VIBRATION RESPONSES OF SIMPLE CURVED PANELS TO HIGH-INTENSITY RANDOM AND DISCRETE FREQUENCY NOISE

Carl E. Rucker

NASA Langley Research Center
Langley Station, Hampton, Va.

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ABSTRACT

Unexpectedly short times to failure for curved panels under acoustic loading led to detailed studies of their dynamic response characteristics with the objective of exploring the reasons for such short times to failure. Nonlinear response characteristics were observed involving significant low-frequency motions due to buckling. Such behavior resulted in a much higher percentage of large strain amplitudes than would be predicted for a normal strain amplitude distribution. The acquisition of joint strain-sound pressure distributions for significant time durations was greatly facilitated by the use of a pulse height analyzer which digitized, classified, stored, and displayed large amounts of information.

INTRODUCTION

The responses of aircraft or spacecraft structures to complex noise inputs involves such important variables as the structural materials, the fabrication techniques, and the related environmental conditions. Analytical procedures have generally not been adequate for predicting such responses, and hence, much reliance has been placed on experiments.

As part of a series of basic research studies of panel responses to noise, the effects of panel curvature have been documented. The purposes of this paper are to present some of the recent panel test analysis results and to briefly describe a unique method of collecting statistical data.

SONIC FATIGUE FAILURES

Sonic fatigue data are shown in figure 1 for three different panel curvatures for comparison. Root-mean-square strain for a strain gage near the panel edge is plotted as a function of time to failure in minutes. The excitation was a broadband random noise from an air jet. Its overall sound pressure level was 155 decibels and the spectrum peaked sharply at about 100 Hz (see refs. 1 and 2). Identical 20- by 20-inch sheets of material were formed to curved fixtures with lap attachments. Three-sixteenth-inch A&N bolts and nuts adjusted to a given torque were spaced on $1\frac{1}{2}$-inch centers with 5/8-inch edge distance. By this means, an attempt was made to minimize the edge-attachment condition differences for the test panels.

One of the main results of the above study was the relatively short times to failure of the 4-foot-radius panels even though the measured strain levels were markedly lower than for the other curvatures. It was originally suggested that significantly different stress concentration factors may have existed. The present paper, however, contains results of other studies relating to the dynamic behavior of the 4-foot-radius panel and which may also be significant in causing shorter times to failure.

Figure 2 is a photograph of a panel which failed due to sonic fatigue while formed into a 4-foot-radius configuration. It is believed that this failure resulted from test conditions for which the panel was buckled. Since buckling is a strong indicator of nonlinear behavior, the panel response was studied for other evidence of nonlinearities.

STRAIN RESPONSES

A series of dynamic response studies involving different intensities of acoustic loading were conducted, and some representative results are presented in figure 3. Overall root-mean-square strains are plotted as a function of discrete driving frequency for sound-pressure levels of 115 and 125 dB impinging on the lower surface of the panel. At the lower excitation level the panel appeared to be responding generally as a linear system. At the higher level, however, there was definite evidence of nonlinear response. The skewness of the response peaks toward lower frequencies represents a soft spring effect; that is, the panel becomes less stiff at large vibration amplitudes.

The mode shape sketches on the figure illustrate qualitatively the modal pattern variation for these two levels of excitation. The sketch at the top suggests a buckling condition such that the center portion of the panel experiences relatively large amplitude motions at frequencies other than the driving frequency.

Both analytical and experimental studies have been made for the modal response of this panel, and the results are presented in figure 4. Frequency in Hz is shown for various modal numbers (number of antinodes). Theoretical calculations assuming both clamped and simply supported boundary conditions (ref. 3) are represented by the solid and dashed curves, respectively. Experimental data obtained using discrete frequency excitation are represented by the circle points. They seem to fall close to the simply supported values at low modal numbers and close to the clamped values at high modal numbers. Note that the frequency for the fundamental mode corresponds closely to that of the sixth mode. As indicated in figure 3, the sixth mode only was excited at a level of about 115 dB, whereas at a 120-dB level and higher, the panel snapped into a buckled condition for which the fundamental and sixth modes are superimposed (see upper sketch of fig. 3).

Additional panel response data for high levels of noise excitation are presented in figures 5 and 6. In figure 5 the mean square bending strains are shown as a function of frequency. The discrete frequency excitation (at a 145-dB level) was provided by a siren for which the harmonic content was at least 40 dB lower in level than the excitation frequency of the figure. The data shown were obtained by means of a 15-second tape loop and narrow band filters. Note that relatively strong responses of the panel occur at frequencies lower than the exciting frequency.

The response of the panel to broadband noise is shown in figure 6. Again, mean-square bending strain per unit bandwidth is plotted as a function of frequency. The spectrum shape of the 150-dB level random noise is shown at the top of the figure. Note that this nearly flat random noise spectrum was generated by a unique jet turbulator nozzle represented by the sketch at the right. A number of relatively strong responses are observed at the low frequencies even in the range where the input spectrum tends to drop off. These latter response data are thus consistent with those of figure 5.

AMPLITUDE DISTRIBUTIONS

In order to study the statistical behavior of the panel, a unique method was made use of to collect and analyze appropriate sound pressure and associated strain data. Figures 7 and 8 illustrate the method used, and figure 9 contains the main results.

