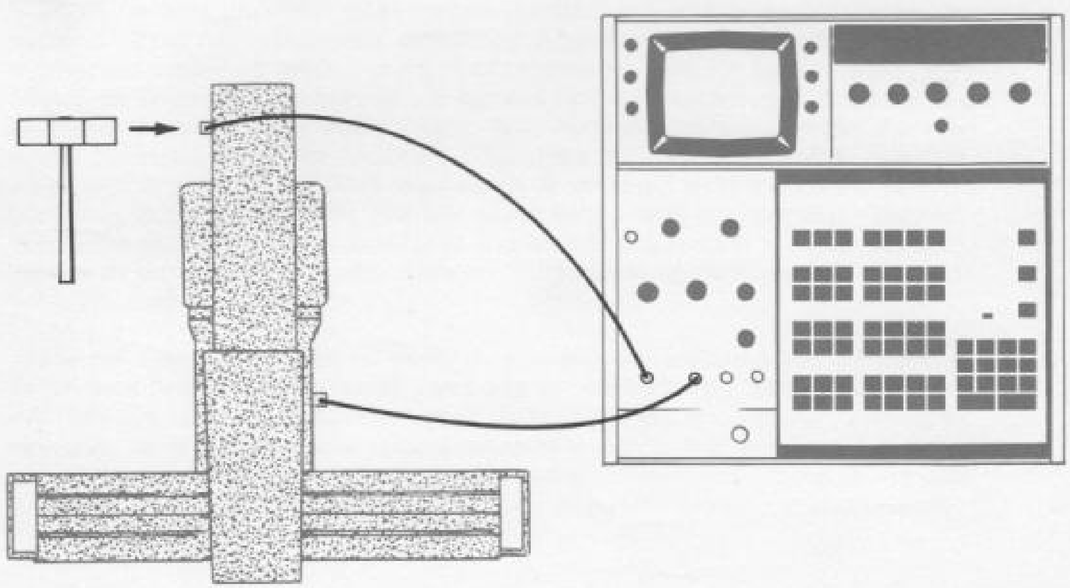
APPLICATION NOTE 140-3

Dynamic Testing of Mechanical Systems Using Impulse Testing Techniques

Summary of this Note

Impulses applied to a structure with a suitable hammer allow the dynamic characteristics of that structure to be rapidly and accurately measured. The HP Fourier Analyzer measures both the force and response so that the true "impedance" can be measured. A 100 fold saving in time over conventional sine sweep tests, with equal accuracy, can result from using this technique.

The simplicity of set up and speed of data reduction should make possible many mechanical investigations that were previously too time consuming and expensive. These techniques are directly applicable in design and testing of all types of machinery, automotive components, aircraft and other types of structures.



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This note is the result of a continuing research program on machine dynamics being performed at the University of Cincinnati. Part of this Application Note consists of a paper presented at the 13th International Machine Tool Design and Research Conference, University of Birmingham, England, September 22, 1972.

The second section of this Note describes how the Hewlett-Packard Fourier Analyzer was actually used in impulse testing.

APPLICATIONS OF PULSE TESTING FOR DETERMINING DYNAMIC CHARACTERISTICS OF MACHINE TOOLS

by

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SUMMARY

The theoretical relationships between the time response of a structure to an input pulse and the dynamic characteristics of the structure have long been known. However, recent improvements in transducers and in data acquisition and computing equipment have occurred which make pulse testing a practical method for determining the dynamic characteristics, frequency response and mode shapes of machine tools.

This paper presents several pulse test applications and correlations relating pulse test and sinusoidal (TFA) test results on machine tools and structures. The development of special techniques for pulse testing are also discussed.

INTRODUCTION

Many technical papers have been written in recent years describing various techniques and corresponding results for determining the dynamic characteristics of machine tools and other mechanical structures and/or electro-mechanical systems. Each has presented a convincing argument for the necessity of detailed information regarding these important characteristics and the form in which they should be presented or obtained. It appears that a presentation in the form of a frequency response curve and corresponding mode shapes is considered the most useful for determining the important parameters affecting the dynamic performance of a linear system. This information is obtained from theoretical analysis and/or experimental test. We feel that experimental tests are essential. They can supply the data required to understand and correct problems in existing machines, as well as provide the basis for mathematically modeling machines and structures for future analysis and design.

There are several experimental methods available for acquiring this information. Some have been in use for many years and are consequently well documented in the literature. Others, because of certain limitations, have not been fully developed. However, as is so often the case, developments occurring in different or allied areas can remove many of the previously existing limitations. Such is the case regarding the application of pulse testing for determining dynamic characteristics.

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†This paper presented on September 22, 1972 at the 13th International Machine Tool Design and Research Conference, University of Birmingham, Birmingham, England.

As pointed out in the conclusions of the paper by Bollinger and Bonesho[1], several factors were imposing definite limitations on the future development of the pulse testing technique. Three of the important factors mentioned were the pulsers, transducers, and the data reduction techniques and systems. We feel that improvements in the transducers and the development of techniques and equipment for data reduction have occurred which make pulse testing a feasible, reliable, and economical method.

Pulse testing involves the application of a transient input to a system and the analysis of this input and the resulting transient response to determine the system's frequency response. The frequency response can then be displayed in any convenient form, such as a polar plot (Nyquist diagram) or a plot of the magnitude and phase angle versus frequency (Bode diagram).

The actual computation of the frequency response, $G(j\omega)$, proceeds directly from its definition as the Fourier Transform of the system's response divided by the Fourier Transform of the input.

$$G(j\omega) = \frac{F[x(t)]}{F[f(t)]} = \frac{\int_{-\infty}^{\infty} x(t)\, e^{-j\omega t}\, dt}{\int_{-\infty}^{\infty} f(t)\, e^{-j\omega t}\, dt} \qquad (1)$$

In practice, the pulse input and the system response are recorded simultaneously, their Fourier Transforms computed, the ratio taken, and the resulting frequency response data obtained. The equipment employed for the tests described in this paper captures the transients, performs the desired signal manipulations and displays the results in a matter of seconds. Since only one pulse is required to obtain the frequency response for most systems, this is the total effort and time required to obtain a complete frequency response plot.

COMPUTATION AND MEASUREMENT PROCEDURES

In the past the major limitation with Fourier Transform methods has been with data reduction. The problem has been two-fold. First, the Fourier Transforms are usually processed in digital form and adequate Analog-to-Digital Converters (ADC) had not existed. The second limitation was with the computational algorithm for computing the Fourier Transform. In recent years, both the hardware (ADC) and the computer algorithm, Fast Fourier Transforms[2,3], have been developed to the point where it is feasible to use Fourier Transforms and transient signals to measure the frequency response of a system. Small, portable, commercial systems are available which incorporate both a good ADC and a dedicated mini-computer using the improved Fourier Transform algorithm.

The University of Cincinnati has a Hewlett-Packard Fourier Analyzer which is capable of computing the frequency response from transient data, or, in fact, from any transformable input data which has frequency content in the range of interest. The system is shown in Figure 1 and consists of a two channel ADC, a small dedicated computer, a display unit with plotting and viewing capabilities plus digital input-output devices. Since a small general purpose computer is a part of the system, the data can be conveniently transformed from a Bode plot presentation to a Nyquist form, etc. The computer can also be used to pick important points off the measured frequency plot and display these points: for example, the mode shapes can be plotted in this manner.

Other limitations were with the pulser to produce a suitable force and the transducers to perform the measurements of the force and the motion. In the past, it was

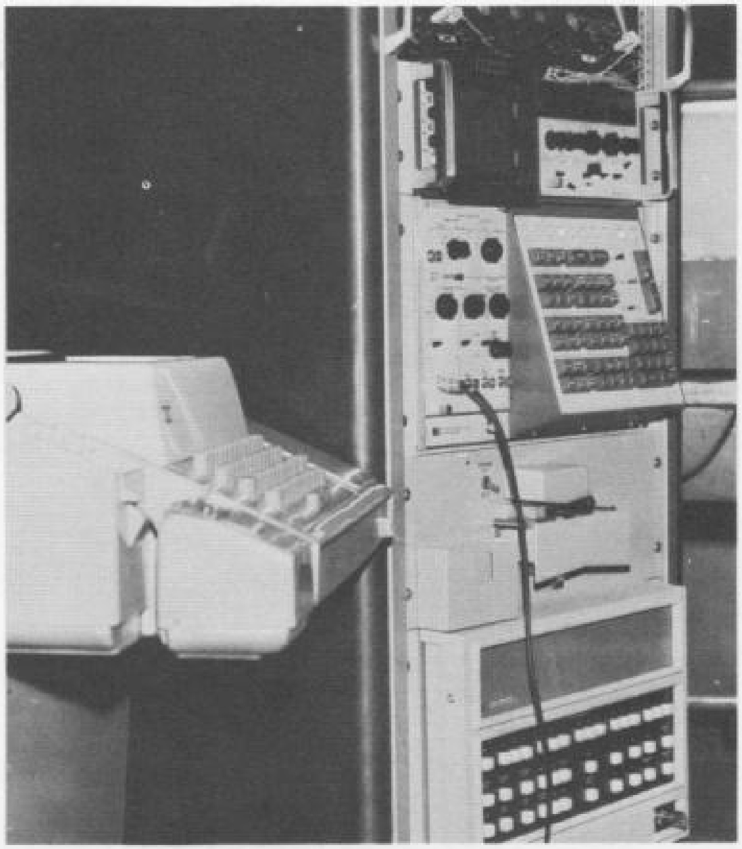

Figure 1. Fourier Analyzer for Pulse Test Data Manipulation

felt that a pulser should supply energy over a broad frequency range and also produce the proper input pulse shape. Since the input is Fourier Transformed, it is not necessary to carefully control the input waveform. Instead, the main consideration is to obtain the right frequency content of the input pulse. In this work the input pulse was produced by a small hammer striking a load cell. A hammer with interchangeable heads was used in order to obtain the proper frequency content. In Figure 2 the Fourier Transform for the input pulse produced by different hammer heads is shown. As can be seen, it is possible to obtain the desired frequency content in a force pulse produced by a hammer blow. Hence, it would be possible to develop a small portable pulser which can provide better control over the peak force value in the pulse in order to investigate non-linearities in the system.

