Acoustical analyses solve vibration, failures in

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Vibration and failure problems with the piping and reciprocating pump internals of an Ecopetrol oil-pipeline pump station in Colombia were solved with acoustical analyses of the suction and discharge systems by a digital computer program.

The program was further used to redesign the piping systems to reduce high pulsation, shown to have been the source of cavitation in the suction system at nearly all operation conditions.

An all-liquid filter was specified in the redesign of the piping modifications, completely eliminating piping failures. The pumps have continually delivered 40,000 b/d since that time with only normal maintenance problems.

Colombian site. Problems occurred with four triplex reciprocating crude-oil pumps operating in parallel at the Dina pumping station, Colombia (Fig. 1). The pumps had a rated speed of 275 rpm with a capacity of 388 gpm. The nominal suction pressure was 60 psig, and the discharge pressure was

Ecopetrol's Dina pump station, Colombia (Fig. 1).

Fig. 2

Measurement locations

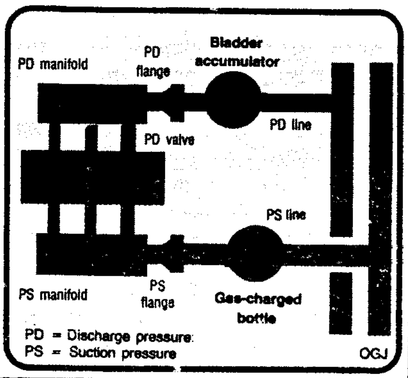


Fig. 3

Cavitation on the suction stroke

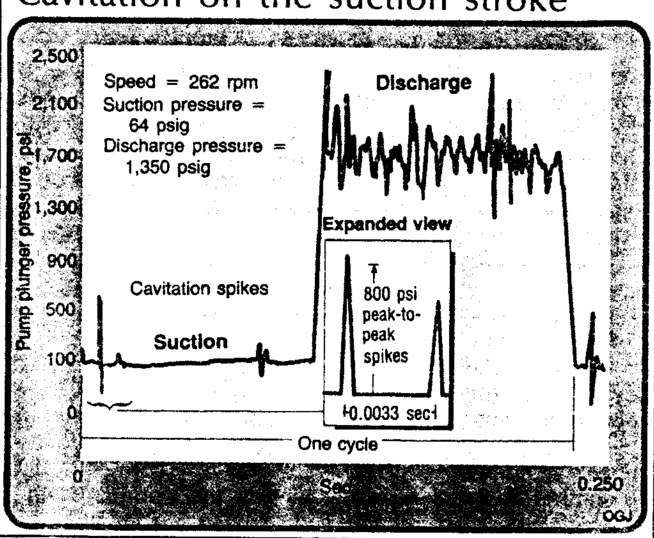
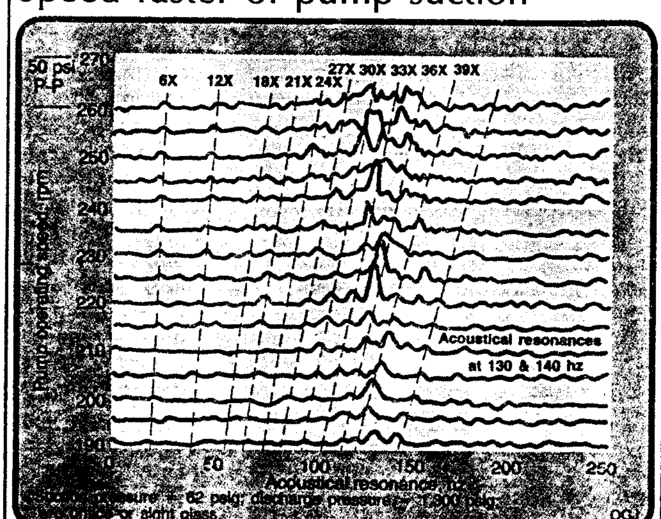


Fig. 4

Speed raster of pump suction *



recip pumps

1,800 psig.

The Delrin pump valves had repeated fatigue failures beginning 3 months after startup. The discharge valve disks were replaced with steel, and the Delrin disks used on the suction valves were replaced every 90 days to avoid fatigue failures.

Valve failures were controlled after the first 9 months of station operation. For the first 4 months, there were no pull-rod failures; however, there were 18 failures in the following 18 months.

Many of these failures required replacement of the crosshead, the guide ways, and on two occasions a broken or bent connecting rod. The suction and discharge piping systems vibrated excessively, resulting in numerous piping fatigue failures. Attempts to control the piping vibrations with pipe clamps and additional supports were unsuccessful.

The four pumps had a common suction header supplied by a charge pump which was capable of supplying pressures up to 90 psi. The discharge of the four pumps fed into a common header which connected to the main pipeline. The original piping design included bladder-type accumulators on both the suction and discharge.

It was difficult to keep the pumps running smoothly because constant maintenance was needed to keep the accumulator bladder pressures charged to approximately 60 to 70% of line pressure. The static discharge could change from 1,600 psig to less than 700 psig in a few minutes if the down-line booster stations started up. When this happened, the accumulator was ineffective.

