Optimized Skid Design for Compressor Packages

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Abstract

The majority of compressor packages are now mounted on steel skids or baseplates. Designing a skid for a new machinery package can be challenging because of these factors:

- The skids must be designed to avoid resonance and vibration (from dynamic machinery forces and couples).
- The industry is looking for lower cost packages. This can drive suppliers to reduce the skid cost and associated stiffness, but an inappropriately designed skid will create vibration and reliability problems. In some cases, skids are considered too flimsy for the required application.
- New designs must consider loading, lifting and transportation issues, as well as weight limitations. Pedestal height can also cause problems.

This paper will outline the issues and approaches involved in skid design for vibrating loads such as reciprocating compressors and pumps.

This paper discusses industry best practices in skid design, including optimized design techniques. Two case studies will be used to illustrate different skid designs and the impact on cost, performance and reliability. This paper will benefit owners, packagers, and engineering companies involved with rotating equipment.

1. Introduction

When designing a structural steel skid (baseplate) for a compressor or pump package, the design must balance stiffness, mass, and cost. High stiffness will help avoid alignment problems due to skid deflection during transportation and installation. Heavier skids tend to have lower overall vibrations, but can have high deflections when lifted. The challenge when optimizing the design is to know where steel can be added or removed to maximize the stiffness and minimize the costs.

2. Skid Loads

There is considerable confusion about dynamics, quasi-static and static analysis. Figure 1 identifies the applicable load frequency ranges and the design criteria for these three analyses.

2.1. Static Loads

Static skid design focuses on evaluating stress and buckling of members under constant loads. (Constant loads can also be described as loads applied at a frequency of 0 Hz.) They can also focus on deflection of skid members, which can affect alignment of equipment.

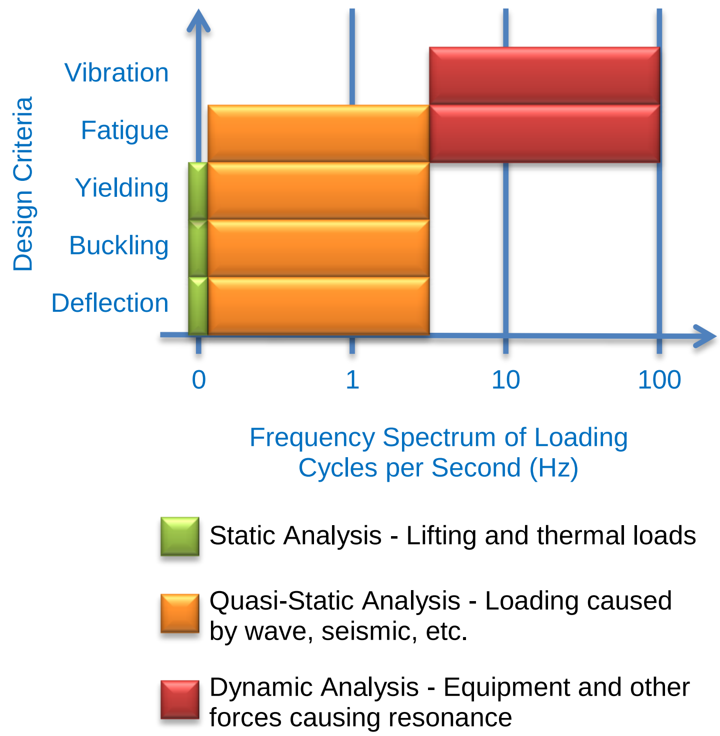


Figure 1. Loading type vs. loading frequency and design criteria

Typically static loads are:

- Dead loads, including weight of permanent equipment.
- Thermal loads which includes forces created by temperature changes and pressure.
- Drive torque of compressors and engines.
- Lifting or dragging loads, when moving the skid with cranes or winches. These loads can include a load factor which considers the impact from sudden stops or motion of the lifting equipment (e.g., offshore lifts). A load factor of 1.15 to 2.0 is common.
- List angle, which creates horizontal loads when a ship leans to one side.

Guidelines for static skid design include American Institute of Steel Construction (AISC) and owner specifications for deflection (e.g., 0.5 inch deflection per 15 feet of skid length when lifting).