In recent years, improvements have also been made in transducer technology. Small, convenient load cells, accelerometers, velocity pickups, and displacement pickups are available. Displacement pickups and servo-accelerometers can be used at very low frequencies and unlike some analog equipment the digital data analysis system can also be used at these low frequencies.

In summary, the limitations of the past have been removed by new technologies.

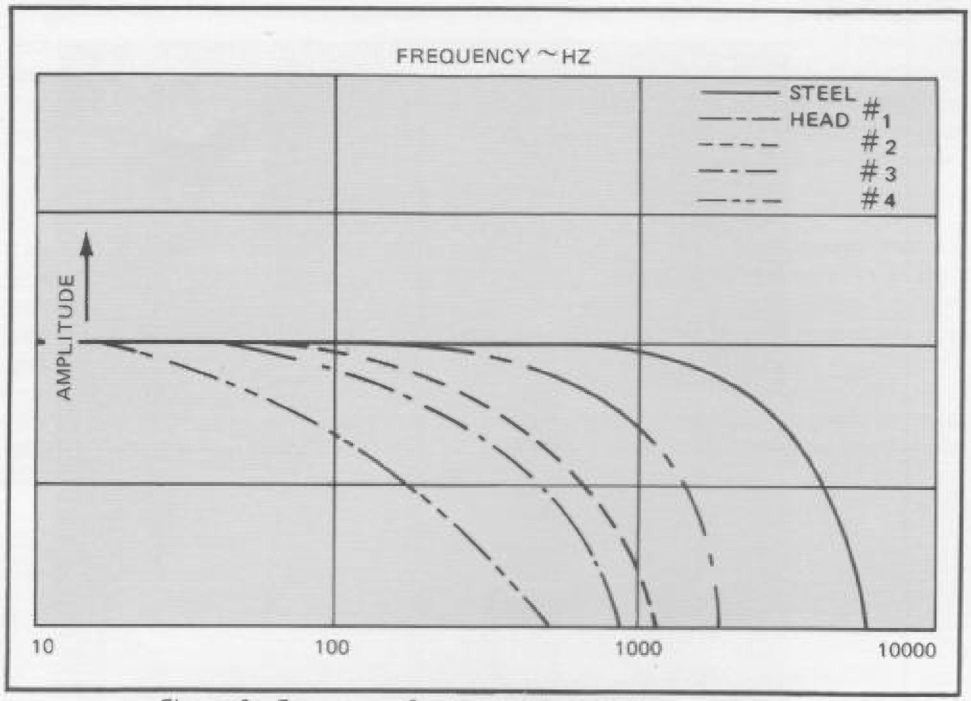

Figure 2. Frequency Content of Typical Hammer Impacts

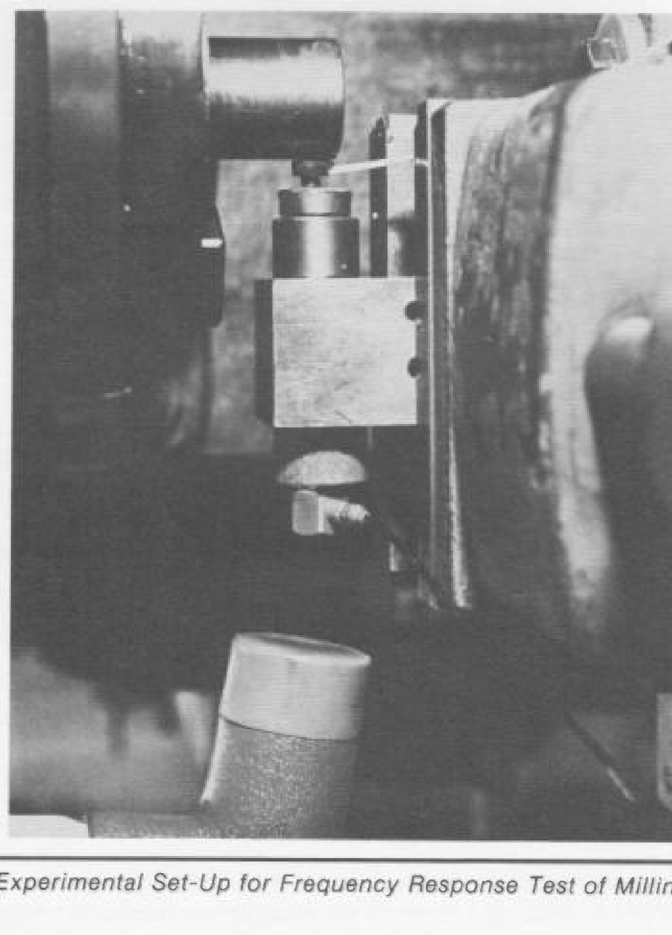

Figure 3. Experimental Set-Up for Frequency Response Test of Milling Machine

APPLICATIONS

The pulse test technique was applied to evaluate the effect of a dynamometer on the characteristics of a universal milling machine. Frequency response plots relating the motion along the three principal axes to forces applied along these axes were obtained for the machine with and without the dynamometer.

The input pulse was generated by striking a load cell mounted in a simulated work piece. The resulting relative deflection between the tool and the work piece was monitored by a displacement transducer mounted in the tool holder. The transducer locations for monitoring the displacement in the transverse direction due to a force in the transverse direction are shown in Figure 3 and the frequency response plots are shown in Figure 4. For comparison, plots with and without the dynamometer are presented. As can be seen, the static stiffness of the milling machine in the transverse direction without the dynamometer is about 8 times greater than with it and the dynamic stiffness in the transverse direction is about 30 times greater without the dynamometer.

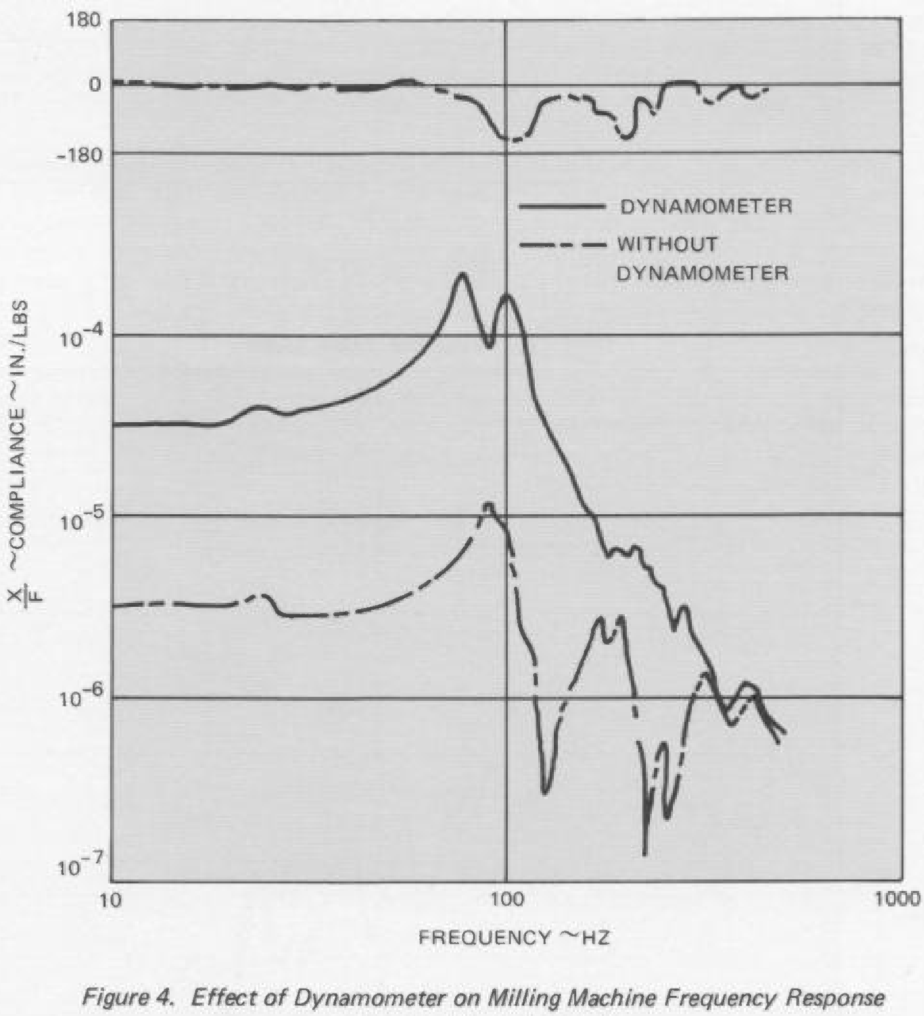

Figure 4. Effect of Dynamometer on Milling Machine Frequency Response as Obtained by Pulse Testing

An important observation concerning the pulse test method in this type of application is its relative simplicity. The experimental set-up required no special fixtures. In practice, the entire program from initiation of the concept to completed evaluation of all three major axes can easily be performed by one person in less than a day.

