EFFECTS OF CRACK PROPAGATION ON DYNAMIC PROPERTIES OF COMPRESSOR BLADES

By

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EFFECTS OF CRACK PROPAGATION ON DYNAMIC

PROPERTIES OF COMPRESSOR BLADES

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PREFACE

The writer wishes to acknowledge indebtedness to the following individuals who made possible this study:

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FOREWORD

This thesis represents an investigation of the effects of fatigue crack propagation on the natural frequency and damping characteristics of jet engine compressor blades. Findings and suggested applications to nondestructive testing of compressor blades which are presented in this thesis represent a portion of the duties of the author while assigned to the Non Destructive Testing Project as a graduate research assistant.

The Non Destructive Testing Project was conducted for the Directorate Materiel Maintenance of the Oklahoma City Air Materiel Command under contract number AF 34(601)-9879.

CHAPTER I

INTRODUCTION

The purpose of this investigation was to devise a simple experimental test which would establish general trends relating dynamic properties with fatigue crack propagation in J-57 jet engine compressor blades. Interpretation of this relationship could then be used to determine the presence of fatigue cracks in a given compressor blade. A vibratory nondestructive test such as this has several advantages over existing methods in that no preparation of the blade surface is necessary and little operator skill is required.

Dynamic properties selected for observation were the first cantilever natural frequency and damping characteristics. Natural frequencies were measured directly, and damping properties were evaluated by analyzing transient vibratory response records of the blades. A crack or flaw in a blade is a physical discontinuity which was expected to affect the dynamic properties measured. The natural frequencies of vibration of blades are dependent on the distribution of mass, stiffness, damping factor, and end conditions. Major contributions to the damping of the free vibrations of a blade are condition of the material, state of the internal stress, interface shear damping, and air coupling. A crack or flaw in a blade can result in a local decrease in stiffness of the blade and an increase in interface shear damping at the crack due to friction caused by the edges of the crack rubbing together.

The analytic determination of natural frequency for compressor blades is very difficult. In the case of damping caused by internal friction, the analytical means are few and limited in applications. The effect on dynamic properties of a small discontinuity such as a fatigue crack or flaw has not been determined analytically. The measurement of these dynamic properties was obtainable with specially constructed apparatus and existing equipment in the mechanical engineering laboratory. Due to these considerations, measurements were made experimentally; and in order to reduce the effect of experimental uncertainties, all results recorded are relative changes of the parameters.

From the interpretation of these measured trends, it was discovered that a simple vibratory nondestructive test could be performed on the compressor blades using comparison of the damping characteristics for cracked and uncracked blades. The expected dependence of natural frequency on crack propagation applicable to a nondestructive test was not definitely established.

CHAPTER II

PREVIOUS INVESTIGATIONS

A considerable amount of work has been done, concerning the vibration of cantilever beams and plates, applicable to the vibration of compressor blades. Due to the physical geometry of the blade which may be thought of as a nonsymmetrical beam of variable cross section with twist from root to tip, the equation of motion of the blade is nonlinear and must be solved by approximate methods. Much is involved in obtaining accurate numerical results. The problem of blade vibration has been of considerable interest since the advent of the steam turbine and is of increased importance concerning the jet engine because of weight considerations, safety factors, and high rotative speeds.

A. Mendelson and S. Glender (1.)[1] investigated analytically the coupled bending-torsion vibrations of cantilever beams. The same investigators (2.) evaluated the effect of twist on vibrations of cantilever beams and verified the results experimentally. D. Rosard (3.) investigated experimentally the effect of twist on the natural frequencies of cantilever beams by a unique transient method and verified the results analytically.

It is interesting to point out that the qualitative application of damping phenomena for testing purposes has long been an

[1]Parentheses refer to Selected Bibliography.

accepted feature of everyday life. The duration of the ring of a wine glass, or a train axle, or of a coin when suitably excited is commonly used to ascertain the soundness of these various objects. R. Canfield (4.) was one of the first to propose measurement of internal friction as a nondestructive test and pointed out that the internal friction in the metal is a property which is very sensitive to physical changes not always easily detectable by other means. A. Graves (5.) mentions the application of vibratory energy to flaw detection in writing on Applications of Vibratory Energy, Apparatus for Determining Elastic Moduli. Recently, W. Stephens (6.), writing on the application of damping capacity for investigating the structure of solids in Progress in Nondestructive Testing, pointed out the difficulties and inaccuracies involved in considering the application of damping measurements to nondestructive testing.

Investigations into the general nature of the actual vibrations of compressor blades were undertaken by J. Schnittger (7.). Qualitative results of the changes in internal damping of gas turbine materials due to continuous vibration were obtained by G. Wilkes (8.). As yet there has been no direct application of the vibratory response of compressor blades to nondestructive tests.

CHAPTER III

EXPERIMENTAL APPARATUS AND EQUIPMENT

Apparatus constructed for use in the investigations will be described in this chapter. Additional equipment existing in the Mechanical Engineering Laboratory used in the tests will be described.

Fatigue Crack Production

Fatigue cracks were produced in the blades by a controlled vibratory displacement mechanism. An electrodynamic vibration excitator previously mounted in a large table constructed especially for vibration work was used to control the blade displacement and introduce the desired fatigue bending stresses. The changes in the natural frequency of the vibrating system, which were used as the criterion for detecting fatigue failure, were monitored by the changes in current required to drive the blade at a constant vibratory frequency and amplitude. The number of stress cycles was recorded by a digital electronic counter. A fine degree of control was possible which allowed rapid production of fatigue cracks to any desired magnitude. The arrangement of equipment is shown schematically in Figure 1 with the actual system pictured in Plates I and II.

Natural Frequency Measurements

Two rotatable vises were mounted on a 3' x 4' kirksite table acquired for use in the experiment. An electromagnet constructed for

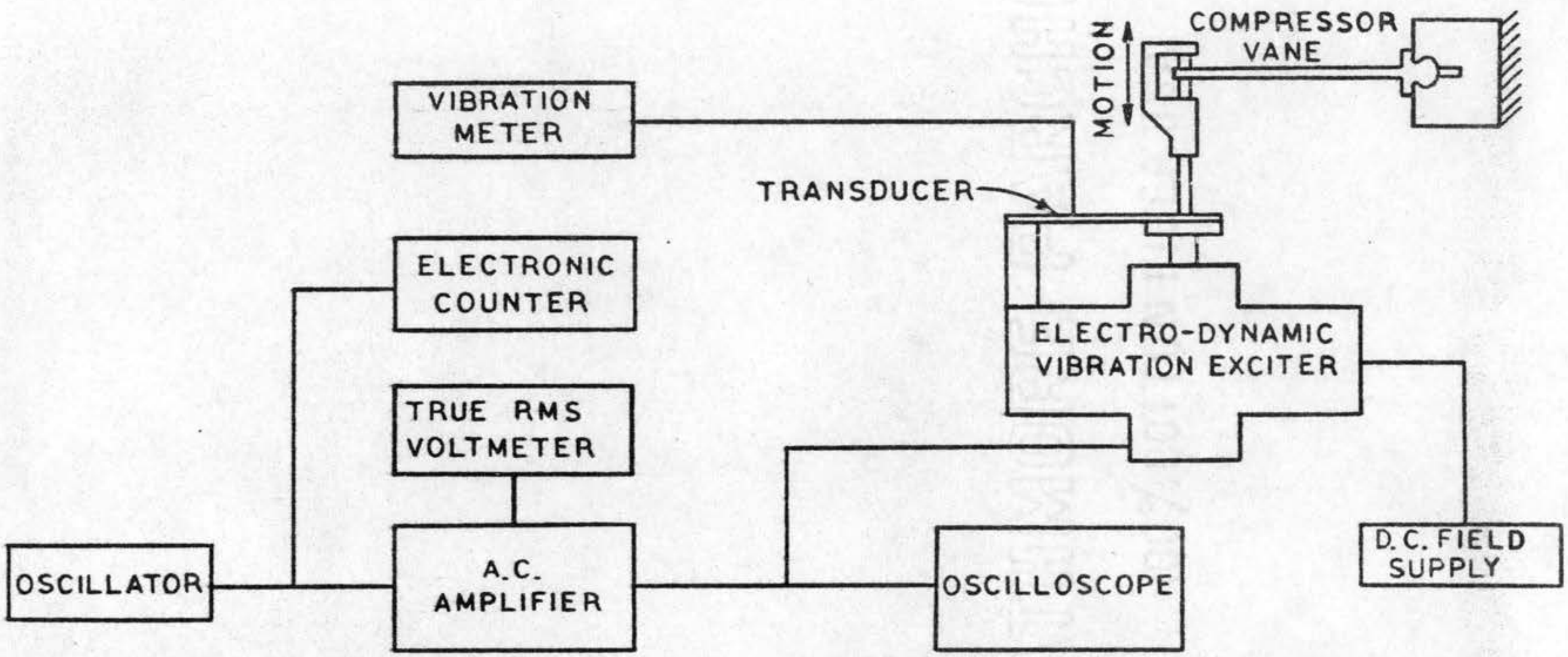


FIGURE I. SCHEMATIC OF APPARATUS FOR PRODUCING AND CONTROLLING FATIGUE CRACKS IN COMPRESSOR BLADES

PLATE I

CRACK PROPAGATION APPARATUS

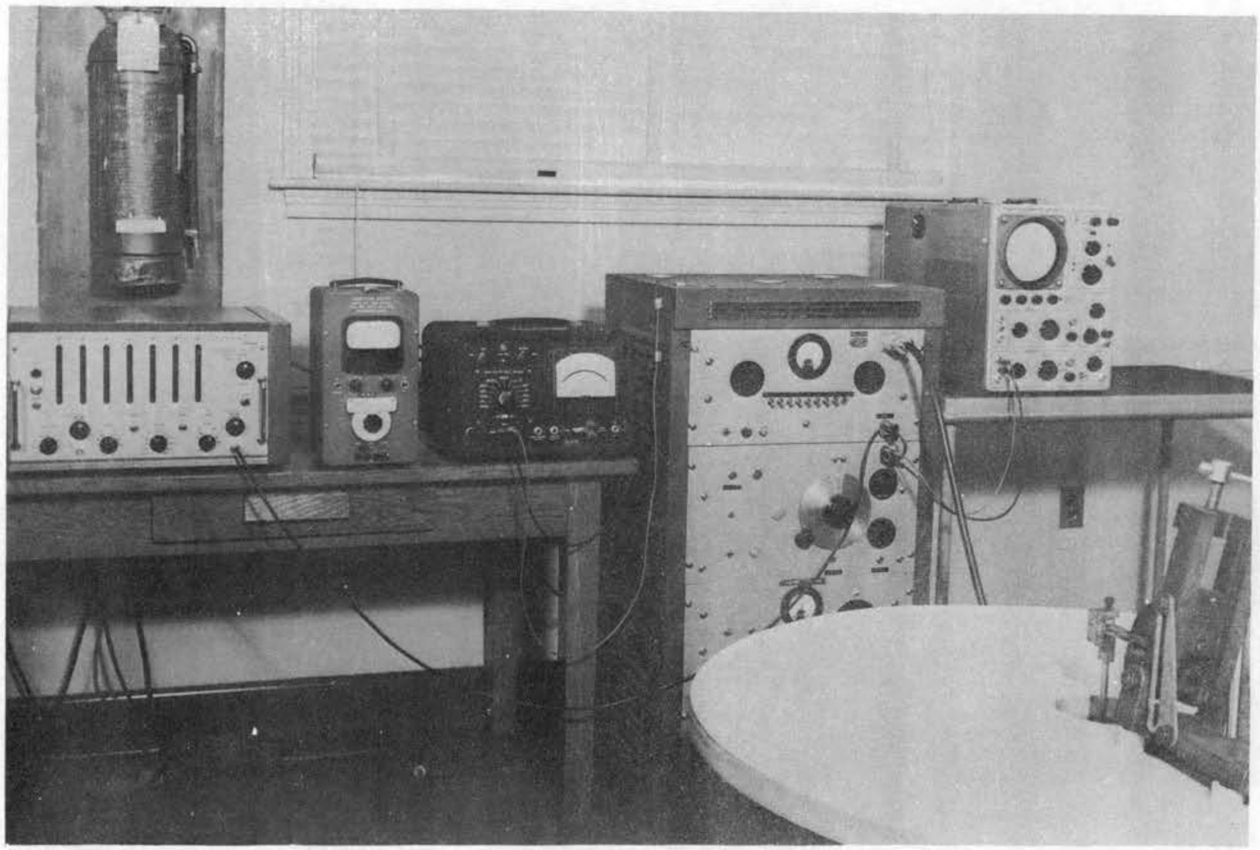
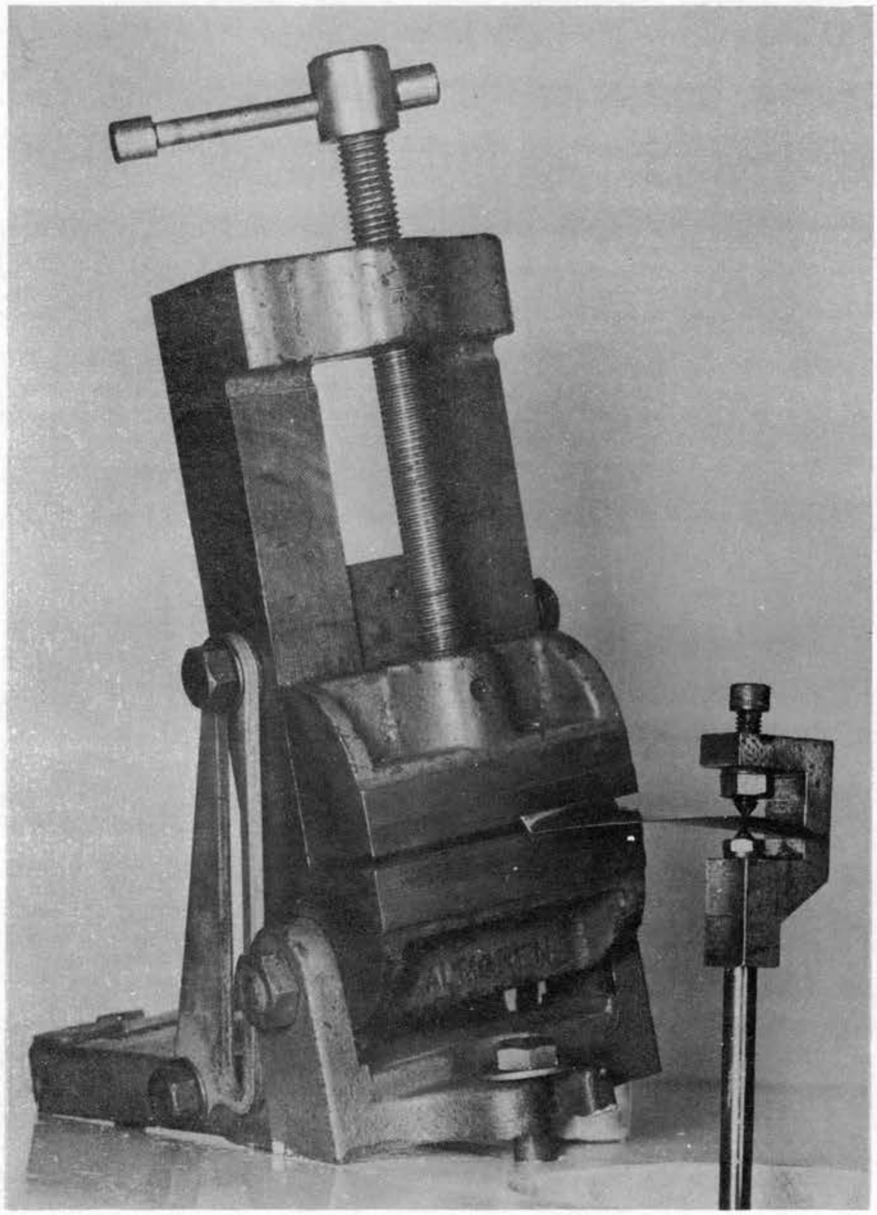


PLATE II

CLOSE UP OF BLADE CLAMPING AND SHAKER HEAD



the experiments was clamped in one vise. A compressor blade and the clamping blocks were placed in the jaws of the other vise. A quartz accelerometer was attached to this vise in a position suitable for the tests. The electromagnet was used to excite the blades at their first cantilever natural frequency. An oscillator signal was amplified and applied to the electromagnet. Oscillator tuning to the natural frequency was controlled by the maximum response of the blade as measured by an amplified voltage reading from the quartz accelerometer. Accurate measurement of the cycles per second was obtained by applying the oscillator signal to a digital electronic counter. Schematic representation is shown in Figure 2, and the actual system is pictured in Plate III. The clamping of the blade was controlled for each blade test by aligning the clamping blocks with the edge of the vise as shown in Plate IV.

