

# **Torsional Vibration: The Value of Field Verification**

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## **ABSTRACT**

Torsional modelling of reciprocating machinery systems is critical to the reliability of the installation. Differences in construction, error in manufacturer information or other variables can, and occasionally do prevent the prediction from matching reality. This has led to major failures and lost production.

This paper examines some variables that can affect the accuracy of torsional modelling:

- Differences in crankshaft data when calculated with industry-standard techniques and finite element analysis.
- Electric motor rotor stiffness.
- Coupling installation.
- Reciprocating compressor loading and performance.

Finally, a case history, where “twin” units have distinctively different torsional vibration is presented. One unit had electric motor rotor and shaft failures, while the other unit has not had any torsional failures. Did the “good twin” have a failure that prevented catastrophic shut down?

## **Background**

Torsional vibration design analysis (TVA) is essential to ensure the reliability of rotating machinery; particularly when the driver or driven machine is a reciprocating engine or compressor. Because of the conversion of reciprocating energy to rotating energy or vice-versa, the required or provided torque curves (torque-effort) are not flat. As a matter of fact, the alternating component of torque is quite often as large as two to three times the mean, or constant torque. Each component in a drive train must be designed to withstand these “rough” torque requirements. Dynamic response due to resonance can amplify these torque levels to where component failures occur. The importance of a TVA is quickly recognized when a system failure has caused unplanned outages.

A TVA at the design stage utilizes vendor data to build a model of the drive train and predict system response. All vendor data has tolerances applied to them and a range of predictions are then available. With conservative design factors and appropriate guidelines, the TVA will prevent most torsional failures.

However, even with a successful design approach, torsional failures do occur. This paper describes how reality can differ from the theoretical model and presents a strong case for field measurement of torsional vibration at startup.

## Variation in Model Inputs

A theoretical model is only as good as the information used to create it. If the input data do not represent the real characteristics of the machinery, the model, in turn, will not either. There have been a number of areas where vendor data has been evaluated for accuracy. The topic of modelling accuracy is not the focus of this paper but is presented to reiterate the importance of accurate input data.

## Compressor Crankshaft Mass-Elastic

The accepted procedures for determining the mass-elastic parameters for compressor crankshafts come from B.I.C.E.R.A. [1] or Ker Wilson [2]. In these procedures, crankshaft geometry is used to estimate inertias and connecting torsional springs using discrete mathematical formulae. With finite element analysis (FEA) readily available and accessible, these crankshaft characteristics can be calculated very accurately. One example shows that the FEA calculated stiffnesses can be as much as 8% higher than those calculated using the mathematical method (Figure and Table 1). Depending on the rest of the system, this difference in stiffness can substantially change a predicted system response.

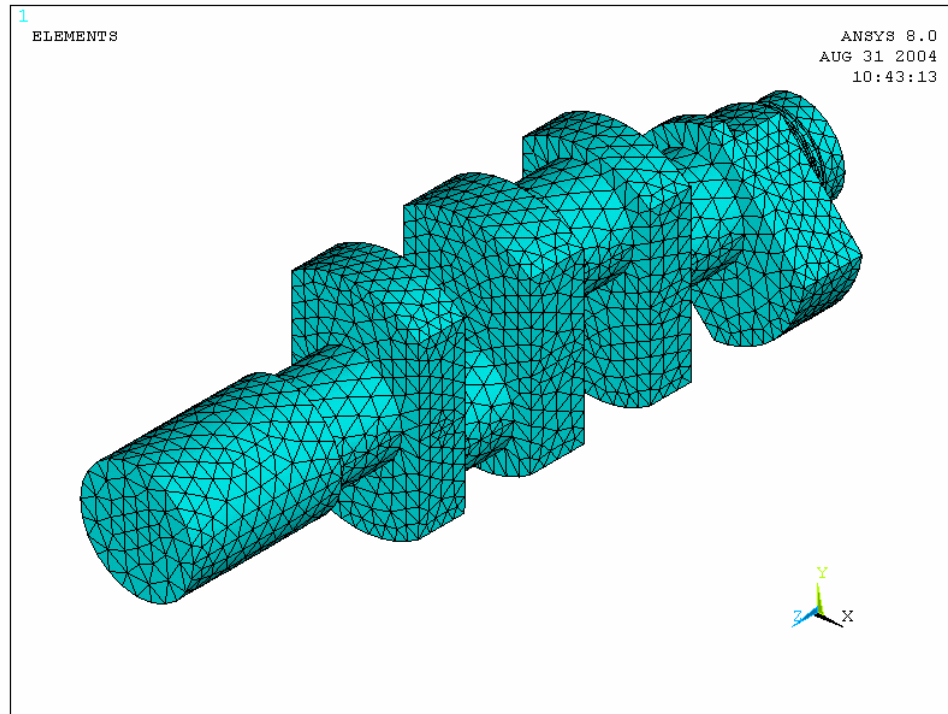


Figure 1: Finite element model of half of a 4 throw compressor crankshaft

**Table 1: Stiffness of selected elements of the crankshaft comparing mathematical and finite element analysis**

	From	To	Mathematical Formulae in-lb/rad x 1E6	Ansys FEA in-lb/rad x 1E6	Difference
K1	End of drive stub	Bearing #1	123.6	136.960	+ 5%
K2	Bearing #1	Throw #1	154.7	184.464	+ 8%
K3	Throw #1	Throw #2	158.3	163.651	+ 2%
K4	Throw #2	Bearing #2	157.4	185.616	+ 8%

## Electric Motor Rotor Stiffness

The shaft in an electric motor has very calculable stiffness characteristics when it is not assembled with the rotor. By definition, however, motor rotors are very massive and quite stiff when compared with the bare shaft. The method in which the rotor is attached to the shaft can have a significant affect on the stiffness of the assembly.

A traditional approach to shrink-fit, keyed rotors has been the one-third penetration estimation – that is, one third of the length of the shaft inside the rotor acts completely unrestrained by the rotor. As one could imagine, this estimation will vary in validity with the amount of shrink in the assembly, lubrication at the shaft-rotor interface, surface finishes on the shaft and rotor, etc. Depending on the relative stiffness of the shaft to the rotor and the shaft to the rest of the drive train, the effect on the system response could be substantial.

