Torsional challenges with
a wide range of compressor performance

By

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Abstract

More and more is expected of our compressor packages. Wide speed ranges, large variations in pressures and flow, varying gas compositions and multiple unloading schemes all present challenges to a proper torsional design of a system where the operating envelope is stretched to encompass numerous operating scenarios.

This paper presents methods to meet some of these challenges and reasons to establish boundaries for the design of reciprocating gas packages. Using the "GMRC Recommended Practice for Control of Torsional Vibrations in Direct-Driven Separable Reciprocating Compressors," the paper discusses how to thoroughly define the expected operating envelope to achieve the widest possible operating range.

The limits of typical design options as well as various alternatives to avoid torsional vibration problems are illustrated with three case studies.

Introduction

A torsional vibration analysis (TVA) is essential to ensure the safe, reliable, long-term operation of reciprocating gas compressor packages. Ignoring torsional vibration at the design stage can result in catastrophic failure of couplings or crankshafts. The Gas Machinery Research Council (GMRC) has developed a guideline to help packagers and owners understand the scope of a torsional analysis. Generally, every *new* compressor and driver configuration requires a TVA. If the packager has experience with a "standard" configuration of a particular engine and compressor combination, then a TVA may not be required. However, the packager should be aware that subtle changes in the configuration or operation of the unit can produce undesirable results. This is particularly true when the intended operation includes deactivation of the head end or crank end of cylinders.

It is recommended that motor-driven compressor packages always have a TVA. This is because there usually are no "standard" motor-compressor combinations. The inertia and stiffness properties of motors can vary widely between manufacturers, and the torsional responses of different motors are not comparable even with the same power and torque ratings. In addition, motor-driven compressor packages lack damping in the system, which can result in very high torsional vibrations if resonance occurs. Engine-driven compressor package generally have higher damping due to the viscous damper on the engine which helps control the amplitude of vibration at resonance. Variable-frequency-drive (VFD) motors have the highest risk of torsional failures, as a wide speed range increases the likelihood of resonance. Particular care should be given if the packager selects a motor with a rating significantly below the rated power of the compressor, as the inherent torques from the reciprocating motion of the pistons are still present, even if they do not contribute to the overall power consumption.

While it is important to consider the full range of operating conditions, the sheer number of possible permutations and combinations can be impractical to analyze. Along with basic parameters such as suction and discharge pressures and speeds, compressors may have variable volume control pockets (VVCPs), and the design may require both single-acting and double-acting conditions.

As a general rule, if a compressor is equipped with VVCPs, designers can expect operators to use them (with or without consulting with their engineering team). As a result, the TVA analyst needs to consider a range of possible loading conditions. On the other hand, the engineering team will likely be involved when considering major changes to the compressor configuration, so the decision whether to consider single-acting operation can be made at the outset of the project.

Note: while unloading the compressor with VVCPs, spacers or single-acting conditions can lower the overall power consumed, it will not necessarily lower the torsional response. Torsional failures are often the result of resonance, which occurs when a harmonic of operating speed coincides with a torsional natural frequency. Changing the loading conditions, especially when done unevenly, can drastically alter the amplitude of torques at different harmonics, which in turn can significantly alter the torsional response.

The GMRC torsional vibration guideline provides a risk assessment table for production and engineering companies to determine the relative risk of compressor designs, considering different compressor configurations and loading scenarios.

Item	Basis	Score to Add	Enter Score for All Items Present on System
Power (Driver Rating)	HP/3000 or kW/2237	Computed	
Speed Range	Constant Speed Variable Speed	0 3	
Motor Type	Induction Synchronous (with driven inertia ≥ driver inertia	1 2	
Number of Compressor Throws	2 Throws 4 Throws 6 Throws	2 1 3	
Compressor Capacity Control	Volume Pockets One, or More, Cylinders can Single Act Infinite Step	1 2 3	
Criticality of Application	Basic Unit Unit Unspared or Essential to Operation Unit Installed in a Remote Location	2 4 4	
Experience with Similar Equipment	Little, or No, Experience with Similar Equipment Experienced (Robust System - No History of Issues) Experienced (Sensitive System - History of Issues)	1 0 3	
Total Score (If ≥ 8.0, A Torsional Vibration Analysis Is Recommended) =			

Figure 1: Risk Assessment Table - Electric Motor Drive

Item	Basis	Score to Add	Enter Score for All Items Present on System
Power (Driver Rating)	HP/3000 or kW/2237	Computed	
Speed Range	Variable Speed (Almost All are Variable Speed)	2	
Number of Engine Cylinders	4 - 8 Cylinders More than 8 Cylinders	2 1	
Number of Compressor Throws	2 Throws 4 Throws 6 Throws	2 1 3	
Compressor Capacity Control	Volume Pockets One, or More, Cylinders can Single Act Infinite Step	1 2 3	
Criticality of Application	Basic Unit Unit Unspare or Essential to Operation Unit Installed in a Remote Location	2 4 4	
Experience with Similar Equipment	Little, or No, Experience with Similar Equipment Experienced (Robust System - No History of Issues) Experienced (Sensitive Systen - History of Issues)	1 0 3	
Total Score (If ≥ 9.0, A Torsional Vibration Analysis Is Recommended) =			

Figure 2: Risk Assessment Table - Engine Drive

At a basic level, the risk assessment table can be used to determine whether or not to conduct a torsional analysis, but it should also serve as a guide to understand the risk factors for torsional vibration.

