DYNAMICAL JOINTS INFLUENCE OF SANDWICH PLATES

Francesco Petrone, Francesca Garescì, Michele Lacagnina and Rosario Sinatra

Abstract
The simulation of joints in structures of sandwich panels is very important in order to obtain values of the analytical natural frequencies and mode shapes that are in accordance with the experimental values.

By means of a FEM program and an experimental set for modal analysis, different assemblies of sandwich plates in aluminium, fit together with corner joints, are tested.

The FEM simulation of the corner joints is made in different ways and the comparison between analytical and experimental results shows that different methodologies of simulation can be applied.

Interesting results on the influence of the element set forming the joint are obtained.

Moreover the non-linear behaviour of the structures due to the thickness of the adhesive in the joint is analysed.

1 Introduction

To predict the dynamic behaviour of box-frames made of sandwich panels, it is important to determinate the natural frequencies and mode shapes.

The study can be made by means of FEM codes that are capable of predicting with good accuracy the dynamical parameters of almost all practical components, no matter how complex their form [1]. However, when such components are assembled by means of joints to form a typical box-frame, the same component models yield a greatly reduced quality of prediction [2].

The complex mechanical characteristics of sandwich plates (anisotropy, non-homogeneity, and non-linear effects due to the core and the adhesive between different layers) involves greater difficulty in the analytical simulation of the dynamical behaviour of the structure.

The simulation of jointed zones, where different materials are present, can be made with good accuracy, but it is not always possible to generalise the results. Therefore the complexity of analytical models is not often justified by the accuracy of results.

The aim of this work is to check, by means of comparison with experimental tests, the possibility to make simplified analytical simulations to use in the dynamical analysis of jointed structures when the complexity of the analytical model can cause numerical and calculation time problems.

We determinate the natural frequencies and mode shapes of two sandwich plates assembled with corner joint in 4 different manners by using an experimental set for modal analysis.

The experimental data are compared with analytical results obtained by using a FE code for modal analysis. Four different FE models are employed to simulate the plates and the joint.

In order to avoid the uncertainly in the constraints realization a free-hanging configuration has been employed.

2 Material and joints description

The structural joints can be made with different methods using adhesive and angular profiles of various materials. The joints used to connect the various components introduce a degree of influence which is often omitted by the coupled structure analysis methods used.

The sandwich panel fabrication technology, as known, provides various types of joints such as:

flat joints, corner joints and T joints. The tests included in this paper relate to plates assembled together by means of corner joints.

Various methods are used to make the corner joints (Fig. 1). In this work the supported by bonded L-section extrusions method has been employed.

The tests were carried out on a simple structure of two jointed sandwich plates with aluminium skins 0.56 mm thick and a core of an aluminium hexagonal honeycomb cell 25.4 mm thick.

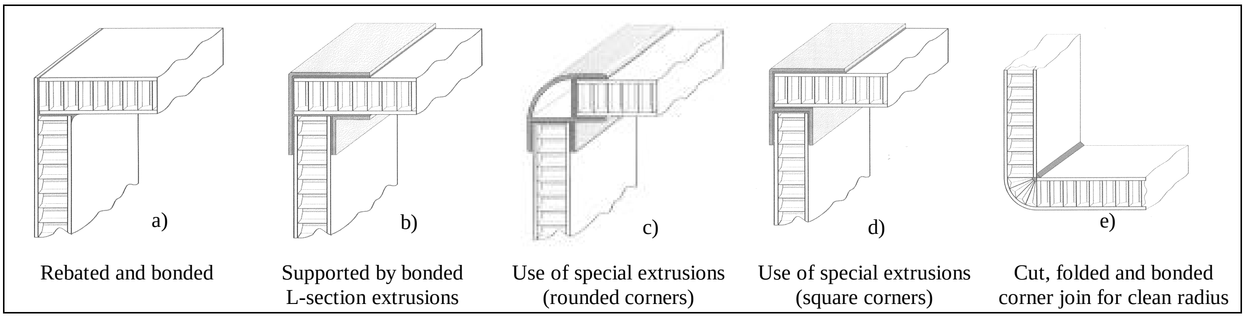


Fig. 1 - Typical corner joints

The corner joints were made using 3 different kinds of bonding aluminium angular profiles and two thickness of adhesive (Table 1) checked by means of steel wires (Fig. 2).

Shape and dimensions are the same for all specimens. They are made with 2 panels whose dimensions are 400x300 mm and 175x300 mm.

