Field Instrumentation and Diagnostics of Pump Vibration Problems

By

J. C. Wachel
J. D. Tison
K. E. Atkins

Engineering Dynamics, Inc.
16117 University Oak
San Antonio, Texas 78249

Presented at

Rotating Machinery and Controls (ROMAC)
Short Course

Vibration Problems in Pumps

University of Virginia
Charlottesville, Virginia

June 6-7, 1983

Field Instrumentation and Diagnostics Of Pump Vibration Problems

By
J.C. Wachel
J. D. Tison
K. E. Atkins

I. Introduction

There have been tremendous improvements in the instrumentation and diagnostic procedures used to analyze pump vibration problems in the last few years. The use of FFT analyzers, multi-channel FM tape recorders and diagnostic software packages in conjunction with microcomputers has vastly improved the engineer's ability to understand the types and causes of vibration problems. There are numerous instruments that can generate waterfall displays, order tracking, Nyquist diagrams, etc. Most vibration engineers find that these tools become a mandatory part of the diagnostic system once they have been exposed to them.

The authors have found that these tools are so valuable that they need to be available at the job-site during the testing. This presents a problem since many of the systems would have a difficult time surviving the severe treatment that is usually encountered in shipping electronic equipment to the field.

Another important consideration regarding field diagnostics systems is the cost of the instrumentation. Most consulting firms charge a daily fee for the use of the equipment in the field. The costs per day typically run from 0.5 percent to 2.0 percent of the replacement cost. If all the testing went according to schedule, this would not normally be a problem; however, Murphy's law says that anything that can go wrong, will, so the typical study ends up with many delays.

Faced with these constraints, many engineers tape record the data and ship the tape back to the diagnostic laboratory systems for the detailed analysis. This can sometimes result in several days of delay in obtaining the detailed analyses and eliminates the ability of the test engineer to interactively modify the test plan.

To solve the problems of durability and costs, our company has developed software to perform vibration diagnostics using a low cost Apple microcomputer. This allows the engineer to perform detailed diagnostic analyses at the job site at a minimum cost to the client since the equipment costs for the microcomputer are less than $5000. In addition, the engineer can make on-the-spot analyses and determine what tests need to be made to more quickly develop a solution.

This paper describes some of the capabilities of the data acquisition system and shows how it has been used in the field to analyze some pump vibration problems. Other field problems, previously studied, are included to show, in general, the diagnostic procedures that have been used to define the causes of pump vibration.

II. Field Equipment and Instrumentation

A. Data Acquistion System

The software for the EDI data acquisition system is written specifically for the Apple II+ with 48K RAM and uses a SSM IEEE 488 interface card, CSD real time clock and calender and an Interactive Structures AI13 12 bit A/D converter. The microcomputer is used to control a Hewlett Packard 3582A two channel FFT Spectrum Analyzer. The results are plotted using a Hewlett Packard 7470A digital plotter.

A photograph of a typical field equipment setup is given in Figure 1. This microcomputer-controlled system for data reduction, analysis, and presentation has been designed to be field-portable so that detailed analysis can be performed at the actual location of the problem machine. This is a time saving advantage over non-portable systems which require tape recording of data for reduction and analysis in laboratory conditions. Hardware costs for this system are much less than presently available non-portable diagnostic systems, yet overall capabilities are comparable. Specific applications include: frequency spectrum plotting, Bode plots, Nyquist plots, cascade or raster presentation of data versus speed, time, or other variables such as flow, modal analysis, multi-channel data logging, reciprocating cylinder performance, multiplane balancing, and a data documentation routine.

Figure 2 gives the general menu for the data acquisition system which lists the functions that can be selected. The details of four of the diagnostic programs are presented in the block diagrams. (The menu selection program is patterned after a program written by Mark Darlow, Reference 1.) The individual, or submenu selections, such as that for the Trim Balance Analyzer program, can be further outlined as in Figure 3.

Examples of the types of data acquisition and presentations are given in Figures 4-11. Figure 4 gives the Bode plots of amplitude and phase versus speed for the four proximity probes on a motor. The Nyquist plots for these vibrations are given in Figure 5. Figure 6 gives the frequency analyses of the vibration plotted versus speed in a Campbell diagram format. The cascade presentation of spectrum analysis versus time is illustrated in Figure 7. An example of the analysis of a structural vibration problem using modal analysis is given in Figure 8. A sample printout from the least squares multiplane balancing program is given in Figure 9. The data acquisition system can also be used to capture transient signals during rapid events, such as startups of synchronous motors. Figure 10 gives a digital plot of the transient stress in a shaft on startup obtained from a telemetry system. The system can also be used to measure the horsepower of reciprocating pumps and compressors using the pressure-time waves as shown in Figure 11.

B. Pump Transducers

Typically pumps are installed with minimum vibration monitoring equipment which generally consists of a velocity pickup on each bearing housing. These systems work particularly well for pumps with rolling element bearings, since the bearings will transmit the rotor forces directly to the case.

On pumps experiencing vibration problems, additional instrumentation is often required to define the problem. It is important to the engineer trying to find the cause of a vibration problem to have the proper instrumentation on the pump. For measuring shaft position and vibration, it is desirable to have two proximity probes 90 degrees apart near each bearing and an axial probe. Accelerometers and velocity probes attached to the bearing housings or case are often used to measure pump vibration; however, if the proximity probes are not installed the data will be limited and it can prevent the diagnosis of some types of problems. Another type of instrumentation that is vital to diagnostic work is a pressure transducer for measuring dynamic pulsations in the piping, impeller eye, diffuser and across flow meters. Accurate flow measurements are necessary to define the pump vibration characteristics as a function of its location on the head-flow performance map.

III. Typical Causes of Pump Vibration Problems

There are many types of problems which can occur in pumps. Therefore, several types of analyses and diagnostic procedures are required to positively identify vibration causes. Listed below are some of the more prevalent causes of excessive pump vibrations and failures. (Ref. 2-7)

A. Rotordynamic

1. Pump Lateral Critical Speeds
2. Torsional Critical Speeds
3. Seal Rubs
4. Unbalance
5. Coupling or Shaft-to-Shaft Misalignment
6. Shaft Instabilities
7. Shaft Misalignment in Journals
8. Inadequate or Inappropiate Bearing or Seal design

B. Fluid Dynamic

1. Hydraulic Instabilities
2. Recirculation
3. Pressure Pulsations and Acoustical Resonances
4. System Flow Distribution Problems
5. Water Hammer

6. Inadequate NPSH
7. Interaction of Pump Head-Flow Curve with Piping Responses
8. Flow Induced Excitations

C. Structural

1. Bearing Housing/Pedestal Resonances
2. Case Distortions Caused by Piping Strain
3. Impeller Resonances
4. Piping Lateral Mechanical Resonances
5. Piping Shell Wall Resonances
6. Foundation Resonances
7. Loose Tie Down Bolts and/or Grout

The vibration "trouble shooter" must develop a theory for the cause of the vibration problem based on the data that can be obtained. Some of the types of problems will only require a spectrum analysis to document the cause; however, other problems may require cascade spectrums over a range of speed, flow, or other variables, such as temperature, pressure, etc. Even when one of the data analysis techniques indicates a possible cause of a problem, additional tests may have to be made to positively identify one particular source as the cause.

Those vibration problems caused by rotordynamic influences may require spectrum analysis, Bode plots, Nyquist plots, plus cascade plots versus speed or time. Those caused by fluid dynamic sources may require spectrum analysis under varying operating conditions and can be difficult to discover if the operating conditions that cause the problem cannot be obtained during the testing. The problems caused by structural resonances can best be identified by modal analysis techniques, either by the use of a mechanical or electrodynamic shaker or by impact excitation to define the natural frequencies and mode shapes.

IV. Diagnosis of Pump Vibration Problems

To show the types of diagnostic procedures that are used to identify the causes of some typical pump problems, several field case histories will be presented.

A. Pump Critical Speed Problem (Ref. 2,3)

A critical speed analysis was performed on a centrifugal pump used as a jetting pump on a pipe laying barge. The critical speed analysis of liquid pumps is complicated by the effect of the liquid seals. The hydrodynamic forces involved affect both the response characteristics and the rotor stability. In liquid seals the liquid flows through the small clearances between the rotating shaft and non-rotating bushings. The fluid film interaction with the shaft gives rise to a load capacity and a set of dynamic stiffness and damping coefficients. Figure 12 gives the predicted unbalance vibration response of the centrifugal pump showing the effects of the seals. Note that the seal effects shifted the critical speed from 1800 to 3700 rpm, which illustrates that the pump critical speed is sensitive to seal stiffness effects. When the seals were considered, the predicted amplitudes at 1800 rpm were reduced by a factor of more than 10 to 1.

Figure 13 gives the measured vibrations on the pump. It can be seen that the vibrations were low until the pump reached 3300 rpm. The vibration sharply increased to 6 mils p-p at 3600 rpm. The design speed was 3600 rpm; however, the pump could not be run at that speed. The pump speed was kept below 3400 rpm so that the vibrations were less than 2 mils p-p. Even with the reduced pump speed, the pipe-laying barge was able to set pipe-laying records in the North Sea.

After a year of operation, the pump vibrations began to increase until the vibrations at 3400 rpm were unacceptable. The pressure breakdown bushing had worn which reduced it's effective stiffness and the critical speed had dropped to 3400 rpm. It was suggested that the pump be operated at a speed above the critical speed. This was tried and the pump operated at 3600 rpm with vibration levels less than 2 mils p-p. The barge remained in service and reset the pipe-laying records during the next season.

The data analysis technique used to determine this critical speed was to use the peak-store capabilities of the real-time analyzer. The tracking filter method would also have been an ideal way to determine this critical speed response since both the amplitude and phase data would have been available. Although it is generally better to have both the Bode and Nyquist plots, for this case they were not required to define the cause of the vibration problem.

B. High Vibrations of a Centrifugal Pump

Critical speed calculations were performed on a three-stage centrifugal pump to determine if a critical speed could be near running speed. Some of these calculations are summarized in Figures 14 and 15, which give the mode shapes for the first and second critical speeds. As discussed in Case 1, the liquid seals can significantly affect the critical speeds. However, for this rotor the seals only increased the critical speed by about 15 percent. This critical speed analysis had to consider a shaft with two bearings and the eight seals, or ten sets of eight stiffness and damping coefficents. Each of these coefficients varied as a function of speed and was included in the unbalance response analysis (Figure 16). A comparison of predicted reponses to measured test stand data is given in Figure 17.

The data acquisition system was used as a tracking filter to plot the amplitude and phase angle versus speed (Bode Plot). The agreement with the measured data was good. This means that the rotordynamic model of the rotor, bearings, and seals was correct. The lack of a significant phase shift through the critical speed could not be explained without a more detailed analysis. As in many field studies, other priorities prevailed, and additional analyses were not possible. Although the critical speed was in the running speed range, the vibration amplitudes were low and any wear of the seals would move it further away from the rated speed of 3600 rpm.

