

# TURBINE ROTORS WHIRL AFTER DYNAMICALLY STABLE DESIGNS ARE UPRATED

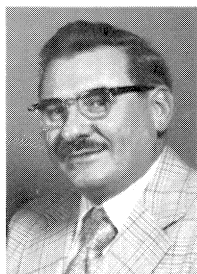
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## ABSTRACT

The problem of steam turbine rotor whirling following uprating of a stable design and the measures taken to define the problem and implement a solution are described.

Nearly identical units installed at several of Union Carbide's process plants exhibited different symptoms of minor instability after rerating. The newest unit, rerated before installation, began to exhibit signs of not-so-minor instability following a very stable and successful startup. The deceptive symptoms, temporary corrective action, final diagnosis of the problem, including field testing, are discussed.

## INTRODUCTION

Severe rotor vibration at subsynchronous speeds began to plague the nearly identical steam turbine drivers of process compressors installed at different chemical process plants following several months of satisfactory operation (Figure 1). Internal rubs, oil whirl, distortion of the casing due to excessive pipe strain and hysteresis whirl were each considered as a possible contributor or cause of the vibration. Bearing design and aerodynamic excitations were also considered and investigated before a solution to the problem was found.

A well-proven design was field modified after a few years of satisfactory performance to meet the new requirements of the expanded process unit. At the same time, a new process plant of the expanded size was being designed and built. Before delivery to the site, its steam turbine was modified and satisfactorily tested at the manufacturer's plant. Following this test, another turbine of the uprated design was built and tested

to replace the compressor drive in another unit that had been operating for several years.

The actual modifications to uprate the turbines from 7000 hp to 9000 hp consisted of the following:

1. Destaged from 5 to 4 to allow a greater steam flow.
2. Installed 2 highlift cams and 2 governor valves with larger port area to increase steam flow to the lower quadrants of the nozzle ring.
3. Manufactured new rotor with same shaft diameter, length, and bearing span but with 4 wheels (shrink fit) instead of 5.
4. Replaced 3 diaphragms in casing grooves. Number 4 diaphragm position was left blank.

All units were manufactured to accept either the 7000 hp or 9000 hp internals, and the 7000 hp, 5-stage parts were retained as spare parts.

## FIRST INDICATIONS OF TROUBLE

The first indication of trouble with any of the machines was noted on the machine that had been field modified when minor nervousness or vibrations interpreted as oil whip were detected. However, one of the conditions that was noted during the time that oil whip was being diagnosed was that, instead of heating the oil to eliminate the cause of vibration, cooling of the oil by increasing the flow of cooling water, and thereby increasing the film thickness, damped or reduced the vibration to an acceptable level. This particular phenomenon was not recognized as significant at that time.

The second unit to exhibit peculiarities and high vibration levels was the machine that had been modified before being tested at the manufacturer's plant. After approximately one year of operation, all of which had been satisfactory, the bearings were inspected during a unit turnaround. During the bearing inspection, everything was found to be normal, clearances were within the recommended values, and no indication of trouble was suspected or detected. After a successful solo and overspeed run, during the attempt to start up the unit after the turnaround, the machine began to vibrate severely as the speed was increased to near the design value. It was impossible to get the unit on the line and, as the turbine was accelerated, the radial vibration levels, as measured with noncontacting probes, would suddenly increase to the full range of the meters, tripping the machine off the line. It was suspected at that time that something during the bearing inspection might have been overlooked and that the possibility of an oil labyrinth or seal rub existed. A spectrum analyzer was connected to the vibration probes before attempting to restart the machine. The frequency of vibration was determined to be above the critical speed that had been measured at the factory during the test. We were concerned over this difference; however, the frequency did not change as the speed of the machine increased or decreased. With the frequency information, it was assumed