The cost of the parts and labor attributable to this problem exceeded $1 million. The operator, the piping designer, and the pump manufacturer began a study to determine the cause or causes of the vibrations and failures. The complex relationship of the system variables, however, made it difficult to develop definite conclusions.

Piping changes. Several changes were made in the piping system in an attempt to improve the vibrations and reduce the failures. These modifica-

Fig. 5

Onset of cavitation in suction pressure

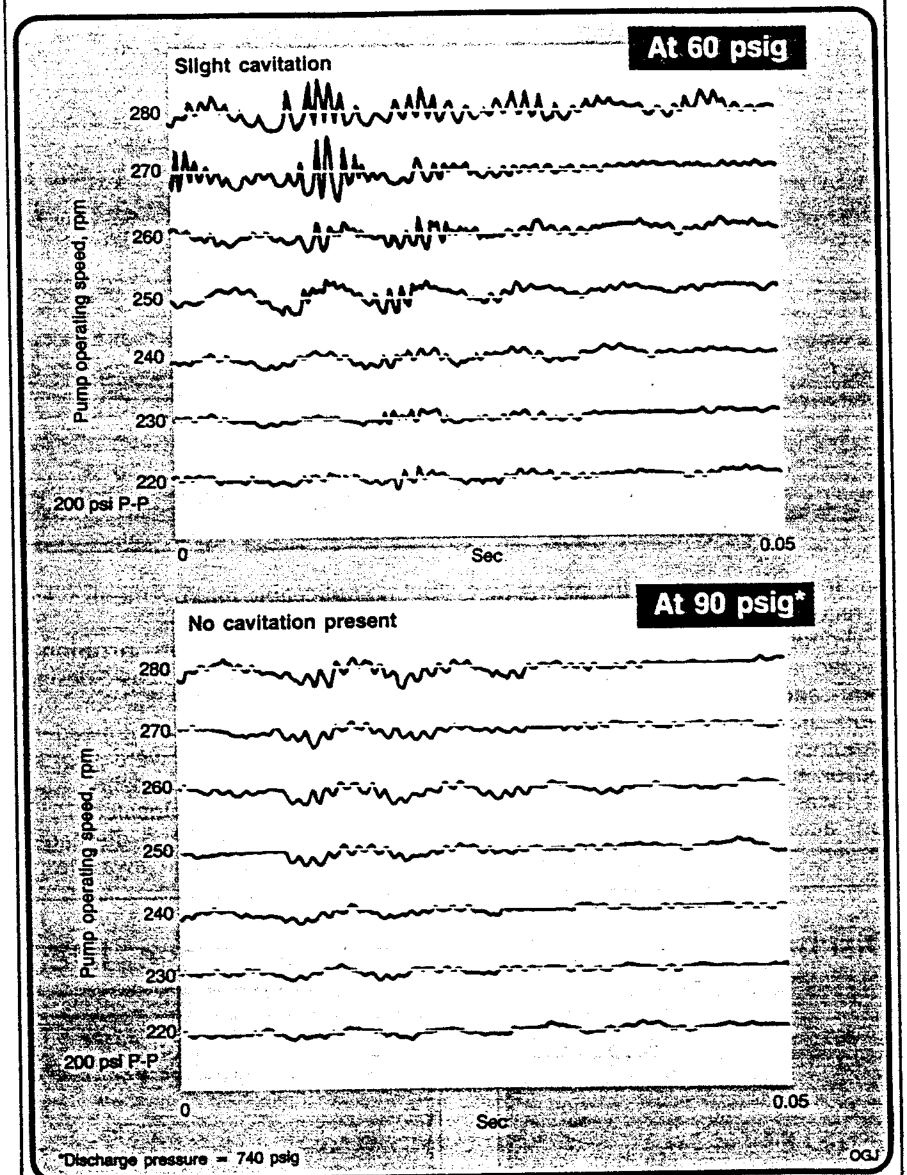


Fig. 6

Pump/piping simulation model

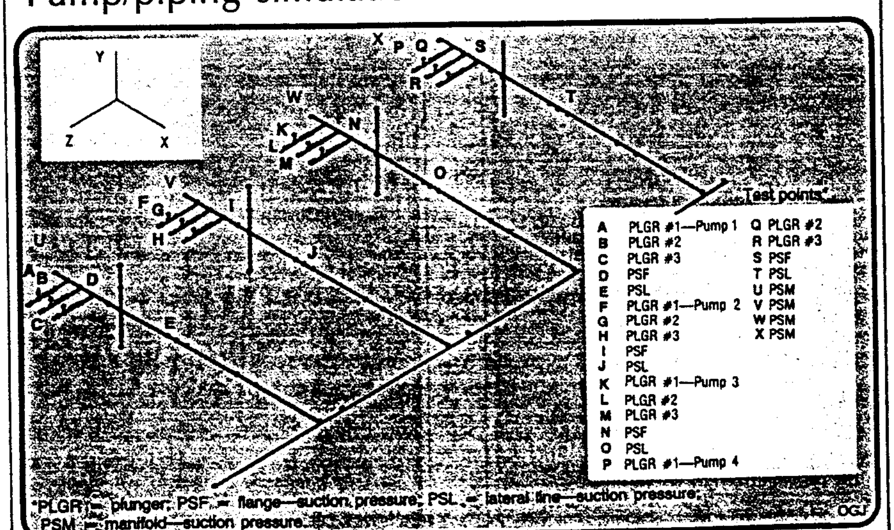
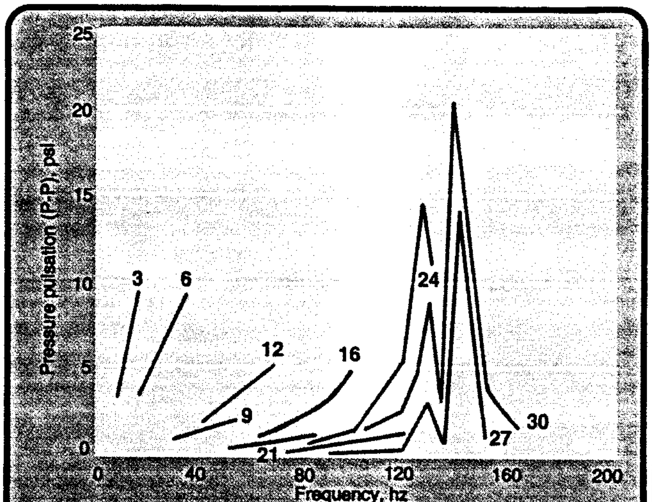


Fig. 7

Predicted pulsations*



*In existing suction piping system, w/gas-charged filter

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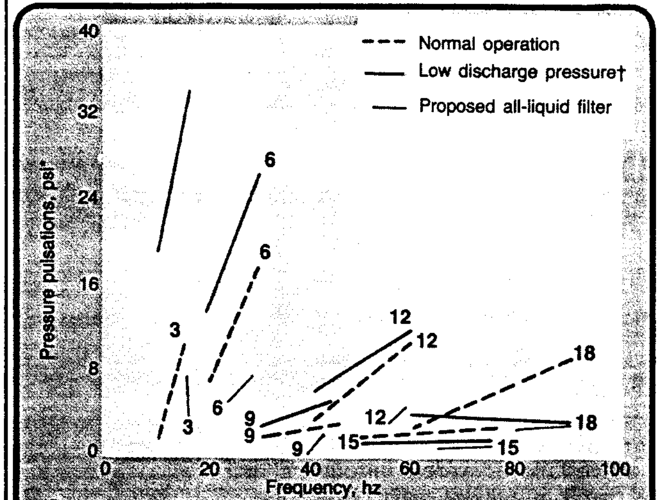
Fig. 8

Predicted discharge pulsations w/filter



