MOTOR COOLING FAN FAILURES SOLVED WITH MODAL AND FINITE ELEMENT ANALYSES

Dr. Jae-Yeol Park[1], Dong-Ju Lee[1], Yong-Jae Jang[1], and Troy Feese[*2]

[1]Samsung Engineering
Seoul, Korea

[2]Engineering Dynamics Incorporated
San Antonio, Texas, USA

Abstract

Various problems occurred on two new synchronous motor – reciprocating compressor units. Torsional testing indicated that the first torsional natural frequency (TNF) of the system was coincident with the 8th compressor order and responsible for a compressor crankshaft failure covered in reference [1]. This paper focuses on failures of the motor external cooling fan and the solution method used to improve the system. At the worst compressor load condition, torsional oscillation of the motor fan had elevated levels at 8× running speed. Modal testing of the motor fan identified a second torsional mode that could be excited by the 12th compressor order. An increase of the compressor flywheel inertia was recommended to detune the first TNF but was predicted to cause insufficient separation margin between the second TNF and the 60 Hz electrical line frequency (11× running speed). An inertia ring was sized using finite element analysis (FEA) and added to the motor external cooling fan to increase the separation margin. Follow-up torsional tests of the modified system confirmed adequate separation margins for the first and second TNFs. The speed fluctuations measured at the modified external cooling fan were significantly reduced and acceptable throughout the entire compressor load range.

INTRODUCTION

Two new hydrogen compressor units, equipped with stepless capacity control, are driven by 22-pole synchronous motors at 327 RPM (5.45 Hz). The motor shaft is bolted directly to the flywheel and compressor crankshaft with a flanged connection (no coupling). The motor has a single outboard bearing. The cooling fan that failed was located on the non-drive end of the motor. The aluminium fan blades failed as shown in Figure 1.

A variable frequency drive (VFD) is only used as a soft starter for the motor. Once at synchronous speed, the motor is switched to across-the-line for constant speed operation and the power factor is automatically controlled. Therefore, significant torsional excitation does not occur during start-up, and synchronous motor excitation was ruled out as a potential cause of the fan failures.

