CRITICAL REVIEW OF COMPRESSOR IMPELLER VIBRATION PARAMETERS FOR FAILURE PREVENTION

by

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ABSTRACT

One of the most difficult diagnostic problems for centrifugal compressors is fatigue failure of an impeller, especially those with covers. The structure of a covered impeller is inherently resistant to alternating forces at normal flow conditions and operation, but only if it is designed using proper parameters over and above steady-state stress limits. Critical parameters are reviewed with respect to excitation and response. Besides well-known disk critical speeds, a fundamental parametric equation first published in 1979 is further explained and emphasized.

Occasional failures that occur usually involve certain design features coupled to added excitation from flow variations, either at the extremes of the operating map or from process and control design features and deviations. Test data for process refrigeration units that experienced numerous failures in the 1970's required actual strain gauge results via slip ring data acquisition to prove that liquid ingestion was the cause of high excitation. These critical data gave much confirmation to assist in future design reliability. In a more recent unusual case, a compressor impeller failed within hours after startup; aspects of the original design and process operation are presented along with improvements.

Some cases require special internal instrumentation to solve recurring failures. Documentation of failures including some with liquid ingestion is discussed, along with suggestions for review to mitigate potential failures of repaired units as well as new designs. Also discussed is confirming evidence from other references where applicable. Design improvements, both for older and more efficient impellers, are shown to reduce risk, especially for those with vaned diffusers.

INTRODUCTION

When considering vibration and fatigue, covered radial impellers are perhaps one of the most reliable components for turbomachinery rotors. If very high mechanical tip speeds are required, such as for high pressure-ratio wheels in plant air compressors and turbochargers, shrouds can be eliminated to reduce centrifugal stresses. Single stage open impeller designs can maintain proper clearance of shroud-side blade tips with nearby casings, whereas multistage compressors could have large changes in operating clearance and greater thrust loads.

Incorporating a rotating shroud greatly increases reliability for avoiding vibration response to all sources. The impeller cover (shroud) in effect couples all blades to give fixed-fixed boundary condition instead of much more responsive cantilevered blades. The hub/shroud/blades structure then results in coupled disk modes that can be designed to be difficult to excite. There still are individual blade modes that have high frequencies, and that typically are only of concern for very high flow, high-speed machines.

Whenever a failure does occur in an impeller, damage can be extensive, so proper design parameters to eliminate critical resonances are necessary. For unshrouded impellers, surge and stall can much more easily cause failures even if there is no resonance of blade or disk modes with inlet or diffuser vanes. Nonuniform flow for volute stages will give excitation at various harmonics of speed, especially away from the design point. The main discussion in this paper is for disk modes that can be excited from nonuniform flow, only if there is a matching of phase angles between excitation and response. This can occur either at disk critical speeds or at other resonant speeds for certain combinations of stationary and rotating blades. Explanation of the theory presented by one of the authors, Kushner (1980), is discussed and confirmed with additional field data. Other publications that have discussed blade/vane interaction are also reviewed. Not all causes of impeller fatigue are reviewed in this paper, e.g., the case described by Price and Smith (1999) with compressor inlet pressure pulsations from excitation of an acoustic mode of the gas, concluded to be the cause of impeller three-diameter mode excitation.

EXPLANATION OF DISK MODE EXCITATION

As shown by Grinsted (1951), NASA (1990), and others, turbomachinery disks can have many vibratory modes wherein there can be significant alternating bending stress. In Figure 1 is shown the fundamental two-diameter mode for out-of-plane motion in the axial direction; while in Figure 2 is a two-circle mode with one node at the bore and one toward the tip. Higher modes have more diameter or circle nodal lines and there also are combinations of diameter and circle nodes, generally more difficult to excite. Modes with one-diameter and also one-circle modes are coupled to other disks and the shaft; for these within machines having high bearing damping there is little concern for problems. For two-diameter modes and higher, vibratory moments of diametral modes are balanced within the disk; amplitude cannot be sensed on the shaft, so that test data require direct measurement. High frequencies typically involve plate modes, generally much more difficult to excite. Also, torsional modes for a cover relative to the hub sometimes require review for a shrouded impeller.

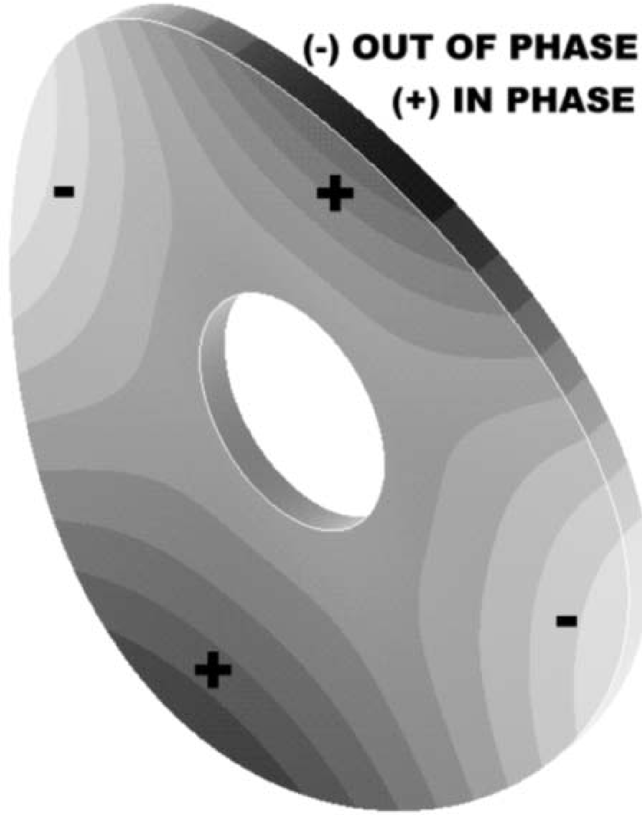

Figure 1. Two-Diameter Mode of Disk Fixed at the Bore.