*Peak to peak. †With bladder-type accumulator.

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tions included changing the piping (at the recommendation of the accumulator vendor) so that the flow would be directed at the bladder. This piping modification did not improve the pulsation characteristics of the system.

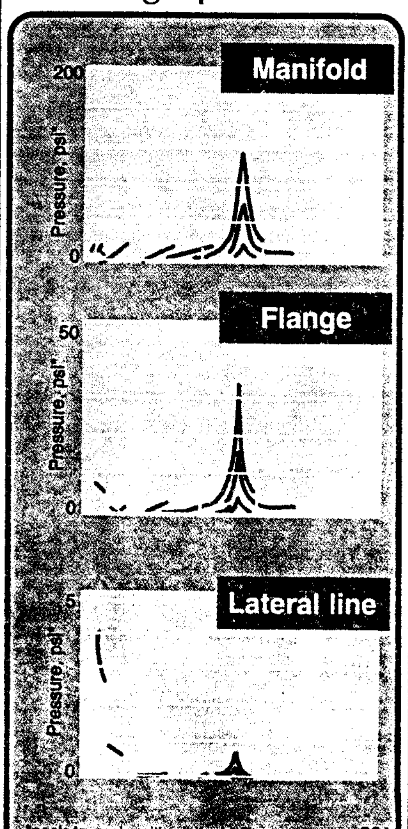
Another modification which was tried on the suction side of Pumps 1 and 3 was replacement of the bladder-type accumulators with nitrogen-charged, flow-through accumulators. But no noticeable improvements were observed.

The severity of the problems brought into question the basic design of the system, since the suction and discharge lead lines from the headers to the pump manifold were shorter than normal for most pipeline stations. The pumps were located 16 ft apart with the suction and discharge headers located 10-12 ft away from the pump flanges.

