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EXPERIMENTAL STUDY OF THE RANDOM VIBRATIONS OF AN AIRCRAFT STRUCTURE EXCITED BY JET NOISE

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FOREWORD

This report was prepared by the University of Southampton, U. K., Department of Aeronautics and Astronautics, under USAF Contract AF 61(052)-332. The contract was initiated under Project No. 7351, "Metallic Materials", Task No. 73521, "Behavior of Metals", and it was administered by the European Office, Air Research and Development Command. The work was monitored by the Materials Central, Directorate of Advanced Systems Technology, Wright Air Development Division, under the direction of Mr. W. J. Trapp.

This report covers work performed during the period of July 1959 to December 1960.

ABSTRACT

Recordings have been made of the strains induced in a full scale rear fuselage test structure of the Caravelle airliner when one jet engine is running at maximum take-off thrust. The structure is a conventional sheet-stringer combination attached to pressed out frames.

The analysis has been concentrated on the strains in the centres of panels. Correlation measurements have indicated that the lower frequencies (up to 500 c.p.s.) are associated with overall vibration modes and have low strain amplitude. The larger panel strains occur at higher frequencies with the frames acting as boundaries. In these measurements the main resonance peak in each panel occurs at about 600-700 c.p.s. and has been identified with the fundamental stringer twisting mode (i.e., adjacent panels 180° out of phase). There are generally two smaller peaks in the 800-1000 c.p.s. range but the modes of vibration have not been completely identified due to lack of information.

An attempt has been made to calculate the panel resonant frequencies theoretically, assuming that the frames act as boundaries. Although this work appears promising it has not yet progressed far enough for any definite conclusions to be drawn from it.

PUBLICATION REVIEW

This report has been reviewed and is approved.

FOR THE COMMANDER:

W. J. Trapp

W. J. TRAPP
Chief, Strength and Dynamics Branch
Metals and Ceramics Laboratory
Materials Central

TABLE OF CONTENTS

1. Introduction

In considering the vibrations of a continuous structure excited by random acoustic pressures, Powell[1] has shown that mathematically the total response can be written down in terms of the response in each normal mode. This form of analysis could be applied directly if the normal modes of vibration of the structure were know. Unfortunately, it is not yet possible to calculate theoretically the frequencies and damping of the normal modes of vibration of a typical aircraft structure. This paper discusses experimental methods which have been used to indicate the type of vibration which is taking place when a structure is subjected to random pressure loads. The response of the structure is multimodal in form and hence for a thorough understanding of the vibration it is essential to know the phase relationships of the deflection or stress at each part of the structure. To do this it is necessary to isolate each resonant frequency and measure its associated mode shape. This knowledge can then be used to assess the validity of the types of mathematical approximations which can be made to render possible the estimation of normal modes and frequencies. This information is also required if damping materials are to be used in the most efficient way.

These phase measurements can be made by using a reference strain gauge say in the centre of a panel and other gauges for comparison on adjacent panels and support structure. To make the comparison between any two gauges the signals must first be filtered so that only one of the resonances is studied at a time. In a random vibration the phase relationship will not in general be constant for the whole of the time

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but will vary randomly such that on the average the signals may be exactly in phase for a certain proportion of the time. This now requires a statistical measurement and the best way to do this accurately is to measure the correlation between the two strains. As we are only interested in the percentage time that the two strains are either in or out of phase the correlation can be normalised to give the correlation coefficient. On this basis a correlation coefficient of + 0·9 for example, means that in the particular frequency band being investigated, the strains are exactly in phase for 90% of the time. If the strains were out of phase the correlation coefficient would be negative.

2. Test Installation

The results and analyses reported in this paper refer to strains induced in the rear structure of the Caravelle aircraft when the engines are run at full thrust for take-off. The test section and engine installation is shown in Figure 1. In this test installation a section of the aircraft consisting of all the structure aft of the pressure dome was mounted at the correct height above the ground and one engine was fitted. This engine was run at full take-off thrust and strain recordings made on magnetic tape. Structure aft of the jet nozzle includes part of the rear fuselage (unpressurized) and also the fin, rudder, tailplane and elevator. This structure is conventional sheet stringer construction as shown in Figure 2. Panel strain gauges were positioned as indicated in Figure 2; and in addition gauges were positioned along the frame flange and stringer flange over several bays in order to pick out any overall vibration which may have been present.

In this position behind the jet the noise pressures have a wide spectrum with peak intensity in the region 700 - 800 c.p.s. Thus a wide range of structural resonances are excited and the structure vibrates in a random manner.

The main part of the analysis referred to in this paper has been devoted to a study of the panel gauge group and this will be discussed in detail to outline the method of analysis and the conclusions which can be drawn from the results.

3. Analysis of Experimental Results

3.1 Method of Analysis

Random strains can only effectively be described by their power spectrum and amplitude distribution. The distribution of strain amplitude has been checked by Sud-Aviation and it has been found to closely approximate to Gaussian. For most applications, therefore, it will be adequate to assume a Gaussian distribution. The power spectrum of any particular measured strain can be found by using narrow band filters and measuring the power in each frequency band throughout the range of interest. An alternative method to that of filtering is to measure the auto-correlation function and then Fourier transform this to give the power spectrum. The relationships governing this transformation are as follows:-

If the two stresses are represented as functions of time, $\sigma_1(t)$ and $\sigma_2(t)$, then the cross correlation $R_{12}(\tau)$ is defined as the average of their product when a time delay τ is inserted in one signal, i.e.,

$$R_{12}(\tau) = \lim_{T \to \infty} \frac{1}{T} \int_0^T \sigma_1(t)\, \sigma_2(t + \tau)\, dt$$

For random stress functions this must be measured by continuously multiplying the two signals together and integrating the output. The integration time must be long compared with the period of the lowest frequency component present in the signal.

The real part of the cross power spectrum $C_{12}(f)$ is now obtained by a Fourier cosine transformation of the cross correlation $R_{12}(\tau)$

i.e. $$C_{12}(f) = 2 \int_{-\infty}^{+\infty} R_{12}(\tau) \cos 2 \pi f \tau \, d\tau$$

In the special case where $\sigma_1(t) = \sigma_2(t)$, $R_{12}(\tau)$ becomes the auto-correlation and $C_{12}(f)$ becomes the power spectral density $w(f)$. Since the auto-correlation curve is symmetrical about $\tau = 0$

$$C_{11}(f) = w(f) = 4 \int_{0}^{\infty} R_{11}(\tau) \cos 2 \pi f \tau \, d\tau$$

The limitations on this process (for obtaining power spectra) are that the transformation involves an infinite integral which requires an infinite time delay. In practice, however, the auto-correlation function $R_{11}(\tau)$ must be truncatated at some point τ_{max}. If the correlogram has reached steady zero values by this point (as in the case for broad band noise), then the transformation can be carried out. However for the type of spectrum which has narrow peaks the correlogram continues to oscillate for a considerable time and it may be necessary for practical reasons to truncate it before steady zero values have been reached. If this is transformed, a mathematical filter, or spectral window, has been effectively introduced, due to the fact that a weighting function has been used on the correlogram.

This function $D(\tau)$ has the form $D(\tau) = 1, \quad 0 < \tau < \tau_{max}$

$$D(\tau) = 0, \qquad \tau > \tau_{max}$$

The result is that the apparent spectral density is

$$P(f) = 4 \int_0^{\tau_{max}} D(\tau)\, R_{11}(\tau) \cos 2\pi f \tau \, d\tau$$

$$= \int_0^{\infty} Q(f)\, w(f)\, df$$

The function $Q(f)$ is sometimes known as a spectral window[2] and has the form shown in Figure 3. The main features of this window are that it has negative values reaching -0·2 for some frequencies, and that the width of the filter is directly proportional to the maximum time delay. The negative portion can be removed at the expense of bandwidth by using Bartlett's[3] weighting function which has the form:

$$D(\tau) = 1 - {}^{\tau}/\tau_{max} \qquad \text{for} \quad 0 < \tau < \tau_{max}$$

and $$D(\tau) = 0 \qquad \text{for} \quad \tau > \tau_{max}$$

To illustrate the use of these relationships and spectral windows, both weighting functions have been used on the strain signals.

3.2 Power Spectra

The characteristics of the forcing noise pressures on the structure are shown in Figure 4. It can be seen that the auto-correlogram does not oscillate and therefore no weighting function is necessary. The power spectrum obtained by Fourier transformation is relatively flat showing that energy is avilable to excite the structure over a wide range of frequencies. This pressure was measured by a microphone adjacent to gauge 2 outside the fuselage. A microphone was also placed in a corresponding position at the other side of the jet in the free field but no significant difference in spectrum was observed.

The auto-correlogram of the strain measured by gauge 2 is shown in part in Figure 5. In this case the auto-correlation function continues to oscillate for a considerable time and it must be truncated before transformation. The resulting power spectra using both types of spectral window are shown in Figure 6. As a result of the negative parts of the first spectral window the power spectrum given by this shows several spurious secondary peaks and troughs. These spurious secondary peaks are eliminated by use of the second spectral window as given by Bartlett's weighting function. We shall therefore call the first spectrum unsmoothed and the second, smoothed. The choice of spectrum to be used now depends on the purpose to which the spectrum is to be put. To give a general picture the smoothed spectrum is probably the best but if it is required to study a narrow peak in detail, then the unsmoothed one may be the most suitable. For example, we can estimate the damping of the 590 c.p.s. resonance by measuring the width of the peak, but the accuracy of this measurement depends critically on the width of filter we have used for obtaining the spectrum and hence the narrower band unsmoothed spectrum is to be preferred. By this method the damping ratio of the 590 c.p.s. mode is estimated to be 0·018. This will be a slight overestimate because of the finite width of the filter.

Figures 7, 8 and 9 show the corresponding strain spectra of panels 5, 7 and 4. It is seen that the principal resonant frequencies are 720, 705 and 720 c.p.s. respectively. The only variation in dimensions of panels 2, 5 and 7 is in the distance between the stringers, and this is not thought to be sufficient to cause the lower frequency of panel 2 when compared with panels 5 and 7.

Generally, it can be seen from the strain spectra that there is very little contribution from the frequency range up to about 500 c.p.s. There is then one main peak at about 600 - 700 c.p.s., and several smaller peaks in the range 700 - 1000 c.p.s.

3.3 Damping of the Normal Modes

The following damping ratios were estimated for the main resonance peaks from the unsmoothed strain spectra.