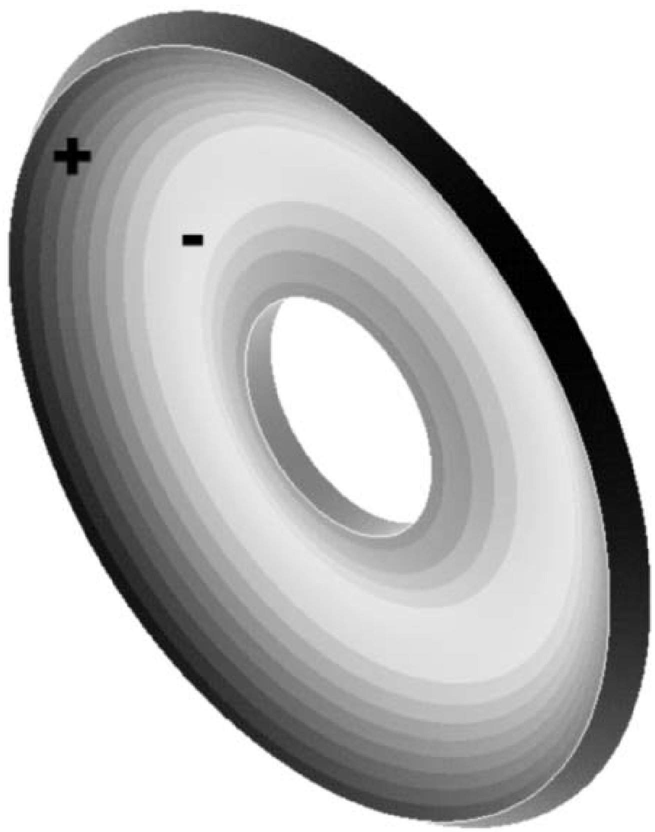

Figure 2. Two-Circle Mode of Disk Fixed at the Bore.

Disk Critical Speeds

Campbell (1924) documented what happened when disks became too thin for steam turbine designs. Resonance occurred at disk critical speeds, defined when rotating speed was equal to the natural frequency of a diametral mode divided by the number of diameters for that mode. In Figure 3 is shown a Campbell diagram where a two-diameter mode at 200 Hz is operating at a disk critical speed of 6000 rpm, or 100 Hz. This speed is also where the "backward wave" frequency ($f_r - 2\omega$) intersects the abscissa. At this speed, relative to any stationary point, there are two relative frequencies of the impeller mode, one at 400 Hz and one of which is zero. In other words, the disk mode shape for zero frequency would "appear" visually to be a standing wave. As the disk is rotating, nodal lines remain fixed to the disk as emphasized by Tobias and Arnold (1957), and also Ewins (1973); there are actually two natural frequencies for each mode due to dynamical imperfection (only one is shown in Figure 3). Another description is by Jay, et al. (1983). Since average amplitudes relative to all circumferential stationary points form a standing cosine wave at disk critical speed, a static nonuniform pressure distribution would excite a disk diametral mode at this speed. It is almost always recommended that disk critical speeds be avoided due to unknowns for overall damping and many effects giving nonuniform flow; for centrifugal compressors to date, avoidance is deemed to be mandatory.

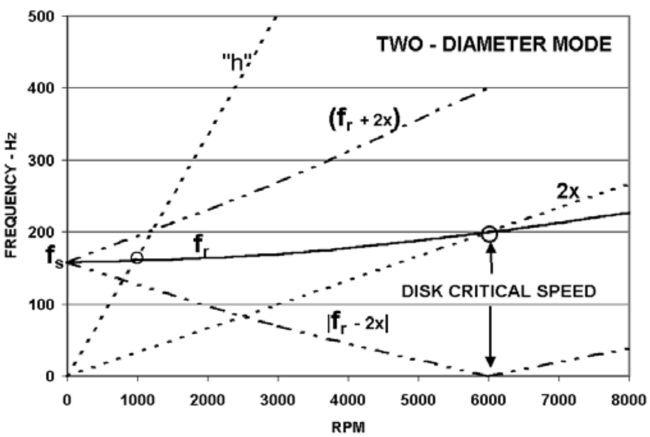

Figure 3. Campbell Diagram for Generic Disk Critical Speed (f_r = natural frequency at speed; f_s = natural frequency at standstill).

Since 1969, a design with disk critical speeds occurred only once in one of the authors' companies—in 1981 for a plant air compressor impeller originally developed by a consortium. In Figure 4 is a Campbell diagram for one of the open (unshrouded) impellers, where the four-diameter mode was resonant with four times running speed, and caused fatigue cracks at the impeller tip. Changes in blade height and outer diameter resulted in a different mode being at a critical speed. Excitation was from nonuniform flow and pressure distribution around the circumference due to the discharge volute; use of vaned diffusers was not sufficient to minimize the feedback from the volute back upstream to the impeller. Harmonic excitation is unavoidable even when proper design methods for volutes are included in centrifugal compressor aerodynamic design methods, e.g., those given by Aungier (2000). The solution was to properly taper the disk profile, also taking into account steady stress changes and selection of blade dimensions for aerodynamic variations. Also shown in Figure 4 is the revised design with the nearest disk critical speed giving sufficient margin. Note that the worst design for this impeller would have been to use four upstream or downstream struts or vanes equal to the number of nodal diameters. Excitation forces at the disk critical speed could have been even higher, even if there was a uniform casing downstream as for interstages on a multistage compressor.

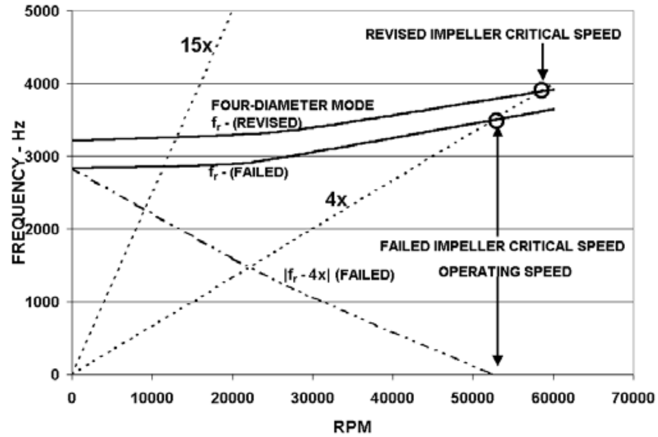

Figure 4. Campbell Diagram for Original and Revised Open Impeller.

Finite Element Procedures

Besides stress analysis and fracture mechanics reviews, finite element analysis (FEA) has provided greatly improved modal analysis. A compressor impeller can be considered a cyclic symmetric structure for FEA. Because of this periodic symmetry only a one-blade sector of the impeller needs to be mathematically modeled. A proper sector represents a pattern that, if repeated B times in cylindrical coordinate space, yields the complete impeller (Figure 5). In this case, B is the number of blades for a complete impeller. Proper boundary conditions are applied depending on the type of analysis or loading on the impeller. By taking advantage of this cyclic symmetry, computer simulations using finite element methods are more efficiently run.

