INCORRECT VALVE SELECTION ON PLUNGER PUMPS RESULTS IN UNDETECTED HIGH FREQUENCY VIBRATION AND COSTLY FAILURES, A CASE STUDY

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ABSTRACT
An amine triplex plunger pump system had been extremely noisy during the four months since its installation. Piping welds and nipples on the discharge piping had failed frequently, causing substantial loss of amine each time. The owner installed discharge pulsation dampers on the triplex pump. This procedure eliminated a severe vibration, but failures of nipples persisted on the discharge piping.

Spectrum analysis and normal piping vibration guidelines did not indicate that the vibration was severe enough to cause such problems. The vibration waveform, however, showed positive and negative 1.0 in/s spikes at irregular intervals. The suction valve closing angles were up to 30 degrees after dead centre and the discharge valve closing angles were up to 23 degrees after dead centre.

On the recommendation of Beta and with the concurrence of the pump supplier, the owner installed valves with stiffer springs to correct the late valve closures. The original suction and discharge valve sleeves were found to have excessive wear. Some of the valve plates and backguards were also worn. Cavitation damage was found on at least one plunger.

When the pump was run with the new valves it was so quiet that operators questioned whether the pump was functioning. The suction and discharge valve closing angles were around 10 degrees after dead centre. The time base plot of the vibration waveform showed the vibration levels to be significantly lower, and the large positive and negative spikes were no longer occurring.

1. THE PROBLEM

An amine triplex plunger pump system had been extremely noisy during the four months since its installation. Piping welds and nipples on the discharge piping had failed frequently, causing substantial loss of amine and plant downtime each time.

The owner then sought third party assistance.

2. THE SYSTEM

The amine system consists of two triplex plunger pumps, with one at 100% standby, as shown in Figure 1. The system is used to remove hydrogen sulfide and carbon dioxide from a natural gas stream. Each plunger pump had six valves, with steel plates.

Motor speed was about 1760 RPM and the pumps were belt driven to run at approximately 337 RPM.

The suction side had a booster pump and a desurger, and the discharge side had a desurger.

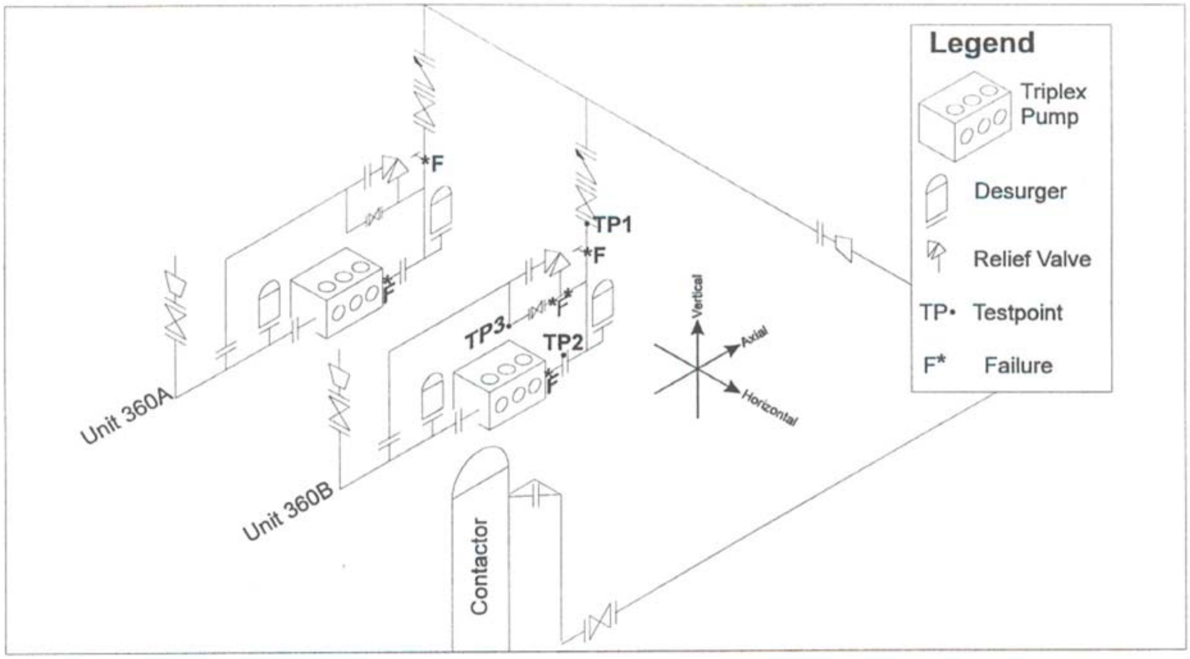

Figure 1 *Two identical plunger pumps, with one on 100% standby, are used to remove hydrogen sulfide and carbon dioxide from a natural gas stream. Both pumps experienced failures on the discharge piping.*

3. THE HISTORY

Figure 1 shows the areas where failures occurred. Both pump discharge system had similar failures.

