EXPERIMENTAL EVALUATION OF BLADDER TYPE PULSATION DAMPENERS FOR RECIPROCATING PUMPS

by

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ABSTRACT

Pulsation dampeners are frequently used to reduce piping pressure pulsations and vibrations in reciprocating pump systems. Despite their extensive use, scientific understanding of how the dampeners work and what design parameters are important needs considerable development. To advance this understanding, the authors' company initiated an extensive dampener evaluation test program in cooperation with five leading manufacturers of commercial bladder type dampeners. All dampeners were tested in a special test loop under identical conditions to analyze the effect of a dampener's design on its effectiveness. Pressure measurements at several points in the piping were made as a function of pump speed, bladder precharge pressure, and suction and discharge line pressures for each dampener. Results indicate that the type and location of dampener have a considerable influence on pulsations. Dampeners are effective over a much broader range of precharge pressures than generally believed. Based on these findings, a number of practical recommendations in the selection and use of dampeners are offered.

INTRODUCTION

Pressure pulsations in piping systems connected to reciprocating pumps are caused by cyclic flow variations into and out of the pumps. Under resonant conditions, these pulsations can be so severe as to require pump shutdown for safety reasons [1, 2]. The traditional remedy to this problem has been the use of one or more pulsation dampeners in the piping system. Dampeners are frequently used in the suction piping and often, but less frequently, in the discharge piping.

While a dampener may attenuate pulsations, the level of attenuation (or effectiveness of the dampener) depends on several factors including type and location of the dampener. In some instances, misplacement of the dampener may actually amplify pulsations [4]. The authors recently came across an example of this anomalistic behavior. In a multiple-pump boiler feed pumping station, piping vibration became much worse when the field personnel moved the suction pulsation dampener from the flanged end to the closed end of the manifold. When the authors analyzed the piping system, analysis results exactly matched field observations.

Normally, high pulsations are caused by coincidence of piping acoustical resonant frequencies with one of the multiples of the pump speed. While such coincidence in theory can be avoided by proper selection of piping geometry and pump speed, in practice it is quite difficult. Calculation of piping natural frequencies depends directly on the speed of sound, whose value is either not precisely known or can change during operation due to changes in composition of the fluid being pumped. Thus, these frequencies are not precisely known. In variable-speed pumps, one or more of the many resonant frequencies are likely to match exciting frequencies at some speed, even when the pump speed varies in discrete steps.

Thus, a practical solution to avoidance of system problems is the use of dampeners along with piping system analysis to guide the selection and overall system design. The importance of proper dampener selection should not be underestimated and the recent trend towards higher speed pumps and economical piping system designs lends even greater urgency to the issue. In response, the authors' company has developed a comprehensive pump piping system simulation computer program [1] which has been successfully used in the problem free design of many installations. In this program, virtually any kind of dampener can be modelled. While the program has been shown to very effective, the validity of dampener modelling has not been directly verified, since to do so, one would need to set up an experiment wherein either the dampener or its characteristics are systematically varied without changing any other variable.

In 1986, one of the authors surveyed major dampener manufacturers in the U.S. and abroad to determine if they would have the needed information. The authors' company initiated a dampener evaluation program in 1987 to test various commercially available dampeners on a uniform basis at a special test facility in Cambridge, Canada.

Most of the major dampener manufacturers in the United States and some in Europe were invited to participate in the program. Virtually all of them responded with enthusiasm to the program and five of them, four domestic and one European, participated in the program on relatively short notice. This study is focused on experimental results and practical insights derived from the program.

DAMPENER DESCRIPTION

The term "pulsation dampener" herein is used in a generic sense. Manufacturers try to differentiate themselves by assigning different names to their products such as filter, accumulator, stabilizer, damper and desurger. All these devices are used to control pulsations. From a technical viewpoint, pulsation filter is probably a better term, although some devices include characteristics of both a filter and an attenuator by intentionally incorporating pressure drop element such as orifices.

Several types of dampeners (Figure 1) (bladder, reactive, resistive) are commercially available, although the bladder type is used most frequently. Bladder type dampeners are pressure vessels appendage mounted to the piping system or incorporated into the piping system itself. These units have direct communication with the pumped liquid. The gas and liquid are separated by a flexible membrane. Miller [3] has illustrated over 30 variations of these three broad categories of dampeners. Each dampener has its own frequency response (or transfer matrix) which for some simple types can be analytically computed with great accuracy.

The APPENDIX shows attenuation characteristics of a typical bladder type dampener. Both calculations and experimental observations indicate that this type of dampener (direct analogy to an electronic filter) is most effective in a certain frequency range and, in fact, can amplify pulsations at other frequencies.

