

FATIGUE FAILURES OF COMPRESSOR IMPELLERS AND RESONANCE EXCITATION TESTING

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ABSTRACT

The similarity of failures of impellers in centrifugal compressors soon after initial start up and failures of impellers in other compressors after about seven years of successful operation and the test programs to define the problems are described.

Early failure of an impeller in a turbine driven air compressor followed by other failures in parallel constant speed process compressors installed at another Union Carbide pro-

cess plant indicated resonance or response at known excitation frequencies. However, fatigue failure of an impeller without a known source of excitation soon after initial start-up was followed by an identical failure about four years later in the same constant speed machine. Concurrently, an impeller failed by fatigue in a turbine driven process compressor after about seven years of satisfactory operation. These failures led to extensive testing and quality control programs at several manufacturers' plants for both new equipment and replacement impellers plus a revitalized series of field tests to identify the excitation sources.

Shaker testing and field testing were effective in determining that, while the resonant frequencies and modal patterns may not correspond with any known source of excitation, the fluid flow at various load and speed conditions can excite a resonance and lead to failure.

Both the shop and field testing, including corrective action, are discussed.

INTRODUCTION

Fatigue failures of centrifugal compressor impellers have occurred all over the world in various services operated by different companies and users, and in machinery built by several manufacturers. Union Carbide's experience with impeller failures is not limited to any one manufacturer or to one type of impeller. We have had failures of riveted, cast, one piece machined, and welded impellers where the obvious causes were one or more of the following: erosion, corrosion, liquid ingestion, overstress, rubbing, manufacturing and design errors. However, our experience, during the past fifteen years, with fatigue failures includes failure of (1) open impeller discs and blades and (2) closed wheel discs, covers, and blades, which have been caused by resonance or responses excited by several different sources.

This experience has led us to a better understanding and more accurate definition of the problems and has also made us more sympathetic to the problems and constraints faced by the manufacturer that prohibit or limit the fabrication of "resonant-free" impellers. On the other hand, it has also led to the strengthening of our Machinery Quality Assurance Documentation program requirements by including detailed certification and inspection requirements for impellers that will help us reduce or minimize the possibility of future failures.

Efforts to improve economy of operation due to intense competition and now the energy shortage have and are continuing to cause producers such as Union Carbide to install

increasingly larger process compressors. These machines must operate at high load factors and have high on-stream times.

Equipment manufacturers have and are extending the range of their machinery models due to fierce pricing competition and increasing manufacturing costs. As a result, little or no margin exists for process upsets or malfunction of components. The monetary losses resulting from unscheduled outages of the large process compressors require that every reasonable precaution be taken to insure the equipment reliability.

Case histories of seven failures involving high capacity, large diameter (36 inch to 45 inch) welded impellers in the 12,000 to 30,000 horsepower range are presented in chronological order.

The expansion of our Machinery Quality Assurance Documentation Program to enhance or improve our equipment reliability by including more detailed impeller inspection and selective vibration shaker testing is also presented. Furthermore, the immediate benefits we have obtained from the program are discussed.

THE PROBLEM OF IMPELLER RESONANCE IS FIRST SUSPECTED

A gas turbine driven process compressor was plagued by continuing excessive casing vibration, high sound level problems and unexplained or vaguely explained impeller cracking beginning immediately after being put in service at one of Union Carbide's chemical process plants in the mid-1960's. The tip speed range of the welded first stage open impeller was from 1251 to 1288 ft/sec. Failures included weld cracking where the vanes joined the disc, cracking of the disc adjacent to the welds both in and out of the heat affected zone, and impact damage due to diffuser vane breakage and ingestion into the impeller. A new open impeller with heavier or thicker vanes was fabricated, but the problems continued.

Many balancing experts from within Union Carbide, from the manufacturer's plant, and from firms supplying vibration and balancing specialists attempted to reduce the levels of vibration and noise. The relief was minimal. The manufacturer and UCC obtained the services of two outside consultants to assist in diagnosing the problems. Some of the conclusions suggested that some form of foundation or system resonance might be a contributor, but none of the conclusions really incriminated impeller resonance as the most probable cause of the problem.

After a year and a half another impeller failure occurred in this compressor which brought the problem into proper view. When the machine was disassembled, extensive damage to the stationary and rotating parts was discovered. The impeller was severely damaged. Nine pieces of metal were missing from the periphery of the impeller disc and several blades were bent or ripped by impact. The impeller damage can be seen in Figures 1 and 2. This impeller failure (the last) with this machine produced the incriminating evidence that pointed directly to impeller resonance and triggered further shaker testing of the wheel.

There was no disagreement between our metallurgist and the manufacturer's metallurgist as to whether fatigue had contributed to the failure; both agreed that it had.

The results of the metallurgists' studies indicated that failure started by fatigue in several locations on the impeller and progressed through a large number of cycles at *low overstress* until radial cracks had formed.

