7. SPECIFIC SOLUTIONS

The previous rules are esentially qualitative and focused on guiding the early stages of the design process to the securing of a product which is reliable in the usual vibration enviroments. When especially harsh conditions are expected, a more rigorous analysis and more powerful specific solutions are necessary.

7.1 RESTRICTION IN MAXIMUM DISPLACEMENT

Since the stress in the different electronic components is essentially given by the displacements in the PCB, it is interesting to relate the fatigue life of the components to the dynamic displacements developed by the board during vibration, trying to find a suitable maximum value for the latest. According to [1], many different types of electronic components can achieve a fatigue life of about 10 million stress reversals in a sinusoidal-vibration excitation, and 20 million for random-vibration enviroment, when the peak dynamic single-amplitude displacement of the PCB is limited by the value given

$$Z = \frac{0.00089\ B}{C\ h\ r\ \sqrt{L}} \qquad (7\text{-}1)$$

where

B = length of PCB edge parallel to component, cm.

L = length of electronic component, cm.

h = height or thickness of PCB, cm.

C = constant for different types of components.
1.0 for standard dual inline package (DIP)
1.26 for a DIP with side-brazed lead wires
1.26 for a pin grid array (PGA) with two parallel rows of wires extending from the bottom surface of the PGA
1.0 for a PGA with wires around the perimeter extending from the bottom surface of the PGA
2.25 for a leadless ceramic chip carrier (LCCC)
1.0 for leaded chip carriers where the lead length is about the same as for a standard DIP

r = relative position factor for components.
1.0 when component is at center of the PCB (0.5 point X and Y)
0.707 when component is at 0.5 point X and 0.25 point Y on a PCB supported on four sides
0.5 when component is at 0.25 point X and 0.25 point Y on a PCB supported on four sides

The parameter C offers difficulties when the type of component used does not appear in the list given. It is the case of the BGA, a frequent and critically fragile component. It is necessary then to use a suitable value, big enough for being secure without causing an excessively expensive design. For example, if a BGA is analysed the LCCC should be used as the most suitable comparison, since both consists on a big chip packaged to the PCB in a configuration which is stiffer than the usual “legged” ones. The value used for the BGA should be in the same range of the one given for the LCCC, with 3 being an advisable one.

For the parameter r it is possible to estimate the respective value according to the position of the electronic component, or use the most suitable from the given. In both cases it is important to use a value in the side of the security.

7.2 DESIRED RESONANT FREQUENCY

The main condition applied to the resonant frequency of the PCB is the fulfillment of the octave rule. However this value is relative, and does not fully assure the reliability of the electronic components. A more accurate estimation can be done using the previous section.

Assuming the PCB acting like a single degree of freedom system at its resonant condition, the peak single amplitude displacement expected at its centre can be estimated with equation

$$Z = \frac{a_{max}}{\Omega^2} = \frac{a_{max}}{4\,\pi^2\, f^2} \tag{7-2}$$

where a_{max} is the maximum acceleration of the vibration, Ω the natural frequency expressed in rad / s and f the frequency expressed in hertz.

With this equation and the result from the previous section it is possible to determine then the minimum PCB resonant frequency to assure the aforementioned 10 million cycles fatigue life:

$$f = \sqrt{\frac{a_{max}\; C\, h\, r\, \sqrt{L}}{0.00356\, \pi^2\; B}} \tag{7-3}$$

where a_{max} is output acceleration, that is, it is related to the vibration of the PCB, and not to the excitation. Often this value is not known before the design stage is complete. If there are approximate values for the transmissibility Q, the last two equations can be expressed then as:

$$Z = \frac{Q\, a_{max,input}}{\Omega^2} = \frac{Q\, a_{max,input}}{4\,\pi^2\, f^2} \tag{7-4}$$

$$f = \sqrt{\frac{Q\, a_{max,input}\; C\, h\, r\, \sqrt{L}}{0.00356\, \pi^2\; B}} \tag{7-5}$$