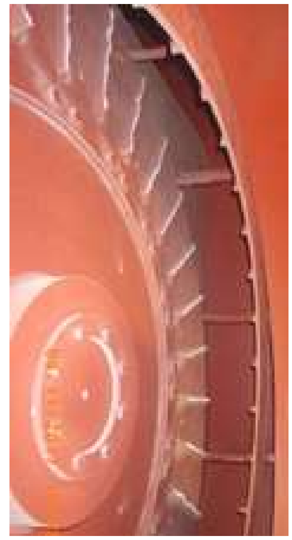


Figure 1 – Failed fan blades

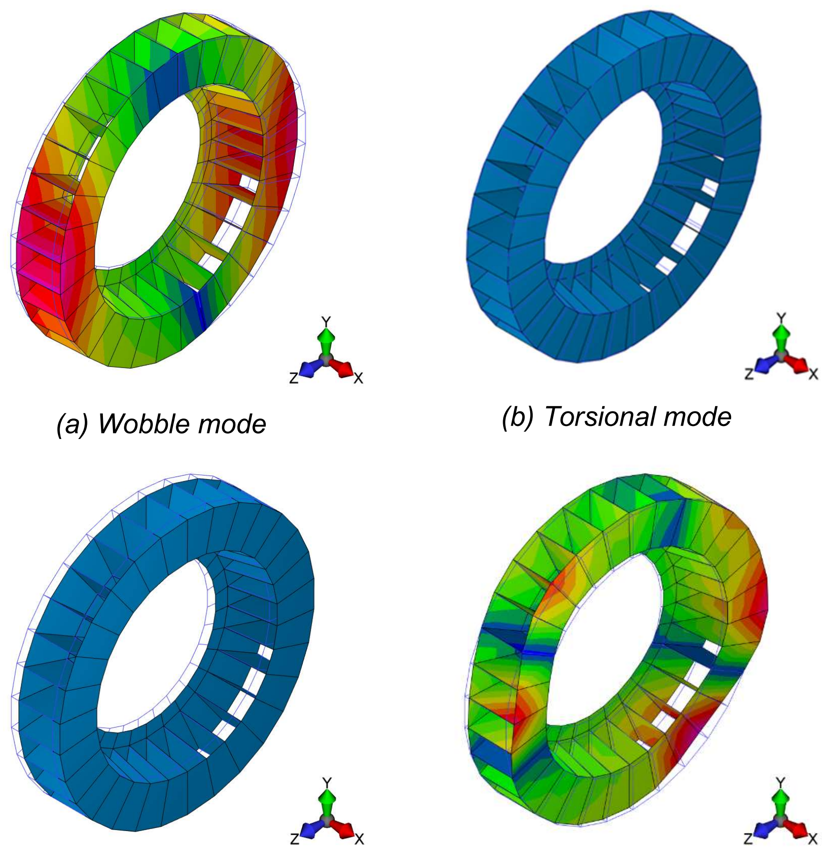
IMPACT TESTS

Figure 2 – External motor cooling fan

With the unit down and locked out, the fan cover was removed (Figure 2) so that the external fan could be tested in situ. Impact tests [2,3] were performed to measure the natural frequencies of the fan. This was accomplished using an instrumented impact hammer and response accelerometer. A small teardrop accelerometer was attached with wax and moved to various locations.

The large aluminium fan had long ring times after each impact indicative of very light damping. An Impact Mode Shape (IMS) model was developed, which utilizes data acquired from many points around the fan to animate the impact response. The scaled model of the fan was created using the Clear Motion™ software package. The IMS data were then input and processed by the software to visualize the relative motion at each of the response frequencies.

Results of several fan modes are shown in Figure 3. Note that the amplitude of motion has been exaggerated for clarity. Colors are used to help visualize the mode shapes shown in Figure 3 and do not represent stress levels.



(a) Wobble mode *(b) Torsional mode*

(c) Umbrella mode *(d) 2-diameter mode*

Figure 3 – IMS results of external cooling fan

Table 1 summarizes the mechanical natural frequencies (MNFs) identified from the impact testing. Typical modes for a large fan include wobble, torsional, umbrella and 2-diameter. The second MNF was of concern because twisting motion would be consistent with the blade failures. In addition, API [4,5] specifies that TNFs have sufficient separation margins from significant compressor orders and the electrical line frequency (60 Hz).

Table 1 – Measured natural frequencies of external cooling fan

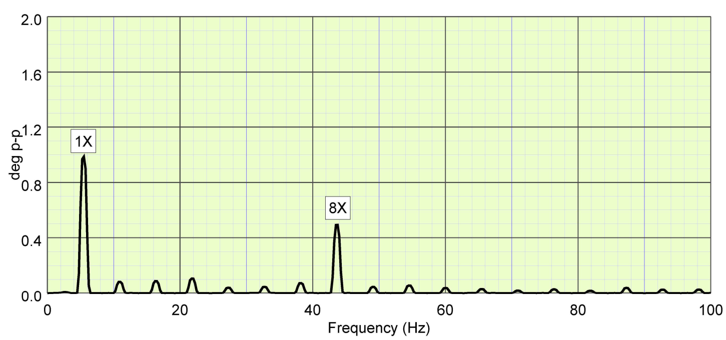
Frequency	**Speed Order**	**Mode Shape**
36 Hz	6.6×	Wobble
64.75 Hz	11.8×	Torsional
78 Hz	14.3×	Umbrella
88 Hz	16.1×	2-Diameter