Shaker or vibration testing of the failed impeller was performed by the manufacturer to establish frequency responses. The nodal patterns were developed by spraying or sprinkling

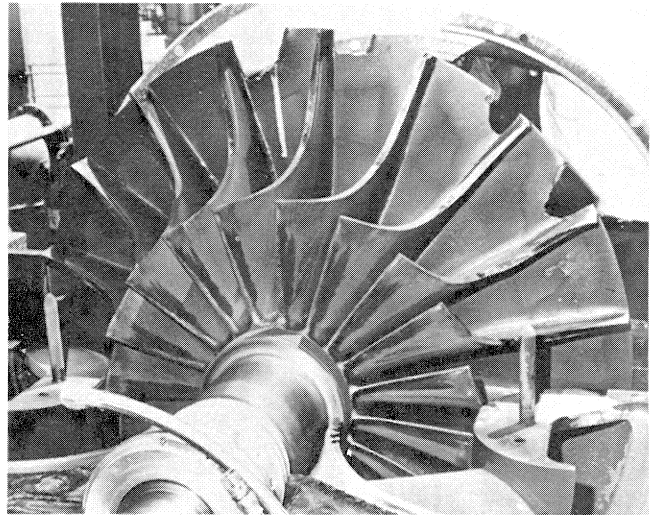


Figure 1. Open Welded First Stage Impeller after Failure as Viewed from Inlet Side.

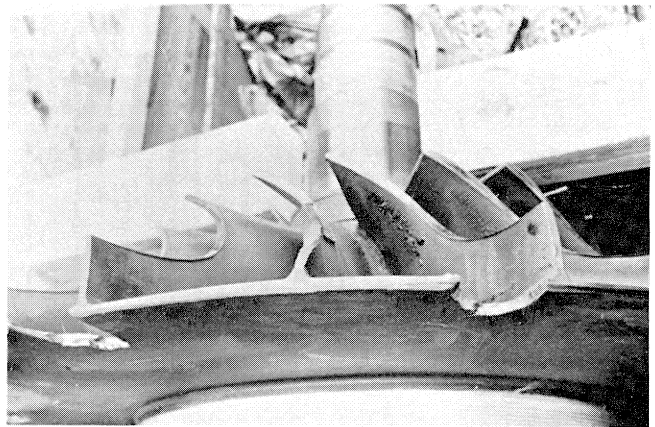


Figure 2. Open Welded Impeller after Removal from Machine Showing Two of the Nine Fracture Surfaces, Impact and Rubbing Damage.

sand or powder on the disc. During the tests, it was observed that the mode shapes were affected by the cracked vanes. However, one mode that matched the frequency of diffuser vane \times RPM had nodes that followed closely the failure pattern described by the metallurgists' study which had indicated the cracking originated in the disc one inch from the tip of the vanes. The conclusions of the test were that there were resonances or responses in the operating range. Diffuser vane \times RPM frequency range was 2389 to 2460 Hz and the wheel resonance was 2443 Hz. It was difficult to accept that if the resonant condition had existed from the time this impeller was installed then why had it not failed earlier instead of after nearly 50 billion cycles. It is noteworthy to mention that the manufacturer had earlier vibrated a similar impeller. Also, the frequency of the high sound level previously mentioned at or near the discharge of the machine was at diffuser vane \times RPM. The manufacturer replaced this compressor with a different model, and it has operated relatively trouble-free for the past 12 years.

SECOND RASH OF FAILURES OCCUR

Late in 1971, prior to initial operation of two parallel motor driven process compressors, we were informed by the

manufacturer that they had identified a second flexural blade critical or resonance between 1380 and 1490 Hz on the open first stage impellers in these machines. Impeller tip speed is 750 ft/sec. There was no anticipation of any problems since no known source of excitation existed. The analysis of vibration, as measured with the installed noncontacting monitoring system, did not reveal a component at the frequencies in question. However, six weeks after initial operation of the first unit, an increase in horizontal radial vibration at running speed which increased still further the next day was experienced. The unit was shut down and the compressor was opened. Two pieces were missing from the cover of the discharge wheel. It can be seen from Figure 3 that the regions where the pieces are missing somewhat resemble the open wheel failure discussed earlier. The rotor with the failed wheel was returned to the manufacturer for impeller replacement and further repairs.

It was immediately suspected that resonance was a contributing factor and requested that this impeller be shaker tested to develop nodal patterns and determine the response frequencies. Additionally, a recount was made of all the speed increaser gear teeth, guide vanes, diffuser vanes, impeller blades and any other interruption that could help create any excitation.

Manufacturing techniques and tolerances were reviewed. Detailed measurements of all impellers that were accessible on the removed and/or spare rotors were made which included the following:

1. Variation in disc and cover thickness between each blade.
2. Radius of the machined junction of each blade to the disc or cover if blades were milled from disc or cover and welded to the other (two-piece wheel).
3. Radius of the fillet where disc or cover is welded to blade (two- and three-piece wheel).
4. Hardness profile of disc, cover and blades.
5. Thickness variation due to balance grinding of disc or cover.
6. Overall finish of machined surfaces.
7. Line-up of gas passages in impeller with diffuser (axial relative position).
8. Line-up or run-out of centerline of gas passages with centerline of impeller bore.

