Acoustic Pulsation Problems in Compressors and Pumps

By
J. C. Wachel
Engineering Dynamics Inc.
San Antonio, Texas

1. Introduction

It is well recognized in industry that a majority of vibration problems are induced by internal fluid pulsations. There is sometimes a tendency to attribute these pulsations solely to the action of pumps or compressors within the piping systems, and to direct remedial steps either toward modifying the pump internals or providing more effective pipe support. While each of these approaches is valid and useful, an alternate or complementary approach lies in modifying the pulsation response of the piping. When pulsation frequencies from a reciprocating pump, for example, coincide with a mechanical vibration resonance of the piping, the process of mismatching the driving force from the normal vibration response modes is quite effective. In addition, pulsation resonant frequencies within the piping can also amplify pulsation effects to the point that mechanical fatigue failures occur in the piping or pump components. Thus, successful treatment of resulting vibration problems can often be achieved merely by mismatching these pulsation resonances from major pump frequencies, and by providing acoustic filters to attenuate pulsation amplitudes over a wide range of frequencies. The acoustic filter approach is particularly applicable to variable speed pump installations where operational flexibility dictates operation over a range of speeds and pulsation frequencies.

Similar pulsation problems exist with centrifugal machinery, but influence of the piping is often even more important in defining both pulsation frequencies and amplitudes. Pulsation problems with reciprocating pumps can now be adequately dealt with, and reliable techniques for predicting and solving such problems have been previously described in the technical literature. Therefore, this paper will deal chiefly with centrifugal pump and compressor pulsation problems.

By far the most difficult problem in analytically describing plant dynamics problems lies in predicting driving forces from the internal flow or pulsation forces which excite the vibrations. These pulsation driving forces are a function of both the pulsation response of the piping system and the source strength and frequencies of the pulsation generating mechanisms.

2. Piping System Pulsation Response

2.1 Piping Acoustics

A piping system employing various discrete lengths, diameters, branches, constrictions, etc., constitutes a complex flow impedance network incorporating dynamic transfer characteristics which will amplify some pulsation frequencies and attenuate others. In flowing piping systems, the acoustic resistance is a direct function of the ρV^2 pipe frictional losses. These far overshadow conventional acoustic dissipation effects such as molecular relaxations for the range of flows normally encountered in industrial piping systems. This distributed piping system, therefore, can amplify low level pulsation up to as much as a factor of 100 depending on frequency, pipe size, flow, termination impedance, etc.

Since the acoustical pulsation characteristics of typical pump and compressor piping system are linear, conventional acoustic theory can be used to define resonant frequencies, mode shapes and filtering characteristics. The non-linear effects of damping only affect the amplitude or sharpness of the resonance peaks. Using elementary physics, resonances of individual piping segments can be very simply described from "organ pipe" resonance theory. The standing wave (pressure) mode shapes for various piping resonances are shown in Figure 1. It will be recalled that such resonance modes are characterized by:

- pressure maxima at closed ends
- velocity minima at closed ends
- pressure minima at open ends
- velocity maxima at open ends.

Since the basic equation defining wave length (λ), pipe length (L) and acoustic velocity (a) is $a = f\lambda$, resonant frequencies for the various modes of the three configurations can be conveniently calculated from the following equations:

Half-Wave Resonance (Pipe closed at both ends or open at both ends)

$$f = N \frac{a}{2L} \text{ where } N = 1, 2, 3 \ldots = \text{ harmonic} \tag{1}$$

Quarter-Wave Resonance (Pipe open at one end and closed at the other)

$$f = (2N - 1) \frac{a}{4L} \text{ where } N = 1, 2, 3 \ldots \tag{2}$$

When piping components are coupled together, more complex resonances can exist (i.e., of several interacting segments) in addition to the pure component resonances, and one component can be situated to detune a resonance within another (e.g., a quarter-wave stub or side branch Helmholtz resonator). These complex system interaction characteristics can best be analyzed by solving the acoustic equations of motion using a digital computer solution.

3. Vibration Excitation Sources

Piping and machinery vibrations are often excited by pulsation forces inside the piping/machine or, secondarily, by mechanical excitation from machinery unbalanced forces and moments at one and two times the running speed. Potential excitation sources are included in the following list and are also summarized in Figure 2.

- Mechanical energy from machinery unbalanced forces and moments
- Pulsations generated by reciprocating compressors and pumps
- Pulsations generated by centrifugal compressors and pumps
- Pulsations generated by flow through or across objects
- Pulsations generated by pressure drop at restrictions
- Pulsations generated by cavitation and flashing
- Pulsations generated by waterhammer and surge

3.1 Pulsation Generating Mechanisms

3.1.1 Reciprocating Compressors and Pumps

The intermittent flow of a fluid through compressor or pump cylinder valves generates fluid pulsations which are related to a number of parameters, including operating pressures and temperatures, horsepower, capacity, pressure ratio, clearance volumes, phasing between cylinders, fluid thermodynamic properties, and cylinder and valve design. Pulsations are generated at discrete frequency components corresponding to the multiples of operating speed.

The pulsation amplitudes depend on the magnitude of the pulsation generated and the reflected amplitudes of the frequency components as they interact with the acoustical resonances in the system.

Pulsation amplitudes can be predicted by modeling the acoustic characteristics of the piping, the pulsations generated by the compressor or pump and the interaction of the two. Digital and analog simulation techniques have been developed to model the piping and the pulsation generating characteristics of compressor and pump systems. The analog technique, which was developed in the 1950's solves the differential equations by building electrical models of the piping and the compressors and pumps. In the digital technique, the differential equations of the acoustic phenomena are solved with complex matrix algorithms using modern high speed computers.

3.1.2 Centrifugal Compressors and Pumps

Pulsation amplitudes generated by centrifugal machines generally occur at one times running speed and blade passing frequency and their multiples. They are a function of the radial vibrations, the radial impeller clearance, seal and wear ring clearances, the symmetry of the impeller, diffuser and case, and the volute characteristics. As operating conditions deviate from the design or best efficiency point, a variety of secondary flow patterns may produce additional pressure fluctuations.

Significant low frequency pulsations can also be produced as a result of dynamic interaction of the acoustical response of the piping, the head-flow curve of the unit, the dynamic flow damping, and the location of the unit in the piping geometry.

3.1.3 Flow Through or Across Objects

Flow through a restriction or past an obstruction or restriction in the piping may produce turbulence or flow-induced pulsations. These flow generated pulsations (commonly called Strouhal excitation) produce noise and vibration at frequencies which are related to the flow velocity and geometry of the obstruction.

The acoustical modes of a piping system and the location of the turbulent excitation have a strong influence on the frequency and amplitude of the vortex shedding. The frequencies generated by the turbulent energy are centered around a frequency which can be determined by the following equation:

$$f_s = \frac{S_n V}{D} \tag{3}$$

where

f_s = Strouhal vortex frequency, Hz
S_n = Strouhal number, dimensionless (0.2 to 0.5)
V = Flow velocity in the pipe, ft/sec
D = Characteristic dimension of the obstruction, ft

For flow over tubes, D is the tube diameter, and for excitation by flow past a branch pipe, D is the diameter of the branch pipe.

3.1.4 Pressure Drop Through Restrictions

Pressure regulators, flow control valves, relief valves, and pressure letdown valves produce pulsations (noise) associated with turbulence and flow separation, and the relatively broad band frequency spectrum is characteristically centered around a frequency corresponding to a Strouhal number of approximately 0.2.

3.1.5 Cavitation and Flashing

Flashing and cavitation can occur in the low pressure region of liquid system pressure control valves when the pressure drops below the vapor pressure. When cavitation occurs, a gas bubble is formed and moves with the flow. As the pressure increases, the pressure rises above the vapor pressure, the gas bubble collapses, and a high amplitude shock pulse results in the fluid.

To avoid flashing after a restriction, sufficient back pressure should be provided by taking pressure drop at several locations. Alternately, the restriction could be located near an open end so that the flashing energy can dissipate into a larger volume.

3.1.6 Hydraulic Waterhammer and Surge

Starting and stopping pumps with the attendant fast opening and closing of valves is a major cause of severe transient pressure surges in piping systems. Increasing the closure time of valves can reduce the severity of the surge pressure. Methods are available to evaluate the severity of waterhammer in a particular piping configuration for various closure rates.

Centrifugal compressors and pumps can sometimes surge when they are operating at a low flow, high-head condition. The flow-versus-head curve can actually cause backflow to occur and significant pulsations can be generated which are a function of the piping acoustical natural frequencies and the overall impedance characteristics.

3.2 Coupling Mechanisms

For vibrations to occur, there must be an energy generating source plus a coupling mechanism to convert the pressure forces into shaking forces. Therefore, in evaluating the piping vibration characteristics of an installation, it is essential to understand the coupling mechanisms which cause shaking forces to occur in the piping system.

Pressure pulsations couple to produce shaking forces at piping bends, closed ends of vessels and headers, discontinuities or changes in the piping diameters and at restrictions, such as orifices, valves, and reducers. In a continuous straight pipe of constant diameter, pulsations will not produce a significant vibration excitation force.

Thus for normal piping systems, the most common coupling point is the piping bend. The shaking force acting on a bend results from the change in momentum due to the change in direction. An indication of the actual shaking force at a bend associated with a given pulsation in a piping run can be computed by considering the acoustic system as a conservative system (the maximum kinetic energy is equal to the maximum potential energy and the total dynamic energy is the same at any point in the given piping run). The magnitude of the shaking force at any frequency component can be shown to be $(2\rho A \cos \frac{\theta}{2})$ where ρ is the maximum pressure pulsation amplitude at that frequency in the piping run, A is the area of the pipe, and θ is the angle between the legs of the bend. Thus, for a 90 degree bend, the shaking force would be 1.414 ρA bisecting the angle, Figure 3.

Another mechanism of an acoustical energy coupling to provide shaking forces occurs for manifold bottles, capped headers, filters, etc. In such elements, the acoustic forces present at each end and at each baffle or restriction, if any, must be considered to determine the total resultant unbalanced force. Thus, not only differences in pulsation amplitude, but also in phasing and the areas on which the pulsations act must be accounted for in obtaining the vector sum for each frequency component. For example, for a simple case of an empty surge bottle or header capped on both ends, the shaking force that would be produced by a pulsation component that is equal in amplitude at each end but opposite (180 degrees) in phase would be $2\rho A$. This condition would exist for the fundamental mode of resonance for any piping component capped at both ends and also for all odd order multiples of this frequency. On the other hand, for the even order modes of resonances for a simple bottle, the dynamic pressures acting on each end of the bottle will be in phase (0 degrees) with each other such that there will be no net resulting shaking force. Figure 4 gives an illustration of shaking forces in a bottle in the first four modes of resonance.

Thus, a knowledge of the pulsations that can exist in a piping system and the mechanism by which shaking forces can result from these pulsations are essential in evaluating the piping vibration characteristics of an installation. It should be re-emphasized that the most significant factor controlling the interactions of a piping system is the coupling of the system excitation forces with the resonances of the system, both acoustical and mechanical.

3.3 Centrifugal Machinery Response to External Pulsations

One of the more interesting phenomena associated with centrifugal pumps and compressors is their ability to either amplify or attenuate pulsations introduced into the piping from an external source such as a nearby reciprocating machine. Such phenomena have been documented both in the laboratory and in industrial plants for gas compressors and strongly evidenced in a variety of liquid pump problems.

Pulsations from a reciprocating pump or compressor are normally fixed in frequency at the reciprocating pump shaft speed and multiples thereof. Pulsation amplitudes are strongly influenced by the dynamic acoustic response of the attached piping system, and if a harmonic of pump speed corresponds to a strong normal mode acoustic resonance of the piping system, severe pulsations can result. Normally, when such pulsations are introduced into the suction or discharge of a centrifugal pump, they will be conducted through the impeller, and standing wave pulsation resonances can exist throughout the pump piping system. Further, under certain design conditions, the compressor can serve to amplify the pulsations to a higher level than would occur in a purely passive piping system.

3.4 Centrifugal Machinery Response to Flow-Induced Pulsations

One type of pulsation that still defies complete quantitative analysis or simulation is that involving flow-induced pulsations. Such problems are typified by flow past an obstruction, side branch or piping discontinuity which produces vortex shedding and which may be either amplified or attenuated by the acoustic properties of the piping network and by interaction of a centrifugal machine with the piping system. For example, the pressure pulsation caused by variation in the flow at the blade passing frequency can be as high as 3-6 percent for some liquid centrifugal pumps.

Action of centrifugal machines in amplifying these relatively low level pulsations up to significant and even destructive levels has been well documented in industrial plant systems, and experimental work in the laboratory has served to define many of the phenomena involved. Field data on a broad spectrum of centrifugal machines has shown strong pulsations at frequencies which, typically, are not harmonically related to (and do not vary with) rotor speed. Investigation has revealed that, in most cases, the problem is not caused simply by pump or compressor characteristics, but instead is the result of dynamic interaction of the passive response of the piping and pump characteristics. The action of the pump or compressor in causing amplification or attenuation of this vortex energy is quite complex, but basically is dependent upon:

1. The head curve slope and operating point.
2. System flow damping in the piping.
3. The existence of strong reactive resonances in the piping, particularly if they coincide with vortex frequencies.
4. The location of the pump or compressor in the standing wave field (i.e., at a velocity maximum rather than a pressure maximum).

Pulsations can build to extremely high levels and result in a variety of machine and piping problems such as destruction of machine internals, large torsional reactions, initiation of cavitation, large force reactions at valves, bends, high noise, etc. Amplitude, of course, is flow dependent and subject to coincidence of various resonances. Thus, pulsations may come and go as conditions change. Detailed testing of the mode shapes of pulsation standing waves both in the field and laboratory show that the resonances observed are characteristic of the normal length resonances of the coupled piping components, and that the pulsation initiating source normally involves flow vortex formation at piping discontinuities (junctions, obstructions, etc.). When the frequency of this vortex formation excites one of the acoustic resonances of the piping system, and when the compressor/pump is situated near a velocity maximum in this resonant system, high amplitude, self-sustaining pulsations can result. Particular resonances can often be destroyed by moving the compressor to a velocity minimum, but often a new pulsation frequency will be generated such that the compressor is again situated near a velocity maximum for this higher mode oscillation. With proper care and detailed analysis of the relative strength of the various pulsation resonance modes of the piping, piping designs can be developed to avoid these strong resonances. Further, by controlling piping stub lengths (e.g., in by-pass piping, etc.) such that their quarter-wave stub resonances are far removed from the preferential vortex frequencies, the basic generation mechanisms can be substantially suppressed.

Several case histories involving pulsation problems in centrifugal pumps and compressors are discussed below.

4. Case 1: Failure of Pump Internals

A four stage centrifugal pump suffered repeated failures of the splitter between the pump stages. A detailed field study revealed that the cause of the problems was an acoustic resonance of the long crossover which connected the second stage discharge with the third stage suction (Figure 1.1). The resonant frequency was a half-wave acoustical resonance.

$$f = \frac{C}{2L} \tag{4}$$

where

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

The speed of sound in water is a function of the temperature and at 310°F, the speed of sound was 4770 ft/sec. The length of the crossover was 5.75 ft. The acoustical natural frequency is

$$f = \frac{4770}{(2)(5.75)} = 415 \text{ Hz} \tag{5}$$

The acoustical natural frequency was excited by the blade passage frequency (7 times running speed). Coincidence occurs at $(415)(60)/7 = 3560$ rpm.

Pulsations measured in the crossover showed pulsation amplitudes of of 100 psi p-p. These pulsations were attenuated to the point that at the suction and discharge flanges, the amplitudes were less than 10 psi p–p.

Since the severe vibrations occurred when the speed was 3560 rpm for a water temperature of 310°F, several possible changes could be made to eliminate the possibility of coincidence of resonances during normal pump operation. One possible change would be to reduce the diameter of the impellers and operate the pump at a higher speed. Another possible change would be to change to 6 or 8 blades to change the blade passage frequency.

The impeller diameter change was the quickest and was carried out in the field, and the failures were eliminated when the excitation frequency did not coincide with the acoustical natural frequency.

5. Case 2: Piping Vibration Excited by Blade Passing Frequency

Three single stage centrifugal pumps driven by a variable speed 1100 HP turbine at 4800–6000 rpm were used on a main oil line from an offshore platform. All of these pumps experienced major problems with noise and piping vibrations. Fatigue failures were experienced on the small diameter pipework, valves, and instrumentation lines. All the pumps also experienced failures of the seals which caused oil leaks. The noise levels were extremely high requiring both ear plugs and muffs to work in the area.

Vibration measurements made on the discharge piping with a velocity transducer showed that the amplitudes were in excess of 0.85 ips, with the major frequency component being at the blade passage frequency which was 5 times running speed (5 blade impeller). The vibrations were measured with an accelerometer and the speed of the pump was varied from 3600 to 6000 rpm (Figure 2.1). This data gives the vibrations every 2.25 seconds as the speed is increased. The vibration data indicated that there were resonances at 360, 420 and 450 Hz, with maximum amplitudes of approximately 30 g's peak to peak.