	Resonant Frequency	Damping Ratio
Panel 2	590 c.p.s.	0·018
Panel 5	720 c.p.s.	0·018
Panel 7	705 c.p.s.	0·018
Panel 4	720 c.p.s.	0·017

An alternative method of estimating the damping of a pronounced resonant peak in a spectrum is to make use of the single degree of freedom theory. When a lightly damped single degree of freedom system is excited by broad band noise the response takes on the form of a sine wave of randomly varying amplitude. The auto-correlation of this strain response has been shown by Crandall[4] to be given by

$$R_x(\tau) = \int_0^\infty h(\tau_1)\, d\tau_1 \int_0^\infty h(\tau_2)\, R_f\,(\tau + \tau_1 - \tau_2)\, d\tau_2$$

where $R_f(\tau)$ is the auto-correlation function of the applied force and $h(\tau)$ is the response of the system to a unit impulse. The impulse response function $h(\tau)$ is given by:

$$h(\tau) = \frac{\omega_n e^{-\zeta \omega_n \tau}}{\sqrt{1 - \delta^2}} \sin \sqrt{1 - \delta^2}\, \omega_n \tau$$

$$\simeq \omega_n e^{-\zeta \omega_n \tau} \sin \omega_n \tau \quad \text{for small damping}$$

where ω_n = undamped natural frequency of the system

ζ = damping ratio

For white noise excitation the power spectral density of the applied force is

$$W_f(\omega) = W_0 \quad \text{(constant)}$$

Thus

$$R_f(\tau) = \frac{1}{2\pi} \int_{-\infty}^{+\infty} W_0 \, e^{i\omega\tau} \, d\omega$$

$$= W_0 \, \delta(\tau) \quad \text{where } \delta(\tau) \text{ is the delta function}$$

$$= 0, \; \tau \neq 0$$

$$= \infty \; \tau = 0$$

The auto-correlation function of the response is therefore

$$R_x(\tau) = \int_0^\infty h(\tau_1)\, d\tau_1 \int_0^\infty h(\tau_2) \; W_0 \; \delta\,(\tau + \tau_1 - \tau_2)$$

$$= W_0 \int_0^\infty h(\tau_1)\, h(\tau + \tau_1)\, d\tau_1$$

$$= W_0 \; \omega_n^2 \, e^{-\zeta\omega_n\tau} \int_0^\infty e^{-2\zeta\omega_n\tau_1} \sin \omega_n \tau_1 \sin \omega_n (\tau + \tau_1)\, d\tau_1$$

$$= W_0 \; \omega_n^2 \, e^{-\zeta\omega_n\tau} \left[\frac{\cos \omega_n \tau}{4 \zeta \omega_n} - \frac{\sin \omega_n \tau}{4 \omega_n} \right]$$

$$\simeq \frac{W_0}{4} \frac{\omega_n}{\zeta} e^{-\zeta\omega_n\tau} \cos \omega_n \tau \quad \text{as a first approximation for small damping}$$

Thus the auto-correlation function of the response of a lightly damped single degree of freedom system to white noise approximates to a damped cosine wave. The damping of this auto-correlogram gives directly the damping ratio of the system. In practice it is not necessary to have true white noise excitation but rather broad band noise which has constant power spectral density over the frequency range of appreciable response of the system.

This principle can be used in the case of a multi-modal response to broad band noise if the resonant peaks in the response spectrum are sufficiently widely spaced to enable them to be separated. As an illustration of this the strain signal from panel 2 was passed through a filter $^1/3$ octave wide, centred at 590 c.p.s. The auto-correlogram of this filter output is shown in Figure 10. The damping ratio given by this curve is 0·016 which compares well with the over-estimate of 0·018 which was obtained by measuring the width of the resonant peak.

3.4 Filtered Cross-Correlation

Figure 11 shows the results of correlating filtered strain signals from pairs of panel gauges. The aim is to filter out each resonant frequency and by measuring the correlation coefficient, determine the phase relationship between the two strains. A pair of matched $^1/3$ octave filters were used to do this preliminary work and the results for the panel group are shown in the two diagrams in Figure 11. The filters are rather wide for this particular spectrum but the main points of interest are essentially:

1. In the two lower frequency bands (160 c.p.s., 400 c.p.s.) the strain correlations are high in both the vertical and horizontal

directions indicating that some overall form of vibration is taking place.

2. In the two higher frequency bands which include the main panel response frequencies the correlation in the vertical direction is much greater than that in the horizontal direction. This indicates that the frames are acting as vertical boundaries but that strains are correlated over several stringer bays in the vertical direction between the frames.

Figure 12 shows similar filtered cross-correlations for the bending moment in the frame mid-way between stringers. The high correlations indicate that in this direction the vibration maintains its dependence over several stringer bays. The negative values indicate where the phase change occurs and so show up the mode shape.

3.5 <u>Cross Spectra</u>

A more elegant and meaningful way of obtaining the correlation and hence phase relationship between two strains at any frequency is to measure the cross-correlation function and then transform this to give the real part of the cross power spectrum. The cross-correlogram for strains 2 and 5 is shown in part in Figure 13. This is unsymmetrical, does not have a maximum value equal to unity, and continues to oscillate for a considerable time. The computed cross-spectrum is shown in Figure 14, and here negative values of the cross-spectral density indicate that the strains are out of phase. No attempt has been made to use any smoothing process in the transformation of the cross-correlogram. The narrow band correlation coefficient is now given by:

$$\sigma_2(f)\,\sigma_5(f) = \frac{C_{25}(f)}{\sqrt{w_2(f)\,w_5(f)}}$$

i.e., by dividing the cross-spectral density by the square root of the product of the power spectral densities of strains 2 and 5. This correlation spectrum for strains 2 and 5 is shown in Figure 15. It is only possible to obtain a correlation spectrum above about 500 c.p.s. because of the inaccuracies in the derived strain spectra at very low spectral density values. Two curves are shown in the Figure - one derived from the unsmoothed strain spectra and the other from the smoothed strain spectra. It appears that the use of the smoothed strain spectra gives a correlation spectrum retaining approximately the correct values but without spurious secondary peaks.

Correlation spectra, both smoothed and unsmoothed, have also been obtained between panels 2 and 7 and between panels 4 and 5, and these are shown in Figures 16 and 17. A comparison as shown in Table I can now be made with the results obtained by using the 1/3 octave filters. A mean value of the correlation spectrum must be taken between the appropriate filter band limits.

TABLE I

Panels	Centre Frequency	Correlation Spectrum Average	Filtered Cross Correlation (1/3 Octave Filter)
2 - 5	630 c.p.s. 800 c.p.s.	-0·42 +0·27	-0·39 +0·10
2 - 7	630 c.p.s. 800 c.p.s.	+0·61 -0·48	+0·48 -0·38
4 - 5	630 c.p.s. 800 c.p.s.	-0·10 +0·20	0 0

These show a good comparison but Figures 15, 16 and 17 show that much information can easily be lost by using the $^1/_3$ octave filters. For example, from the filtered results, it appears that there is only +0·10 correlation at 800 c.p.s. between panels 2 and 5. In actual fact it can be seen from the correlation spectrum that at 710 c.p.s. (near the lower limit of the 800 c.p.s. filter band) the correlation is −0·78 while at 830 c.p.s. the correlation is +0·96. Thus the value +0·10, indicating very little correlation across stringers in the 800 c.p.s. band, is completely misleading.

4. Normal Mode and Frequency Estimation

The strain spectra and correlation spectra indicate that the lower resonant frequencies up to about 500 c.p.s. belong to overall modes of vibration of the whole fuselage. The modes having frequencies in the 550 – 750 c.p.s. region appear to be continuous across the stringers and, since panels are alternatively in and out of phase, probably constitute the fundamental stringer-twisting mode. The mode shapes for the frequency range 750 – 1000 c.p.s. have not been exactly determined but it can be seen that panels 2 and 5 are generally in phase while panels 2 and 7 are out of phase. There seems to be little correlation across the frames at these higher frequencies and they can, therefore, be regarded as boundaries. On this basis, theoretical work due to Lin[5] was first applied to the problem. This theory assumes a row of identical panels simply supported by two frames, but unfortunately it only considers two fundamental mode shapes, and the harmonics of these:-

(a) The fundamental stringer-twisting mode (i.e., adjacent panels out of phase).

(b) The fundamental stringer-bending mode (i.e. adjacent panels all in phase).

Using Lin's approach, with the dimensions of panel 5, the following results are obtained:

(a) The fundamental stringer-twisting mode

Calculated frequency	626 c.p.s.
Measured frequency	720 c.p.s.

(b) The fundamental stringer-bending mode

Calculated frequency	1128 c.p.s.

The calculated and measured frequencies compare favourably in the fundamental stringer-twisting mode. The difference is possibly due to the fact that there is, in the structure, some fixing at the frames.

There is no marked peak in the measured spectrum which can be attributed to the fundamental stringer-bending mode, although there are a number of very broad peaks in the region 750 - 1100 c.p.s. which are not accounted for.

Since the correlation results indicate that the next measured mode above the fundamental stringer-twisting mode has a wavelength equivalent to more than two panel-widths, it was thought that an energy approach might be more fruitful. Again, a row of identical panels simply supported between two frames has been assumed for the analysis. A series of mode shapes may be assumed, the first having a wavelength equivalent to two panel-widths, the second having a wavelength equivalent to three panel-widths and so on. The n^{th} mode, of course, is equivalent to Lin's fundamental stringer-twisting mode which has all of the panels in phase. Using this method, with the dimensions of panel 5, a few frequencies have been calculated.

Equivalent wavelength	Calculated frequency
2 panel-widths	727 c.p.s.
4 panel-widths	937 c.p.s.
6 panel-widths	1020 c.p.s.
n panel-widths	1090 c.p.s.

5. Discussion of Results

The major part of the analysis has been limited to the mid-panel strains and from this it is possible to see the form of skin vibration which is taking place. The strain spectra for the mid-panel positions all show a similar form. There is very little strain in the frequency range up to about 500 c.p.s. then the main panel resonance appears in the range 500 to 700 c.p.s. At higher frequencies the spectrum is relatively flat suggesting a combination of several closely spaced resonant frequencies. In the low frequency range the panels are all in phase, suggesting that the vibration is of an overall nature. In the panel response frequency range, however, the strains are associated with modes of vibration which are essentially interframe. In this form of vibration adjacent panels across a frame are vibrating almost independently of one another whilst several panels in a row between the frames show marked phase relationships. These results now justify the theoretical simplification that only a row of panels vibrating between two frames need be considered in estimating the panel response frequencies of such an aircraft structure.