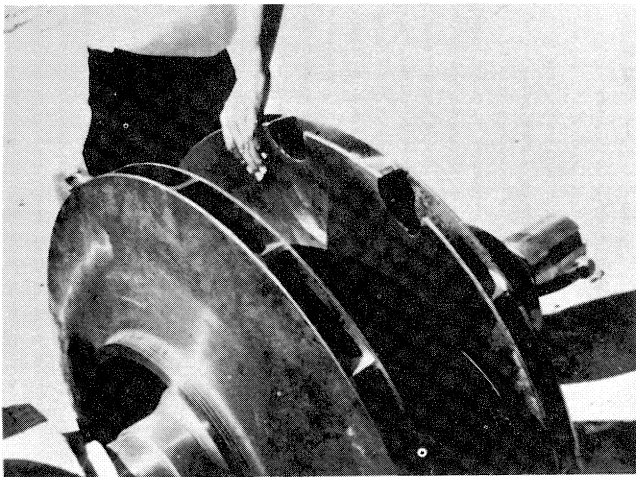


Figure 3. Closed Discharge Stage Impeller after "Half-Moon" Pieces at Two Adjacent Blades had Separated from the Cover.

Field shaker tests of the spare and another assembled rotor were conducted by us at the plant site. The spare rotor which had a discharge impeller with a thicker disc and cover was installed and the unit restarted.

The results of the field and shop shaker testing and measurements indicated that the failed impeller cover was resonant at or near the blade pass frequency of 980 Hz. The main source of excitation was the stationary discharge tongue or "cut water" that each blade of this discharge stage wheel passed. The mode shape was a scalloped pattern between each blade and followed very closely the crack path and can be seen in Figure 4. It was also interesting to note that material hardness had a significant and more than expected effect on the response of the wheel to a given excitation. The frequency of the resonance was not materially changed by variation in hardness, but the change in response was dramatic. The wheels, heat treated to a higher hardness, exhibited much higher amplitudes and extended decay times. Also, the variations in disc and cover thickness appeared to enhance the response of the impellers to a much broader range of excitation frequencies.

The radii of the blades to disc and cover or absence of radii also acted as stress risers to help initiate cracking. The manufacturer modified his drawings and tooling to eliminate the problems.

A new impeller was manufactured with special attention to quality control, i.e., very generous machined radii at blade to cover, large radii at fillet welds where blades join disc, uniform thickness of disc and cover between blades, attention to heat treating and maintaining hardness at the low end of the tolerance range. This and dynamic balancing of each individual impeller were followed by their inspectors, as well as our own resident inspector. The attention to quality control plus increasing the disc and cover thickness shifted the 13 scallop mode (once per blade) frequency from 975-980 Hz on the failed discharge wheels to 1210 Hz on the replacement impellers.

PREDICTABILITY AND THE UNEXPECTED

Union Carbide stated that it was expected the other machine would fail in a given length of time after its start-up. The manufacturer's engineers stated, however, that they thought the problem was due to the lack of a generous machined radius at the blade-to-cover intersection and another failure should not be anticipated since they assumed the other machine's last stage impeller to be properly manufactured. A different impeller failed in this same compressor only one

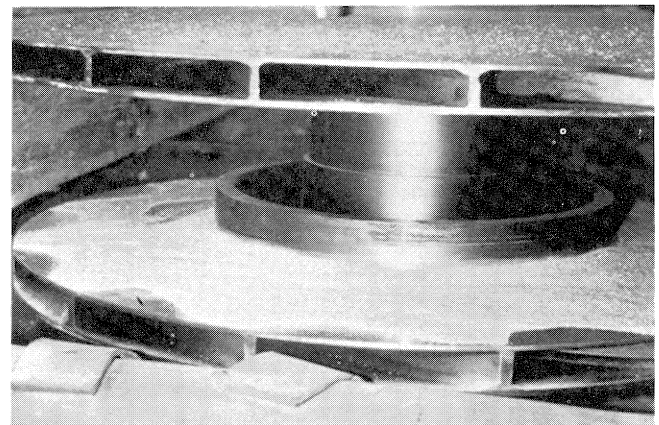


Figure 4. Closed Discharge Stage Impeller with Sand Pattern Formed by 980 Hz Excitation During Field Shaker Test. Note the "Half-Moon" or Scallop Shape.

month and 12 days after the discharge wheel failed. This time we did not get the "minor increase" in vibration; the vibration level instantly increased to 26 mils and the monitoring system tripped the machine.

Expecting to find the same type failure, only more of the same, it was a surprise to remove the cover and find that this time the first stage open impeller had failed. This impeller had blades that were supposed to be resonant at 1380-1490 Hz. One blade had cracked from the disc, apparently in the weld. While not being separated from the disc, the blade had deflected and touched the stationary inlet shroud causing the machine to vibrate violently and further destroy all the labyrinths and seals as it coasted down through its critical frequency after being tripped by the protection system.

The original rotor was already at the manufacturer's plant for the discharge impeller replacement, and now the spare rotor (common to both machines) had been wrecked as well. After the rotor was removed from the machine, a segment containing the cracked blade was sawed out of the wheel and hand carried to the manufacturer's plant to expedite the examination by their metallurgist and our metallurgist for cause of failure while the rotor was en route.

We received the original rotor with its new discharge impeller installed just five days later. The manufacturer again responded by applying the maximum effort in manufacturing and engineering support.

Conclusions based on the metallurgical examination indicated that the crack apparently started in the leading face of the blade, about 1/8 inch above the weld to the hub and 5/8 inch from the inlet edge. It then progressed into the weld and the wheel disc. No visual abnormalities were disclosed except a small unrelated gas bubble. The fracture indicated *low stress-high cycle fatigue* of several million cycles. There was complete agreement between our metallurgist and the manufacturer; nothing was wrong metallurgically.

