

# Low Frequency Acoustics of Open Plan House

By Bohdan Raczynski

Architectural style of today reflects our relaxed lifestyle. As a result of this, open layout of several interconnecting rooms or areas is very common. This type of dwellings allow the air and sound to freely flow from one room to another (no doors) and as such, one large, complex shape cavity is created. An example of such complex cavity is shown below. Here we have a floor plan of a house comprising of an entry hall, sitting room to the left, formal dining room to the right, and kitchen and living room further down the hall. Our task is to determine lowest (<100Hz) modal frequencies of the dwelling and pressure pattern distribution for those frequencies. For the purpose of analysis, it was assumed that walls absorb negligible amount of acoustic energy.

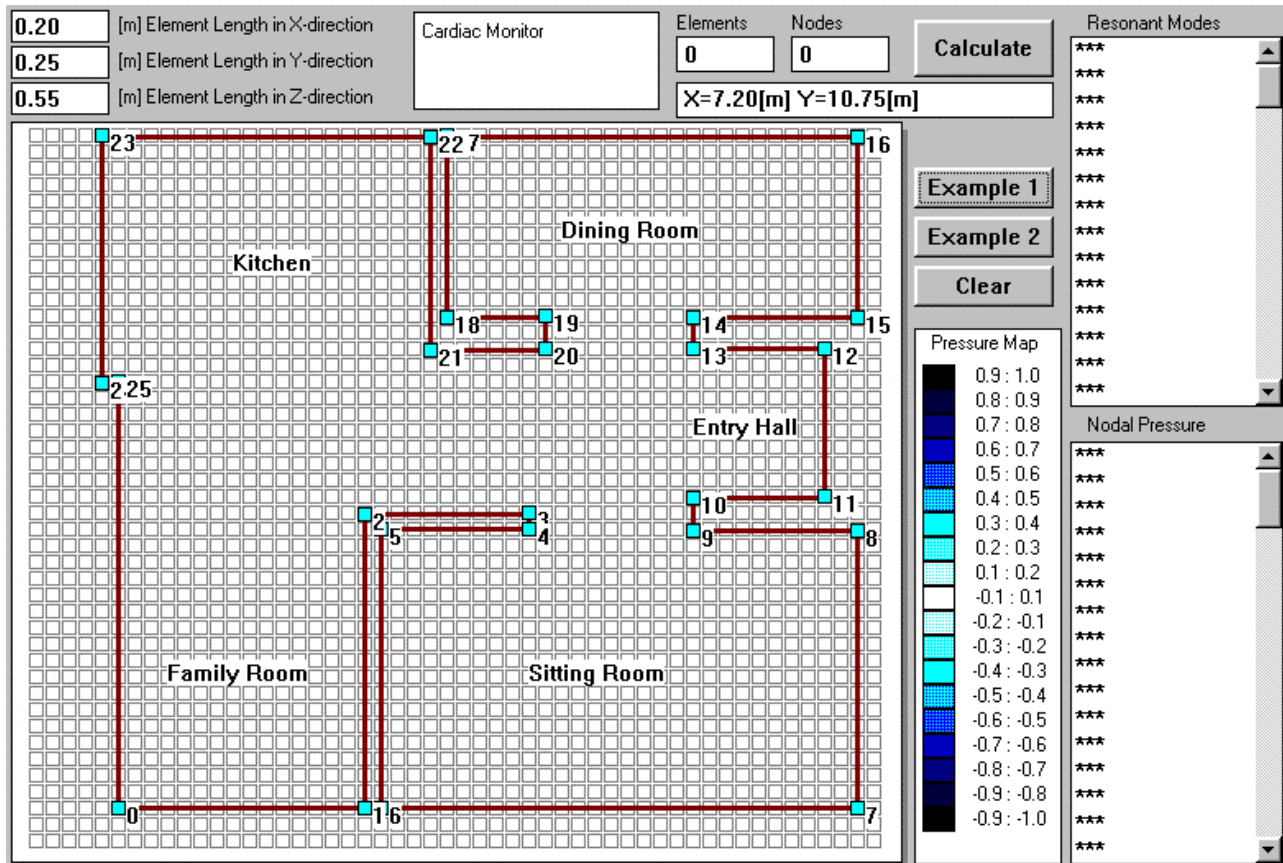


Figure 1. Floor plan of the house to be analysed.

## Method of Analysis

The longest distance between the most distant walls is typically much larger than the distance between floor and ceiling. Therefore the lowest modes will be tangential modes - these are modes NOT involving floor or ceiling reflections. It makes sense to analyze tangential modes only in X-Y (horizontal) plane using Tangential Modes Technique.

This way, program capacity for handling 1350 nodes can be allocated to only one layer of nodes, so that high nodal density and resulting high accuracy of analysis can be attained. As one can easily observe, the final process adopted by the program takes into account three somewhat conflicting factors: (1) maximum number of nodes, (2) processing capacity of the computer and (3) CAD mechanism for entering geometry of your open space living area into the computer. In order to keep FEM processing time within 30min limit (on Pentium™ 200MHz machine), the number of nodes has been limited to 1350 for both types of analysis; tangential or oblique.

If the oblique analysis is required, the 1350 nodes are allocated to 7 layers of nodes (6 layers of elements) offering 192 nodes per layer. Delaunay Triangulation process is employed to subdivide the floor plan into a number of small triangular areas. The triangles are considered to be linear elements with 3 nodes - one node at each vertice. In the next step, a 6-node "wedge" elements are formed by adding identical triangle offset in vertical direction (Z-axis).

The height of this 3D "wedge" element is determined by the user from the available data field. In order to calculate the height of the element, you need to measure the floor-wall distance and divide it by 6. For example: distance between floor and wall is 3.30 meter. Therefore the height of the "wedge" element is  $3.30/6 = 0.55$  meter. The smallest increments (resolution) along the X and Y axes is also determined by the user. This way, a large and complex floor plan may include thin walls in the model - this feature is essential in modeling complex floor plan of the house.

The Delaunay Triangulation process is fully automated and only requires that the you provide the floor plan as the input. This is accomplished by drawing the perimeter of the area being analyzed as a Planar Straight Line Graphic (PSLG). The graph must be entered in an anticlockwise direction. Now, we can review the critical elements of the whole analytical process.

## 1. Vibration - basics

One of the types most common types of analysis for structures are: (1) the natural frequency analysis or eigenvalue analysis and (2) steady state response of the structure to a harmonic force at a selected frequency. We are going to apply both method to analyze a "listening room"acoustics. The general equation of motion describes the balance required for all inertial and damping forces in the equation.

$$M[] * \frac{\partial^2 D[]}{\partial t^2} + C[] * \frac{\partial D[]}{\partial t} + K[] * D[] = F[]$$

Where:

M[][] represents the structure mass matrix,

C[][] is the structure damping matrix,

K[][] is the structure stiffness matrix,

F[] is time varying nodal load vector and

D[] is the nodal displacement vector with its acceleration and velocity.

