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UCID- 20102

DYNAMIC TESTING OF NOVA LASER SWITCHYARD TOWER

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JUNE 1, 1984

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Work performed under the auspices of the U.S. Department of Energy by the Lawrence Livermore National Laboratory under Contract W-7405-Eng-48.

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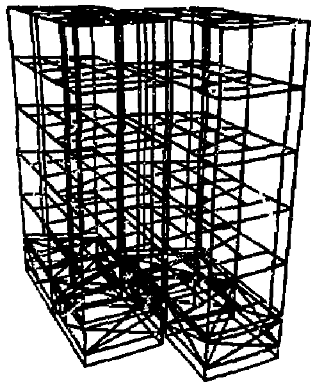

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ABSTRACT

NOVA is the latest in a series of powerful laser systems designed to study the feasibility of initiating a controlled fusion reaction by concentrating several laser beams on a small fuel target. The laser components, turning mirrors and target chamber are all mounted on large steel frame structures. These structures were first analyzed via finite element models to access their seismic integrity as well as their overall vibrational stability. When construction was completed, a modal analysis was performed on the structures to verify and improve the finite element models. This report discusses the linking of the analytical and experimental studies for the NOVA switchyard tower structure.

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INTRODUCTION

The NOVA laser system is currently being constucted at Lawrence Livermore National Laboratory to perform laser fusion feasibility research. Physics requirements call for ten large diameter (74 cm) laser beams to be focused on a small diameter (200 μm) fusion pellet in a symmetrical (4π) manner. The physical realization of such a laser system requires that the laser components (amplifiers, spatial filters, optical rotators, lenses, etc.) be mounted on large spaceframe steel structures. In the NOVA system there are three main types of structures; the laser spaceframe, the switchyard towers, and the target chamber tower. The laser spaceframe is used to mount the laser amplifiers, spatial filters and optical isolation equipment. The switchyard towers are used to mount large optical turning mirrors and laser beam diagnostic equipment. Finally, the largest structure is the target chamber tower which is used to mount the target chamber as well as additional steering mirrors.

As the structures were designed a finite element model of each was constructed. These models were then used to examine both the static and dynamic expected loads on the structures. Upon completion of construction of the structures a dynamic (modal) test was performed to determine the accuracy of the finite element models. This report discusses the testing of one of the switchyard tower structures and a comparison of these results to those predicted by the finite element codes.

SWITCHYARD TOWER TEST PLAN

The NOVA switchyard room contains several large steel frame towers. Two of the towers are used to mount laser beam diagnostic equipment while the others are used to mount large turning mirrors. In this section we describe the modal analysis that was performed on one of

Figure 1. Mirror tower to be dynamically tested.

the mirror towers (see Fig. 1). A transfer function model (see Appendix A) of this tower was constructed by locating nodes at the intersection of the main (outside) vertical and horizontal beam

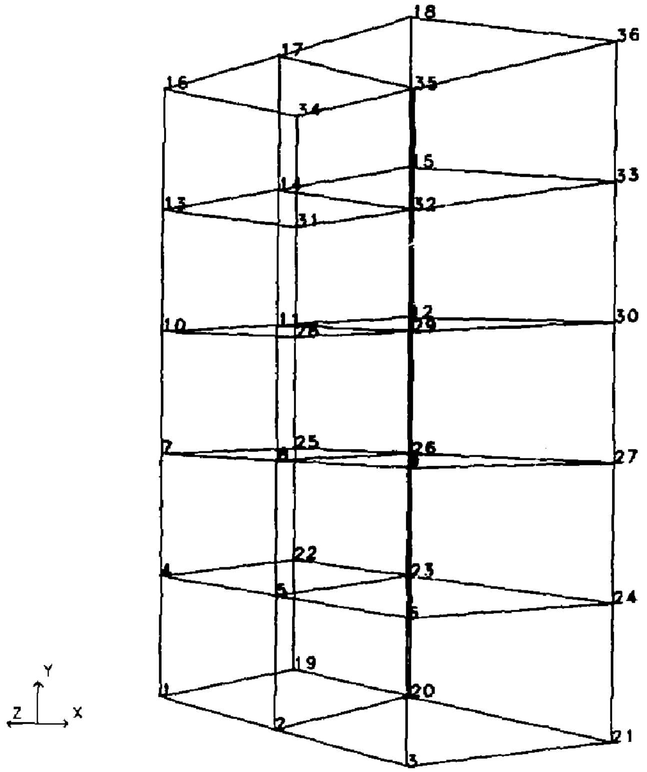

Figure 2. Transfer function model of mirror tower.

members of the structure. This model is shown in Fig. 2. The input excitation was accomplished via an electromagnetic shaker which was suspened from an overhead crane and attached to the structure at the

4th level[†]. Since the shaker was mounted to a horrizontal beam which subtended a partial diagonal across the structure, the excitation was in a non-orthogonal (to x or z) direction. The details of the shaker mounting are shown in Figs. 3 and 4. As can be seen in these figures, a load cell was mounted between the end of the shaker pushrod and a rectangular aluminum plate. The aluminum plate in turn was fastened to the structure using C-clamps. The shaker was driven by a random noise Gaussian signal with a bandwidth of 50 Hz. At each designated node of the structure the resulting motion in the x and z directions (see Fig. 2) was measured using piezo-resistive accelerometers which are shown in Figure 5. The signals from both the accelerometers and the load cell were fed into a dual channel spectrum analyzer which automatically calculated the resulting transfer function between the input excitation and the resulting output motion (see Appendix A equation A-g). These transfer functions were obtained over a frequency domain which ranged from 1 to 42 Hz. To increase the frequency resolution to 0.025 Hz. we used a recursive zoom technique. That is to say, we divided the overall frequency domain into four subdomains and then captured the full 400 spectral lines of data for each subdomain.

To improve the signal to noise level we used the standard averaging techniques. In the switchyard tower room the environment was reasonably benign and consequently we required only 3-4 averages to obtain clean transfer functions. The calculated transfer functions were first displayed on the spectrum analyzer and once we were satisfied with their quality, they were transferred to a floppy disk by way of an IBM personal computer (PC). At the completion of the test all the transfer function data were uploaded to the Livermore Time Sharing computer System (LTSS) for subsequent data processing (on the CRAY machines).

