Model Verification of Printed Circuit Boards for Environmental Vibration Testing in Mounted Condition

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Abstract

This work adopts finite element analysis (FEA) and experimental modal analysis (EMA) to perform model verification for the printed circuit Boards (PCB) that is specially designed for environmental vibration test and mounted to the fixture. For the purpose to quantify the designed PCB, a suitable and reliable mathematical model for PCB is desired to predict and study its dynamic response. More importantly, such for chip failure or any fault condition during testing can be identified in advance and compared to actual tests. The equivalent FE model of PCB in free boundary has been successfully constructed and verified. This work will consider the practical mounted condition of the PCB in testing. Both FEA and EMA are performed, respectively, to determine the modal properties of the PCB. Via the optimization procedure to verify modal properties obtained from FEA with those from EMA, the equivalent FE model for the PCB in mounting can be validated. Results show both FEA and EMA agree reasonably well. In particular, modal characteristics of the PCB can be identified and beneficial to design consideration. The developed finite element model can also be used for response prediction, model modification or sub-structuring analysis of the PCB.

Key words: finite element analysis, experimental modal analysis, printed circuit boards.

1. Introduction

The design of PCB is a quite important issue for its miniature and resulting in thermal effect. Dynamic failure due to environmental factors such as transportation can affect the PCB life cycle; therefore, dynamic analysis in the design of PCB is crucial.

FEA is frequently adopted to construct the FE model of the structure to perform theoretical analysis and design. The PCB consists of glass fiber board and chips. The overall material mechanical properties can vary for different layout of chips on boards. To construct an effective FE model is difficult. EMA can practically determine the structure modal parameters, including natural frequencies, mode shapes and damping ratios, for the structure without the pre-known material properties. However, the flexibility in experiments to study the change in physical properties such as dimension or layout is limited. Both FEA and EMA can be adopted and combined to adjust the limitations due to each others. By the comparison of modal properties obtained from FEA and EMA, the FE model can be easily modified to fit the experimental results, and so forth the equivalent FE model to the practical structure can be validated and helpful to further dynamic design analysis.

Yang *et al.* [1] studied the vibration characteristic of PCB due to the effect of a Plastic Ball Grid Array (PBGA) PCB assembly. They adopted EMA and FEA to construct the equivalent FE model and analyzed the fatigue failure of solders of the PBGA module in vibration test. Pitarresi *et al.* [2] used FEA to analyze the main board response of a PC due to mechanical shock and random vibration excitation and validate the equivalent FE model. Both works show good examples in using FEA and EMA for PCB design.

PCB is a compound material structure which analysis technique is required. Gibson [3] performed EMA on the compound material PCB. The quasi-linear approach can well interpret the nonlinear properties of PCB from experiments. Low *et al*. [4] studied the vibration characteristics of square plates in six different boundary conditions and different types of mass distribution loading. Ma and Hung [5] utilized Amplitude Fluctuation Electronic Speckle Pattern Interferometry (AF-ESPI) to investigate the vibration of isotropic square plates in four kinds of boundary conditions. The effect of boundary

conditions must be considered for design of the PCB vibration analysis. He and Fulton [6] theoretically studied the vibration response of non-linear PCB in simply-supported boundary due to free and forced vibration loads. They presented the theoretical analysis of layered plate vibration in linear and non-linear assumptions and verified the theoretical results with the experiments.

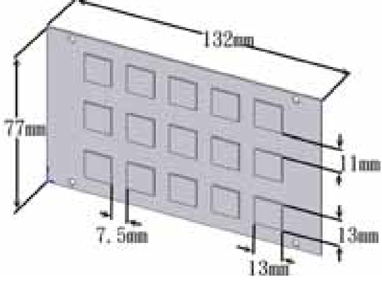
Yang *et al.* [7] carried out EMA for the PCB with PBGA chips in different boundary conditions and showed that many factors can affect the test results, such as the transducer loading, the planning of measurement points and the fixed force applied on PCB.