Included in figure 7 are schematic representations of the root-mean-square sound pressure and total panel strain time histories. In the course of this study the amplitude distributions were obtained at several arbitrary input noise loading levels. Such a procedure could be accomplished by the reading of oscillograph records at the proper times, as indicated schematically in figure 7. In order to automate the process of accumulating data of this type, however, a pulse-height analyzer was used in the manner suggested by the diagrams of figure 8. Records such as those of figure 7 were digitized about 300 times per second during their 80-minute durations (time to failure of the panel). The analyzer operates in such a way that all strain values associated with a given sound pressure are grouped together. Thus, it is possible to determine amplitude distribution directly from the analyzer.

Such a display is illustrated in figure 8 which contains a cathode ray oscilloscope presentation of the strain and sound-pressure data. The abscissa represents panel strain, the zero value being in the center and the negative and positive values being to the left and right, respectively. The ordinate is root-mean-square sound pressure; the vertical coordinate represents the number of measurements for given values of sound pressure and panel strain. Thus, at a given sound-pressure value, the display indicates the number of strain samples at each strain value for the entire time of the data recording. The type of display illustrated in the figure is useful qualitatively, but the numerical data are obtained directly from the tabulation circuits.

Sample strain amplitude distribution data as measured with the pulse-height analyzer are presented in figure 9 for the 4-foot-radius panel. The distribution shown is for a sound-pressure level of 157 dB and contains over 79,000 samples. Also shown is a solid curve representing a normal or gaussian distribution. The probability of being equal to or less than a given value of strain is plotted on the vertical scale for various multiples of standard deviation (σ). It can be seen that the measured data generally follow the normal distribution curve up to nearly 2σ and then deviate from the normal distribution curve at higher values. This result implies that a greater percentage of the panel lifetime is spent at strain values above 3σ than would be the case for a normal distribution of strain amplitudes. Although not shown on the figure, similar data for the other two panel curvatures of figure 1 fell generally along the gaussian curve at the higher σ values. The implication from these data is that the 4-foot curvature panel, probably because of its nonlinear behavior characteristics, experienced an abnormally high number of high strain values. These strain peaks may account for the shorter times to failure of these panels.

CONCLUDING REMARKS

Unexpectedly short times to failure for curved panels under acoustic loading led to detailed studies of their dynamic response characteristics. Nonlinear response characteristics were observed involving significant low-frequency motions due to buckling. Such behavior resulted in a much higher percentage of large strain amplitudes than would be predicted for a normal strain amplitude distribution.

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2. P. M. Edge, Jr., and Carl E. Rucker, "Response-to-Noise Studies of Some Aircraft and Spacecraft Structures," Acoustical Fatigue in Aerospace Structures, W. S. Trapp and D. M. Forney, Jr., eds., Syracuse Univ. Press, 1965, pp. 404-408.
3. John L. Sewall, "Vibration Analysis of Cylindrically Curved Panels With Simply Supported or Clamped Edges and Comparisons With Some Experiments," NASA TN D-3791, Table IVb, p. 38, and Figure 5, p. 49, 1967.