2.2. Quasi-Static Loads

Quasi-static loads are loads which are periodic, but at a low enough frequency (relative to the natural frequencies of the equipment package) so the inertia effects of the structure do not come into play. They tend to have a frequency of less than 3 cycle per second or 3 Hertz (Hz).

Typical quasi-static loads are:

- Environmental loads like live loads, wind, current, wave, earthquake, ice, earth movement, and hydrostatic pressure. These can occur in any direction.
- Construction loads including loadout, transportation, and installation.

Quasi-static skid design focuses on evaluating stress and buckling of members, similar to static stress design. However, fatigue analysis may be done on loads like those caused by waves.

Guidelines for quasi-static skid design include API RP 2A-WSD and International Building Code (IBC). Owners and equipment manufacturers may also have standards and specifications for both static and quasi-static loads and deflections.

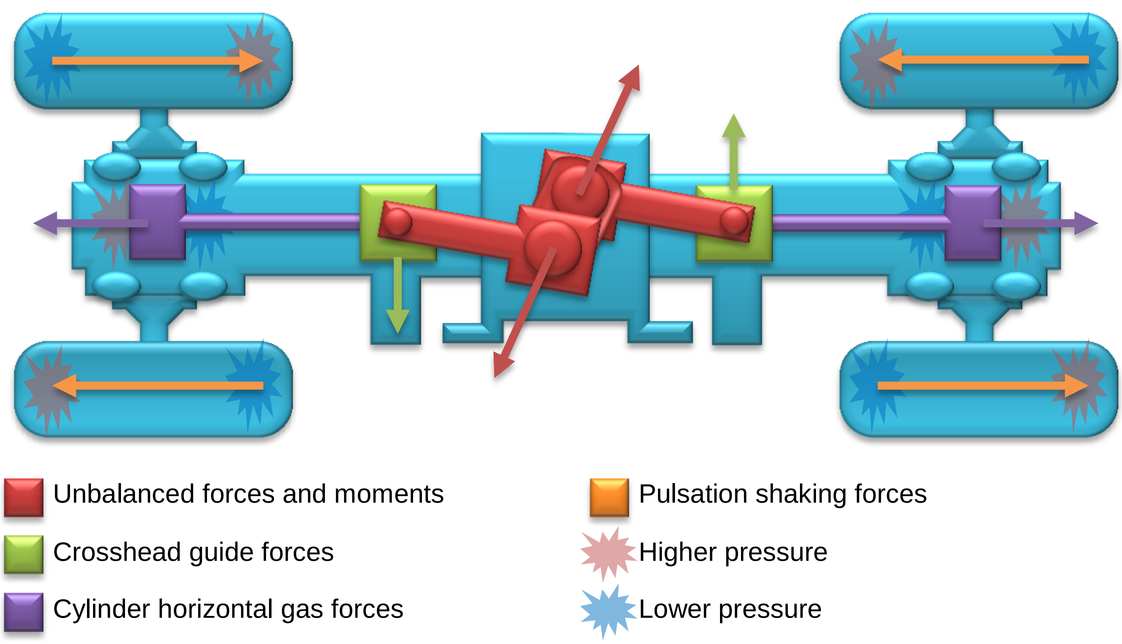


Figure 2. Common Reciprocating Compressor Dynamic Forces

2.3. Dynamic Loads

Dynamic loads can be caused by waves, wind, earthquake or machinery, but it is typically loads by the machinery itself that concern the skid designer, as it is most likely to cause resonance. Resonance is the condition when the frequency of the dynamic force is within +/-10% of the mechanical natural frequency (MNF) of skid, vessels, piping, and structure/foundation. (At the design stage, +/-20% is typically used to account for modeling and fabrication uncertainties.)

Figure 2 shows common reciprocating compressor dynamic forces, which include:

- Unbalanced forces created by rotating and reciprocating weights like crankshafts and piston assemblies. If the forces are offset, they can create unbalanced moments on the equipment. These can be obtained from the compressor or engine/motor manufacturers.

- Horizontal cylinder gas forces, created by the differential pressure between the head end of a cylinder and the crank end.
- Vertical forces on the crosshead guides, created when rotating motion is converted into reciprocating motion.
- Pulsation-induced shaking forces, created at elbows and changes in pipe diameter due to pressure pulsations.
- Misalignment of the compressor and driver.
- Rolling torque on engines, which can occur at higher orders of engine runspeed (e.g., 4x, 8x, … or 3x, 6x, …).
- Torsional vibrations, which may cause horizontal vibrations of the compressor frame.