In some motors a so-called spider rotor is utilized. The rotor in such a motor is mounted to the shaft via a frame-work of radial bars which are fastened at the outer side to the rotor and at the inner side to the shaft. The design of spider attachments varies widely, as does how they are affixed to the shaft and rotor. The effective stiffness of the motor assembly, therefore, varies as well. For instance, if the spider bars are slender, the stiffening of the motor shaft may be minimal. However, if the spider framework is more robust, the rotor stiffening effect may be substantial.

The American Petroleum Institute (API) and others have developed ways of approximating the stiffening effect using an equivalent diameter for the length of shafting through the rotor. FEA can be used to determine whether these equivalent diameter methods are valid. The results, however, will depend greatly on the connectivity between the spider elements and the rotor and the stiffness of the rotor itself. Because of unavoidable variances in motor construction, it is hard to imagine that any calculation method will provide an accurate stiffness, or one that is repeatable between two “identical” motors. It is recommended that a range of rotor stiffnesses be considered in a TVA. Table 2 demonstrates predicted first mode natural frequencies assuming no rotor stiffening and essentially a rigid rotor.

There is field evidence to show that a robust, properly-constructed spider rotor, behaves essentially as a rigid element. On the other hand, in cases where the rotors themselves have failed due to poor construction, the rotor has behaved like there is no rotor stiffening of the shafting.

**Table 2: Torsional natural frequencies with stiff and soft motor rotors****System: 2 Throw Compressor + Flex Disk Coupling + 650 HP Induction Motor**

TNF Prediction	No Rotor Stiffening		Stiff Rotor	
	TNF (Hz)	TNF (Hz)	TNF (Hz)	Order of Runspeed @990 RPM
Low	112.0	117.3	117.3	6.8
Nominal	115.2	120.8	120.8	7.0
High	117.4	123.2	123.2	7.1

### Coupling Installation

Usually the stubs on the drive or driven shafts have the smallest diameters of all the shafting in a drive train. Because these components are relatively small diameter, they can be the softest “spring” in a torsional system. Of course the stiffness of these springs varies with the length of the element. The longer the shaft, the less stiff it is.

When a shaft is inserted into a coupling hub, part of that shaft effectively becomes part of the hub; therefore much stiffer. The remaining part of the shaft retains its original stiffness properties. Typical practice for the installation of coupling hubs is to mount the hub flush with the end of the shaft. Occasionally, however, a fabricator may choose to install the hub further onto the shaft or not all the way on. Table 3 shows the difference in the stiffness of a motor drive stub for two different coupling hub mounts. Differences like this can change the torsional response of the system. In one case [3], this meant the difference between a resonant and non-resonant design.

**Table 3: Difference in motor drive stub stiffness with two different coupling hub mounts**

Coupling hub mount description	Stiffness (lb-in/rad x 1E6)
Flush with the end of the shaft	198.9
End of shaft extending 3” beyond the hub flange (ie. into the coupling center section)	295.6

### Actual Operating Conditions

It is essential that the operating envelope for a reciprocating compressor be fully understood for the TVA. The unit torque-effort, the alternating and mean torques required to turn the compressor, depends not only on the compressor geometry, but also on the conditions under which the machine will operate. Variations in speed, pressures, temperatures, cylinder configuration or gas composition can affect the torque-effort greatly. A seemingly small difference such as unloading cylinder # 3 instead of #1 can be the difference between success and failure (See Figures 3 to 7). In the design stage, it is often uncertain what the compressor operating point will be. It is advisable to consider a larger envelope than expected and re-

evaluate the TVA if operating conditions deviate from design. For further machinery reliability, it is advisable to design a system to withstand reasonable upset conditions, or at least test the system to see how sensitive it is to upset conditions. Upset conditions may include conditions where cylinder ends are unloaded to simulate the effect of damaged or inoperative valves. For engines upset conditions may include misfiring cylinders or worn engine dampers.