## 2. Eigenvalue Analysis

Assuming, that all nodes vibrate in a sinusoidal manner with a peak displacement  $P$ , we can write:

$$D[] = P * \sin(\omega t)$$
$$\frac{\partial D[]}{\partial t} = P \omega \cos(\omega t)$$
$$\frac{\partial^2 D[]}{\partial t^2} = -P \omega^2 \sin(\omega t)$$
$$\omega = 2\pi f \quad j = \sqrt{-1}$$

Also, we shall assume no damping of the structure, which implies  $C[][] = 0$  and no external forces, so that  $F[] = 0$  as well. The general equation of motion simplifies now to:

$$(K[][] - k^2 M[]) P[] = 0$$
$$k = \frac{\omega}{c}$$

The above equation can now be solved as a standard eigenvalue and eigenvector problem. Theoretically, we imply that if the structure is deformed into a mode and then released, it would continue to vibrate indefinitely due to the absence of damping. However, practical structures always have some damping, so that the vibrations will eventually decay.

## 3. Frequency Response Analysis

In order to model harmonic acoustic behavior of an enclosed space with a sound source, the following equation is sufficient:

$$(K[][] - k^2 M[] + j \omega C[]) p[] = j \rho_0 \omega v[]$$

Where  $K[][]$  is called acoustic stiffness matrix,  
 $M[][]$  is the acoustic mass matrix,  
 $C[][]$  is the damping of the system,  
 $p[]$  is the sound pressure vector,  
 $v[]$  is the excitation vector in cubic meters per second,  
 $f$  is the test frequency and  
 $c$  is the speed of sound.

For the source of the sound one could use a “point source”, which is convenient, as it can be located in any of the mesh nodes. Larger sources would have to be accounted for as a part of the room boundary. Mathematically, the problem now reduces to assembly of the  $K[][]$  and  $M[][]$  matrices

and inverting the expression in the brackets. This way, vector  $p[]$  can be found for any frequency and location of the sound source represented by excitation vector  $v[]$ . Solving this equation for different discrete input frequencies determines the frequency response of the structure. When the test frequencies are equal to the natural modes, the solution algorithm fails due to division by zero.

#### 4. Stiffness and Inertia Matrixes

Derivation of stiffness and inertia matrixes is beyond the scope of this article, however, I would like to describe briefly the elements involved. Our interest will focus on a volume  $V$  enclosed by a surface  $S$ . The surface is acoustically hard, so that normal velocity on the surface is zero.

$$\frac{\partial p}{\partial n} = 0$$

Under those conditions the pressure  $p$ , must satisfy the wave equation

$$\nabla^2 p + \left( \frac{\omega^2}{c^2} \right) p = 0$$

Variational principle, equivalent to the above equations can be prescribed as follows:

$$\delta \int_V \frac{1}{2} \left[ (\nabla p)^2 - \left( \omega^2 / c^2 \right) p^2 \right] dV = 0$$

In order to maintain continuous pressure between the nodes, an 8-node “brick” elements can be used to fill the volume  $V$ . Elements  $k[][]$  of the stiffness matrix  $K[][]$  can be expressed as follows:

$$k_i[][] = \int_V B_i^T [] B_i [] dV$$

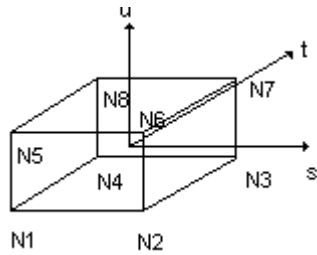
Where:

$$B_i = \begin{bmatrix} \frac{\partial N_i}{\partial x} \\ \frac{\partial N_i}{\partial y} \\ \frac{\partial N_i}{\partial z} \end{bmatrix}$$

of the inertia matrix  $M[][]$  is expressed as follows:

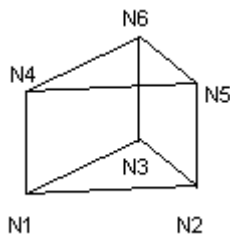
$$m_i[][] = \int_V N_i^T [] N_i [] dV$$

Both elemental matrixes for 8-node element are 8x8 in size. Vector  $N[]$  represents eight linear shape functions defined as below:



$$\begin{aligned}
 N1 &= (1 - s)(1 - t)(1 - u)/8 \\
 N2 &= (1 + s)(1 - t)(1 - u)/8 \\
 N3 &= (1 + s)(1 + t)(1 - u)/8 \\
 N4 &= (1 - s)(1 + t)(1 - u)/8 \\
 N5 &= (1 - s)(1 - t)(1 + u)/8 \\
 N6 &= (1 + s)(1 - t)(1 + u)/8 \\
 N7 &= (1 + s)(1 + t)(1 + u)/8 \\
 N8 &= (1 - s)(1 + t)(1 + u)/8
 \end{aligned}$$

For the 6-node “wedge” element, vector  $N[]$  represents six linear shape functions defined as below:



$$\begin{aligned}
 N1 &= s(1 - u)/2 \\
 N2 &= t(1 - u)/2 \\
 N3 &= (1 - s - t)(1 - u)/2 \\
 N4 &= s(1 + u)/2 \\
 N5 &= t(1 + u)/2 \\
 N6 &= (1 - s - t)(1 + u)/2
 \end{aligned}$$

With the above in mind, the variational principle can be expressed in matrix form as:

$$(K[[]] - k^2 M[[]]) P[] = 0 \qquad k = \frac{\omega^2}{c}$$

Due to complexity of the resulting expressions for element  $k[][]$  and  $m[][]$  they are best evaluated by using numerical integration techniques. Integration can be facilitated by Gauss integration scheme, requiring integration limits of +1 and -1. We can change integration from x,y,z space to s,t,u space with the help of Jacobian  $J[][]$  of the transformation resulting immediately in the volume element  $dV$  being expressed as follows:

$$dV = |J| ds dt du$$

Where  $|J|$  is the determinant of Jacobian matrix  $J[][]$ .