C. Pump Seal Rub

During the testing of the above pump, a seal rub was experienced. Seal rubs can be caused by high synchronous vibrations and sometimes by hydraulic or rotordynamic instabilities. Figure 18, which is a cascade presentation of the spectrum analysis versus time, shows the beginning of a seal rub at synchronous frequency.

Many times a seal rub can be caused by hydraulic instabilities. To verify that the problem was not caused by hydraulic instabilities, the cascade spectrum analysis was plotted from minimum to maximum flow for a constant speed (Figure 19). Hydraulic instabilities most often occur at reduced flow rates and can cause subsynchronous shaft vibrations at frequencies which are usually less than 30 percent of synchronous speed. Figure 19 does not show any large subsynchronous vibrations, indicating that the seal rub problem was not due to hydraulic instabilities.

D. Pulsation Induced Vibrations (Ref. 6)

A four-stage centrifugal pump suffered repeated failures of the splitter between pump stages. A detailed field study revealed the cause of the problems was an acoustic resonance of the long cross-over which connected the second stage discharge with the third stage suction (Fig. 20). The resonant frequency was a half-wave acoustic resonance.

$$f = \frac{A}{2xL}$$

where

A= speed of sound, ft./sec.
L= length, ft.

The speed of sound in water is a function of the temperature, and at 310 °F was calculated to be 4,770 ft/sec. The length of the cross-over was 5.75 feet. The acoustic natural frequency was

$$f = \frac{4770}{2 \text{ X } 5.75} = 415 \text{ Hz}$$

The acoustic resonant frequency was excited by the blade passage frequency (7 times running speed). Coincidence occurred at (415) (60)/7 = 3,560 rpm.

Pulsations measured in the center of the cross-over showed pulsation amplitudes of 100 psi peak-to-peak. The pulsations at the suction and discharge flanges were less than 10 psi peak-to-peak, which agreed with the mode shape of the half-wave acoustic resonance.

There were two possible changes that would eliminate the coincidence of blade passage frequency and the resonant frequency and reduce the vibration levels that occurred when the speed was 3560 rpm. One possibile change was to reduce the diameter of the impellers and operate the pump at a higher speed. Another possibility was to change to 6 or 8 blades to alter the blade passage frequency. The impeller diameter modification was the quickest and most economical and was carried out in the field, and the splitter failures were eliminated.

To properly diagnose problems with acoustical resonances, it is useful to plot cascade diagrams of pulsations versus speed. Peak-store frequency analysis spectrums of the pulsations over the entire speed range can also define the acoustical resonances.

The above example also illustrates the importance of selecting proper test points when measuring pulsations. For example, if the acoustical resonance is in the cross-over, then the pressure transducer should be installed near the center of the cross-over length, rather than at the ends. When measuring pulsations, another problem that is often encountered is the acoustical quarter-wave stub frequency that is excited by flow past the open end. This is analogous to blowing past a soda bottle. When a transducer is mounted in a pipe, there will be some length from the transducer diaphragm to the inside surface of the pipe. This length is acoustically open on one end and closed at the pressure transducer. The first acoustical natural frequency excited by the flow past the open end will be A/4L where A is the velocity of sound in feet per second and L is the length of the stub in feet. For an installation where the speed of sound was 4500 ft/sec. and the transducer was mounted one foot from the inside surface, the stub (quarter-wave) frequency would be 1125 Hz. A fictious pressure pulsation component will be measured at this frequency and also at 3X, 5X, 7X, etc. The pulsation exists in the stub; however, it may not be present in the main pipe. If this stub frequency is close to the vane or blade passing frequency, the measured amplitude at the stub frequency will not be valid since it would be amplified by the acoustical resonance. The acoustical amplification factor can be as high as 300. To measure high frequency pulsations, the transducer should be mounted flush to the inside surface of the pipe.

Sometimes the cross-over or cross-under passage can have an acoustical resonance that can be so severe that it can excite the shaft and result in high shaft vibrations (Ref. 2). This is illustrated in Figure 21 which shows that pulsation levels in the cross-under were over 250 psi and caused approximately 0.5 mils of shaft vibration at the acoustic natural frequency.

E. Shaft Failures Caused by Hydraulic Forces

Repeated shaft bending fatigue failures were experienced in a high pressure ash water pump (Figure 22). The failures exhibited the classical fatigue beach marks with the failures occurring straight across the shaft at the sharp corners at the change in diameters. The pump was instrumented with proximity probes, pressure transducers, and accelerometers. Measurements were made over a wide range of startup and flow operating conditions. The pumps had a double-volute casing which was supposed to balance the radial forces on the impeller; however, measurements made of the shaft centerline by measuring the DC voltage on the proximity probes showed that the impeller was being forced up against the casing. This caused a large bending moment on the shaft as it rotated. Figure 23 gives the pump shaft center location on startup and during recycle near the inboard and outboard bearings. It can be seen that there was 6 mils differential movement across a fairly short distance. The wear patterns on the impellers were consistent with the major axis of the orbit and the direction that the shaft moved. The large movement in the bearing journals only occurred under certain start-up conditions; therefore it was possible to modify the start-up procedures to insure that the large hydraulic forces would not cause shaft failures.

This example illustrates that hydraulic forces can cause shaft failures, therefore it is good practice to determine if the shaft is properly aligned in its journals under all operating conditions.

F. Pump Instability Problem

A high speed pump experienced high vibrations in the process of startup at the jobsite. Originally, the problem was thought to have been due to a lateral critical speed causing increased synchronous vibration levels when full speed was reached.

Analysis of this problem was particularly difficult due to the extremely short startup time of the motor-driven pump and the even more rapid rate at which the vibration levels increased as the pump approached rated speed. To analyze the problem, an FM recording of a startup was analyzed while running the recorder playback at 1/8 of recorded speed, which in effect, caused the startup period during playback to be 8 times as long. Figure 24 shows a cascade plot of the vibration data. The plot showed that just before trip of the unit at 21960 rpm, an instability vibration component occurred near 15000 cpm. From this and other data, it was determined that the high vibrations were caused by:

(1) A sudden increase in nonsynchronous vibration as the unit approached full speed resulting in shaft bow.

(2) A sudden increase in unbalance due to the shaft bow and, as a result, a rapid increase in synchronous vibration levels as the nonsynchronous components disappeared.

After the problem source was identified using the above data analysis technique, computer simulation of the rotor led to a solution consisting of bearing modifications. The stability analysis of the pump rotor predicted that the pump had an unstable mode at 15000 cpm with a negative logarithmic decrement of 0.01 for a simulated fluid dynamic loading of 1000 lb/in at the impellers (Ref. 7,8). The pump rotor with the modified bearings was predicted to have a positive logarithmic decrement of 0.10. Figure 25 shows the rotor vibrations after the modifications were made. The nonsynchronous vibration component was no longer present and the unit has since operated successfully.

REFERENCES

1. Darlow, Mark, "An Interactive Menu Routine", Dept. of Mechanical Engineering, Rensselaer Polytechnic Institute, Troy, NY 12181

2. Wachel, J. C., "Prevention of Pump Problems", Worthington 2nd Technical Pump Conference, Oct. 1981.

3. Szenasi, F.R. and Wachel, J.C. "Vibration and Noise in Pumps", Pump Handbook , McGraw-Hill, 1976, Section 9.5, pp. 9-87 - 9-97.

4. Wachel, J.C. and Szenasi, F.R., "Displacement Pump Performance, Instrumentation and Diagnostics", Pump Handbook , 2nd Edition, McGraw Hill (to be published - 1983).

5. Sparks, C.R., Szenasi, F.R., Wachel, J.C., "Pump Noise" Pump Handbook , 2nd Edition, McGraw Hill (to be published - 1983).

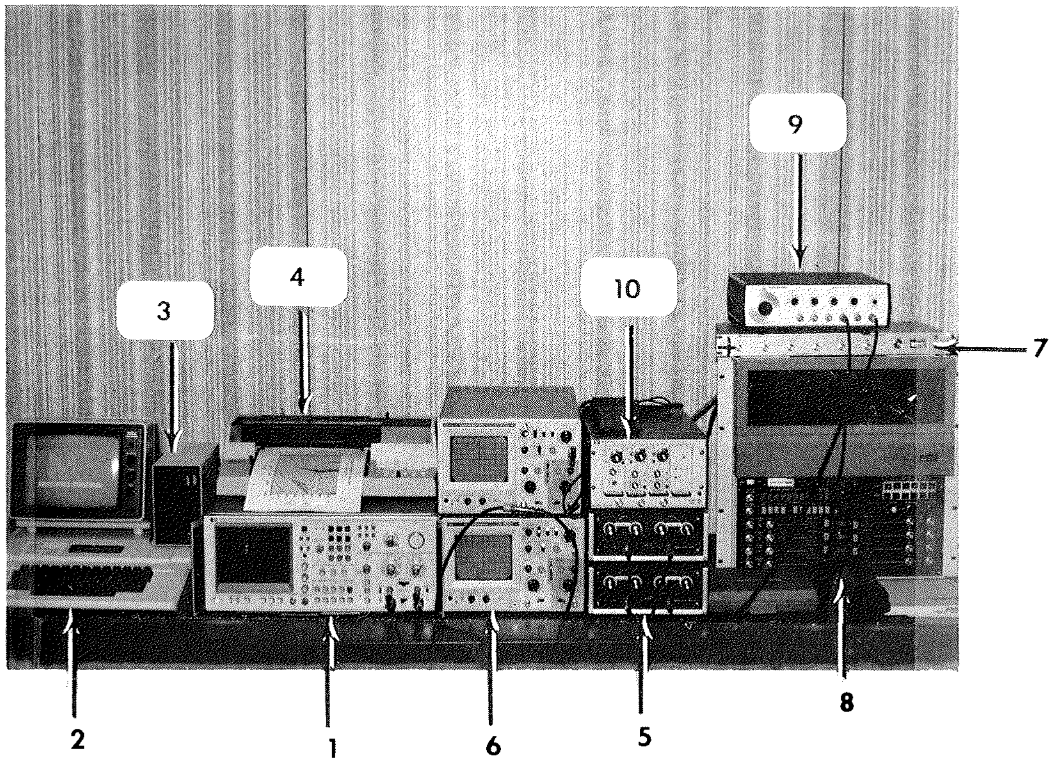
6. Sparks, C.R. and Wachel, J. C., "Pulsation in Centrifugal Pump and Piping Systems", Hydrocarbon Processing , July 1977, pp. 183-189.

7. Wachel, J.C. "Rotordynamic Instability Field Problems" Rotordynamic Instability Problems in High-Performance Turbomachinery, May 1982, NASA Conference Publication 2250, p. 1.

8. Allaire, P.E., Branagan, L.A., Kolur, J.A., "Aerodynamics Stiffness an Unbounded Eccentric Whirling Centrifugal Impeller With a Definite Number of Blades," Rotordynamic Instability Problems in High-Performance Turbomachinery, NASA Conference Publication 2250, p.323.

