

Experimentally Based Statistical Forced Response Analysis for Purpose of Impeller Mistuning Identification

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2005 in order to develop practical tools for controlling the vibratory response of IBR/Blisks as well as conventional bladed disks. Applications for these new technologies include military engine IBRs and turbines, damping coatings, and power generation turbines for oil, gas and nuclear generators.



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ABSTRACT

A Root Cause Failure Analysis (RCFA) for repeated fourth stage impeller blade failures in a five stage centrifugal propane compressor used in Liquefied Natural Gas service is expanded upon by examining the amount and effect of impeller mistuning by using an intact fourth stage impeller taken from service. Mistuning refers to the fact that the blades of an impeller are not identical, but have slightly different frequencies. Mistuning can cause mode localization, higher vibratory stress, and reduced safety factors for high cycle fatigue. This paper describes both the experimental and analytical assessment of the effect of mistuning on the vibratory response of a fourth stage impeller taken from service.

INTRODUCTION

In 2011, White, Laney and Zorzi reported on the Root Cause Failure Analysis (RCFA) for repeated impeller blade failures in a five stage centrifugal propane compressor (White, et al., 2011). This type of propane compressor was part of a 4.7



Jerry Griffin received his Ph.D. in Applied Mechanics in 1973 from the California Institute of Technology. From 1980 to 2009 he was Professor of Mechanical Engineering at Carnegie Mellon University and was awarded the William J. Brown Chaired Professorship in 2004. He was Founder and Director of the GUIDe Consortium for Bladed Disk Structures

Research and a Fellow of ASME. His research areas focused on developing practical solutions to the mistuning problem in bladed disk vibration and on predicting friction damping. He has authored or coauthored over 50 journal papers. The computer codes developed by Professor Griffin's research group at CMU are widely used by the gas turbine industry. Professor Griffin founded Blade Diagnostics Corporation in

Mtpa LNG plant. A cross section of the propane compressor casing and rotor is shown in Figure 1. It is a straight through centrifugal unit with a horizontally split casing. Five impellers are mounted upon the 6.5 m long shaft. Three side loads add gas upstream of the suction at impellers 2, 3 & 4. The impellers are mounted to the shaft with an interference fit. The return channel and side stream inlets have vanes to align the downstream swirl angle. The impeller diffusers are all vaneless. The vane and rotating blade count are as indicated in Table 1.

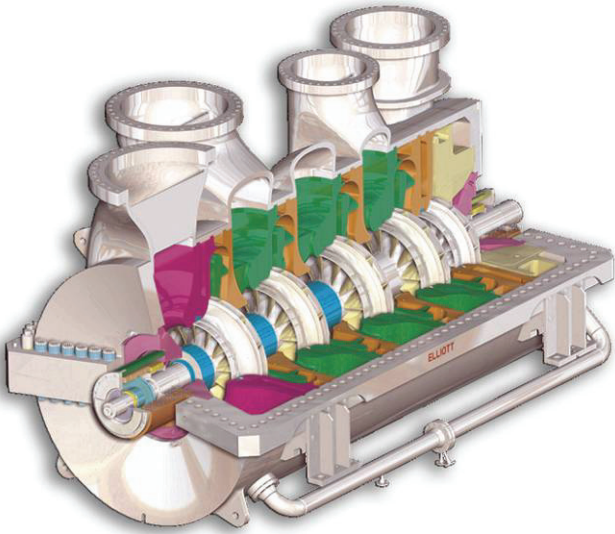


Figure 1. RasGas Train 3, 4 & 5 Propane Compressor

Table 1. Fourth Stage Stationary & Rotating Vane/Blade Count

Component	No. of Vanes / Blades
Return Channel # 3 (Upstream of Fourth Impeller)	16
Side Stream # 3 (Upstream of Fourth Impeller)	14
Fourth Impeller (Tip Dia. = 1.23m)	13

The first failure was believed to have occurred in February 2007 after 26000 hours of operation as evidenced by a vibration step change. The compressor continued to operate until June 2007 (approximately 2200 hours) when it was shutdown and opened up for examination. A large crack was found in a blade in the third impeller and two large pieces released from adjacent blades on the fourth impeller.

The second failure was believed to have occurred in October 2008 after 11000 hours of operation as evidenced by another vibration step change. The compressor continued to operate until May 2009 (approximately 5100 hours) when it was again shutdown and opened up for examination. A blade crack was found at the fourth impeller. The crack initiated at the leading edge of the blade on the shroud side. It follows the toe of the weld before turning within the blade and returning

back to the leading edge near the mid span of the blade. There were no defects at the crack initiation site and welds were to specification.



Figure 2. Fourth Impeller Released Pieces from Blade Leading Edge



Figure 3. Fourth Impeller Released Piece at Blade Leading Edge

Due to the two consecutive failures, an RCFA was conducted as discussed by White, et al., 2011. Results of the RCFA identified three potential root causes likely in some combination with each other:

- Resonant Frequency Mode and/or Mistuned Blade Leading Edge Mode
- Mist Carry-Over
- Choked Operation

This paper will focus further on the effect of mistuning on the fourth stage impeller along with a discussion of impeller design methodology.

Further information on mist carryover, choked operation,

and failure mitigation efforts can be found in White, et al. (2011).

IMPELLER MECHANICAL DESIGN AND EVALUATION

A review of traditional methods of impeller mechanical design have been presented by numerous authors including by Kushner, et al. (1980, 2000, 2004), Singh, et al., (1988, 2003), and Konig, et al., (2009). As mentioned by Konig. . . “regarding the state-of-the-art of evaluation of impeller excitation, shrouded centrifugal impellers should be distinguished from semi-open centrifugal impellers and from the airfoils of axial compressor stages. The sensitivity to excitation and the statistical failure frequency is highest for axial blades and lowest for shrouded centrifugal impellers.” The following are typically examined:

- Mean Stress
- Disk Critical Speeds
- Leading Edge Impeller Blade Resonance
- Disk Interaction Resonance

Mean Stress

Impeller mechanical design typically begins with an evaluation of the mean stress and bore growth. The maximum mean stress must remain within safe limits during an API 617 impeller overspeed test (API, 2002). In addition, the impeller bore-to-shaft fit must safely be able to transmit the required horsepower even when considering the bore deformation under overspeed condition.