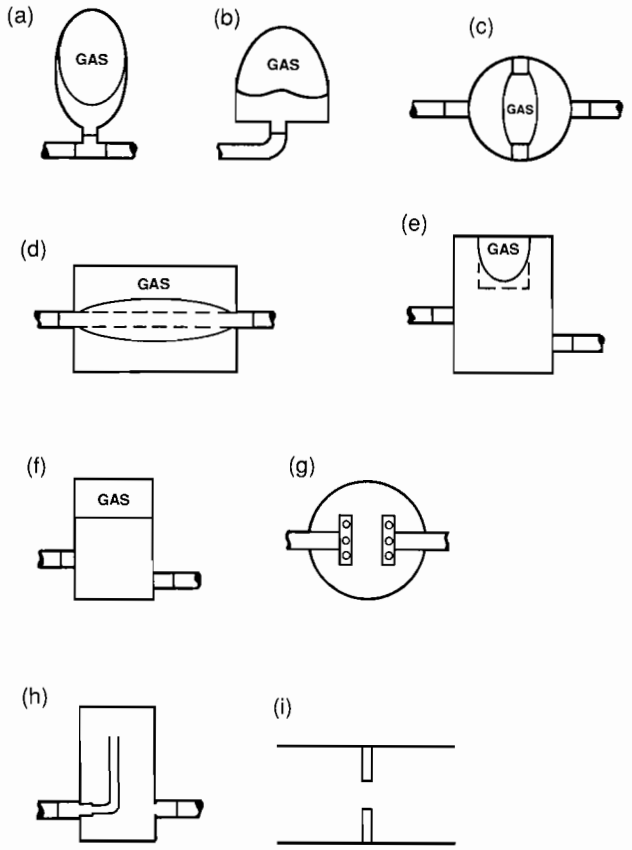

Figure 1. Type of Pulsation Dampeners: Bladder (a-e); Gas-charged (f); Reactive (h); Resistive (g, i). (i) is a simple orifice.

The five dampeners tested under the program were all bladder type, although the shape of and access to the bladder were different from one dampener to the other. The bladder, filled with an inert gas, substitutes for a large liquid volume which otherwise would be necessary to provide equivalent compliance. Liquid volume V_l equivalent in compliance to gas volume V_g, can be calculated from the following relationship:

$$V_l = \frac{K}{\gamma P} V_g$$

where

K is the liquid's bulk modulus, γ the specific heat ratio, and P the absolute line pressure.

For instance, at 100 psig gas pressure, the ratio of water to gas volume, is 1900, i.e., a liquid tank has to be about 1900 times larger in volume than a gas bottle for equivalent compliance. At high discharge pressure, an all liquid filter with that large a volume and pressure vessel rating can be prohibitively expensive to procure as well as to install. Bladder type dampeners are, therefore, popular because they are less expensive and smaller relative to other types. However, they are only effective as long as the bladder precharge pressure lies in a certain range of the line pressure. The operation of bladder type pulsation dampeners is discussed in greater detail by Wachel [4]. One of the main thrusts of the test program was to determine this effective range.

Three of the five test dampeners had a small neck at the inlet of the dampener (Figure 1a) while the other two had open access to the bladder (Figure 1 b and c). The neck, acting like an impe-

dance in electrical filters or a mass in mass spring systems, can increase attenuation at certain frequencies, similar to response of a Helmholtz resonator [4]. On the other hand, open-access dampeners can respond over a wider frequency range, reducing the importance of matching peak-response frequency to the main excitation frequency. The characteristics of the five test dampeners are shown in Table 1.

Table 1. Test Dampener Characteristics

Test Dampener No.	Location	Type (See Fig. 1)	Est Dampener Liquid Volume (Gal.)	Est Bladder Gas Volume (Gal.)
D-1	Suction	1e	5	2.1
	Discharge	1a	5	4.2
D-2	Suction	1a	2.6	2.4
	Discharge	1a	2.6	2.4
D-3	Suction	1a	20	18.4
	Discharge	1a	20	18.4
D-4	Suction	1c	40	5.0
	Discharge	1c	40	5.0
D-5	Suction	1b	10	7.2
	Discharge	1b	10	7.2

Note: When two dampeners were installed in one location, the volumes shown are the combined total.

TEST LOOP

Ingersoll-Rand Pump Test Facility located in Cambridge, Ontario, Canada, was used for conducting the dampener tests. A special test loop was fitted with a reciprocating Ingersoll-Rand 7MP400 triplex mud pump. This test loop is not necessarily representative of a typical mud pump field installation, yet it includes many of the piping elements generally used in field piping systems. In addition, the test loop design purposely included some bad piping practices (lack of pipe supports, too many short radius bends, etc.), commonly encountered in field by pump and dampener manufacturers. The piping elements in the test loop could be readily changed, offering test flexibility.

The test loop schematic including piping dimensions is shown in Figure 2. The suction piping starts from the bottom of a large, elevated, open 1300 ft^3 tank and ends at the suction manifold through a centrifugal booster pump. The pump was driven by an independent electric motor running at 1750 rpm in order to provide about 80 psig suction pressure under all operating conditions. Without the charge pump, the suction head was about 21 ft with full tank. The pump would not operate satisfactorily operated at 120 rpm and above without the charge pump, even when fitted with a suction dampener.

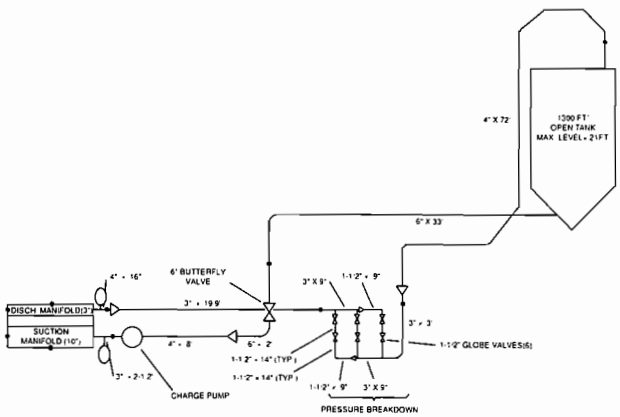

Figure 2. Test loop schematic. Label four inches × eight feet stands for a four inches diameter, eight feet long section of the pipe. o marks transducer sections.

