Parametric Modal Study of Multilayer Composite Electronic Boards

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Abstract

Printed circuit boards made of composites in working environments such as avionics or transportation are susceptible of structural failure or irreversible damage under the vibration of the equipment. Structural modal/harmonic finite element models in ANSYS are integrated in this study to enable the predictions of resonance frequency shifts and failure stress for vertically clamped parallel circuit boards with inclusion of series of mounted electronic modules (chips). The board is modeled as a thin plate made of layers of E-glass composite with different fiber orientations and inter-fiber angles. Appreciable differences in maximum failure stress and resonance frequency shifts are observed by changing the above parameters in the fabrication of the boards. Mathematical approximations (metamodels) are provided to describe the complex trend of resonance frequencies in terms of angular design parameters. The model equations would simplify the parametric effects on frequency shifts and provide insight into any subsequent design optimization in the absence of access to FE codes, or readiness to devote more resources on FE analysis. Overall this study presents feasible ANSYS models that render successfully an optimal design for composite circuit boards under vibration loads, and without weight increase by modifications or material additions. This would lead to board designs that are more durable and reliable under noise, shock, and vibration loads.

Introduction

Printed electronic boards (PCB) and electronic composite boards (ECB) are subject to significant stress levels caused by temperature changes as well vibration and shocks. These conditions frequently have contributed to serious damages to the boards and loss of functionality in indispensable and diverse systems such as those in aerospace, automotive, and defense industries. In particular the CB's used in aircrafts and space crafts can experience frequencies ranging from 5-3000 HZ, and possible shock levels up to 30 g [1]. Thus the structural dynamic analysis of circuit boards is a major step in approaching more reliable and robust designs that are tuned for specific applications and environments.

Previous work has been done to produce structurally stronger boards and interconnections between the components on the board. Adding ribs, thickening the boards, or adding vibration isolation fixtures are among examples of ideas in board reinforcement. However these efforts can cause considerable increases in weights of equipments and may consume valuable space especially in aerospace or spacecraft systems. Recently smart materials and active damping have shown favorable results in control and reduction of vibration intensity in electronic enclosures [2-4].

To be able to analyze the composite boards and its components (e.g. connections, chips, etc), the designers need to access realistic mechanical material properties. This has presented a challenge to researchers and for reasons such as immense diversity of composites, accuracy in measuring the orthotropic properties, effects of frequency/amplitude/ temperature on damping properties, and the complexity of board's components. Although several of studies have been done on the characterization of damping in composites, the majority deal with limited applications, geometries, and specific composites other than those used in ECB's. Chandra et. al. [5] summarized the status of research in damping with emphasis on the damping characterization in polymer composites. Their review included the macro/micromechanical models on laminate and interphase damping and damage, as well as items of needed research in this field, such as the absence of 3D laminate models and effects of damage on damping. In a later study the authors [6] studied orthotropic loss factors based on finite element analysis and strain energy as they correlated the results to micromechanical theories. They also reported the effect of interphase volume fraction on the damping properties. An experimental study on unidirectional glass and Kevlar composites [7] addresses an observation regarding the change of damping loss factor with the fiber angle. The trend of damping found in that study will be incorporated in the model of present study. A recent paper on methodology for finding damping constants [8] is of interest since it utilizes simple bending vibration experiments and finite element models. This paper also discusses the use of proper damping inputs such as Rayleigh (proportional) mass damping vs. stiffness damping. The study [8] concludes some typical values for SDC (specific damping capacity or ψ), and loss factor (η) in woven glass (GFRP) polymer composites. A few related studies on damping characterization in composites have presented the loss factor vs. frequency using viscoelastic lamina models [9], the effect of carbon fiber orientation on specific damping capacity (ψ) [10], and the effect of temperature and aging on SDC [11]. But again it should be mentioned that the efforts cited above are very specific to the types of composites, experiments, and conditions used in the characterization process. Furthermore the nature of damping input in FE analysis is challenging as it becomes more intricate with using geometrically complicated models with anisotropic material input. To date, there seems to be an absence of studies on ECB's or PCB's under harmonic loads that address general guidelines in the selection of damping parameters, the use of appropriate damping values, the evaluation of damage causing parameters, and the prediction of failure stresses.

The present study is an effort to address two issues as related to a generalized model of an ECB made of unidirectional E-glass composite plies: (1) what are the effects of fiber orientation on shifting the resonance frequencies in a multilayer laminate board? The answer to this question is important in design for a robust product possessing natural frequencies distant to the applied excitation frequency, and (2) what are the typical trends in the board's maximum failures stress vs. fiber angle considering the existence of material damping? Analysis efforts regarding the issues presented above may lead to parametric board designs that can be more reliable without adding reinforcement ribs, isolation enclosure mounts, or installation of complex adaptive damping fixtures.