The correlation spectra for the panels have in effect been found by two methods and it is evident that a far more comprehensive picture is obtained by measuring the cross-correlation function. The $^1/_3$ octave filters provide, perhaps, a suitable method for initially determining the general pattern of vibrations present, but they are unsuitable for the more exact determination of particular modes. Considering, then, the spectra shown in Figures 15, 16 and 17, it is seen that the smoothed correlation spectrum is more suitable than the unsmoothed one in all three cases considered. Using such spectra, the correlations may be determined for any particular frequency, giving the mode shape, providing only that sufficient spectra are available.

The analytical method developed by Lin and also an energy method have both been used in an attempt to calculate the resonant frequencies of the panels. It would seem that the energy approach is the most promising, since it allows for so many different modes, but the accuracy of the method is, as yet, uncertain. However, the mode shapes which have been investigated (i.e., with equivalent wavelengths of 2, 4, 6 and n panel-widths) indicate a series of modes, with in this case, increasing frequency. It is thought that the mode shapes not yet investigated (i.e., with equivalent wavelengths of 3, 5, etc. panel-widths) may have resonances which fit logically into the previous series. This theory agrees well with the strain spectra obtained for the panels, but there were insufficient strain gauges on the specimen to pick out the individual modes of vibration in this frequency range.

6. Conclusions

This method of investigating the form of vibration taking place in an aircraft structure subjected to random pressure loads has proved successful in picking out the main panel responses. The associated damping ratios and mode shapes may also be completely determined provided that sufficient data is available. It is felt that the strains in the frames and stringers could be analysed in the same way thus determining the overall vibrations present.

The particular structural resonances which are excited depend on the characteristics of the pressure field. Further experiments should be made in which the pressure correlation is measured and as a result it should become clear why particular modes are being excited.

Since the analysis has shown the essential nature of the response of conventional sheet-stringer construction, more detailed experiments are now clearly justified to give fuller information on the mode shapes. This work could well be supplemented by resonance tests on specimens which have a minimum of three frame bays and many stringer bays in order to reproduce the interframe type of vibration.

REFERENCES

1. Powell, A. — 'On the Fatigue Failure of Structures due to Vibrations Excited by Random Pressure Fields.' Journal of the Acoustical Society of America, Vol.30, 1958, p.1130.

2. Blackman, R.B. and Tukey, J.W. — 'The Measurement of Power Spectra from the Point of View of Communications Engineering.' Bell Systems Technical Journal, Vol. 37 1958.

3. Bartlett, M.S. — 'Periodogram Analysis and Continuous Spectra.' Biometrika 37 (1950) p.1-16.

4. Crandall, S.H. (Editor) 'Random Vibration' Technology Press, The Massachusetts Institute of Technology. 1958.

5. Lin, Y.K. 'Free Vibrations of Continuous Skin-stringer Panels.' To be published in the Journal of Applied Mechanics.

Acknowledgments

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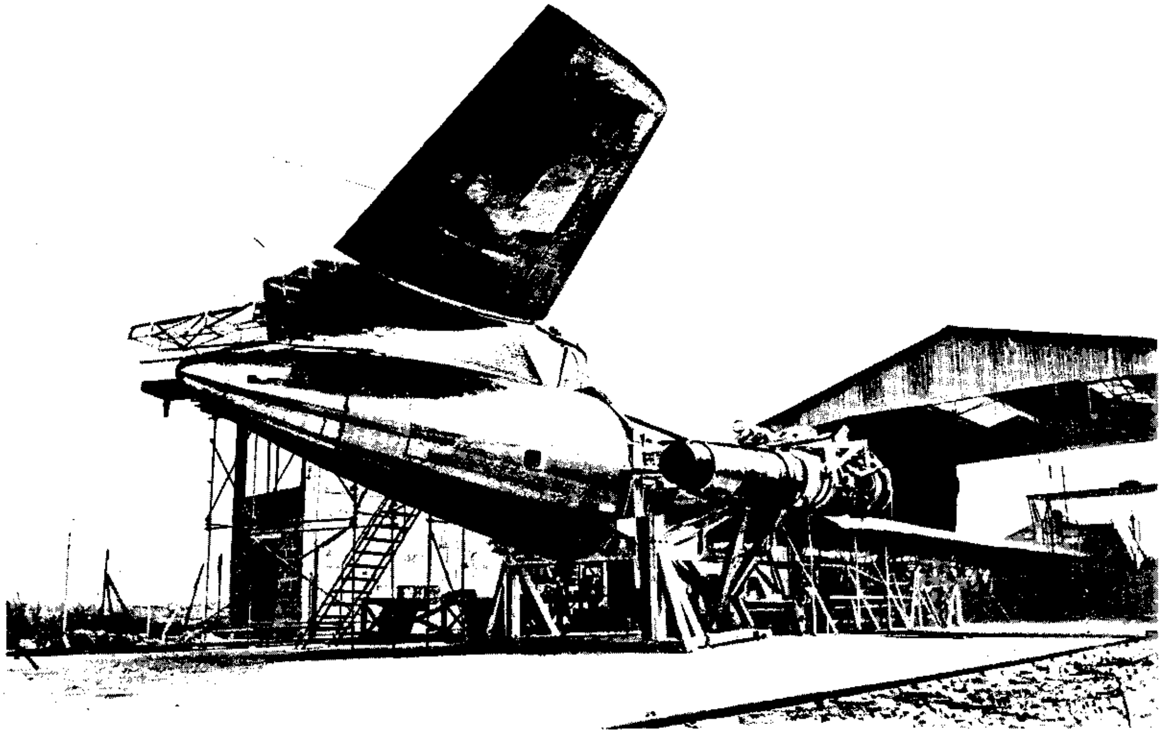