The discharge piping consisted of a 20 ft section of straight piping from the discharge manifold to a pressure breakdown junction and then up to the tank where it discharged freely into the tank. The pressure breakdown junction is a combination of three parallel legs each fitted with two orifices (1.5 in globe valves) in series. These valves were used to set discharge pressure during the tests. The pump was driven by a variable speed Caterpillar diesel engine through a fluid coupling, allowing any desired speed between 60 and 180 rpm.

The pump characteristics are as follows:

Type:	Reciprocating mud pump
Number of cylinders:	3
stroke:	7"
Piston Diameter:	5½"
Valve Type:	Poppet
Engine RPM:	477-1430 (Test Range)
Pump RPM:	60-180 (Test Range)
Fluid:	Recirculated potable water

TEST PLAN

The primary objective of the test program was to develop an understanding of the dampener behavior at different gas charge pressures and different pump speeds.

The test plan was formulated to answer three basic questions:

- What are the pressure pulsations at different points in the piping with and without dampeners? How does the variation in pump speed affect these pulsations?
- What happens to these pulsations at different dampener charge pressures? This question is important, because the ratio of charge pressure to the line pressure can vary in the field for several reasons. These reasons include changes in the line pressure due to variations in piping system resistance, change in pump speed, gas bladder leakage or rupture, etc. Most dampener manufacturers recommend a precharge pressure range (typically 50 percent to 70 percent of the line pressure) for optimum performance.
- Are some dampeners consistently more effective than others over a broad range of operating conditions? The variable precharge tests were run by gradually varying either the suction or discharge dampener precharge pressure from typically 120 percent of the line pressure to atmospheric pressure in a series of steps.

The pressure transducer locations in suction and discharge piping are shown in Figure 2. Up to eight pressure transducers were used to record pulsations at the following points:

- Upstream and downstream (i.e., both sides) of the suction and discharge dampeners. The transducers were mounted six inches away from the dampener flanges. The terms upstream and downstream are defined according to the flow direction. For example, the downstream suction transducer was typically mounted in the middle of the suction manifold, while the upstream suction transducer was mounted between the charge pump and the dampener.
- Discharge and suction line locations as shown in Figure 2.
- Suction and discharge manifold closed end.
- Middle cylinder.

Participating manufacturers, following a review of the test loop, supplied dampeners and recommended installation locations and procedures. These procedures were carefully followed. Some participants were present during the test of their dampeners. Each participant was given relevant test data for their dampener following the tests. One manufacturer recommended use of two dampeners each in suction and discharge piping, one at the flanged end and other at the closed end of the suction and discharge manifolds. All others recommended only one dampener, installed near the manifold to piping flange connection.

The test procedure for each dampener consisted of following tests:

- Run the pump without any dampener, called a no dampener test. Run the tests at a fixed discharge pressure of about 1500 psig (maintained by controlling breakdown orifices). Vary rpm from 180 to 60 in steps of 30 (i.e., 180, 150, 120, 90, 60).
- Repeat the above test but with suction and discharge dampeners in place. The dampeners are to be properly precharged per the manufacturer's recommendations.
- Maintain discharge line pressure at 1000 psi and pump rpm at 180. Vary the discharge precharge pressure of the dampener from 110 percent of the line pressure to atmospheric pressure in five to six equal steps and acquire data at each step. Maintain proper precharge pressure in the suction dampener.
- Repeat the above test while gradually varying the precharge in the suction dampener from 110 percent of the suction line pressure to atmospheric, while maintaining the discharge dampener at the proper precharge pressure. The discharge line pressure is maintained at 1000 psi and pump rpm at 180.

TEST DATA AND INSTRUMENTATION

Pressure measurements were made at several points along the piping using strain gauge or piezoelectric pressure transducers. The strain gauge transducers have the advantage of providing D.C. signal (i.e., static pressure) besides the A.C. (i.e., dynamic pressure) signal. The primary purpose of static pressure measurements was to monitor the general pressure level in the pumps and to provide a check for the pressure gauge readings. However, the focus of this investigation centered in the measurement of dynamic pressure fluctuations. Following is a list of the instrumentation used to acquire, process and analyze data:

- Strain Gage Transducers
  Schaevitz Type P723-0025
  Range 0-10000 psi and 0-500 psi
- Piezocrystal Transducer
  Kistler Type 607F122, 0-1500 psi
- Tape Recorder
  Racal D300-3-14DS, 16 Channel
- Strain Gage Amplifier
  Vishay Model 2110
- Voltmeter
  Fluke 8010A
- Dynamic Signal Analyzer
  HP 3572A, Dual Channel, 800 lines
- Tachometer
  Concorde Photo Reflective
- Flowmeter
  Badger 4″ Turbine
- Flowmeter Readout
  Dynapac MTJR-1-0
- Key Phaser
  Magnetic Pick-up from Crank Shaft Key.

During each test, data from all pressure transducers and the key phaser was simultaneously recorded. Data used in dampener evaluation was recorded at the following locations:

- Upstream of the discharge dampener: Center of the discharge manifold.
- Downstream of the discharge dampener: About six inches away from the dampener flange.
- Upstream of the suction dampener: About eight inches away from the suction flange.
- Downstream of the suction dampener: Center of the suction manifold.