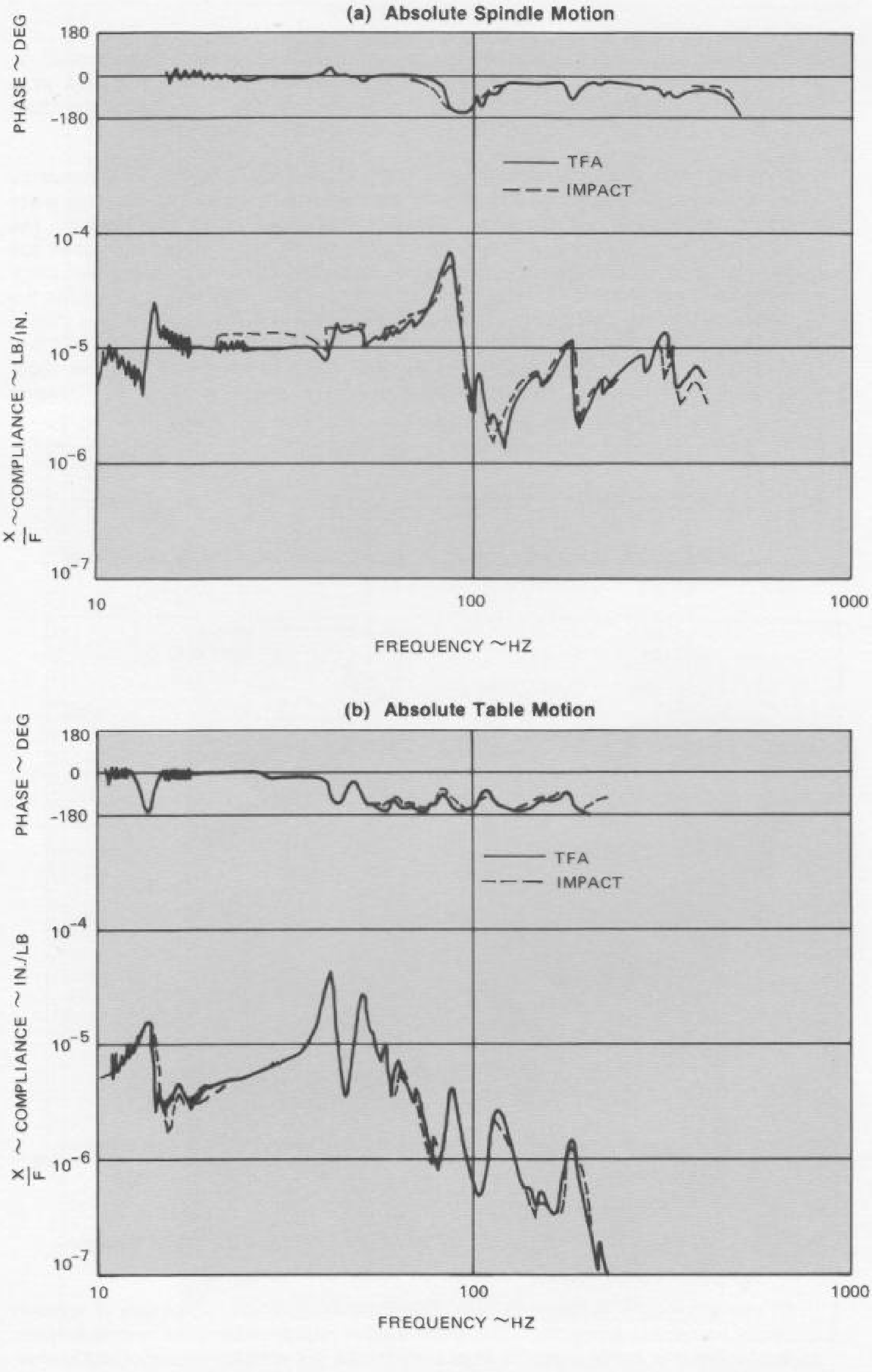

Figure 5. Comparison of Frequency Response From Pulse and Sinusoidal Tests

The pulse test approach was also used to determine the frequency response and mode shape data for a small vertical milling machine. In this case, both the frequency response and mode shapes were also determined using traditional sinusoidal excitation. For comparison, driving point frequency response plots in the feed direction obtained using both methods are shown in Figure 5. The table motion and the spindle motion are shown in Figures 5a and 5b respectively. These curves illustrate the high degree of correlation obtained by the pulse test with respect to sinusoidal excitation tests.

Mode shape data can also be obtained using pulse tests. In the present case, the mode data were obtained by applying the pulse input to those selected points on the structure where the modal displacements were to be measured. The transient response in each case was measured at a common point. In effect, a set of cross frequency response plots was obtained relating motion at the common point to force inputs at various selected input points. At a natural frequency, the amplitude of the response at the common point due to a force at a selected input point will be equal to the response of the selected point due to a force applied at the common point. Therefore, the relative amplitudes of those points selected for the mode shape plot are obtained directly from the set of frequency response plots. Further, it is clear that once the required set of frequency response plots is obtained, the data are available for all of the structures modes. In other words, using the pulse test techniques data for all the modes can be obtained simultaneously.

It should also be evident that the mode data could be obtained by impacting at a common point and measuring the response at various selected points on the structure in the same manner. In either case, it is not actually required to record the entire frequency response for each point, but only the response at the natural frequency for which the mode shape is desired. In practice, these frequencies are pre-selected and the computer programmed to identify and record only the desired data.

Mode shapes obtained from pulse tests for the small vertical milling machine are shown in Figure 6a. Mode shape data for only one mode are shown but data for modes at twelve frequencies were recorded simultaneously. The mode shape obtained using sinusoidal excitation is also shown in Figure 6b for comparison and illustration of the close agreement between the two methods.

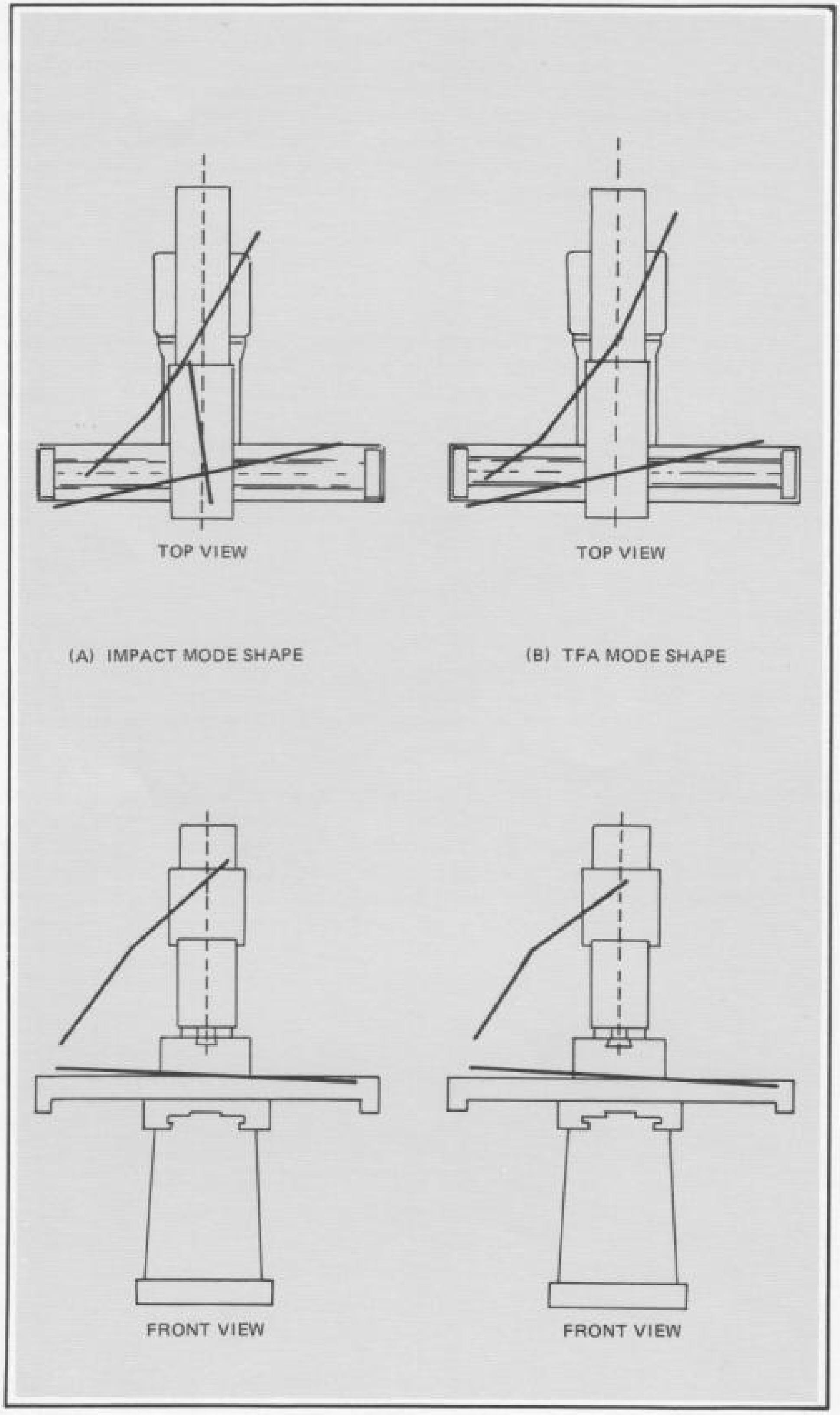

Figure 6. Milling Machine Mode Shapes Obtained by Pulse and Sinusoidal Tests

A third application involved determining the frequency response and mode shapes for a large roll grinder. The relative motion between the wheel and workpiece due to a force acting between them is shown in Figure 7. In this case, the relative motion was obtained by impacting the wheel and the workpiece separately and obtaining their absolute motions. The relative motion was then obtained as the difference between their absolute motions. The frequency response obtained using sinusoidal excitation is also shown. A comparison of the frequency responses obtained by the two methods indicates good correlation.

