BETA
Machinery Analysis

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Troubleshooting Machinery Problems

Machinery troubleshooting assignments come in an incredible variety of shapes and sizes. Problems may involve lack of throughput, inefficiency, chronic failures, poor product quality, failure to meet guarantees, excessive vibration, noise, short component life, excessive energy consumption - just about anything. Add to this the broad range of machinery being operated in industry: gas turbines, steam turbines, engines, motors, reciprocating and centrifugal compressors, paper machines, pulp refiners, fans, blowers, all kinds of pumps, ….; the list goes on and on.

Clearly, competence in machinery troubleshooting requires a broad range of tools, expertise and experience. Various recorders, sensors and analyzers are needed to acquire operating data like pressures, temperatures, speeds, flow and vibration. Knowledge of technical disciplines including thermodynamics, fluid mechanics and solid mechanics is essential to interpreting the data. Of course, experience with similar problems is a great help. But more critical to successfully resolving the next problem is a disciplined, methodical approach.

BMA field engineers generally follow a procedure that includes:

- interviewing customer personnel to collect input about the problem; much of the information supplied is based on impressions of the situation; it is treated as evidence that must be corroborated and expanded. This stage includes familiarization with the machine and process.
- reviewing any records relevant to the problem, such as maintenance history and operating logs.
- planning and executing a test program, the objective being to observe and record the technical data associated with the problem; i.e pressures, temperatures, flow rates, vibration, etc., quite often over a range of operating conditions which explore both the problem operating area and other conditions where operation is normal. What can actually be accomplished is often limited by operating constraints, safety, and/or time. Ongoing evaluation of results during testing often leads to adjustments in the planned testing.
- analysing of test results and all other evidence; evaluation against possible root causes usually results in a high confidence diagnosis; sometimes further testing is required to validate diagnosis.
- recommending problem solution; varying from an immediate fix in the field to a major rework requiring careful analysis and planning.

Over the years, Beta has been involved in many hundreds of troubleshooting assignments, covering the types of problems and machinery mentioned above. A few have bordered on the bizarre, such as the laundry facility where the floor occasionally vibrated at an amplitude that sent customers running out the door! Some have defied satisfactory, practical solutions; recalling a certain coal pulverizer installation that regularly tore up gearboxes, piping, etc. This situation became the domain of lawyers rather than engineers. By far the majority have been what we regard as routine and have been resolved efficiently and expeditiously.

A few observations and suggestions from all this experience:

- there is frequently a lack of consensus about the circumstances surrounding the problem; try to capture all the information about what was seen and heard; the way the machine was operating, etc.
- rarely is there enough recorded data; try to capture and preserve data on operating conditions; i.e. pressures, temperatures, flows, speeds, settings and so on.
- if there is a failure involved, preserve the broken components.
- avoid the temptation to jump to conclusions and try ad hoc solutions. A quick fix (ie. bearings are replaced) may not solve the problem and might incur unnecessary expense and lost time. Generally, bringing in specialists sooner rather than later is a more cost effective solution.

INSIDE:

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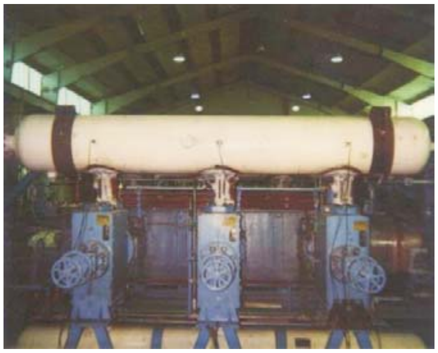
Case Study: Unruly Suction Bottles

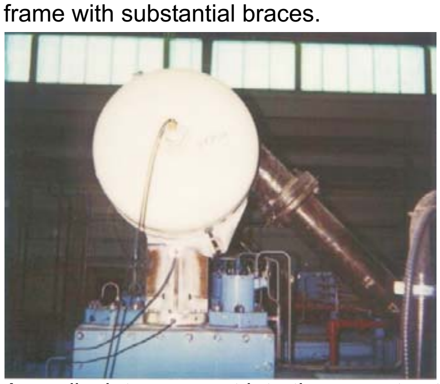


Machine Description:

- Eight, six throw separable compressors
- Single stage
- Natural gas storage service
- 2485 HP engine drive
- 600 - 900 RPM speed range
- Multiple operating configurations with wide range of conditions and automatic load step control
- Suction bottles have 4" XXH nozzles, gussets and repad

Vibration Problems and Nozzle Failures

At the start up in 1996 the suction bottles exhibited high vibration and developed cracks in the gusseted suction nozzles. Responding to these problems, the bottles were braced back to the compressor frame with substantial braces.