The pulsations in the discharge piping of the pump were measured and are plotted versus speed in Figure 2.2. It can be seen that the pulsation energy for exciting the piping vibrations comes from the component at the blade passing frequency (BPF) at 5 times running speed. The acoustical resonances occurred at 420 and 460 Hz and pulsation amplitudes of 150 psi peak to peak were measured. The discharge pressure was 550 psig; therefore, the pulsations were significant. The noise was measured and it can be seen that the highest energy was at the same frequencies as the vibration (Figure 2.3).

To define the maximum amplitude of the pulsations, it is possible to track individual orders of running speed with a computer-based data acquisition system. The acoustical resonances can be clearly seen by tracking the blade passing frequency (fifth order) pulsation data for a speed range of 4500 to 6000 rpm as shown in Figure 2.4. The maximum amplitude was 180 psi peak to peak at 455 Hz. The amplification factor can be determined using the amplification bandwidth method. (The amplification factor is equal to the natural frequency divided by the bandwidth, where the bandwidth is defined as the width of the frequency at the 0.707 times the maximum amplitude.) Using this method, the amplification factor was 45 for the pulsations at 415 Hz and was 35 for the 455 Hz resonance.

Since the piping vibration problems were caused by pulsation resonances, it was decided to try an acoustical modification in the field to determine if the acoustical system could be changed. An orifice plate was installed in the discharge flange of the pump and data obtained without and with the orifice plate. As can be seen from Figure 2.5, the acoustical resistance of the orifice plate reduced the pulsations from 480 psi at 375 Hz to 105 psi at 335 Hz. For the pulsation resonance at 450 Hz, the amplitude was reduced from 140 to 50 psi. The orifice plate worked better on the lower frequency mode since it was associated with the piping, whereas the 450 Hz mode appeared to be associated with the pump internals. Note that, at this operating condition, the pressure pulsations were near 500 psi at the lower frequency resonance.

The orifice plate selected was approximately one-half the pipe diameter of 6 inches. This 0.5 ratio is a good starting point to make an acoustical change in a system. Unfortunately, the orifice plate had a high pressure drop and could not be used as a permanent solution.

The interference diagram in Figure 2.6 illustrates the range of the acoustical natural frequencies and the excitation energy at 5 times running speed. Another possible solution that is evident to change the impeller from 5 blades to 7 blades. This would move the blade passing frequency out of the range of the acoustical natural frequencies for the operating speed range. The impeller modification was selected since it would not require extensive piping modifications and would only involve an impeller changeout. When the 7 blade impeller was installed, the piping and pump vibrations were significantly reduced and the failures eliminated. The pump vibration data after the impeller was changed are tabulated in Figure 2.7.

6. Case History 3

Several centrifugal pumps (Figures 3.1-3.2) had high noise levels and high vibrations. EDI was asked to measure the vibrations, noise and pulsations to determine the cause of the excessive noise and make recommendations to alleviate the problems.

The centrifugal pumps had a double-suction, 7 vane impeller as shown in Figure 3.3. Pump 1 ran at 4160 rpm and the speed of pump 2 varied from 3320 rpm to 4240 rpm. The pulsation data taken in the suction and discharge piping is given in Figures 3.4 and 3.5. The noise level is given in Figure 3.6. The discharge pulsations are given as a function of pump speed in Figure 3.7. The data indicated that the piping shell wall had mechanical natural frequencies in the 2 times blade passing frequency range which amplified the pulsation energy in this frequency range and contributed to the high noise.

The data shows that the maximum pulsations, noise and vibrations occurred at the blade passing frequency and its multiples. The maximum amplitudes were at the two times blade passing frequency (14X). Over 300 psi p-p was measured at 4050 rpm at a frequency of 940 Hz.

Since the maximum energy was at multiples of the blade passing frequencies, one way to reduce the energy is to minimize the interruption of flow as the vanes pass the cutwater. One possible way this can be accomplished is by staggering the vanes on the impellers as shown in Figure 3.7. Impellers with the vanes staggered about one-third were installed and tested in June 1992. These results are given in Figures 3.8-3.10. The data shows that the pressure pulsation levels were reduced significantly and the noise levels were reduced by 5 dB, with nearly 20 dB reduction of the energy at 2 times blade passing frequency (94 dB to 74 dB).

7. Case History 4

A Dehydration system operates with two systems in operation and one unit in regeneration, Figures 4.1-4.3. The regeneration cycle was 400 minutes long, consisting of 20 minutes of depressurization, 180 minutes of heating, 180 minutes of cooling and 20 minutes of depressurizing. During the heating cycle, the temperatures of the Regen gas to the cooler increases from approximately 100°F to 530°F.

High vibrations of the valves and the actuators occurred during the regeneration process and caused repeated fatigue failures of the instrumentation tubing, valve components, gages, etc. Since the fluid is a hydrocarbon at approximately 500 °F, the situation was very dangerous. The vibrations were at 20–30 Hz with no known excitation sources in this frequency range.

The piping was modified by the user in an attempt to eliminate and/or reduce the severity of the problem. The modification changed one section of the piping from 8″ to 14″. The modification did not work and further failures were experienced.

EDI was requested to investigate the problems and make recommendations to solve the problem. EDI's Don Smith and Phil Grazier performed a field test on May 1–7, 1992. As a result of the testing and the analysis, it was found that the excessive vibration and noise levels were primarily due to the low level pulsation in the Regen gas outlet piping and attached stub piping. The maximum pulsation levels occurred during the heating cycle of unit C, Figure 4.4. The pulsation levels were not considered to be excessive with regard to shaking forces.

The vibration levels measured on the major larger diameter piping (Figure 4.5) were considered to be satisfactory and should not result in fatigue failure. However, the vibration levels measured on the valve instrument panels (Figure 4.6) were considered to be excessive and resulted in fatigue of the instrumentation and attached tubing, and loosening of the bolts.

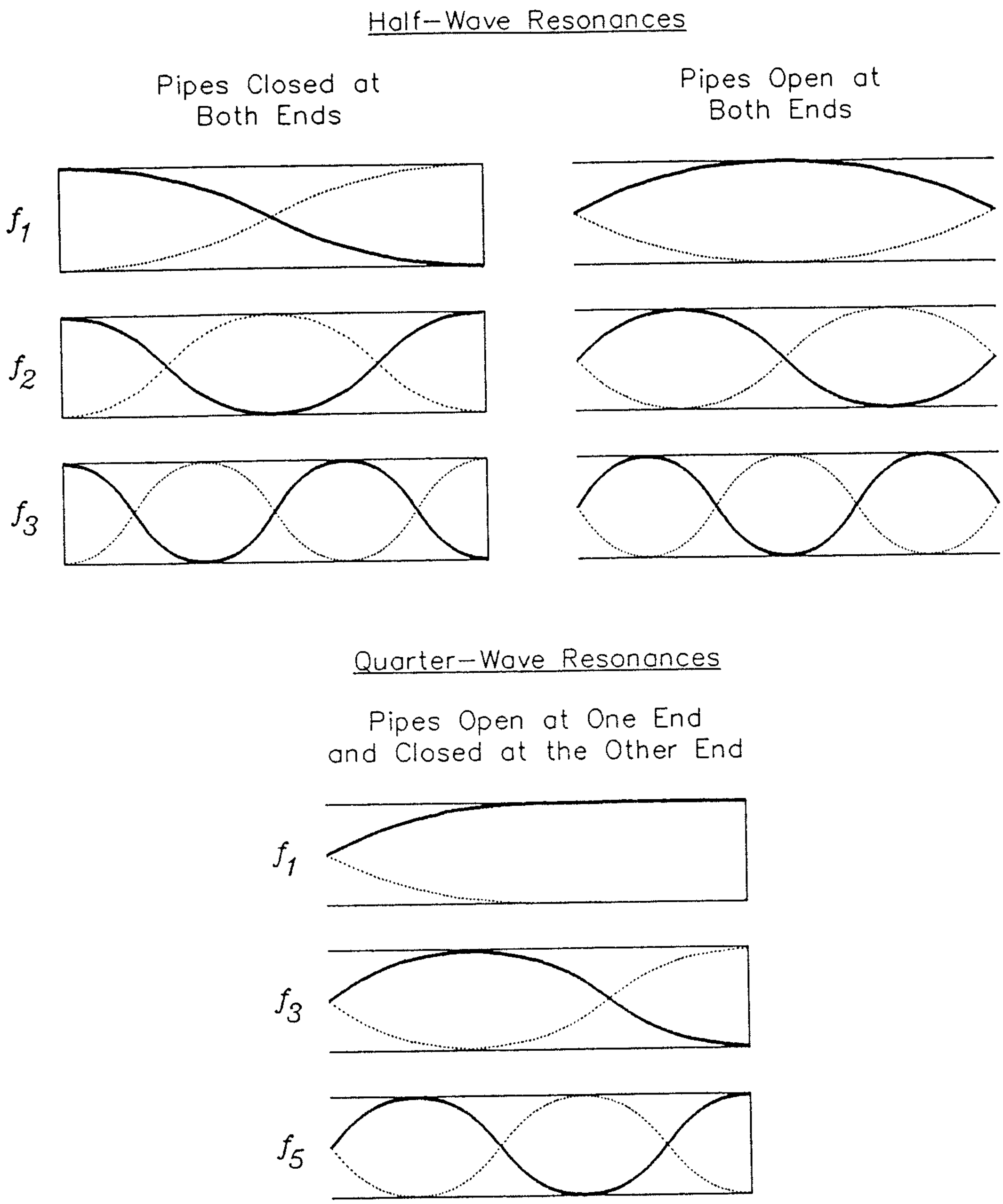
The pure tone pulsations between 25 and 30 Hz were due to the coincidence of the Strouhal frequency of the vortices generated at the openings of the side branch piping to valves off the Regen outlet piping and the acoustical natural frequency of these stubs (Figures 4.7–4.8). The interesting phenomenon was that the Strouhal excitation frequency varied as the flow increased with the increase in temperature. The change in the speed of sound as the temperature increased also increased the quarter-wave stub frequency from the main pipe to the closed valve; however, the slope of the change was slightly less as can be seen from Figure 4.9. When the Strouhal excitation frequency matched the quarter-wave acoustical resonance of the side branch, the coupling of the energy with the resonance caused the pulsations to build up and the coincidence with the mechanical natural frequencies caused the excessive vibrations.

Data recorded by the user was reported to be unrepeatable. However, EDI's data showed repeatably at the same conditions; however, it was necessary to record the data over the entire regen cycle to ensure that the conditions were the same.

The major factor in the high vibrations was that there were many mechanical natural frequencies of the structure and the valve instrument panels which were near the pulsation frequencies. The fairly low pulsation levels and shaking forces caused vibrations; however, coincidence with the mechanical natural frequencies amplified the vibration levels to excessive levels.

Since the pulsation levels were low, the recommended method of solution was to brace the piping and instrumentation lines to increase the mechanical natural frequencies above the Strouhal excitation frequencies and to absorb the energy. It would have been difficult to attenuate the pulsation levels since it would have required major changes in the piping.

When these braces and supports were installed, the vibration levels were successfully attenuated and the failures eliminated. The braces not only raised the mechanical natural frequencies, but they also added damping.



Half-Wave: $f = \dfrac{N\,c}{2L}$

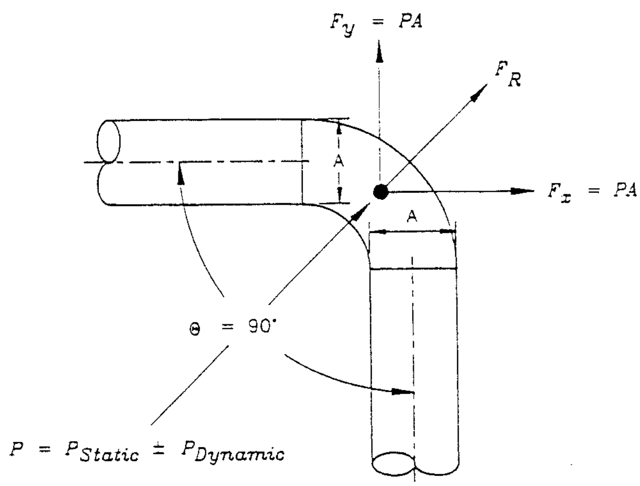
Quarter-Wave: $f = \dfrac{(2N-1)\,c}{4L}$

where
c = velocity of sound, ft/s (m/s)
L = length, ft (m)
$N = 1, 2, 3, \ldots$
f = frequency (Hz)

Figure 1: Organ Pipe Resonant Pressure Mode Shapes

Generation Mechanism	Description of Excitation Forces	Excitation Frequencies		Piping Response	Typical Problems
1. MECHANICAL INDUCED					
A. Machinery Unbalanced Forces & Moments	High Level, Low Frequency	$f_1 = \frac{1N}{60}$ $f_2 = \frac{2N}{60}$		Mechanical and/or Piping Resonance of Piping System	Foundation Resonances
B. Structure — Bourne	Low Level	$f = \frac{N}{60}$			Vents & Drains Instrumentation Lines
2. PULSATION INDUCED					
A. Reciprocating Compressors	High Pressure Pulsations, Low Frequency	$f = \frac{nN}{60}$	n = 1,2,3, ... (modes) N = Speed, rpm	Mechanical and/or Acoustic Resonance of Piping System	Piping System Fatigue Failures, Excessive Loads to Rotating Equipment, Damaged Supports/Restraints
B. Reciprocating Pumps	High Pressure Pulsations, Low Frequency	$f = \frac{nNP}{60}$	P = Number of Pump Plungers	Mechanical and/or Acoustic Resonance of Piping Systems	Cavitation on Suction Piping Fatigue Failures
C. Centrifugal Compressors & Pumps	Low Pressure Pulsations, High Frequency	$f = \frac{nN}{60}$		Complex Vibration Modes	High Acoustic Energies (Noise) Piping System Failures, Excessive Loads to Rotating Equipment, Small Branch Connection Failures
		$f = \frac{nBN}{60}$ $f = \frac{n\nu N}{60}$	B = Number of Blades ν = Number of Volutes or Diffuser Vanes		
3. GASEOUS FLOW EXCITED					
A. Flow Through Pressure Letdown Valves or Restrictions/Obstructions	High Acoustic Energy, Mid to High Broad Band Frequencies	$f = S\frac{V}{D}$	S = Strouhal Number 0.2 – 0.5 V = Flow Velocity ft/sec D = Restriction Diameter, ft.	Complex Vibration Modes in Both Longitudinal and Circumferential Directions	Fatigue Failures of Large Diameter Piping Downstream of High Capacity Pressure Letdown Valves, Small Branch Connection Failures, Flange Leakage
B. Flow Past Stubs	Moderate Acoustic Energy Mid to High Frequencies	$f = S\frac{V}{D}$	S = 0.2 – 0.5 D = Stub Diameter, ft.	Acoustic Resonance of Short Stubs	Fatigue Failure of Stub Connection to Main Run, Valve Chatter
4. LIQUID (OR MIXED PHASE) FLOW EXCITED					
A. Flow Turbulence Due to Quasi Steady Flow (e.g. Fluid Solids Lines)	Random Vibrations, Low Frequency	f = 0 – 30 Hz	(Typically)	Low Frequency Line Movements at Mechanical Natural Frequencies	Excessive Loads on Piping Supports and Restraints
B. Cavitation and Flashing	High Acoustic Energy, Mid to High Frequencies	Broad Band		Complex Vibration Modes in Both Longitudinal and Circumferential Directions	Fatigue Failures, Small Branch Connection Failures
5. PRESSURE SURGE/ HYDRAULIC HAMMER	Transient Shock Loading	Discrete Events		High Impact Loads to Piping and Restraints	Excessive Piping/Structure Loads Due to Quick Valve Closures or Rapid Pump Starts/Stops

Figure 2
Piping Vibration Excitation Sources.