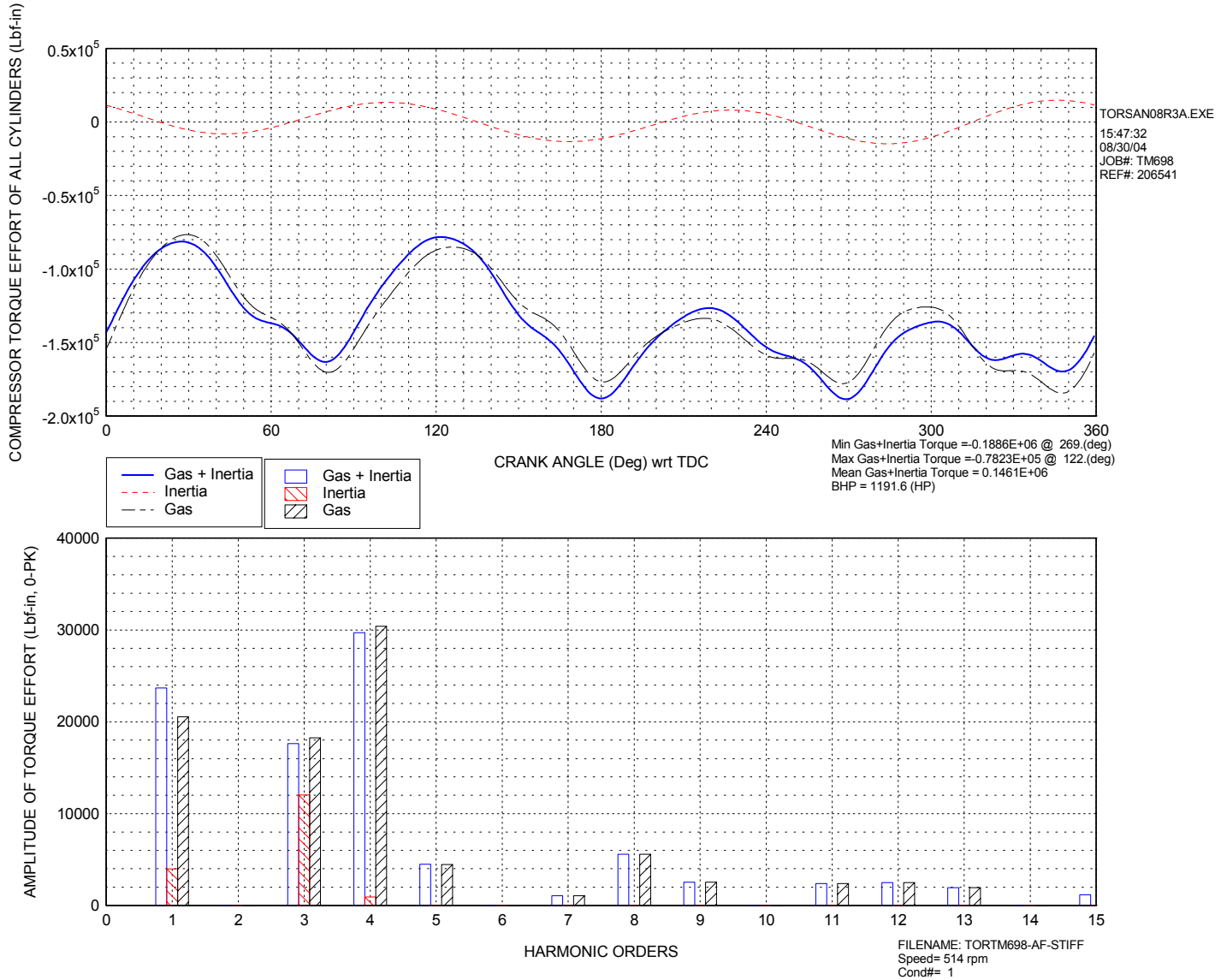


Figure 2: 1 Stage 4 Throw - All cylinders double acting

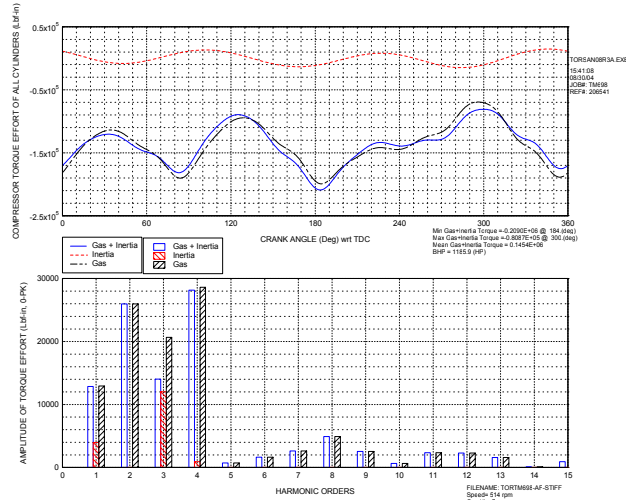


Figure 3: SACE1

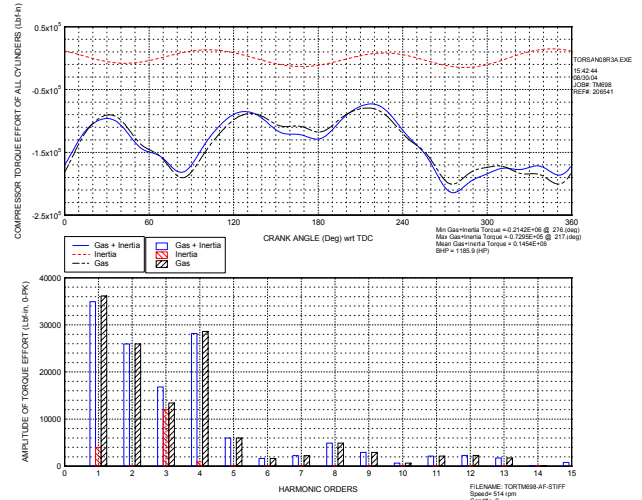


Figure 5: SACE3

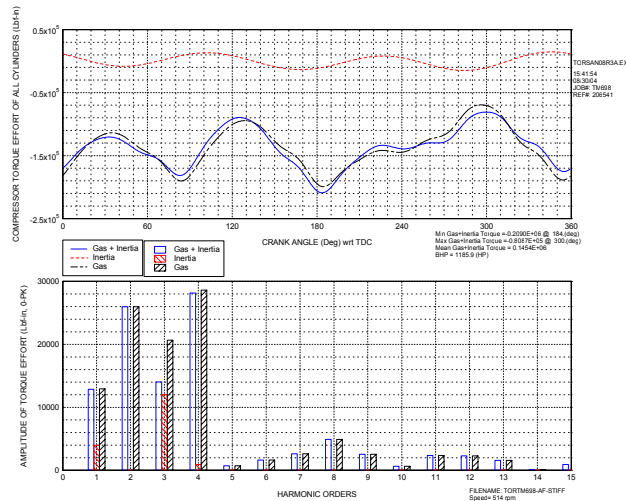


Figure 4: SACE2

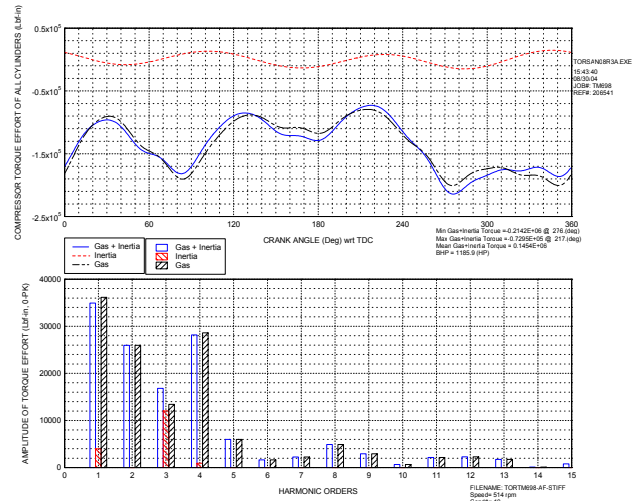


Figure 6: SACE4

## Case History: 'The Evil Twin'

### Machinery Description

2 "identical" 2 throw, single stage field booster compressors with flexible disk couplings and fixed speed induction motors.