Type	Internal profile	External profile	Adhesive thickness
#1	L 20x20	L 20x40	1 mm
#2	L 30x30	L 30x60	1 mm
#3	L 30x30	L 30x60	3 mm
#4	L 50x50	L 50x50	1 mm

Tab.1 - Dimensions of tested joints

The builder gives the core mass density as equal to 83 kg/m^3 with a $\pm$ 10% tolerance and the values of shear modulus as equal to 4.4E8 N/m^2 and 2.95E8 N/m^2 respectively in the longitudinal and transverse direction.

The aluminium of the skins has the following mechanical characteristics:

- Young modulus: E = 6.9E10 N/m^2
- shear modulus: G = 2.62E9 N/m^2
- mass density: ρ = 3180 kg/m^3

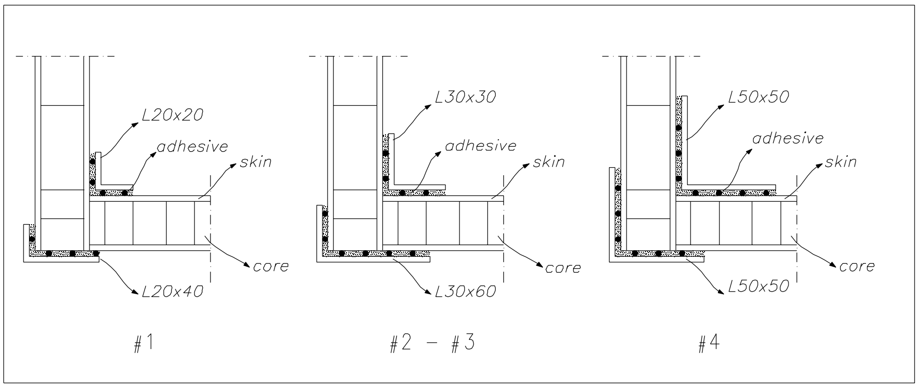


Fig. 2 - Configuration of tested joints

3 Experimental methodology

Dynamical analysis consists in the investigation of the frequencies and mode shapes of the structure using modal analysis technique. The results of different analytical simulations and experimental tests are compared.

A MIMO technique has been used by exciting the panels in different mesh-points using a hammer equipped with a force transducer. The transient signals of the accelerometers were acquired in the time domain with rectangular weighting. The length of the acquisition window has been selected in order to the signal has effectively died away to zero by the end of the record.

The time record was converted in the frequency domain by means of a 4 channel FFT analyser, with a software to obtain the FRF; this technique can be expressed as the sum of the responses of n single degree-of-freedom system [3]:

$$\mathrm{FRF} = \Sigma\, H_{ij}(w) = \Sigma\, x_{ij}/\mathbf{F}_j$$

The calculation of the eigenvectors in the frequency domain comes to the minimization of the difference between the measured FRF and a combination of some FRF related to a sDOF system.

The experimental mesh is composed by 21 nodes, 12 located on the horizontal panel and 9 on the vertical one (Fig. 3).

In order to avoid recording double modes or placing the accelerometer on a nodal point two accelerometers were used.

To verify the accuracy of experimental results various factors characterising the modal analysis have been taken into account: the geometrical correlation between two different modal deformations, the relationship between the real and imaginary part of modal deformation, the statistical variation of the phase angle in each eigenvector of average value, that in the real mode shape should be close to 0%.

The constraint used in the tests was free-free in order to avoid the influence of the constrain realization which can introduce some factor of uncertainty as observed by other AA. [4]. In every case the FRF was recorded in a single test discarding the signals that were unclear.

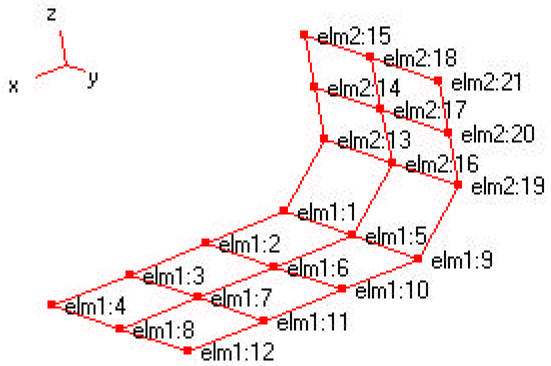


Fig. 3 - Experimental mesh

4 Numerical simulation

A pre and post-processor (FEMAP) and a finite element code (NASTRAN), were used for the FEM simulation.

