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Torsional Vibration Case Study Highlights Design Considerations

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Abstract:

A large compressor installation in a remote location of Russia experienced rapid crankshaft failures. Due to the very high costs and logistics involved, a root cause failure analysis was conducted. Field measurements obtained at site identified an unexpected situation where the first two torsional natural frequencies occurred at essentially the same speed. This resulted in a double resonance condition – something very rare in the field of Torsional Analysis. Detailed modeling using the field data confirmed the cause of the failure and was instrumental in finding a quick solution.

This technical paper highlights a number of important considerations for the torsional system in new or revamped compressors, as well as design philosophies to assess and mitigate the risk of torsional failures.

1 Introduction

The authors of this paper represent the manufacturer of the compressor, the packager of the unit, and the third party consultant that provided assistance to solve the problem.

The unit was a reciprocating compressor driven by a motor, as described in more detail below (the delivered unit):

- fixed speed, induction motor, 1500 RPM synchronous speed, 1480 loaded, 500 kw;
- shim pack coupling;
- flywheel (32,724 lbf in^2, original design);
- horizontally opposed reciprocating compressor, 4 throw, 3 stage.

The cause of failure was clearly torsionally-induced alternating stress. Refer to Figure 1 for a view of the failed surface at the drive end of the compressor. The classic 45 degree crack is consistent with a torsional failure. The crack appears to have initiated at the end of the keyway, close to the flywheel. Having the keyway causes a slightly greater stress concentration than without (2.85 versus 2.2).

Figure 1: Broken Shaft End

It would be logical to assume that the presence of the keyway was the cause of the failures. Evidence given below shows that the presence of the keyway is not sufficient to explain the failures. Stresses predicted for the original system without the keyway are still high enough to cause a failure. By way of explanation, stresses will be high at a node (defined as a point of zero motion). Typically, a node for the torsional deflections and, therefore, a plane of maximum stress, will occur near the flywheel. Refer to the mode shape plot in Figure 2 (shown for the model tuned to field conditions for Scenario B). A node occurs for the second mode close to the flywheel.

The sharp change in the slope of the second mode line at the flywheel also suggests a stress maximum is present at that location. The actual stress contribution depends on amplitudes of vibration at or near the second mode. The node for the first mode is closer to the motor, but there is a significant change in slope for the first mode at the flywheel, indicating that a significant contribution to stresses at the failure location could also come from torsional vibrations at or near the first mode.