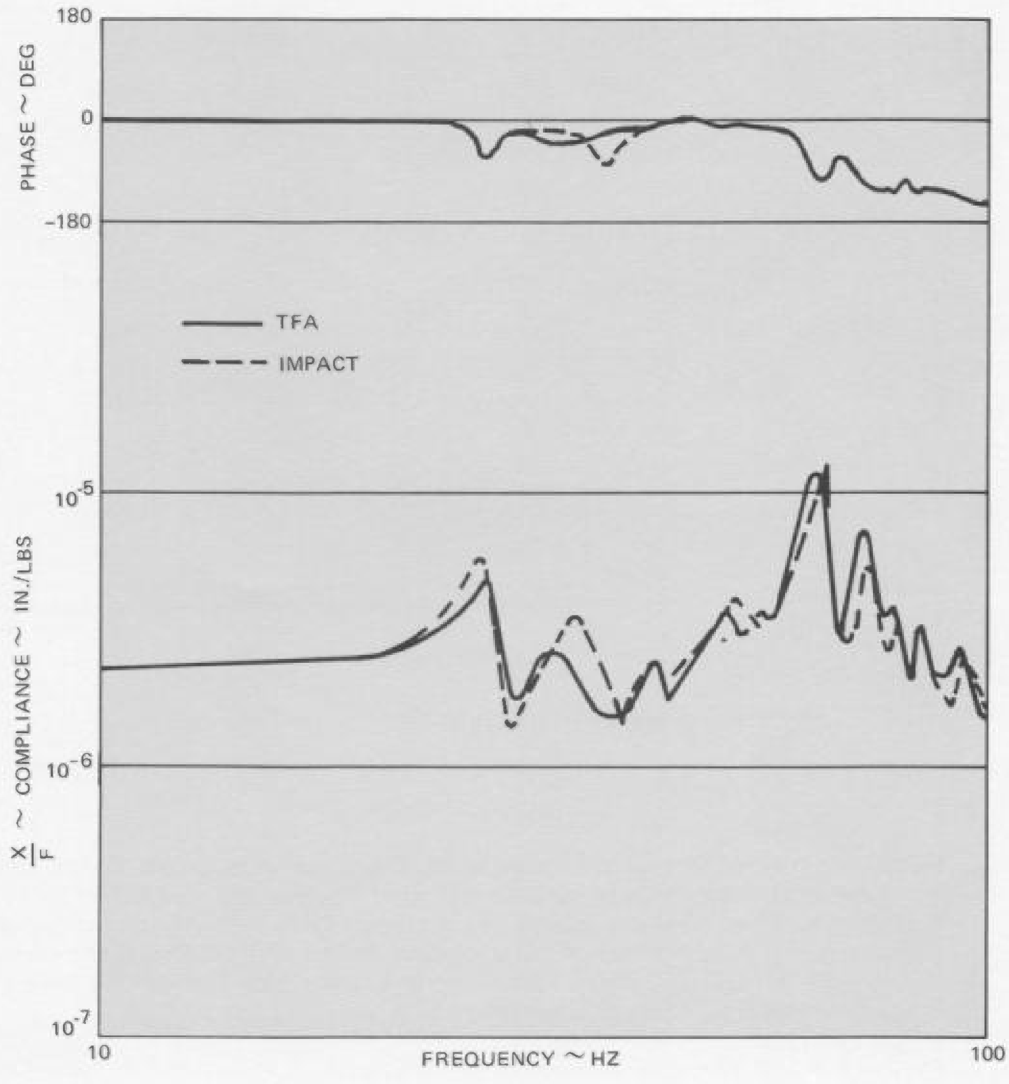

Figure 7. Comparison of Grinder Frequency Response Plots From Pulse and Sinusoidal Tests

The last application selected for discussion involves the response of a machine tool isolation system. The frequency response obtained from the pulse test is shown in Figure 8 and illustrates the ability of the pulse test to define low frequency characteristics.

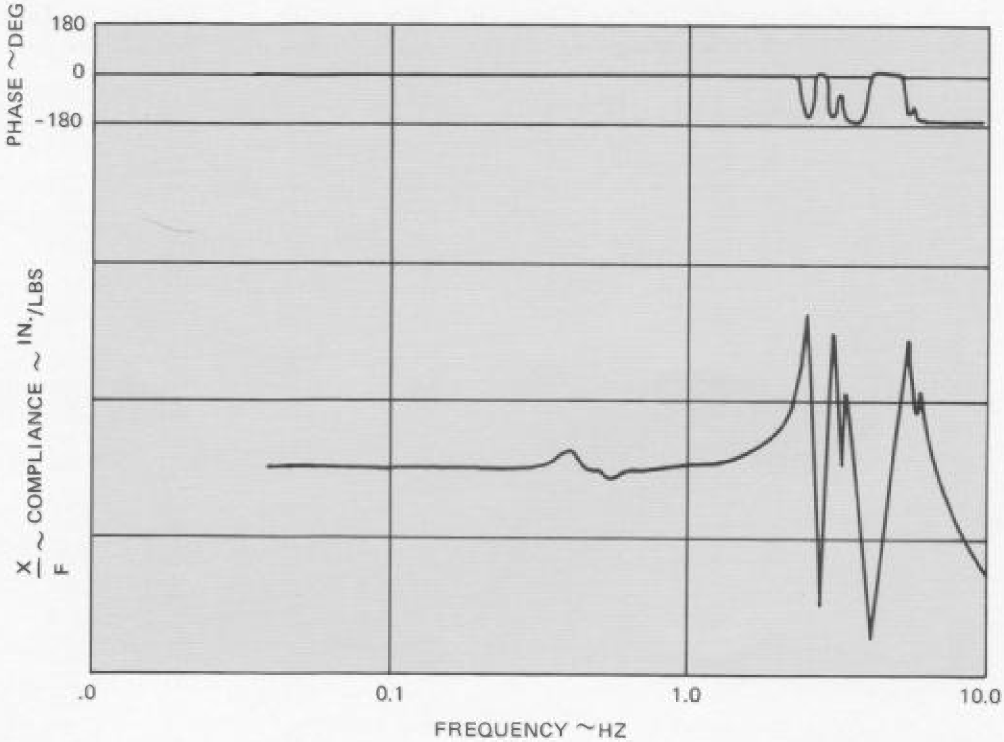

Figure 8. Frequency Response of Machine Tool Isolation Block

CONCLUSIONS

Pulse testing is a practical alternative to traditional methods for the evaluation of the dynamic characteristics of systems typical of machine tool structures. Indeed, pulse testing offers many advantages over existing procedures. Among the primary advantages are the simplicity of the excitation system and the virtual elimination of the need for special fixtures, and an order of magnitude reduction in the time required to obtain a frequency response plot.

By using signal averaging or correlation techniques, it is possible to obtain the response of the system under noisy operating conditions. Since the pulser does not have to be fixed to the machine, it should be possible to obtain the frequency response of operating components (for example, a rotating spindle).

It should be noted that the techniques described in this paper are not limited to only pulse type inputs but that they can be applied to linear systems with any measurable, Fourier transformable input which has adequate frequency content in the range of interest.

To summarize the results briefly, it appears that pulse testing of machine tools or any structure is a currently practical technique. It also appears particularly attractive for those problems where time and fixturing expenses are important.

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UTILIZING THE HP FOURIER ANALYZER FOR IMPULSE TESTING

I. INTRODUCTION

The availability of an easy-to-use digital Fourier Analyzer makes the impulse method of testing mechanical structures a practical reality. Using the Fourier Analyzer quantitative data derived from an input force and system response yield the dynamic properties of a structure quickly and easily. Involved experimental set-ups are greatly simplified or eliminated. Structures previously difficult or impossible to characterize due to size, location or operating environment (i.e., rotating machinery) easily lend themselves to the impulse method. This note describes how the Fourier Analyzer is used to measure a structural model using this method.

II. SET-UP

A typical set-up for impulse testing a mechanical system is shown in Figure 1. A force applied to point x is measured with a load cell and signal amplifier and sent into channel A of the Fourier Analyzer. The response (acceleration, velocity, or displacement) is measured at point y, amplified, and sent to channel B. A suitable hammer is used to strike the load cell and apply an impulse to the system.