[†] For reference, the top level is denoted as level 5

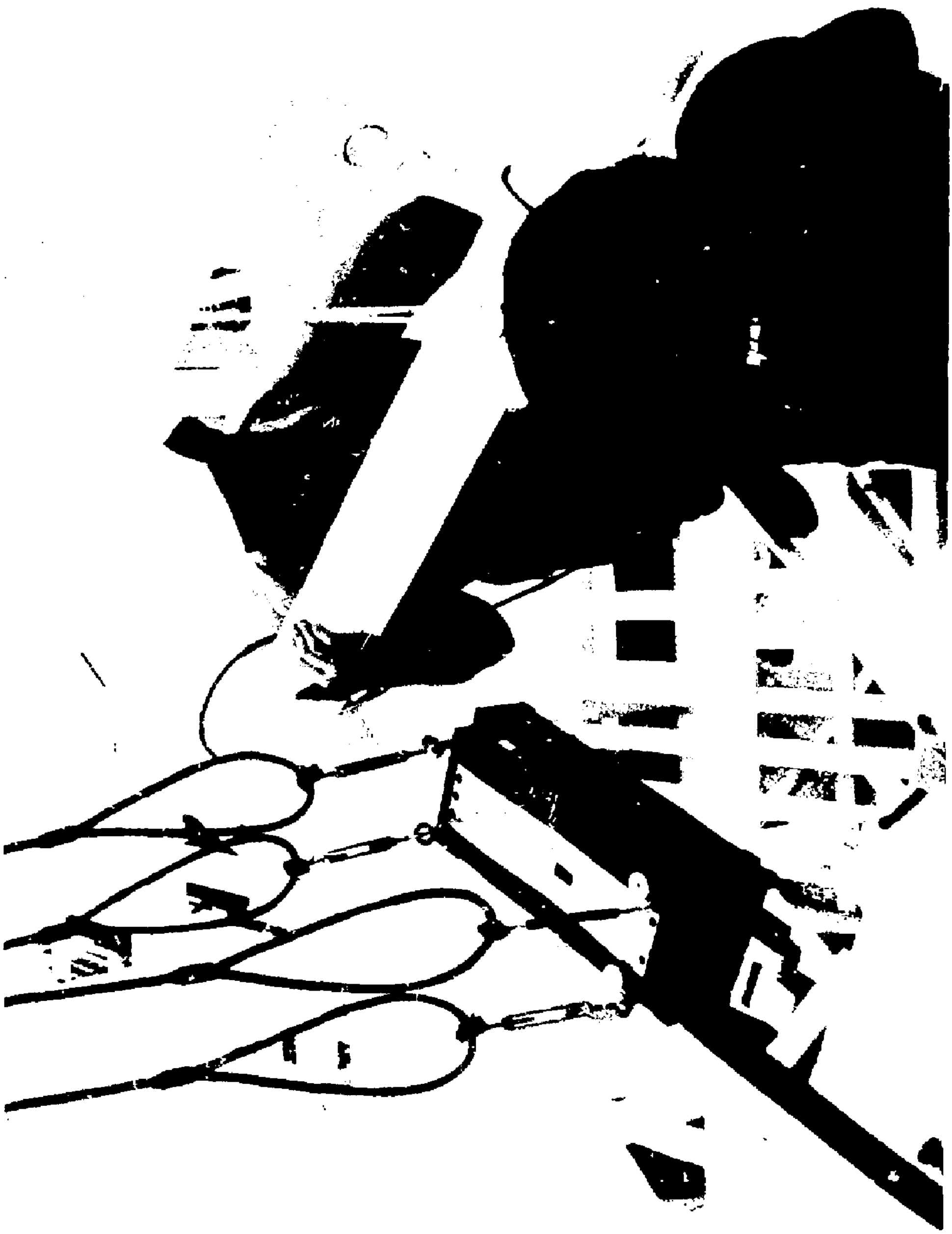

Figure 3. Shaker mounting scheme.

Figure 4. Expanded view of shaker mounting system.

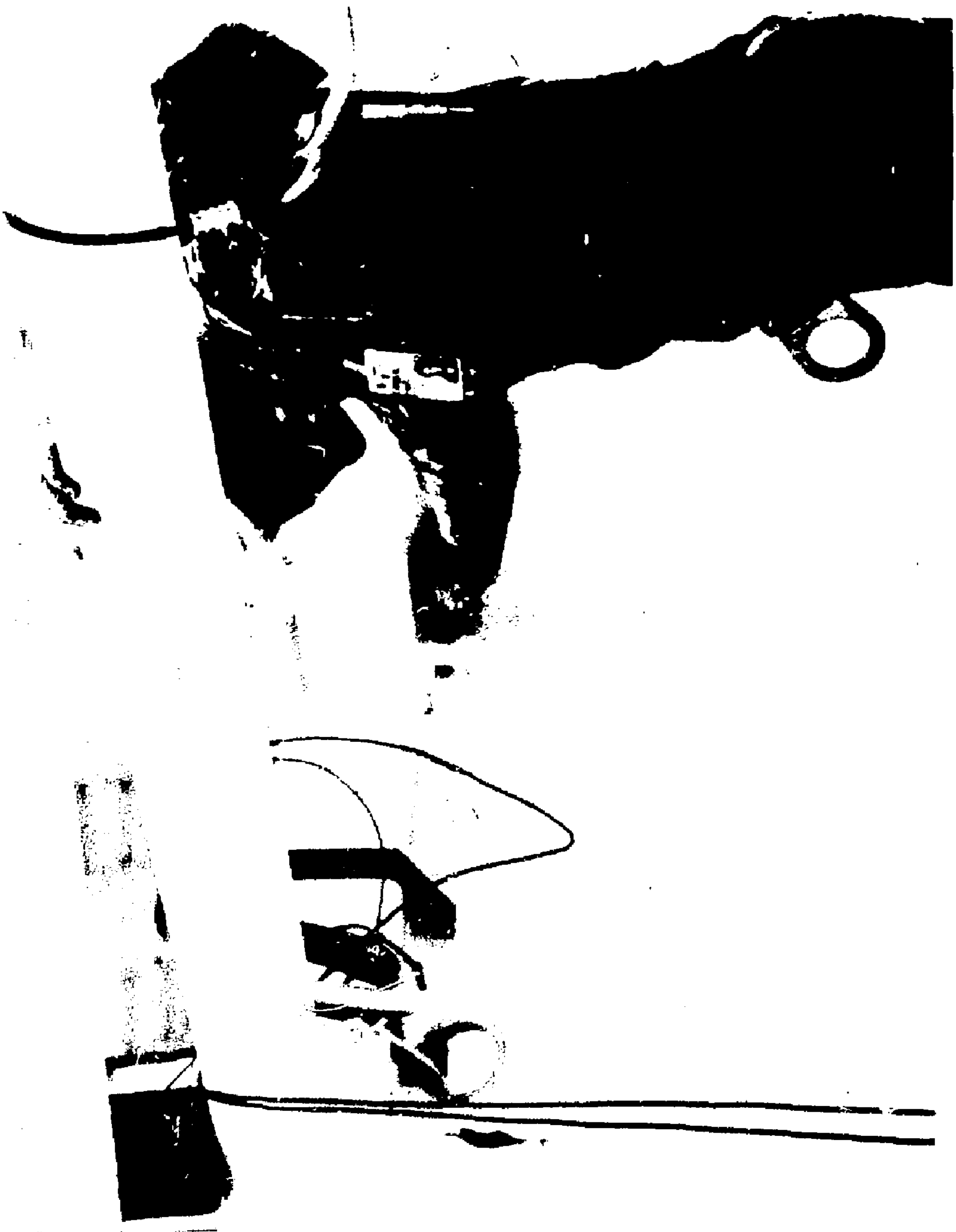

Figure 5. Accelerometers used to measure response motions.

SWITCHYARD TOWER TEST RESULTS

In Fig. 6 we show the four (zoomed) subdomains for a typical transfer function (node 18X). Each peak in the transfer function corresponds to a natural, or resonant, frequency of the structure. To obtain an accurate estimate of these frequencies, as well as the associated damping values, we use the modal parameter extraction code *TRANSF* (developed by the Modal Analysis Group). This code was used on each of the 72 transfer functions and the results averaged to obtain the data presented in Table I. Note that in this table we have also characterized the type of spatial behavior that the modes exhibit.

Table I. Switchyard mirror tower modes from 1 to 40 Hz.

Mode	Frequency	Damping	Characterization
SY1	5.6 Hz	0.65%	Cantilever (1st)
SY2	6.3 Hz	0.47%	Cantilever (1st)
SY3	8.6 Hz	0.32%	Twisting (1st)
SY4	19.0 Hz	0.13%	Cantilever (2nd)
SY5	21.4 Hz	0.16%	Cantilever (2nd)
SY6	28.5 Hz	0.16%	Twisting
SY7	29.9 Hz	0.09%	"Folding"
SY8	34.5 Hz	0.31%	Cantilever (3rd)
SY9	35.0 Hz	0.47%	Cantilever (3rd)
SY10	36.1 Hz	0.51%	Cantilever (3rd)
SY11	38.4 Hz	0.08%	Twisting (2nd)

To fully appreciate the significance of these modes we must examine their spatial characteristics. In other words, we must consider their mode shapes. This can best be accomplished by viewing the 16 mm computer movies that were generated (by MODAL[†]) and show the tower vibrating in each of its determined modes. Figures 7 through 17 show

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[†] MODAL is a family of codes (developed by the Modal Analysis Group) that are used to analyze the modal parameters obtained from a dynamic test.