For the impeller in question, the mean stress values and bore deflections were considered acceptable. The maximum equivalent mean (von Mises) stress, due to impeller rotation, was located at the leading edge cover fillet. A value of 85.3 ksi (588.1 MPa) was predicted at the maximum continuous speed versus the minimum material yield strength of 110 ksi (758.4 MPa), therefore the impeller was well designed from a static loading perspective. Furthermore, the API required spin test was shown to induce a residual compressive stress of -20 ksi (-137.9 MPa), which further increased the safety margin.

Disk Critical Speed Avoidance

The term disk critical speed was defined by Wilfred Campbell (Campbell, 1924). The resonant operating speed is equal to the natural frequency divided by the number of diametrical nodal lines. The resonant frequency versus nodal diameters is shown in Figure 4. Large margins from disk critical speeds are present. At a disk critical speed, the relative amplitude of the disk around the circumference in stationary coordinates would form a standing wave in stationary space. The modes of concern are up to the number of rotating impeller blades divided by two. Manufacturers have particularly avoided the second through sixth nodal diameters. Overall,

there have not been many disk critical speed problems since Campbell as all manufacturers try to avoid them.

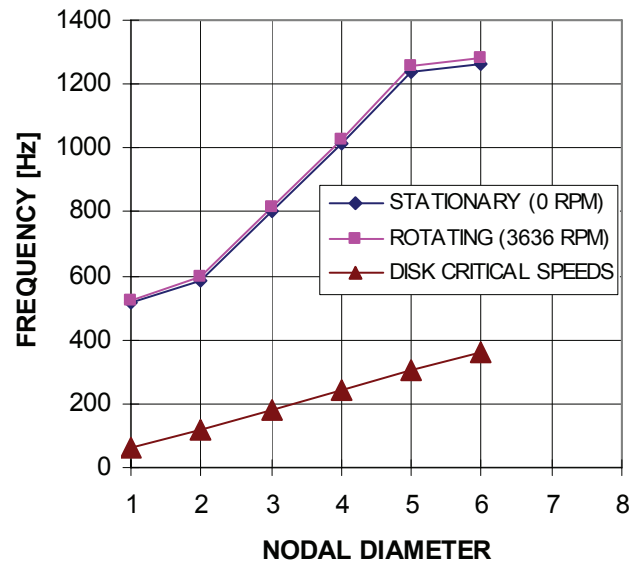


Figure 4 – Fourth Impeller Disk Critical Speeds

Leading Edge Impeller Blade Resonance

Blade resonance for the impeller blade’s leading edge is a concern especially for impellers having a full inducer. Figure 5 shows the Campbell Diagram for the first leading edge blade mode. Large separation margin was present from the 16 upstream return channel vanes as well as the 14 sideload guide vanes.

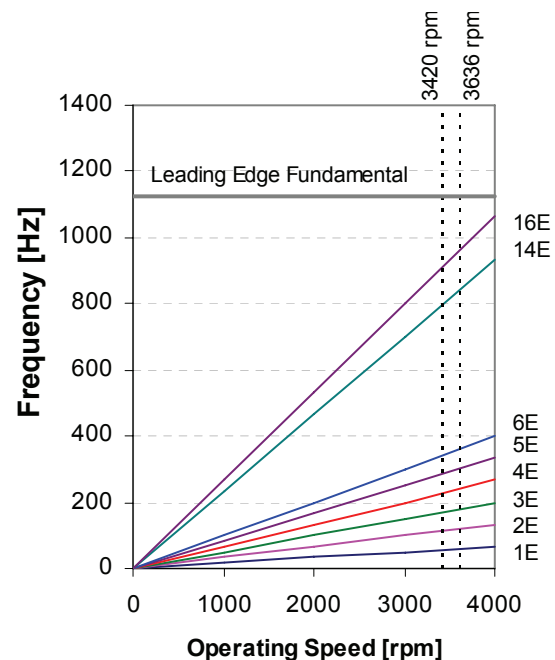


Figure 5 – Fourth Impeller Campbell Diagram

Avoidance of Rotating Blade/Stationary Vane Interaction Resonance

Besides disk critical speeds, there also can be excitation at harmonics of speed, especially from upstream and downstream vanes. Kushner (1980) developed a set of parametric equations for evaluating interaction resonance using the number of rotating blades to determine the excitation and response potential. For certain combinations of the two numbers, the coupled blade/disk modes with diametric nodal lines are either phase cancelled or excitable when an excitation source coincides with a resonant mode. These equations are represented as follows:

When not at a disk critical speed,

$$|y \cdot S| \pm |z \cdot B| = n$$

$$y \cdot S = h$$

$$f_r = y \cdot S \cdot \omega$$

Whereas when at a disk critical speed,

$$\text{For } B > 1$$

$$y \cdot S = h = n$$

$$f_r = n \cdot \omega$$

where:

B = Number of rotating blades

S = Number of stationary elements

f_r = Natural frequency at speed, Hz

h = Harmonic of speed

n = Number of diameter nodal lines

y & z = Integers > 0

ω = Rotating speed, Hz

Interaction resonance requires a certain combination of numbers besides having operation at a specific resonant speed. An interference diagram as applied to impellers is essentially a graphical representation of these equations. This type of interference diagram is also called a "SAFE diagram" (Singh, 2003). Such methods are used to determine whether or not a certain number of stator vanes could excite the impeller to resonant vibrations.