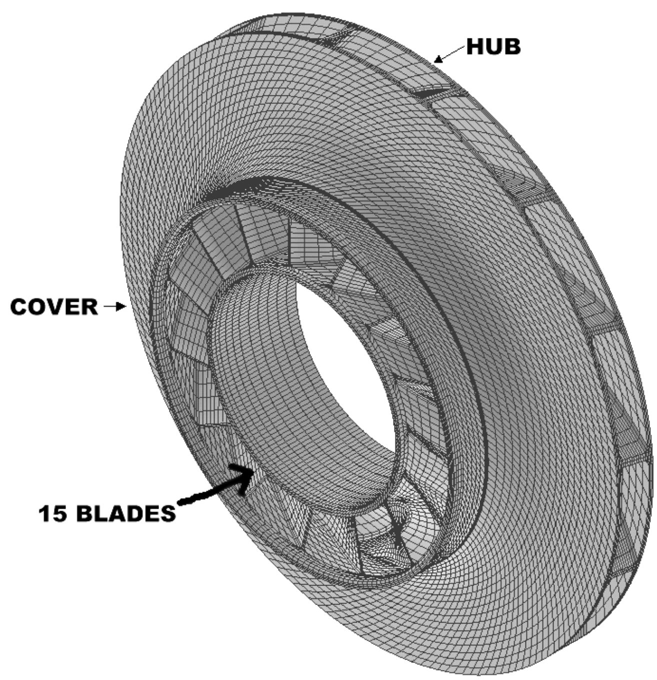

Figure 5. Finite Element Model of a Covered Impeller With a Full Inducer.

For symmetric structural loading such as centrifugal loads due to angular rotation, cyclic symmetric boundary conditions are applied to each edge of the basic sector. The hub is constrained on both cut edges in the tangential direction. Radial and axial deflections on one cut edge are set equal to radial and axial deflections on the opposite cut edge. For covered impellers, the radial, tangential, and axial deflections on one cut edge of the cover are set equal to radial and axial deflections on the opposite cut edge. These boundary conditions can also be used to calculate blade normal modes and both impeller circular and torsional disk modes.

Modal analysis of a cyclic symmetric structure to determine nodal diameter frequencies can also be computed by modeling only one sector (Kohnke, 1999). Nodal diameter mode shapes contain lines of zero out-of-plane displacements crossing the entire disk (Figure 6). These lines of zero displacements are commonly called nodal diameters. For complex structures with cyclic symmetry, such as a compressor impeller, lines of zero displacement might not be observable in a mode shape. The mathematical definition of nodal diameter is more general and does not necessarily correspond to the number of lines of zero displacements through the structure. The number of nodal diameters (*n*) is an integer that determines the variation in the value of a single degree of freedom at points spaced at a circumferential angle equal to the basic sector angle (*Theta*). For a number of nodal diameters equal to *n*, this variation is described by the function, *cos [n (Theta)]*. This definition allows a varying number of waves to exist around the circumference for a given nodal diameter.

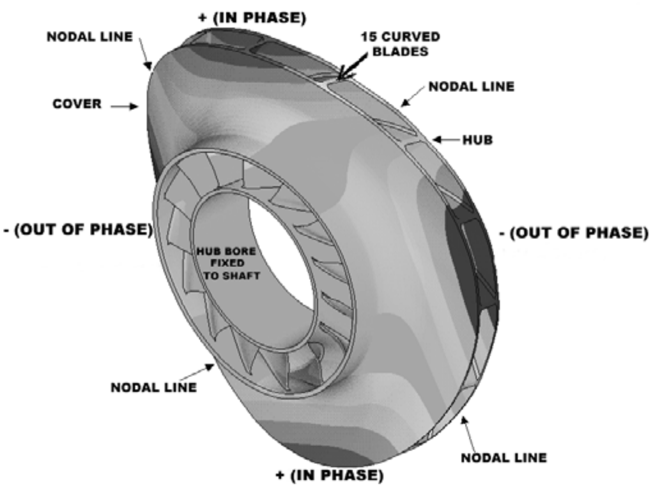

Figure 6. Two-Diameter Mode of a Covered Impeller With a Full Inducer.

In the ANSYS® finite element computer code, Kohnke (1999) also describes a method to perform modal analyses of cyclically symmetric structures. Constraint relationships can be defined to relate the displacements of one edge of the cut boundary to the other edge. This allows for calculation of natural frequencies related to a given number of nodal diameters. The basic sector is used twice to satisfy the required constraint relationships and to obtain nodal displacements. The analysis results will show pairs of frequencies for each nodal diameter solution. Using this technique it is possible to obtain solutions for up to *B/2* nodal diameters. For an even number of blades, *B*, this would be *B/2*, for an odd number of blades this would be *(B − 1)/2*. Methods are given to evaluate these frequencies for both static and dynamic (stress stiffened) conditions. Higher modes require a model of the entire impeller.

For covered impellers, there is usually less concern for reaching disk critical speeds as compared to open impellers, due to inherent stiffening effect of the combined cover/blade/hub construction. However, with the general increased speed for performance and with lighter weights for rotordynamics, vigilance is mandatory. In Figure 7 is a Campbell diagram for a recent covered impeller design used for Figure 5. Maximum design speed of 9550 rpm shown is based on the highest strength materials currently provided for steady-state limits, but as seen in Figure 7, there is still much margin for avoiding the closest disk critical speed near 44,000 rpm.

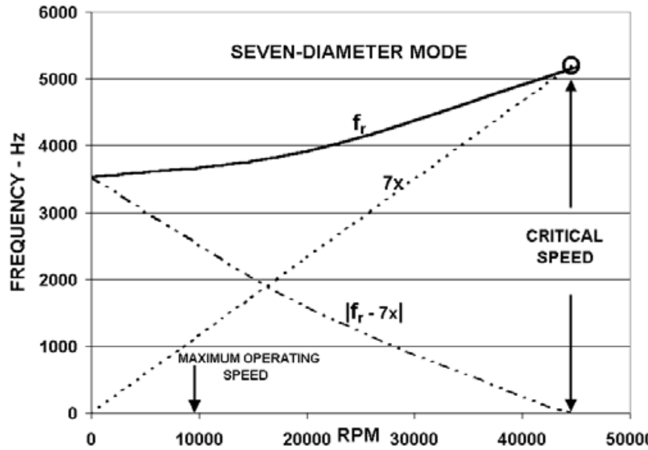

Figure 7. Campbell Diagram for a Covered Impeller Showing Disk Critical Speed Well Above Maximum Design Speed.

