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The Estimation of Vibrational Energy of Two Coupled (Welded) Plates Using Statistical Energy Analysis

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Abstract

This paper deals with a method called Statistical Energy Analysis that can be applied to the mechanical and acoustical systems like buildings, bridges and aircrafts …etc. S.E.A as a tool can be applied to the resonant systems in the circumstances of high frequency or/and complex structure». The parameters of S.E.A such as coupling loss factor, internal loss factor, modal density and input power are clarified in this work ; coupled plate sub-systems and explanations are presented for these parameters. The developed system is assumed to be resonant, conservative, linear and there is an equipartition of energy between all the resonant modes within a given frequency band in a given sub-system. The aim of this work is to find the energy stored in the sub-systems for two coupled (welded) plates in rectangular angle systems and study the effect of changing sub-systems dimensions, the results shows that as surface area of directly driven plates A1 increases energy level of plate 1 increases while a reduction in the energy level of indirectly driven plate (plate 2) is noticed. This is because of the strength of coupling decreases towards the weak coupling condition and this leads to a reduction in the power transferred from plate 1 to plate 2 and consequently a lower energy level for plate 2. In addition the effect of changing the internal loss factor for a range of (0.00001-0.1) causes a reduction of the values of energy level in these sub-systems. because the increasing of internal loss factor values led to the increasing of the material resistance and that will dissipate the energy flow across those sub-systems. A comparison is made between S.E.A models built by FORTRAN program and Finite Element model solved by ANSYS package.

Keywords: *- Type I diabetes, Backstepping, Bergman's model, oral glucose tolerance test.*

1. Introduction

Of the techniques used to predict vibration levels in mechanical structures, are those based on the assumption of idealized mathematical in which the properties of the structures and the excitation are known exactly, and proceed effectively by numerical solutions of the equations of motion. For many structures, however the very detailed physical properties required such a model may not be available, or relevant, for the particular problem at hand. The vibrational behavior of complex structures involves modes high up in the modal series at frequencies of practical interest, and is very sensitive to structural detail. Variations of detail between a structure and its mathematical model, or between different realizations of nominally the same structures, may then account for significant quantitative differences in response. The computational effort involved in direct numerical solution may also be prohibitively large for these structures. Under these circumstances, a more appropriate technique is often one which provides an understanding of broad features of the vibration levels and transmission, given only a relative coarse and uncertain description of the structure and its excitation. Such technique is Statistical Energy Analysis (S.E.A) Ref.[1] and Ref.[4]. The method Statistical Energy Analysis (S.E.A) involves the division of a complex system into a number of inter-connected sub-systems with similar characteristics, for example plates, beams, acoustic spaces etc. It is based on power flow between sub-systems, the power dissipative by damping and the excitation power input within the system of interest Ref.[9].

In this paper, the energy stored in sub-systems is found by using Statistical Energy Analysis, and a comparison is made between analytical solution by using Statistical energy Analysis (S.E.A) and numerical analysis by using Finite Element Method (F.E.M) built by (ANSYS 5.4) . For the case of the two coupled (welded) plates joined along a straight edge, one of the plates is excited and that excitation is harmonic at radian frequency ω and of constant amplitude F_o. The Statistical Energy Analysis gives us the energy in average manner over all the points of the plate and does not give any information about concentration of energy in particular parts of the sub-systems. The Finite Element Method gives the response in any particular point. Therefore, The ANSYS model is built with an application of point harmonic force at different locations on the driven sub-system. Results of the response (energy levels), for a given force location, are predicted at several locations on coupled sub-systems. An average of response at different locations that ensemble averages over the different force locations is obtained.

2. S.E.A Theory

In an energy distribution model the sub-system energies and input powers are related by the following matrix equation,

$$\{\overline{P}_{in}\} = w\,[A]\{\overline{E}\} \qquad \ldots(1)$$

Equation (1) gives the relationship between the power injected matrix $\{\overline{P}_{in}\}$, and the total energy matrix at center frequency of the band $\{\overline{E}\}$, noticing that, $[A]$ is a matrix of energy influence coefficients in the relevant frequency band and the symbol $(\overline{\ })$ indicates time average quantity. In S.E.A a power balance equation is written for each sub-system, so that, for sub-system i,

$$\overline{P}_{in,i} = \overline{P}_{diss,i} + \overline{P}_{trans,ij} \qquad \ldots(2)$$

The power balance equation means that the input power into sub-system i equals to the summation of power dissipated due to damping in sub-system i, $\overline{P}_{diss,i}$ and the power transmitted from sub-systems i to sub-system j, $\overline{P}_{trans,ij}$, due to coupling loss factor between sub-systems.
The equation for the dissipation of power in a subsystem can be written as Ref. [5] :-

$$\overline{P}_{diss,i} = w h_i \overline{E}_i \qquad \ldots(3)$$

Where, h_i is the damping loss factor of sub-system i, $\overline{E}_i$ represents the total stored energy in sub-system i and w is the center frequency of the band of interest. The equation for the net power transmitted from sub-systems i to j is:-

$$\overline{P}_{trans,ij} = w h_{ij} \overline{E}_i - w h_{ji} \overline{E}_j \qquad \ldots(4)$$

Where, h_{ij}, h_{ji} is the coupling loss factor from sub-systems i to j and from sub-systems j to i respectively and it depends on the type of connection between sub-systems, the material properties, the dimensions of the system and the center frequency of the band, ω (rad/sec) Ref. [5].

The power flow relationship of a structure consisting of two sub-systems is shown in fig.(1).

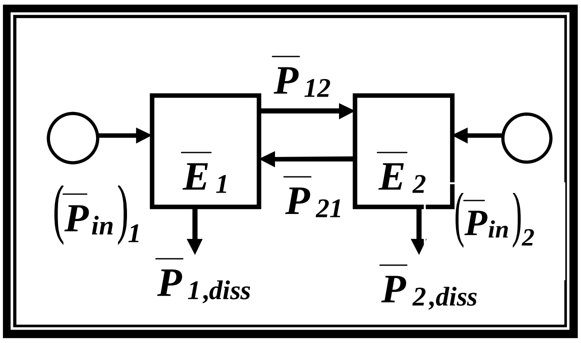

Fig. 1.Basic S.E.A model Ref.[3].

Where $\overline{E}_1$ and $\overline{E}_2$ are the total stored energy of sub-systems (1) and (2) respectively, $\overline{P}_{in}$ is the power injected into the first sub-system, $\overline{P}_{1,diss}$ and $\overline{P}_{2,diss}$ are the power dissipated in sub-systems due to internal damping and $\overline{P}_{12}$ and $\overline{P}_{21}$ are the power transmitted between sub-systems.