The station capacity was 39,900 b/d when 3 of the 4 pumps were operating at their rated capacity of 388 gpm. This results in a fluid velocity of 3.3 fps in the 12-in. schedule-40 suction manifold and 6.9 fps in the 10-in. schedule-XS discharge manifold. The flow velocities in the individual pump piping were 1.1 fps in the 12-in. standard-weight suction

Fig. 9

Discharge pulsations*



*peak to peak, with all-liquid filter

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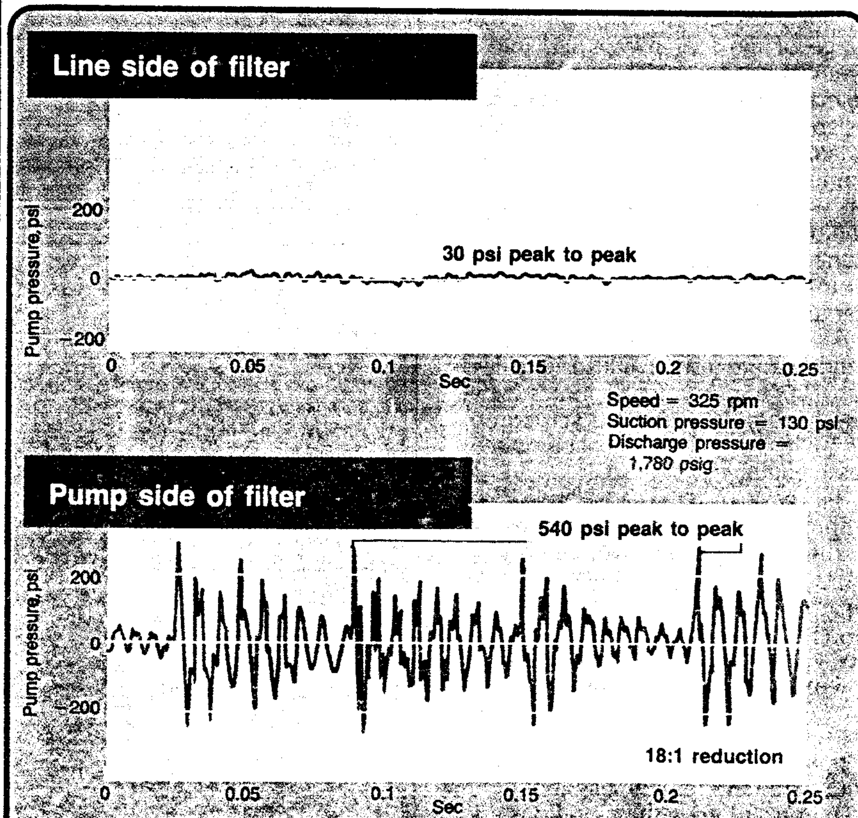
Fig. 10

Discharge flange, lateral line pulsations*



*Input & output of all-liquid filter on pump discharge

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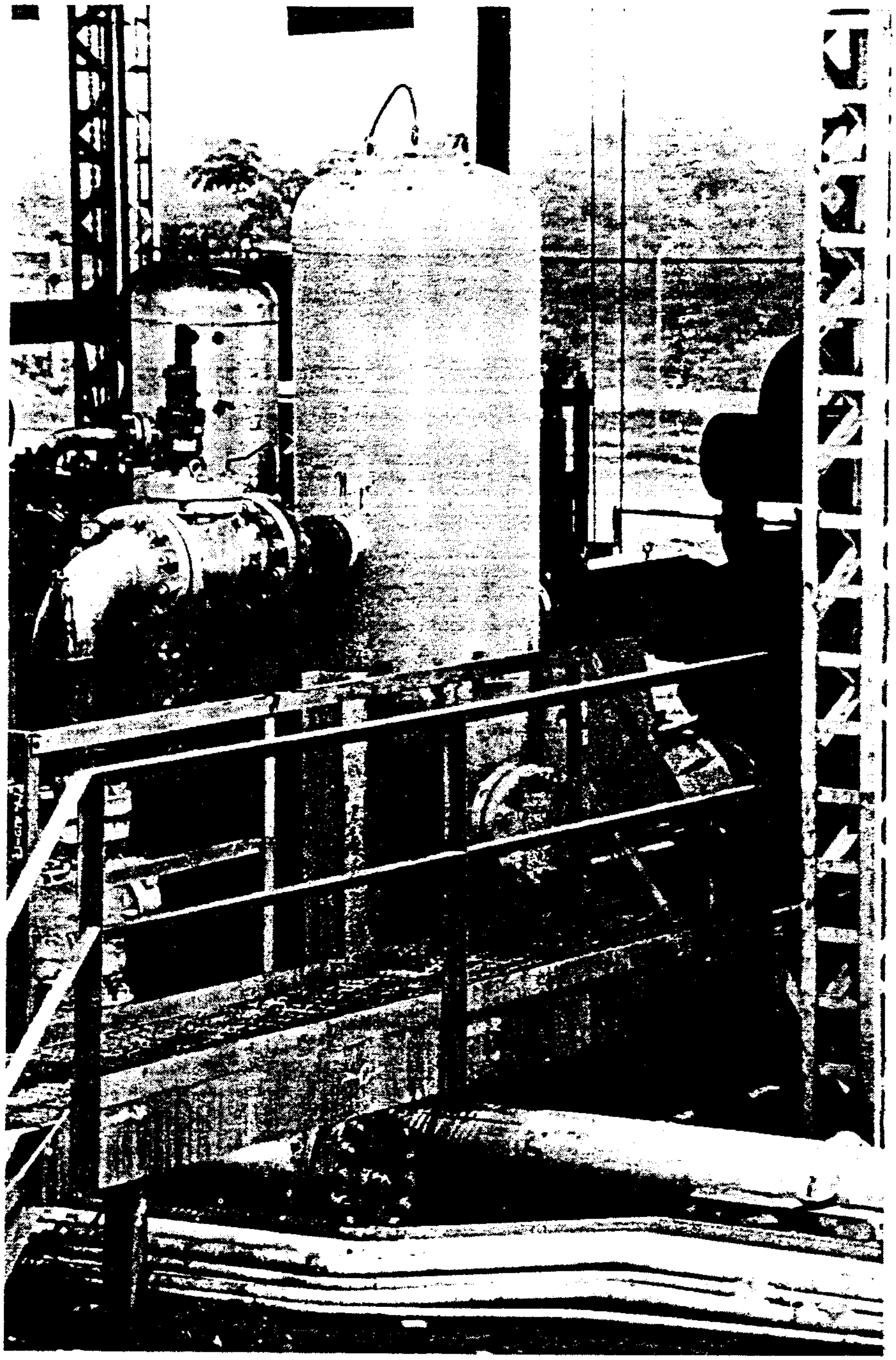