Since the impulse is being used to excite the system over a broad range of frequencies, it is important to observe the frequency response of the force input before proceeding with the test. The frequency content of the impulse should go out to the frequency of interest, but not beyond this point to avoid aliasing problems. At the same time it should have a low enough frequency input to excite the system sufficiently at low frequencies. A softer or harder hammer head may be indicated by the observed frequency content of the impulse.

Optionally, a step input may be applied in place of the impulse by preloading the desired point on the mechanical system to some preset load or displacement and subsequently releasing the load rapidly in order to induce system vibration. This technique may be preferred on a delicate mechanical structure where even very small impulses are too severe. For instance, exciting a small scale model with a hammer may induce a nonlinear response not characteristic of its typical performance. On the other hand, a step input is necessary when an impulse will not cause sufficient disturbance in a very large mechanical structure. Here, preloading through a cable may be easily realized.

III. COMPUTING THE TRANSFER FUNCTION (IMPEDANCE)

The data may be processed several ways, depending upon the amount of background noise, e.g., running motors or fans. If the system is quiet, except for the impact, then the straight ratio of the Fourier transforms of the two signals may be used to obtain the system response.

That is,

$$H(j\omega) = \frac{S_y(j\omega)}{S_x(j\omega)}$$

where $S_x(j\omega) = F[x(t)]$ and $S_y(j\omega) = F[y(t)]$

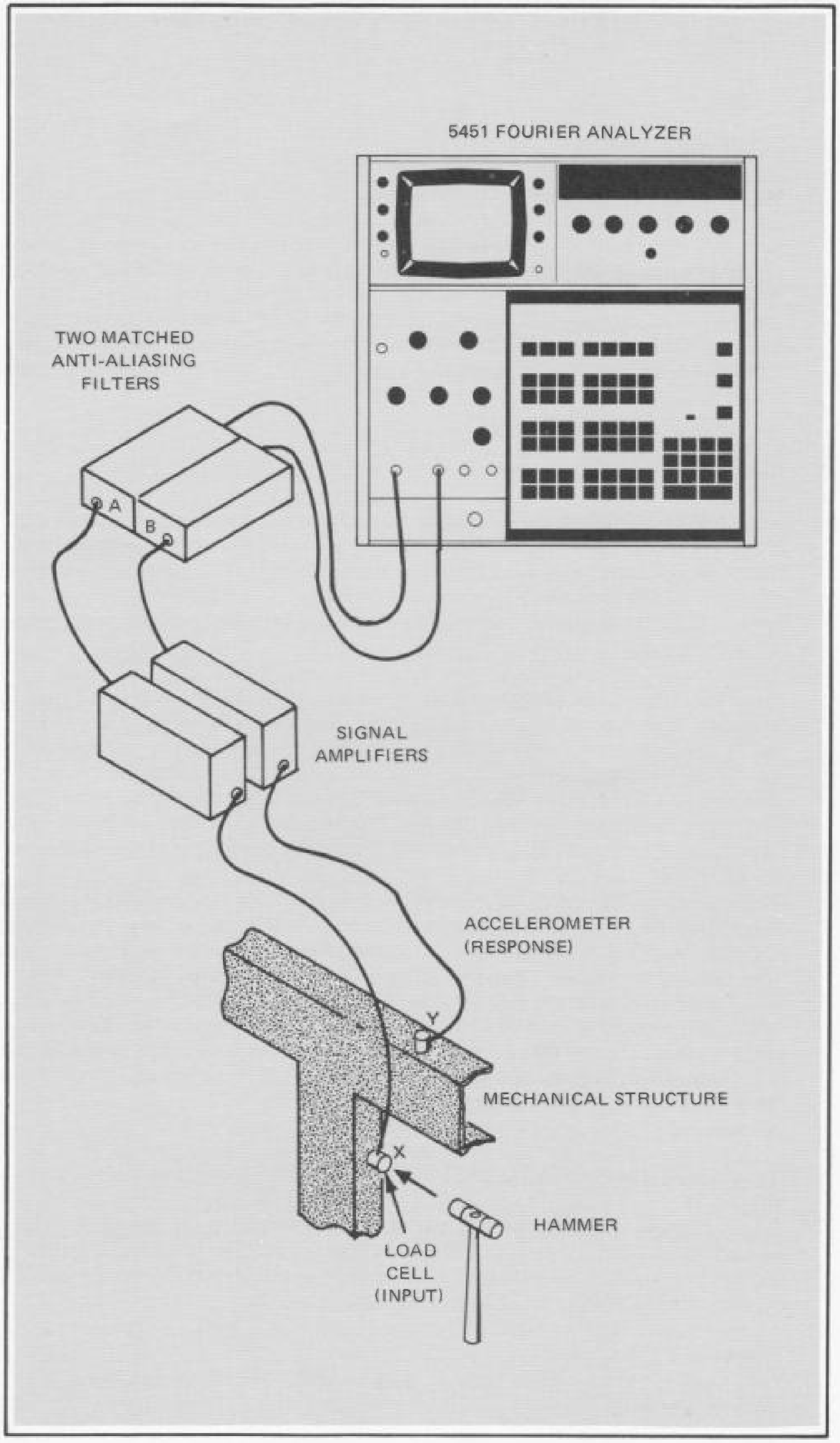

Figure 1. Typical Impulse Test

Where there is considerable background noise, then statistical methods may be employed and the system response can be calculated as:

$$H(j\omega) = \frac{\overline{G_{yx}}(j\omega)}{\overline{G_{xx}}(j\omega)}$$

where $\overline{G_{yx}} = \overline{S_y S_x^*}$ and $\overline{G_{xx}} = \overline{S_x S_x^*}$

With averaging, the signal to noise ratio will improve as the square root of the number of averages.[1]

An important measure of the quality of the measurement is the coherence function defined as:

$$\gamma^2 = \frac{|\overline{G_{yx}}|^2}{|\overline{G_{xx}}|\,|\overline{G_{yy}}|} \qquad 0 \leq \gamma^2 \leq 1$$

when $\overline{G_{yy}} = \overline{S_y S_y^*}$

This tells the percentage of energy in the response that was due to the input. At points where this value is 1 or nearly 1, good data can be assumed. But where the coherence is low, there is a direct indication of a poor measurement. This very powerful "self-checking" feature can be calculated at the same time as the system response.[2] A sample program, in Appendix I, shows how this may be done.

IV. MODE SHAPE PLOTTING

A convenient transfer function to characterize a mechanical structure is the compliance X/F. When the force is applied at any point on the structure, the resulting displacement may be measured at that point or any other point. The mode shape may be determined by measuring the displacement at several points along the structure caused by the force input at one reference point.

The compliance transfer function $H(j\omega)$ is a complex quantity having real and imaginary parts. For lightly damped systems, the imaginary (quadrature) part at a resonance equals the compliance for that mode of vibration. The real (coincident) part is zero at each resonance. Because the structure is excited with a broadband input (impulse), the mode shape information is conveniently obtained for all modes simultaneously at each point, making this technique extremely fast. A keyboard routine may be set up on the Fourier Analyzer to print out the real and imaginary values of the compliance around each resonance. Then, for each resonant frequency, the imaginary values corresponding to points along the structure are used to plot the structure's vibration mode shape.

Making use of Maxwell's reciprocal relations, the same result may be obtained by moving the impact point along the surface and measuring the response at a single point. This latter technique may be preferred in some cases.

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[1]See reference 1.

[2]See references 2 and 3.

V. TRIGGERING

The acquisition of the data ideally commences as the force impulse becomes infinitesimally greater than zero. By triggering the ADC on the force signal a portion of this signal may be lost due to the internal triggering level set by the user. This problem is not serious and good results are obtained in spite of its presence. Optionally though, data acquisition may be started manually prior to the impulse on an asynchronous basis at the expense of losing sampling time after the impulse.

VI. WINDOWING

A rectangular window is sufficient in many damped systems, but may cause leakage[1] problems in the response transform when a very lightly damped system is analyzed. This happens when the response oscillations have not decayed to zero before the sampling interval ends. The best method is to extend the sampling time sufficiently to enable the response to reach a zero value. The following example measurement uses a rectangular sampling window and a five-second total sampling interval which is adequate for the system measured.