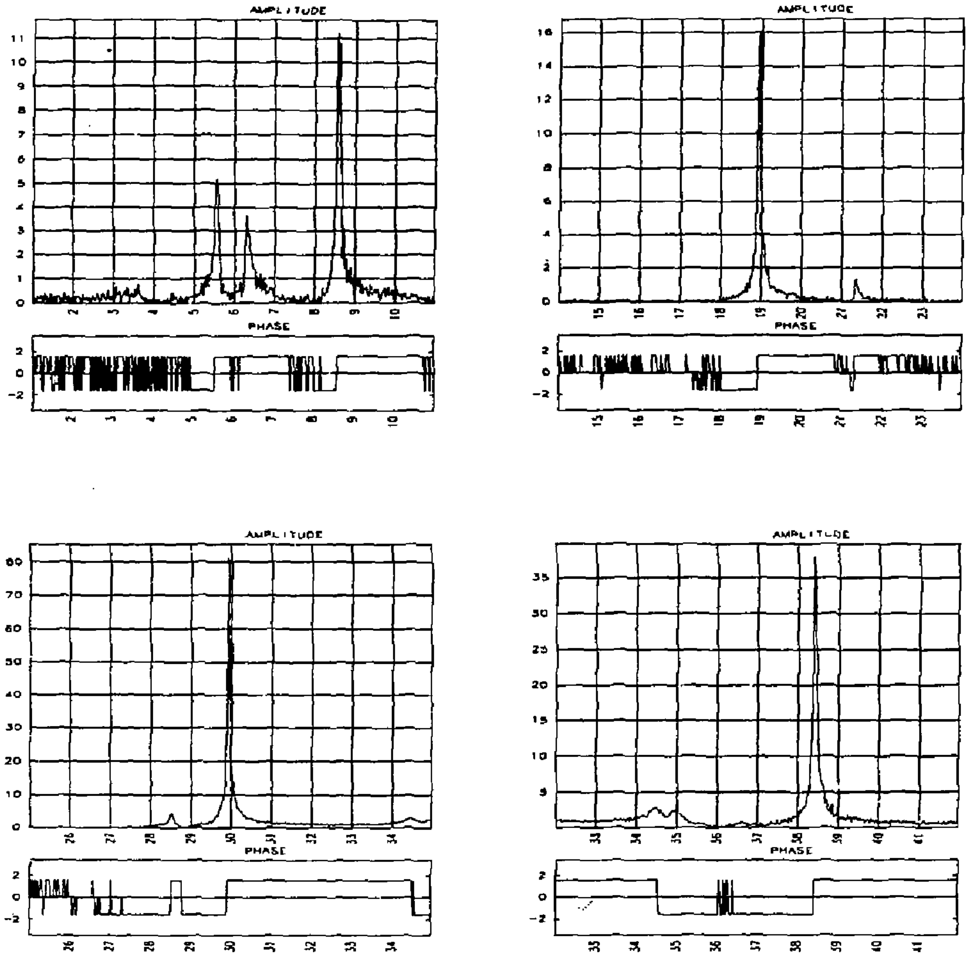

Figure 6. Typical transfer function for switchyard tower.

selected frames from this movie. In all of these figures the structure is viewed along each of its coordinate axes. In addition, the linear perspective view appears at the top left. The thin line in these figures represents the static or undisplaced state of the tower. The thick line shows the displaced state of the tower for that particular mode.

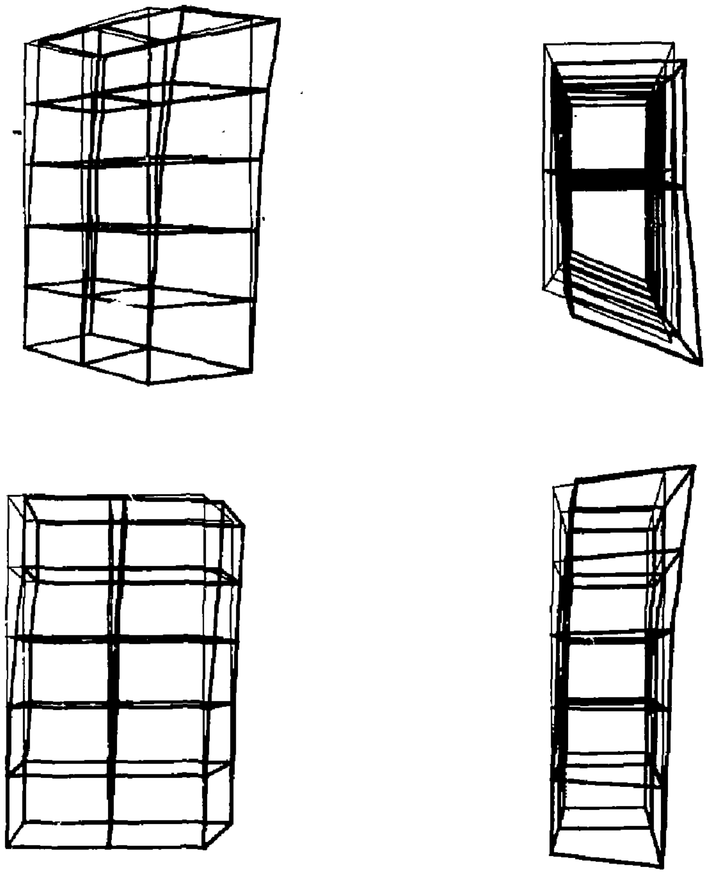

Fig. 7. Mode shape SY1 (5.6 Hz and 0.65 percent damping).

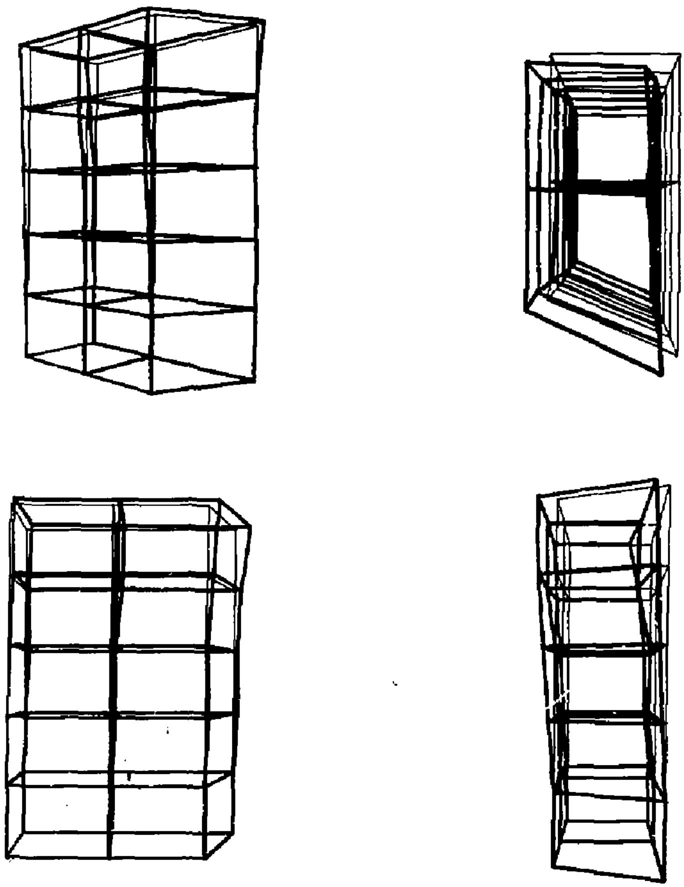

Fig. 8. Mode shape SY2 (6.3 Hz and 0.47 percent damping).