As an example, Figure 6 shows a Campbell Diagram of the complete fourth stage impeller showing all tested modes. The Campbell diagram indicates that it is essentially impossible to avoid every mode in this case. However, the equations of Kushner (1980) would indicate that only certain modes are of concern as the rest would experience phase cancellation.

Figure 7 shows the interference diagram based on FEA predictions for the fourth impeller disk modes. The left vertical axis represents the natural frequencies in Hertz (Hz). The horizontal axis represents the nodal diameter (ND) mode shape of the impeller. The right vertical axis shows excitation harmonics of running speed. The blue diagonal lines represent maximum continuous speed times multiples of the harmonics of running speed shown on the right vertical axis.

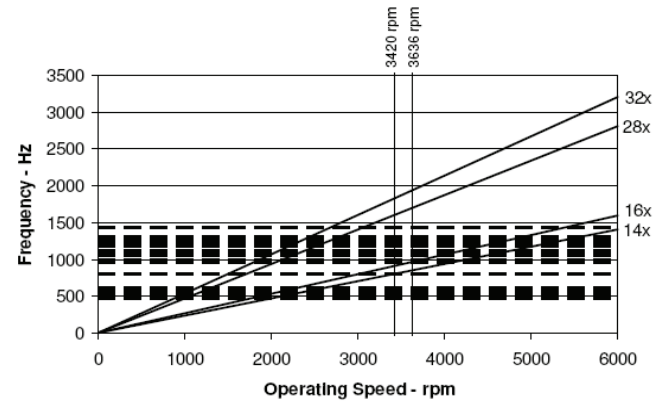


Figure 6. Fourth impeller Campbell Diagram from 2009 failure

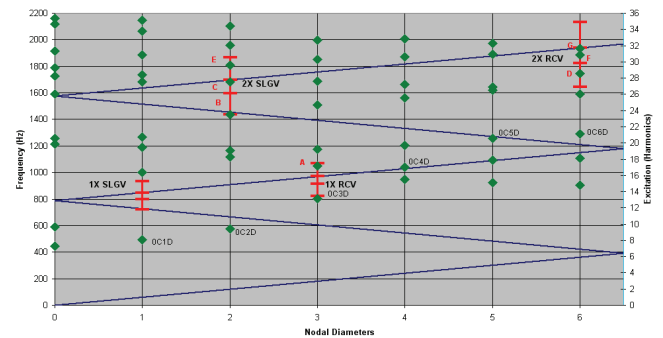


Figure 7 – Fourth Impeller Interference Diagram

The red vertical lines show potential excitation from the 14 side load guide vanes and the 16 return channel vanes upstream of the impeller that has 13 rotating blades. Lines are shown for both one time and two times excitation. Because the compressor can be operated over a variable speed range, the red lines are shown with an upper and lower limit represented by the bar at the top and bottom of the line. The upper limit is for 10% above maximum continuous speed and the lower limit is based on 10% below minimum continuous speed. The plus and minus 10% values are used to account for potential differences between predicted and actual natural frequencies. The inner red bars represent actual continuous operating speed range from 3420 to 3636 rpm.

The green diamonds represent calculated natural frequencies versus nodal diameter mode shape. As an example of how this chart is used, zero nodal diameter modes are not of any concern as number of vanes is not equal to number of rotating blades.

The points labeled OCxD represent the fundamental impeller nodal diameter frequencies. Based on this data seven potential resonant points were identified and are labeled A through G.

Acoustic Resonance

The methods of Kushner (1980) and Singh (1988) are

correct, but they may not be sufficient to explain why some impellers still continue to fail, especially when considering vane wake excitation alone. Different excitation sources such as vortex-shedding and Tyler/Sofrin-modes could be present and coinciding with an acoustic eigenfrequency and mode shape (Konig, et al., 2009 and Petry, et al., 2010) thereby causing amplified impeller excitation. Of course, Kushner's (1980) equations for interaction resonance or Singh's (1988) interference diagram could be extended to consider the acoustic modes as well.

Mistuning

Another potential explanation of impeller failures is mistuning. In a perfectly tuned bladed system, each blade is identical, and cyclic-symmetry can be applied. This leads to a notable characteristic described by He, et al. (2007): each system mode shape consists of identical motion in each sector except for a fixed sector-to-sector phase difference, which is called an interblade phase angle for bladed disks. For the tuned bladed disk, this leads to nodal lines across the disk called nodal diameters, and the system modes are referred to as nodal diameter modes. Mistuning refers to the fact that turbomachinery components, such as fan, compressor and turbine blades, are not identical, but have slightly different frequencies. The frequency differences can be the results of material or dimensional variation inherited from manufacturing tolerance and assembly differences. Corrosion, erosion, deposits, or foreign object damage during machine operation can also increase these frequency differences (Kushner, 2004, McCloskey, 2002). While the advances in design and manufacturing techniques have been able to produce better bladed systems with increased uniformity, i.e. the frequency differences are very small, smaller mistuning does not necessarily translate into smaller amplification factors in the forced vibration cases.

While the deviation of the natural frequency for each blade is small, the mistuning effect could potentially lead to large local vibration level around one or a limited number of blades, increase the stresses of blades compared to the perfectly tuned system, and increasing the likelihood of a high cycle fatigue failure.

Although mistuning can be harmful to a machine, it is not always so. Whitehead (1966) and Ewins (1969) published the first studies of mistuning effect on flutter stability and forced response. In many cases, it was found that mistuning can have a beneficial effect on blade flutter stability in gas turbines. The use of intentional mistuning of bladed disks to reduce their sensitivity to unintentional random mistuning has been well adapted in aircraft gas turbine engines as discussed by Nowinski and Panovsky (2000), and Panovsky and Kielb (2000).