A = Projected Area

$$F_{Dynamic} = P_{Dynamic}\,A = F_x = F_y$$
$$F_R = 2\,P_{Dynamic}\,A\cos(\theta/2)$$

Figure 3
Shaking Forces in Piping Bends

16117 University Oak San Antonio, TX 78249 (512) 492-9100 FAX 492-9586

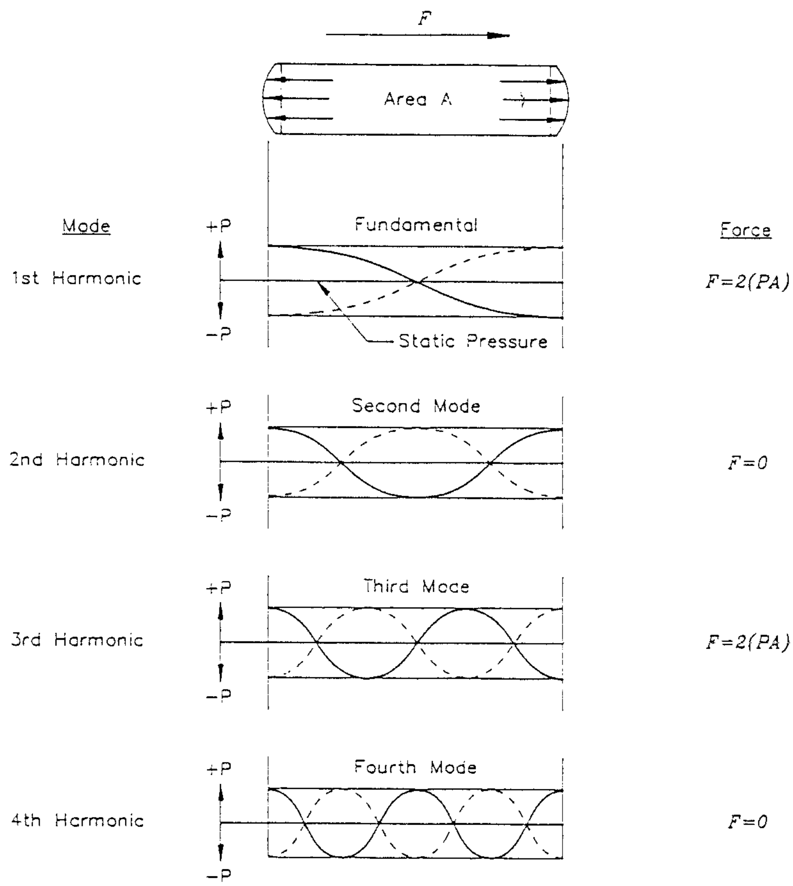
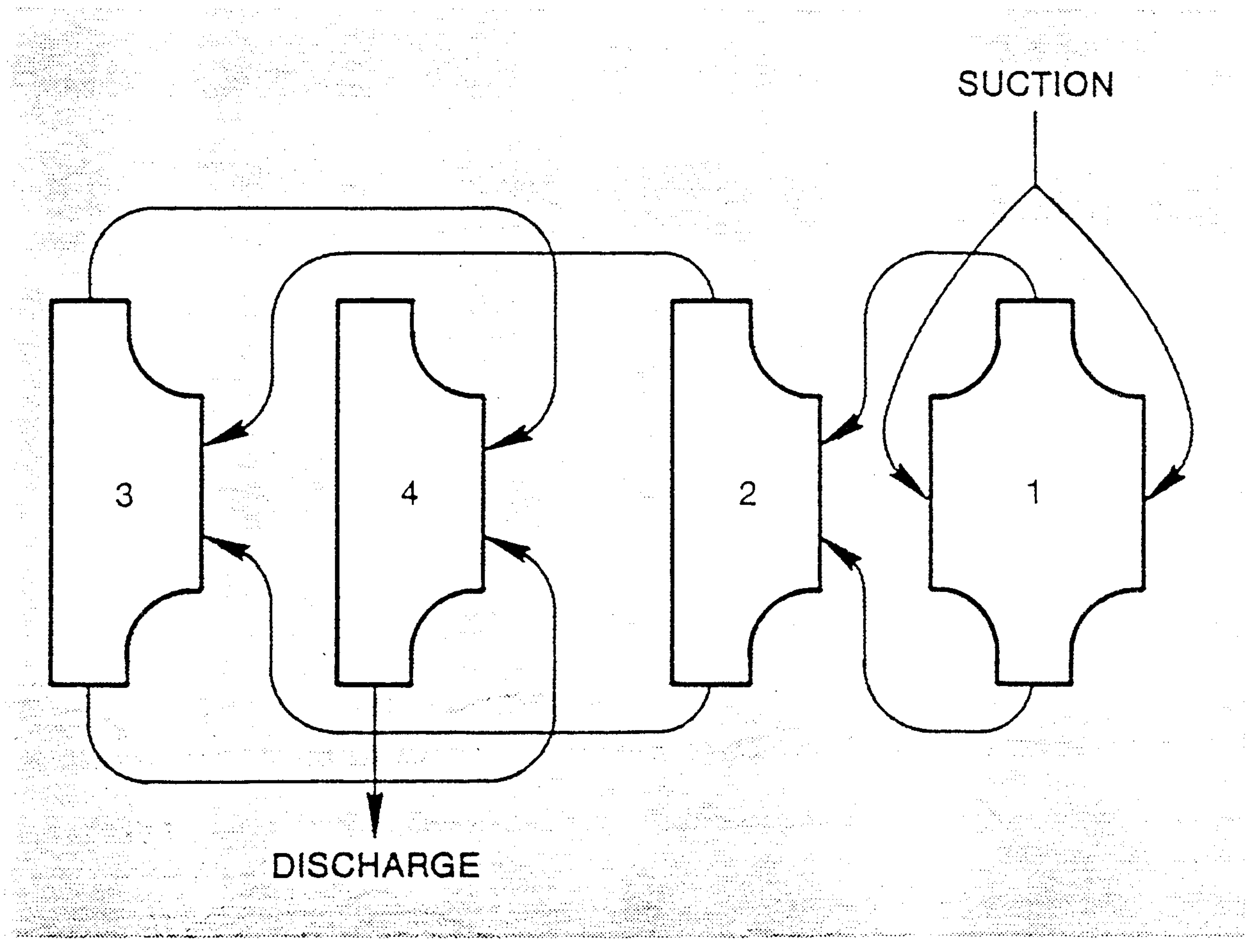
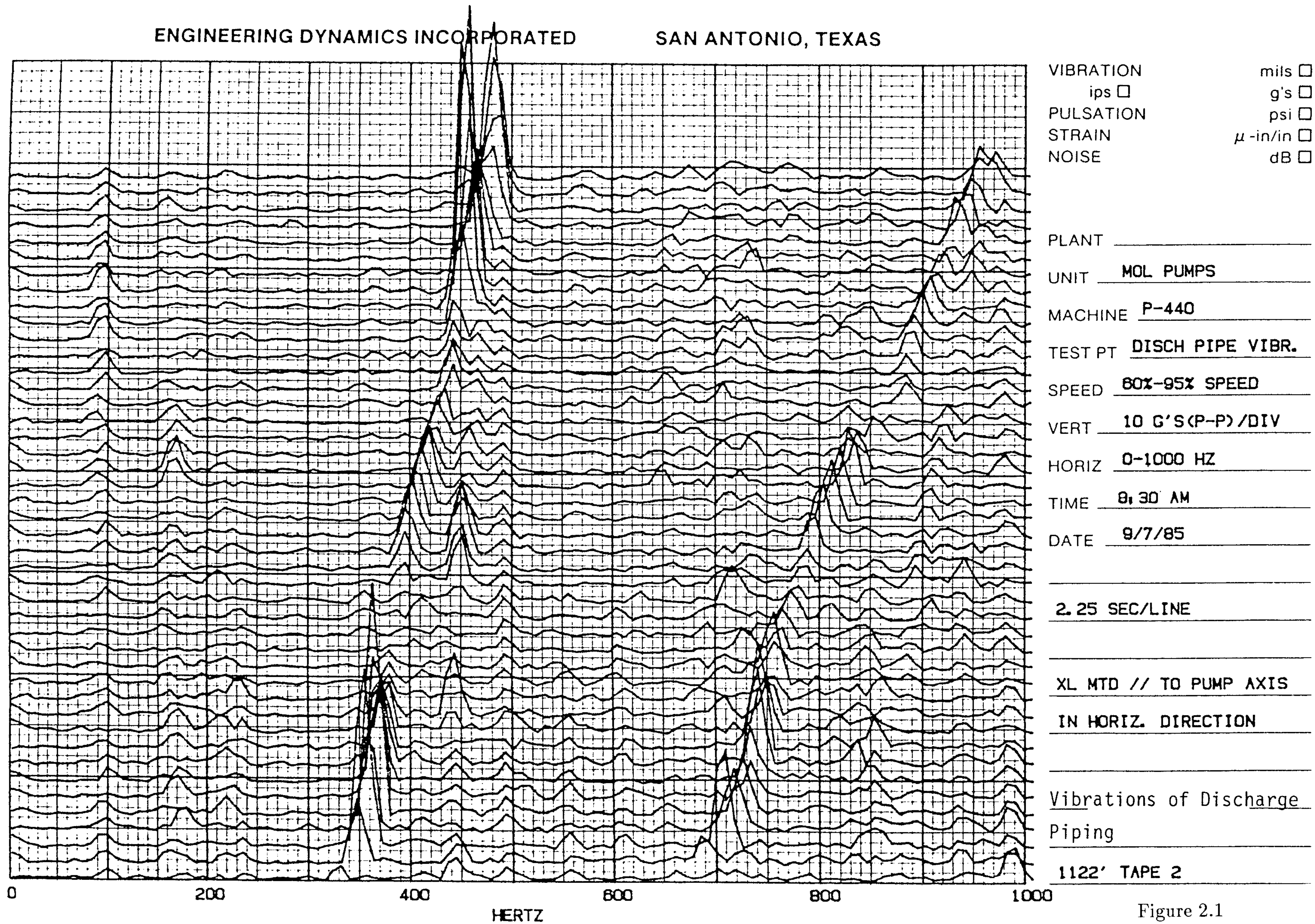


Figure 4
Phase and Amplitude of Dynamic Pressures in a Bottle



Flow schematic of a four-stage pump.

Figure 1.1


ENGINEERING DYNAMICS INCORPORATED
SAN ANTONIO, TEXAS
VIBRATION
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STRAIN
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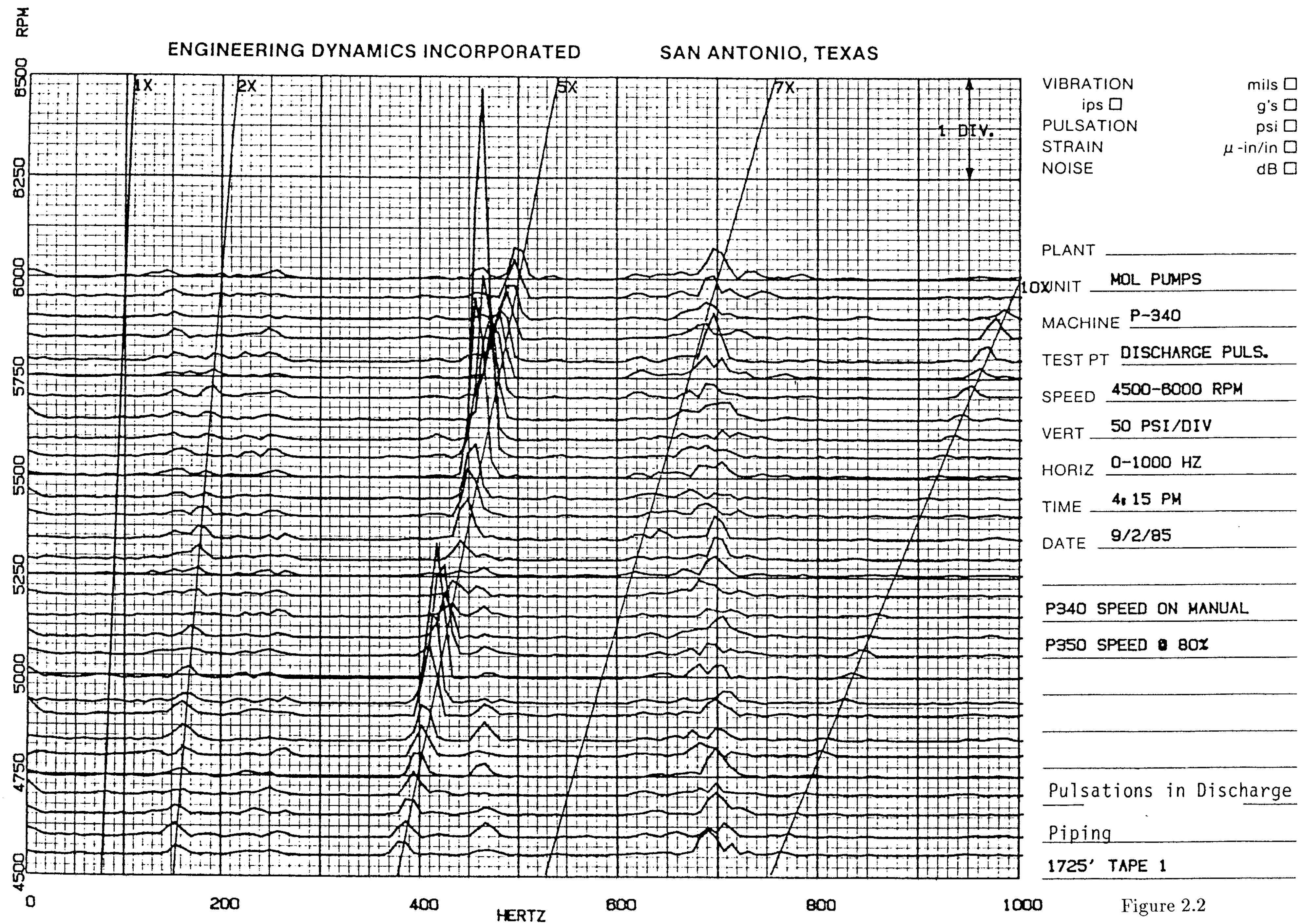
PLANT
UNIT MOL PUMPS
MACHINE P-440
TEST PT DISCH PIPE VIBR.
SPEED 80%-95% SPEED
VERT 10 G'S(P-P)/DIV
HORIZ 0-1000 HZ
TIME 8:30 AM
DATE 9/7/85
2.25 SEC/LINE
XL MTD // TO PUMP AXIS
IN HORIZ. DIRECTION
Vibrations of Discharge
Piping
1122' TAPE 2
0
200
400
600
800
1000
HERTZ

Figure 2.1



ENGINEERING DYNAMICS INCORPORATED
SAN ANTONIO, TEXAS
RPM
8500
8250
6000
5750
5500
5250
5000
4750
4500
0
200
400
600
800
1000
HERTZ
1X
2X
5X
7X
10X
1 DIV.
VIBRATION
ips
PULSATION
STRAIN
NOISE
mils
g's
psi
μ-in/in
dB
PLANT
UNIT MOL PUMPS
MACHINE P-340
TEST PT DISCHARGE PULS.
SPEED 4500-6000 RPM
VERT 50 PSI/DIV
HORIZ 0-1000 HZ
TIME 4:15 PM
DATE 9/2/85
P340 SPEED ON MANUAL
P350 SPEED @ 80%
Pulsations in Discharge
Piping
1725' TAPE 1

