Vibration and Performance Design Considerations for Multi Unit Reciprocating Compressor Gas Storage Facilities

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ABSTRACT

A typical gas storage facility is a long term investment that relies heavily on the efficient and reliable operation of multiple high horsepower reciprocating compressors. The compressors must be capable of operating over a very wide range of operating conditions. An additional factor is that gas control demands may dictate that a unit starts and stops a number of times in a single day.

This paper looks at a number of the vibration and performance design aspects of a multi-recip compressor storage facility and provides suggestions for initial layout and goals for the necessary design studies. Some of the issues discussed are:

- Ensuring pulsation control design will accommodate extremes in compressor loading
- Common sense approaches to unit layout, on-skid and off-skid piping layout and support strategies with an emphasis to minimize vibration interaction between units, problems with instrumentation and improved operator comfort.

1. INTRODUCTION

A gas storage facility has compression equipment that must:
- work very hard,
- cover a wide range of operating conditions,
- perform efficiently, and
- last a long time.

Beta Machinery Analysis has contributed to the acoustical and mechanical design of compression equipment for a number of multi-unit gas storage facilities with 20,000+ total compression horsepower. Beta has also provided vibration and pulsation troubleshooting for other installations. In the process it has been learned that by keeping some fairly basic and common sense design considerations close at hand during the design, a more robust and trouble free installation will result. This paper presents viewpoints on certain aspects of the design of these facilities.
2. A TYPICAL GAS STORAGE FACILITY

A gas storage facility consists of four main components. See Figure 1 and the descriptions following.

- **Storage cavern** - Typically a depleted reservoir or salt deposit with suitable formation characteristics and a network of wells. Working capacities will be in the range of billions of cubic feet.

- **Pipeline systems** - High capacity pipelines for flexible supply and distribution of gas. Moving gas quickly is important. These systems will be capable of flowing a billion standard cubic feet per day or more. The well head piping system will typically tie in multiple wells accessing the storage cavern.

- **Compression equipment** – There will usually be two stages of compression capable of injection from a pipeline supply pressure in the area of 400 to 1100 psig to a maximum reservoir pressure in the area of 2500 even 2900 psig. There will be times that gas can free flow into the storage cavern as well as times when the compressors will be operating at very low compression ratios. These compressors will also be used in withdrawal mode when cavern pressures dip below sales pipeline pressures. How much horsepower is required at a particular site will depend on the injection rates that need to be achieved. Multiple units are best to handle incremental flow requirements efficiently. Five to ten units are typical.

- **Withdrawal de-hydration/processing systems** – Gas flowing out of the reservoir will usually need to be cleaned up before heading out the sales pipeline.

![Figure 1: Schematic of a gas storage facility](http://www.BetaMachinery.com)
3. COMPRESSION EQUIPMENT (and some thoughts on each)

The Compressor

The typical compressor in gas storage service will be a large six throw separable frame in the range of 1800 – 4500 horsepower operating at 750 – 1200 RPM. Most commonly the compressor will be configured to switch automatically between single stage and two stage operation as needed. Less common are installations with dedicated single stage units operated in series or parallel as needed.

Acoustically, these machines will be strong pulsation sources. A low speed of 750 RPM means the lowest frequency of pulsation will be 12.5 Hz, strongest when single acting; this can be managed with reasonable pulsation control techniques. Operating the compressors any slower begins to require significantly larger pulsation bottle volumes.

Mechanically, cylinder motion due to gas loads is significant. With the expected range of operating conditions, the harmonic content of the cylinder motion is extremely variable. This means that not only the pulsation bottles but all the various components attached to the bottle or cylinder will see varying degrees of vibration excitation at frequencies up to ten times compressor speed. Risk of problematic vibration due to cylinder motion is significant.

The photograph in Figure 2 shows four electric motor drive, six throw compressors operating in parallel. They are automatically configured as one stage or two stage operation depending on supply and reservoir pressures. Capacity control is achieved with head end bypass unloaders.

