Case Histories of High Pinion Vibration When Eight Times Running Speed Coincides with the Pinion's Fourth Natural Frequency

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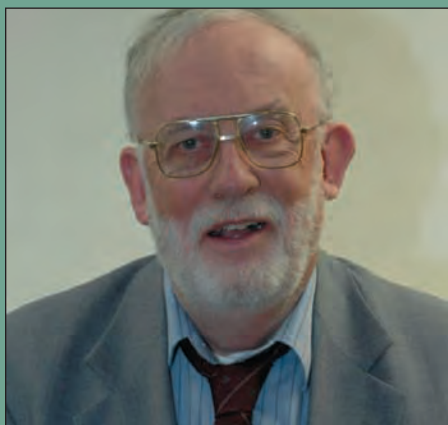
INTRODUCTION

Gear sets needed modifications to the pinion during full-load, full-pressure string tests of three centrifugal compressor trains [1], each manufactured for a different end user. Two trains were driven by the contract gas turbine and one by a shop steam turbine. Total horsepower for the compressors ranged from 22000 to 31050 BHP. Gear ratios ranged from 2.14 to 2.44. The vibration frequency of concern was at 8 times (8X) running speed of the pinion (high-speed) and this coincided with the fourth natural frequency of the pinion. This 8X frequency was found only on load tests such as ASME PTC 10 Class 1. The first pinion was modified by taking weight off the free end and the other two pinions were modified by adding weight to the free end. This OEM of these compressors now requires the gear manufacturer to show by rotor dynamic analysis that certain multiples of pinion speed will not coincide with the natural frequencies of the pinion. Such analysis should be made during the design phase rather than finding an issue during full-load testing or field operation. Test engineers should set up the Fast Fourier Transform (FFT) to look for these higher frequencies.

The Lateral Analysis, in section 2.6.2 of API 613 fifth edition specification for gears [2] does not address this situation. It requires an undamped (forward synchronous) analysis to find the critical speeds in the range of 0 to 125 percent of trip speed, which is well below eight times running speed. It suggests that there are only three lateral critical speeds of concern, but the fourth critical speed is sometimes also of concern. It says that if one of the first three critical speeds causes concern, then a damped unbalance response analysis should be made, but it need only go from 0 to 125 percent of trip speed. The damped synchronous response analysis should not be extended to speeds at higher multiples of running speed to find the higher critical frequencies of pinions because neither the bearing coefficients nor polar inertia terms will be correct.

The first four critical frequencies should be determined with no limitation on percentage of the critical frequencies above trip speed. The polar inertia term effects at higher speeds in a synchronous undamped critical speed map are not appropriate for the higher modes of a pinion running at a fixed speed much lower than the frequency of the higher mode. A nonsynchronous undamped critical speed map should be completed, either for the pinion running at its own speed or running at zero speed (*i.e.*, a planar critical speed map). The fixed speed and planar modes can be calculated as in reference [3]. Instead of or in addition to this special critical speed map, a damped eigenvalue analysis should be generated with the bearings in the loaded condition at speed. A ten percent margin of the fourth critical frequency from eight times pinion running speed may be appropriate for the bearings at the loaded condition.



ABSTRACT

Three case studies are described to illustrate pinion vibration during full-load, full-pressure string tests of centrifugal compressor trains. The vibration frequency of interest in the studies was eight times the speed of the pinion, which was near the pinion's fourth lateral natural frequency. Simple modifications to the pinion lowered the vibration to appropriate levels. Proper use of the undamped critical speed map for higher modes well above synchronous speed will be discussed. Modifications to the dynamics paragraphs of the API 613 Gear Specification will be presented for consideration.

UNDAMPED CRITICAL SPEEDS

The standard forward synchronous critical speed map will be shown to be inaccurate for higher modes such as the fourth when the fourth is well above the running speed. The planar critical speed map for the shaft at rest provides sufficiently accurate results for tuning the fourth natural frequency so that it does not occur at 8 times running speed. This map should be easy to plot and does not involve the use of bearing properties, bearing programs, or damped eigenvalue programs. The undamped natural frequency calculation uses a moment equation [3] with the term:

$(I_P\,\omega - I_T\Omega)\,\Omega\,\theta$ (Formula 1)

where ω = shaft speed, Ω = whirl frequency,

I_P = polar moment, I_T = transverse moment, and θ = slope

Synchronous forward: $\omega = \Omega$ and then the term becomes: $(I_P - I_T)\,\Omega^2\,\theta$

Planar or shaft at rest: $\omega = 0$ and then the term becomes: $-\,I_T\,\Omega^2\,\theta$

Synchronous backward: $\omega = -\Omega$ and then the term becomes: $-\,(I_P + I_T)\,\Omega^2\,\theta$

Synchronous forward frequency > Planar frequency > Synchronous backward frequency

The equations can be solved when ω = shaft speed is fixed

For the higher frequencies when $\Omega >> \omega$ then:

- The critical frequencies are very close to the planar frequencies
- The forward synchronous map should not be used

In explanation, from the term (Formula 1) in the moment equation, if the whirl frequency Ω is much higher than the shaft speed ω, then the term $I_P\,\omega$ can be ignored, *i.e.* the calculated frequencies are almost the same as for planar = shaft at rest.