These forces are typically harmonic and occur at discrete multiples of equipment runspeed. In variable speed machines (e.g., motors with VFDs), the frequency of the force will change with the speed of the equipment.

The majority of these forces occur at the first and second order of compressor runspeed, so raising the mechanical natural frequencies (MNFs) of major components on the equipment package above 2x runspeed is an effective strategy for avoiding resonance. Intertuning (between 1x and 2x) or detuning (below 1x) may also be possible in selected cases.

In rotating equipment like motors, centrifugal compressors, and screw pumps, the dynamic forces are usually just unbalance forces and flow-induced pulsations, which tend to be low. Standards like ISO 1940/1 give recommended residual unbalance for various rotating equipment.

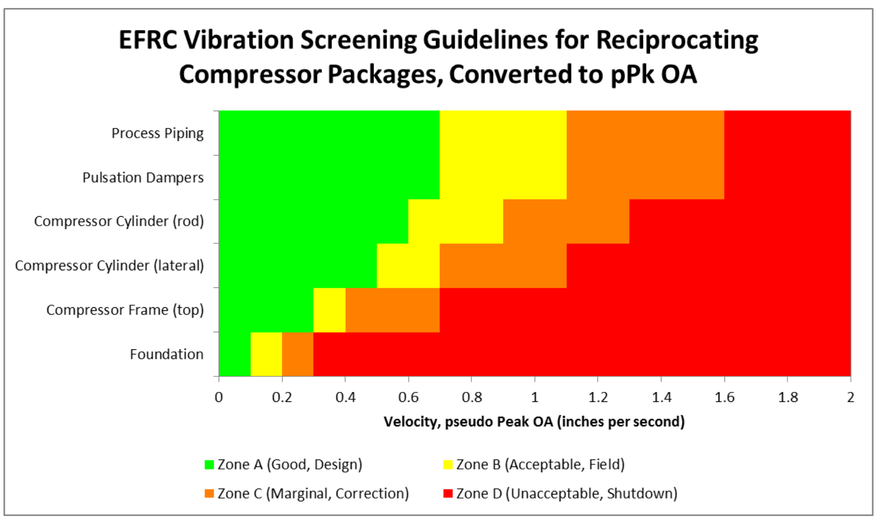


Figure 3. EFRC Vibration Guidelines Converted to pseudo Peak Overall Velocity

Dynamic skid design focuses on evaluating vibrations and fatigue. Limiting vibrations of skid members is important to limit the vibration of the equipment, vessels and piping that are attached to it. If a skid member has high vibrations then the components attached to it will likely have high vibrations also. It is important when taking vibration measurements that all components along the load path (described in Section 3.1) are measured and compared to guideline.

When discussing guidelines, it is important to distinguish been spectral and overall vibration measurements. Overall vibration measurements are the actual deflections (or velocity) of the component versus time. Spectral vibration measurements typically require specialized measuring equipment, which break the vibration down into peaks at discrete frequencies. It is used for troubleshooting to identify and evaluate problem areas.

Vibration guidelines for skid members can vary depending on the situation. Many equipment vendors specify a maximum allowable vibration at the equipment mounting locations. A spectral guideline of 0.1 in/s peak (25.4 mm/s peak) is common. Figure 3 shows the European Forum Reciprocating Compressors (EFRC) overall (OA) screening guideline for different areas of a reciprocating compressor. (The guideline is in pseudo Peak because it has been converted from an RMS guideline.) Note that the components that are closer to the foundation typically have a lower allowable vibration guideline.

3. Best Practices

3.1. Load Path

The structure and foundation underneath a reciprocating compressor are used to absorb the energy created by the dynamic forces. When the foundation is heavier, the vibrations are lowered due to the formula: *acceleration = force / mass*. When the support structure is stiffer, the vibrations are lowered due to formula: *deflection = force / stiffness*. Therefore, it is desirable to have a massive foundation and stiff structure. (Damping can also lower vibrations, but it is usually not practical to add damping to a structure without reducing the stiffness.)

