Engineering the Reliability of Reciprocating Compressor Systems

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Introduction

Excessive vibration in reciprocating compressor piping is a major cause of machinery downtime in refineries and petrochemical plants and is often the result of inadequate pulsation control. Often, this is brought about by a misunderstanding of the importance of proper acoustical design procedures and the desire to minimize costs of the design, fabrication and installation of the equipment. Compromises in the acoustical design can result in excessive pulsation-induced shaking forces in the piping system. It often means that, upon startup, extensive bracing of the piping is needed on an expedited basis to allow operation, or worse yet, the system must be completely shutdown.

API 618 Design Approaches 2 and 3 [1] are specified for most high horsepower, high pressure units to ensure acceptable dynamic stresses. Many companies specify that Design Approach 3 studies be performed; however, sometimes they insist that only a simple surge-type pulsation bottle be used on the suction and the discharge manifold systems, which contradicts the guidelines laid down in the API standard. For low molecular weight gases, such as gases with high hydrogen content, it is often possible to achieve acceptable pulsation levels with simple surge volumes and orifice plates. However, applying Design Approach 3 for other systems often requires the use of volume-choke-volume (Helmholtz) filter designs in order to achieve adequate filtering of the pulsations that are transmitted into the piping system. This is particularly true for multiple units in parallel and for units which have variable speed and loading conditions.

Numerous design evaluations and field tests have proven that that the philosophy of specifying a "One Bottle Design" to lower initial costs is in poor judgement. Consultants who claim that "Two Bottle Filter Designs" are rarely or never needed for control of pulsation are putting their clients at unnecessary risk, either as a result of inaccurate modeling techniques or lack of experience in actual field installations. Actually, in many systems, it can be difficult to obtain acceptable pulsation levels, even when two volume filter systems are used.

Data from case histories will show the consequences of inadequate pulsation control. Improved pulsation attenuation and control of existing one bottle systems can sometimes be achieved by the use of orifices; however, the operating company will have to pay for the excessive pressure drop over the life of the plant. The initial capital cost of the extra bottle in a filter system can be trivial when the costs associated with pressure drop, downtime, pipe supports, long-term maintenance and safety are considered.

Design Procedures

In order to engineer the reliability of reciprocating compressor systems, it is necessary to use acoustic simulation techniques to develop the required pulsation control systems. Figure 1 is a block diagram describing the (computer-based) acoustic simulation procedure developed and tested by Engineering Dynamics Incorporated (EDI) over the past 13 years. The simulation code is based on the one-dimensional, damped wave equation which is derived from the momentum, state and continuity equations. Surrounding the simulation code are various input file structures, pre-processors and post-processors which facilitate the use of the system.

This analysis tool is used to simulate 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 comparisons of various typical bottle designs for a compressor piping systems. The adequacy of the pulsation control system depends upon the knowledge and experience of the analyst.

The pulsation amplitude and phase for any frequency can be animated to help understand the resonant mode shapes for any piping configuration. The amplitude of the pulsations 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 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 in the diameter a decrease in pressure.

Due to its importance in pulsation control, the volume-choke-volume filter system acoustical characteristics will be discussed first.

Volume-Choke-Volume Filter Characteristics

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 2 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 3 shows animated mode shapes for the low-mode filter frequency, and of the choke tube pass-band frequency.

Volume-choke-volume filters have, in addition to two compliance components (two volumes), a choke tube which acts as an acoustical inertance to resist changes in velocity of the fluid contained in the choke tube. As for the single surge volume, these lumped compliance and inertance properties are valid at frequencies below the open-open resonant frequencies of the choke tube length, and the closed-closed resonant frequencies of the bottle lengths. The mechanical analogy of such a filter is a high flexibility (volume) in series with a large mass (choke) and another high flexibility (volume). At frequencies above the resonant frequencies of the mass spring system, the piston motion is isolated (filtered) from the fluid in the pipe, with the degree of isolation increasing with frequency due to the momentum characteristics of the choke tube fluid. The acoustic filter is similar to L-C filters used in electrical systems.

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 [2,3].

The type of pulsation suppression device required depends on the degree of pulsation control and bottle unbalanced force control desired. Figure 4 shows typical volume bottle designs used in compressor installations.

In certain cases, a surge volume (Figure 4 — Type I) 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, various degrees 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 can be higher than that of a single surge volume in some cases. The actual size of the surge volumes or filters depends on the speed range of the compressor or pump, flow capacity, the stage pressure ratio, the thermo-physical 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 inch bore with a 6 inch 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 5.

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 1x and 2x running speed).

Case 7: A volume-choke-volume acoustic filter with the Helmholtz frequency tuned between 1x and 2x 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 1x 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 2x running speed — from 23.3 to 33.3 Hz). The simulation predicts a maximum amplitude at 2x running speed of approximately 32 psi p–p. The high amplitude pulsations at 2x 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 2x 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 2x running speed is lowered to as 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 2x 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 (4x 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 2x 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 1x and 2x 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 in case 5. Note that the maximum predicted response at 2x 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 2x 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 2x 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 1x and 2x 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 1x 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 5. 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.

Cost Comparisons of Filter Systems

This parametric study showed the relative improvements that can be obtained with the use of two vessel volume-choke-volume filters. Many designers who try to minimize initial capital costs in the installation phase of a project try to accomplish the pulsation control through the use of a one bottle system. The initial cost evaluation of a one versus two bottle filter system may show that the one bottle system is less expensive; however, it is important to consider other cost factors.

A cost comparison was made between a one and two bottle solution to a vibration problem on a three cylinder compressor system [4]. This analysis showed that the original discharge bottle used 20 feet of 42 inch pipe with two baffle plates with an exit on the end of the bottle. The costs of this bottle was compared to a two bottle solution which used two smaller 30 inch diameter bottles that were each 20 feet long. The cost analysis which included the material and labor costs for the fabrication showed that the two bottle system was actually approximately $800 less expensive than the original one bottle system. In addition to the lower initial costs, the two bottle system had lower pressure drop, less costly bottle supports and tie-downs, lower weight loads on the compressor cylinders, higher mechanical natural frequencies and comparable physical space requirements. This analysis shows that the use of one bottle designs may not be cost effective when all the various costs are considered.

API 618 Design Approach 3 [1] also recognizes that the two bottle filter design is the preferred design approach. The need for this approach is not new to industry. Barta [5] in 1971 stressed the importance of using a two bottle design when laying out the initial design of packaged compressor units.

When evaluating the costs of a one bottle filter design, the possible additional costs should be considered, such as the cost of possible delays in startup and downtime. In addition, the safety aspects alone are often sufficient justification for the two bottle design.

Note that one bottle filter designs can often be successfully used in low mole weight systems due to the low gas density. Also, large bottle sizes would be required to place the Helmholtz below the first order or between the first and second order. However, the one bottle filter system designs for low mole weight gases do require that orifice plates be installed at selected locations to control the pulsation resonances.

There are some main line gas transmission companies that have used single bottle filter systems with success for several years by installing additional piping supports when vibration problems occurred. When new compressor units are added as the demand for gas increases, this design approach reaches a limit since the pulsation levels in the piping are a result of the combination of the residual pulsations from all the units in parallel. At this limit, it is not possible to properly restrain the piping. It is sometimes necessary to completely change out the one bottle filters with two bottle designs on all the units to achieve safe, reliable operation.

Case Histories

Case histories will be used to help illustrate the importance of designing reliability into the compressor systems in the design stage.

Case History Number 1

A pulsation analysis was performed on a 3300 horsepower, two stage reciprocating compressor which compressed 41 mmscfd of ethane from 385 psi to 1400 psi after the pulsation bottles had been ordered. The bottle design used a nozzle "pipe-back" arrangement in the two cylinder suction and discharge bottles as shown in Figure 4 — Type II. This type of arrangement creates a volume-choke-volume system to filter energy transmitted to the line piping; however, it can result in manifold systems with high shaking forces.

When this system was analyzed, the calculated shaking forces in the discharge bottle were significant at the first nine harmonics as can be seen from the data in Figures 6 and 7 for the double acting and single acting conditions. For double acting conditions, the shaking force at 2 times running speed was 2999 lbs, peak to peak. The shaking forces for the single acting conditions were 1043 lbs p-p at 1 times running speed and 2770 lbs p–p at 2 times running speed. Shaking forces greater than 500 lbs p–p were present at several harmonics.

When the unit was put in operation, pulsation and vibration levels in the piping were severe. Under single acting conditions, it was not possible to operate the machine due to the extremely high vibrations. These high vibrations were experienced due to the high shaking forces on the compressor manifold system and piping. Even after improvement of the supports and clamps, the vibrations were excessive. Since reliability of the unit was vital to the plant operation and profitability, it was decided that the pulsation bottles would be redesigned using two vessel volume-choke-volume filter systems which would reduce the shaking forces in the suction and discharge bottles and lower the forces transmitted to the line piping.

The system was analyzed by EDI and the second stage discharge bottle design as shown in Figure 8 was developed. The frequency analyses of the pulsation and the shaking forces are given in Figures 9 and 10. It can be seen that the shaking forces were significantly reduced with the maximum being only less than 100 lbs p–p. When these pulsation bottles were installed, the vibrations were satisfactory and the unit has performed with low vibration levels for several years.

Note that the discharge bottle design shown in Figure 8 used a volume-choke-volume filter design; however, it was all contained in one longer bottle as shown in Figure 4 — Type III. The second stage suction bottle design that was developed utilized two separate volume bottles as shown in Figure 11. Note that the external portion of the choke tube between the primary and secondary bottle is straight (no elbows) to minimize the shaking forces that can occur.

Case History Number 2

High vibrations of a separator at a compressor station prevented its continuous operation. Pressure pulsations near the scrubber (Figure 12) were measured over a speed range. The data indicated a pulsation resonance excited by 2x running speed near 250 rpm. This second harmonic was order-tracked and indicated that the pulsation resonance was at 8.25 Hz (Figure 13). The pulsation amplitude was 13 psi p-p, and due to the large area of the separator, the shaking force caused approximately 75 mils p–p of vibration at the pulsation resonance (Figure 14).

The pulsation analysis of the existing system indicated that the shaking force near the separator could be as high as 1200 lbs p-p (Figure 15a). This shaking force coupled with a nearby mechanical natural frequency caused the high vibrations.

The station had also had other problems with reliability associated with the compressors and piping. The pulsation levels exceeded the API 618 allowables and the pressure drop through the existing pulsation bottles exceeded the allowable, partly due to the increase in throughput.