Figure 1

Typical Field Equipment Setup



ITEM	DESCRIPTION
1	2 Channel FFT Analyzer
2	Micro-computer
3	Floppy Disk Drive
4	Digital Plotter
5	Tuneable Filters
6	2 Channel oscilloscope
7	Transducer Signal Conditioner and Power Supply
8	8 Channel FM Tape Recorder
9	Function Generator
10	Strain Gage Amplifier and Frequency Demodulator

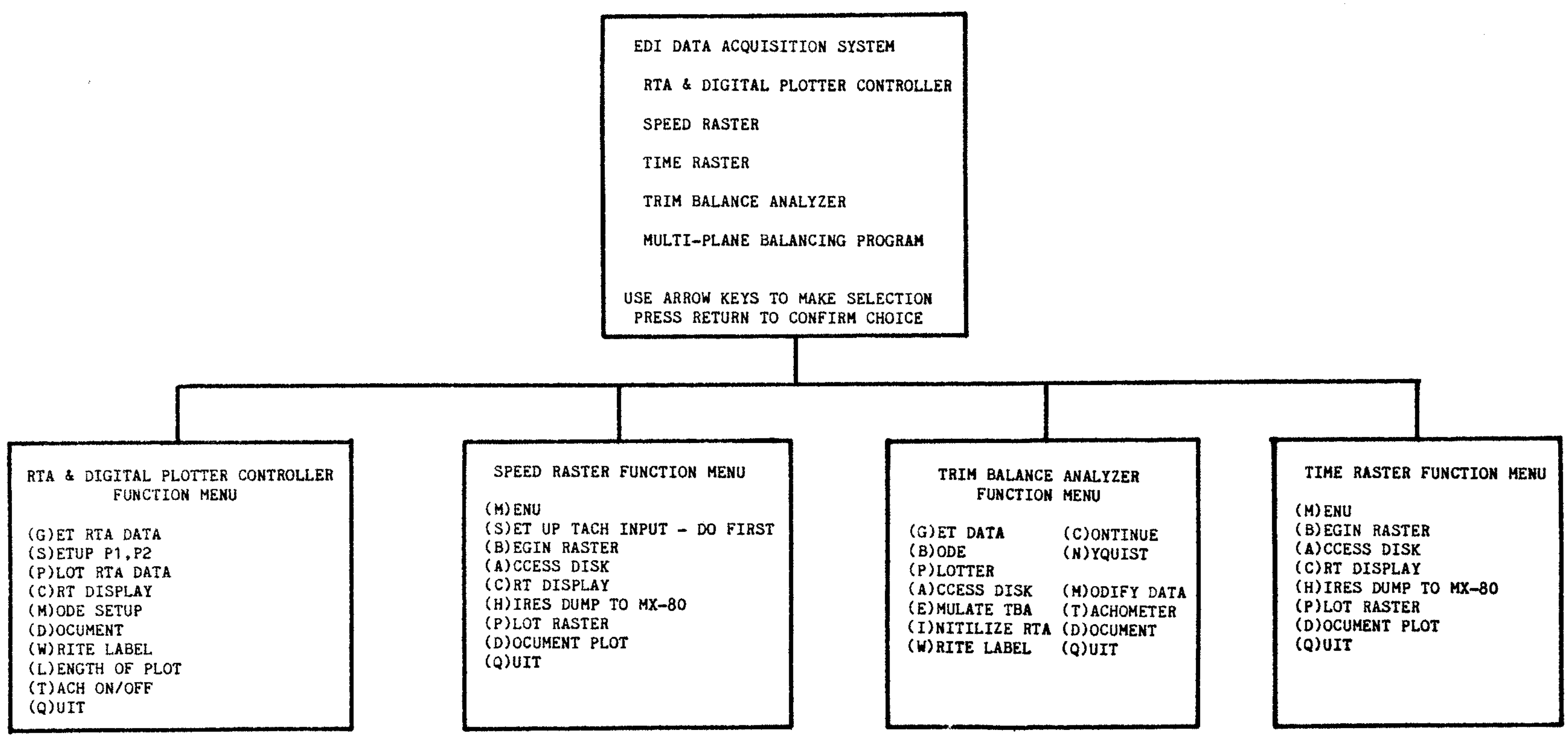


Figure 2

SAMPLE MENU FOR DATA
ACQUISITION SYSTEM

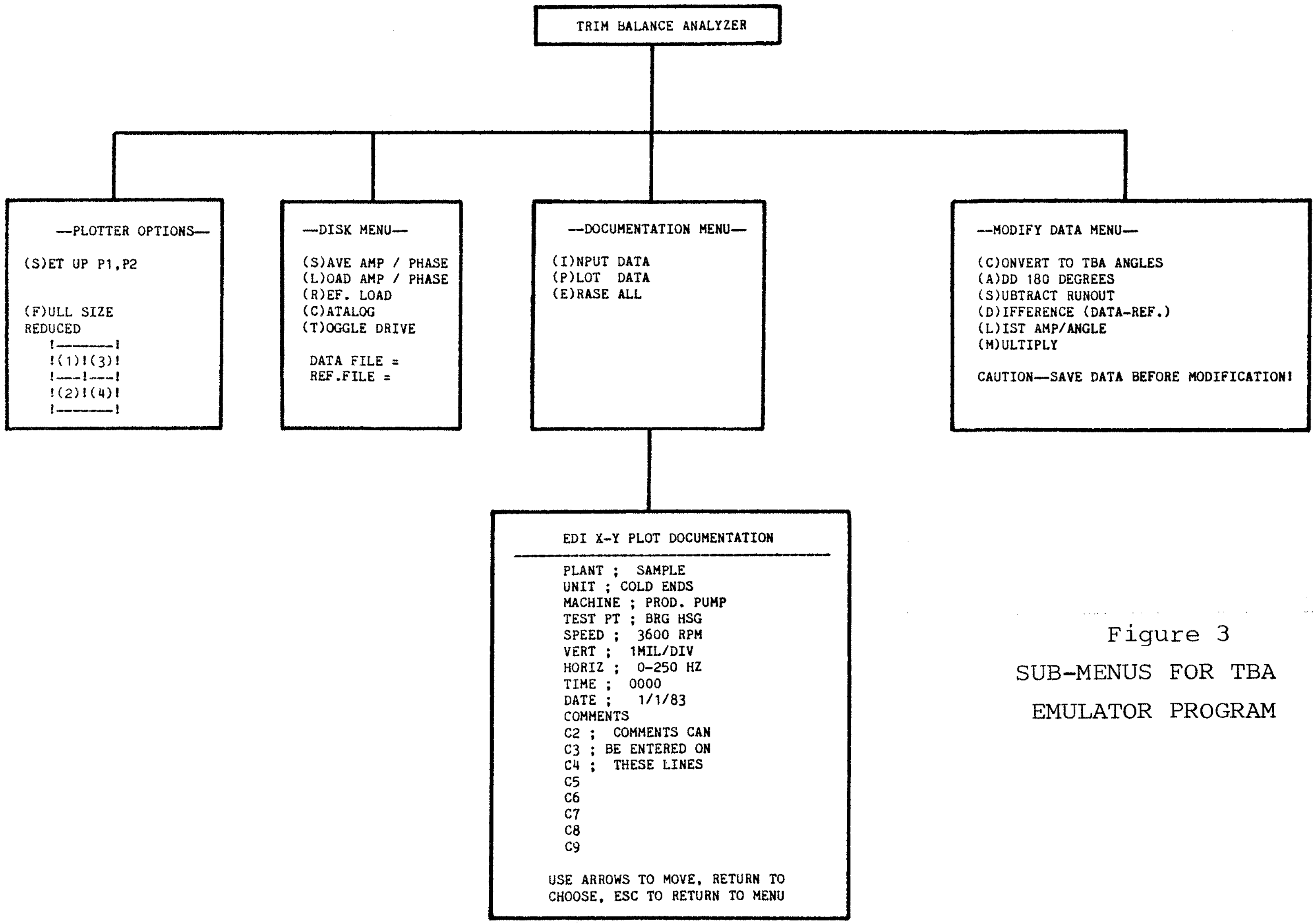


Figure 3

SUB-MENUS FOR TBA EMULATOR PROGRAM

ENGINEERING DYNAMICS INCORPORATED

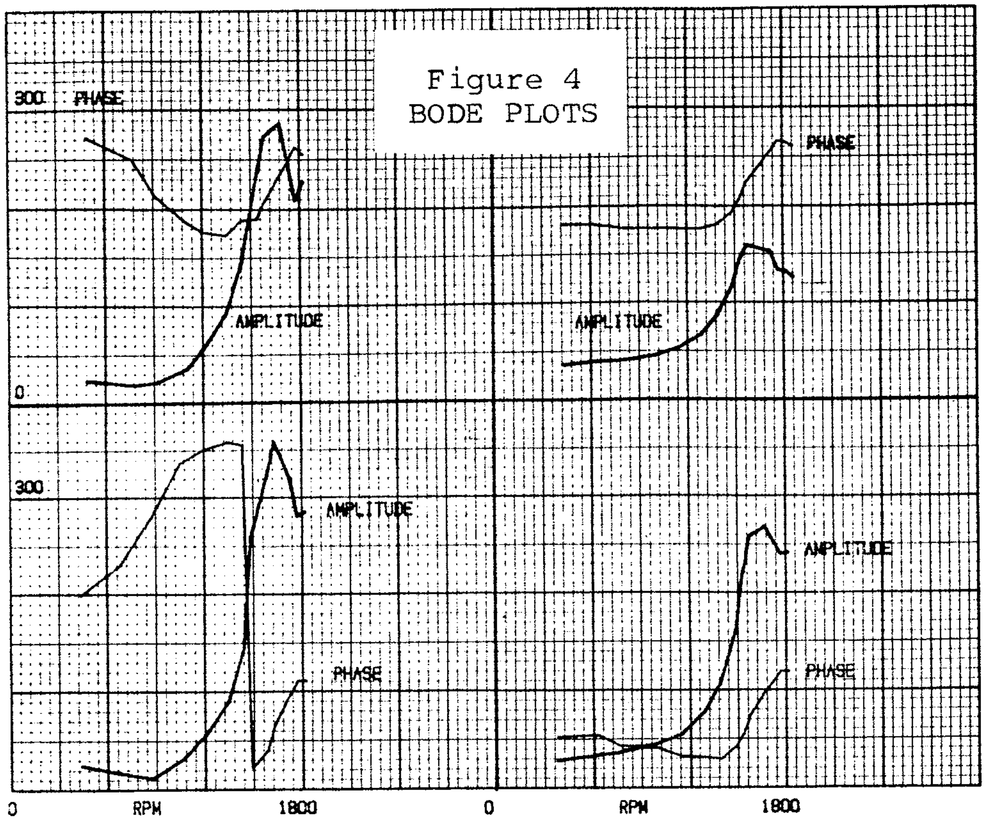


Figure 4
BODE PLOTS

VIBRATION ips ☐ mils ☒ g's ☐
PULSATION psi ☐
STRAIN μ-in/in ☐
NOISE dB ☐

PLANT
UNIT EXTRACTION PLT.
MACHINE D 451 A
TEST PT NOTED BELOW
SPEED START UP
VERT 2 MILS/DIV.
HORIZ 0-50 HZ
TIME 11:15 AM
DATE FEB 2 1989
COMMENTS

