For more information,
please contact
Clance Doelling
3030 LBJ Freeway,
Box 60
Dallas, Texas 75234
Phone 214/241-PCRC
Fax 214/620-8518

8TH INTERNATIONAL RECIPROCATING MACHINERY CONFERENCE
Denver, Colorado
September 20-23, 1993

Technical Paper No. 4

ACOUSTICAL & MECHANICAL DESIGN CONCEPTS UNDERSTANDING AND CONTROLLING PULSATION

presented by

James D. Tison
Kenneth E. Atkins

Engineering Dynamics Incorporated
San Antonio, Texas

Pipeline & Compressor Research Council est 1952

Acoustical & Mechanical Design Concepts — Understanding and Controlling Pulsation

By

James D. Tison
Kenneth E. Atkins
Engineering Dynamics Incorporated

ABSTRACT

A technique for displaying acoustic mode shapes of piping systems using a computer-based simulation system is described. Traveling and standing waves are demonstrated in simple acoustical systems, as well as more generalized acoustic mode shapes in actual gas compression systems, using the computer based graphics. These graphical techniques help illustrate the vibratory nature of fluid in piping to improve one's understanding of piping acoustics. Various techniques for pulsation control are explained and compared, and the important relationships between the mechanical and acoustical characteristics of piping systems are demonstrated with a case study.

INTRODUCTION

This paper describes how a computer-based acoustic simulation technique is used to understand the overall acoustic characteristics of compressor piping systems and thereby aid the analyst in developing piping systems that control pulsation. The software allows the analyst to display the characteristics of very large piping systems since pulsation amplitude and relative phase can be animated simultaneously for a large number of points in the system. While describing the utility of the software, fundamental concepts of steady-oscillatory flow in one-dimensional systems are developed and demonstrated.

In addition to using the software to describe the mode shapes of various acoustic systems, comparisons of various typical bottle designs for a single-cylinder compressor discharge piping system are made using the simulation procedure. The all-important interdependent relationship between acoustical and mechanical designs are also described using a case history.

Figure 1 is a block diagram describing the acoustic simulation procedure used to generate the various examples described in this paper. The software represented in this diagram is the result of over ten years of development and testing by Engineering Dynamics Incorporated (EDI). The simulation code is based on the one–dimensional, damped wave equation which is derived from the momentum and continuity equations. Surrounding the simulation code are various input file structures, pre–processors and post–processors which facilitate the use of the system.

Animations of various acoustic mode shapes are displayed during the oral presentation of the paper. Since it is not practical to show time–based animations in print, figures which correspond to each of the "live" animations are included which show a superposition of all the animation "frames". The animations of pulsation amplitude and phase in time are represented by a circle whose diameter varies with dynamic pressure amplitude. A constant diameter circle represents a point in the system (located at the the center of the circle) which has little or no dynamic pressure variation (a pressure node). An increase in diameter represents an increase in pressure, a decrease a decrease in pressure.

ACOUSTIC MODE SHAPES IN SIMPLE SYSTEMS

Several simple acoustic systems are used to illustrate the technique by which acoustical mode shapes are displayed: an infinite length (non-resonant) pipe, a pipe which is closed on both ends, and a pipe which is closed on one end and open on the other.

Traveling Waves in Infinite Length Pipe

Figure 2 shows the computer model of a simple infinite–length line with a sinusoidally varying mass-flow excitation at one end. This excitation is equivalent to a piston being displaced back and forth with a constant amplitude at angular frequency ω in a steady-oscillatory fashion, with the piston motion described by:

$$x = X\sin(\omega t)$$

Since the pipe is infinitely long, no acoustic reflection occurs and the system is non-resonant. The animation shows the relationship in time of the dynamic pressure amplitude and phase at each of the test point locations. A continuous train of compressions and rarefactions travels along the *infinitely* long tube, with each point undergoing a full cycle of dynamic pressure. In the absence of damping, the amplitude at each point would be identical. The wavelength of the pressure wave is related to the acoustic velocity c, and the frequency, f, by the relationship:

$$\lambda = \frac{c}{f}$$

where:

c = speed of sound, fps
$f = \frac{\omega}{2\pi}$ = frequency, Hz

Half-Wave (Closed-Closed) Resonance

Consider the closed-closed system of Figure 3. The pipe is assumed to be 30 feet long, the diameter is 4.0″, and the speed of sound is 1200 ft/s. Using the equation for the half-wave resonance frequencies:

$$f_n = \frac{nc}{2l},\ n = 1,2,3,\ldots,$$

the resonant frequencies of the system are

$$f_n = 20, 40, 60, 80, \ldots \mathrm{Hz}$$

Assume that the piston undergoes oscillation $x = X\sin\omega t$ so that the peak piston velocity ($\dot{x} = X\omega\cos\omega t$) amplitude $X\omega$ is constant, and that the frequency f of the piston is varied slowly from a very low frequency to 100 Hz. The velocity amplitude $X\omega$ for the example is arbitrary, but constant over the frequency range 0–100 Hz.

If we plot the amplitude of pulsation in psi p–p at a particular point along the pipe (A, B, or C) over the frequency range of excitation (0–100 Hz), we obtain the frequency response amplitude of the pulsation over that frequency range. (Note that the pulsation spectra amplitudes are normalized.) The mode shape of the fundamental mode at 20 Hz of this resonant frequency is also shown in Figure 3. A node is apparent at the midpoint as the circle diameter is constant. Antinodes occur at the ends as the diameter of the circle varies from zero to a maximum value.

Quarter Wave (Open-Closed) Resonance

Figure 4 shows similar results for a quarter-wave stub (right end open). The resonant frequencies for this system are:

$$f_n = \frac{nc}{4l}, \; n = 1, 3, 5$$

$$f_n = 10, 30, 50, \ldots$$

Figure 4 also shows the animated mode shape for the fundamental resonant frequency at 10 Hz. The pressure variation is zero at the right, open end. Therefore, the diameter of the circle at this point is constant. The left end is a pressure antinode.

Mode Shapes of Resonances Associated with a Volume-Choke-Volume Filter

An acoustic filter consists of two volumes connected by a relatively small diameter pipe (choke tube). The volumes of the two chambers serve as acoustic compliances, while the fluid in the choke tube serves as an acoustic inertance. The combination of these acoustic elements in this manner produces a "low pass" filter which attenuates pulsation at frequencies above its "cutoff" frequency. Figure 5 shows an acoustic model of a volume-choke-volume filter, and the passive frequency response of the system at the one-quarter point of the choke tube. The resonant peak at frequency $f_H = 10.5$ Hz, is referred to as the Helmholtz frequency of the two-chambered filter, and will *amplify* pulsations at that frequency. In addition to the Helmholtz resonance of a two-chambered filter, internal resonances of the filter elements can have the effect of "passing" particular frequencies. The choke tube acts as an open–open pipe such that a pass band occurs at the half wave length resonance of the choke at 112.5 Hz. Figure 6 shows animated mode shapes for the low–mode filter frequency, and of the choke tube pass–band frequency.

ACOUSTIC MODE SHAPES IN REAL COMPRESSOR SYSTEMS

An understanding of the overall characteristics of the acoustic resonant frequencies is important in order to make effective and efficient design decisions regarding the control of pulsation in the piping. One of the most important tools available to this end is the ability to display acoustic mode shapes of actual systems. Examples of simulation of actual systems are presented in this portion of the paper.

Hydrogen Compressor Suction Piping System

The simulation of this system serves as an example of the type of data generated during an acoustic simulation, as well as the effectiveness of orifice plates in certain systems.

Figure 7 is the piping model diagram of the suction system of two parallel Hydrogen compressors, each consisting of a single cylinder. Pulsation test points are documented on the diagram. Test points are shown at each bottle, each suction riser, midway between the two units in the header, in the header near the scrubber, and in the scrubber itself.

The upper graph in Figure 8 gives the pulsation spectrum at the cylinder flange test point of compressor A for single acting operation of the compressor cylinders. Since the compressors run at 450 RPM, the fundamental excitation frequency is 7.5 Hz, and high pulsation levels (120 psi p-p) occur at both compressors near this frequency. The simulation considers a speed range (±10%) so that response peaks can be identified. Note that the shape of the response curve indicates a peak response or resonance just above 7.5 Hz.

The passive frequency response (middle graph – Figure 8) shows various resonant frequencies, with a dominant response at 8.15 Hz. Animation showing the relative amplitude and phase relationships of the 8.15 Hz resonance (Figure 9), indicates that the pulsation amplitudes are (180 degrees) out of phase at the bottles of the two compressors, with a pressure node near the header midway point between the compressors. This mode shape data gives insight into effective locations for orifice placement since orifice plates are most effective when located at velocity maximums.

Based on the simulation results, it can be determined that orifice plates would be most effective when placed in the suction lead lines or in the header of the two compressors as opposed to locating one at the scrubber vessel outlet flange. The lower graph of Figure 8 shows simulation results with orifice plates installed at the suction bottle inlet flange of each compressor. Pulsation levels at 1× running speed were reduced by a factor of approximately 10. (Note the y-axes are auto-scaled so that the scales of the upper and lower graphs are different.)

Suction Piping of Four Natural Gas Compressors

Figure 10 shows the computer generated piping model diagram of the suction piping of four six–cylinder, single–stage compressors in a common header. Figure 11 shows the passive frequency response (spectrum) at the cylinder flange of the far-right compressor due to a flow excitation at one cylinder of the same compressor. A representation of the animation of the lowest acoustic natural frequency near 1.7 Hz is also illustrated. The mode involves oscillatory flow of the header between the two end compressors such that the maximum pressure modulation occurs at the end compressors. Note that the pulsation levels are low in the header itself. Each of the peak response frequencies shown in the passive analysis has a characteristic acoustical mode shape which can be demonstrated in a similar manner. The second mode near 1.9 Hz involves oscillatory flow of the header portion between the two compressors on the far left. The third modes near 3.5 Hz and 5.4 Hz are associated with the primary and secondary bottles, and the mode near 20 Hz is associated with the frequencies of the primary bottle itself.

COMPARISON OF PULSATION SUPPRESSION DEVICES

A thorough understanding of the acoustical mode shapes of resonant frequencies in piping systems can be valuable for determining orifice plate locations, effective locations for pipe diameter changes for tuning of resonant frequencies, etc. However, usually the bottle (or filter) design is the most important element available to the designer. This section of the paper deals with the effect of bottle design on pulsation control.

The type of pulsation suppression device required depends on the degree of pulsation control desired. In certain cases, a surge volume may provide adequate attenuation of pulsations. However, for maximum attenuation over a wider frequency range, an acoustic filter should be designed. Even with an acoustic filter, several levels of pulsation control are possible depending on the size of the elements (bottle, choke) used. Because of the need for bottle internals or a two bottle design to achieve the acoustic filter, the cost of these devices is normally higher than that of a single surge volume. The actual size of the surge volumes or filters depends on the speed of the compressor or pump, the stage pressure ratio, the thermophysical properties of the fluid, and the geometry of the piping system.

In order to illustrate the relative effect of the different types of pulsation devices, several devices were analyzed using the digital acoustical simulation program. The discharge piping of a typical compressor cylinder was assumed in conjunction with various size surge volumes and acoustic filters. A case assuming no pulsation control device in the piping system was also analyzed. The compressor cylinder simulated has a 9.25″ bore with a 6″ stroke and an operating speed range of 700–1000 rpm. The suction pressure is 267 psia and the discharge pressure is 567 psia. To eliminate the effect of piping acoustic resonance, the attached piping is assumed to be infinite length. Note that this assumption ignores the interaction of the compressor/bottle with the attached piping, and therefore compares only the relative effects of the various bottles themselves.