Figure 2: System with four units in parallel
The Driver; Electric Motor vs Engine Drive

Comparison of the economics of motor vs engine is complicated. Maintenance costs are higher on engines than on motors. Energy costs should be higher for motors. There are some circumstances where the price of power has been offered to the end user at an unusually low rate. In this circumstance the balance of the cost could be higher for the engine.

From an acoustical perspective, a fixed speed motor drive compressor is the simplest system to work with. A common complaint about multiple fixed speed units operating in close proximity is an annoying tendency of the vibration in some areas to rise and fall over a period of time. This behavior develops because the dynamic forces (primarily acoustical forces in the piping, but also structure borne mechanical forces) from each unit will either add or subtract depending on the phasing of the sources. Engine drive units operating at nominally the same speed can also exhibit this behavior. However, the actual speed difference between machines is usually greater - resulting in a more rapid variation in phasing between units. The result is that a vibration interaction is not quite as discernible.

Choosing variable frequency drive (VFD) for electric motor speed control will typically offer a reasonably efficient 50% capacity reduction. However, slowing a 750 RPM machine down to 375 RPM means that pulsations will be as low as 6.25 Hz. The vessels required to filter pulsation will need to be significantly larger.

Cylinder Unloading

In order to provide the maximum automated flexibility in capacity control, some manner of cylinder unloading will be utilized. Pneumatic suction valve finger unloaders are commonly utilized. Stepless suction valve unloaders, automated fixed volume pockets, and head end bypass un-loaders are other methods of unloading and offer a combination of increased flexibility and greater efficiencies. Manual head end variable volume pockets may also be seen.

The customer will want total flexibility with regard to cylinder loading. In some cases a specific loading sequence may offer significantly better acoustical, mechanical, or torsional system behavior and should be discussed with the customer. It will be beneficial to work with whomever is responsible for setting up the compressor control system. Choices will have to be made regarding loading sequence and staging change over points. It may be practical to avoid acoustically marginal operating conditions.

Compressor Packaging

Typically the compressor and driver, perhaps with secondary volumes and minimal piping will be installed on a fabricated steel skid. Current practice will have several identical compressor packages sharing the same concrete foundation and steel frame
building. Each compressor will have its own cooling system. Keeping the piping systems identical on each unit ensures consistent results and also reduces the acoustical and mechanical design burden.

Concrete filled ladder skids are typical and entirely suitable for this service without extensive design requirements. Designs with widely spaced compressor and engine main skid beams are preferred.

The skid design should be able to accommodate large diameter primary discharge bottles and have room for additional length in the pulsation bottle or a secondary discharge pulsation vessel. On suction systems a similar accommodation is needed. Secondary suction volumes in the form of vertical scrubber type vessels offer a more compact skid arrangement for a given volume requirement than an overhung multi-chamber suction bottle. Horizontal secondary suction and discharge bottles fit well with the layout of a single staged machine. See Figure 3 below. One stage/two stage skids are not amenable to the use of horizontally oriented secondary volumes.

4. DESIGN STUDIES

There are four important aspects of the design of the compression equipment installation that require careful consideration:

- Acoustical behavior of the piping system
- Mechanical vibration behavior of,
  - equipment, pulsation bottles and piping near the compressor
  - piping away from the compressor
  - small bore piping and instrumentation
  - pulsation bottle internals
• Thermal behavior:
  o Compressor-to-cooler piping and lateral piping to headers
  o Long header piping
• Torsional vibration behavior of driver and compressor driveline.

**Acoustical Behavior**

It is critical to manage the acoustical behavior of the piping system. That means developing a clear understanding of the pulsation generated by the compression equipment and how it interacts with the piping and vessels attached to it. An analysis should be conducted to develop this understanding and to design appropriate pulsation control measures in consideration of the following points:

• API 618 Acoustical Simulation Study M2 or equivalent is an appropriate level of analysis. This study predicts the pulsation characteristics throughout the piping system and facilitates the critique and design of the pulsation control elements.

• A computer based digital pulsation simulation is preferred over an “analog” simulation. A digital simulation is really the only practical way to handle the extent of the multi-unit piping system and the range of operating conditions that need to be considered. The software should be capable of both a frequency and time domain simulation of the pulsation behavior in the proposed piping system to conduct a credible and efficient analysis. Accurate representation of the acoustical excitation sources from the various cylinder loading schemes is important and will likely be handled with closely held proprietary methods. Look for a supplier with a proven ability to conduct field verifications of pulsation behavior.