Unloading: None

1500 HP induction motor at 890 RPM.

Operating Envelope:

Ps = 600 - 1400 psig, Pd = 2500 - 2750 psig, Pockets 0 - 100% Open

Delivering 14 - 41 mmscfd

### Failures:

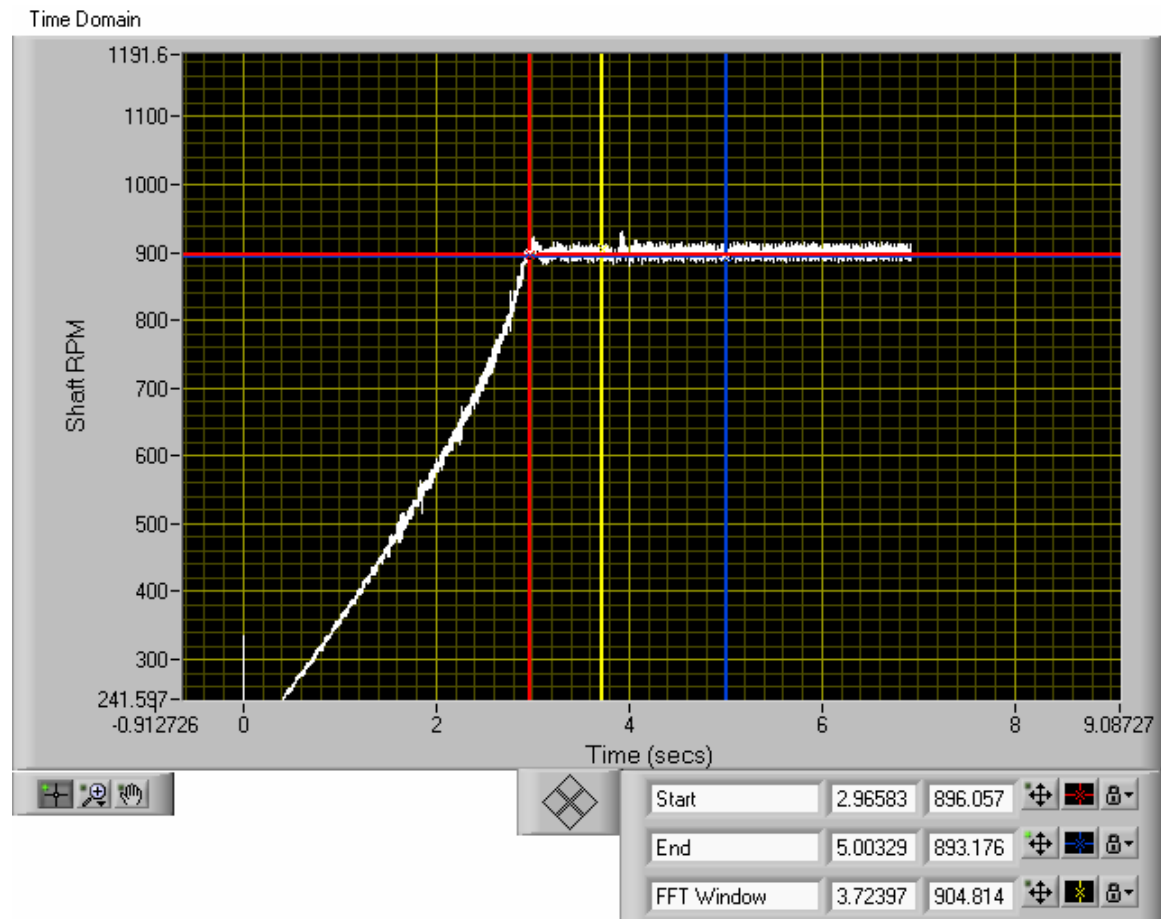
Boosters A and B had been running off and on for about 6 months. Booster B experienced a rotor bar failure which in turn, caused a catastrophic destruction of the rotor and stator. The motor was replaced with one of the "same" specifications and the machine was restarted. Shortly thereafter, the motor shaft failed on Booster B. All the while, Booster A experienced no drive train failures. Why not?

### Probable Causes:

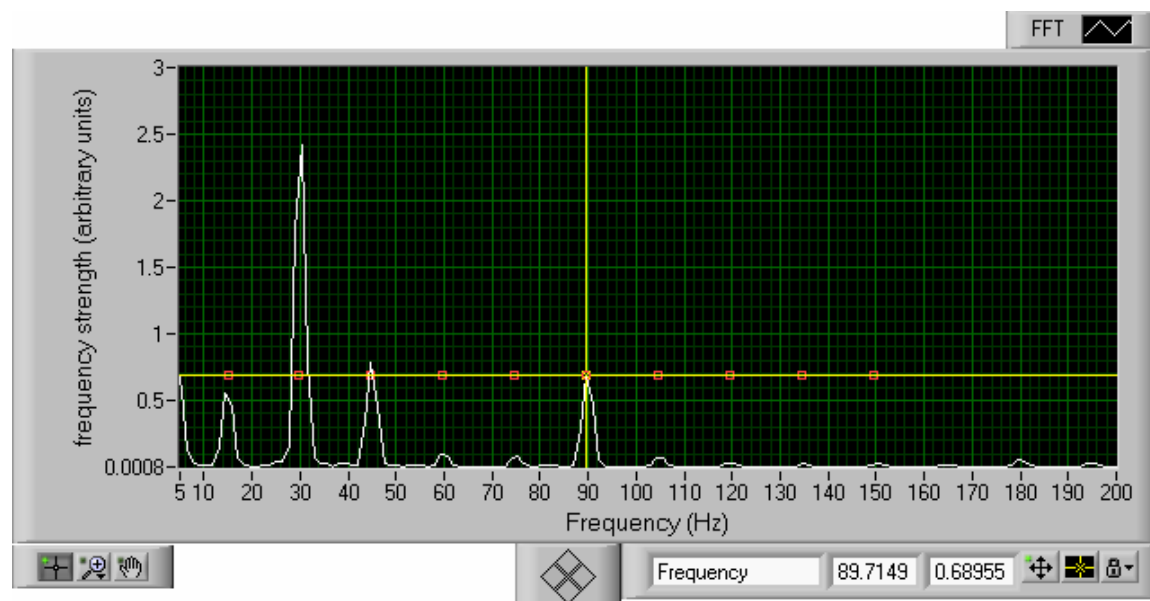
There was conjecture that a bearing failure may have lead to the shaft failure. Further investigation was necessary. The original TVA completed by another service provider held no explanation for the shaft failure (albeit the motor was treated as a simple one mass one stiffness model) but a field and simulation study was ordered. Prior to the field study, Unit B motor was replaced again. The replacement consisted of the good shaft from the first failed motor and the good rotor core from the second failed motor. A metallurgical failure analysis also proceeded concurrently with the field and simulation torsional work.

