BETA
Machinery Analysis

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Avoiding Vibration Problems on High Speed Reciprocating Compressors

Many compressor packagers and owners have observed an increase in the number and scope of vibration problems on high speed reciprocating compressor installations. High speed reciprocating compressors are those with a maximum operating speed of 1200 to 1800 rpm. This article sheds some light on the technical issues regarding the design of these packages.

Basic Vibration Concepts

Simply stated, vibration is defined by:

Vibration = Dynamic Force/Dynamic Stiffness

The result of this relationship is that as the force increases for a given stiffness, the vibration will increase. Also, if the stiffness decreases for a given force, the vibration increases.

All components have a dynamic stiffness; that is the stiffness varies with frequency. Dynamic stiffness is a minimum at a mechanical natural frequency (MNF). When there is a force at a frequency equal to an MNF, very large vibration can result for a small force amplitude; this condition is called resonance.

The mechanical natural frequency of the component can be defined as:

MNF = $(\text{Stiffness/Mass})^{1/2}$

Therefore, to double the mechanical natural frequency, the stiffness would need to increase by a factor of four, or the mass would need to be reduced by a factor of four.

There are many different dynamic forces inherent to a reciprocating compressor. In general, the highest forces occur at frequencies that are the same speed as the compressor and at twice the compressor speed. These forces can be from mechanical unbalance, cylinder gas forces or pressure pulsations, and can be minimized through proper design but cannot be eliminated.

Pulling these three concepts together results in a design goal for a reciprocating compressor installation. The vibration on a compressor package can be minimized if mechanical natural frequencies are higher than twice the compressor speed. Meeting this design goal will ensure there is adequate dynamic stiffness to resist most dynamic compressor forces.

Application of these Concepts

The early reciprocating compressor installations were typically large, slow speed integral compressors, usually operating to 360 rpm maximum. Achieving the design goal of having all mechanical natural frequencies above two times 360 rpm or 12 Hz (2 x 360 rpm/60) can be easily done without much consideration of the dynamic characteristics. Typical system designs required to meet the static and operational design requirements were usually sufficient to meet this mechanical design goal.

The reciprocating compressor industry next evolved into using separable compressors with typical maximum operating speeds of 900 rpm. This increase in compressor speed significantly raised the design challenge. A speed change from 360 to 900 rpm (2.5 times) means the stiffness of a component must increase by more than six times to achieve the same mechanical natural frequency and vibration level (assuming the same mass and dynamic force). However, this goal is relatively easy to attain with implementation of incremental changes to skid construction, vessel design and pipe layout.

In recent years, larger compressors operating at higher speeds have become common. Beta has been involved in over 700 design projects with units greater than 1500 HP and operating at 1200 rpm or higher. The change from 900 to 1200 rpm means the system must have an increased stiffness of 1.8 times to achieve the same vibration assuming a fixed mass and dynamic force. It is possible to achieve this increase in stiffness but it impacts all phases of a compressor package from design through to operation and maintenance.

One of the assumptions stated above is that the dynamic force remains constant as the speed increases. This assumption is not accurate with respect to the compressor mechanical unbalanced forces and moments. These dynamic forces increase with the square of the compressor speed. Therefore, the stiffness must increase by more than 3 times for a compressor operating at 1200 rpm compared to a compressor operating at 900 rpm to achieve the same vibration. Figure 1 illustrates how the increase in speed requires an increasing amount of resources (design, material, installation) to achieve an acceptable design.

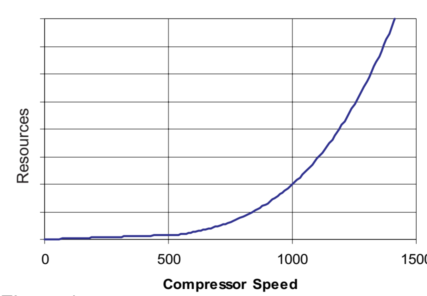

Figure 1

(Continued on page 4)

INSIDE:

Case Study: Vibration Diagnosis at a Distance

Abstract

Complex and powerful reciprocating compressors are shipped world wide. Design efforts to mitigate the potential for vibration problems with these machines are thorough, but cannot account for every possible circumstance. When the “As Built” machine differs from design a vibration problem may result. When the machine is a world away, solving a vibration problem is far from routine.