All-liquid filter installed at the pump flange (Fig. 10).

pipe and 2.7 fps in the 8-in., extra-heavy discharge pipe.

Field investigation

Engineering Dynamics Inc. (EDI) was requested to investigate and make recommendations to alleviate the problems. The investigation included two phases.

The first was to model the acoustical characteristics of the suction and discharge piping systems with digital simulation. The second phase was a detailed field investigation to evaluate the pulsation and vibration characteristics of the pumps.

Solutions were then developed to eliminate the problems and installed in January 1985. A follow-up field study was made in March 1985 to determine if the modifications were adequate for long-term reliability.

Procedures. Piezoelectric pressure transducers and accelerometers were used to measure the pressure pulsations and the vibrations. A sketch of the pump suction and discharge piping illustrating some of the pressure test points is shown in Fig. 2.

The pulsation and vibration signals were analyzed for frequency content with a two-channel Hewlett-Packard 3582A FFT analyzer and documented on an HP 7470A digital plotter. The analyzer and instruments were controlled by an Apple II+ microcomputer using EDI software to analyze the vibration and pulsation data.

Cavitation. The initial vibration surveys revealed high vibration amplitudes on the suction piping, indicating large excitation forces present in the piping systems. Analysis of the pressure pulsation waveforms revealed severe cavitation in the suction piping system.

For liquid reciprocating pumps, the static pressure in the suction system must be adequate to compensate for frictional pressure-drop losses, the required acceleration head, and the pulsations present in the system. This capacity ensures that the pressure remains above the vapor pressure. The vapor pressure of the oil was less than 2 psia.

When pulsations exist in a system, they will consist of a positive peak of pressure which will be added to the static pressure and a negative peak which will be subtracted from the static pressure.

If the negative peak of the pulsation, when subtracted from the static pressure, reaches the vapor pressure, the fluid will cavitate, resulting in high-pressure spikes as the liquid vaporizes and then collapses as the pressure increases above the vapor pressure.

To illustrate the formation of cavitation, Fig. 3 presents the plunger pressure-time wave which shows that cavitation occurs on the suction stroke. Note than when the cavitation portion of the waveform is expanded, the pressure spikes are approximately 800 psi with a time period of approximately 0.00025 sec.

The presence of cavitation can usually be observed on the complex wave because pulsations, generally sine-shaped waves, will "square-off" at the trough of the waves when the vapor pressure is reached.

The effect of the static pressure on the cavitation was investigated by an elevation of the suction pressure to the maximum possible (90 psig). The increase in suction pressure alone was not sufficient to eliminate the cavitation. Severe pulsations were found with levels in excess of 200 psi peak-to-peak.

At a suction pressure of 76.5 psia, pulsations of approximately 75 psi zero-peak are required to cause cavitation. This value is the difference between the negative pulsation peak and the static pressure. Since pulsations greater than 75 psi were always present at the higher speeds, cavitation always occurred.

The presence of cavitation makes it difficult to evaluate the influence of variables, such as the effect of other units, speeds, and the accumulator design. A reduction of the pressure pulsations was necessary in order to obtain meaningful test data on the units.

Acoustical resonances. The major suction pulsation components occurred at acoustical resonances at frequencies between 110 and 150 hz with pulsation amplitudes of approximately 100-150 psi peak-to-peak (p-p), which, when combined with the pulsation at the lower pump harmonics, caused the overall static pressure to drop below the vapor pressure.