LOAD TESTS

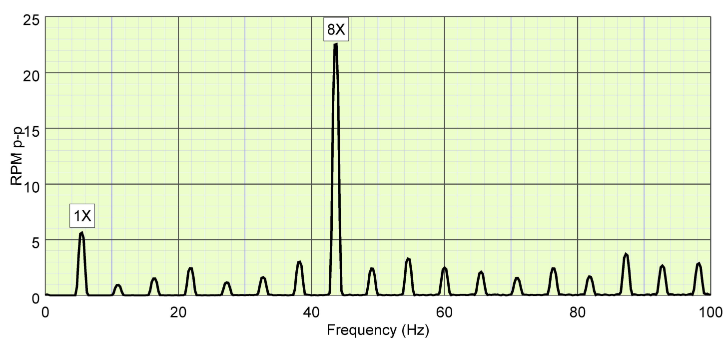
After completing the impact tests, the compressor was instrumented for operational testing. Motor shaft angular deflection was measured near the external motor fan using a high-speed laser and "zebra tape" as shown in Figure 4. These data were gathered as the compressor load was varied from 20% to 70% using the stepless capacity control system.

At 40% compressor load, the measured angular oscillation had significant response peaks at 1× and 8× running speed, with 8× running speed being the second largest order as shown in Figure 5a. However, when converted to velocity, 8× running speed becomes the highest order with over 22 RPM p-p of speed fluctuation (Figure 5b). This exceeds the maximum recommended speed fluctuation of 1.5%. Note that the oscillation data measured on the motor shaft would not necessarily indicate if any of the fan natural frequencies identified from the impact tests were being excited and creating increased vibration of the fan components.

Figure 4 – Laser and zebra tape installed on the motor shaft between bearing and external cooling fan



(a) Angular oscillation of motor fan



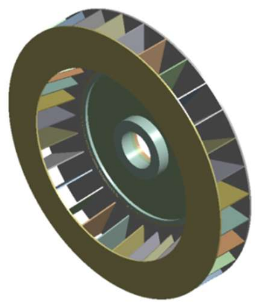
(b) Speed fluctuation of motor fan

Figure 5 – Frequency spectra at 40% compressor load

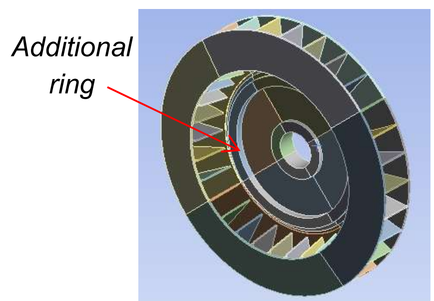
FINITE ELEMENT ANALYSIS

A finite element (FE) model was created as shown in Figure 6. The external cooling fan was modelled in two ways: 1) as an independent component and 2) as part of the overall compressor torsional system. Unforced modal analysis was performed. The results in Figures 7 and 8 show agreement with the impact tests when the fan was modelled as part of the overall system. Note that colors are used to help visualize the mode shapes and do not represent stress levels.

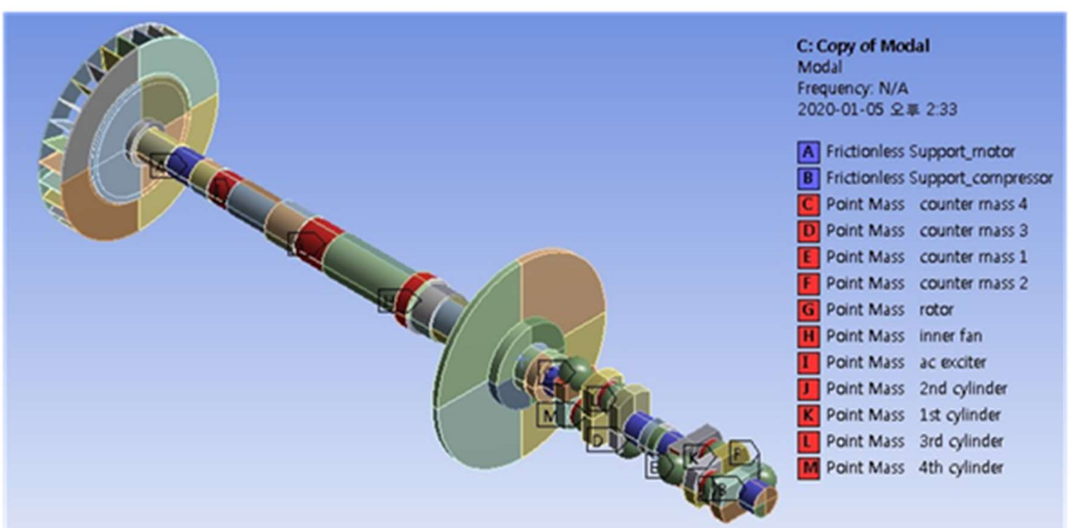
The results from the cooling fan only model were significantly different than the measurements. Without the shafts included, the predicted natural frequencies of the fan were higher. It is therefore necessary to include the motor and compressor shafts to simulate proper boundary conditions when evaluating the dynamic design of the cooling fan. The predicted natural frequencies are summarized in Table 2.