Modeling Issues-Modal Shifting

Any methodology in the design and fabrication for a reliable product under periodic excitations necessitates the knowledge of natural frequencies and mode shapes. The challenge is that there are multitudes of parameters that can affect the resonance status of a board. The board's supports, mass of components and their structural strengths, component interconnections and soldering, fiber orientations, fiber volume fraction, and modal damping are examples of those parameters. Hence it has been highly challenging to create mathematical/FE/experimental models that can (1) address commonly and accurately the effect of these parameters, and (2) be utilized in creating general and efficient methodologies for optimum board design. This study is of no exception; however it offers the feasibility of steps to create FE and math models that can explain the effects of more pertinent structural parameters, namely θ, the fiber orientation relative to a reference coordinate, and θ_i, the inter-fiber angles in multi-ply laminates. In previous parametric studies, ANSYS models were used successfully to evaluate the thermal stresses in a board similar to the one in the present study [12-13].

The simulated electronic modules are mounted on vertical parallel boards made of a common composite material. The reference material fiber orientation, $\theta = 0°$, is aligned in x direction. Four rectangular areas were assigned a typical added mass and rigidity representing chips mounted on the board. Three modes of vibration and associated resonance frequencies were found through ANYS models. Polynomial multivariable metamodels were calculated and presented for the variation of stress with fiber angles and for 2-layer and 3-layer composite boards.

Modeling Issues-Harmonic Excitation

In the second part of this study, effort is made to investigate the effect of fiber orientation and material damping property on the peak failure stresses caused by typical cyclic excitations of the board's base support (or enclosure). The challenges here are threefold: (1) the appropriate category of damping and the lack of guidelines in literature in selection of practical damping constants for composites, (2) the absence of values for different damping constants related to the anisotropic/orthotropic material behavior, and (3) the challenge of finding spatially the realistic maximum amplitudes of failure stresses as they are made of stress components that are not in phase with each other. The assumptions used for models in this study tries to address the above issues. The previous work [8] on the damping type recommends proportional damping, especially mass proportional one, as defined by α, for the harmonic analysis of composite beams. In addition, based on the existence of a practical operational range for ξ, the damping ratio, that has minimal effects on mass and stiffness damping, a "frequency-independent" damping is an appropriate input. The latter was utilized in this study, namely the "damping ratio" (MPDRAT) as a solution input to ANSYS. A typical trend found in [7] explaining anisotropic nature of the composite damping, and an upper limit for the damping ratio equal to .05, deemed to be the appropriate inputs for the FE models, as shown in Figure 1. Furthermore we investigated the effect of damping on phase shifts of each stress component. This was needed to estimate more accurately the peak values of stress intensity (SINT) or equivalent stress (SEQV). In the absence of relevant algorithms in ANSYS, the authors found a helpful technical memo [14] inclusive of an APDL routine that calculates the failure stresses vs. the times in a cycle. It was found that due to the low value of damping ratio the failure stresses are almost in phase with other stresses and ANSYS "Time-History" plots can be used to estimate the peak stress intensity with adequate confidence.