The GMRC torsional vibration guideline also provides a flow chart outlining the responsibilities for a torsional analysis:

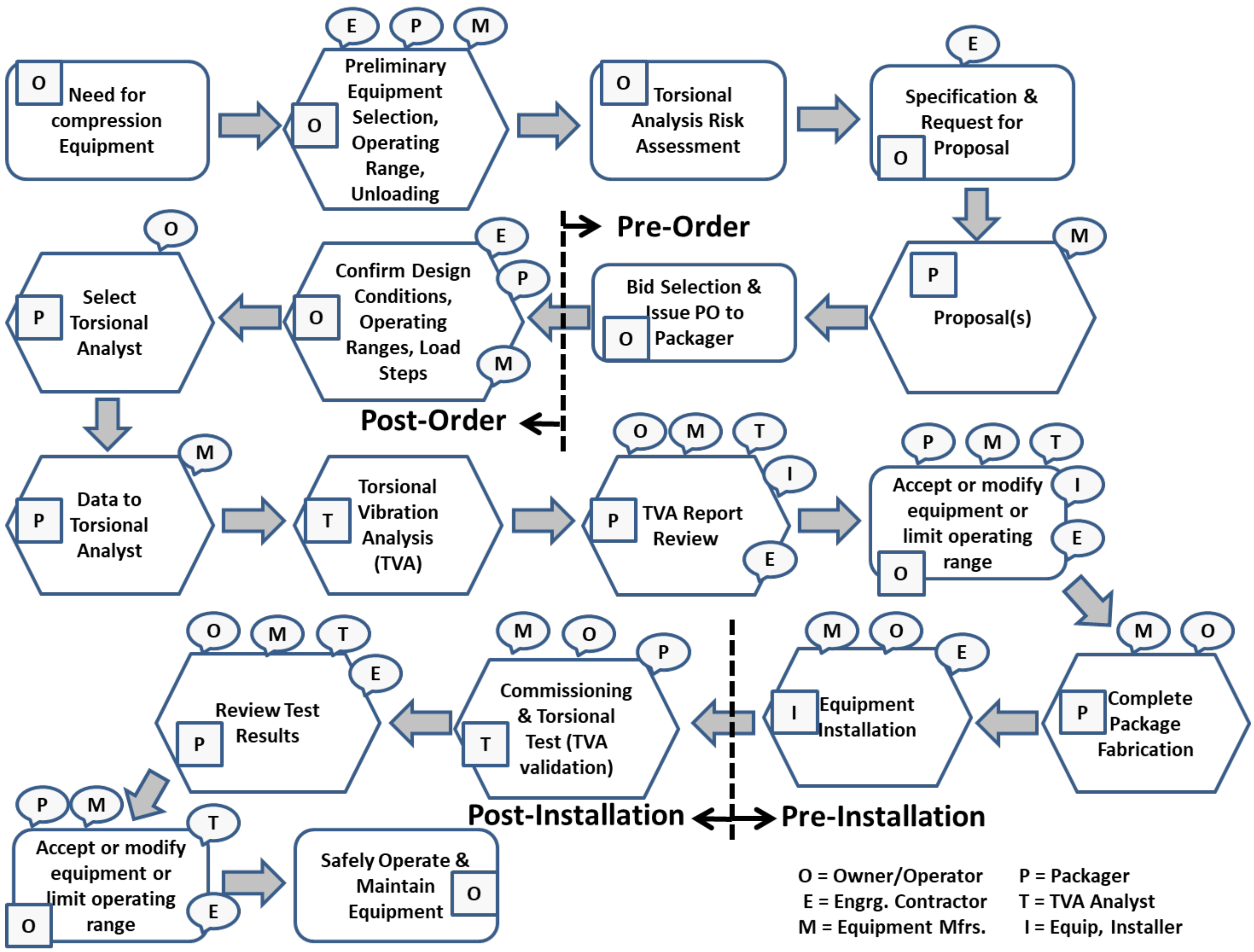


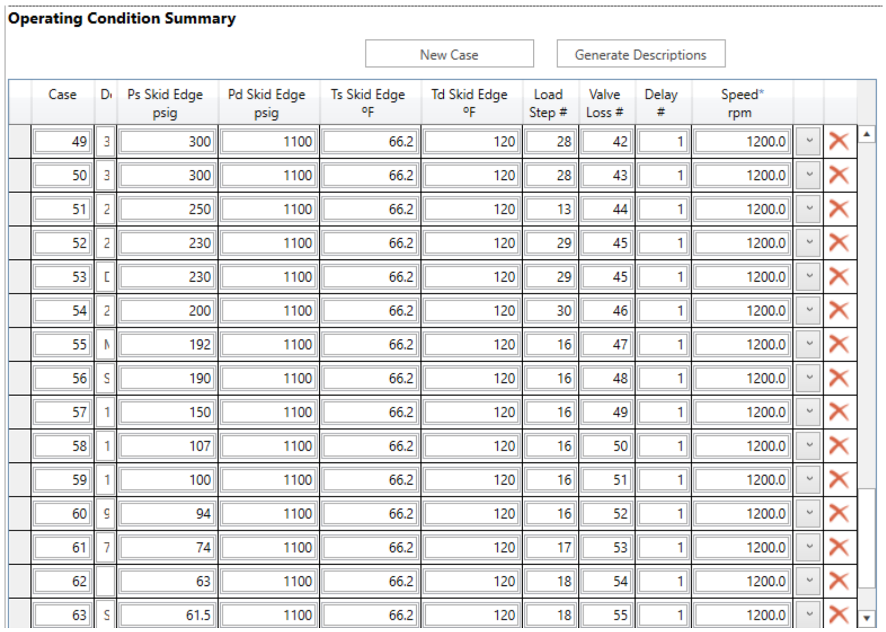
Figure 3: Torsional analysis flow chart (GMRC)

While the flowchart appears to be quite complex, it helps to illustrate who is directly responsible for each of the steps in a torsional analysis.

In this flow chart, the risk assessment should be done prior to bid requests being sent to the packagers. Furthermore, the confirmation of design conditions, operating ranges and load steps occurs after the project has been awarded but before the torsional analysis work has started. However, as demonstrated in the following case studies, not paying attention to risk factors when confirming design conditions can lead to delays in the analysis and costly solutions.

Case study 1: selecting the right analysis conditions

The packager / EPC provided 63 operating scenarios for an Ariel JGK/4 two-stage compressor driven by VFD motor.


Operating Condition Summary

Case	D	Ps Skid Edge psig	Pd Skid Edge psig	Ts Skid Edge °F	Td Skid Edge °F	Load Step #	Valve Loss #	Delay #	Speed* rpm