Specifically, failures occurred on

- the relief valve nipple on the vertical riser directly after the pump discharge,
- the pressure tap nipple at the same location, and
- the top side of the discharge nozzle socket weld.

The pump manufacturer explained that the plunger pump had been designed for 33 psig suction and 675 psig discharge but was operating at 60 psig suction and 337 psig discharge.

Originally, the discharge system had been installed without desurgers, which was an oversight. The owners recognized this shortcoming after a few hours of operation, and installed discharge desurgers where fittings were available. Improvement was noted. Later, they installed a larger desurger for each pump, as close as possible to the pump's discharge. There was a substantial reduction in ***perceptible*** vibration. However, piping failures continued to occur, each time causing downtime and loss of amine.

4. THE MEASUREMENTS

The unit was pumping 103 US gal/minute, compared to the rated 106 US gal/minute.

Vibration was measured at several points in the vicinity of the pump. See figure 1, Testpoints 1, 2 and 3. Particular attention was paid to the discharge system, where breakages had occurred.

As a rough screening guideline for piping vibration, we used an empirically determined figure of 1.0 in/s pk, for any peak in the spectrum. This figure is based on close to thirty years of field experience and has usually proven to be a conservative level.

Piping attachments had vibration levels of up to 2.0 in/s pk (Figure 1, TP 3). This level is double the guideline, and explains why the nipples

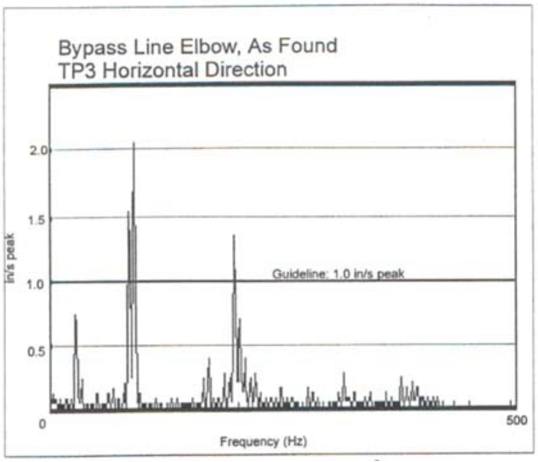

Figure 2 *Vibration on the piping attachments was double the guideline.*

failed. This vibration was not readily apparent to the human touch, and it was later determined to be a high frequency vibration. See Figure 2.

Vibration on the main piping, on the other hand, was well below guideline, based on spectrum analysis. The discharge riser (TP 1) had approximately 0.5 in/s pk vibration in the horizontal and vertical directions. The pump discharge nozzle flange (TP 2) showed vibration of 0.3 in/s at 2X plunger passing frequency. See Figure 3.

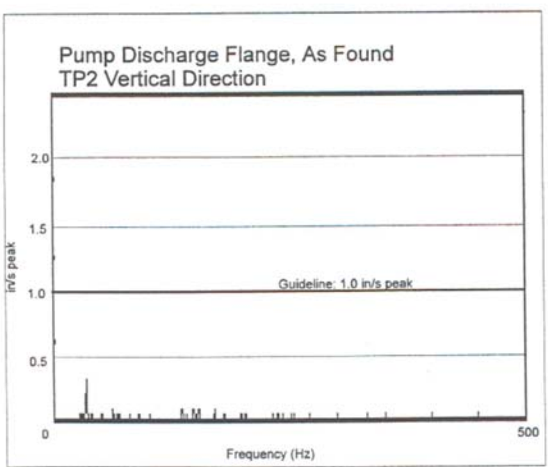

Figure 3 *According to the spectrum, vertical vibration on the pump discharge flange was well within the guideline, even though the weld had broken at the pump.*

Detection Versus Diagnosis

Vibration analysis has two distinct aspects, detection and diagnosis. For diagnosing a problem, spectrum analysis is superior, since it gives you the opportunity to identify which machine element is causing the unacceptable vibration.
For pure detection, however, spectrum analysis is occasionally misleading. For example, if the waveform is "peaky", the Fourier Transform is a long series of harmonics, with the total energy in the signal distributed among them. The actual vibration could easily be excessive even though each spectrum peak is well within guidelines. For problem detection, therefore, it is advisable to use a "true peak" overall detector or to review the time domain plot.