Figure 2.2

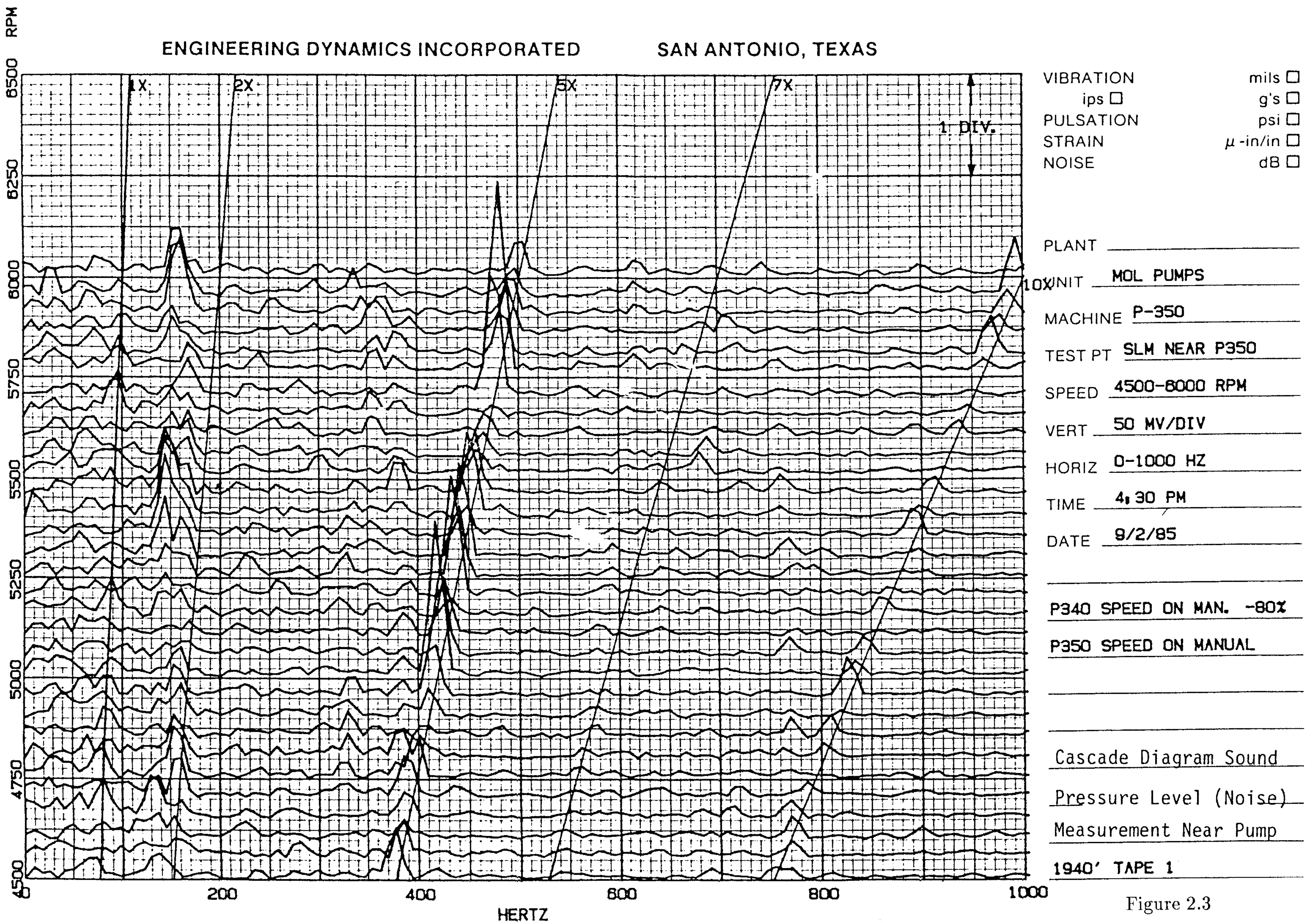
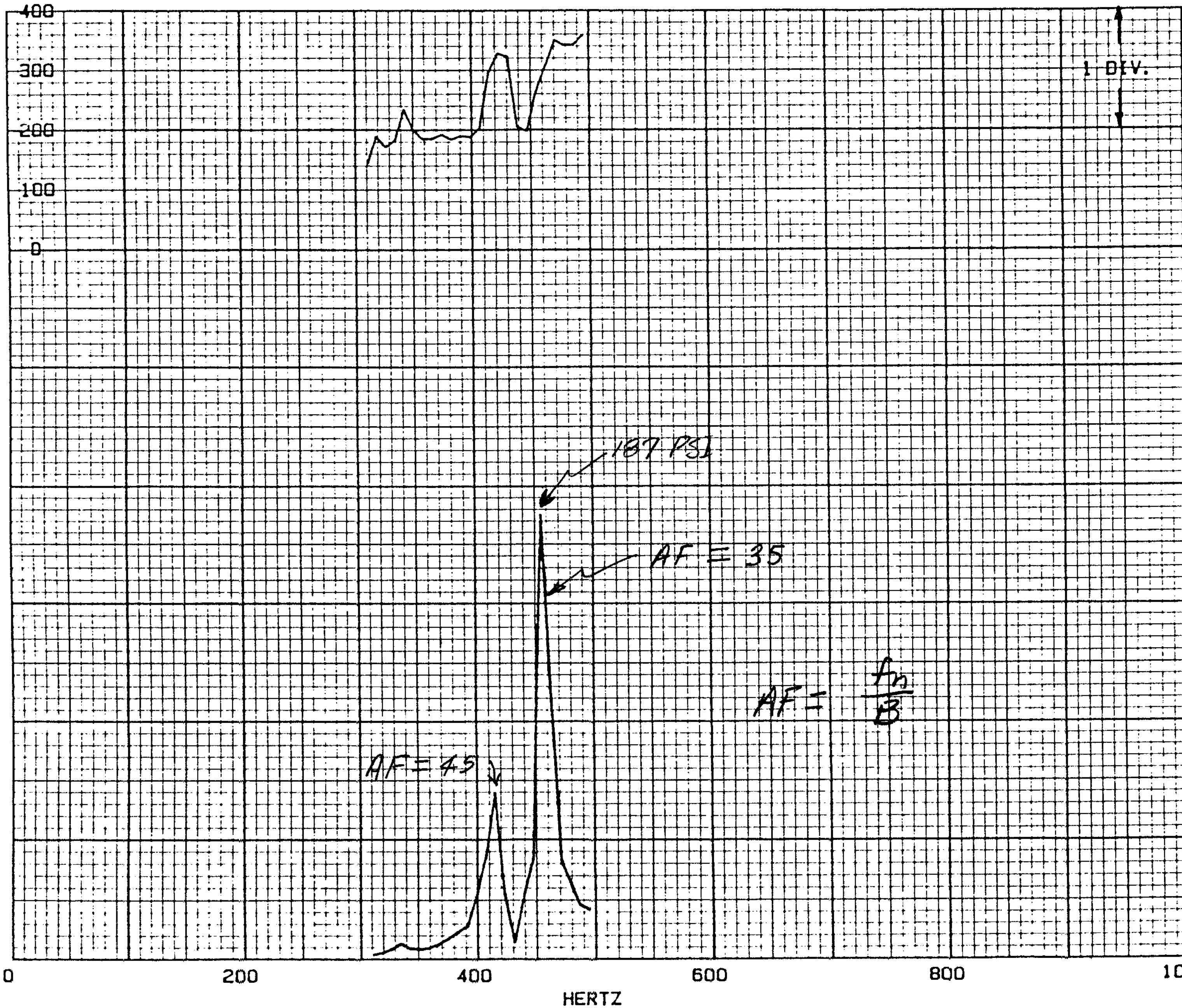

ENGINEERING DYNAMICS INCORPORATED
SAN ANTONIO, TEXAS
RPM
6500
6250
6000
5750
5500
5250
5000
4750
4500
1X
2X
5X
7X
10X
1 DIV.
0
200
400
600
800
1000
HERTZ
VIBRATION
ips
PULSATION
STRAIN
NOISE
mils
g's
psi
μ-in/in
dB
PLANT
UNIT MOL PUMPS
MACHINE P-350
TEST PT SLM NEAR P350
SPEED 4500-6000 RPM
VERT 50 MV/DIV
HORIZ 0-1000 HZ
TIME 4:30 PM
DATE 9/2/85
P340 SPEED ON MAN. -80%
P350 SPEED ON MANUAL
Cascade Diagram Sound
Pressure Level (Noise)
Measurement Near Pump
1940' TAPE 1

Figure 2.3

ENGINEERING DYNAMICS INCORPORATED SAN ANTONIO, TEXAS



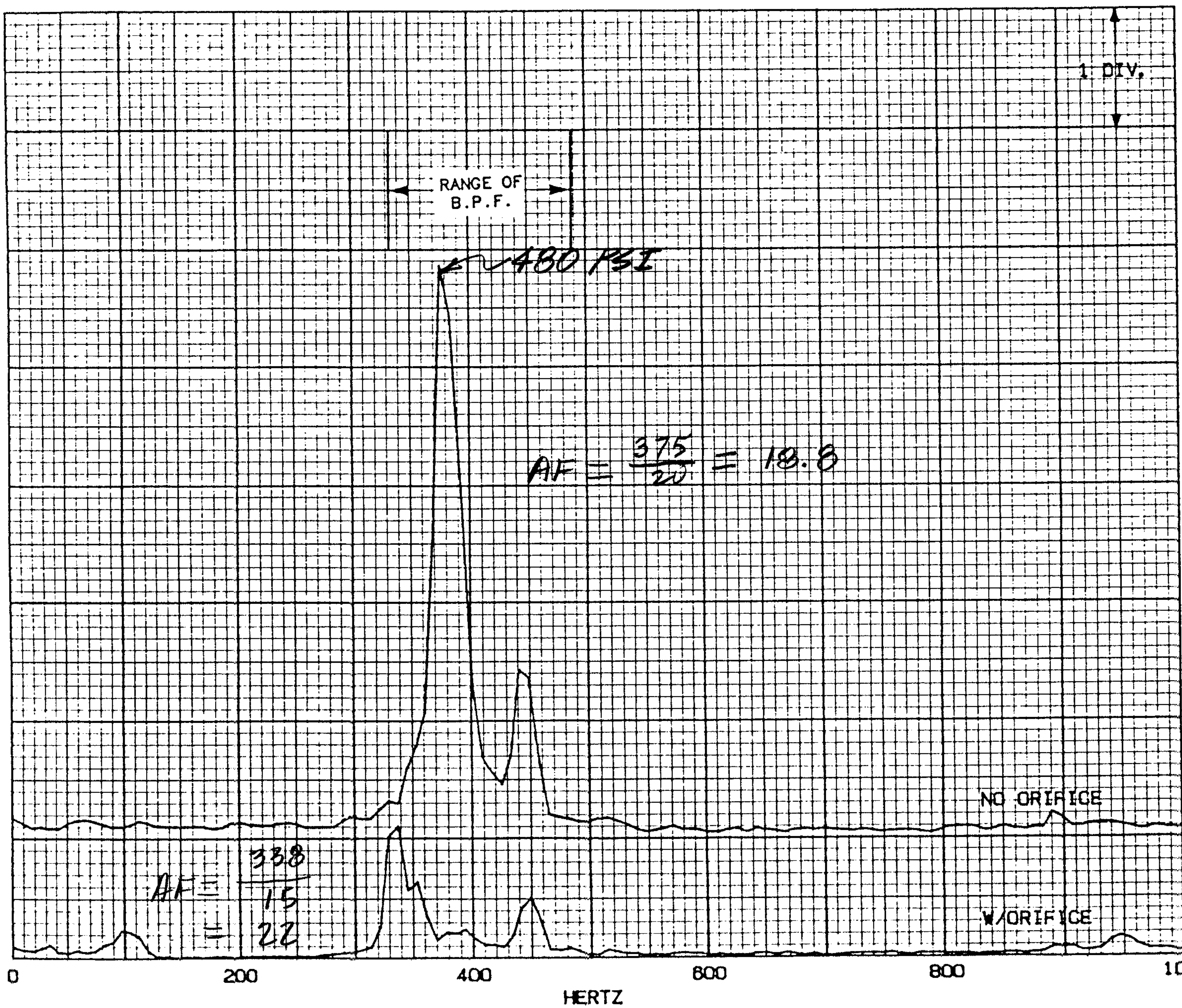
VIBRATION ☐ ips ☐ mils ☐ g's ☐
PULSATION ☐ psi ☐
STRAIN ☐ μ-in/in ☐
NOISE ☐ dB ☐

PLANT
UNIT MOL PUMPS
MACHINE P 340
TEST PT PULS DSCH
SPEED 4500-6000 RPM
VERT 50 PSI/DIV
HORIZ 0-1000 HZ
TIME 4:30 PM
DATE 9/2/85
P 350 SPEED AT 80%

Pulsations at the Blade Passing Freqency (5th order)

Figure 2.4

ENGINEERING DYNAMICS INCORPORATED SAN ANTONIO, TEXAS



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

PLANT

UNIT MOL PUMPS

MACHINE P-450

TEST PT DISCHARGE PULS.

SPEED 4000-5850 RPM

VERT 100 PSI/DIV

HORIZ 0-1000 HZ

TIME

DATE 9/6/85

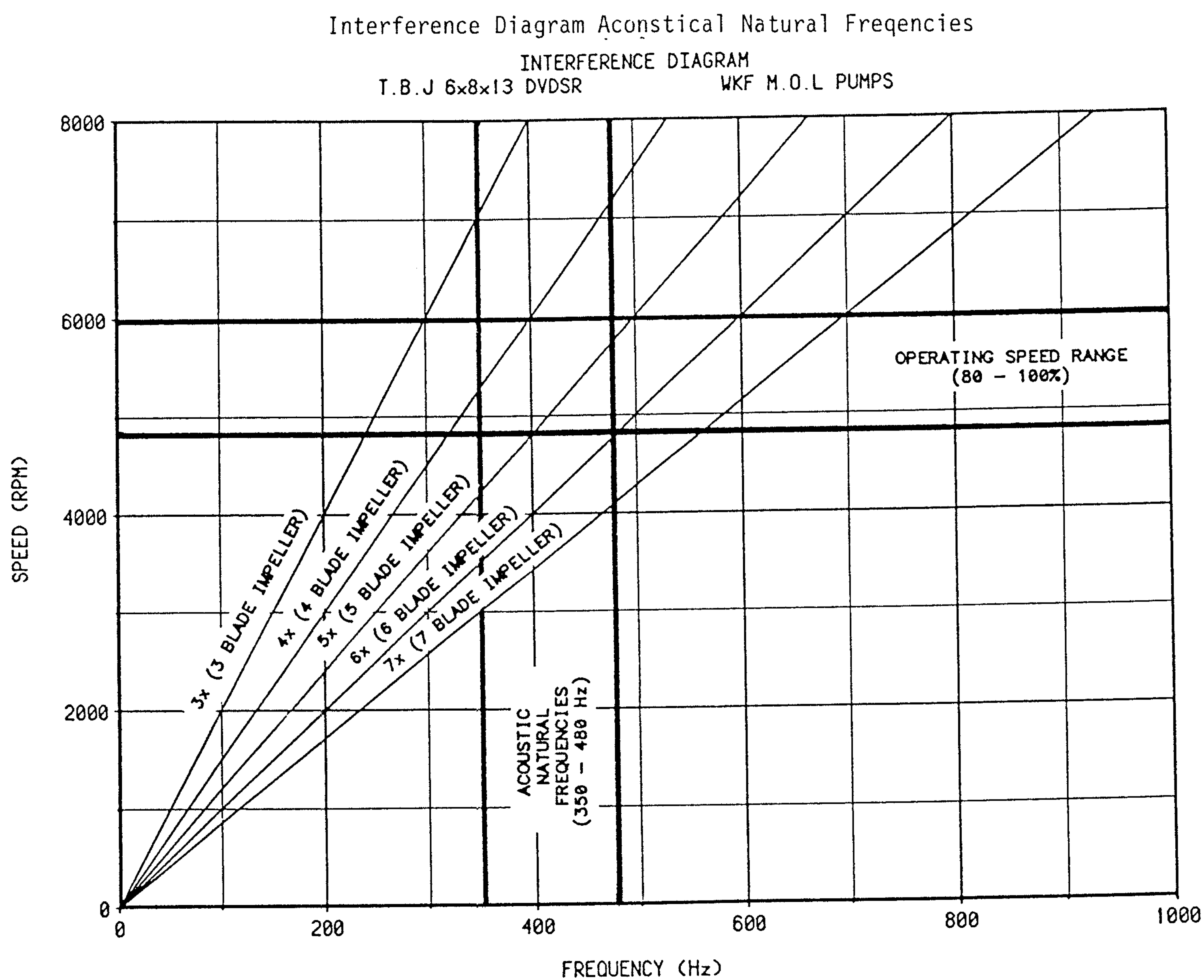
OTHER PUMPS ON AUTO

COMPARISON OF PEAK-HOLD WITH AND W/O ORIFICE

Effect of Orifice Plate on Discharge Pulsations

813' + 980' TAPE 2

Figure 2.5



Interference Diagram Aconstical Natural Freqencies
INTERFERENCE DIAGRAM
T.B.J 6x8x13 DVDSR
WKF M.O.L PUMPS
SPEED (RPM)
FREQUENCY (Hz)
0
2000
4000
6000
8000
0
200
400
600
800
1000
3x (3 BLADE IMPELLER)
4x (4 BLADE IMPELLER)
5x (5 BLADE IMPELLER)
6x (6 BLADE IMPELLER)
7x (7 BLADE IMPELLER)
ACOUSTIC
NATURAL
FREQUENCIES
(350 - 480 Hz)
OPERATING SPEED RANGE
(80 - 100%)

Figure 2.6

Figure 2.7

Pump Vibrations at Blade Passing
Frequency (7x) on Bearing Housing

Power Turbine Speed (Percentage)	Direction	Inboard (in/sec)	Outboard (in/sec)
79	X	.03	.02
	Y	—	—
	Z	—	—
84	X	.03	—
	Y	.04	—
	Z	—	.02
88	X	.06	—
	Y	.04	—
	Z	.02	.04
90	X	.11	.01
	Y	.03	—
	Z	.02	.03
93	X	.11	.02
	Y	.03	.03
	Z	.03	.09

Vibration at 1× and 2× at same speeds were also recorded and were .02–.04 with a maximum reading of .08 in/sec.

