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ARTICLE *in* COMPUTATIONAL MATERIALS SCIENCE · FEBRUARY 2011

Impact Factor: 2.13 · DOI: 10.1016/j.commatsci.2010.08.005

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Computational Materials Science 50 (2011) 1263–1268

Contents lists available at ScienceDirect

Computational Materials Science

journal homepage: www.elsevier.com/locate/commatsci

Dynamic behaviour of sandwich plates containing single/multiple debonding

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ARTICLE INFO

Article history:
Received 5 October 2009
Received in revised form 13 May 2010
Accepted 3 August 2010

Keywords:
Debonding
Free vibration analysis
Sandwich plates
Finite element method

ABSTRACT

Debonding is one of most severe defects associated with sandwich composite structures. The damage inflicted on debonding may lead to the significant degradation of the load carrying capacity of sandwich structures and affects their mechanical behaviours. Free vibration analyses of sandwich plates with honeycomb and PVC foam cores containing single/multiple debonding are carried out. Dynamic characteristics such as natural frequencies and corresponding mode shapes of sandwich plates with and without debonding are calculated using the commercial finite element code ABAQUS. Parametric studies over a wide range of size, location and number of debonding zones to examine the influence of these parameters on the overall dynamic behaviour of the damaged plates are performed. As a consequence, natural frequencies and mode shapes are being calculated as functions of these parameters that enable to draw conclusions concerning the debonding influence.

1. Introduction

The advanced properties of sandwich structures, widely-used in aerospace and aircraft industries, such as high stiffness and strength at relatively low weight can be significantly reduced due to a local separation between the face sheet and the core. This partial disbond in sandwich structures is similar to delamination in multilayered composites and is also known as core-to-face sheet debonding. Such damage is identified as a pre-failure mode of sandwich construction elements that determines integrity and safety of a whole construction [1,2].

Dynamic parameters of a construction such as natural frequencies, mode shapes, and damping parameter are often used for vibration-based monitoring techniques as a non-destructive method for detection and quantification of invisible defects within the construction. Cawley and Adams [3] successfully demonstrated this approach to define the location and severity of damage within carbon-fibre-reinforced plastic plates. A full review on vibration-based health monitoring techniques recently used can be found in [4]. Thereby, a dynamic analysis to examine debonding effects on dynamic responses of sandwich structures in applications for damage detection is an important engineering task. Recently reported studies on the dynamic behaviour of debonded sandwich structures in most cases are confined to one- and two-dimensional models of beams and plates. In contrast to the perfect bonded sandwich structures the interface assumption implying continuous displacements and traction across the interface is no longer valid for structures damaged by debonding. Ramkumar et al. [5] were first, who studied this problem for composite beams with delamination. They modelled a beam by the split model that comprises of four Timoshenko beams connected at delamination edges. Moreover, due to the debonding presence, debonded sandwich structures no longer remain balanced even though they were originally balanced and symmetrical. Wang et al. [6] proposed an analytical model of the unsymmetrical composite beam based on the classical beam theory including the effect of the coupling between flexural and longitudinal motion of delaminated layers. An analytical approach involving the split sandwich beam model with the bending–extension coupling to evaluate the effect of the extent of debonding on the flexural stiffness and on the natural frequencies compared with experimental observations was studied in [7]. Also, changes in forms of the frequency response functions due to debonding were investigated from the viewpoint of the debonding identification by using natural frequencies and damping ratios.

In general, the vibration of sandwich structures with debonding is accompanied by contact problems between the interfaces of the debonded parts. Several analytical and numerical models have been proposed to study the contact boundary problem. The local contact phenomena, denoted as clapping in debonded sandwich beams due to forced oscillations was studied in [8] by using both the finite element method and analytical approach. The impact-like contacts along the debonded surfaces were simulated with two clamped at the one edge separated parts with different cross-sections. Depending on the frequency of excitation, a various bifurcation scenarios were investigated. The dynamic analysis of a