This work assumes that the PCB material is isotropic. The FE model of PCB is first constructed to theoretically determine vibration modal parameters, and the EMA is also performed to obtain the experimental ones. By the adoption of optimization procedure supported by ANSYS software, the material parameters can be optimally fitted such that the equivalent FE model can be validated and applicable to further design analysis, such as the failure evaluation of the integrated or parts of PCB.

2. Problem Description and Analysis Objective

The PCB in fixed boundary and its dimension are shown in Figure 1. The chips on the test PCB are constructed with lead free solder and designed specifically for the use in dynamic testing according to JEDEC [8].

(a) **Fixed** boundary PCB



(b) Dimension of PCB

Figure 1. Fixed boundary PCB and dimension

To build an effective and equivalent theoretical model for the PCB can be beneficial to the response prediction during vibration test and lead to failure analysis and chip design for the integrated PCB or parts. The objectives of this work are:

(1) To apply ANSYS software to construct the FE model for the PCB and perform both modal and harmonic response analysis, so as to obtain theoretical natural frequencies, mode shapes and frequency response functions (FRFs).

(2) To perform EMA on the PCB to get the experimental FRFs and modal parameters, including natural frequencies, mode shapes and damping ratios by curve-fitting process.

(3) To conduct the optimization procedure to validate the FE model of the fixed PCB and get the equivalent theoretical model.

The conceptual idea of model verification is shown in Figure 2. Both FEA and EMA are, respectively, carried out to obtain the theoretical and experimental modal parameters. Based on the experimental results, the FE model parameters are optimally determined via ANSYS optimization module, so as to obtain the equivalent FE model to the realistic PCB in fixed boundary.

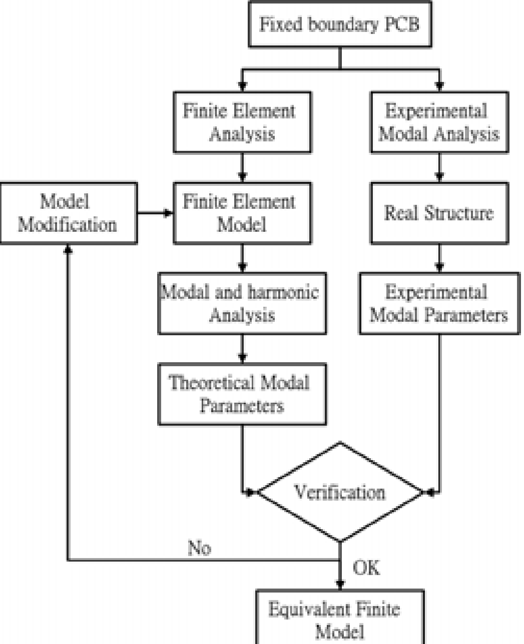


Figure 2. Conceptual idea of model verification

3. Finite Element Analysis

This work adopts ANSYS software to conduct modal analysis and harmonic response analysis. The FE model for the PCB is constructed as shown in Figure 3 and shown the convergence [9]. The board is assumed to be orthotropic material, and the chips are isotropic. Table 1 shows the related material properties.

Figure 4 shows the FE model of the fixed PCB. The linear spring-damper element (COMBIN 14) is adopted to simulate the fixed boundaries in the four corners by screwing. The damping effect of screw is neglected, therefore the only spring effects are included and covered over the screw area. There are 9 elements in each fixed location. The spring constant is assumed to be uniform and to be determined by the following optimization procedure.