49	3	300	1100	66.2	120	28	42	1	1200.0
50	3	300	1100	66.2	120	28	43	1	1200.0
51	2	250	1100	66.2	120	13	44	1	1200.0
52	2	230	1100	66.2	120	29	45	1	1200.0
53	[	230	1100	66.2	120	29	45	1	1200.0
54	2	200	1100	66.2	120	30	46	1	1200.0
55	N	192	1100	66.2	120	16	47	1	1200.0
56	S	190	1100	66.2	120	16	48	1	1200.0
57	1	150	1100	66.2	120	16	49	1	1200.0
58	1	107	1100	66.2	120	16	50	1	1200.0
59	1	100	1100	66.2	120	16	51	1	1200.0
60	9	94	1100	66.2	120	16	52	1	1200.0
61	7	74	1100	66.2	120	17	53	1	1200.0
62		63	1100	66.2	120	18	54	1	1200.0
63	S	61.5	1100	66.2	120	18	55	1	1200.0

Figure 1: Operating condition summary

Some conditions were only intended to be used over a certain speed range, but with VVCPs on the two first-stage cylinders and the provision for either head-end or crank-end single-acting conditions and valve spacers on the two second-stage cylinders, the multi-run file had almost 3000 operating points per scenario for a total of 185,000 unique scenarios:

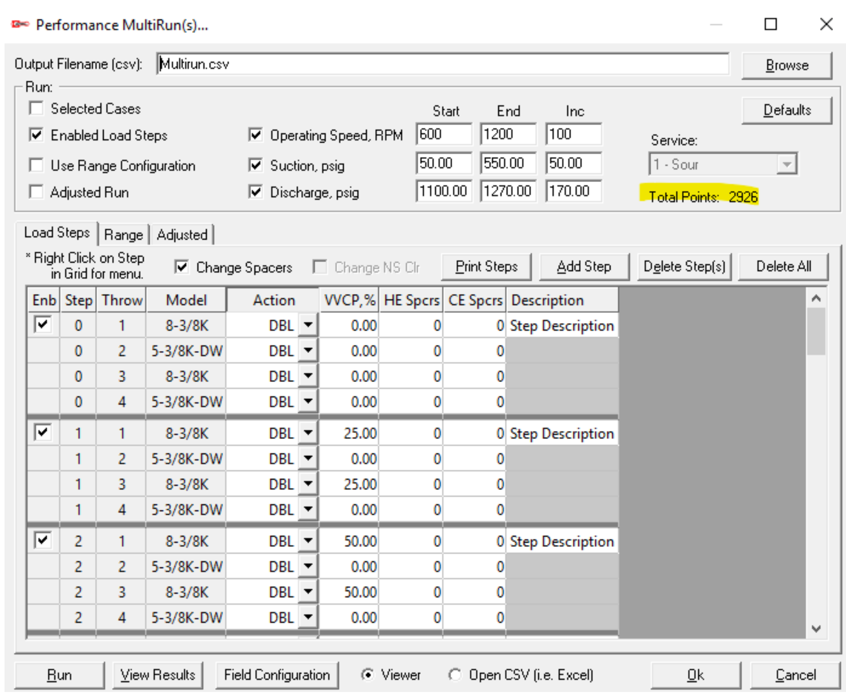


Figure 2: Multirun analysis

A number of these operating scenarios were added at the request of the packager / EPC to explore a 'possible' operating scenario, not necessarily an 'expected' operating condition.