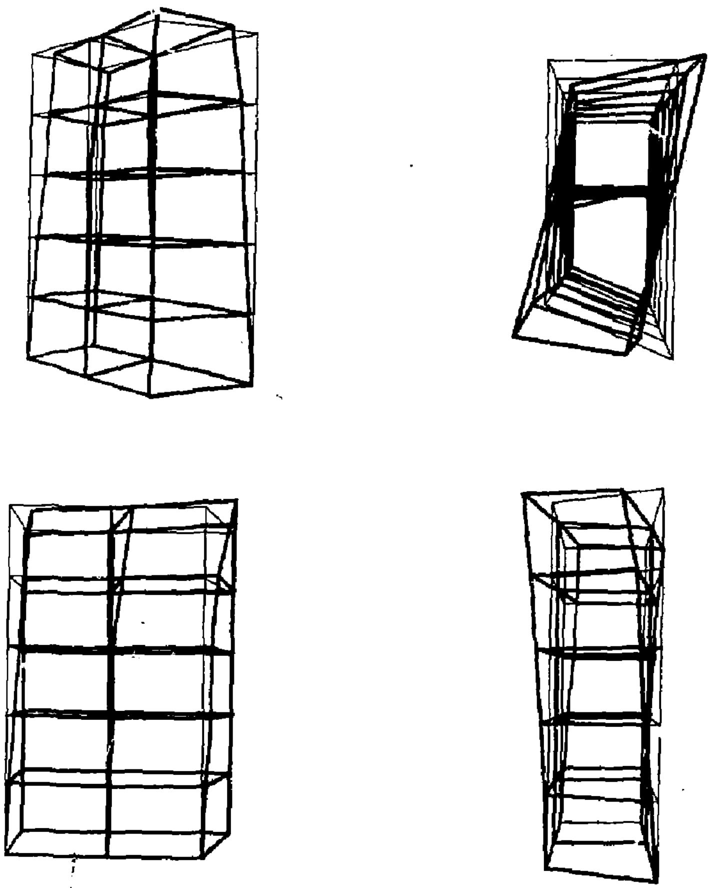

Fig. 9. Mode shape SY3 (8.6 Hz and 0.32 percent damping).

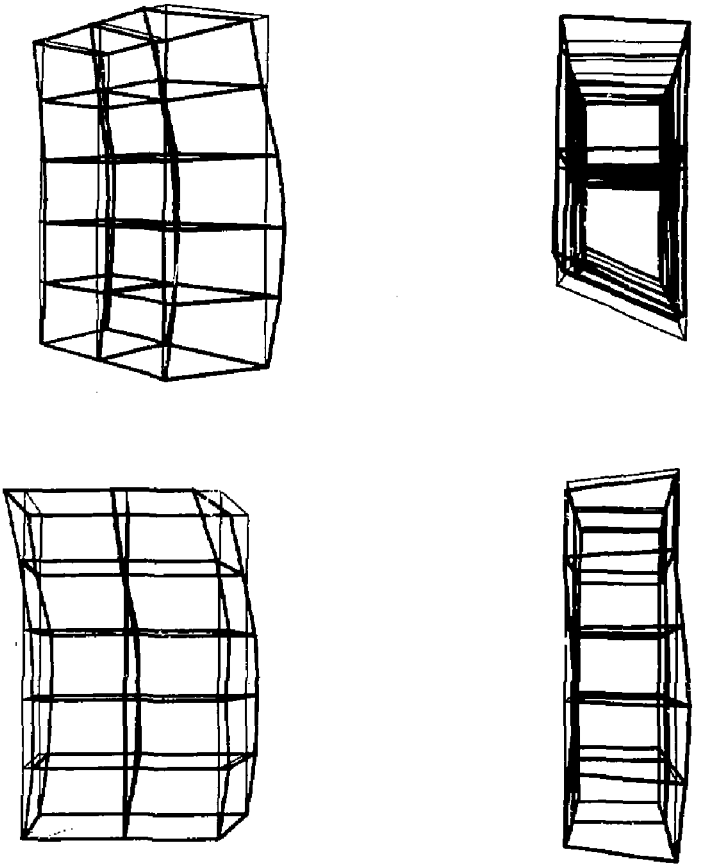

Fig. 10. Mode shape SY4 (19.0 Hz and 0.13 percent damping).

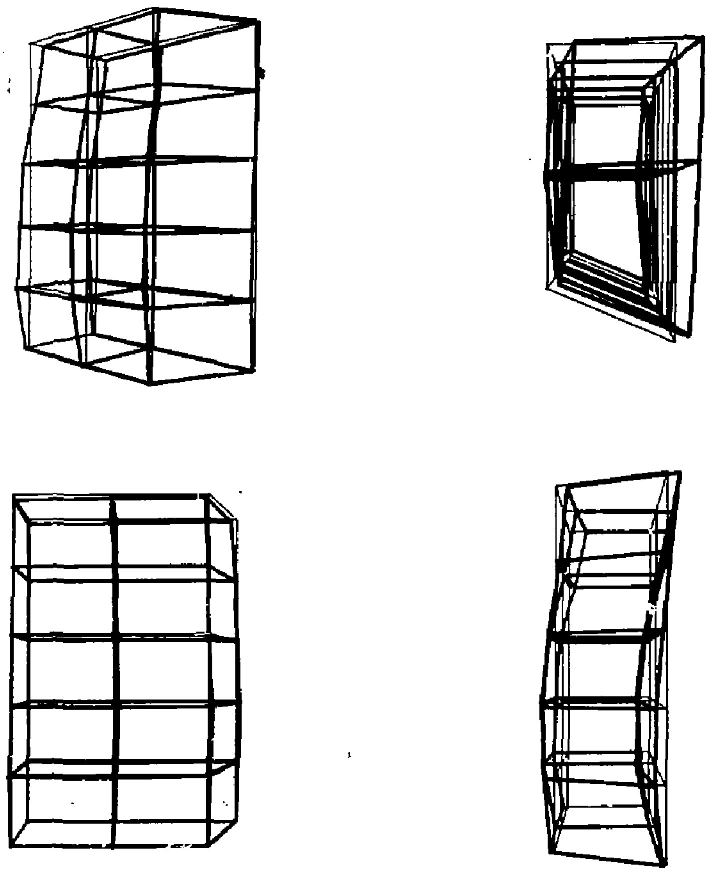

Fig. 11. Mode shape SY5 (21.4 Hz and 0.16 percent damping).

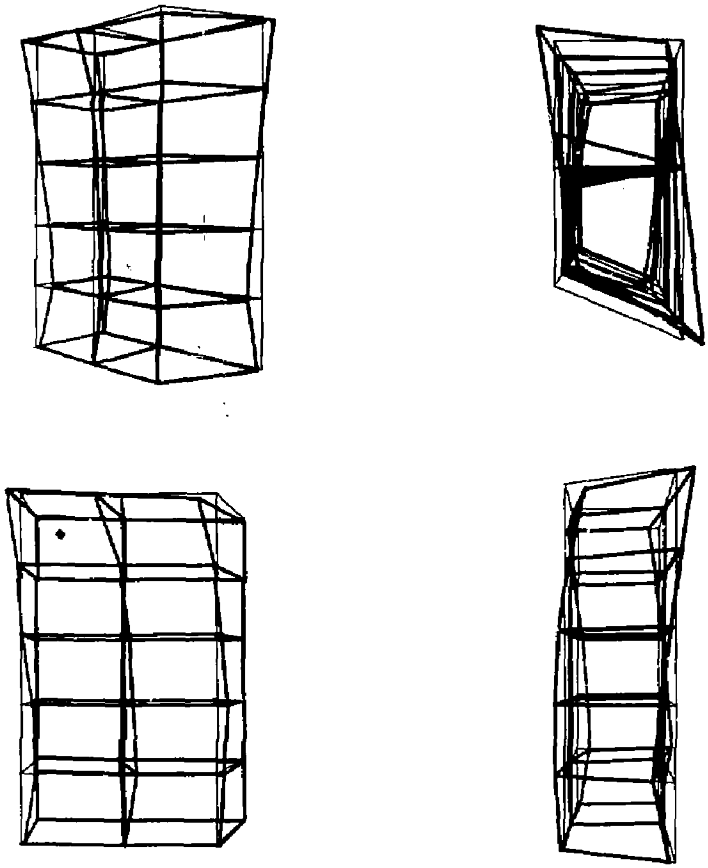

Fig. 12. Mode shape SY6 (28.5 Hz and 0.16 percent damping).