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doi:10.1016/j.commatsci.2010.08.005

delaminated composite beam and debonded sandwich plate by employing a piecewise linear spring model inserted in the imperfect interfacial region were performed in [9,10], respectively. Finite element analysis (FEA) with the surface-to-surface contact model satisfying the certain contact–impact conditions was presented in [11], where the dynamic behaviour of debonded cantilever sandwich beams excited at an impact load at the free edge was examined. A numerical study of dynamic responses in delaminated composite beams was performed using ANSYS code in [12]. In the work the bilinear contact elements with the tension-only option were utilized for modelling contact delamination conditions. A high-order theoretical approach for the free vibration analysis of debonded unidirectional sandwich panels with a compressible core taking into account the conditions of with and without contact was presented in [13]. The progressive failure process of debonding during oscillations of a sandwich beam was considered by using Dugdale–Barenblatt cohesive zone model in [14]. A three-dimensional (3D) FE method to analyze the dynamic behaviour of delaminated composite plates was presented in [15]. Free vibration analysis of sandwich plates containing single debonding based on the 3D FE model and piecewise spring contact in the debonding zone was carried out in [16]. The sandwich plates with circular, rectangular and elliptic forms of debonding under various types of boundary conditions were investigated.

In this work, 3D FE modelling of free vibrations of sandwich plates with honeycomb and PVC foam cores containing *a priori* embedded single/multiple disbonds between the core and the face sheet is presented. The influence of size, location and number of debonded zones on dynamic characteristics of sandwich plates is investigated.

2. Finite element modelling

The finite element method is used to compute modal parameters such as natural frequencies and corresponding mode shapes for sandwich plates without or with debonding. A number of assumptions within the framework the FE simulation were adopted to analyze the debonding influence on oscillating sandwich plates. Firstly, debonding is modelled as an artificial flaw embedded into interface between the skin and the core of the sandwich plates. Secondly, debonding is assumed to exist before the vibration starts and to be constant during oscillations, i.e. the propagation of debonding was not allowed. Thirdly, the debonded surfaces are in contact vertically, the piecewise spring model is applied. Fourthly, the complicated debonding geometry is idealized by a regular circle form. And fifthly, underlying cores are assumed to be intact in all cases and the honeycomb core is treated as an orthotropic homogeneous material, whereas an isotropic material presences the foam cores.

A 3D FE model of a sandwich plate with circular interfacial disbond between the core and the face sheet was developed as it was to simulate as close as possible a physically real situation. No artificial adjustment of the material or geometry was made at the debonding location. The reduced integrated continuum shell elements (`SC8R`) and solid elements with incompatible mode (`3D8I`) were applied for discretizing the face sheets and the core, respectively. It is important to notice that the homogenization technique based on the unit cell conception [17] was used to treat the honeycomb core as a homogeneous orthotropic material with equivalent material constants and density. While, the cores made of PVC foams of various densities were considered as homogeneous isotropic materials with data given from the manufacturer [18].

The meshing of sandwich plates and the free vibration analysis were carried out using commercial software ABAQUS/Standard 6.6 [19]. A parametric input file was developed to enable a representation of various debonding configurations, thereby, the modal characteristics over a wide range of the debonding size, form and location were obtained. The general mesh of the FE model of the sandwich plate was subdivided into three different zones: the refine meshed debonded zone (I), the next zone (II) surrounding debonding with gradually decreased mesh density, and the coarse meshed fully bonded zone (III) was used to minimize a CPU time.

The connection between FE elements of the face sheet and the core in the fully bonded region of the sandwich plate was simulated by imposing multi-point constraints (MPC) in general nodes, which are denoted as single nodes. The lack of adhesion between the elements in the debonded interface was modelled by removing of those constrains and, as a consequence, the double nodes appear in this zone, Fig. 1a. The disbond was created by a real small gap (about 1% of the face sheet thickness) in the double nodes of the face sheet and the core. To prevent the debonded face sheet from overlapping with the core and to model opening and closing of the interfacial damage in the vibration state, the 3D spring elements `SPRING2` were introduced between the double nodes in the debonded area, Fig. 1a. The spring stiffness was taken as an arbitrary value between zero in the tension and very big its magnitude in compression when relative transverse displacement goes to zero, Fig. 1b. Because the influence of the friction properties on the dynamic responses is negligible [11], the contact friction was assumed as equal to zero.