Mistuning also affects the ability for phase cancellation to take place. For turbine disks with relatively short blades in packets, mistuning is often present and the natural frequencies

of each packet for a particular mode can be found a few percent different from packet to packet. The packets can be excited individually and the nodal diameter pattern may not be important due to the phase cancellation of the excitation force. In such a case, phase analysis described by Kushner (2004) is suggested to be used in conjunction with nodal diameter disk mode interference graphing.

The full available literature on the subject of mistuning is quite extensive, especially as it relates to bladed disks used in axial flow machinery. Much less research has been published on mistuning in impellers. Hattori et al. (2008) showed that a large scatter in blade vibration response was observed in the measured data. FEA + CFD simulations for both the tuned system and mistuned system were conducted. The simulations related well to the test results and demonstrated that blade mistuning resulted in the phenomenon of the scattered vibration responses.

One reason that the mistuning of impellers may have received less attention is that it is known that, in the case of axial flow hardware, mistuning is less important if you have fewer blades and impellers often have fewer blades than axial flow stages. For example, both Whitehead (1966, 1998) and Kenyon (2004) found that the maximum increase in vibratory response that you can get from mistuning is proportional to \sqrt{N} , where N is the number of blades on the rotor. As a result, this suggests that an axial flow, high compressor stage with 52 airfoils would potentially have a factor of two higher vibratory stress enhancement from mistuning than this propane machine's fourth impeller that only has 13 blades.

Mistuning is a problem that is essentially probabilistic in its nature. One way to examine this problem would be to perform a vast series of FEA studies, but such computational time and cost can be prohibitive to the design process. Methods to estimate the statistics of the forced response for a population of randomly mistuned bladed disks having the same nominal design include accelerated Monte Carlo simulation, using finite element-based reduced-order models (ROMs). This has been an active area of investigation by a number of researchers including Yang and Griffin (2001) who presented an important reduced-order modeling technique that they called the subset of nominal modes (SNM) method. This was further developed by Feiner and Griffin (2002) for the case of an isolated family of blade-dominated modes. A basic requirement for any study regarding the effect of mistuning on the vibratory response of impellers is to develop methods for measuring and interpreting their vibratory response.

In the next section, a method used to measure the vibratory response of the impeller is described. The extraction of experimental modes from the test data and quantitative comparison with tuned modes calculated from the cyclic symmetric finite element analysis is discussed. It will be shown that the process also provides a method for estimating the stresses in the experimental modes. Furthermore, the approach can be generalized so that it can use displacement

frequency response data to estimate stress frequency response. The approach is validated by applying it to a benchmarked numerical test case. Lastly, the implications of the results will be discussed.

METHOD OF MISTUNING MEASUREMENT

The impeller was mounted to a hub matching the shaft design then secured to a granite table. This was to ensure that the bore of the impeller would be stiffened and provide a closest approximation to the constraints the part experiences in the field. The impeller has 13 blades and was excited by 13 speakers uniformly positioned around the inlet of the impeller, Figure 8. The speakers were positioned such that they excited from the backside of each impeller blade, Figure 9. The vibratory response of the impeller was measured at 650 points using a scanning laser vibrometer. The 650 points corresponded to 50 measurement points per sector: 36 points were located on the blade, and 14 points were located on the cover plate, Figure 10.