Stationary Vane/Rotating Blade Interaction

For resonance at disk critical speeds, there can be high response no matter how many rotating blades are attached to the disk. Besides disk critical speeds, there also can be excitation at harmonics of speed, especially from upstream and downstream vanes. For these cases, it is imperative to consider interaction of rotating blade number with the number of stationary vanes. There was no explanation found in references, e.g., Swearingen and Mafi (1969), for this interaction effect prior to use of following parametric equations, originally given with simplified numerical proof by Kushner (1980):

- Equation (1)—Not at disk critical speeds:

$$|y \cdot S \pm z \cdot B| = n \tag{1a}$$

$$y \cdot S = h \tag{1b}$$

$$f_r = y \cdot S \cdot \omega \tag{1c}$$

- Equation (2)—At disk critical speeds:

$$\text{For } B > 1 \tag{2a}$$

$$y \cdot S = h = n \tag{2b}$$

$$f_r = n \cdot \omega \tag{2c}$$

where:
B = Number of rotating blades
S = Number of stationary elements
f_r = Natural frequency at speed, Hz (includes centrifugal and mass loading with thermal effects)
h = Harmonic of speed
n = Number of diameter nodal lines
y & z = Integers > 0
ω = Rotating speed, Hz

For Equation (1), the parameters of natural frequency at speed for a specific number of diameters for a mode and the required number of stationary vanes and rotating blades all must occur to have a net unbalanced force at resonance due to stationary vanes. Otherwise, there is phase cancellation of forces. The blades transfer nonuniform forces into the disk as they do with the steady forces for transfer of torque. Either of the integers “y” and “z” are typically set equal to one or two, a factor not explained by Gill, et al. (1999), using a similar equation. With both factors equal to one, for all other conditions being equal, response is maximum as each rotating blade effectively has excitation with first harmonic of vane passing frequency for each vibration cycle. With either "y" or "z" greater than one, excitation and response are lower. Equation (1b) is to emphasize that excitation at the harmonic, *h*, is from stationary vanes, not rotating blades. Equation (1) was referred to by Wang, et al. (1999), for an open impeller failure from corrosion fatigue for blade-coupled modes with diametral patterns. However, test data were not presented to show consideration of loss of symmetry for blade-coupled modes from mistuning as was shown by Straub, et al. (1993).

Due to phase cancellation, circle and torsion modes ($n = 0$) cannot be excited, except when $y \cdot S = z \cdot B$. Although indicated but not explained, Gill, et al. (1999), show values that include 10 for stationary elements in a Campbell diagram for a 10-bladed impeller. Having the number of rotating blades equal to vanes as in some emergency sirens could be extremely noisy as well. Two-diameter modes, i.e., with $n = 2$, are generally the most responsive, so that it is optimum to have one number, B or S, even and the other odd; usually the optimum difference is one. A prime number for the number of rotating blades is generally the best choice if aerodynamic performance is not compromised. When the plus sign is used in Equation (1a), the result is typically a very high value of "n," for which modes with many diameters are about impossible to find, let alone excite, at least for current centrifugal compressors. Equation (2) is a special case to emphasize extreme excitation at disk critical speeds; however excitation can also be from other pressure/flow nonuniformities, e.g., from inlet casings and discharge volutes, i.e., setting $S = 1$ and $y =$ *"n."*

For new centrifugal impeller designs and modifications, FEA analysis utilized for modal analysis ensures there is no infringement of speed limits based on design loads, temperatures, H_2S service, etc. Occasional modal testing is only required for a check of FEA or actual manufacturing variations. Thus alternating stresses from speed excursions, startups/shutdowns, and rare, short-duration surging and upsets can easily be accommodated. It is mainly when there is severe amplification of unsteady aerodynamic forces that gives the greatest concern, with much knowledge gained in the 1970's on parameters that must be considered.

FIELD EXPERIENCE CONFIRMING CRITICAL PARAMETRIC EQUATION

Cases Prior to Discovery of the Parametric Equation

In the time period before 1970, some covered impeller failures were ascertained to be due to pure high cycle fatigue, rather than other cause(s), e.g., weld and casting defects, stress corrosion, hydrogen embrittlement, liquid or particle erosion, or corrosion fatigue. As tip speeds were gradually increasing, use of fillet and slot welding along with brazing compared to riveted construction had greatly improved reliability (Cameron and Danowski, 1973). Use of vaned diffusers was rare for large process compressors, and usually the true cause of the failure was unknown. Disk mode resonance had been evaluated using shakers to produce sand patterns, but it was difficult to believe that inlet vanes could excite a disk mode enough to cause fatigue, and blades were fairly short with very high natural frequencies. Usually no changes were made for repairs other than perhaps additional care with welding and improved materials, along with users being more careful with operation—especially number of and time period of surging. Most did not recur; for one case where a second failure did occur it was decided to arbitrarily change the number of rotating blades and it solved the problem. Test data of diametral modes eventually helped verify Equation (1) as:

- For the failed impellers, using the Campbell diagram in Figure 8, the seven-diameter mode was resonant near 4750 rpm, and the absolute value of difference between 25 blades and two times 16 inlet vanes was seven; and
- For the revised impeller, increase from 25 to 29 thinner blades still had resonance of the seven-diameter mode, but the difference between blades and two times inlet vanes was no longer seven. Other modes of the revised impeller were also free of response based on not conforming to Equation (1).

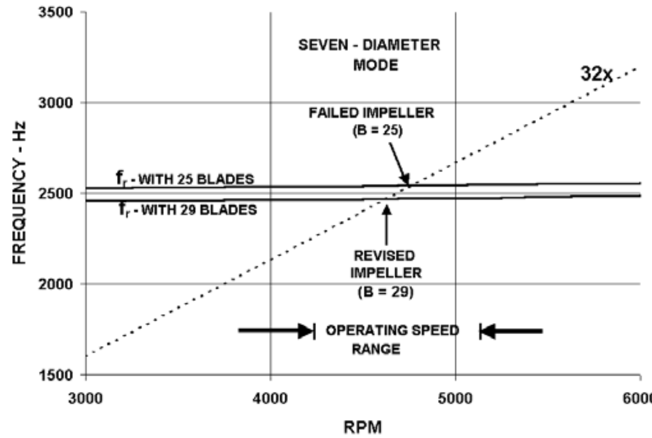

Figure 8. Campbell Diagram for Failed Impeller (25 Rotating Blades) and Modified Impeller (29 Rotating Blades).

Confirming Field Data—Case A

Numerous repetitive failures of four identical refrigeration compressor trains operating independently at a plant in the 1970's eventually required actual strain gauge data on impellers to prove that liquid ingestion was the cause of severe excitation. The four covered impellers per high-pressure compressor rotor were all of the same family of designs, with a steam turbine variable speed drive. Until strain gauge data were obtained on two rotors, the cause eluded proof from previous investigations including assistance from several consultants, including the renowned Professor Den Hartog (1954) who discusses disk modes. Cracking both near the eye and near the periphery, or sometimes only at one location, compounded the failure analysis. Use of internal dynamic pressure probes had not shown potential excitation such as from a high number of stall cells rotating at a slow speed. Fracture mechanics analysis of crack surfaces showed propagating stress range up to 65 ksi, based on striation measurement; with striation spacing per cycle versus stress intensity being verified by special tests on welded sections simulating impeller construction. The gas mixture had many high molecular weight components that gave ease of formation of condensate in the piping and casing. In fact, initially the four larger low-pressure machines had some fatigue cracks that did not repeat.