3. Test calculations

For verification of the proposed finite element model with a damaged region, a sandwich beam cored by foam with rectangular cross-section containing damage at the middle span was considered (Fig. 2). Material properties of the sandwich beam were the same as described in [13] and they are listed in Table 1. The first six natural frequencies of this sandwich beam were analytically obtained in [13]. These data were compared to the numerical calculations found with ABAQUS's model discussed above. As can be seen in Table 2, the close results are demonstrated.

Convergence studies were carried out to obtain values of natural frequencies as accurately as possible at the minimum number of elements required with the view to optimize a computational cost. A perfect bonded rectangular sandwich plate used for this

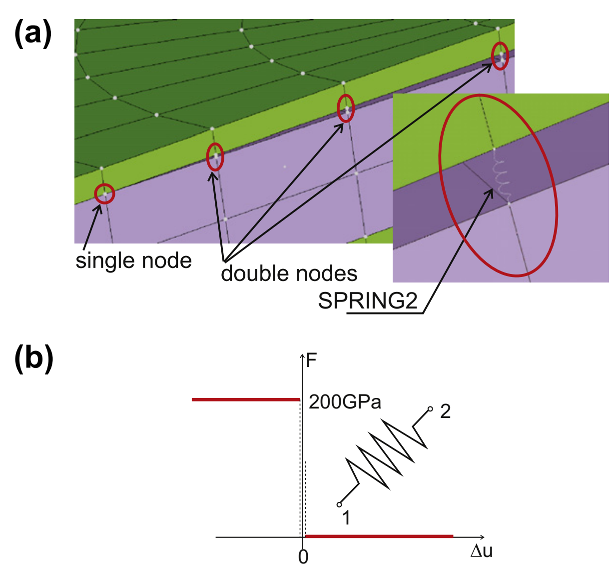

Fig. 1. Details of the finite element mesh: (a) meshing in the debonded zone and (b) behaviour of the spring element.

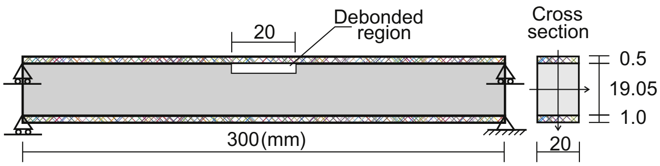

Fig. 2. Model of the damaged sandwich beam with foam core.

Table 1
Material properties of the foam cored sandwich beam.

Components	Elastic constants
Core	E_c = 50 MPa, G_c = 21 MPa, ρ_c = 52 kg m^{-3}
Face sheet	E_f = 36 GPa, ρ_f = 4400 kg m^{-3}

Table 2
Natural frequencies (Hz) of the intact and debonded foam cored sandwich beam.

Mode	Intact		Damaged	
	[13]	FEA	[13]	FEA
1	289.3	293.46	288.98	293.07
2	683.3	707.09	388.32	433.67
3	1096.9	1106.7	1093.2	1093.2
4	1151.6	1495.8	1146.9	1132.0
5	1778.2	1818.7	1771.3	1769.9
6	1895.3	1918.1	1842.2	2080.2

purpose consisted of unidirectional CFRP face sheet material and aluminum honeycomb core. The fibre orientation angle of CFRP was zero degree to the longitudinal axis. The dimensions of the sandwich plate are pointed out in Fig. 3.

The material properties used for numerical simulation are shown in Table 3. For the free vibration analysis perfectly free boundary conditions were imposed around of all plate edges. The experimental data of the first torsional and bending modes of this sandwich plate were investigated in [20] and, thereby, were used to compare against the numerical calculations performed with ABAQUS. The FE mesh of the plate finally accepted due to convergence studies contained 2500 elements and 15,000 nodes. Comparisons between numerical and experimental results for first torsional and bending eigenfrequencies are shown in Table 4. Good agreements indicating the successful modelling of the eigenvalue problem can be seen.