These vibration levels are normally considered acceptable and therefore, the failure that occurred at the pump discharge nozzle weld would not have been expected. Further investigation was required.

In an attempt to determine why breakage occurred at the pump discharge flange even though vibration was apparently acceptable we measured time waveforms. These revealed a much different picture. On the pump discharge flange in the vertical direction, for instance, (same location as shown in Figure 3), "spiky" positive and negative 1.0 in/s peaks occurred. See Figure 4.

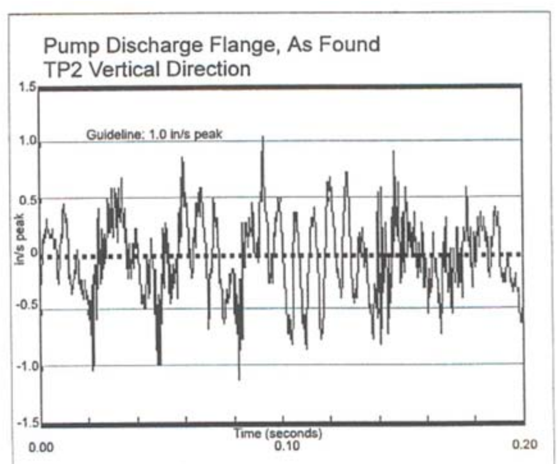

Figure 4 *The time domain pattern showed sharp peaks, with much greater amplitude than that shown in the spectrum*

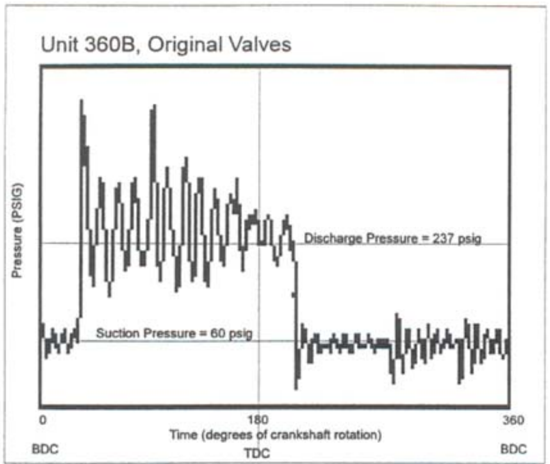

Figure 5 *The As Found Pressure-Time curve showed that the valve events were significantly late.*

Now, the question was “What phenomenon could generate sharp vibration peaks?” Experience led us to question the valve timing on the pump. To check, we measured a pressure-time curve, shown in Figure 5.

The suction valves closed up to 30 degrees after dead centre, as compared to the expected angle of less than 10 degrees. Similarly, the discharge valves closed up to 23 degrees after dead centre.

5. THE THEORY

Good Valve Behaviour

In plunger pumps, valves open and close sequentially: suction valves close, then discharge valves open, then discharge valves close, then suction valves open. A suction opening event ***cannot*** happen until discharge closing has occurred; a discharge opening event ***cannot*** happen until the preceding suction closing has occurred. Valve opening events occur largely due to pressure differential across the valve; closing events occur largely due to spring force (combination of spring stiffness and preload).

Good pump performance for low compressibility liquids such as water or amine is obtained when valve closing events occur no more than 7 to 10 degrees after dead centre. In these cases, plunger velocity is quite low and therefore the pressure spike which occurs during the valve closing event is minimized. Other things being equal, vibration levels are reasonable because forces in the system are low.

Similarly, if valves are performing well, opening events are at a relatively low velocity and the plates does not contact the backguard at all.

Near theoretical capacity is achieved, because the plunger is pushing liquid during almost all of its stroke.

Cavitation does not occur when valves are good (given adequate suction pressure) because the valve opens before the chamber pressure goes below the vapour pressure of the fluid. (Cavitation happens when a liquid reaches vapour pressure, allowing bubbles to form and then implode when the pressure rebounds.

The Effects of Late Valve Events

Since the valve events were so late, we considered what effects they might have.

Valve events that are late occur after the plunger has had significant time to accelerate, and therefore the plunger velocity is greater.

When closing occurs, the sudden change from high to zero velocity of the fluid through the valves causes shocks similar to water hammer in the liquid system. The shocks are very sharp pressure peaks that produce shaking forces in the system piping, which in turn cause piping vibration.