Fig. 1. General View of Ground Test Installation

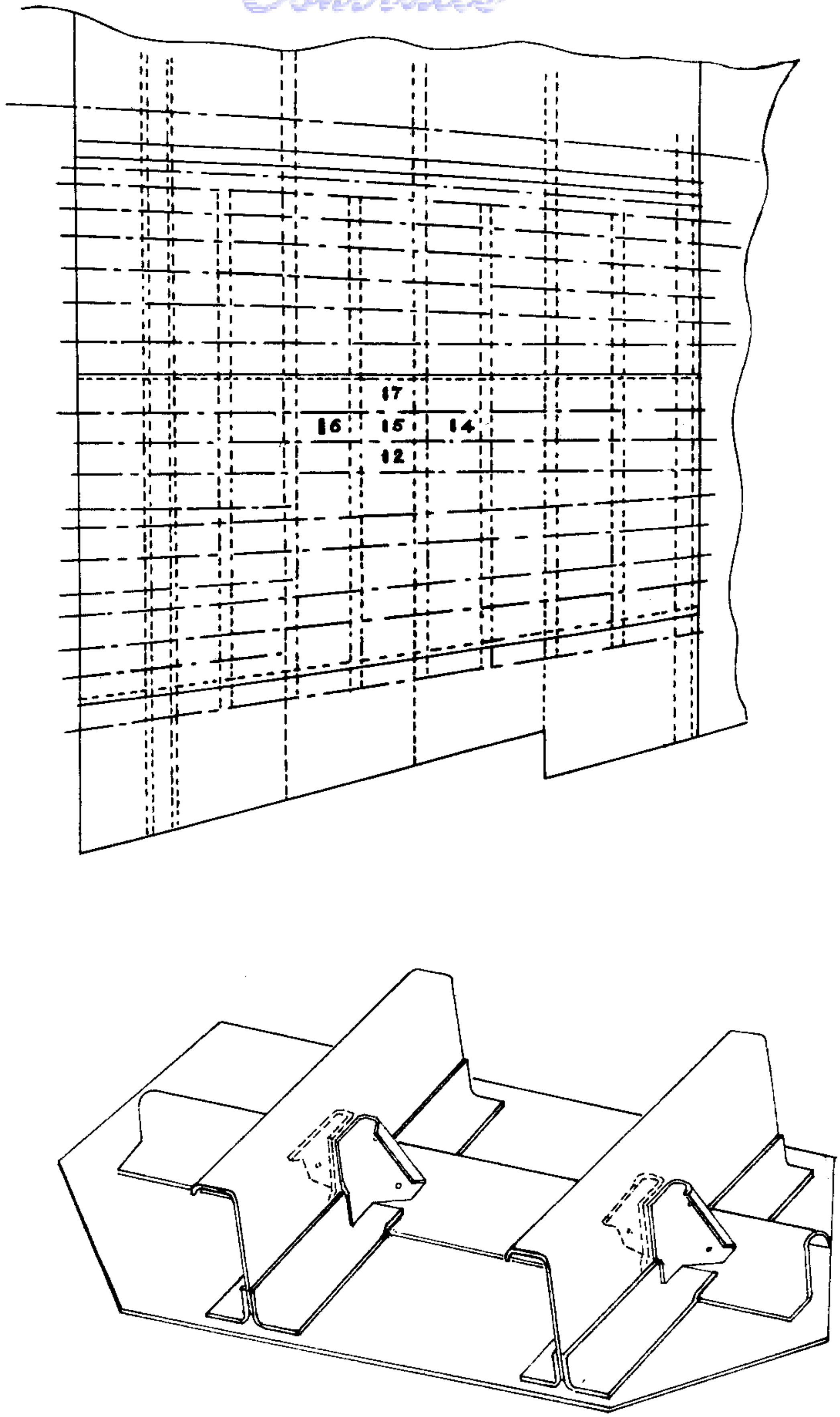


Fig. 2. Positions of Gauges and Details of Structure

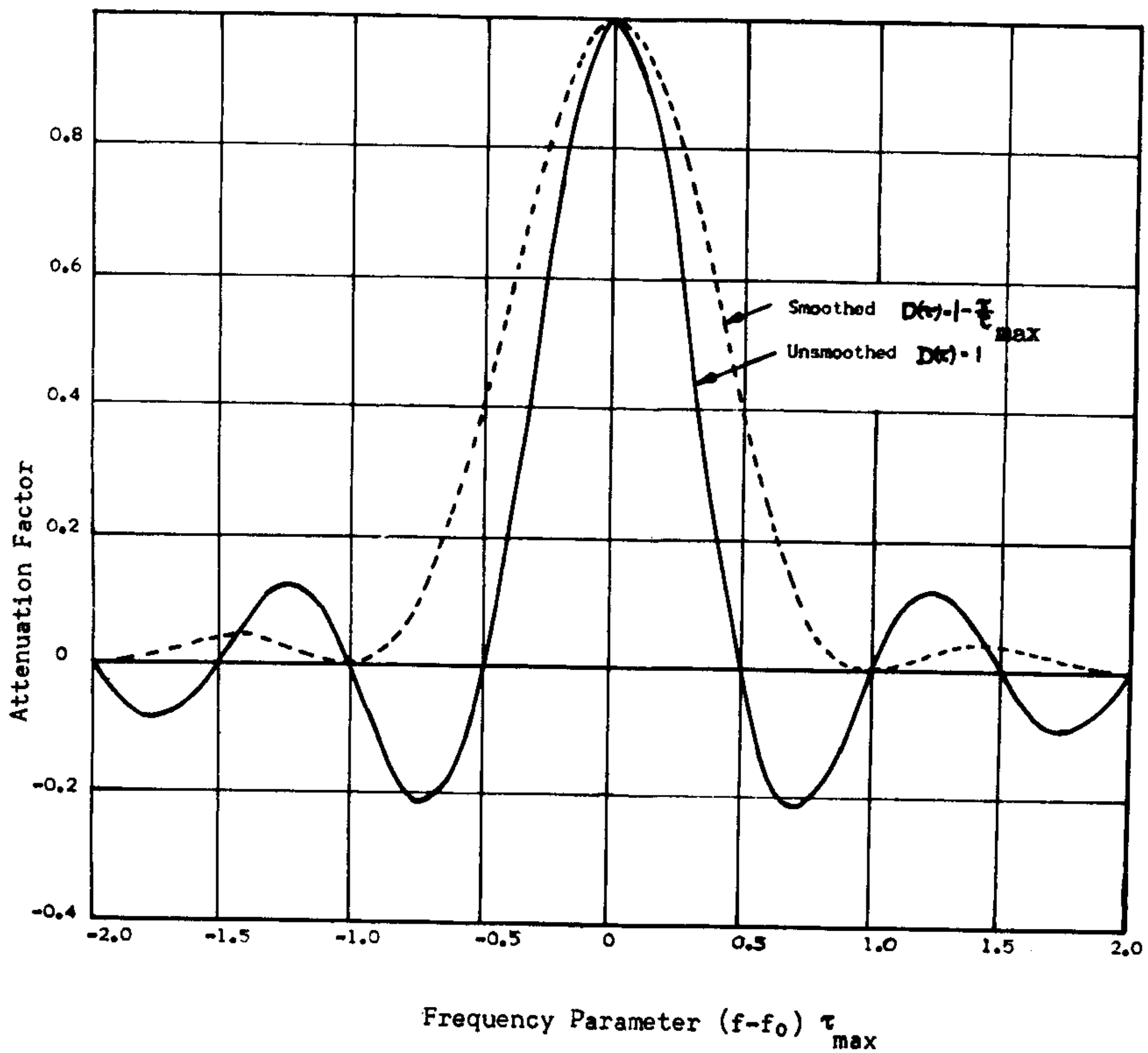
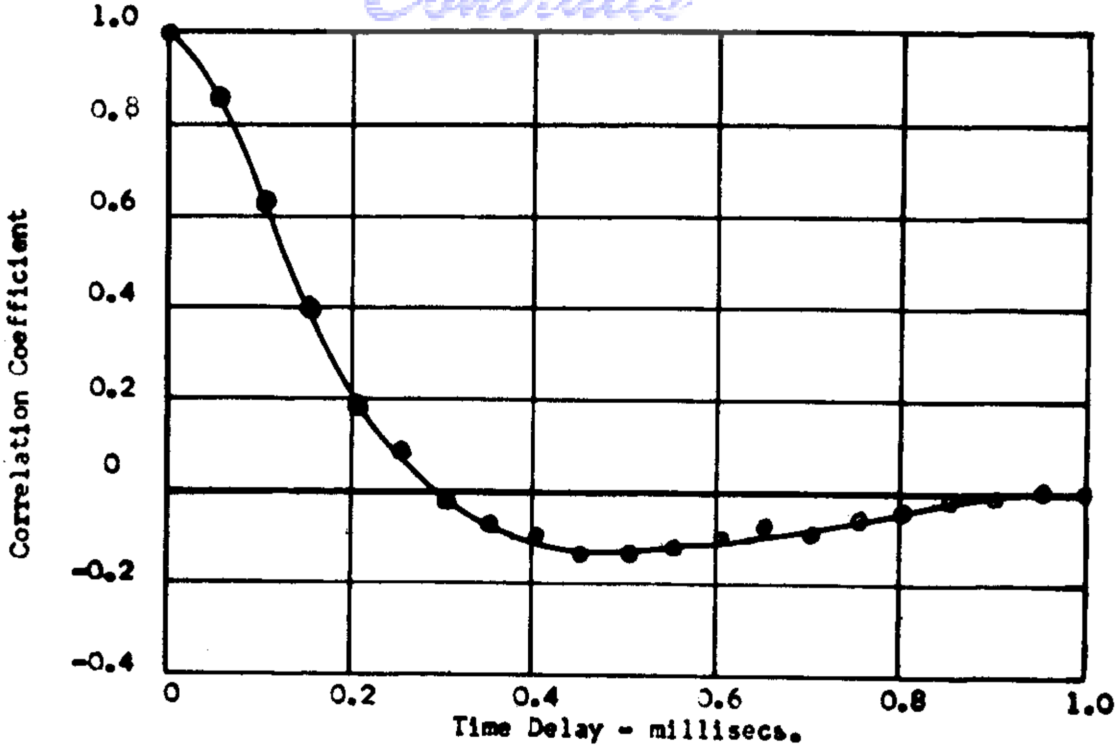
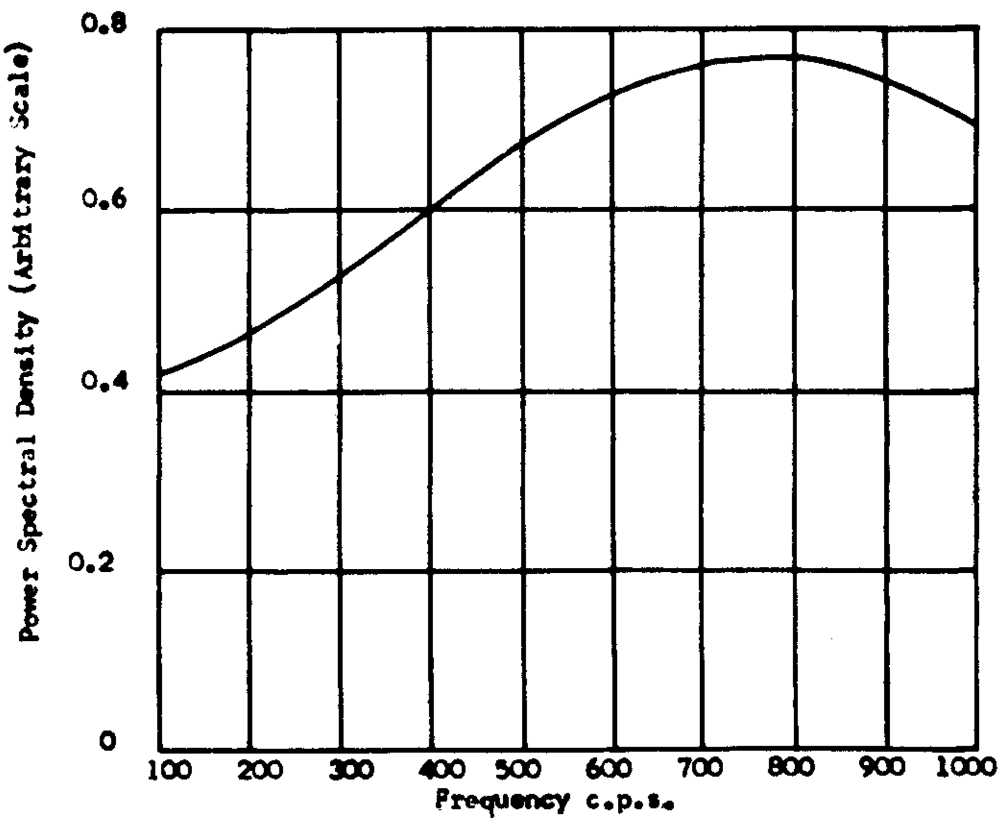


Fig.3. Spectral Windows Q (f)