The compressor OEM's original critical speed program only calculated forward synchronous frequencies. It was based on an earlier version of the undamped critical speed program [3]. The compressor OEM modified the original program to add the capability to calculate frequencies for backward synchronous, forward and backward for a fixed shaft speed, and planar = shaft at rest. The planar option is used for the modeling of the modal ring test of rotors. Modifications for the fixed speed option were verified by comparison with the results from the Lund stability program [4,5] for the same model. The program also was verified by comparing results using the planar option vs. modal ring test results for an actual rotor.

In the first case study the original and modified pinion were analyzed for each of the above models noted above. Forward synchronous, forward for a fixed speed, planar, backward for a fixed speed, and backward synchronous analyses were run to determine which method possessed the best correlation to the physical test data. In the second case study, only forward synchronous, planar, and backward syn-

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chronous analyses were run for the original and modified pinion. In the third case study, only the planar analysis was run for the original and modified pinion.

FIRST CASE HISTORY

A sketch of the train for the first case history is shown in Figure 1. This train was string tested in 2002 at the gas turbine manufacturer's facility on full-load, full-pressure using hydrocarbon gas with the contract gas turbine used as the driver. Three identical trains were shipped to an offshore platform. The compressors were designed for natural gas injection at a final discharge pressure of 4510 PSIA (311 BAR). The total BHP is 31040 for the compressors. The low-speed range is 3840 to 5040 RPM and the high-speed range is 8214 to 10780 RPM with a gear ratio of 2.14. The gear is a double helical design and the pinion (high-speed) bearings are of the tilt-pad type.

A vibration frequency issue was found during the full-load string test and it coincided with 8 times the running (high) speed. See Figure 2, a waterfall plot of amplitude from the test. An amplitude of twenty-seven (27) microns is well over the 20 percent allowable limit for any discrete, nonsynchronous vibration in 4.3.2.2.9 of API 613 fifth edition [2]. There also were measurements of up to 10 G's (acceleration) at 8X running speed of the pinion measured on the gearbox. Ten (10) G's is well above the 4 G's limit for overall operation as defined in 2.7.1.3 of API 613 fifth edition [2].

Figure 3 is a calculated forward synchronous critical speed map for the original pinion, where … = shaft speed = Ω = whirl frequency in Formula 1. This map does not indicate a good match to the test data. By extending the bearing stiffness lines, the calculated fourth natural frequency appears to be about 90000 cpm. This is much higher than the 76500 cpm determined in the full load test. The tilt-pad bearing coefficients were determined by the compressor manufacturer by using the tilt-pad bearing progam as described in [6].