The equation for the power flows between the sub-systems 1 and 2 under typical conditions is expressed as follows Ref. [3] : -

$$\left(\overline{P_{in}}\right)_1 = w h_1 \overline{E}_1 + w\left(h_{12}\overline{E}_1 - h_{21}\overline{E}_2\right)$$
$$\left(\overline{P_{in}}\right)_2 = w h_2 \overline{E}_2 + w\left(h_{21}\overline{E}_2 - h_{12}\overline{E}_1\right) \qquad \ldots(5)$$

So the coupling loss factors must be related by the consistency relation Ref. (5),

$$n_i h_{ij} = n_j h_{ji} \qquad \ldots(6)$$

Where n_i and n_j are the modal densities of sub-systems i and j respectively. So, the equation of input power injected into sub-systems can be written as Ref. [10]:-

$$\left(\overline{P_{in}}\right)_1 = wh_1\overline{E}_1 + wh_{12}n_1\left(\frac{\overline{E}_1}{n_1} - \frac{\overline{E}_2}{n_2}\right)$$

$$\left(\overline{P_{in}}\right)_2 = wh_2\overline{E}_2 + wh_{21}n_2\left(\frac{\overline{E}_2}{n_2} - \frac{\overline{E}_1}{n_1}\right) \quad \ldots(7)$$

A frequently encountered situation is when one sub-system is directly driven by an external force and the other sub-system is driven only through the coupling. The equation then reduced to Ref. (10),

$$\left(\overline{P_{in}}\right)_1 = wh_1\overline{E}_1 + wh_{12}n_1\left(\frac{\overline{E}_1}{n_1} - \frac{\overline{E}_2}{n_2}\right)$$

$$0 = wh_2\overline{E}_2 + wh_{21}n_2\left(\frac{\overline{E}_2}{n_2} - \frac{\overline{E}_1}{n_1}\right) \quad \ldots(8)$$

Equation (7) and (8) for the two sub-system results cannot be extended to more than two sub-systems, although the power flow energy difference relationship is true even when the two sub-systems are connected through a third un damped sub-system Ref. [11]. By expanding our way of thinking in the previous section, it is possible to formalize, in the same way, the power flow relationships of a structure consisting of multiple sub-systems as shown in fig. (2).

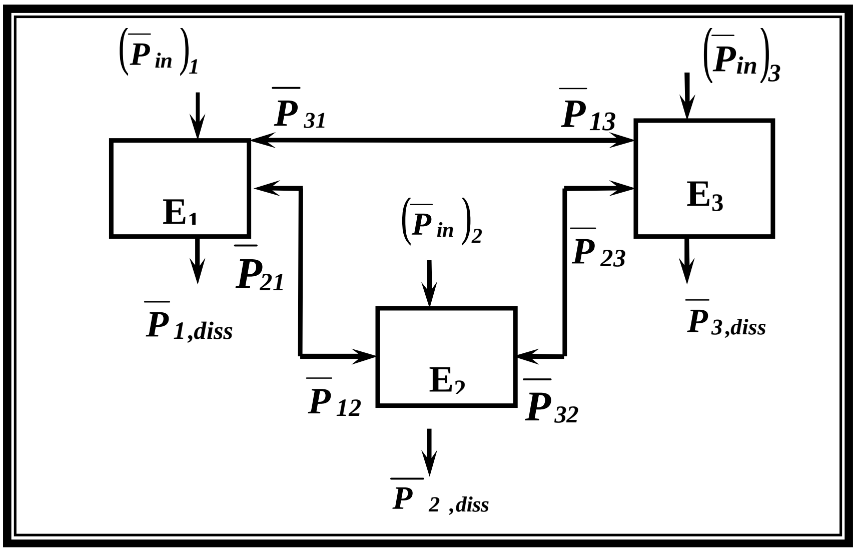

Fig. 2. Three coupled sub-systems Ref[11].

The power flow equation for a structure made up of N sub-systems is expressed with the following equation in matrix form,

$$w\begin{bmatrix} \left(h_1 + \sum_{i\neq 1}^{N} h_{1i}\right)n_1 & -h_{12}n_1 & \cdots & -h_{1N}n_1 \\ -h_{21}n_2 & \left(h_2 + \sum_{i\neq 2}^{N} h_{21}\right)n_2 & \cdots & -h_{2N}n_2 \\ \vdots & \vdots & \ddots & \vdots \\ -h_{N1}n_n & \cdots & \cdots & \left(h_N + \sum_{i\neq N}^{N-1} h_{Ni}\right)n_N \end{bmatrix} \times \begin{bmatrix} \overline{E}_1/n_1 \\ \overline{E}_2/n_2 \\ \vdots \\ \overline{E}_N/n_N \end{bmatrix} = \begin{bmatrix} \overline{P}_{i1} \\ \overline{P}_{i2} \\ \vdots \\ \overline{P}_{iN} \end{bmatrix} \quad \ldots(9)$$

Where $\frac{\overline{E}_i}{n_i}$ is the energy density of sub-system i. Equation (9) represents a set of N-simultaneous linear equations with the independence variable of interest $\frac{\overline{E}_i}{n_i}$. This set of equations (matrix equation) can be solved to obtain the stored energies in the coupled sub-systems Ref. [3].

3. System Modeling and S.E.A Parameters

SEA models are all based on the balance of energy and power among groups of natural modes in a system. Complex systems can be modeled as coupled mode groups. There are three main steps in defining an SEA model. The first is to divide the system into groups of components with similar characteristics, for example plates, beams, acoustic spaces and etc.

Similar subsystems are used because they have similar modes, similar structures also carry similar wave types like compression, shear and bending for example, a beam has the main mode of bending, as do all other beams. The second step is to define the physical connection between the subsystems for example line connection, bolt connection or weld connection. The last step is to determine or define the external excitation of the system like the noise or vibration generated by wind, engine, transmission, tire ...etc.