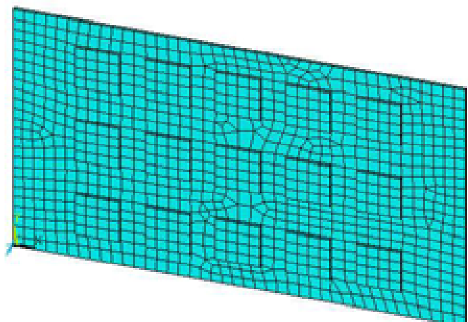


Figure 3. FE model PCB in Free-free boundary [9]

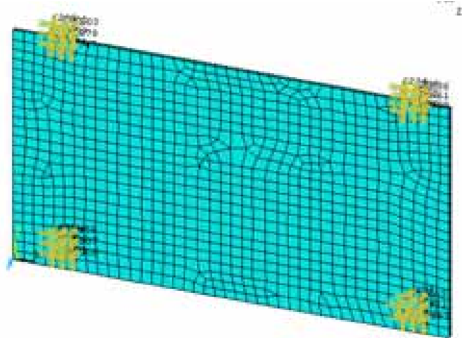


Figure 4. FE model PCB in Fixed boundary

Table 1. Material Properties of PCB [9]

	Properties
Board Young's Modulus $E_{X,B}$	9.42×10^{9} (N/m^3)
Board Young's Modulus $E_{Y,B}$	9.25×10^{9} (N/m^3)
Board Shear Coefficient $G_{XY,B}$	3.12×10^{9} (N/m^3)
Board Poisson's Ratio $v_{XY,B}$	0.25
Board Density ρ_B	2050(Kg/m^3)
Chip Young's Modulus E_C	20
Chip Poisson's Ratio v_C	0.4×10^{9} (N/m^3)
Chip Density ρ_C	1840(Kg/m^3)
Accelerometer Mass *m*	0.0015 (Kg)

This work utilizes the ANSYS optimization module to solve for the optimum spring constant *K*. The defined optimization problem is as follows:

(1) Design variables: The only design variable is the spring constant (*K*) that is to simulate the screw-fixed boundary.

$$X = (K) \tag{1}$$

(2) Objective functions: The design variable (*K*) must be determined to minimize the sum of square errors between theoretical and experimental natural frequencies. The objective function is shown defined as:

$$f(x) = \sum_{n=1}^{N} \varepsilon_n^2 \tag{2}$$

Where $$\varepsilon_n = \frac{f_{n,F} - f_{n,E}}{f_{n,E}} \times 100\% \tag{3}$$

(3) Constraints: The relative error between natural frequencies from FEA and EMA is specified with in $\pm10\%$, i.e.

$$|\varepsilon_n| < 10\% \tag{4}$$

The optimized spring constant *K* can be solved and to be 2.78×10^{6}(N/m) according to the above optimization formulation. More detail results will be discussed in Section 5.

4. Experimental Modal Analysis

Figure 5 shows the experimental setup for performing EMA. The conventional EMA procedure is adopted. The accelerometer is fixed and the impact hammer is moving. The FRFs between the acceleration and force can be experimentally measured and, input to the curve-fitting software, ME'Scope VES, to extract structural modal parameters, including natural frequencies, mode shapes and damping ratios.

Figure 6 shows the grid of measurement points. In previous work [9], there are 17X29 divisions in both x and y directions. The test results were good and well characterized the plate mode shapes. To expedite the experiments only those points on the four straight lines, i.e. A1~A4 as shown in figure 6, are taken into account. There are 88 points. The accelerometer is fixed near the plate left corner as denoted by in Figure 6 ■.

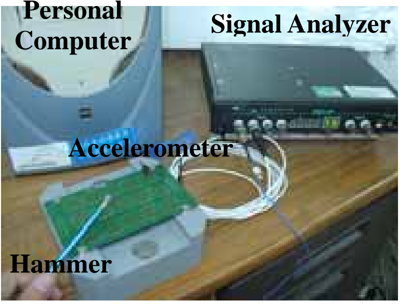


Figure 5. Experimental setup

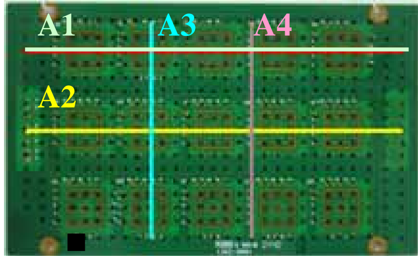


Figure 6. Measurement points.