The closed discharge wheel in this machine failed after 890 hours of operation. The open inlet wheel failed after approximately 950 hours. The closed wheel failure was caused by a cover resonance corresponding with the blade passing frequency, aggravated by the thinner than optimum cover thickness and the smaller than optimum radii at the blade-to-cover intersection.

The cause of the open wheel failure was unknown. It was thought that it must have resulted from mechanical deficiencies aggravated by hydraulic influences which were either internal or external to the machine. The investigation plan called for wheel vibration tests, cumulative damage analyses, and the influences of manufacturing tolerances and material physical properties on wheel mechanical performance.

Since the facilities for shaker testing, especially the mapping of nodal patterns on complex surfaces such as vanes was limited and the manufacturer produced documentation of past good experience, further shaker testing was not thought to be justified. Moreover, the manufacturer presented shaker data on a similar impeller and no resonance near a source of known excitation seemed to exist.

The conclusion was that it was a fatigue failure, but not due to an obvious excitation, especially in light of four billion cycles (blades x RPM x operating hours).

PREDICTIONS COME TRUE

Earlier predictions were that the discharge impeller in the other compressor string would fail after a given number of hours of operation. It happened two days or about 48 hours after it was expected. The horizontal radial vibration level in-

creased and the machine was immediately shut down to remove its rotor. When the cover was removed from the machine, one piece was missing from the cover of the discharge impeller, and except that only one piece was missing, instead of two, it was exactly the same as the other machine's discharge wheel failure. However, the machined radii at the blade-to-cover intersections were slightly larger than the other unit, but the failure still occurred.

The manufacturer was already fabricating replacement discharge impeller number three for the spare rotor. These two rotors were quickly refurbished with new discharge impellers and one also received a replacement open first stage impeller. The open impeller design and fabrication methods remained unchanged.

ANOTHER UNEXPLAINED FAILURE

Nearly 3½ years after the three failures of the above mentioned constant speed machines, another open impeller failed in the same machine that had experienced the open wheel failure previously. This failure was more severe and the damage was further aggravated by two attempts to restart the machine to gather additional vibration data which caused deeper ingestion of the fragments.

It can be seen from Figures 5 and 6 that all of the blades left the disc. One blade marked #1 failed in fatigue. All the rest were torn off by impact overloading and associated rupture. The metallurgists' conclusion indicated that the crack of Blade #1 formed and propagated at the nose or leading face near the weld to the hub. Progression was due to *low overstress-high cycle fatigue* for about four inches and then separated from the disc and impacted on and separated the other blades from the disc.

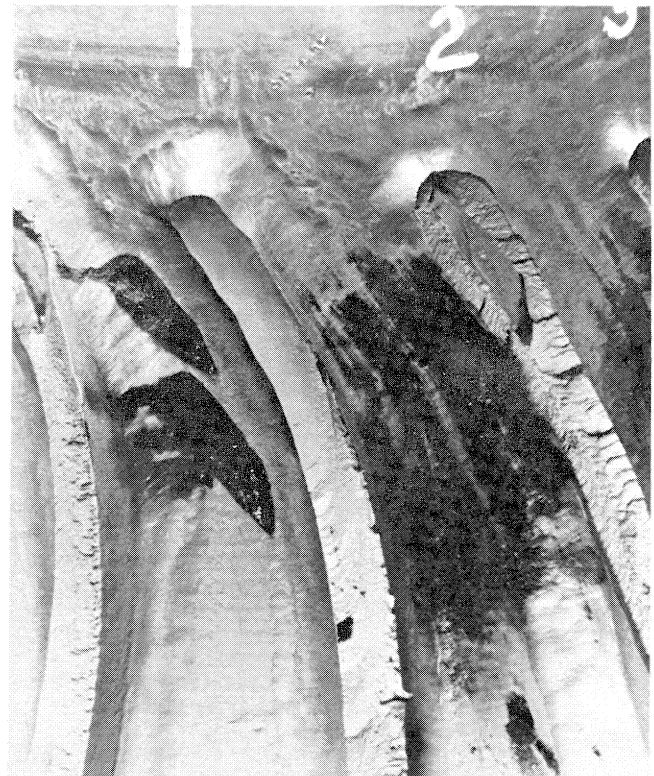


Figure 5. Open Impeller (Second Open Type Fatigue Failure in the Same Compressor) Showing Fracture Surface at #1 Blade Location.



Figure 6. Open Impeller Showing Fracture Surfaces where Other Blades Separated from the Disc.

Once again, our metallurgist and the manufacturer's metallurgist were in complete agreement. Some type of flow phenomena was still suspected as the cause for excitation, especially near surge. We were not running this machine at 100% capacity; however, we were operating well into what was considered a stable region (14% above predicted surge). The manufacturer's new operating recommendations placed severe horsepower penalties on us since we now were required to operate at an additional 10% margin above where surge had been determined.

A tentative test program to try to determine the source of the excitation was developed and preparations were made to implement the program. The cumulative damage analysis, started earlier, had never been completed and now additional manpower became available to finish the study. It was thought that this would provide a means to identify the excitation or forces leading to the wheel failure. Other studies relating to the effects of manufacturing procedures, tolerances, geometry, etc., on wheel mechanical security were discussed, but firm conclusions were not reached.