VII. EXAMPLE MEASUREMENT

Figure 2 shows an arbitrary mechanical system constructed for laboratory demonstration purposes. A force transducer and accelerometer are used with

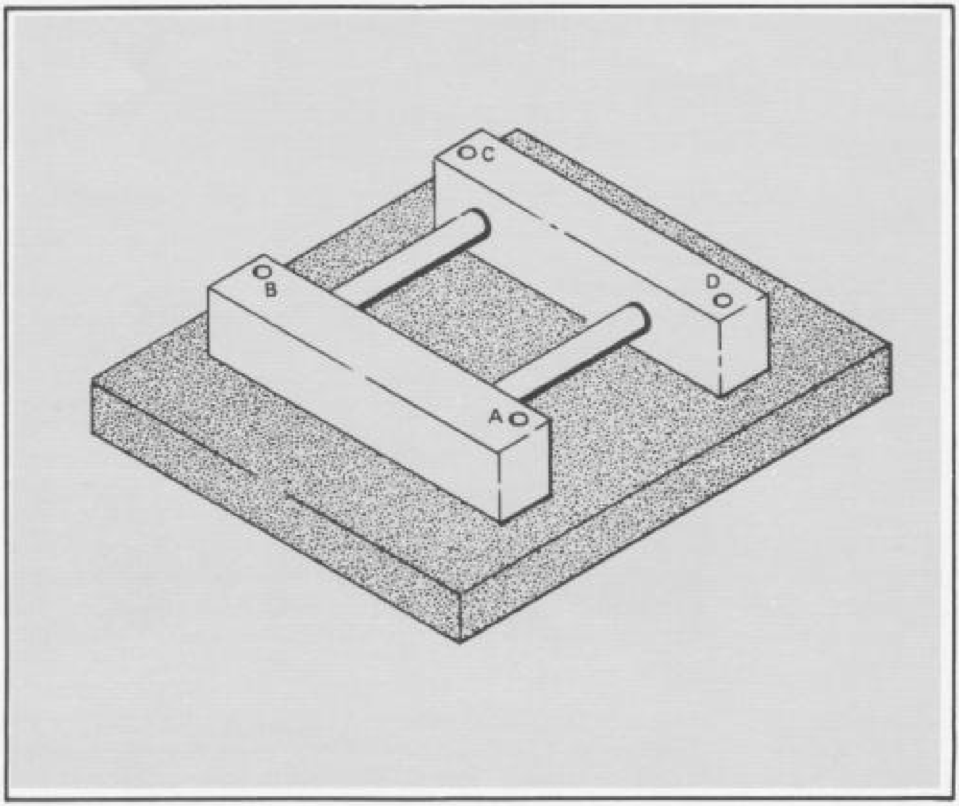


Figure 2. Laboratory Demonstration Mechanical System

[1]See Fourier Analyzer Training Manual AN140-0.

two signal amplifiers to obtain the required signals. The transducers are set up and checked for proper operation, followed by some initial measurements on the specimen to get a feel for the response that is to be expected. This involves making some hammer blows, looking at their Fourier transforms to see what range of excitation they produce and choosing a hammer head for proper frequency content.

Figure 3 shows the time record of the impulse and accelerometer response.

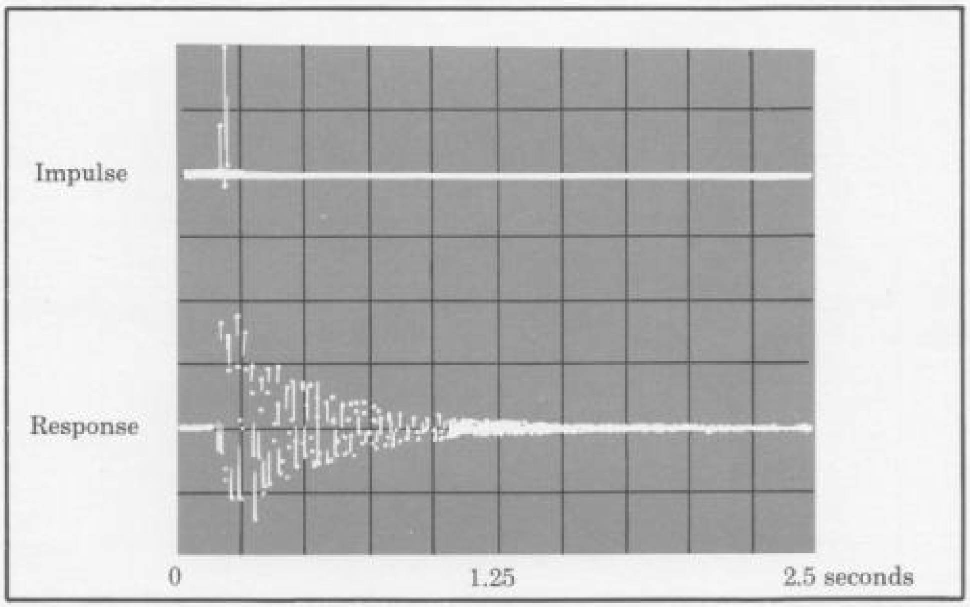


Figure 3. Time Waveforms of the Impulse and System Response

The log magnitudes of their Fourier transforms are shown in Figure 4.

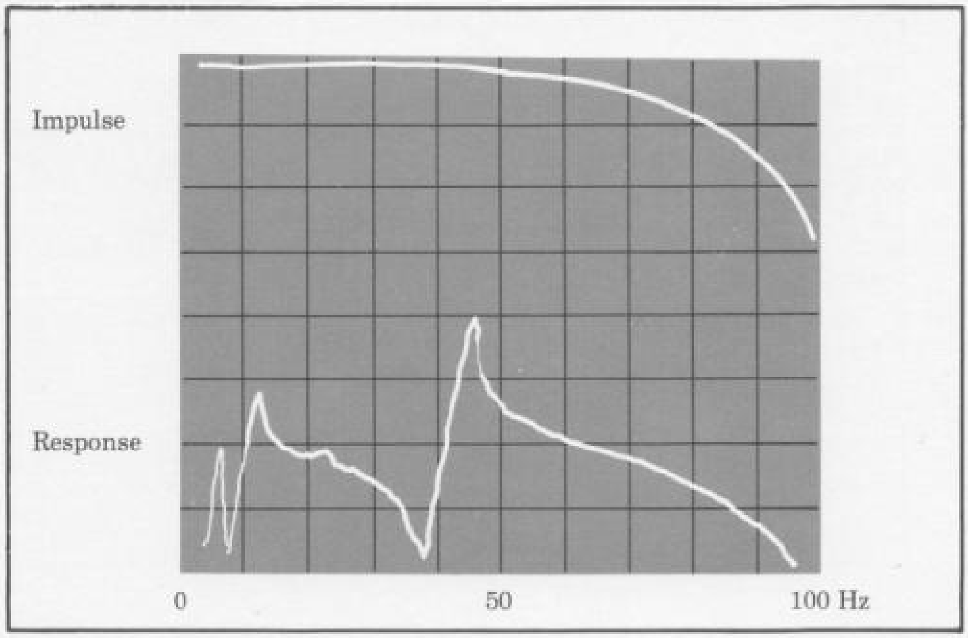


Figure 4. Frequency plot of the Impulse (top) and Response (bottom)

Note that the impulse provides excitation over the frequency range of the three predominant modes in the acceleration trace.

Some initial transfer function (impedance) calculations are then made and from these a number of resonances identified (Figure 5). The top trace is the coincident (real) part and the bottom trace is the quadrature (imaginary) part.

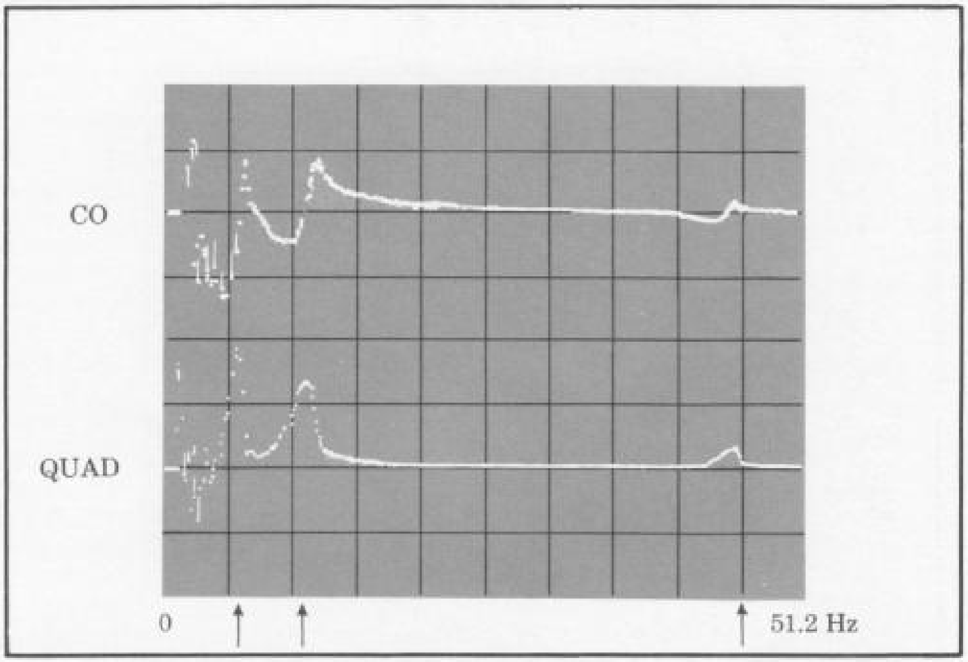


Figure 5. Co-Quad Plot of the Transfer Function X/F at point A.

Of most interest were resonant modes at 6.0, 11.8, and 46.2 Hz, as identified by their channel numbers. A keyboard program was then written to calculate the transfer function and print out the values around each desired resonance (Appendix II). The program was written in such a way that following the impact, the impact itself is observed on the screen followed by the response of the system. If these look proper, the program is continued and the calculation and print out proceeds automatically. Figure 6 shows expanded displays of the real and imaginary parts of the transfer function around the 46.2 Hz mode. This information at points A, B, C, and D on the mechanical specimen allow a mode shape plot to be made as illustrated in Figure 7. Similarly, modes at 6.0 Hz and 11.8 Hz are plotted as shown in Figures 8 and 9 respectively. Measurements at other points on the specimen made no significant contribution to the mode shape plot.