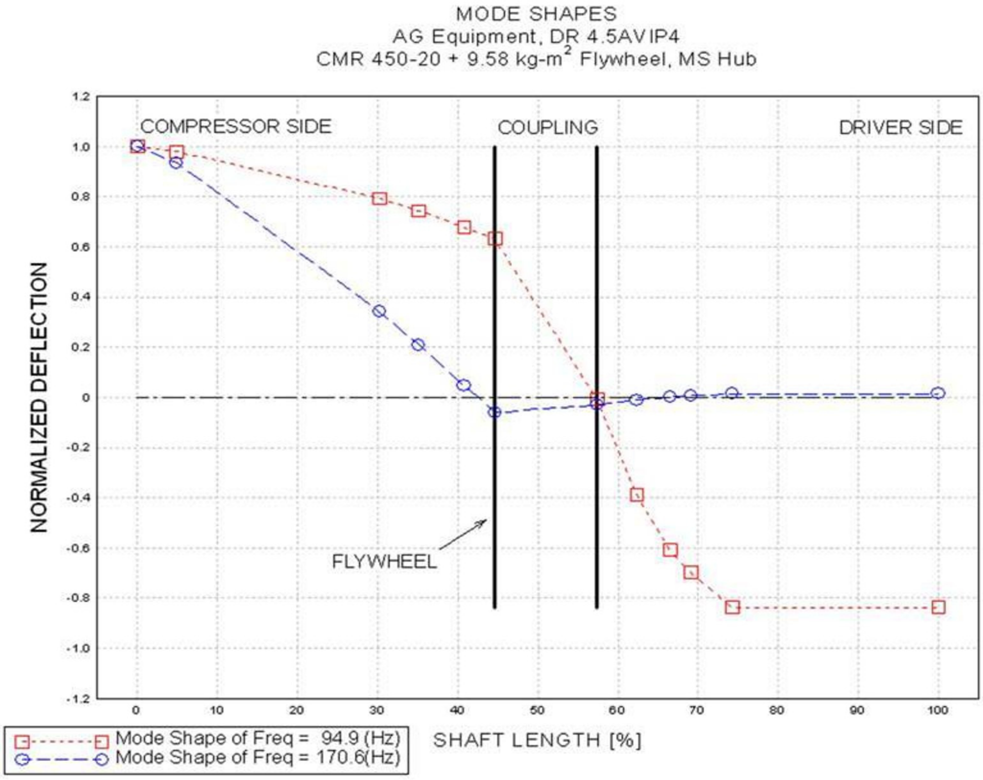

Figure 2: Mode Shape Plot

2 Case History

2.1 Original Design and Operation Resulting in Failures

The original design analysis resulted in a system with a flywheel on the compressor side of the coupling. The calculated torsional natural frequencies (TNFs) were placed between 3 and 4 times run speed for TNF1, and between 7 and 8 times run speed for TNF2. See Figure 3.

Despite the predictions of non-resonant performance, the system design factors (DFs) were assessed assuming that resonance would occur. This is a conservative design philosophy that is usually sufficient to avoid torsional failures. The stresses assuming resonance at TNF1 or resonance at TNF2 were found to be acceptable. Since the expected operation was not going to be resonant at either TNF, the design was accepted.

In due course, the system was packaged and shipped to Siberia for installation and startup. The first failure occurred after 375 hours of operation. (Note that many of those hours were under very light load while the process was being purged.)

After the failure, it was noted that there was a keyway in the compressor crankshaft. This had not been intended, but the DF with the keyway was still very good. It was decided that the failure was likely a rogue failure and a replacement crankshaft was shipped to site and installed. The replacement also had a keyway, but was considered acceptable as no alternative spare was available in a short timeframe and since calculations indicated that the keyway was not critical.

The second failure occurred after another 111 hours of operation (about 32 hours were under full load).

2.2 Post-Failure Design Audit

A third crankshaft was shipped to site. This one did not have a keyway. However, in an effort to prevent a third failure, a second opinion was requested from a consulting engineering organization with extensive experience analyzing torsional response of reciprocating compressor systems.

The first observation from the design audit was that the first TNF was predicted to be resonant at 4 times run speed. See Figure 3. (Note that the analysis methods used by the consultant included tolerances for system stiffnesses and inertias, as described in papers found in the Bibliography at the end of this paper. Scenarios A, B, and C are calculated for the low, the average and the high natural frequencies based on the tolerances for stiffness and inertia.) The apparent reason for the difference in the predictions of TNF1 was the way the motor rotor was modelled. However, even at resonance for the first mode, the design factor (DF) for the system was acceptably high.

The second TNF was not observed to be close to resonance. Note that in both models (original design and design audit) there was no added inertia for the oil pump. The oil pump used in this model of compressor is driven directly through a coupling at the outboard end of the compressor crankshaft.

In a normal modelling job in the design stage, the reaction to having predicted operation at resonance would have been to change the system to avoid this resonant condition. Uncertainties associated with

- defining damping values,
- defining the torque effort from the compressor under upset conditions, as well as
- the unknown level of acceptable torsional vibrations for auxiliary equipment, such as motor fans and compressor oil pumps,

generally lead to a conservative design philosophy of avoiding resonant operation, if practical, on fixed speed motor driven units.

In this case, however, due to the lack of a clear indication from the design audit models as to why the failures had occurred, it was concluded that a visit to the field to measure the actual TNFs and amplitudes of torsional vibrations in operation was required to determine the root cause of failure.

Due to the remoteness of the site and the delays that would have ensued getting travel documents, it was decided that the consultant would train and equip the packager's representative (already possessing travel documents) to collect the required data. The data were transferred via electronic mail to the consultant's office for analysis.