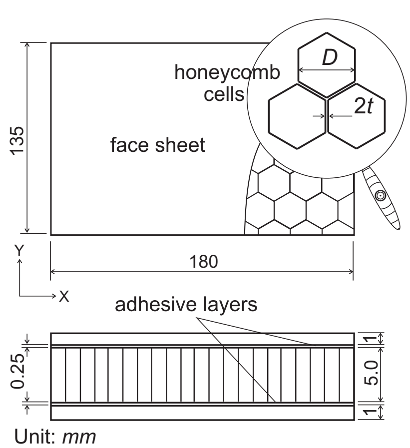

Fig. 3. Honeycomb sandwich plate model.

Table 3
Material properties of the honeycomb sandwich plate.

Components	Elastic constants
Honeycomb core	E_{11} = 0.461 MPa, E_{22} = 0.461 MPa, E_{33} = 1.494 GPa, G_{12} = 0.194 MPa, G_{13} = 341.7 MPa, G_{23} = 192.1 MPa, ρ = 57.17 kg m^{-3}
CFRP face sheet	E_{11} = 140 GPa, E_{22} = E_{33} = 10 GPa, G_{12} = G_{13} = 4.6 GPa, G_{23} = 3.8 GPa, ρ = 1650 kg m^{-3}
Adhesive resin	E = 1.5 GPa, G = 0.5 GPa, ρ = 141.3 kg m^{-3}

Table 4
Comparison between experimental and numerical results.

First torsional mode (Hz)			First bending mode (Hz)		
FEA	[20]	Error (%)	FEA	[20]	Error (%)
637.7	603.4	5.7	1300.1	1212.8	7.9

4. Numerical results

Effects of debonding on vibration responses of damaged sandwich plates were assessed by comparing numerical results of free vibration analysis between intact plates and plates containing single/multiple debonding.

4.1. Effects of single debonding presence

To investigate the effects of debonding on natural frequencies and mode shapes, sandwich plates with single circular debonding located at the center were considered. The material properties, plate geometry and boundary conditions were the same as adopted in the previous convergence study. Generally, the numerical results demonstrated that the interfacial local damage in sandwich plates leads to the significant shifts of the natural frequencies and changes of the mode shapes with respect to intact plates. In most cases the frequencies of debonded plates decrease due to loss in stiffness caused by initial debonding, and the mode shapes contain along with a global shape and a local deformation in the discontinuity region. And the influence of debonding was becoming more visible with increase in the debonding size. Besides, the frequencies change more rapidly as mode number increases. Although this trend in changing of frequencies was violated due to local thickening phenomenon caused by debonding, that for some modes made the frequencies of the damaged plates even higher than intact ones.

Moreover, the investigations showed that decreasing the natural frequencies with increasing the debonding size is greatly dependent on boundary conditions. Four kinds of constrains for the honeycomb plate were considered, namely, F–F–F–F (all free), S–S–S–S (all simply supported), S–F–S–F (two free and two supported) and C–C–C–C (all clamped). It was found that the more strongly the plate is restrained, the greater this effect on its frequencies. Also the results of calculations revealed that core types of simply supported sandwich plates strongly affect their dynamic responses. It was seemed that more sensitive to the debonding presence were the stiffer honeycomb sandwich plates than the plates made of light PVC H 100 and H 200 foams, data for which are provided by the manufacturer [18]. The studies of the effect of debonding shapes demonstrated that the various shapes as a circle, rectangle, ellipse and through-the-width debonding influenced in different way the different vibration modes.