The response to a serious vibration problem at a remote location will often involve a lengthy shutdown while the problem is discussed. Lack of data, differences in time zones, languages, etc. all contribute to a stressful situation. Ultimately, someone with expertise in a broad range of competence will have to be sought out and deployed to the remote site with sophisticated equipment. This approach will be costly and time consuming, in many cases unnecessarily so.

This article relates a case where a major vibration problem was solved quickly, efficiently, and with a significant cost savings because the experts stayed home.

This troubleshooting effort made full use of the following resources:

- The design background; acoustical and mechanical models of the compressor package.
- A simple vibration data collector.
- A mechanic committed to travel to site and willing to collect and transmit vibration and operational data as directed by email.
- Graphical data presentation techniques. “Profiling” was used to quickly review, share and analyze vibration data.
- Computer modelling tools to explore the problem and develop recommendations.

During the analysis:

- Vibration data was successfully transmitted from the remote site and served to describe the problem.
- Field data was used to calibrate computer models.
- Mechanical recommendations to resolve the problem were developed with the aid of the updated computer model.
- Vibration data transmitted from the remote site verified the successful implementation of the modifications.

The problem was solved remotely, quickly and at a cost savings.

Introduction: Two 3000 HP natural gas engine driven three stage compressors were packaged in Canada, and shipped to China.

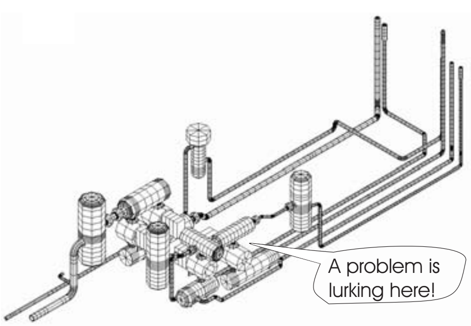

Figure 1: Computer model of complete compressor piping system

The design of these machines had been scrutinized with appropriate API 618 acoustical and mechanical studies. The study and testing verified that the machines were fit and ready for their intended service. However, due to a combination of circumstances, a significant problem was shipped to China with the compressors.

Word of a problem: During the plant start up after 30 hours of run time a cylinder nozzle on Unit 100 cracks at the gusset to bottle weld. Both Units 100 and 200 are shut down. Owner contacts packager who in turn contacts us.

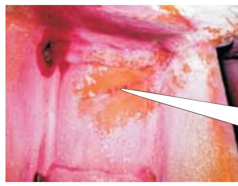

Figure 2: Unit 100 3rd stage suction nozzle crack

Initial response: Experience, backed by a review of acoustical and mechanical models quickly determined that the nozzle failure was most likely due to a mechanical resonance of the suction bottle, in response to cylinder motion due to gas loads.

Potential fixes were available on the basis of modelling alone. However, the high profile nature of this problem warranted the effort to: a) secure field data to absolutely verify nature of the problem and; b) verify that repairs and modifications were effective and would cause no further delays.

The option of sending an expert troubleshooter was considered but rejected after review of availability, cost, and timing, but ultimately because a remote diagnosis was a viable alternative.

A plan: The plan for a remote vibration analysis was simple. It had to be. The two identical machines, Unit 100, with the failed nozzle on the third stage suction bottle and Unit 200, intact, were both currently shut down. The information required to understand the problem conclusively resided in the vibration behavior of Unit 200. Discussions with the end user and packager resulted in agreement on the following response:

1. Establish and lay out the necessary vibration data testpoints and prepare vibration equipment for travel.
2. Train the packager’s field technician (already preparing for departure) in use of equipment and data file transmission.
3. Test Unit 200 in the “As Found” state, measuring vibration at key points during a brief run test.
4. Transmit the vibration data back by email for immediate analysis.
5. Compare field findings to characteristics predicted in mechanical models.
6. Evaluate and present potential modifications within capacity of shop services accessible from the remote site in China.
7. Modify one unit.
8. Run the modified unit, collect vibration data again and send data back for analysis.

We anticipated a quick and decisive convergence on the best solution.

The Data-Trap and a rapid training program: A key element in the plan was the ability to get useful vibration data back from the field without a vibration specialist.

The vibration equipment would have to be simple to use and rugged. We selected a single channel vibration data collector called the “Data-Trap” and “Profiling” software.

The tri-axial vibration testpoints necessary to describe the problem were identified on a sketch of the third stage suction system. The test points were then created in the Profiling software to be loaded onto the mechanic’s portable computer.