Damping Measurements

The same table and the vise with the quartz accelerometer used in natural frequency measurements were utilized in obtaining damping measurements. Previous investigations by others (9.) show that reproducibility of the end condition of the blade will have a large effect on the reproducibility of the response. This problem was solved by clamping the blade with a simple machine steel wedge block as pictured in Plate VI. The blade was excited by displacing it a measured amount and releasing it, as in plucking a guitar string. The magnitude of displacement was controlled by an optical measuring device shown in Plate VI, which measured relative displacement from equilibrium. The transient response of the blade was picked up by the quartz accelerometer, amplified, and displayed on an oscilloscope

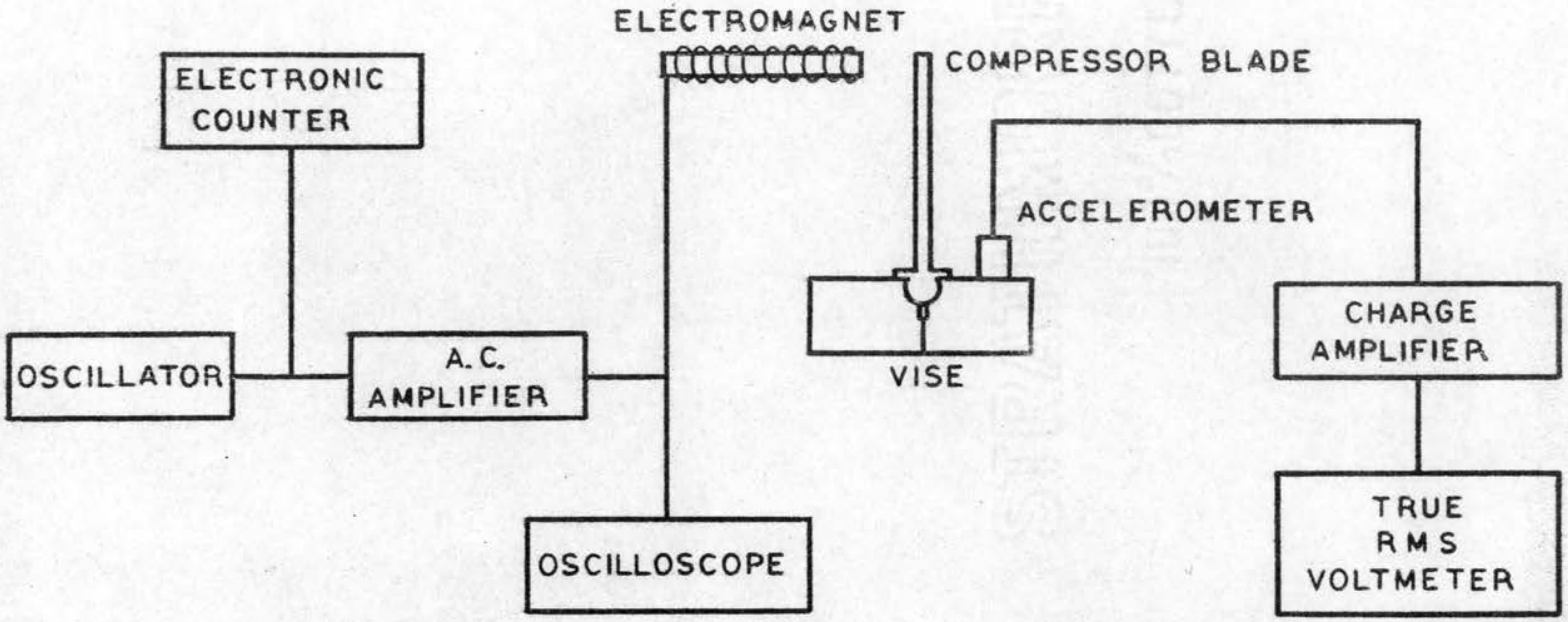


FIGURE 2. SCHEMATIC OF APPARATUS FOR DETERMINING THE NATURAL FREQUENCY OF COMPRESSOR BLADES

PLATE III

NATURAL FREQUENCY MEASUREMENT APPARATUS

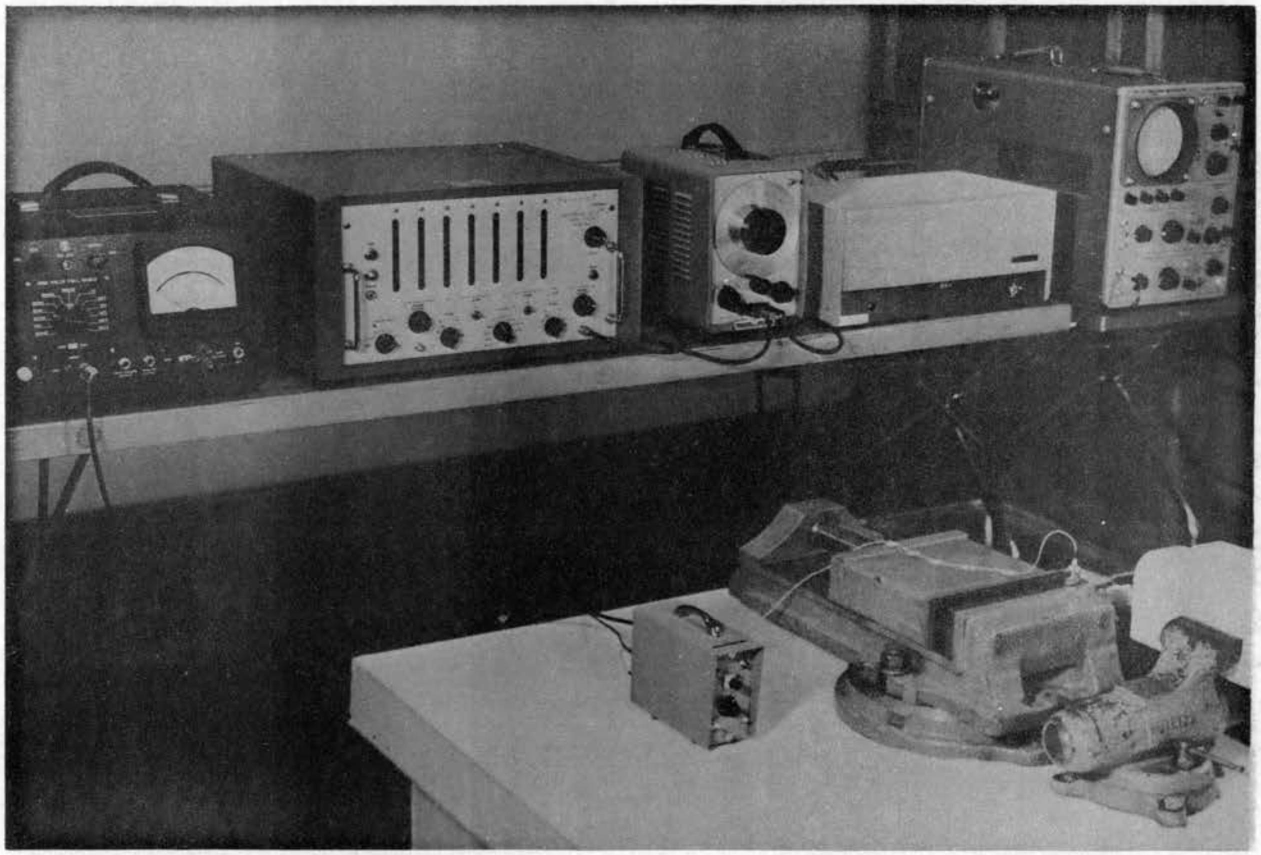
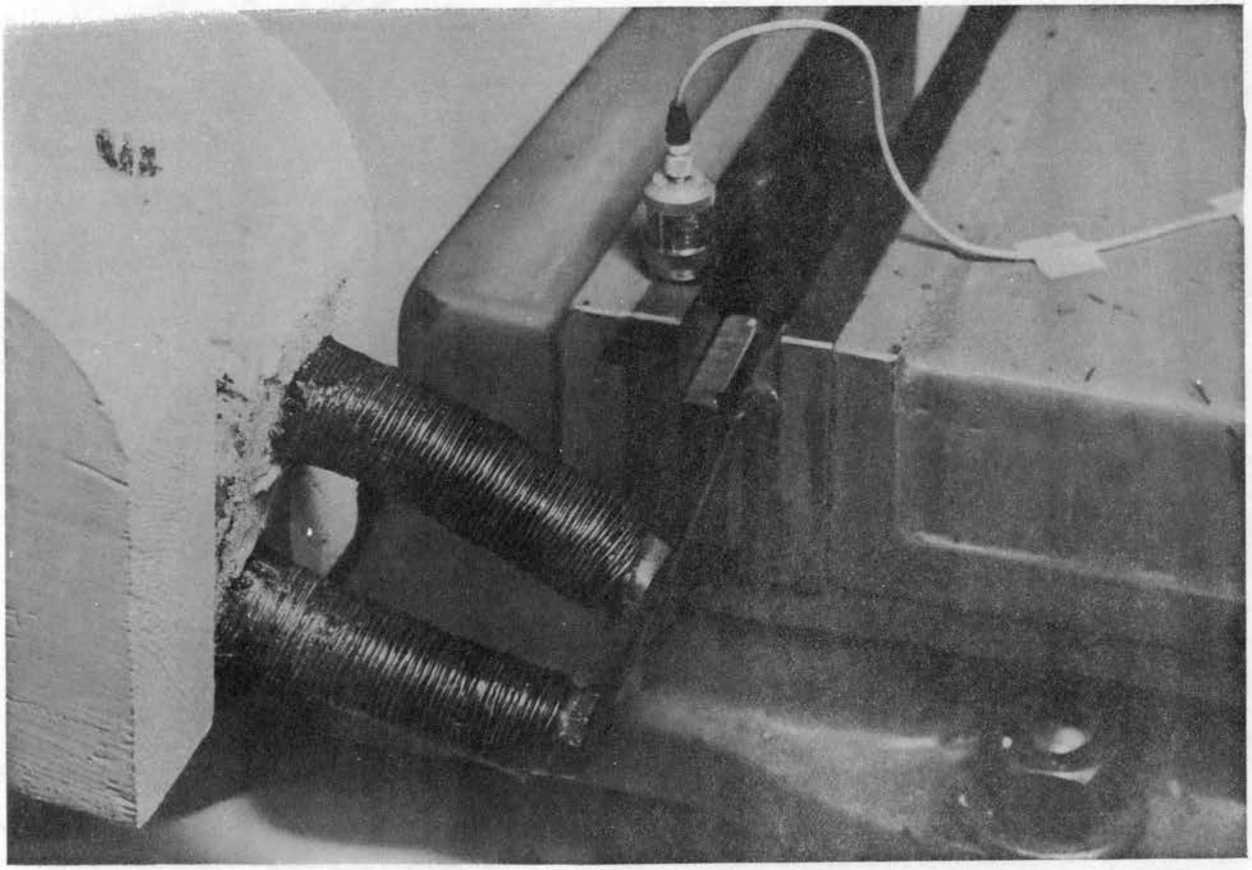


PLATE IV

CLOSE UP OF ELECTROMAGNET, CLAMPING BLOCKS, AND ACCELEROMETER



screen. Time exposures of the response were taken with an oscilloscope camera for a permanent record to be used in analysis. The system arrangement is shown schematically in Figure 3 with the actual apparatus pictured in Plate V.