• There will likely be a number of iterations in the required performance of the compression equipment between the initial planning stages to the actual operation of a facility. Hence the need in the early stages to design in some “headroom” for unforeseen situations (e.g. supply pressures being lower or higher than planned, operation at very low compression ratios in both single and two staged modes, etc.). As long as normal compressor operational parameters like rod load and discharge temperature are within bounds it is conceivable, even likely, that someone will operate the compressor there.

• Carefully review the operating conditions. The customer should be submitting hundreds of performance points. If not, read between the lines and extrapolate a reasonable range of operation to discuss with the customer as a more realistic and thorough basis for design purposes. The extreme ends of operating envelopes may not be the “worst case” condition for pulsation design at every harmonic. Selection of acoustical (and mechanical) design conditions should be based on a thorough review of pulsation and gas load harmonic distributions.

• Model detail should include all off-skid piping to well head and pipeline tie ins. As a minimum the model should include full detail of the pulsation bottles of all units on-line as well as the source unit.
• Be sure to examine the pulsation performance of each unit at all design conditions. Pulsation from units at different positions along the header will typically result in quite different header and lateral piping unbalanced forces.

• A good starting point for vessel design is to develop filters with a low frequency cut-off of 0.7x run speed, for instance 10.5 Hz for a 900 RPM compressor that would operate with one or more cylinders single acting.

• Design guidelines should follow API 618 as well as include conservative unbalanced force and dynamic pressure drop guidelines. Account for the worst case interaction of pulsation from all units.

• Consider adjusting header lengths to minimize shaking forces at 1x run speed for the “average” proposed operating condition.

Mechanical Vibration Behavior

The dynamic mechanical vibration behavior of the equipment is also important. In this case it means understanding the dynamic forces acting on the compressor, cylinders, pulsation bottles (internals too!) and piping as well as how these components will respond to the various dynamic forces. It is appropriate to break this examination into four logical blocks:

• Mechanical analysis of components near the compressor
  o API 618 study option M6 (Compressor Manifold System Vibration and Dynamic Stress Analysis) as an extension of the M5 study (mechanical natural frequency predictions) is recommended. The goal of this dynamic analysis is to ensure that the cylinders, bottles, vessels and nearby piping are not mechanically resonant and that the predicted vibration and dynamic stresses in the bottles and piping components are within target values.
  o Boundary condition representations are very important. Model detail should include discharge bottle supports, compressor frame, inboard/outboard support and skid flexibilities, close coupled piping and support details for useful predictions.
  o Include residual compressor dynamic forces, rod load forces affecting cylinder motion and acoustical forces. Select design conditions for worst case forces at all harmonics up to 10x run speed.
  o Pay particular attention to the proximity of suction pulsation bottle cantilever mode to 3x run-speed on six throw machines with three cylinders/bottle. The 3x rod load forces are in phase and excite this mode readily.

• Mechanical analysis of components away from the compressor
  o An API 618 study option M7 (Piping System Dynamic Stress Analysis) of any potential problem areas due either to marginal pulsation forces and/or geometry as an extension of an M5 study (mechanical natural frequency predictions) of the header section and/or lateral piping system in question is recommended. The goal of
this dynamic analysis is to ensure that the piping and support system is not mechanically resonant and that the predicted vibration and dynamic stresses in the piping or support components are within target values.

- Boundary condition representations are again very important. Model detail should include a generous section of piping around the area of interest, pipe clamp and support details for useful predictions.
- Include the worst case shaking forces predicted in the pulsation analysis.

- **Mechanical analysis of pulsation bottle internals**
  - An API 618 study option M8 (Calculation of Dynamic and Static Stresses on Pulsation Suppressor Internals) is recommended.
  - Boundary condition representations in this situation are relatively simple but no less important.
  - Model detail should include accurate representation of the internal bottle assemblies and weld details used. Figure 4 shows a portion of the three choke tubes and one of three baffles pre-assembled for insertion into a multi-chamber discharge bottle.