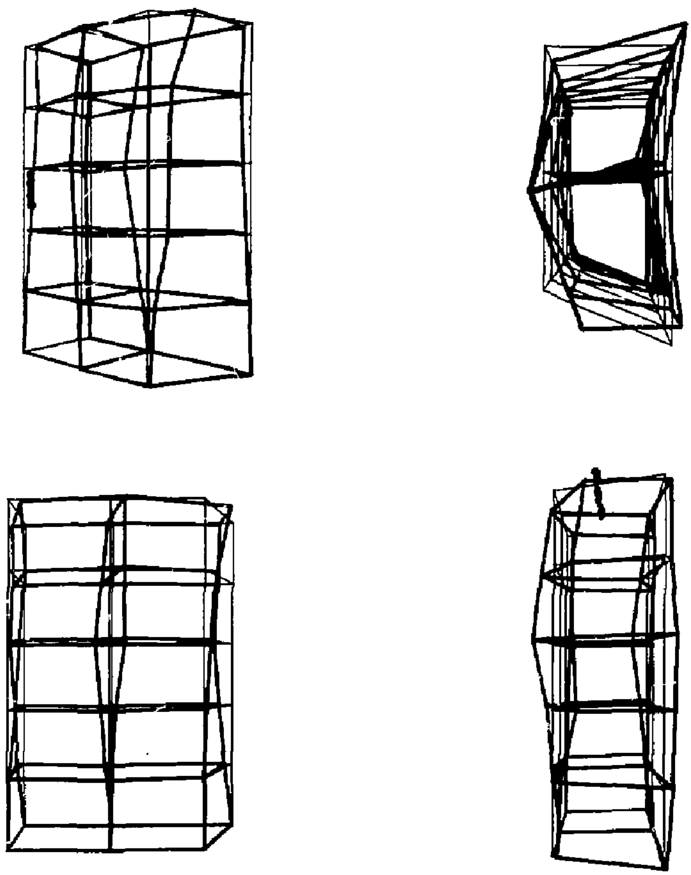

Fig. 13. Mode shape SY7 (29.9 Hz and 0.09 percent damping).

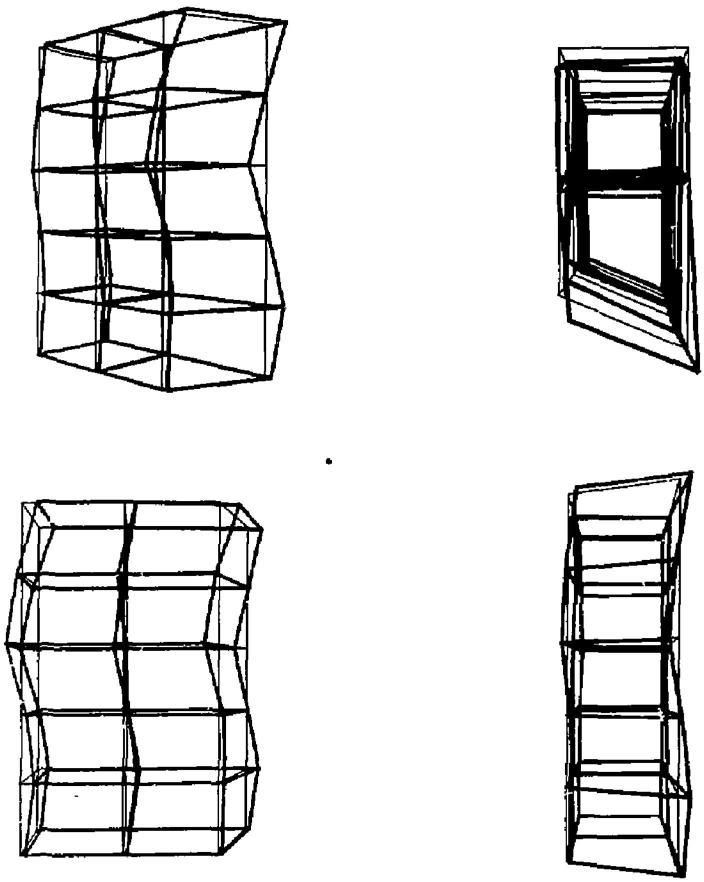

Fig. 14. Mode shape SY8 (34.5 Hz and 0.31 percent damping).

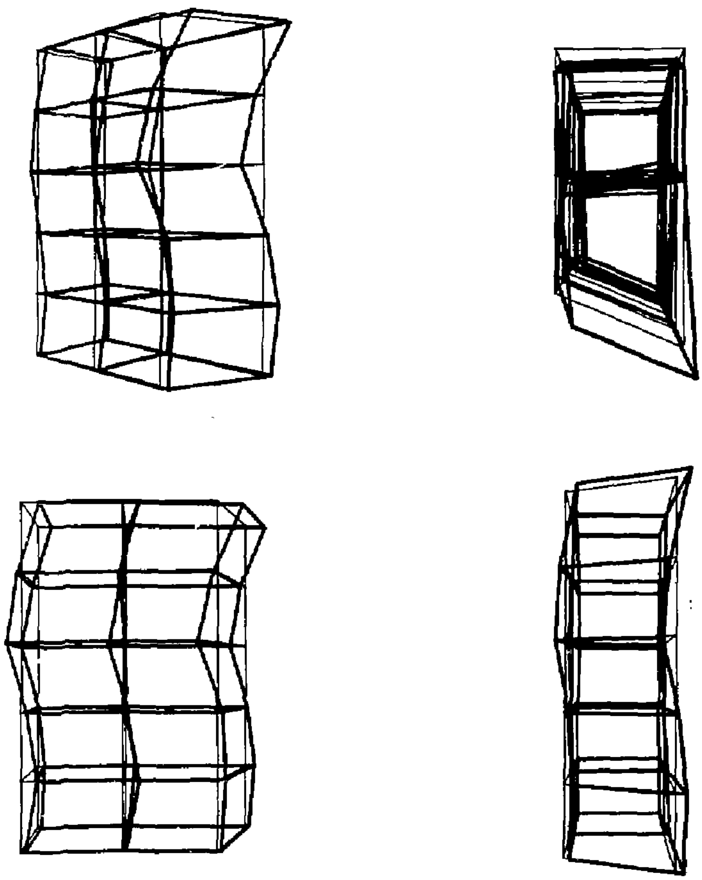

Fig. 15. Mode shape SY9 (35.0 Hz and 0.47 percent damping).

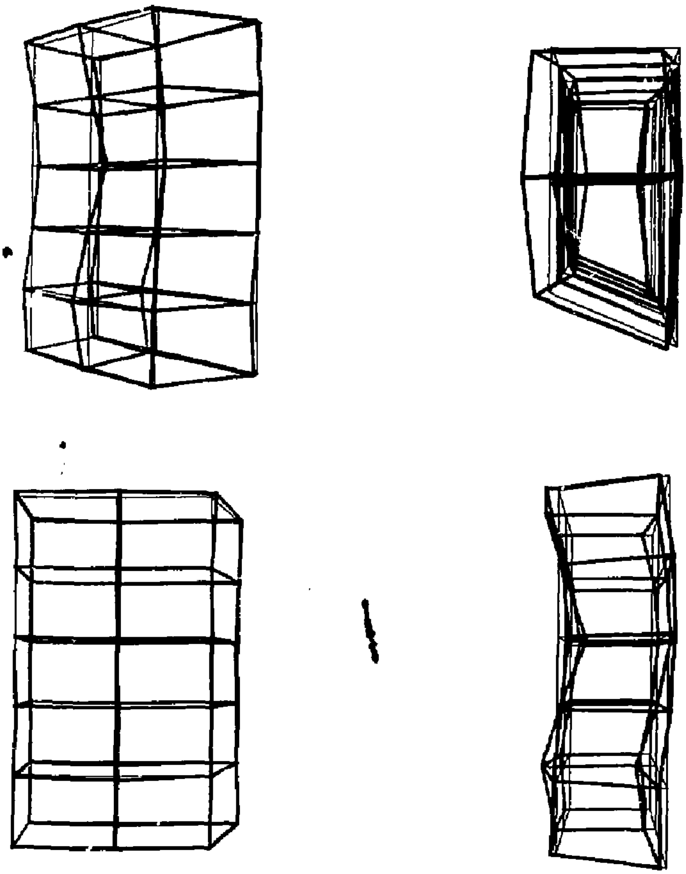

Fig. 16. Mode shape SY10 (36.1 Hz and 0.51 percent damping).