It is important to understand that at a fundamental level, a compressor is designed to move a certain quantity of gas from a lower pressure to a higher pressure. There are many ways to finetune the operation of a compressor to get to a precise design condition but exploring every possible way of achieving a set condition creates an unreasonable number of operating conditions. There are practical considerations around how many conditions can be analyzed without producing an overwhelming amount of data. So how do we reduce the number of conditions to a more manageable set that still represents all expected operating scenarios?

One way to reduce the number of operating conditions is to simplify the scenarios by ensuring all cylinders on a particular stage are evenly loaded. Maintaining the same VVCPs for every cylinder on a particular stage balances the torques in the system. Also, capacity between single-acting on the head end vs the crank end is typically similar, and whatever small differences remain can be equalized with speed control. Thus, it makes sense to only consider single-acting on the crank end. (A head-end-only operation can lead to problems with a lack of rod reversal.)

There can be differences in the torque effort curve between single-acting crank end (HEVR) and single-acting head end (CEVR) (see Figures 1 and 2) but those differences are minor compared with the effects of only one single-acting cylinder (see Figure 3). For a four-throw compressor with a 90° phase lag between throws and reasonably balanced loads on each cylinder, the phased sum of the torque harmonics at 2x, 6x, 10x, etc. cancel each other out. If one cylinder is single-acting, however, harmonics appear at 2x, 6x, 10x. If a torsional natural frequency coincides with 6x operating speed (not uncommon), a torsional resonance will be excited, which can cause damaging torsional stresses.

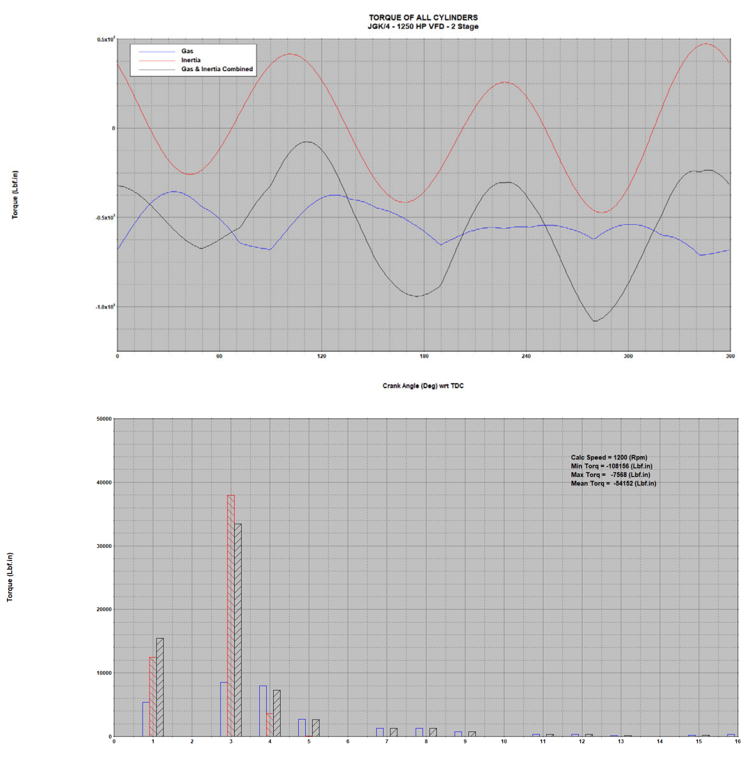


Figure 3: Single-acting crank end (HEVR)