![Figure 4: Typical pulsation bottle internals being prepared for installation into multi-chamber discharge bottle.](http://www.BetaMachinery.com)

- Include the worst case shaking forces predicted in the pulsation analysis for dynamic loading and the appropriate maximum static pressures.

- **Small bore piping and instrumentation review**
  - Modelling is not being suggested.
Examine all small bore piping and instrumentation connections to pulsation bottles and process piping. Apply the “Three R’s” of small bore piping connections near reciprocating compressors:

- REDUCE the number to absolute minimum.
- RELOCATE away from the immediate compressor area if at all possible.
- RE-CONFIGURE the connection to avoid an unsupported cantilevered mass.

Thermal Behavior

The typical gas storage facility has long header runs with multiple lateral connections to the compressors. A thermal analysis is essential to ensure that piping and support stresses due to thermal expansion are acceptable.

- **Thermal analysis of headers, unit piping**
  
  - ANSI B31.3 is a typical piping design code for the thermal analysis of this piping.
  - If possible utilize the same model for both mechanical dynamic analysis and thermal analysis. Integrating the two allows improved efficiency when considering the divergent requirements of the thermal and dynamic designs. The benefit is reduced design cycle time and cost.
  - The extent of the models should be determined by the particular behavior being analyzed and be generous enough that boundary condition termination detail in the area of interest is appropriate and not over conservative. Multiple models will likely be required.
  - Select thermal load cases carefully to ensure they cover the site conditions and realistic gas temperature possibilities completely.
  - Primary goal is to ensure that the piping system has the appropriate flexibility in the critical thermal growth directions while being stiff enough in the critical vibration directions to meet the basic mechanical requirement that mechanical natural frequencies exceed 2.4x wherever possible.

Torsional Vibration Behavior

The torsional vibration behavior of any pairing of a reciprocating compressor with an engine or motor should be well understood.

- **Torsional analysis of driveline**
  
  - API 618 Section 2.5 refers to both lateral and torsional critical speeds.
  - An analysis to predict the torsional natural frequencies of the driveline is the first step. Critical speeds should be separated from operating speed by a margin of 10% and from harmonics of operating
speed by a margin of 5%. If ideal separation is not possible the analysis should be extended to a forced response study to predict shaft stresses and ensure a minimum safety factor is met.

- In practice, lateral studies of reciprocating compressor drivelines with engines or electric motors are seldom done.
- Always conduct a torsional vibration forced response analysis.
- Use a tolerance band method to allow for variability in the supplied information to avoid over and under conservative recommendations.
- As with the acoustical design, be conservative with the design at the onset. Look for any choices that give any extra headroom. Use good shaft material, large motor shaft diameters, generous radius on all steps in shaft and, of course, do not use keyways!
- When faced with unwieldy numbers of conditions, examine all of them and select conditions with the highest contributors at each harmonic at each throw as a minimum for a thorough study.
- Review loading sequencing choices. Loading the compressor from the auxiliary end will typically be better.
- If new conditions arrive…be sure to check them.

5. TYPICAL RECOMMENDATIONS

Acoustical

- Be conservative with the pulsation control especially in shared lateral and header piping. This is a very serious aspect of the design and typically means that it is necessary to filter pulsation at the source. Various configurations for constructing the filter are practical.
  - For suction systems, a chambered pulsation bottle with a close coupled scrubber offers the most volume for a given footprint. The extra volume will help create an effective acoustical filter when striving for the minimum pulsation away from the compressor within practical constraints of mechanically sound primary and secondary pulsation vessels. When less volume works, a single overhung multi-chamber pulsation bottle is practical. Designs with long piping runs between primary bottle and secondary volume should be avoided.

- For discharge systems, large multi-chambered bottles can work quite well if sufficient room is available to accommodate a single bottle with enough volume (diameter and length) to achieve the appropriate filter characteristics without incurring excessive pressure drop. Figure 5 shows a unit with a long four chamber discharge bottle.
- Use end treatments on choke tube entrances and exits to minimize pressure drop. Figure 6 shows one way that these details can be easily achieved.