## Field Evidence:

-Booster B resonant at 90 Hz (6x runspeed)



**Figure 7: End of startup ramp measured with a shaft encoder at the opposite drive end of the motor. Windowed data bound by the red and yellow vertical lines.**



**Figure 8: Frequency content showing strong 6<sup>th</sup> order resonance near 90 Hz.**

## Field Evidence Continued:

Stress too high for Booster B, especially with 1035 shaft material

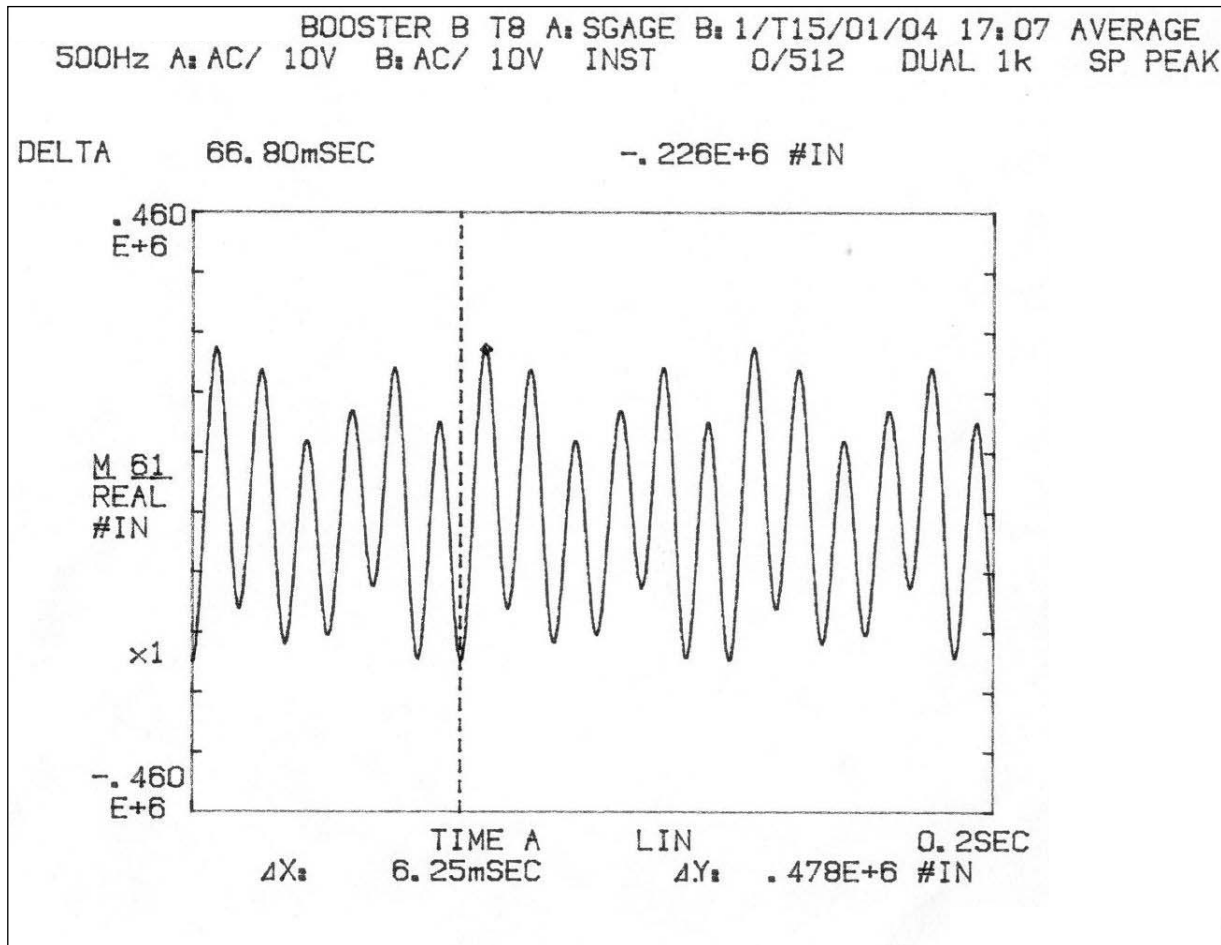
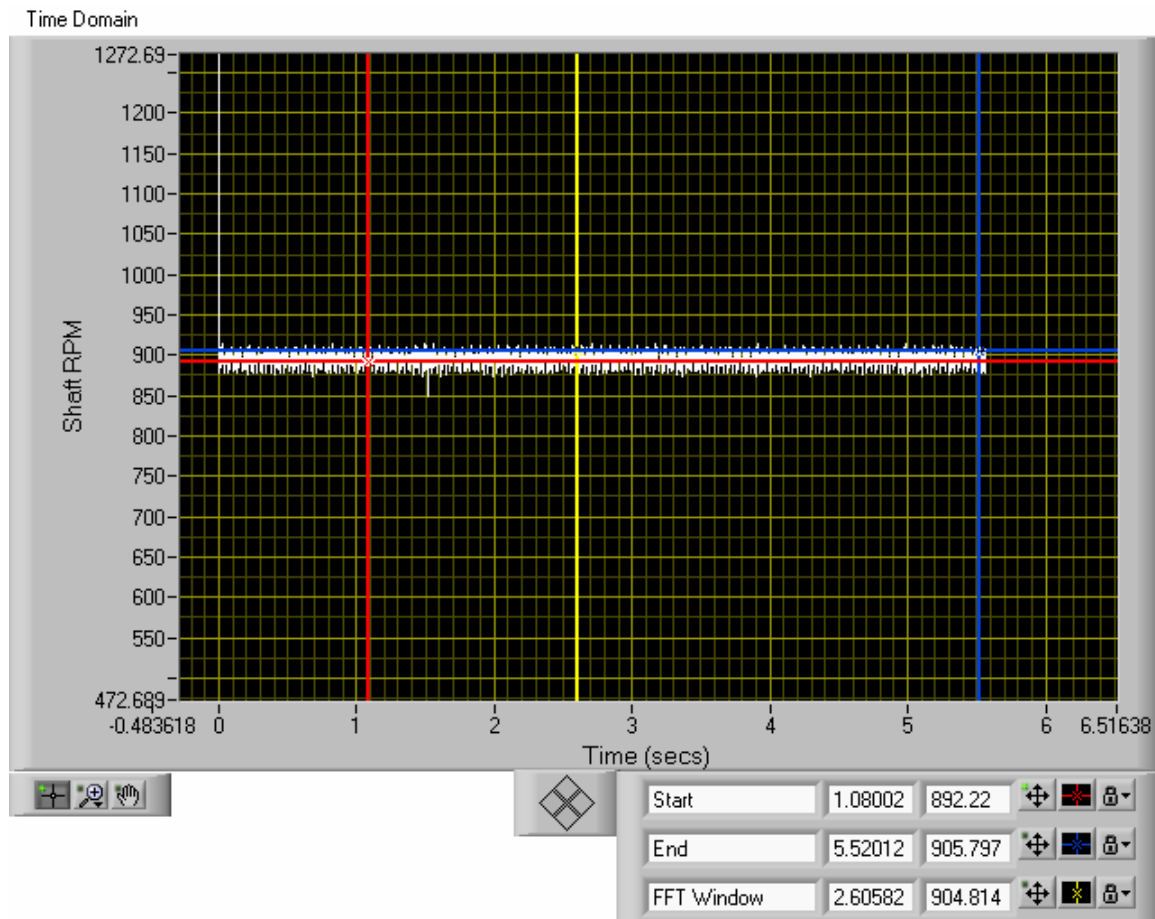