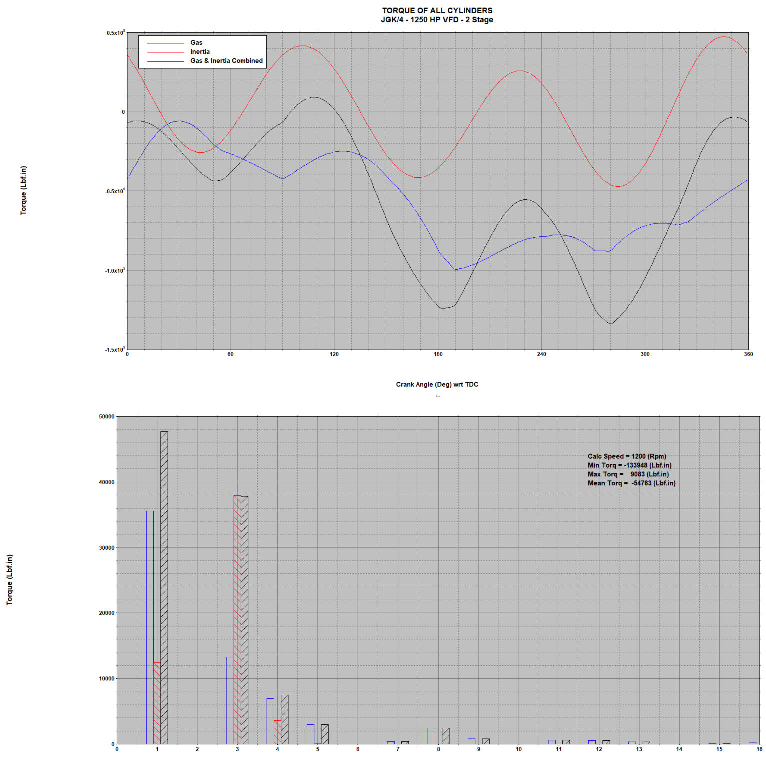


Figure 4: Single-acting head end (CEVR)

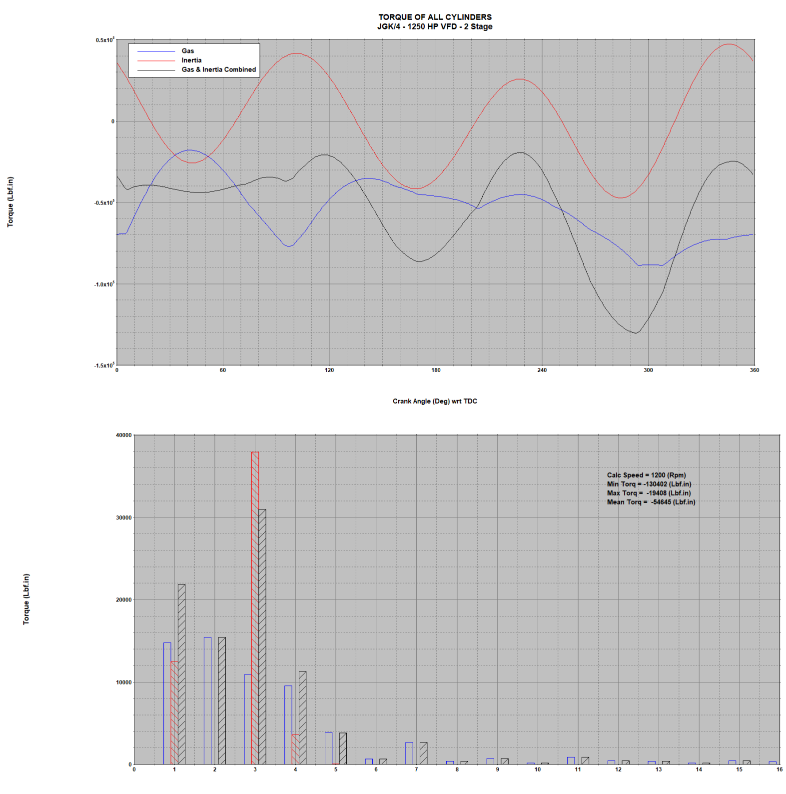


Figure 5: Only one cylinder HEVR

In the end, through consultation with the packager/EPC, the total number of operating scenarios was reduced from 185,000 to 7000. This was done by eliminating scenarios that were not considered to be realistic for operation or were redundant for capacity control as well as simplifying the loading scheme by avoiding scenarios that would likely increase the torsional vibration. However, getting agreement on the condensed set of operating conditions resulted in a delay of several weeks in the analysis.

Case study 2: expanding the operating conditions to identify the worst scenario

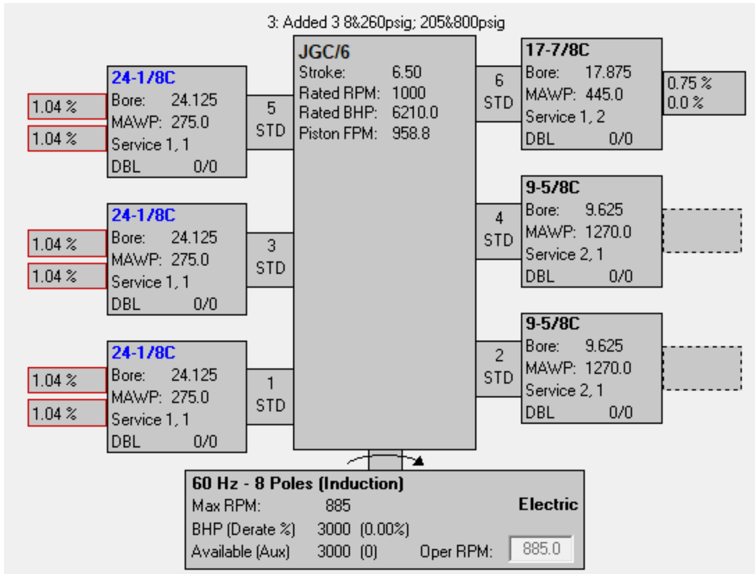


Figure 1: JGC/6 three-stage, two-service

This case involves an Ariel JGC/6 compressor (three-stage, two-service). The problem with two-service machines is when the two services are allowed to be operated independently. Since the torsional response of the whole system is determined by the phased sum of the torques of each individual throw, the range of conditions consists of all possible operating parameters of service 1 multiplied by all possible operating parameters of service 2. In this case, the 8 design points that were provided for the analysis grew to be 64 design conditions to be analyzed.