If the transmissibility is unkwon, it is possible to use one of the approximations given before. For example, considering the relation $Q = \sqrt{f_n}$

$$f = \left(\frac{a_{max,input}\; C\, h\, r\, \sqrt{L}}{0.00356\, \pi^2\; B} \right)^{\frac{2}{3}} \tag{7-6}$$

7.3 RELATIVE DEFLECTION AND BOUNDARY CONDITIONS

It was explained in SITUATION OF THE ELEMENTS ON THE PCB that, although the usual procedure is to place the critical electronic components near the edges of the board, the optimum configuration is to place them in the areas where the flexure moment is zero. These may coincide with the edges of the board (such as in simply supported conditions), but can also be in another place of the PCB (clamped conditions).

The analysis of the flexure moment has traditionally been done through strength of materials and structures theory. The moment is the space derivative of the shear stress, and both are usually easily calculated for different configurations of beams (an example is shown in Fig. 7.1). The two-dimensional case of a board is usually much more complex, although nowadays it has been greatly simplified thanks to computers and FEM.

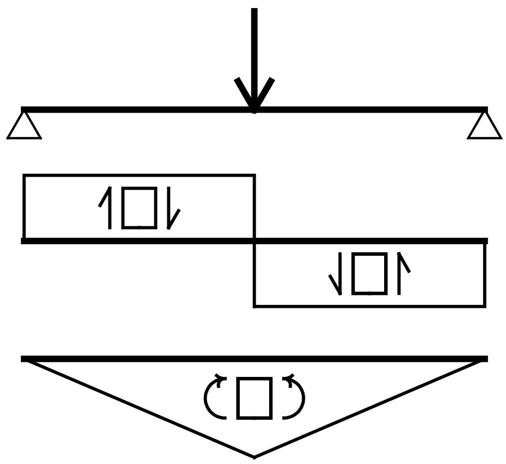

Fig. 7.1.- Stress analysis in simply supported beam.

The suggested procedure is then to conduct a FEM analysis, which will provide the areas of the board in which the moment is minimum for the excited vibration mode. The shape, size or position of these areas will depend on the actual configuration of the electronic structure, and placing an electronic component there will assure that the relative deformation with the PCB is minimized, regardless of the final excitation levels.

It might be then possible to design the boundary conditions of the PCB in order to provide a large area with low flexure moment, instead of the usual procedure of maximizing the resonant frequency of the board explained in GENERAL SOLUTIONS. This can be especially interesting in applications where one electronic component is much more sensible than the rest, such as a board with a big BGA and several small and more reliable legged components. The new design will provide a lower general reliability, but more protection for the critical one.

Clamped boundary limits impose the condition of zero displacement in the edge of the PCB, while with simply supported conditions it is its first derivative which equals zero. Although clamped conditions also provide an area of zero momentum in the board, it is generally smaller than the one generated with simply supported edges, and its position will depend on the vibration mode, so it might be difficult to accurately locate it. On the other hand, the reduction of the total deflection of the PCB is higher in the case of clamped edges.

Using a dynamic FEM analysis it is possible to compare the shapes of the PCB under the same vibration conditions and different boundary limits, and decide which configuration will be more suitable.

The reduction of the resonant frequency associated with supported conditions should provide also with a reduction of the transmissibility, but this effect is difficult to assess, so it is not suggested to rely on its efect as an improvement.

Two important considerations must be done. First, it is important to assure that the boundary limits are really simply supported, and not a loose guide allowing rotation, given the effect this would have (see LOOSE PCB GUIDES). The tecnical solution could involve the same holder than for the clamped conditions, but with an axis or a ball joint allowing rotation.

Secondly, this solution can be dangeous when the frequency of the expected excitation is near the resonant frequency of the PCB with simply supported boundary limits. In resonance conditions a small miscalculation result in critical failure and a FEM may not be accurate enough.

