STATIC ANALYSIS OF SIMPLY SUPPORTED FOAM CORE SANDWICH PLATE

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Abstract

This paper presents a static analysis for simply supported sandwich plate with foam core by using refined plate theory. Face layers of the sandwich plate are assumed isotropic, core of the sandwich is assumed foam. In the bending of sandwich plates, the bending load is carried by face layers while the transverse shear loads are carried mainly by the core. There are many papers in the literature on the analysis of the bending, buckling and vibration of sandwich structures. Classical lamination theory, first order shear deformation theory and various shear deformation theories are applied for analysis of sandwich plates. In this study static analysis for simply supported sandwich plate is investigated. Governing equations are derived from the principle of virtual displacements. A closed-form solution is found by using the Navier's method. Numerical results are performed parametrically. The effect of density and pore size of foam on stress distribution through thickness are investigated.

Key words: *sandwich plates, naviers solution, bending, stress, shear strain*

1. INTRODUCTION

Sandwich structures are defined as a thick core layer between two thin layers which are fairly tough (Fig.1). The material of the face layer may be isotropic, composite or laminated composite (Vinson 1999). Main functions of the face layers are to keep the structure together and transfer the load to the core (Carlsson & Kardomateas 2011).

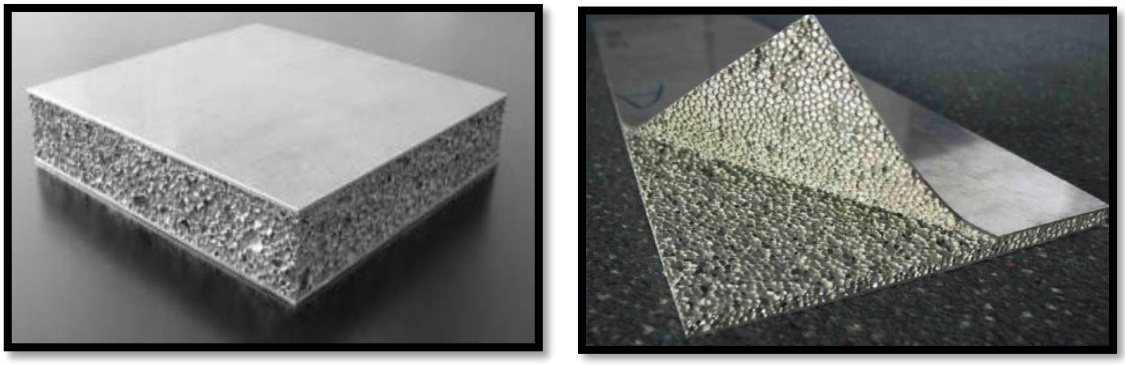


Fig. 1. Sandwich structure with metal foam core (Demirhan & Taskin, 2015)

Nowadays sandwich structures have become more popular in many industrial fields such as aerospace, naval, automotive (Fig.2) etc.



Fig. 2. Automotive parts (Demirhan 2009)

The materials for the core of sandwich (Fig.3) can be chosen such as wood, honeycomb or cellular metals (Banhart et al. 2008).



Fig. 3. Sandwich core materials (Demirhan & Taskin, 2015)

Metal foams are cellular metals which known to combine the low density with high strength (Taskin et al. 2010a). Due to the unique properties such as

- inflammability,
- sound insulation,
- vibration damping,
- electricity insulation,
- thermal insulation,
- low weight
- relatively low price metal foams are very attractive for the core material (Taskin et al. 2011b).



Fig. 4. Metal foam (Demirhan 2009)

Mechanical properties of metal foam are mainly related to the density (Taskin et al. 2011a). Ashby (2000) is figured out the relation between the elasticity modulus of metal foam with the relative density (density of foam / density of solid metal) of foam (Demirhan et al. 2011). E_f represents metal foam elasticity modulus, E_s is the solid metal's modulus, ρ_f and ρ_s are represent the density of foam and solid metal respectively.

$$E_f = (0.1 - 1.0)E_s\left[0.5\left(\frac{\rho_f}{\rho_s}\right)^2 + 0.3\left(\frac{\rho_f}{\rho_s}\right)\right] \quad (1)$$

With reducing relative density is obtained low weight but the elasticity modulus of metal foam is become smaller (Taskin et al. 2010c). Due to achieve desired strength, metal foams are to be used as a core of the sandwich structure.