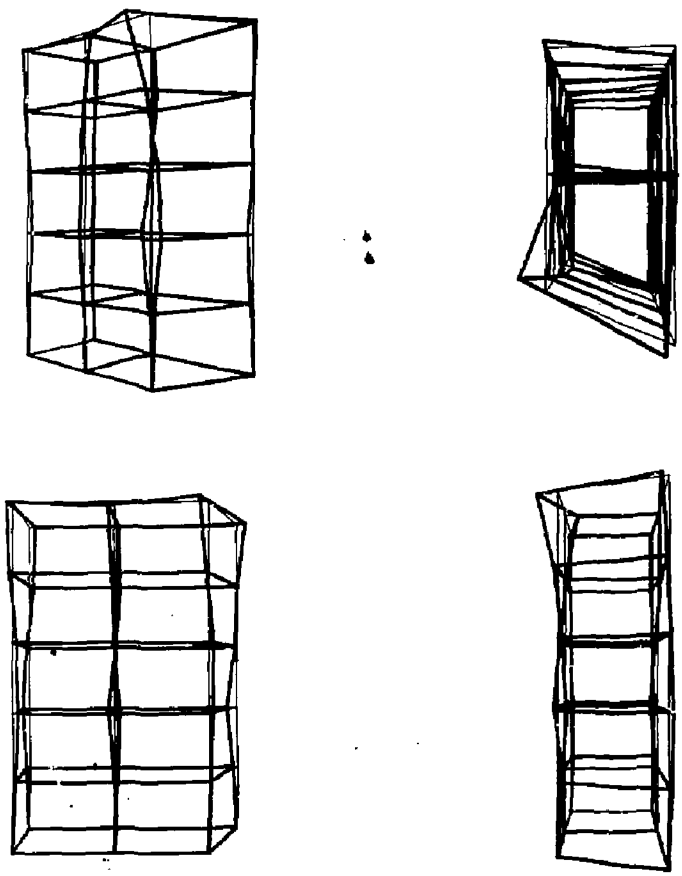

Fig. 17. Mode shape SY11 (38.4 Hz and 0.08 percent damping)

COMPARISON TO FINITE ELEMENT MODEL

The existing SAP IV finite element model (generated by the Y-Division Engineers) of the selected tower was modified and the mirror masses removed. It was then run on the new finite element code GEMINI which calculated the first five natural frequencies and mode shapes. In ascending order these frequencies were found to be 6.5, 7.1, 9.2, 21.0, and 23.2 Hz. Before we compare these to those obtained via the dynamic testing (Table I) it is instructuve to examine their associated spatial characteristics or mode shapes. These are shown in Figs. 18 to 22. As can be appreciated by comparing these figures to Figs. 7 - 11, the second finite element model mode corresponds to the first mode obtained by testing. Conversely, the first finite element model mode corresponds to the second tested mode. With this information noted we can now compare the finite element and modal test results. This is accomplished in Table II.

Table II. Comparison of finite element and modal test results.

Mode	Modal Test	Fin El Model	%-Difference[†]
SY1	5.6	7.1	23.6
SY2	6.3	6.5	3.2
SY3	8.6	9.2	6.8
SY4	19.0	21.0	10.0
SY5	21.4	23.2	8.0

[†] The percent difference between two numbers A and B is defined as $2|A-B|/(A+B)$ times 100.

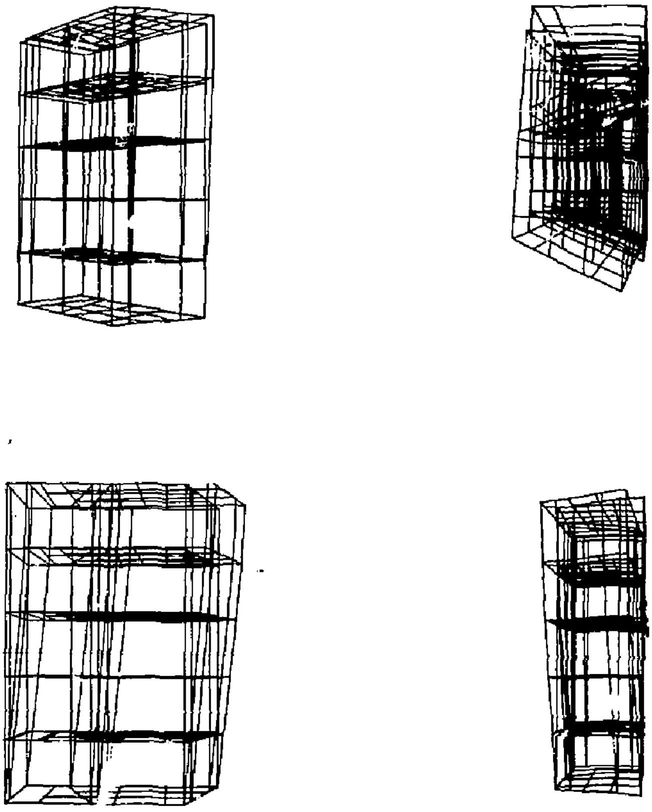

Fig. 18. GEMINI Code mode shape A (7.1 Hz).

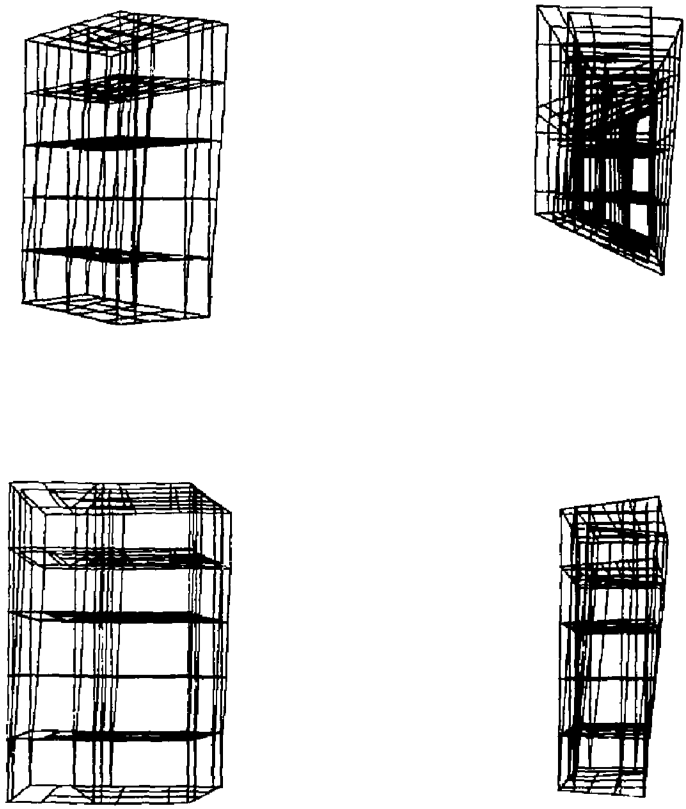

Fig. 19. GEMINI Code mode shape B (6.3 Hz).