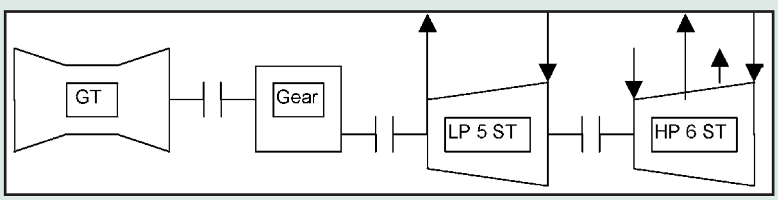

Figure 1 - Train Sketch - Case History 1

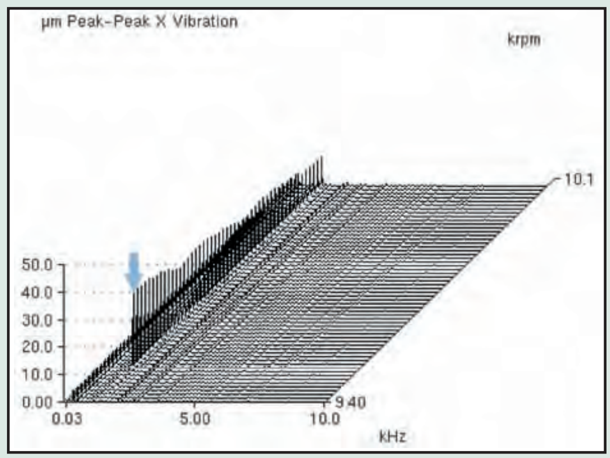

Figure 2 - Waterfall Plot - Case History 1 - Original Pinion

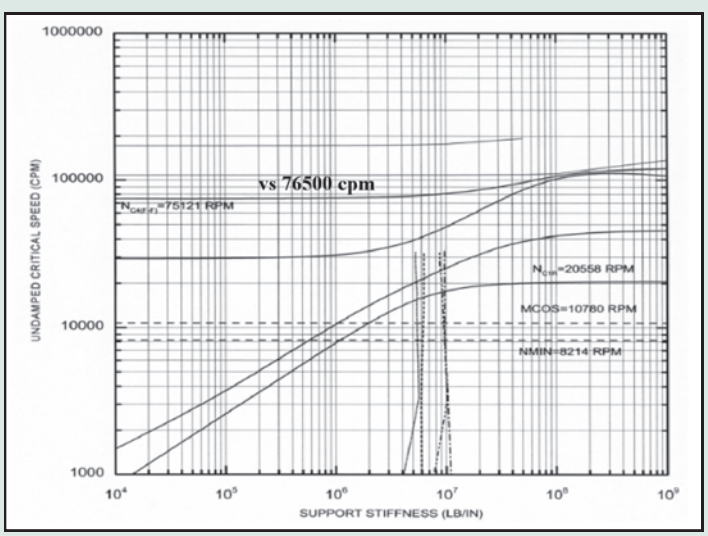

Figure 3 - Synchronous Forward Critical Speed Map Case History 1 - Original Pinion

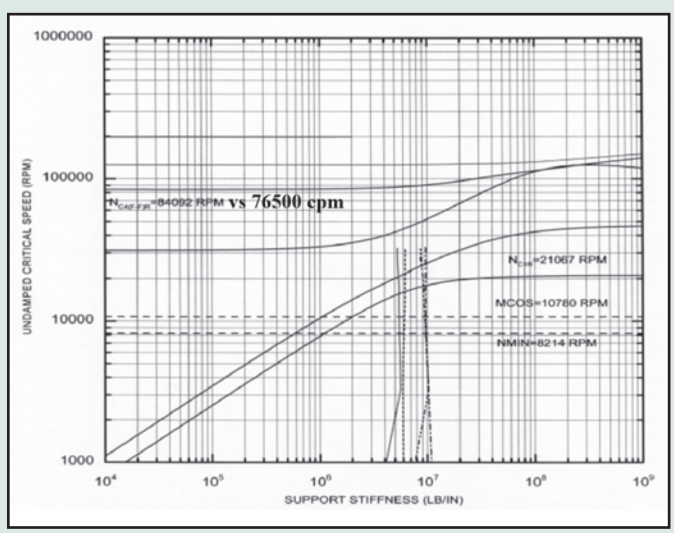

Figure 4 - Fixed Speed = 10780 RPM Forward Critical Speed Map Case History 1 - Original Pinion

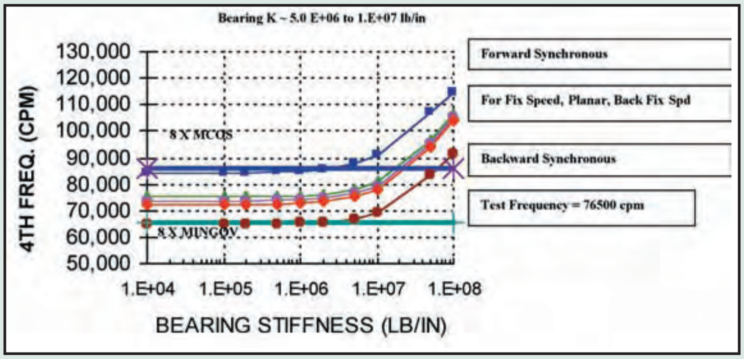

Figure 5 - Original Pinion - Case History 1 - Fourth Undamped Natural Frequencies

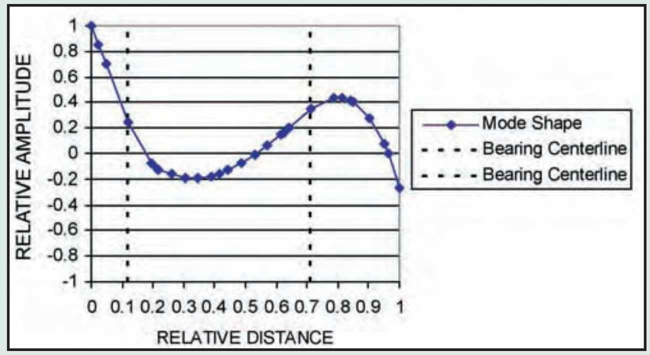

Figure 6 - Original Pinion - Case History 1 - Fourth Forward Mode = 78106 CPM at Fixed Speed

Figure 4 is a calculated forward critical speed map for a fixed speed of 10780 RPM, which is the maximum continuous speed. This is where ω = shaft speed is fixed at 10780 RPM and Ω = whirl frequency is independent of ω in Formula 1. This map indicates much better correlation between the tested frequency of 76500 cpm and the calculated fourth frequency on this map of about 80000 cpm.