----PROBES----
MOB V MIB V
MOB H MIB H

ENGINEERING DYNAMICS INCORPORATED

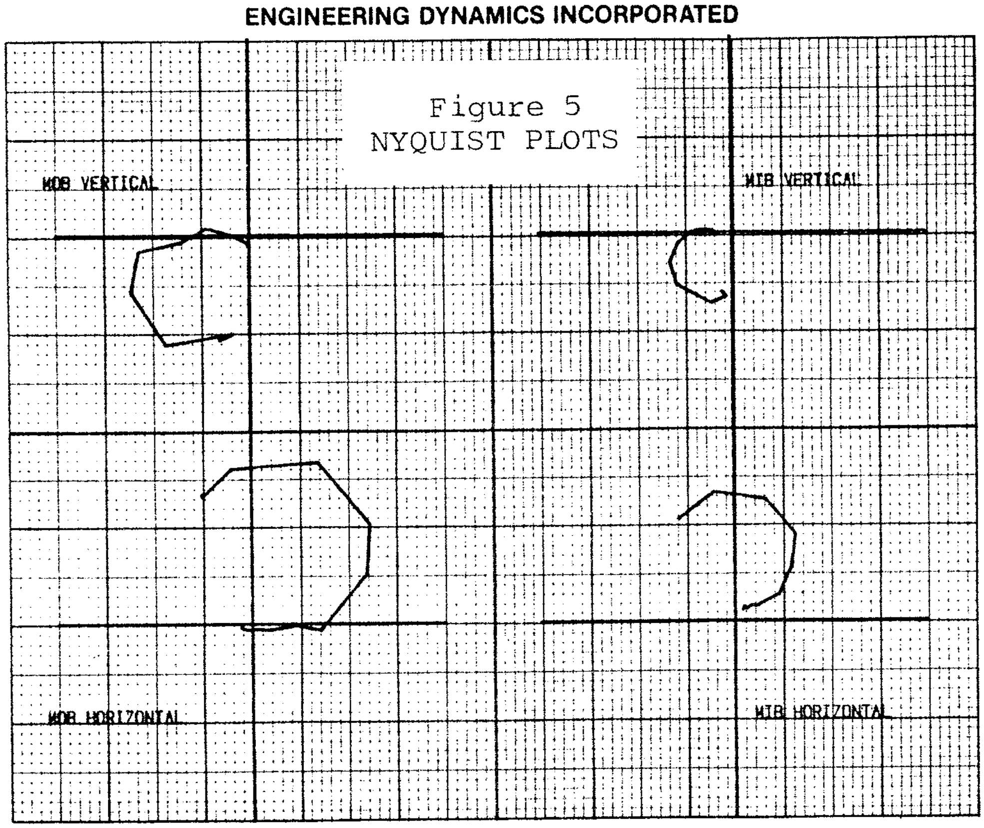


Figure 5
NYQUIST PLOTS

VIBRATION ips ☐ mils ☒ g's ☐
PULSATION psi ☐
STRAIN μ-in/in ☐
NOISE dB ☐

PLANT
UNIT EXTRACTION PLT.
MACHINE D 451 A
TEST PT NOTED BELOW
SPEED START UP
VERT 4 MILS/DIV.
HORIZ POLAR
TIME 11:15 AM
DATE FEB 2 1989
COMMENTS

