BETA
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VERTICAL FORCES CAUSE VIBRATION IN A RECIPROCATING COMPRESSOR

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ABSTRACT

It is well known that forces that cause vibrations in reciprocating compressors arise from a number of sources. These include unbalanced reciprocating forces and moments, piping system pulsations, and cylinder stretch. Reliable and efficient reciprocating compressor installations will result from including the effects of all forces in the design calculations.

Evidence from field trouble-shooting indicates that force mechanisms operating in the vertical direction in horizontal cylinder assemblies can cause significant vibration. Therefore, a design that ignores these potential sources of vibration problems (as is common) may be inadequate.

One such vertical force is caused by pressure pulsations acting on the suction and discharge cylinder nozzle areas and bottles. While the nozzle areas are generally equal, the differing pulsation amplitudes and phases result in net vertical forces on the cylinder. This paper presents a proposal for enhancing the API 618 guideline for pulsation control in reciprocating compressors. Currently, the compressor side guideline is higher than the line side guideline and disregards frequency. The proposed guideline incorporates frequency and adds a limit based on vertical shaking forces that could be generated in a cylinder.

A second vertical force in a horizontal cylinder assembly arises because the connecting rod motion is not purely horizontal. The slider-crank mechanism generates a time-varying force perpendicular to the motion of the piston through the crosshead guide. Which varies with the crankshaft angle, the l/r (connecting rod to crankshaft pin radius) ratio, the gas forces, and the reciprocating inertia. These vertical crosshead forces are sometimes contained by the foundation, but in some applications the stiffness under the crosshead guide is too low, and high vibrations result. In many cases, a rigid body analysis of these forces shows that the vector sum of the forces from all of the crossheads is zero. Such an analysis can be misleading.
This paper describes the origin of these forces, discusses their practical implications, and presents examples.

INTRODUCTION

Recent field trouble-shooting experiences have shown severe vertical vibration problems in certain reciprocating compressor installations. The severity of these vibration problems was made worse by the presence of mechanical resonance.

In the first case, lack of attention by the designers to the potential for large vertical shaking forces between cylinders and pulsation dampener bottles aggravated the problem.

In the second case, vertical forces acting through the crosshead guides appear to have contributed to high vibrations.

PRESSURE PULSATION GUIDELINES

The shaded areas on each graph (Fig. 2 – 7) indicate acceptable pulsations based on the American Petroleum Institute standard (API 618).

Compressor Side

The time domain curves show the compressor side pulsation limit (CPL), which shall be the lesser of

- 7% of average absolute line pressure, OR
- 3R% of average absolute line pressure, where R is stage compression ratio.

The purpose of the CPL is to avoid pressure-volume curve distortion. It does not address concerns about vibration or compressor valve life.

Lineside

The frequency domain curves show the API lineside design pulsation limit (DPL):

$$\textbf{DPL} = 3\bullet\sqrt{\frac{P}{ID \bullet f}}$$

Where DPL = maximum allowable peak to peak level of pulsation at discrete frequencies,
P = average absolute line pressure,
ID = inside pipe diameter (inches),
f = pulsation frequency (Hertz).
The purpose of the DPL is to avoid high unbalanced forces due to pressure pulsations.

CASE 1 – SINGLE STAGE FOUR THROW COMPRESSOR

The Problem

Two four throw compressors had been experiencing failures of pistons and rods over some years. On a visit to the site in February, 1994, the following problems were found:

- cylinder supports were loose or broken
- discharge bottle wedge supports were broken
- joints had fretted, shown by red dust (iron oxide) present at most of the joints in the cylinder supports
- anchor bolts and grout under the crosshead guide had broken

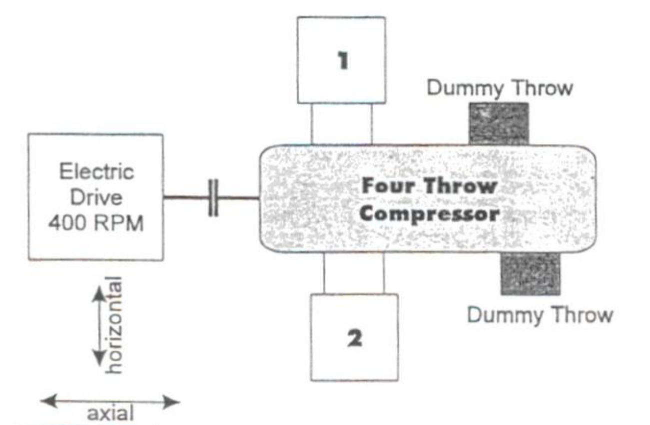

Figure 1 *Rod and piston failures over many years made this unit uneconomic to run.*

- high vibrations were observed on the concrete pedestal under the outboard cylinder support.