### 5. Jacobian of the Transformation

$$\begin{bmatrix} \frac{\partial N_i}{\partial s} \\ \frac{\partial N_i}{\partial t} \\ \frac{\partial N_i}{\partial u} \end{bmatrix} = [J] \begin{bmatrix} \frac{\partial N_i}{\partial x} \\ \frac{\partial N_i}{\partial y} \\ \frac{\partial N_i}{\partial z} \end{bmatrix}$$

Where  $J[]$  is a **Jacobian** of the transformation expressed as follows:

$$[J] = \begin{bmatrix} \frac{\partial N_1}{\partial s} & \frac{\partial N_2}{\partial s} & \frac{\partial N_3}{\partial s} & \frac{\partial N_4}{\partial s} & \frac{\partial N_5}{\partial s} & \frac{\partial N_6}{\partial s} & \frac{\partial N_7}{\partial s} & \frac{\partial N_8}{\partial s} \\ \frac{\partial N_1}{\partial t} & \frac{\partial N_2}{\partial t} & \frac{\partial N_3}{\partial t} & \frac{\partial N_4}{\partial t} & \frac{\partial N_5}{\partial t} & \frac{\partial N_6}{\partial t} & \frac{\partial N_7}{\partial t} & \frac{\partial N_8}{\partial t} \\ \frac{\partial N_1}{\partial u} & \frac{\partial N_2}{\partial u} & \frac{\partial N_3}{\partial u} & \frac{\partial N_4}{\partial u} & \frac{\partial N_5}{\partial u} & \frac{\partial N_6}{\partial u} & \frac{\partial N_7}{\partial u} & \frac{\partial N_8}{\partial u} \end{bmatrix} \begin{bmatrix} x1 & y1 & z1 \\ x2 & y2 & z2 \\ x3 & y3 & z3 \\ x4 & y4 & z4 \\ x5 & y5 & z5 \\ x6 & y6 & z6 \\ x7 & y7 & z7 \\ x8 & y8 & z8 \end{bmatrix} =$$

$$\begin{bmatrix} \sum_{i=1}^8 \frac{\partial N_i}{\partial s} x_i & \sum_{i=1}^8 \frac{\partial N_i}{\partial s} y_i & \sum_{i=1}^8 \frac{\partial N_i}{\partial s} z_i \\ \sum_{i=1}^8 \frac{\partial N_i}{\partial t} x_i & \sum_{i=1}^8 \frac{\partial N_i}{\partial t} y_i & \sum_{i=1}^8 \frac{\partial N_i}{\partial t} z_i \\ \sum_{i=1}^8 \frac{\partial N_i}{\partial u} x_i & \sum_{i=1}^8 \frac{\partial N_i}{\partial u} y_i & \sum_{i=1}^8 \frac{\partial N_i}{\partial u} z_i \end{bmatrix}$$

Now, the derivatives of the shape functions in respect to x,y and z coordinates can be calculated by inverting the Jacobian matrix  $J[][]$ .

$$\begin{bmatrix} \frac{\partial N_i}{\partial x} \\ \frac{\partial N_i}{\partial y} \\ \frac{\partial N_i}{\partial z} \end{bmatrix} = [J]^{-1} \begin{bmatrix} \frac{\partial N_i}{\partial s} \\ \frac{\partial N_i}{\partial t} \\ \frac{\partial N_i}{\partial u} \end{bmatrix}$$

## 6. Gaussian Integration

Example of Gaussian integration scheme for the 8-node "brick" element and N=2 is being used and the location of the points together with the associated weights is shown below.

Node ----- Weight ----- s ----- t ----- u-----

A1 -----	1.0	-0.57735	-0.57735	-0.57735
A2 -----	1.0	0.57735	-0.57735	-0.57735
A3 -----	1.0	0.57735	0.57735	-0.57735
A4 -----	1.0	-0.57735	0.57735	-0.57735
A5 -----	1.0	-0.57735	-0.57735	0.57735
A6 -----	1.0	0.57735	-0.57735	0.57735
A7 -----	1.0	0.57735	0.57735	0.57735
A8 -----	1.0	-0.57735	0.57735	0.57735

## 7. Assembly of System Matrixes

Local nodes of individual elements are always numbered as 1 to 8 for "brick" elements and 1 to 6 for "wedge" elements. However, when the complete mesh has been generated, coordinates of all nodes and global numbering scheme for all components become available, and at this point of time we know which elements share the same nodes. It is therefore possible to assign the global node numbering and the global element coordinates to the "local" nodes of each element. The process is called mapping and can sometimes be "tricky".

Assembly of the stiffness  $K[][]$  and inertia  $M[][]$  system matrixes involves calculation of derivatives of the shape functions  $N_i$  and can be quite complex process. Also, local node numbers of each element need to be mapped onto the global node numbering scheme.

## 8. Solution of Eigenvalue problem

Eigenvalue problem can be solved using Householder Algorithm and Tridiagonal QLI Algorithm. However, this process requires symmetrical matrixes for its input. Assuming that matrixes  $K[][]$  and  $M[][]$  are symmetric of order  $n$ , the eigenvalue problem can be stated as:

$$(K[][] - \lambda M[][])X[] = 0$$

The  $X[]$  is a vector called eigenvector and  $\lambda$  is a scalar called eigenvalue. The above can be converted to standard eigenvalue problem by pre-multiplying both sides by inverted matrix  $M[]$ .

$$A[] = M^{-1}[] * K[]$$

However, matrix  $A[]$  is often non-symmetrical despite  $M[]$  and  $K[]$  being both symmetrical. If the method adopted for solving the eigenvalue problem requires  $A[]$  to be symmetrical, the following procedure may be used:

1. Using Choleski method, decompose  $M[]$ , so that only upper triangular  $U[]$ , matrix will be used:

$$M[] = U^T[] * U[]$$

2. Formulate  $A[]$  as:

$$A[] = (U^T[])^{-1} * K[] * (U^{-1}[])$$

Matrix  $A[]$  is now symmetrical and suitable for Householder algorithm and Tridiagonal QLI Algorithms.

## 9. Generating Mesh

Generating mesh is a large subject in its own right. Mesh is the way to translate information about the physical parameters of the object being analysed into a set of numbers understood by the program. Vastly simplified version is presented here, however, good results can be achieved for a number of applications.

## 10. Delaunay Triangulation Algorithm

The process commences with the Planar Straight Line Graphic (PSLG) description. In simple terms, the PSLG it is the closed contour (perimeter) encompassing the area to be triangulated. The coordinates (nodes) are entered as a graph oriented counterclockwise, and using mouse pointer. The next step is the Delaunay Triangulation process. It needs to be noted, that final results of the triangulation is expected to be a uniform mesh with a predetermined maximum triangle size, suitable for Finite Element Method mesh.