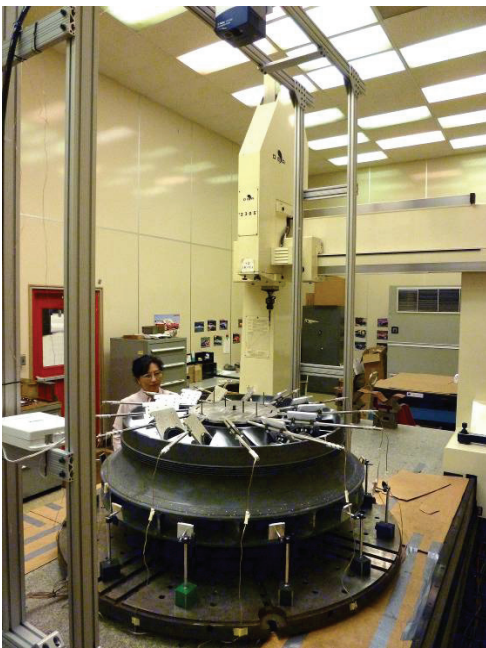


Figure 8 – Traveling Wave Test Setup



Figure 9 – Speaker Positioning

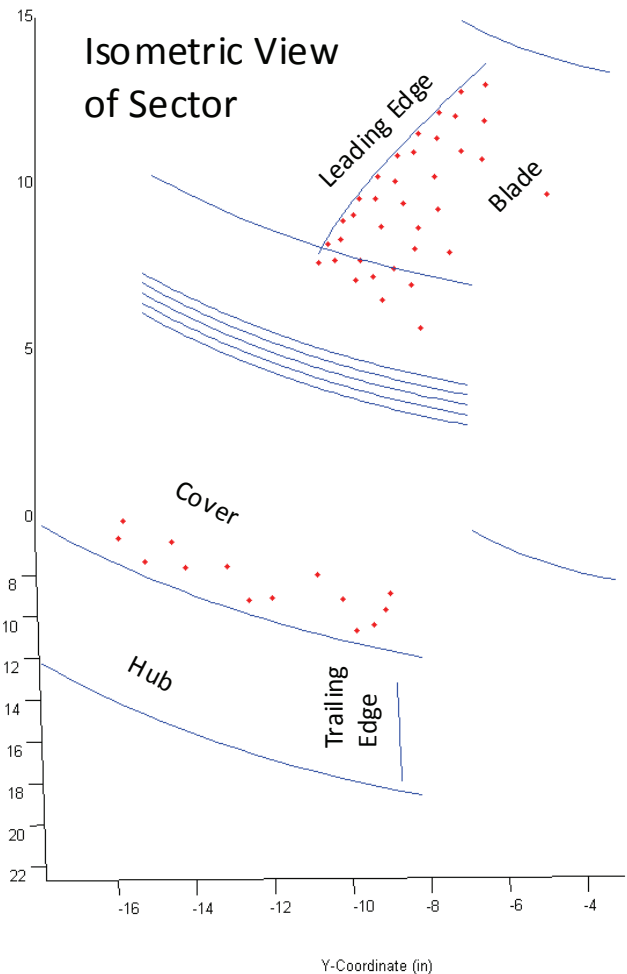


Figure 10 – Measurement Points Indicated By Dots

The vibratory response of the impeller is measured with a traveling wave test. Basically, the impeller remains stationary during the test and is excited acoustically with a set of speakers. The speakers and the input to the speakers are phased to simulate the type of traveling wave that would be caused in the field by the part rotating behind stationary structures. The frequency responses of the blades and cover to the traveling wave excitation are measured at 650 points using a scanning laser vibrometer.

A complete series of “NxN” tests were conducted. The terminology “N” refers to the idea of exciting blade 1 and measuring the vibratory responses of all N blades (N equals 13 in this case), then exciting blade 2 and measuring the vibratory responses of all N blades, then exciting blade 3, etc. In its simplest form, there is only one measurement point on each blade so you have N sets of N measurements when you are finished, hence the NxN terminology. In the case of the covered impeller, we actually did an Nx50N experiment since we excited the system with 13 different speakers and measured 50x13 points for each speaker. The value of NxN testing is that it provides a complete set of tests - in that you can adjust for

phase effects and superimpose the resulting frequency response functions to get the response to any possible traveling wave excitation. A representative set of 650 transfer frequency response functions are shown in Figure 11. Note that the strength of the input to the speakers was chosen to ensure a high signal-to-noise ratio in the output and that the speakers were in their linear range and would not overheat. In addition, the temperature in the laboratory was controlled so that the natural frequencies of the impeller did not drift during the extensive NxN tests that lasted hours.

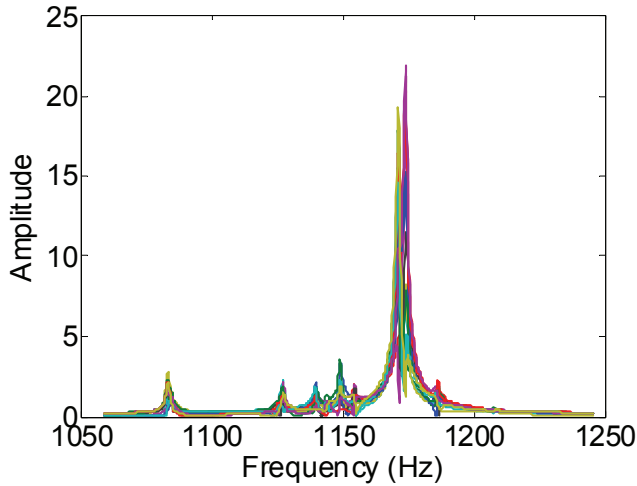


Figure 11 – 650 Frequency Response Functions

A Polytec 400 scanning laser vibrometer was used to make the measurements. Selected nodes were mapped onto the test piece so that the measurements could be directly compared with predictions from the finite element model. In addition, custom firmware determines a unit vector, \vec{u}_m , in the direction of the laser beam at each measurement point so that it knows the direction of the velocity component that is being measured at each point.

Method of Mistuning Characterization

Mistuning identification technology is used to measure the part, analyze the vibration data, and determine the structural properties of the impeller. The properties are used in a reduced order structural model to predict the vibratory response of the tested impeller. It is confirmed that the vibratory response predicted by the reduced order model is consistent with the test data. Both the reduced order structural model and a 360 degree, full rotor, finite element model are used in Monte Carlo simulations to calculate the vibratory responses of a sample of impellers in which the blade frequencies are randomly mistuned. The finite element model requires hours to do a single impeller calculation; however, the reduced order model can calculate the vibratory responses of 100 impellers in one second. As a result, the finite element calculations are used to provide an additional benchmark to confirm that the reduced

order structural model is providing the correct statistical trends. Then the reduced order model is used in a more extensive Monte Carlo simulation to assess how manufacturing variations affect the vibratory response of impellers.

A measure of the statistical behavior of the impeller is the 99th percentile amplitude A99, i.e. 99 impellers out of 100 will have peak amplitudes less than A99. In the literature, it is found that A99 increases as the amount of mistuning increases, eventually reaching a maximum and then typically decreases as the level of mistuning increases, Griffin (2011). This phenomenon occurs to varying degree.

Categorization of Modes

ANSYS was used to calculate mode shapes and natural frequencies of the impeller using a single sector model and applying cyclic symmetric boundary conditions to the edges of the cover plate and base plate. The modal displacements from the FE calculation are vectors, $\vec{\tau}_n$, that prescribe all three components of the motion at each node in the FE model. The scanning laser vibrometer only measures the displacement in the direction of the laser beam at a limited number of measurement points. Since we want to compare the experimental and FE modes, we reduce the resolution of the FE modes to match the experimental modes. This is done by keeping only the displacements of points that were measured and by using the modal displacement vectors to compute the displacement component in the direction of the laser beam. Then the simplified finite element modes, $\vec{\tau}'_n$, have the same number of components as the experimental modes.