It was decided to design new volume-choke-volume filter bottles to reduce the pulsations and pressure drop. The basic design of the two bottle filter on the suction system is shown in Figure 16. The secondary bottle had to be carefully designed to be located in the available space.

The summary of the improvement in pulsation levels for the existing and modified systems is given in Figure 17. For one unit operating, the maximum shaking force was reduced to 62 lbs p-p at 2 times running speed (Figure 15b). Even when all four units were operating, the predicted shaking force at the separator was only 182 lbs p–p at 2 times running speed (Figure 15c).

Figure 18 gives the vibration of the separator after the modifications were installed. It shows that the vibration has been reduced to approximately 3 mils p–p.

This case history illustrates that redesigns to solve specific vibration problems can also result in an improvement in compressor performance, reduced pressure drop, reduced vibration levels and improved reliability.

Conclusions

To ensure maximum reliability of reciprocating compressor systems, it is generally desirable to use the two bottle volume-choke-volume filter designs. This type of design results in the lowest shaking forces in the compressor manifold system and piping. Many people assume that the one bottle design is the lowest cost design; however, when other costs, such as possible delays and downtime are considered, the two bottle design may even be less costly.

Another reason for using the best pulsation control techniques is to minimize the need for piping supports, such as in skid mounted compressors intended for offshore installations. One recent offshore application which used one bottle filter systems required significant support retrofits after installation, which was very costly since it required several weeks of shutdown.

At plants with many units in parallel, the two bottle pulsation control design prevents (limits) the pulsation cross-talk between units. In reciprocating compressor plants that add centrifugal units, it is easier to ensure that the pulsations from the reciprocating units will not affect the performance of the centrifugal units.

With the modern acoustical analysis tools available to industry today and the ever increasing search for increased reliability, performance, and safety, it is important that the best pulsation control techniques be utilized.

References

1. "Reciprocating Compressors for Petroleum, Chemical, and Gas Industry Services," API Standard 618, American Petroleum Institute, Washington, D.C. (June 1995).

2. Tison, J. D. and Atkins, K. E., "Acoustical and Mechanical Design Concepts—Understanding and Controlling Pulsation," 8th Annual Reciprocating Machinery Conference, Denver, CO (1993).

3. Wachel, J. C. and Tison, J. D., "Vibrations in Reciprocating Machinery and Piping Systems," *Proceedings of 23rd Turbomachinery Symposium*, The Turbomachinery Laboratory, Texas A&M University, College Station, Texas (1994).

4. "One Bottle vs. Two Bottle Designs for Pipeline Applications," Prepared by Staff of the PCRC Design Facility at Southwest Research Institute, San Antonio, TX, PCRC Journal (August 1995).

5. Barta, M. L. and Bass, T. P., "Gas Piping Design for High Speed Reciprocating Compressor Units," ASME Paper 71-Pet-3 (1971).

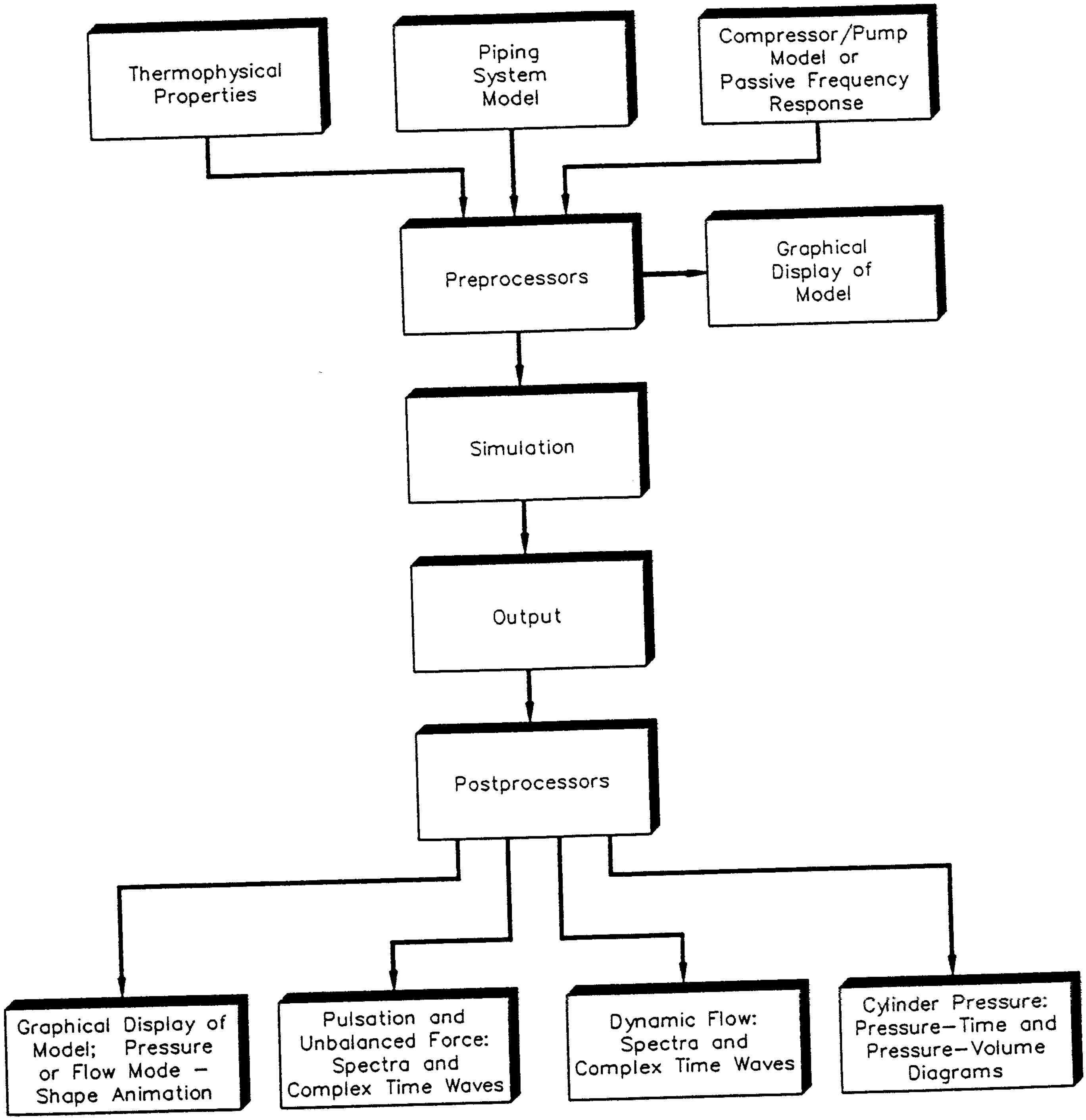
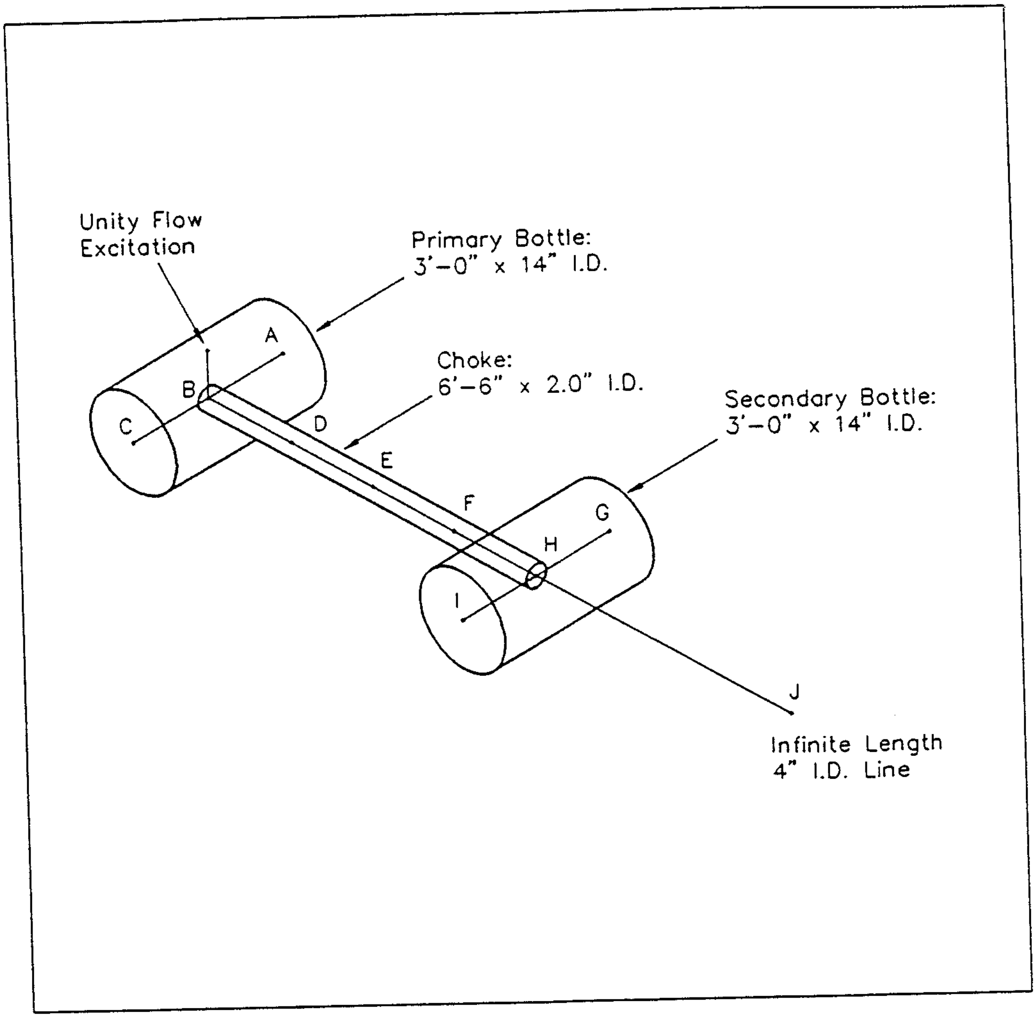