Measurements

Excessive rod loads were ruled out as a cause of the breakage, because rod loads during the tests were within or at the manufacturer's guidelines.

Pulsations were measured in the time domain at the suction and discharge valve caps (Fig. 2 & 3).

Suction side pulsations were within the compressor side guideline (CPL), while discharge side pulsations were about 120% of the guideline.

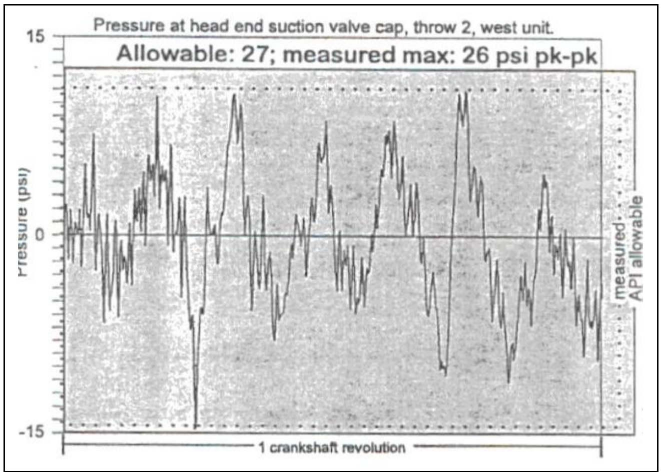

Figure 2 *Suction side overall pulsation was within guideline.*

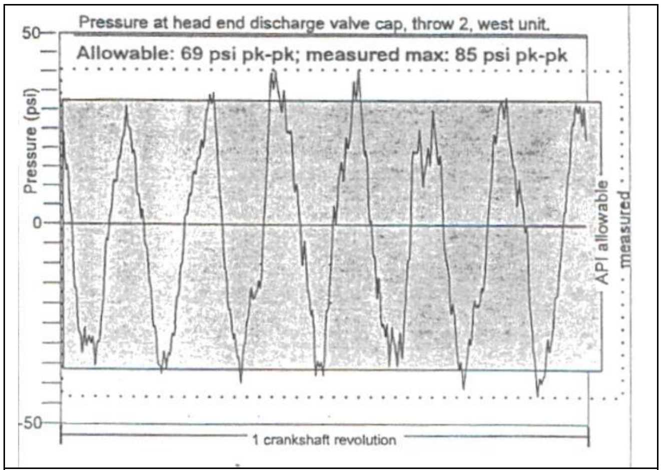

Figure 3 *Discharge side overall pulsation was about 120% of guideline.*

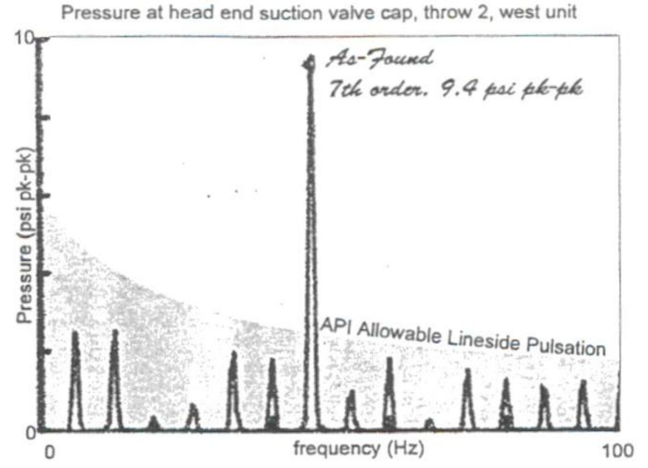

Figure 4 *As-found spectrum on the suction side shows high pulsation at 7X shaft speed.*

A spectrum of the as-found suction valve cap pulsation is shown in Fig. 4, and of the discharge valve cap pulsation in Fig.5.