Example

A critically sensible electronic component is placed on a PCB of rectangular shape, 97 x 24 x 1.6 mm of free space, made of FR-4 and modelled with a density of 3487 kg / m^3 to simulate the effect of the components. The excitation expected is sinusoidal at 150 Hz, and the rest of the electronic elements are assumed to be reliable enough as long as resonance is avoided. The board is to be supported by the two shorter edges, and it is desired to study the best configuration in order to obtain to maximize the reliability of the critical component.

Two different configurations are considered: clamped in both sides, and clamped-simply supported, with the critical component being near the latter. The resonance frequencies are predicted as 406.22 and 278.83 Hz respectively. This means that in both cases resonance conditions are avoided.

The following step is to evaluate the shape of the board in its response to the expected excitation. A dynamic FEM analysis is conducted, and the following figures show the total transversal displacement (including the motion of the base).

The maximum deflection is about 2.5 times higher in the case of the case of the supported-clamped board, and the respective shapes are as predicted by theory.

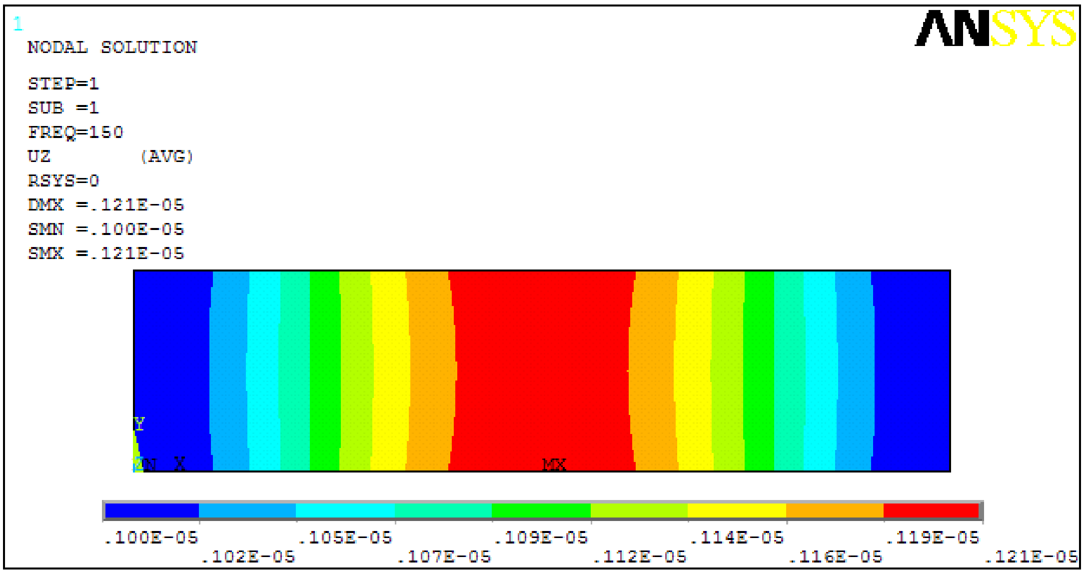

Fig. 7.2.- Total transversal displacement for clamped-clamped PCB.