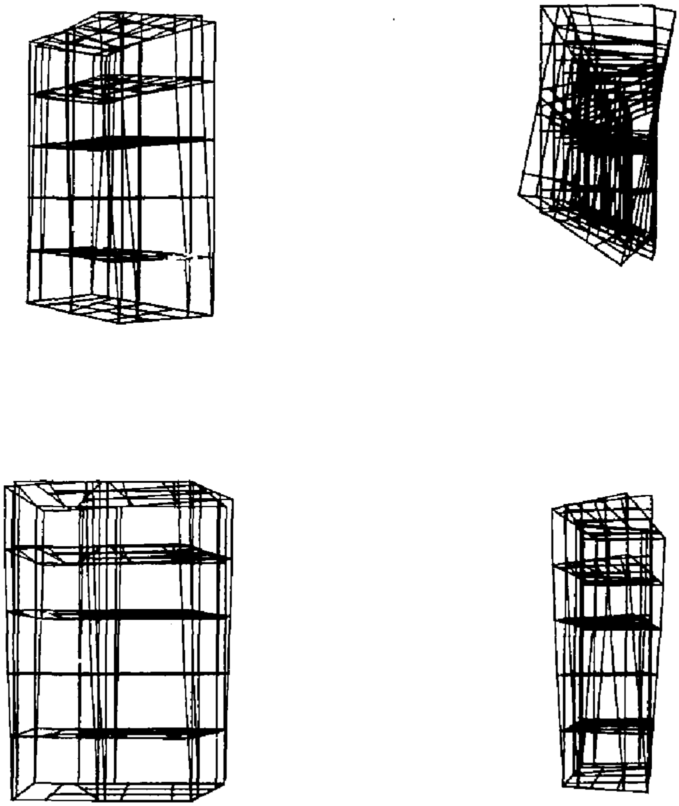

Fig. 20. GEMINI Code mode shape C (9.2 Hz).

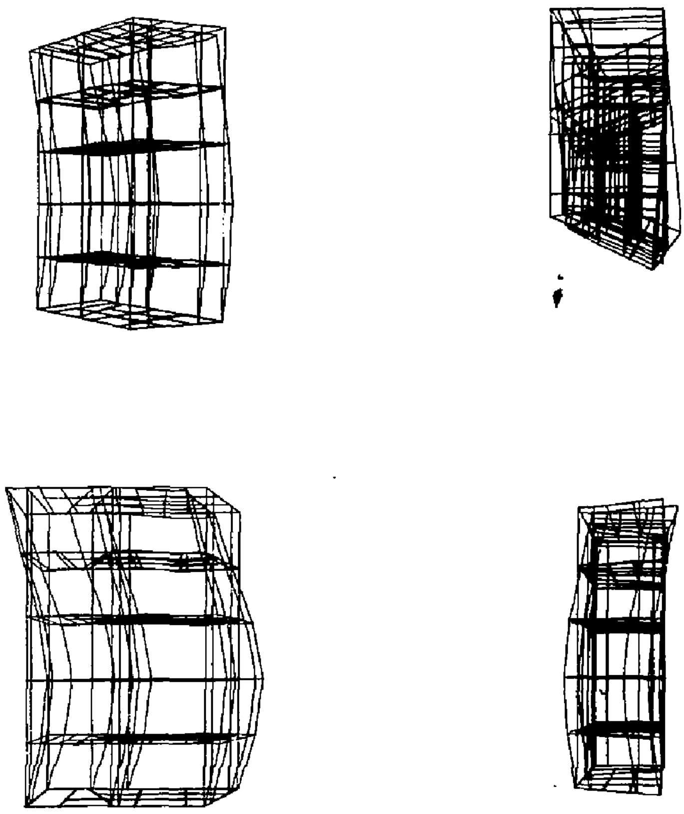

Fig. 21. GEMINI Code mode shape D (21.0 Hz).

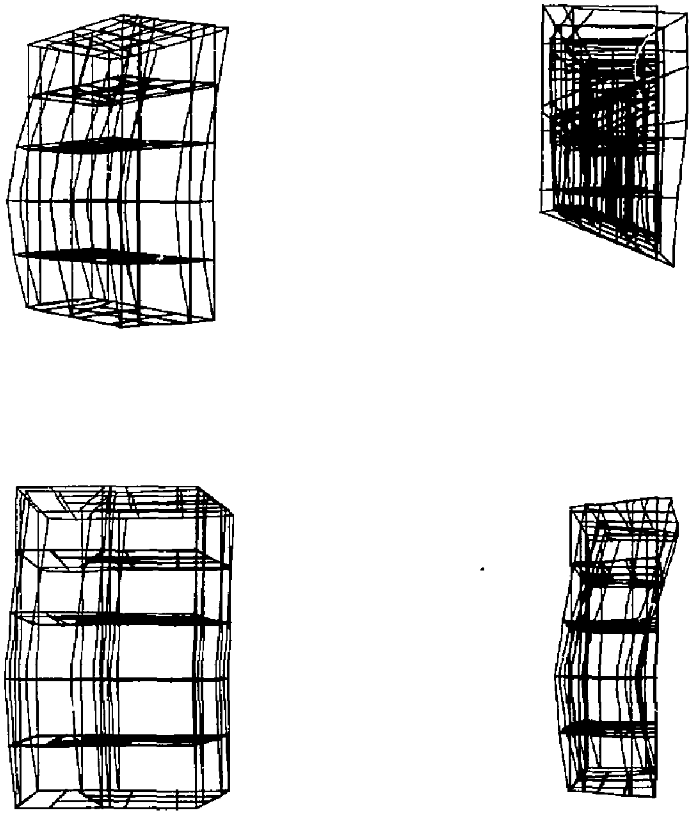

Fig. 22. GEMINI Code mode shape E (23.2 Hz).

SUMMARY AND CONCLUSIONS

With the exception of mode SY1, the agreement between the finite element model and modal analysis results is quite good This disagreement has been attributed to the fact that the original finite element model did not take into account the effect of the mirror mounts. These mounts are rectangular in shape and welded into the corners of the structure in a plane parallel to the z (short) direction. In addition to serving as a means of attaching the large turning mirrors, they also act as gussets and greatly contribute to the stiffness in this direction.

The damping values determined by the modal analysis add new information that is not available in the finite element model analysis. These damping values are certainly withing the range that would be expected for welded steel structures. Since these values are low, any excitation of the structure will cause vibrational motions that will be in evidence for several seconds after the source of excitation has been removed.

ACKNOWLEDGMENTS

The authors would like to express their appreciation to Chuck Hurley and Gary Bradley of Y-Program for their help and cooperation during the testing of their structure.

We are also grateful to Dick Miller (Y-Program) who constructed the original SAP IV finite element model for his interest and help during and after the test.

Finally, thanks to Bob Murray and Tom Nelson of the Structural Mechanics Group for their expert assistance in comparing the test results with the finite element results.

MATHEMATICAL DESCRIPTION OF DYNAMIC TESTING METHOD

To accomplish the dynamic testing of any structural system we must first construct a mathematical *transfer function model* of the structure. That is to say, we divide the structure into a series of nodes. Physically, these nodes are defined as any location where the *structure is either excited* or where the resulting *response motion* is *measured*. These transfer function nodes are analogous to the nodes of *a finite element model*, constructed for a structural analysis code (such as GEMINI).

The input excitation at any node will cause resulting response *motions at all of the other nodes. Mathematically, the response at* node i ($x_i(t)$) due to *an excitation* force applied at node j ($f_j(t)$) is described by the following expression

$$(A\text{-}a) \qquad x_i(t) = \mathcal{S}_{ij}[f_j(t)].$$

This expression tells us *that* the structure acts as a system which *maps, or converts*, the input function $f_j(t)$ at node j to a resulting output motion $x_i(t)$ at node i. When the *input function* $f_j(t)$ is an *impulse (or delta) function* $\delta(t)$, then we call the output motion at node i the impulse response motion and denote it as $h_{ij}(t)$. In terms of Eq. A-a we have

$$h_{ij}(t) = \mathcal{S}_{ij}[\delta_j(t)].$$

Using the convolution, or Duhamel, integral [1,6], we can use *this impulse response function* to describe the motion at node i due to a general input function $f_j(t)$ at node j, i.e.,

$$(A\text{-}b) \qquad x_i(t) = \int_{-\infty}^{\infty} h_{ij}(t-\xi) f_j(\xi) d\xi.$$

Equation A-b is usually written in the notationally simpler form

(A-c) $x_i(t) = h_{ij}(t)*f_j(t)$.