----PROBES----
MOB V MIB V
MOB H MIB H

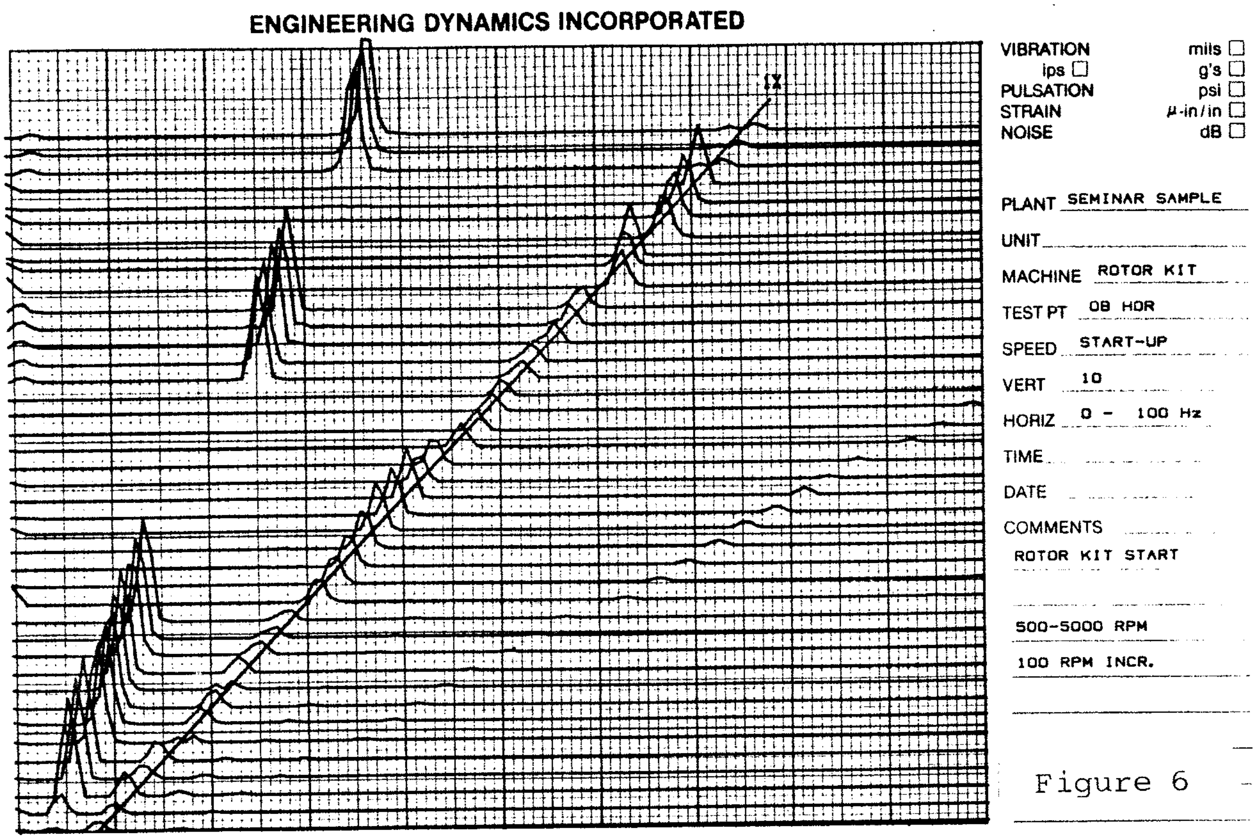


Figure 6

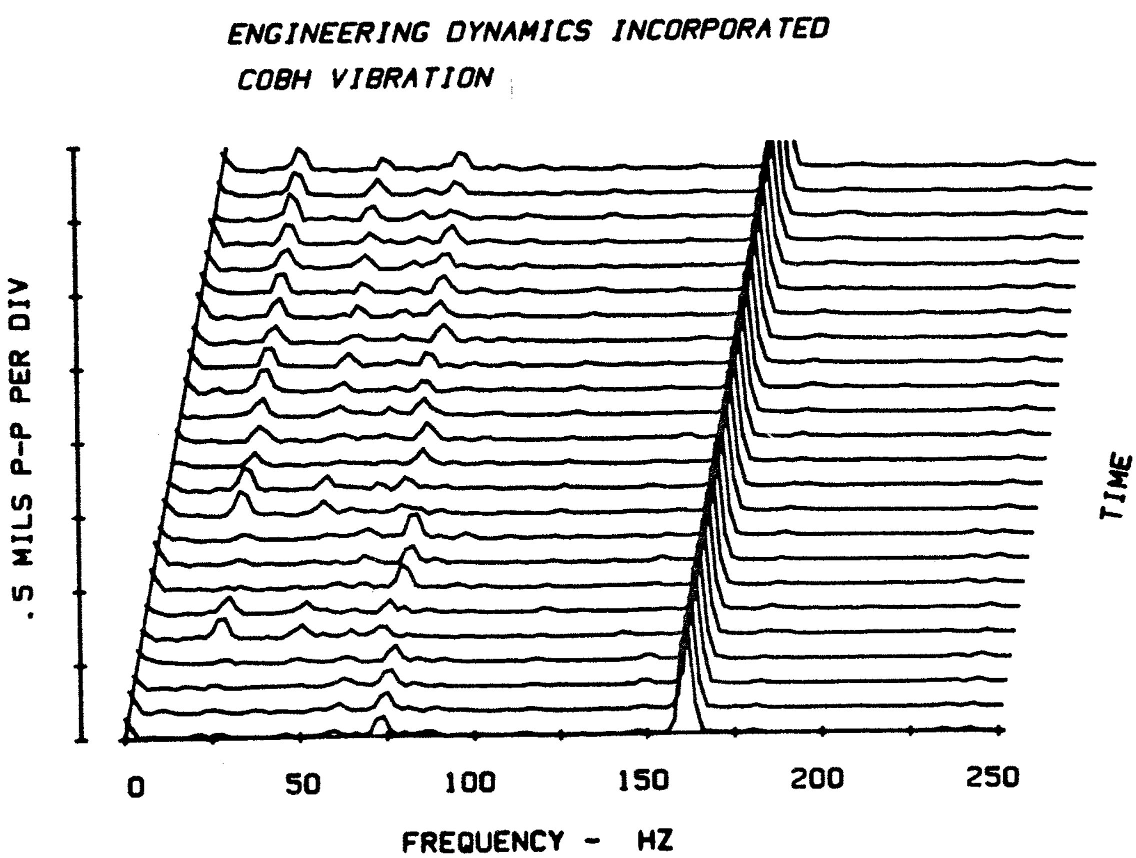


Figure 7

Figure 8

SAMPLE OUTPUT FROM

MODAL ANALYSIS SYSTEM

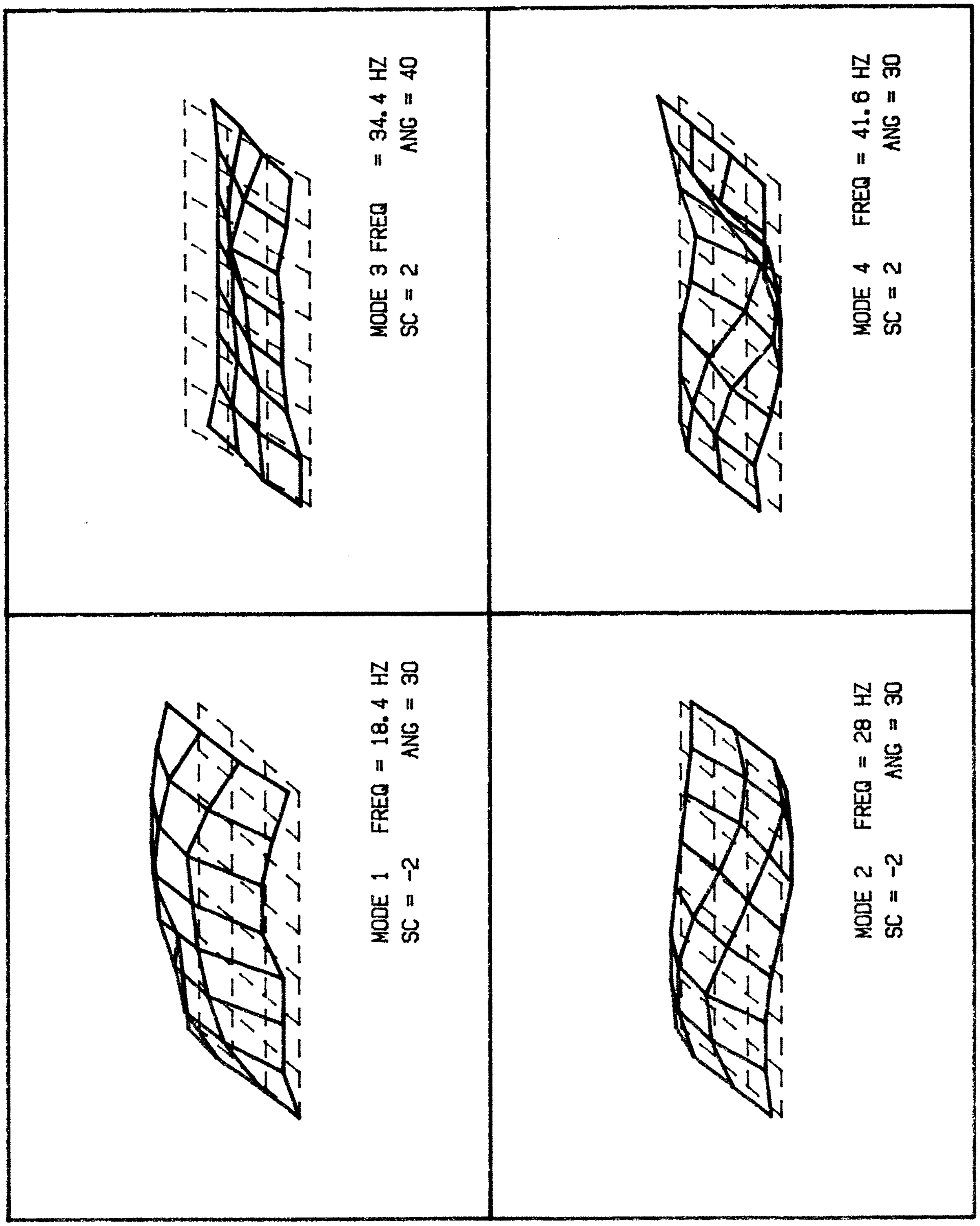


Figure 9

ENGINEERING DYNAMICS INCORPORATED
MULTI-PLANE BALANCING PROGRAM

PLANT:
UNIT: 1-A
MACHINE: I. D. FAN

PROJECT:
DATE:
ENGINEER: KEA/WRF

BASELINE DATA - PLANE #1
(FILENAME = BLID1A)

TEST POINT DESCRIPTION		SPEED (RPM)	AMPLITUDE (MILS P-P)	PHASE (DEG)
1ST STG HORIZ	# 1	900	6.5	184
2ND STG HORIZ	# 2	900	5.7	184

TRIAL WEIGHT DATA - PLANE #1
(FILENAME = TWID1A15)
TRIAL WEIGHT = 73.0 OZ AT 272 DEG

TEST POINT DESCRIPTION		SPEED (RPM)	AMPLITUDE (MILS P-P)	PHASE (DEG)
1ST STG HORIZ	# 1	900	4.4	106
2ND STG HORIZ	# 2	900	4.0	106

VIBRATION TO BE MINIMIZED

TEST POINT DESCRIPTION		SPEED (RPM)	AMPLITUDE (MILS P-P)	PHASE (DEG)
1ST STG HORIZ	# 1	900	4.4	106
2ND STG HORIZ	# 2	900	4.0	106

CORRECTION WEIGHTS

PLANE NO.	AMPLITUDE (OZ)	PHASE (DEG)
1	45.82	155

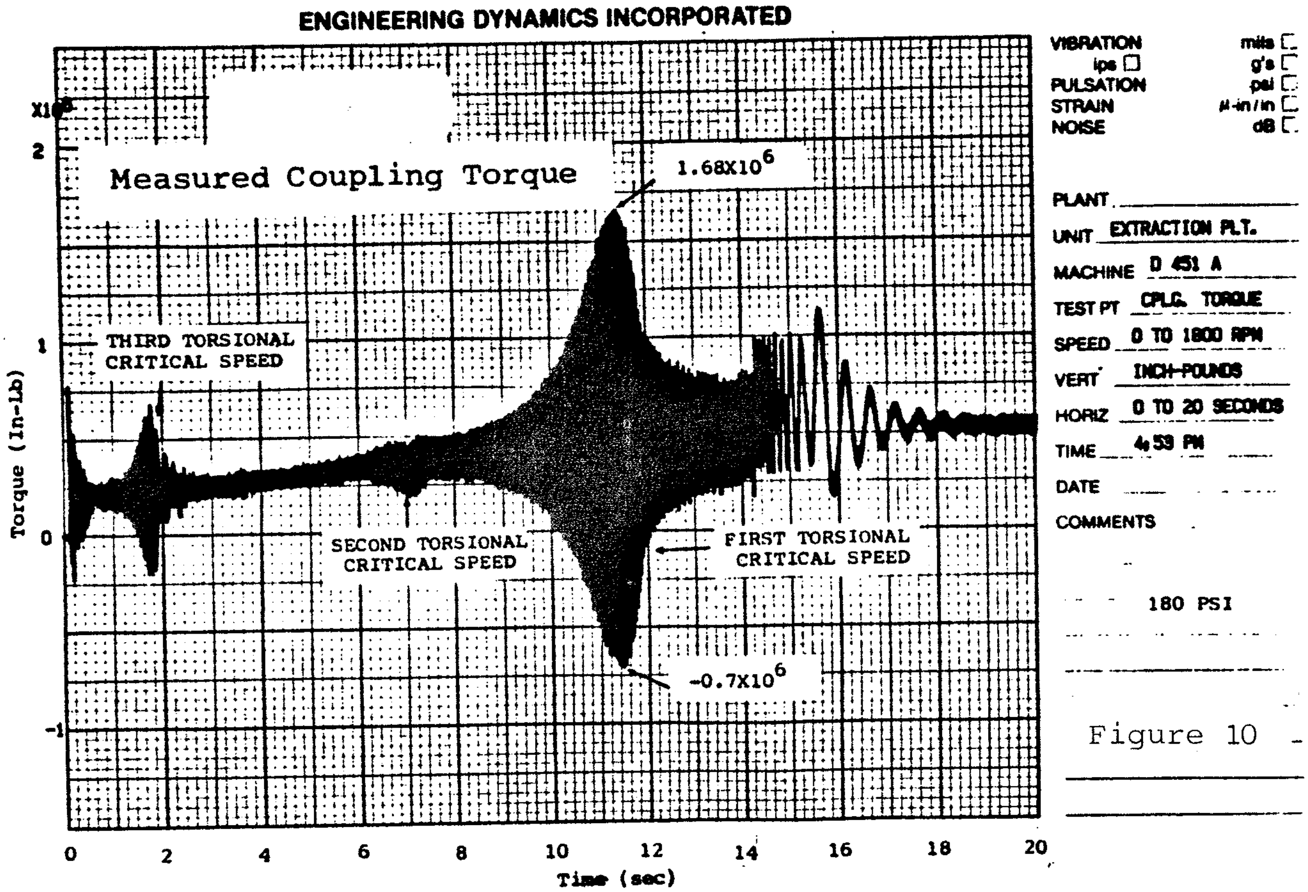


Figure 10

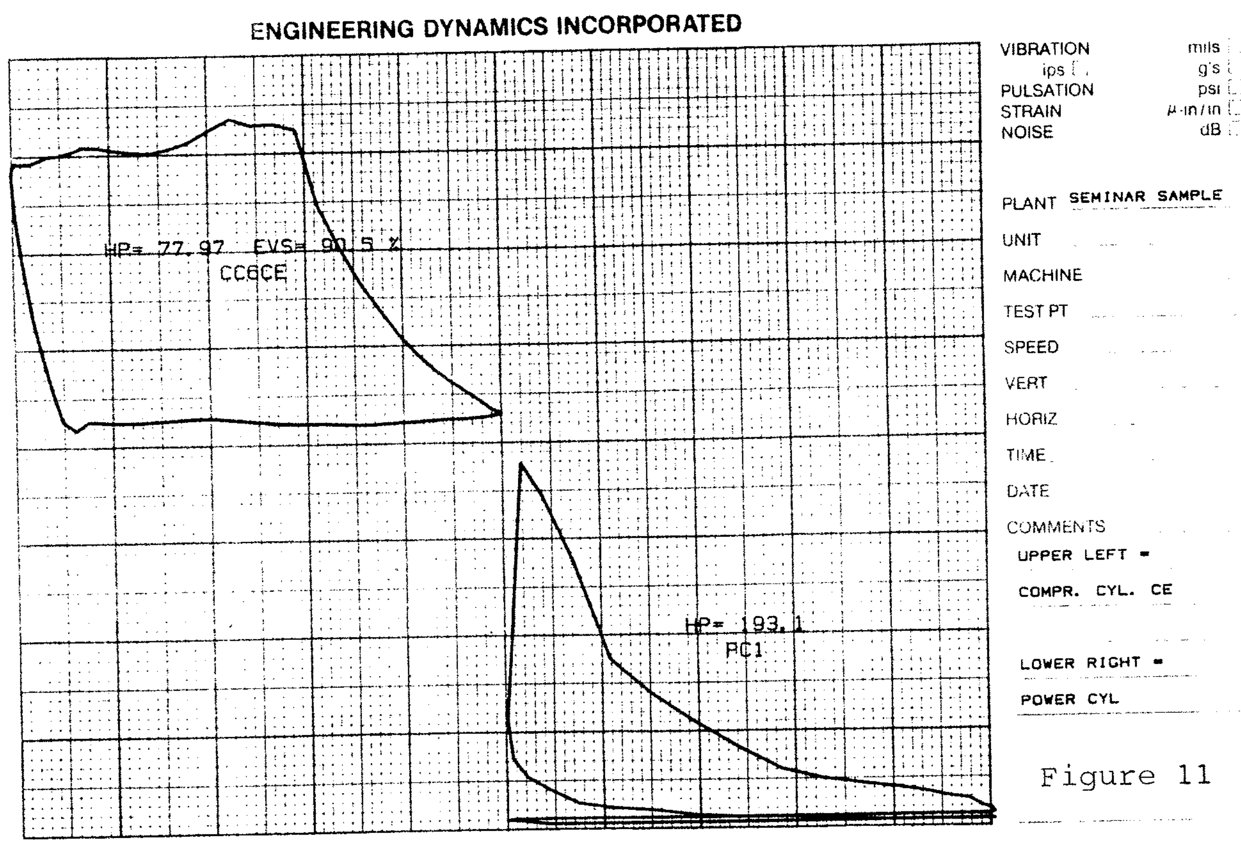
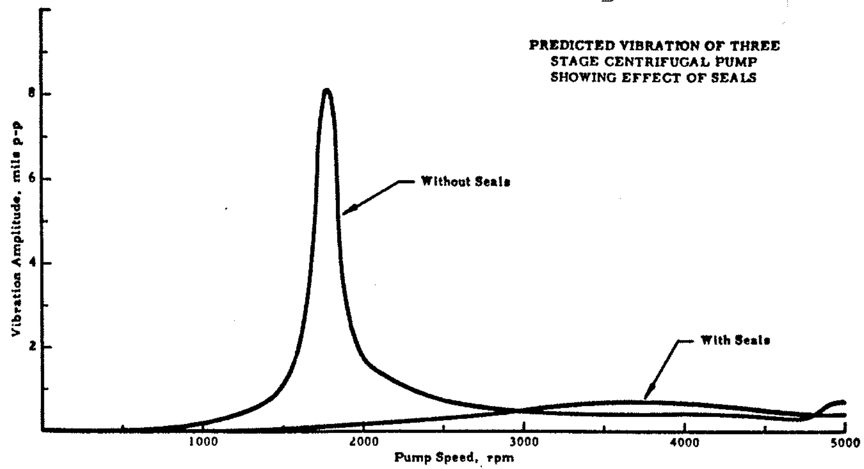