2. METHOD

In the study a sandwich plate with the length of *a*, width *b* and thickness of *h* is considered. The coordinate axes are placed at the mid-plane of the sandwich plate as shown in the Fig. 5 and Fig. 6.

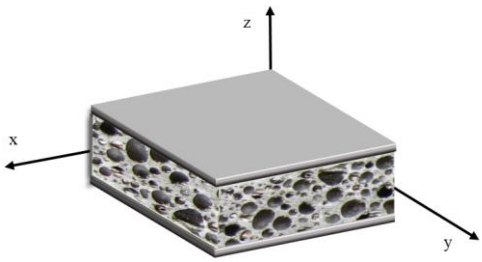


Fig. 5. Sandwich plate with foam core



Fig. 6. Thickness of sandwich plate

Elasticity modulus of face layer is E_f and elasticity modulus of core is E_c

$$h = t_c + 2t_f \quad (2)$$

$$E(z) = \begin{cases} E_f & -\frac{h}{2} < z < -\frac{h}{2} + t_f \\ E_c & -\frac{h}{2} + t_f < z < \frac{h}{2} - t_f \\ E_f & \frac{h}{2} - t_f < z < \frac{h}{2} \end{cases} \quad (3)$$

Face layers are assumed to be isotropic. Metal foam is chosen for the core material of the sandwich plate. Elasticity modulus of metal foam is related to density. It is assumed also isotropic.

$$E_c = (0.1)E_s\left[0.5\left(\frac{\rho_f}{\rho_s}\right)^2 + 0.3\left(\frac{\rho_f}{\rho_s}\right)\right] \tag{4}$$

E_s : Elasticity modulus of solid materials which metal foams are made from

ρ_s : density of the solid material

ρ_f : density of metal foam

Refined plate theory is studied (Shimpi 2002, Shimpi & Patel 2006, Kim, Thai & Lee, 2009, Mechab et al. 2010, Abdelaziz et al. 2011). Displacement components u, v and w along the x, y and z directions, respectively, the following displacement field of the high order deformation theory can be written (Zenkour 2005):

$$u = -z\frac{\partial w_b}{\partial x} + \left[\frac{h}{\pi}\sin\frac{\pi z}{h}\right]\frac{\partial w_s}{\partial x} \tag{5a}$$

$$v = -z\frac{\partial w_b}{\partial y} + \left[\frac{h}{\pi}\sin\frac{\pi z}{h}\right]\frac{\partial w_s}{\partial y} \tag{5b}$$

$$w = w_b + w_s \tag{5c}$$

Strain displacement relation is shown in Eq. (6a)

$$\varepsilon_x = \frac{\partial u}{\partial x} - z\frac{\partial^2 w_b}{\partial x^2} + \left[\frac{h}{\pi}\sin\frac{\pi z}{h}\right]\frac{\partial^2 w_s}{\partial x^2} \tag{6a}$$

$$\varepsilon_y = \frac{\partial v}{\partial y} - z\frac{\partial^2 w_b}{\partial y^2} + \left[\frac{h}{\pi}\sin\frac{\pi z}{h}\right]\frac{\partial^2 w_s}{\partial y^2} \tag{6b}$$

$$\varepsilon_z = 0 \tag{6c}$$

$$\gamma_{xy} = \frac{\partial u}{\partial y} + \frac{\partial v}{\partial x} - 2z\frac{\partial^2 w_b}{\partial x \partial y} + 2\left[\frac{h}{\pi}\sin\frac{\pi z}{h}\right]\frac{\partial^2 w_s}{\partial x \partial y} \tag{6d}$$

$$\gamma_{yz} = \left[1 + \cos(\frac{\pi z}{h})\right]\frac{\partial w_s}{\partial y} \tag{6e}$$

$$\gamma_{xz} = \left[1 + \cos(\frac{\pi z}{h})\right]\frac{\partial w_s}{\partial x} \tag{6f}$$