3 Field Measurements and Design of Solution using Tuned Model

Field data were collected by the packager's representative. A team effort, including the end user's personnel, the packager's representative and the consultant, was required to overcome various hurdles. For example, the only location available to measure torsional response with the test equipment available was the opposite drive end (ODE) of the motor. A special adapter had to be manufactured in the field to accomplish this measurement. The compressor ODE was not available due to looseness in the coupling (by design) between the crankshaft and the oil pump.

The first TNF was measured close to resonance at the 4th order of run speed - consistent with the design audit model (Scenario C, see Figure 3). However, the second TNF was found to be close to resonance at the 7th order – not consistent with the audit model.

An interesting aside: It was observed that TNF1 was about 3 Hz (2.8%) higher when the system was started cold, compared to starting the system in a "warm" condition. Consistent with this observation, the amplitude of torsional vibrations at 4th order increased as the system warmed up under load. These behaviors are consistent with the motor rotor being stiffer when cold – an observation that has been made in the past by the consultant's personnel.

The results from the field measurements immediately pointed toward the idea that having both TNFs resonant at the same speed would lower the design factor for the system. How to model this simultaneous resonance was a problem.

Experience with modelling other compressor packages suggested that the addition of inertia for the oil pump might correct the second TNF discrepancy between field and model. Addition of a representatively small inertia to the model at the compressor ODE lowered the predicted second TNF to resonance with the 7th order without changing the first TNF. The design factor with both TNFs resonant at the same speed was at a failure level (0.9 with and 1.1 without the added stress concentration due to a keyway!). The technical explanation of the failures (the root cause) was now understood.

Using the tuned model, the size of the flywheel was changed to see if resonance could be avoided at both TNFs. In fact, both modes were sensitive to the removal of inertia from the flywheel. The flywheel inertia was reduced in the model to about 50% of the original flywheel size. This change was predicted to move the TNFs midway between orders and raise the design factor above the target minimum of 2 for all conditions.

This change was implemented in the field and no crankshaft failures have been reported since.

Figure 3 below summarizes model and field TNF and DF results for the three project phases described in the case history.

Note the design factors for the design audit (Phase 2). The DF goes up as the predicted TNF1 approaches resonance. At the same time, the TNF2 gets further away from resonance. This is an indication that resonance at the second TNF is of more concern than at the first.

For Phase 3, the design factors are given for Scenario C, which was the worst for the tuned model.

Design Phase	Design Factor*2	TNF1 (Hz/Order*3) 1480 rpm	TNF2 (Hz/Order*3) 1480 rpm
Phase 1 design before construction			
Design	2.4	88 / 3.6	186 / 7.5
Resonant at TNF1*1	2.7	Resonant at 4X 1320 rpm	
Resonant at TNF2*1	2.0		Resonant at 8x 1394 rpm
Phase 2 design audit after two failures			
Scenarios A/B/C	2.1 2.2 2.2	89.8 / 3.64 94.4 / 3.8. 98.1 / 3.98	183.0 / 7.42 186.8 / 7.57 189.1 / 7.66
Phase 3 field measurements and tuned model			
Field data*4	----	105/ 4.25(C)*4 102 / 4.1(W)*4	177 / 7.17
Tuned Design*6 As Built Scenario C	0.9 1.1*5	98.6 / 4.0 98.6 / 4.0	172.7 / 7.0 172.7 / 7.0
Modified *6,7 Scenarios A/B/C	2.0*5 2.1*5 2.2*5	103.0 /4.17 107.8 / 4.37 111.7 / 4.53	176.9 / 7.17 180.6 / 7.32 183.0 / 7.42

Figure 3: Model and Field TNF and DF Summary

Notes for Table in Figure 3:

*1. "Worst case" calculation assuming one TNF is resonant at the nearest order of run speed.

*2. Design Factor (DF) is the minimum for the compressor crankshaft, based on overall stresses. DF > 2 is good, and DF <1 is a guarantee of failure. Unless otherwise noted,

- a keyway was included in the design factor calculation,
- a flywheel (inertia = 9.58 kg-m^2) was present at the compressor drive end, and
- no inertia was present to represent the oil pump at the compressor outboard end.