AUTOCORRELOGRAM



SPECTRUM

Fig. 4. Noise Pressure on Fuselage Side

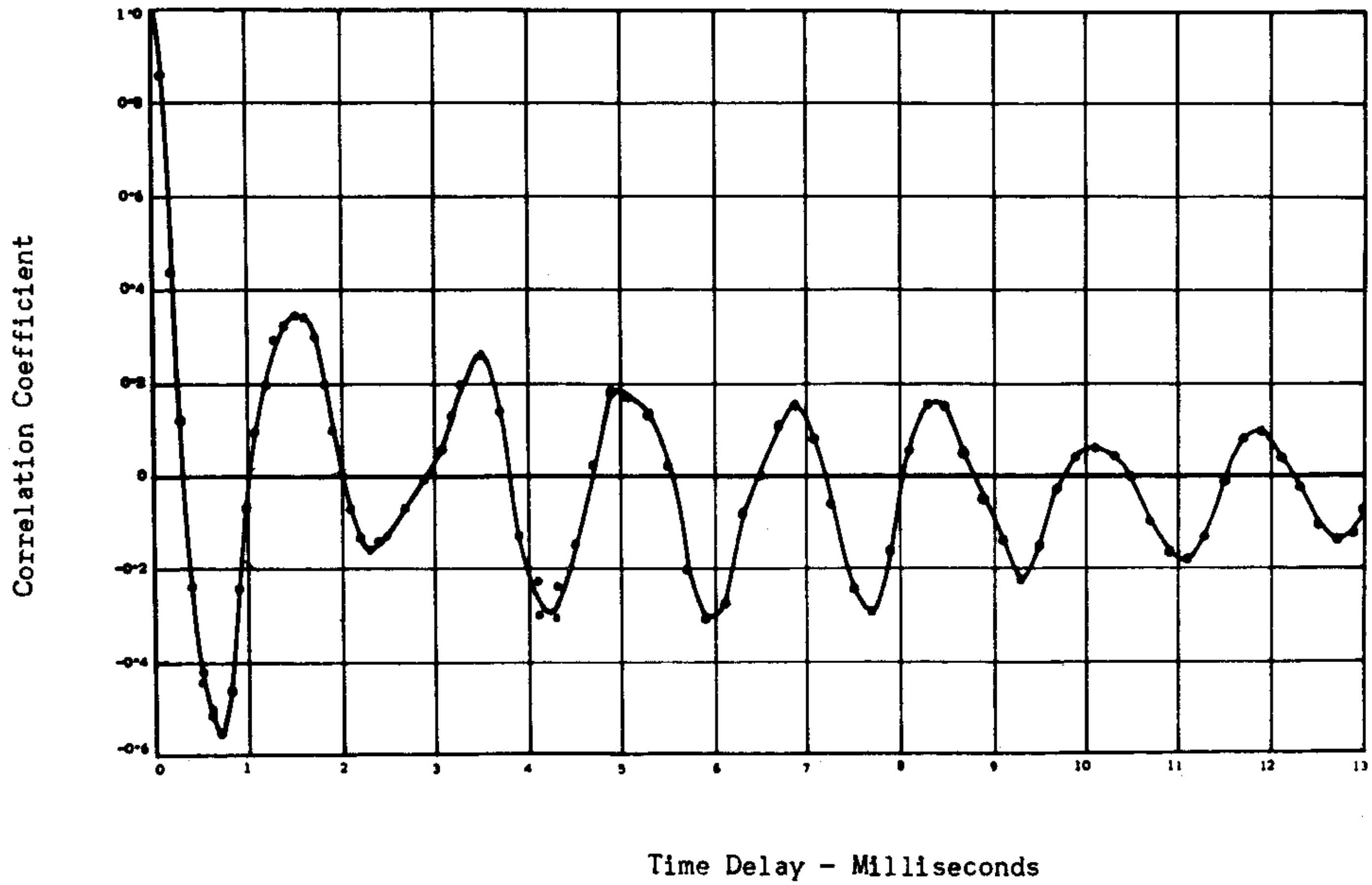


Fig.5. Autocorrelogram of Strain at Panel Position 2

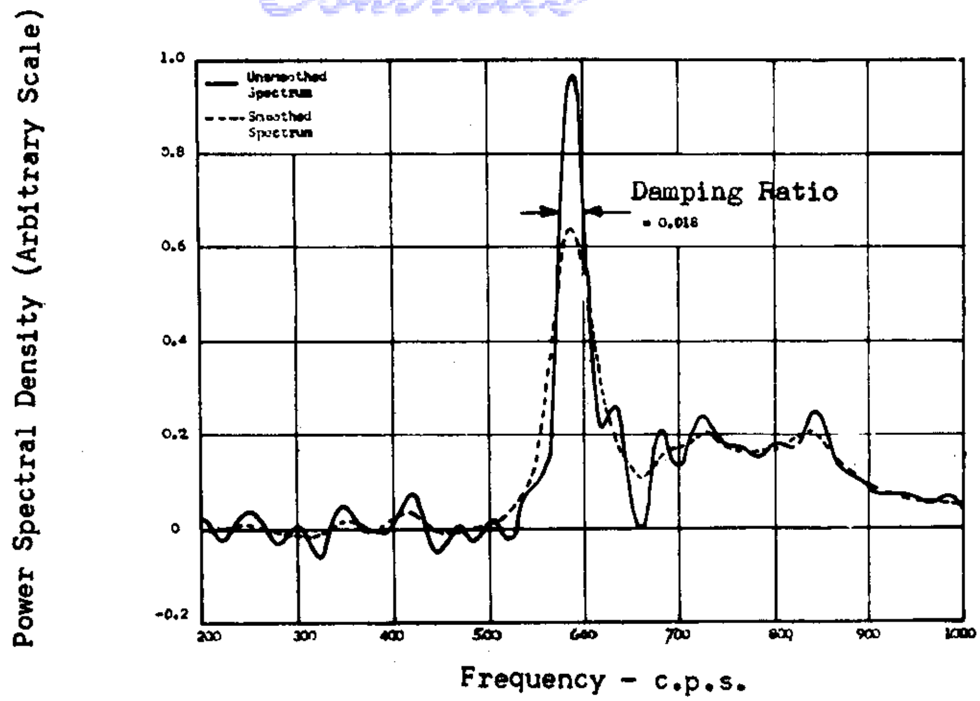


Fig.6. Strain Spectrum at Panel Position 2

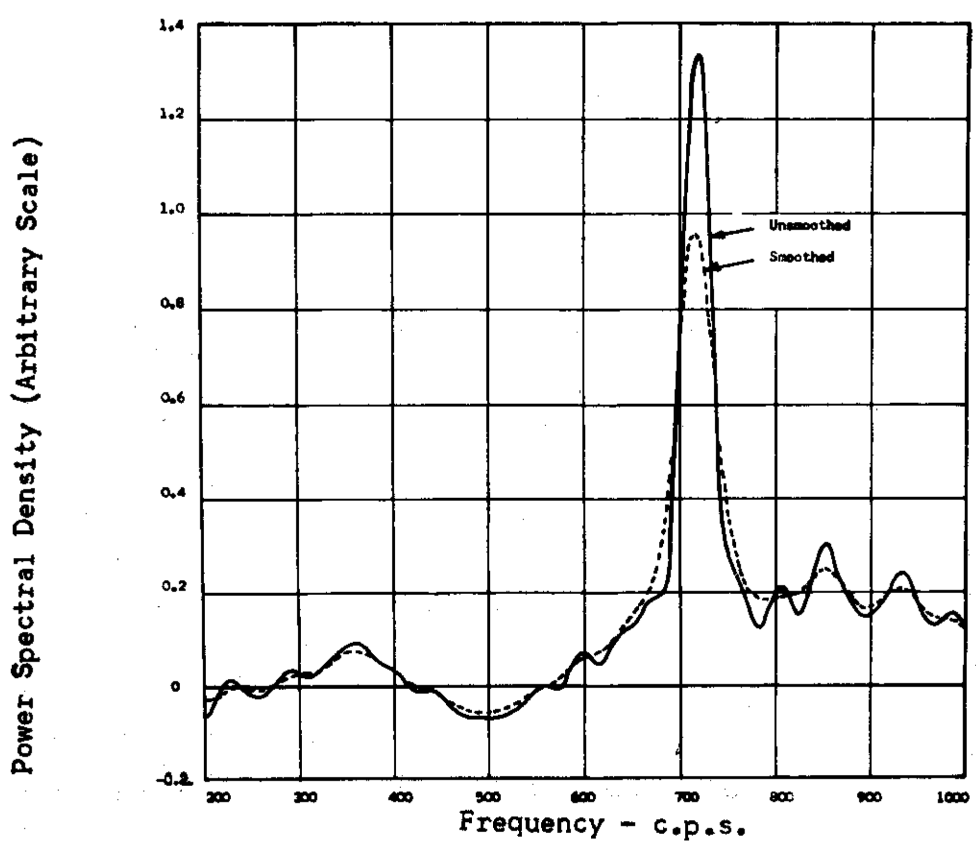


Fig.7. Strain Spectrum at Panel Position 5

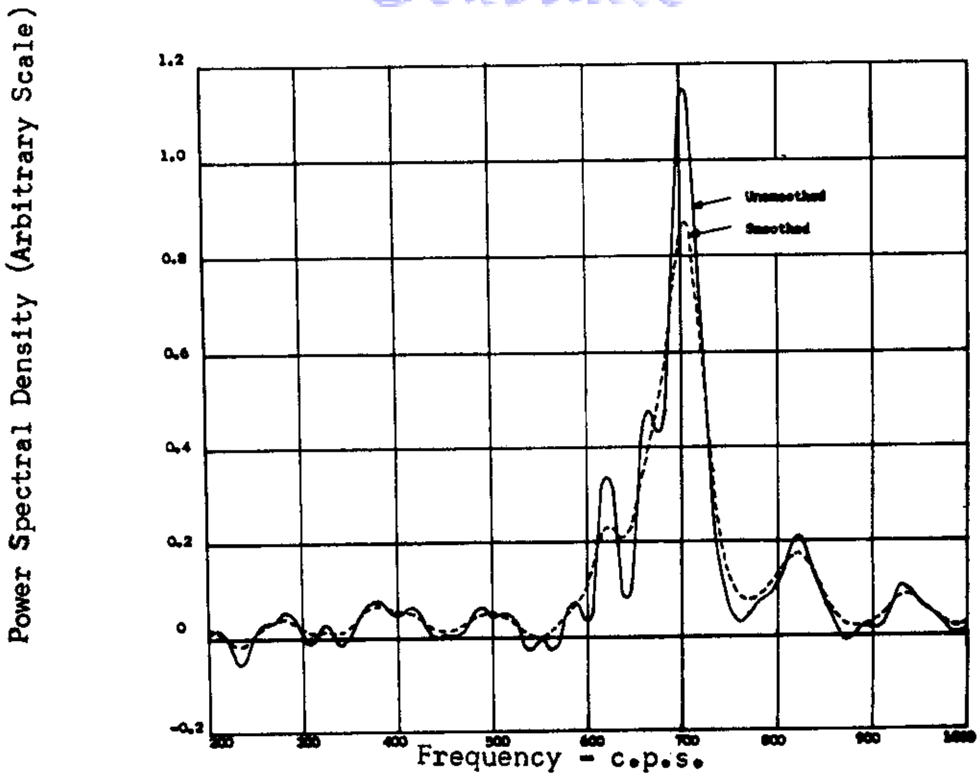
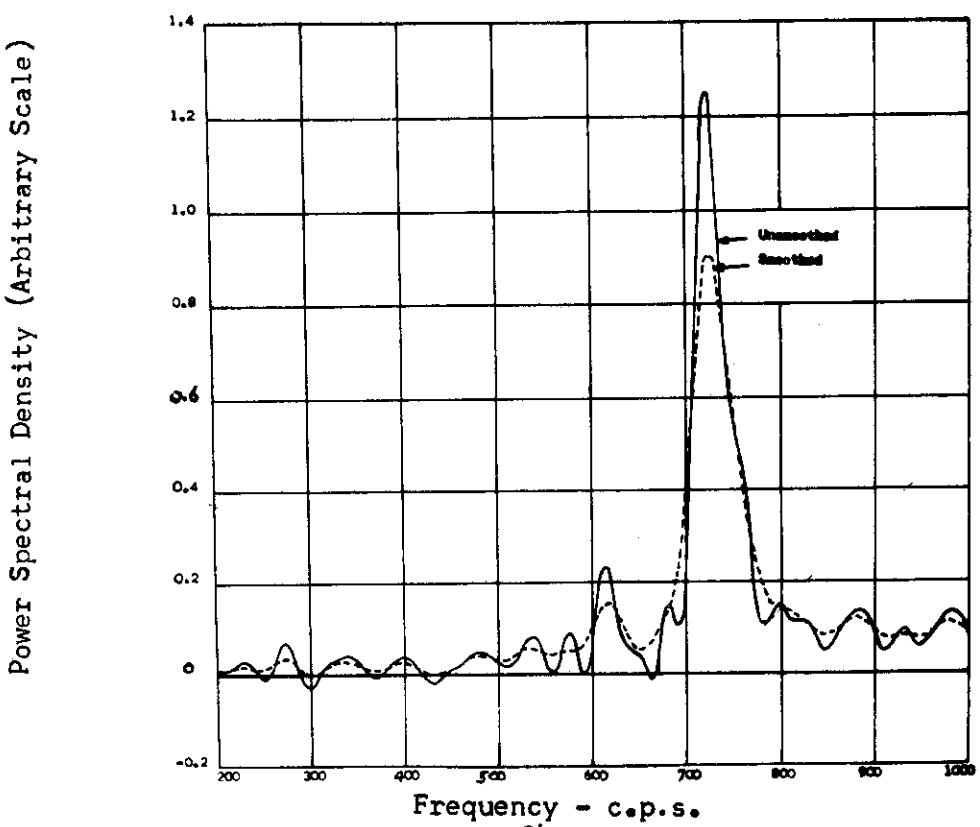