Strain gauge real-time monitoring showed that the high-pressure machines required liquids to cause crack initiation for the five-diameter mode resulting in cracking toward the periphery. These data proved beyond a doubt that Equation (1) is correct. In Figure 9 is the Campbell diagram for the last stage impeller, showing that the five-diameter mode was resonant with the second harmonic of inlet vanes in the preceding return channel, i.e., $y = 2$. A match is verified for Equation (1), i.e., with $S = 16$, $y = 2$, $B = 27$, $z = 1$; then $(2 \cdot 16 - 1 \cdot 27) = 5$. It was only with liquids intentionally ingested in large quantities for test purposes, and especially while approaching surge, where dynamic stresses peaked at very high levels. Testing with conditions as dry as possible resulted in much lower stresses including flow near the surge point. Five-diameter mode resonance was very sharp (amplification factor near 300), and it required a careful slow speed sweep to reach maximum value. The first stage impeller, also with 27 blades, had 14 inlet guide vanes to straighten flow from the 90 degree inlet. It never cracked as it did not satisfy Equation (1): the 13-diameter mode, $(1 \cdot 27 - 1 \cdot 14) = 13$, and even the one-diameter mode, $(2 \cdot 14 - 1 \cdot 27) = 1$, were both far removed from resonance.

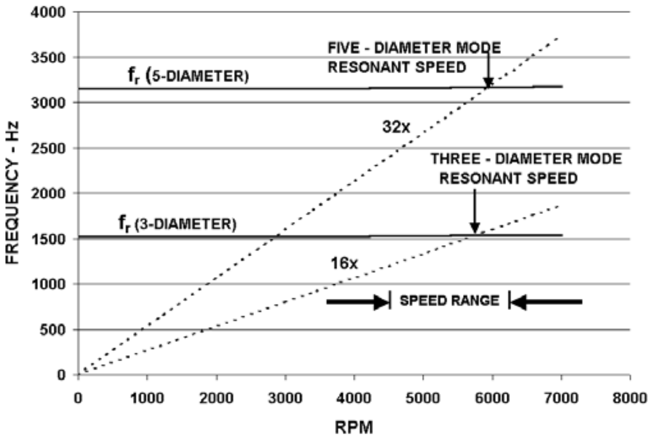

Figure 9. Campbell Diagram for Case A, Responding Five-Diameter Mode and Vibration-Free, Three-Diameter Mode.

Also verified by strain gauge data, other modes did not respond at resonance with or without liquid, including the three-diameter mode since for $y = z = 1$, $(S - B) = (16 - 27) \neq 3$, with resonance in the speed range with the first harmonic of vane passing frequency (Figure 9). At first, there were failures of the last three impellers, each absorbing about 7500 hp; process operation including incidence and time period of surging then was gradually improved. With better operation along with strength modifications, using full penetration welds with modified weld rod and shot peening, the last failures were only on the fourth stage impeller and sometimes the third, showing that effect of liquids extends downstream delaying vaporization. Cause of the other cracks initiated near the eye of impellers at stages 2, 3, and 4 was shown by strain gauge data to be from transient response of the two-diameter mode at maximum flow conditions. This occurred only for some tests when opening the recycle valve after intentionally accumulating liquids in the piping upstream of the valve. The final solution for both modes was to remove the potential for large ingestion of liquids especially the transient at overload, along with better avoidance of surging. Concurrently the number of rotating blades was changed eliminating vane/blade interaction confirmed by Equation (1), with a semi-inducer impeller that also reduced stresses at the eye.

Mass loading can affect disk natural frequencies; a test of a small aluminum, unshrouded impeller for a pump had a 20 to 25 percent reduction when immersed in oil with a density of 55 lb/ft^3. Review of strain gauge data for Case A showed negligible change in natural frequencies due to mass loading for gas density near 2.5 lb/ft^3, compared to tests in air (0.07 lb/ft^3). This is in contrast to the greater than 35 percent reduction reported by Gill, et al. (1999), for a CO_2 compressor impeller with estimated gas density of 15 lb/ft^3. Perhaps that report giving FEA values assumed very tight clearances or a seal at impeller tips, forcing the frequencies down so low; otherwise it is difficult to understand. No other reference found, including Borer, et al. (1997), contains information on frequency reduction for high gas density compressors.

More Recent Field Experience—Case B

In 1997, a newly installed propylene refrigeration compressor tripped on high vibration after being in service for just 20 hours. The unit was a refurbished, two-section, centrifugal design with new diaphragms and a new (spared) rotor driven by a constant-speed, induction motor and gear. During this 20-hour period the compressor was operating entirely on false load with low vibration levels, and nitrogen was bled from the receiver drum for the first couple of hours of operation to remove any inert gas from the closed loop system. The first suction and side load suction pressure and temperature were significantly higher than normal. The compressor ran under these conditions for the remainder of the 20-hour duration before tripping on high radial vibration at both load bearings. The decision was made to inspect the compressor internals prior to restarting.

Inspection of the compressor found that the cause of the high vibration was a failure of the fifth stage (last stage) impeller. As shown in Figure 10, a triangular section of the impeller back plate was missing, apparently due to a crack that initiated from a weld on one of the impeller vane tips. Several cracks were found near each vane tip on the toe of the fillet welds, and resonance of an impeller natural frequency was suspected to be the cause. The failed impeller was a "cutback" design, meaning that the blade tips did not extend all the way to the outside diameter of the cover plate and back plate. This is often done to reduce head from an impeller without having to change the diaphragm design; in this case the amount of cutback used was near the vendor's maximum limit. The immediate plan was to cut down the diameter of the back plate and cover plate on the spare rotor impeller so that the OD of the impeller is flush with the vane tips, thus stiffer to give higher resonant frequencies. The vendor agreed and drawings were initiated for impeller and diaphragm modifications; however, it was decided to leave enough material to eliminate machining in the welded areas of the impeller. It was agreed to also analyze both the failed impeller and the spare impeller before implementing any modifications.

Figure 10. Damage at Tip of Cutback Impeller for Case B.

Both failed and spare impellers were flown to the vendor for modal frequency analysis before conducting a dimensional and metallurgical analysis. No metallurgical problems were found with the impellers; the failure was probably due to fatigue propagation. Numerous cracks were found at the impeller cutback area on both hub and cover sides of the blade, with initiation points at the toe of the weld, as shown in Figure 11. It was shown that cracking was not due to any preexisting defects, with an example shown in Figure 12. Steady-state FEA stress analysis resulted in very low values, especially at the cutback area, with 15.4 ksi at the crack initiation point. FEA modeling and modal analysis testing using a two-channel fast fourier transform (FFT) system did indicate several plate mode natural frequencies near blade passing frequency and upstream return channel vane passing frequency. There were no vanes in the diffuser to excite the tip of the impeller. For both the failed and modified impellers:

- Disk critical speeds were much higher than actual speed.