5. Results and Discussions

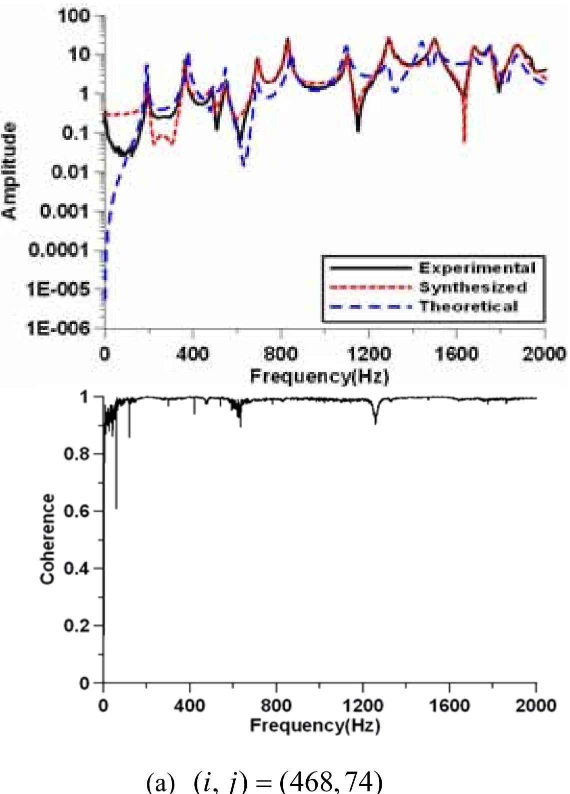
5.1 Frequency Response Functions

Figure 7 shows the FRFs and their corresponding coherence functions for $(i,j)=(468,74)$ and $(i,j)=(468,358)$, respectively. Some observations are discussed as follows:

(1) That both the experimental and synthesized FRFs watch very well indicates the success in curve-fitting process.

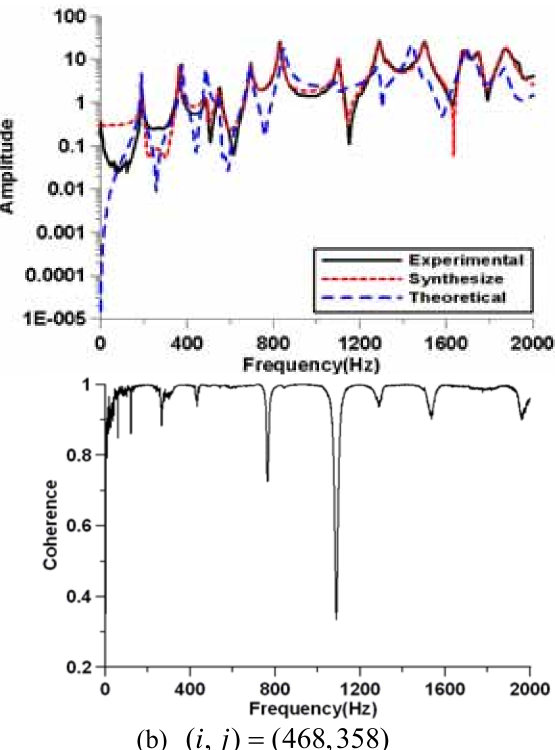
(2) The theoretical FRFs that are determined from the optimized FE model reveal reasonable agreement with the experimental ones.

(3) The coherence functions appear close to 1 mostly except where the antiresonance occurs. This indicates the reliability of the experiments.



(a) $(i,j)=(468,74)$

Figure 7. FRFs and their corresponding coherence functions (cont.)



(b) $(i,j)=(468,358)$

Figure 7. FRFs and their corresponding coherence functions (Cont.)