The pulsation control devices considered are listed below, with the results of the analyses shown in Figure 12.

Case 1: No pulsation control.

Case 2: A surge volume with one-half the volume required by API 618.

Case 3: A surge volume with the full API 618 recommended volume.

Case 4: API 618 surge volume with an orifice plate at the bottle inlet (cylinder discharge) flange.

Case 5: A volume-choke-volume acoustic filter with its Helmholtz or cut-off frequency near 4 times running speed (50 Hz).

Case 6: A volume-choke-volume acoustic filter. The Helmholtz or cut-off frequency of this filter is approximately 22 Hz (between 1× and 2× running speed).

Case 7: A volume-choke-volume acoustic filter with the Helmholtz frequency tuned between 1× and 2× running speed (22 Hz). The larger bottle and choke tube in this design result in the same Helmholtz frequency as Case 6, but with less pressure drop.

Case 8: A volume-choke-volume acoustic filter. The Helmholtz or cut-off frequency of this filter is approximately 9 Hz (below 1× running speed).

Frequency spectra of the pulsation in the discharge pipe (downstream of the pulsation suppression device) are presented in a graphical data format for each case.

Each of the frequency spectra plots contain multiple curves representing frequency response at each individual harmonic of compressor speed. The amplitude of pulsation in psi p–p of each harmonic is plotted versus frequency in Hertz for the full speed range for the simulation. For example, for a compressor speed range of 700–1000 rpm, the first harmonic would sweep the 11.67–16.67 Hz range, the second harmonic would sweep the 23.34–33.33 Hz range, etc. These data formats are important for evaluating the actual predicted pulsation amplitudes and frequencies of individual harmonic components. A knowledge of both amplitude and frequency are important in evaluating the acceptability of piping designs from a vibration standpoint.

For case 1, in which no pulsation control is assumed, a high response is predicted at the second harmonic of running speed (referred to as 2× running speed – from 23.3 to 33.3 Hz). The simulation predicts a maximum amplitude at 2× running speed of approximately 32 psi p–p. The high amplitude pulsations at 2× running speed is expected because the cylinder is operating in a double acting mode. This causes two pulses to be generated each cycle. Note that the actual pulsation trace in the line measured with a dynamic pressure transducer would be some combination of all of the harmonics of running speed.

In case 2, a surge volume with one-half the volume recommended by API 618 is connected to the discharge flange of the cylinder. The peak pulsation at 2× running speed is lowered from 32 psi p–p in case 1 to approximately 12 psi p–p with the surge volume. However, note that a new response is predicted at 115 Hz with a maximum amplitude of approximately 6 psi p–p. This response is a result of the "nozzle" resonance between the cylinder and the bottle. Pulsation is transmitted through the surge volume into the discharge line at this frequency. Note that the amplitudes are much higher at the cylinder valves at this frequency than at the piping test point.

In case 3, a larger surge volume with the volume recommended by API 618 is connected to the discharge flange of the cylinder. In this case, the maximum predicted pulsation at 2× running speed is lowered to approximately 8 psi p–p. The nozzle resonance still occurs, but near 110 Hz and with a maximum predicted amplitude of 6 psi p–p.

Case 4 is identical to case 3, except that an orifice plate has been added to the cylinder discharge flange. The orifice plate has a pressure drop of 0.71 psi (1/8% of line pressure). In comparing case 3 to case 4, note that the addition of the orifice plate did not affect the predicted amplitude at 2× running speed. However, the predicted amplitude at 110 Hz (the nozzle resonance) is reduced to below 2 psi p–p.

Case 5 represents a volume-choke-volume filter with the Helmholtz frequency set to approximately 60 Hz (4× running speed). Note that the total volume of the filter (both chambers) is equivalent to the API 618 recommended surge volume used in cases 3 and 4. This filter could be created by adding a baffle to divide the API 618 surge volume into two chambers and connecting them with a choke tube through the baffle as shown. The pressure drop through the choke tube is approximately 2.5 psi (0.44%). The pulsation spectra for case 5 looks almost identical to that of case 4. The effect of the choke tube is to attenuate pulsations above the Helmholtz frequency (60 Hz). The nozzle resonance near 110 Hz (See case 3) is attenuated with the filter; however, the pulsation at 2× running speed is unaffected. In comparing case 4 to case 5, essentially the same pulsation control can be obtained using an orifice plate having a pressure drop of 1/8% of line pressure as can be obtained by adding a choke tube with 2.5 psi pressure drop.

Case 6 represents a volume-choke-volume filter (using a total volume equal to the API 618 recommended volume) with a Helmholtz frequency which has been set to approximately 22 Hz (between 1× and 2× running speed). This is accomplished by using a longer choke tube with a slightly larger diameter. The resulting pressure drop through this choke tube is 2.4 psi (0.42%), which is approximately the same as in case 5. Note that the maximum predicted response at 2× running speed actually increases in this case to approximately 11 psi p–p (compared to cases 3,4, and 5). Note also that the shape of the harmonic curve has changed, with the maximum amplitude occurring at the lowest frequency (approximately 23.3 Hz, corresponding to 700 rpm compressor speed). This increase in predicted pulsation at 2× running speed is caused by its proximity to the Helmholtz frequency of the filter. Pulsations greater than this frequency are generally attenuated, but pulsations near this frequency can be amplified. Note, however, that the predicted pulsation at 2× is still less than that predicted in case 1 with no pulsation control.

Case 7 shows a volume-choke-volume filter where the Helmholtz frequency has, as in case 6, been tuned between 1× and 2× running speed. However, in this case, the bottle volume and the choke diameter have been increased. This results in practically the same attenuation characteristics, but only requires a pressure drop of 1.3 psi (0.23% of line pressure) through the choke tube.

Finally, case 8 represents a volume-choke-volume filter with a Helmholtz frequency which has been set to approximately 9 Hz (below 1× running speed). This is accomplished with large volumes and longer choke tubes. The filter can be built as one large bottle with a baffle and choke tube, or it can be built using primary and secondary bottles, with the choke tube connecting them as shown in the schematic in Figure 12. The pressure drop through the choke tube is again 2.4 psi (0.42%) for this case. The predicted pulsations are significantly attenuated at all harmonics of running speed. No harmonic is predicted to have pulsation levels greater than 1 psi

INTERDEPENDENCE OF ACOUSTICAL AND MECHANICAL DESIGNS

When designing a compressor installation, it is important that both the mechanical and acoustical aspects be considered to ensure acceptable dynamic stresses and vibration levels. This is the basic goal of API 618 Design Approach 3. The following case history is used to illustrate the interdependence of the mechanical and acoustical design parameters for a single stage natural gas transmission station.

The compressor station under consideration involves an existing facility with two compressors operating in parallel. A third unit was to be added, and it was decided to design pulsation filters with their primary Helmholtz frequencies below the first order (1× compressor speed). The intent of this design was to acoustically isolate the new unit from the existing compressors and minimize the pulsation levels in the yard piping and filter-separator areas. The new unit was a two-cylinder single stage compressor rated for 4600 bhp at 330 rpm with a design flow of 492 MMSCFD. The suction pressure was 810 psig and the discharge pressure was 980 psig. The suction system will be described here.

The design goal of filtering below 1× running speed required that relatively large diameter bottles be used to minimize the pressure drop. The two-cylinder, single stage design typically dictates that baffled primary bottles be used for force balancing. Otherwise, excessive high frequency shaking forces could exist in the bottle.

Baffling the primary bottle creates a volume-choke-volume resonance which must be tuned to the proper frequency to avoid coincidence with mechanical natural frequencies of the compressor cylinder-manifold system. One possibility would be to place this internal Helmholtz resonance between the first and second order. However, this would have required an impractically large bottle diameter and/or excessive pressure drop in the internal choke tube. Therefore, the internal Helmholtz resonance had to be placed on the third order, which is a relatively weak pulsation order and resulted in more practical bottle sizes and pressure drop. This has the added benefit of improving the filtering of the internal Helmholtz passband, since it was further above the *main* filter frequency. Therefore, the pulsation levels in the yard piping were actually reduced by placing the primary bottle's internal Helmholtz frequency at 3× rather than between the first and second orders. The suction filter layout is shown in Figure 13. Also shown in Figure 14 is a shaking force spectrum which shows the force levels generated by the internal Helmholtz resonance.

The compressor manifold system mechanical analysis showed that the original "low mode" mechanical natural frequency was near the internal Helmholtz frequency, and thus the frequency of the predominant bottle unbalanced force, Figure 14. Therefore, dynamic cylinder restraints, (Figure 15) which raise the low mode mechanical natural frequency, helped to minimize the size of the primary bottle, minimize pulsation transmission to the yard piping, and control vibration and stress of the cylinder-manifold system. The predicted vibration response for the modified cylinder-manifold system is shown in Figure 16.

CONCLUSIONS

The ability to animate acoustical modes in real piping systems has proven to be a valuable aid in understanding the responses in piping systems. The use of this technique allows the analyst to more efficiently determine potential modifications to control excessive pulsation levels. In addition, the mechanical and acoustical aspects must be considered simultaneously to obtain the optimum design for vibration control in reciprocating compressor piping systems.