In order to transmit the dynamic forces through the structure to the foundation, the path between the dynamic forces and foundation must be as stiff as possible. Since the stiffness of a collection of components in series is dominated by the stiffness of the weakest link, care must be taken to avoid any excess flexibility in the structure. In other words, the path the dynamic load must take to get to a region of high mass or high stiffness must be as direct and stiff as possible.

Steel plate is significantly less stiff in bending than in tension or compression, therefore bending of steel plate should be avoided. Full depth gussets should be used in wide flange steel under flanges that take dynamic loads.

Table 1. Important Design Considerations Depending on Foundation Type

Type of Foundation	Impact on Skid Design
Concrete foundation or block-mounted	• Grouting ensures stiff connection between skid and foundation.
Gravel pad	• Full gravel bed desired. • Effective if contact between skid and gravel is ensured through packing and pad design.

Pile-mounted	• Pile locations must be appropriately placed for dynamic loads, in addition to static and quasi-static loads.
	• Important dynamic load locations include:
	    ○ Under compressor inboard cylinder supports,
	    ○ Under compressor cylinder head-end,
	    ○ Under scrubbers.
	• Connection between pile and skid is critical.

Offshore structure	• Weight limitations so concrete may not be permitted. • Connection between skid and structure may not be rigid. Plug weld if required. • Platform or FPSO deck may not be designed for reciprocating compressors. • Important that main full depth deck beam pass perpendicularly under compressor frame. Compressors should be located close to vertical columns.

3.2. Risk

A finite element analysis (FEA) of the dynamic forces and associated vibrations should be done in certain cases including:

- New compressor frame or driver/frame combination.
- Higher horsepower compressor on an existing skid design.
- Less massive foundations or less stiff structure, like pile-mounted units or offshore units mounted on platforms or ship decks.

Table 1 above outlines the impact the foundation type has on the skid design.

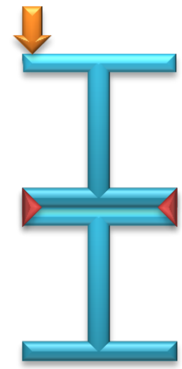
3.3. Concrete

Concrete can be used inside the skid itself to add mass and stiffness. It is most effective when it is rigidly connected to skid beams using rebar or nelson studs, especially the top flange where vibration equipment is attached.

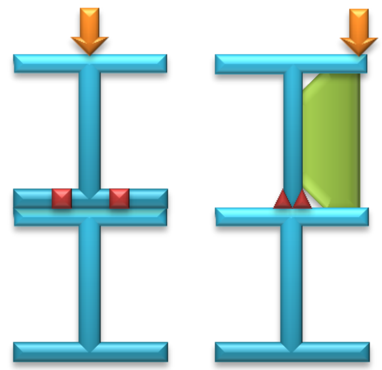
Key locations for concrete are in the compressor and engine/motor pedestal, in the main skid underneath the pedestals and scrubbers.

3.4. Beams

It is usually more advantageous to use taller beams than use beams with thicker flanges and/or webs. Taller beams are much stiffer in bending than shorter heavier beams.



Loads and welds on beam flanges not recommended



Plug welds or T-sections are recommended. Loads should be in the plane of the beam web or gussets should be installed

Figure 4. Recommended design for connecting stacked beams

When stacked beams must be used, the connection between the two beams is important. It is a stiffer design to weld a T-section to the top of a wide flange beam (taking care to line up the webs) or plug weld beams than to weld two wide flange beams on top of each other at the edge of the flange (Figure 4). The loads on beams should be in the plane of the beam web, or gussets should be used.

4. Modeling Considerations

When modeling reciprocating compressor skids using finite element analysis (FEA), different levels of detail are required for different types of loads. For static loads like lifting loads, one-dimensional (1D) beam elements are all that is required. Point masses can be used for equipment. For quasi-static analysis, 1D beam elements can also be used, but more detail needs to be added to the model because quasi-static loads can be horizontal, and thus act on components like piping and vessels. Dynamic analyses often require shell (2D) or solid (3D) elements for certain parts of the model to accurately account for local flexibility (e.g., the flexing of a beam flange or the impact of gussets). Local flexibility can play a large role in determining mechanical natural frequencies and vibration amplitudes of equipment packages.