As well, slots were cut into the gussets at the top and bottom.

Vibration was reduced, but cracks developed near the root of the slots in the gusset to nozzle welds.

At this time Beta Machinery was retained to analyze the problem and recommend a solution.

Field Analysis

A field analysis was conducted with the goal of determining the mechanical natural frequency of the suction bottle and the operating deflected shape of the suction bottle and cylinders.

The key findings were that the existing system with braces and slots in the gussets was in trouble because:

1. The horizontal mechanical natural frequency of the suction bottle was 47 Hz. This is mode sensitive to the in-phase cylinder forces at 3x compressor speed. (3x 900 rpm = 45 Hz.)
2. Motion of the suction bottle relative to the cylinder was in the range of 4.6 mils pk, and was causing distinct kinks (high stress) at the locations of the slots cut into the gussets.

Finite Element Analysis

A finite element analysis was conducted to assess the situation. A model was constructed and verified with field MNF and motion data. The analysis determined that the displacement of the bottle relative to the cylinder resulted in high cyclic stresses in the cylinder nozzles.

Although the finite element analysis calculations were very sensitive to the location of the cuts, it was obvious the slotted gusset geometry has a high fatigue crack potential. The lowest stress range calculated was 4150 psi pk, while the highest was 9700 psi pk. See Figure 1 for a schematic of the relative motion of the bottle relative to the cylinders and Figure 2 for the predicted stress in the lower gusset slot. Stress values on the plot are in psi pk.

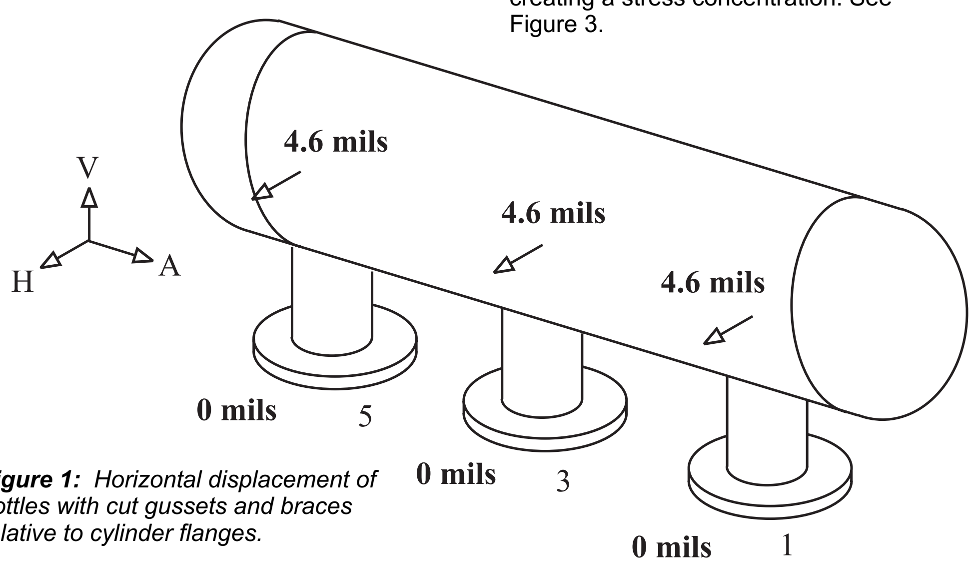


Figure 1: Horizontal displacement of bottles with cut gussets and braces relative to cylinder flanges.

A guideline we typically use for dynamic stresses is:

- 1500 psi pk is acceptable;
- between 1500 psi pk and 3000 psi pk is a likely problem;
- Over 3000 psi pk is unacceptable.

At 900 rpm, a compressor will undergo well over 100 million cycles in less than 80 days of run time. The fact that cracks took months or even years to form confirms that the excessive cyclic stresses were not occurring on every cycle, but that specific load steps and operating conditions did accumulate fatigue damage.

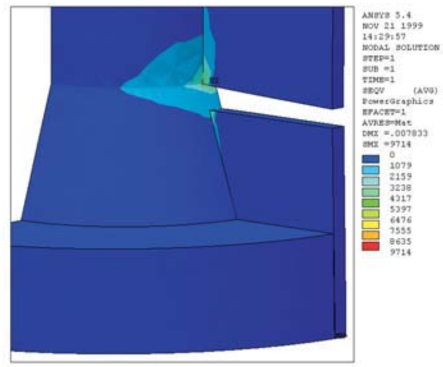


Figure 2: Stresses at lower gusset cut with bottle displaced 4.6 mils (horizontal) relative to flange.

Intermediate conclusion #1 and recommendation:

Modification of nozzle gussets in a manner that creates stress concentrations is not desirable.