It can be derived constitutive relations for isotropic functionally graded materials as follows.

$$\begin{Bmatrix}\sigma_x\\ \sigma_y\\ \tau_{xy}\end{Bmatrix}=\begin{bmatrix}Q_{11} & Q_{12} & 0\\ Q_{12} & Q_{22} & 0\\ 0 & 0 & Q_{66}\end{bmatrix}\begin{Bmatrix}\varepsilon_x\\ \varepsilon_y\\ \gamma_{xy}\end{Bmatrix} \tag{7}$$

$$\begin{Bmatrix}\tau_{yz}\\ \tau_{xz}\end{Bmatrix}=\begin{bmatrix}Q_{44} & 0\\ 0 & Q_{55}\end{bmatrix}\begin{Bmatrix}\gamma_{yz}\\ \gamma_{xz}\end{Bmatrix} \tag{8}$$

$$Q_{11}=Q_{22}=\frac{E(z)}{1-\nu^2} \qquad Q_{12}=\frac{\nu E(z)}{1-\nu^2} \qquad Q_{44}=Q_{55}=Q_{66}=\frac{E(z)}{2(1+\nu)} \tag{9}$$

Governing equations are derived using virtual displacement principles.

$$\delta(U-W)=0 \tag{10}$$

$$W=\int_\Omega qW d\Omega \tag{11}$$

$$U=\int_{-h/2}^{h/2}\int_\Omega(\sigma_x\varepsilon_x+\sigma_y\varepsilon_y+\tau_{xy}\gamma_{xy}+\tau_{yz}\gamma_{yz}+\tau_{xz}\gamma_{xz})\,d\Omega dz \tag{12}$$

$$\begin{aligned}\delta(U-W)=\int_\Omega\Bigg[&N_x\delta\left(\frac{\partial u}{\partial x}\right)+N_y\delta\left(\frac{\partial v}{\partial y}\right)+N_{xy}\delta\left(\frac{\partial u}{\partial y}+\frac{\partial v}{\partial x}\right)-M_x^b\delta\left(\frac{\partial^2 w_b}{\partial x^2}\right)-M_y^b\delta\left(\frac{\partial^2 w_b}{\partial y^2}\right)\\&-M_{xy}^b\delta\left(\frac{\partial^2 w_b}{\partial x\partial y}\right)-M_x^s\delta\left(\frac{\partial^2 w_s}{\partial x^2}\right)-M_y^s\delta\left(\frac{\partial^2 w_s}{\partial y^2}\right)-M_{xy}^s\delta\left(\frac{\partial^2 w_s}{\partial x\partial y}\right)+S_{yz}^s\delta\left(\frac{\partial w_s}{\partial y}\right)\\&+S_{xz}^s\delta\left(\frac{\partial w_s}{\partial x}\right)\Bigg]d\Omega-\int_\Omega q(\delta w_b+\delta w_s)d\Omega=0\end{aligned} \tag{13}$$

Force and moment resultants can be derived as follows.

$$N_x=\int_{-h/2}^{h/2}\sigma_x dz,\quad N_y=\int_{-h/2}^{h/2}\sigma_y dz,\quad N_{xy}=\int_{-h/2}^{h/2}\tau_{xy}dz \tag{14}$$

$$M_x^b=\int_{-\frac{h}{2}}^{\frac{h}{2}}\sigma_x z dz,\quad M_y^b=\int_{-\frac{h}{2}}^{\frac{h}{2}}\sigma_y z dz,\quad M_{xy}^b=\int_{-\frac{h}{2}}^{\frac{h}{2}}\tau_{xy}z dz \tag{15}$$

$$M_x^s=\int_{-\frac{h}{2}}^{\frac{h}{2}}\sigma_x\left[\frac{h}{\pi}\sin\frac{\pi z}{h}\right]dz,\quad M_y^s=\int_{-\frac{h}{2}}^{\frac{h}{2}}\sigma_y\left[\frac{h}{\pi}\sin\frac{\pi z}{h}\right]dz \tag{16}$$

$$M_{xy}^{s} = \int_{-\frac{h}{2}}^{\frac{h}{2}} \tau_{xy} \left[\frac{h}{\pi} \sin\frac{\pi z}{h}\right] dz \tag{17}$$

$$S_{xz}^{s} = \int_{-h/2}^{h/2} \tau_{xz} \left[1 + \cos(\frac{\pi z}{h})\right] dz \ , \quad S_{yz}^{s} = \int_{-h/2}^{h/2} \tau_{yz} \left[1 + \cos(\frac{\pi z}{h})\right] dz \tag{18}$$