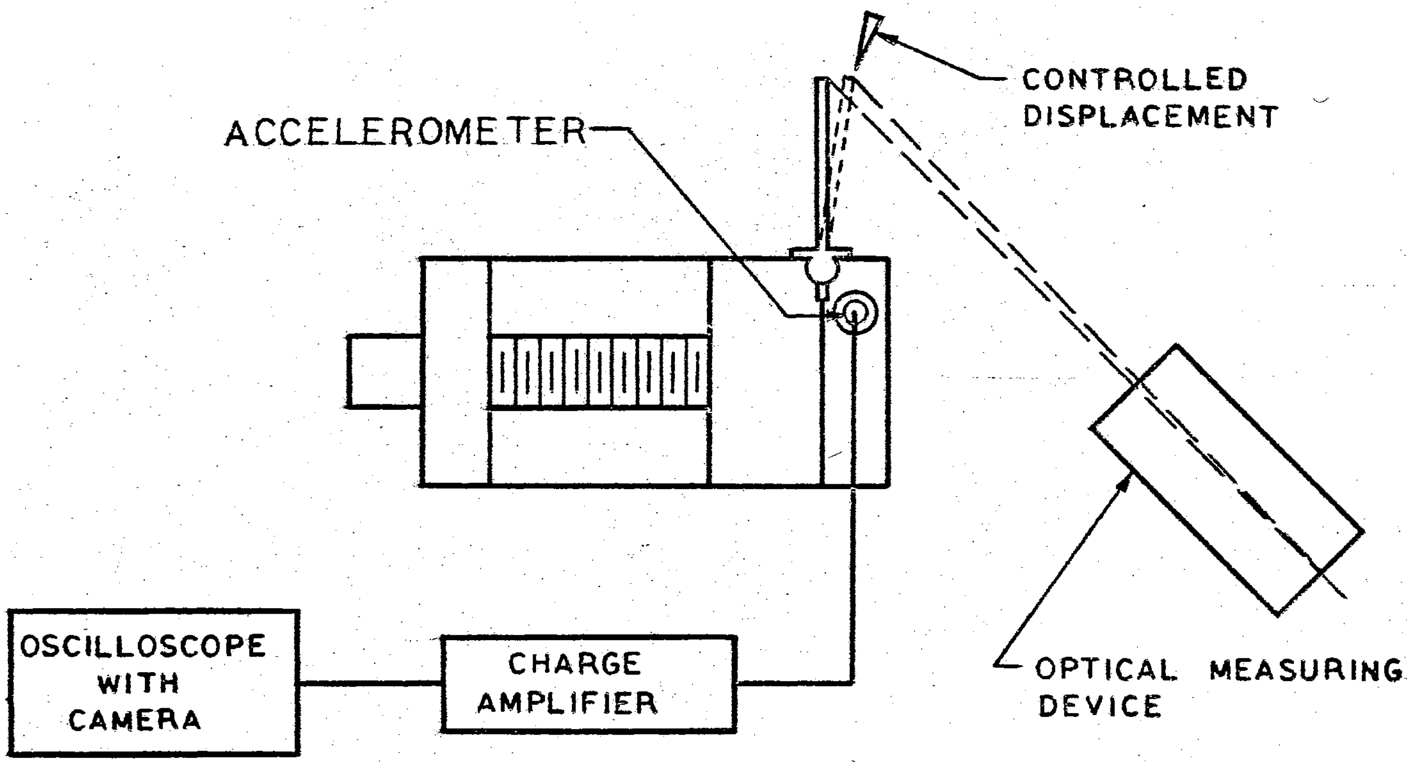


FIGURE 3. SCHEMATIC OF DAMPING MEASURING SYSTEM

PLATE V
DAMPING MEASUREMENT APPARATUS

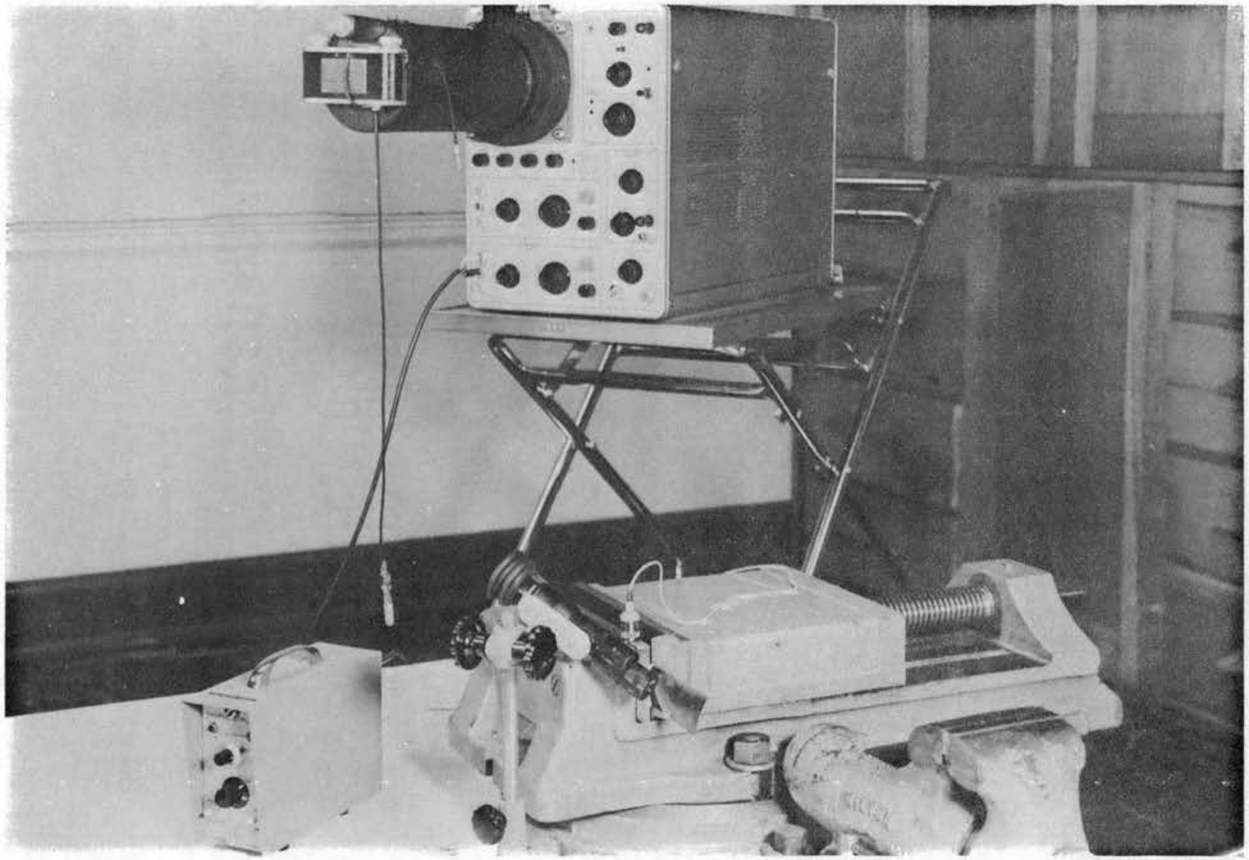
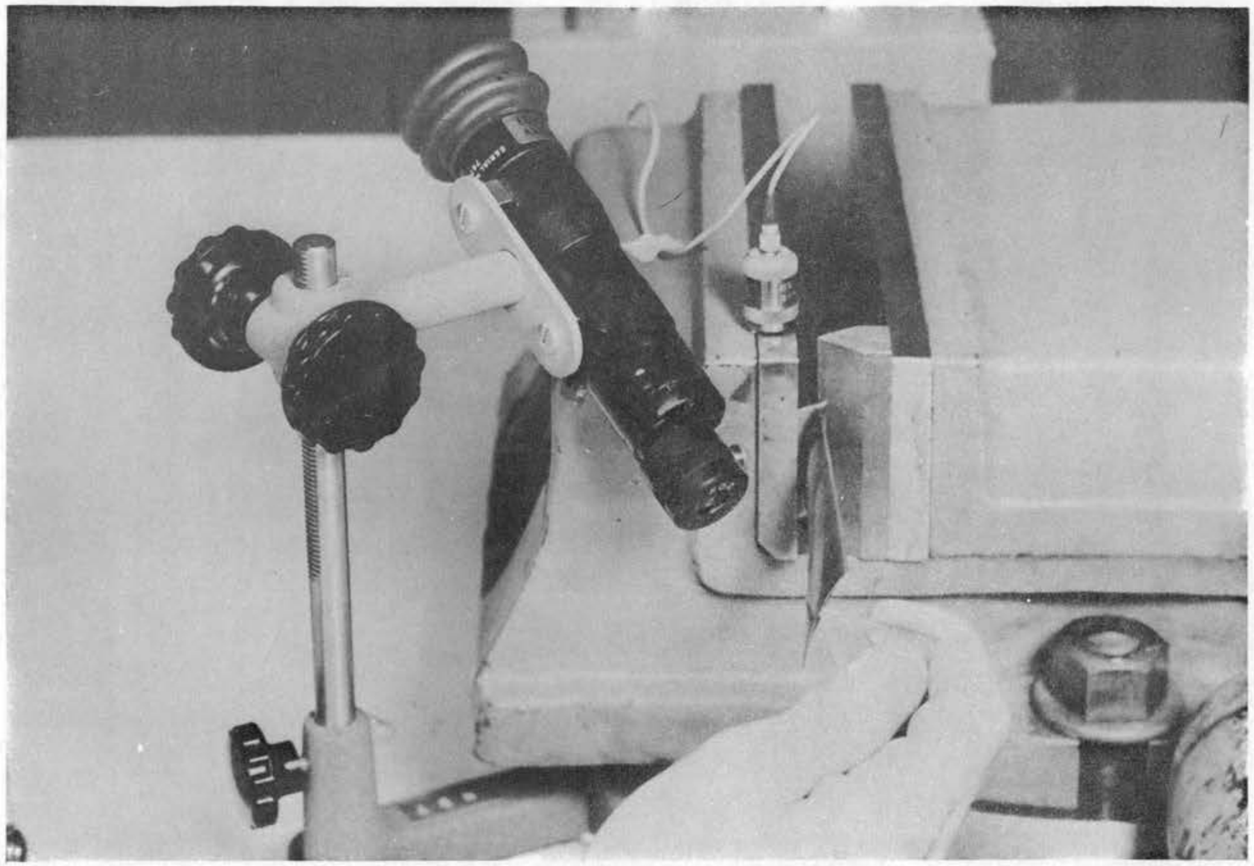


PLATE VI
CLOSE UP OF OPTICAL MEASURING DEVICE, CLAMPING BLOCK,
ACCELEROMETER, AND DISPLACEMENT METHOD



CHAPTER IV

EXPERIMENTAL PROCEDURE

Compressor blades to be tested were selected from stages six, seven, eight, and nine because of difficulty in obtaining feasible crack propagation in other stages with existing apparatus. Because they were available in large amounts and less difficult to crack, a larger number of blades from stage nine were selected for testing. This selection made possible a check on random variation of blade dynamic properties.

The natural frequency and oscilloscope recordings of transient response were obtained with the previously explained apparatus for each blade before any stresses were introduced with the fatigue apparatus. After a desired number of fatigue cycles or change in current required to drive the vibrating system on the shaker was obtained, the blade was again tested for natural frequency and the response was recorded. A schematic diagram of the experimental procedure is shown in Figure 4.

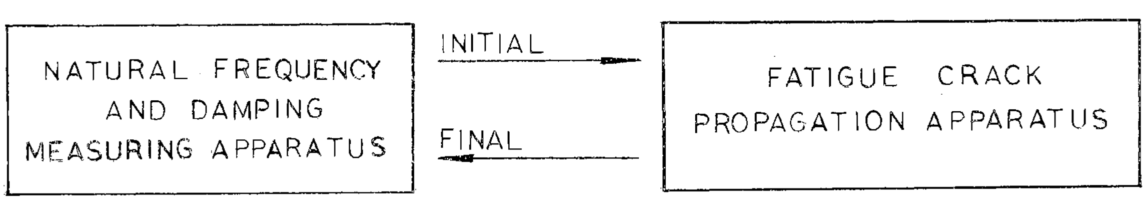


FIGURE 4. SCHEMATIC OF EXPERIMENTAL PROCEDURE

Propagation of fatigue cracks was monitored in the following manner. Blades to be fatigued were clamped firmly, and care was taken to insure good contact at the shaker head. All vibratory fatigue tests were conducted at a frequency slightly lower than the natural frequency of the system for this reason. At these frequencies a small change in stiffness or internal damping of the blade resulted in a large change in the system impedance. Therefore, the presence of a fatigue crack or flaw was made known by a noticeable change in driving current.

After reaching a certain change in current required to drive the system, the blades were examined closely during vibration to determine the presence of fatigue cracks. Parameters recorded for use in accurate control of fatigue crack propagation were: driving frequency, amplitude of vibration, RMS acceleration of the shaker head, initial and final readings of current required to drive the system, and number of vibratory cycles. Vibratory stresses were maintained well above the endurance limit of the blade material to insure production of fatigue cracks in a reasonable length of time. Using a strain gage instrumentated blade, peak dynamic strains were found to be approximately 4 to 5 milli inches per inch.

Apparatus described in the previous chapter was used to determine natural frequencies as follows. Oscilloscope monitoring of the amplifier output insured proper gain settings for sinusoidal wave form. Oscillator tuning to the natural frequency was controlled by noting maximum voltage output of the accelerometer. Measurement of natural frequencies to one decimal place was made possible with the electronic counter by noting the number of oscillator cycles in ten seconds and dividing this number by ten. Blade natural frequencies were found by multiplying the oscillator frequencies by two. This was done because the electromagnetic driving force frequency was twice the frequency of the applied voltage.

Transient vibratory response records were obtained as follows. Blade position was controlled by alignment of the clamping block with the edge of the vise and adjustment of the blade in height to touch a 0.5" gage block on the flat surface between the vise jaws. Practice excitations of the blade were made until results were satisfactorily reproduced. Oscilloscope pictures were made for each of two succesive excitations of every blade to insure obtaining a valid record of response. All oscilloscope records were made with the same settings of the oscilloscope and charge amplifier. All blades were excited by a 0.01" displacement controlled by the optical measurement device, except for thirteen blades tested from stage nine which were displaced 0.0125".

CHAPTER V

EXPERIMENTAL CONSIDERATIONS

Some error exists in all experimental investigation. Observation of the behavior of the test apparatus and equipment throughout the investigation gave some idea of the type and magnitude of errors introduced in these tests. This chapter will discuss certain relevant experimental observations.

The vise used for clamping the blades in the fatigue crack production apparatus was tilted, as shown in Plate II to reduce the effect of torsional coupling with lateral bending of the blade. In addition tests in locating the approximate shear center of the blade were conducted so that the shaker head could be attached near the shear center, thereby reducing coupled bending effects. It was observed that galling of the contact points in the shaker head had an effect on the current required to drive the system. This effect decreased the degree of accuracy of the crack monitoring process. Therefore, heat treatment of the tips in the shaker head was necessary to prevent galling of the tip material and resulting movement of the shaker head along the blade surface.

Several tests were conducted before selecting a location for the accelerometer on the vise. Since it was desirable to test a large number of blades, the accelerometer was mounted in a position which yielded maximum response but did not interfere with rapid changing of blades.

The electromagnet was clamped and held in place using wooden blocks. This method was selected because of the high damping and non-magnetic properties of wood. Experimentation with variable blade position vertically in the blocks and at distances varying from 0.125 to 0.25 inches from the electromagnet revealed little effect on natural frequencies measured. Tests carried out using varying clamping pressure revealed that values of clamping torque above 25 foot-pounds would give reproducible results. Natural frequency measurements at high and low resonant amplitudes were found to be the same within the margin of experimental error.

Degeneration of the amplifier wave form from a purely sinusoidal form produced electromagnetic forces containing beats and complex forcing functions. If a blade was vibrated in this manner, complex transmitted forces picked up by the accelerometer made oscillator tuning to the natural frequency difficult due to fluctuations of the voltmeter readings. Therefore, an oscilloscope was connected to the power amplifier output to insure that amplitude controls and impedance matching were such that a pure sinusoidal wave form of the applied voltage appeared at the electromagnet.

The natural frequencies of badly cracked blades were difficult to obtain because blade stiffness was decreased to the extent that large amplitudes were encountered. During vibration at these amplitudes, the electromagnetic driving force (which is inversely proportional to amplitude) varies considerably. Therefore, since the blade under these conditions is being vibrated with a nonlinear amplitude dependent force, natural frequencies obtained are questionable.

Experiments were conducted using various methods of exciting the blades before the displacement procedure was chosen. It was felt that the displacement method was simpler and more easily controlled with existing apparatus. The problem was to excite the blades at the lowest mode with the least amount of higher frequency transients so that a larger portion of the response could be used for damping analysis with good reproducibility of results. In exciting the blades an excessive amount of higher frequency transients were readily detectable by characteristic overtones of the natural frequency of vibration. Practice with the apparatus made possible the release of the blade from its deflected position with little scraping of the tip surface and consequent little excitation of higher modes of vibration. Deflection of the blade tip was controlled by the optical measuring device to plus or minus 0.0013 inches. Variations within these tolerances from the standard deflection set for each stage were not detectable in the recorded response.

Due to the extremely small voltage levels of response recorded in damping measurements, sixty cycle amplifier noise (which was rated at plus or minus one millivolt) was apparent in all oscilloscope data. (See Plate X). Experiments were carried out in grounding all equipment and shielding instrument leads without affecting the noise signal.

Different trigger level settings of the oscilloscope were made in order to determine the effects on the response obtained. These tests indicated that the oscilloscope was triggered at the first zero velocity position of the blade after release in every instance. The oscilloscope used was equipped with an automatic sweep lockout which enabled photographing of one sweep only. This feature eliminated the possibility of photographing multiple sweeps of the transient response.

CHAPTER VI

RESULTS

Qualitative and quantitative results of this investigation showing the effect of fatigue crack propagation on natural frequency and damping of compressor blades are contained in this chapter.

Qualitative Results

Tabulated results show natural frequencies of the blades before and after crack introduction. Representative oscilloscope records of the transient response of the blades are presented in Plate VII for the before and after cracking conditions. The remainder of the data taken is on file because it is too bulky to include here. Results of an experiment to determine accuracy in reproduction of blade response are included in Appendix A.

Data recorded in Tables I and II tend to show that natural frequency of the blade is affected by the presence of a fatigue crack. The sensitivity of blade damping to crack propagation is evidenced by the fact that recorded transient response data show dependence on the number of fatigue cycles even without microscopic detection of a fatigue crack in the blade surface. Response records were found to change markedly with the introduction of a fatigue crack in a blade. Representative microphotographs showing the size of cracks detectable by observing blade transient response are pictured in Appendix C.