FIRST STAGE CLOSED WHEEL FAILURE

Before the field test program recommended for the open wheel machines could be implemented fully, a turbine driven process compressor in another plant tripped on high vibration in November 1975. This machine had been operating in parallel with another identical unit for over seven years without problems.

The three-piece type, first stage impeller had lost part of one vane, as can be seen in Figure 7. The fragment was ingested deeper in the machine causing more extensive damage. The blades had a leading or inlet lip at the eye and were stitch welded to the disc and the cover. This impeller operated at a tip speed range of 945 to 990 ft/sec.

After so long a time, resonance would not be suspected; however, this was confirmed as another fatigue failure. Nevertheless, the following was also investigated:

1. Liquid slugging.
2. Vapor entrainment.
3. Operating in or near surge.
4. Foreign object damage.
5. Impeller rubbing.
6. Instability.
7. Impeller overstressed.
8. Resonance.

Resonance and overstress were the most probable causes for failure.

Discussions with the Operating and Maintenance engineers and specialists at the plant location revealed that some of them had occasionally heard the compressor make an intermittent buzzing or vibration sound during the two years preceding failure and that it could be changed or eliminated with minor alterations of speed and/or load. This statement applied to both compressor trains.

The manufacturer conducted a finite element analysis. As a result of the finite element analysis, the manufacturer increased the length of the stitch welds near the inlet eye where the blades join the disc and the cover from 1½" to 4½". Although many suggestions were analyzed, this was the only modification that was made to the new impeller. The old impeller was forwarded to the Acoustics and Dynamic Measurements Laboratory in Charleston, West Virginia, for shaker testing.

The shaker tests indicated that the natural frequencies of the unfailed blades with 1½ inch welds fall in the range of 700-905 Hz and that by extending the weld to 4½ inches, as



Figure 7. Closed First Stage Impeller Blade Failure Near the Leading Edge after Seven Years of Operation can be Seen at 10 O'Clock. Impact Damage is also Apparent on the Leading Edge of the Other Blades.

done on the new impeller, this range shifted to 928-1017 Hz. All the modes had nodal points near the welds, but the 845 Hz mode was strongest and was shifted to 928 Hz by the welding, as can be seen in Figures 8 and 9.

The inlet guide vane passing frequency range was 800 to 841 Hz and was assumed to be the source of the excitation since additional field tests indicated that the intermittent buzzing was near 840 Hz and could be brought on by adjusting load more than speed. The inlet vanes, however, were far upstream, and it was not expected that trailing wakes would excite the impeller blades — but they did.

PLANNED ROTOR REPLACEMENT PREVENTS WRECK

It was suspected that the other turbine driven unit might be due for failure and an orderly shutdown was planned to exchange its rotor as soon as one that had a new wheel with the extended welds was received. When removed early in 1976, the first stage impeller of the original rotor was found to be cracked exactly as in the first unit except pieces had not left the wheel.

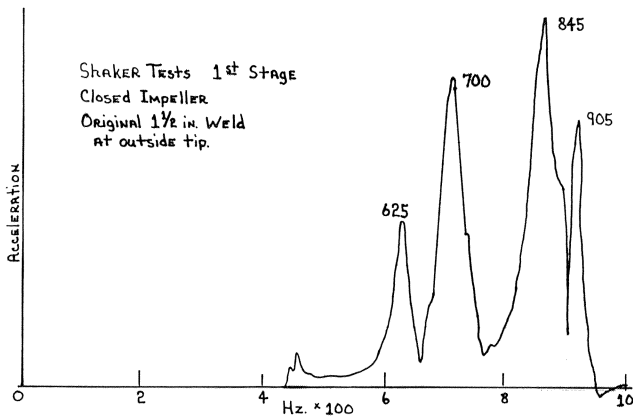


Figure 8. Plot of Acceleration vs Frequency of Closed Impeller Blades Indicating Response Range and 845 Hz Peak. Welds at Disc and Cover and 1½ Inches Long as Originally Manufactured.

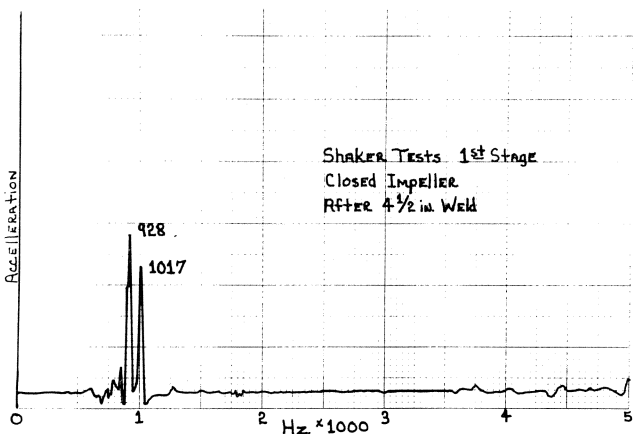


Figure 9. Plot of Acceleration vs Frequency of Closed Impeller Blades Indicating Change in Response and Shift of 845 Hz Peak to 928 Hz after Increasing Welds at Disc and Cover to 4½ Inches.

FIELD TESTING PROGRAM IS IMPLEMENTED

Armed with the most recent success of establishing that blade vibration could be brought on by changes in gas flow in very narrow bands led to the next series of field tests. Additional impeller and blade shaker tests and compressor testing that nearly duplicated a wheel by wheel field performance test were planned for the machine that had failed two open first stage impellers. The casing had been drilled for a wheel by wheel test.