Fig. 8. Strain Spectrum at Panel Position 7

Fig. 9. Strain Spectrum at Panel Position 4



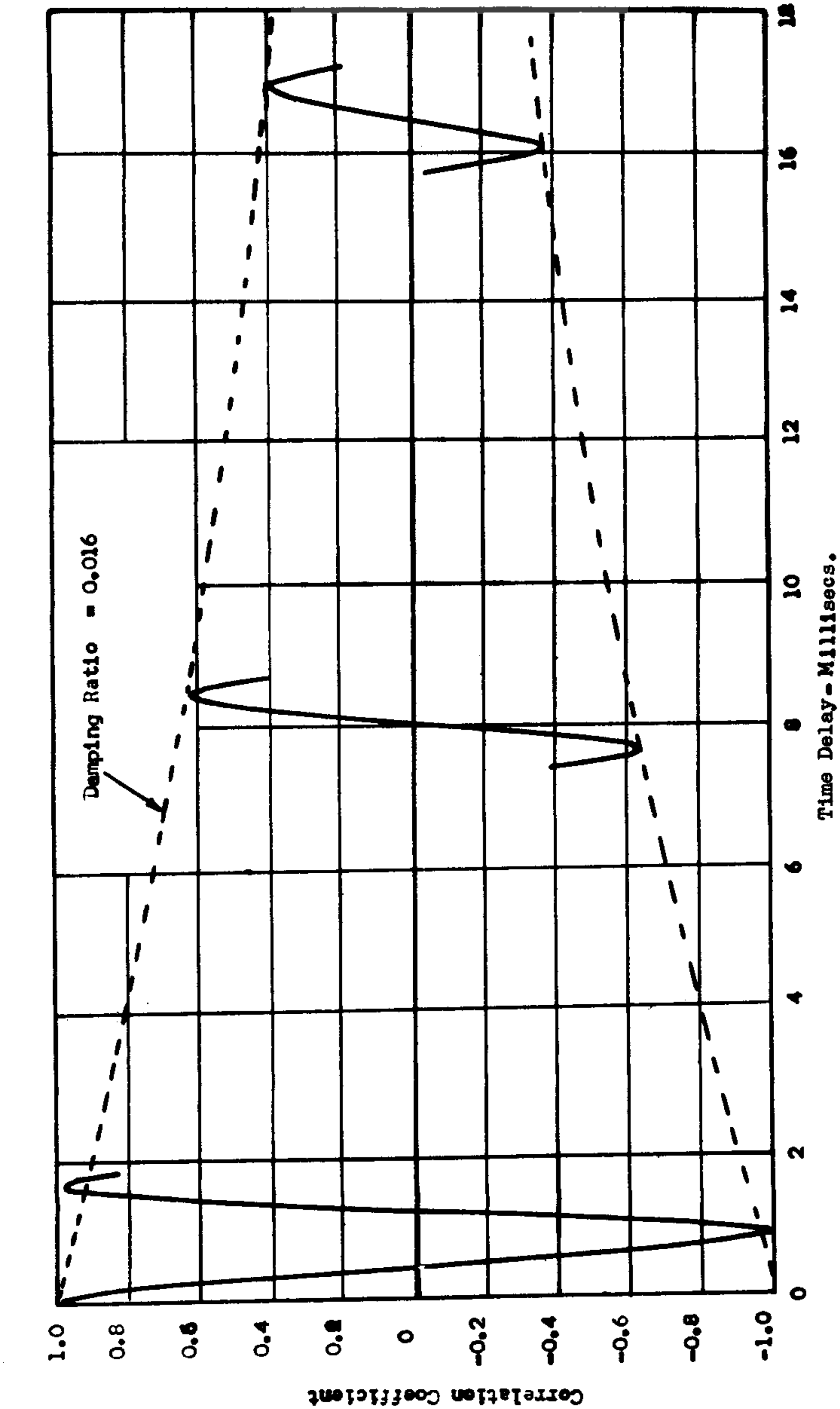


Fig. 10. Determination of Damping from Autocorrelogram

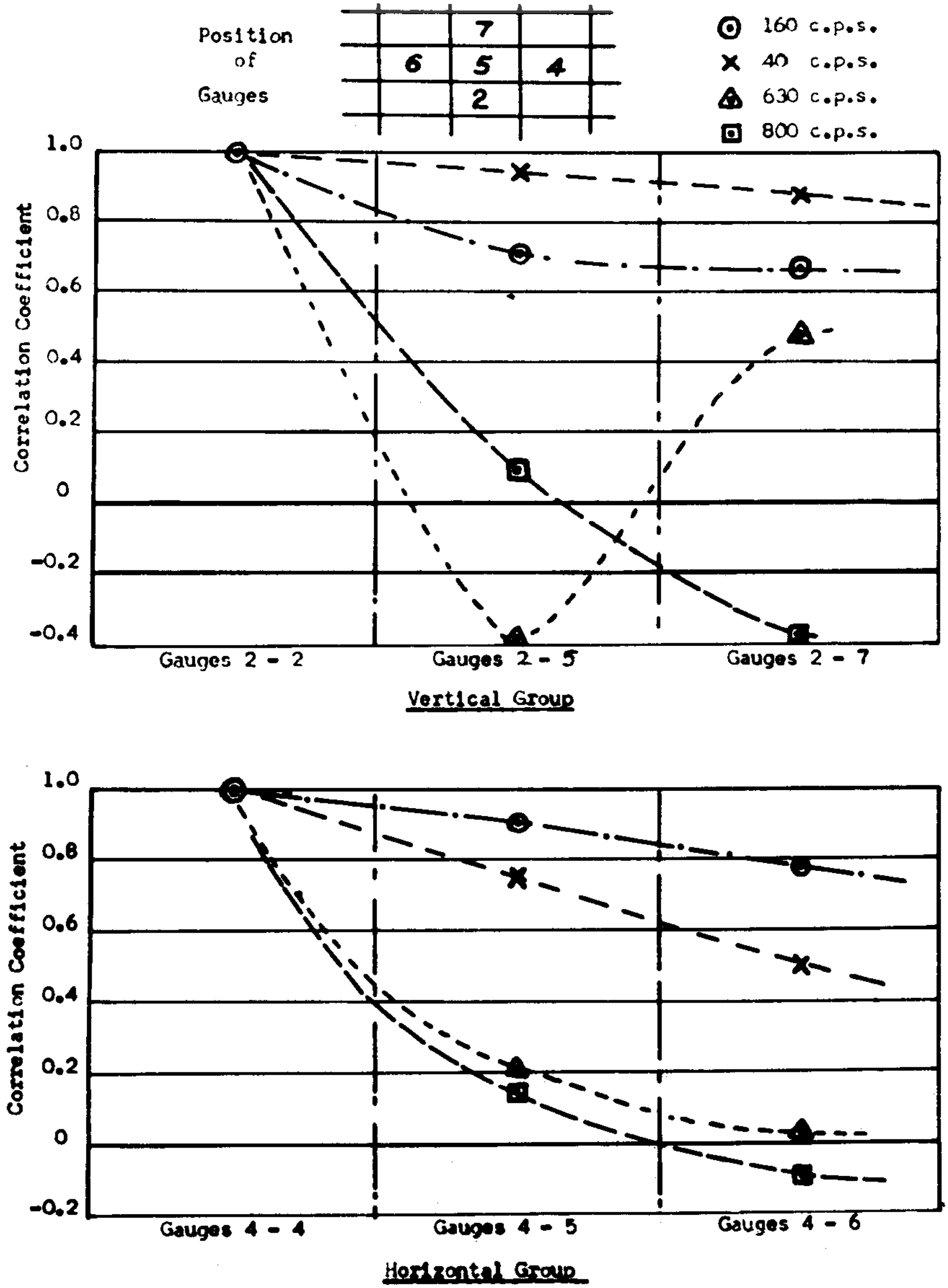


Fig. 11. 1/3 Octave Band Cross Correlations, Mid Panel Strain