Figure 11

Figure 12



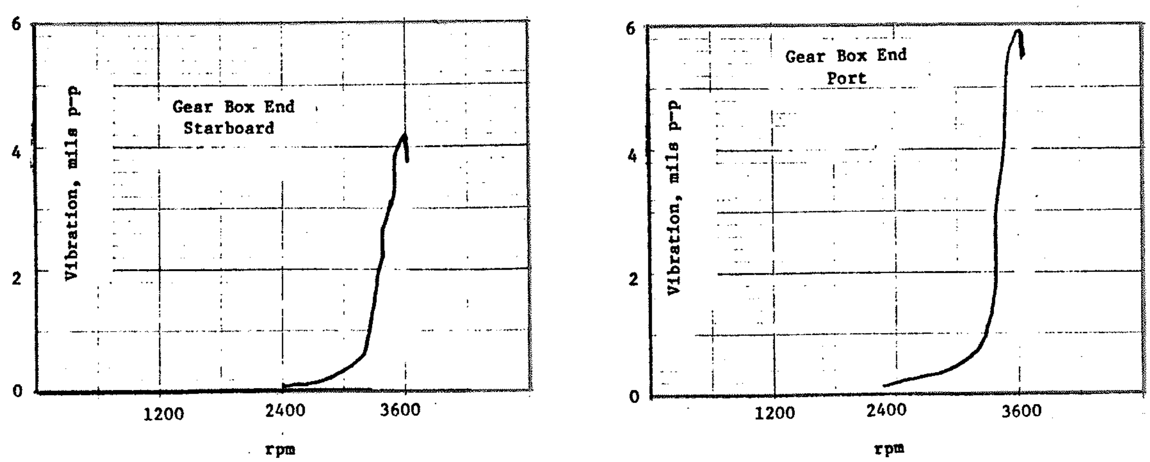


Figure 13

JETTING PUMP VIBRATIONS

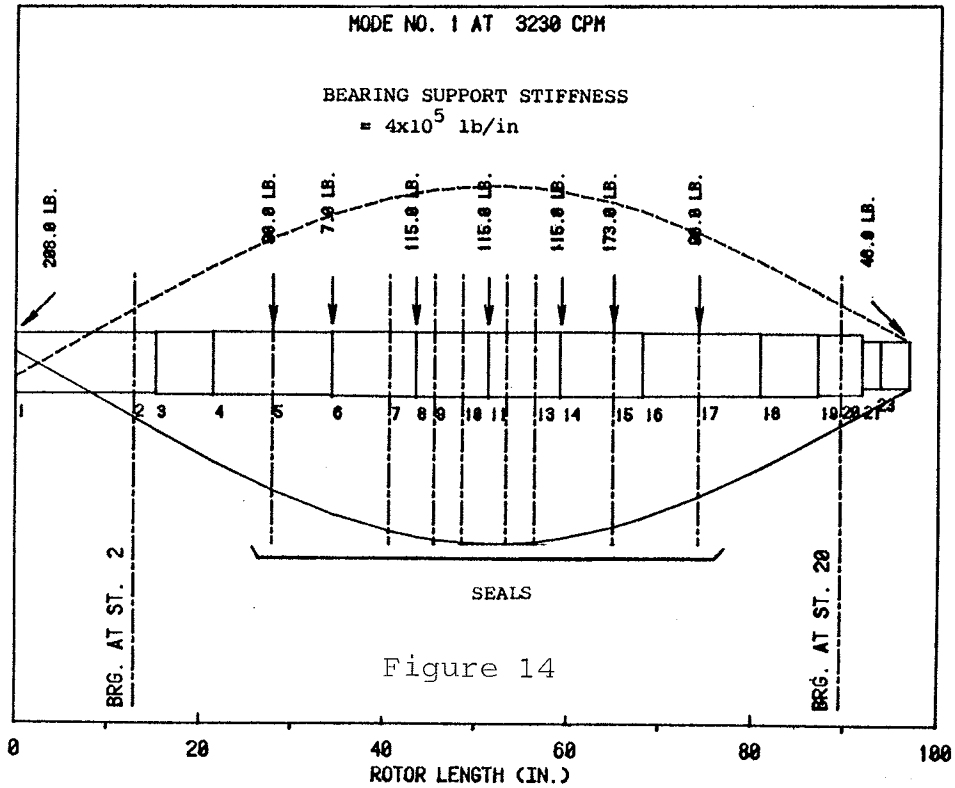


Figure 14

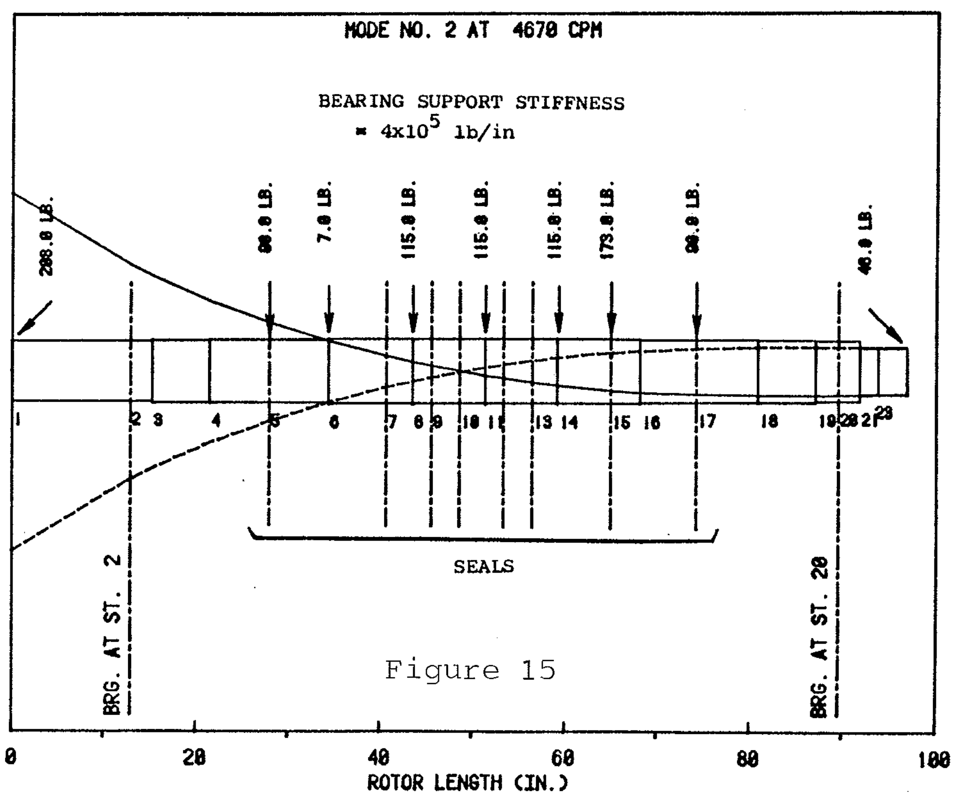
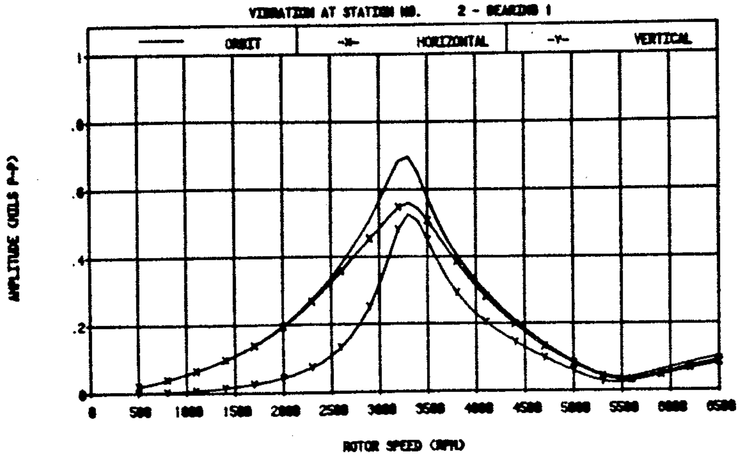


Figure 15

ENGINEERING DYNAMICS INCORPORATED
UNBALANCE RESPONSE - PUMP - EDI NO. 82
5SLOP BEARINGS DESIGN CLEAR. CLAD=0 MILS (NOMINAL) - NEW SEALS

UNBALANCE AT STATION NO. 11 - IMPELLER = 4.610 IN-OZ AT 0 DEG.