Figure 1: Block Diagram of Digital Computer Based Acoustic Simulation System



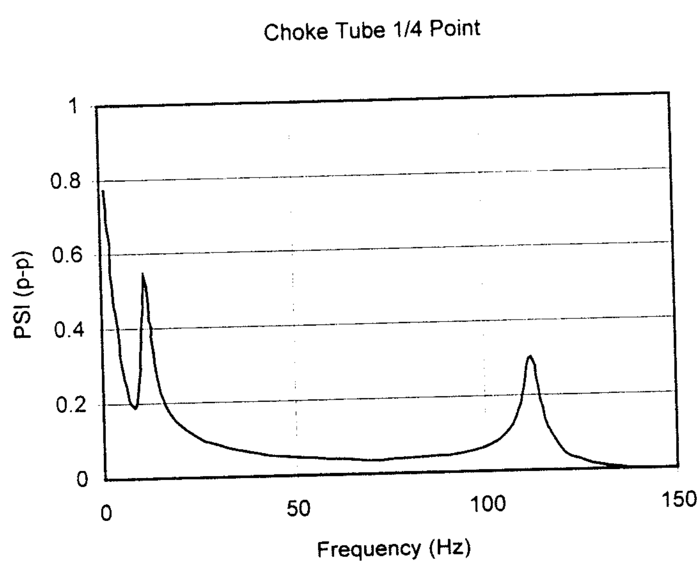


Figure 2: Volume-Choke-Volume Filter (Top) and Frequency Response to Unity Flow Excitation (Bottom) ($F_H = 10.5$ Hz, $F_P = 112.5$ Hz)

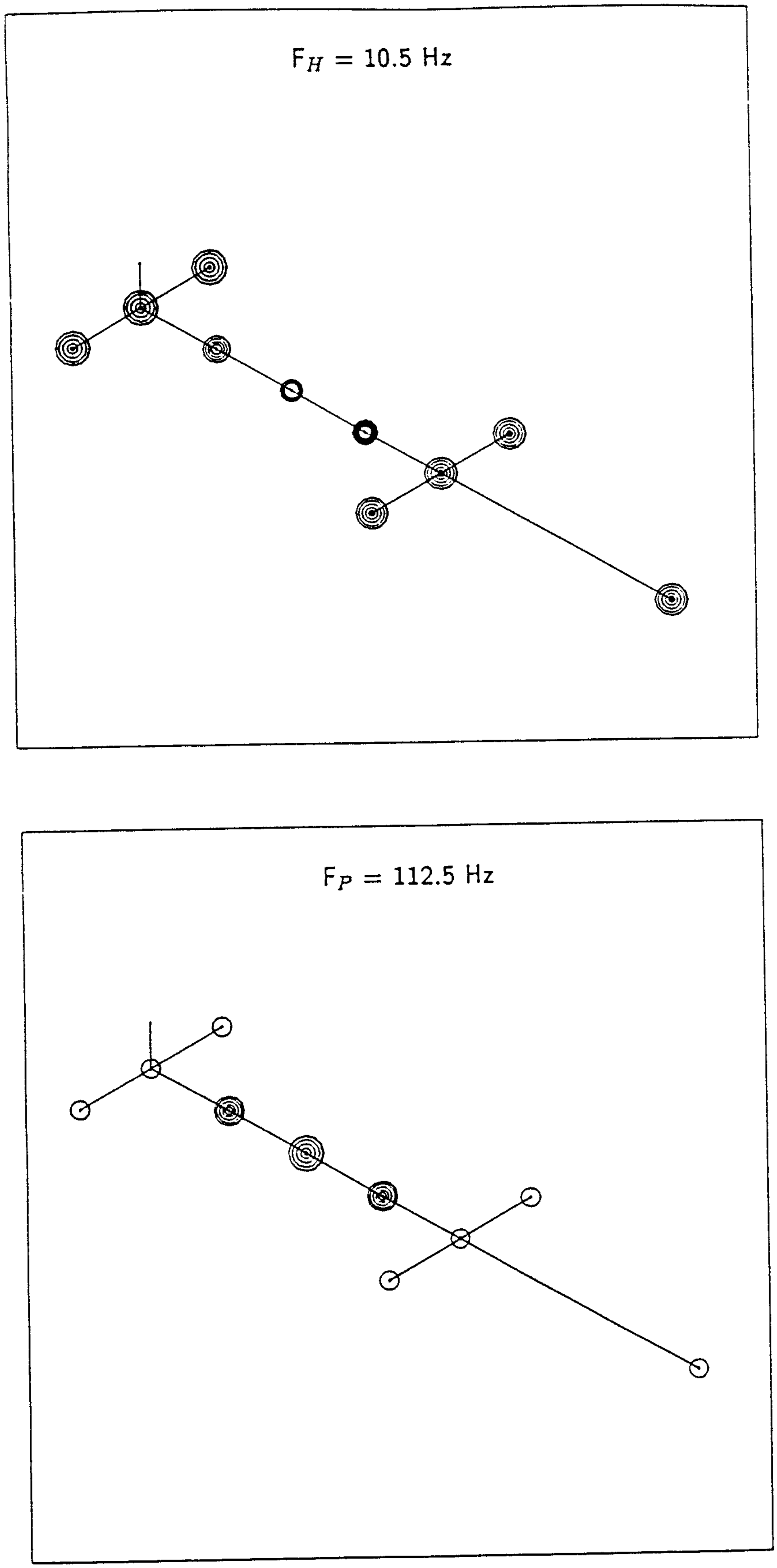
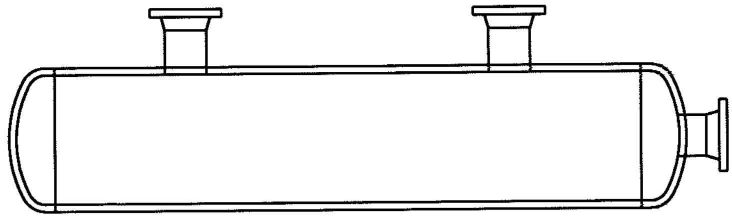


Figure 3: Animation of Low-Mode and Choke Tube (Open-Open) Mode of Volume-Choke-Volume Filter of Figure 2



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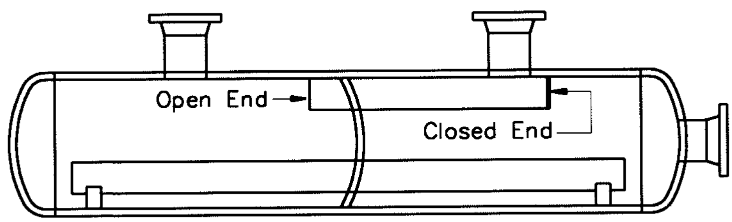
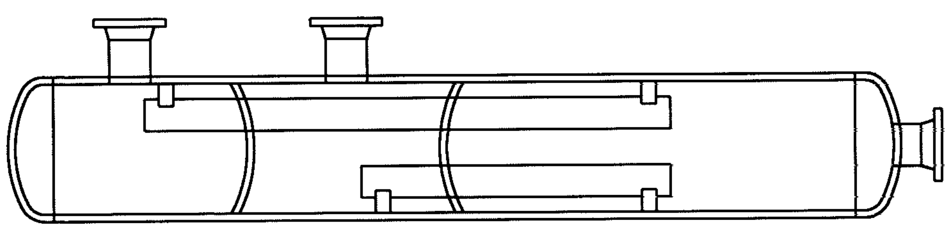


TYPE II



TYPE III

Figure 4: Common Discharge Filter Designs

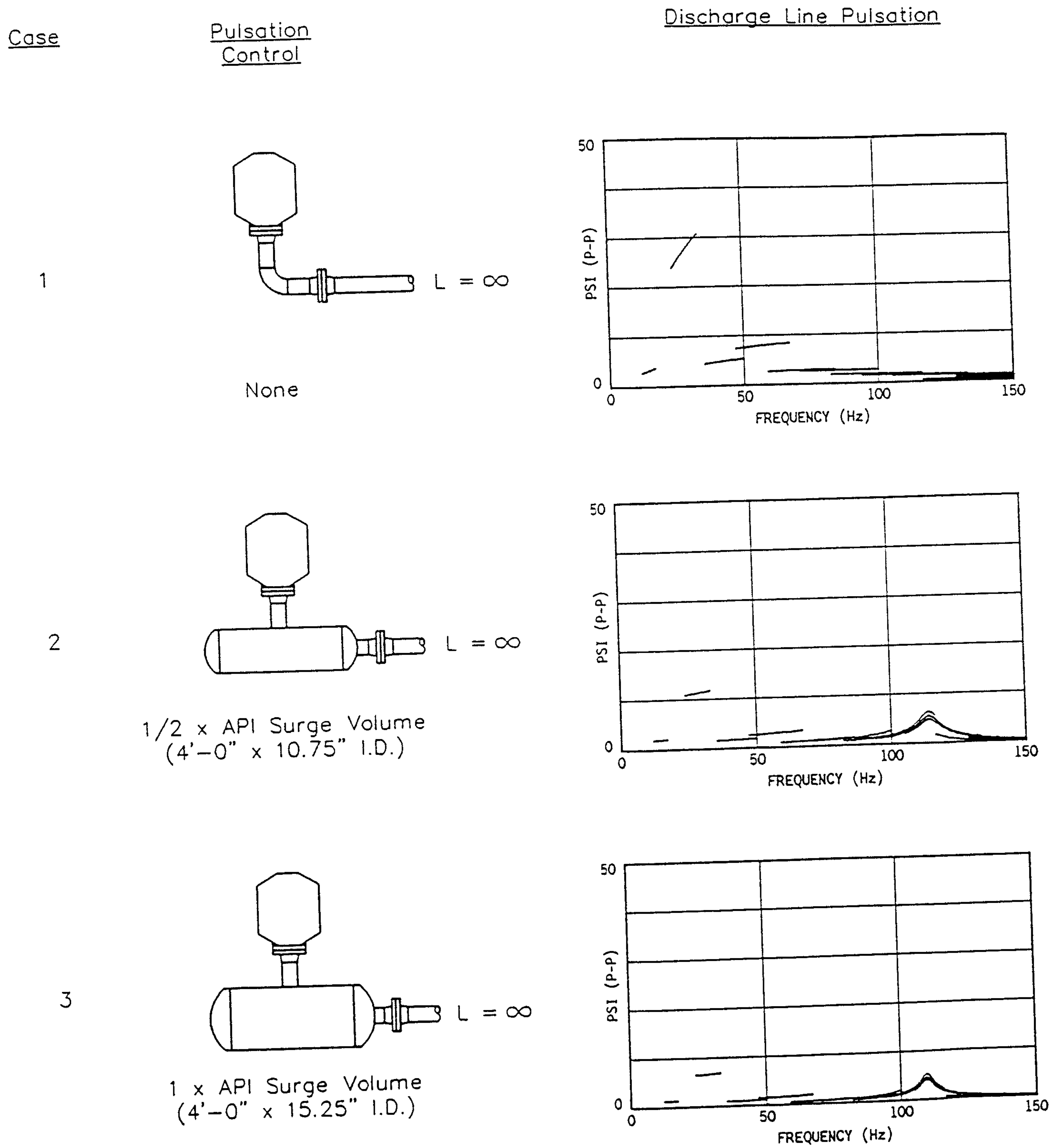


Figure 5: Comparison of Pulsation Control Devices for (Single Cylinder) Compressor Discharge Piping System