In the time domain the evaluation of this convolution equation is rather tedius. On the other hand, in the frequency domain it can be written in a very straightforward and convenient manner [1]. This fact can be appreciated if we make use of the convolution theorem which briefly paraphrased states, *"If $H(w)$ and $F(w)$ are the Fourier transforms of the functions $h(t)$ and $f(t)$, respectively, then the Fourier transform of the convolution product $x(t)=h(t)*f(t)$ is simply the product of the individual transforms, i.e., $X(w)=H(w)F(w)$."* Using this theorem we can rewrite Eq. A-b or A-c as

$$\text{(A-d)} \quad X_i(w) = H_{ij}(w)F_j(w),$$

where $X_i(w)$ and $F_j(w)$ are the Fourier transforms of $x_i(t)$ and $f_j(t)$, respectively. $H_{ij}(w)$ is called the transfer function between nodes i and j and is also the Fourier transform of the impulse response function $h_{ij}(t)$.

If we assume that the structure in question behaves (reasonably) linearly, then the impulse response motions $h_{ij}(t)$ and, consequently, the transfer functions $H_{ij}(w)$, can be described by the following simple equations

$$\text{(A-e)} \quad h_{ij}(t) = \sum_{k=1}^{N} {}_{ij}A_k e^{-\alpha_k t}\cos(2\pi w_k t+\varphi_k), \qquad \text{and}$$

$$\text{(A-f)} \quad H_{ij}(w) = \sum_{k=1}^{N} \frac{{}_{ij}A_k e^{i\varphi_k}}{\sigma + 2\pi i(w-w_k)}$$

$$w>0$$
$$i,j = 1, \ldots ,N.$$

In Eqs. A-e and A-f, N is 3 times the number of *nodes*[*] *used to*

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[*] The number of mode shapes that can be obtained is limited by the number of nodes used to model the structure. However, the number of modal damping and frequencies that can be obtained is not limited by

describe, or model, the structure in the x,y and z directions. Note that, as we have already mentioned, Eqs. A-e and A-f are Fourier transform pairs. That is to say, $H_{ij}(w)$ is the Fourier transform of $h_{ij}(t)$ or, equivalently, $h_{ij}(t)$ is the inverse Fourier transform of $H_{ij}(w)$. For the sake of illustration, we show a single term of the

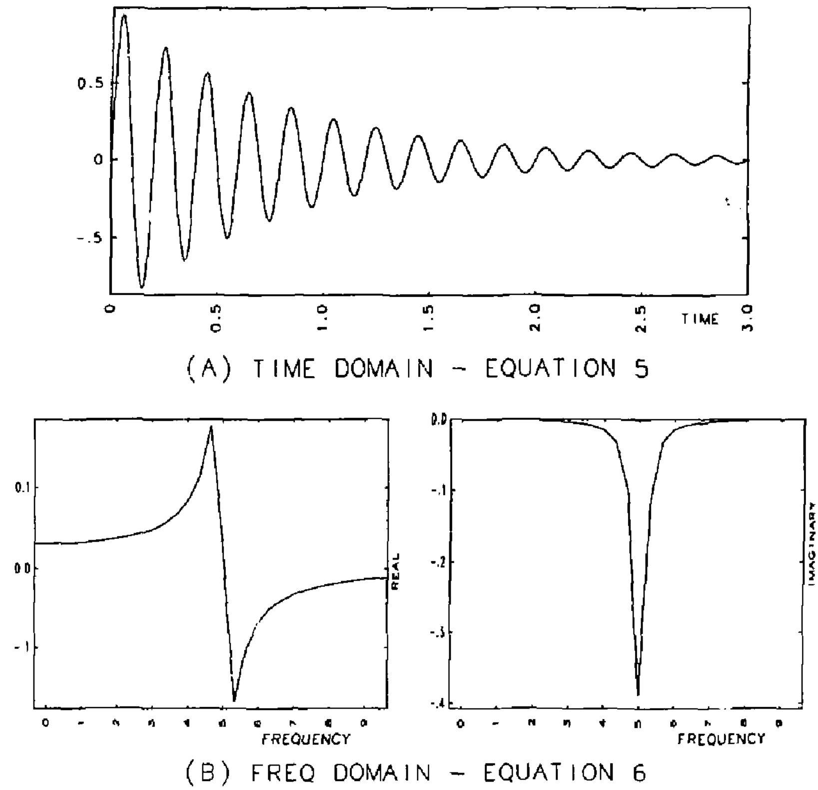

Figure 23. Damped sinusoid and its corresponding transform.

summation of Eq. A-e and its transform Eq. A-f in Fig. 23.

In our work we will deal mainly with the transfer function, Eq. A-f. In this equation there are 3 main sets of constants or parameters.

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the number of nodes but instead only by the accuracy of the acquired measurement data.

Two of them are real valued (σ_k and w_k), and the other is complex (${}_{ij}A_k \exp[i\varphi_k]$) with a real and imaginary portion. For mechanical systems these parameters have physical significance. Namely, w_k is the kth modal frequency; σ_k is the kth modal damping; and ${}_{ij}S_k = {}_{ij}A_k \exp[i\varphi_k]$ is a measure of the relative strength of the kth mode between nodes i and j. When combined for all values of i,j =1,...,N, these ${}_{ij}S_k$ parameters provide a spatial description of the kth mode (mode shape).

When we test a structure, the first task is to obtain the transfer functions between the nodes. To accomplish this we excite the structure at node j with a forcing function $f_j(t)$ that is typically a short pulse or hammer blow. We then place accelerometers at node i to measure the resulting response $x_i(t)$. Both $f_j(t)$ and $x_i(t)$ are recorded, fed into a digital computer, and Fourier transformed to obtain $F_j(w)$ and $X_i(w)$. The transfer function $H_{ij}(w)$ is then (digitally) obtained by dividing $F_j(w)$ into $X_i(w)$ (see Eq. A-d), i.e.,

$$\text{(A-g)} \qquad H_{ij}(w) = \frac{X_i(w)}{F_j(w)} \quad .$$

Since we are assuming that the structure behaves linearly, we expect this transfer function to be described by some form of Eq. A-f. That is, there exist parameters ${}_{ij}S_k$, w_k and σ_k that when used in Eq. A-f will match the measured transfer function $H_{ij}(w)$. To find these parameters we use a computer code called *TRANSF* which is available on the Livermore Time Sharing System (LTSS) [4]. This code uses a least-squares-fit technique to find the set of parameters that best matches the measured data.

At first glance it would appear that we require N^2 measurements to obtain all the transfer functions $H_{ij}(w)$ (i,j =1,...,N). However, due to certain symmetries [2], this is not the case and, in fact, we only require N measurements. There are several ways in which these N measurements can be made. Typically, for large structures, we apply the excitation at a single node (call it M) and then measure the response at the other nodes (including node M). This technique gives us the N transfer functions $H_{1M}(w)$, $H_{2M}(w)$, ..., $H_{NM}(w)$. The remaining $N^2 - N$ transfer functions can then be generated from this limited set as per the rules described in Ref. 2.

In summary, to dynamically test a structure we proceed as follows:

(1) Determine nodal locations of the structure (see following sections

for a description of NOVA nodal locations).

(2) Excite the structure at a convenient node and measure the response motions at all the other nodes.

(3) Using Fourier transform techniques, determine the transfer functions between the nodes.

(4) From these transfer functions, extract the modal parameters of the structure using the *TRANSF* computer code.

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