(a) External cooling fan (original)

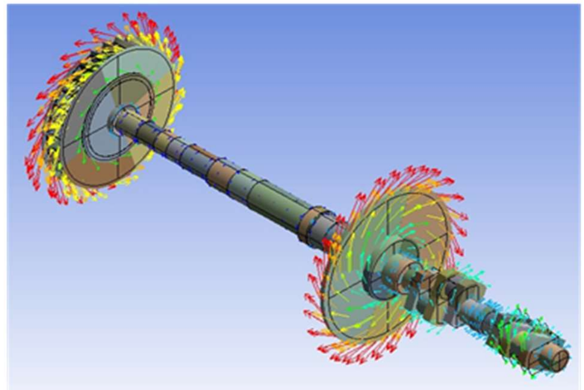


(b) External cooling fan (modified)

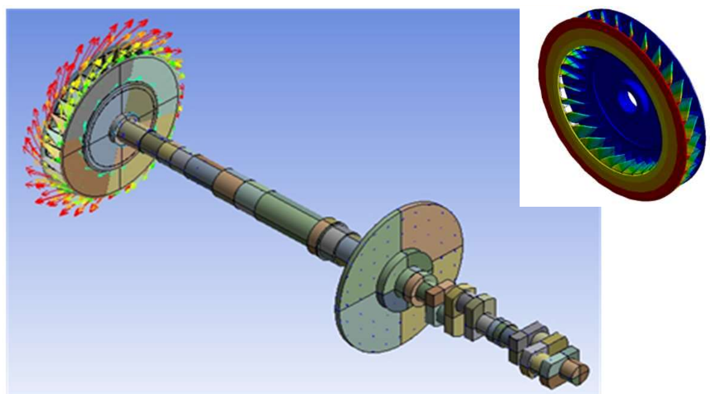


(c) Torsional system consisting of crank and motor shaft with additional inertia elements including external cooling fan

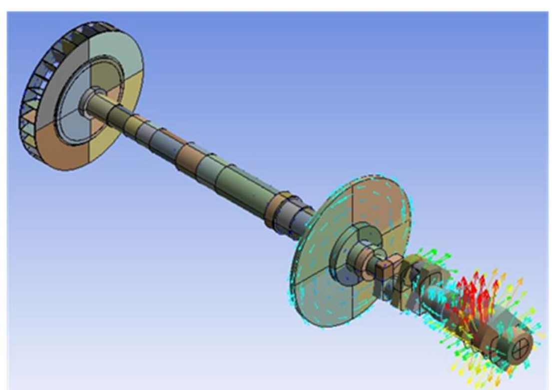
Figure 6 – Finite element model of the entire torsional system



(a) 1st torsional mode at 45.0 Hz (40.0 Hz after modifications)