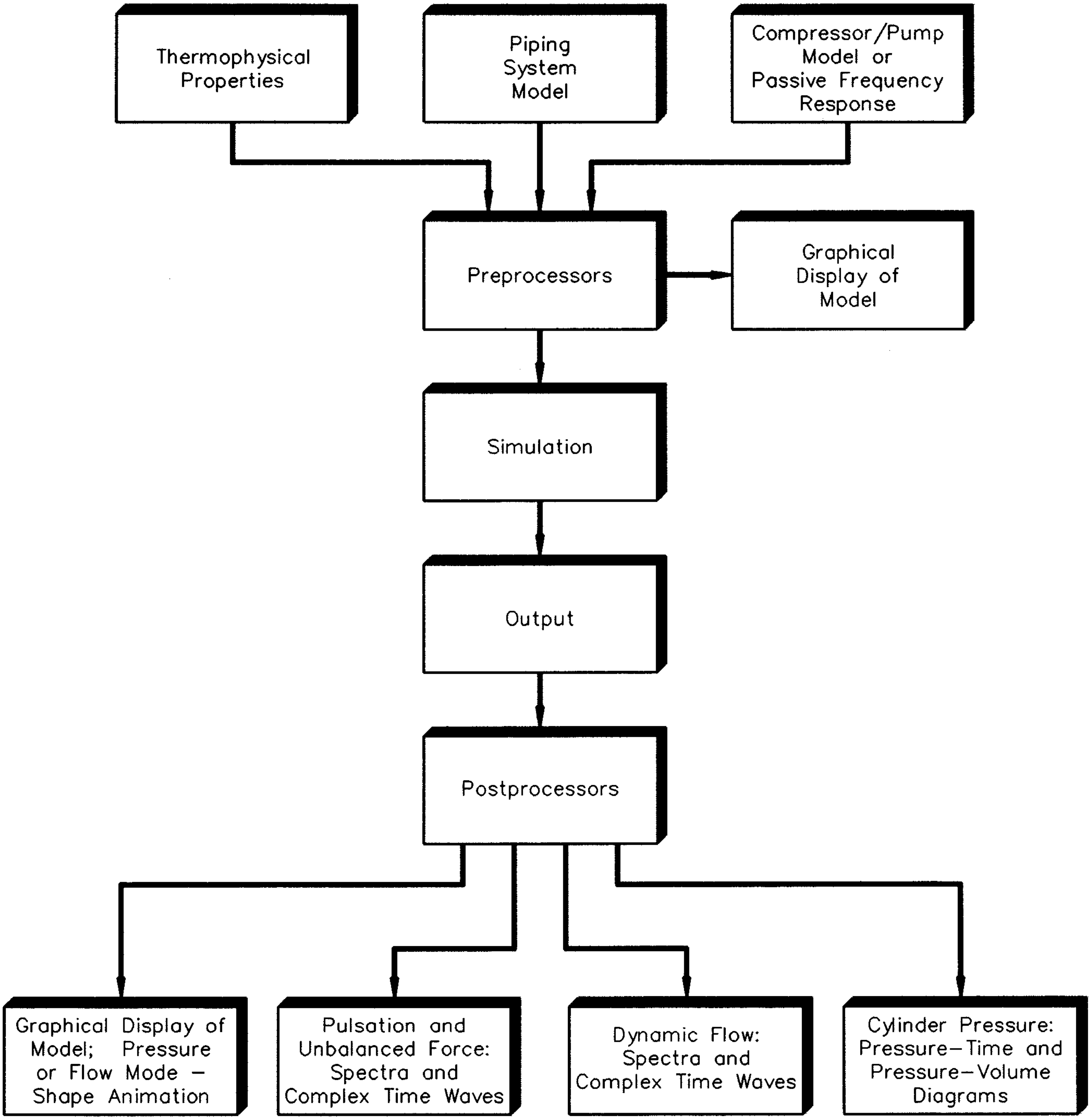


Figure 1: Block Diagram of Digital Computer Based Acoustic Simulation System

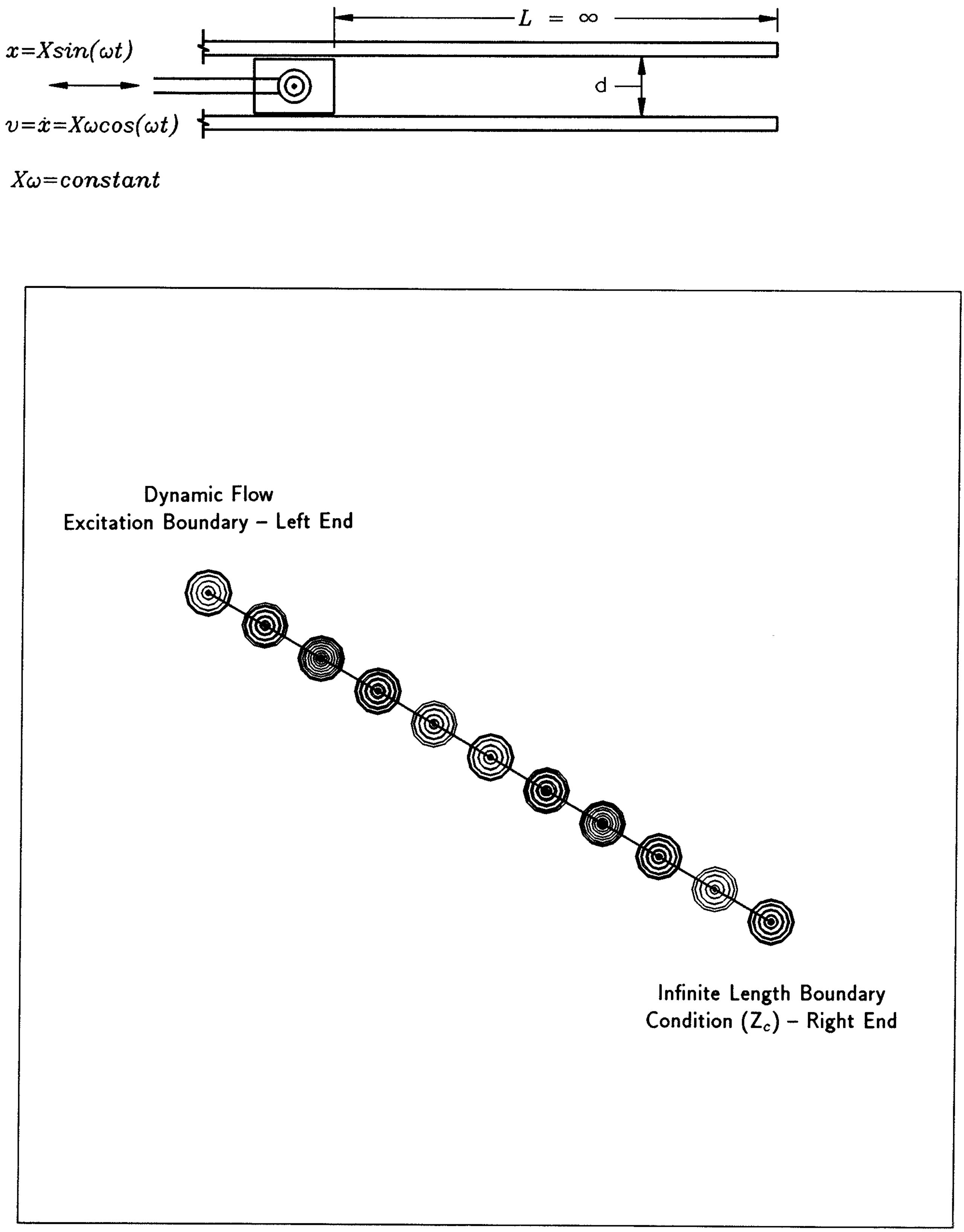


Figure 2: Animation of Traveling Wave in Infinite-Length, Constant Diameter Pipe

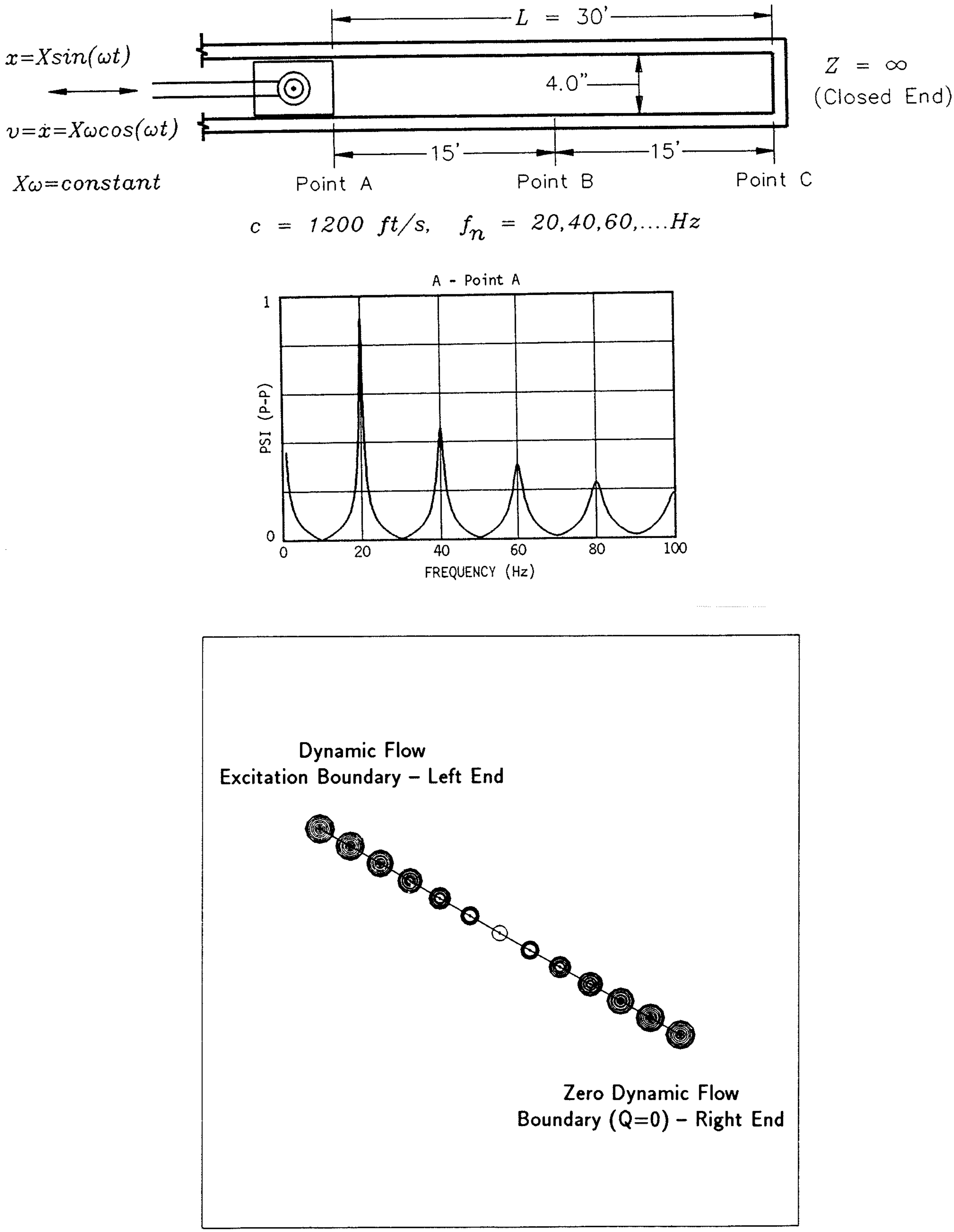


Figure 3: Frequency Response (Middle) and Animation of Fundamental Mode of Closed–Closed (Constant Diameter) Pipe