Figure 3.1

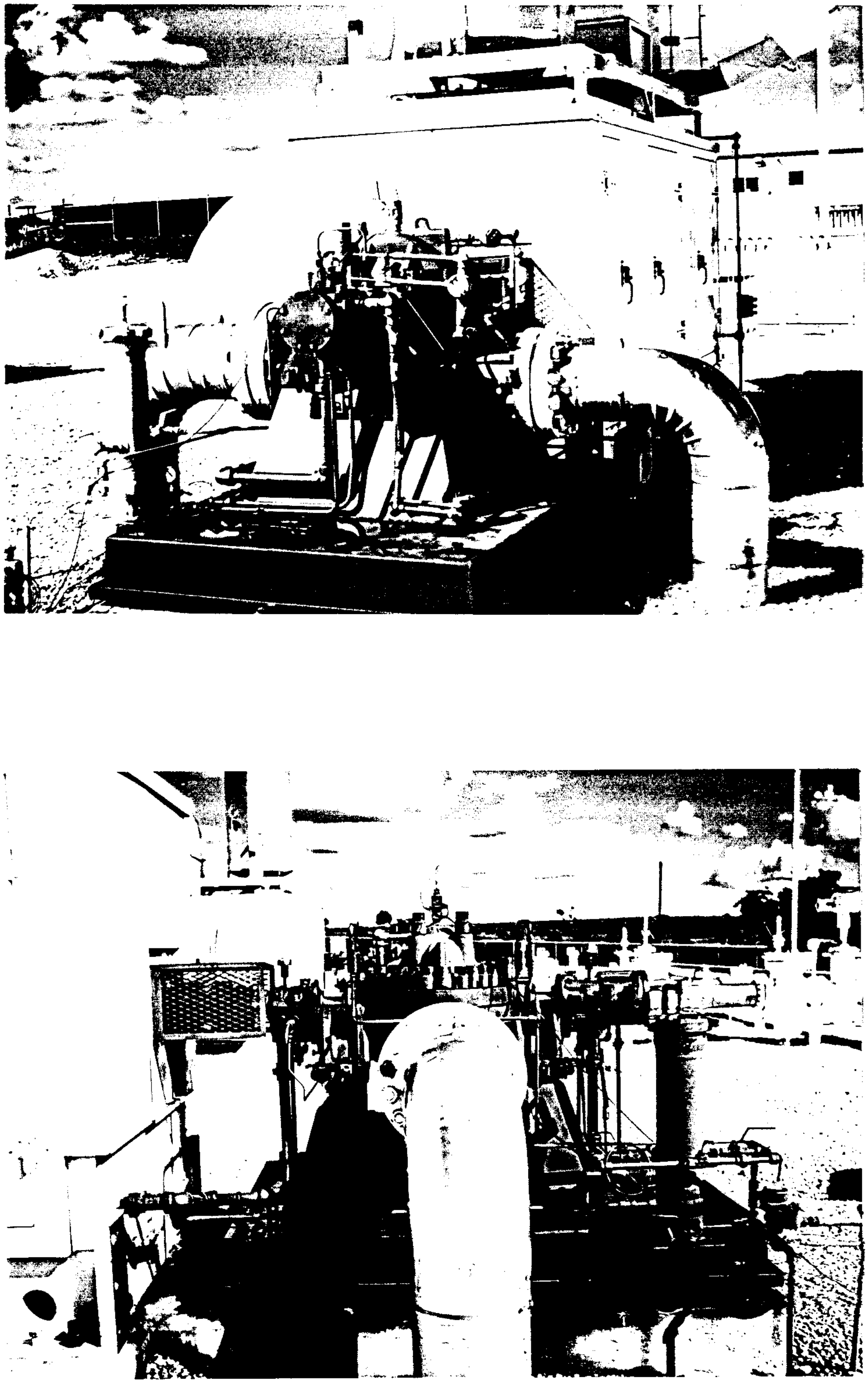
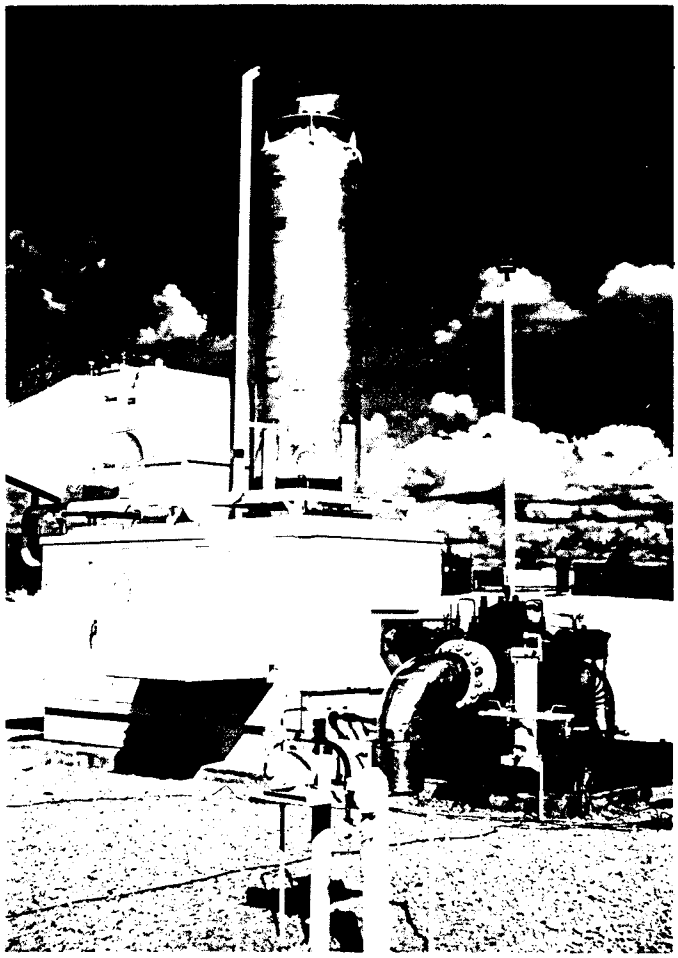
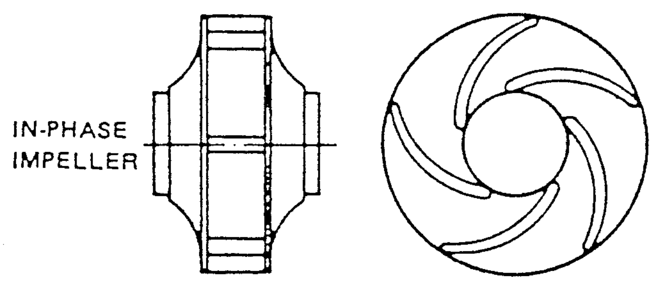


Figure 3.2





Original Impeller
Double Suction
Single Discharge

Figure 3.3

ENGINEERING DYNAMICS INCORPORATED SAN ANTONIO, TEXAS

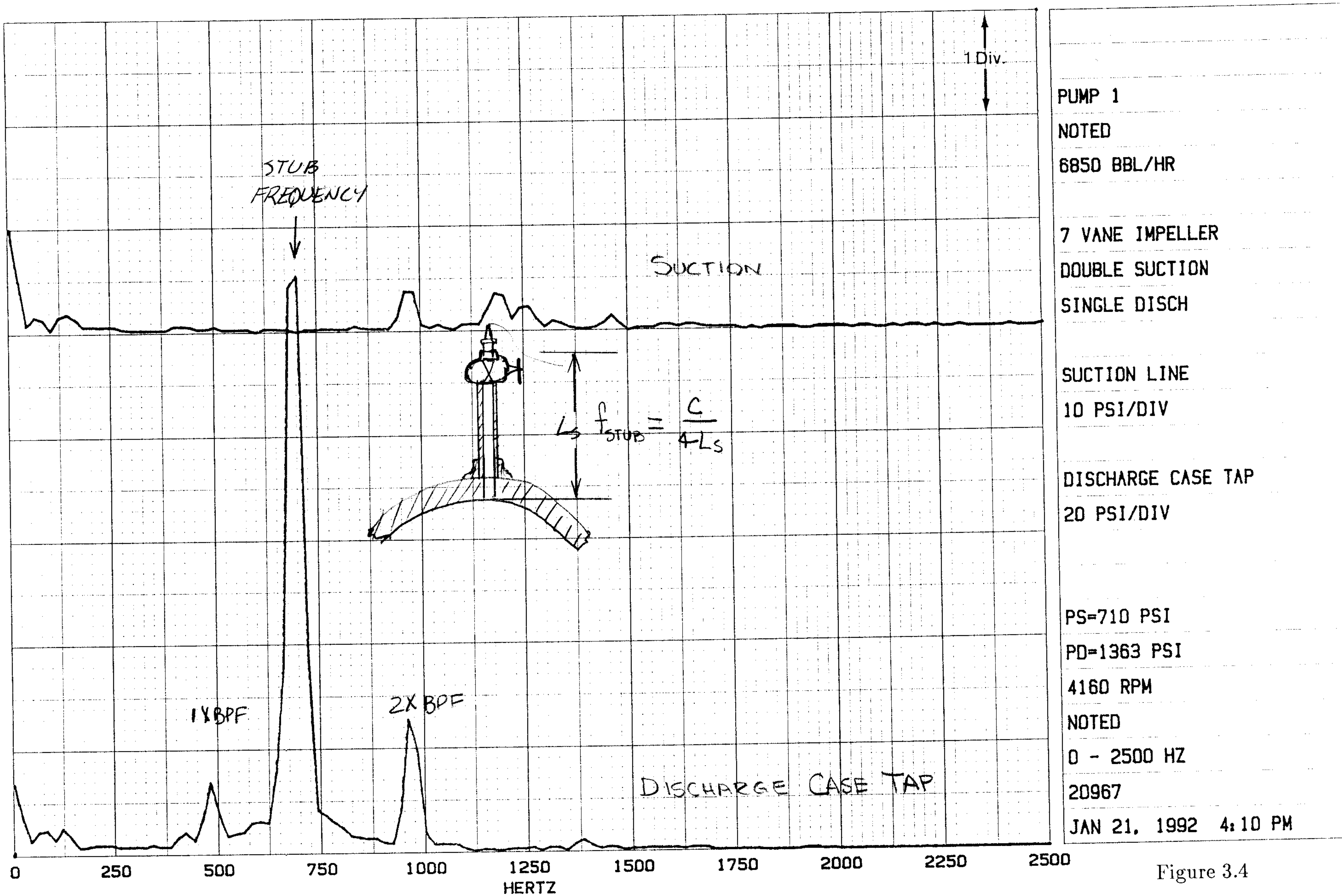


Figure 3.4

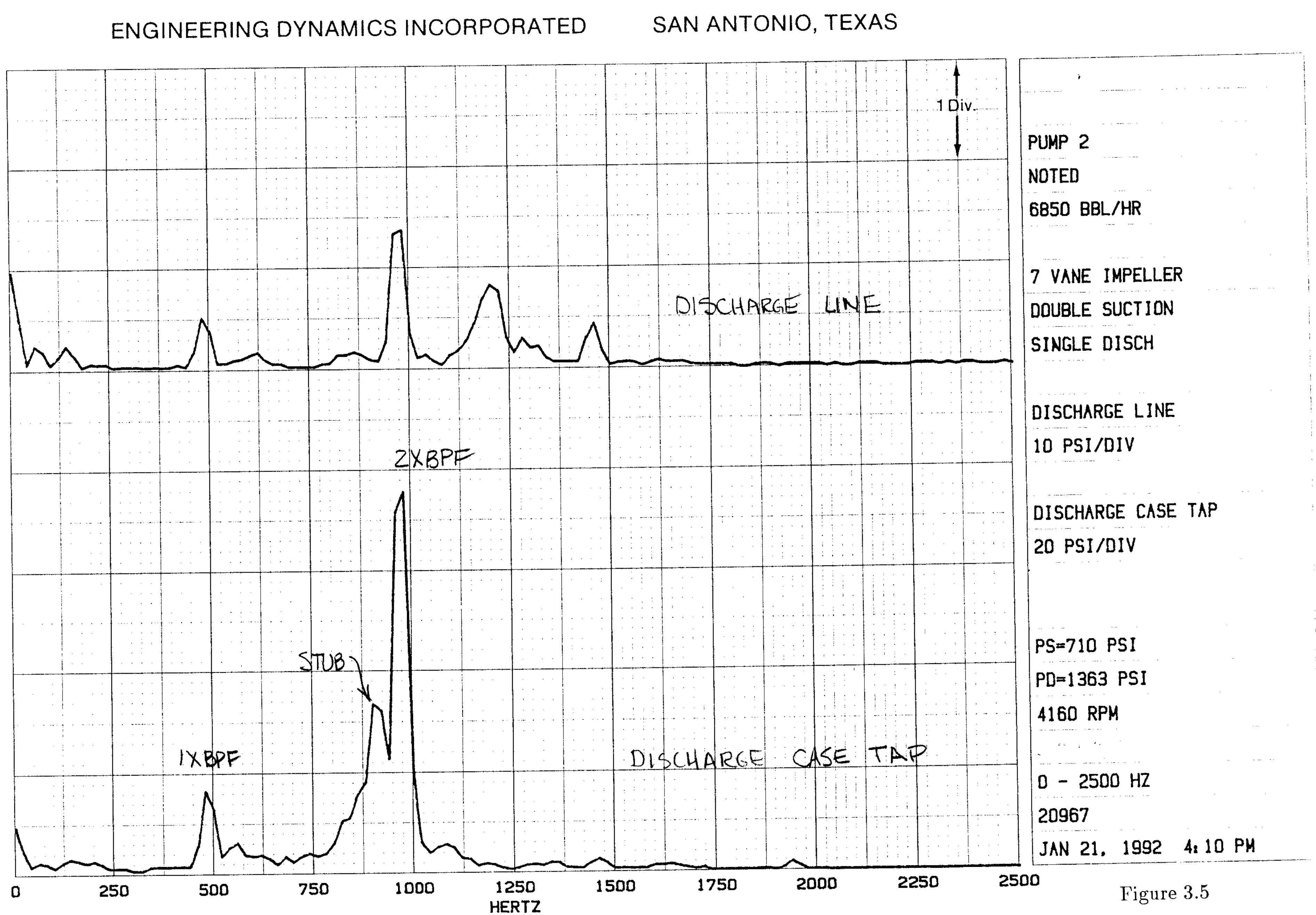
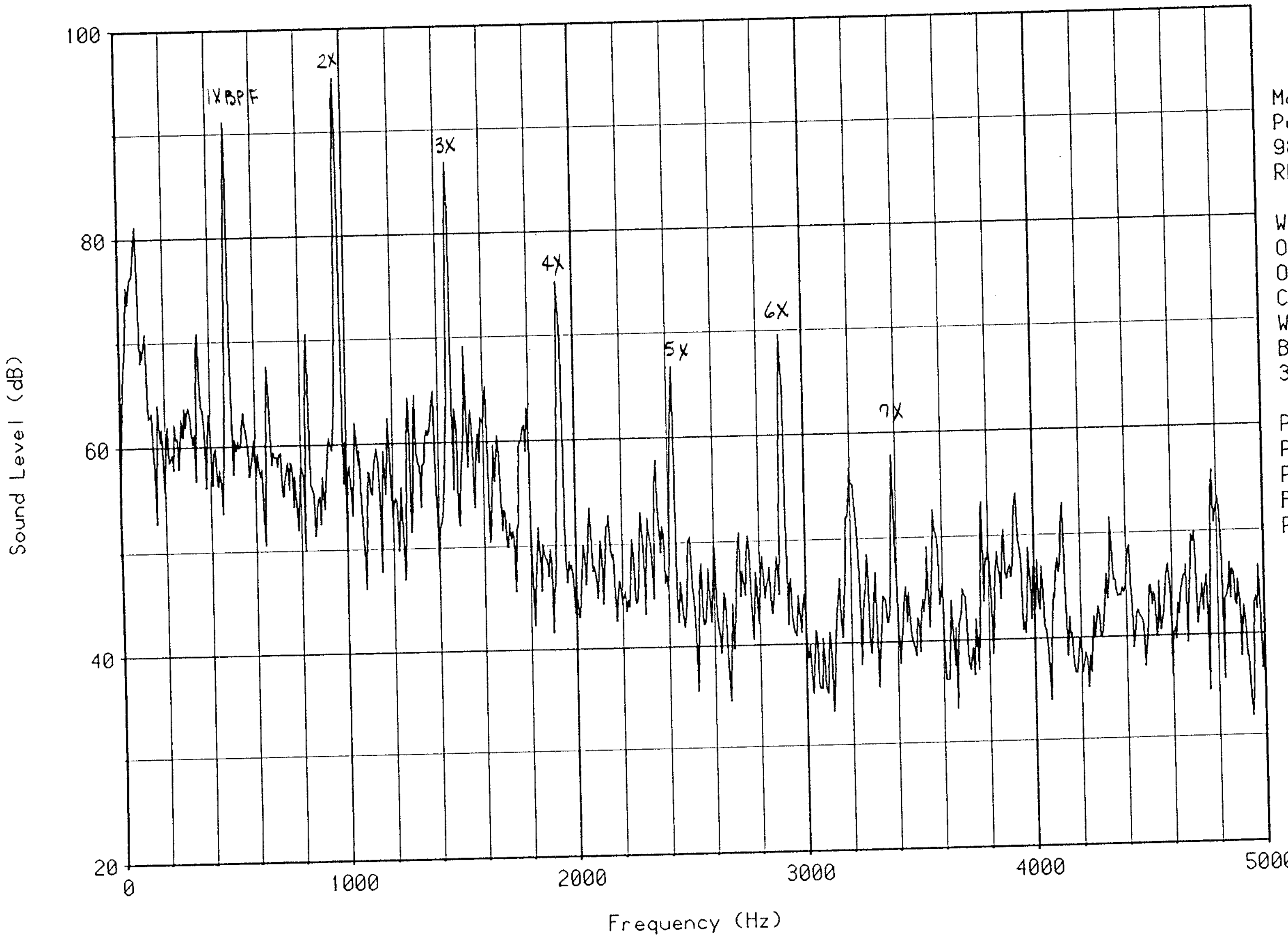

ENGINEERING DYNAMICS INCORPORATED
SAN ANTONIO, TEXAS
1 Div.
DISCHARGE LINE
2XBPF
STUB
1XBPF
DISCHARGE CASE TAP
0
250
500
750
1000
1250
1500
1750
2000
2250
2500
HERTZ
PUMP 2
NOTED
6850 BBL/HR
7 VANE IMPELLER
DOUBLE SUCTION
SINGLE DISCH
DISCHARGE LINE
10 PSI/DIV
DISCHARGE CASE TAP
20 PSI/DIV
PS=710 PSI
PD=1363 PSI
4160 RPM
0 - 2500 HZ
20967
JAN 21, 1992 4:10 PM