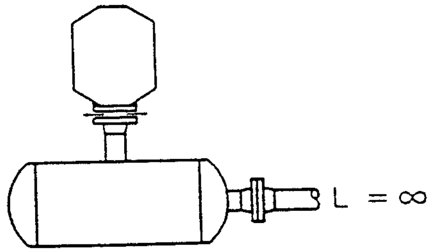
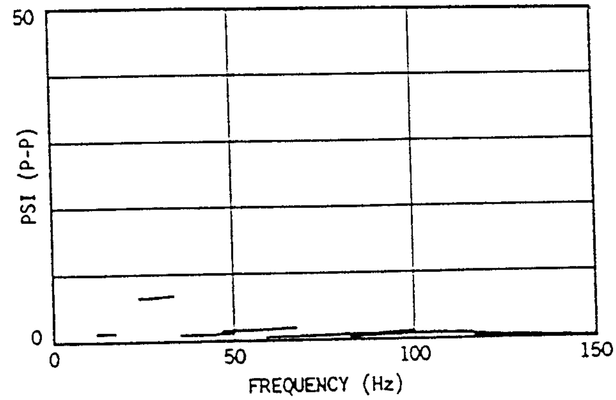
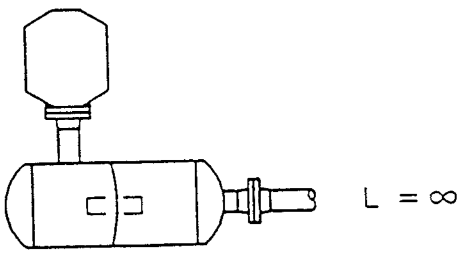
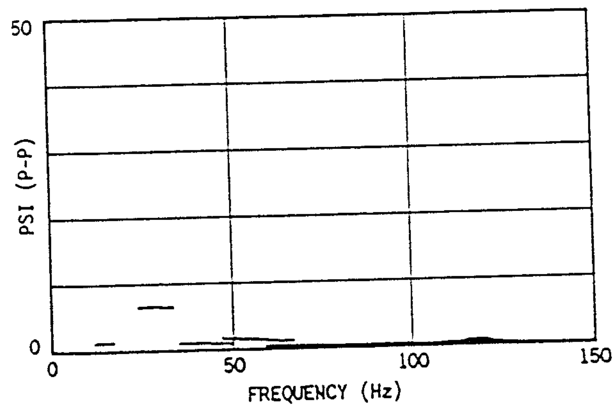
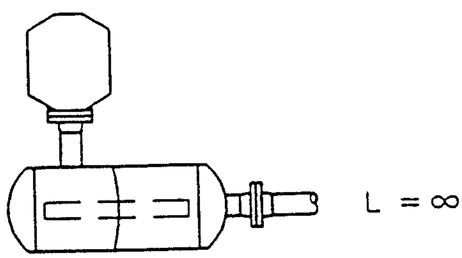
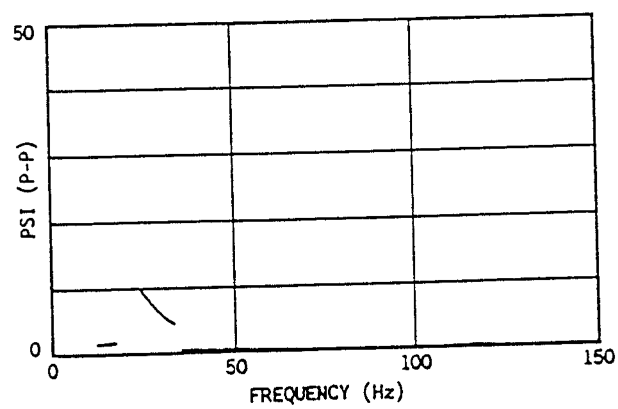
Case	Pulsation Control	Discharge Line Pulsation
4	 1 x API Surge Volume (4'–0" x 15.25" I.D.) 0.125% ΔP Orifice @ Cyl. Nozzle	
5	 Volume–Choke–Volume ($f \approx 4x$) Total Vol. = 1 x API Surge Volume (4'–0" x 15.25" I.D.) 5" x 2.624" I.D. Choke (0.44% ΔP)	
6	 Volume–Choke–Volume ($1x<f<2x$) Total Vol. = 1 x API Surge Volume (4'–0" x 15.25" I.D.) 3'–0" x 2.728" I.D. Choke (0.42% ΔP)	

Figure 5: (Continued)

Case	Pulsation Control	Discharge Line Pulsation

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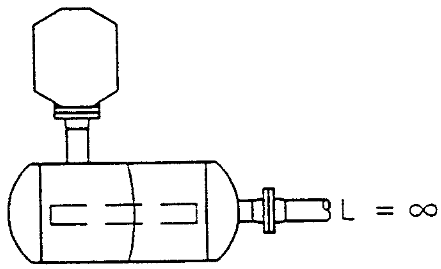
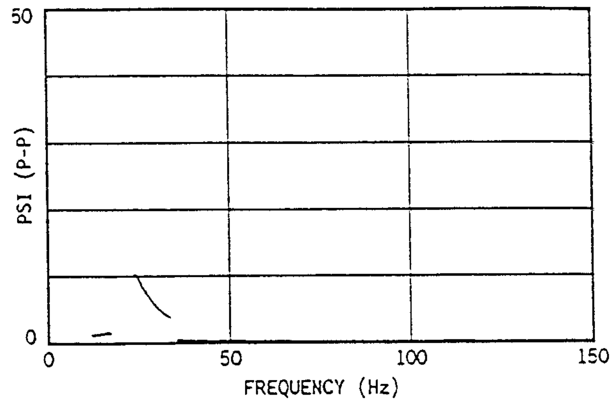


Volume–Choke–Volume ($1x<f<2x$)
Total Vol. > 1 x API Surge Volume
(4'–0" x 17.94" I.D.)
3'–0" x 3.152" I.D. Choke
(0.23% ΔP)



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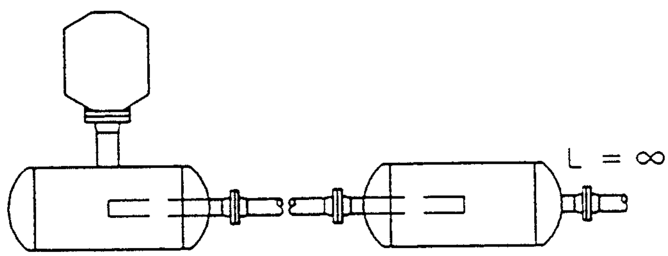


Volume–Choke–Volume ($f<1x$)
Vol. Each Bottle > 1 x API Surge Volume
(5'–0" x 15.25" I.D.)+(5'–0" x 15.25" I.D.)
10'–0" x 2.9" I.D. Choke
(0.42% ΔP)

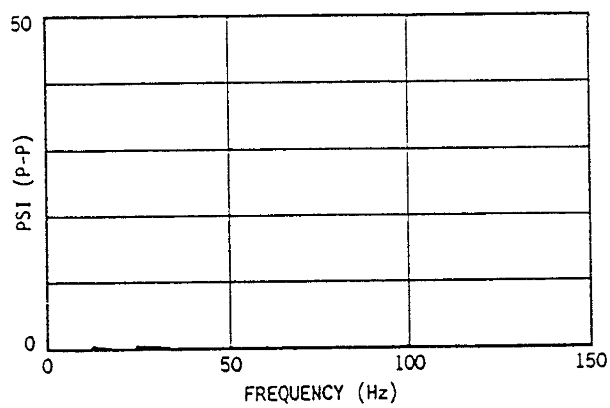
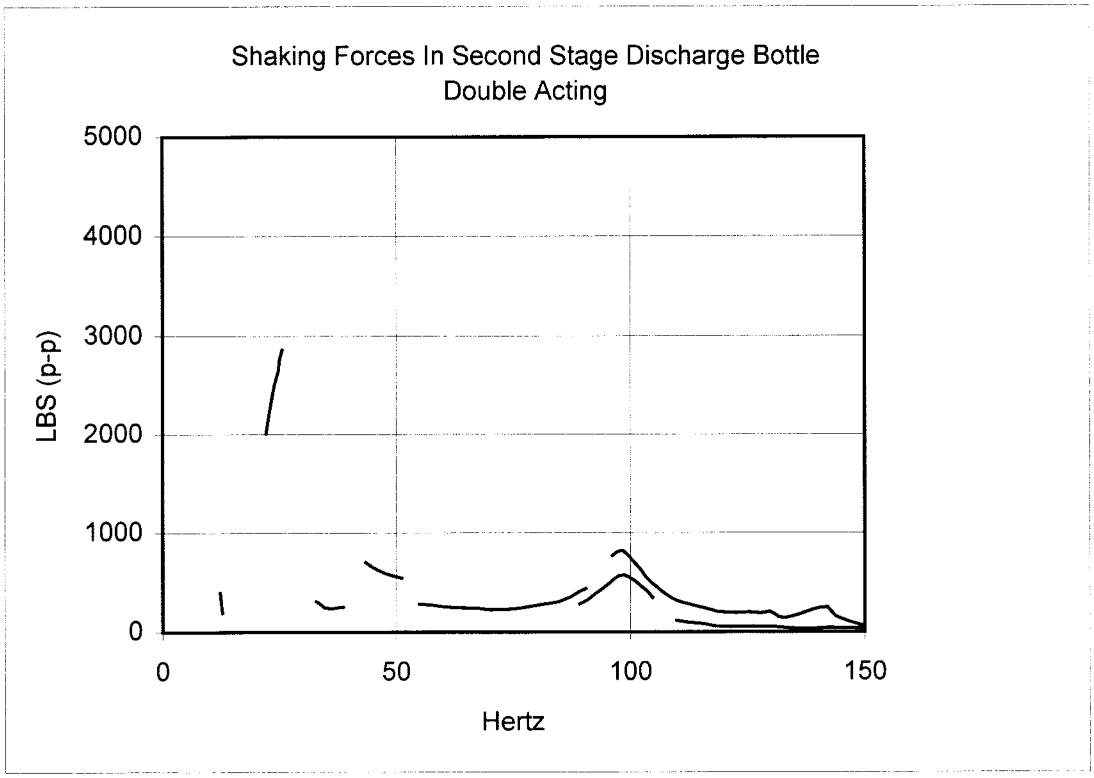