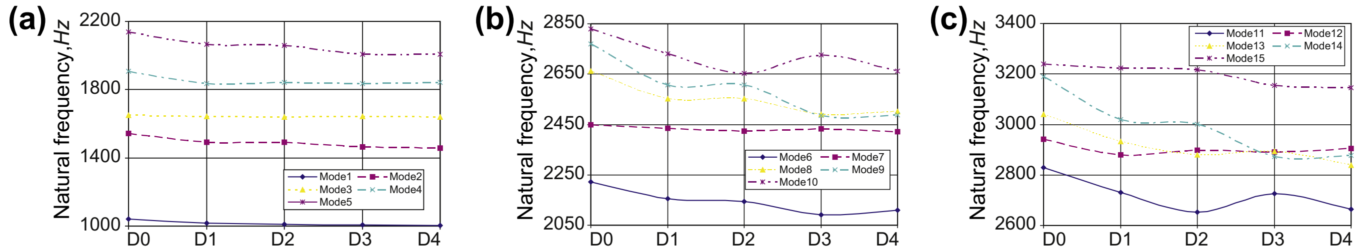

Fig. 4. Natural frequencies of intact and single debonded plates with different debonding location: (a) the first five frequencies; (b) the second five frequencies; and (c) the third five frequencies.

4.2. Effect of single debonding location

Next, numerical investigations were performed to study the effect debonding location. The simply supported rectangular sandwich plate cored by the PVC H 100 foam with dimensions 270 mm length and 185 mm width was considered. The face sheets and the core thicknesses were 2.4 mm and 50 mm, respectively. The material properties were the same as presented in Table 1. The plate had single circular debonding zone of radius 29 mm that corresponded to the planar damage parameter $D_{\%} = 10\%$, denoting a ratio of the debonding area to the total area of the sandwich plate. The debonding location within the plate area was assumed at four different places such as a center (D1), near longitudinal (D2) and transverse (D3) edges and at the corner (D4). The symbol D0 refers to the plate without debonding. From Fig. 4a it can be seen that the first five natural frequencies of the debonded plates are more shifted from the intact one when debonding is situated near the corner and less for centrally located debonding. The locations of debonding near the edges are intermediate cases between the mentioned above positions. Although the location of debonding at the transverse edge more affects the frequencies. The higher modes follow the similar patterns of changing of the frequencies with some exceptions for the several high modes, Fig. 4b and c.

4.3. Effect of number of debonding zones

The effects of multiple debonding can be examined using the same finite element model that was developed for studying sandwich plates with single debonding. Thus, sandwich plates with two equal sized circular debonding zones were considered. The material properties and plate geometry were the same as in the previous calculations of the simply supported foam cored sandwich plate. The two debonding zones were symmetrically located with respect to in-plane midlines of the plate and it was assumed that debonding occupied the following four positions: symmetrically located on the transverse (DD1) and longitudinal (DD2) midlines and symmetrically located at the each corner side, transversely (DD3) and longitudinally (DD4). The damage inflicted these two debonding was accepted the same magnitude as in the previous case with single debonding zone, i.e. it was equal to 10%. The comparative results of frequencies for the intact plate and plate with two debonding are presented in Fig. 5. In general, the natural frequencies highly depend on the location of these two debonding within the plate area and the appropriate mode shapes, as well. As seen from Fig. 5 the two debonding, located along the longitudinal midline (DD2), the most affects the frequencies of the plate. The other cases of the debonding location influence the frequencies in different ways and in a lesser degree. In general, this tendency is retained for all modes from the low to high ones.

Simultaneously with the study of the two debonding locations, it was possible to evaluate the contribution of more than one debonding zone on the natural frequencies and mode shapes. For this aim the cases of one and two debonding zones, inflicted the same damage ($D_{\%} = 10\%$) and considered in the previous calculations, were compared. Natural frequency shifts versus mode numbers for the debonded plates with one and two debonding zones located in different ways are presented in Fig. 6. It one can see that the frequency shifts are more affected by single debonding than the equally sized symmetrically located two debonding zones. Although Fig. 6a shows, that most of modes do not follow the made conclusion. This signifies the importance of the multiple debonding locations within a sandwich plate which affect some of modes more than others.

By comparing the cases D4 for a single debonding with damage of 10% and DD3 and DD4 for two equal sized debonding with total damage 10% and 20% it was found that the frequency shifts are more pronounced for the case with two debonding zones inflicting damage 20% for all cases of debonding locations, Fig 7.