Rigidity matrix is derived in Eq. (16)

$$\begin{bmatrix} N_x \\ M_b \\ M_s \end{bmatrix} = \begin{bmatrix} A & B & B_s \\ B & D & D_s \\ B_s & D_s & H_s \end{bmatrix} \begin{bmatrix} \frac{\partial u_0}{\partial x} \\ z\frac{\partial w_b}{\partial x} \\ z\frac{\partial w_s}{\partial x} \end{bmatrix} \tag{19}$$

$$[Q] = [A_s] \left[\left[1 + \cos(\frac{\pi z}{h})\right] \frac{dw_s}{dx}\right] \tag{20}$$

Rigidity matrix coefficients can be derived as shown in Eq. (18)

$$A_{ij} = \int_{-\frac{h}{2}}^{\frac{h}{2}} Q_{11} dz \qquad B_{ij} = \int_{-\frac{h}{2}}^{\frac{h}{2}} z Q_{11} dz \qquad D_{ij} = \int_{-\frac{h}{2}}^{\frac{h}{2}} z^2 Q_{ij} dz \tag{21a}$$

$$B_{ij}^{s} = \int_{-\frac{h}{2}}^{\frac{h}{2}} \left[\frac{h}{\pi} \sin\frac{\pi z}{h}\right] Q_{ij} dz \quad D_{ij}^{s} = \int_{-\frac{h}{2}}^{\frac{h}{2}} z \left[\frac{h}{\pi} \sin\frac{\pi z}{h}\right] Q_{ij} dz \quad H_{ij}^{s} = \int_{-\frac{h}{2}}^{\frac{h}{2}} \left[\frac{h}{\pi} \sin\frac{\pi z}{h}\right]^2 Q_{ij} dz \tag{21b}$$

$$(i, j = 1,2,6)$$

$$A_{ij}^{s} = \int_{-\frac{h}{2}}^{\frac{h}{2}} \left[1 + \cos(\frac{\pi z}{h})\right]^2 Q_{ij} dz \qquad (i, j = 4,5) \tag{21c}$$

Equilibrium equation can be obtained as

$$\frac{\partial N_x}{\partial x} + \frac{\partial N_{xy}}{\partial y} = 0 \tag{22}$$

$$\frac{\partial N_{xy}}{\partial x} + \frac{\partial N_y}{\partial y} = 0 \tag{23}$$

$$\frac{\partial^2 M_x^b}{\partial x^2} + 2\frac{\partial M_{xy}^b}{\partial x \partial y} + \frac{\partial^2 M_y^b}{\partial y^2} + q = 0 \tag{24}$$

$$\frac{\partial^2 M_x^s}{\partial x^2} + 2\frac{\partial M_{xy}^s}{\partial x \partial y} + \frac{\partial^2 M_y^s}{\partial y^2} + \frac{\partial N_{xz}^s}{\partial x} + \frac{\partial N_{yz}^s}{\partial y} + q = 0 \quad (25)$$

