich beam.

4. 2. 4. E_f/E_{f0} Figure 10 shows a decrease in the first natural frequency when the ratio of E_f/E_{f0} decreases for the length L1. We observe also a decrease in frequency (9.48%). This curve shows the decrease in natural frequency which can be an important tool to detect the deterioration of the sandwich material.

5.CONCLUSION

In this study, free vibration of the symmetric sandwich beam is presented by finite element method. By applying the model of Khalili et al. for different configurations: clamped-free (C-F) and simply supported (S-S). The element stiffness matrix K_e^{FEM} and the element mass matrix M_e^{FEM} are 6×6 symmetric matrix with three degrees of freedom per node. The squared natural frequencies are calculated from the eigen-values of (KM^{-1}) where K and M, the global of stiffness and mass matrix, are obtained by using assembly method. We used a Matlab sofware for programming this method. Parametric studies were presented to show the effect of the ratio of some parameters: ρ_c/ρ_f, h_f/h_c, G/G_c and E_f/E_{f0} for different lengths of beams (L1= 711.2 mm, L2= 914.4 mm and L3= 1218.7 mm) on the dynamic behavior of sandwich beams. These studies indicated that:

- Increasing the core/face density ratios (ρ_c/ρ_f), decreases the first natural frequency of sandwich beam for all configurations studied.
- Decreasing the face Young's modulus ratios (E_f/E_{f0}), decreases the first natural frequency of sandwich beam for all configurations studied.
- Increasing the face/core thickness ratios (h_f/h_c), increases the first natural frequency of sandwich beam for all configurations studied.
- Increasing the core shear modulus ratios (G/G_c), increases the first natural frequency of sandwich beam for all configurations studied.

A comparison is established between three different lengths for each configuration. For all parameters studied, this study shows that there is an effect of length when we calculated the first natural frequencies. There is a good agreement between the results obtained by the present study with the results obtained by Khalili et al.

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Effects of Material and Geometrical Parameters on the Free Vibration of Sandwich Beams

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چکیده

هدف از مطالعه حاضر، بررسی اثر پارامترهای مختلف از قبیل طول، ضخامت، چگالی، مدول برشی از هسته و مدول یانگ از پوست برای شرایط مختلف مرزی، کلمپ-رایگان و ساده، با استفاده از مدل خلیلی و همکاران برای تجزیه و تحلیل ارتعاشات آزاد پرتوهای ساندویچ از روش المان محدود است. چگالی هسته مورد بررسی قرار می‌گیرد. ارتعاشات خمشی تیرها توسط روش المان محدود با استفاده از ماتریس سختی و جرم عنصر پرتو با سه درجه آزادی در هر گره مورد تجزیه و تحلیل قرار می‌گیرد. سه فرکانس طبیعی برای اولین بار با استفاده از نرم افزار تجاری Matlab محاسبه می شود. برای هر پیکربندی، مقایسه‌ای بین سه طول مختلف انجام می‌شود. توافق خوبی بین نتایج به دست آمده از مطالعه حاضر با نتایج به دست آمده توسط خلیلی و همکاران وجود دارد.

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