The parameters that define the sub-systems are the average frequency spacing between modes (center frequency of third octave band), the average modal damping factor can be taken from table (1) that shows some typical values of structural loss factor for some common materials presented by Ref. [8], and Ref. [6]. The modal density of flat plate in flexural vibration is given by Ref.(3)**:-**

$$n(w) = \frac{A\sqrt{12}}{2pC_L t} \quad \text{…(10)}$$

Where;

$$C_L = \sqrt{\frac{Y}{r(1-u^2)}} \quad \text{…(11)}$$

Where**,** C_L is the longitudinal wave speed, Y is Young's modulus, u is Poisson's ratio, A is the surface area of plate, and, t is the plate thickness. The time average input power can be written as,

$$P_{in} = \frac{1}{2}\left|\tilde{F}\right|^2 \mathrm{Re}\left\{\tilde{Z}_m^{-1}\right\} \quad \text{…(12)}$$

The real part of drive–point mechanical impedance of an infinite plate of thickness t and mass per unit area ρ_a in flexural vibration Ref. [6], is**:-**

$$\mathrm{Re}\left\{\tilde{Z}_m\right\} = 8\sqrt{\frac{Y t^3 r_a}{12\left(1-u^2\right)}} \quad \text{…(13)}$$

And the coupling loss factor between mode groups for line junction is evaluated by Lyons and Cremer et al., and it is conveniently given in terms of the wave transmission coefficient, t_{12} (the ratio of transmitted power to the incident power) for a line junction, L Ref. [7]. It is**:-**

$$h_{12} = \frac{2C_B L t_{12}}{pw A_1} \quad \text{…(14)}$$

Equation (14) reveals that the coupling loss factor depends on the surface area, A_1 and the bending wave speed, C_B of the first plate for two connected plates with line junction as a function of center frequency, f Ref. [6]. Where;

$$C_B = \sqrt{1.8 C_L t f} \quad \text{… (15)}$$

Table 1,
Structural loss factor for some common materials Ref. [6].

Material	Structural Loss Factor
Aluminum	$10^{-4}\times1.0$
Brick, Concrete	$10^{-2}\times1.5$
Cast Iron	$10^{-3}\times1.0$
Copper	$10^{-3}\times2.0$
Glass	$10^{-3}\times1.0$
Plaster	$10^{-3}\times5.0$
Plywood	$10^{-2}\times1.5$
PVC	$10^{0}\times0.3$
Sand (dry)	0.02-0.2
Steel	$10^{-4}\times1$-6
Tin	$10^{-3}\times2.0$

And the normal incidence transmission coefficient for two coupled flat plates at right angles to each other is given in Ref. [6]**:-**

$$t_{12}(0) = 2\left\{y^{1/2} + y^{-1/2}\right\}^{-2} \quad \text{…(16)}$$

Where;

$$y = \frac{r_1 C_{L1}^{3/2} t_1^{5/2}}{r_2 C_{L2}^{3/2} t_2^{5/2}} \quad \text{…(17)}$$

The random incidence transmission coefficient t_{12} is approximated by Ref. [6]**:-**

$$t_{12} = t_{12}(0)\frac{2.754X}{1+3.24X} \qquad \ldots(18)$$

Where;

$$X = t_1 / t_2 \qquad \ldots(19)$$

These parameters must be modeled correctly because they are the coupling of the sub-systems that determines how much energy is transmitted from one subsystem to another.

When all these steps are defined correctly and submitted into the major equation of S.E.A eq(9), the conclusion will be easy to estimate when we find the energy stored in each sub-system. It should be noticed that the results of vibrational energies are obtained in decibels (dB) rather than any other physical units because it can show any small variation in the energy level and also it can mark the sensitivity of the model quite easily. This is done by using the formula Ref. [2],

$$\overline{E}(dB) = 10\,Log_{10}\left(\overline{E}(N.m) / E_{ref} \right) \qquad \ldots(20)$$

Where E_{ref} =10E-7 N.m

The above equations give the most desired results of Statistical Energy Analysis (S.E.A) model.

4. System Details

The system considered here is shown in fig. (3). It comprises two thin, flat, uniform, plates joined along a straight edge, each plate is forming one subsystem. The "Thin" assumption implies that the effects of shear deformation and rotational inertia are both negligible. The length of the coupling is *L* and the areas of the plates are A_1 and A_2. It is assumed that only plate 1 is excited and that the excitation is harmonic at frequency ω and of constant amplitude F_o. And the material of the system is steel. The developed system is assumed to be resonant, conservative, linear and there is an equpartition of energy between all the resonant modes within a given frequency band in a given sub-system.

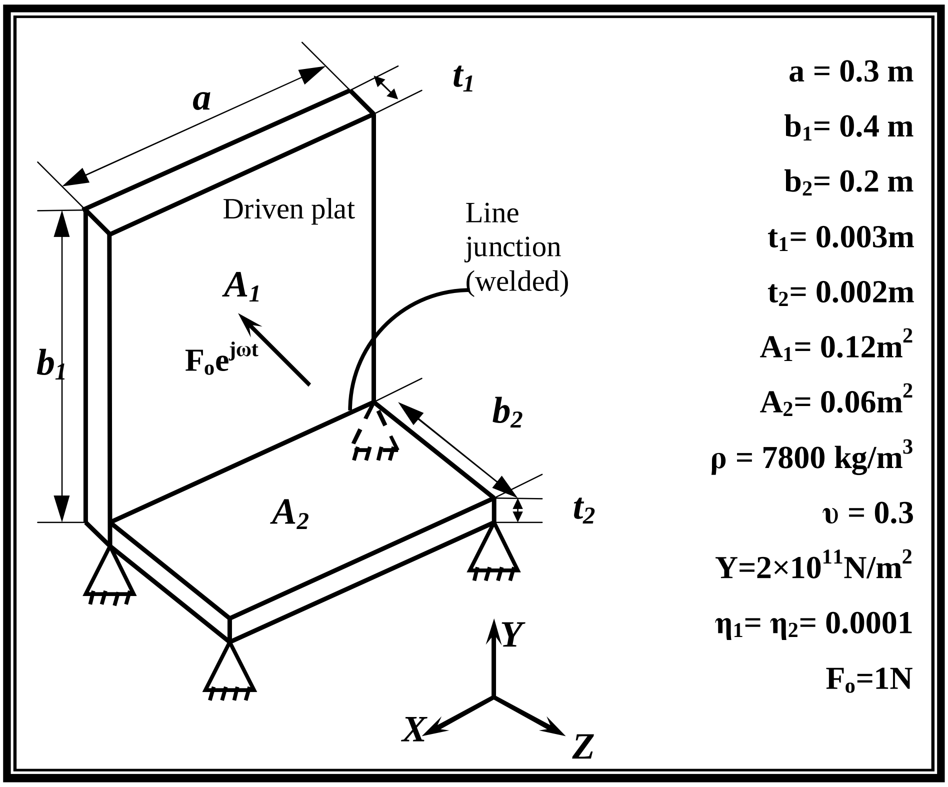

Fig. 3. Two Coupled Plates System.