For this system, the highest torques occurred at a condition not included in the original eight design conditions. The maximum predicted torques in the coupling for the worst-case condition was 10% higher than any of the base case scenarios. By expanding the range of possible operating conditions, the worst condition was identified, and the analysis proceeded from there.

The GMRC torsional guideline does not explicitly include multiple service machines in its risk assessment process. That is an oversight that ought to be corrected in future editions.

Case study 3: operating a compressor well outside its intended application

A torsional analysis was required on an existing Ariel JGK/4 three-stage compressor that was greatly oversized for the application. Because of production declines, the unit was only running at about 25% of the engine-rated horsepower. In order to reduce capacity through the compressor, the head ends of all cylinders had been deactivated and the unit was slowed down to its minimum 800 rpm.

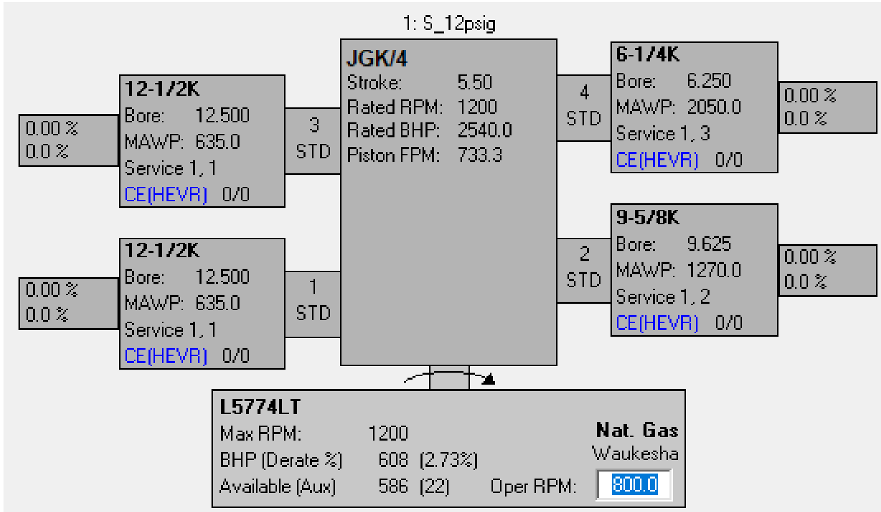


Figure 1: JGK/4 three-stage unloaded

Based on results from the TVA, the cylinder on throw 1 was deactivated and replaced with a dummy cylinder. This improved the torsional response; however, the unit is still operating well below its intended capacity. While a torsional failure was avoided in the short- to medium term, continued operation at partial loads and reduced speed will likely accelerate the build-up of carbon deposits in the engine cylinders and affect the unit's long-term reliability.

Case study 4: how to correct for torque imbalances

A torsional analysis was done on a Dresser-Rand 6HOS6 compressor driven by a Waukesha 12V275 engine from 750-1000rpm. The initial results showed that the stresses in the compressor shaft and the torques in the coupling were higher than acceptable when operating at a resonance speed.