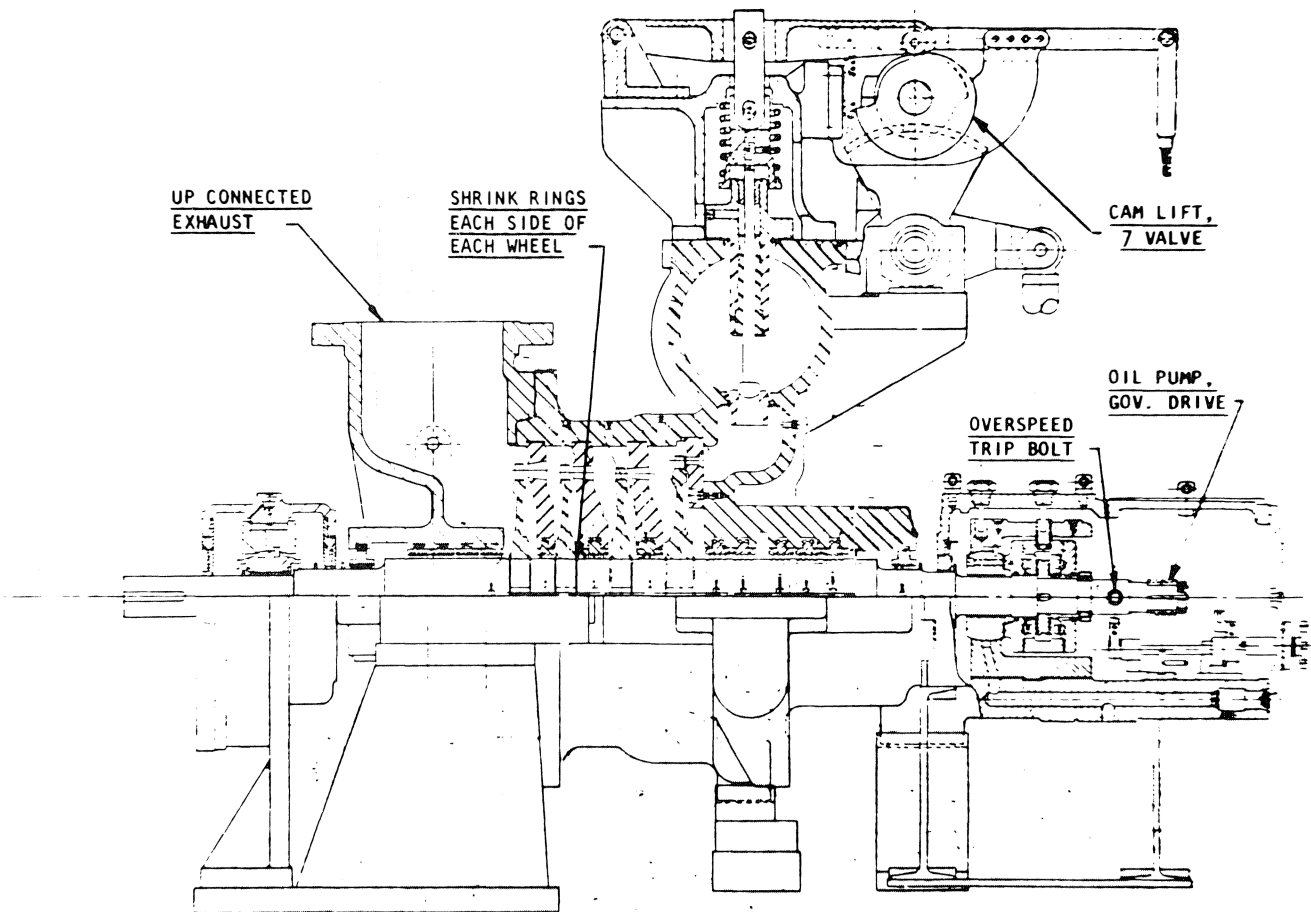


Figure 1. Cross-Section of Four-Stage Multi-Valve Steam Turbine with Built-Up Rotor.

that a bearing oil labyrinth rub was causing the problem and tended to make the shaft vibrate at its first natural critical frequency, the same way that the chatter or stick-slip vibration caused by drawing the bow across a violin string will cause it to vibrate at its fundamental frequency. The bearing covers were removed, and it was evident that the coupling end bearing oil labyrinth had indeed been in contact with the shaft. This was corrected by increasing the clearance and scraping the bronze labyrinths to knife edges, after which the bearing cover was reinstalled; and the machine was started up without incident.

About a month later, during a unit process upset, the machine was tripped off. When the upset was corrected, attempts to start the turbine and compressor resulted in high vibration again measured at the assumed first natural frequency of the turbine shaft. Since there was no reason to believe that we had an oil labyrinth rub and the machine had been operating satisfactorily, we searched for other causes for the vibration.

The start-up sequence was next investigated. The normal method for starting the machine was to roll the cam shaft for the governor throttle valves to wide open and then slowly open the trip throttle valve and bring the machine up to minimum governor speed, after which the throttle valves would partially close, the trip throttle valve would then be fully opened, and the speed control transferred to the governor. Starting the machine in this manner resulted in nearly 360° nozzle admission of steam. This is normal and considered to be a proper

method. It was thought that we might possibly be getting some sort of an aerodynamic phenomenon by starting up in this fashion. The next attempt to start was to close the governor valves down manually to the point where the first valve was just ready to crack, then open the trip throttle valve wide open so that we were starting the machine with only a small portion of the nozzle ring admitting steam. This attempt was successful and the machine was put back on the line.