Figure 9: Torque at the motor stub measured with a strain gage telemetry system.

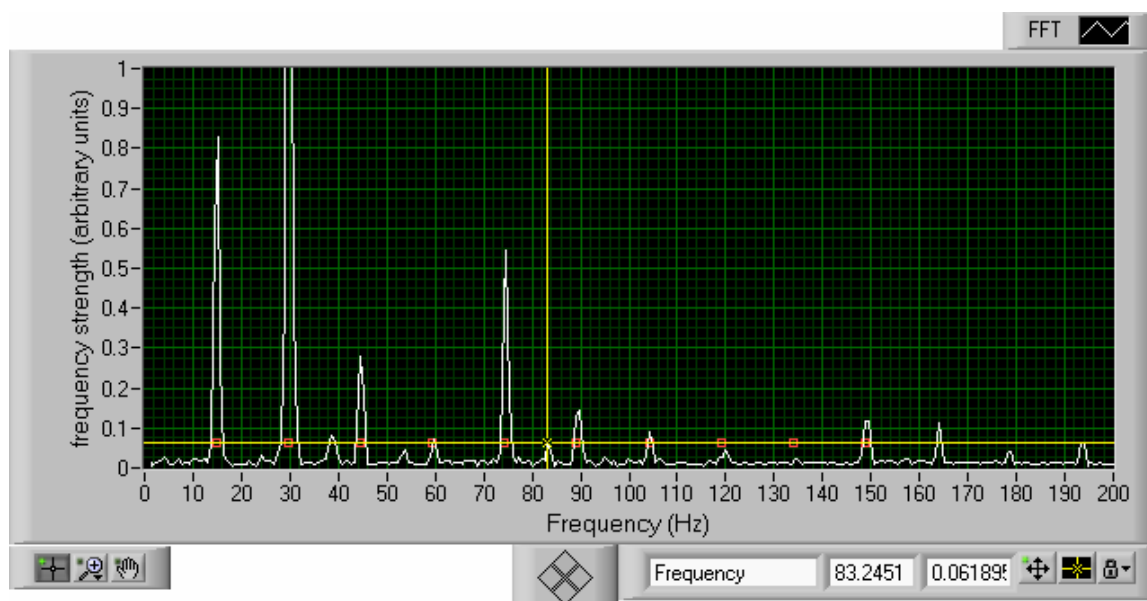
At a diameter of 5.9 inches, 478,000 lb<sub>f</sub>-in pk-pk torque creates a peak stress of 5925 psi. The endurance limit of the 4140 motor shaft is 18,730 psi. With a shrink stress concentration factor of 2.2, the design factor for the motor stub is calculated to be 1.4. Technically any design factor over 1.0 means that the shafting will have infinite life. However at resonance a design factor as low as 1.4 will accommodate very little increase in excitation torque. This is because at resonance there is very high dynamic amplification of alternating torques - the amplitudes are only limited by damping, which is usually low for motor driven reciprocating compressors. Excitation of 6<sup>th</sup> order torque effort could easily be increased by losing a suction or discharge valve, or some other upset, or unexpected operating condition.

## Field Evidence Continued:

-Booster A not resonant



**Figure 10: Torsional data at runspeed measured with a shaft encoder at the opposite drive end of the motor. Windowed data bound by the red and yellow vertical lines**