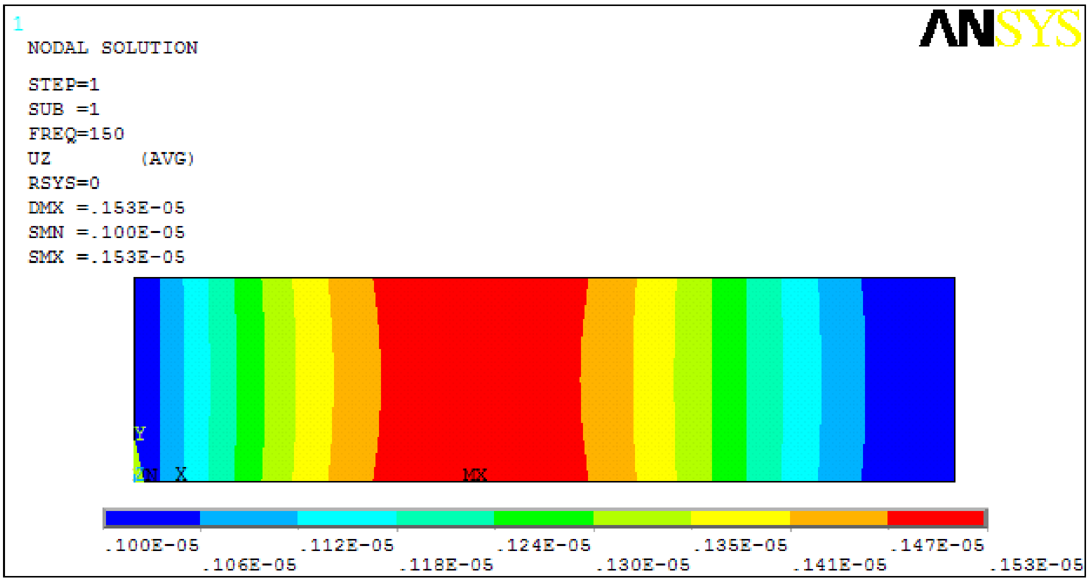

Fig. 7.3.- Total transversal displacement for clamped-simply supported PCB.

Next figure shows the relative displacement of the central line of the PCB, from the edge with different conditions in each case to the middle point. A black line has been added to show that, although the deflection is much higher for the clamped-supported conditions, the relative deformation nead the edge is smaller in that case. Between 0.15 and 0.25 mm from the clamped edge there is another zero momentum area, although smaller than in the other case.

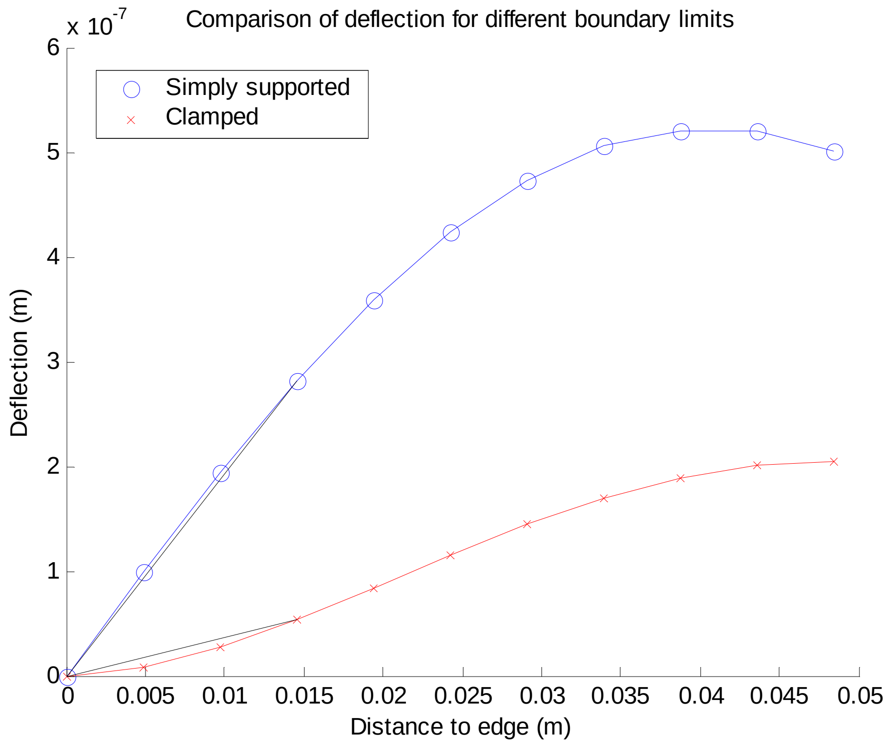

Fig. 7.4.- Relative deflection of the central line of PCBs in different conditions

The final decision should be done taking into consideration the type of PCB and electronic components on it, but it is clear that both configurations can be considered.