If installed gussets are found to be a problem, then deactivate them by cutting a curved scallop from the gussets. This will result in a smooth variation of section and will reduce the stiffness without creating a stress concentration. See Figure 3.

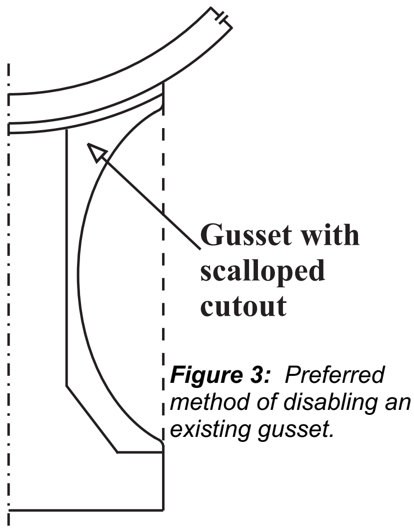


Figure 3: Preferred method of disabling an existing gusset.

The finite element analysis also determined that the horizontal mechanical natural frequency of the original bottle design, (intact gussets, no braces) was 43 Hz. This design proved to be inappropriate for this system because:

- MNF is in the 3rd order of run speed range, 3x 900 rpm = 45 Hz.
- This mode is very sensitive to in phase cylinder motion at 3x.
- Third order cylinder gas forces are greater than 10% of rated rod load (55,000 lbs) for some operating conditions, and in phase.

Intermediate conclusion #2 and recommendation:

Three nozzle suction bottle horizontal mechanical natural frequencies should not be at or near 3x run speed on six throw compressors.

Design stage analysis of multi-nozzle bottles should include both modal and forced response FEA and consider:

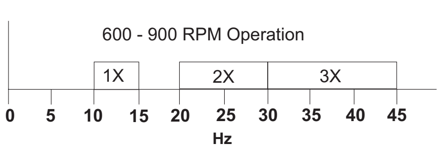
- cylinder motion excitation @ 1x, 2x, 3x, and 4x run speed as a minimum
- full range of configurations, speed, operating conditions and load steps
- potential for restrictions in operation, speed range or load steps

Designing a Solution:

With the history of high vibration and results from the finite element analysis there was strong motivation to change the nozzles.

But how???

First we considered the first three harmonics of the run speed range. Where could a horizontal bottle MNF hide?



There is a 5 Hz band between 1x and 2x that was potentially available.

Try it!

The finite element analysis evaluated the mechanical response of the suction bottle with no gussets and no braces.

Surprise!

The predicted horizontal MNF was 22 Hz, not bad for a simple change. The bottle twisting mode was predicted to be 50 Hz, above 3x run speed at 900 RPM and less sensitive to 3x forces. The axial bottle mode was predicted to be above the top of 2x run speed.

The forced response analysis results were promising. Table 1 compares a summary of the predicted vibration response of the various bottle configurations. The bare nozzle without bottle braces is the clear winner.

Description	Horizontal Mode MNF, Hz	Response to Cyl. Motion*
Gussets Cut with Braces (as found)	50	Medium @ 3x
Gussets Cut without Braces	38	**Large** @ 3x
Gussets Not Cut without Braces	43	**Large** @ 3x
No Gussets without Braces	22	Small @ 1x Medium @ 2x Small @ 3x

**Response is based on calculated ratio of bottle displacement to cylinder displacement:*

<u>Response</u>	*<u>bottle displ./cyl. displ.</u>*
Small	*<1.5*
Medium	*>1.5 and <2.5*
Large	*>2.5*

***Table 1:** Mechanical Natural Frequencies and Response to Cylinder Motion*

Build it!

The results look a bit strange, with rather spindly nozzles compared to the bottle diameter. Especially when you consider that gusseted and braced bottles were a problem.

NOTE: The first two bottles were modified by mechanically removing the existing gussets. Subsequent bottles were modified by completely replacing the nozzles.

Run it!

The suction bottles on one unit were modified by grinding off the gussets. Then field vibration tests were conducted. The behavior was a distinct improvement over the previous designs.

Vibration at 900 RPM is excellent and stays that way down to 750 RPM where it becomes questionable. At 750 RPM, 4x cylinder motion excites the twisting mode at 50 Hz and results in a maximum bottle vibration of 3 mils pk relative to cylinder 1.

The finite element analysis was used to evaluate the stresses in the new nozzles caused by the displacements of the actual running vibration. Figure 4 depicts the maximum relative vibration displacement seen in the run test of the new nozzles. Figure 5 shows the maximum stresses (psi pk) estimated by our models for the maximum relative displacement.