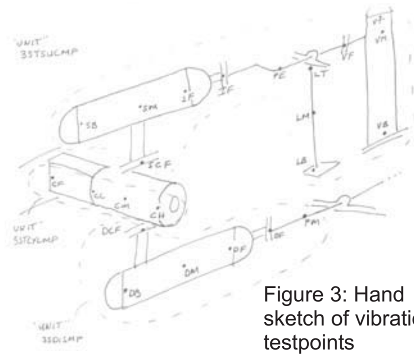

Figure 3: Hand sketch of vibration testpoints

After a brief training period, (less than four hours) the field mechanic demonstrated the ability to:

- Load the Data-Trap with the pre-planned test points.
- Identify all the testpoints and test orientations on the hand sketch of 3rd stage suction systems.
- Collect data.
- Dump the vibration data from the Data-Trap to his computer.
- Email the data file.

The Analysis

Cost considerations: The total engineering and equipment cost of the remote vibration troubleshooting effort was $22,700Cdn. This is a 70% cost savings compared to the estimated cost of equivalent field troubleshooting without immediate access to the mechanical modelling resources important to the success of this project.

Conclusion: It is possible to quickly and efficiently troubleshoot vibration problems at remote locations at much reduced cost. The main element for success is using a team approach to:

- Identify the problem and the concise data requirements.
- Provide simple tools to allow in-field resources to collect and relay reliable data.
- Use intuitive graphical data presentation tools to turn the data into information that can be shared rapidly with the field and packager.
- Evaluate vibration behavior with computer modelling tools to quickly test modifications to determine the best solutions.
- Implement recommendations and at each step immediately share concise machinery behavior with all members of the team to make quick and informed decisions.

The Remote Vibration Analysis at a Glance

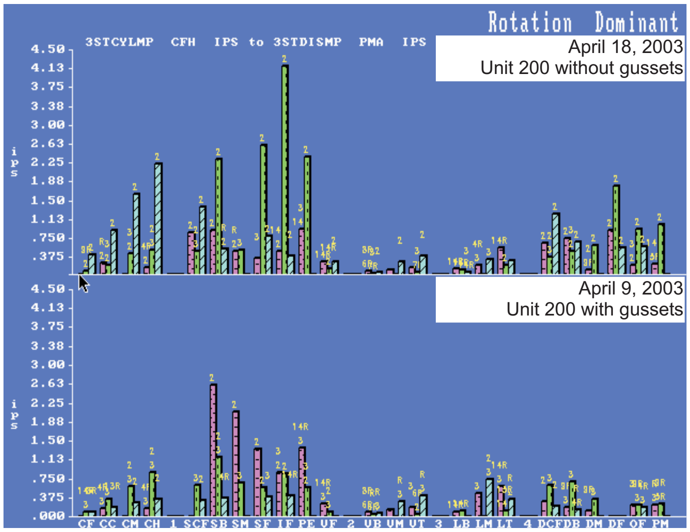

Figure 4: Unit 200, rotation dominant vibration profiles with and without gussets on cylinder nozzle.

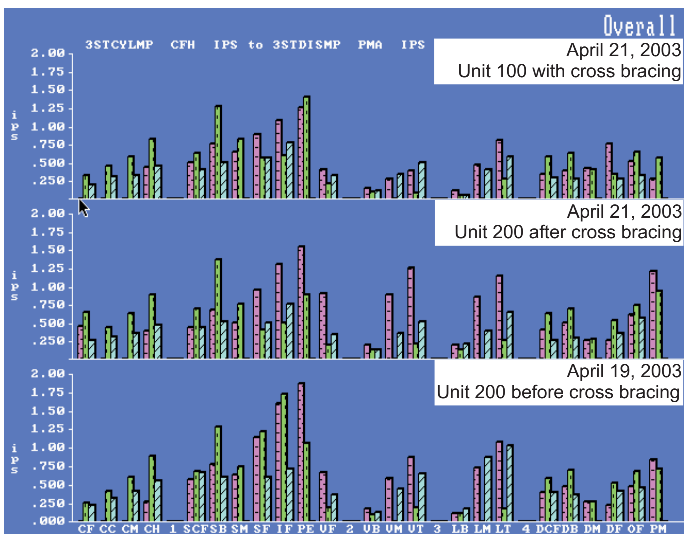

Figure 7: Overall vibration profiles on Unit 200 before and after stiffening inlet flange vertical brace and Unit 100 with similar modifications implemented. Note reduced scale.