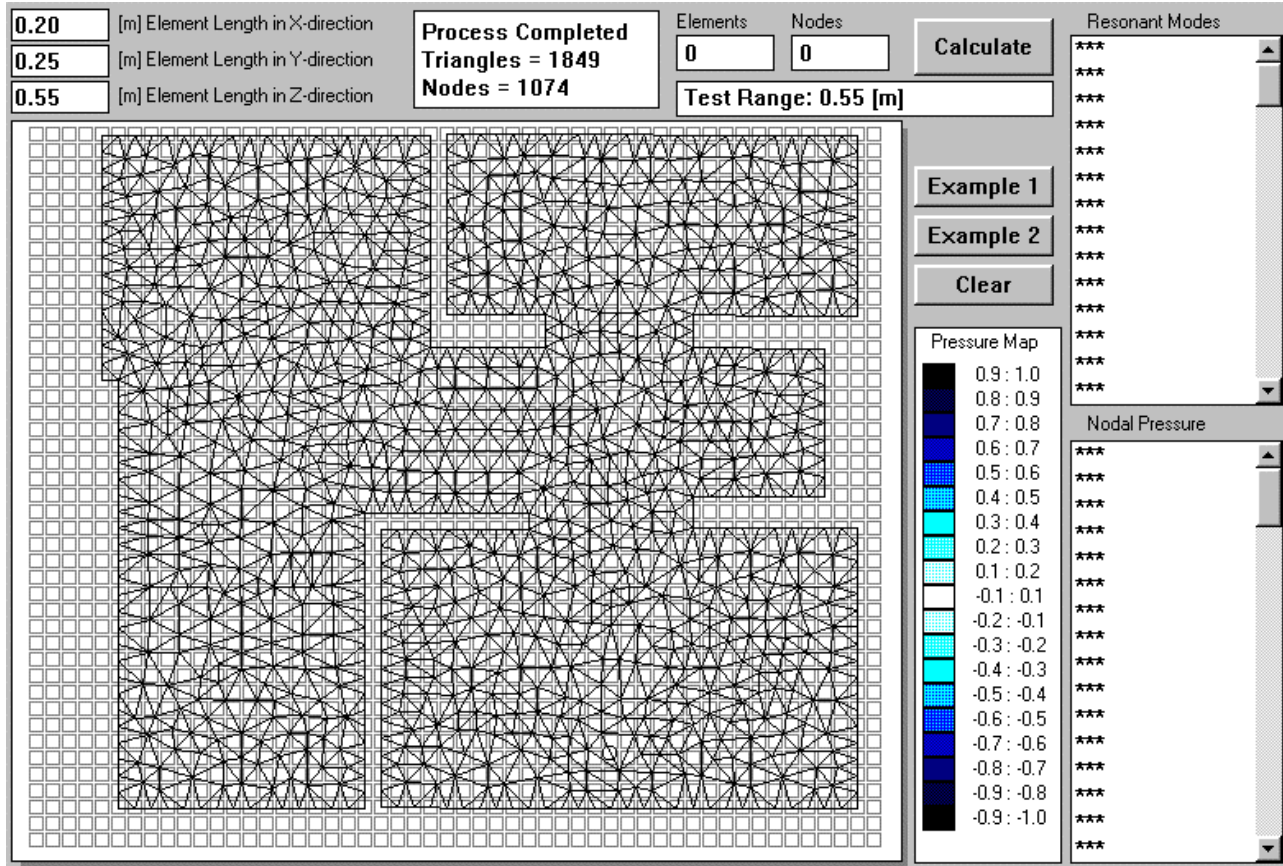


Figure 2. Delaunay Triangulation processor finished.

Delaunay triangulation of a point set  $P$ , is a triangulation with the property that no point in the point set  $P$  falls in the interior of the circumcircle (a circle that passes through all three vertices) of any triangle in the triangulation.

Initially, the point set  $P$  is the set of nodes of the PSLG, as determined by the user. The point set  $P$  is then immediately expanded by splitting each edge of the PSLG into smaller sections whose length is determined by the maximum triangle size, as specified by the user. The resulting point set  $P$  is then triangulated. At this moment, there is no points inside the PSLG, therefore the first pass of the triangulation produces very long and “skinny” triangles. The record of the longest edge  $L$ , is maintained for future reference.

In the next loop, the longest edge is compared against the required triangle size and if a triangle with a longer side is found, the triangle is destined to be split by inserting an extra point into the offending triangle. Once all triangles are checked and all extra points are inserted, the whole point set  $P_{new}$  is then re-triangulated. The process continues until there is no more extra points to be inserted – see Figure 1.

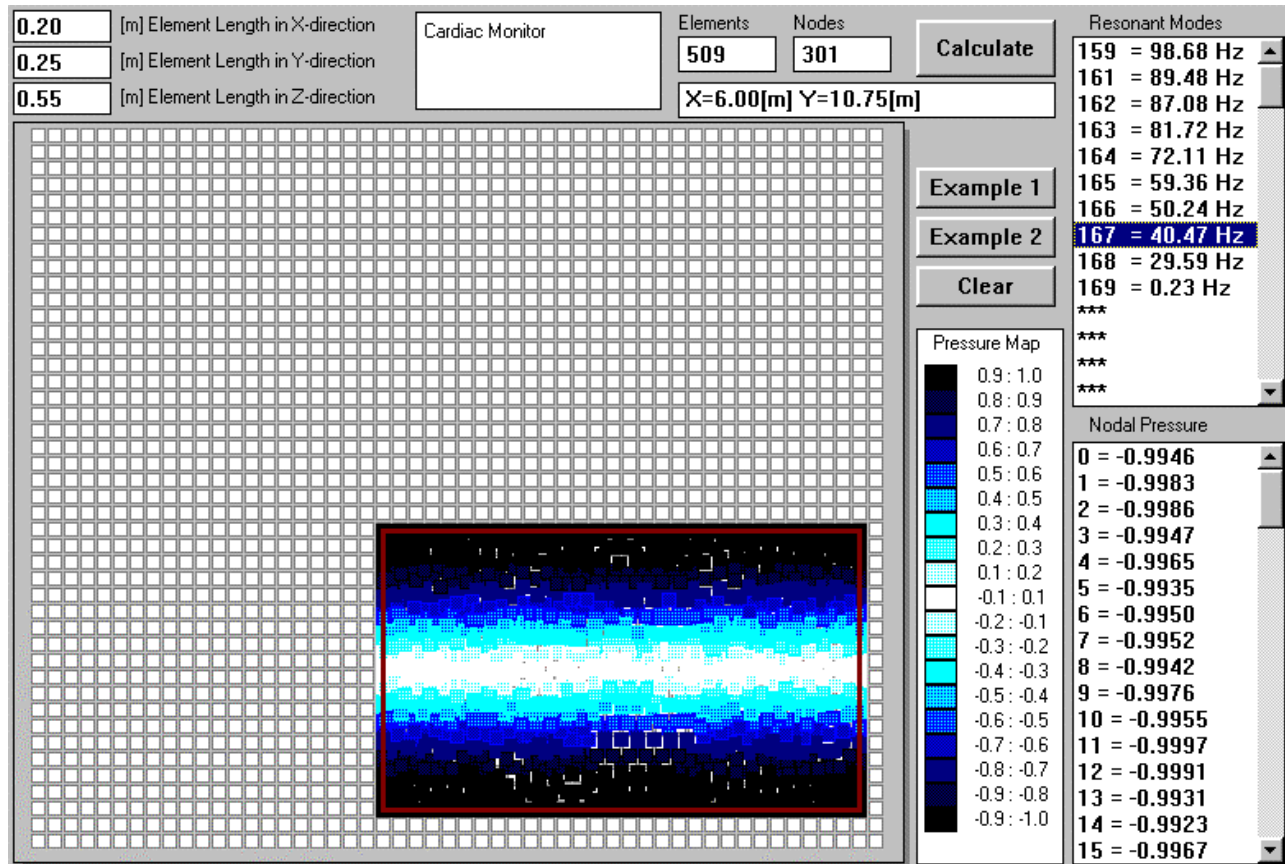


Figure 3. Fully isolated sitting room

Each triangulation pass produces “denser” set of points from which Delaunay triangles are build. However, it is wasteful to check the whole point set  $P_{new}$  for the purpose of building Delaunay triangles. Only the points located no further from the current “seed” edge than longest edge  $L$ , need to be considered for potential new Delaunay triangle.