LIST OF FIGURES

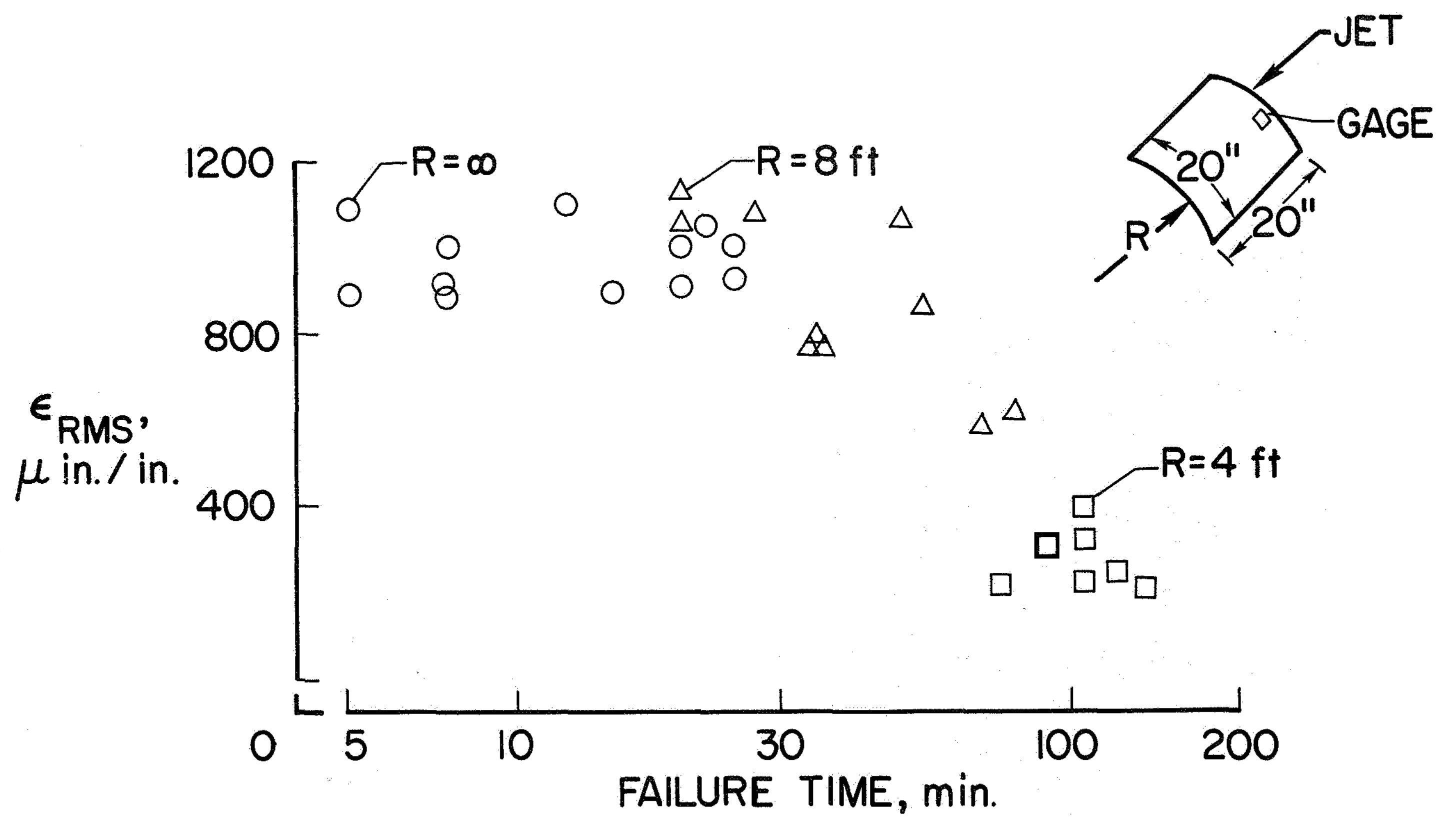

Figure 1.- Time to failure for 0.020-inch-thick aluminum-alloy panels of three curvatures in a random noise field having an overall sound pressure level of 160 dB.

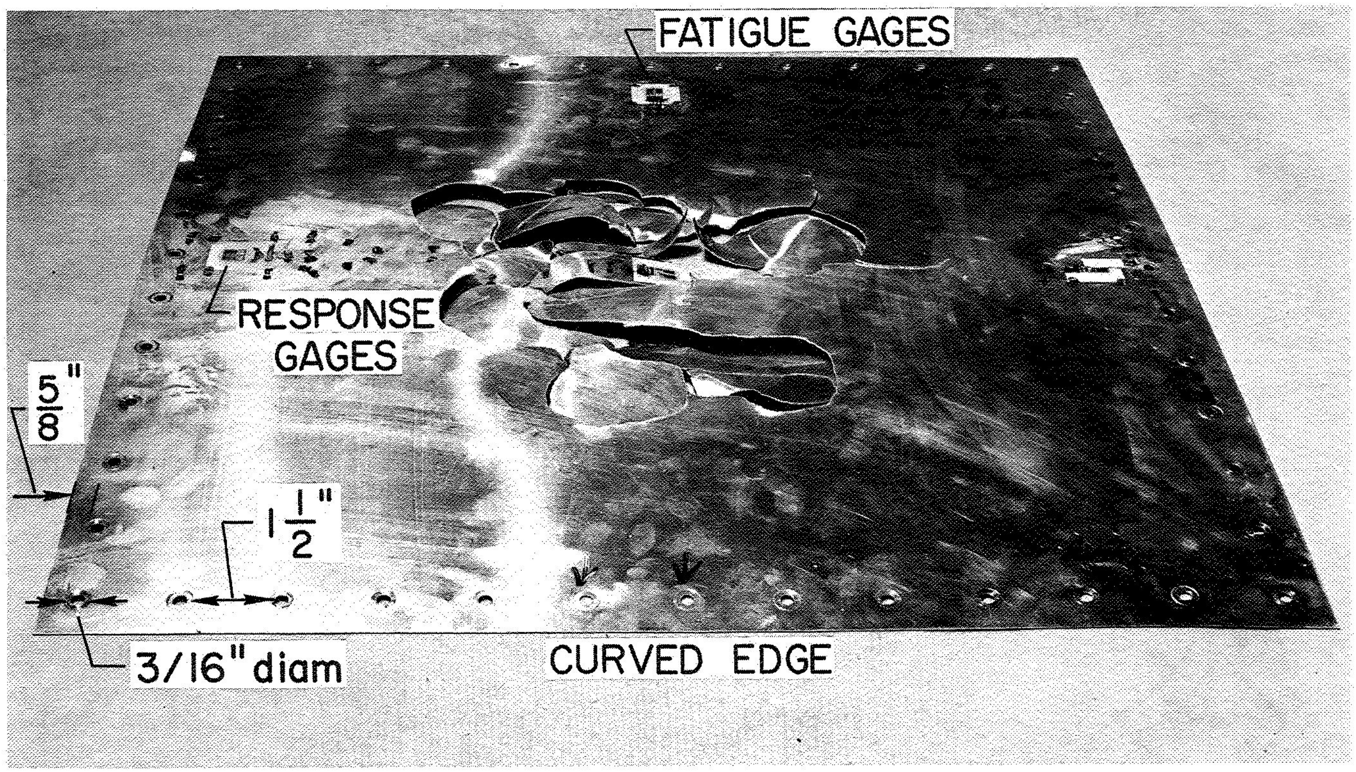

Figure 2.- Photograph of 0.020-inch-thick aluminum-alloy panel of 4-foot radius after sonic fatigue failure due to siren excitation.