Finally, the numerical calculations were carried out to study the natural frequency shifts for plates with two non-equal sized debonding located symmetrically with respect to midlines of the plates (the cases of DD1–DD4). The trends in natural frequencies seem to be similar to the cases for the two equal sized debonding. The same conclusion can be drawn for the frequency shifts of plates with two equal sized debonding located non-symmetrically with respect to midlines of the plates. However, in theses all cases

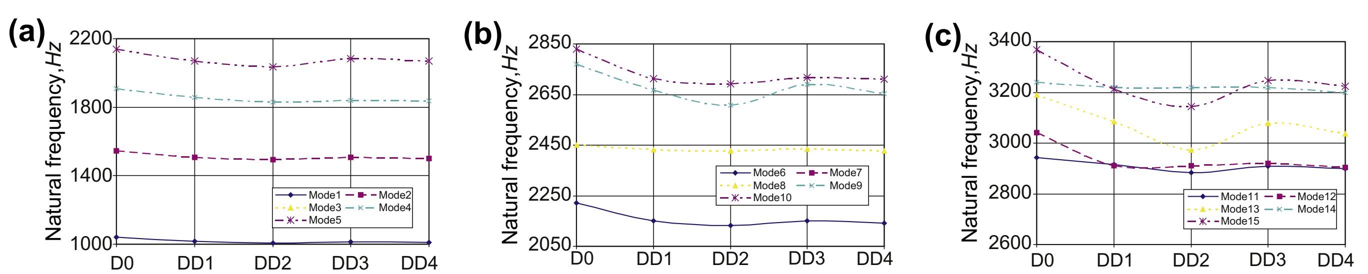

Fig. 5. Natural frequencies of intact and double debonded plates with different debonding location: (a) the first five frequencies; (b) the second five frequencies; and (c) the third five frequencies.