5. Finite Element Modeling

In this paper, the energy stored in sub-systems is found by using Statistical Energy Analysis, and a comparison is made between analytical solution (S.E.A) and numerical analysis (F.E.M) ,built by ANSYS 5.4 for the case of the two coupled (welded) plates. Because, the Statistical Energy Analysis gives us the energy in average manner over all the points of the plate and does not give any information about concentration of energy in particular parts of the sub-systems and that the Finite Element Method gives the response in any particular point. Therefore, the loading of harmonic forces will apply at different points on the first plate (five different points) and the average responses over different locations (ten different points for each plate) are obtained. A job name and analysis title is specified from the file option on the Utility menu and clarifies the type of analysis (structural analysis). Then, PREP7 preprocess is used to define the element types, element real constant, material properties for steel after consider it as isotropic material, and the model geometry as shown in figure (4). their properties will be:

Element type = Shell Elastic.
Young modulus of elasticity, $Y=2\times10^{11}$ (N/m^2).
Poisson's ratio, $\upsilon=0.3$.
Mass density, $\rho=7800$ (Kg/m^3).

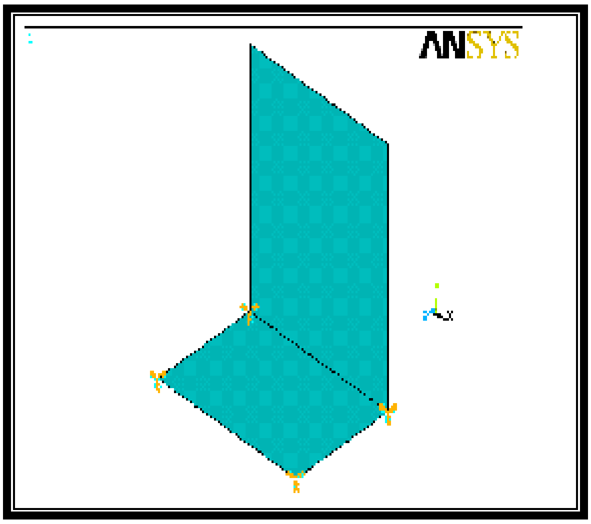

Fig. .4.A Tested model of Two Coupled Plates Built by ANSYS Package.

In applying the Finite Element method, it is necessary to divide the structure into a set of interconnected elements sufficiently small to adequately model, the structural geometry and to adequate represent loads and structural deformations. So, the whole system was divided and meshed (free mesh) as follows:
Time of the run of program= 105 minutes.
Number of elements for the first sub-system (plate 1)=192.
Number of elements for the second sub-system (plate 2)=96.

Figure (5) shows the final model after it is meshed and the load is applied to it, and by that the model building is completed.

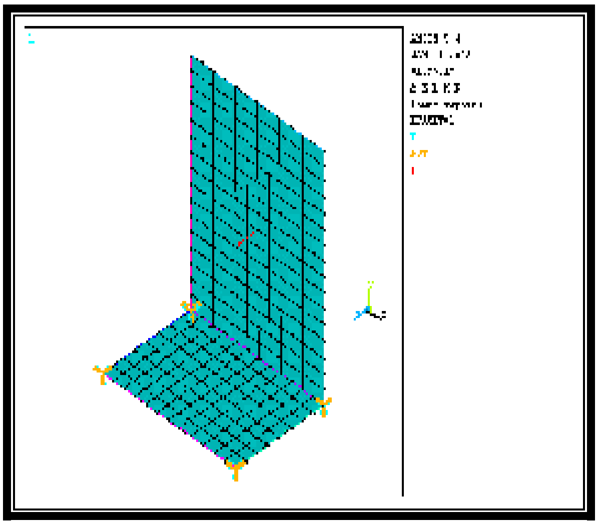

Fig. .5. The Final Model Built by ANSYS 5.4.

6. S.E.A Results And Discussion

6.1. The General Behavior Of Energy Flow

The trend of all energy versus frequency curves is similar for all cases studied in this work. As the center frequency of third octave band increases the energy content of all sub-systems decreases. An obvious explanation for this behavior is due to the fact that as frequency increases the amplitude of vibration of any structural members decreases and consequently a reduction in the level is observed.

6.2. Effect of Variation of Surface Area on Energy Level

Figures (6) and (7) demonstrate the effect of variations of the surface area of the directly driven plate A_1 on the energy levels of the coupled plate-plate system. The figures display the variation as a function of center frequency of 1/3 octave bands. It shows that as A_1 increases energy level of plate 1 increases while a reduction in the energy level of the indirectly driven plate is noticed. This is because of the fact that as A_1 increases the strength of coupling decreases towards the weak coupling condition. This leads to a reduction in the power transferred from plate 1 to plate 2 and consequently a lower energy level for plate 2. Figures (8) and (9) show that as the surface area of the indirectly driven plate A_2 increases its energy level increases and a reduction in energy level of the directly driven plate is noticed. This is due to the same reason described above (i.e.:- coupling strength). Table (2) displays some of the predicted results at 500 Hz center of frequency of 1/3 octave band.

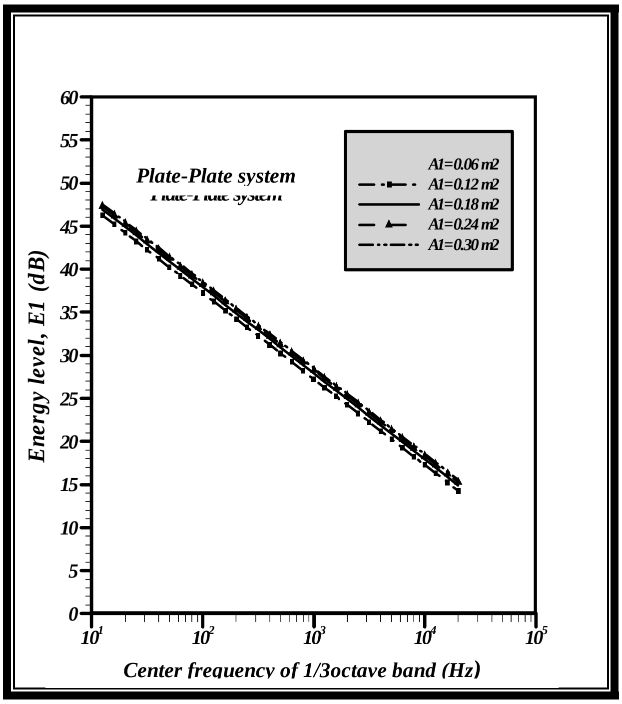

Fig. .6. The Effect of Variation Of Surface Area of Directly Driven Plate on First Plate Energy Levels.