In the finite element code analysis, the core simulation with two different shear moduli was fundamental to obtain the exact correspondence of vibration mode shapes at various frequencies; contrary to what literature shows in studies on static analysis on sandwich panels, whose cores are simulated with an average value of shear modulus G, in the present study the core anisotropy was not neglected to confirm the analytical mode shapes.

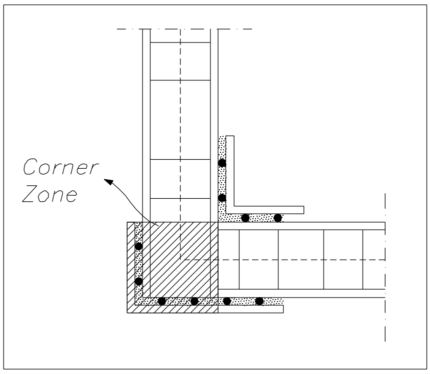


Fig. 4 - Corner zone of the joint

We used different simulations of the sandwich panels and of the jointed zone using bi-dimensional FE referred to the medium surface or volume elements to simulate the real geometry. In the plane simulations the resistance sections are considered like a laminate composed by different layers:

- aluminium angular profile 0.002 m thick
- adhesive 0.001 or 0.003 m thick
- skins 0.00056 m thick
- core 0.0254 m thick.

Four simulations #A, #B, #C, #D are made using bi-dimensional laminate FE for plates and joints; they differ in the modelling of the joints.

The third simulation (#C) is different from the previous because the adhesive is considered by adding other layers.

In the fourth simulation (#D) the corner zone is simulated by the 3 layers sandwich adding, on the external surface, the adhesive and the angular profile, without considering the real discontinuity.

Finally, the last simulation (#E) is made with volume elements for all materials to reproduce the real geometry (Fig. 5).

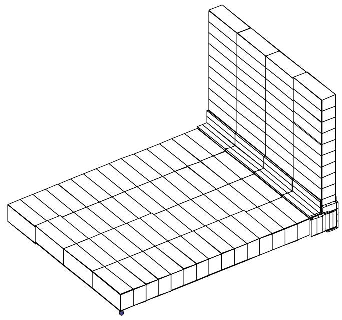


Fig. 5 - Analytical mesh using 3D FE

5 Results

The results of modal analysis obtained with the 5 different simulations and from the experimental tests are compared in the first four natural frequencies (Table 2).

The pattern of the percent deviation from analytical and experimental values (Fig. 6) points out an almost analogous behaviour of the simulations #B and #C, as predictable, since only the adhesive layers differentiate the FE used in the pattern of plates and corner joint.

In all simulations the best convergence is found in the values of the 2^{nd} natural frequency (torsional mode).

In all analytical simulations an inversion between the third and fourth mode shape with respect to experimental tests was detected (Fig. 7).

The percent deviations for the #3 joint (3 mm of adhesive) are slightly higher than the ones of the #2 joint (1 mm of adhesive). It is probably due to the non-linear behaviour of the stiffness of the joint caused by the variation in the thickness of the adhesive.

With the more accurate simulation made with volume elements (#E) the best results are reached when the joint is assembled by using angular profiles which guarantee the correct overlapping of the joint zone (specimen #2 and #3). When using profiles which are either too small (#1) or badly arranged (#4), no advantage in the use of more complex model is recorded.

The simplest model (#A) without joint schematization, shows a good accuracy on average.

The results of this work lead to the rejection of the #D simulation that appears as the most intuitive to simulate technically the presence of the adhesive and the angular profiles in the joint.