Figure 5: (Continued)


Shaking Forces In Second Stage Discharge Bottle
Double Acting
LBS (p-p)
5000
4000
3000
2000
1000
0
0
50
100
150
Hertz

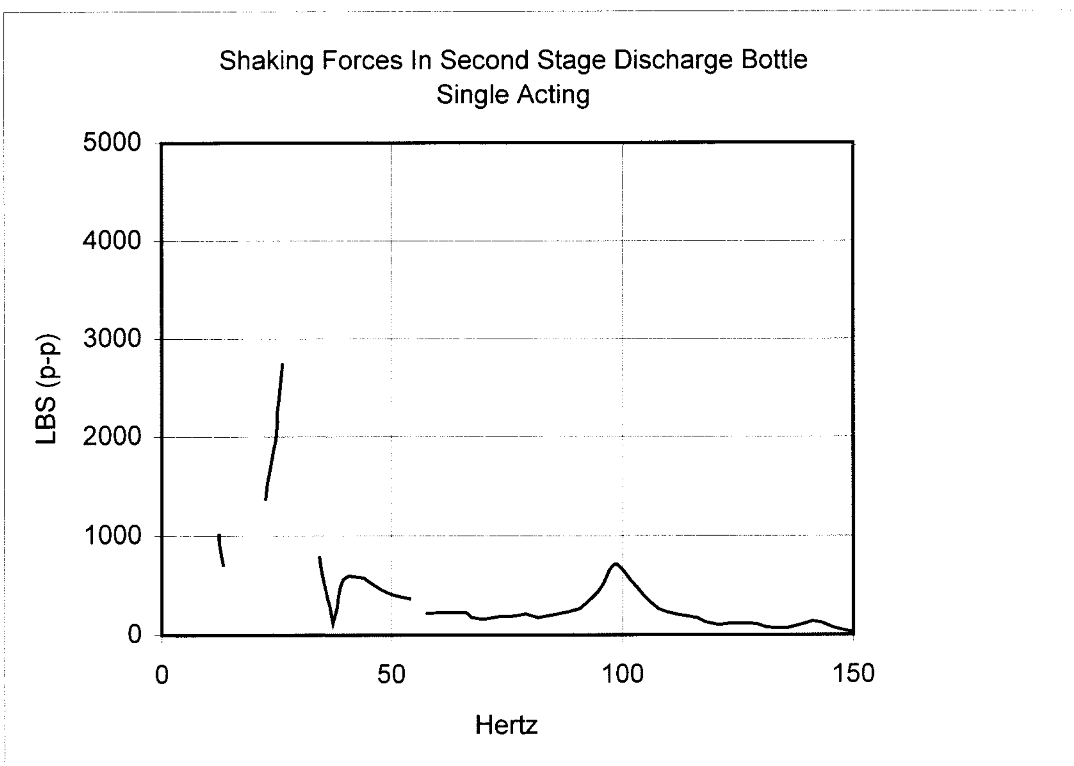

Shaking Forces In Second Stage Discharge Bottle
Single Acting
LBS (p-p)
5000
4000
3000
2000
1000
0
0
50
100
150
Hertz

Figure 6

All Cylinder D.A. - 706 RPM (+/- 10%)

Maximum Pulsation Amplitudes (PSI Peak-Peak)

Test Point	Harmonic 1	2	3	4	5	6	7	8	9
A - Cyl 3 Flange	3.99	137.17	19.77	28.04	9.74	5.35	4.57	3.27	2.56
B - Cyl 1 Flange	5.60	98.72	34.44	38.83	10.06	5.66	4.95	3.35	3.37
C - Choke Midpoint	3.73	8.72	3.74	6.78	1.42	2.60	3.23	2.25	2.74
D - TP 1	2.98	2.04	0.40	0.37	0.04	0.03	0.04	0.02	0.02
E - TP 2	2.79	1.98	0.33	0.28	0.03	0.02	0.02	0.01	0.01
F - TP 3	3.54	1.81	0.28	0.22	0.02	0.01	0.01	-	-
G - TP 4	2.44	1.65	0.25	0.17	0.01	-	-	-	-
H - PSV 855	3.33	9.58	0.29	0.17	0.05	-	-	-	-
I - TP 5	3.19	1.94	0.23	0.16	0.01	-	-	-	-
J - Cooler Inlet	1.70	0.69	0.10	0.05	-	-	-	-	-
K - Cooler Exit	0.83	0.26	0.01	-	-	-	-	-	-

Maximum Shaking Forces (LBS Peak-Peak)

Test Point	Harmonic 1	2	3	4	5	6	7	8	9
a - Discharge Bottle	174	2999	306	719	270	274	535	579	827
b - Discharge Baffle	1065	2176	1106	1321	160	52	115	132	162

All Cylinders S.A. - 706 RPM (+/- 10%)

Maximum Pulsation Amplitudes (PSI Peak-Peak)

Test Point	Harmonic 1	2	3	4	5	6	7	8	9
A - Cyl 3 Flange	23.40	122.64	55.81	23.49	8.91	4.19	3.21	2.95	2.14
B - Cyl 1 Flange	33.15	81.03	101.67	32.65	9.63	4.47	3.40	2.96	2.28
C - Choke Midpoint	20.87	7.64	12.40	5.70	1.30	2.78	3.29	2.30	2.44
D - TP 1	17.82	1.64	1.22	0.31	0.04	0.03	0.04	0.02	0.02
E - TP 2	16.44	1.43	1.02	0.24	0.03	0.02	0.02	0.01	0.01
F - TP 3	23.13	1.23	0.89	0.18	0.02	0.01	0.02	-	-
G - TP 4	14.89	1.09	0.68	0.14	0.01	-	-	-	-
H - PSV 855	20.06	5.81	0.70	0.15	0.04	-	-	-	-
I - TP 5	20.83	1.25	0.71	0.14	0.01	-	-	-	-
J - Cooler Inlet	11.11	0.61	0.32	0.05	-	-	-	-	-
K - Cooler Exit	5.04	0.24	0.03	-	-	-	-	-	-

Maximum Shaking Forces (LBS Peak-Peak)

Test Point	Harmonic 1	2	3	4	5	6	7	8	9
a - Discharge Bottle	1043	2770	958	602	246	217	393	728	790
b - Discharge Baffle	6766	1576	3472	1110	154	40	82	164	161

Figure 7: Pulsation and Shaking Force Amplitudes

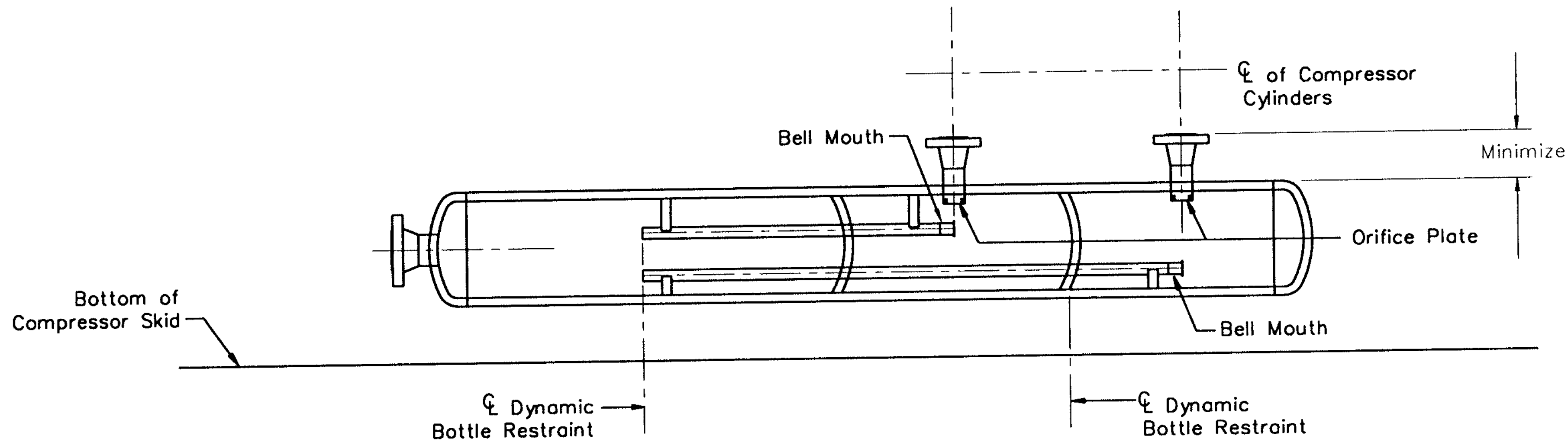
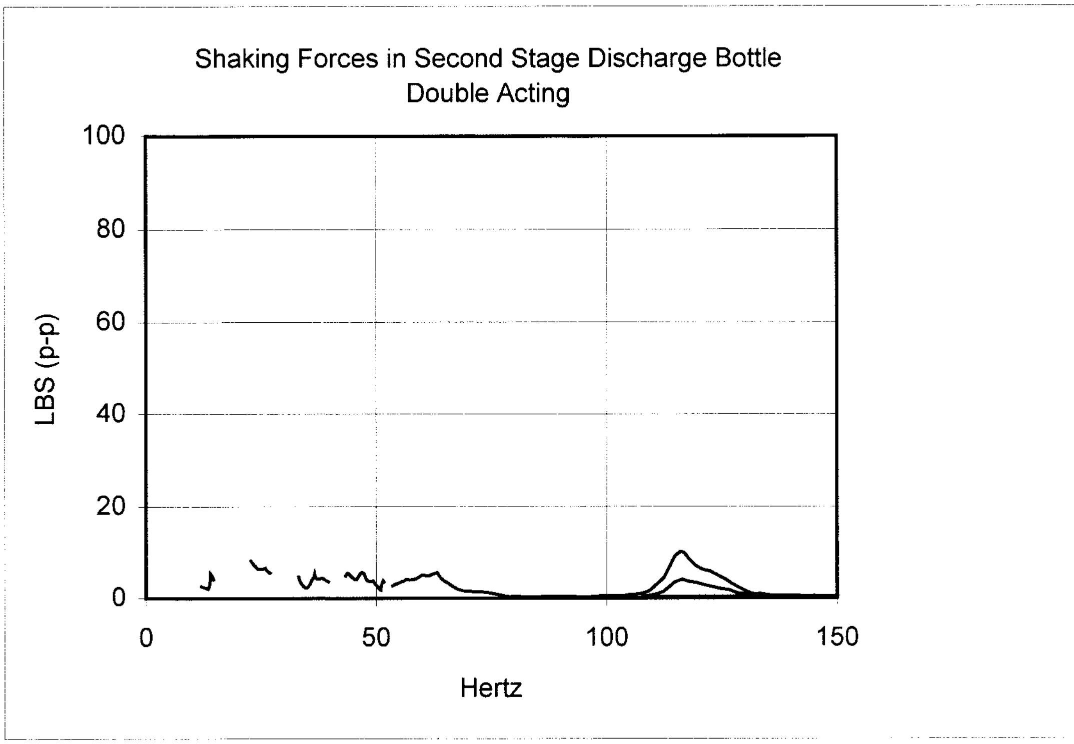