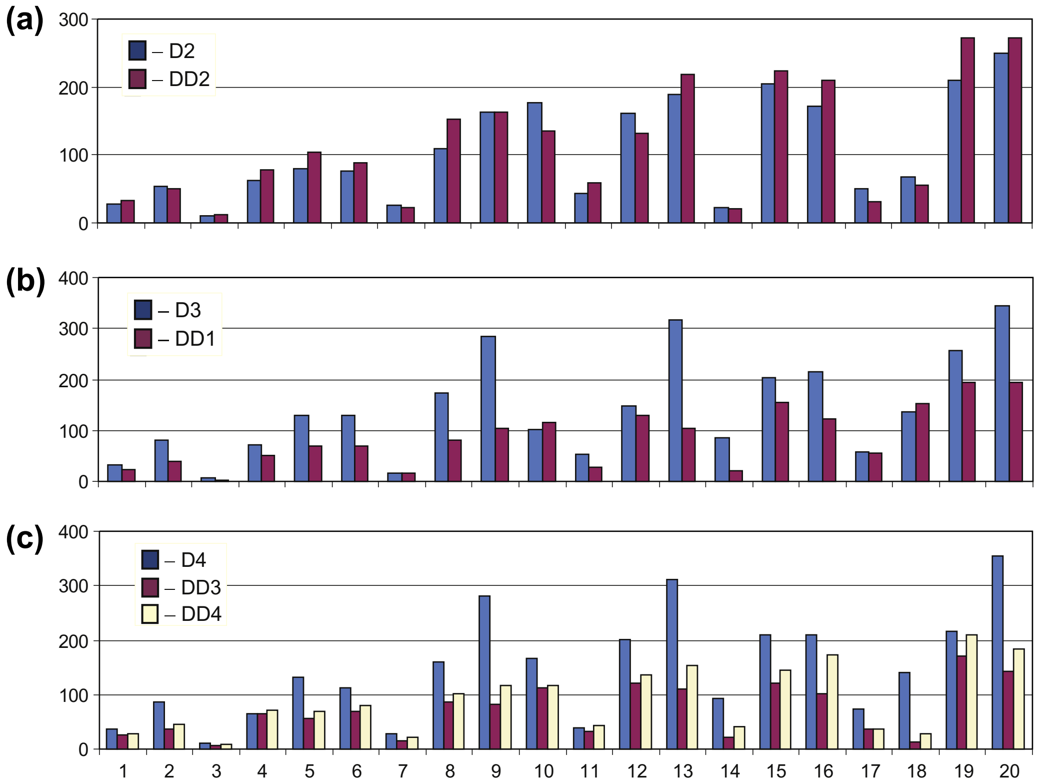

Fig. 6. Natural frequency shifts of debonded plates with a single and two debonding zones located in different ways: (a) near the longitudinal edge and symmetrically with respect to the longitudinal midline; (b) near the transverse edge and symmetrically with respect to the transverse midline; and (c) at the corner and symmetrically with respect to the midlines.

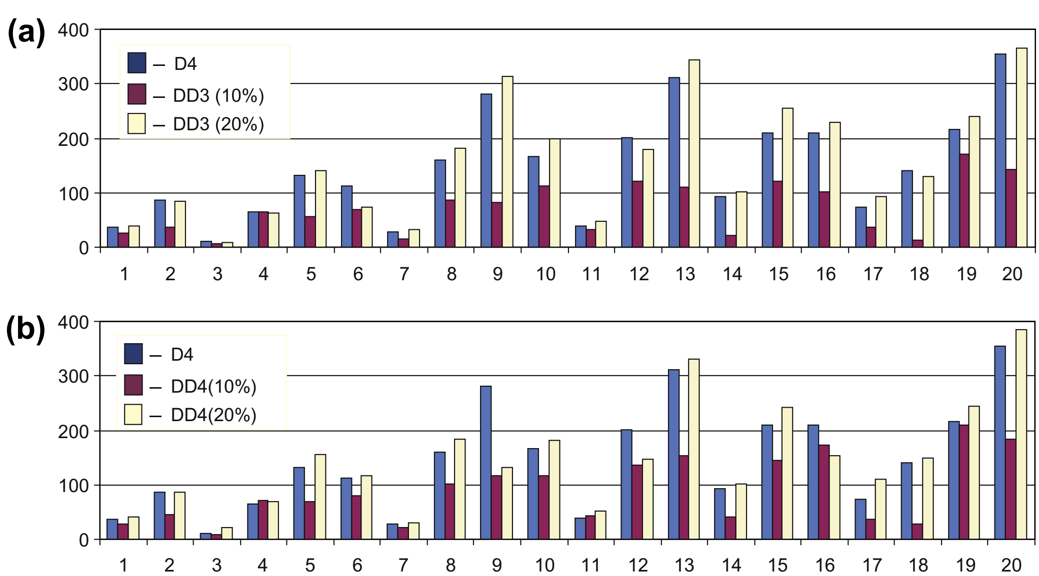

Fig. 7. Natural frequency shifts of debonded plates with a single and two debonding zones of different sizes and locations: (a) at the corner edge and symmetrically at the each corner side in a longitudinal direction and (b) at the corner edge and symmetrically at the each corner side in a transverse direction.

the debonding existence more easily can be observed from the mode shapes of sandwich plates containing the non-equal sized and non-symmetrically located multiple debonding zones.

5. Conclusions

In summary the following conclusions from the viewpoint of sensitivity of dynamic characteristics to the presence of debonding can be drawn. Firstly, both changes in natural frequencies and mode shapes are sensitive to the debonding presence within sandwich plates. Natural frequencies of debonded plates usually decrease with debonding increasing. Secondly, the higher natural frequencies and mode shapes are more sensitive to the debonding presence. Thirdly, the sensitivity of natural frequencies and mode shapes are affected by boundary conditions, core type and debonding locations. Fourthly, the presence of two debonding zones changes natural frequencies and mode shapes of sandwich plates comparing with the single debonding existence. These changes mostly depend on the location of the two debonding zones within the plates than of their quantity. Thus, by comparisons between effects for a single and double debonding it is shown that the natural frequencies are poorly sensitive to the number of the debonding zones.

Acknowledgement

This research was supported by the European Union within the 7th Framework Programme, call FP7-REGPOT-2009-1, Grant Agreement No. FP-7 24547.

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