One challenge in comparing FE and experimental modes is to know which modes should be compared. If we had a finite element model that perfectly represented the test impeller, then we could order the modes by increasing frequency, assign them a mode number, and compare FE and experimental modes with the same mode number. This won't always work in practice because the frequencies can be very closely spaced and the modes can appear in a different sequence in the FE model than in the actual part. So, instead, we will compare modes that have the most similar shapes. We accomplish this by representing the experimental modes as a linear combination of a subset of the FE modes. If an experimental mode is denoted by the vector $\vec{\varepsilon}$ and the simplified tuned modes by the vectors $\vec{\tau}'_n$ then approximate the experimental mode as a limited sum of the simplified tuned modes, i.e.

$$\vec{\varepsilon} \approx \sum_{n=1}^N \mathbf{c}_n \vec{\tau}'_n \quad (3)$$

This representation of the experimental modes as a sum of tuned modes is essentially the same as the Subset of Nominal Modes (SNM) approach used to develop reduced order models of mistuned bladed disks (Yang, et al., 2001). SNM is an

analytical approach in which blade frequency mistuning is used to calculate the mistuned modes and natural frequencies of the system. In this paper, we find the modal content, \mathbf{c}_n , of a particular experimental mode by using a standard, least squares fit that minimizes the square of the error. We refer to the resulting approximation, i.e. the right hand side of equation (3), as the composite modal vector.

Estimation of Modal Stresses

The following procedure was developed for relating the displacement measurements to stresses. Once the values of \mathbf{c}_n are determined that best fits an experimental mode with a sum of tuned modes, equation (3), the same modal content coefficients are used to estimate the experimental modal stress, i.e. if the modal stress vectors of the tuned modes are $\bar{\sigma}_n$ then the experimental modal stress vector can be estimated as:

$$\bar{\sigma}_{Exp} \approx \sum_{n=1}^N \mathbf{c}_n \bar{\sigma}_n \quad (4)$$

For example, one of the experimental modes from the low frequency regime was decomposed in terms of the FE modes and the resulting modal content. Modal Assurance Criteria (MAC) is used to correlate between the experimental mode shape and the composite modal vector. A value of 100% indicates perfect agreement.

RESULTS

Measurement Objectives

The objective of the measurement was to determine the amount of mistuning at the modes of greatest concern including any effect on phase cancellation assumptions for those modes where phase cancellation was being relied upon. In addition, the mistuning of individual blades needed to be determined along with the additional amplification caused by mistuning.

Disk Mistuning and Confirmation of Phase Cancellation

The 3-ND mode at 784 Hz is shown to split, Figure 12. The difference between the split modes is only 0.2 Hz; therefore, the mistuning level is

$$\frac{f_i - f_{avg}}{f_{avg}} \quad (5),$$

which is a disk mistuning level of only 0.01%. Corresponding mode shapes are indicated in Figure 13.

Most tuned FE modes that are calculated from an axisymmetric finite element model occur in pairs that have the same frequency. These pairs of modes have the same number

of nodal diameters and are often referred to as the “sine” and “cosine” modes. Because the natural frequencies of the two modes are repeated, the mode shape is non-unique, but can always be represented as a linear combination of two linearly independent mode shapes. That is what is happening here; the experimental mode is a linear combination of two repeated sine and cosine modes. Practically, this means that if the modal content has only two components that are right next to each other in the modal content plot, then the experimental mode shape looks like a tuned mode and agrees with the mode shape predicted by the FE model. Mode shapes in the lower frequency range tended to agree fairly well with the mode shapes predicted by the finite element model.

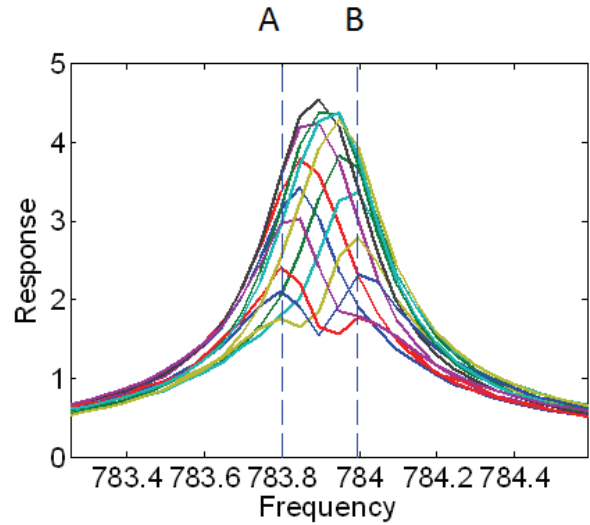


Figure 12 – Impeller Response Around The Cover Plate For 16X Excitation of the 3-ND Disk Mode

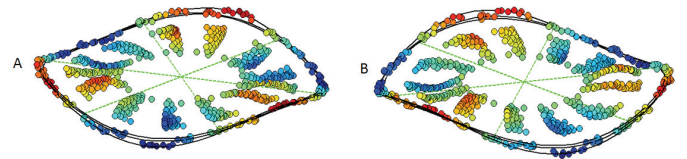


Figure 13 – Impeller Mode Shapes

An examination of the phase cancellation or aliasing effect depicted in Figure 14 shows how a difference of 4 between the number of stationary and disk modes can cancel out the 3-ND mode. This was demonstrated by using the traveling wave testing technique. The amplitude is reduced by a factor of roughly 10 as depicted in Figure 15. Within the mistuning community, phase cancellation is known to work when mode pairs split, but are not distorted. Figure 16 shows that the next nearest nodal diameter frequency is 8% higher, which is quite significant. This would require very large levels of mistuning to cause 3-ND modes to interact with other nodal diameters and for the modes to be significantly distorted.

Blade Mistuning and Examination of the Leading Edge Blade Mode

The process outlined for mistuning characterization, the cumulative probability for blade mistuning, was carried out for the leading edge blade modes. This process begins with the data, Figure 17. Statistical simulations are performed to determine the standard deviation of mistuning using commercial software, EzID, derived from the methods previously documented by Feiner, et al., 2004.