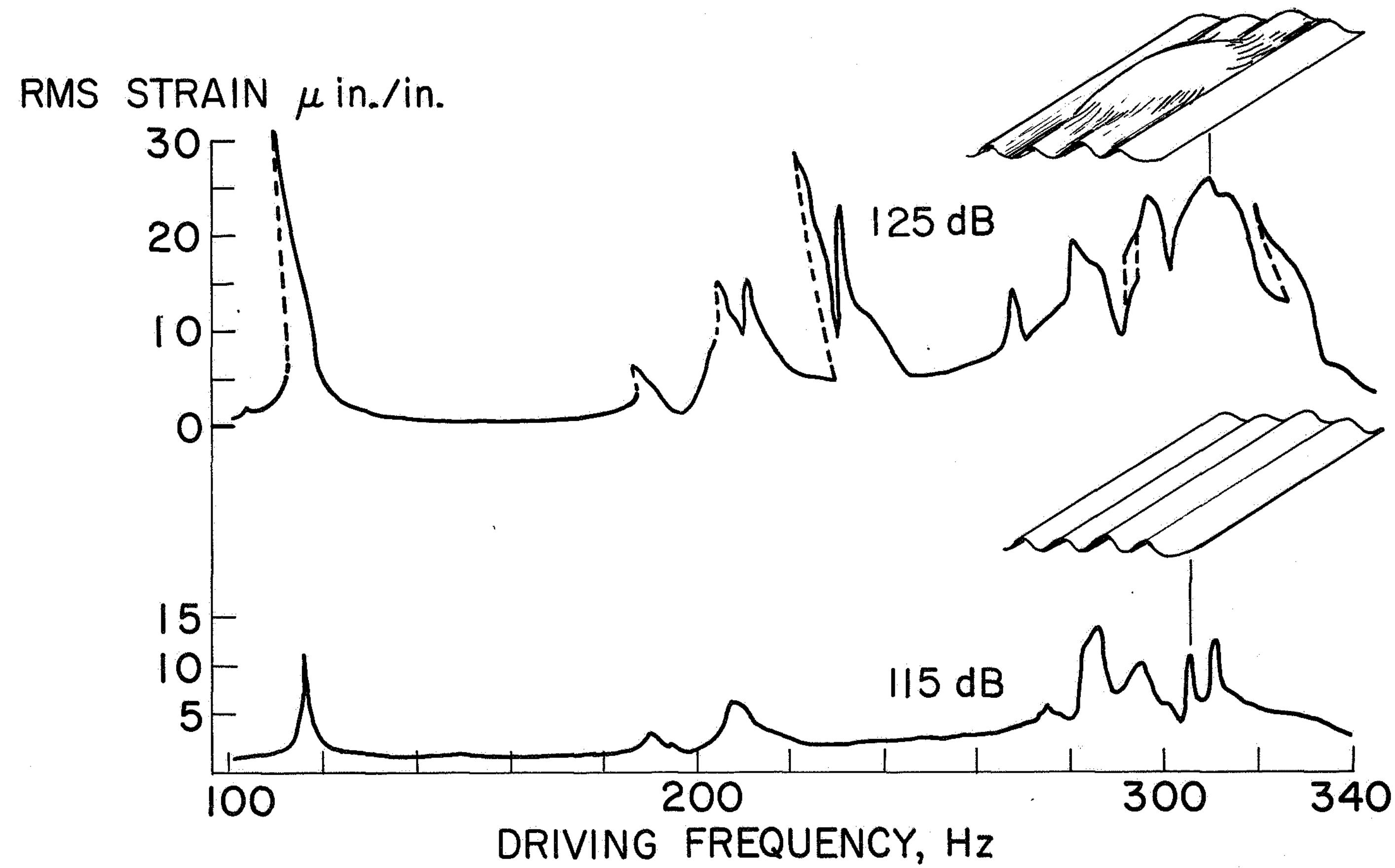

Figure 3.- Overall strain responses of a 4-foot radius panel as a function of driving frequency for two different levels of discrete frequency excitation.