Figure 5 pertains to the original pinion and plots the calculated fourth undamped natural frequency vs. bearing stiffness for the five different ways to run the undamped critical speed program (forward synchronous, forward fixed speed, planar, backward fixed speed, and backward synchronous). The fixed speed is the maximum continuous speed of the pinion. The curve for the fourth natural frequency is pulled from a critical speed map for each of these conditions and plotted on one map. Also shown on Figure 5 are 8 times the minimum governor speed and 8 times the maximum continuous speed of the pinion. The best way to correlate between 8 times = 76500 CPM and the calculated frequencies is to use forward fixed, planar, and backward fixed, all of which provide similar results. The bearing stiffness is approximately 5 to 10 million LB/IN at 31000 HP.

Figure 6 displays the calculated fourth undamped mode of the original pinion at the fixed speed of 10780 RPM (= maximum continuous speed) at 5 million bearing stiffness. The frequency is 78106 CPM, which gives a good match to the tested value of 76500 CPM. The blind end is on the left side of the plot. Because the highest amplitude is on the blind end, the most effective way to change the location of the natural frequency is to either add or remove weight from the blind end.

The pinion was modified to raise the fourth natural frequency by removing mass from the blind end of the pinion. Figure 7 is a sketch of the modified pinion. On a subsequent load test the modified pinion did not indicate the high G's acceleration issue nor did it indicate a vibration frequency issue at eight times running speed.

Figure 8 pertains to the modified pinion and displays the calculated fourth undamped natural frequency vs. bearing stiffness for the five different ways to run the undamped critical speed program (as stated above). The fixed speed is the maximum continuous speed of the

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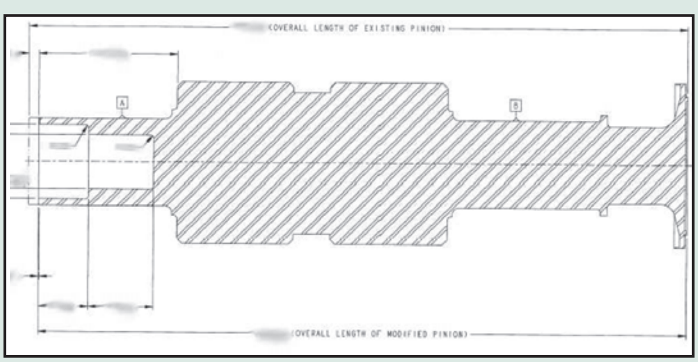

Figure 7 - Sketch of Modified Pinion - Case History 1

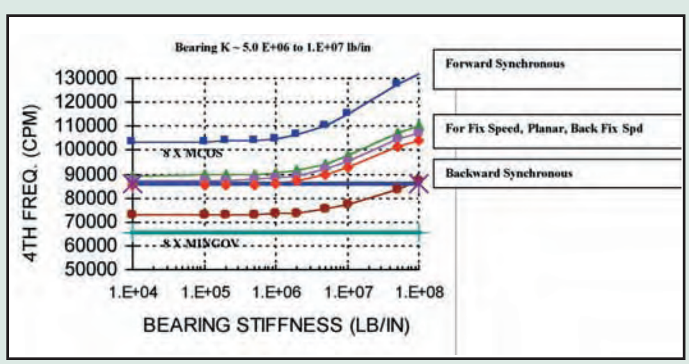

Figure 8 - Modified Pinion - Case History 1 - Fourth Undamped Natural Frequencies

pinion. As the bearing is loaded, the fourth natural frequency (for forward fixed, planar, and backward fixed speed = maximum continuous speed), is raised above 8 times the running speed range of the pinion.

A damped eigenvalue analysis was conducted by the compressor OEM using the Lund stability program [4-5] and the tilt-pad bearing program [6]. For the original pinion, the fourth damped natural frequency was calculated to be 77,000 CPM with a log dec of 0.7 to 1.0. The calculated frequency agrees very closely with the tested value of 76500 CPM. But this calculated log dec of 0.7 to 1.0 was not high enough to avoid excitation at 77000 CPM because during the load test there was substantial horsepower going through the pinion. One way to think of this is to view the amplification factor as approximately Pi/log dec (in the range 4.5 to 3.1); thus the frequency is not critically damped (amplification factor < 2.5) as per 2.6.1.2 of [2]. For the modified pinion the fourth damped natural frequency was calculated to be 100,000 to 106,000 CPM with a log dec of 0.4 to 0.6. Note that 8xMCOS = 8x10780 = 86240 and 100000 CPM for the modified pinion is well above this..