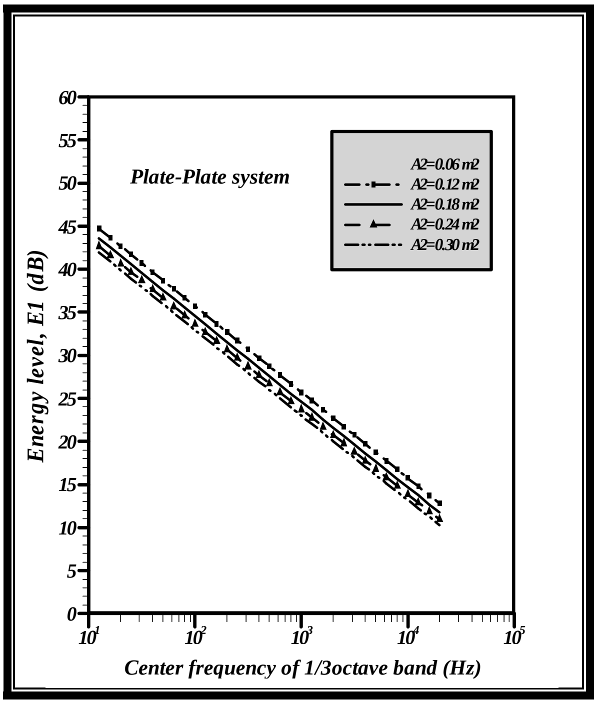

Fig. 8.The Effect of Variation of Surface Area of Indirectly Driven Plate on First Plate Energy Levels.

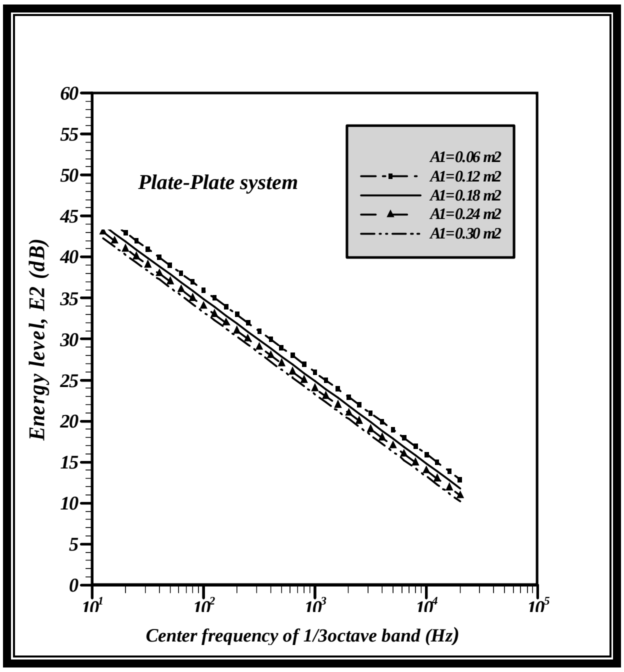

Fig. 7. The Effect of Variation of Surface Area of Directly Driven Plate on Second Plate Energy Levels.

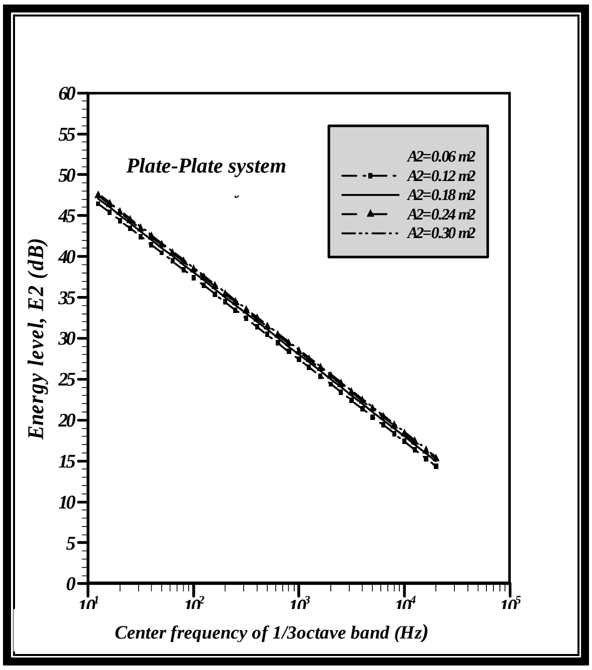

Fig. 9 The Effect of Variation of Surface Area of Indirectly Driven Plate on Second Plate Energy Levels.

Table 2,
Effect of Surface Area Variation on the Energy Level at 500 Hz.

Values of Area M^2		0.06	0.09	0.12	0.15	0.18
ΔA_1	E_1	28.701640	30.247600	30.915370	31.292170	31.535020
	E_2	30.444710	28.980370	27.887220	27.014640	26.288390
ΔA_2	E_1	30.247600	28.712280	27.588370	26.702860	25.973140
	E_2	28.980370	30.437570	31.056880	31.403130	31.624950

6.3. Effect of Variation of Internal (Damping) Loss Factor on Energy Level

The second part of the parametric study is concerned with the effect of variation of material dissipation loss factor on the predicted results of total vibrational energy for case studied in this work. This is done by increasing (alternatively) the dissipation loss factor (η_i) of one sub-system in each case while other parameters kept fixed. Figures (10)-(13) present the effect of variation of internal loss factor for a range of (0.00001-0.1). This variation causes a reduction of the values of the energy levels for the two plates and that is because the increasing of internal loss factor values led to increasing of the material resistance and that will dissipate the energy flow across that sub-system. Of course the amount of the energy dissipated depends on the sub-system that the variation lies on. Table (3) displays some of the predicted results at 500 Hz center of frequency of 1/3 octave band.

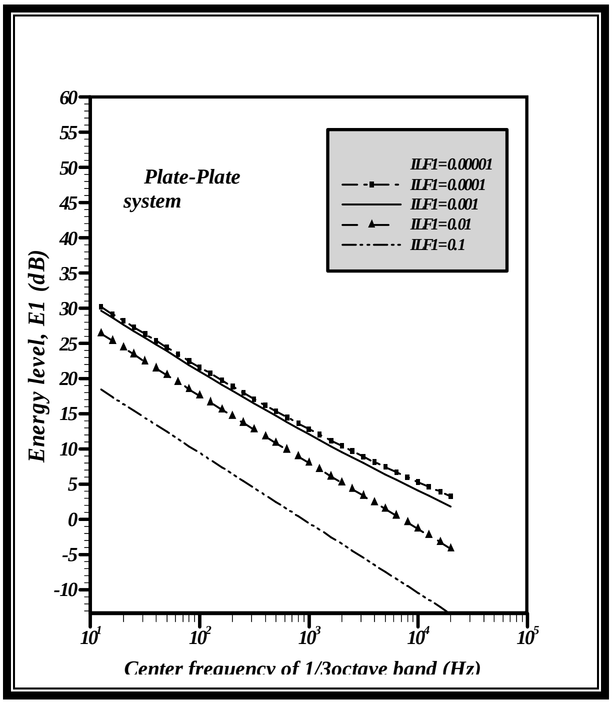

Fig. 10. The Effect of Variation of Directly Driven Plate Internal Loss Factor on the Plate 1 Energy Levels.