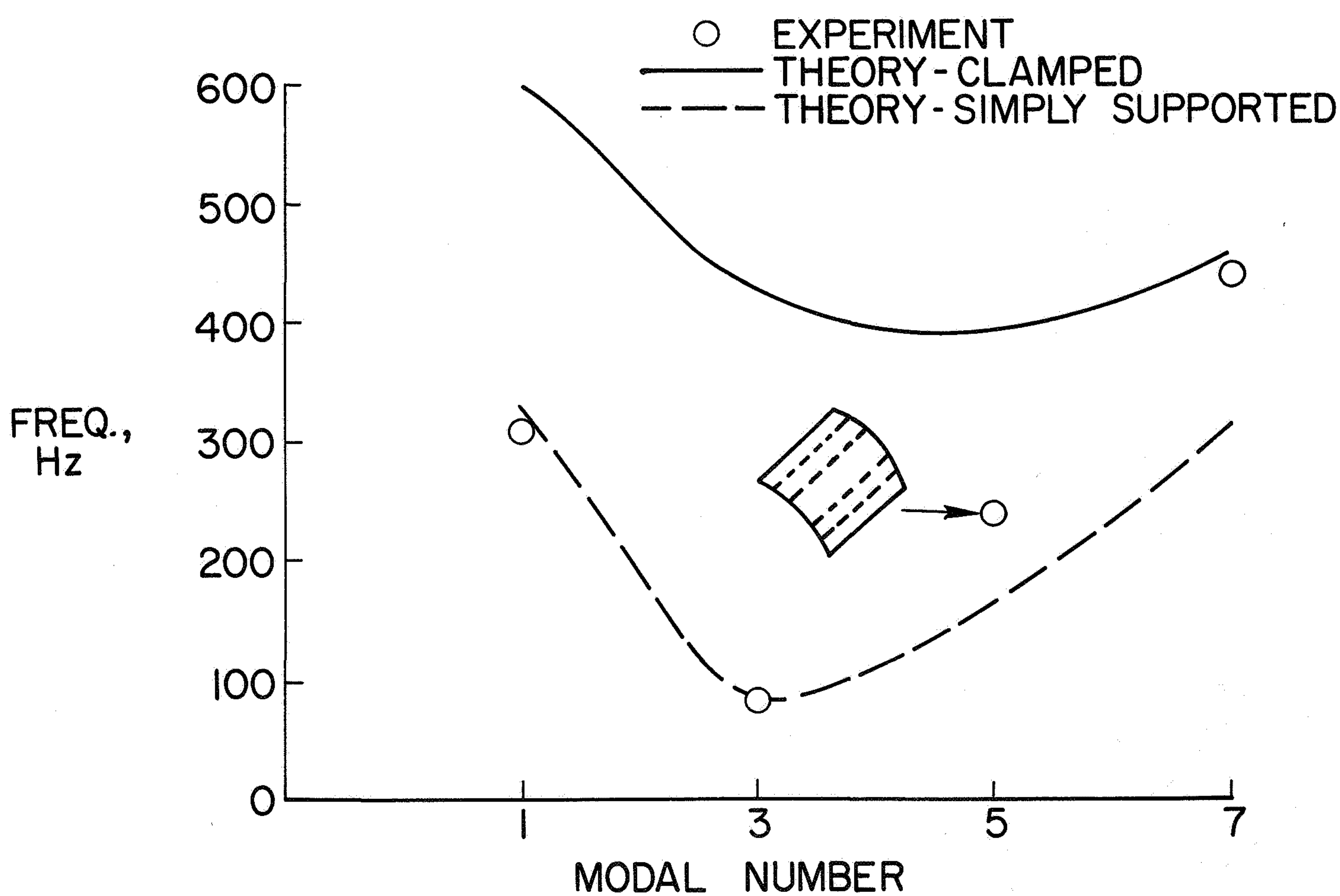

Figure 4.- Some modal responses of curved aluminum panel (R = 4 ft) excited by discrete frequency noise.