5. Case Studies

5.1. Replacement of Existing Driver Pedestal

Figure 5. Existing engine pedestals (left) and new engine rails (right)

A customer was doing a field retrofit to replace an engine driving a reciprocating compressor. The turnaround time was fairly quick, but the customer wanted to avoid vibration problems. There were no skid drawings, so the deck plate had to be removed and the beams field-measured. The goal was to remove the four existing engine mounting pedestals and replace them with two new engine rails (Figure 5) that could handle the higher horsepower Caterpillar G3608.

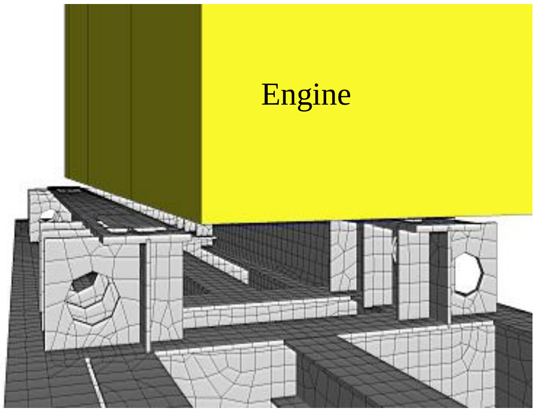


Figure 6. Early FEA model (with holes in gussets)

A finite element model was created, using ANSYS software, to try several different designs (Figure 6). The design that was eventually installed was a W18x106# beam for the two main engine rails. The rail was sunk about 3.5” below the top of the main skid, and the top flange of the main skid was coped so the engine rail could dropped down and be welded directly to the skid beam web (Figure 7). Also, this allowed the concrete poured in the main skid to cover the bottom of the engine pedestal rail, adding stiffness and transmitting the engine dynamic forces into the concrete.

No cross beams were possible in the engine pedestal due to the oil pan on the G3608, so gussets had to be installed on the outside of the engine rails. Gussets were added until the mechanical natural frequency (MNF) of the engine was above 2.4 times maximum engine runspeed. This was done to avoid resonance due to the coincidence of the engine MNFs and the engine primary (1x) and secondary (2x) forces and moments.

This optimized approach helped focus the design on the areas that required stiffness, and avoided excess costs and time in order to meet the tight revamp schedule. The engine was installed and has been running without issue since 2011.

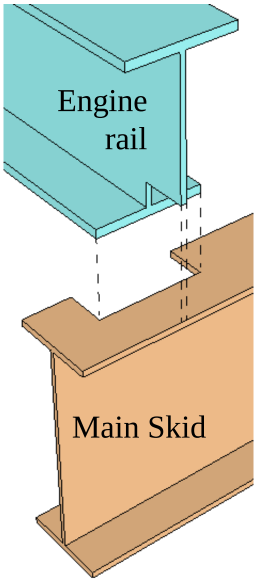


Figure 7. Engine rail design

5.2. Dynamic Skid Analysis & Consequences

This example illustrates a dynamic skid analysis, and the consequences of not following the design optimization recommendation.

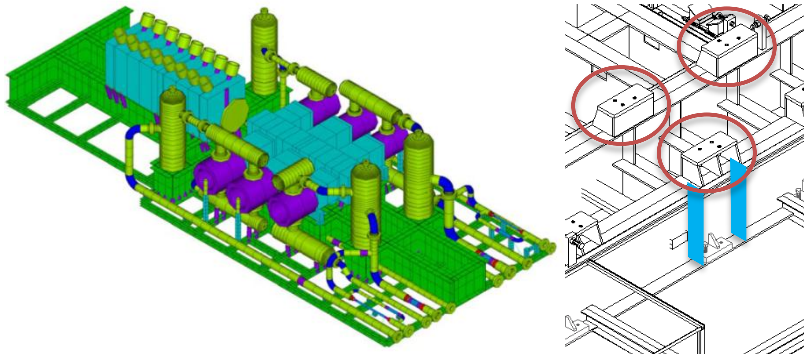


Figure 8. Compressor dynamic skid model (left) and engine mounts (right)

The skid design was created using ANSYS finite element (FE) software (Figure 8 left). The skid dynamics were evaluated and both vibration and stress amplitudes were compared to guidelines. To reduce vibrations to guideline levels, the skid design was optimized by adding additional anchor bolts in key locations and adding gussets near the engine mounts (Figure 8 right). The engine mounts are circled in Figure 8 and the recommended gussets are in blue.