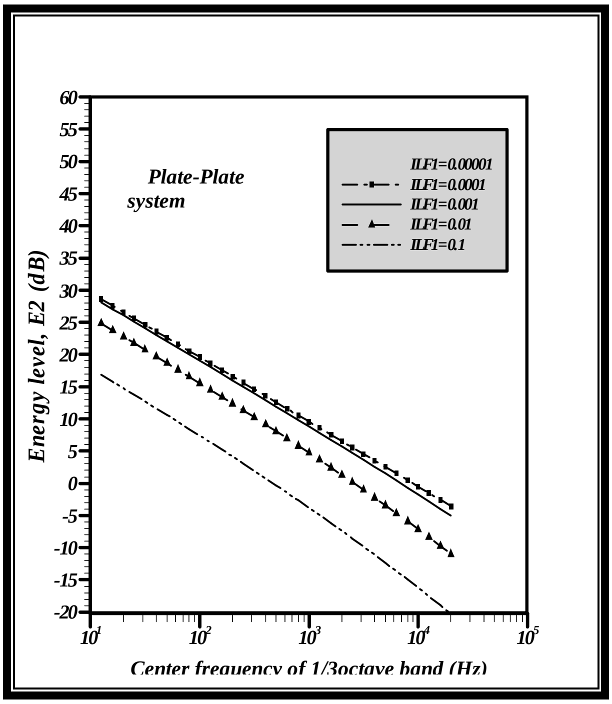

Fig. 11. The Effect of Variation of Directly Driven Plate Internal Loss Factor on the Plate 2 Energy Levels.

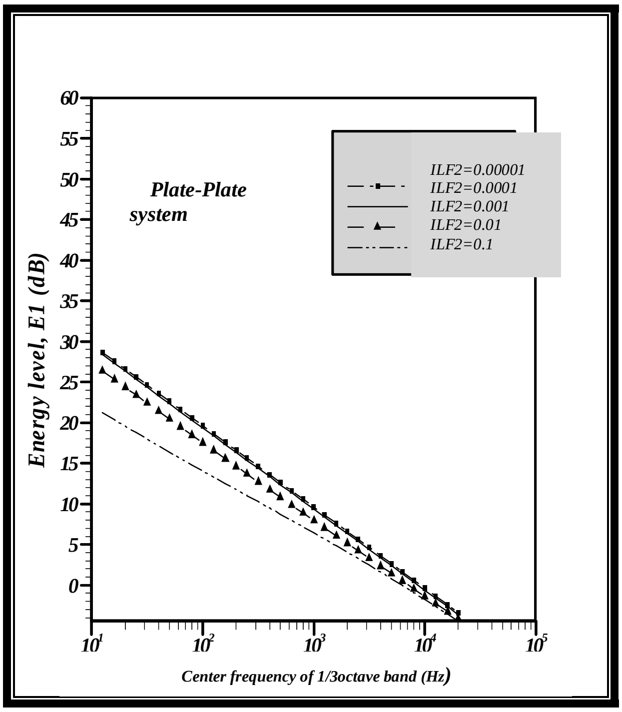

Fig. 12. The effect of variation of indirectly driven plate internal loss factor on the plate 1 Energy levels.

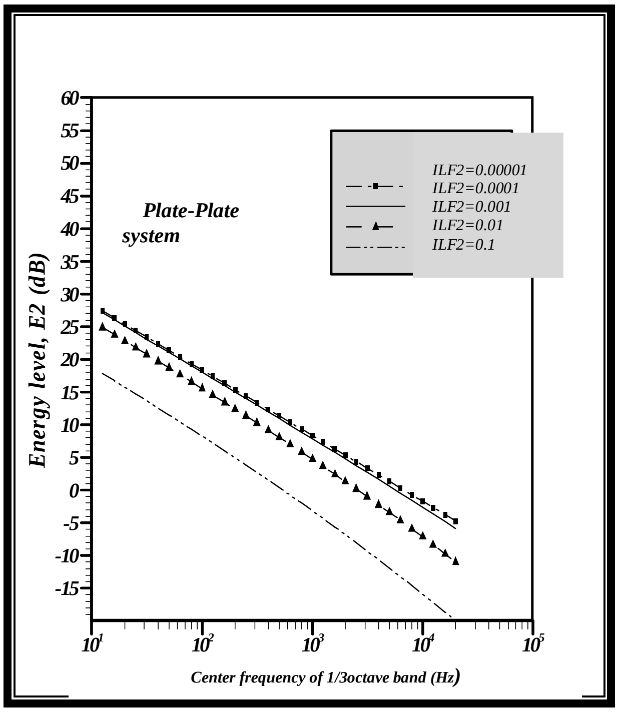

Fig. 13. The Effect of Variation of Indirectly Driven Plate Internal Loss Factor on The Plate 2 Energy Levels.

Table 3,
Effect of Dissipated Loss Factor Variation On The Energy Level at 500 Hz.

Values of Internal Loss Factor		0.00001	0.0001	0.001	0.01	0.1
$\Delta\eta_1$	E_1	15.409160	15.336340	14.668310	10.819890	2.445565
	E_2	12.662200	12.589380	11.921350	8.072927	-0.301393
$\Delta\eta_2$	E_1	12.667110	12.638050	12.368280	10.819890	8.751205
	E_2	11.415940	11.370820	10.943660	8.072927	0.411249

6.4. Comparison Between S.E.A and F.E.M Results

Figures (14)-(18) present the average of energy levels predicted by Statistical Energy Analysis and Finite Element averages over different response locations due to an applied load at a specific location (node). Figure (19) shows the comparison between the average energy levels predicted using S.E.A and the ensemble averages of energy levels over the different force and response locations using F.E.M. The noticed differences between results of the two methods are due to the fact that S.E.A obtains the response for the whole sub-system without going into the details of the modes of vibration involved while F.E.M results involve mode by mode calculations. The comparison shows that at high frequency domain the results of the two methods become more comparable because of the fact that at high frequency regimes the number of modes becomes higher than that at low frequency regimes. The higher the number of modes means that the coupled sub-system in that region of frequency has high modal overlap factor which is the condition that should be available when using S.E.A. This comparison proved that Statistical Energy Analysis is more acceptable tool of analysis at high frequencies than that at low frequencies.