TABLE I

RECORDED NATURAL FREQUENCY DATA

Blade Designation Stage	Number	Number of Stress Cycles	Natural Frequency (cycles per second)	Comment
6	1	0	264.4	
6	1	10,803	260.0	
6	4	0	265.3	
6	4	22,088	224.0	
6	5	0	266.1	
6	5	30,720	262.0	
6	6	0	272.2	
6	6	14,988	269.8	No Visible Crack
6	6	18,484	265.4	Small Crack
6	9	0	268.8	
6	9	32,743	260.4	
7	1	0	265.3	
7	1	19,568	261.0	Very Small Crack
7	4	0	263.4	
7	4	66,170	250.0	No Visible Crack
7	6	0	247.6	
7	6	34,867	248.2	No Visible Crack
7	6	44,263	248.0	
7	7	0	257.4	
7	7	55,809	253.8	No Visible Crack
7	7	58,612	236.2	
8	2	0	271.6	
8	2	59,514	267.0	
8	4	0	279.6	
8	4	43,657	280.2	
8	5	0	267.0	
8	5	37,369	263.6	
8	10	0	272.2	
8	10	41,783	195.8	Blade Nearly Broken
9_w	1	0	293.4	
9_w	1	25,343	291.8	
9_w	2	0	269.1	
9_w	2	19,440	253.6	
9_w	3	0	282.6	
9_w	3	15,878	280.6	No Visible Crack
9_w	3	20,481	270.0	
9_w	5	0	303.2	
9_w	5	38,218	287.0	No Visible Crack
9_n	1	0	296.8	
	1	23,548	293.8	
	2	0	285.0	
	2	10,244	259.0	
	3	0	298.6	
	3	33,562	293.8	

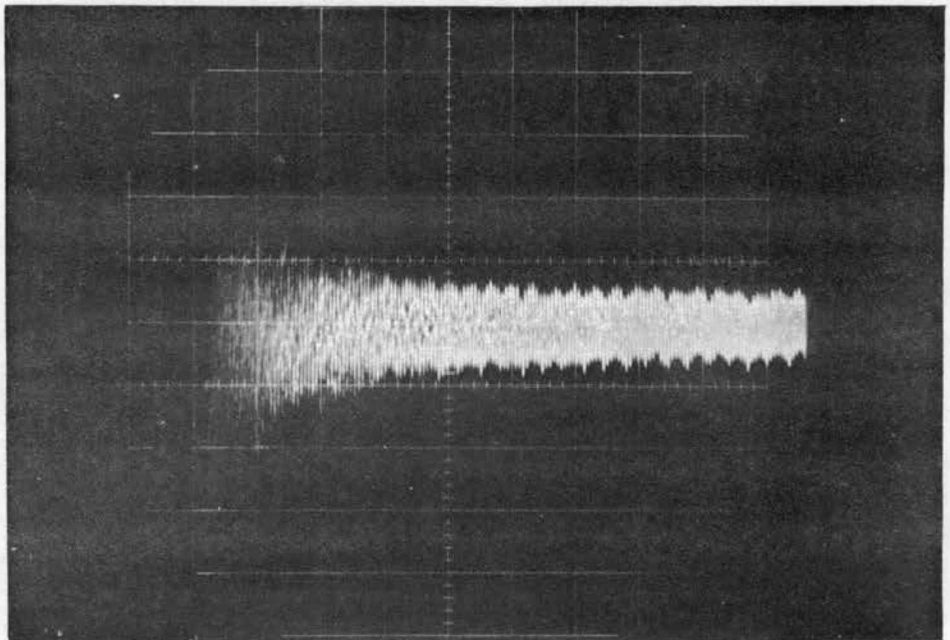
TABLE I (CONTINUED)

Blade Designation Stage	Number	Number of Stress Cycles	Natural Frequency (cycles per second)	Comment
9_n	5	0	288.6	
	5	10,897	279.6	
	6	0	288.2	
	6	23,055	274.8	
	7	0	283.8	
	7	12,785	194.6	Blade Nearly Broken
	8	0	296.0	
	8	11,042	294.4	No Visible Crack
	9	0	289.2	
	9	11,368	277.4	
	12	0	286.2	
	12	35,461	279.0	
	14	0	295.2	
	14	37,431	285.4	
	15	0	280.4	
	15	15,952	270.6	
	17	0	288.0	
	17	24,443	274.0	
	18	0	285.4	
	18	33,185	278.6	
	19	0	283.2	
	19	21,714	264.2	
	35	0	292.8	
	35	12,202	284.8	
	40	0	284.0	
	40	23,253	278.4	
	43	0	286.2	
	43	52,710	278.6	
	46	0	287.2	
	46	48,744	282.4	
	48	0	298.1	
	48	48,737	282.0	
	51	0	286.0	
	51	85,026	256.6	
	55	0	283.8	
	55	62,030	270.0	
	56	0	286.4	
	56	25,665	283.2	
	58	0	293.6	
	58	16,165	283.8	
	62	0	281.2	
	62	57,609	281.4	No Visible Crack
	72	0	285.0	
	72	30,824	282.0	

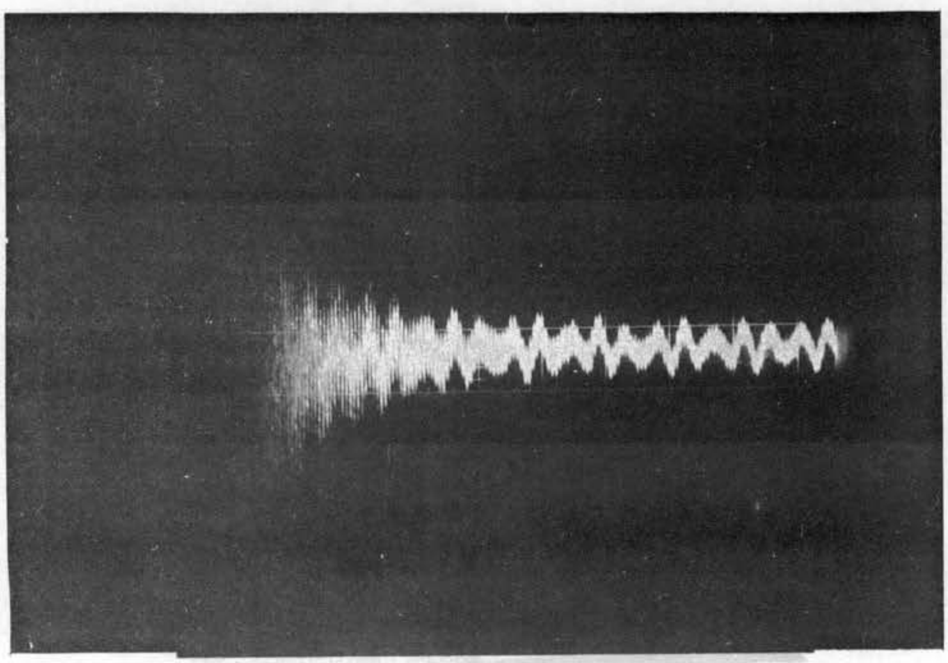
KEY: 9_w — Stage 9 Wide Base, 9_n — Stage 9 Narrow Base

PLATE VII

REPRESENTATIVE RECORDED RESPONSE DATA

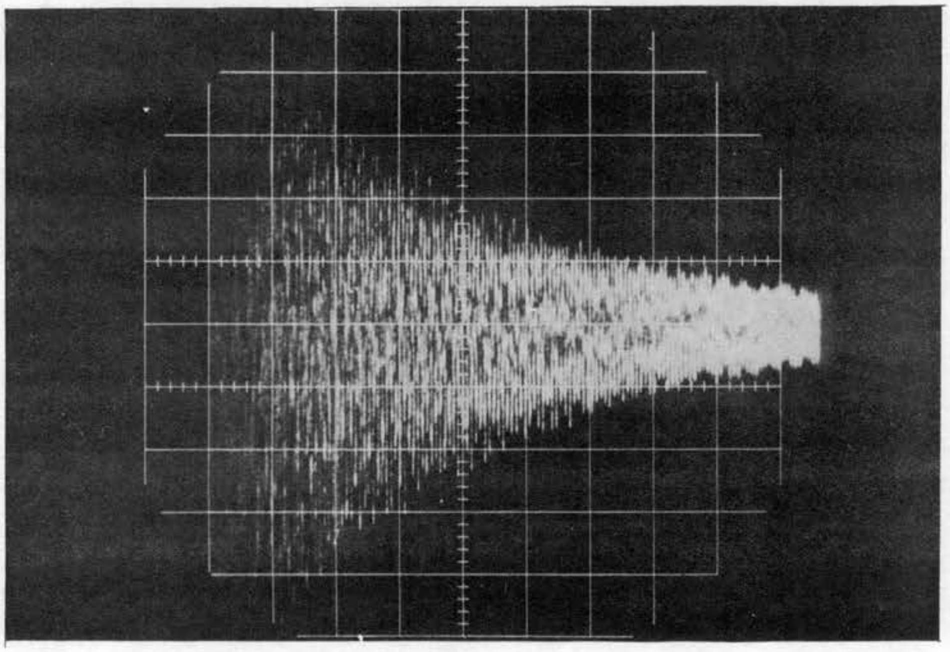


1. Blade 9_W - 3, Zero Stress Cycles

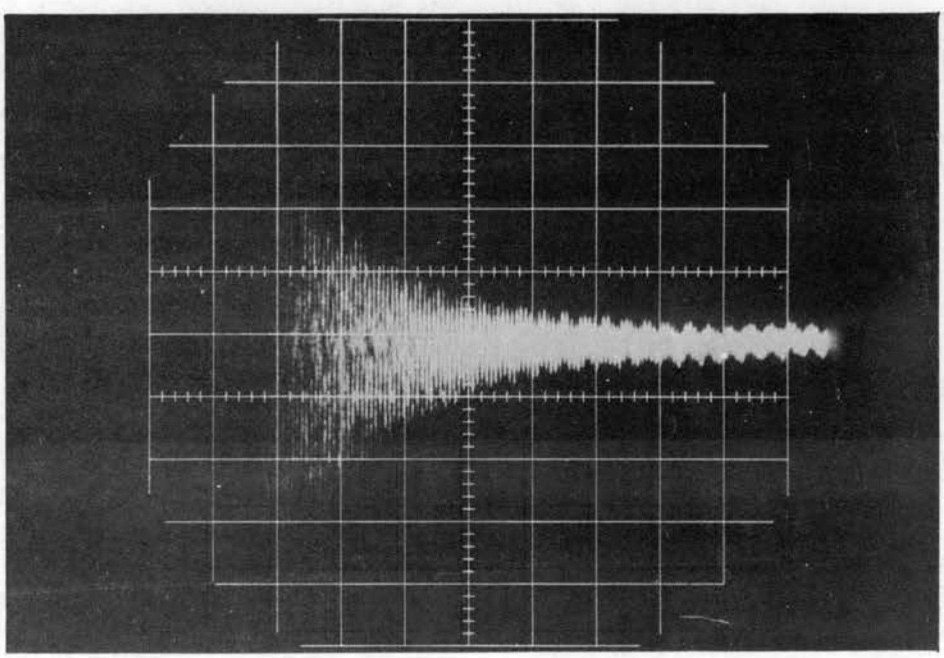


2. Blade 9_W - 3, 33,421 Stress Cycles

PLATE VII (CONTINUED)

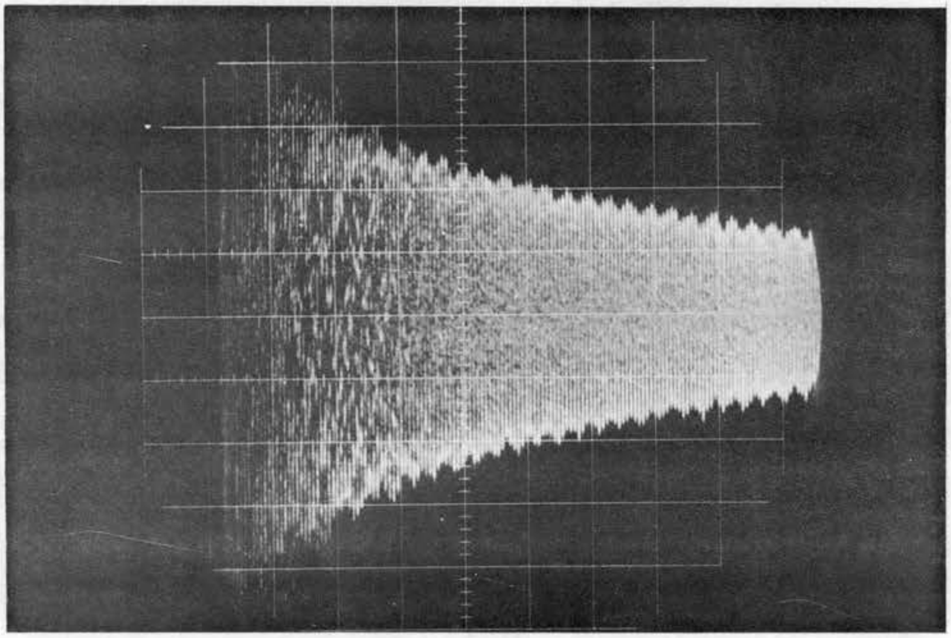


3. Blade 6-9, Zero Stress Cycles

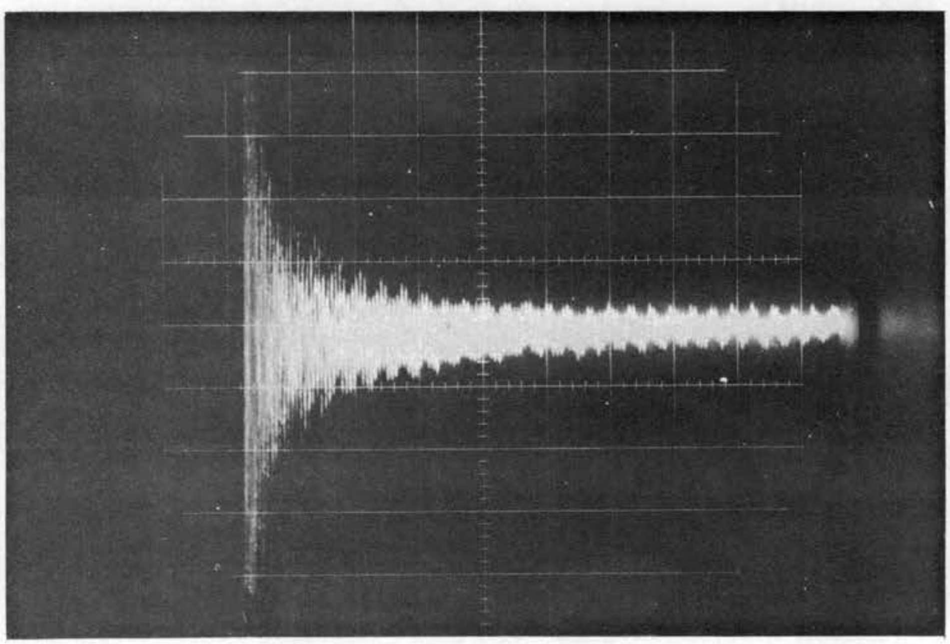


4. Blade 6-9, 32,743 Stress Cycles

PLATE VII (CONTINUED)



5. Blade 9_n^* - 43, Zero Stress Cycles



6. Blade 9_n^* - 43, 52,710 Stress Cycles

Quantitative Results

The analytical analysis of the response curves presents a problem because vibrations of the blades, especially cracked blades, are decidedly nonlinear. Comparisons based on logarithmic decrement are not possible because of amplitude dependence. Since the enlargement scale of different oscilloscope response pictures is not constant, absolute values of response will not give a true representation of the results.

These considerations led to an analysis based on a plot of logarithm of amplitude ratio of the response at a defined amplitude versus relative amplitude. Relative amplitudes are calculated by taking the ratio of the number of amplitude units and the number of units per screen division or,

$$\text{Relative Amplitude} = \frac{\text{units of amplitude}}{\text{units per screen division}} = \frac{A_n}{k} .$$

The damping property used is defined as damping index or,

$$\text{Damping Index} = \text{Log}_e \frac{A_{n-1}}{A_n} = \text{Log}_e \frac{D_{n-1}}{D_n}$$

where $D_n =$ Double Amplitude and $n = 1, 2, 3, \cdots$

Amplitudes on the response curve were measured starting two screen divisions from the initial trigger point in order to reduce the possibility of transient response errors.