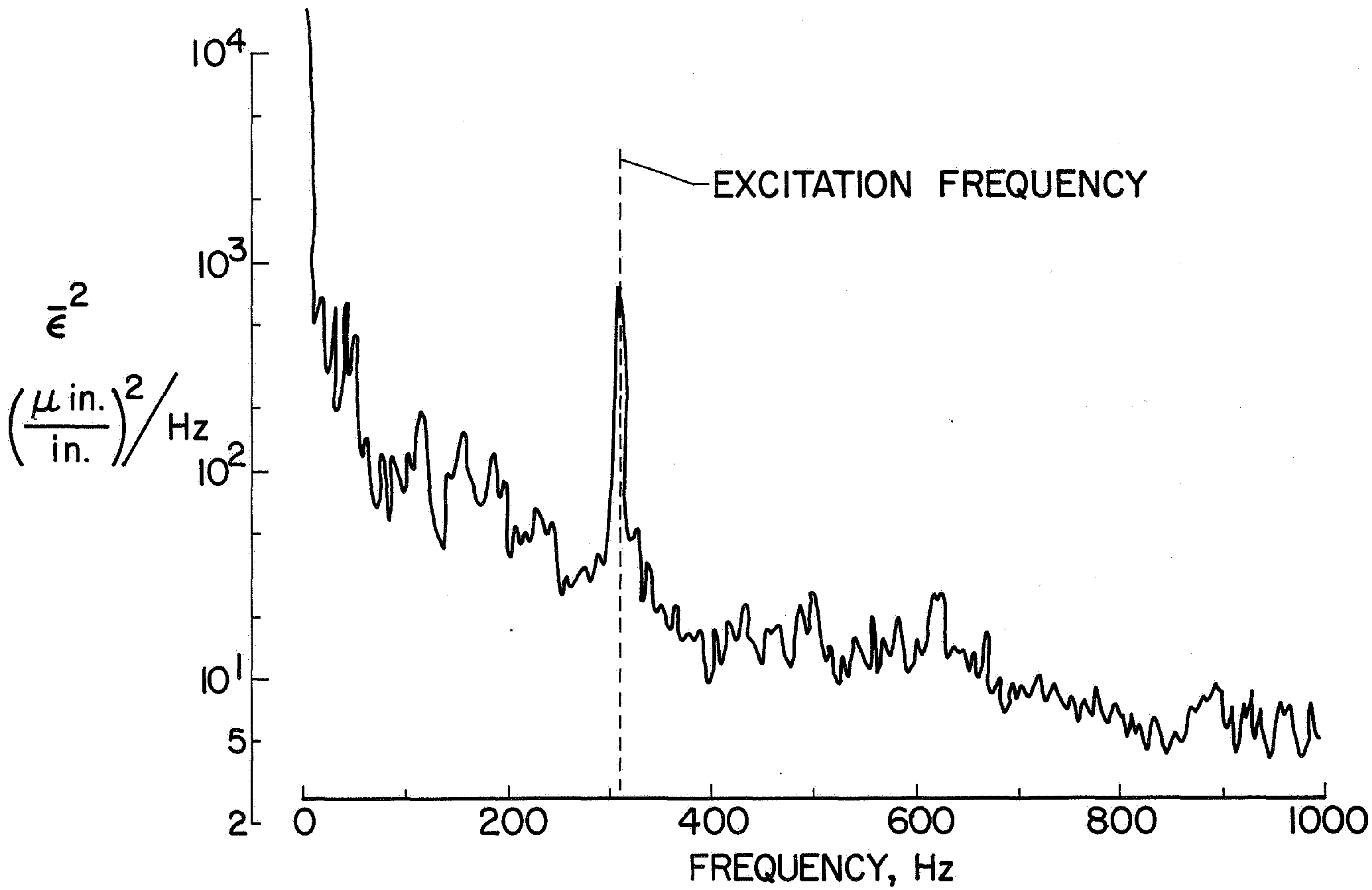

Figure 5.- Spectrum of bending strain response of 4-foot radius curved panel due to discrete frequency excitation at 145-dB sound pressure level.

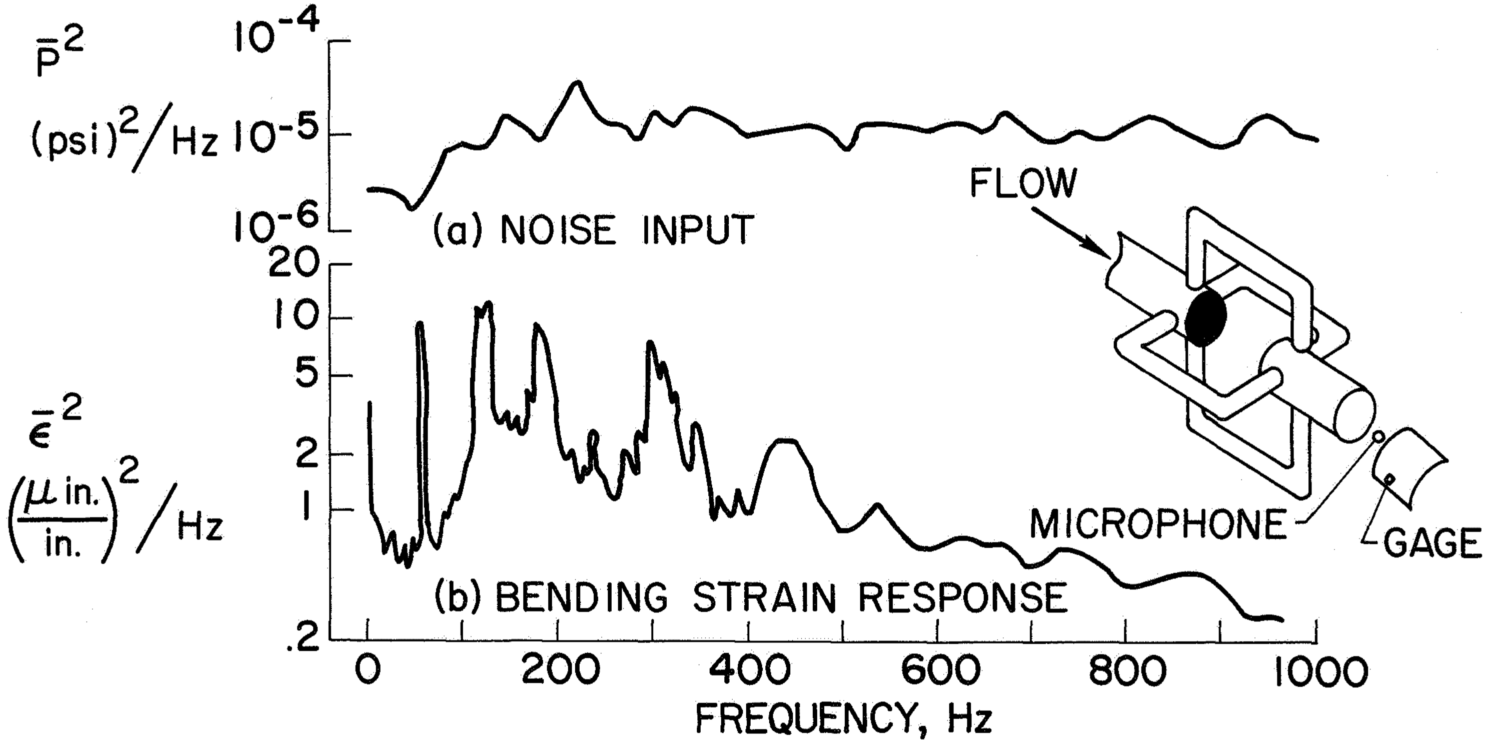

Figure 6.- Bending strain response spectrum of a 4-foot radius panel due to random noise input from a four-branch air jet noise source.