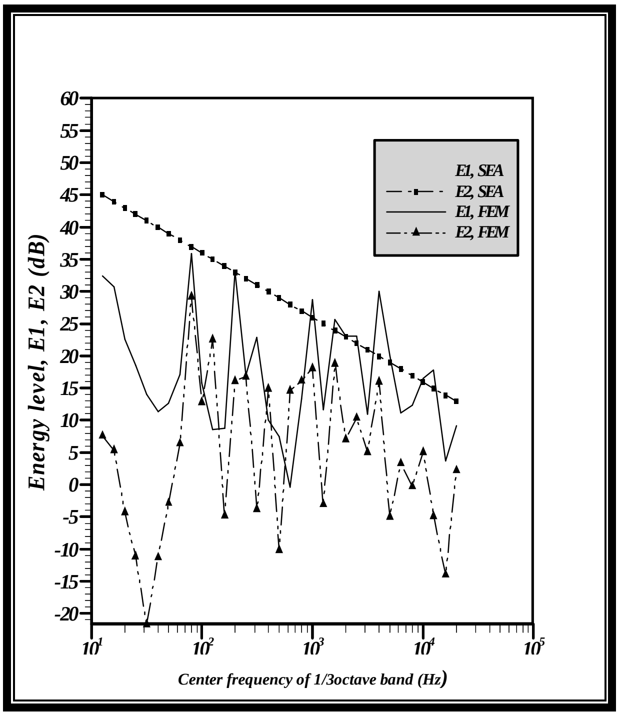

Fig.14. The comparison between S.E.A results and F.E.M average results due to a load applied in the center point of the first sub-system.

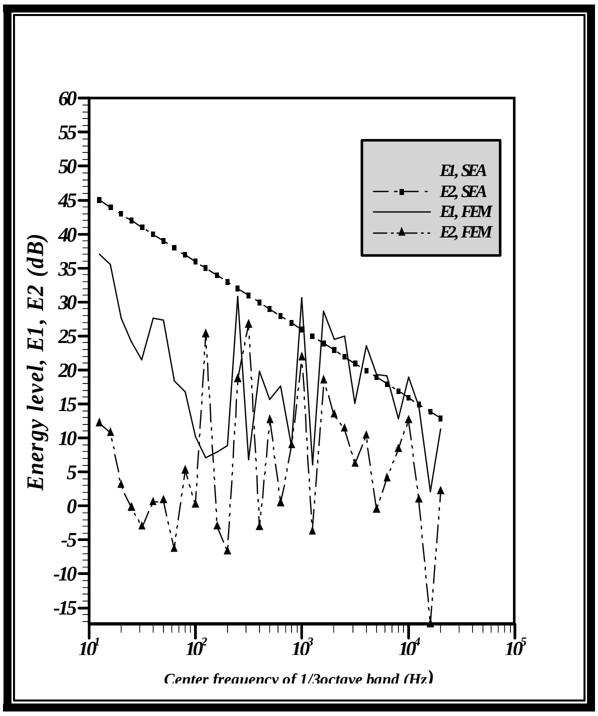

Fig.16. The comparison between S.E.A results and F.E.M average results due to a load applied in the upper left corner of the first sub-system.

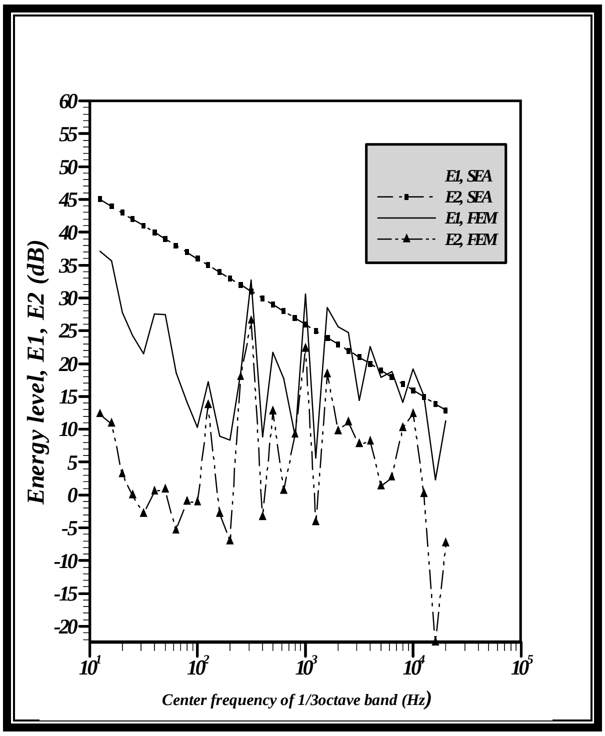

Fig. 15. The Comparison Between S.E.A Results and F.E.M Average Results due to a Load Applied in the Upper Right Corner of the First Sub-System.

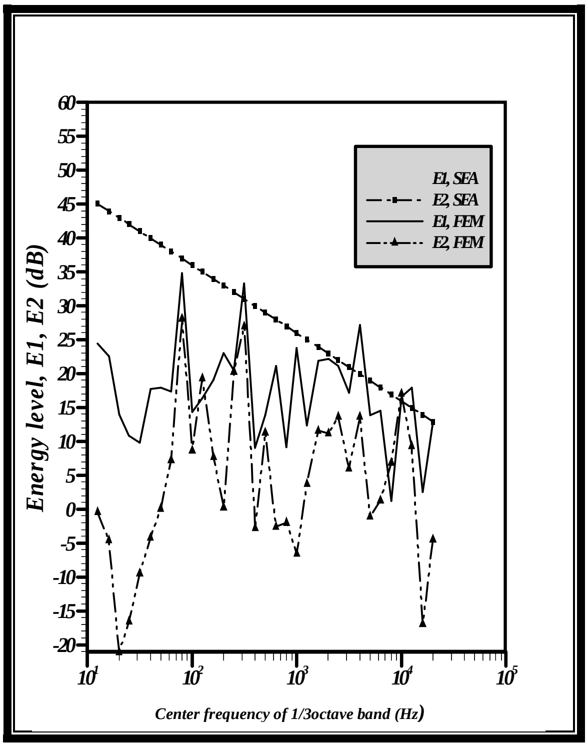

Fig. 17. The Comparison Between S.E.A Results and F.E.M Average Results due to a Load Applied in the Lower Right Corner of the First Sub-System.

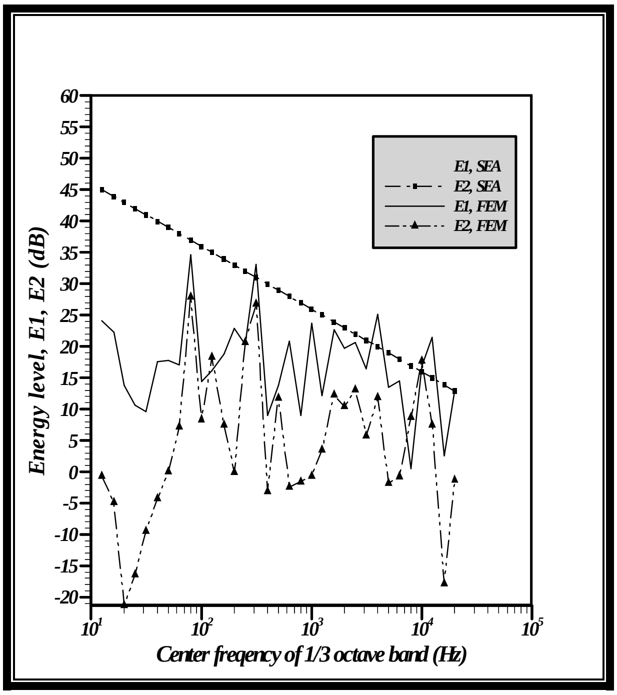

Fig.18. The Comparison Between S.E.A Results and F.E.M Average Results due to a Load Applied in the Lower Left Corner in the First Sub-System.