Force and moment resultant defined as

$$A_{11}\frac{\partial^2 u}{\partial x^2} + A_{66}\frac{\partial^2 u}{\partial y^2} + (A_{12} + A_{66})\frac{\partial^2 v}{\partial x \partial y} - B_{11}\frac{\partial^3 w_b}{\partial x^3} - (B_{12} + 2B_{66})\frac{\partial^3 w_b}{\partial x \partial y^2} - B_{11}^s\frac{\partial^3 w_s}{\partial x^3} - (B_{12}^s + 2B_{66}^s)\frac{\partial^3 w_s}{\partial x \partial y^2} = 0 \quad (26)$$

$$(A_{12} + A_{66})\frac{\partial^2 u}{\partial x \partial y} + A_{66}\frac{\partial^2 v}{\partial x^2} + A_{22}\frac{\partial^2 v}{\partial y^2} - (B_{12} + 2B_{66})\frac{\partial^3 w_b}{\partial x^2 \partial y} - B_{22}\frac{\partial^3 w_b}{\partial y^3} - (B_{12}^s + 2B_{66}^s)\frac{\partial^3 w_s}{\partial x^2 \partial y} - B_{22}^s\frac{\partial^3 w_s}{\partial y^3} = 0 \quad (27)$$

$$B_{11}\frac{\partial^3 u}{\partial x^3} + (B_{12} + 2B_{66})\frac{\partial^3 u}{\partial x \partial y^2} + (B_{12} + 2B_{66})\frac{\partial^3 v}{\partial x^2 \partial y} + B_{22}\frac{\partial^3 v}{\partial y^3} - D_{11}\frac{\partial^4 w_b}{\partial x^4} - 2(D_{12} + 2D_{66})\frac{\partial^4 w_b}{\partial x^2 \partial y^2} - D_{22}\frac{\partial^4 w_b}{\partial y^4} - D_{11}^s\frac{\partial^4 w_s}{\partial x^4} - 2(D_{12}^s + 2D_{66}^s)\frac{\partial^4 w_s}{\partial x^2 \partial y^2} - D_{22}^s\frac{\partial^4 w_s}{\partial y^4} + q = 0 \quad (28)$$

$$B_{11}^s\frac{\partial^3 w_s}{\partial x^3} + (B_{12}^s + 2B_{66}^s)\frac{\partial^3 u}{\partial x \partial y^2} + B_{22}^s\frac{\partial^3 v}{\partial y^3} - D_{11}^s\frac{\partial^4 w_b}{\partial x^4} - D_{22}^s\frac{\partial^4 w_b}{\partial y^4} + (B_{12}^s + 2B_{66}^s)\frac{\partial^3 u}{\partial x^2 \partial y} - H_{11}^s\frac{\partial^4 w_s}{\partial x^4} - H_{22}^s\frac{\partial^4 w_s}{\partial y^4} - 2(D_{12}^s + 2D_{66}^s)\frac{\partial^4 w_b}{\partial x^2 \partial y^2} - 2(H_{12}^s + 2H_{66}^s)\frac{\partial^4 w_s}{\partial x^2 \partial y^2} + A_{55}^s\frac{\partial^2 w_s}{\partial x^2} + A_{44}^s\frac{\partial^2 w_s}{\partial y^2} + q = 0 \quad (29)$$

Simply supported boundary conditions for the refined theory are;

$$v(0,y) = w_b(0,y) = w_s(0,y) = \frac{\partial w_b}{\partial y}(0,y) = \frac{\partial w_s}{\partial y}(0,y) = 0 \quad (30)$$

$$v(a,y) = w_b(a,y) = w_s(a,y) = \frac{\partial w_b}{\partial y}(a,y) = \frac{\partial w_s}{\partial y}(a,y) = 0 \quad (31)$$

$$N_x(0,y) = M_x^b(0,y) = M_x^s(0,y) = N_x(a,y) = M_x^b(a,y) = M_x^s(a,y) = 0 \quad (32)$$

$$u(x,0) = w_b(x,0) = w_s(x,0) = \frac{\partial w_b}{\partial x}(x,0) = \frac{\partial w_s}{\partial x}(x,0) = 0 \quad (33)$$

$$u(x,b) = w_b(x,b) = w_s(x,b) = \frac{\partial w_b}{\partial x}(x,b) = \frac{\partial w_s}{\partial x}(x,b) = 0 \quad (34)$$

$$N_y(x,0) = M_y^b(x,0) = M_y^s(x,0) = N_y(x,b) = M_y^b(x,b) = M_y^s(x,b) = 0 \quad (35)$$