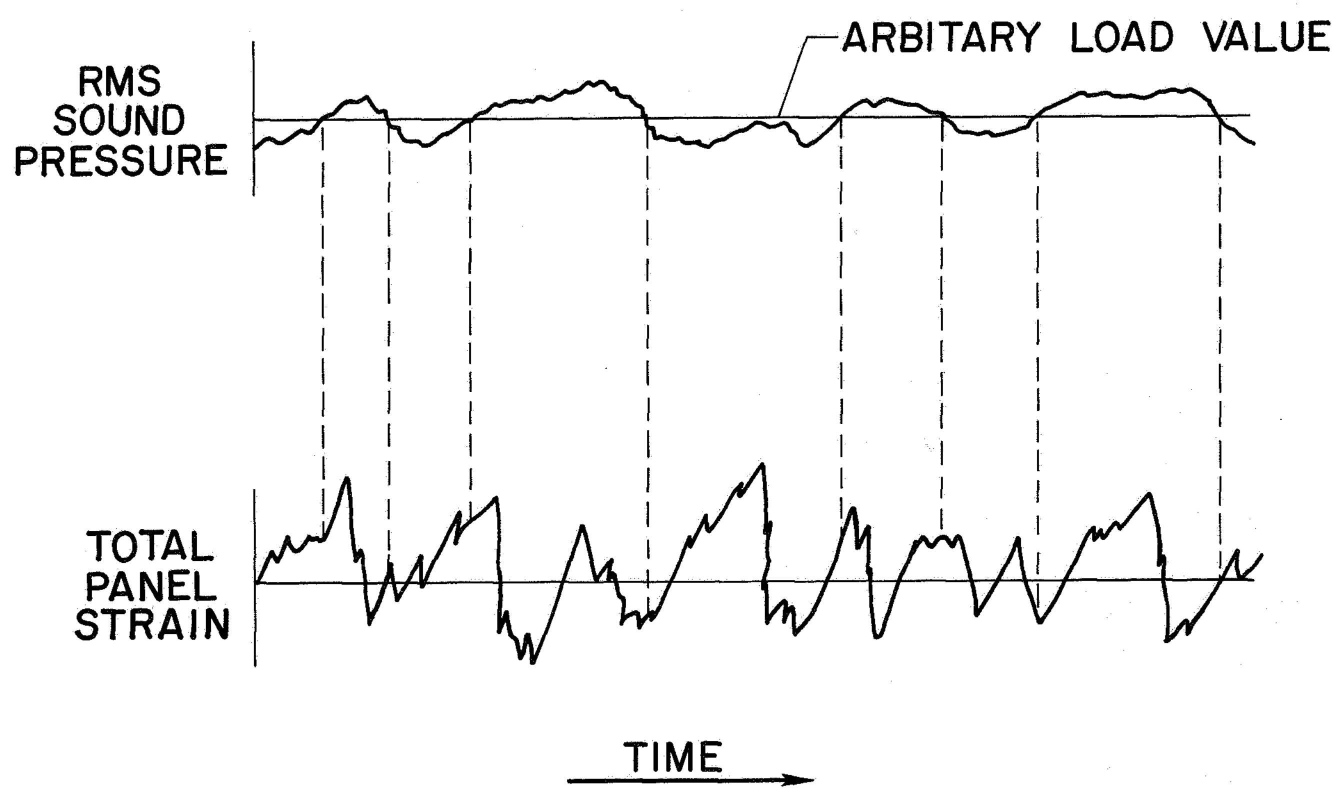

Figure 7.- Schematic diagram of the time histories of the root-mean-square sound pressure and associated total panel strain. Vertical dashed lines indicate points at which strain values were determined for an arbitrary sound pressure load value.