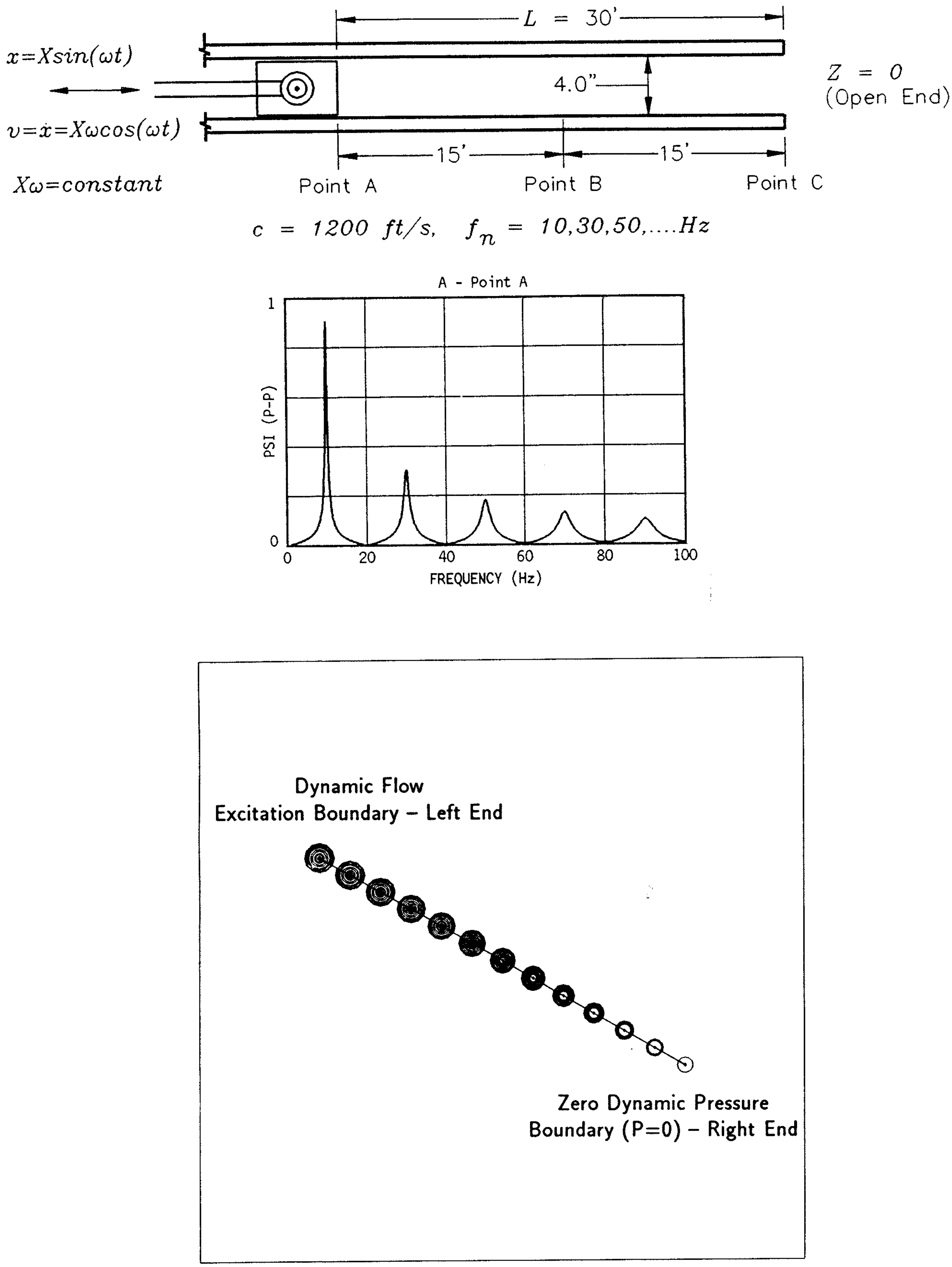


Figure 4: Frequency Response (Middle) and Animation of Fundamental Mode of Closed–Open (Constant Diameter) Pipe

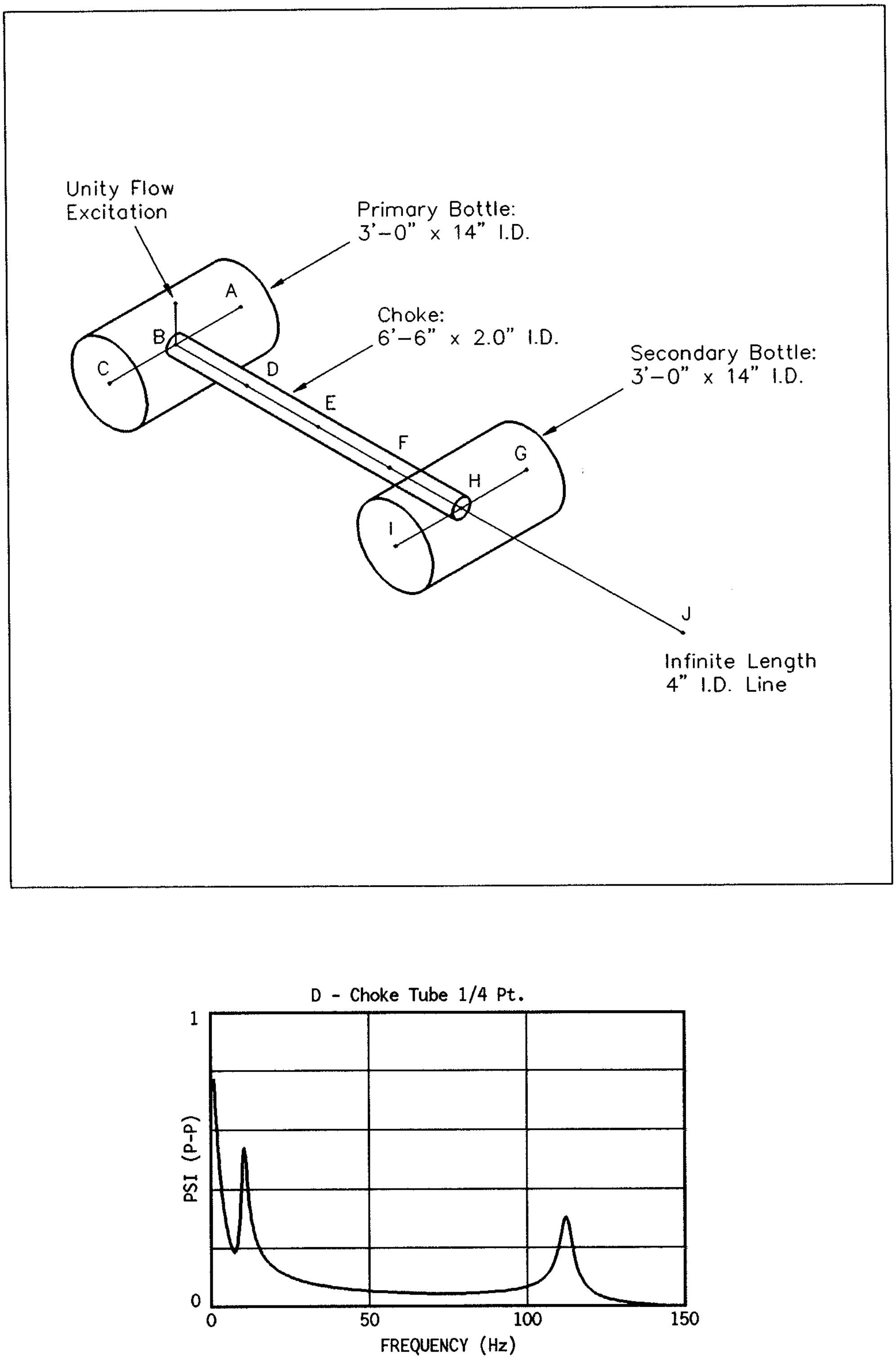


Figure 5: Volume-Choke-Volume Filter (Top) and Frequency Response to Unity Flow Excitation (Bottom) ($F_H = 10.5$ Hz, $F_P = 112.5$ Hz)

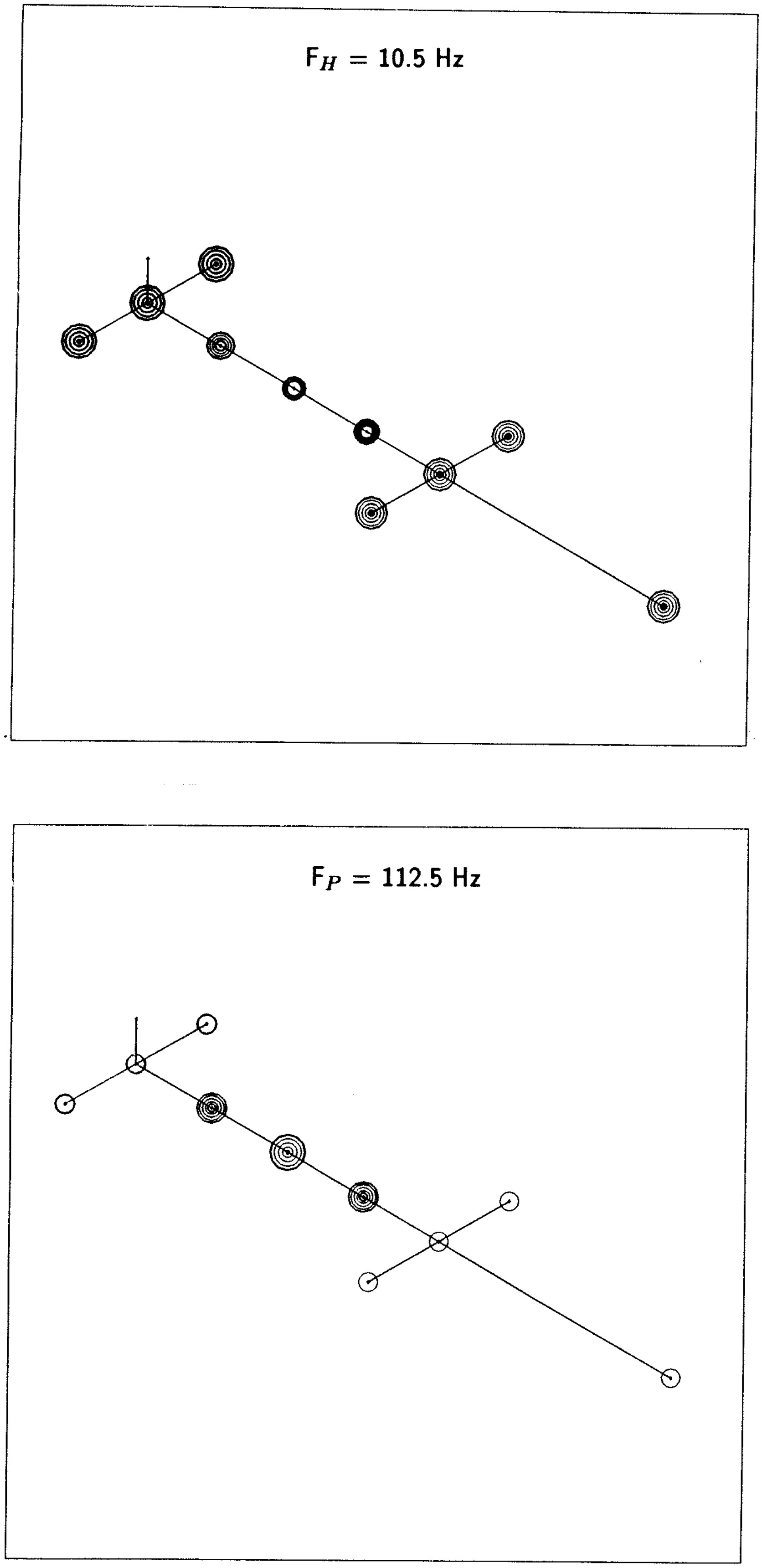


Figure 6: Animation of Low-Mode and Choke Tube (Open-Open) Mode of Volume-Choke-Volume Filter of Figure 5