In the no dampener tests, data was still recorded at about the same locations (within six to eight inches) as if a dampener had been present. A difference of six to eight inches is inconsequential at frequencies below 100 Hz where the pulsation wavelengths exceed 50 ft.

RESULTS AND DISCUSSION

Pulsations Without Dampeners

The pump was initially run without a booster pump, the way many mud pumps are run in the field. However, above 120 rpm, high suction pulsations made continuous operation unsafe, even when a suction dampener was installed. Without the dampener, a boost pressure of nearly 60 psi was required to maintain vibration free operation at the maximum speed of 180 rpm. Time signal and spectral analysis plots are shown in Figure 3 of the pressure signal in the suction manifold at 180 rpm. The maximum peak-to-peak pulsation is about 130 psi, implying that the suction pressure needs to be at least 65 psia just to avoid cavitation in the manifold. An additional five to ten psi is required to overcome valve losses and local acceleration head, boosting minimum required suction pressure to 75 psia or 60 psig.

Furthermore, accounting for friction losses in the suction piping, a minimum of 170 ft absolute head is required at the tank, compared to 36 ft of head that was available in the test loop. Standard calculations of acceleration head losses using Hydraulic Institute procedures [5] place absolute head requirement at the tank of 120 ft. However, our test results indicated that this was insufficient to prevent cavitation. Computer predictions indi-

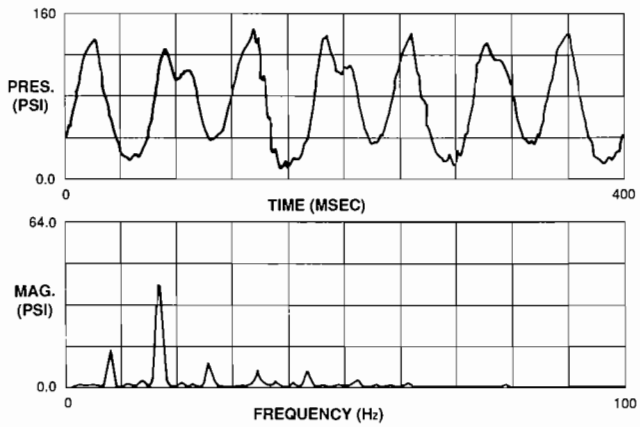

Figure 3. Time-Signal and Frequency Plot of Pulsations in Suction Manifold, without Dampener. The vertical axis in the frequency plot shows peak-to-peak amplitude.

cate that the sixth harmonic pulsations are amplified by a quarter-wave resonance set between the pump and the open tank.

Time signal and frequency plots are shown in Figure 4 of pulsations in the discharge manifold at 180 rpm. The pulsation amplitude is 165 psi (half of peak-to-peak 330 psi), about 12 percent of the mean discharge pressure of 1400 psi. Larger discharge pulsations are caused by a higher flow velocity in the smaller diameter discharge manifold as compared to the lower pulsations due to lower velocities in the larger suction manifold. Discharge pulsation levels are shown in Figure 5 as a function of pump speed. Pressure levels in this and other similar figures are shown as a percentage of the line pressure in the related piping system. Dashed lines joining discrete points are for clarity only and should not be used for interpolation. Since suction and discharge pressures were kept constant during each test, percentage pulsation levels also provide a direct measure of pulsation magnitudes.

It may seem surprising that discharge pulsations in the no dampener case stay virtually constant with increase in pump speed since pulsation levels generally tend to increase with speed. The explanation for this odd result can be found in the piping system setup. In the test loop, the discharge pressure is maintained constant by gradually increasing the piping resistance through pinching of the globe valves. Therefore, any expected decline in pulsations due to reduction in flow at lower pump speed is balanced by the growth of pulsations as a result of increase in piping resistance. In essence, the pulsation level as a ratio of the line pressure essentially remains constant.

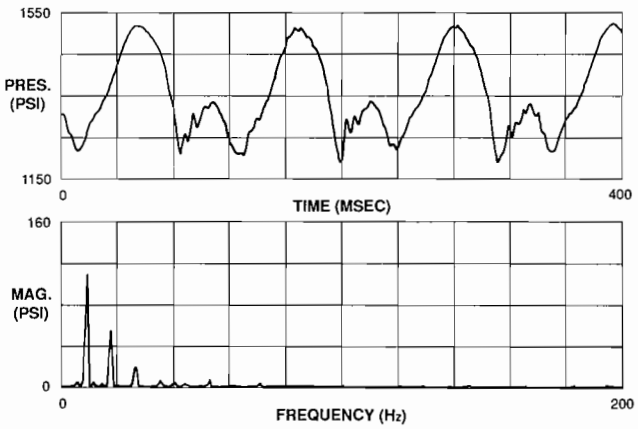

Figure 4. Magnitude and Frequency Plot of Pulsations in Discharge Manifold, without Dampener. The vertical axis in the frequency plot shows peak-to-peak amplitude.