For opening events, a similar pressure spike occurs due to the sudden change from zero to high velocity. In addition, the increased velocity causes the valve plate to move much faster and to have a greater chance of contacting the backguard. Such contact drastically increases wear and tear on the valve, and should not occur.

Another effect of late valve closing is reduced pump capacity due to backflow. On the suction

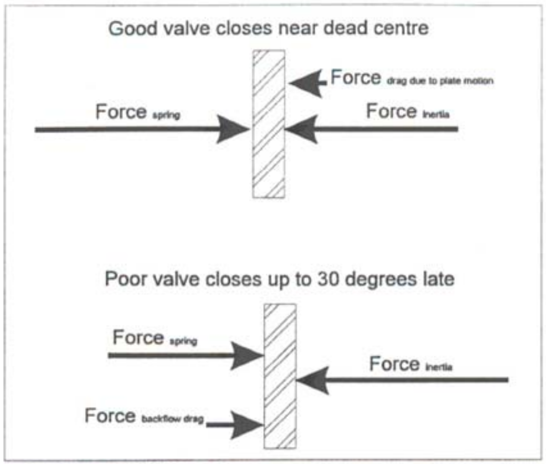

Figure 6 *Exaggerated comparison of forces acting to close the valve. If the contribution of the sprig to valve closure is less, that of the drag force must be greater.*

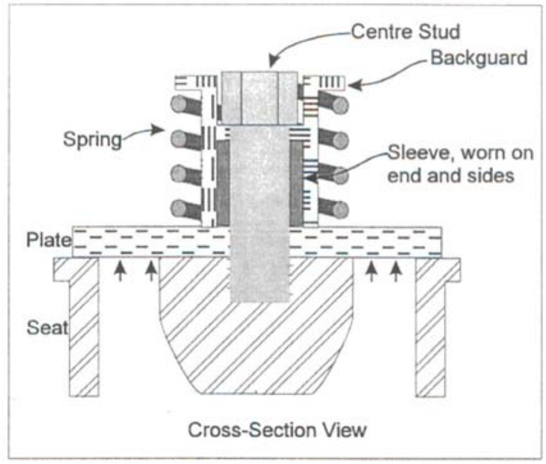

Figure 7 *The structure and wear pattern of the valve is an important piece of the puzzle.*

side, liquid gets pushed back into the suction line as long as the valve remains open after dead centre. On the discharge side, liquid gets drawn back into the pump chamber as long as the valve remains open after dead centre.

Suction cavitation may occur with the later discharge valve closure because the chamber pressure falls faster. The inertia of the suction valve tends to prevent it from opening until after vapour pressure is reached.

The Cause of Late Valve Closure

The next question, then, is “What could cause the valve event to be so late?”

Consider valve spring stiffness and assume the same free length and installed length (preload). Insufficient spring stiffness could cause the suction valve to close late because there is less contribution from the spring toward forcing the valve to close. To compensate for the lack of force from the spring, drag force from backflow is required to close the valve. See Figure 6. The discharge valve opening is late accordingly, since it must happen after suction closing.

6. THE PROOF

On Beta’s recommendation, and with the concurrence of the pump supplier, the owner installed valves with stiffer springs and plastic plates to correct the late valve closures. When the pump was run with the new valves, it was so quiet that operators questioned whether the pump was functioning!

Wear Patterns

When the original valves (with the relatively flexible springs) were removed, we examined them carefully to see if they confirmed our theory. A sketch of the valve cross-section is shown in Figure 7. Typically, the centre bolt was loose and the cylindrical sleeve in the centre of the valve showed excessive wear on both its ends and its sides. Some of the valve plates and backguards were worn. Cavitation damage was found on at least one plunger.

These patterns and modes of failure would suggest that the plate opened more violently than it should have and by travelling farther than it should have, hit the backguard. The repeated impacts could have loosened the nut, allowing the sleeve to move radially. That movement could have allowed the wear that was found on the ends and sides of the sleeve. (Valve flutter and squirming in relation to the guide can also cause sleeve wear, but that wear is usually on the sides of the sleeve, not the ends.)