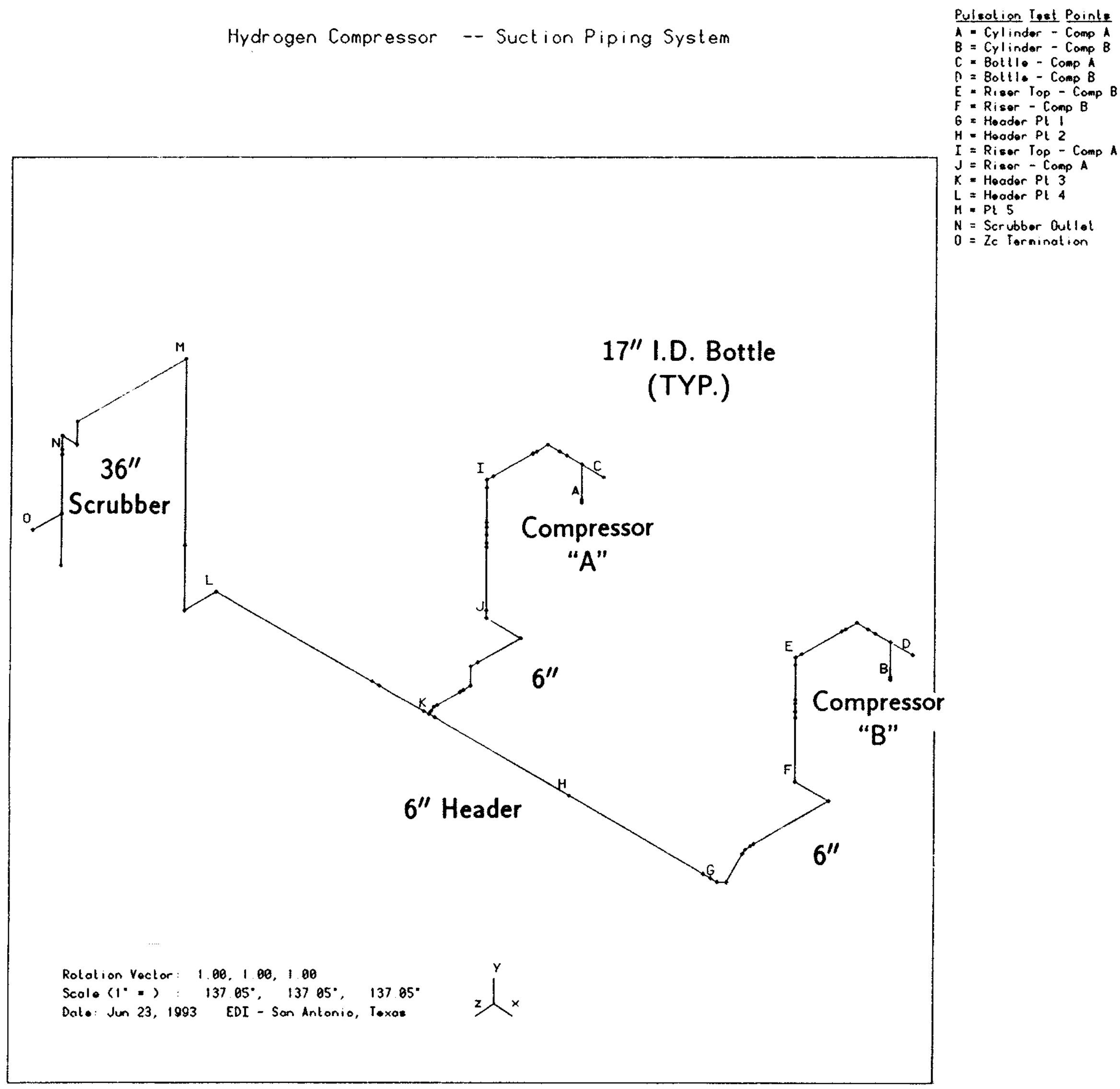


Figure 7: Suction Piping System of Two H_2 Compressors

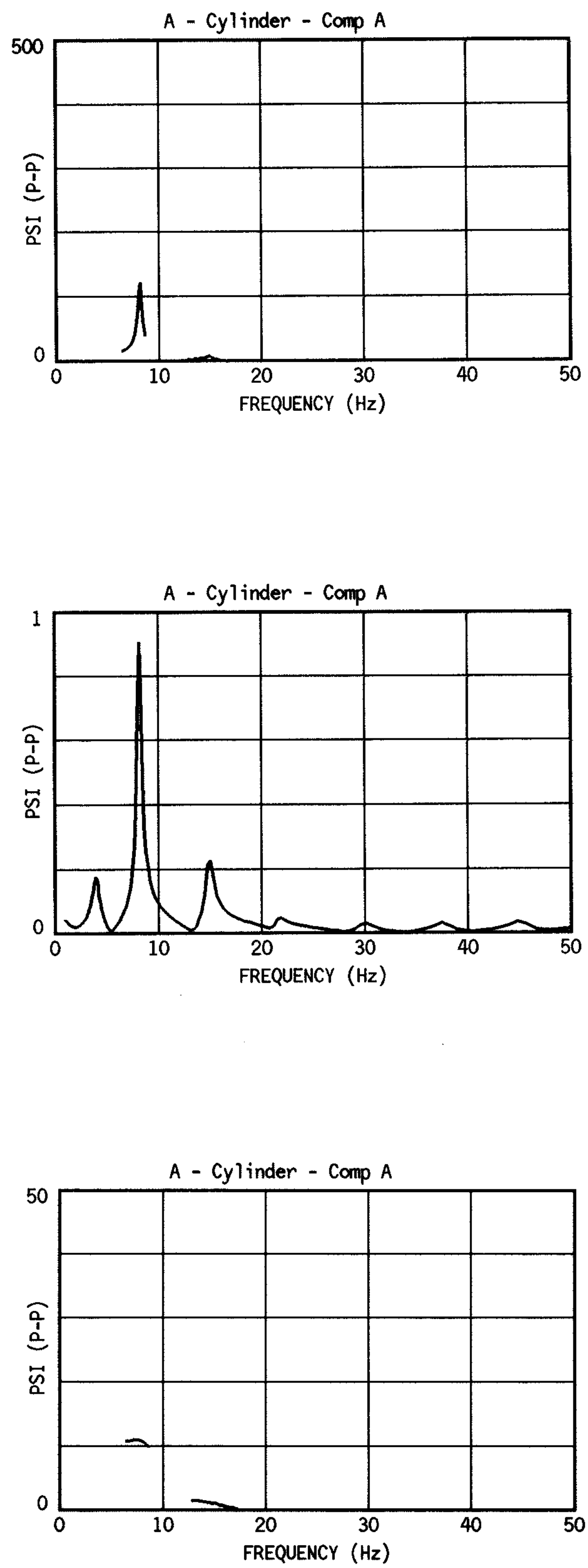


Figure 8: Pulsation Spectra at Compressor A Cylinder Flange for Two, 450 RPM, Single Acting H_2 Compressors/Suction Piping System

Top — Predicted Pulsation at 1× = 120 psi p–p at Cylinders
Middle — Passive Frequency Response – Resonance at 8.15 Hz
Bottom — Orifices Installed at Suction Bottle Inlets

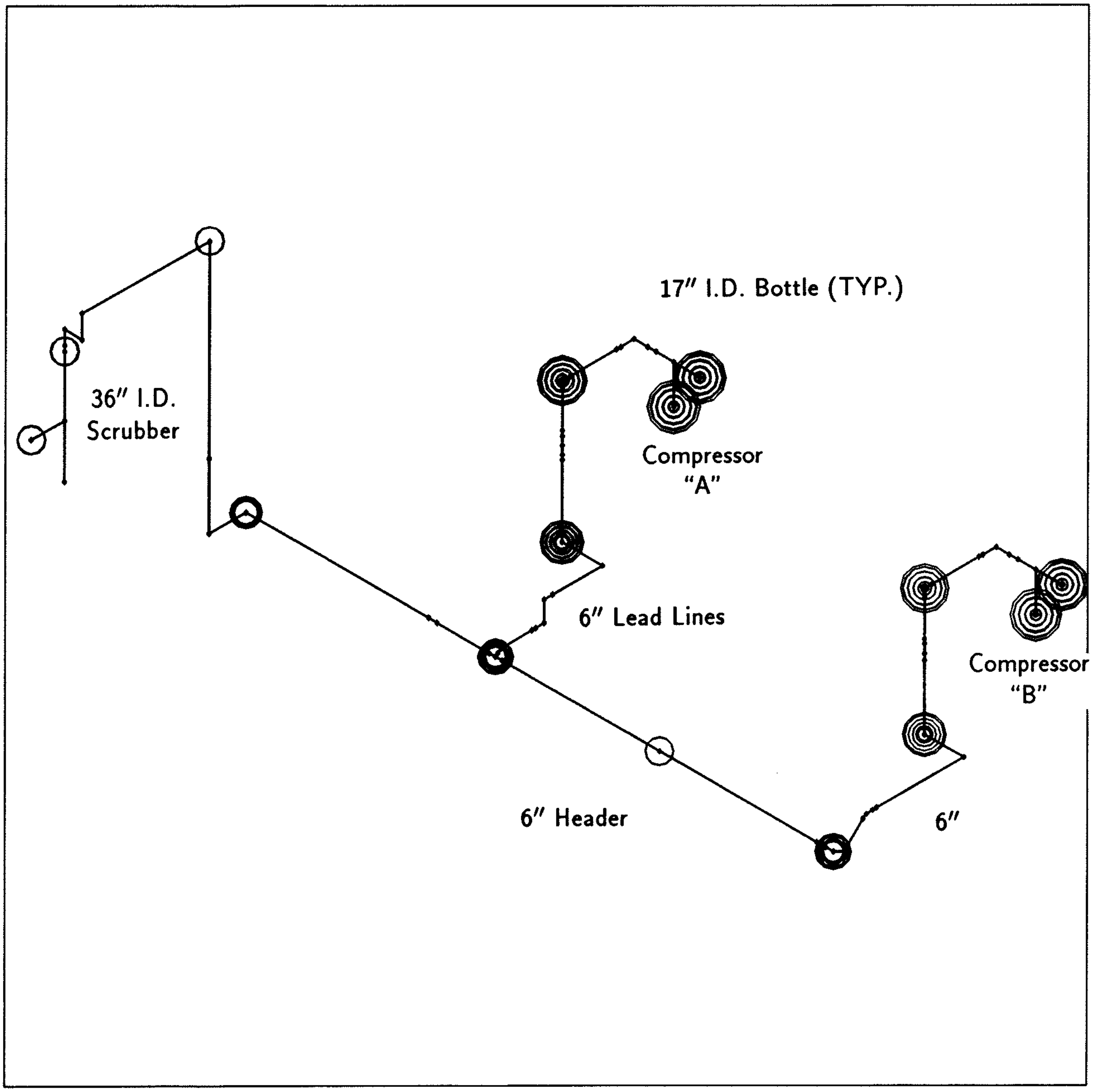
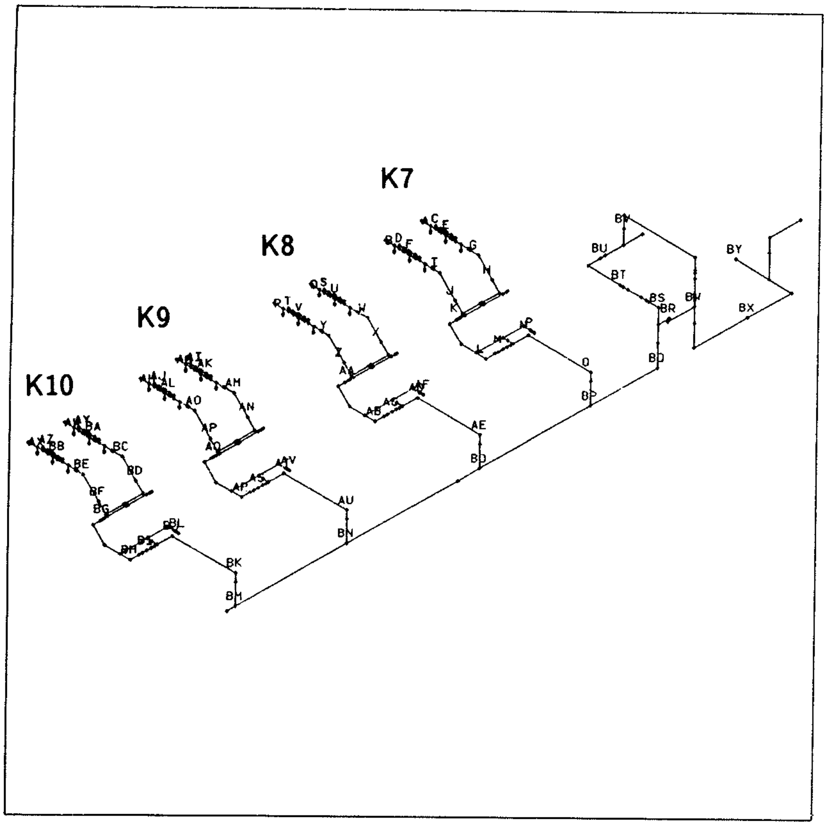