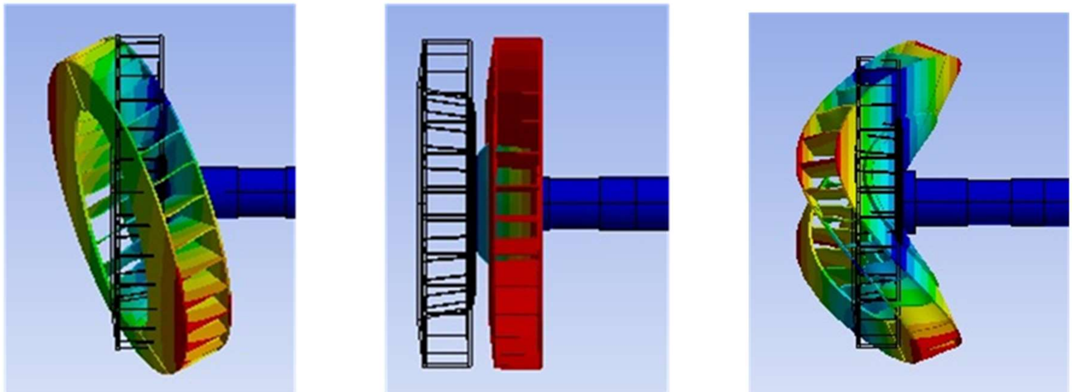


(b) 2nd torsional mode at 58.4 Hz (56.1 Hz after modifications)
(Torsional mode of blades with isolated fan model at 88.7-88.5 Hz)



(c) 3rd torsional mode at 72.3 Hz (69.8 Hz after modifications)

Figure 7 – Torsional modes predicted in the original and modified compressor system (additional inertia from GD^2=7500 to 12000 kg-m^2 and from GD^2=1560 to 1900 kg-m^2 applied to the modified system for flywheel and external cooling fan, respectively)



(a) Wobble mode at 37.4 Hz (40.9 Hz) *(b) Umbrella mode at 82.1 Hz (90.6 Hz)* *(c) 2-diameter mode at 91.9 Hz (109.7 Hz)*

Figure 8 – Natural modes predicted in the original (and modified) motor cooling fan (additional ring applied to external cooling fan increasing inertia from GD^2=1560 to 1900 kg-m^2)

Table 2 – Summary of predicted natural frequencies (Hz) using FEA

Modes	**Original Fan**		**Modified Fan w/ Inertia Ring**	
	Cooling fan only model	**Fan+shaft model w/ orig. FW**	**Cooling fan only model**	**Fan+shaft model w/ new FW**
1st Torsional	N/A	45.0	N/A	40.0
2nd Torsional	88.7	58.4	88.5	56.1
3rd Torsional	N/A	72.3	N/A	69.8
Fan Wobble	49.0	37.4	42.0	40.9
Fan Umbrella	105.0	82.1	90.0	90.6
Fan 2-Diameter	106.0	91.9	110.0	109.7

PROPOSED SYSTEM

As discussed in reference [1], with a new larger sized compressor flywheel, the first TNF was predicted between 7× and 8× running speed and should meet the API recommended separation margin. API also recommends maintaining a separation margin from the electrical line frequency, which for a 22-pole motor is 11× running speed. There was concern that the modified flywheel could shift the 2nd TNF to 59.4 Hz, which would be near the electrical line frequency of 60 Hz.

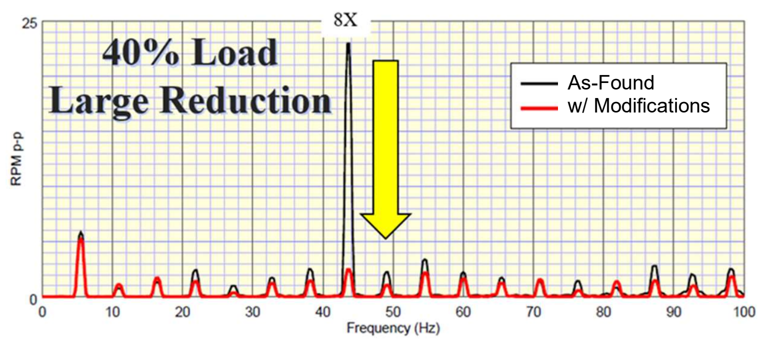
Torsional calculations can have some uncertainty [6] and the actual TNFs of the system could be closer to excitation frequencies than predicted. This torsional system has very low damping so that compressor harmonics above 10× could still be of concern. It was therefore proposed that the external cooling fan also be modified. Increasing the fan inertia with a bolt-on ring of GD^2 = 1900 kg-m^2 was predicted to provide additional separation margin from the 2nd TNF.