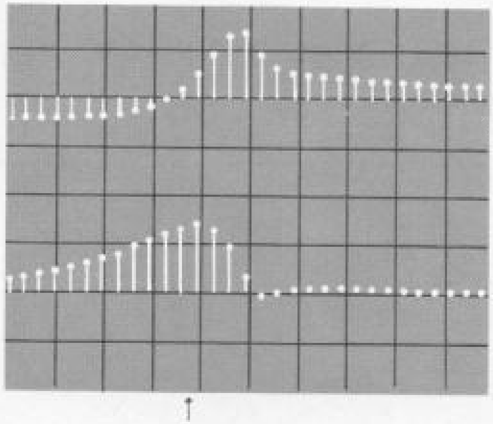
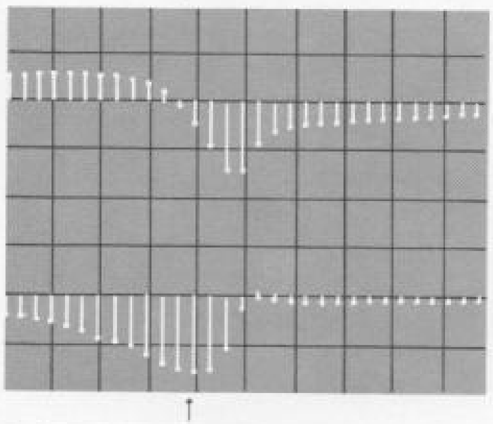
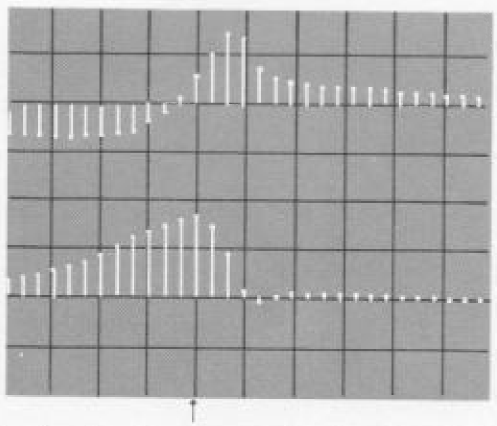
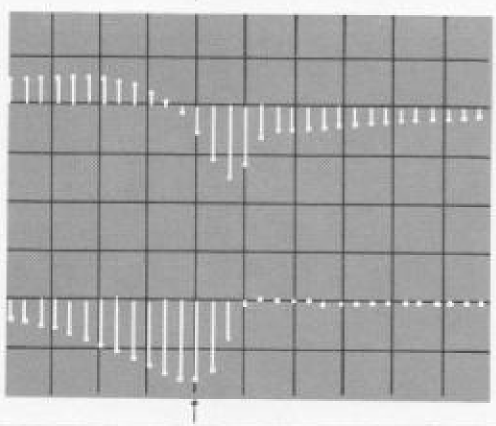
DISPLAYS	PRINTOUTS			
	SF (	-5 231)	4 7Ø	55 3Ø19
Point A (6a)				
	SF (	-5 231)	4 2Ø	55 -33ØØ
Point B (6b)				
	SF (	-5 231)	4 1Ø	55 3239
Point C (6c)				
	SF (	-5 231)	4 -3Ø	55 -321Ø
Point D (6d)				

Figure 6. Co-Quad Plots of the 46.2 Hz Mode of the Transfer Function X/F

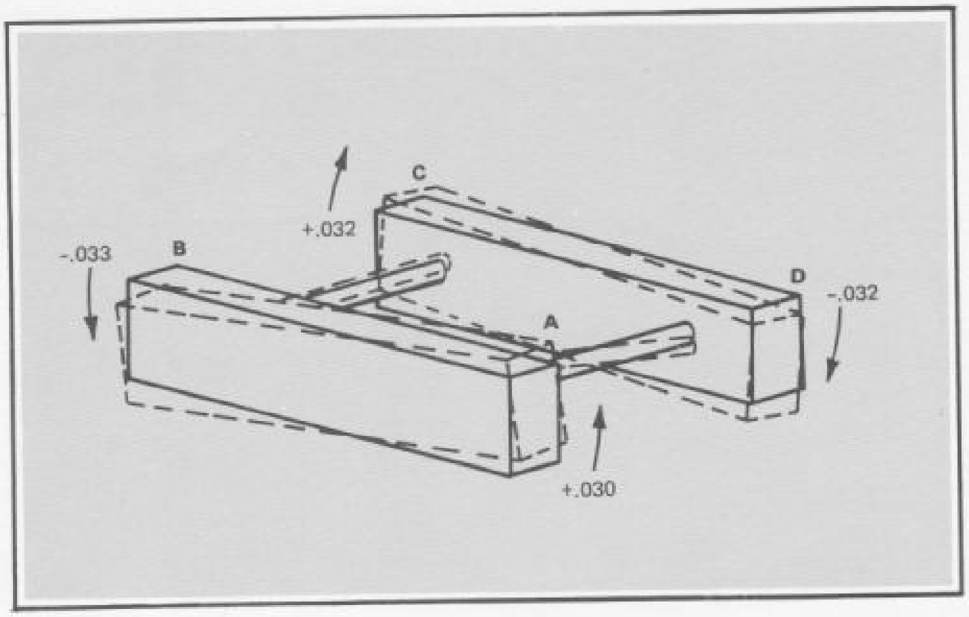


Figure 7. 46.2 Hz Twisting Mode (Values are X/F Inches Per Pound)

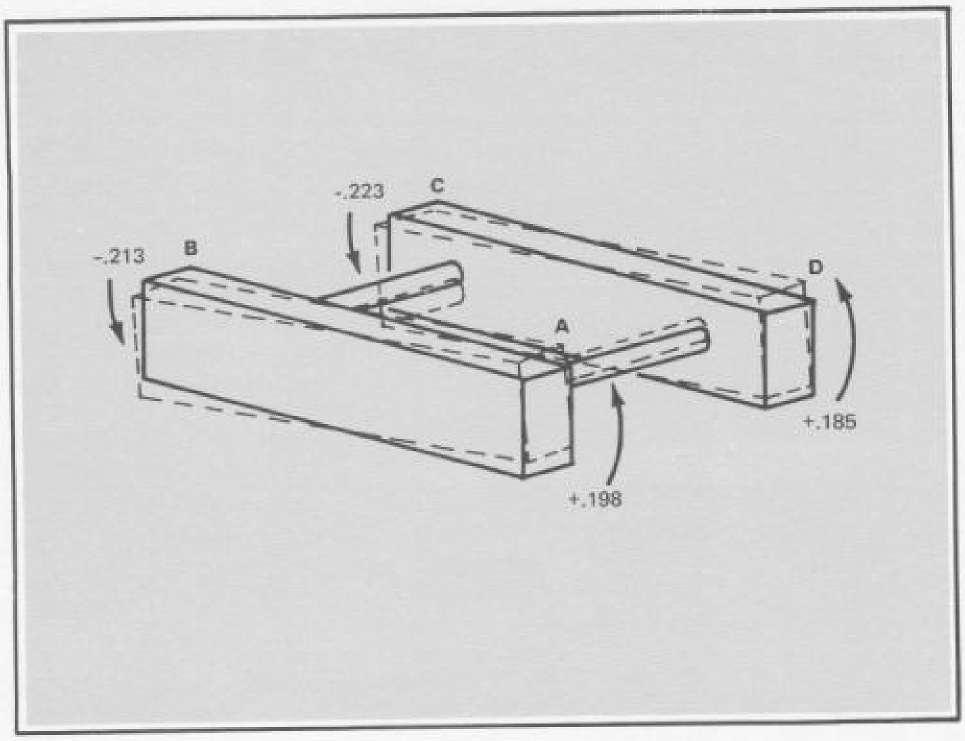


Figure 8. 6.0 Hz Rocking Mode

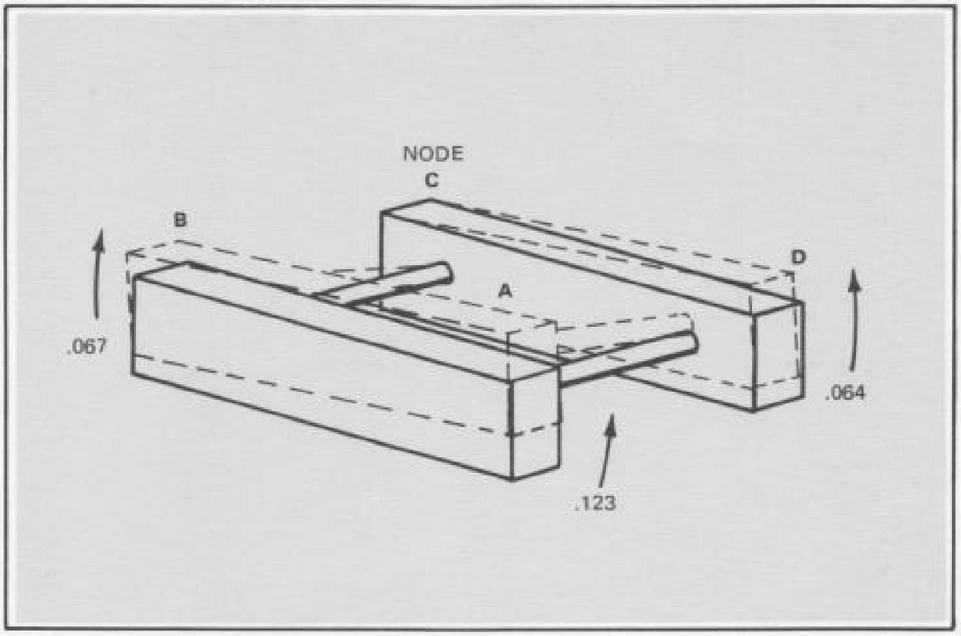


Figure 9. 11.8 Hz Tilting Mode

VIII. CONCLUSION

This technique is extremely fast and can result in a time savings of 100 to 1 or more over conventional sine sweep tests. Also, sine sweep shaker tests do not lend themselves to the application of rotating or reciprocating machinery, as does the impulse technique.