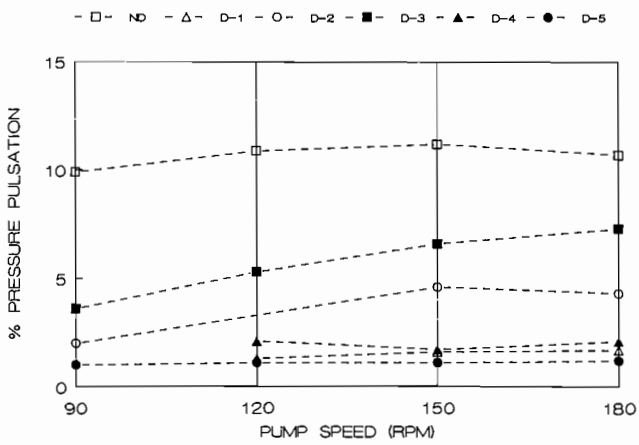

Figure 5. Variation of Pressure Pulsations in Discharge Manifold with Pump Speed. ND stands for the no dampener case, D-1 to D-5 for the five dampeners tested. The percent pressure pulsation in this and other plots is defined as the percent ratio of half the peak-to-peak pressure to the line pressure.

Discharge Pulsations—With Dampeners

For the dampener tests, discharge and suction dampeners were installed and charged according to the manufacturers' instructions. All tests were run at about 80 psig suction pressure and 1500 psig discharge pressure. A comparison of pressure pulsations in the discharge manifold is shown in Figure 5, as a function of pump speed for the five dampeners designated D-1 to D-5 and for no dampener. Pulsation level here is defined as half of the maximum peak-to-peak pulsation in a pressure-time signal averaged over at least 10 cycles. With all dampeners, pulsations in the discharge manifold are reduced from a level of nearly 10 percent without dampener to less than five percent when the dampener is installed. Lowest levels are seen with the D-5 dampener actually is a set of two dampeners, one installed at the flanged end and the other at the closed end of the pump discharge manifold. Reduced pulsations at the manifold result in smoother discharge valve operation, decrease in plunger dynamic loading and a small improvement in pump efficiency have been identified by Perry, Miller, and Wachel [2, 3, 4].

Downstream of the dampener, pulsation levels for different dampeners show greater variation than those at the manifold, upstream of the dampener (Figure 6). In the case of D-2 and D-3 dampeners, pulsations are higher than those in the manifold. Generally, one would expect lower pulsations past the dampener since one primary purpose of installing a dampener is to isolate piping from the pump. However, this data indicates that pulsation levels depend on the complete piping system and results seen in one kind of piping system cannot be blindly transported to another system.

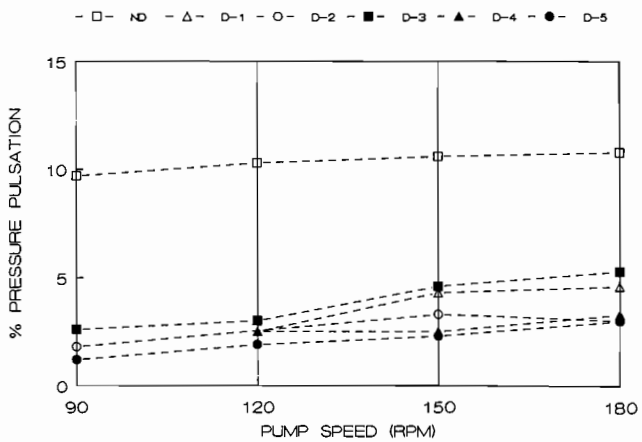

Figure 6. Variation of Pressure Pulsations Just Downstream of Dampener with Pump Speed.

Typical pulsations are shown in Figure 7 upstream and downstream of a discharge dampener at 180 rpm. When compared to the no dampener case, the plots show many more oscillations, indicating presence of higher frequency pulsations. This higher frequency is a result of interaction between the dampener and pump.

Variation in downstream pulsation levels is shown in Figure 8, with bladder precharge pressure for various dampeners. For

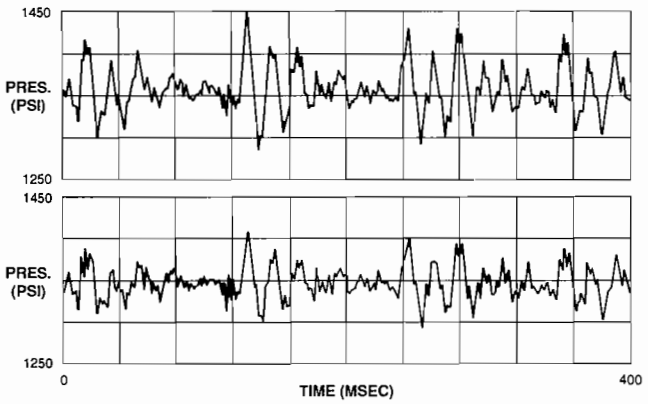

Figure 7. Pressure Pulsations Upstream and Downstream of the D-3 Discharge Dampener at 180 RPM.

these tests, discharge pressure was maintained at 1000 psig, while precharge pressure was gradually varied. Another series of tests, not reported here, was run in which bladder precharge pressure was kept constant at the manufacturer recommended level (typically 50 percent to 70 percent of the line pressure) and the line pressure was gradually varied. Both tests indicated that over a wide range, 10 percent to 90 percent, precharge pressure had little impact on the discharge pulsation levels. Computer predictions tend to verify this type of broad-range effectiveness, resulting from high compliance of even a small amount of gas in the bladder.

Compliance of the bladder material itself is relatively insignificant in the frequency range of interest to reciprocating pump designers.