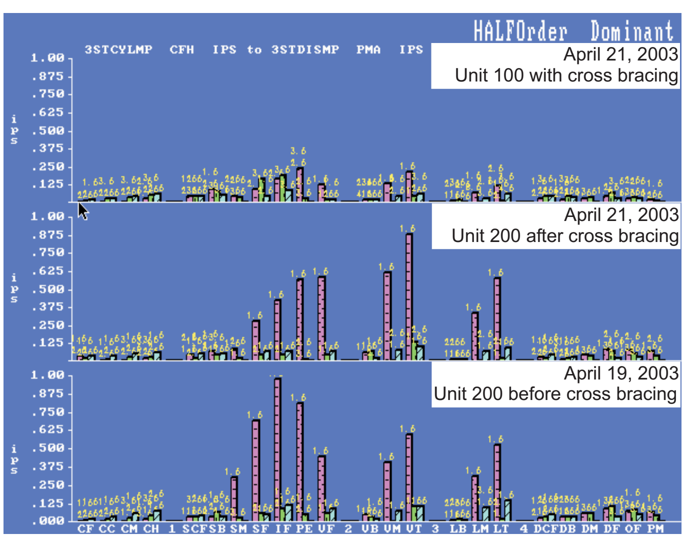

Figure 8: Half order dominant vibration profiles.

Note: The three profiles shown at left group tri-axial vibration data above each testpoint as vertical bars; red bars indicate horizontal vibration (parallel to piston motion), green bars - vertical vibration, blue bars - axial vibration (parallel to compressor crankshaft).

The data comes back:

April 9th - As Found look at Unit 200 3rd stage cylinder, suction bottle and piping shows high overall vibration in the horizontal direction clearly visible on the suction bottle at test points SB, SM, SFE, IF and PE.

The rotation dominant profiles in Figure 4 lower plot identify the major component of the horizontal bottle vibration at 2x run speed. The 2x amplitude pattern across testpoints SB to PE shows a combination horizontal - twisting mode of the suction bottle. The back end of the suction bottle (SB) vibrates most, diminishing to the piping testpoint (PE) just upstream of the inlet flange, where 3x vibration dominates.

The computer model verifies that high stresses will be seen at the gusset to bottle connection. The first recommended modification is to remove the cylinder nozzle gussets.

April 18th - Unit 200 is examined with cylinder nozzle gussets removed. Horizontal vibration is now acceptable but a vertical tipping mode of the suction bottle is excited along with an increase in axial cylinder vibration. Figure 4 upper plot shows the dominant vibration is again at 2x run speed. The pattern is consistent with a tipping of the suction bottle in the axial direction.

Note the highest vertical vibration is at the suction bottle inlet flange (IF), less at the front seam (SF) of the bottle, much less at the middle of the bottle (SM) and high again at the back seam (SB) of the suction bottle.

The appearance of the axial tipping mode is not unexpected and the computer model is used to quickly evaluate various bracing configurations. The resulting recommendations are a bottle brace, see Figure 5 and a vertical support at the inlet flange of the third stage suction bottle, see Figure 6.

Figure 5: Bottle brace as implemented

Figure 6: Inlet flange Vertical support with cross braces

April 19th - With added bottle brace, and vertical support on inlet flange, Unit 200's vibration results show distinct improvement but a review of the vibration results, Figure 7 lower plot, and the computer model show that cross bracing the inlet flange support will reduce horizontal vibration of the bottle, hence stress on the nozzle.

April 21st - Both Unit 100 and 200 are examined with cylinder nozzle gussets removed, the bottle brace, and the cross braced inlet flange support are installed. All vibrations are acceptable, see top two profiles in Figure 7. A bonus: Unit 200 data shows probable engine misfire, see half order dominant profiles in Figure 8. Site representatives are directed to confirm engine misfire and find a fault in the ignition wiring.

In the end, a successful and well understood solution to a problem half a world away was achieved.