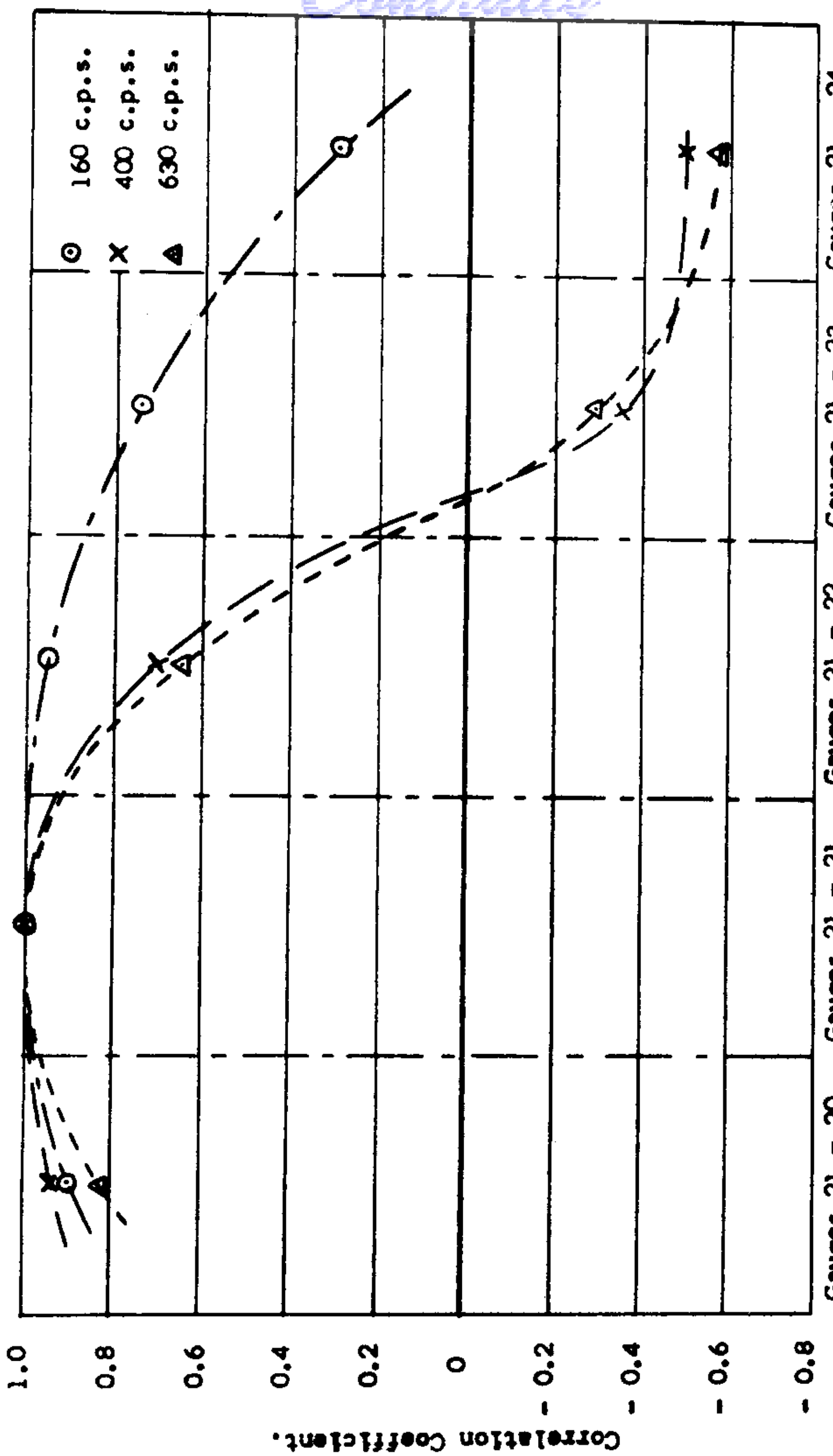


Fig. 12. 1/3 Octave Band Cross Correlation of Frame Strains

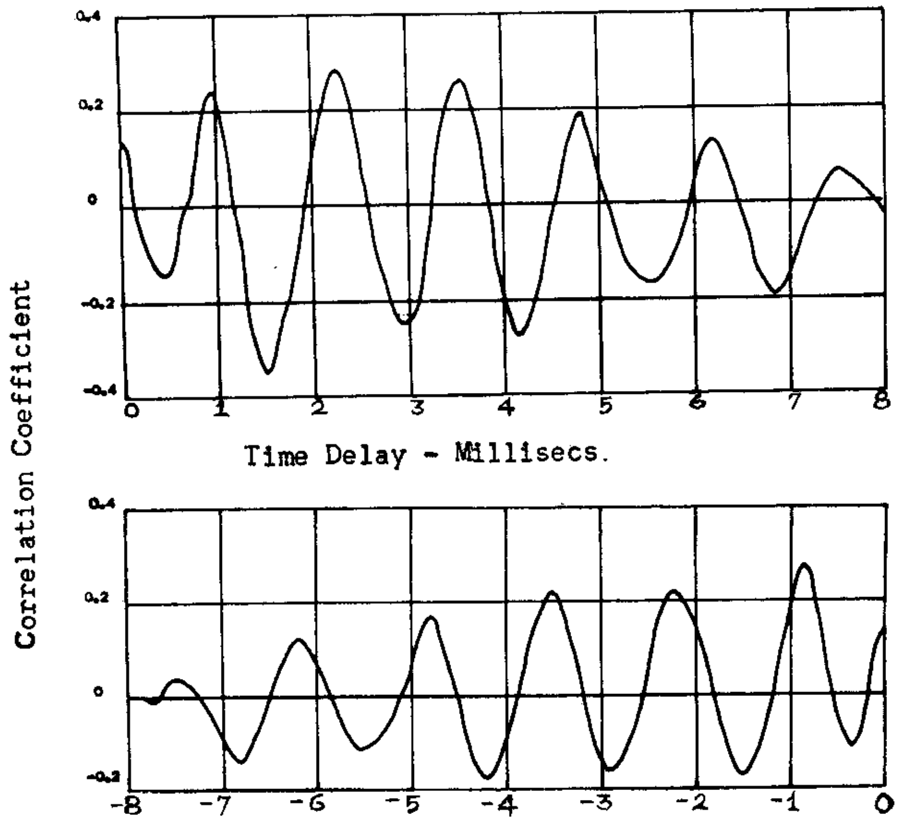


Fig. 13. Cross Correlation of Strains at Panel Positions 2 and 5

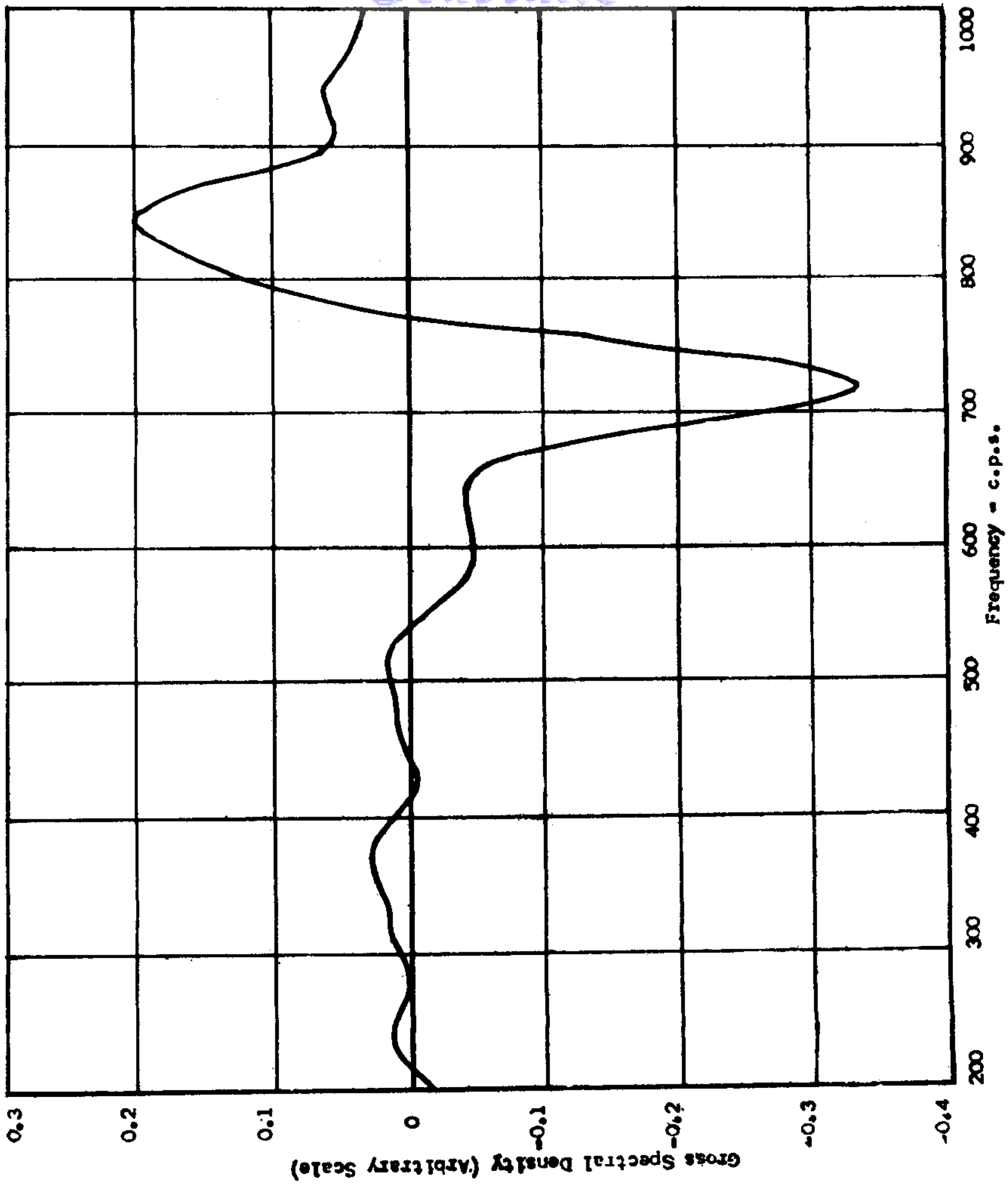


Fig. 14 Cross Power Spectrum of Strain, Gauges 2 and 5.