SECOND CASE HISTORY

A sketch of the train for the second case history is shown in Figure 9. This train was string tested in 2003 at the compressor manufacturer's facility at full-load, full-pressure with hydrocarbon gas with the contract gas turbine as the driver. Four identical trains were shipped to an offshore platform. The compressors were designed for natural gas injection at a final discharge pressure of 6621 PSIA (457 BAR). The total BHP is 30427 for the compressors at the certified condition. The low-speed range is 3360 to 5040 RPM and the high-speed range is 7739

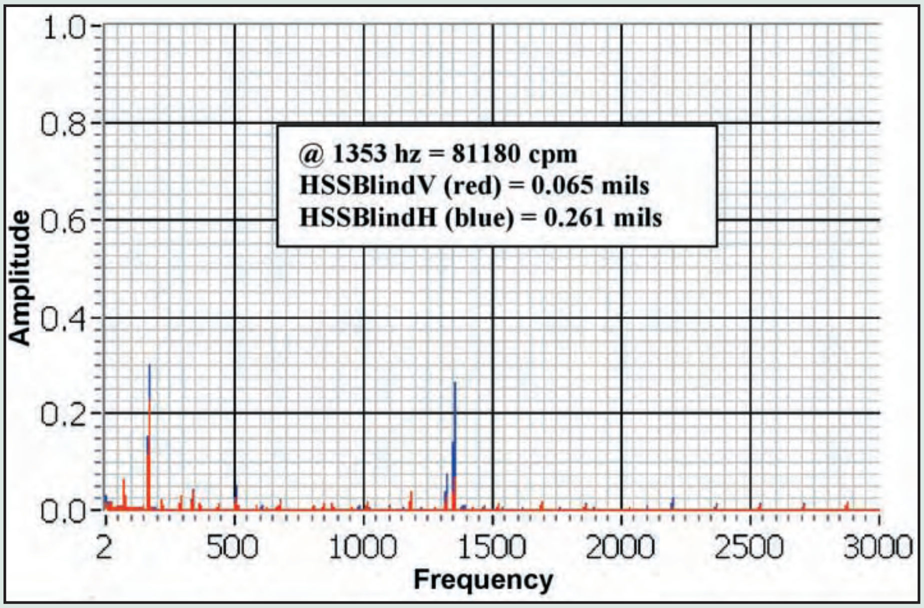

Figure 10 - Vibration Spectrum - Case History 2 - Original Pinion

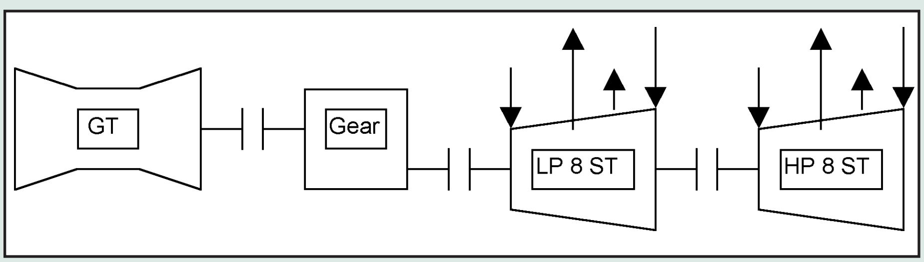

Figure 9 - Train Sketch - Case History 2

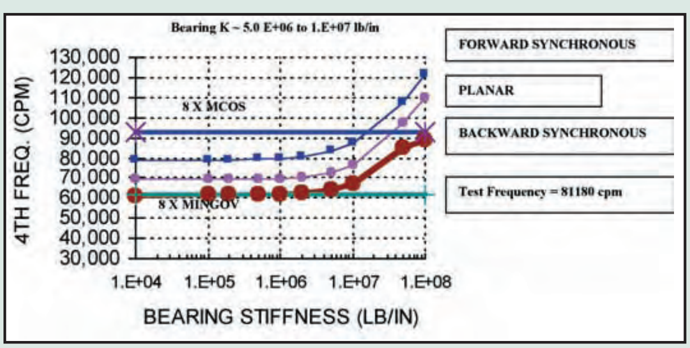

Figure 11 - Original Pinion - Case History 2 - Fourth Undamped Natural Frequencies

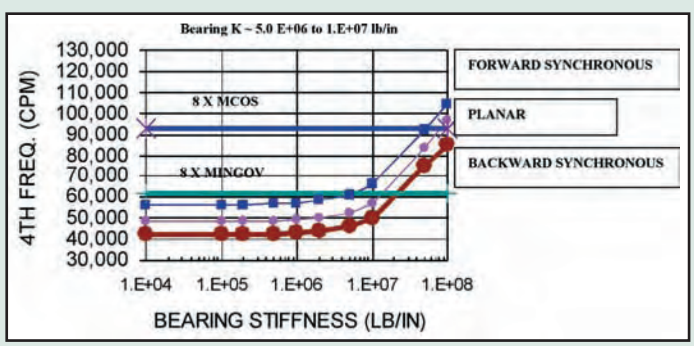

Figure 12 - Modified Pinion - Case History 2 - Fourth Undamped Natural Frequencies