The Navier solution procedure is used to determine the analytical solutions for a simply supported plate. The external force is defined with double trigonometric series as Navier's method (Eq. 36).

$$q(x,y) = \sum_{m=1}^{\infty}\sum_{n=1}^{\infty} q_{mn}\sin(\alpha x)\sin(\beta y) \qquad m = n = 1,\ \ q_{11} = q_0 \tag{36}$$

Displacement components are defined as in Eq.(37)

$$\begin{Bmatrix} u \\ v \\ w_b \\ w_s \end{Bmatrix} = \sum_{m=1}^{\infty}\sum_{n=1}^{\infty} \begin{Bmatrix} U_{mn}\cos(\alpha x)\sin(\beta y) \\ V_{mn}\sin(\alpha x)\cos(\beta y) \\ W_{bmn}\sin(\alpha x)\sin(\beta y) \\ W_{smn}\sin(\alpha x)\sin(\beta y) \end{Bmatrix} \tag{37}$$

U_{mn}, V_{mn}, W_{bmn}, W_{smn} unknown constants are found with solving the Eq. (35). K is the coefficient matrix, F is the external forces vector.

$$K\Delta = F \tag{38}$$

$$\Delta^T = \{U_{mn},\quad V_{mn},\quad W_{bmn},\quad W_{smn}\} \tag{39}$$

$$F^T = \{0,\quad 0,\quad -q_{mn},\quad -q_{mn}\} \tag{40}$$

$$K = \begin{bmatrix} A_{11}\alpha^2 + A_{66}\beta^2 & \alpha\beta(A_{12}+A_{66}) & -\alpha[B_{11}\alpha^2 + (B_{12}+2B_{66})\beta^2] & -\alpha[B^s_{11}\alpha^2 + (B^s_{12}+2B^s_{66})\beta^2] \\ \alpha\beta(A_{12}+A_{66}) & A_{66}\alpha^2 + A_{22}\beta^2 & -\beta[(B_{12}+2B_{66})\alpha^2 + B_{22}\beta^2] & -\beta[(B^s_{12}+2B^s_{66})\alpha^2 + B^s_{22}\beta^2] \\ -\alpha[B_{11}\alpha^2 + (B_{12}+2B_{66})\beta^2] & -\beta[(B_{12}+2B_{66})\alpha^2 + B_{22}\beta^2] & D_{11}\alpha^4 + 2(D_{12}+2D_{66})\alpha^2\beta^2 + D_{22}\beta^4 & D^s_{11}\alpha^4 + 2(D^s_{12}+2D^s_{66})\alpha^2\beta^2 + D^s_{22}\beta^4 \\ -\alpha[B^s_{11}\alpha^2 + (B^s_{12}+2B^s_{66})\beta^2] & -\beta[(B^s_{12}+2B^s_{66})\alpha^2 + B^s_{22}\beta^2] & D^s_{11}\alpha^4 + 2(D^s_{12}+2D^s_{66})\alpha^2\beta^2 + D^s_{22}\beta^4 & H^s_{11}\alpha^4 + 2(H^s_{12}+2H^s_{66})\alpha^2\beta^2 + H^s_{22}\beta^4 + A^s_{55}\alpha^2 + A^s_{44}\beta^2 \end{bmatrix} \tag{41}$$

3. NUMERICAL RESULTS

For numerical results, the material properties of face layers are given in Table 1.

Table 1. Face Layer Material Properties

	Aluminum
Elasticity modulus	70 GPa
Poisson ratio	0.33
Density	2.701 g/cm^3

Core material is chosen metal foam which is made from the metal of face layers. Properties of core layer are changed with Eq.4. Elasticity modulus change of core layer is given in Fig. 7.