Successive amplitudes were measured at two screen division intervals giving four amplitude and three damping index values per response curve. Plots of damping index versus relative amplitude are presented for each stage tested showing envelope of points for cracked and un-

TABLE II

CALCULATED NATURAL FREQUENCY DATA AND RESULTS

Blade Designation Stage	Number	Number of Stress Cycles	Change in Natural Frequency (cycles per second)	Comment
6	1	10,803	4.4	
6	4	22,088	41.3	
6	5	30,720	4.1	
6	6	14,988	2.4	No Visible Crack
6	6	18,484	6.8	Small Crack
6	9	32,743	8.4	
7	1	19,568	4.3	
7	4	66,170	13.4	
7	6	34,867	-0.6	No Visible Crack
7	6	44,263	-0.4	
7	7	55,809	3.6	No Visible Crack
7	7	58,612	21.2	
8	2	59,514	4.6	
8	4	43,657	-0.6	
8	5	37,369	2.4	
8	10	41,783	76.4	Blade Nearly Broken
9_w	1	25,343	1.6	
9_w	2	19,440	15.5	
9_w	3	15,878	2.0	No Visible Crack
9_w	3	20,481	12.6	
9_w	5	38,218	16.2	No Visible Crack
9_n	1	23,548	3.0	
	2	10,244	26.0	
	3	33,562	4.8	
	5	10,897	9.0	
	6	23,055	13.4	
	7	12,785	89.2	Blade Nearly Broken
	8	11,042	1.6	No Visible Crack
	9	11,368	11.8	
	12	35,461	7.2	
	14	37,431	9.8	
	15	15,952	9.8	
	17	24,443	14.0	
	18	33,185	6.8	
	19	21,714	19.0	
	35	12,202	8.0	
	40	23,253	5.6	
	43	52,710	7.6	
	46	48,744	3.8	
	48	48,737	16.1	
	51	85,026	29.4	
	55	62,030	13.8	
	56	25,665	3.2	
	58	16,165	9.8	
	62	57,609	-0.2	No Visible Crack
	72	30,824	3.0	

KEY: 9_w — Stage 9 Wide Base, 9_n — Stage 9 Narrow Base

TABLE III

REPRESENTATIVE CALCULATED RESPONSE DATA AND RESULTS

Blade Designation		Number of						RELATIVE AMPLITUDE						DAMPING INDEX		
Stage	Number	Stress Cycles	2k	D_o	D_1	D_2	D_3	$(A_o)_r$	$(A_1)_r$	$(A_2)_r$	$\frac{D_o}{D_1}$	$\frac{D_1}{D_2}$	$\frac{D_2}{D_3}$	$Ln \frac{D_o}{D_1}$	$Ln \frac{D_1}{D_2}$	$Ln \frac{D_2}{D_3}$
6	1	0	86	200	139	110	94	2.325	1.617	1.280	1.443	1.262	1.172	0.365	0.231	0.157
6	1	10,803	78	200	117	70	44	2.562	1.500	0.898	1.172	1.678	1.593	0.537	0.513	0.464
6	5	0	82	238	188	152	130	2.902	2.293	1.854	1.266	1.237	1.169	0.236	0.213	0.156
6	5	30,720	82	190	106	61	37	2.317	1.293	0.744	1.792	1.738	1.649	0.583	0.553	0.500
6	9	0	80	289	174	110	74	3.613	2.175	1.375	1.661	1.582	1.486	0.507	0.459	0.396
6	9	32,743	80	170	61	26	20	2.125	0.763	0.325	2.787	2.346	1.300	1.026	0.854	0.262
7	1	0	82	252	176	129	101	3.073	2.146	1.573	1.432	1.364	1.277	0.359	0.310	0.245
7	1	19,568	84	271	157	95	61	3.226	1.869	1.131	1.726	1.653	1.557	0.546	0.503	0.443
7	4	0	82	257	170	130	111	3.134	2.073	1.585	1.512	1.308	1.171	0.413	0.269	0.158
7	4	66,170	76	244	148	100	74	3.211	1.947	1.316	1.649	1.480	1.351	0.500	0.392	0.301
7	7	0	73	240	161	114	88	3.288	2.205	1.562	1.491	1.412	1.295	0.399	0.345	0.259
7	7	55,809	84	193	113	71	50	2.298	1.345	0.845	1.708	1.592	1.420	0.535	0.465	0.351
7	7	58,582	76	68	35	21	15	0.894	0.461	0.276	1.943	1.667	1.400	0.664	0.511	0.337
8	4	0	88	128	93	80	71	1.455	1.507	0.909	1.376	1.163	1.127	0.319	0.151	0.111
8	4	43,658	84	148	100	70	56	1.762	1.190	0.833	1.480	1.429	1.250	0.392	0.357	0.223
8	5	0	84	129	92	77	70	1.536	1.095	0.916	1.402	1.195	1.100	0.338	0.178	0.095
8	5	37,369	84	160	104	71	54	1.905	1.238	0.845	1.538	1.465	1.315	0.431	0.382	0.274
8	10	0	84	145	89	75	69	1.726	1.060	0.893	1.629	1.187	1.087	0.488	0.171	0.083
8	10	41,783	84	56	23	12	11	0.667	0.274	0.143	2.435	1.917	1.091	0.888	0.651	0.087
9_w	1	0	84	169	96	62	50	2.012	1.143	0.738	1.760	1.548	1.240	0.565	0.437	0.215
9_w	1	25,343	84	106	52	30	22	1.262	0.619	0.357	2.038	1.733	1.364	0.713	0.550	0.310
9_w	3	0	86	156	92	66	56	1.814	1.070	0.767	1.696	1.384	1.179	0.538	0.332	0.165
9_w	3	15,878	84	166	87	50	37	1.976	1.036	0.595	1.908	1.740	1.351	0.646	0.554	0.301
9_w	3	20,481	78	152	81	44	32	1.949	1.038	0.564	1.877	1.841	1.375	0.630	0.610	0.319
9_w	5	0	84	180	102	66	52	2.143	1.214	0.786	1.765	1.545	1.269	0.568	0.435	0.238
9_w	5	38,218	84	48	18	15	13	0.571	0.214	0.179	2.667	1.200	1.154	0.982	0.182	0.143
9_n	46	0	84	121	76	61	53	1.440	0.904	0.630	1.592	1.246	1.151	0.465	0.220	0.141
9_n	46	48,744	76	71	33	20	15	0.934	0.434	0.263	2.152	1.650	1.333	0.766	0.501	0.287
9_n	62	0	84	164	87	68	60	1.952	1.036	0.810	1.885	1.279	1.133	0.634	0.246	0.125
9_n	62	57,609	76	92	52	42	36	1.211	0.684	0.552	1.769	1.238	1.167	0.570	0.214	0.154
9_n^*	15	0	76	177	120	86	63	2.328	1.578	1.131	1.475	1.395	1.365	0.389	0.333	0.311
9_n^*	15	15,952	76	42	25	17	15	0.552	0.328	0.223	1.680	1.471	1.133	0.519	0.386	0.125

KEY: 9_w — Stage 9 Wide Base, 9_n — Stage 9 Narrow Base, 9_n^* — Stage 9 Narrow Base 0.0125" Relative Displacement, D_n = Double Amplitude, $(A_n)_r = \frac{D_n}{2k}$.

Remainder of data too bulky to include, will be kept on file.