The DPL is included as a reference in each plot, since there is currently no recognized pulsation limit based on discrete frequencies for the compressor side piping. In both cases, the DPL has been greatly exceeded at seven times crankshaft speed (7X).

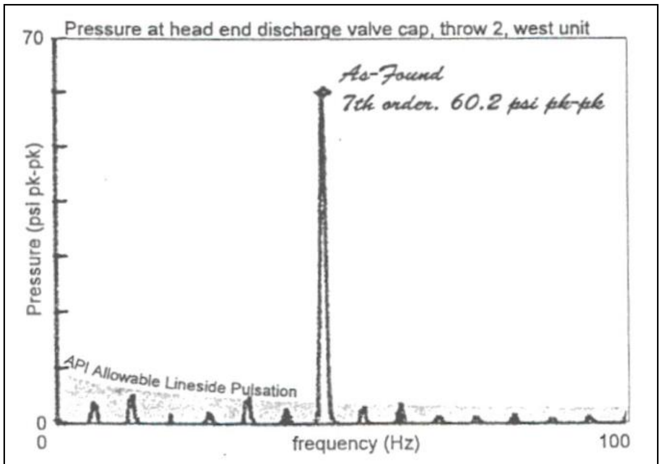

Figure 5 *As-found spectrum on the discharge side shows pulsation at 7X shaft speed is about 15 X DPL.*

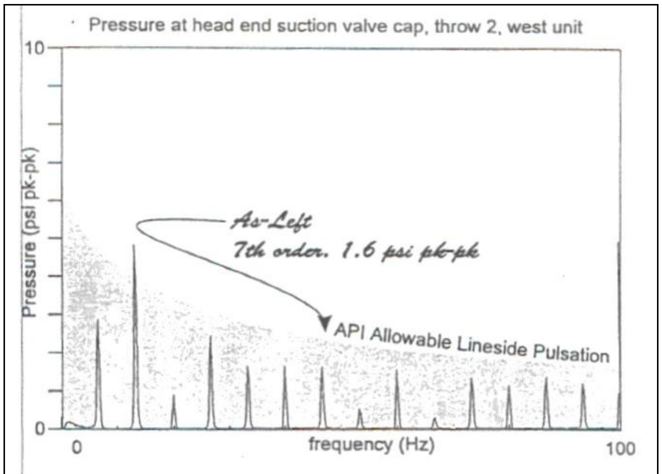

Figure 6 *The orifice plate significantly reduced pulsation levels.*

Modifications to the System

As-left readings (Fig. 6 & 7) were taken after orifice plates were installed in both the suction and discharge nozzles. As shown in Fig. 7, the as-left pulsation at the seventh order on the discharge side still exceeded the line side guideline by about 3X, even though it was only about 20% of the as-found reading.

Three years later, the compressor is still working well.

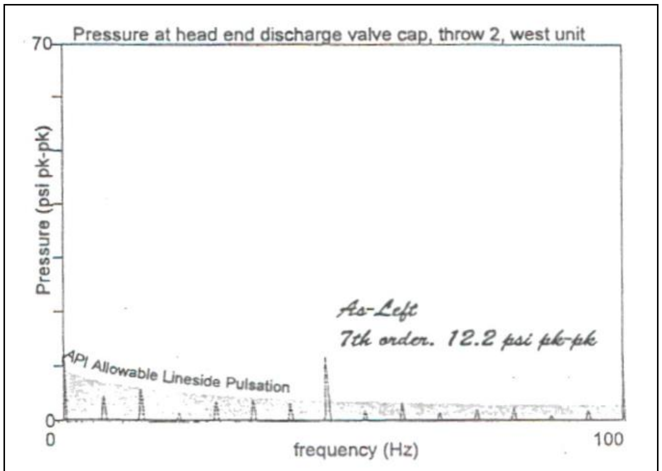

Figure 7 *The orifice reduced pulsation at 7X to about 20% of as-found; still three times DPL.*