Previous testing had been performed but was inconclusive in regard to resolving the question of first stage impeller blade failure. More comprehensive field tests were conducted in a final effort to resolve the cause of the open wheel failures. The compressor performance was varied over a wide range in order to identify excitable natural frequencies of the impeller blades, which might lead to an explanation of the failure, and to determine the lower operating range of the compressor without surge.

The compressor was instrumented to measure and record case and shaft vibrations and high frequency pressure pulsation in the diffusers, especially the first section, as the flow was varied through the first section. The flow was varied from the maximum obtainable down to where first section incipient surge was detected by case vibratory acceleration. At this point, a component of the compressor case acceleration at 1450 Hz became obvious from the on-line continuous frequency analysis.

Others have successfully experimented with case accelerations and frequency measurements to detect incipient surge, and they report that at near surge conditions the flow incidence across the impeller blade can excite the blade natural frequency. At an average or high flow, the 1450 Hz acceleration was barely noticeable. At a low flow-high pressure ratio condition the 1450 Hz acceleration amplitude increased considerably.

Subsequent shaker testing of the spare compressor rotor first stage impeller revealed that a strong natural flexural mode of the impeller blades fell in the 1460 to 1500 Hz band.

These tests seem to be conclusive in pinpointing that at a low flow-high pressure ratio condition, the open wheel blades are excited at the 1450 Hz mode, and eventually a blade will fail under *low overstress-high cycle* fatigue. The tests were inconclusive, however, in determining if a backward traveling wave existed, that is, that the first wheel was excited by a backward traveling wave from the second wheel at its blade pass frequency.

The field shaker testing definitely established the blade resonance. The set-up showing some of the instrumentation, the impeller and blades is shown in Figures 10, 11 and 12. The results of the compressor testing using high frequency pressure transducers and accelerometers indicated that the machine will develop a vibration at the measured resonant frequency of the first stage blades. The first section (first two impellers) seems to be approaching incipient surge even though the overall machine is operating well to the right of the predicted surge line. Figures 13 and 14 indicate the changes in the vibration peaks at the blade resonant frequency. Operating conditions have since been modified to avoid the condition and to allow the first section to operate at higher capacity.

IMPELLER TESTING FOR NEW PROJECTS

The earlier impeller failures prompted us to strengthen our established Quality Assurance Documentation Program by

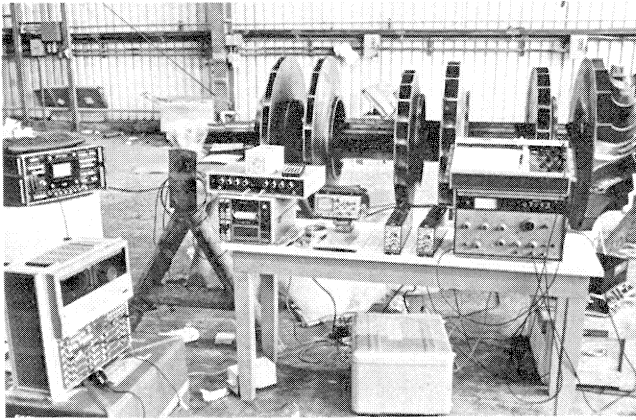


Figure 10. Rotor in Horizontal Position for Excitation Testing of Open Impeller Blades.

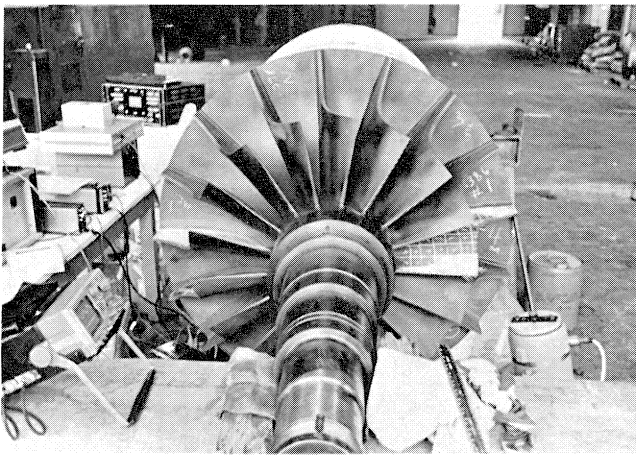


Figure 11. Inlet End View of Rotor Showing Shaker and "Stinger" on Right Side with Blade Laid Out in a Grid Pattern on Convex Side.

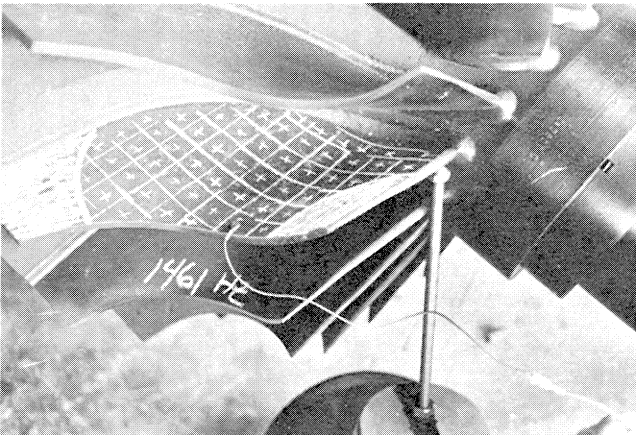


Figure 12. Blade of Open Impeller with Grid Pattern on Concave Side and Chalked Mode Shape after Survey. Shaker and "Stinger" Moved from Position Shown in Figure 11.

including a more detailed testing and inspection program for the impellers in any centrifugal machine for new projects.