*3. An "Order" is an integer multiple of run speed. Run speed is 1500 to 1480 rpm.

*4. Measured TNFs reported here were at "cold" (C), (ambient), and "warm" (W) temperatures.

*5. A keyway was not included in the stress calculation for this model.

*6. An inertia representative of the oil pump at the outboard end of the compressor was added to the model.

*7. The flywheel inertia was changed in this model to 4.83 kg-m^2)

4 Philosophy for Deciding to Perform a Torsional Design Study on a New Unit

End users and packagers of reciprocating compressor systems need to make the decision whether or not to perform a torsional dynamic design study for each new compressor package that is ordered.

The following considerations are offered from the consultant's experience with fixed speed electric motor driven reciprocating compressors. If the answer to any of the following questions is yes, then a design study should be performed.

1. **Is the unit critical?** A definition of a critical unit is required. It is suggested that the cost of one day of down time or lost production ($lp) is compared to the cost of performing the design study ($ds). Usually, this ratio ($lp/$ds) is greater than one. Problems caused by torsional issues with a unit will cost many days of lost production, as well as other costs. Therefore, a conservative definition of a critical unit would be one for which the above mentioned ratio is greater than one. Remote sites will always have a high cost per day of down time, if only for the repair process. New installations are likely to have subtle differences that only a torsional study can identify.

2. **Is the new package a different combination of compressor and driver from what has been used in the past?** For example, a change in motor manufacturer between an existing package and a new package could justify a model study, even if the motor is nominally the same size. There are usually subtle differences.

3. **If the new package is not a new combination, are there significant differences in cylinder size, staging, operating conditions, or load steps, compared to existing packages?** A definition of significant is required. How can the significance of the differences be determined? One way is to compare the predicted compressor torque effort of the new package to that of the existing package. Both overall torque and individual orders of torque should

be compared with an emphasis on torque orders close to the first and second TNFs. A simple ratio of the old to new torque efforts times the DF of the existing system will be useful as a guide.

4. **If none of the other questions have been positive, are speeds going to be different for the new package, compared to the existing package?**

5 Conclusions

Tolerances associated with fabrication cannot be avoided and must be considered in the design stage when tuning of TNFs is required.

In the design stage, it is important to make "worst case" assumptions to avoid bad surprises after startup. In the case study provided, a failure would have been predicted in the design stage if a "worst case" design approach (a model with resonance occurring at both TNFs at the same speed) had been used.

Clients who have not had experience with torsional failures may be critical of the excessive conservativeness of the "worst case" approach to design. This conservativeness has to be weighed against the huge costs from lost production, problem identification and repairs that can result after a failure.

If predicted torsional problems are

- expensive to eliminate as a possibility in the design stage,
- destructive if they do occur, but
- easily corrected in the field once the exact response is known (for example, by addition of mass rings on the crankshaft to avoid resonance),

then, torsional vibration measurements at startup should be performed.

The cost-effectiveness of this design philosophy must be judged on a case by case basis. Consideration should be given to the cost of

- an extended outage in the event of a failure, versus
- making expensive design changes (typically a soft coupling), versus
- torsional measurements at startup with fine-tuning at site, as required.

The actual system can turn out to be satisfactory after torsional measurements are made. Then, no additional costs beyond the testing will be incurred. However, if a condition that will lead to failure is detected at startup, relatively inexpensive changes (compared to the cost of a failure) can be implemented.

As demonstrated in the case study;

- torsional systems are complex and sensitive to small errors,
- the feedback of data from field measurements to designers can lead to improved design methodology which will help prevent future problems, and
- cost-effective ways are available to allow packagers and end users to perform baseline startup checks instead of waiting for expensive failures to occur.

The job of the designer should be to identify those systems for which the expense of a torsional analysis check at startup is justified. This job can be accomplished by assessment of system tolerances and by using "worst case" analysis methods.

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