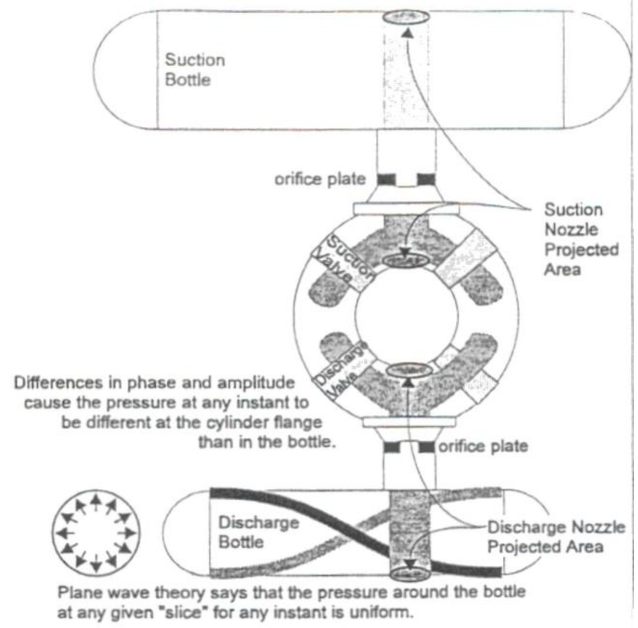

Figure 8 *Vertical forces due to pulsation can develop between the cylinder and the bottle.*

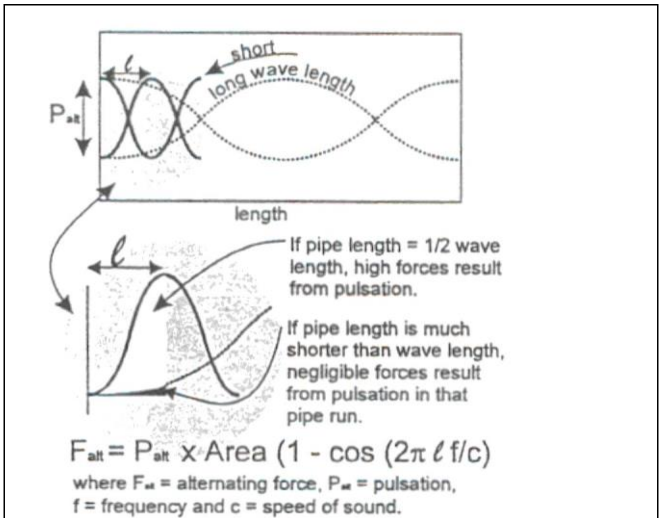

Figure 9 *The maximum unbalanced force occurs when pipe length equals half wave length.*

Vertical Forces Between Bottle and Cylinder

Figure 8 shows a cross-section of the suction and discharge bottles and cylinder on one side of the machine. The standing wave pattern of the pressure pulsation carries from the valves through the cylinder gas passages and the cylinder nozzle into the bottle, on each side of the cylinder. There is no direct connection between the two standing waves, because the suction and discharge valves are never open at the same time.

There is a difference between the amplitude and phase of the pulsation at different points on the standing wave.

Varying pressure acting on an area produces a varying force. In the vertical direction, the areas of concern are the projected areas of the inside diameters of the suction and discharge nozzles (Fig. 8) on the bottle and the cylinder passages.

A computer model was used to predict pulsations, and the model was verified with field measurements (Figs. 5 – 7). The worst vertical forces in the discharge nozzles were calculated to be about 7000 pounds peak to peak (pk – pk) and in the suction nozzles about 1100 pounds pk – pk) and in the suction nozzles about 1100 pounds pk – pk, both at the seventh order. It is clear that these forces are sufficient to cause the problems found.

Note that over the short length between the cylinder and the bottle, there is very little change in pressure amplitude at low frequencies (Fig. 9). The unbalanced force generated in short pipe runs by pressure pulsations at low orders of run speed (long wave lengths) will be much lower than at higher orders, all other things being equal.

Proposal for Additional API 618 Guideline

That proposed guideline prevents design conflicts that arise between the CPL and the DPL in the piping between the bottles and the cylinder.

In Beta's experience, the DPL is too restrictive to be applied to the piping between the bottles and the cylinders. It is generally acceptable to have low frequency (long wave length) cylinder side pressure pulsations that meet CPL but exceed DPL. If the DPL were applied, unnecessary and uneconomic design changes would be required in many cases.