### THE PROBLEM PROGRESSES

Every time thereafter that this machine was shut down, it became more and more difficult to get it restarted without problems of vibration. This continued until the next annual turnaround of the unit when the bearings were again examined, and the upper half of the casing was removed permitting a thorough examination of the internals. Just before the turnaround, we had tried to put the machine in service and the same high vibration occurred forcing us to schedule the opening of the turbine.

We found that one of the eight shrunk-on lock rings, located upstream and downstream of every wheel on the shaft to prevent the wheels from moving axially, had fractured, passed on through the downstream stages of the machine, and nicked the buckets. It was suspected that perhaps friction between the wheels and the lock rings was the cause of the problem. All lock rings were then cut from the shaft, as the wheels all appeared

to be tight on the shaft with no evidence of axial movement. Also suspected was casing distortion due to pipe strain, because during the turnaround there very definitely was evidence of seal rubs between each stage. Whether or not this was the cause or the effect we did not know, but the symptoms were beginning to point to a self-excited or hysteresis whirl problem. The rotor, diaphragm, and casing alignment was improved and the interstage steam seal clearances were increased.

It was also suspected that the steam was entering the machine through the nozzle ring unevenly because of the differences in valve port sizes and that a larger amount of steam could enter through one lower nozzle quadrant causing the turbine rotor to actually lift in the bearings (which were already very lightly loaded) and tend to initiate oil whip. In order to counteract this, new bearings were designed. They were shorter to reduce the area and of the four-lobe type to increase the stability and to prevent the onset of oil whip.

Pipe strain plus the casing distortion was also investigated, and differences in the piping arrangement at each of the locations were investigated. Again, following this turnaround, we had difficulty bringing the machine on the line despite a near perfect solo and overspeed run.

Massive amounts of data including but not limited to vibration, bearing and oil temperature, alignment (internal and external), casing thermal movement, compressor load, speed and vibration, and steam conditions were recorded during the next four months. Only four major sources of excitation that could contribute to the problem were recognized. They were as follows:

1. Excitation caused by the oil pump - speed governor drive shaft and worm gear.
2. Hysteresis whirl possibly caused by load coupling friction or the shrunk-on rotor parts.
3. Internal diaphragm seal rubs caused by casing distortion.
4. Aerodynamic cross-coupling or flow induced whirl due to rotor to stator eccentricity.

#### SIMILARITIES TO A WELL DOCUMENTED PHENOMENON

In the 1920's, Dr. B. L. Newkirk [1] and A. L. Kimball [2] concluded that internal friction or hysteresis of the metal and/or shrunk-on discs could contribute to rotor whirling at a rate equal to or near the first critical speed. The facts concluded from the early testing and listed by E. J. Gunter [3] are as follows:

1. The onset speed of whirling or whirl amplitude was unaffected by refinement in rotor balance.
2. A well balanced rotor sometimes required an external disturbance to initiate whirl.
3. Whirling always occurred while rotating above the first critical speed.
4. The whirl threshold speed could vary widely between machines of similar construction.
5. The precession or whirl speed was constant regardless of unit rotational speed.
6. Whirling was encountered only with built-up rotors.
7. Increasing the foundation flexibility would increase the whirl threshold speed.

8. Introducing damping into the foundation would increase the whirl threshold speed.
9. Increasing the axial thrust bearing load would increase the whirl threshold speed.
10. Distortion or misalignment of the bearing housing would increase stability.

Aerodynamic cross-coupling or variations in the clearance between the rotor and stator either at the periphery of the wheel or in the labyrinth steam seals can act as a source of excitation and set a sensitive rotor to whirling at its first critical frequency as described by J. S. Alford [4]. However, he stated that the frequency of whirl can be from 35% to 50% below rotational speed and is more prominent during high pressure and power levels.

#### AMBITIOUS EFFORTS PRODUCE MINIMAL GAINS

The next move a few months later was to again remove the upper half of the turbine and change out the worn gear type coupling. Since it was not a spacer type, the hubs could not be removed without removing either the compressor or the turbine shaft. The coupling was changed from a 25° pressure angle to a 40° pressure angle since this was the spare replacement that had been supplied by the coupling manufacturer. We now feel that this was a mistake and that the higher friction forces in the 40° pressure angle coupling acted to increase the possibility of hysteresis or friction excited whirl.

We also noticed with axial position monitors another characteristic as the machine was being brought up to speed that, generally at the instant the direction of thrust of the compressor rotor would shift from thrust toward the turbine to thrust toward the compressor, the machine would begin to vibrate uncontrollably. Since both shafts moved nearly as a unit, we thought we had a problem with the coupling, that it was almost locking up in the axial direction. We elected to reduce the thrust bearing clearances for two reasons: one, to prevent as nearly as possible the movement during this thrust shift; and two, to make a damper of the turbine bearing by reducing the thrust clearances to the point that the thrust bearing, while not being tight enough to burn up, might act as a damper to help reduce the levels of vibration in the event that the rotor was excited. This also proved to be of no value.