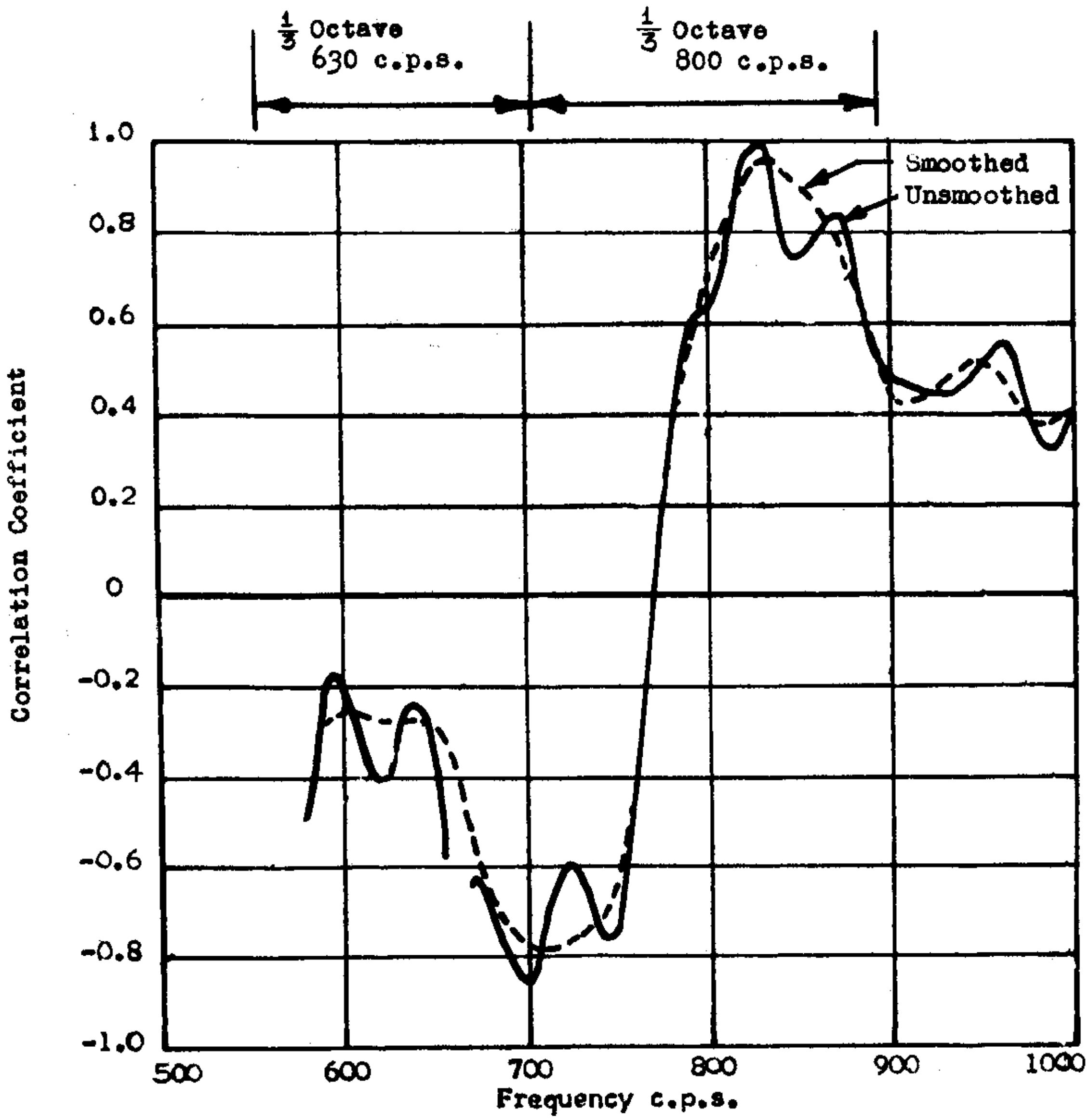
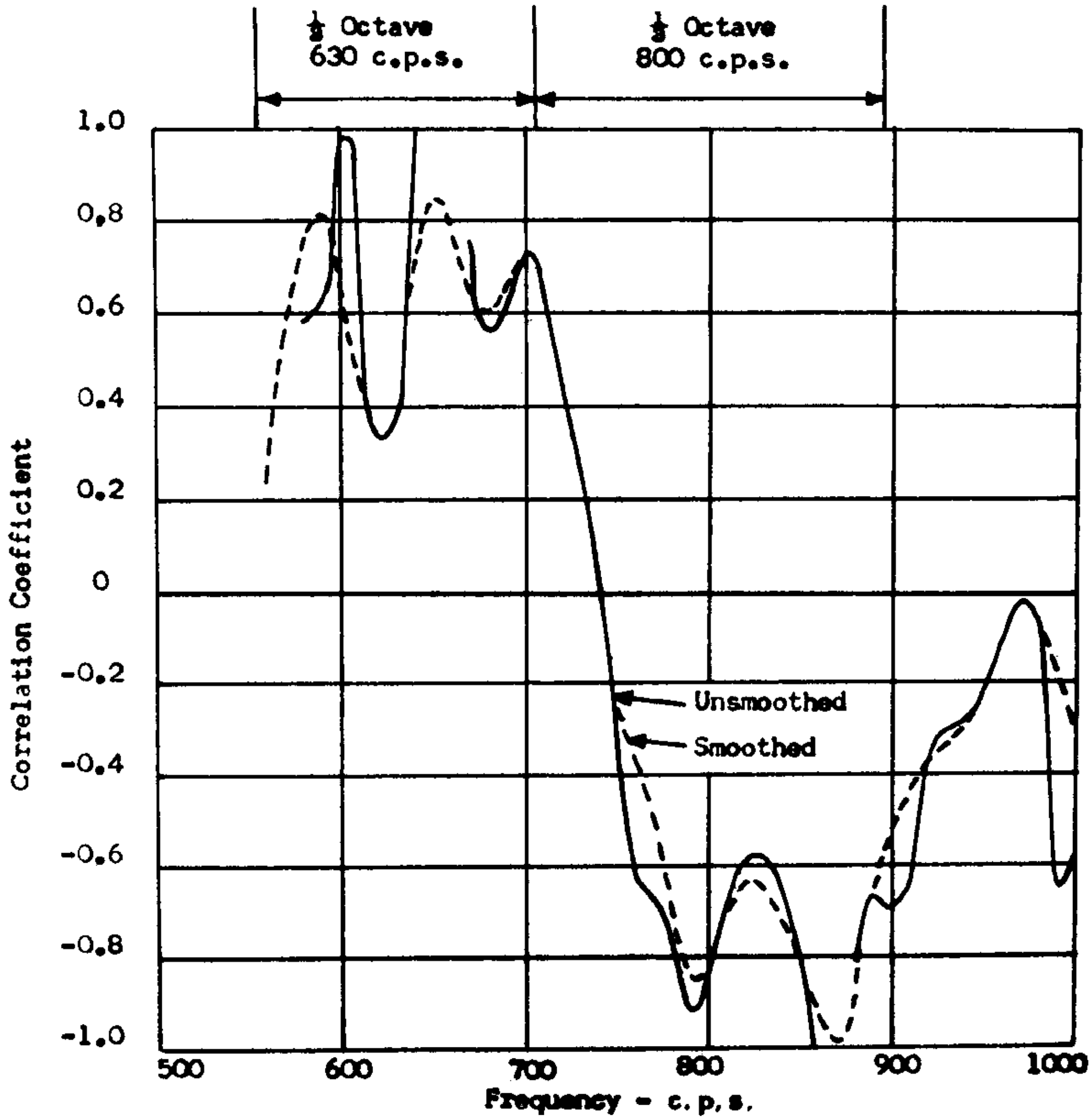
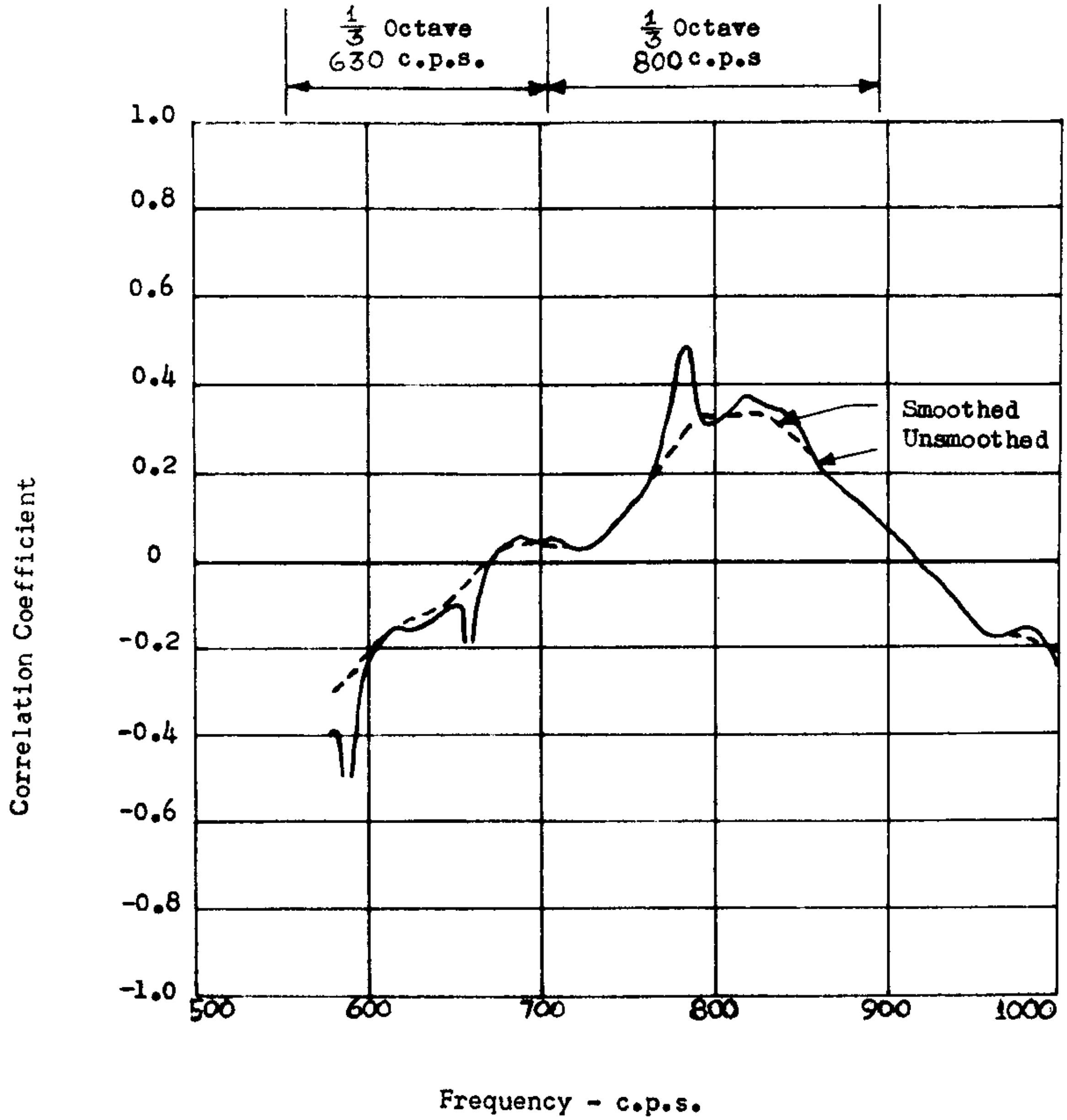


Fig. 15. Correlation Spectrum Between Panels 2 and 5



Correlation Spectrum between Panels 2 and 7

Fig. 16.



Correlation Spectrum between Panels 5 and 4

Fig. 17