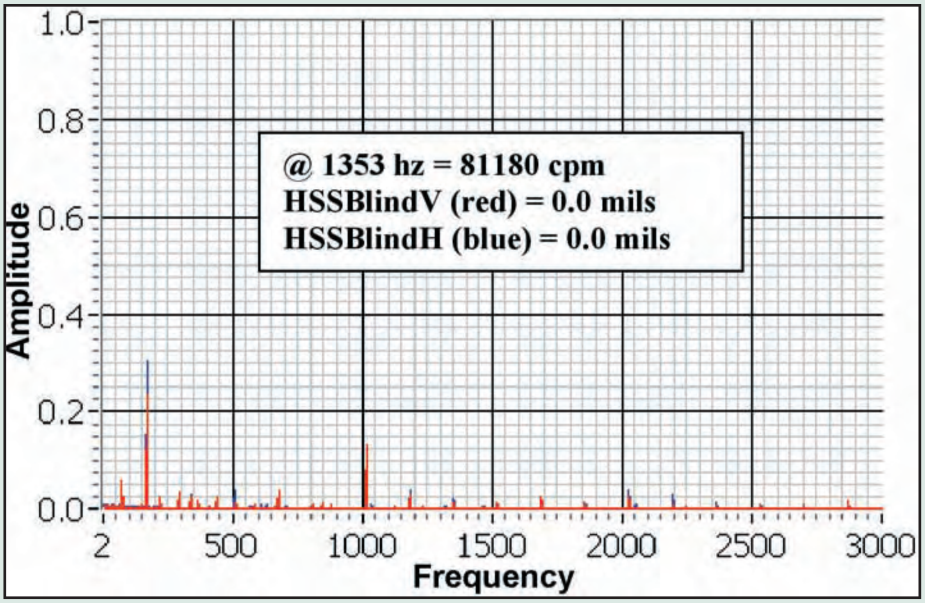

Figure 13 - Vibration Spectrum - Case History 2 - Modified Pinion

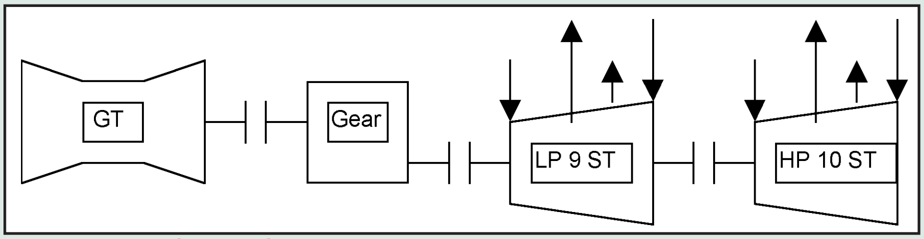

Figure 14 - Train Sketch - Case History 3

to 11608 RPM with a gear ratio of 2.30. The gear is a double helical design and the pinion (high-speed) bearings are of the tilt-pad type.

A vibration frequency issue was found during the full-load, full-pressure string test and it coincided with 8 times the running (high) speed. See Figure 10, a vibration spectrum of amplitude vs. frequency from the test. The peak on the right at 1353 hz shows an amplitude of 0.26 mils of 8X frequency. The running speed of the pinion was 169 hz = 10148 CPM and 1353 hz/8 = 169 hz. The vibration amplitude of 0.26 mils is just over the 20 percent allowable limit for discrete, nonsynchronous vibration in 4.3.2.2.9 of API 613 fifth edition [2].

Figure 11, for the original pinion of Case History 2, is the same type of plot of the calculated fourth undamped natural frequency vs. bearing stiffness as displayed in Figure 5 for Case History 1, except that it does not show the calculated frequencies for fixed-speed forward and fixed-speed backward, as they are very close to the planar. The test vibration frequency of 81180 CPM is equal to the calculated fourth planar frequency at 8 times speed. As noted above, the test data was derived from a pinion speed of 10148 RPM, which is below the maximum continuous speed of 11608 RPM.

After a weight was added to the free or blind end of the pinion, the calculated fourth natural frequency shifted to a frequency below 8 times running speed range. The added weight is negligible compared to the load that is transmitted to the pinion bearings from the gear.

Figure 12, for the modified pinion of Case History 2, plots the calculated fourth undamped natural frequency vs. bearing stiffness for forward synchronous, planar, and backward synchronous. The frequency of concern is the planar. The calculated fourth frequency is below the 8X running speed range.

Figure 13 shows a vibration spectrum of amplitude vs. frequency for a load test of the modified pinion; in which the vibration frequency issue at 8X running speed is not indicated.

THIRD CASE HISTORY

A sketch of the train for the third case history is shown in Figure 14. This train was string tested in 2005 at the compressor manufacturer's facility on full-load, full-pressure with hydrocarbon gas with a shop steam turbine instead of the gas turbine as the driver. The train was shipped to an offshore platform. The compressors were designed for natural gas injection at a final discharge presssure of 3612 PSIA (249 BAR). The total BHP is 22112 for the compressors at the certified condition. The low-speed range is 4080 to 5040 RPM and the high-speed range is 9960 to 12304 RPM with a gear ratio of 2.44. The gear is a double helical design and the pinion (high-speed) bearings are of the tilt pad type.