Figure 9: H_2 Compressor Suction System — Animated Mode Shape at 8.15 Hz

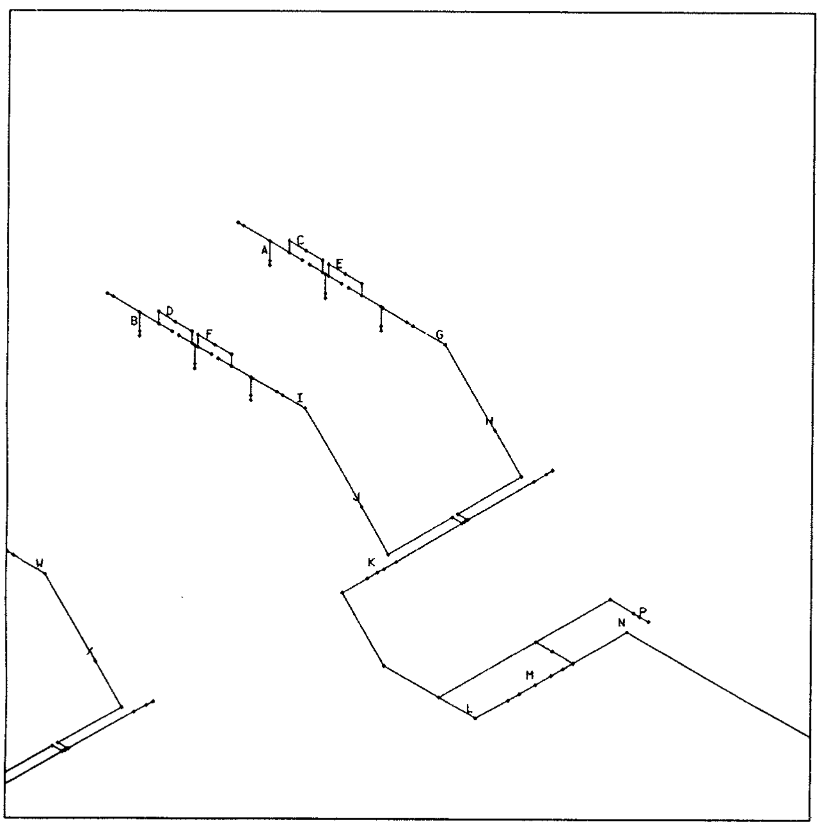
1st Stage Suction



Pulsation Test Points

A = K7 Cyl 1 Flg
B = K7 Cyl 2 Flg
C = K7 Chk 1A Mid
D = K7 Chk 2A Mid
E = K7 Chk 1B Mid
F = K7 Chk 2B Mid
G = K7 Bottle 1 45 Elbow
H = K7 Sec Chk Mid 1
I = K7 Bottle 2 45 Elbow
J = K7 Sec Chk Mid 2
K = K7 Sec Bot
L = K7 Lead TP 2
M = K7 Lead Valve
N = K7 Lead Midpt
O = K7 Lead TP 3
P = K7 Bypass
Q = K8 Cyl 1 Flg
R = K8 Cyl 2 Flg
S = K8 Chk 1A Mid
T = K8 Chk 2A Mid
U = K8 Chk 1B Mid
V = K8 Chk 2B Mid
W = K8 Bottle 1 45 Elbow
X = K8 Sec Chk Mid 1
Y = K8 Bottle 2 45 Elbow
Z = K8 Sec Chk Mid 2
AA = K8 Sec Bot
AB = K8 Lead TP 2
AC = K8 Lead Valve
AD = K8 Lead Midpt
AE = K8 Lead TP 3
AF = K8 Bypass
AG = K9 Cyl 1 Flg
AH = K9 Cyl 2 Flg
AI = K9 Chk 1A Mid
AJ = K9 Chk 2A Mid
AK = K9 Chk 1B Mid
AL = K9 Chk 2B Mid
AM = K9 Bottle 1 45 Elbow
AN = K9 Sec Chk Mid 1
AO = K9 Bottle 2 45 Elbow
AP = K9 Sec Chk Mid 2
AQ = K9 Sec Bot
AR = K9 Lead TP 2
AS = K9 Lead Valve
AT = K9 Lead Midpt
AU = K9 Lead TP 3
AV = K9 Bypass
AW = K10 Cyl 1 Flg
AX = K10 Cyl 2 Flg
AY = K10 Chk 1A Mid
AZ = K10 Chk 2A Mid
BA = K10 Chk 1B Mid
BB = K10 Chk 2B Mid
BC = K10 Bottle 1 45 Elbo
BD = K10 Sec Chk Mid 1
BE = K10 Bottle 2 45 Elbo
BF = K10 Sec Chk Mid 2
BG = K10 Sec Bot
BH = K10 Lead TP 2
BI = K10 Lead Valve
BJ = K10 Lead Midpt
BK = K10 Lead TP 3
BL = K10 Bypass
BM = K10 Header Tie-in
BN = K9 Header Tie-in
BO = K8 Header Tie-in
BP = K7 Header Tie-in
BQ = Yard TP 1
BR = Yard TP 2
BS = Yard TP 3
BT = Yard TP 4
BU = Filter Outlet
BV = Filter Inlet
BW = Yard TP 5
BX = Yard TP 6
BY = ZC End

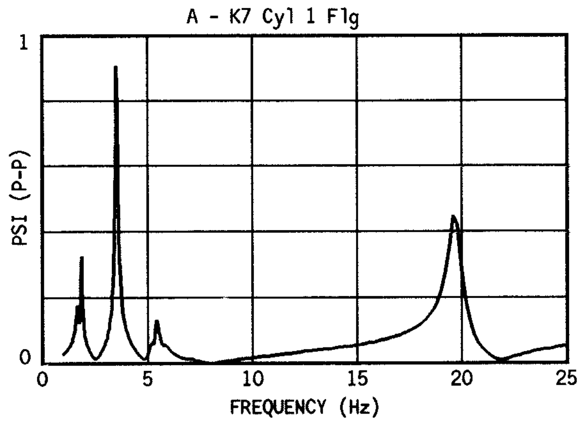
Top – Complete Model



Bottom – "Zoom" of Compressor "K7" Piping

Figure 10: Suction Piping Model of Four Natural Gas Compressors

Pulsation Spectra



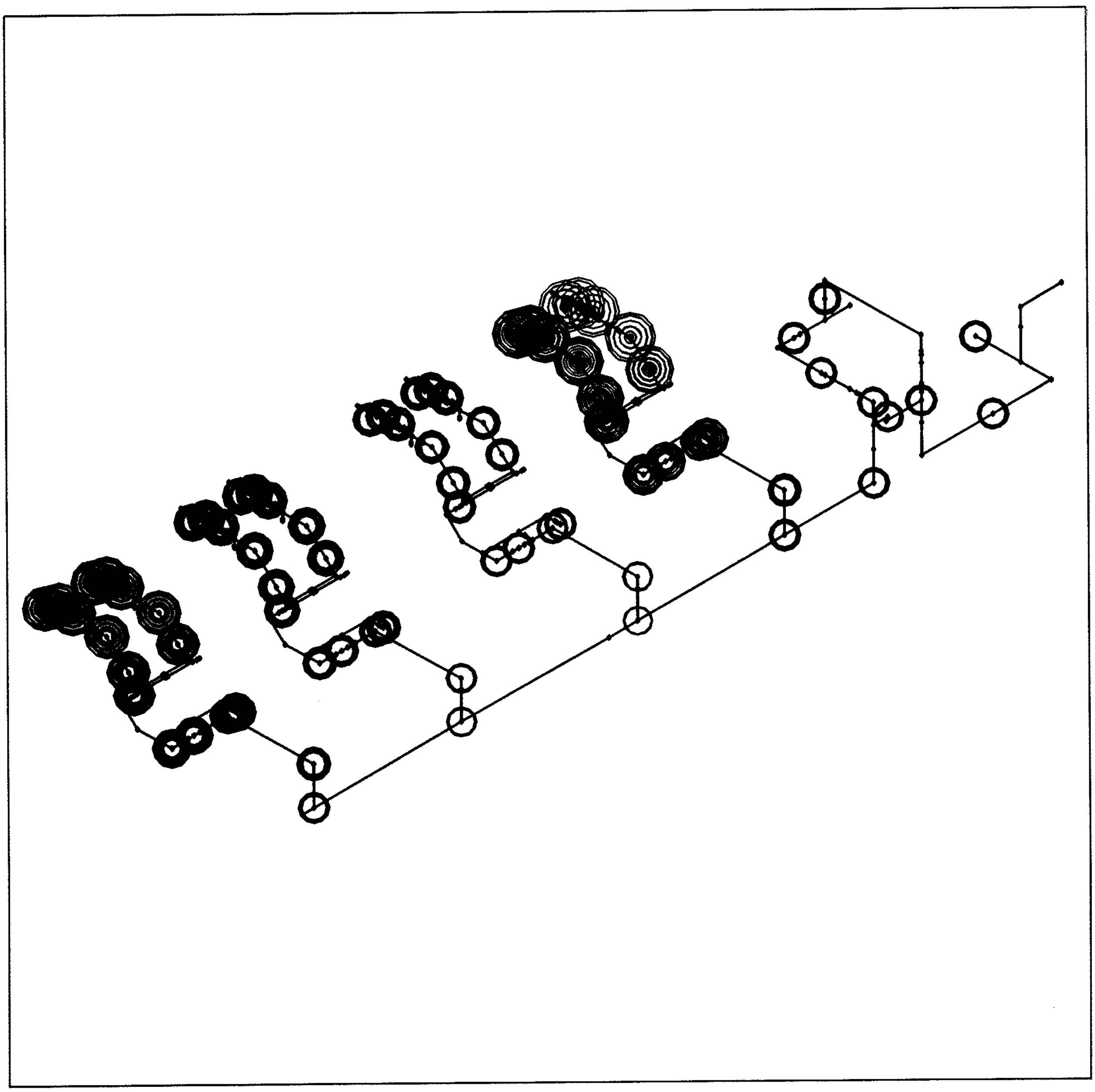


Figure 11: Passive Frequency Response (Top) and Animation of Lowest Acoustic Natural Frequency at 1.7 Hz (Bottom)