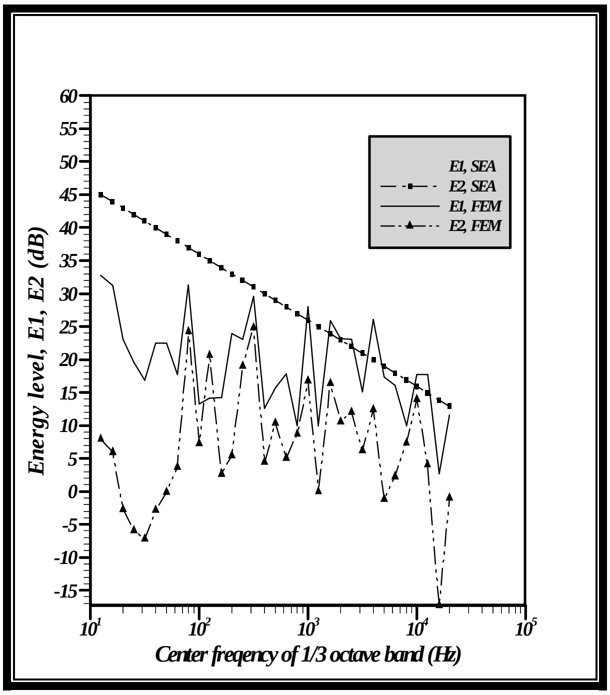

Fig. 19. The Comparison between S.E.A Results and F.E.M Ensemble Average.

Table 4,
The Values of Energy at Frequencies of Octave Band for S.E.A and F.E.

FREQUENCY HZ		1000	2000	4000	8000	16000
S.E.A	E_1	27.240470	24.234590	21.230570	18.229050	15.231060
	E_2	25.965870	22.949580	19.930900	16.908710	13.881680
F.E.M	E_1	28.000000	23.200000	26.100000	9.9900000	2.6500000
	E_2	16.800000	10.600000	12.400000	7.4100000	1.7300000

7. Conclusions

The main conclusions drawn from this work based on the results obtained in the present study are summarized as follows**:-**

1. Statistical Energy Analysis is a quite powerful tool in analyzing the dynamic behavior of coupled systems without going into great technical details as other approaches.
2. The main thing to remember is that Statistical Energy Analysis uses power flow from higher sub-system energy to lower sub-system energy for the basis of its calculations. It is also important to remember that all of the parameters are based on averages and that; this method cannot calculate an exact response at an exact location.
3. The general trends of all energy reverse frequencies decrease with increasing frequency. This behavior is due to the fact that by increasing frequency, the amplitude decreases and consequently causes a reduction in energy levels.
4. The energy flow between sub-systems is increased by increasing the coupling loss factor (strong coupling). While decreasing the coupling loss factor decreases the energy flow between sub-systems (weak coupling).
5. Changing of sub-system dimensions affects the coupling loss factor and modal density and

consequently affects the energy stored in that sub-system.

6. The energy stored in particular sub-system decreases with increasing the internal loss factor.
7. The suggested analytical solution gives good agreement with the numerical solution at the high frequency domain.

8. References

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تقدير طاقة الاهتزاز لصفيحتين متصلتين (ملحومة) باستعمال طريقة تحليل الطاقة الإحصائي

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الخلاصة

هذا البحث يتعامل مع طريقة تسمى طريقة تحليل الطاقة الإحصائي والتي يمكن تطبيقها على المنظومات المهتزة الميكانيكية والصوتية مثل البنايات, الجسور, والطائرات.....الخ. هذه الطريقة كأداة يمكن تطبيقها على المنظومات الخاضعة للظروف (ترددا عالي أو/ و هيكل معقدا). العوامل التي تعتمد عليها هذه الطريقة (معامل خسارة الارتباط, معامل الخسارة الداخلي, الكثافة الطورية والقدرة الداخلة إلى النظام المهتز تم توضيحها لصفيحتين متصلتين وتم تقديم شرح لهذه العوامل في هذا البحث و بالإضافة إلى ذلك الشروط والخطوات التي يجب أتباعها للنمذجة باستخدام هذه الطريقة تم تقديمها في هذا البحث. المنظومة المهتزة تم أفتراض أنها تهتز بصورة رنانة, و يتم حفظ الطاقة فيها, و الأهتزاز يتم بصورة خطية و هناك توزيع متساوي للطاقة بين كل الأجزاء المهتزة عند تردد معين. الهدف من هذا البحث هو أيجاد الطاقة المخزونة في الأجزاء المهتزة الصفيحتيين مرتبطتين (ملحومة) بزاوية قائمة ودراسة تأثير تغيير أبعاد (طول وعرض) هذه الأجزاء, النتائج تبين أن عند زيادة المساحة السطحية للصفيحة المثارة بطريقة مباشرة (الصفيحة الأولى) يزيد مستوى الطاقة للصفيحة الأولى بينما يقل مستوى الطاقة للصفيحة التي لم تثار بطريقة مباشرة (الصفيحة الثانية). وهذا بسبب أن قوة الأتصال بين الصفيحتين يتجه نحو الأرتباط الضعيف و هذا يقود الى قلة الطاقة المنتقلة من الصفيحة الأولى الى الصفيحة الثانية و بالتالي مستوى طاقة قليل للصفيحة الثانية. بالإضافة إلى ذلك دراسة تغيير معامل الخسارة الداخلي لمدى من (٠,٠٠٠١-٠,١) هذا التغيير يسبب في قلة مستوى الطاقة للصفيحتيين و ذلك بسبب أن زيادة معامل الخسارة الداخلي يؤدي الى زيادة المقاومة الداخلية للمادة و هذا سيضيع الطاقة المنتقلة بين الصفيحتيين. وتم عمل مقارنة بين طريقة تحليل الطاقة الإحصائي التي تم بناء نموذج لها باستعمال برنامج فورتران وطريقة العناصر المحددة التي تم عمل نموذج لهل باستخدام برنامج ANSYS 5.4.