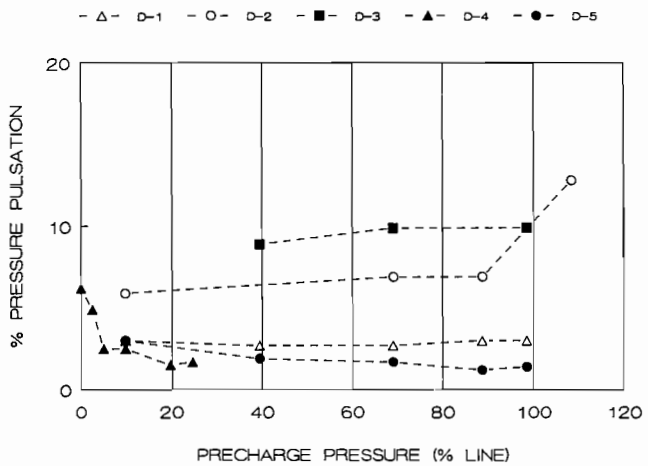

Figure 8. Variation of Pressure Pulsations with Bladder Precharge Pressure. Only discharge dampener precharge pressure was varied; suction dampener precharge pressure was maintained at the manufacturer recommended level. In D-5, precharge was varied in only one of the two dampeners.

Suction Piping Pulsations—With Dampener

For all tests, suction line pressure was maintained at approximately 80 psig. Pulsations in the suction manifold are shown in Figure 9 at various pump rpm, with different dampeners. In all cases, pulsations are reduced from the no dampener case but by varying amounts. Once again, D-4 and D-5 dampeners show the best performance, D-2 the worst, with only a minor reduction from the no dampener level. These levels are important, since they determine effective NPSHR for the pump in the test loop piping system. These results are consistent with field experience, where sometimes simple replacement of one dampener type by another can solve pulsation problems. The D-5 suction dampener, like its discharge counterpart, is a set of two dampeners, one mounted at the flanged end and the other at the closed end of the manifold.

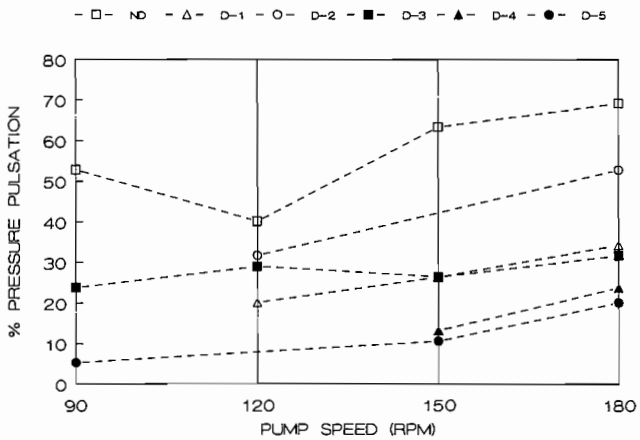

Figure 9. Variation of Pressure Pulsation in Suction Manifold with RPM. ND stands for no dampener case; D1-D5 for the five dampeners tested.

A plot of pulsations upstream and downstream is shown in Figure 10 of the D-5 suction dampener at 180 rpm. Without a dampener, the spectral plot is dominated by first few harmonics of the pump rpm times the number of cylinders, e.g., third, sixth, ninth harmonics for a triplex (Figure 3). These harmonics are still present in Figure 10 but the dominant harmonic lies at a much higher frequency of 140 Hz. This harmonic arises from a new resonance created by insertion of the dampener in the piping system. Near resonant conditions, pulsation amplitudes can vary substantially, depending on the resonance frequency and associated system damping. This explanation accounts for the wide divergence of levels seen in Figure 9.

Pulsation levels upstream of the dampener are shown in Figure 11. These levels are lower then those in the manifold shown in Figure 9 but do not change dampener effectiveness ranking. These pulsations are transmitted to suction piping and can be a source of piping vibrations. Whether pulsations cause high pip-

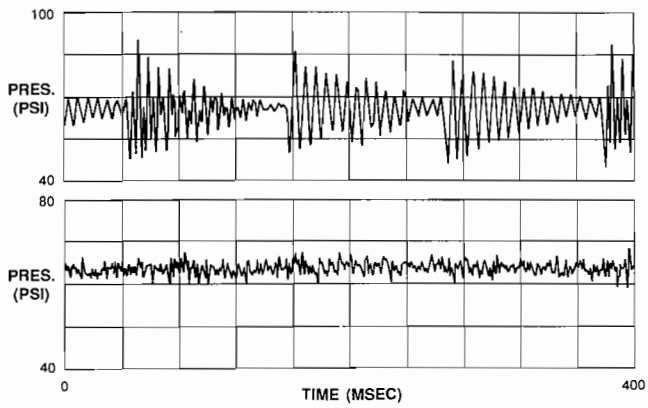

Figure 10. Pressure Pulsation Upstream and Downstream of D-3 Suction Dampener at 180 RPM.

ing vibrations depends on a host of other factors such as piping design, including supports and mechanical resonant frequencies of the piping system. A complete piping simulation program should include both mechanical and acoustic analyses.