**Figure 11: Frequency content showing possible TNF near 83 Hz**

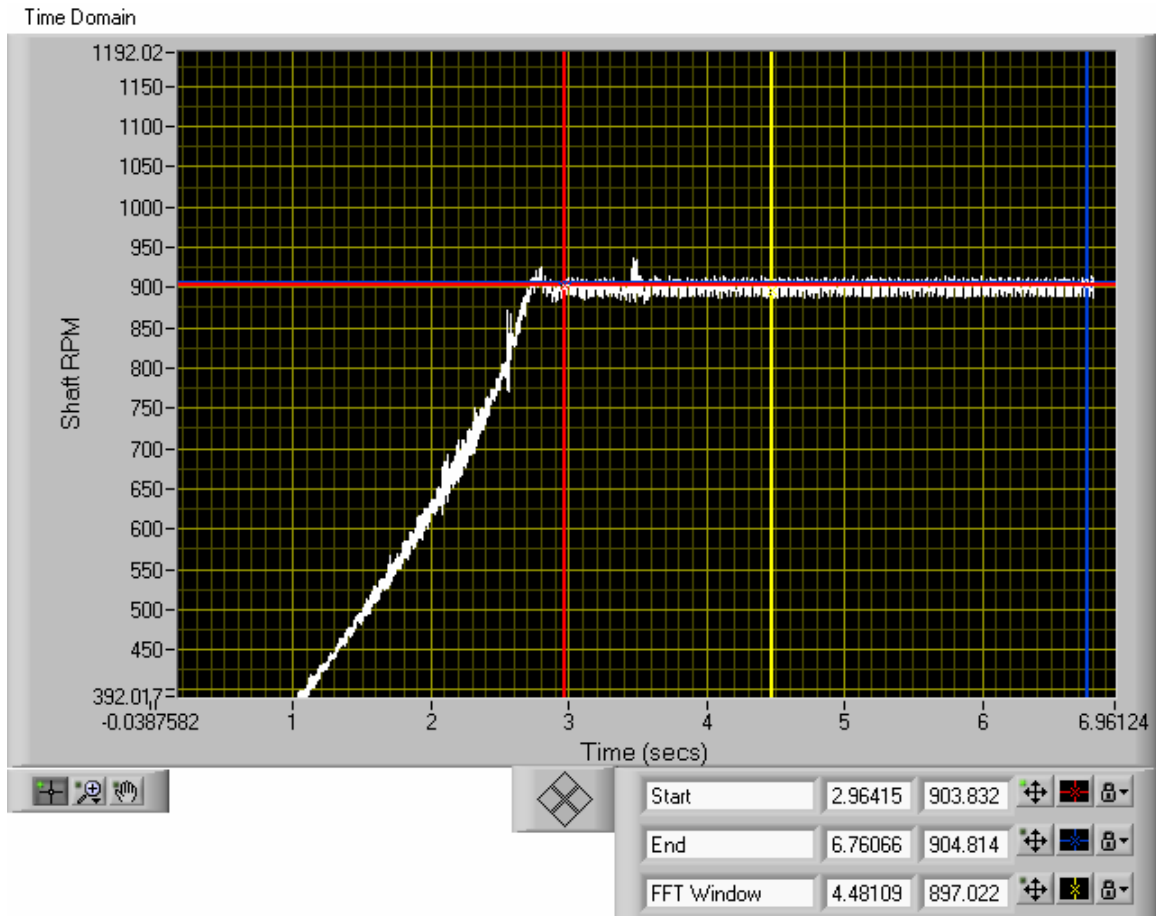


Figure 12: End of startup ramp

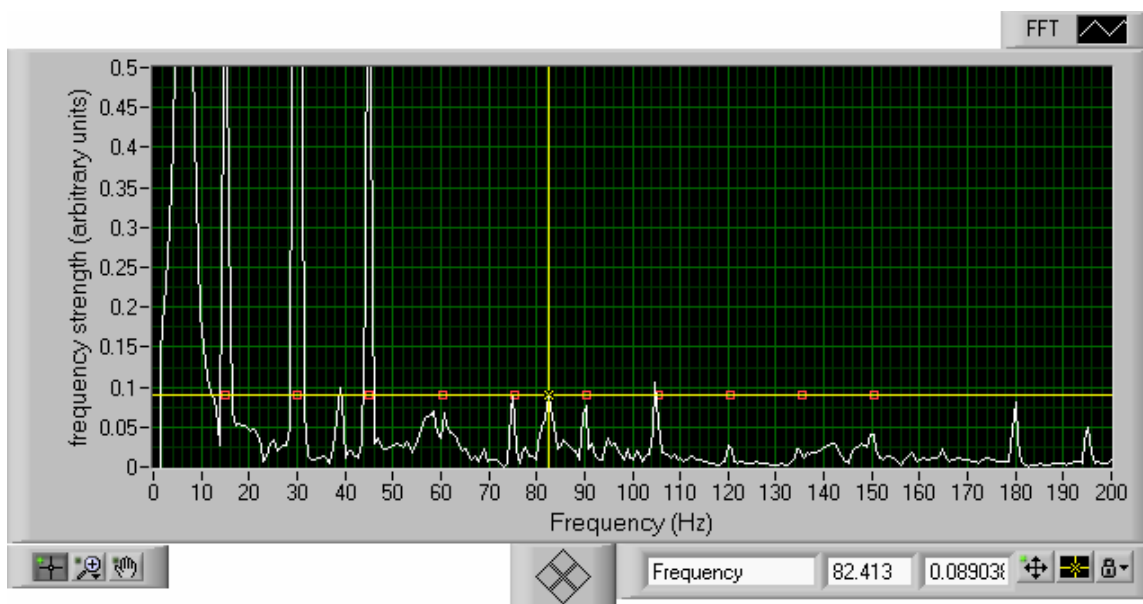


Figure 13: Expanded Y axis. The red squares are orders of runspeed. The only realistic peak away from an order of runspeed is at 82.5 Hz (5.5x runspeed). Therefore the TNF is likely between 82 and 83 Hz.