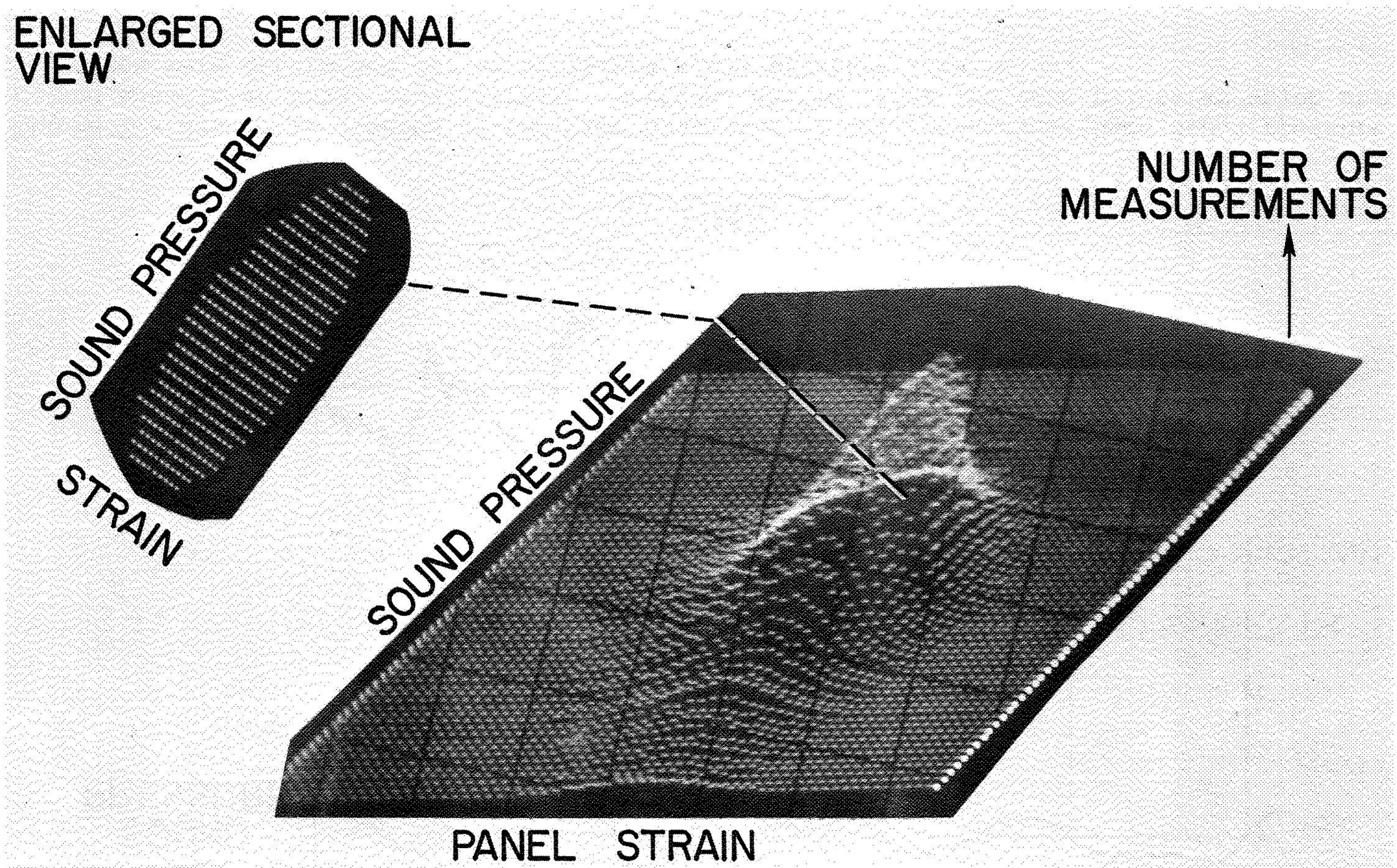

Figure 8.- Photograph of oscilloscope display of a pulse height analyzer used for accumulation of joint probability data.

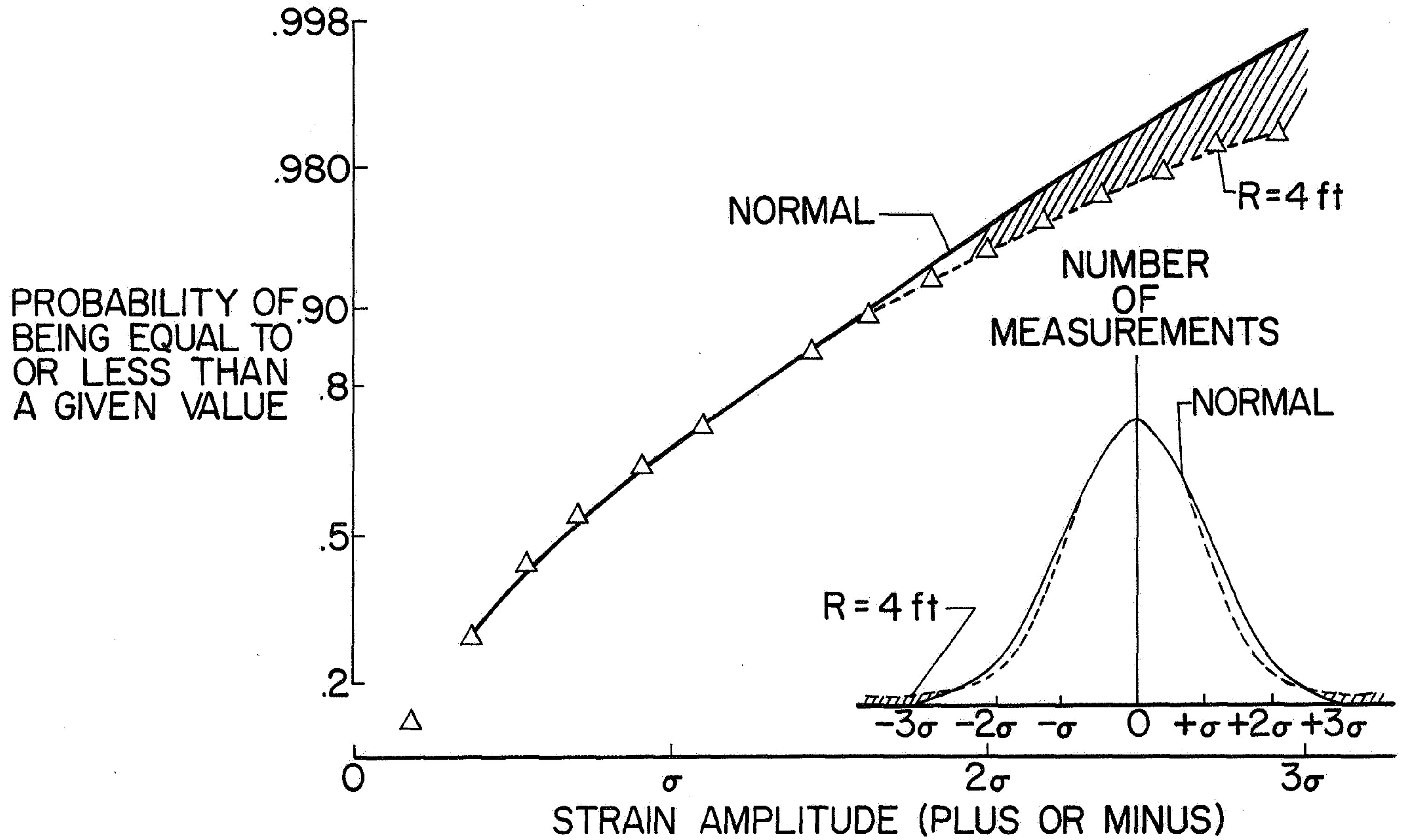

Figure 9.- Total strain amplitude distribution for 4-foot radius panel due to random noise loading compared to a normal distribution.

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