Figure 8: 2nd Stage Discharge Filter Design


Shaking Forces in Second Stage Discharge Bottle
Double Acting
LBS (p-p)
100
80
60
40
20
0
0
50
100
150
Hertz

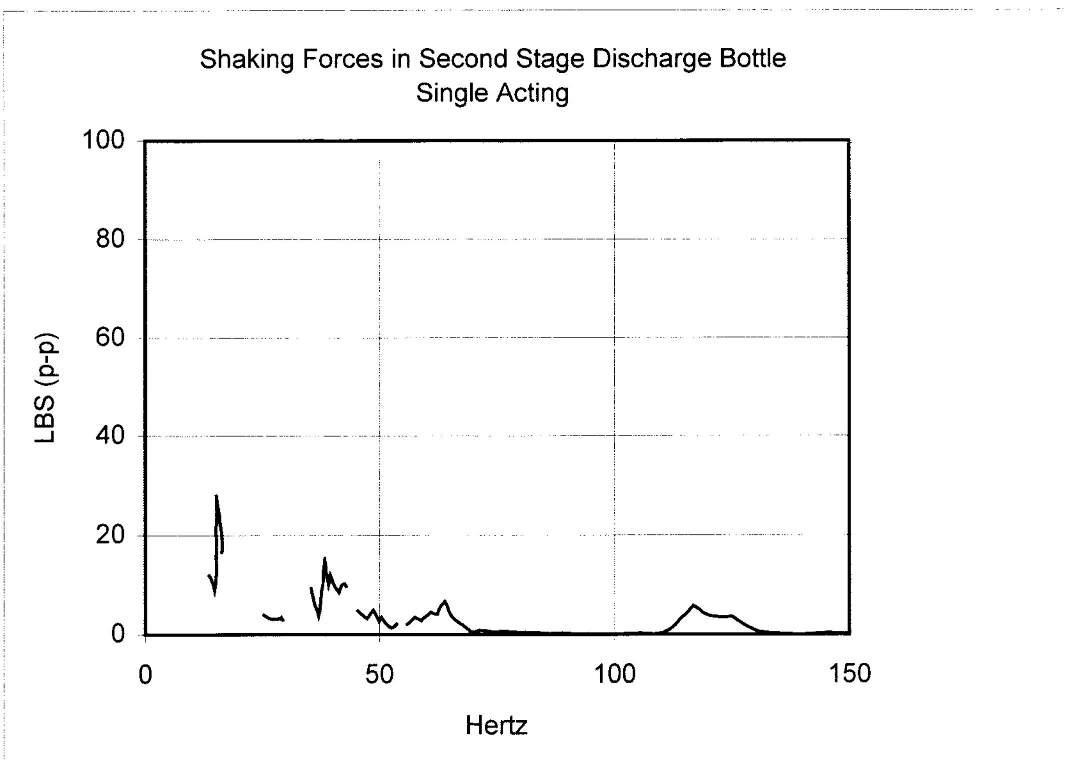

Shaking Forces in Second Stage Discharge Bottle
Single Acting
LBS (p-p)
100
80
60
40
20
0
0
50
100
150
Hertz

Figure 9

All Cylinder D.A. - 706 RPM (+/- 10%)

Maximum Pulsation Amplitudes (PSI Peak-Peak)

Test Point	Amplitude @ Harmonic 1	2	3	4	5	6
A - Cyl 1 Flange	63.50 @ 2x	35.05 @ 3x	33.58 @ 4x	10.01 @ 5x	7.93 @ 6x	5.27 @ 7x
B - Cyl 3 Flange	62.93 @ 2x	34.97 @ 3x	33.61 @ 4x	10.01 @ 5x	7.93 @ 6x	5.27 @ 7x
C - 2D Pri Bottle	0.14 @ 1x	0.05 @ 4x	0.04 @ 3x	0.03 @ 2x	-	-
D - 2D Chk Mid Pt	9.45 @ 2x	3.28 @ 3x	2.79 @ 4x	1.67 @ 5x	1.50 @ 6x	1.44 @ 1x
E - 2D Chk2 Mid Pt	10.21 @ 2x	6.19 @ 4x	4.12 @ 3x	1.45 @ 1x	1.24 @ 5x	0.21 @ 6x
F - TP 1	0.14 @ 1x	0.09 @ 3x	0.07 @ 4x	0.04 @ 2x	-	-
G - TP 2	0.40 @ 1x	0.09 @ 3x	0.06 @ 4x	0.04 @ 2x	-	-
H - TP 3	0.16 @ 1x	0.09 @ 3x	0.06 @ 4x	0.04 @ 2x	-	-
I - TP 4	0.13 @ 1x	0.07 @ 3x	0.05 @ 4x	0.04 @ 2x	-	-
J - PSV 855	0.42 @ 1x	0.31 @ 4x	0.10 @ 3x	0.08 @ 2x	-	-
K - TP 5	0.13 @ 1x	0.10 @ 3x	0.05 @ 4x	0.05 @ 2x	-	-
L - TP 6	0.28 @ 1x	0.05 @ 4x	0.05 @ 3x	0.05 @ 2x	-	-
M - TP 7	0.23 @ 1x	0.07 @ 3x	0.05 @ 2x	0.03 @ 4x	-	-
N - TP 8	0.06 @ 3x	0.06 @ 1x	0.04 @ 4x	0.02 @ 2x	-	-
O - Cooler Inlet	0.10 @ 1x	0.04 @ 3x	0.02 @ 4x	0.02 @ 2x	-	-
P - Cooler Exit	0.08 @ 1x	-	-	-	-	-

Maximum Shaking Forces (LBS Peak-Peak)

Test Point	Amplitude @ Harmonic 1	2	3	4	5	6
a - 2D Pri Bottle	11 @ 9x	10 @11x	9 @ 2x	6 @ 5x	6 @ 4x	6 @ 1x
b - 2D Pri Baffle A	10224 @ 2x	2071 @ 3x	1391 @ 1x	253 @ 5x	204 @ 6x	81 @18x
c - 2D Pri Baffle B	5175 @ 2x	1495 @ 3x	1044 @ 1x	945 @ 4x	178 @ 5x	105 @ 6x

All Cylinder S.A. - 706 RPM (+/- 10%)

Maximum Pulsation Amplitudes (PSI Peak-Peak)

Test Point	Amplitude @ Harmonic 1	2	3	4	5	6
A - Cyl 1 Flange	107.30 @ 3x	37.32 @ 2x	30.42 @ 4x	7.86 @ 5x	5.19 @ 6x	4.28 @ 7x
B - Cyl 3 Flange	107.00 @ 3x	36.97 @ 2x	30.49 @ 4x	7.86 @ 5x	5.19 @ 6x	4.28 @ 7x
C - 2D Pri Bottle	0.73 @ 1x	0.13 @ 3x	0.05 @ 4x	0.02 @ 2x	-	-
D - 2D Chk Mid Pt	9.88 @ 3x	7.32 @ 1x	5.56 @ 2x	2.55 @ 4x	1.96 @ 5x	1.39 @ 6x
E - 2D Chk2 Mid Pt	12.74 @ 3x	7.62 @ 4x	7.31 @ 1x	6.02 @ 2x	1.09 @ 5x	0.14 @ 6x
F - TP 1	0.75 @ 1x	0.22 @ 3x	0.06 @ 4x	0.02 @ 2x	-	-
G - TP 2	2.10 @ 1x	0.23 @ 3x	0.05 @ 4x	0.02 @ 2x	-	-
H - TP 3	0.87 @ 1x	0.22 @ 3x	0.05 @ 4x	0.02 @ 2x	-	-
I - TP 4	0.67 @ 1x	0.16 @ 3x	0.05 @ 4x	0.02 @ 2x	-	-
J - PSV 855	2.19 @ 1x	0.28 @ 4x	0.22 @ 3x	0.05 @ 2x	-	-
K - TP 5	0.70 @ 1x	0.21 @ 3x	0.05 @ 4x	0.03 @ 2x	-	-
L - TP 6	1.50 @ 1x	0.15 @ 3x	0.05 @ 4x	0.03 @ 2x	-	-
M - TP 7	1.21 @ 1x	0.15 @ 3x	0.03 @ 4x	0.03 @ 2x	-	-
N - TP 8	0.31 @ 1x	0.17 @ 3x	0.03 @ 4x	0.01 @ 2x	-	-
O - Cooler Inlet	0.55 @ 1x	0.11 @ 3x	0.02 @ 4x	0.01 @ 2x	-	-
P - Cooler Exit	0.40 @ 1x	0.01 @ 3x	-	-	-	-

Maximum Shaking Forces (LBS Peak-Peak)

Test Point	Amplitude @ Harmonic 1	2	3	4	5	6
a - 2D Pri Bottle	29 @ 1x	16 @ 3x	7 @ 5x	7 @ 9x	6 @11x	6 @10x
b - 2D Pri Baffle A	7056 @ 1x	6007 @ 2x	5925 @ 3x	199 @ 5x	133 @ 6x	75 @18x
c - 2D Pri Baffle B	5308 @ 1x	4278 @ 3x	3040 @ 2x	858 @ 4x	140 @ 5x	69 @ 6x

Figure 10: Pulsation and Shaking Forces (Modified System)