The following can be noted:

1. The algorithm maintains a list of all edges that were already used as “seeds” for constructing Delaunay triangles. These edges are never used again in a current pass.
2. The Delaunay triangulation uses **Orientation Test** - Check whether or not a point lies to the left of a vector. If point “C” lies to the left of the vector AB, the cross product of the vectors  $AB * AC$  will be larger than 0. If points “A”, “B”, and “C” are colinear, the cross product of these two vectors will be equal to 0. If point “C” lies to the right of the vector AB, the cross product of the vectors  $AB * AC$  will be smaller than 0.
3. Also, the Delaunay triangulation uses **In Circle Test** - Check whether or not a point lies inside the circumcircle of a triangle. The algorithm uses the InCircle Test to make sure that there is no point that is located inside the circle which passes through the three points it found. These three points can be now connected to form a Delaunay triangle.
4. Finally, the algorithm checks for “crossed edges” of triangles.

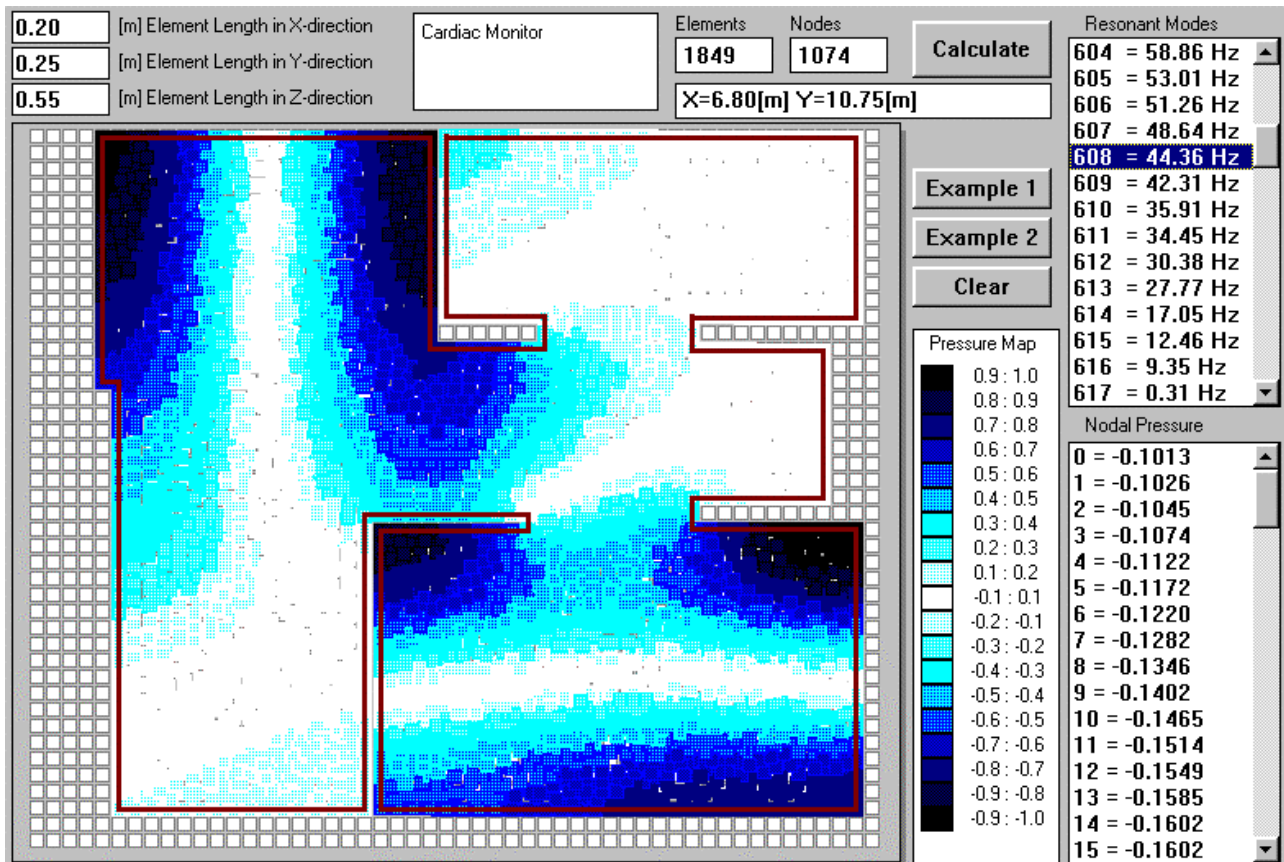


Figure 4

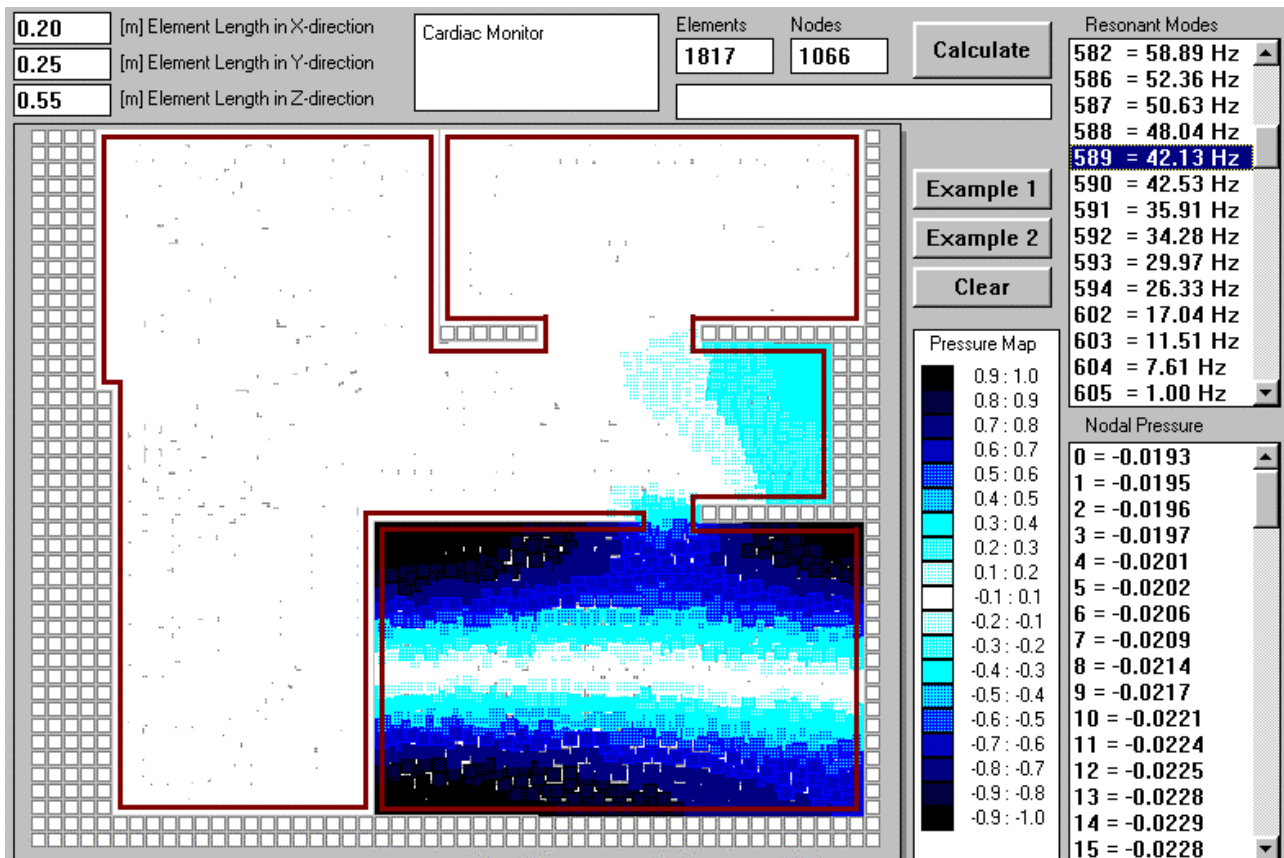


Figure 5

## Results of the Analysis

The lowest 11 modes in our example are all tangential (see table below). Since larger number of nodes per layer can be allocated to tangential scheme, it follows, that modal frequencies calculated this way would be more exact. Finally, graphical presentation of the pressure patterns is more readable using tangential model.

Tangential [Hz] ----- Oblique [Hz]

9.35	-----	9.76
12.46	-----	13.19
17.05	-----	17.27
27.77	-----	29.23
30.38	-----	30.62
34.45	-----	34.70
35.91	-----	36.36
42.31	-----	42.97
44.36	-----	44.85
48.64	-----	49.20
51.26	-----	51.79
51.95		oblique
52.80		oblique

and so on.....

Since the modal frequencies are solely determined by the geometry of the enclosed space being analysed, it can be easily observed, that better isolated rooms having different sizes often maintain many of their own fundamental modal frequencies - despite being interconnected with other rooms (cavities). Our sitting room for example, maintains its own lowest tangential modes: 30.38Hz and around 60.0Hz along the longer dimension (x-axis). There is no openings in the opposite walls along the x-axis and the modal frequencies and pressure distribution pattern along this wall is quite similar to the pressure pattern of the isolated room.

However, there is an opening in the opposite walls of the sitting room along y-axis. The fundamental mode along this wall is affected by the existence this passage. The modal frequency is shifted from 40.47Hz to 44.36Hz and pressure pattern is distorted. Please compare the two images shown on Figure 3 and Figure 4, where the influence of the whole house on the sitting room pressure pattern is quite noticeable.

Figure 3 shows fundamental mode of fully isolated sitting room along the y-axis (4.25meters).

Figure 4 shows fundamental mode of the sitting room connected to the rest of the house, along the y-axis (4.25meters).

However, if the opening between the sitting room and the rest of the house was made smaller (by narrowing the passage from 2meter wide to 0.6meter wide), the sitting room becomes much more isolated at the lowest modal frequencies from the rest of the house. The previously recorded modal frequency of 44.36Hz is now reduced to 42.12Hz and the pressure pattern is almost identical to that of the isolated room. Please review the image shown on Figure 5.