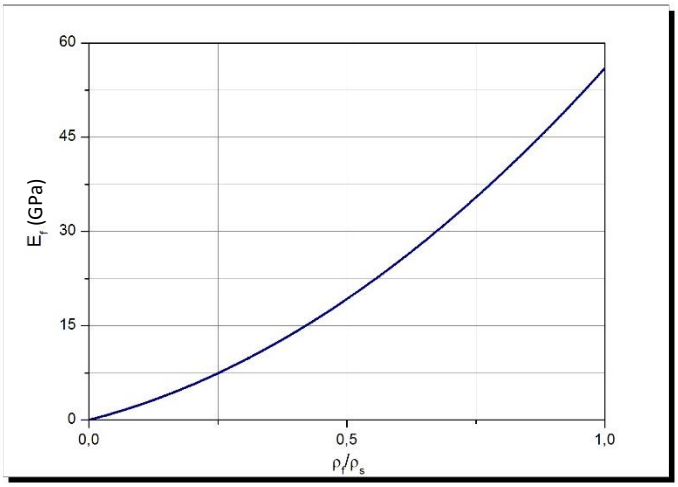


Fig. 7. Variation of elasticity modulus ratio

In the Fig. 7, the variation of elasticity modulus ratio of solid and metal foam with respect to relative density is illustrated. Depending on the increase in relative density, the elasticity modulus ratio of solid and metal foam also increases.

The deflection parameters are compared for the ratio of face layer thickness (t_h) according to the total thickness (h) of sandwich plate. It is obvious from the Fig. 8 that since the ratio decreases, the maximum deflection of the plate increases adversely.

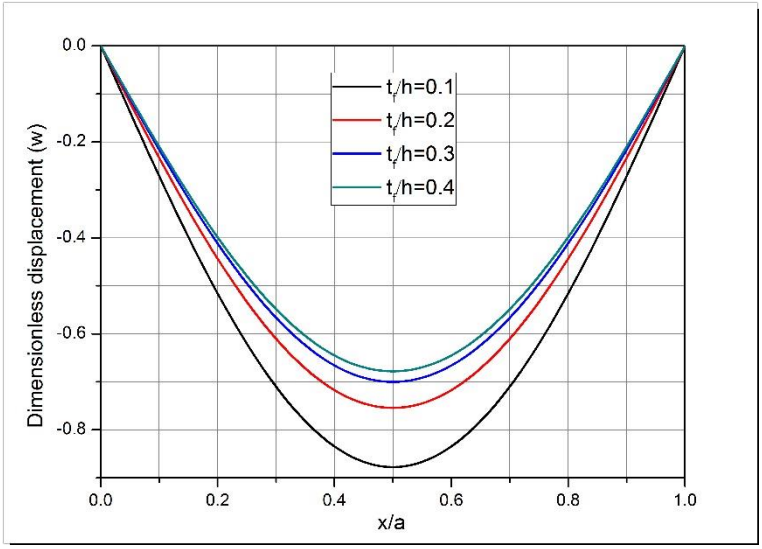


Fig. 8. Variation of the deflection

In the Fig.9, variation of the deflection parameters with respect to the relative density is illustrated. Depending on the increase in relative density, the maximum deflection of the plate decreases.

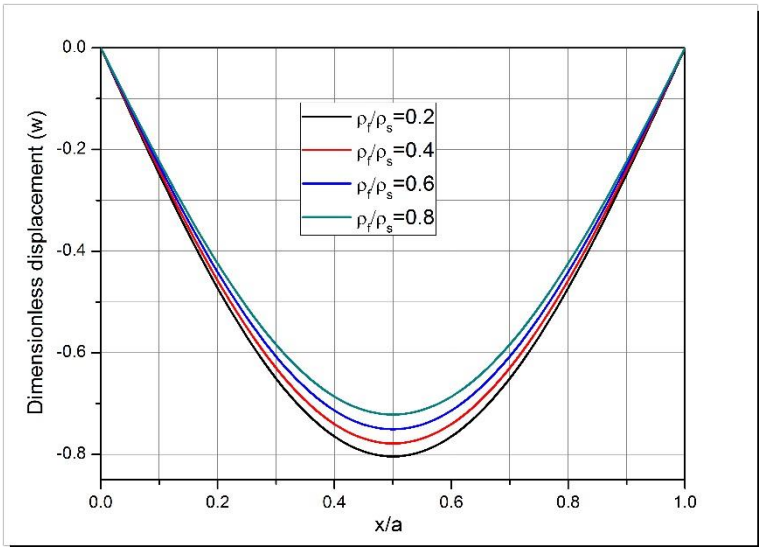


Fig. 9. Variation of the deflection with respect to the relative density

Fig. 10 shows normal stress of sandwich plate through thickness direction. Linear normal stress distribution is shown in face layers. Core stress is also linear but it shows small slopes.