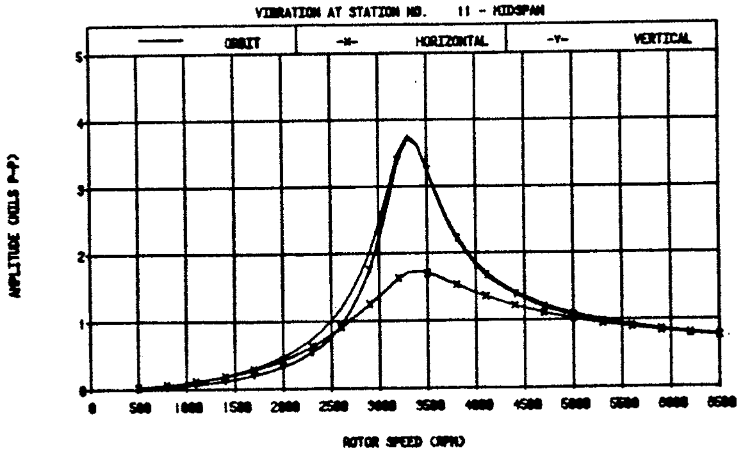
VIBRATION AT STATION NO. 2 - BEARING 1



ENGINEERING DYNAMICS INCORPORATED
UNBALANCE RESPONSE - PUMP - EDI NO. 82
5SLOP BEARINGS DESIGN CLEAR. CLAD=0 MILS (NOMINAL) - NEW SEALS

UNBALANCE AT STATION NO. 11 - IMPELLER = 4.610 IN-OZ AT 0 DEG.

VIBRATION AT STATION NO. 11 - MIDSPAN



ENGINEERING DYNAMICS INCORPORATED
UNBALANCE RESPONSE - PUMP - EDI NO. 82
5SLOP BEARINGS DESIGN CLEAR. CLAD=0 MILS (NOMINAL) - NEW SEALS

UNBALANCE AT STATION NO. 11 - IMPELLER = 4.610 IN-OZ AT 0 DEG.

VIBRATION AT STATION NO. 20 - BEARING 2

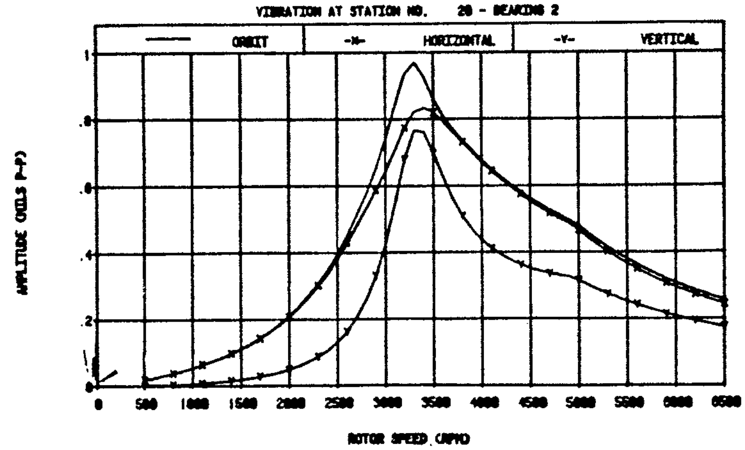


Figure 16

Figure 17

COMPARISON OF PREDICTED AND

MEASURED PUMP VIBRATIONS

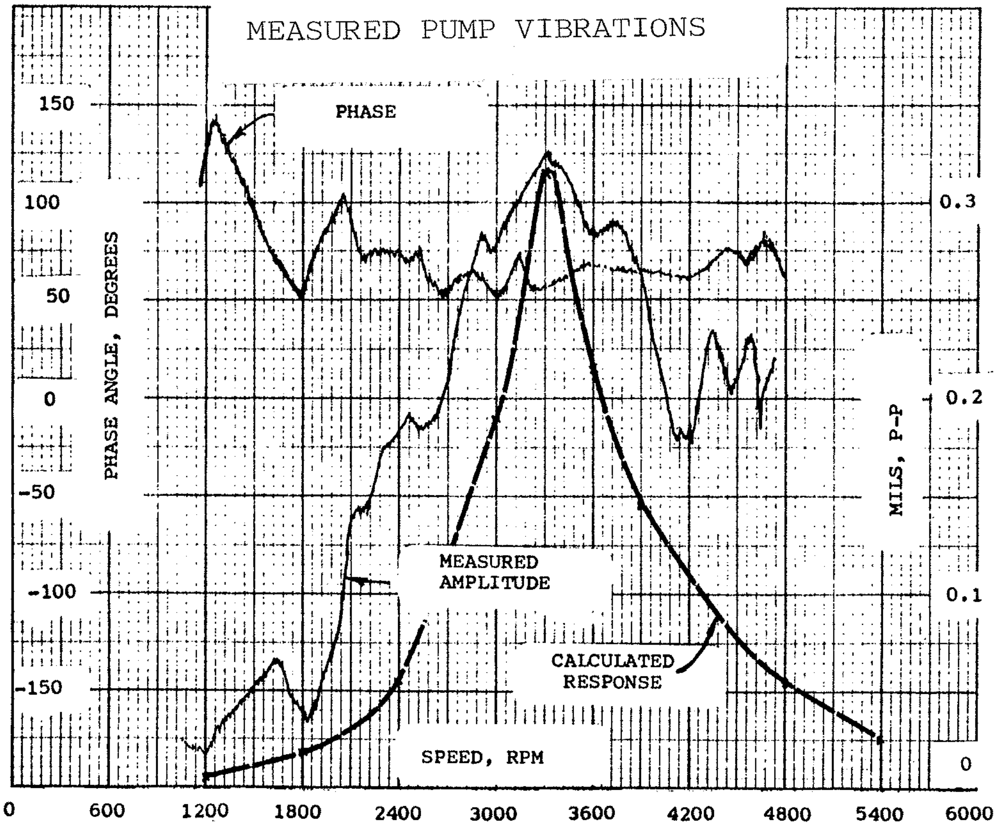


Figure 18

CASCADE PLOT VERSUS TIME

ILLUSTRATING SEAL RUB

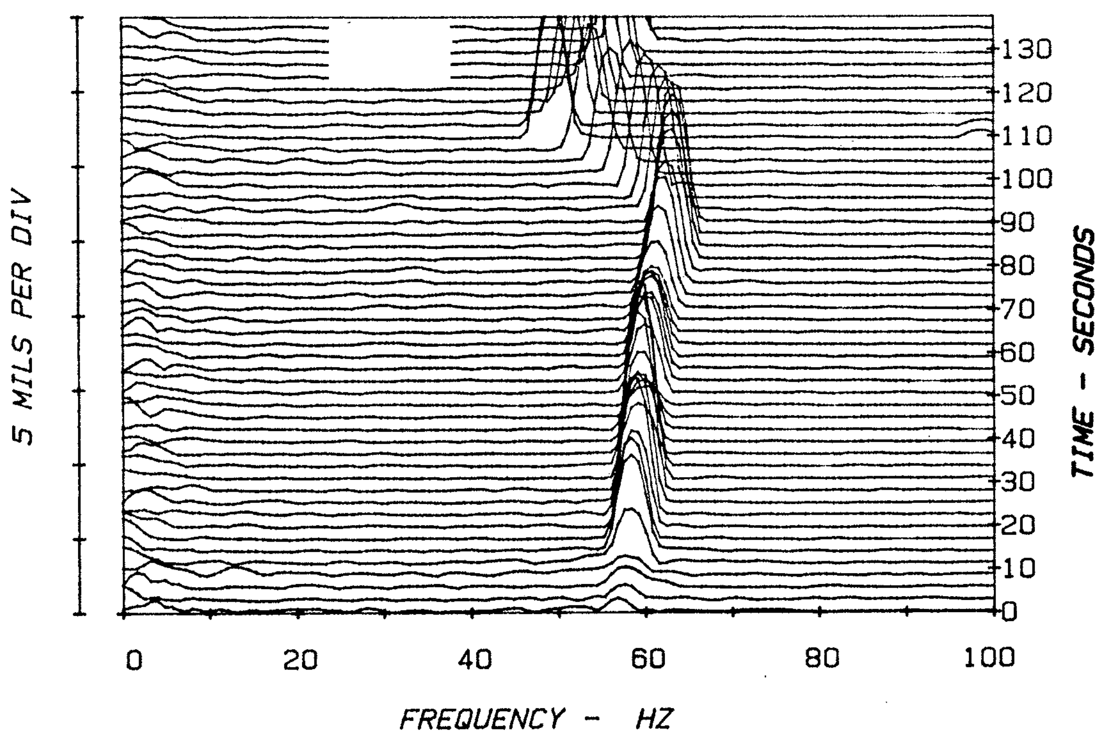
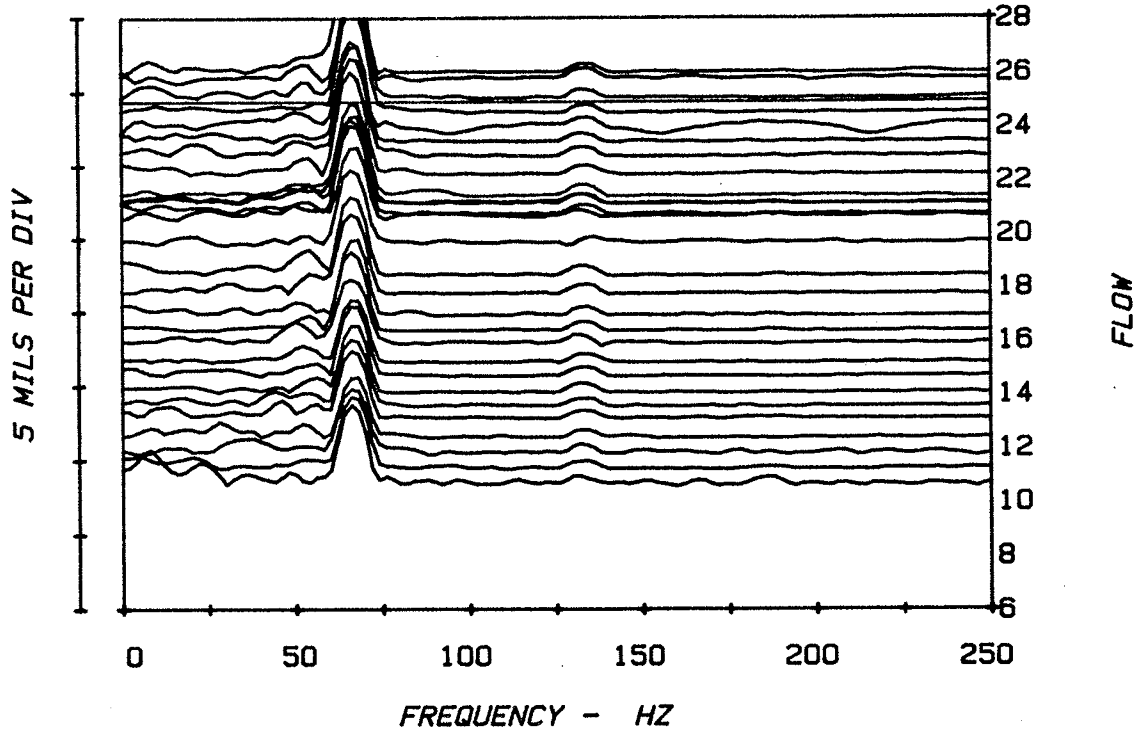
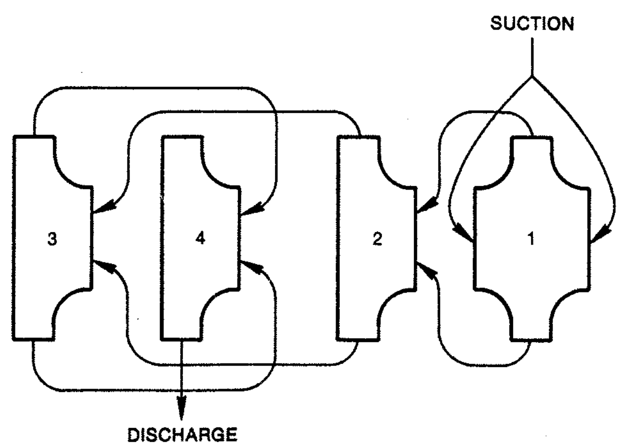