A vibration frequency issue was found during the full- load, full-pressure string test that coincided with 8 times the running (high) speed. See Figure 15, a vibration spectrum of amplitude vs. frequency from the test. The peak on the right at 1582 hz exceeds 1 mil amplitude, well over the 20 percent allowable limit by 4.3.2.2.9 of API 613 fifth edition [2].

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There were up to 4 G's of acceleration at 8 times running speed of the pinion measured on the gear box. The running speed of the pinion was 198 hz = 11865 cpm and 1582 hz/8 = 198 hz and is shown as the peak on the left.

Weight was added to the free or blind end of the pinion to lower the fourth natural frequency. Figure 16 shows a similar plot of the calculated fourth natural frequency vs. bearing stiffness as displayed before, except that it shows only the calculated planar frequencies and the results for the original and modified pinion are on the same plot. The test vibration frequency of 95000 CPM observed with the original pinion is close to the calculated fourth planar frequency at 8 times running peed for the original pinion. For the modified pinion with the added weight, the calculated fourth natural frequency has shifted below the 8X running speed range.

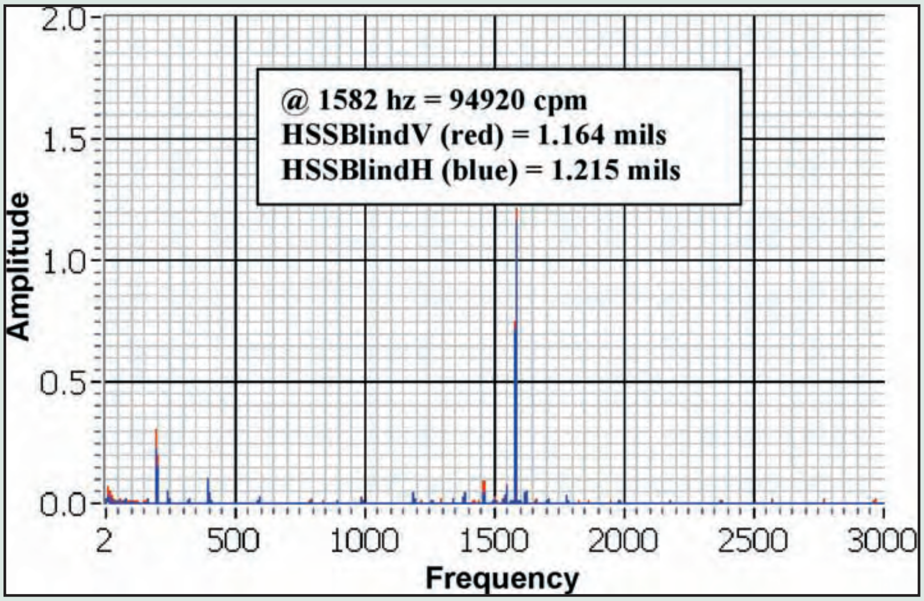

Figure 15 - Vibration Spectrum - Case History 3 - Original Pinion

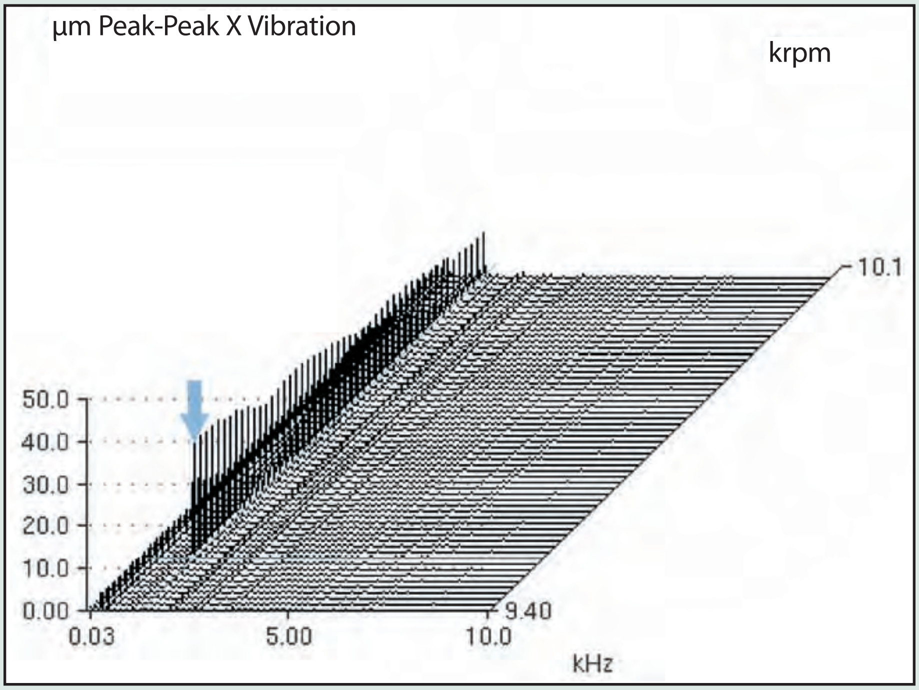

Figure 16 - Original vs. Modified Pinion - Case History 3 Fourth Undamped Planar Natural Frequency

Figure 17 shows a vibration spectrum of amplitude vs. frequency for a load test of the modified pinion, in which the vibration frequency issue at 8X running speed is not indicated.