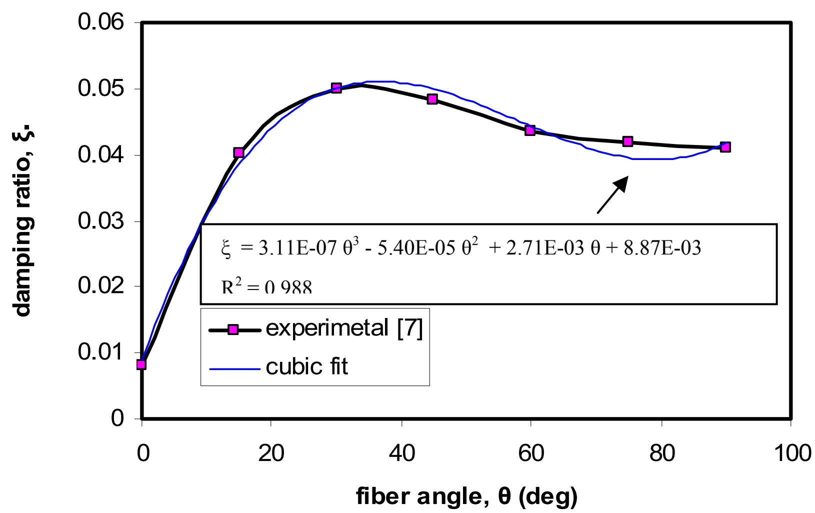


Figure 1. Typical damping ration for a glass composite

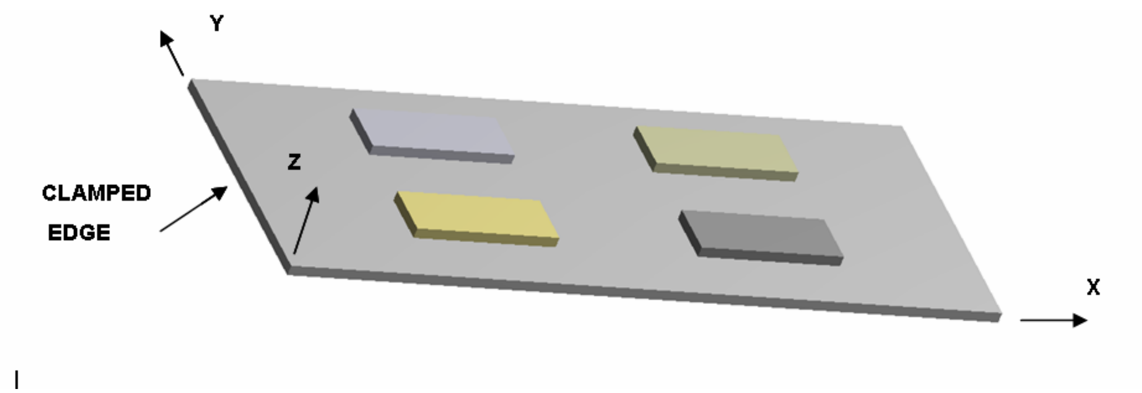


Figure 2. Physical model and placement of modules

Physical/F.E. Model

The physical model for the printed composite board is assumed to have dimensions of L=20 cm, w=7.5 cm and t=0.3 cm (Figure 2). In particular the board is made of an E-glass epoxy laminate with the assumed properties as follow: for glass fibers, E_f = 71 GPa, G_f = 6 GPa, ρ_f =2560 kg/m^3, and Poisson's ratio, υ_f =0.22. For the epoxy matrix, E_m = 4.66 GPa, G_m = 2 GPa, ρ_m =1120 kg/m^3, and υ_m =.29. The board contains four symmetrically located modules. Each module is represented by a rectangular area equal to 1.5cmX4cm, adding to the mass and stiffness of the board. The properties assumed for each module were close to those for the board. They were: t = .3 cm, E_c= 71 GPa, ρ_f =2200 kg/m^3, and υ_f =0.28.

For the modal analysis in the first part of this study, the board is assumed to be made of two and three unidirectional plies that can possess fiber volume fraction (V_f), reference fiber orientation (θ) and inter-fiber angle (θ_i). The laminate properties were found by the rule of mixtures [15, 16]:

$$\rho_c = \rho_f V_f + \rho_m V_m \quad (1)$$

$$\nu_{12} = \nu_f V_f + \nu_m V_m \quad (2)$$

$$\nu_{23} = \frac{E_{22}}{2G} - 1 \quad (3)$$

$$E_{11} = E_f V_f + E_m V_m \quad (4)$$

$$E_{22} = \frac{E_m}{1 - \sqrt{V_f}(1 - E_m / E_f)} \quad (5)$$

$$G = \frac{G_m}{1 - \sqrt{V_f}(1 - G_m / G_f)} \quad (6)$$

The structural ANSYS model representing the circuit board and the rectangular modules (chips) is rendered using shell quad elements (shell 181). The DOF's at the nodes of chips are coupled to those on the board to simulate the bonding effect. Structural boundaries include a vertically clamped boundary representing the inserted or clamped left edge of the board and three other unconstrained edges. At the clamped edge, it is assumed that UX=UY=UZ=ROTY =0. The investigation of natural frequencies included a diverse range of orientation angle θ =0 to 90°, and the inter-fiber angle, θ_i =0 to 90° , in increments of 10°. These ranges would cover all possibilities for the bending mode shapes due to the symmetry. The fiber volume fraction is set at 0.55 in all cases. The configuration of layers for the 3-layer board is symmetrical, e.g. (10^o -50^o -10^o).

The second part of study incorporates the same model as the above, with the exception of replacing the transverse motion of the clamped edge with UZ=10^{-3}Sin(ωt) *m* , thus changing the model analysis to a damped harmonic analysis. The frequency-independent damping values as presented earlier were input to the "Solution" module in ANSYS with "MPDRAT" command. Peak stress intensity in a unidirectional-fiber board vs. frequency of excitation was observed in "Time History" module. Then the values of stress peaks vs. fiber orientations were plotted for the purpose of finding an optimum angle. One of the features of ANSYS that was helpful in this model was the capability of coupling (CP command) the DOF's of nodes on the chips with those on the boards.