Figure 3.5



Main Line Between
Pump 1 and Pump 2
98.1 dB
RMS 4 Averages (CSI)

Weather:
Outside Temp. = 57 F
Outside Humid. = 39%
Clear Sky
Wind from N @ 5 mph
Barometric Press. =
30.26 inches Hg rising

Pump Conditions:
Psuct = 766 psig
Pdisch = 1419 psig
Flow Rate = 6850 bbl/hr
Pump Speed = 4160 rpm

Time: 13:50:06
Date: 01-09-1992

Figure 3.6

ENGINEERING DYNAMICS INCORPORATED - SAN ANTONIO, TEXAS

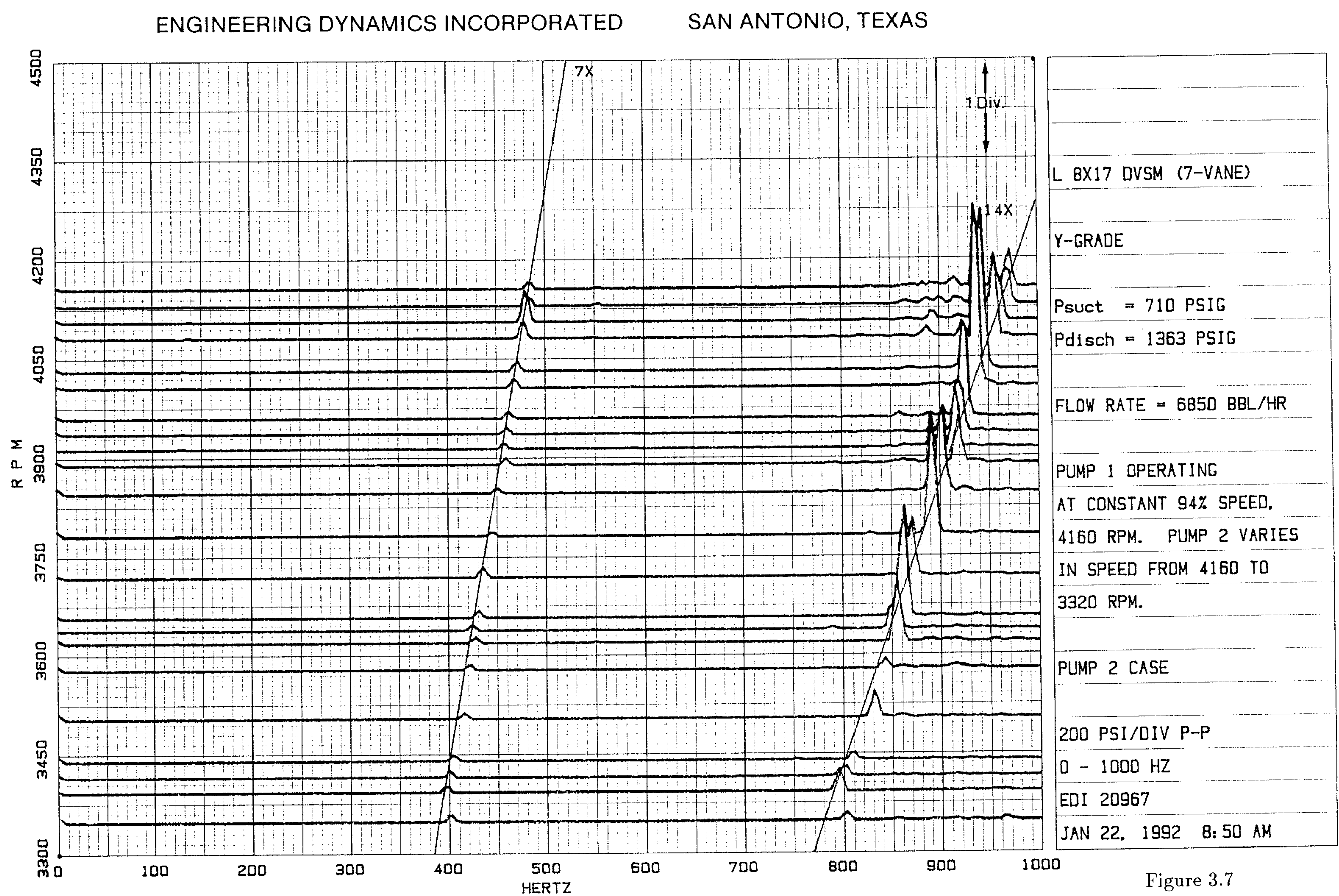
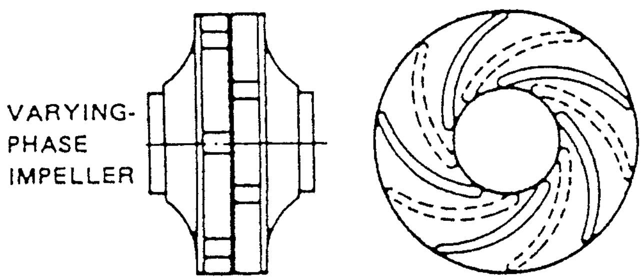

ENGINEERING DYNAMICS INCORPORATED
SAN ANTONIO, TEXAS
R P M
4500
4350
4200
4050
3900
3750
3600
3450
3300
0
100
200
300
400
500
600
700
800
900
1000
HERTZ
7X
14X
1 Div.
L 8X17 DVSM (7-VANE)
Y-GRADE
Psuct = 710 PSIG
Pdisch = 1363 PSIG
FLOW RATE = 6850 BBL/HR
PUMP 1 OPERATING
AT CONSTANT 94% SPEED,
4160 RPM. PUMP 2 VARIES
IN SPEED FROM 4160 TO
3320 RPM.
PUMP 2 CASE
200 PSI/DIV P-P
0 - 1000 HZ
EDI 20967
JAN 22, 1992 8:50 AM

Figure 3.7



Modified Impeller
Double Suction
Double Discharge with staggered vanes

Figure 3.8

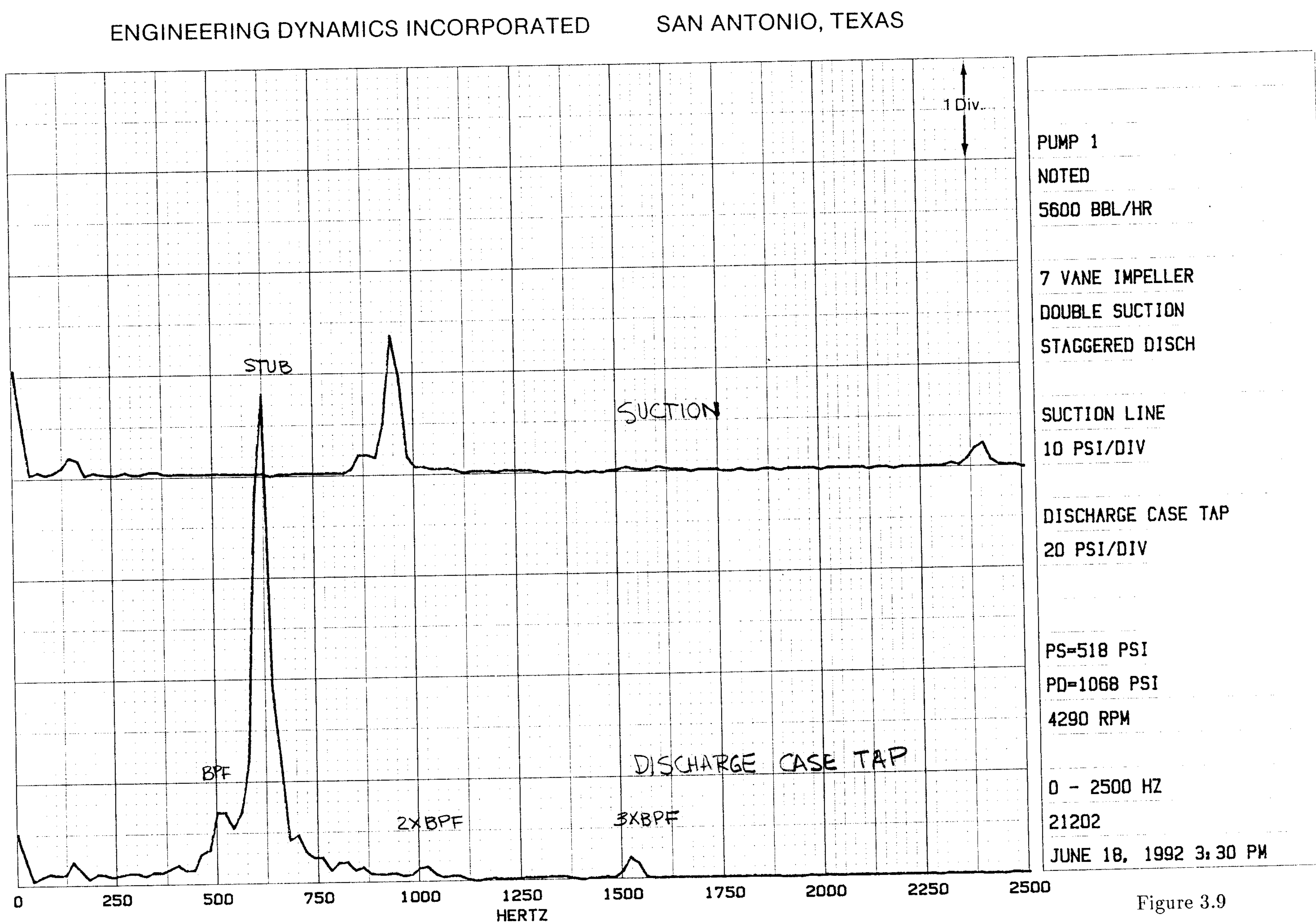
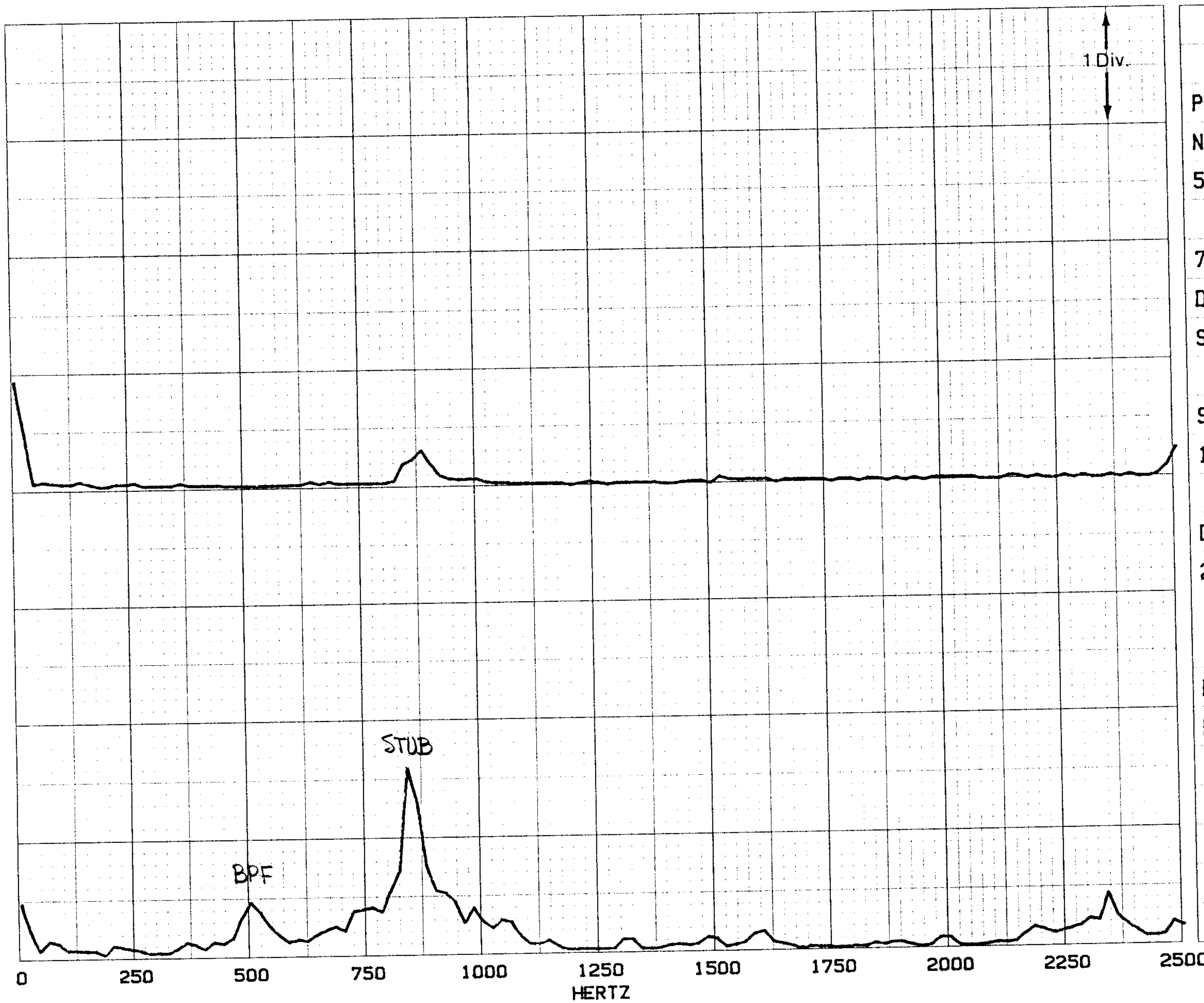

ENGINEERING DYNAMICS INCORPORATED
SAN ANTONIO, TEXAS
1 Div.
STUB
SUCTION
BPF
2XBPF
3XBPF
DISCHARGE CASE TAP
0
250
500
750
1000
1250
1500
1750
2000
2250
2500
HERTZ
PUMP 1
NOTED
5600 BBL/HR
7 VANE IMPELLER
DOUBLE SUCTION
STAGGERED DISCH
SUCTION LINE
10 PSI/DIV
DISCHARGE CASE TAP
20 PSI/DIV
PS=518 PSI
PD=1068 PSI
4290 RPM
0 - 2500 HZ
21202
JUNE 18, 1992 3:30 PM