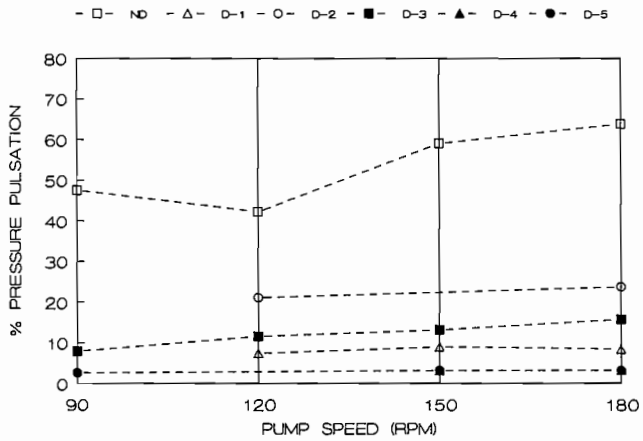

Figure 11. Variation of Pressure Pulsation Upstream of Suction Dampener (between Dampener and Charge Pump) with RPM.

The effect of bladder precharge pressure on suction dampener effectiveness is shown in Figure 12. Like the discharge case, dampener performance virtually remains unchanged over a wide range of precharge pressure. Pulsation levels are shown in Figure 13 upstream of the D-2 dampener at varying suction line pressures. These levels virtually remain unchanged from 10 psig to 60 psig precharge pressure. The line pressure is 80 psig.

A review of suction and discharge piping pulsation data and Table 1 indicates that dampener performance is not directly related to total dampener or bladder gas volume, i.e., larger dampeners do not necessarily perform better. This result, as expected, implies that the performance depends on the dampener's transfer function or frequency response which is a function of many factors (Appendix A) including dampener and bladder volumes.

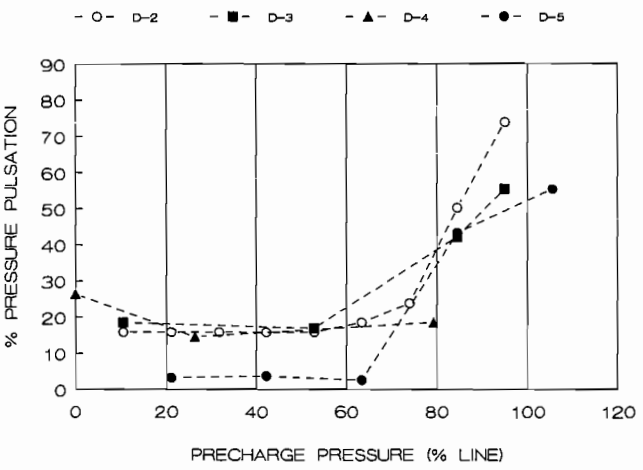

Figure 12. Variation of Pressure Pulsations with Bladder Precharge Pressure. Only suction dampener precharge pressure was varied; discharge dampener precharge pressure was maintained at the manufacturer recommended level. In D-5, precharge was varied in only one of the two dampeners.

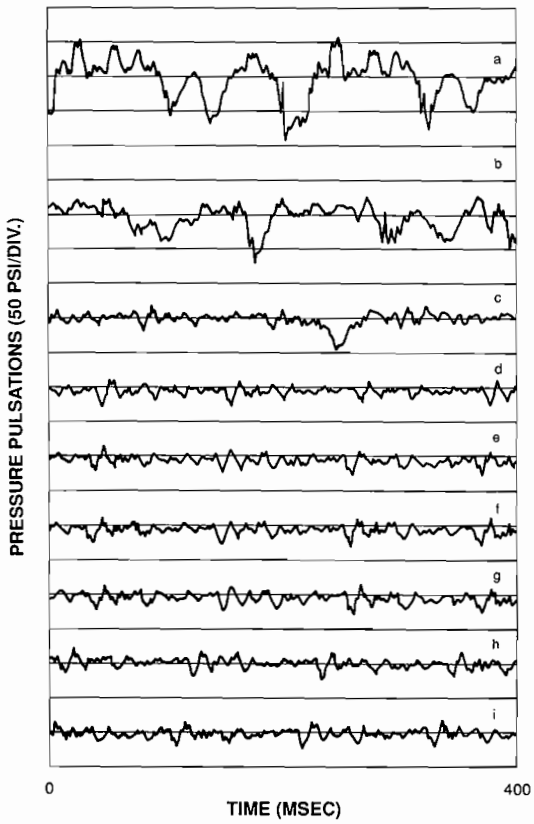

Figure 13. Pressure Traces Upstream of the D-4 Suction Dampener at Bladder Precharge Pressures Varying from 90 PSIG (a) to 10 PSIG (i) in Steps of 10 PSI. Line pressure = 80 psig.

CONCLUSIONS AND RECOMMENDATIONS

Five bladder type dampeners were tested under controlled conditions in a special test loop. Pressure pulsations were recorded at several points in the loop. Based on the review of this data, the following conclusions are highlighted.

- Pulsation dampeners can be effective in reducing pulsations throughout the piping system, although the extent of reduction depends on the type and design of the dampener. Significant variations in dampener effectiveness, measured by the attenuation of pulsations due to the dampener, were seen at all tested pump speeds.

- While dampeners caused significant reduction in the primary harmonics (3rd, 6th, 9th), their interaction with the pump manifold generated new, higher (60-100 hz) frequency resonant pulsations. Thus, the overall effect was a shift in the frequency spectrum towards these higher frequencies. It should be noted that under most conditions shifting pulsations to higher frequencies is beneficial, since excitation forces at these frequencies are lower.