Pressure Pulsations Near Separator

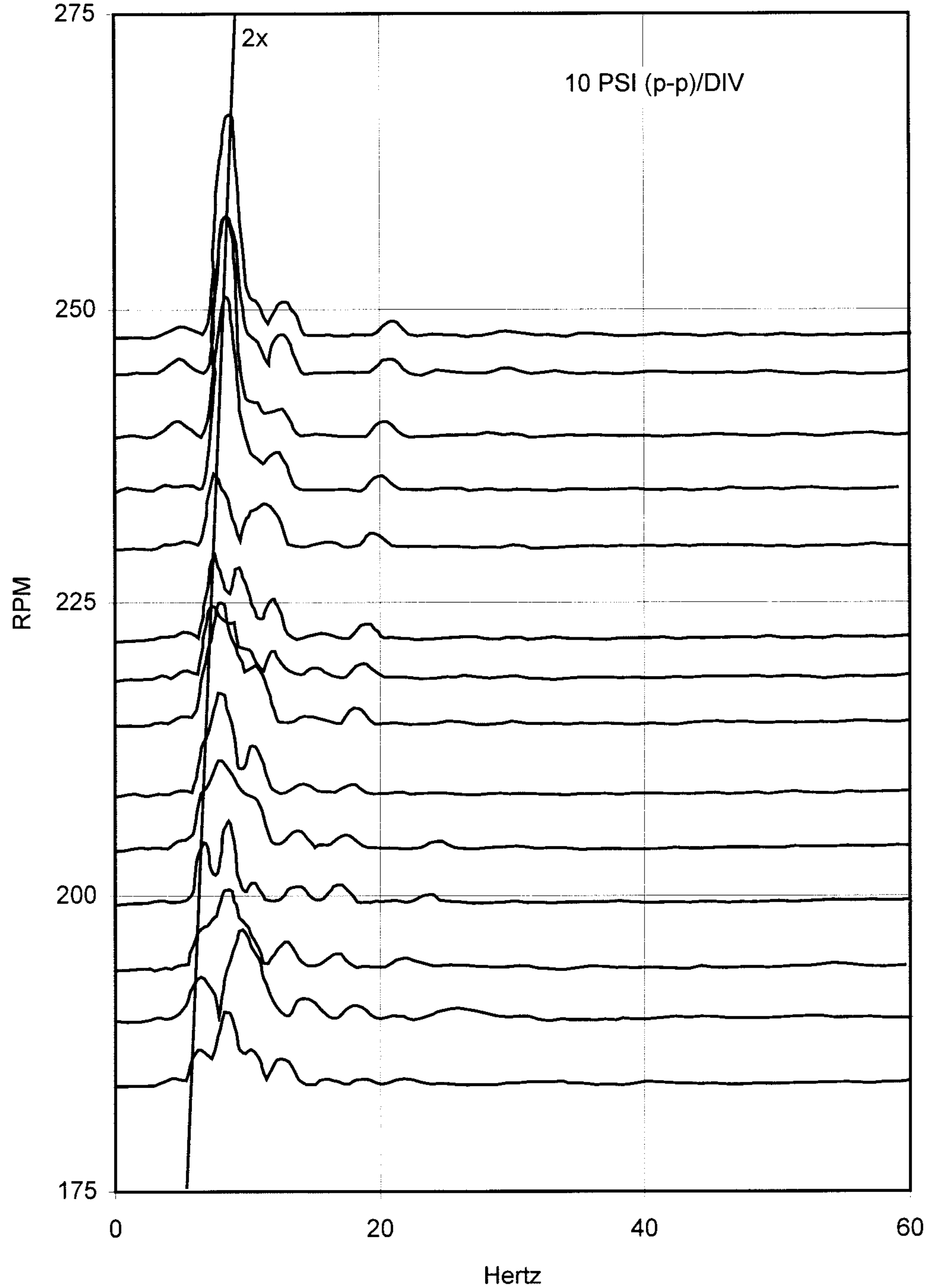


Figure 12

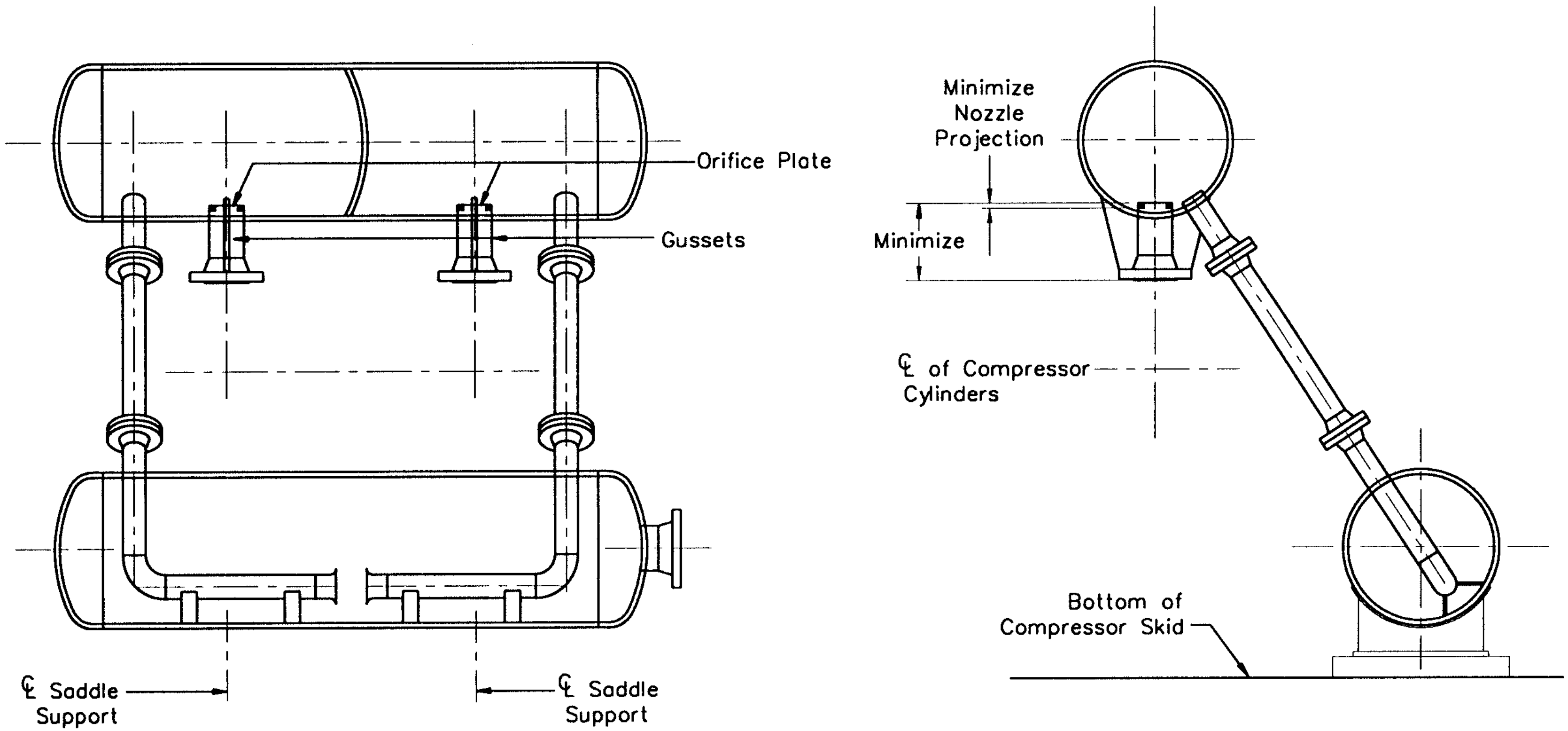


Figure 11: 1st Stage Suction Filter Design

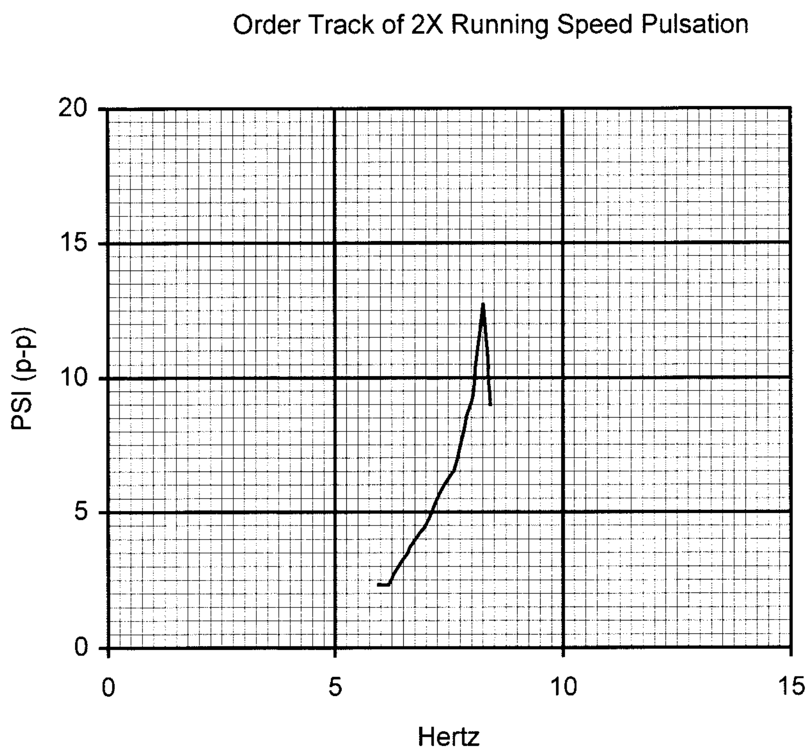

Order Track of 2X Running Speed Pulsation
20
15
10
5
0
PSI (p-p)
0
5
10
15
Hertz

Figure 13

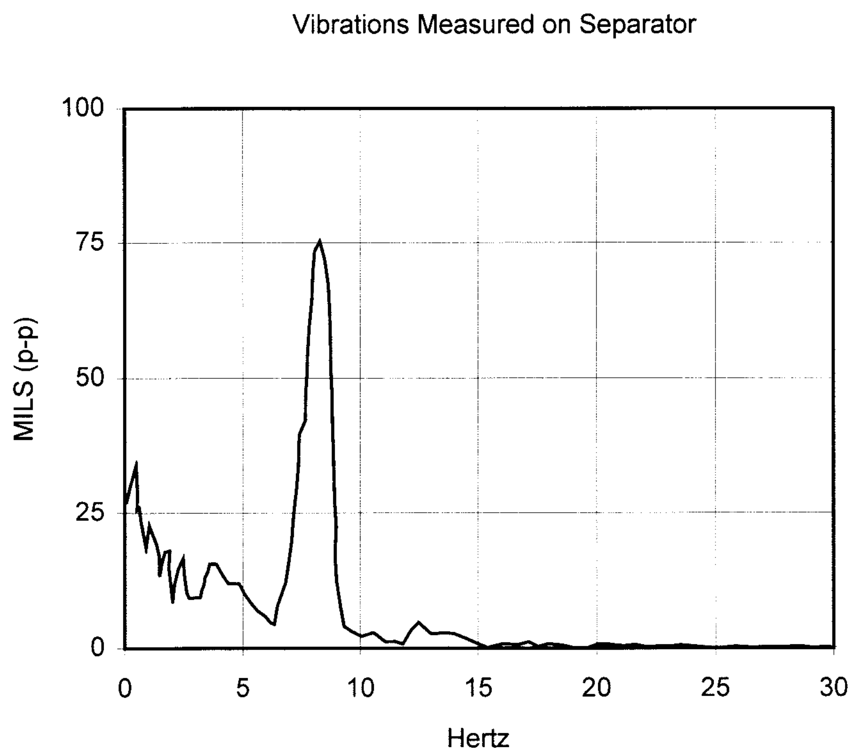

Vibrations Measured on Separator
100
75
50
25
0
MILS (p-p)
0
5
10
15
20
25
30
Hertz