MODIFICATIONS TO CONSIDER FOR API 613 [2]

- An undamped analysis for fixed speed or planar (0 speed) and at a minimum to 10 times trip speed can be added to Section 2.6.2.1 - Undamped Analysis.
- The number of modes can be changed from three to four in Section 2.6.2.4 - Modes of Concern.
- A required separation margin of lateral critical frequencies from multiples of running speed, such as eight times running speed, can be stated in Sections 2.6.2.4.1 or 2.6.2.4.2 - Separation Margins.
- A new section can be added on Stability Analysis, including a requirement to provide the first four modes and their log dec as a function of load.
- For mechanical tests, as in Section 4.3.2.2.9, the sweep is at a minimum to four times synchronous speed of the pinion. For load tests it should at a minimum be to ten times synchronous speed.

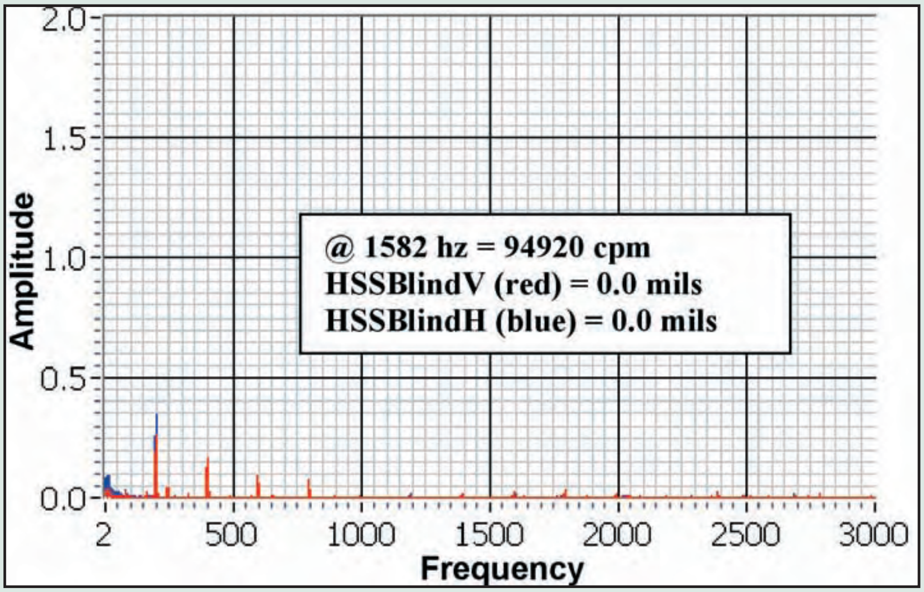

Figure 17 - Vibration Spectrum - Case History 3 - Modified Pinion

CONCLUSIONS

The case histories above demonstrate that gearbox pinions can be analyzed and designed to prevent a vibration issue associated with 8 times pinion speed coinciding with the fourth natural frequency of the pinion, by simply tuning the pinion.

Modifications to the dynamics requirements of the API 613 gear specification [2] may be considered in order to avoid the vibration issues as outlined in the case histories.

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The information contained in this paper includes factual data, technical interpretations and opinions which, while believed to be accurate, are offered solely for information purposes. No representation, guarantee or warranty, of any kind, is made concerning such data, interpretations and opinions including the accuracy thereof.

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BIOGRAPHY

Ed Memmott is a principal rotor dynamics engineer and has been with Dresser-Rand since 1973. He has written sixteen other papers on rotor dynamics, most of them on high-pressure centrifugal compressors. In October 2005 at the 23rd Machinery Dynamics Seminar of the CMVA in Edmonton, he presented a three-hour short course, “A Review of the Dynamics Paragraphs of API 617 for Centrifugal Compressors.” He was on the API subcommittee that wrote the dynamics paragraphs of the Seventh Edition of API 617, and on the task force that wrote the Second Edition of API 684, “API Standard Paragraphs Rotordynamic Tutorial.” He belongs to the ASME, the Vibration Institute, and the MAA. He received an A.B. from Hamilton College (Phi Beta Kappa), an A.M. from Brown University, and a Ph.D. from Syracuse University all in the field of mathematics.