Included in our purchase specification are our Quality Assurance Documentation requirements for the machines with additional impeller requirements.

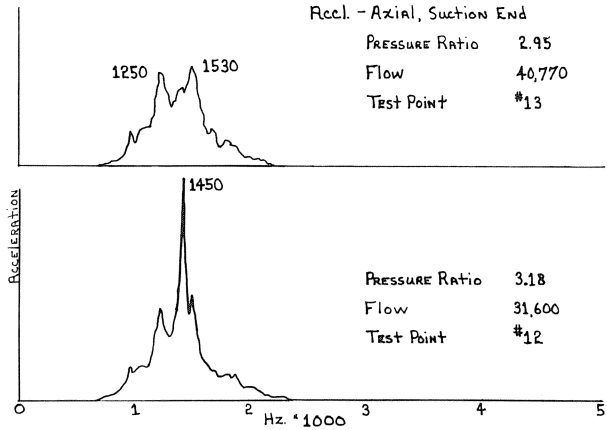


Figure 13. Compressor Case Axial Acceleration vs Frequency Plots of Two Test Runs Indicating an Increase of the 1450 Hz Component with a Flow Reduction of 22.5%.

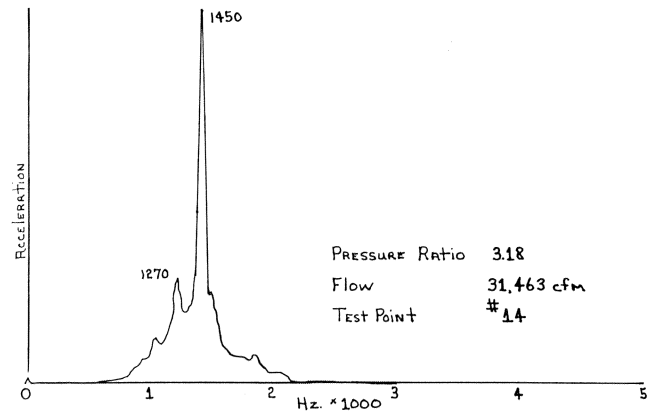


Figure 14. Compressor Case Axial Acceleration vs Frequency Plot at Further Reduction in Flow to 22.8%. Note the Dramatic Response of the 1450 Hz Component to Such a Small Change in Flow.

These additional requirements are:

1. The manufacture analogous user's list of all impellers proposed for the job. RPM, blade-vane count, pressure, diameter, etc.
2. Numbers of blades in each impeller for our proposed compressors.
3. Numbers of guide vanes upstream of each impeller for our proposed compressors.
4. Numbers of diffuser vanes downstream of each impeller even if around the U-bend of the diaphragm wall for our proposed compressors.
5. Definition of experience and available past shaker test data to permit us to decide on whether or not to require additional shaker or resonance tests which include impact (soft hammer) excitation and electromagnetic excitation.
6. Manufacturing tolerances that will be maintained and an opportunity to review impeller manufacturing drawings at the manufacturer's plant.
7. Assurance that documentation and records of impeller quality requirements will be maintained and forwarded

to us and our inspector or our designated agent. This includes:

- 7.1 Certification of material physical and chemical properties.
- 7.2 Heat treat certification of rough stock and after fabrication.
- 7.3 Disc thickness between blades at periphery and at mid-point between hub and outside diameter.
- 7.4 Cover thickness between blades at periphery and at mid-point between eye and outside diameter.
- 7.5 Blade thickness for each blade measured one inch from the tip and from the eye.
- 7.6 Radius of juncture of blades to the disc and the cover.
- 7.7 Wheel-to-shaft shrink or interference fit.
- 7.8 Run-out of disc, cover and gas passage centerline with respect to hub bore.
- 7.9 Provisions so that balance grinding will not alter the impeller's characteristics.
- 7.10 Dynamic balance for each impeller.
- 7.11 Overspeed for each impeller.
- 7.12 Nondestructive testing results for each (witnessed).

The above program has been carried out on over 70 impellers and has resulted in the replacement of impellers by two manufacturers before shipment to correct manufacturing defects.

Shaker tests were performed on 13 of the 70 impellers, and two additional impellers were modified to change a suspicious response and tested again to confirm that the modifications were effective.

SHAKER AND IMPACT TESTS

Tests of individual impellers, usually at the manufacturers' plants have been accomplished by recording data from two sets of excitations before the impellers are assembled on the rotor.

A list of frequencies for *detailed search* is developed by a prior review to identify possible excitations including but not limited to: all shaft speeds in the compressor train (motor, gear, turbine, compressor, etc.) impeller blade pass, inlet guide vane pass, and diffuser vane pass. All of the tests are usually witnessed and in some cases have been performed at the manufacturers' plants by Union Carbide personnel using their own equipment or the vendor's equipment.