Results of FE Analysis

Modal Shifting : Understanding the trends in modal characteristics of the boards can be valuable for structural design optimization in instances where noise or vibrations are present. To achieve more control in the design of the boards, unidirectional multilayer laminates with different fiber orientations can be used in fabricating the board. In this study, 2-layer and 3-layer laminates are investigated for the effects of fiber orientations and for the 3 lowest natural frequencies: one frequency belongs to the tensional mode and the other two are related to the bending modes. Figure 3 and 4 demonstrates the resonance frequencies as functions of fiber orientations, θ, and the increment fiber angles, θ_i. Overall the frequency shifts up to 72% can be observed depending on the fiber angles. It is interesting to notice the similarity of trends for the frequency surface plots in the bending modes (3a and 3c) as compared to the plot for the torsional or twisting mode (3b). It is also observed that for the 2-layer board both θ and θ_i significantly affect the shifts, but for the 3-layer board θ, and not θ_i, is significantly responsible for shifting the resonance frequency.

Polynomial Models-modal Shifting

For convenience the ANSYS results were fitted to polynomial expressions that were established using stepwise regression of a cubic (in one case quadratic) polynomial using *p = 0.05* as a criterion to enter a term in the model. These models can be construed as modal characteristics of a board and can be of assistance in the design of the board without using any FE simulation. The following six equations represent the stepwise regression that fits to the data from the surface plots in Figures 3a to 4c respectively. The frequencies are indicated by the subscripts where the first index denotes the mode of the vibration and the second index denotes the number of layers in the board.

$$f_{12} = 56.7 - 0.586\theta - 0.344\theta_i + 3.83\times10^{-3}\theta^2 + 2.45\times10^{-3}\theta_i^2 + 4.03\times10^{-3}\theta\theta_i - 1.71\times10^{-5}\theta\theta_i^2 \qquad \mathrm{R}^2_{\mathrm{adj}} = 0.915 \tag{7}$$

$$f_{22} = 156 + 2.75\theta + 2.67\theta_i - 0.0578\theta_i^2 - 0.163\theta\theta_i + 2.06\times10^{-3}\theta\theta_i^2 + 2.22\times10^{-3}\theta^2\theta_i - 1.28\times10^{-5}\theta^2\theta_i^2 - 1.08\times10^{-3}\theta^3 + 3.21\times10^{-4}\theta_i^3 - 7.94\times10^{-6}\theta^3\theta_i - 7.03\times10^{-6}\theta\,\theta_i^3 + 8.19\times10^{-6}\theta^4 \qquad \mathrm{R}^2_{\mathrm{adj}} = 0.874 \tag{8}$$

$$f_{32} = 362 - 2.93\theta - 2.74\theta_i + 0.0203\theta_i^2 + 0.0353\theta\theta_i - 1.26\times10^{-4}\theta\theta_i^2 - 7.75\times10^{-5}\theta^2\theta_i + 1.77\times10^{-4}\theta^3 \qquad \mathrm{R}^2_{\mathrm{adj}} = 0.944 \tag{9}$$

$$f_{13} = 59.2 - 0.771\theta + 5.86\times10^{-3}\theta^2 \qquad \mathrm{R}^2_{\mathrm{adj}} = 0.983 \tag{10}$$

$$f_{23} = 164 + 2.73\theta - 0.0450\theta^2 + 1.45\times10^{-4}\theta^3 \qquad \mathrm{R}^2_{\mathrm{adj}} = 0.916 \tag{11}$$

$$f_{33} = 363 - 3.20\theta + 8.47\times10^{-4}\theta\theta_i + 2.12\times10^{-4}\theta^3 \qquad \mathrm{R}^2_{\mathrm{adj}} = 0.988 \tag{12}$$