Figure 14

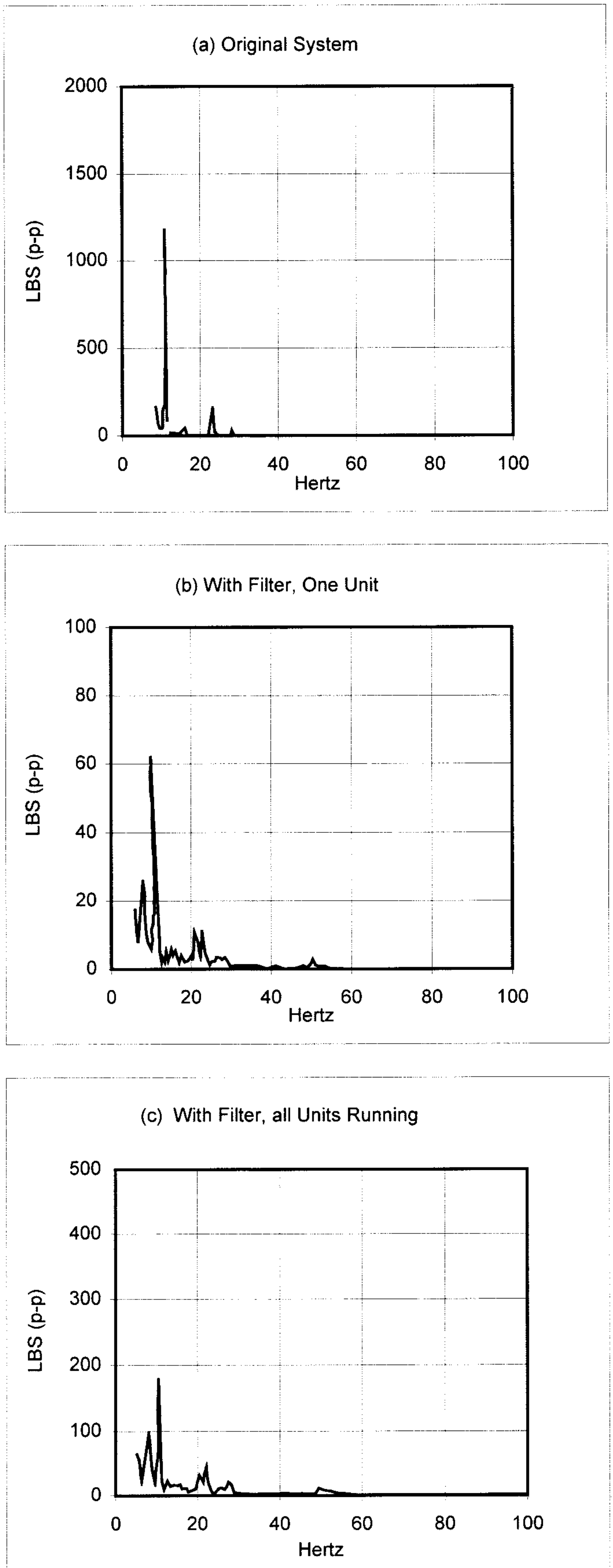


Figure 15: Shaking Forces At Separator

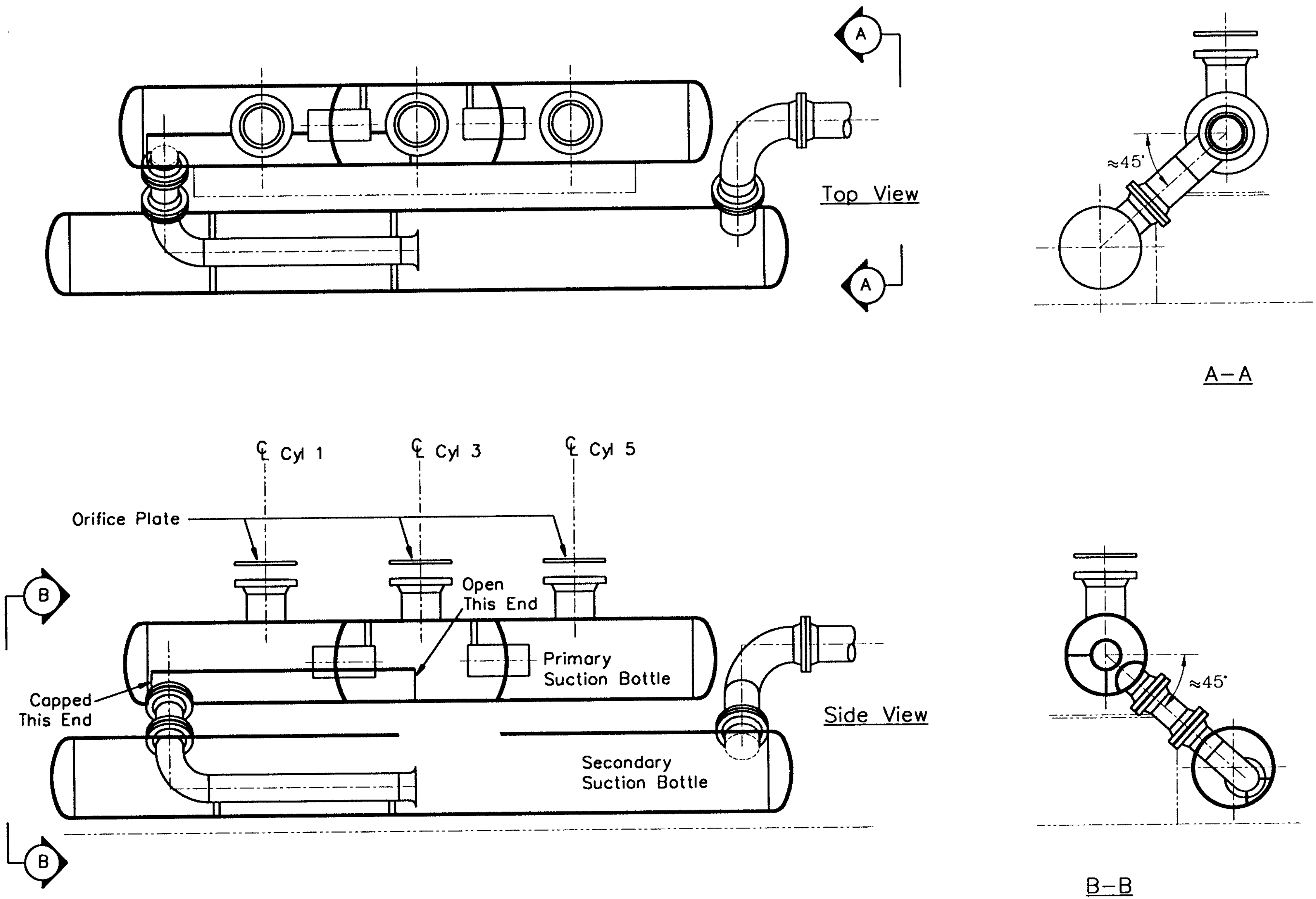


Figure 16: Suction Filter Design

Test Point	Pulsation (psi p–p)				
	Existing System		Modified System		API
	Pockets Closed	Pockets Open	Pockets Closed	Pockets Open	
Suction Source					
1	8.0 @ 2×	9.2 @ 2×	0.8 @ 1×	1.0 @ 6×	7.0 @ 2×
2	9.5 @ 2×	12 @ 8×	1.9 @ 6×	3.3 @ 6×	5.0 @ 8×
3	3.7 @ 2×	3.9 @ 2×	0.9 @ 1×	0.9 @ 1×	7.5 @ 2×
4	3.0 @ 2×	3.6 @ 2×	0.5 @ 1×	0.5 @ 6×	9.0 @ 2×

Figure 17: Comparison of Pulsation Amplitudes

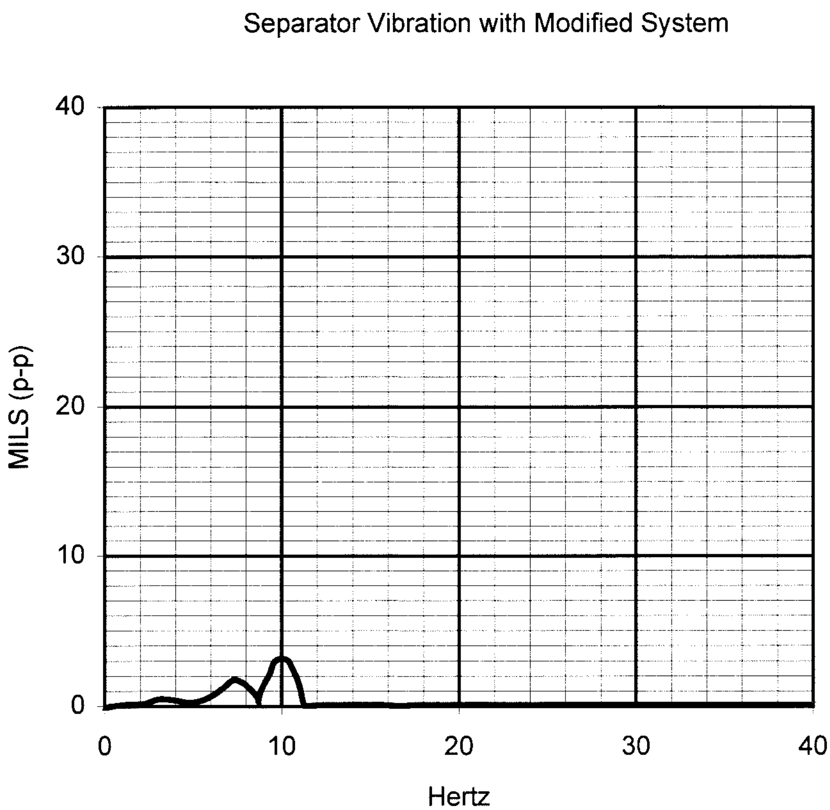

Separator Vibration with Modified System
40
30
20
10
0
MILS (p-p)
0
10
20
30
40
Hertz

Figure 18