Acoustical resonances amplified the pulsations whenever one of the harmonics of the pump speed passed through the resonant frequency. The acoustical resonance near 130 hz was associated with the 9-ft length of the suction manifold, the accumulator, and the pump internal passage volumes.

When an acoustical resonance is encountered in a system, the pressure pulsations can be reduced by elimination of the resonance or by attenuation of the amplitudes through the addition of a resistive element, such as an orifice.

Therefore, an orifice plate was installed at the suction flange in an attempt to attenuate the pulsation amplitudes and reduce or eliminate the cavitation. A diameter ratio (orifice diameter-to-inside diameter of pipe) of approximately 0.4 was used.

When the orifice plate was installed, the pulsations were reduced, not sufficiently, however, to eliminate completely the cavitation.

Interaction. All the other pumps were shut down and Pump 1 was run to determine if the cavitation was caused by interaction with the other pumps or was a function of the individual piping design.

These tests indicated that the pulsations were primarily caused by the individual pumps and that the major factor was the acoustical resonances near 130 hz. This test also gave evidence that the location of the pump in the piping system was not a major factor in the cavitation.

This latter conclusion was verified by the fact that cavitation occurred on Units 1 and 3 at the exact same speed under the same operating conditions. Units 1 and 3 were separated by 32 ft with Unit 2 midway between them. If the location of the pump in the header were a prime factor, there would have been different pulsation and cavitation characteristics.

To investigate further the interaction of the other pumps, tests were made with Pump 1 on the verge of cavitation and the adjacent Unit 2 was swept through the entire speed range to determine if it affected the speed at which cavitation occurred. This test showed that the adjacent unit did not significantly influence the cavitation.

In an attempt to determine whether the acoustical resonance was associated with a piping length from the other units, the suction block valve was pinched momentarily to see if a pressure drop taken on the upstream side of the accumulator would affect the resonances in the 130 hz range. The pressure drop of approximately 10 psi in the block valve did not have a significant effect.

After the orifice plate was installed and the nitrogen-charged accumulator bottle on the suction system had the maximum gas charge, the cavitation was eliminated over much of the speed range, making possible study of the effect of varying system parameters.

The normal procedure for the testing involved establishing a set of steady-state conditions (such as suction pressure, gas volume in the bottle, or charge pressure in the bladder accumulator, speeds on the other pumps, etc.) then changing the pump speed from 190 rpm to 290 rpm.

During the speed run, the pulsations at several locations in the suction and discharge piping were tape recorded. The resulting data presentation for the speed variation is given in Fig. 4, showing pulsation pressures in the suction manifold of Pump 3 over the speed range.

The data show that the primary cause of the cavitation was the high-level pulsations at the acoustical natural frequencies in the system near 130 and 140 hz which were excited by the 21st through the 30th harmonics of pump speed.

Speed effects. The effect of speed on cavitation at the suction valves can be seen in Fig. 5 which gives the complex pressure waves for speeds from 220 to 270 rpm for a suction pressure of 60 psig. Pulsations generally increase with speed unless there are acoustical resonances.

As shown, when the speed increased to more than 250 rpm, the pulsations increased to the point that the negative pressure pulsation amplitude was near the vapor pressure and the wave became flattened on the trough.

As the speed was further increased, the cavitation became more severe.

Static-pressure effects. When the static suction pressure was increased to 90 psig, the pulsation amplitudes were reduced and the unit could be run at 280 rpm without cavitation (Fig. 5). The higher suction pressure seemed to inhibit the amplitude of the pulsations.

The results of these tests indicated that the cavitation could be reduced by increasing the suction pressure to the maximum possible, installing an orifice plate to reduce the pulsation amplitudes, and ensuring that the accumulator was properly charged.

The effectiveness of the gas-charged, flow-through accumulator was strongly influenced by the volume of the nitrogen gas in the accumulator bottle. The increased gas-charge volume eliminated pulsation components of 46 psi at 12 hz and 22 psi at 65 hz.

Discharge piping. The measured field data showed high amplitude pulsations in the discharge piping with levels exceeding 1,000 psi peak-to-peak in some tests (Fig. 3).

An investigation was made to determine if the discharge pulsations were affected by the cavitation on the suction side. The complex pressure waves at the suction and discharge valves were captured simultaneously during the time that severe cavitation was present and showed that the discharge side was isolated from the suction side.

Although the pulsation amplitudes were very high, they were not caused by the cavitation on the suction. The pulsations were a function of the energy output from the plungers and were strongly influenced by the valve ringing and the acoustical resonances as dictated by the acoustical properties of the bladder-type accumulator and the piping system.

Whenever the station discharge pressure dropped below the charge pressure in the bladder of the discharge accumulator, a noticeable increase in the pulsations occurred.

The pressure pulsations of 100-200 psi peak-to-peak measured in the discharge lateral caused shaking forces of 4,500 to 9,000 lb at the bends in the discharge piping. Forces this large are difficult to control with normal pipe clamps and support. These forces caused excessive vibration resulting in repeated fatigue failures.