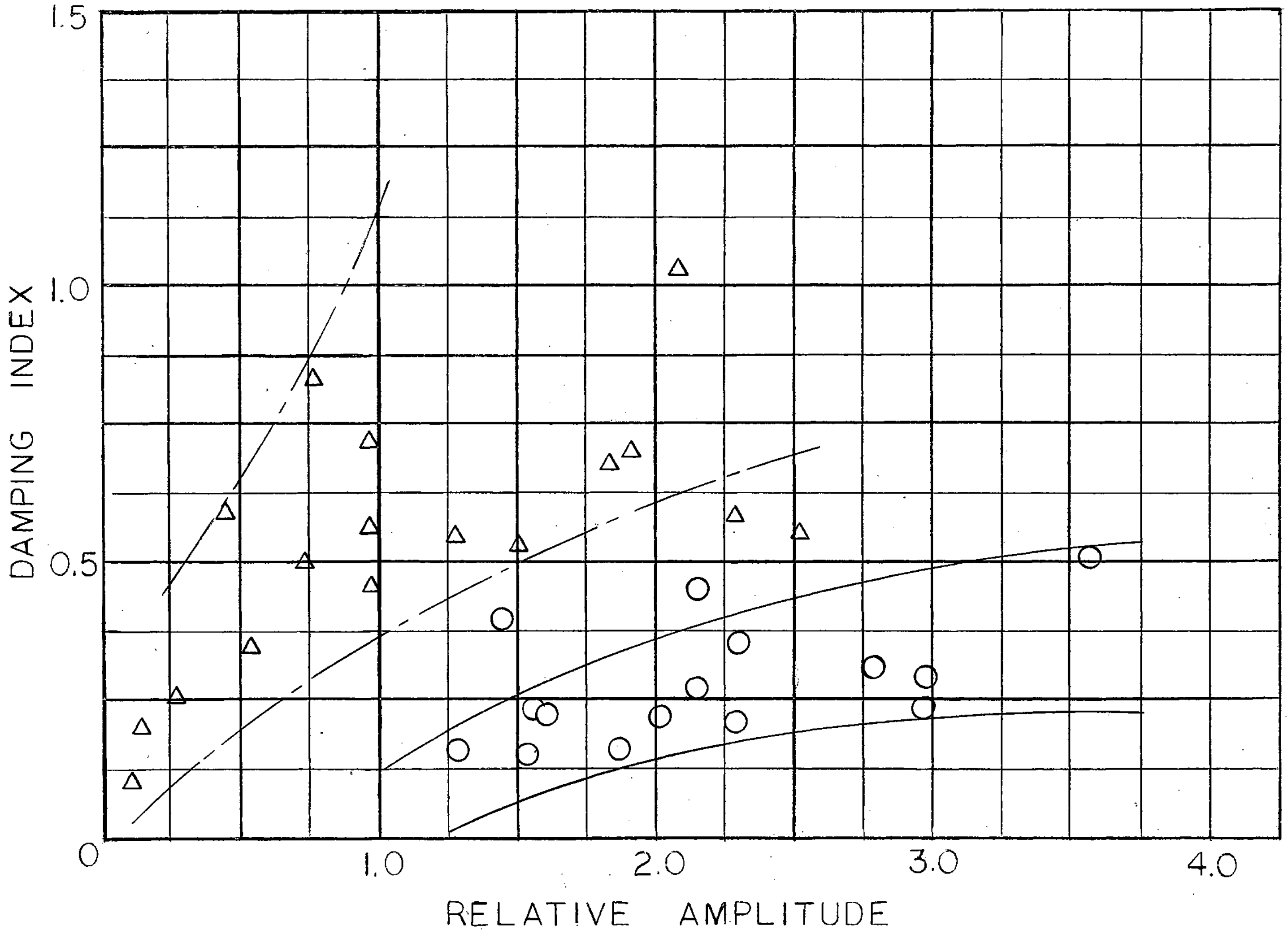


FIGURE 5. DAMPING INDEX VERSUS RELATIVE AMPLITUDE FOR STAGE 6

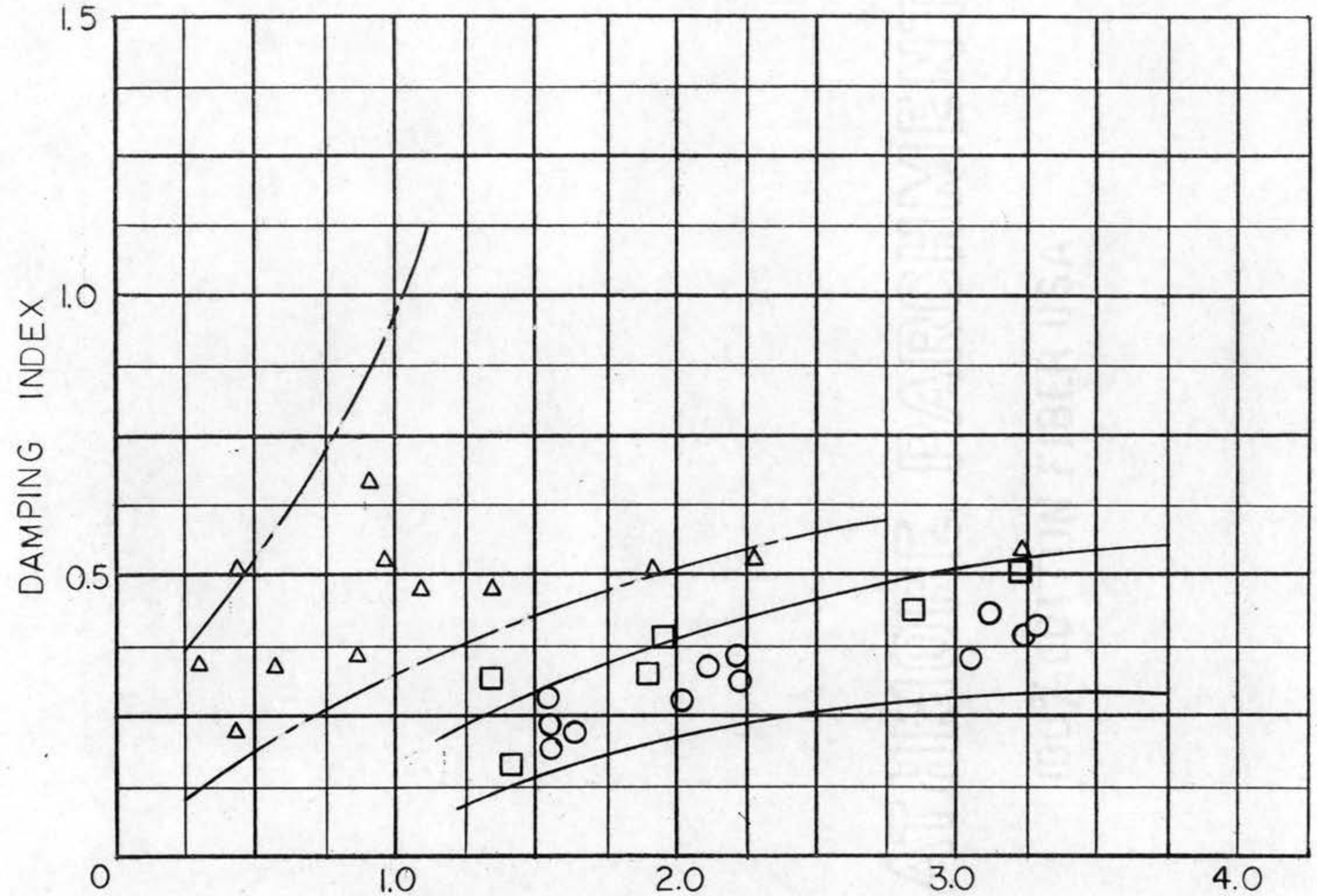


FIGURE 6. DAMPING INDEX VERSUS RELATIVE AMPLITUDE FOR STAGE 7

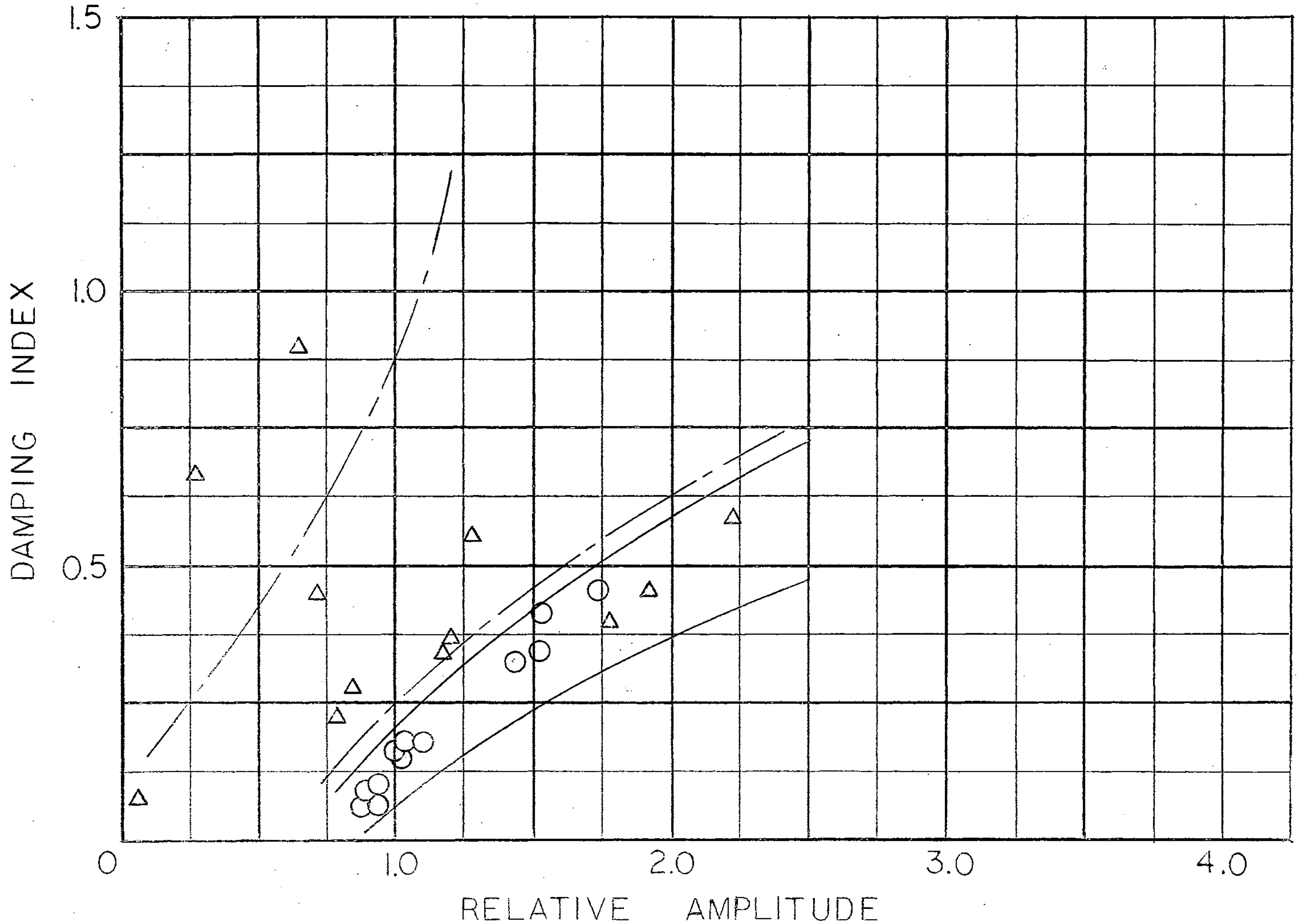


FIGURE 7. DAMPING INDEX VERSUS RELATIVE AMPLITUDE FOR STAGE 8

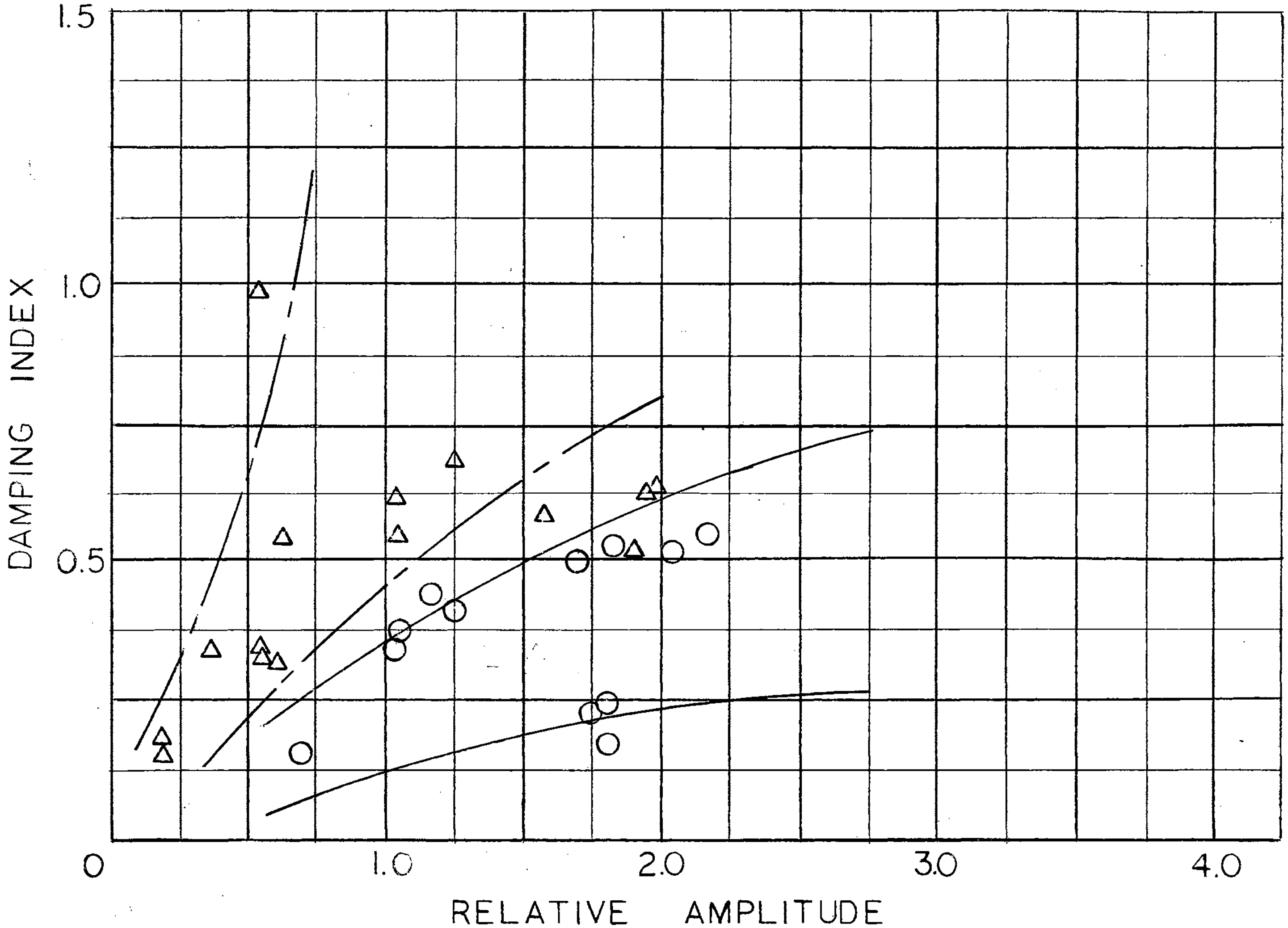


FIGURE 8. DAMPING INDEX VERSUS RELATIVE AMPLITUDE FOR STAGE 9 (WIDE BASE)

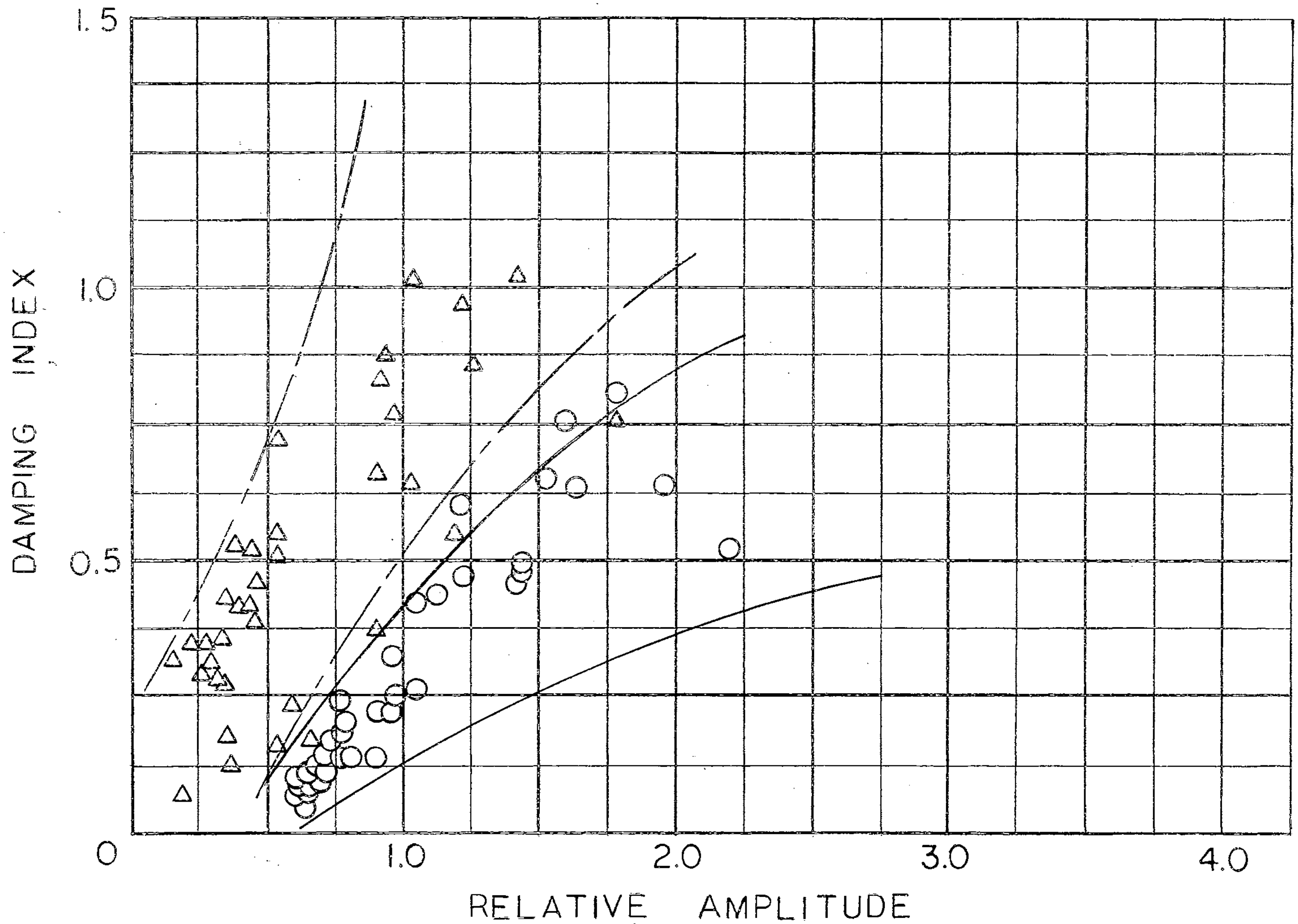


FIGURE 9. DAMPING INDEX VERSUS RELATIVE AMPLITUDE FOR STAGE 9 (NARROW BASE, 0.01" DISPLACEMENT)

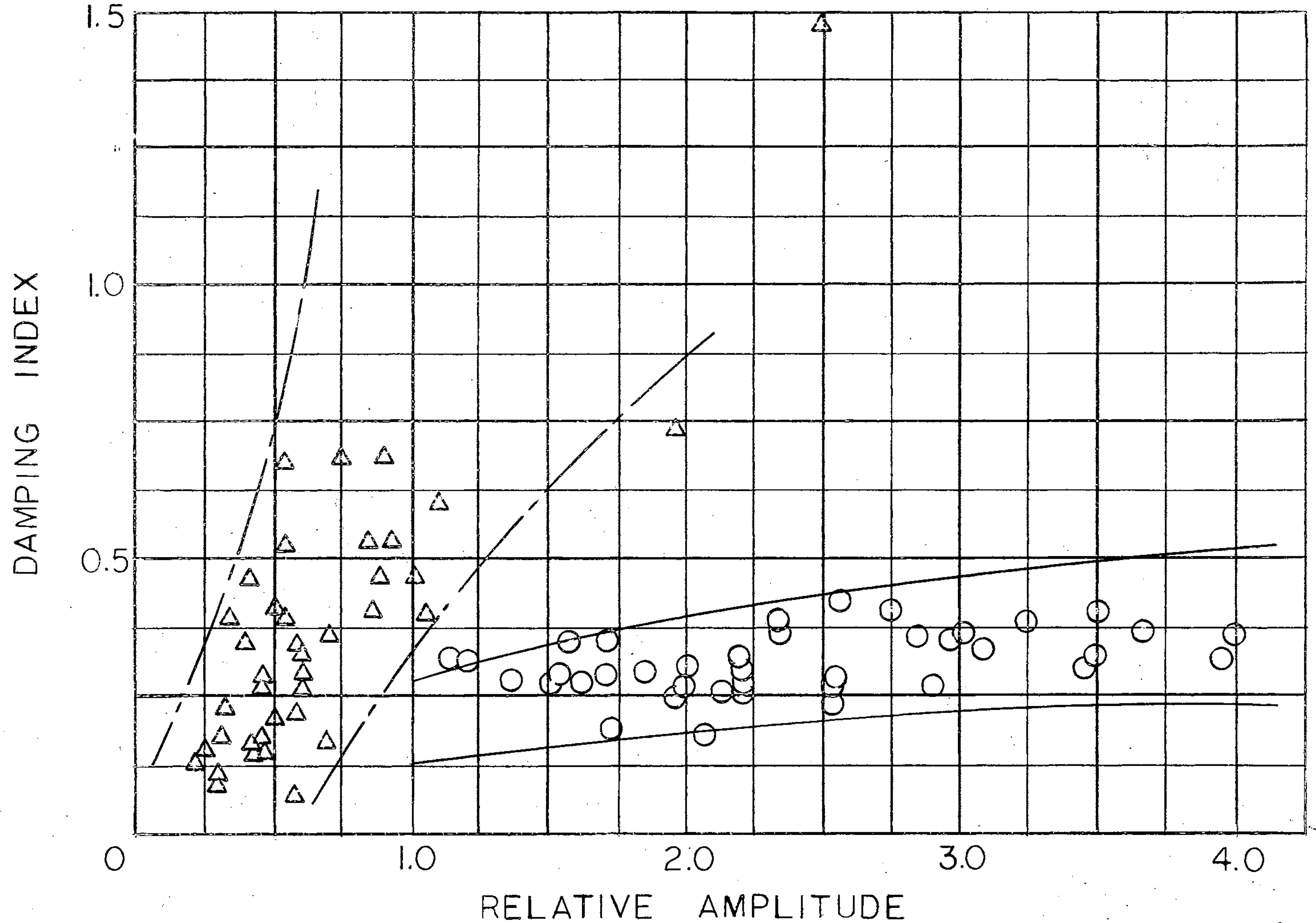


FIGURE 10. DAMPING INDEX VERSUS RELATIVE AMPLITUDE FOR STAGE 9 (NARROW BASE, 0.0125" DISPLACEMENT)

CHAPTER VII

ANALYSIS OF RESULTS

In general the natural frequency of all blades tended to decrease as high level fatigue stresses were introduced by the fatigue apparatus. This effect was probably due to a local decrease in stiffness and increase in internal damping in the vicinity of fatigue cracks propagated in the blades. However, the magnitude of this decrease in natural frequency was not sufficient to distinguish from the normal variation of natural frequency for uncracked blades. One reason for the small change in natural frequency can be attributed to the fact that although local yielding occurs in the vicinity of a fatigue crack, strain hardening of adjacent areas may be of sufficient magnitude to offset this effect.