Kitchen + family room (being practically one cavity) also maintains its own fundamental resonance and the same can be stated about the dining room. The pressure pattern for frequencies (below 40-50Hz in our example), in each case is almost the same as one would expect from the room being analyzed as a completely isolated and enclosed cavity.

Higher modes (above 40-50Hz in our example) do not follow this mechanism. As soon as their wavelengths become smaller and comparable with the dimensions of interconnecting passages, the whole enclosed space becomes one, very complex shape resonating cavity. It is also important to observe where the passages are and correlate their location with the pressure patterns. It can be observed, that modal pressure patterns are more distorted if the opening happens to be situated on one of the opposite walls responsible for this mode.

In this higher frequency range, pockets of high/low pressure are found scattered all over the space and pressure pattern become very complex. Also, modal density (number of modes per frequency band) for this particular geometry is quite high.

Frequency Band [Hz]	Modal density(whole house)	Modal density(isolated sitting room)
90-100	13	5
80-90	12	4
70-80	10	3
60-70	8	1
50-60	10	3
40-50	4	2
30-40	3	0
20-30	1	1
10-20	2	0
0-10	1	0

The table above clearly shows the dramatic increase in modal density as a result of the sitting room being connected with the rest of the house.

**Observations**

1. When analyzing the acoustics of open-living house, one may expect to find three overlapping frequency ranges.



2. The lowest modal frequencies are determined by the distance between most separated walls. Typically, these frequencies are in the sub-sonic range or less than 15Hz. In the example shown, the sitting room is involved in the longest distance path, but in general the room we are investigating may not be involved.

3. In the “low-bass” frequency range, individual cavities (rooms) are exhibiting many of their own modes and if the interconnecting passage is small, the rooms can often be treated or analyzed as a free standing, isolated rooms.

4. In the “high-bass” frequency range, the half-wavelength becomes comparable with the dimensions of the passages and the whole open-living house becomes single resonating cavity of a very complex shape

5. The degree of separation between a room and the rest of the house is determined by the dimensions of the interconnecting passages. If the cross-sectional dimensions of the passage are much smaller than dimensions of the room, than pressure pattern of the lowest modes of the isolated room will be much less affected by the passage and the existence of rest of the house.

6. The actual modal frequencies and the distribution of the modal pressure pattern in the room connected with the rest of the house is affected by the cross-sectional dimensions of the passage and the location of the passage.

I guess, to some extent I confirmed the obvious - if you live in the “open plan” house, expect the sound to fill all the rooms. However, proper analysis requires more than that. Our task, stated at the start of this article, was accomplished and the example of the analysis given above provides us with:

1. numerical data of actual modal frequencies of quite complex house and.
2. modal pressure distribution patterns for the lowest modes in the whole house.