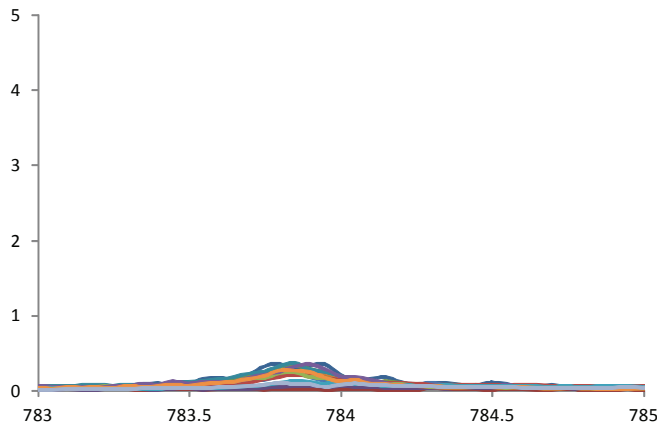


Figure 14 – Impeller Around the Cover Plate For 17X Excitation of the 3-ND Disk Mode.

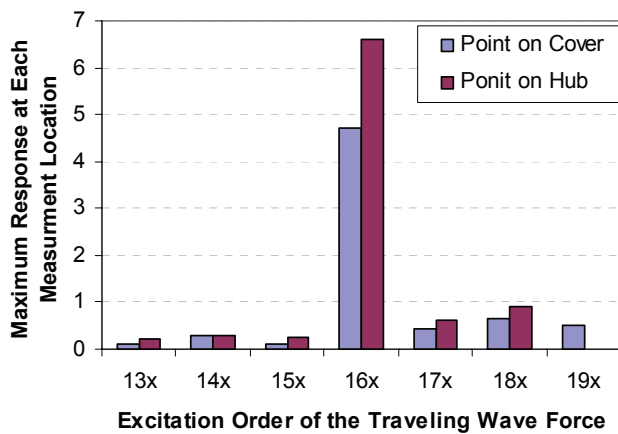


Figure 15 – Maximum Response at Each Measurement Location.

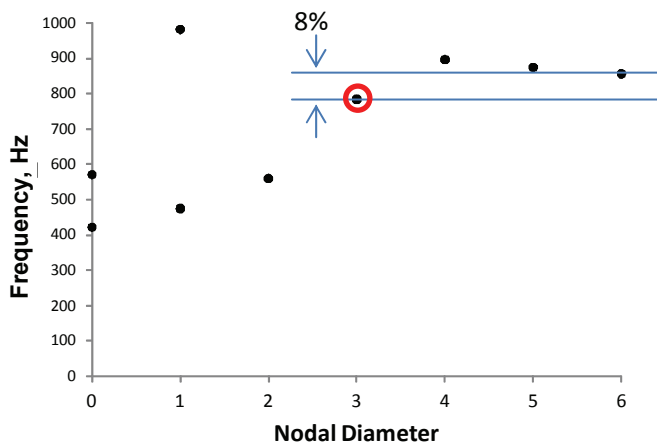


Figure 16 – Percent Margin from the Next Nearest Mode

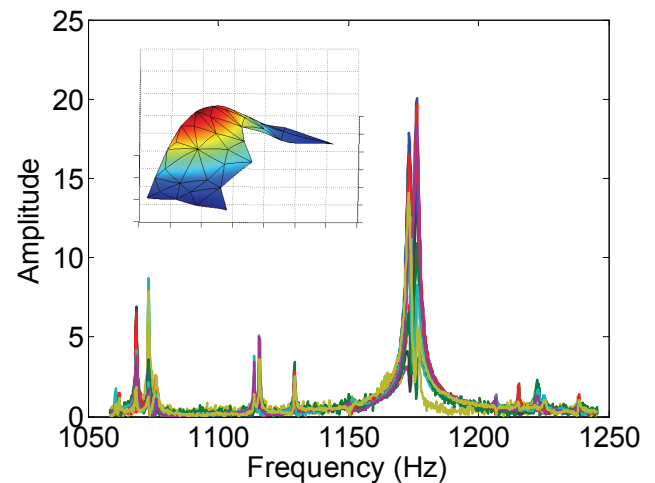


Figure 17 – Blade Test Data

The blade mistuning in this large 48" fourth stage impeller was 0.5%, which is on the level typically seen for blisks (one-piece bladed disks). Based on this 0.5% standard deviation of mistuning, the probability distribution curve can be determined from reduced order modeling techniques where the peak amplitude is picked from the maximum responding blade from each impeller. From this, the vibratory response of 10,000 impellers with 0.5% mistuning can be simulated. Benchmark finite element analyses are also performed using full, 360 degree impeller models to simulate vibratory response of 10 impellers with 0.5% mistuning and picking the peak amplitude of the maximum responding blade of each impeller.

It should be noted that previous blade leading edge modal rap tests identified the blade mistuning level to be 1.8% on an intact fourth impeller, and 1.5%, 2.7%, and 5.6% for failed fourth impellers with cracks. Use of varying amounts of clay to dampen out the excitation of adjacent blades is suspected to contribute to mistuning in the measurements as well as difficulty in performing modal identification by rap test. The impellers with missing pieces had generally higher amounts of mistuning, which is to be expected.

The probability distribution calculated using the reduced order model for the acoustically tested impeller blades is shown in Figure 18 for 0.5% level of mistuning. This indicates a 1% chance that an impeller built from this process will have a blade

amplitude that exceeds 1.35x tuned response. The average impeller would have a 1.15x tuned response. The more detailed FEA (Figure 19) corroborated up to a 1.2x tuned response for the same level of mistuning. Figure 20 shows an A99 curve, which provides the 99th percentile of amplitude magnification as a function of the mistuning. The worst case amplitude magnification is 1.48.

The Goodman Diagram in Figure 21 from White et al., (2011b) had calculated alternating stress values based on Fluid Structure Interaction Analysis (FSI) for the modes of concern. The FSI results, which were used to generate the Goodman Diagram assumed 4.3% aerodynamic modal damping, which was based on typical literature values adjusted for gas density. In reality, the aerodynamic damping could be less, especially during transient operation with excessive mistuning at or near choke. Loss of aerodynamic damping when approaching choke especially with liquids is known in the literature; it can become very low and even negative thereby causing cracks to initiate from the increased response (Kushner, 2004).