Figure 3.9

ENGINEERING DYNAMICS INCORPORATED   SAN ANTONIO, TEXAS



PUMP 2
NOTED
5600 BBL/HR

7 VANE IMPELLER
DOUBLE SUCTION
STAGGERED DISCH

SUCTION LINE
10 PSI/DIV

DISCHARGE CASE TAP
20 PSI/DIV

PS=518 PSI
PD=1068 PSI
4290 RPM
NOTED
0 - 2500 HZ
21202
JUNE 18, 1992 3:30 PM

Figure 3.10

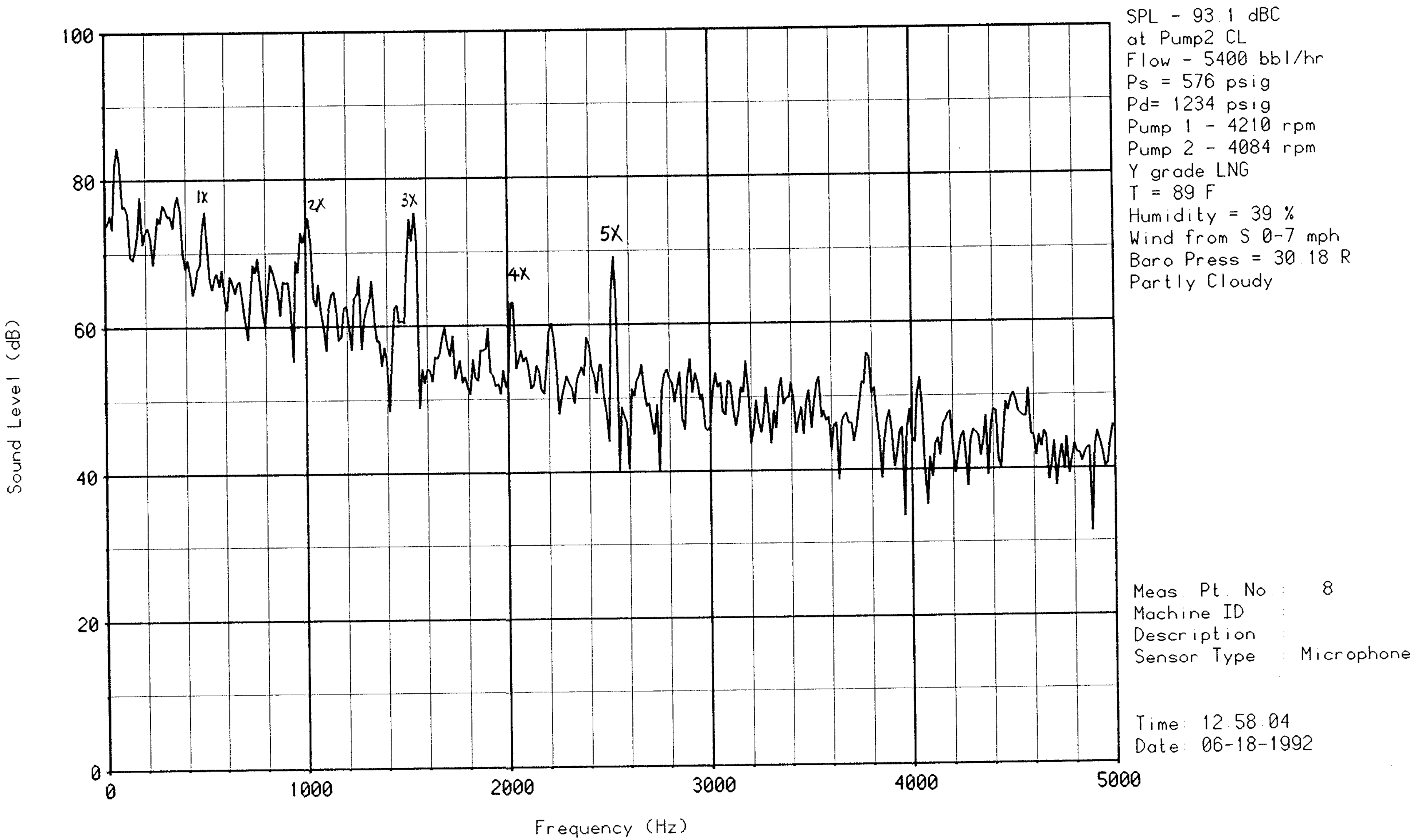
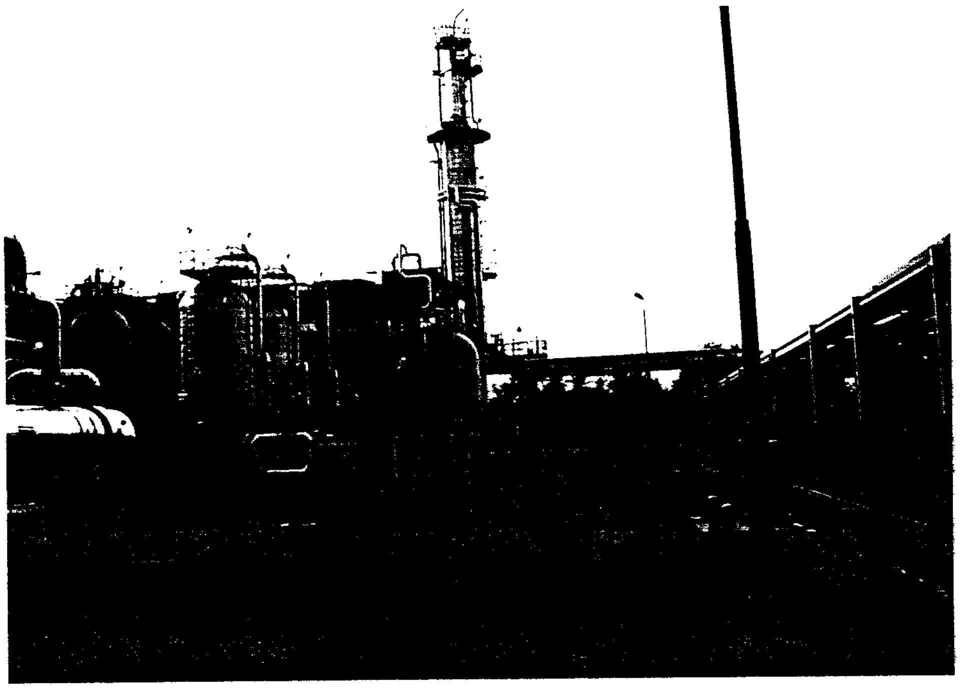


Figure 3.11

ENGINEERING DYNAMICS INCORPORATED - SAN ANTONIO, TEXAS

Figure 4.1

Dehydration System



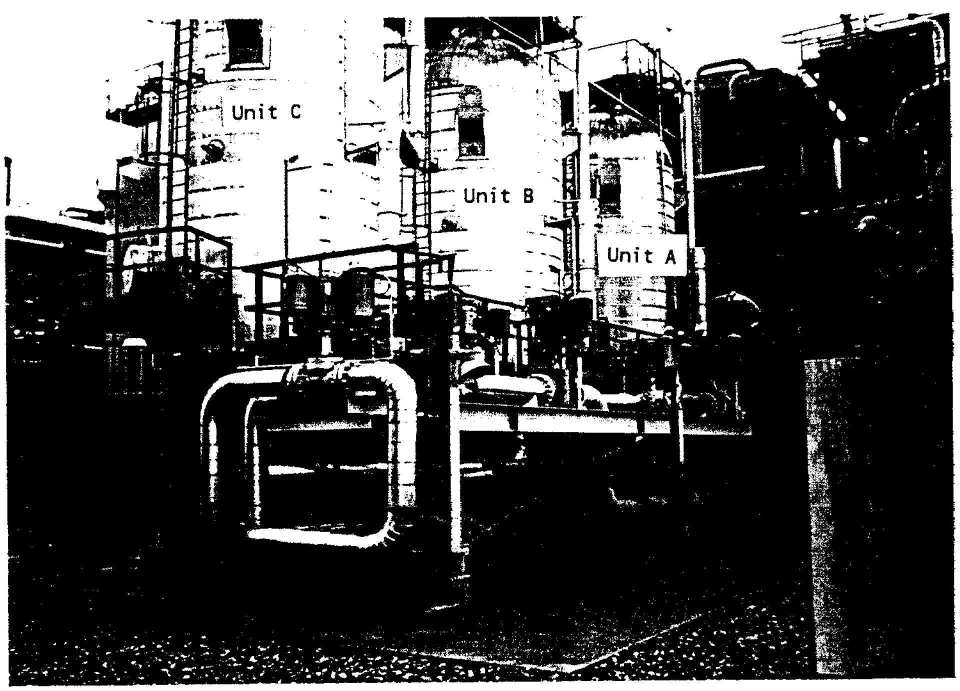
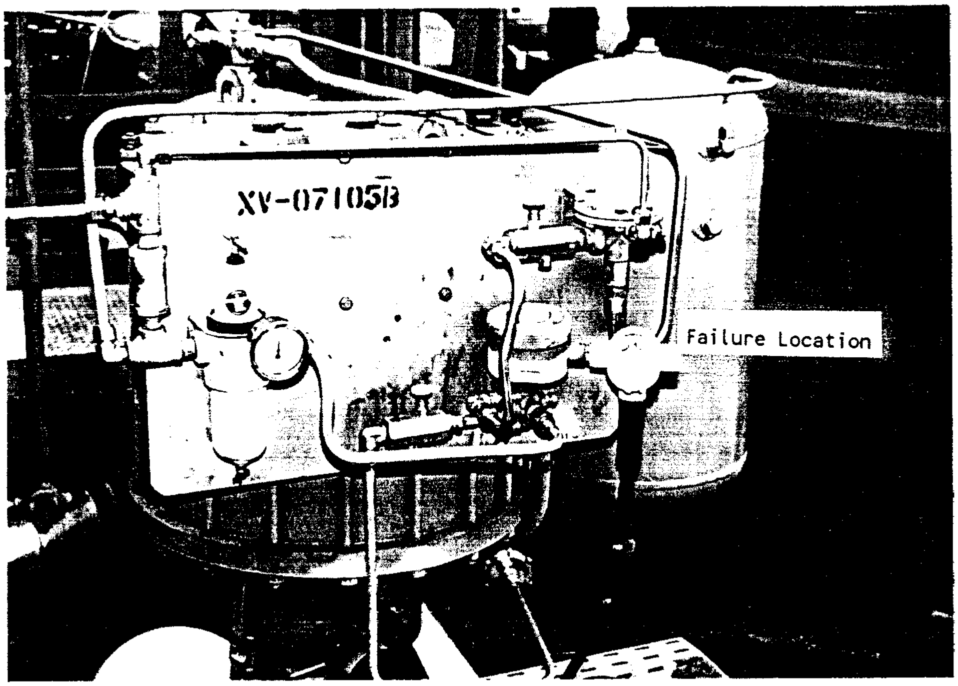


Figure 4.2

Typical Tubing Fatigue Failure



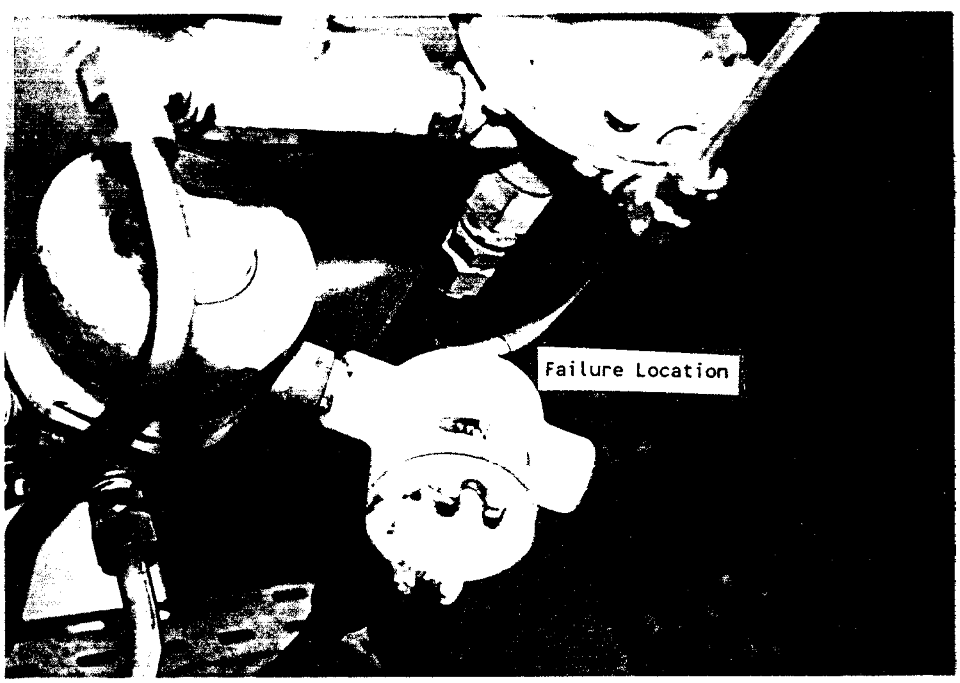
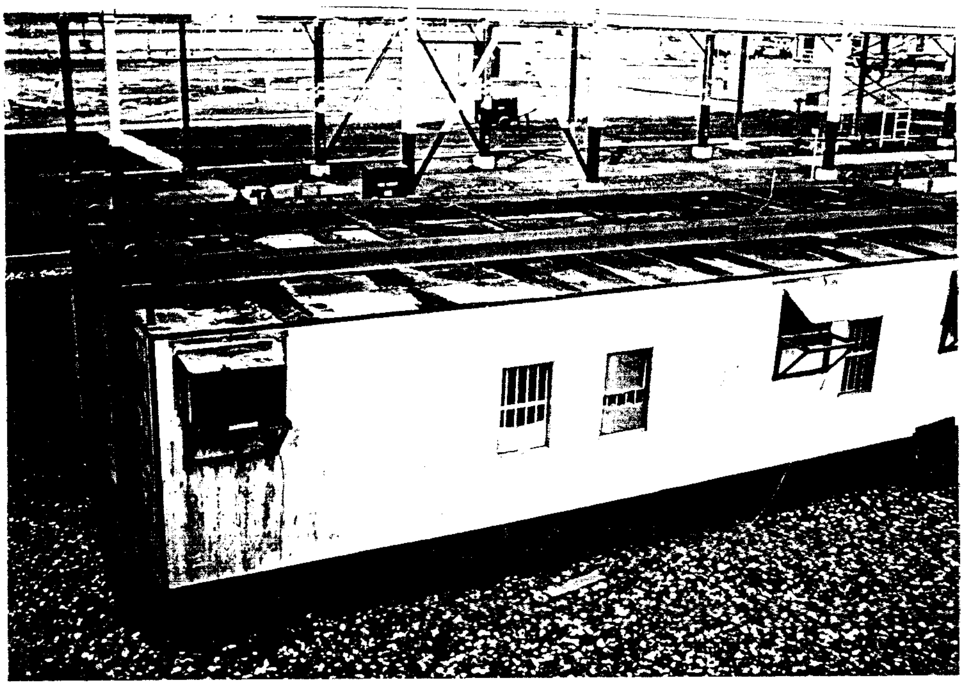


Figure 4.3

Instrumentation Setup in Portable Building



ENGINEERING DYNAMICS INCORPORATED SAN ANTONIO, TEXAS

DEHYDRATION SYSTEM

PULSATION IN REGEN GAS LINE TO COOLER

UNIT C
END OF DEPRESSURE CYCLE AND 54 MINUTES INTO HEATING CYCLE

9:21 PM TO 10:18 PM
PLOTTED EVERY 3 MINUTES

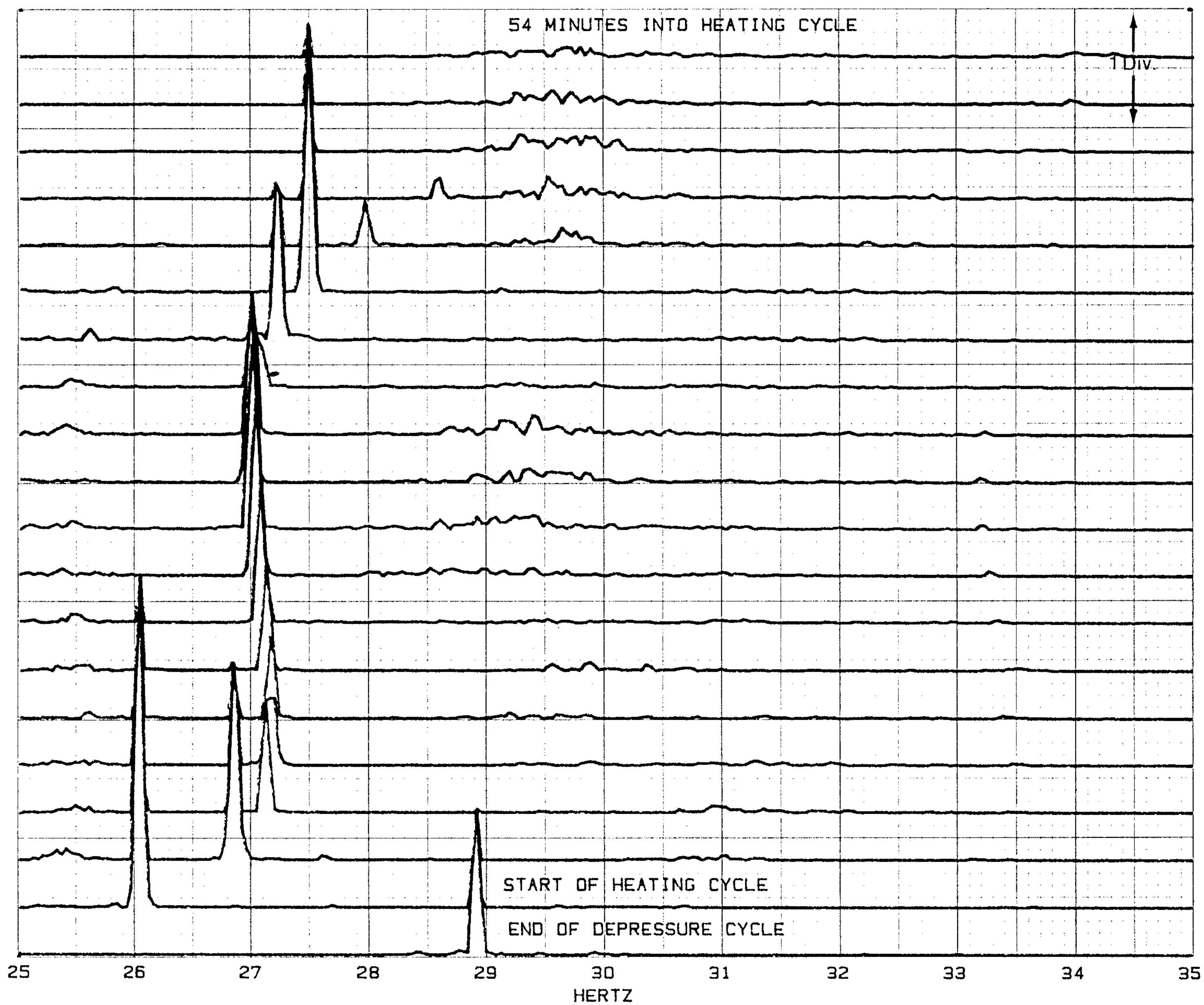
NOTE PULSATION AMPLITUDE AND FREQUENCY VARIED DURING THIS TIME.

TAPE 2-1 300-600'
0.2 PSI/DIV P-P
25 - 35 HZ

MAY 5, 1992

Figure 4.4

ENGINEERING DYNAMICS INCORPORATED SAN ANTONIO, TEXAS



DEHYDRATION SYSTEM

VIBRATION OF REGEN GAS LINE
TO COOLER. LOWER ELBOW
BELOW VALVE XVO7130 (E-W).

UNIT C
END OF DEPRESSURE CYCLE
AND 54 MINUTES INTO HEATING
CYCLE.