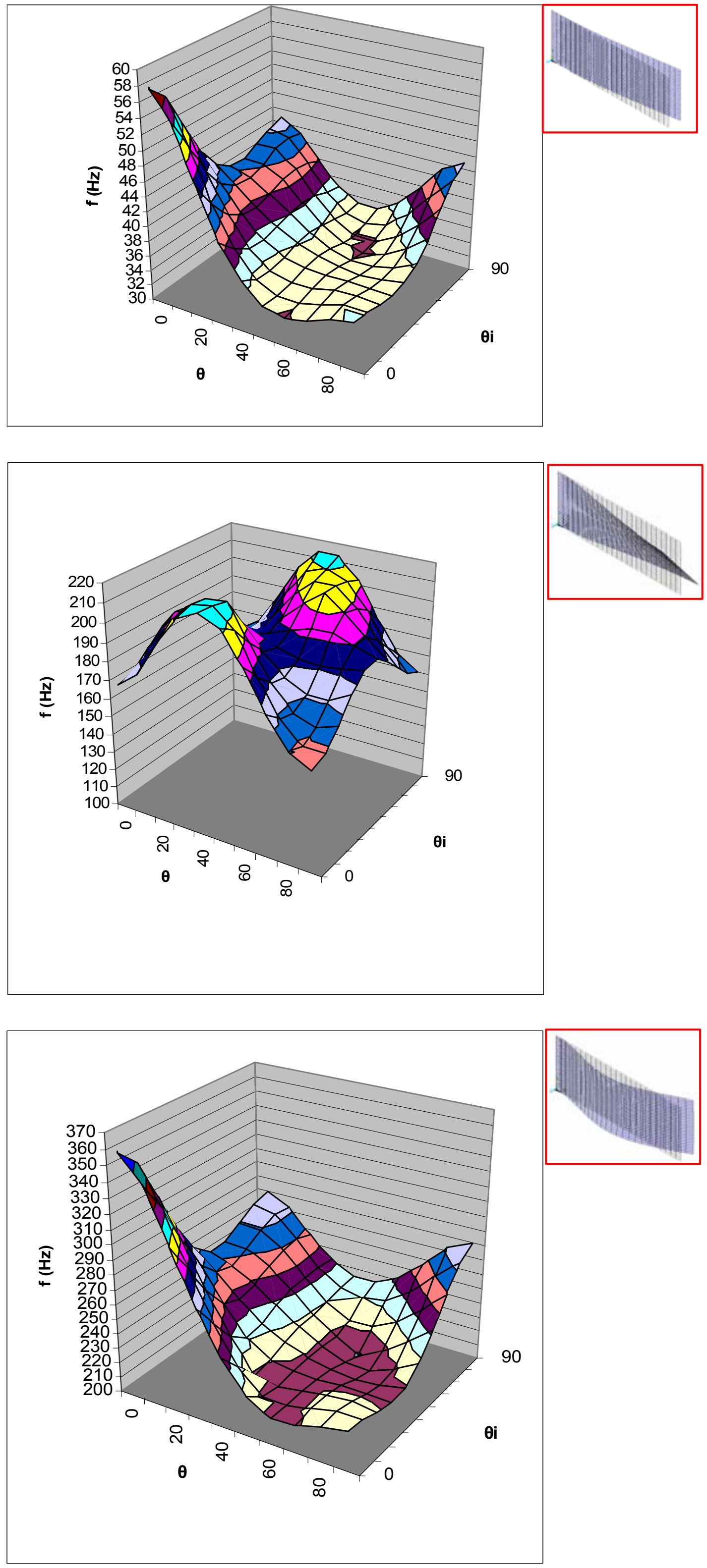
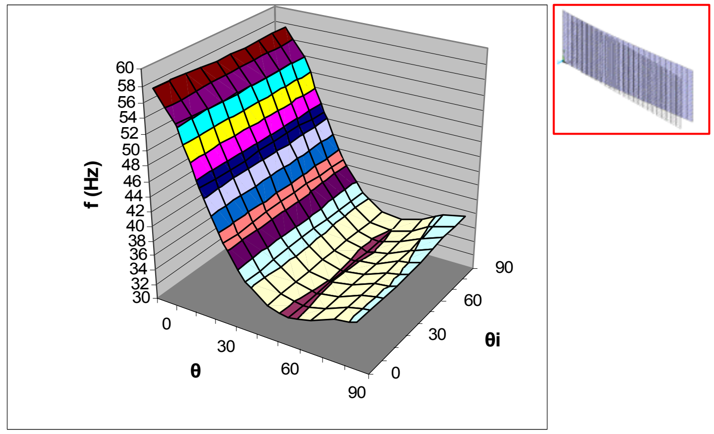
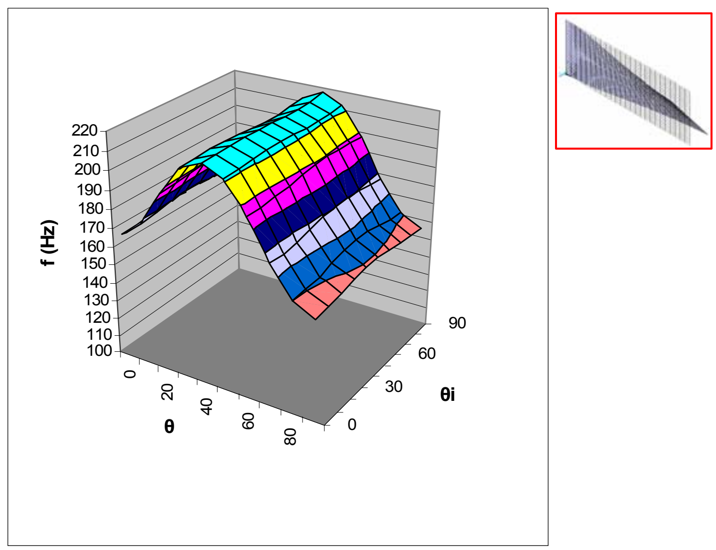