Pressure pulsations ***by themselves*** do not shake the compressor. It is the forces generated because of pulsations that cause vibration problems. For this reason, we propose that compressor designs should limit potential transverse (vertical in horizontal cylinders, horizontal in vertical cylinders) unbalanced forces, not just pulsations. This force guideline is a compromise between ignoring the transverse forces on the compressor side of the pulsation bottle and using the more restrictive (and likely uneconomic) DPL guideline. It should be used with the CPL.

Unbalanced Force Guideline

Unbalanced forces are calculated according to the equation below, (where the pressures are vectors).

Force = (area 1 x pressure 1) – (area 2 x pressure 2)

On the basis of empirical studies, the following staged unbalanced force guidelines are proposed:

- Avoid coincidence between the vertical mechanical natural frequency of the cylinder bottle system and any peaks in the vertical unbalanced force spectra.

- Limit unbalanced forces in the cylinder nozzles at any discrete frequency as follows:

  1) the lesser of 500 lbs. Or 50 X nominal diameter (inches) maximum pk – pk on either the suction or the discharge side.

  2) if 1) is exceeded, the arithmetic sum of the suction and discharge forces should be less than 1000 lbs. Or 100 X nominal diameter pk pk.

  3) If 2) is exceeded, the vector sum of suction and discharge forces should be less than 1000 lbs or 100 X nominal diameter pk – pk.

- If 3) is exceeded, the forces must be reduced.

To illustrate, one can calculate from the equation in Fig. 9 the allowable pulsations for a particular system, as shown in Fig. 10. The CPL is too high to control vibration; the DPL is too stringent.

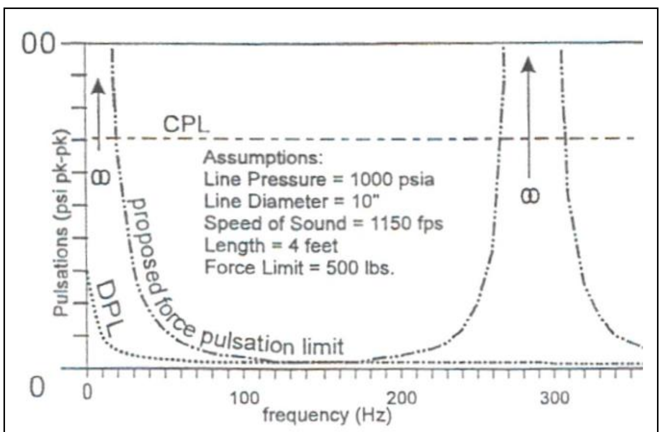

Figure 10 *The proposed frequency-based compressor side pulsation limit is a compromise.*

CASE 2 – SINGLE STAGE, SIX THROW COMPRESSORS

The three units (Fig. 11) are large variable speed horizontal compressors, mounted on skids bolted to 3 ft. thick concrete foundation blocks. The foundation blocks are embedded into the sandy soil. No piles were used.

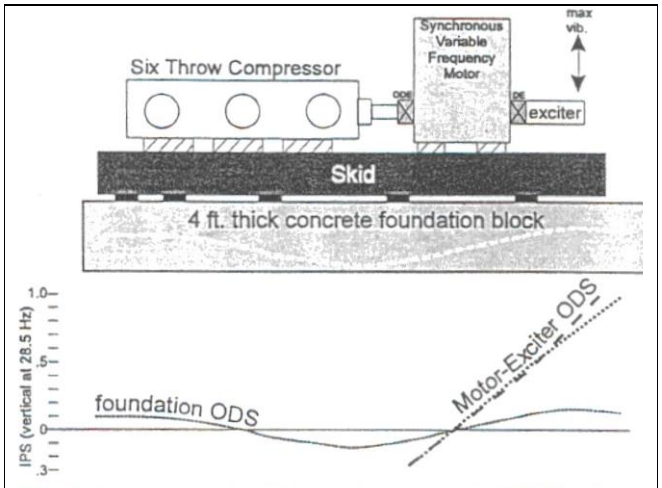

Figure 11 *Elevation view, with operating deflected shapes (ODS) at resonance.*

The Problem

The units are equipped with vibration sensors, which indicated excessive vibration. Supporting the exciter to the foundation had not helped.