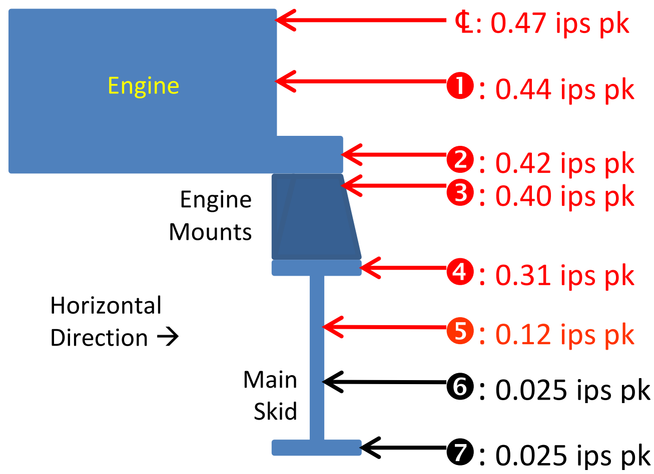


Figure 9. Skid operating deflected shape (ODS) of non-drive end of engine

The unit was commissioned and high horizontal vibrations were detected. Vibration on the non-drive end (NDE) of engine, at crankshaft height, was 0.47 in/sec peak at 16.7 Hz, which exceeded the Caterpillar engine guideline of 0.26 in/sec peak. Since the engine was running at 1000 RPM, this vibration occurs at 1x engine runspeed. Horizontal vibration was measured at seven test locations on the engine, engine mounts and main skid (Figure 9 and Figure 10). The test locations in red have vibration above BETA's skid vibration guideline of 0.1 in/sec peak. It was noted that there was a significant increase in vibration between test points ❺ and ❻. The mechanical natural frequency (MNF) of the engine was also checked, and found to be 18.5 Hz. As described in the previous case study, the engine/pedestal assembly MNF should be above 2.4x engine runspeed (40 Hz) however leaving this mode intertuned may be acceptable if the engine does not run below 620 RPM.

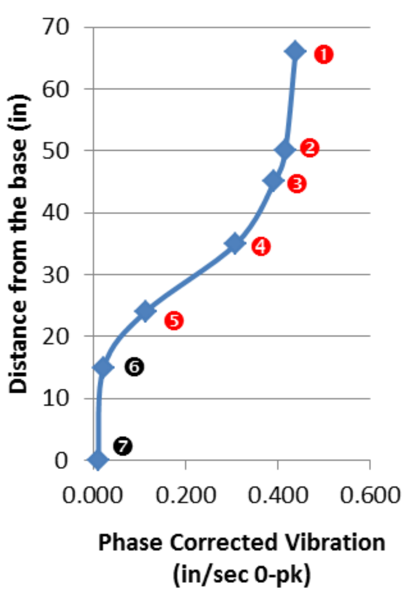


Figure 10. Plot of skid ODS

There appeared to be high flexibility along the load path from the engine to the main skid beam. Upon review, it was discovered that the recommended 0.75" thick full depth gussets near the engine mounts were not installed during fabrication. This flexibility dominated the other well designed and stiff components and thus lowered the overall stiffness of the entire engine mounting system.

6. Conclusion

Skid designs must consider static and quasi-static forces, as well as dynamic forces. In most cases, a detailed finite element analysis is required to evaluate skid designs.

The main dynamic forces on a reciprocating compressor are the unbalanced forces created by the compressor frame, as well as cylinder horizontal gas forces and vertical crosshead guide forces. Avoiding having these forces resonant with the main natural frequencies of the equipment, vessels, piping, skid, and structure/foundation will lower the chance of resonance and keep skid vibrations below the recommended spectral guideline of 0.1 in/s peak.

The dynamic forces can only be controlled if the load path from their sources to the foundation is as direct and stiff as possible.

The two examples presented in this paper describe the challenges related to the proper design of a skid support structure. They illustrate the need to properly account for the flexibility of the different parts of the equipment package skid, in order to achieve an acceptable design.

7. References

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