White, et. al., (2011a, 2011b) concluded that the responding mode was the mistuned blade mode. Based on the tests, the average impeller blade will have 15% more alternating stress on average due to mistuning as compared to a perfectly tuned impeller. This is well within the allowable limits. For worst case situation corresponding to more than 3% mistuning, and accounting for a probability of 99% of cases being less than the amplitude ratio, the amplitude ratio is less than 1.48. However, even if the vibratory stress were enhanced by 48%, it still does not cross over the Goodman line for the failed case having compressive residual stress, thus showing that the original design had adequate safety factors for normal operation.

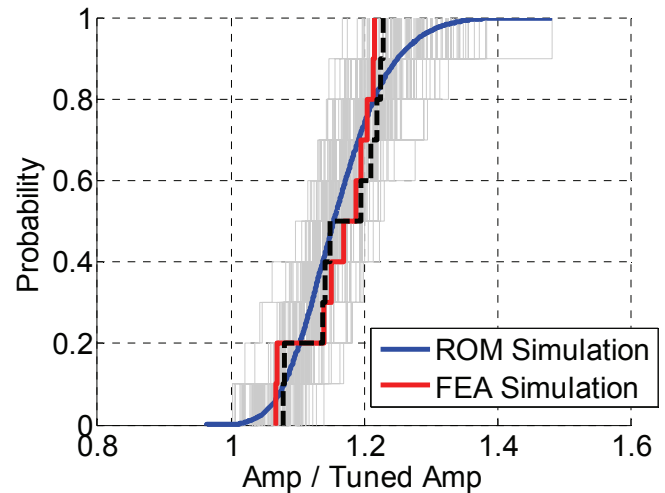


Figure 19 – FEA Versus Reduced Order Model.

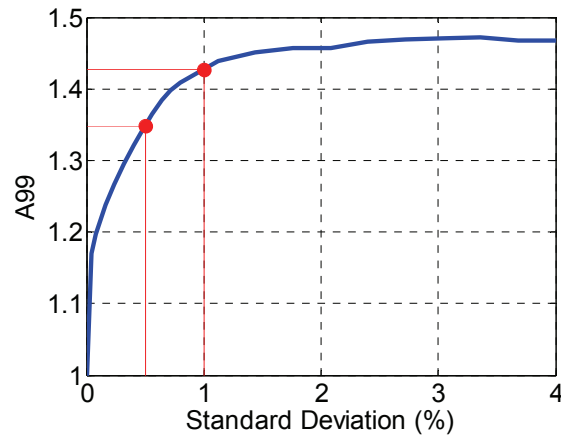


Figure 20 – A99 Curve

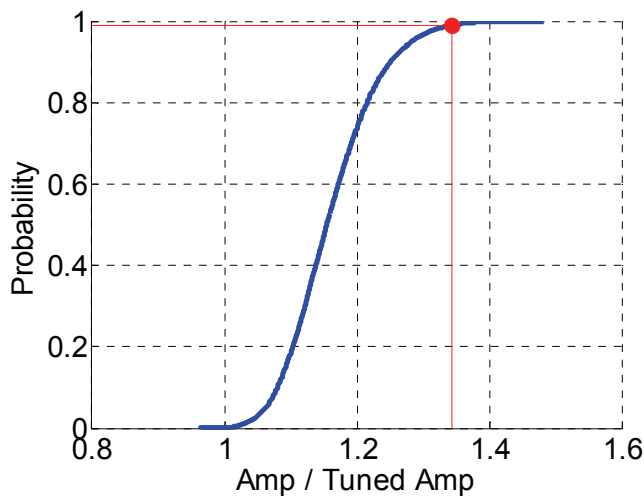


Figure 18 – Impeller Cumulative Distribution

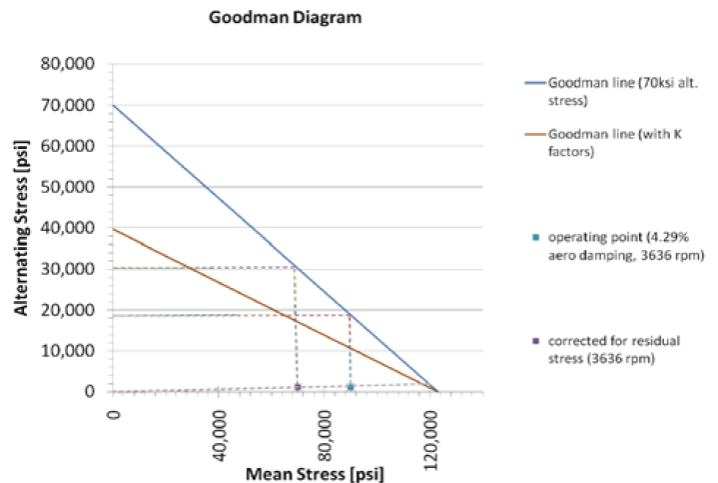


Figure 21 – Fourth Impeller Goodman Diagram (White, et al., 2011b)

CONCLUSIONS

- Experimental measurements for vibratory modes and mistuning were successful.
- Phase cancellation was demonstrated for 784 Hz, 3-ND mode in confirmation of Kushner (1980) and Singh (1988) analysis methods.
- Disk mistuning levels were notably small for the 3-ND mode, near 0.01%.
- Blade mistuning levels for the fundamental leading edge mode were measured to be near 0.5%.
- Analytical results imply that mistuning alone is not an issue for the leading edge blade mode for this impeller, and confirms the conclusions of White, et al. (2011a, 2011b) that a combination of factors were required for these failures to have occurred.
- A probabilistic based approach to designs should consider the mistuning as an item to factor into the dynamic stress calculation.

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