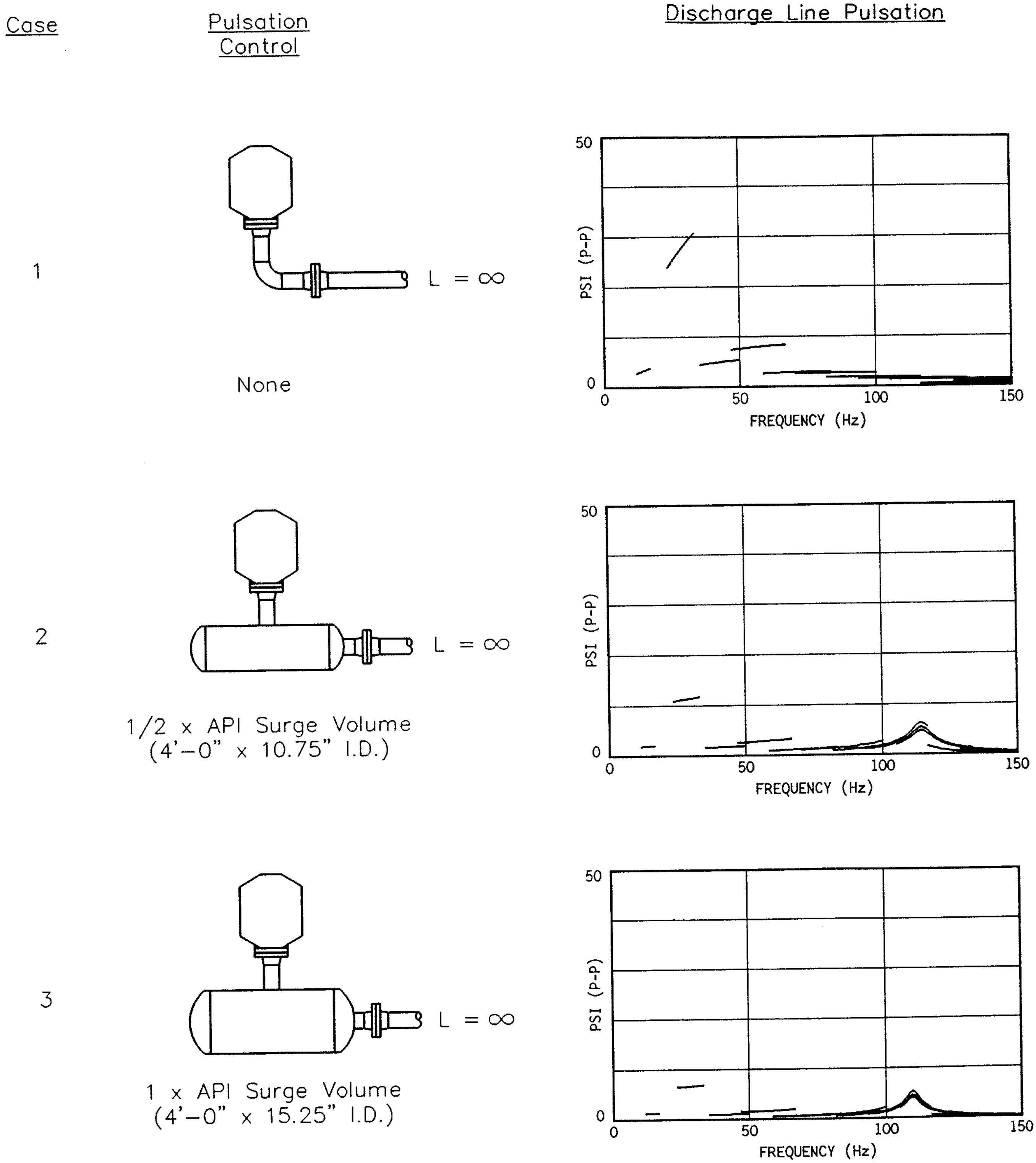
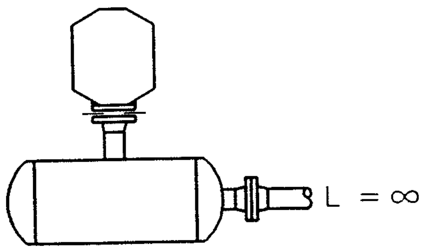


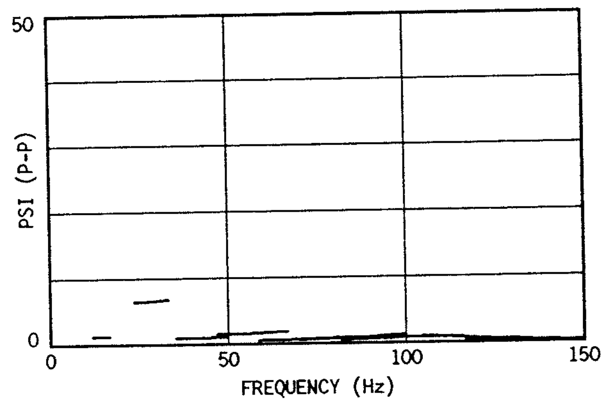
Figure 12: Comparison of Pulsation Control Devices for (Single Cylinder) Compressor Discharge Piping System

Case	Pulsation Control	Discharge Line Pulsation

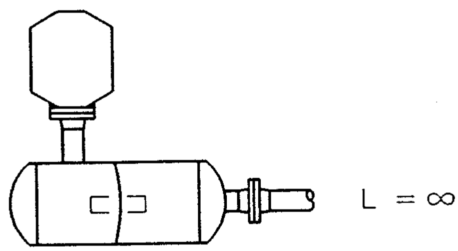
4



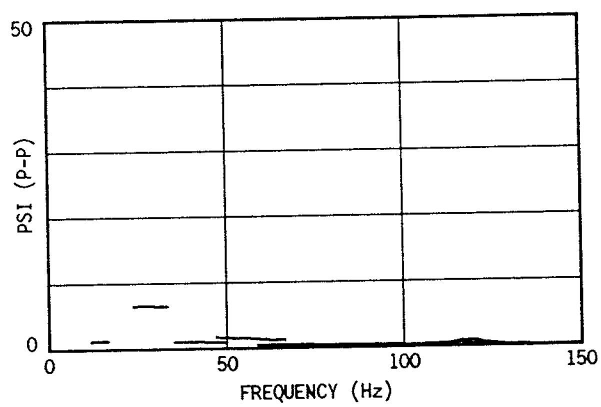
1 x API Surge Volume
(4'–0" x 15.25" I.D.)
0.125% ΔP Orifice @ Cyl. Nozzle



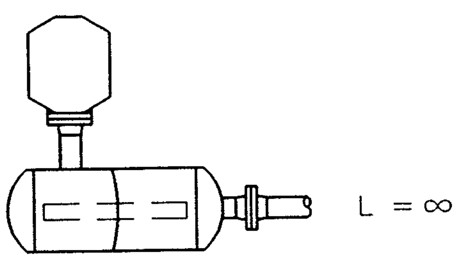
5



Volume–Choke–Volume $(f \approx 4x)$
Total Vol. = 1 x API Surge Volume
(4'–0" x 15.25" I.D.)
5" x 2.624" I.D. Choke
(0.44% ΔP)



6



Volume–Choke–Volume $(1x<f<2x)$
Total Vol. = 1 x API Surge Volume
(4'–0" x 15.25" I.D.)
3'–0" x 2.728" I.D. Choke
(0.42% ΔP)

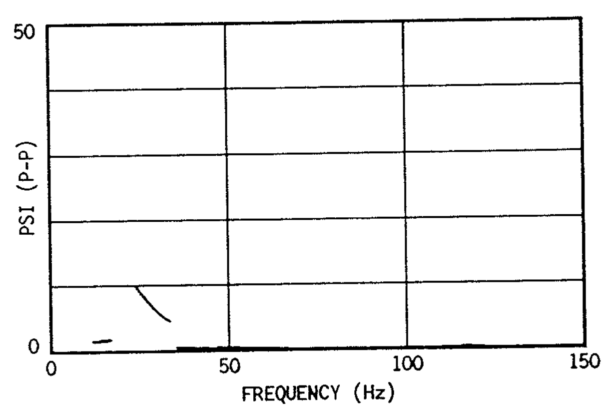
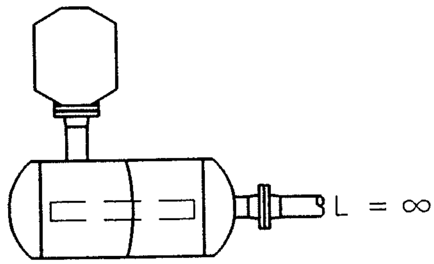


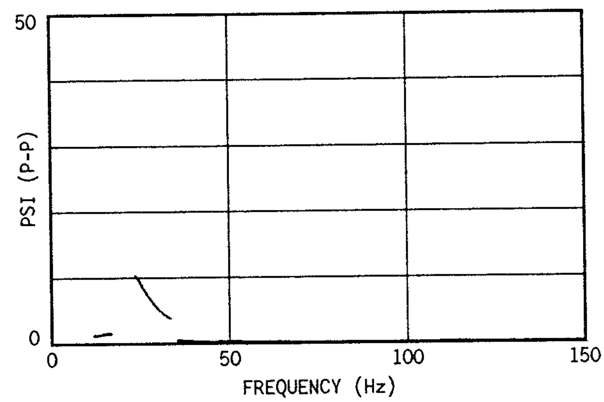
Figure 12: (Continued)

Case	Pulsation Control	Discharge Line Pulsation

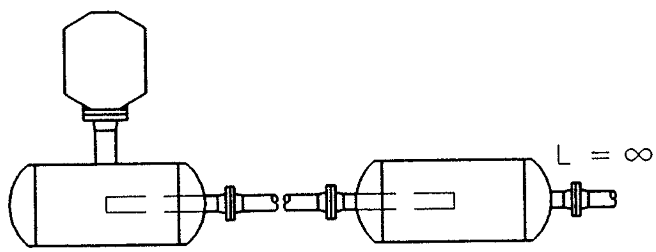
7



Volume–Choke–Volume ($1x<f<2x$)
Total Vol. > 1 x API Surge Volume
(4'–0" x 17.94" I.D.)
3'–0" x 3.152" I.D. Choke
(0.23% ΔP)



8



Volume–Choke–Volume ($f<1x$)
Vol. Each Bottle > 1 x API Surge Volume
(5'–0" x 15.25" I.D.)+(5'–0" x 15.25" I.D.)
10'–0" x 2.9" I.D. Choke
(0.42% ΔP)