Figure 19

CASCADE PLOT VERSUS FLOW

FOR CONSTANT SPEED

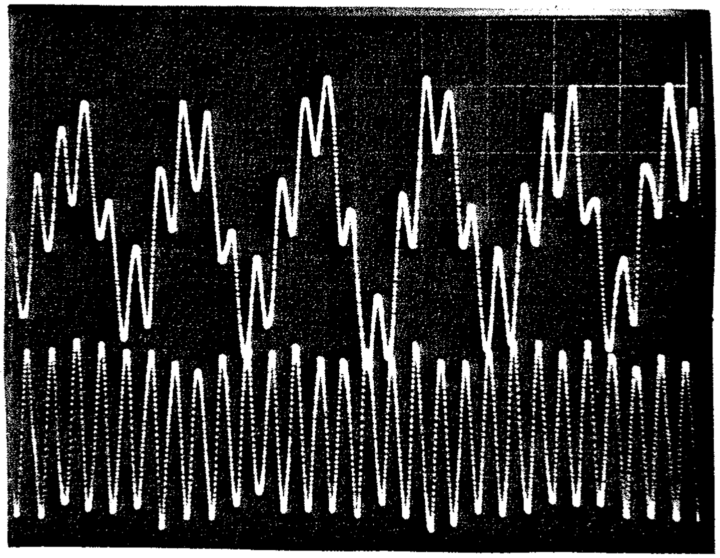




Flow schematic of a four-stage pump.

Figure 20

Figure 21
ACOUSTIC RESONANCE
CAUSES SHAFT VIBRATION



SHAFT VIBRATION

CROSSUNDER
PULSATIONS

Figure 22
SINGLE STAGE
OVERHUNG PUMP

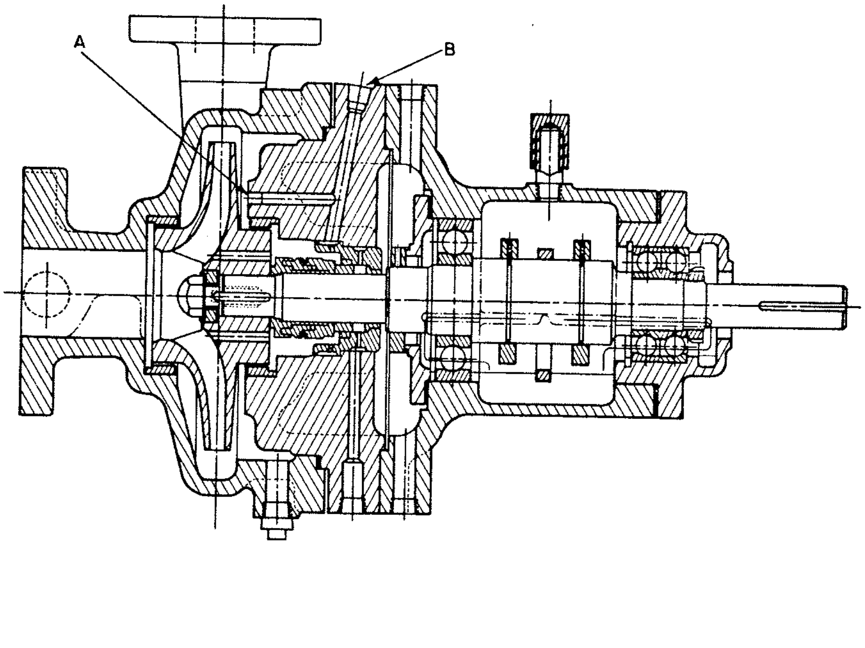
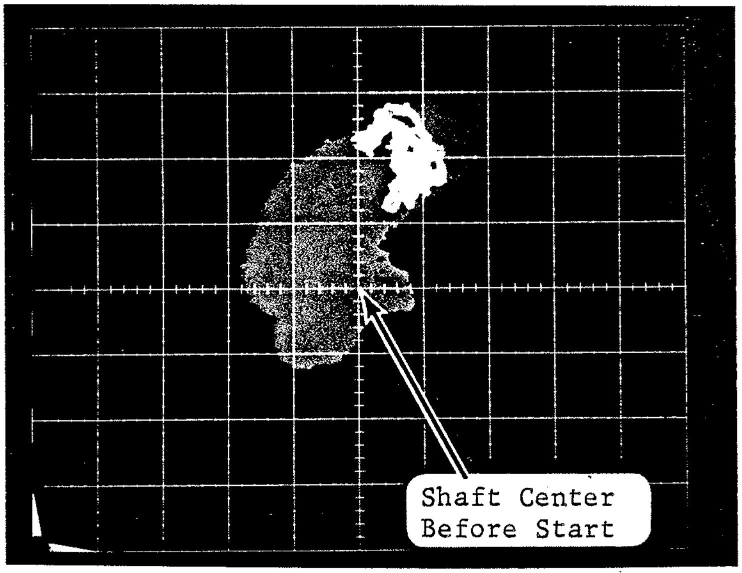


Figure 23

Pump Shaft Vibration Orbits

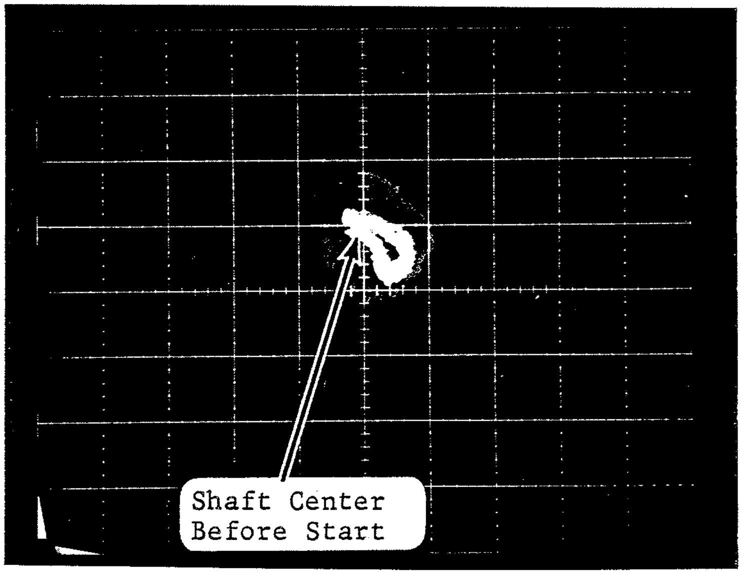
Viewed from
Suction End

2 mil/div



Pump Shaft Center
Startup and Recycle

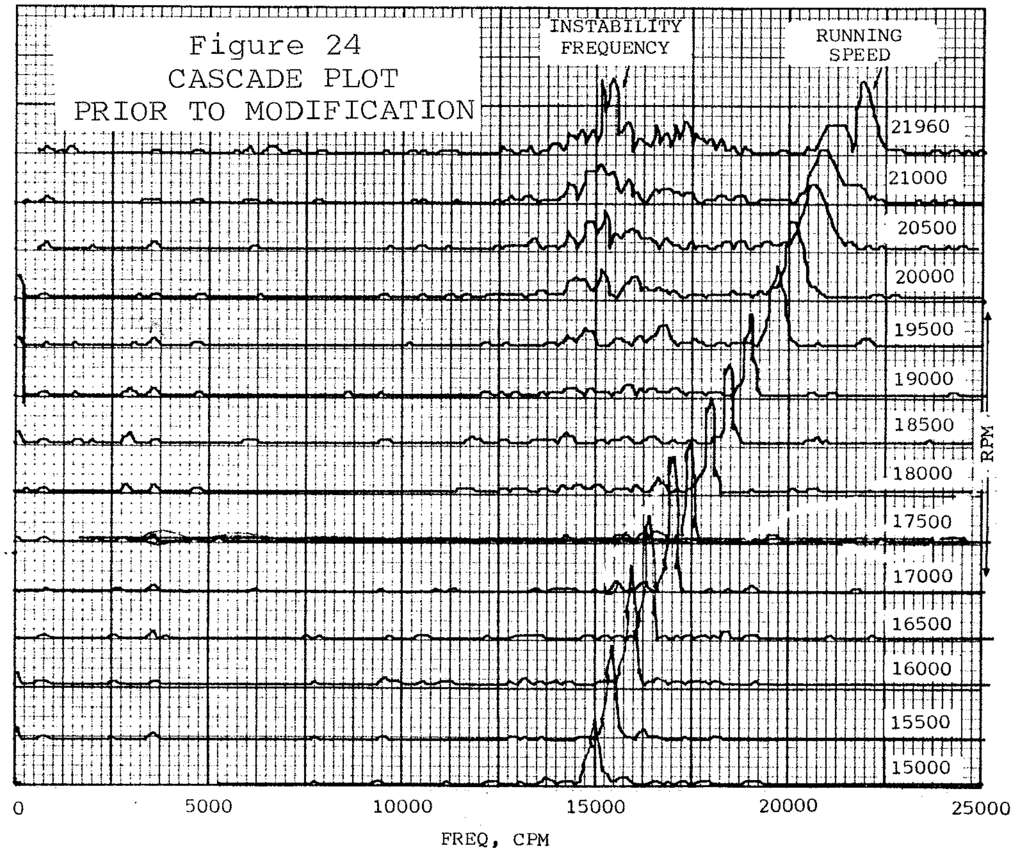
Note: (1) During recycle the shaft is offset 6 mils from the initial position at rest.



Pump Shaft Inboard
Startup and Recycle

Note: The vibration amplitude and DC position of the orbit is much less than the shaft center position.

ENGINEERING DYNAMICS INCORPORATED



ENGINEERING DYNAMICS INCORPORATED