7.4 REDUCING THE SPRING RATE OF THE ELECTRICAL WIRES

Sometimes it is useful to reduce the spring rate of the electrical lead wires. Although it can seem contradictory, it is due to the special conditions on the electric components. Usually in structural analysis the forces acting over the structure are defined, so increasing the spring rate of the different elements will reduce both the defformation and the stress. In the case of most electronic components, the defformation is assumed to depend almost exclusively on the behaviour of the PCB, that is, independent of the effect of the electronic components, so according to Hooke's law

$$F = k\,x \qquad (7\text{-}7)$$

a reduction of the spring rate k for a given defformation x will produce a reduction of the force F. Although the electroctic component and its wires are not exactly a mass-spring system, the process has been proved useful, and the analogy is accurate enough for a first analysis.

The way to reduce the spring constant will depend on the kind of mounting and its way of deforming. In the typical “legged” application the electrical lead wires are forced to bend, so the stiffness involved is

$$k = \frac{E\,I}{L^3} \tag{7-8}$$

Since E is a material property, the stiffness can be reduced both by increasing the length L, whose effect will be high due to the cubic factor, or decreasing the moment of inertia I. This is accomplished by coining the lead wire into a flat thin metal strip perpendicular to the defformation plane. This way the moment of inertia is reduced while mantaining the cross-sectional area.

Effect on response

The reduction in spring rate should not imply a noticiable increase in the dynamic response of the system. Using a spring-system model as seen in Fig. 7.5, where x and y are the displacement of the electronic component and the PCB, respectively, the magnitude of the response for the usual sinusoidal excitation is

$$X = \frac{k}{k - m\,\omega^2}\,Y = \frac{1}{1 - \left(\frac{\omega}{\Omega}\right)^2}\,Y \tag{7-9}$$

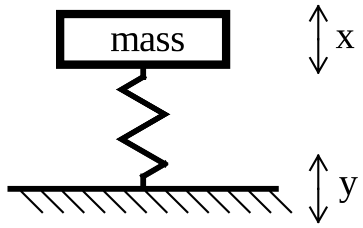

Fig. 7.5.- Spring-mass model

The possible influence of a variation in the spring rate will only affect the natural frequency Ω of the system. However this effect is practically neglectable, as resonance frequencies of electronic components are usually very high. Taking 5000 Hz as a suitable value, and an excitation frequency of 1000 Hz, a reduction of factor 2 in the spring rate will produce an increase in the response of only 1.0435 times. Other values can be found in Table 7-1.

Reduction of spring rate	Increase in response
1.5	1.0213
2	1.0435
5	1.2000
10	1.6000

Table 7-1.- Reduction of spring rate and its effect on response

Due to the typical shape of fatigue curves, the hypothetical reduction of strain will be amplified when translated into an increase in fatigue life.

Other types of mounting

The method applies also to other mounting systems, such as BGA, LCCC or poke-through. In applications that do not use lead wires the reduction must be done on the shear stiffness of the solder joints. According to [1], the greatest success in this direction is provided increasing the heigh of the solder joint, while the attempts to affect on cross-sectional area of the shear modulus have not been fully successful.

More recent studies have been produced, but mainly focusing on thermal cycling, as in [8]. The possible use of the results to mechanical loading is still not clear.

7.5 LOCAL STIFFNERS

A local increment on the stiffness of the PCB can provide a reduction of the relative deflection in a certain area whitout affecting the whole dynamic properties of the board. This can be achieved adding local stiffners elements near especially sensible electronic components. Components such as large DIPs will allow a metallic sheet to be epoxy-cemented to the PCB under it, which makes the process simple and therefore especially interesting. Other components such as BGA will probably require applying the stiffner in the other side of the board, with the corresponding lost of space increasing the production costs. The effect of the metallic sheet should be studied in an electronic point of view, since new generation high frequency processors could induce in them capacitor properties, apparently neglictable but important enough to produce critical delays in the circuit. Possible solutions might involve the use of stiff but non metallic elements.