It should now be easier to visualize, which modes can be dealt with by modifying the acoustics of the sitting room alone and which modes would require modifications to the whole house – in other words, how well (if at all) is your sitting room's modal response is isolated acoustically from the rest of the house.

Alternatively, one can take opposite approach by actually using “room gain” and room modes. Taking advantage of the “room gain” at low frequencies is quite easy – just place the loudspeakers in the corners of the room and you will gain 9-10dB more SPL output in the low frequency range. Secondly, you may also wish to experiment with room modes. For example, your speaker system starts rolling off at  $f_{3dB}=35\text{Hz}$ , but still provides some output at 30Hz. Incidentally, you have a fundamental room mode at 30Hz between opposite walls A and B. You can take advantage of this mode by placing the speakers along wall A and locating your listening area along wall B – the loudspeakers are now placed to fully energize the 30Hz mode. The “room gain” affect and room modal response will combine to produce quite strong output at 30Hz. A word of caution here: you are trading-off increased irregularities in frequency response in other bands for extending very low frequency SPL output - but then again, this may be your ultimate goal.

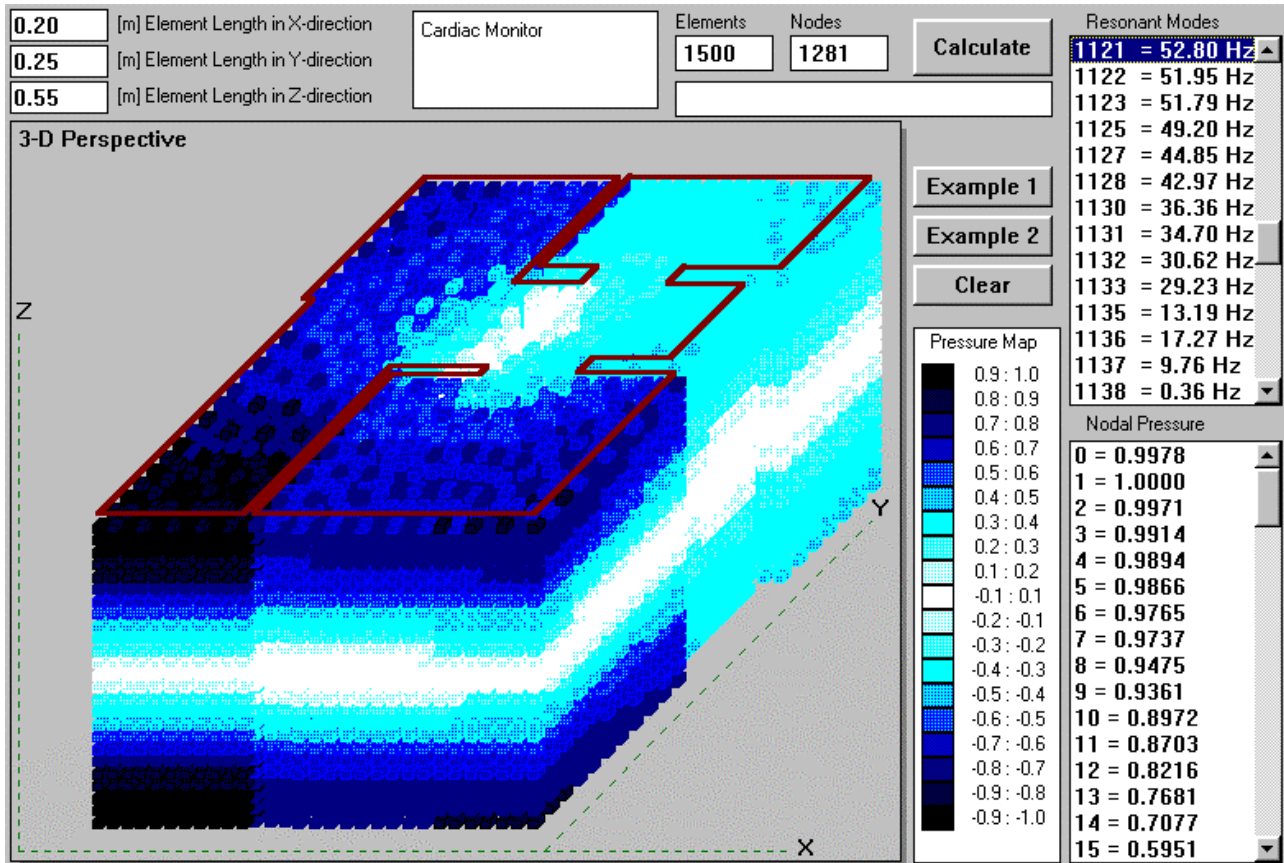


Figure 6

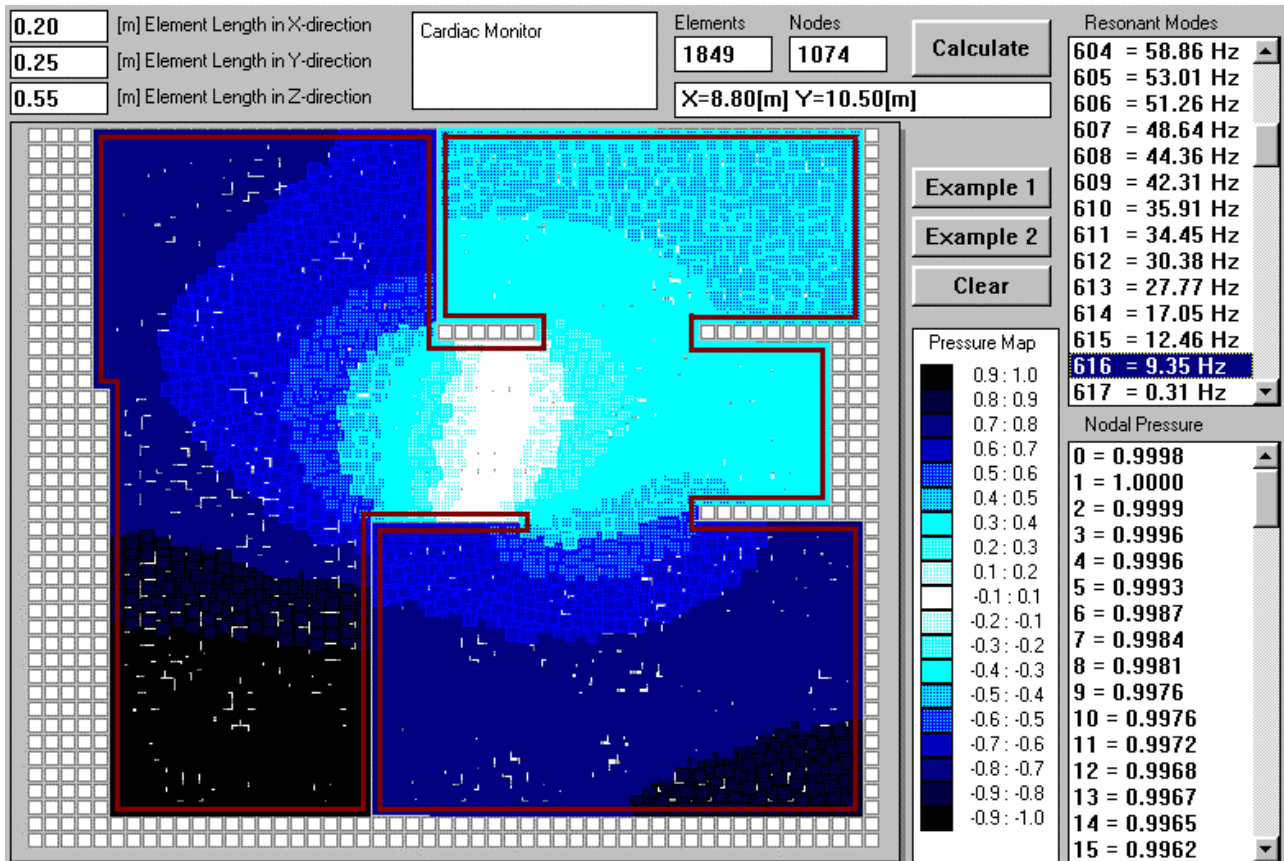


Figure 7

Regardless of your choice of approach, I suggest using the capabilities of the program the following way:

1. Run 3D analysis of your house using 3D Delaunay Modal Analyzer. The output of this process shows all modes existing in your open layout area and enables you to determine which modes are tangential and which are oblique – see Figure 6
2. Run 2D analysis of your house using 2D Delaunay Modal Analyzer. The output of this process shows all tangential modes existing in your open layout area. The number of nodes, the resulting accuracy and presentation is much better than the output of 3D analysis - see Figure 7