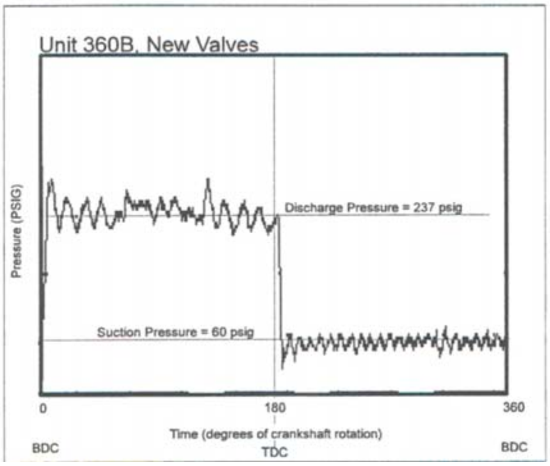

Figure 8 *The Pressure-Time curve for the new valves showed closure within 6 degrees of dead centre. Plunger velocity is still slow, and therefore impacts are low.*

Operations

After the new valves were installed, the following improvements were noted:

- the flow rate of the pump had increased to 106 US gal/min, the rated capacity.
- the suction and discharge valve closing angles were about 6 degrees after dead centre. See Figure 8.
- high frequency vibration had decreased. The spectrum showed very little high frequency energy. See Figure 9. The time domain plot showed the vibration to be 0.3 in/s positive and negative, and the large (over 1 in/s) spikes were no longer occurring. See Figure 10.

The lowest cylinder pressure during suction was 15 psig; therefore, cavitation will not be a problem, since the vapour pressure of amine is well below that level.

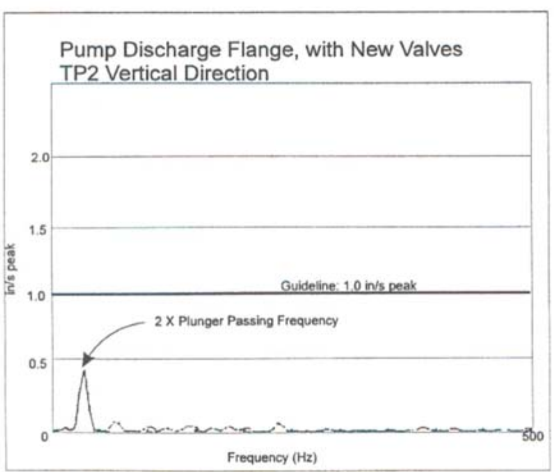

Figure 9 *The spectrum related to Figure 9 has little energy at higher harmonics.*

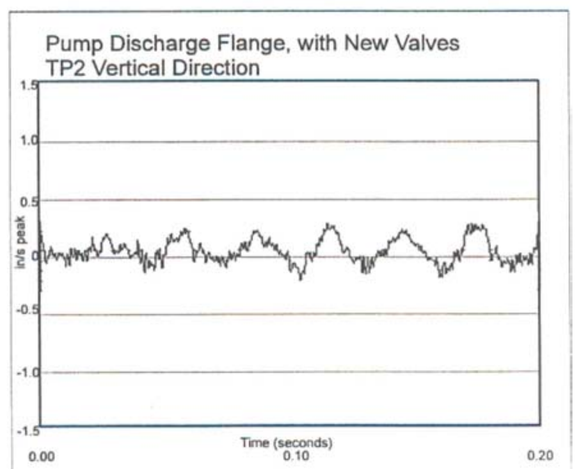

Figure 10 *Vibration taken after the valves were replaced showed acceptable levels.*

7. CONCLUSIONS

1. Something as apparently minor as valve selection, which is usually left to the supplier (who may not have the best information) may cause very expensive failures. In this case, the directly resulting problems were down-time and loss of amine due to breakages, and reduced capacity due to backflow.

Pulsation and vibration control on plunger pumps requires proper valve as well as proper design of pulsation controls (such as bladder desurgers or hard element filters with choke tubes).

2. The source of the problems described in this paper is difficult to detect without specialized analysis. Some serious problems are simply not apparent to human perception.

Conventional spectrum analysis is not enough to detect vibration problems in all cases. In this case, for an example, an overall true peak detectors or a look at the time domain pattern was required.

Moreover, vibration analysis alone would not have been sufficient to diagnose the cause. Time-based pressure readings were also required.

Appropriate measurements and appropriate acceptance guidelines must be used. Oversimplification can lead to a false sense of security.

3. Trial and error problem solving can be much more expensive than state-of-the-art design methods such as computer modelling of system dynamics, since there is a strong possibility that the "trial" may not work or that it may be an incomplete solution.

8. FURTHER RESEARCH

Further work is required on operating deflected shapes using time domain data, to clarify the limits of spectrum analysis.

9. REFERENCES

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10. THE AUTHORS

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