- With 18 upstream return channel vanes and 15 blades, using Equation (1) with both integers "y" and "z" set to one, the most responsive mode is the three-diameter, but it was far from resonance. Even when using two for either of the integers that results in less excitation and response, the resulting six- and twelve-diameter modes also did not have resonance.

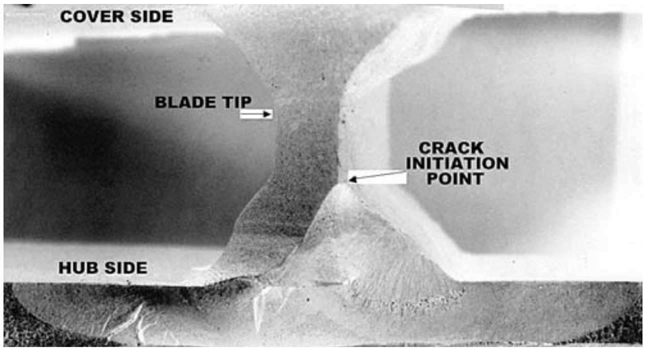

Figure 11. Crack Initiation Area of Cutback Impeller for Case B.

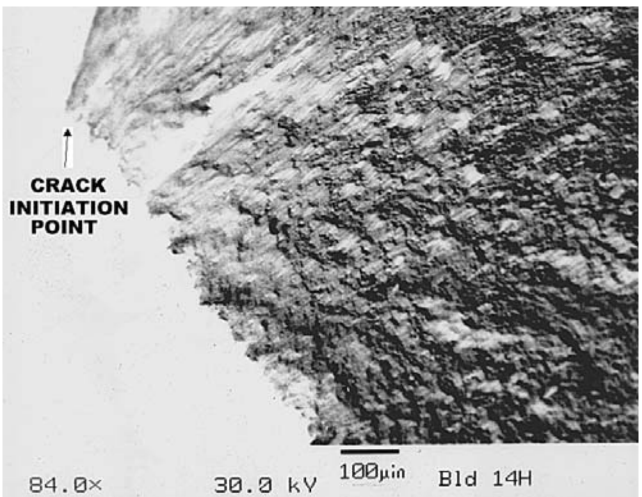

Figure 12. One of the Crack Locations of Case B Cutback Impeller Showing No Preexisting Defects.

The spare impeller was machined, over-speed tested, and balanced; the spare rotor was reassembled, diaphragm modified, and the compressor was reassembled and successfully placed in service only 11 days after the failure. A completely new spare impeller was tested and obtained within about three months. In the meantime, operational review indicated that the compressor had been run in an overload state. Rated power of the motor was much higher than compressor design requirements, so it could be a common spare with that of another compressor train at the site. A review of the process data indicated that motor amps, inlet pressures, and inlet temperatures were high; however, the discharge pressure, discharge temperature, and flowrates were low. These data did not make sense, so that condition of all instrumentation around the compressor was checked. These checks revealed that the measured flowrates and the compressor discharge temperature were incorrect. The remaining pressure, temperature, and load data were used to determine the operating point of each compression section.

After reviewing the vendor's preliminary failure analysis report, it was necessary to fully understand why the high load condition did not correlate with the aerodynamic design curves. A closer review of the operating data prior to the failure indicated that the density of the propylene was very high at the first suction and side load suction. Inlet pressures and temperatures were both elevated relative to design, but the impact of the increased pressure was more significant than the elevated temperature. Realizing that the high densities could possibly explain the higher loads, a new set of design curves was provided for each compression section based on the inlet conditions prior to failure.

The new compressor curves did show a substantial increase in the load and differential pressure curves, but the head and efficiency curves changed only slightly. For a given inlet flow, required power increased 50 to 60 percent in the first section and 25 to 30 percent in the second section. Using the available data proven to be valid by the post-failure instrument inspections, a closer correlation was found between the new curves and the operating conditions prior to failure. The estimated inlet flow to the first suction was 5500 icfm. This flow position, which is on the design curve, is 22 percent higher than the rated flow and 2 percent lower than end of curve flow. The estimated flow through the second compression section is 7950 icfm. This flow position is 5 percent beyond the end of the curve and 15 percent higher than the original rated flow; performance curves are in Figures 13 and 14. Final checks of calculated and measured motor powers were within 2 percent of each other.

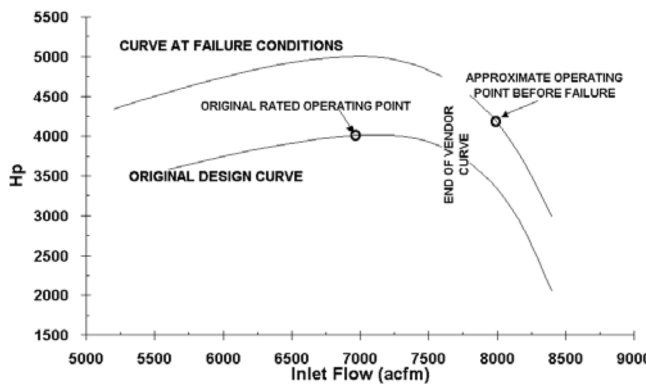

Figure 13. Case B, Compressor Second Section—Power Versus Inlet Flow.

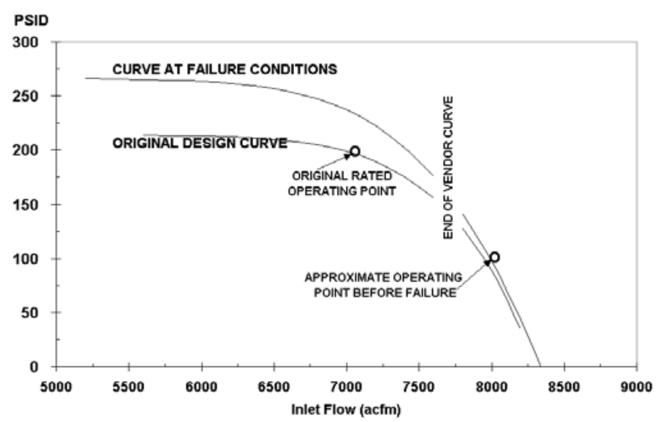

Figure 14. Case B, Compressor Second Section—Pressure Rise Versus Inlet Flow.