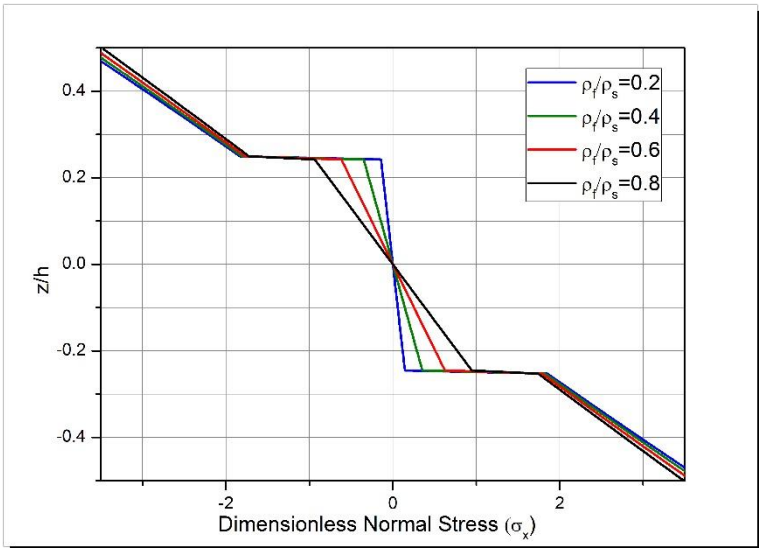


Fig. 10. Variation of the dimensionless normal stress with respect to the relative density

Fig. 11 shows shear stress of sandwich plate through thickness direction. Shear stress of face layers is become linear from parabolic with increase ρ_f / ρ_s ratio. Shear stress of core is decreases with density of foam. Also it shows nearly linear distribution with decreasing relative density.

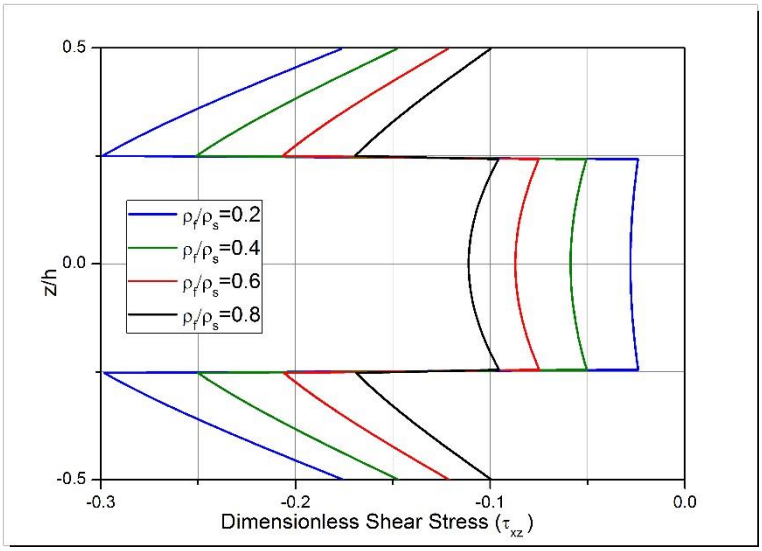


Fig. 11. Variation of the dimensionless shear stress with respect to the relative density

4. CONCLUSION

In this study bending of simply supported sandwich plate is studied. The effect of relative density on deflection is investigated. It is shown that the maximum deflection of the plate decreases with increasing relative density. Normal and shear stress distributions of face and core layers are presented. It's obtained that face layer normal stress is linear. Normal stress of core is linear too but slope of line is smaller. It's found that shear stress of face layers is become linear with increasing ρ_f/ρ_s ratio and shear stress of core is decrease with density of foam. In the future works, various boundary conditions and loadings can be applied.

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