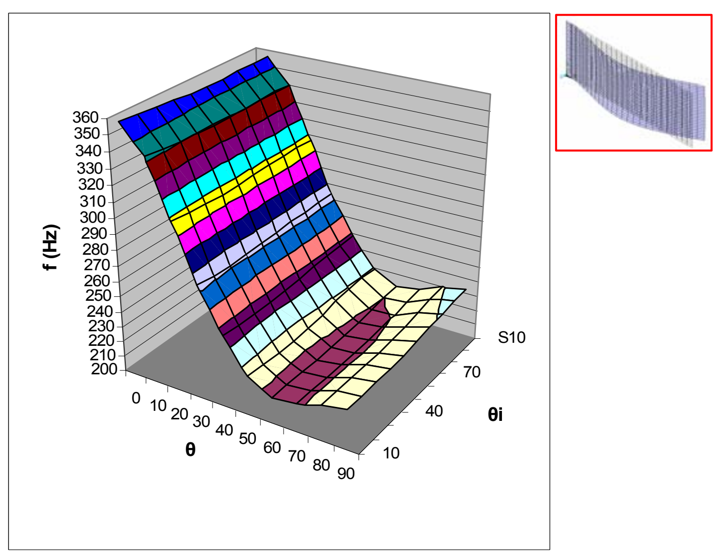
Figure 3. Resonance freq. for 2-layer board (a) 1st mode - bending, (b) 2nd mode- torsional, and (c) 3rd mode-bending



(a)



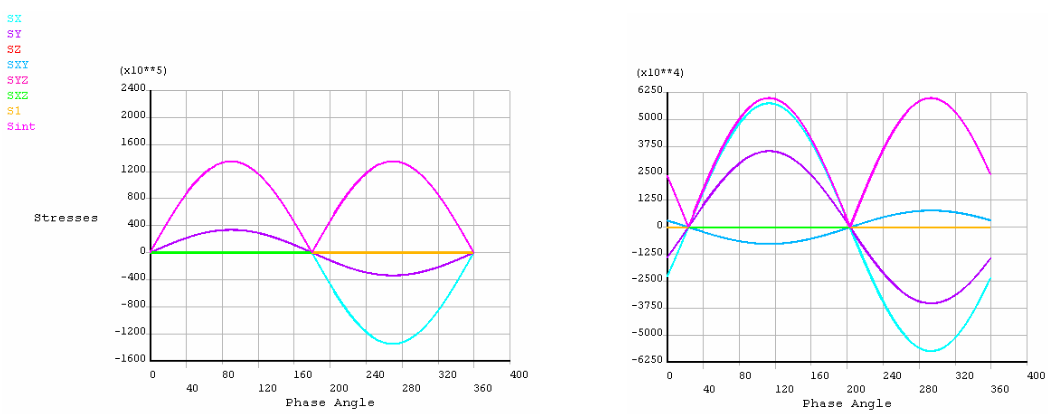
(b)



(c)

Figure 4. Resonance freq. for 3-layer board (a) 1st mode - bending, (b) 2nd mode- torsional, and (c) 3rd mode-bending

Harmonic Base Excitation: Effort was made to address the feasibility of finding an optimum design that can provide more robustness for the board. This can be achieved by reducing maximum failure stresses in the boards as the fiber angle changes. The results in this part include the effect of fiber angle on maximum stress intensity in a single-ply board. Future work can be extended to study the added parameters in 2- or 3-layer boards similar to those in the modal shifting as presented above. The failure stress chosen was maximum stress intensity (SINT). This stress is combination of principal stresses and its maximum usually occurs at a node on the clamped edge. However the location on the edge varies depending on the fiber orientation. It was discerned that the amplitudes of harmonic stress components found by ANSYS are out of phase depending on the level of damping. This would possibly render a somewhat inaccurate prediction of maximum stress intensity in the board since the principal stresses may not be in phase with each other. It has been reported that minimal damping ratios (less than 0.1) may not create a significant phase change for stress components depending on the model and loading conditions. Consequently it was decides to check the significance of the phase changes by calculating the stress magnitude throughout the cycle, $\omega t = 0 \rightarrow 360°$, and compare them to the results presented in the time-history module of ANSYS. An ensuing research directed us to a helpful technical memo [14] that provided an APDL macro to find the combination stress components at different times in the cycle. Figure 5 presents samples of results obtained by running the macro, as they conclude that the variations of components of stress are all very close in phase and the values of peak stress intensity provided by Time-History module can be used with confidence.