3. If the analysis at this point indicates, that the room in-question is sufficiently isolated ( just like in the example above, when the passage was narrowed to 0.6m ) run 3D analysis of your listening room alone using 3D FEM Analyzer. This analyzer is very flexible and can accept very complex geometries for the input - see Figure 8.

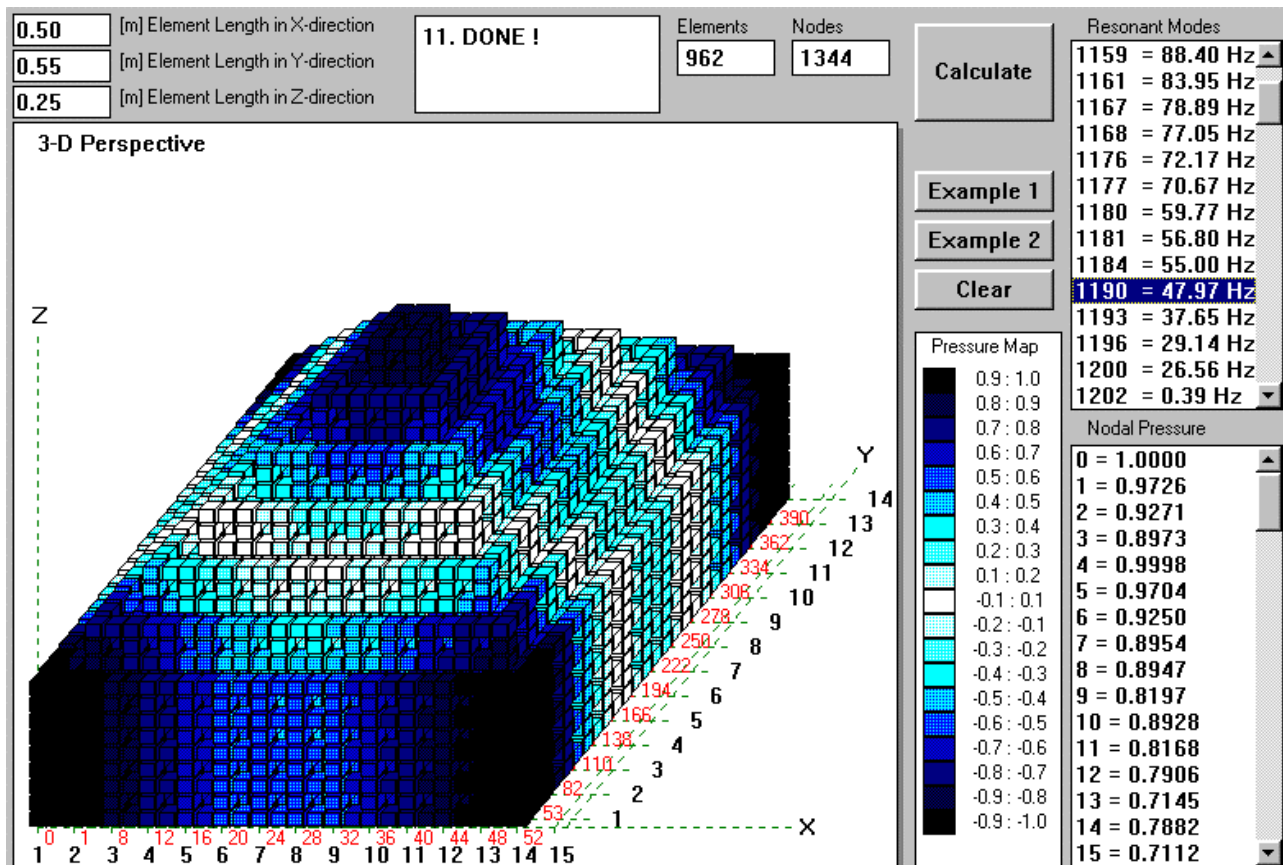


Figure 8

Data generated by all 3 processes are needed to evaluate the complexity of the problem at hand. The analysis presented above heavily depended on the availability of the computer program capable of handling complex floor plans of “open living” interior spaces.



The example chosen, focused on the sitting room interaction with the rest of the house and as the results would indicate that ( except for the first mode along the longest wall ) the room is reasonably well “coupled” to the rest of the house. This could be attributed to the particular geometry of the space being analyzed. It was also observable, that reducing the area of the passage connecting this room with the rest of the house, “uncoupled” the room for higher modes as well.

Finite Element Method has been successfully implemented to accomplish the task because the method lends itself particularly well to complex geometry of the space being analyzed. All essential mathematical components of the whole analytical process have been nominated and briefly explained. Attempt to solve a problem of this nature without a computer would prove to be too formidable task to undertake, if only because of the shape complexity of the volume being analyzed.

Bodzio Software provides computer tools capable of assisting you in a situation like this. Data generated by the computer is intended to assist you in making educated decisions about the loudspeaker-listener interaction in your individual listening environment.