Other Issues (cont. from page 1)

One characteristic of high speed compressors is that they are frequently designed to operate over a larger speed range. When a compressor operates over a large speed range, there are no speeds to tune a resonance so that it will not be excited by forces at the compressor speed (fundamental speed) or two times the compressor speed (second order speed). For example, a compressor that operates from 600-900 rpm allows for tuning of vibration problem modes between the top of the compressor fundamental speed and the bottom of the second order speed, between 900 and 1200 cpm in this example. However, if the unit operates from 600 to 1200, all frequencies will be excited. The compressor package must be carefully designed, constructed, installed and operated to avoid resonance and high vibration at the fundamental and second order of compressor speed.

This discussion has so far been limited to discussing the forces at the compressor fundamental and second order speeds. A reciprocating compressor causes forces at all the orders of compressor speed. These forces generally decrease as the order increases. However, as the compressor speed increases, the frequency corresponding to a particular order also increases. For example, the 6th order of a 900 rpm compressor is 90 Hz compared to 120 Hz for a 1200 rpm compressor. A mechanical natural frequency that is at a high frequency may not be excited in the case of a low speed compressor since the force at the order that is resonant with the mechanical natural frequency is low. However, for a high speed compressor, a lower order with a higher force may be resonant with the mechanical natural frequency at a high frequency. Determining solutions to vibration problems at high frequencies is typically much more difficult than solving vibrations at the compressor fundamental and second order speed.

Large power (>1500 HP), high speed, wide operating range separable compressors are becoming more common, particularly for pipeline and storage applications. These applications represent a significant challenge for the compressor manufacturer, compressor packager, and designer to develop solutions which will minimize vibration. These units often require extensive control of the acoustical response of pressure pulsations in the piping to minimize the forces caused by the pulsations. "Extensive control" means large primary pulsation bottles with complex internals and secondary pulsation bottles. More complex pulsation control requires additional support design, fabrication and installation time. The typical fast-track compression project will not allow for the required level of analysis to be done. Early involvement of the acoustical and mechanical designer (such as Beta) in the project as well as building some additional time into the project schedule to allow for design, fabrication and construction are key to a successful project.

Summary

In an ideal world, all reciprocating compressors would operate at a single speed, with low HP per cylinder and would always operate with double acting compressor cylinders. This configuration would result in a unit that could be designed to have very low vibrations. However, its operating capacity would be limited.

Proper design, construction, installation, operation and maintenance will result in the smooth long term operation of a high HP, high or variable speed reciprocating compressor. Achieving this goal is a co-operative effort between the compressor manufacturer, packager, design team, owner and operator.

News & Notes

Mark Deutscher presented "Troubleshooting Pulsation/Vibration Problems" at the GMRC Engine Analyzer and Reliability Workshop, held in Nashville in July.

BMA instructors were in Houston in September providing our "Improving the Reliability of Reciprocating Compressors" course for the GMRC. As usual, the GMRC attracted the best students and the class was very enjoyable for the instructors. Thanks to Marsha Short and this year's attendees.

Shelley Greenfield attended the API 68th Fall Refining Meeting in Denver, CO September 15 - 17, 2003. Over the past several years Beta Machinery has been participating in the pulsation and vibration control task force which has been working on a recommended practices (RP-688) document to be published with the 5th edition API 618. Both documents will tentatively be published in the spring of 2004.

The annual Gas Machinery Conference will be in Salt Lake City from October 6 - 8, 2003. Please drop by our booth to see us. For further information on the conference visit www.gmrc.org.

Beta will be attending the CMVA Annual Meeting and Trade Show to be held in Halifax from October 29 - 31, 2003. The theme of the conference is "21st Century Solutions for Condition Monitoring". Please contact Val Zacharias at cmvaed@shaw.ca or go to their website at www.cmva.com for further information.

Vibration Troubleshooting Seminar: This Fall Beta is adding a new seminar to the ½-day courses we regularly offer. Vibration Troubleshooting will be offered for the first time November 5 in our Calgary classroom, there is no charge to our customers for this training. This course will benefit people operating or designing equipment by helping them gain an understanding of what is involved in troubleshooting a vibration problem. Typical reciprocating compressor case studies will be used to demonstrate: frequency and amplitude characteristics of vibration and pulsation behavior, dynamic forces, mechanical natural frequencies, and mechanical resonance.

Our regular seminars, Torque Talk, and Pulsation Vibration, will be offered in our Calgary office in the coming months as well. Onsite training may be arranged. E-mail us at info@betamachinery.com or call Sarah at 403-245-5666 to register or obtain further information on our training opportunities.

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