Acoustic simulation

The rapid advances of digital computers have made it more practical to analyze digitally the acoustical characteristics (pulsations) of piping systems.

A comprehensive computer program has been written by EDI to predict pulsation levels for piping systems with liquid pumps or gas compressors. The program can be used to design pulsation filters or to evaluate the effectiveness of systems with liquid/gas accumulators.

The program is based on basic fluid-dynamics relationships (the equation of motion, the continuity equation, and the thermodynamic equation of state), from which the plane-wave equation is developed.

This one-dimensional flow assumption adequately simulates the motion of pressure disturbances and the acoustic response of typical piping systems found in most industrial plants. The effects of flow on damping and of pipe wall flexibility on the speed of sound are included in the analysis.

The program is written in a general manner so that any piping network can be simulated by combinations of distributed or lumped elements and any number of flow excitation points may be specified. Terminating boundary conditions of flow, pressure impedance, or non-reflective impedance can all be specified.

In liquid pump systems, the flow wave generated by the plunger is the source of pulsation. The flow rate is a function of piston velocity and the valve behavior.

The computer program generates the plunger combined flow-time wave, and the acoustical response of the piping system to the harmonic content is then computed over the specified speed range.

Suction piping modeled. The acoustic response of the piping system was simulated digitally. The suction system was modeled as shown in the piping geometry (Fig. 6.) Note that all four pumps can be simulated in the analysis.

The results of the computer analysis of the original suction piping system with the gas-charged filter are given in Fig. 7. Pulsation at any selected location in the piping system can be predicted and presented. The predicted pulsations at each harmonic of pump speed from minimum to maximum speed are calculated and plotted. The harmonic numbers are indicated adjacent to the appropriate curve.

The interaction of the individual harmonics with the acoustic resonant frequencies at 130 and 140 hz can be seen. These data can be compared to the measured pulsations given in Fig. 4. Generally, there was good agreement with the acoustical resonances at 130 and 140 hz; but the calculated amplitudes were lower.

It must be remembered in the assessment of the field data that cavitation was still occurring on the suction side and the amplitudes measured would be expected to be higher than calculated for the steady-state operating conditions. Note that the lower order harmonics, 3X, 6X, 9X, 12X, etc., are close to the calculated values.

While investigators were in the field, an orifice plate was installed in the suction flange and was successful in reducing the pulsations and cavitation. This modification agreed with the computer analysis which showed a reduction in responses.

The acoustical analyses showed that if the installed gas-charged, flow-through liquid accumulator were located 2 ft closer to the pump flange, pulsation levels inside the suction manifold would be lower. Therefore, all four pumps were modified to include the flow-through, gas-charged accumulator on the suction piping, mounted as closely as possible to the suction flange.

Discharge piping analyses. The pulsation characteristics of the discharge piping system were analyzed and indicated that the bladder-type accumulator was not effective for some of the station operating conditions.

For example, the discharge pressure would sometimes fall to 700 psi from the normal 1,600 psig.

When this happens, the bladder becomes fully expanded, blocking off the entrance of the accumulator and voiding the beneficial effects of the gas volume.

Also, the high-level pulsations caused bladder failures which then eliminated the gas cushion on the back side of the bladder, making the accumulator ineffective.

Fig. 9 compares the predicted pulsation for the normal conditions and the case with low discharge pressure. It can be seen that the pulsation amplitudes at the lower harmonics significantly increased when the discharge pressure reduced and the accumulator entrance was blocked.

An all-liquid, low-pass, acoustic filter was designed for the discharge system which significantly lowered the pulsation levels (Fig. 8). An all-liquid pulsation filter system consists of volume-choke-volume or a volume-choke-piping arrangement which is specially designed to attenuate the pulsations above specified frequencies (Fig. 9).

An all-liquid filter design has the advantage that it need not depend upon a gas charge to operate effectively.

The specially designed all-liquid filter was installed on the discharge piping as shown in Fig. 11. The discharge filter was designed so that it could be flanged between the pump manifold flange and the discharge piping valves.

The discharge filter used a bottle that was 8-ft long with an inside diameter of 31.5 in. and an 8-ft long, 2-in. choke tube. The choke tube was placed inside the bottle due to space limitations. The pressure relief and bypass valves were incorporated into the discharge bottle design.

Follow-up. After all the modifications were installed, the plant personnel reported that significant reductions in the piping vibration and overall noise were obvious.

A follow-up field study was made in March 1985 to document the pulsations and vibrations. For the suction system, the suction pressure was increased to 145 psia by a new booster pump in order to reduce the cavitation.

Significant reduction in the pulsations transmitted through the gas-charged accumulator was measured.

As long as the suction pressure was maintained at more than 100 psi, the cavitation was minimal. Vibrations on the suction piping were less than 5 mils peak-to-peak overall.