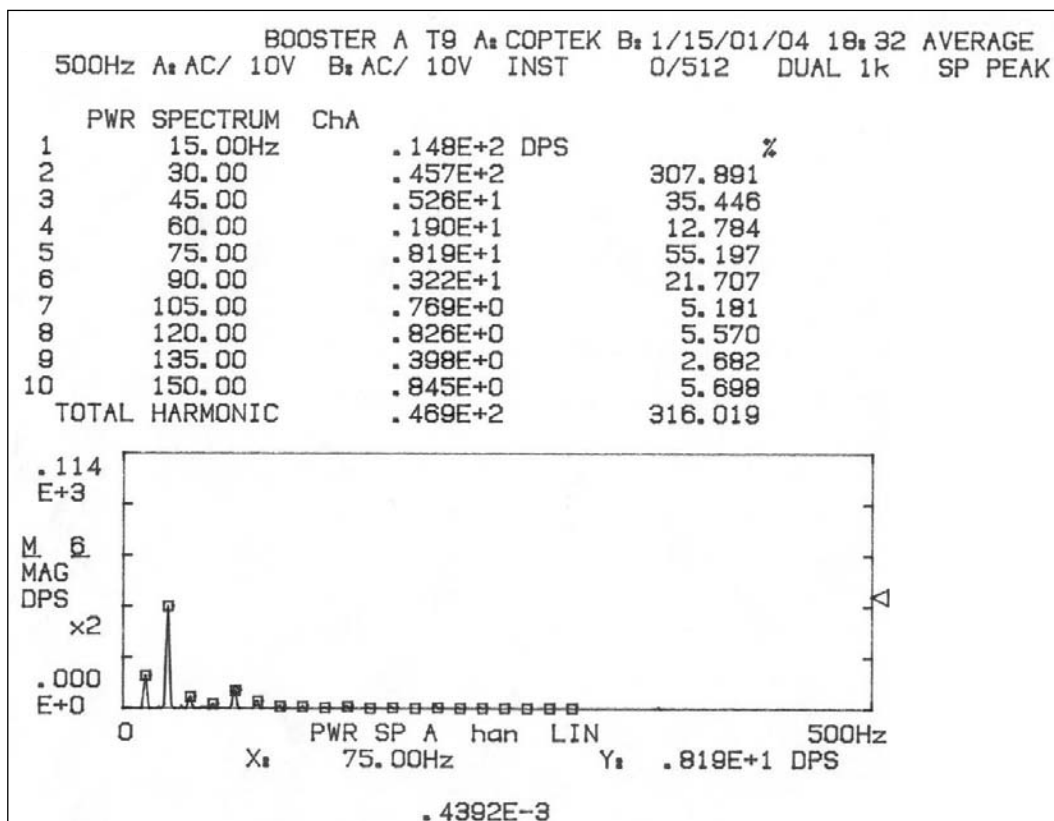


Figure 14: Harmonic content of Booster A at motor ODE, loaded

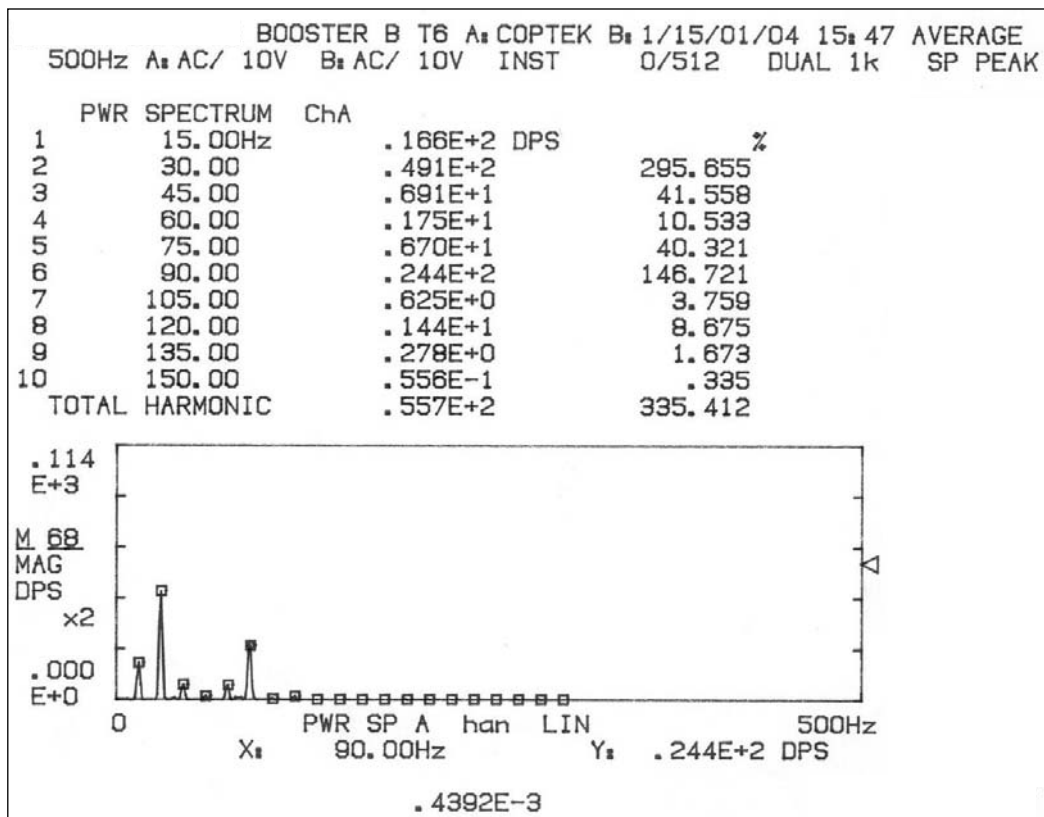


Figure 15: Harmonic content of Booster B at motor ODE, similarly loaded. The content is almost the same except at the 6<sup>th</sup> order where Booster B is significantly higher. Even if the TNF of Booster A is not clear, it is clear that Booster B is resonant and Booster A isn't.

## Shaft metallurgy

A metallurgical analysis of the failed replacement shaft indicated a 1035 material spec instead of 4140 (the original shaft was 4140). The calculated endurance limit of the 1035 shaft was about 30% less than the 4140 shaft.

## Simulation Results

**Table 4: 1<sup>st</sup> Torsional natural frequencies of Booster B**

TNF Prediction	Calibrated to match field data		With no rotor core stiffening	
	TNF (Hz)	Order of Runspeed @990 RPM	TNF (Hz)	Order of Runspeed @990 RPM
Low	86.5	5.8	83.5	5.6
Nominal	89.5	6.0	86.5	5.8
High	91.3	6.1	88.2	5.9

A difference of nearly 7 Hz between the measured 1<sup>st</sup> TNF of Boosters A and B is unexpected on two “identical” units. A TNF between 82 and 83 Hz for Booster A is even lower than the model predicts if there is no shaft stiffening due to the rotor, as shown in Table 4. Very low or virtually no stiffening of the motor shaft may occur if there is looseness or high relative motion between the motor shaft and the rotor assembly.

**Table 5: Worst Design factors of Booster B with 1035 and 4140 motor shafts – assuming the same TNFs and forced response between the two shafts - only the material spec differs**

Worst Condition	Motor DSF with 1035 Shaft	Motor DSF with 4140 Shaft
Design	0.1	0.4
Field	0.4	1.2

So, Why is Unit A OK?

Possibilities?

- considering that Booster B first suffered motor rotor damage, could have the same occurred in Booster A but to a lesser degree?
- could Booster A system have been resonant like Booster B, but incurred only enough damage to change the shaft stiffness between the shaft and the rotor, effectively detuning the system
- was Booster B always resonant but Booster A wasn't, due to differences in rotor construction?
- did Booster B motor originally fail in a non resonant way, and only the replacement motors were resonant?

More detailed field testing is warranted to determine the reason why these two systems are different. Presently both these units are fully utilized and unable to be shutdown again for further testing. Additionally, it will be almost impossible to conduct any meaningful tests on the failed motors or motor parts, beyond the metallurgical tests already completed.

## **Conclusions**

Many variables comprise a simulation study. Appropriate modelling techniques will reduce the risk of torsional failures. A field analysis can provide additional insurance for a reliable system, particularly for systems that are sensitive to design uncertainties and cannot be made insensitive without incorporating expensive modifications. The level of field analysis will be determined by the criticality of the machinery and the degree of sensitivity to the design uncertainties.

## **References**

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