Measurements

The highest vibration occurs near the top of the speed range at twice shaft speed in the vertical direction outboard on the exciter.

The foundation exhibits a resonance near the top of the runspeed range. The modal damping of the foundation was estimated from vibration measurements to be about 3% of critical damping. The foundation vertical vibrations had node points located at the mid points of the motor and the compressor.

Operating deflections (Fig. 11) showed that the exciter was amplifying the motion of the motor-compressor-foundation system due to geometry effects.

Vibrations were not significantly different between loaded and bypass conditions. Therefore, pulsations were ruled out as a cause of the problem.

The shaft alignment of the unit was checked and found to be acceptable. A small improvement in alignment had no effect on vibration levels.

Modelling of the System

The motor-compressor-skid system had been modelled in the design stage. The foundations had been assumed to be rigid, an assumption which proved to be incorrect.

A new finite element model of the system was created following start-up, and after the problem had been identified. Using the model to test the effect of changes showed that the skid-foundation resonance is very hard to change without major modifications.

Modelling proved to be very valuable in the sense that it indicated that doing a major change that made intuitive sense would not have worked. The skid has a weak shear connection to the concrete. It seemed obvious that a continuous grout connection between skid and concrete would stiffen the system, raise the natural frequency, and solve the problem.

The model quickly showed that the stiffening effect of the grout would be counteracted by moving the mass of the motor and compressor farther away from the neutral axis of motion. The rotational inertia of the motor and compressor would be made more influential on the mode, lowering the natural frequency and counteracting the stiffening effect.

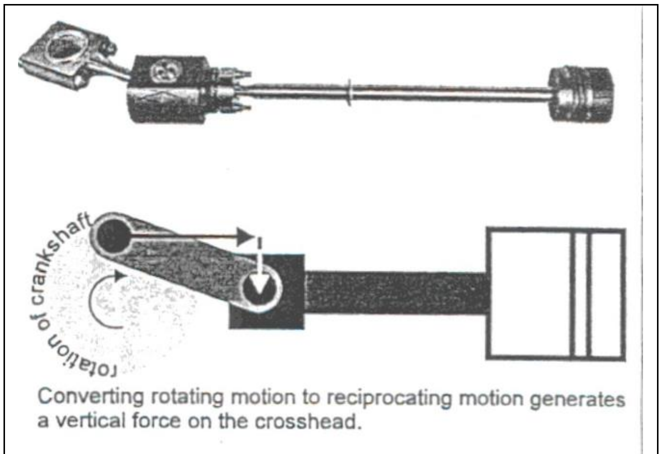

Figure 12 *Slider-crank mechanism generates vertical force.*

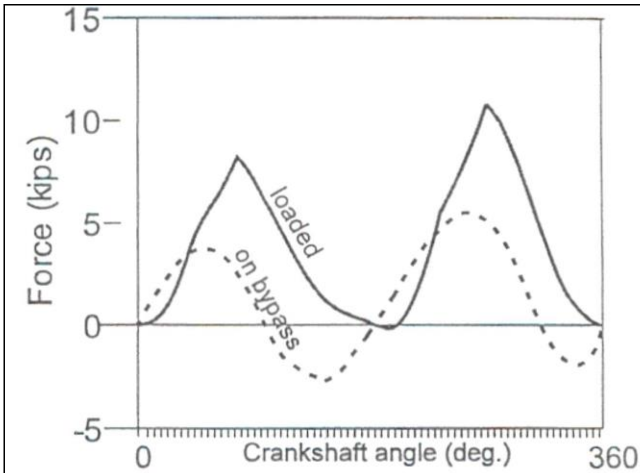

Figure 13 *Substantial vertical forces are generated by the motion of the crosshead*

Analysis

It was unclear what force could be present in the system at 2X shaft speed that could cause such vibrations, even with the foundation resonance since it was reasonably well damped.

Figure 12 illustrates the source of vertical force at the crosshead. Figure 13 shows the vertical force versus crank angle for this compressor at the cylinder 1 crosshead. These vertical forces were considered and initially rejected as a source of the problem, since they are all equal in magnitude and vectorially cancel due to the phasing of the 6-throw crankshaft (120 degrees between throws).