5.2 Modal Parameters

Table 2 summarizes and compares natural frequencies and their corresponding mode shapes between FEA and EMA. Discussions are as follows:

(1) There are 15 flexible-body modes below 2000 Hz as shown, expect that for the 12th mode only FEA mode is observed. The undetected mode in EMA can be the reason that the sensor is located at the nodal point.

(2) The comparison of natural frequencies is quite good. All of modes are within ±2% errors.

(3) From the FEA results, the 3D views of mode shapes are shown and reveal the typical plate modal characteristics as indicated by (m, n) related to (x, y) directions.

(4) The displacement mode shapes in lines A1~A4 for both FEA and EMA are also shown. One can observe that they agree well. Only a few modes that MAC values lower than 0.6 are not completely match. However, for most of modes the MAC values are pretty higher than 0.9. This indicates the very well agreement between the predicted and experimental mode shapes.

Table 2. Natural frequencies and mode shapes

No.	Freq.(Hz) / Diff.(%)		FEA result	MAC / (m, n)
E01 vs. F01	194 vs. 191.5			0.989
	-1.3			(1, 1)
E02 vs. F02	364 vs. 379.5			0.985
	4.3			(1,2)
E03 vs. F03	487 vs. 491.2			0.957
	0.9			(2,1)
E04 vs. F04	549 vs. 552.8			0.949
	0.7			(3,3)
E05 vs. F05	695 vs. 695.5			0.935
	0.1			(4,1)
E06 vs. F06	833 vs. 849.9			0.657
	2.0			(4,2)
E07 vs. F07	845 vs. 859.6			0.841
	1.7			(3,3)
E08 vs. F08	1100 vs. 1100			0.933
	0.0			(5,1)

Table 2. Natural frequencies and mode shapes (Cont.)

No.	Freq.(Hz) / Diff.(%)		FEA result	MAC / (m, n)
E09 vs. F09	1290 vs. 1290			0.800
	0.0			(4,3)
E10 vs. F10	1320 vs. 1311			0.888
	-0.7			(5,4)
E11 vs. F11	1500 vs. 1448			0.448
	-3.5			(5,4)
F12	1517	X		X
	X			X
E12 vs. F13	1670 vs. 1699			0.392
	1.7			(5,4)
E13 vs. F14	1750 vs. 1757			0.580
	0.4			(4,4)
E14 vs. F15	1880 vs. 1874			0.464
	-0.3			(5,2)

Table 3 shows the experimentally determined modal damping ratios. Because the exponential window is applied in impact modal testing, the obtained damping ratios are modified according to single degree of freedom (SDOF) assumption. The accumulated averaged damping ratios are also shown and can be adopted to specify the structural damping in FEA for ANSYS application.

Table 3. Modal damping ratios

Mode	Experimental damping ratios(%)	Modified damping ratios(%)	Accumulated averaged (%)
1	1.550	1.431936821	1.431936821
2	0.761	0.698076218	1.065006519
3	1.740	1.692968672	1.378987595
4	1.700	1.658280042	1.518633819
5	0.810	0.777044235	1.147839027
6	0.799	0.771503893	0.959671460
7	3.180	3.152894371	2.056282915
8	0.595	0.574177948	1.315230432
9	0.725	0.707244762	1.011237597
10	0.905	0.887648290	0.949442944
11	0.880	0.864730495	0.907086720
12	0.682	0.668284876	0.787685798
13	0.772	0.758911853	0.773298826
14	0.940	0.927816885	0.850557855

6. Conclusions

This work combines FEA and EMA to perform model verification of PCB base on structural modal characteristic. In addition to the realization of vibration properties of PCB, most importantly, the reasonable effective and equivalent FE model of PCB can be decided and helpful to the future applications, such as response prediction and failure analysis in vibration tests. The methodology presented in this paper can be adopted for other types of PCB as well.

7. References

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