From the FEA results summarized in Table 2, it was anticipated that adding the inertia ring to the external cooling fan could affect other natural frequencies. For example, while the 2nd TNF was predicted to decrease for the modified system, the natural frequencies corresponding to fan wobble, umbrella, and 2-diameter modes were raised since the ring also reinforced the backplate of the fan and increased the lateral stiffness.

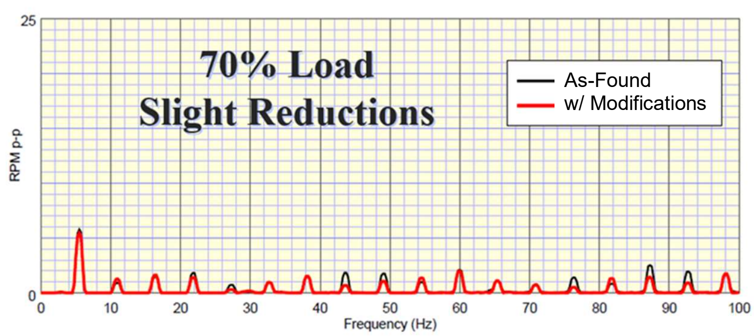
FOLLOW-UP TESTING

Follow-up tests of the modified system showed the first TNF frequency was near 7.4× running speed and the second TNF was near 10.5× running speed. With the modified flywheel and extra inertia ring on the cooling fan, the TNFs were moved away from the compressor harmonics and electrical line frequency as shown in Table 3.

The speed fluctuations measured at the external cooling fan were significantly reduced and acceptable throughout the operating load range of the compressor. For example, when operating at 40% compressor load, the speed oscillation at the fan was reduced from 6% to 1% at 8× running speed as shown in Figure 9. There was only a slight reduction at 70% load, which was already considered low-level before the modifications were implemented.



(a) 40% compressor load



(b) 70% compressor load

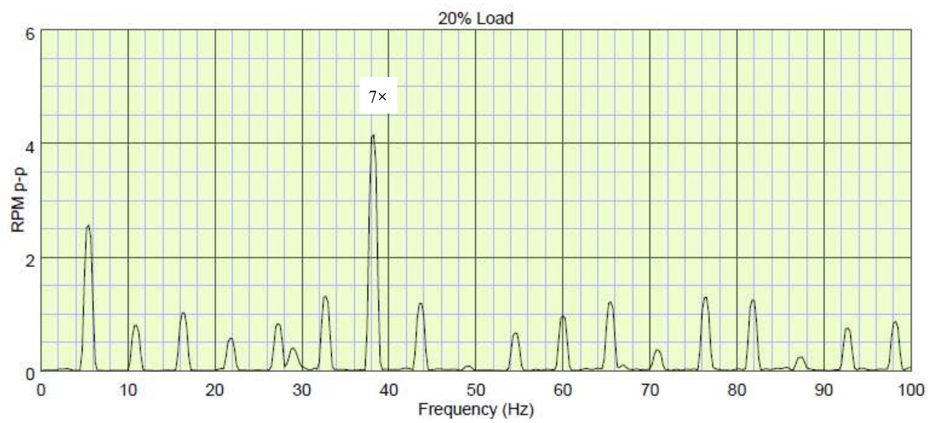
Figure 9 – Speed fluctuation on motor shaft near fan (original vs modified)