(a) (b)

Figure 5. Stress components vs. time (phase angle) in a cycle: (a) board with θ=0 ° (b) board with θ=50 °

The detection of location, value, and frequency regarding the maximum stress intensity in the board was carried out as follow:

- The ANSYS command HRCPLX was used to study the contours for the maximum stress intensity amplitude. A typical plot is shown in Figure 6. The node at which the stress is maximum is recognized and noted down.
- APDL macro (above) was used to confirm the same-phase assumption for SINT at that node.
- The peak values of the SINT was found from the stress-frequency plots/listings in the Time-History module for boards with $\theta = 0 \rightarrow 90°$ in 10° increments.

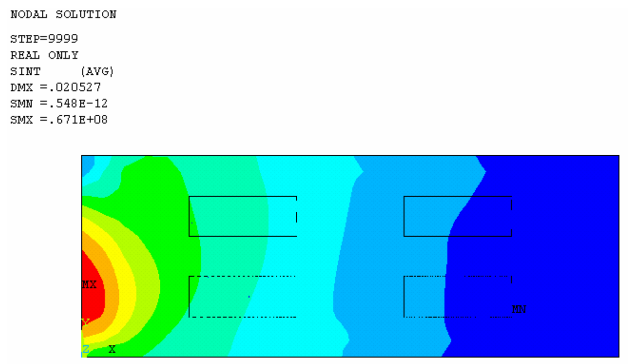


Figure 6. Stress components vs. time (phase angle) in a cycle: (a) board with θ=0° (b) board with θ=50°

Figure 7 summarizes the effect of θ, fiber orientation, on the stress intensity (SINT) under the assumption that damping ratio is a typical constant regardless of fiber orientations and in another case where the damping ratio follows an observed trend as seen in Figure 1. The value for the constant damping ratio is the average of values in Figure 1, as it equals to 0.39.

There are few important findings regarding the effects of θ in Figure 7. There is a consensus of minimum around θ = 60 ° for both sets of stress data. According to these results, for a board with properly oriented fibers the max stress intensity in the board can reach significant reductions of about 49 % under the assumption of constant damping and 92 % in the case of variable damping. Another finding as observed in Figure 8 can partially justify the minima mentioned above. It is noticeable that the resonance frequency also has a minimum at the same fiber angle, supporting the fact that the max stress intensity is directly influenced by the natural frequency of the board.