Impulse testing should find wide application in structural dynamic testing including machine tools, automotive products, aircraft and missiles. This technique has even been applied to buildings to determine the transfer function between forces from the equipment to be installed on the roof to other points in the building.

Mike Howell
Santa Clara Division
August, 1972

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APPENDIX I

Sample Program Utilizing Averaging Techniques

This program is configured for the force signal at input A and the response signal at input B. Internal triggering is required on the force signal to begin data acquisition.

1 L	Ø		
4 CL	3		
7 CL	4		
1Ø CL	5		
13 L	1		
16 RA	1	1	
2Ø D	1		
23 D	2		
26 F	1	2	
3Ø CL	1	Ø	
34 CL	2	Ø	
38 SP	1	2	
42 #	1	5	Ø
47 CH	1	2	
51 •			

PROGRAM COMMANDS	CONTENTS BLOCK 0	CONTENTS BLOCK 1	CONTENTS BLOCK 2	CONTENTS BLOCK 3	CONTENTS BLOCK 4	CONTENTS BLOCK 5	PURPOSE OF COMMAND
LABEL 0							ESTABLISHES INITIAL LABEL POINT
CLEAR 3				CLEARED			INITIALIZE STORAGE
CLEAR 4				〃	CLEARED		〃
CLEAR 5				〃	〃	CLEARED	〃
LABEL 1				〃	〃	〃	ESTABLISH LOOP LABEL
ANALOG IN 1 SPACE 1		CURRENT FORCE TIME RECORD	CURRENT RESPONSE TIME RECORD	〃	〃	〃	ACQUIRE DATA
DISPLAY 1		〃	〃	〃	〃	〃	
DISPLAY 2		〃	〃	〃	〃	〃	
FOURIER 1 SPACE 2		SPECTRUM OF FORCE	SPECTRUM OF RESPONSE	〃	〃	〃	CONVERT TO FREQUENCY DOMAIN
CLEAR 1 SPACE 0		FORCE SPECTRUM MINUS DC COMPONENT	〃	〃	〃	〃	CLEAR DC TERM
CLEAR 2 SPACE 0		〃	RESPONSE SPECTRUM MINUS DC COMPONENT	〃	〃	〃	〃
PWR SPEC 1 SPACE 2		〃	〃	AUTO PWR SPEC. FORCE	AUTO PWR SPEC. RESPONSE	CROSS PWR SPECTRUM	COMPUTE PWR SPECTRAL QUANTITIES
COUNT 1 SPACE 5		〃	〃	SUM FORCE PWR SPEC	SUM RESPONSE PWR SPEC	SUM CROSS PWR PWR SPEC	LOOP BACK TO LABEL 1 5 TIMES
TRANS FUNC 1 SPACE 2		TRANS FUNC RESPONSE/ FORCE	COHERENCE	〃	〃	〃	COMPUTE COMPLIANCE AND COHERENCE
END		〃	〃	〃	〃	〃	

APPENDIX II

Sample Program From Text Example

This program is configured for the accelerometer signal at input A and the force signal at input B. Data acquisition begins when the "CONTINUE" button is pushed on the Fourier Analyzer, at which time an impact on the mechanical specimen follows (see Figure 3 in the text).

```
 1 L     Ø
 4 RA    Ø        Ø
 8 D
1Ø D     1
13 F     Ø        1
17 CL    Ø        Ø
21 CL    1        Ø
25 :     2
28 :     1
31 *     Ø    19558
35 :     Ø     1ØØØ
39 CL    Ø        Ø     2Ø
44 W     Ø       26     33
49 W     Ø       55     62
54 W     Ø      23Ø    237
59 D     Ø       2Ø     7Ø
64 D     Ø      22Ø    25Ø
69 J     Ø
72 •
```

PROGRAM COMMANDS	CONTENTS BLOCK 0	CONTENTS BLOCK 1	CONTENTS BLOCK 2	PURPOSE OF COMMAND
LABEL 0				ESTABLISHES INITIAL LABEL POINT
ANAL IN 0 SPACE 0	CURRENT RESPONSE TIME RECORD	CURRENT FORCE TIME RECORD	f^2	ACQUIRE DATA
DISPLAY 0	"	"	"	
DISPLAY 1	"	"	"	
FOURIER 0 SPACE 1	SPECTRUM OF RESPONSE	SPECTRUM OF FORCE	"	CONVERT DATA TO FREQUENCY DOMAIN
CLEAR 0 SPACE 0	RESPONSE SPECTRUM LESS DC	"	"	CLEAR THE DC COMPONENT
CLEAR 1 SPACE 0	"	FORCE SPECTRUM LESS DC	"	"
DIVIDE 2	SPECTRUM OF DISPLACE-MENT	"	"	OBTAIN DISPLACEMENT FROM ACCELERATION SPECTRUM (SEE APPENDIX III)
DIVIDE 1	TRANSFER FUNCTION X/F	"	"	CALCULATE THE COMPLIANCE
MULT 0 SPACE 19558	"	"	"	MULTIPLY DATA BY SCALE FACTOR
DIVIDE 0 SPACE 1000	"	"	"	"
CLEAR 0 SPACE 0 SPACE 20	TRANS FUNC LESS UNDESIRED MODE DATA	"	"	SCALE UP DESIRED MODES
PRINT 0 SPACE 26 SPACE 33	"	"	"	PRINT OUT THE DESIRED INFORMATION
PRINT 0 SPACE 55 SPACE 62	"	"	"	"
PRINT 0 SPACE 230 SPACE 237	"	"	"	"
DISPLAY 0 SPACE 20 SPACE 70	"	"	"	EXPANDED DISPLAY OF MODES 1 AND 2
DISPLAY 0 SPACE 220 SPACE 250	"	"	"	EXPANDED DISPLAY OF MODE 3
JUMP 0	"	"	"	REPEAT PROGRAM
END	"	"	"	

APPENDIX III

The following data sampling parameters were used in the text example:

DC-coupled ADC
data block size = 1024
frequency resolution $\Delta f = 0.2$ Hz
maximum frequency $F_{max} = 102.4$ Hz
T sampling time = 5 sec
24 db/octave antialiasing filter fc = 75 Hz (48 db/octave preferred)

The incoming signals were calibrated as follows:

force F = 10 lbf/volt

acceleration A = 20g/volt

It is desired to obtain the transfer function X/F in inches per pound from A/F. Because

$$|X| = \frac{A}{\omega^2} = \frac{A}{4\pi^2 f^2}$$

and

$$1\ g = 386.06\ \text{in/sec}^2,$$

X/F may be obtained by dividing A/F by f^2 and applying the conversion factor as follows:

$$\frac{X}{F} = \frac{A}{4\pi^2 f^2 F} = \frac{(A\ \text{g/volt})\ \frac{386.06\ \text{in/sec}^2}{g}}{39.478\ f^2 \text{Hz}^2\ (F\ \text{lbf/volt})}$$

$$= 9.779\ \frac{A\ (\text{g/volt})}{F\ (\text{lbf/volt})}\ \frac{\text{inches}}{\text{lbf}}$$

For the calibrations used in the text example:

$$\frac{X}{F} = 9.779\ \frac{20\ (\text{g/volt})}{10\ (\text{lbf/volt})}\ \frac{\text{inches}}{\text{lbf}}$$

$$= 19.558\ \frac{\text{inches}}{\text{lbf}} \times \text{(observed reading)}$$

The value of f^2 may be calculated easily and stored in a data block (block 2) for ready division into the A/F transfer function. The following keyboard steps will produce f^2.

```
CL

K Ø 1 1Ø23
K -4 4
2ØØØ

$

*_

X > 2
```

(Note that the value entered is equal to the resolution Δf.)

PROGRAM COMMANDS	CONTENTS BLOCK 0	CONTENTS BLOCK 1	CONTENTS BLOCK 2	PURPOSE OF COMMAND
CLEAR	CLEARED			
KEYBOARD 0 SPACE 1 SPACE 1023	"			BLOCK FILL CHANNELS 1 THROUGH 1023
KEYBOARD -4 SPACE 4	"			10^{-4} SCALE FACTOR IN FREQ DOMAIN
2000	0.2 IN CHANNEL 1 THROUGH 1023			2000×10^{-4}
INTEGRATE	f			CALCULATE f FOR EACH CHANNEL
CONJUGATE MULTIPLY	f^2			CALCULATE f^2 FOR EACH CHANNEL
STORE 2	f^2		f^2	STORE FOR FUTURE DIVIDE

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