Table 3 – Measured torsional natural frequencies

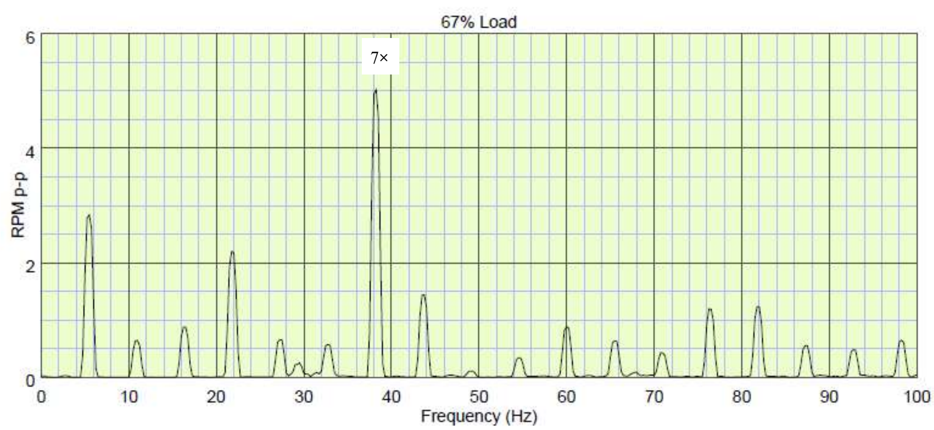
Torsional Modes	Original		Modified	
1st TNF (Crankshaft)	44.5 Hz	8.1×	39.1 – 40.3 Hz	7.2× – 7.4×
2nd TNF (External Fan)	59 Hz*	10.8×	56.8 – 57.6 Hz	10.4× – 10.6×

*Lower frequency was measured with motor shaft on bearing oil film compared to impact test at rest.

During the final test, the highest speed fluctuation on the motor shaft near the cooling fan was approximately 5 RPM p-p at 7× running speed. Figure 10 shows a comparison of 20% compressor load (upper plot) with 67% compressor load (lower plot). Note that the speed fluctuation of the fan was much lower than the original as-found condition (23 RPM p-p at 8× running speed before making the modifications).



(a) 20% compressor load



(b) 67% compressor load

Figure 10 – Speed fluctuation of motor shaft near fan in final test

CONCLUSIONS AND RECOMMENDATIONS

This paper shows how the failure problem with the motor external cooling fan was solved using a combination of field tests and analytical techniques.

- Initial torsional tests showed that the fan failures were likely caused by high angular oscillation from the compressor and amplified by TNFs.
- Impact tests of the motor fan identified various natural frequencies. A torsional mode was found to have very low damping and a mode shape which could induce excessive cyclic stresses in the fan blades and cause fatigue failure.
- The simple FE model of the fan only did not match the test results. Therefore, the fan and compressor shafts need to be included in the FE model to simulate the proper boundary conditions.
- Adding the inertia ring to the fan stiffened the backplate thus changing other natural frequencies. It is important to consider all fan modes such as wobble, umbrella and 2-diameter when making modifications.
- The torsional oscillation at the motor external cooling fan was reduced to an acceptable level with the following modifications:
  - The first TNF of the system was detuned using a larger flywheel.
  - The second TNF was detuned by adding an inertia ring to the fan.
- The modifications were verified by follow-up field testing. The motor fan and the torsional system in general are no longer sensitive to compressor loading by the stepless capacity control.
- This case study demonstrates the importance of maintaining adequate separation margins from significant compressor orders and electrical line frequency as recommended in the API Standards.
- Although unnecessary for this case, additional measurements could have been performed to evaluate strain levels in the fan. This would have required installing strain gages on select blades near the failure locations. When conducting the impact tests, the response could be measured through the strain gage in addition to the accelerometer. Strain could also be evaluated during operation using slip rings or a telemetry system. For example, the last author used these methods when troubleshooting a similar failure problem with other fans [7,8].

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