7.6 CONFORMAL COATING

A usual solution for protecting electronic components in harsh conditions is to apply a conformal coat, usually made of solutions of silicone rubber or epoxy. Although the intention is usually the protection from corrosion or other external agents, it can probably increase the stiffness of solder joints in legged components. Its application to BGA is still to analyse.

7.7 INCREASING THE RESONANT FREQUENCY OF A PCB

This is usually the most effective method for improving the fatigue life, since displacementes are reduced very quickly. Appart from the general rules given in GENERAL SOLUTIONS other three different procedures are presented here.

Drilling an orifice

In some applications, such as aeroespace systems, it is important to increase the resonant frequency of a board without augmenting the total weight of the system. If spatial requirerements are not critical, it is possible to do so just by drilling an orifice in the area of less stiffness, which given the usual boundary conditions is normally the center of the PCB. For example, [9] shows that increases of more than a 10 % in frequency can be obtained when the diameter of the hole is bigger than the fifth of the side of a square PCB. This method is however probably not suitable for tools where the spatial distribution requires a full exploitation of the suface of the board.

New support conditions

A simple technique for maximizing the resonant frequency of vibrating structures is to modify the support conditions of the board, as explained in [10]. Either if the number of point supports is increased or the current ones are barely readjusted, it is important to optimize the locations for the maximum increase in the board's fundamental frequency.

The key is to place the supports at positions so as to prevent the maximum number of lower modes of vibration modes, forcing the structure to converge to the next lowest mode. The ideal positions are not the points of maximum displacement of the lower vibration modes, as it could be supposed, but the nodes of the desired mode, that is, the next one to the ones that it is intend to prevent. Forcing those points to have no displacement will sweep out the lower modes.

For example, adding a support in the free end of a clamped beam will increase its resonant frequency. This gain is however smaller than the one that could be obtained following the method displayed, since placing a support exactly in the node of the second vibration mode will make the beam converge with that mode, which is a higher improvement.

The application of this method to electronic equipment is immediate. Once the supports necessary to keep control of both the position and the alingment are placed, any other additional support should be place following this procedure.

Example

A 97 x 24 x 1.6 mm PCB is made of FR-4, and it is modelled with a density of 3487 kg / m^3 to simulate the effect of the electronic components. It is clamped in the two shorter sides, and its first five natural frequencies are:

Resonant frequency (Hz)
406.22
519.86
1120.6
1293.9
1933.0

It is desired to obtain a first resonant frequency over 500 Hz by adding new supports. The first step is to obtain the first vibration modes of the structure.