Joints Type	Modes	Analytical Frequencies - Percent Deviation										Experimental Frequencies
		#A		#B		#C		#D		#E		
		Hz	%	Hz	%	Hz	%	Hz	%	Hz	%	Hz
#1	1°	462	18,65	399	2,47	405	4,01	494	26,87	459	17,88	389
	2°	544	5,98	538	4,81	537	4,62	543	5,79	535	4,23	513
	3°	939	8,38	915	5,61	920	6,19	992	14,50	1023	18,08	866
	4°	1052	7,54	977	-0,13	976	-0,23	1101	12,55	1044	6,72	978
#2	1°	462	-3,60	402	-16,12	403	-15,91	513	7,04	478	-0,26	479
	2°	544	-0,97	518	-5,70	517	-5,88	545	-0,79	549	-0,06	549
	3°	939	-10,57	905	-13,81	903	-14,00	1010	-3,81	1081	2,95	1050
	4°	1052	-3,49	917	-15,87	920	-15,60	1140	4,59	1102	1,10	1090
#3	1°	462	-2,59	402	-15,24	404	-14,82	513	8,16	478	0,78	474
	2°	544	-0,32	518	-5,09	516	-5,45	545	-0,14	539	-1,24	546
	3°	939	-9,71	905	-12,98	900	-13,46	1010	-2,88	1088	4,62	1040
	4°	1052	-2,59	917	-15,09	924	-14,44	1140	5,56	1113	3,06	1080
#4	1°	462	7,34	418	-2,88	416	-3,34	523	21,52	490	13,85	430
	2°	544	3,43	548	4,19	549	4,38	558	6,09	559	6,28	526
	3°	939	1,06	951	2,35	948	2,03	1040	11,93	1107	19,14	929
	4°	1052	5,20	974	-2,60	977	-2,30	1147	14,70	1138	13,80	1000

Tab. 2 – Natural frequencies and percent deviation from experimental values

6 Conclusions

Using a system of acquisition and analysis of signals for the modal analysis, the natural frequencies and mode shapes of the jointed sandwich are pointed out.

The experimental values were compared to the analytical results carried out by a FE commercial calculation code. Different simulations of the joints, using plane elements and volume elements, were carried out to obtain a better simulation, introducing or not the adhesive layer, or neglecting the presence of the angular profiles in the joint zone.

The comparison shows that a more accurate simulation, with greater complexity of analytical model and numerical problems, does not always produce more accurate results.

Certainly stiffer joints are better simulated with volume elements. When the manufacturing is not very accurate and the joints are more flexible, a simplified simulation produces good results with less numerical problems.

The effect of the adhesive thickness is better simulated with volume elements. Moreover, when the adhesive thickness increases a non linear behaviour of the joints occurs.