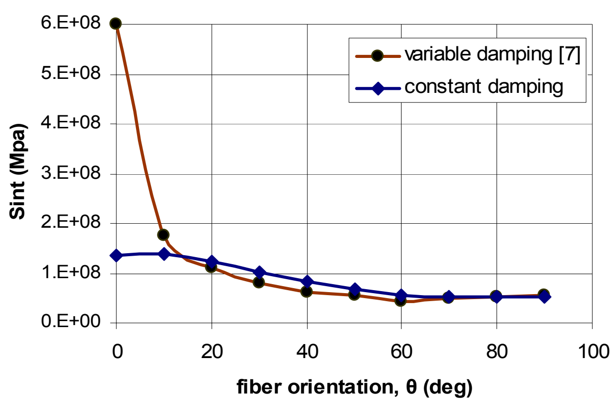


Figure 7. Effect of fiber orientation on max stress intensity

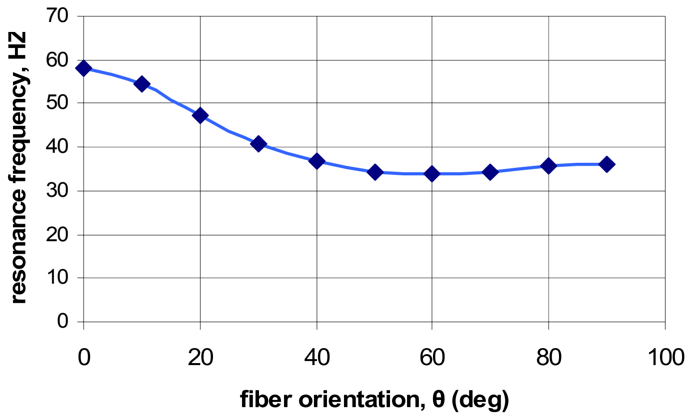


Figure 8. Effect of fiber orientation on resonance frequency

Conclusion

This study presents efforts regarding modal/harmonic modeling in ANSYS as they may provide feasible optimal designs for a composite circuit board under vibration loads. These designs are valued since they do not use addition of ribs, mass damping, complicated active/adaptive damping, or any modification involving increase in the weight. The first part of this work addresses parametric studies of the effect of θ, fiber orientation, and θ_i, the inter-fiber angle, on resonance frequencies for 2-layer and 3-layer composite electronic boards. It was found that the trend of these variations depend on the mode of vibration as well as the number of board layers. For the 3-layer board the fiber orientation is very dominant in shifting the resonance frequency. Multivariable metamodels summarizing the mentioned effects were calculated and presented. The metomodel equations can be helpful to the designer of the board in case of unavailability of FE codes.

An effort was made to explain the effect of fiber orientation on the maximum failure stress in an electronic board under cyclic excitations. The sinusoidal motion of the supportive edge, normal to the board, causes stresses that can be detrimental to the board or its components. Furthermore it is assumed that the damping property is frequency independent and varies with the fiber angle. The stress results by ANSYS are encouraging in the design of a board that provides minimum failure stresses without addition of mass or adaptive damping. It was found that for the simulated board in this study, an optimum design with the fiber orientation of about 60° offers a minimized value for the maximum failure stress (SINT). Future work will include the consideration of other parameters such as harmonic analysis of multilayer boards with changing inter-fiber angles, orthotropic/realistic damping ratio values, and different chip layouts/interconnections.

Reference

1) D.S. Steinberg, Vibration Analysis for Electronic Equipment, 1988, Wiley, New York

2) B. Bhattacharya, B.R. Vidyashankar, D. Patsias, G.R. Tomlinson, 2000 “Active and passive vibration control of flexible structures using a combination of magnetostrictive and ferromagnatic alloys,” Proceedings of SPIE -The Int. Society for Optical Engineering, v 4073, p 204-214

3) B. Esser, D. Huston, J. Miller, 2003, “Aerospace electronic weight reduction through the use of active mass damping , Proceedings of SPIE -The Int. Society for Optical Engineering, v 5052, p 433-444

4) A. Bronowicki, N.S. Abhyankar, S.F. Griffin, 1999, “Active vibration control of large optical space structures,” Smart Mat and structures, v 8, n 6, p 740-752

5) R. Chandra, S.P. Singh, K.Gupta, 1999, “Damping studies in fiber-reinforced composites-a review”, Composite Structures, v 46,p 41-51

6) R. Chandra, S.P. Singh, K.J. Gupta, 2003 , “A study of damping in fiber-reinforced composites, Journal of Sound and Vibration, v 262,p 475-496

7) J. M. Berthelot, Y. Sefrani, 2004, “Damping analysis of unidirectional glass and Kevlar fiber composites, Composites Science and Technology, v 64, p 1261 1278

8) C. Kyriazoglou, F. J. Guild, 2006,“Finite element prediction of damping of composite GFRP and CFRP laminates, Composites Science and Technology, v 66, p 487-489

9) C. Kyriazoglou, D. Korontzis, 2003, “A new method for the determination of viscoelastic properties of composite laminates: a mixed analytical-experimental approach”, Composites Science and Technology, v 63, p 1441-1452

10) R.D. Adams, M.R. Maheri, 2003, “Damping in advanced polymer-matrix composites,” Journal of Alloys and Compounds, v 355, p 126-130

11) S.C. Sharma, M Krishna, A. Shashishankar, S.P. Vizhian, 2004, “Damping behavior of aluminum/short glass fiber composites,” Material Science and Engineering, v 364, p 109-116

12) A. Khalilollahi, 2004, “Finite element stress evaluation of a composite board using 2-d and 3-d convection models”, Proceedings of ANSYS 2004 Conference, Pittsburgh, PA, (CD format)

13) A. Khalilollahi, 2005 , “Thermal stress reduction and optimization for orthotropic composite boards”, Proceedings of 2005 ASME Summer Heat Transfer, San Francisco, CA, (CD format)

14) S. Imaoka, 2004, “Sheldon’s ANSYS tips and tricks: Post processing harmonic results”, ansys.net memo No. STI:05/03, Aug. 22, 2004

15) K. K. Chawla, 1998, Composite Materials, Science and Engineering, Springer Verlag, Berlin.

16) C. C. Chamis, , 1983, “Simplified Composite Micromechanics Equations For Hygral, Thermal And Mechanical Properties”, NASA Tech. Memo NASA-TM-83320