Both sections of the propylene compressor were operating at or beyond the end of vendor's estimated "end of curve" inlet flow. Although the flows were higher than design flows and near or in choke conditions, the user's experience with operating multistage centrifugal compressors with enclosed impellers in or near choke is not normally detrimental to the compressor, unlike low flow or surge. The amount of superheat in the gas was also substantially higher prior to the failure when compared to design conditions, so liquid or excessive vapor ingestion were considered unlikely. The vendor did not deviate from their design criteria with this impeller, and the user (and others) does have many compressors in service with cutback impellers that have not failed. The failure was probably caused by excitation of an impeller natural frequency, but the performance data alone could not conclusively determine the origin of the exciting force. The vendor confirmed that the last stage impeller would be the one that would be choked at the entrance, i.e., at the throat of the blades; in Figure 15 is shown that the operating point was at the end of the flow curve. The last impeller is also in a volute stage where nonuniform flow would also inherently be higher.

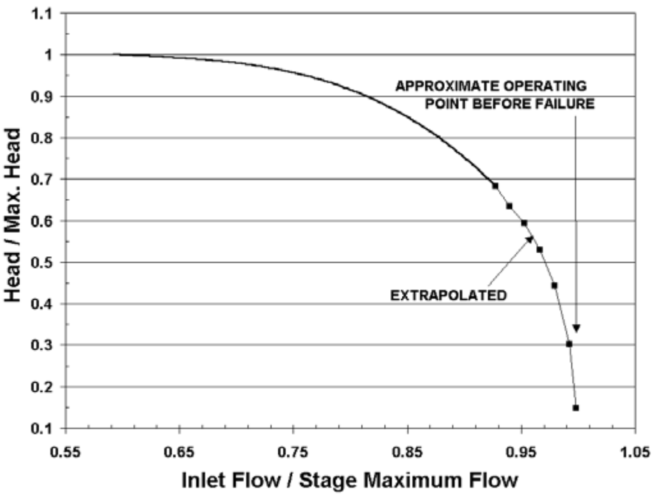

Figure 15. Case B, Head Ratio Versus Inlet Flow Ratio for Last Stage Impeller.

The only resonance found was for plate modes excited by return vanes at the impeller inlet, for which wake effects should dissipate greatly by the time flow reaches the cutback area that has high vibration. There were many plate-type modes at the impeller tip, but the most likely plate mode that is resonant with upstream vanes is shown in Figure 16, and for which almost all the vibratory motion is toward the periphery. It would be difficult to have high inlet vane excitation for this mode unless there were liquids that aggravated the wakes. The corollary is the field experience previously described above for Case A, where toward the end of time period of failures it was the last stage impeller that suffered most and worst failures due to severe liquid ingestion. Liquids can transgress through upstream stages that add heat input, reaching the last stage to greatly increase excitation. As 10 minutes time equates to over a million cycles for 18 times rotating speed excitation, a short time period may have at least initiated the cracks with liquids inadvertently added to increase flow, and with recycle operation not really required since instruments were in error. There were no demisters in the knockout drums and there also may have been buildup upstream of recycle valves as there was in Case A, possibly exciting the two-diameter mode. A constant speed, motor drive would help to ensure operation at the exact resonance point also assisting with the failure occurring in such a short time. Dynamic stress for this mode is maximum value at the actual fatigue initiation points as shown in Figure 17. The pattern shown in Figure 16 has hub sections out of phase with the cover at the tip, with a basic three-diameter pattern. There will be tangential blade motion for this mode, with interaction indicated by Equation (1) as there are 18 upstream return vanes and 15 rotating blades: *(18 − 15 = 3)*. The revised impeller avoids all resonance possibilities, and field instrumentation and controls were modified for the first restart and operation, with no problems during the past three years of continuous operation.

Some Lessons Learned From Case B

- Although a cutback impeller design produces a weaker mechanical structure at the tip, there is extensive problem-free operating experience with this type of impeller; however more care must be used in reducing stress concentrations and also in operation, especially with potential for liquid ingestion.

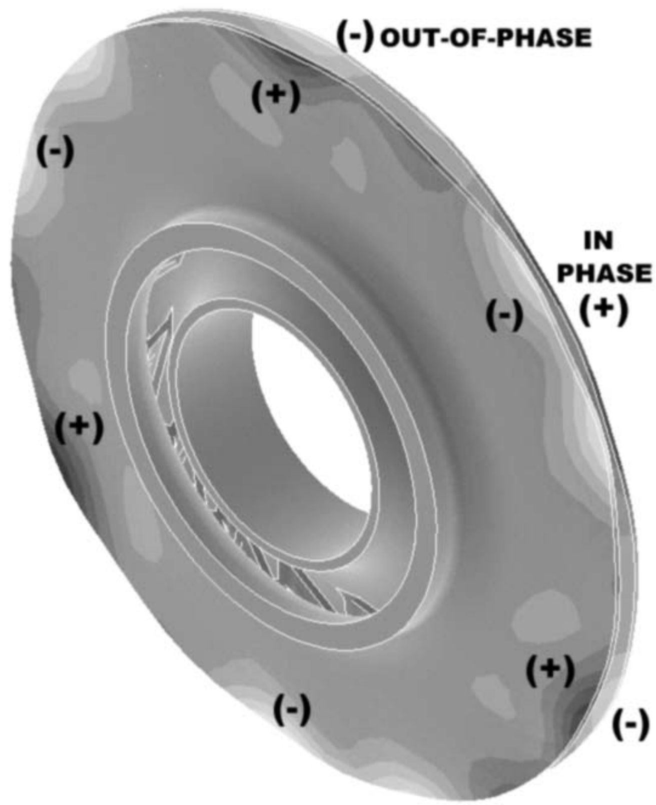

Figure 16. Vibratory Mode Shape During Plate-Type Resonance for Cutback Impeller.

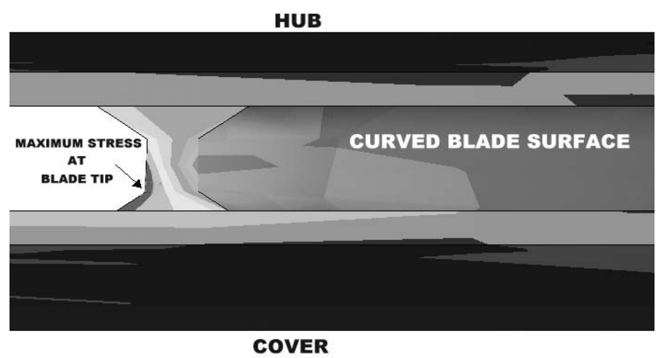

Figure 17. Dynamic Stress Pattern at Tip of Cutback Impeller for Plate-Type Resonant Mode.