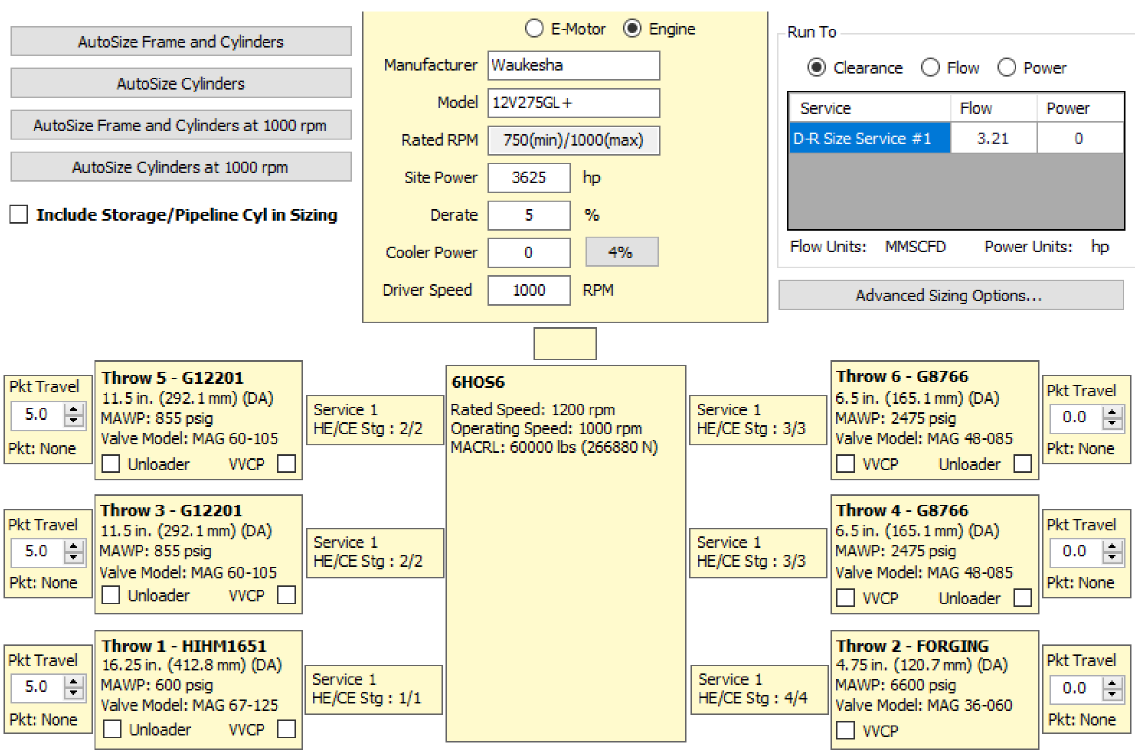


Figure 2: Compressor configuration

The high torques and stresses in the system were primarily due to imbalances in the power (and hence torques) distribution between the throws:

Stage number at (HE/CE)		1/1	2/2	2/2	3/3	3/3	4/4
Frame Ext/Cyl. Bore	#/IN	1/16.250	3/11.500	5/11.500	6/ 6.500	4/ 6.500	2/ 4.750
Cylinder Pattem		HIHM1651	G12201	G12201	G8766	G8766	FORGING
RDP	PSIG	545.0	775.0	775.0	2250.0	2250.0	6000.0
MAWP	PSIG	600.0	855.0	855.0	2475.0	2475.0	6600.0
Cyl. Action/HERod Dia.	Act./IN	DA/ 0.00	DA/ 0.00	DA/ 0.00	DA/ 0.00	DA/ 0.00	DA/ 2.50
Brake power/ Cylinder	HP	267.4	628.7	628.7	585.6	585.6	640.9
Capacity Wet/ Cylinder	MMSCFD	3.14	8.92	8.92	8.92	8.92	17.84

Figure 3 - Horsepower summary

A 6-throw compressor should normally produce torques only at 3x, 6x, 9x, etc orders of operating speed. However, if the torques are not evenly balanced between throws, additional harmonics will appear at other orders which can lead to problems if those orders happen to coincide with a torsional natural frequency of the system producing a resonance.