**Figure 6: Choke tube end treatments**

- Pre-machined bell mouth entrance
- Bell mouth entrance welded to choke tube and sand blasted
- Pre-machined diffuser exit
- Diffuser exit welded to choke tube. Weld penetration will be removed for a flush inner surface prior to sandblasting.
• Orifice plates at the bottle-to-cylinder connections will typically be beneficial.
• Orifice plates should be avoided in the shared piping system.
• While piping changes are not typically required, there may be dead legs in the piping sections containing the automatic valves controlling one stage/two stage operation that will cause problems with the acoustical design. It may be beneficial to relocate the valve to an optimal location. In an extreme case, two valves could be used to eliminate the dead leg altogether.
• Install discharge isolation valves downstream of unit check valves to prevent check valve chatter in no flow situations.

Mechanical
• Use a “full rail” grout of main skid beams, pipe support beams and wedge support beams in the connection of the skid to the concrete foundation. This differs from a “full bed” grout procedure in that the grout pours are concentrated along the main structural members and not wasted in the large voids between skid members. Furthermore this type of a grout detail can be designed to resist oil contamination better than a “full bed” grout. Locate anchor bolts at the key points. Stiffen the flanges at the anchor locations and ensure long anchor bolt stretch length.
• Note: Maximum dimension of “full rail” grout pours will be limited by how much the grout will contract. Incorporate appropriate joints in the grout pours to control cracking. Alternate methods can be used but must ensure a gap free connection is achieved at all important areas including areas directly under the machines.
• Avoid gussets on pulsation bottle cylinder nozzle connections. Cylinder nozzle configuration should be critiqued closely and should focus on providing a clean geometry.
• Avoid bracing designs that result in “tight” over constrained geometry in the area of the cylinders and pulsation vessels.
• Anticipate the possibility that a head end cylinder support may be needed. Do not locate piping through the area directly beneath the outboard end of the cylinders. Figure 7 shows a unit fit with outboard cylinder supports.
• Design of secondary pulsation vessel supports should be considered “machine quality”. Remember that the supports for these vessels will have to deal with residual shaking forces as well as thermal growth. Avoid integral feet on horizontal secondary pulsation bottles.

• Above ground piping and headers should be installed as low as practical. Structural steel supports should be generously sized with minimum of two piles supporting the main wide flange beam, depending on the number and size of piping on the header. Locate piles under or near the largest line. Figure 8 below shows a good example of piping installed low to the ground.
• Ideally, with appropriately controlled acoustical forces in headers, header piping clamp support requirements will be minimal and satisfying thermal requirements will be straightforward. Where clamps are required use a flat bar style clamp with separate bolts. U-bolts are not recommended.

• For operator comfort it is best not to support a stairway or walkway to any process piping support, this includes cooler frames. Where necessary provide separate piles/foundations for stairways and walkways. Figure 9 below is a good example of this.

![Walkway over header piping on separate support](http://www.BetaMachinery.com)

**Figure 9: Walkway over header piping on separate support**

• Ancillary piping and electrical and instrumentation racks should not be connected directly to process piping supports.
• Piping support to lateral piping should be independent from the header rack.
• Maximize unit to unit physical separation within the building.
• Skid foundation blocks should be constructed independently of each other and be integral with a sunken pit for lateral piping and any necessary valves.
• Inter-unit flooring should be poured separately from the skid foundations using ½” felt separation from skid foundation blocks. Under no circumstances should inter-unit flooring be used for support of process piping.
• Minimize small bore attachments. Any vital small bore attachments to reciprocating compressor process piping should be located as far from the compressor as possible, near planned process piping supports and/or large valves and oriented in the direction parallel to anticipated vibration. Use minimal mass and projection (use studded flanges) or as a minimum, gusset any small flanged nozzles or long weld necks with two gussets in the directions parallel and perpendicular to the piping. Figure 10 below shows a good gusset detail used to correct a problem appendage.
In piping over 8” nominal, maximize the process piping wall thickness at least in the area of the small bore attachment; ½” minimum w.t. up to 12” nominal pipe, 1” minimum w.t. over 12” nominal pipe.