- Calibration errors affect the ability of the surge control code to properly operate the compressor. With instrumentation in good working order, the surge control should be sufficient protection to keep the compressor in a good operating flow range, but the addition of low section pressure ratio alarms could be an easy change that would allow additional machine monitoring.
- Calculation of compression section head and efficiency is very difficult when side-loads are present because of the uncertainty of the mixing conditions, but the pressure ratio across each section is stable regardless of suction conditions.
- Pressure ratio might not always be a good indicator of surge conditions since the ratio changes little with flow at low section flows, but the pressure ratio is sensitive to flow changes at high

section flows. Therefore, adding low volumetric flow alarms can give better protection against surge and low section pressure ratio alarms are better for protection against severe choke. Motor load alarms would also normally be effective, but some services can produce significant density changes at the compressor suctions. Suction density changes will produce a variety of load requirements at the same volumetric flow condition. It is recommended that this monitoring philosophy be considered for startups and operational code logic to improve reliability, especially where system resistance can be low to force a stage into deep choke.

OTHER DOCUMENTATION OF THEORY

Since there are not many documented cases found for complete impeller failure analysis, it is difficult to extract all information required especially relative to blade and vane numbers and effects of blade mistuning for open impellers. For example, the papers by Bultzo (1975), VanLaningham and Wood (1979), and that on impeller overload by Borer, et al. (1997), do not give Campbell diagrams. Some references however do give additional direct or inferred information.

• Results given by Straub, et al. (1993), for an unshrouded impeller for air service with about 1600 ft/sec tip speed show coupled blade modes, where mistuning explained response of resonant blade modes as the likely cause of cracks at the inducer. But there is no explanation for higher disk modes that could have caused the other failure shown at the impeller tip, with vanes in the scroll downstream.

• For failures at volute stages, the potential for disk critical speeds, or matching of a diameter mode with nonuniform flow pattern has not always been fully documented. Besides offering aerodynamic efficiency advantages, using vaned diffusers can potentially solve problems, either by reducing interaction of the volute and thus lower harmonic nonuniform loads, or by moving the choke point to the diffuser throat; but Equation (1) must be checked. Diffuser vane excitation, including interaction, is also a subject discussed in general descriptions on the Internet for problems solved. However, conclusion by Jansen and Fetfatsidis (1999) for an open impeller is in direct conflict with use and interpretation of Equation (1), as resonance for 15-diameter mode shown at diffuser vane frequency is plotted at the "backward moving wave" frequency. Not discussed is the interaction effect for the 14-diameter mode that also could have been in resonance with two times diffuser vanes, satisfying Equation (1); i.e., with $S = 15$, $y = 2$, $B = 16$, and $z = 1$, then $(2 \times 15 - 16) = 14$, a match for the 14-diameter mode.

• Axial flow turbines or compressors rarely meet Equation (1), due to much higher blade numbers, and usually higher values for difference between blade and vane numbers. However, a case described by Jay, et al. (1983), shows strain gauge results for turbine disks where the difference between inlet vanes and rotating blades corresponded to excite modes with four diameters, confirming Equation (1).

• For open impellers, individual blade modes are just as important as disk and coupled-blade modes, both for industrial applications (Kushner, 1996) and for university research designs (Jin, et al., 1995). Forces from and frequencies of rotating stall are much more of a risk compared with those for covered impellers. Fixed-fixed blade frequencies for designs with covers are much more difficult to excite, including modern designs with full inducers such as described by Hardin and Boal (1999).

• Baade (1998) discusses fan blade coupled modes using phase analysis. Included is nonuniform flow excitation of four-bladed fans for:

- Third harmonic excitation of one-diameter modes,
- Second harmonic excitation of the two-diameter mode, and
- Fourth harmonic excitation of zero-diameter (one circle) mode.

An equation similar to Equation (1) is presented, referring to ANSI/ASHRAE Standard 87.1 (1992). Results for accelerometers on the housing showed peaks at four per revolution (blade pass frequency) for resonance of the one-diameter and zero-diameter, but not the two-diameter mode, which had vibratory moments balanced within the disk.

• NASA (1990) gives a treatise on possible causes of failure of a high-pressure turbopump impeller, with a cover and combination of long, medium, and short rotating blades. Conclusions were that some modes were the possible cause due to matching for difference between diffuser vane and rotating blade numbers, but not for modes excited by inlet vanes. Results were analytical and the reference does not provide final modifications or the fluid mass loading effect on frequencies.

CONCLUSIONS

• It is doubtful that unshrouded impellers could survive much of the most severe off-design operation that covered impellers have withstood over the years in process applications.

• Cantilevered blade modes on open (unshrouded) impellers are always critical to review; covered impellers have fixed-fixed modes that have much higher frequencies. Blade-coupled modes and mistuning also present added considerations.

• Disk critical speeds are usually more of a potential design problem for open impellers especially if there is nonuniform flow such as due to a downstream volute. Severe excitation could occur when number of or harmonics of stationary elements matches number of diameters for the particular diameter mode.

• Interaction of rotating blades and stationary vanes must always be analyzed if there is a failure of either blade or disk surfaces. Disk modes can cause cracking initially confined to blade surfaces, so that mode shape patterns of dynamic stresses must be analyzed. Typically, failures at mid-height of a disk or near the tip are due to higher nodal diameter or plate-type modes; conversely, cracking at or near the eye (or bore of a turbine disk) leads to the two-diameter mode as the prime suspect mode. Patterns around the circumference are difficult to use for correlation, as nodal lines move slightly as cracks progress, and there are two modes close together for each diametral mode from dynamical imperfection.

• Inlet vane excitation at the impeller eye is usually of extremely low risk for excitation of disk modes for covered impellers, unless there are unusually high loads such as from liquids. Many compressors have resonance meeting Equation (1) that is not detrimental, thankfully, since impeller and return channel families of designs are mixed giving many combinations of rotating and stationary blade numbers. Potential for excitation of the two-diameter mode that has higher transmissibility at the eye is reduced by having one number even and the other odd. Excitation at first stage impellers can be greatly reduced for compressors having 90 degree turn inlets designed using CFD analysis.

• There are and will continue to be unknown causes of some impeller failures, due to the extreme difficulty in gathering data for vibration that is confined to the disk for modes with two or more nodal diameters. Using equations given above can assist in defining most likely causes. Whenever there is absence of running at a disk critical speed, no potential interaction resonance per Equation (1), or prime resonance of an individual blade mode, then transients (upsets) should be suspected for causing excessive excitation.

• For gas processes, system resistance normally prevents severe impeller overload at high flow. Compressors in parallel and other operations with reduced system resistance, including startup, where an impeller can actually choke should be avoided for significant time periods accumulating high number of cycles. This is especially true if there is likelihood of liquid ingestion.

• For centrifugal compressor designs with vaned diffusers and similarly for inlet vanes near the tip of radial inflow turbines, there are optimum numbers of vanes that should always be selected based on both mechanical and aerodynamic limits. Diffuser vanes can be properly designed to minimize effects of volutes on upstream impellers and force the choke point to the diffuser instead of inside the impeller. It is essential to avoid high response resonance at diffuser vane passing frequency and harmonics based on Equation (1).

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