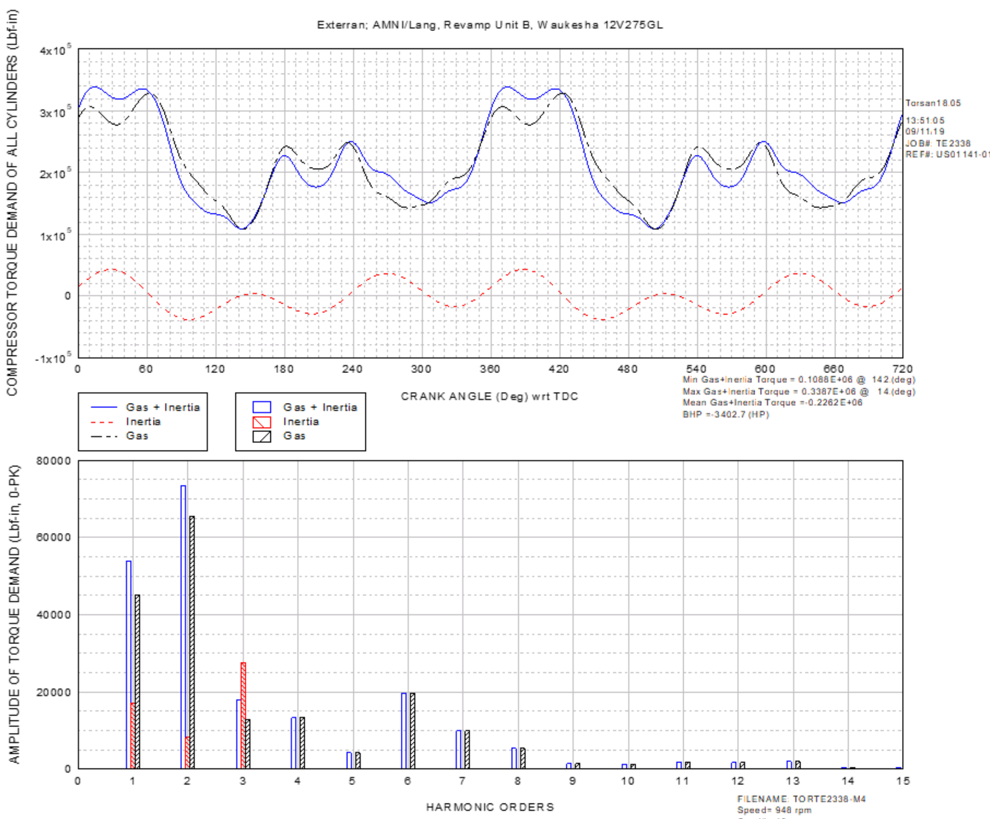


Figure 4 - Torque effort plot

A number of possible solutions were evaluated, but in the end, it was decided to swap the position of the cylinders. Moving the first-stage cylinder on Throw #5 to Throw #1 improved the torsional results. The reason for this is due to the fact torques acting near the 'free' end of the compressor have a greater effect than torques near the center. Referring to the mode shape of the response at the first TNF, the node of the system is close to the coupling. Torques acting at a node have very little effect on the system. That is why changing to unloading cylinder #1 first produced better results. The unevenness of the torques was mitigated by the fact that those torques were acting near a node.

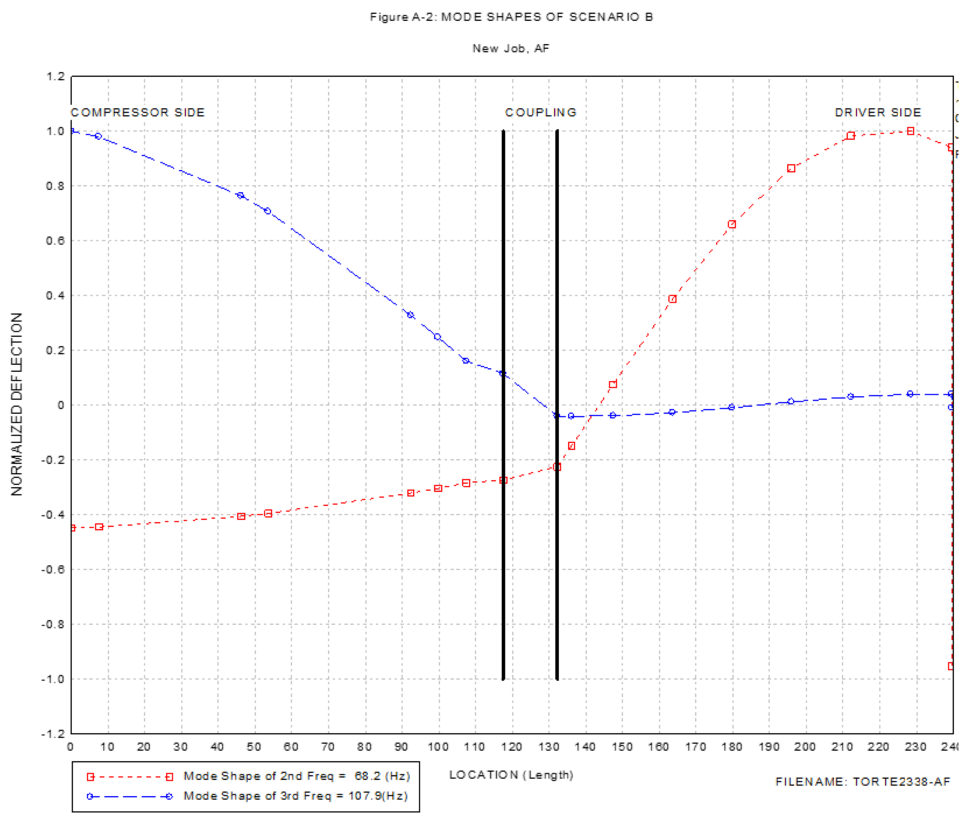


Figure 5: Torsional mode shape

Conclusions and recommendations for owners, packagers and EPCs

The main challenge with designing compressors for a wide range of operating conditions is to identify what conditions represent a reasonable operating envelope. This can be done by eliminating redundant conditions, or, alternatively, by creating additional conditions that were not considered in the original operating set. Furthermore, operating an existing compressor well outside its intended parameters can reduce its long-term reliability.

When selecting analysis conditions, ensure they represent likely operating scenarios. When considering unloading scenarios, reduce torsional vibration risks by unloading cylinders evenly, or, if that is not possible, unload them from the drive end to the auxiliary end.

The GMRC risk assessment process does not currently include a risk factor for multiple service operation. This should be corrected in future editions.

If there is a variable-speed driver, using speed control rather than unloading schemes may be a more reliable option. Lastly, understand the risk factors for torsional vibration and use that information to refine the unit's operating envelope during the initial design to eliminate potentially risky operating scenarios and ensure a smooth torsional analysis.

References

GMRC Guideline and Recommended Practice for Control of Torsional Vibrations in Direct-Driven Separable Reciprocating Compressors – June 2015