On the discharge side, the all-liquid filter was very effective in attenuating the pulsations transmitted to the discharge piping and was not sensitive to speed or pressure. Fig. 10 gives the discharge flange pulsations and the lateral line pulsations for the highest speed and pressure.

The attenuation of the complex wave, peak-to-peak pulsations across the all-liquid filter was greater than 10:1 over the full range of pressure and speeds. Vibration levels measured on the discharge system were less than 10 mils peak-to-peak throughout the speed range. Pulsations were still relatively high in the pump plungers and discharge manifold, primarily due to the all-liquid filter. This mode can be attenuated by the use of an orifice at the discharge flange.

The pump manufacturer's maximum allowable rod load was 30,000 lb, which occurred at a discharge pressure of 1,900 psia. The pulsations caused the peak rod load to exceed 44,000 lb and was one of the causes of the power end failures. The maximum peak rod load after the filter modification was 34,500 lb and was considered to be a safe value by the manufacturer.

Since the installation of the all-liquid filter and the suction modifications in January 1985, the pumps have delivered 40,000 b/d with only one significant failure. The piping failures have been completely eliminated due to the elimination of the large shaking forces in the piping.

Conclusions

The results of the field tests and the acoustical analyses led to these conclusions:

1. Suction pulsations that cause the instantaneous pressure level to drop below fluid vapor pressures result in cavitation. Cavitation can contribute to failure of pump parts such as valves, crossheads, rods, etc.

Severe cavitation can cause high piping vibrations and failures, including vents, drains, and gauge lines. The forces can be so high that normal pipe clamps and supports may be ineffective in controlling the vibrations.

2. Testing revealed that an increased static suction-pressure level lowered the level of the pulsations in the pump manifold and reduced the cavitation. The testing also showed that the pulsation levels increased with speed.

3. Pulsations generated by the flow modulation from the pump plungers were amplified by the acoustical resonances of the piping system.

One of the major acoustical resonances measured was caused by resonances associated with the pump-manifold internals and the accumulator. This occurred on both the suction and discharge manifold systems.

The frequencies and amplitudes at these resonances were a strong function of the type and effectiveness of the accumulator.

4. The piping failures in the discharge piping were caused by large shaking forces which were caused by high pulsation levels.

5. The failure of the pump components was caused by the high pulsations in the pump discharge manifold and by the cavitation in the suction system.

6. When the design recommendations for this case were installed, the excessive pulsations were reduced and piping vibration and failures were eliminated.

7. The field testing showed that the pulsations generated for this system were primarily a function of the individual suction and discharge piping and the basic pump design and not strongly influenced by the interconnecting piping with the other units. Other systems tested have shown that the pulsation can be influenced by the piping from other pumps and the location of the pumps in the system.

8. Many vibration and failure problems in reciprocating pumps in oil pipelines or other applications are caused by system-related acoustical resonances which cause high-level pulsations in the suction and discharge piping. Proper filter design can eliminate system-related acoustic problems where the piping design and pumps are not at fault, only the interaction between the two.

This case is an excellent sample of this phenomenon because the pump and piping remained the same; only new or improved accumulators or filters were inserted between the two.

9. Pulsations in pump/piping systems are a function of the piston plunger velocity and the interaction of the flow modulations with the system acoustics and the valve dynamics. These valve dynamics' effects can be simulated by including the valve and spring equations with the acoustical equations in a time-domain iteration analysis.

10. The acoustical characteristics of pump and compressor piping systems can be analyzed with a digital simulation procedure which considers the fluid acoustic properties, the pump internals, and piping geometry. Multiple pumps can be simulated so that interaction between pumps can be studied. Design changes which will reduce pulsation levels can be determined.

11. All-liquid filters utilizing a volume-and-choke tube can be designed to reduce pulsations to acceptable levels, thus reducing piping vibration and failure. All-liquid filters need to be designed specifically for each installation to prevent acoustic interaction with the entire piping system and to prevent excessive pulsations in the plungers and pump manifolds.

12. All-liquid filters are not sensitive to large changes in pump speed, pressure, or flow and should be considered whenever large variations in such parameters are expected. Practically no maintenance is required, which can result in economic benefits, if labor maintenance costs are considered.

13. Piping bends in the suction and discharge piping should be reduced since they serve as coupling points for the shaking forces from pulsations. Piping should be clamped and supported rigidly near each bend since the system can have very high pulsations and shaking forces.

The natural frequencies of the individual spans should be as high as practical, which can be achieved by short spans between supports. U-bolt types of clamps are usually not effective; saddle clamps provide better vibration control.

14. Acoustical pulsation simulation of the pump and piping system should be performed in the design phase to develop pump and piping systems that will not experience excessive vibration, piping failures, or pump component failures.

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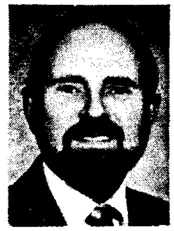
Wachel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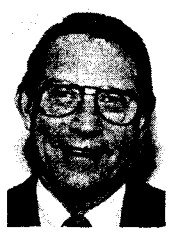
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