Response curves were observed to change markedly with fatigue crack propagation. The increase of damping in highly stressed blades may, in part, be attributed to high energy dissipation due to irreversibility of the plastic deformation within the material.

Plots of damping index versus relative amplitude revealed considerable scatter in the variables. This was probably due to experimental errors coupled with errors encountered in the response curve analysis. The method of controlling excitation of the blades was probably the largest source of experimental error. Charge amplifier noise signal contributed to error because of difficulty in obtaining true values of response amplitude for analysis. A large numerical error was encountered when taking the logarithm of the ratio of two nearly equal amplitude values. (See Appendix B).

The best results were obtained using a 0.0125" displacement excitation for stage nine blades. A large separation of the envelope of points for cracked and uncracked blades was observed. These results indicate that the damping mechanisms encountered in a cracked blade are highly amplitude dependent.

CHAPTER VIII

CONCLUSIONS AND RECOMMENDATIONS

The purpose of this investigation was to determine if crack propagation in selected J-57 jet engine compressor blades had any measurable effect on the vibratory parameters of natural frequency and damping. From a definite establishment of this relationship, nondestructive tests could be performed on compressor blades by observing the deviation of natural frequency and damping from a set standard for the blades.

Since a sufficient number of compressor blades containing fatigue cracks incurred in actual service was not available for testing, the artificial means of producing and controlling fatigue cracks described in previous chapters were used. Therefore, the experimental results obtained are directly applicable only to the extent that fatigue stresses introduced in the blades by the vibratory fatigue apparatus in these tests approximate the actual fatigue stress life of blades in actual service. The low frequency, high level fatigue stresses used in crack propagation may have had adverse effects such as follows on the parameters measured: (a) Excessive strain hardening, (b) Excessive plastic deformation, and (c) Excessive rate of crack propagation.

However, the results obtained using the previously described crack propagation apparatus lead to the following conclusions:

1. The natural frequency of the blades was influenced by fatigue crack propagation, but the dependability and degree of this relationship was not definitely established.

2. The transient response records and related damping index of the compressor blades were definitely affected by the number of fatigue cycles experienced by the blades and the magnitude of the fatigue crack. Therefore, the presence of faults or cracks near the base of the blades tested was detectable without preparation of the blade surface by a simple vibratory nondestructive test utilizing comparison of transient response records or damping index.

It is recommended that investigations be carried out to determine the effect of different fatigue crack propagation methods on the change in natural frequency and damping of the compressor blades. Specifically, fatigue crack propagation using lower stress levels and higher frequencies with resultant increase in number of stress cycles should be investigated. The use of lower stresses in fatigue crack propagation would result in less rapid strain hardening which might give better results for the natural frequency tests.

Vibratory nondestructive tests of compressor blades can be performed without preparation of the blade surface, without requiring visual detection of a defect, and without special operator skill. Because of these advantages and because of trends established in this investigation, it is recommended that further natural frequency and damping research applied to the nondestructive testing of compressor blades be carried out.

Due to the sensitivity possible, more research in methods of application of damping measurements to nondestructive testing in general should be made. Possibilities exist for application of damping comparisons to the nondestructive testing of any machine part made in large quantities with close manufacturing control of mass, composition, and geometry. Rejection of a part could be based on the deviation of its transient response from a set standard. One suggested area of research is investigation of possible methods of reproducible excitation of test items applicable to mass production nondestructive testing.

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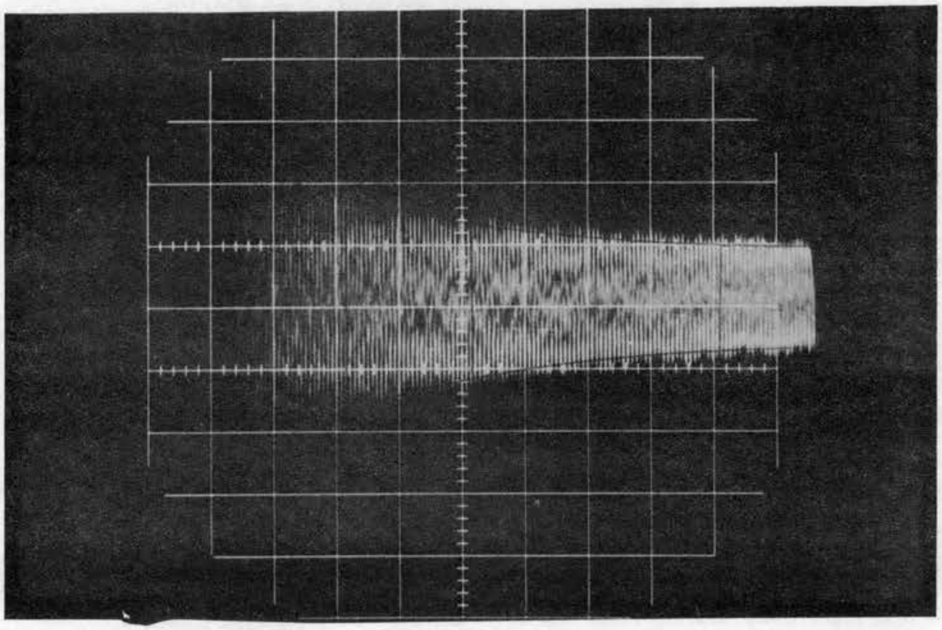
9. Lowery, R. L. et al., "Vibratory Testing of J-57 Compressor Vane and Shroud Assemblies", Oklahoma State University Engineering Research, Nondestructive Testing Project, Interim Report No. 3, April 1, 1962, pp. 4.

APPENDIX A

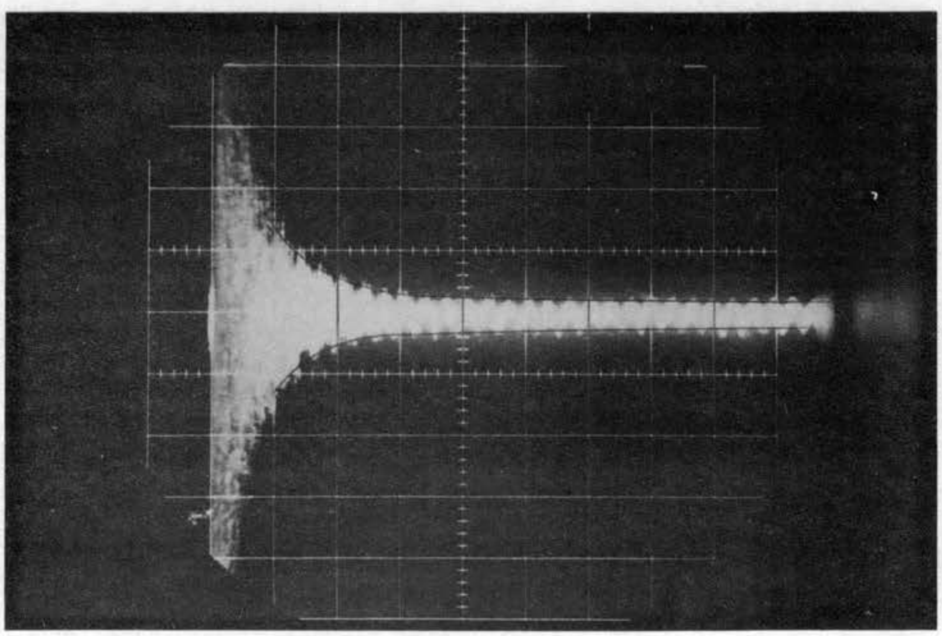
REPRODUCIBILITY OF BLADE RESPONSE

PLATE VIII

REPRESENTATIVE REPRODUCIBILITY DATA



1. Blade 6-2, Zero Stress Cycles



2. Blade 6-2, 206,000 Stress Cycles

TABLE IV

CALCULATED RESPONSE REPRODUCIBILITY DATA AND RESULTS

Blade Designation Stage	Blade Designation Number	Number of Stress Cycles	2_k	D_o	D_1	D_2	D_3	RELATIVE AMPLITUDE $(A_o)_r$	$(A_1)_r$	$(A_2)_r$	$\frac{D_o}{D_1}$	$\frac{D_1}{D_2}$	$\frac{D_2}{D_3}$	DAMPING INDEX $\ln \frac{D_o}{D_1}$	$\ln \frac{D_1}{D_2}$	$\ln \frac{D_2}{D_3}$
6	2_a	0	88	153	112	86	73	1.738	1.272	0.977	1.365	1.302	1.188	0.311	0.264	0.172
6	2_b		86	156	108	82	70	1.814	1.256	0.954	1.444	1.317	1.171	0.367	0.275	0.158
6	2_c		86	156	110	83	71	1.814	1.280	0.965	1.418	1.327	1.170	0.349	0.283	0.157
6	2_d		86	149	102	75	63	1.733	1.187	0.872	1.460	1.360	1.190	0.378	0.301	0.174
6	2_e		86	146	102	76	64	1.698	1.187	0.884	1.432	1.342	1.188	0.359	0.294	0.172
6	2_f		86	148	104	76	62	1.721	1.210	0.884	1.421	1.369	1.226	0.351	0.314	0.204
6	2_g		86	151	106	80	68	1.759	1.232	0.931	1.422	1.325	1.178	0.352	0.281	0.164
6	2_h		86	143	98	75	65	1.666	1.140	0.872	1.458	1.307	1.152	0.377	0.267	0.142
6	2_i		86	146	102	81	71	1.698	1.187	0.942	1.432	1.259	1.141	0.359	0.228	0.132
6	2_j		86	147	108	85	74	1.709	1.256	0.989	1.362	1.270	1.148	0.309	0.239	0.138
6	2_a	206,000	86	34	22	20	19	0.395	0.256	0.233	1.545	1.100	1.053	0.435	0.095	0.052
6	2_b		86	38	22	20	19	0.442	0.256	0.233	1.727	1.100	1.053	0.546	0.095	0.052
6	2_c		86	35	23	20	19	0.406	0.267	0.233	1.522	1.150	1.053	0.420	0.140	0.052
6	2_d		86	33	21	18	17	0.383	0.244	0.209	1.571	1.166	1.058	0.452	0.154	0.056
6	2_e		86	38	22	20	19	0.442	0.256	0.233	1.727	1.100	1.053	0.546	0.095	0.052
6	2_f		86	44	22	20	19	0.512	0.256	0.233	2.000	1.100	1.053	0.693	0.095	0.052
6	2_g		86	51	26	21	20	0.593	0.302	0.244	1.961	1.238	1.050	0.674	0.214	0.049
6	2_h		86	47	24	22	22	0.546	0.279	0.256	1.958	1.091	1.000	0.672	0.087	0
6	2_i		86	49	25	22	21	0.569	0.291	0.256	1.960	1.136	1.047	0.673	0.128	0.046
6	2_j		86	46	24	21	19	0.534	0.279	0.244	1.916	1.143	1.105	0.650	0.134	0.100

KEY: a, b, c, ..., j Represents Different Excitations, D_n = Double Amplitude, $(A_n)_r = \frac{D_n}{2k}$

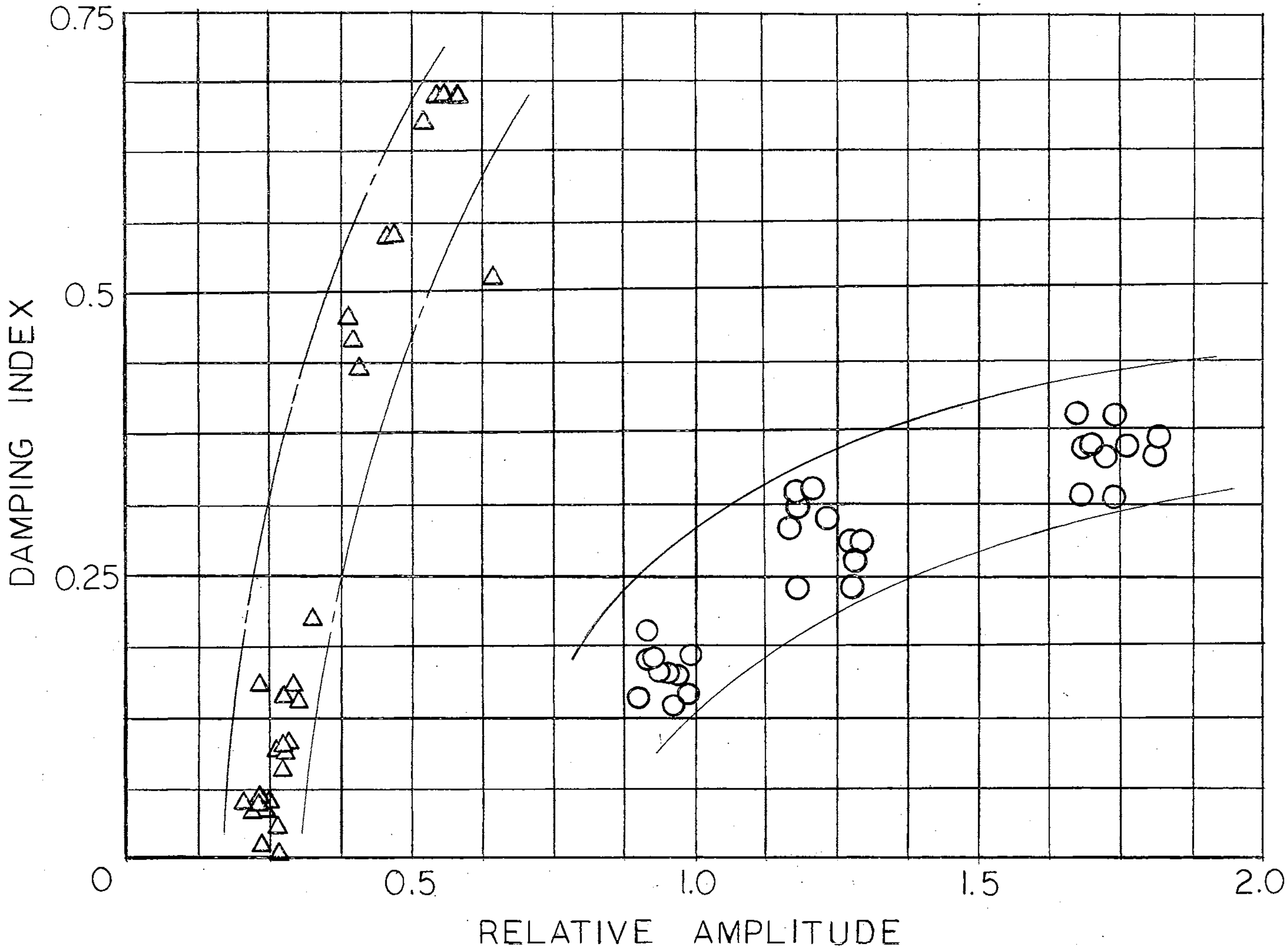


FIGURE 11. DAMPING INDEX VERSUS RELATIVE AMPLITUDE FOR SEVERAL EXCITATIONS OF A SINGLE BLADE

APPENDIX B

NUMERICAL ERRORS IN RESPONSE ANALYSIS

The presence of a steady noise signal from the charge amplifier on all oscilloscope records leads to a large percentage of error in amplitude measurements at low amplitudes. The damping index is defined as the natural logarithm of the ratio of selected amplitudes; and if the selected amplitudes are nearly equal, it is apparent that an error in the measurement of either amplitude will result in a magnified error in the damping index. Therefore in order to reduce the possibilities of gross errors in the analysis, amplitude measurements were made at two screen division intervals—where in general an appreciable difference in amplitude was realized—and all calculations were carried to three decimal places.

A response analysis based on amplitude measurements at one screen division intervals for two different blades is presented in this Appendix to show the effects of errors discussed above. It can be seen by following the path of points plotted in Figure 12 that deviation from a smooth curve caused by errors is more pronounced at lower amplitudes.