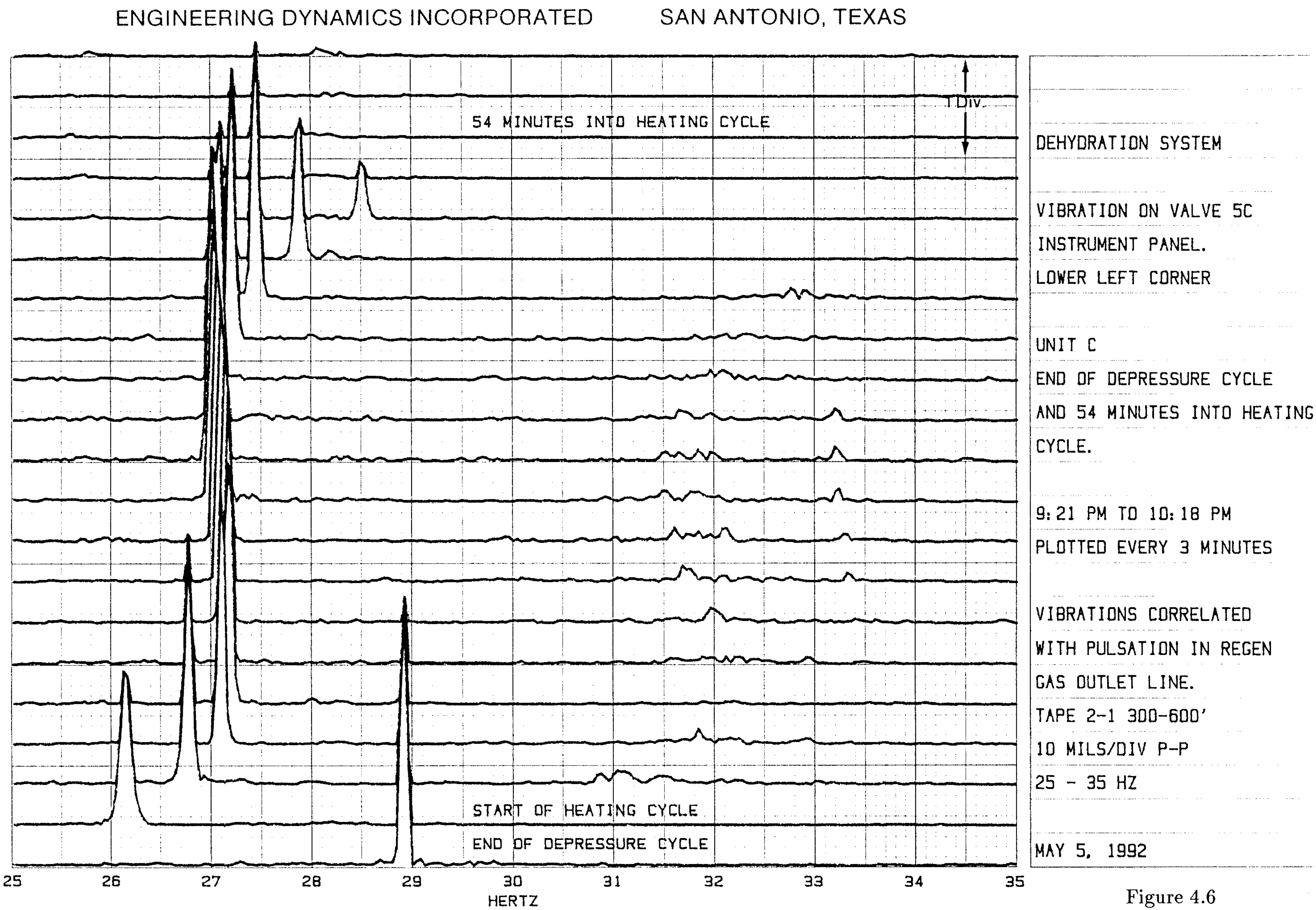
9:21 PM TO 10:18 PM
PLOTTED EVERY 3 MINUTES

VIBRATION DATA CORRELATED
WITH PULSATION DATA.

TAPE 2-1 300-600'
1 MILS/DIV P-P
25 - 35 HZ

MAY 5, 1992

Figure 4.5

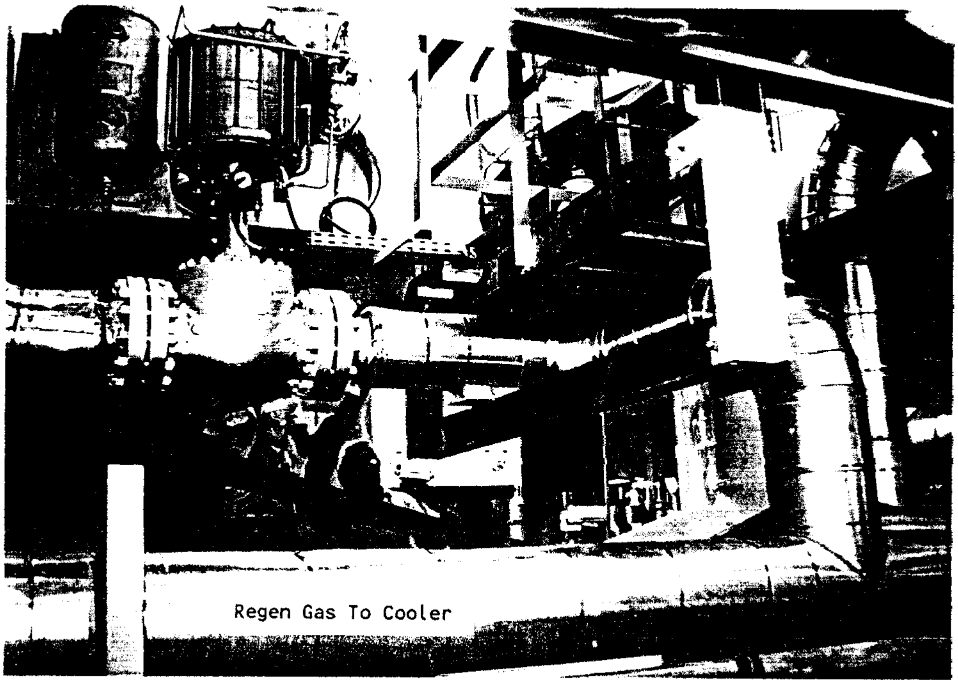


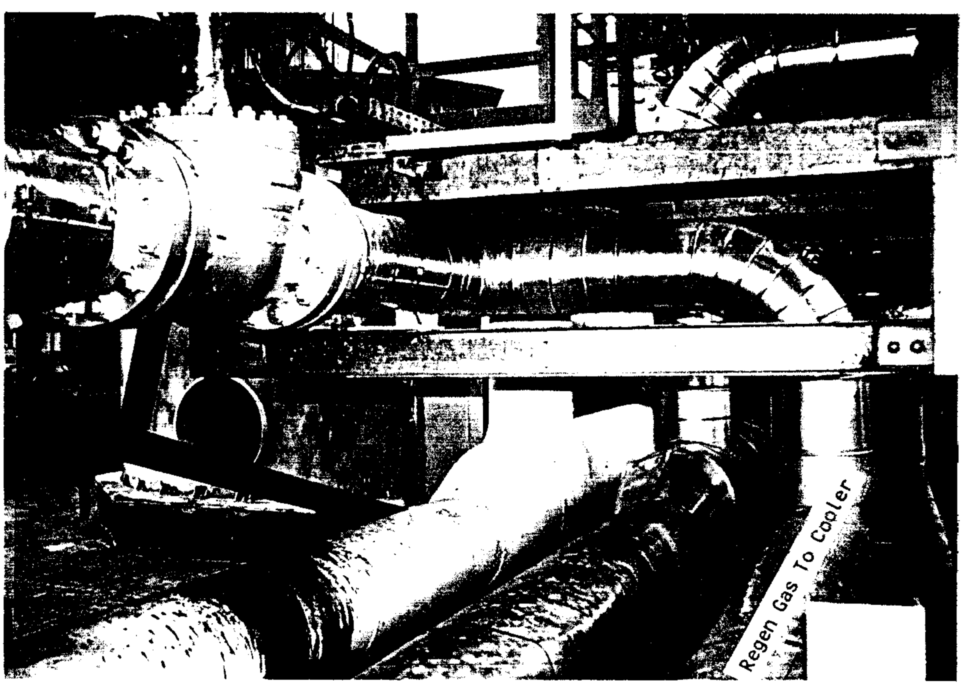
ENGINEERING DYNAMICS INCORPORATED
SAN ANTONIO, TEXAS
1 DIV.
54 MINUTES INTO HEATING CYCLE
START OF HEATING CYCLE
END OF DEPRESSURE CYCLE
25
26
27
28
29
30
31
32
33
34
35
HERTZ
DEHYDRATION SYSTEM
VIBRATION ON VALVE 5C
INSTRUMENT PANEL.
LOWER LEFT CORNER
UNIT C
END OF DEPRESSURE CYCLE
AND 54 MINUTES INTO HEATING
CYCLE.
9:21 PM TO 10:18 PM
PLOTTED EVERY 3 MINUTES
VIBRATIONS CORRELATED
WITH PULSATION IN REGEN
GAS OUTLET LINE.
TAPE 2-1 300-600'
10 MILS/DIV P-P
25 - 35 HZ
MAY 5, 1992

Figure 4.6

Figure 4.7

Piping Stubs Between Regen Gas Outlet Piping and Valves 07103A, 07103B, and 07103C





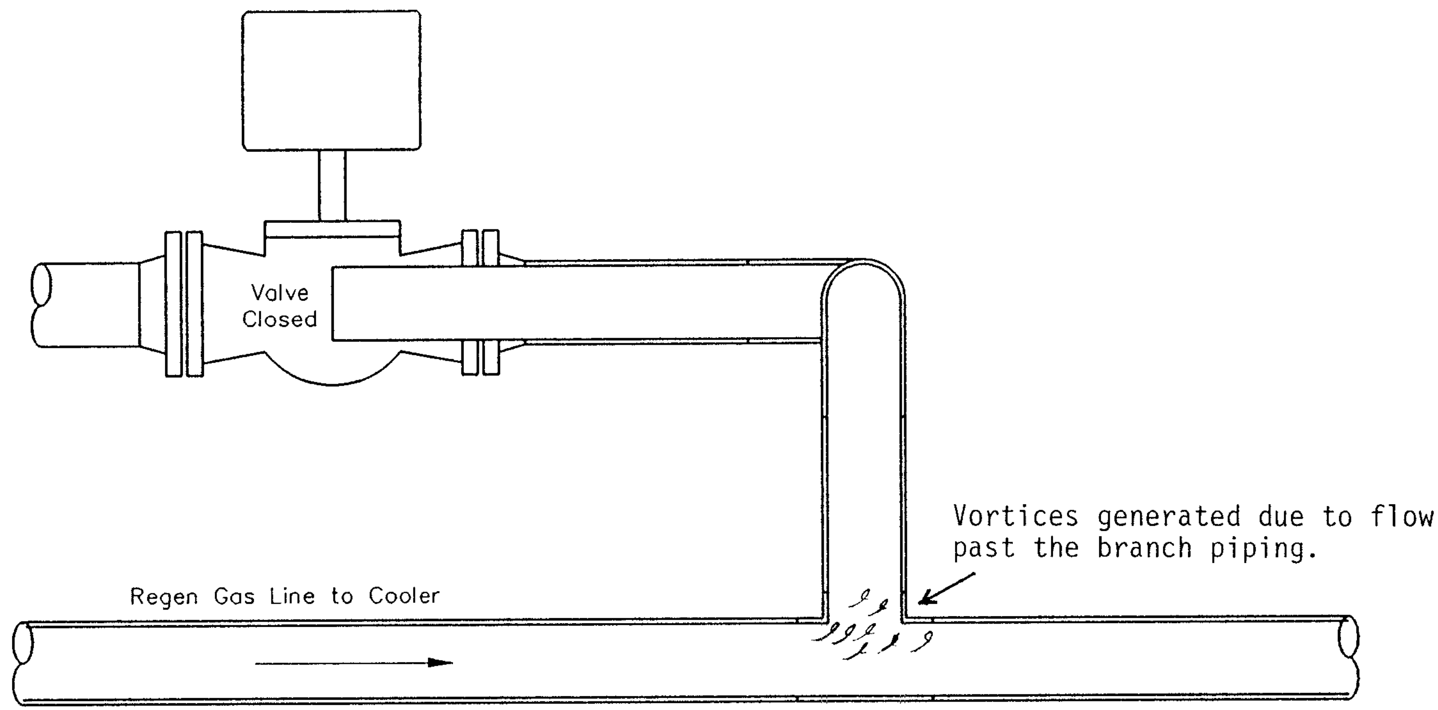
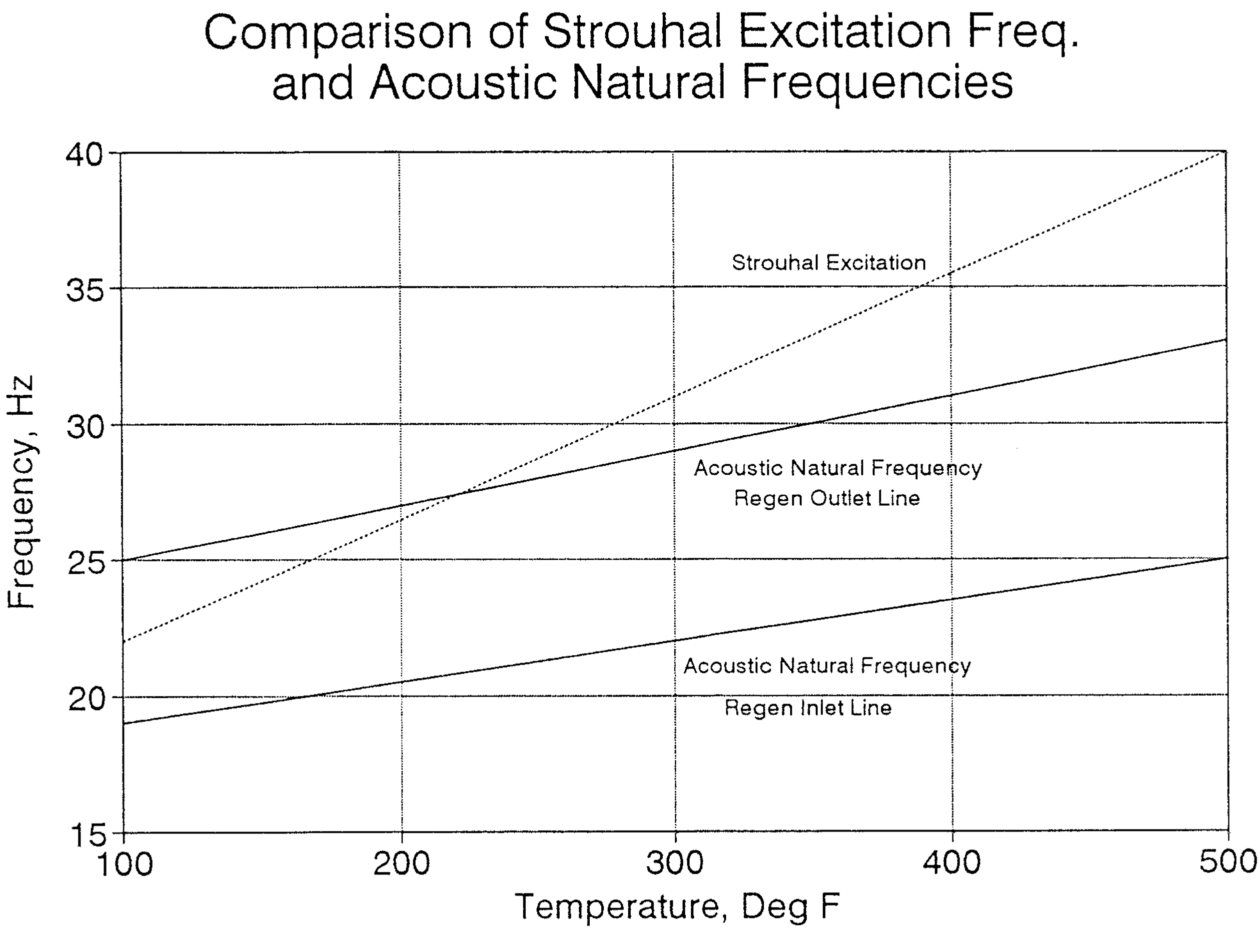


Figure 4.8

REV.	DESCRIPTION	DATE	
			ENGINEERING DYNAMICS INCORPORATED
0	For Review	7/01/92	Regen Gas Line to Cooler and Branch Piping to Valve
			DRWN: MAB
			SCALE: 1:20
			DWG NO: 2099013

Figure 4.9



Comparison of Strouhal Excitation Freq.
and Acoustic Natural Frequencies
40
35
30
25
20
15
Frequency, Hz
100
200
300
400
500
Temperature, Deg F
Strouhal Excitation
Acoustic Natural Frequency
Regen Outlet Line
Acoustic Natural Frequency
Regen Inlet Line