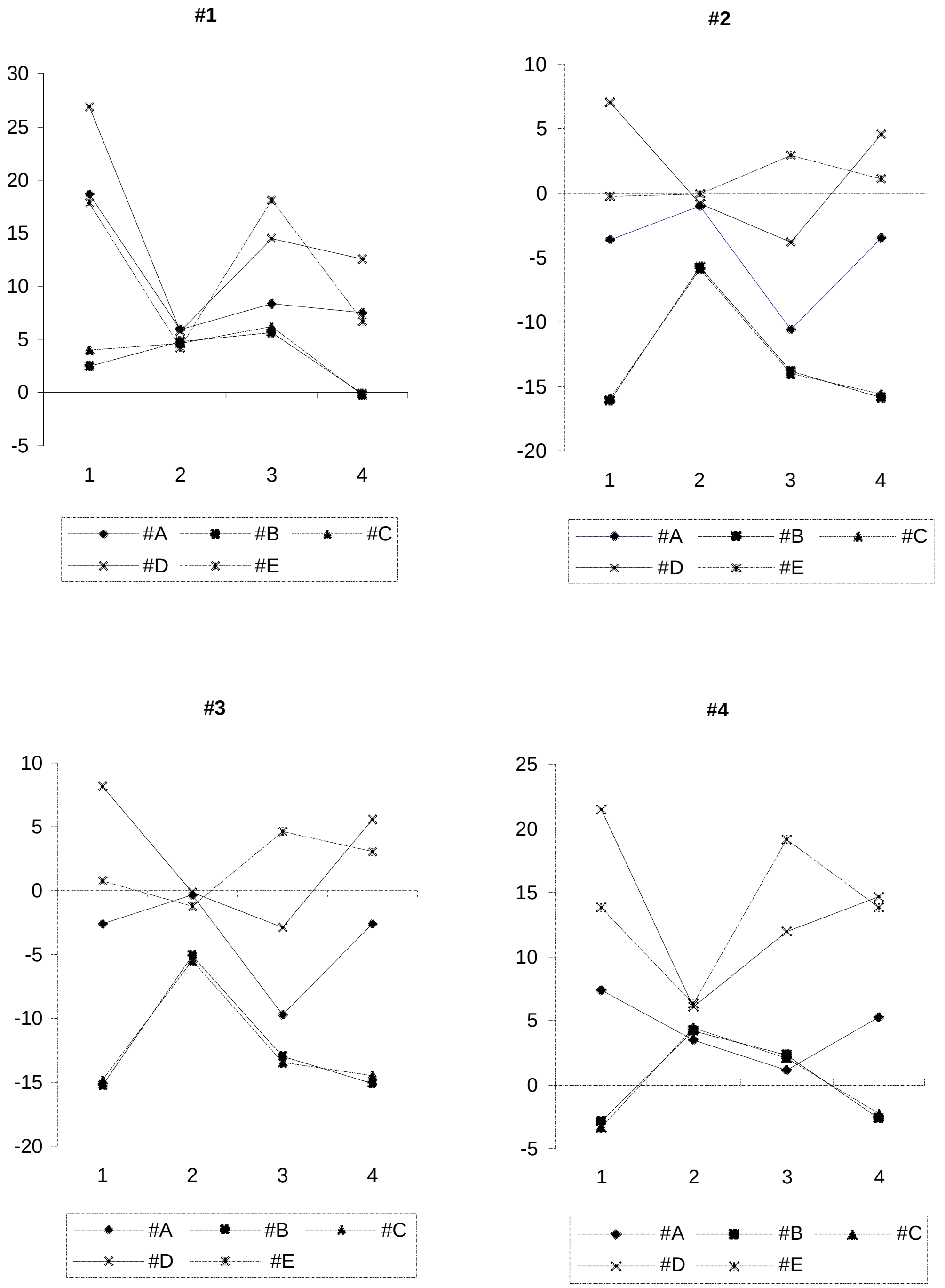


Fig. 6 - Percent deviation of analytical values from experimental

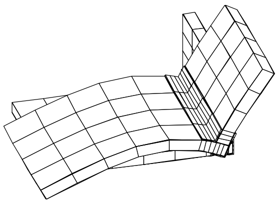
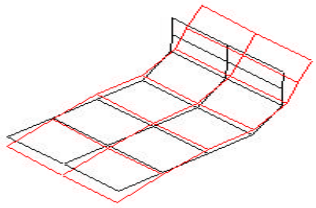
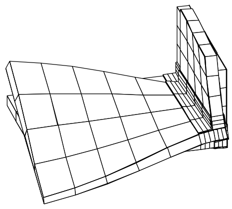
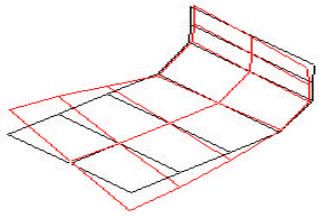
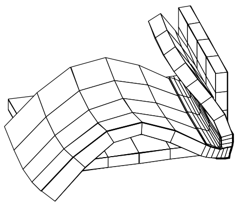
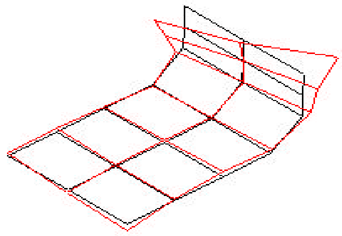
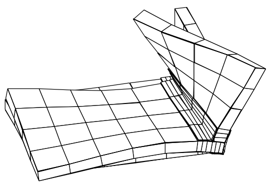
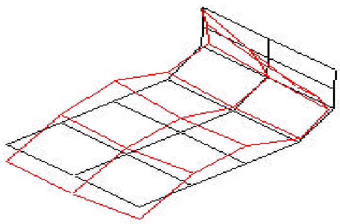
Modes	**FEM Simulation**	**Experimental**
#2 1°	478 Hz 	479,24 Hz 
#2 2°	549 Hz 	549,32 Hz 
#2 3°	1081 Hz 	1050 Hz 
#2 4°	1102 Hz 	1090 Hz 

Fig. 7 - Comparison of FE simulation and experimental natural frequencies and mode shapes of the specimen #2

References

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Francesco Petrone
Istituto di Macchine
Università di Catania
v.le Andrea Doria 6
95125 Catania Italy
fpetrone@im.ing.unict.it
www.cdc.unict.it

Francesca Garescì
Dipartimento di Meccanica
Politecnico di Milano
p.zza L. da Vinci 32
30133 Milano Italy
fgaresci@im.ing.unict.it
www.cdc.unict.it

Michele Lacagnina
Istituto di Macchine
Università di Catania
v.le Andrea Doria 6
95125 Catania Italy
mlacagnina@im.ing.unict.it
www.cdc.unict.it

Rosario Sinatra
Facoltà di Ingegneria
Università di Messina
Via Sperone 31
98166 Messina Italy
rsinatra@im.ing.unict.it
www.cdc.unict.it