Symbols used on Figure 12 are:

○ ----------Zero high stress cycles for blade $9_n^* - 18$

⬡ ----------Zero high stress cycles for blade $9_n^* - 46$

△ ----------Known number of high stress cycles with fatigue crack in blade surface for blade $9_n^* - 18$

□ ----------Known number of high stress cycles with fatigue crack in blade surface for blade $9_n^* - 46$

TABLE V

CALCULATED RESPONSE DATA AND RESULTS FOR AMPLITUDES MEASURED AT ONE SCREEN DIVISION INTERVALS

Blade Designation Stage	Number	Number of Stress Cycles	2_k	D_0	D_1	D_2	D_3	D_4	D_5	D_6	D_7	RELATIVE AMPLITUDE $(A_0)_r$	$(A_1)_r$	$(A_2)_r$	$(A_3)_r$	$(A_4)_r$	$(A_5)_r$	$(A_6)_r$
9_n^*	18	0	76	266	216	180	151	128	112	96	84	3.500	2.842	2.368	1.990	1.685	1.472	1.261
9_n^*	18	33,185	76	58	35	29	25	21	19	19	17	0.763	0.461	0.381	0.330	0.276	0.253	0.253
9_n	46	0	84	121	90	76	66	61	58	53	50	1.440	1.073	0.904	0.870	0.726	0.691	0.637
9_n	46	48,744	76	71	44	33	26	20	19	15	13	0.934	0.581	0.434	0.342	0.263	0.252	0.198

TABLE V (CONTINUED)

Blade Designation Stage	Number	$\frac{D_0}{D_1}$	$\frac{D_1}{D_2}$	$\frac{D_2}{D_3}$	$\frac{D_3}{D_4}$	$\frac{D_4}{D_5}$	$\frac{D_5}{D_6}$	$\frac{D_6}{D_7}$	DAMPING INDEX $\ln\frac{D_0}{D_1}$	$\ln\frac{D_1}{D_2}$	$\ln\frac{D_2}{D_3}$	$\ln\frac{D_3}{D_4}$	$\ln\frac{D_4}{D_5}$	$\ln\frac{D_5}{D_6}$	$\ln\frac{D_6}{D_7}$
9_n^*	18	1.230	1.199	1.191	1.181	1.143	1.168	1.143	0.207	0.182	0.175	0.166	0.134	0.155	0.134
9_n^*	18	1.662	1.205	1.160	1.190	1.105	1.000	1.118	0.507	0.186	0.148	0.174	0.100	0	0.112
9_n	46	1.341	1.192	1.152	1.080	1.051	1.095	1.061	0.293	0.174	0.140	0.077	0.049	0.91	0.058
9_n	46	1.613	1.331	1.270	1.300	1.051	1.272	1.155	0.476	0.285	0.239	0.263	0.049	0.239	0.144

KEY: 9_n — Stage 9 Narrow Base, 9_n^* — Stage 9 Narrow Base 0.0125" Relative Displacement, D_n = Double Amplitude, $(A_n)_r = \frac{D_n}{2k}$.

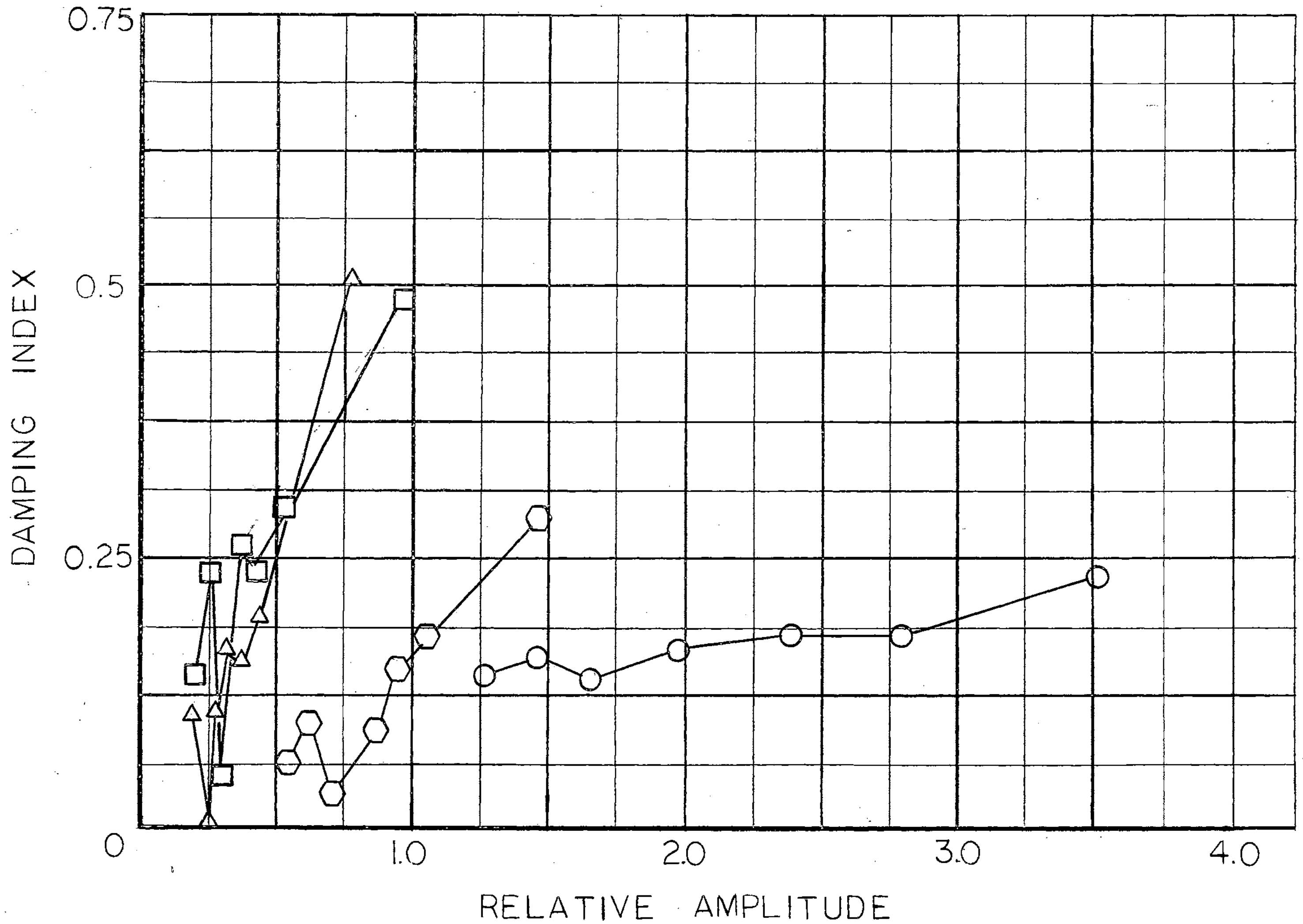


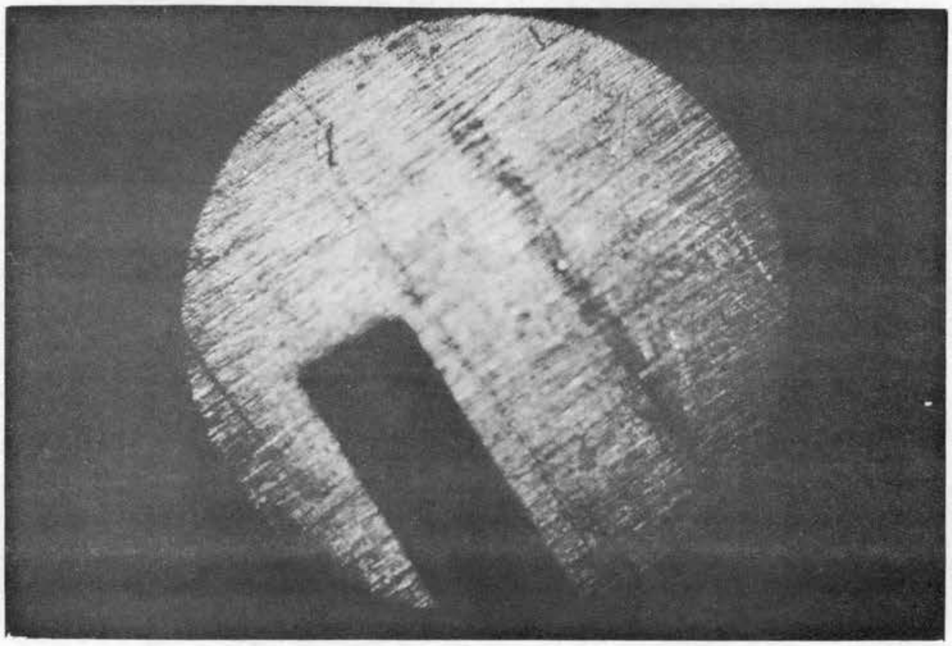
FIGURE 12. DAMPING INDEX VERSUS RELATIVE AMPLITUDE FOR AMPLITUDES MEASURED AT ONE SCREEN DIVISION INTERVALS

APPENDIX C

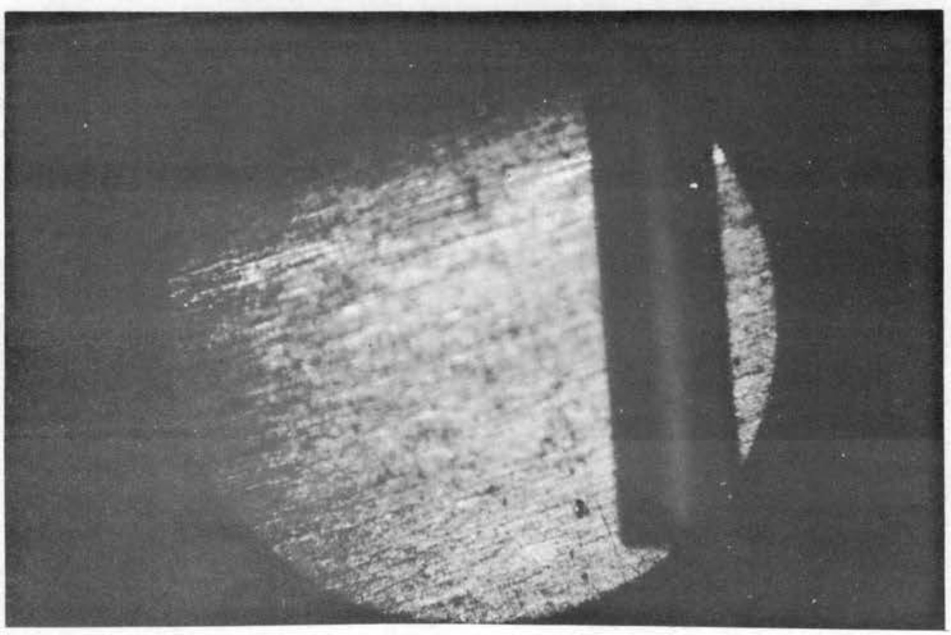
REPRESENTATIVE MICROPHOTOGRAPHS OF FATIGUE CRACKS DETECTED

PLATE IX

REPRESENTATIVE MICROPHOTOGRAPHS OF FATIGUE CRACKS DETECTED



1. Blade 9_n^* -14, 37,431 Stress Cycles, 25X Black Object is Fine Pencil Lead

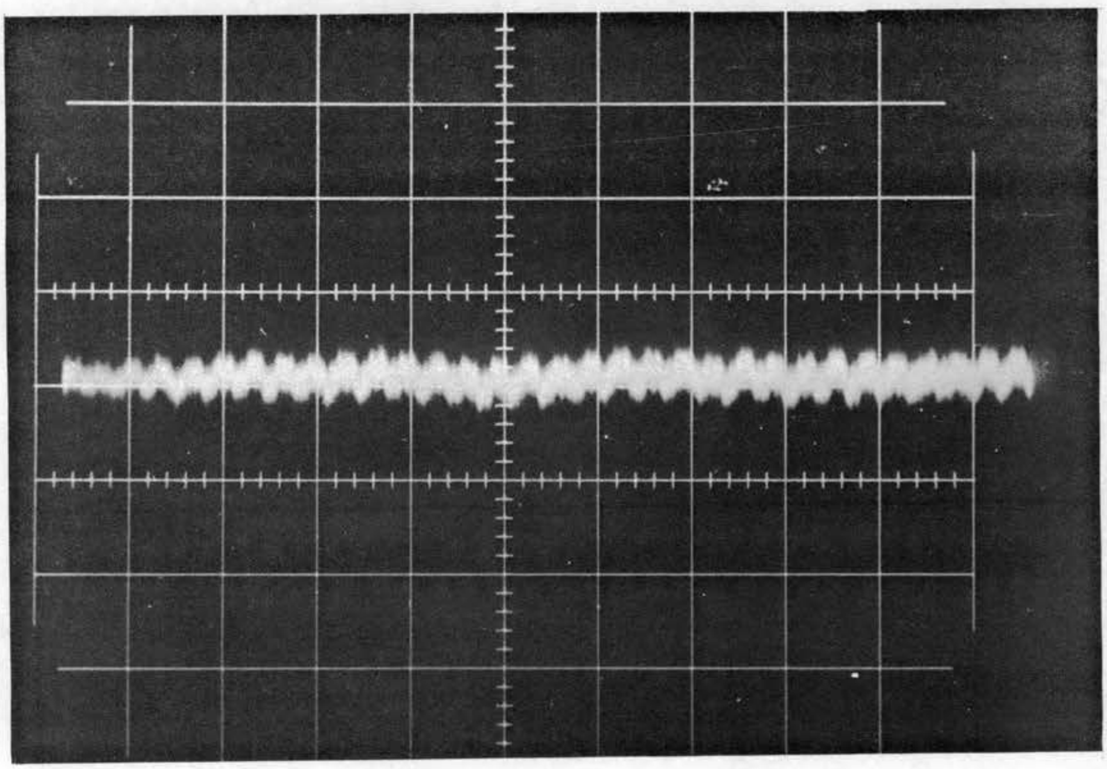


2. Blade 9_n - 48, 48737 Stress Cycles, 25X, Black Object is Fine Pencil Lead

APPENDIX D

PLATE X

TYPICAL OSCILLOSCOPE NOISE SIGNAL



APPENDIX E

TEST INSTRUMENTS

1. Electrodynamic Vibration Exciter: Manufacturer, MB Electronics; Model C-11D

2. Vibration Meter: Manufacturer, MB Electronics; Model M6

3. Digital Electronic Counter: Manufacturer, Beckman; Model 7360

4. True Root-Mean-Square Voltmeter: Manufacturer, Ballantine Laboratories Inc.; Model No. 320

5. Dual Beam Oscilloscope: Manufacturer, Tektronix, Inc.; Type 502

6. Oscilloscope Camera: Manufacturer, Allen B. Dumont Laboratories, Inc.; Type 271-A

7. Audio Amplifier: Manufacturer, Bogen Presto Co.; Model MO 100

8. Oscillator: Manufacturer, Hewlett-Packard; Model 200CD Audio

9. Electromagnet: Manufacturer, Oklahoma State University Mechanical Engineering Laboratory

10. Quartz Accelerometer: Manufacturer, Kistler Instrument Corporation; Model 812

11. Charge Amplifier: Manufacturer, Kistler Instrument Corporation; Model 568

12. Erect Image Filar Telescope: Manufacturer, Edmund Scientific Co.; Serial No. 39088

VITA

Robert Curtis Ikard

Candidate for the Degree of

Master of Science

THESIS: EFFECTS OF CRACK PROPAGATION ON DYNAMIC PROPERTIES OF COMPRESSOR BLADES

MAJOR FIELD: Mechanical Engineering

BIOGRAPHICAL:

Personal Data: Born in Liberal, Kansas, March 7, 1938, the son of Burton and Erma Ikard.

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