6. ADDITIONAL CONSIDERATIONS

Shop “Bump Test” for mechanical natural frequency verifications

Visit at least the first unit fabricated in the shop to conduct “Bump Testing” to verify mechanical natural frequencies predicted by computer models. It can be of excellent value to conduct a test of the bare compressor frame and cylinders on skid prior to installing pulsation bottles to confirm that cylinder and frame as it is installed on the skid beams behaves as predicted and that impact of various construction details are well understood. This is an opportune time to make minor changes that may be required.

Field “Bump Test”, start up performance verifications and vibration baselines.

Recognize that the design and modelling world has limitations. Plan to conduct a comprehensive mechanical natural frequency test of the first unit installed in the field. Prepare at least one unit for performance checks. Collect vibration baseline data on all machines.

Future possibilities….

It may become practical to have unit-to-unit phase control for optimized relationship between units to minimize the effect of pulsation interaction and elimination of beat phenomenon in fixed speed motor drive applications. A potential benefit of this
approach would be the ability to use smaller pulsation bottles and/or reduce pressure drop. It is conceivable that an optimal phase relationship could be determined by the pulsation analysis or by a realtime vibration feedback control program to minimize vibration in header piping at the lower harmonics, 1x – 4x. Initial estimates suggest that for a fixed speed installation of 5 units an overall reduction of total forces in the order of 30% is theoretically achievable along with a complete elimination of the cyclic throbbing component inherent to multiple units at slightly different speeds.

7. SUMMARY

In order to allow storage facility operators the flexibility and high throughput capability they need to respond to marketing demands to store and deliver gas efficiently and reliably, it is important to ensure that the acoustical and mechanical design of the installation is suitable for all operating scenarios. The acoustical design needs to offer as much unit isolation as practical while managing pressure drop. Extra emphasis is required in the mechanical design of the piping and support structures, pulsation vessels, small bore piping and instrumentation etc. All of these components must be capable of withstanding the widely varying forces that accompany the wide range of operating conditions. The interaction of residual acoustical and mechanical forces from multiple units operating in close proximity will highlight any weaknesses in the design.

While many important design recommendations will result from adherence to the suggested design studies, give consideration as early as possible to common sense approaches to unit layout, on-skid and off-skid piping layout and support strategies that can help minimize vibration interaction between units, problems with instrumentation and improved operator comfort.

KEY POINTS:

- The acoustical simulation study must conduct an in depth examination of a representative selection of operating conditions out of a wide potential operating range. It is unlikely that the full range of potential operation can be precisely nailed down in advance. Therefore some proactive measures are prudent to provide a robust acoustical design.

- There is a significant advantage in larger unit-to-unit spacing. This reduces structure borne mechanical vibration interaction between units. Where each unit has its dedicated cooler, it may be that a minimum cooler-to-cooler spacing to prevent shrouding will dictate unit-to-unit spacing. The importance of this detail is that it allows each unit to have identical configurations for the cooler and lateral piping runs. Identical piping in these areas will reduce the design cycle time and cost and result in a design with less risk of installation compromises.
• Integrating the thermal study of both the on-skid and unit lateral piping and the header piping with the dynamic assessment allows improved efficiency when considering the divergent requirements of the thermal and dynamic designs. The benefit is reduced design cycle time and cost.

• Give small things like small bore piping and instrumentation on process piping and near compressor cylinders extra scrutiny in the early stages of the design. Reduce the number of these connections. Relocate them to “quieter” areas. Reconfigure them to eliminate an unsupported cantilevered mass.

• The typical new installation will have identical high power (2000 + HP) machines set up to operate as one or two staged automatically as needed. It is important to consider stage configuration switching valve locations to avoid causing problematic acoustical interactions related to dead-legs.

• Don’t forget the torsional analysis. As with the acoustical design, be conservative with the design at the onset. Look for any choices that give any extra headroom. Include good shaft material, large motor shaft diameters, generous radius on all steps in shafts, clean and accurate assembly of couplings and flywheels, and of course no keyways please. Review loading sequencing choices. Loading the compressor from the auxiliary end will typically be better. If new conditions arrive… check them.

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