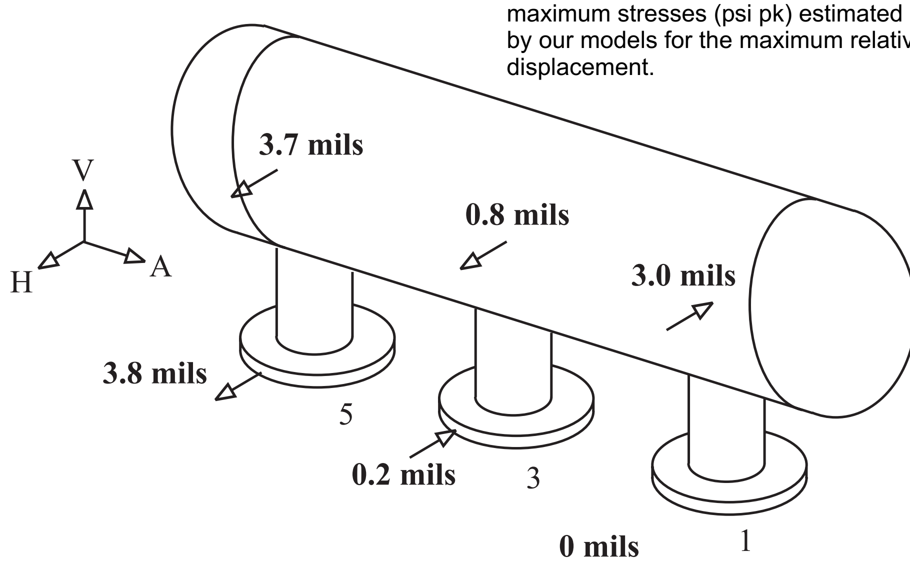


***Figure 4**: Combined bending and torsional motion of the bottles without gussets or braces relative to cylinder 1 at 750 rpm*

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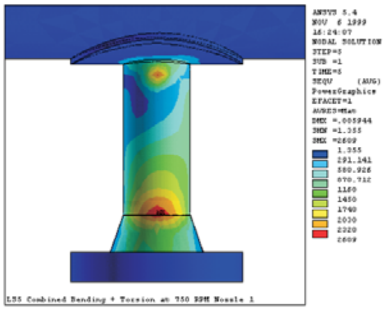

***Figure 5**: Stresses in nozzle without gussets due to bending (horizontal) and torsion at 750 rpm*

The stresses at 750 RPM were in the range of potential problems therefore it was agreed to limit the low end of the loaded operating speed to 800 RPM.

A crack developed in the cylinder 1 nozzle of the suction bottle shortly after modification. The location was at the base of a gusset that had been removed. Subsequently the bottles were modified by completely replacing the nozzles. Over two years later there have been no further problems on the eight units modified.

Final Conclusions

Specifically:

Although strange looking, the suction bottle design with the bare cylinder nozzles and no braces is appropriate for long term use in this multi-purpose compressor application. The solution is non-typical in an industry where it is commonly felt that "stiffer is better".

Summary:

- Controlling the dynamic behavior of multi-nozzle suction bottles is critical.
- Design efforts must include modal and forced response analysis and account for the complex interaction of the mechanical behavior of the pulsation bottles and cylinders and the cylinder motion that results from normal cylinder forces.
- Avoid stress risers in cylinder nozzle geometry. Gussets should be avoided.
- For six throw compressors the mechanical guideline that the MNF of components near a reciprocating compressor should be > 2.4x run speed needs to be augmented with also avoiding pulsation bottle horizontal modes @ 3x run speed.
- It is possible to have successful suction bottle designs that include locating a primary MNF between 1x and 2x run speed.

Did you ever wonder what Beta's engineers do on a weekend off?

Bryan Long was part of a team running this hot rod at Bonneville Speedweek. It ran 192 mph!!

News & Notes

Beta was there.....

- John Harvey and Frank Fifer presented "Improving the Reliability & Performance of Reciprocating Compressors" for the GMRC Workshop in Houston September 10 and 11, 2002.

- The Canadian International Petroleum Conference was held in Calgary June 11 - 13, 2002. A paper entitled "Improving Bottom Line Results Through Asset Management of Production Machinery" was presented by Bryan Long.

Beta will be there.....

- Beta will be at the annual Gas Machinery Conference planned for Nashville in October. Please drop by our booth as we celebrate Beta's 35th Anniversary. Please contact Marsha Short at mshort@southerngas.org for registration and program information.

- The 4th International Pipeline Conference will be held in Calgary from September 29 - October 3, 2002. Brian Howes and Shelley Greenfield will be presenting "Guidelines in Pulsation Studies for Reciprocating Compressors".

- Beta Engineers have presentations scheduled for the CMVA Annual Meeting and Trade Show to be held in Quebec City in October. Please contact Val Zacharias at cmvaed@shaw.ca for further information.

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