Figure 14 shows how the crosshead forces can cause the compressor to rock. Had there been a torsional mode about the centreline of the crankshaft with a node in the middle of the compressor, the couple shown in Fig. 14 would have caused high vibration.

Upon reflection, it was realized that the offset between cylinders (Fig. 15) causes the forces in the opposed pairs (e.g. throws 1 & 2 or 5 & 6) to produce another couple, this one about an axis parallel to piston motion. If the foundation behaved as a rigid body, the three couples from the three pairs of throws would cancel.

In this case, the out-of-phase motion of the foundation at twice crankshaft speed allows the couples to excite the mode.

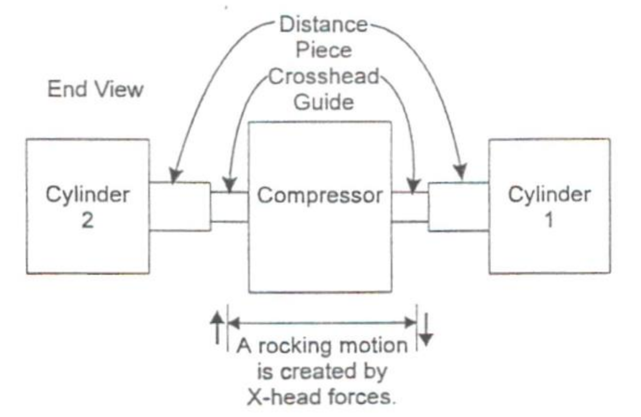

Figure 14 *Crosshead forces produce a couple about the centreline of the crankshaft.*

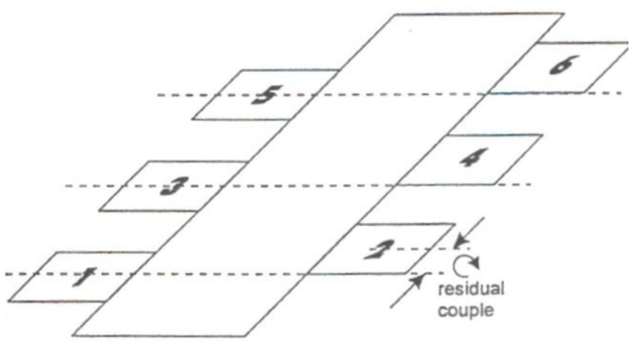

Figure 15 *Side view of compressor – offset cylinders generate moments.*

Modification

The amplitude of the vibrations was reduced by 30% by stiffening the connection of the motor end bells to the skid, using jacking bolts. This modification did not change the frequency of the peak response. It appears that no complete solution is possible except by changing the foundation natural frequency.

Conclusions – Case 2

The forces and moments in a reciprocating compressor do not necessarily cancel. In particular, when the unit is not mounted on a rigid body, modal excitation may result. Care must be taken when designing compressor foundations to avoid natural frequencies at one and two times crankshaft speed.

It is also clear that vertical forces at crossheads should be considered when trouble-shooting vibration problems in the field.

SUMMARY

Vertical forces in compressor cylinder-to-bottle nozzles need to be considered in the design stage. Since there is no suitable design guideline currently available for these forces, one is proposed.

Slider crank mechanisms produce forces transverse to the crosshead guide or cylinder. These forces should be considered in the design of reciprocating machinery systems.

FURTHER WORK

Consideration should be given to enhancing the DPL guideline to include unbalanced forces, or even to replacing the DPL guideline with an unbalanced force guideline. In many instances, the DPL by itself is too strict or too lenient, depending on how the pulsations couple with the pipe lengths.

REFERENCES

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THE AUTHORS

The two authors are employed by Beta Machinery Analysis Ltd. Brian Howes obtained his M.Sc. in Mechanical Engineering in 1972. His experience includes trouble-shooting on a wide variety of high end equipment, research and development in pulsation and vibration of piping systems, and analysis of mechanical and structural systems so as to ensure acceptable static strength and dynamic response. He is Beta's Chief Engineer.

Valerie Zacharias obtained her M.A. in Communications in 1978, and did a postgraduate year in Computer Science in 1979. Her experience includes acoustical modelling of reciprocating compressors, and ten years in the predictive maintenance business, primarily in customer service, training, and communications. Mrs. Zacharias handles Customer Service.