- The gas bladder charge pressure level does not have a significant influence on the pulsation level as long as the bladder does not become completely ineffective. The latter happens when the bladder pressure is very low (<10 percent of the line pressure— collapsed bladder) or very high (>100 percent of the line pressure—overcharged bladder).

Based on this research and further, on their extensive experience in analysis and solution of pulsation problems, the authors offer following recommendations on the use and installation of dampeners.

• Bladder type dampeners offer an economical solution to pulsation problems in reciprocating pump piping systems. With the exception of small pumps (<25 hp), dampeners, particularly in the suction piping, can reduce or eliminate the requirement for high suction pressure.

• Dampener response depends on the piping system in which it is installed. While experience and empirical techniques provide useful guidance for dampener selection, they are not a substitute for piping acoustic analysis using well proven techniques. For critical pump applications high energy pumps and multiple-pump stations, the need for analysis is virtually imperative.

• The dampener should be installed, when analysis is not available as a guide, as close to the system piping end of the manifold as possible. Dampener manufacturers should be consulted for the proper selection and installation procedure.

• While dampener manufacturers generally recommend precharge pressures that range from 50 percent to 70 percent of the line pressure, dampeners are effective over a much broader range. The manufacturer still should be consulted about the optimum precharge pressure since it may have an impact on bladder life.

• Dampeners should not be arbitrarily moved from one location to another to meet space or logistical requirements. In some cases, such a move may be worse than not having a dampener at all.

APPENDIX

Dampener Effectiveness

A standard method for evaluation of dampener effectiveness or ranking does not exist. Part of the problem is that a dampener's response cannot be isolated from the piping system in which it is installed. In acoustic theory, transmission loss (TL) is often used to characterize acoustic filters.

$$T.L. = 20 \log_{10} \frac{P_t}{P_i} \tag{1}$$

Where P_i and P_t are incident and transmitted acoustic pressures. For a bladder type dampener (Fig. 14) attached to an infinite pipe, transmission loss can be shown to be,

$$T.L. = 20 \log_{10} \frac{\left(\frac{\rho c}{2 A_c} + \frac{8\pi\mu L A_1}{A^3}\right)^2 - \left(\frac{M\omega}{A^2} + \frac{\gamma P}{\omega V}\right)^2}{\left(\frac{8\pi\mu L A_1}{A^3}\right)^2 - \left(\frac{M\omega}{A^2} - \frac{\gamma P}{\omega V}\right)^2} \tag{2}$$

where ρ = fluid density,
μ = fluid viscosity,
C = the speed of sound in the fluid,
A_c = Area of the infinite pipe,
A = Area of the bladder chamber,
A_1 = Area of the pipe connecting bladder chamber to main pipe, when applicable.
L = Length of the connecting bladder chamber to main pipe.
M = Equivalent mass of fluid in the bladder chamber and connecting pipe.
V = Bladder gas volume at the line pressure P.
ω = Angular frequency of acoustic pressure.
τ = specific heat ratio

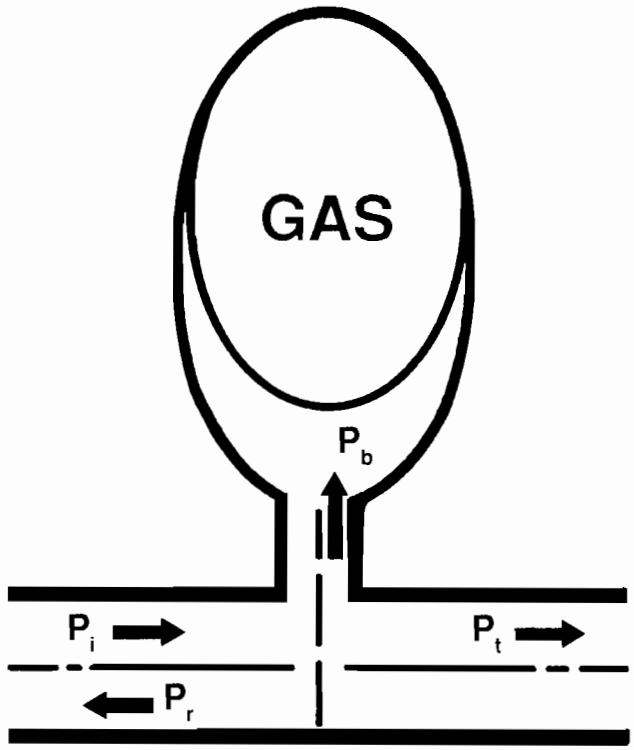

Figure 14. Bladder Type Dampener Schematic.

Equation (2) assumes plane-wave acoustics and linear bladder response. These assumptions are valid over the frequency range and bladder precharge pressures of interest in reciprocating pumps. However, TL's utility in dampener selection is limited since the incident pressure, P_i, itself is affected by the presence of dampener.

Insertion loss (IL), a measure of pulsation attenuation at some point in the piping before and after installation of the dampener, is defined as,

$$I.L. = 20 \log_{10} \frac{P_{nd}}{P_d} \tag{3}$$

where P_{nd} and P_d are acoustic pressures at some measured point before and after installation of dampener. TL, widely used in automotive muffler analysis, is a more useful indicator for pump piping systems. Insertion loss should be calculated at several points in the region of interest to avoid confusion between pulsation node and low pulsation levels. In this study, insertion loss approach has been adopted by showing actual pulsation levels before and after dampener. Substitution of these levels in Equation (3) will yield insertion loss.

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