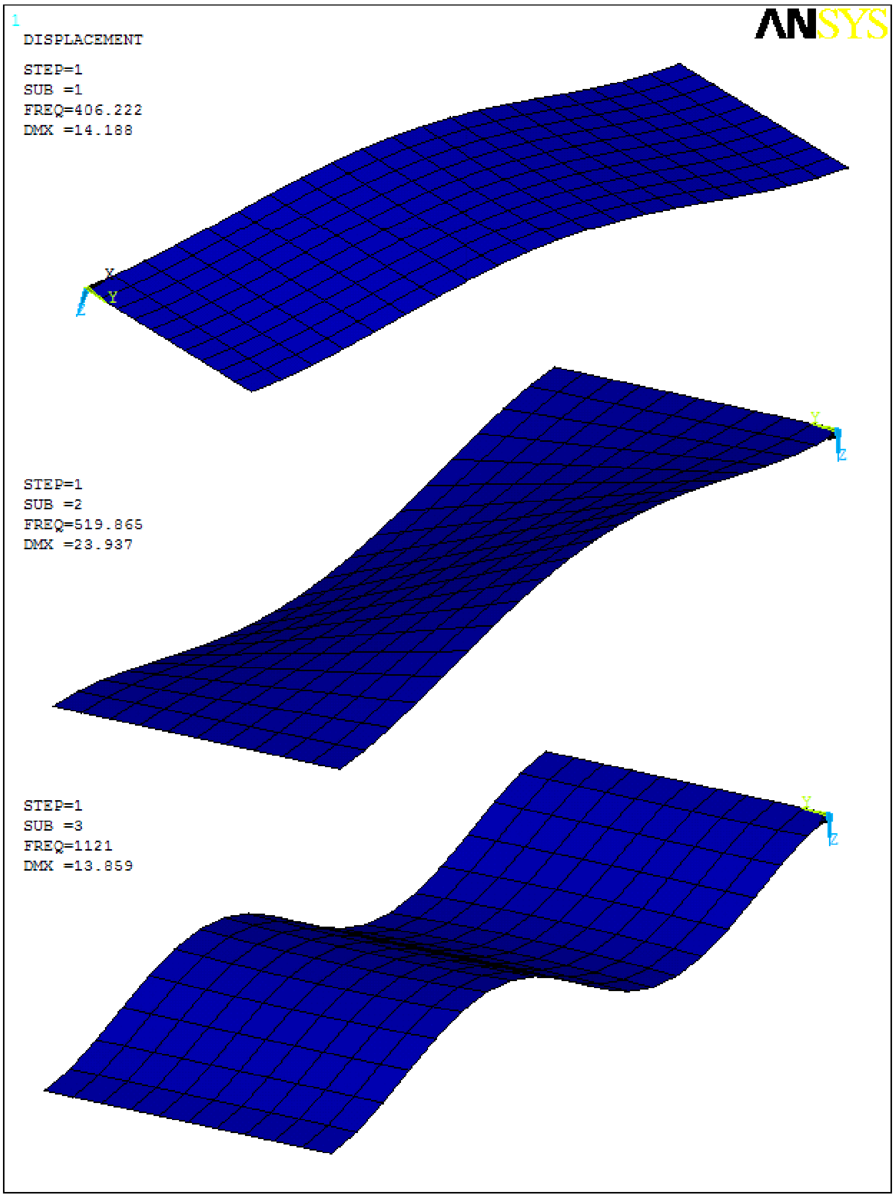

Fig. 7.6.- Vibration modes of the board

Since the third resonant frequency is higher than 500 Hz, the objective is to prevent the two first modes. It is easy to see that the desirable addition will be a support all along the transversal middle line of the board, which has zero displacement in the third mode. The first natural frequencies obtained doing so are now

Resonant frequency (Hz)
1120.6
1293.9
1636.1

Table 7-2.- Results of first improvement.

With this modification the first two natural modes have been completely prevented, producing an increase of factor 2.75 in the first resonant frequency, which is a substantial improvement.

A modification like that, while relatively small, can however be complicated from a constructive point of view, given that PCBs are usually placed in narrow locations, with great difficulties to implement a support like the former. This should not imply that the method is completely not applicable, since even when only the two endings of the middle line are supported, the natural frequencies of the board are

Resonant frequency (Hz)
1120.6
1293.9
1366.9

Table 7-3.- Results of second improvement.

The two first new modes are virtually the same that with the former modification, so the new solution is technically much simpler and equally effective.

The latest result should not suggest that any approximation to the ideal solution is valid. Simple but careful analysis should be done to verify the effectiveness of the solution finally used.

Ribs

The use of ribs is a classic stiffening method in the field of electronic design. They are usually thin and made of steel or other metallic elements, and cemented or soldered to the copper of the PCB. They are probably of little effect in the small boards usually found in hand tools, but when applied properly in a large PCB, they can be virtually divide it into independent sections, producing a high increase of the resonant frequency.

The analysis of the actual effect of the rib would depend on the configuration of the PCB and the method used to fasten it to the board. It can be done using standard structural analysis, and more details should be easily found in electronic design books, since it is a well known technique.