The instrumentation required for testing is:

1. Shaker table with power amplifier.
2. Multi-channel oscilloscope.
3. Multi-channel recording oscillograph (for damping determination).
4. Electronic filter.
5. Accelerometers and/or velocity pickups with long cables.
6. Charge amplifiers.
7. X-Y plotter.
8. Camera and film.

9. Beat frequency oscillator.
10. Multi-channel tape recorder.
11. Microphone.

NOTE: A two channel "Transfer Function Analyser" with phase capability can be used to replace most of the recording and readout equipment listed above.

Impact (soft hammer) tests are the first sets of excitations of the disc, cover and blades and are as follows:

1. Impeller (disc or cover side up as determined before test) is suspended from overhead by a cable or rod connected to a flanged plug or mandrel slip fitted into the hub bore.
2. Microphone is hand held near impeller.
3. Impeller is struck with a soft (plastic, rawhide, lead, etc.) hammer at disc, cover and blades.
4. Output of microphone is recorded on tape for playback through a wave analyser. This is repeated several times.
5. Oscilloscope photographs and X-Y plots of frequency vs. amplitude are made for each excitation.

NOTE: The above has also been performed using accelerometers in addition to or instead of the microphones. Contacting the impeller with the accelerometer must be done while exercising care so as not to damp the vibrations.

Electromagnetic excitation follows a review of the impact excitation test results. Hopefully, predominant frequencies disclosed by impact excitation will be different than those scheduled for detailed search. Nevertheless, they will be added to the detailed search list for development of nodal patterns. The additional steps necessary to determine the mode shapes and responses are:

1. Impeller is suspended the same as for the impact tests.
2. Shaker table is connected to the disc or cover (whichever is down) between the blades by a "stinger" rod to transmit the excitation.
3. Pick-up (velocity type or accelerometer) is mounted vertically at the edge of disc or cover (whichever is up) 180° from excitation. This is repeated at 90° from excitation.
4. Output of pickup is converted to amplitude.
5. Frequency sweep normally is from 0 to 5000 Hz. Graph of frequency vs. amplitude is made.
6. Sand patterns developed at peak amplitudes are photographed.
7. Impellers with surface irregularities or curvature which prohibit sand patterns are laid out in a checked grid. A dual trace oscilloscope is then used to relate the phase of the exciter to the phase of the impeller in each grid section. In phase is marked (+) and out of phase is marked (-). After the complete grid network has been surveyed, approximate nodal lines can be drawn and the mode shape determined.

Impellers mounted as part of a rotor assembly are excitation tested using the same procedures for unmounted impellers except for the support.

Suspend the rotor vertically from hoist or crane with disc

or cover side up as determined before test. Leave the rotor hanging from hoist to facilitate the shaker height adjustment.

NOTE: Have cribbing on hand for supporting the rotor as required between testing periods. Also prepare to test the rotor in the horizontal position, supported at the bearing journals if necessary.

CONCLUSION

We, as a user, have experienced many impeller fatigue failures in addition to the *low overstress-high cycle* failures presented here and have attempted to pass along the highlights of some of our experiences with welded impellers of large diameter operating in high horsepower compressors to other users.

It is also of benefit to note that, regardless of how detailed the study and design audit of a new compressor train and its components, it is possible to be surprised by failure after several years of reliable operation. Factors that can cause impellers to respond and fatigue-to-failure long after start-up and beyond the warranty period are:

1. Balance grinding on impeller covers and disc surfaces creating thick-thin sections between blades.
2. Balance grinding at the disc and cover edges (non-uniform scalloping) at the outside diameter between blades. (Not only does this alter the frequencies, it also disrupts the gas flow passage.)
3. Erosion and corrosion of the disc, cover or blades due to liquid and solids carry over materially altering the original response frequencies.
4. Installing smaller outside diameter or larger outside diameter impellers in a proven machine to change capacity for a new operating condition without reviewing past history and any available shaker data.
5. Changing speed range.
6. Changes in properties of the gas.

7. Entraining liquid in the gas stream.

8. Installing a replacement rotor with impellers that are supposed to be exactly like those on the rotor removed but are not since they may have been manufactured several years later to revised drawings and by a different group of craftsmen employing revised techniques.

9. Installing a replacement rotor or diaphragms with different axial spacing, thereby disrupting the gas flow path.

Although we have covered welded impellers only, it is not the intent to infer that riveted impellers are better. Overall, the welded impeller seems to provide greater security than the riveted. We also wish to stress that welded impellers are better regarding stress distribution and dynamic loading; however, they do appear to have a weakness due to less damping and energy absorption as compared to the riveted types.

It is impossible to design every wheel to be resonance or response free in all conceivable excitation ranges. Manufacturers do not design to avoid impeller resonance, as they consider this to be (1) impossible over a range of wheel diameters, contours, speeds, etc., (2) uneconomical, and (3) unnecessary. On the other hand, they acknowledge that some resonant modes can and should be avoided on constant speed machines.

Union Carbide feels that the cumulative effect of all potentially detrimental factors must be minimized, whether they be flaws, stress concentrations, high amplitude resonances, machining errors or other factors. We, therefore, encourage manufacturers via our Quality Assurance Documentation program to design to avoid resonance and to require documentation of such to minimize the chance for random failure. Moreover, when that random failure does eventually occur, complete open communication between the user, the manufacturer, and any consultant is necessary to properly define the problem. When the problem is properly defined, the assignment of responsibility for failure will go hand-in-hand with that accurate definition.