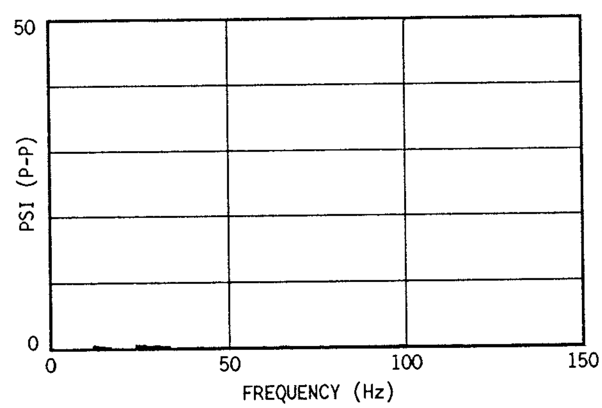
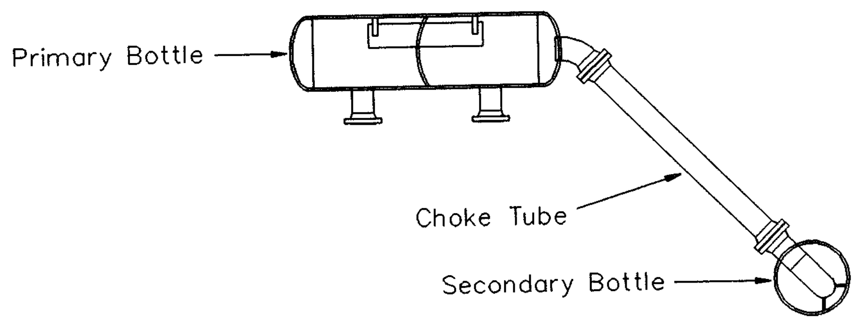
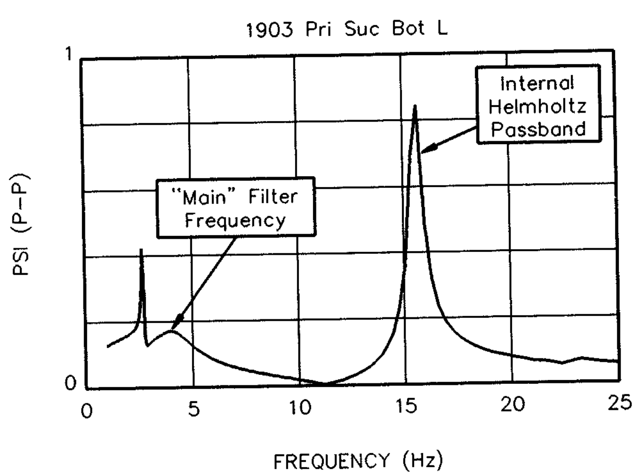


Figure 12: (Continued)

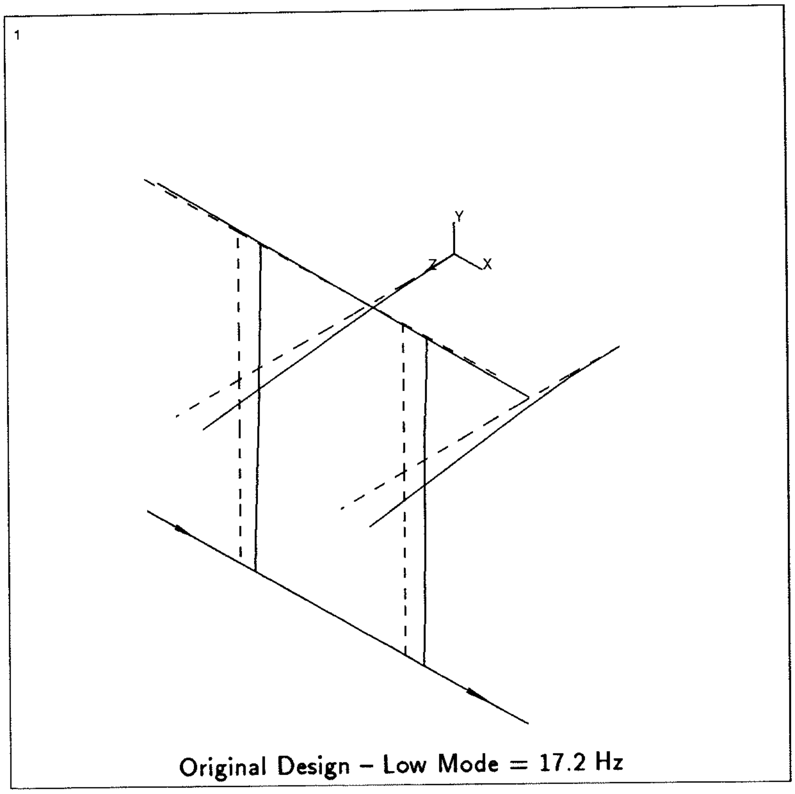
Suction Filter Layout





Passive Frequency Response in
Primary Suction Bottle

Figure 13: Suction Filter Layout and Passive Response



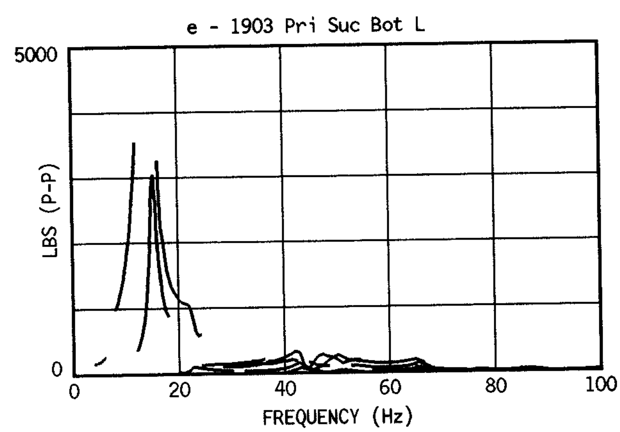
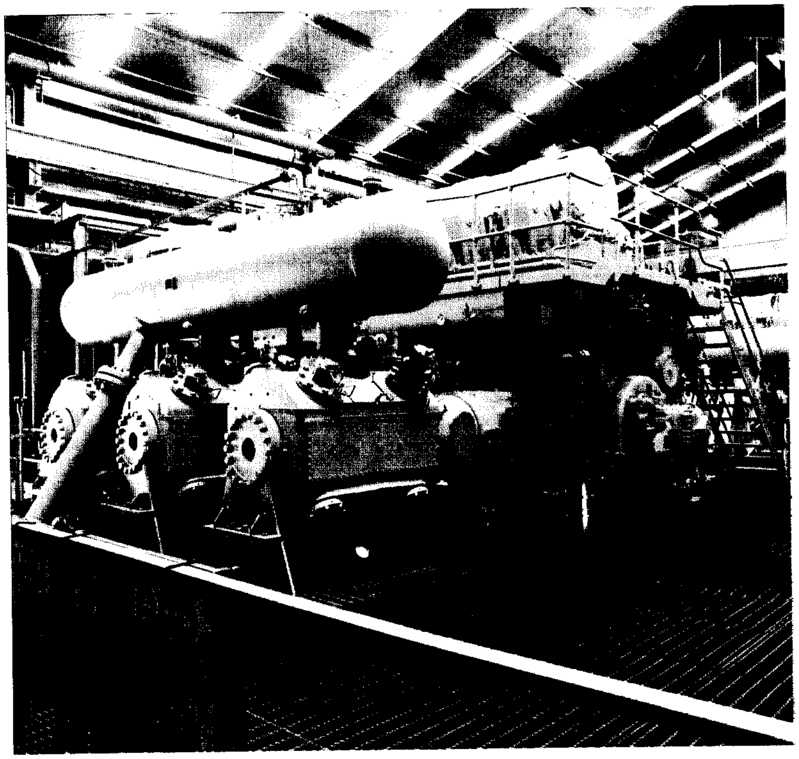
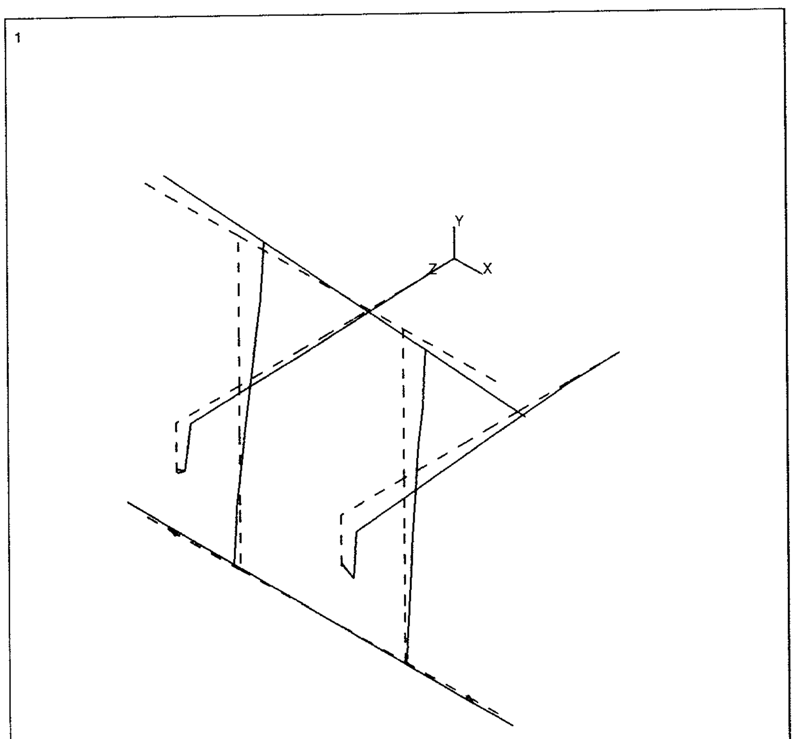


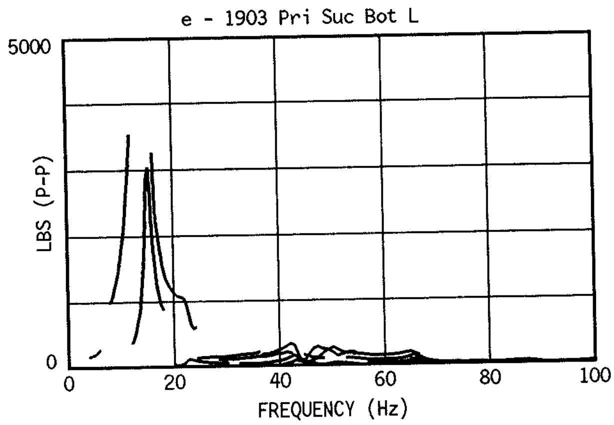
Figure 14: Calculated "Low Mode" Mechanical Natural Frequency Near Internal Helmholtz Resonance (~16 Hz)





With Cylinder Supports – Low Mode = 31.7 Hz

Figure 15: Dynamic Cylinder Supports Used to Raise Low Mode Frequency



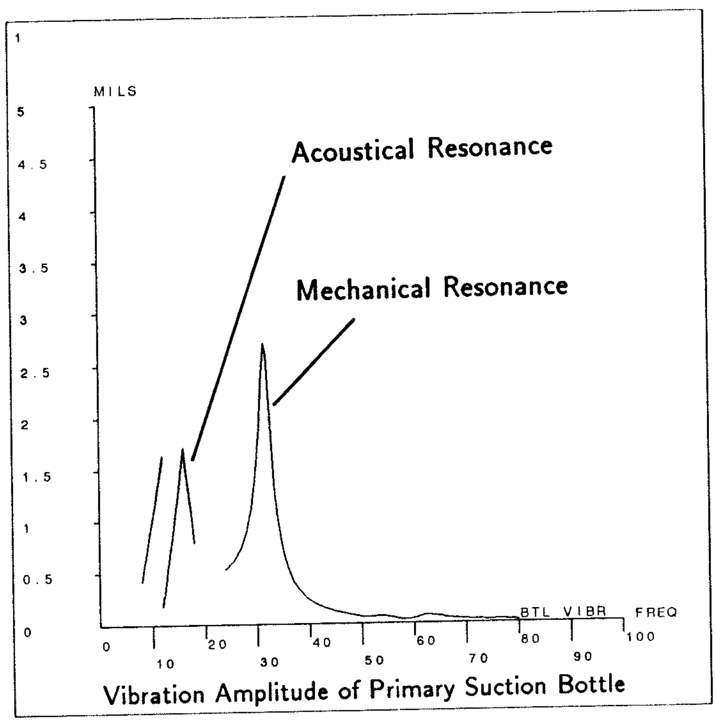


Figure 16: Predicted Vibration Amplitude Due to Shaking Force Spectrum Applied to Primary Suction Bottle