Our own analysis using computer programs indicated that the response to unbalance for this particular unit was not unsatisfactory and that the rotor was basically stable; however, we obviously knew from experience that the unit was not stable. The manufacturer's response had been good, and the manufacturer's field support at all locations had been outstanding. The manufacturer's analysis indicated that it was not a critical speed problem, and they also assumed that the bearings were satisfactory, but the bearing area was decreased to increase the load since the bearings were lightly loaded. This was in addition to the four-lobe type tried earlier.

The nozzle clearance between the rotor and the nozzle ring was decreased, then increased to change the effect of downstream wakes from the trailing edges of the nozzle ring.

The coupling alignment had been reviewed time after time. The turbine and compressor manufacturers came to the same conclusions that we did — there was nothing wrong with coupling alignment.

During this investigative period, we also clamped intentional unbalance on the coupling and on the steam end of the turbine shaft at the overspeed bolt to attempt to counteract the

onset of hysteresis whirl, thinking that perhaps the unit would have less of a chance to whirl at its critical if it were unbalanced and forced to vibrate at synchronous speed. This did have a stabilizing tendency, and we were able to run for a slightly longer period.

All the time that we were working with this one extremely difficult machine, the other units were tending to show further signs of nervousness. More and more often, the "oil whip" problem would appear at one of the other locations. Consultants were called in for a bearing stability review. Nothing could be found that would indicate any evidence of instability with the bearings.

Following a unit trip, it was again impossible to start the machine. Although we thought we had tried everything, we had not attempted to counteract casing distortion or actually distort the casing to make the machine run. Since we had tried most everything else, we removed the insulation from the bottom of the turbine so that the lower half would run cooler, tending to make the machine bow up in the middle and possibly counteract the weight of the exhaust line if indeed the weight of the exhaust line was acting on the turbine. We also put a screw jack under the center of the turbine. With a force on the jack, an attempt to restart the machine was made and low and behold it was a successful start! The unit ran beautifully for three months with the jack under the uninsulated belly of the turbine.

Three months following the placement of the jack under the center of the machine, we were again in trouble. At that time massive pipe supports were installed to ensure that the turbine had zero pipe load in all directions while cold and hot. It ran for three days, started to vibrate and tripped off. The turbine was opened, the 7000-hp, 5-wheel rotor and diaphragms were reinstalled. A stable but underpowered machine was the result. The 4-wheel, 9000-hp rotor was stripped to try and determine the cause of instability.

A task force was gathered consisting of outside consultants and UCC consultants from the Chemicals and Plastics and Linde Divisions. We had meetings onsite, meetings at our Technical Center in Charleston, West Virginia, and also meetings with the manufacturer at his plant. Further unbalance response calculations again indicated that the rotor was not sensitive to unbalance, in spite of the basically unstable operation.

While the investigation to identify the problems with this machine and the nervousness of the other turbine was in progress, one of the other process units was shut down by a process upset. The process was tripped but the machine was not. The turbine speed was reduced until it approached its critical frequency. By the time it was tripped by hand, it was a basket case. The machine was opened, the rotor shaft was destroyed, but the wheels were reusable. The labyrinths had cut deeply into the shaft all on one side, and it was clearly evident that the unit had been rotating at its first resonant frequency. It was necessary to rebuild this rotor by making a new shaft for it in order to get the unit running again since the 7000-hp spare rotor had been reinstalled in the problem unit, and the rotor removed from it was thought to be a "freak." At the time that this shaft was made, a very thorough investigation of the shrink fits of the wheels to the shaft was performed, the shrink fits were increased from 0.006 in./0.008 in. to 0.008 in./0.010 in. After reassembly, the rotor was reinstalled in the machine and started up successfully but not without difficulty. It did tend to exhibit the rotor whirling type of activity at the first critical frequency of the rotor. The stripped-down "freak," 9000-hp rotor was also rebuilt with a new shaft and heavy shrink fits.

## MEETINGS TO DETERMINE THE CAUSE

The meetings with the consultants went in several directions. The manufacturer was working on the problem from one direction and we on another. Two of the consultants were working with transient analysis programs to determine if we had an unstable system, and we were working to gather additional information or data that might be helpful from the units in operation in the field.

All of the technical papers that were available to us on hysteresis whirl, self-excited whirl, friction-excited whirl, damped critical programs, etc., were collected. Certainly, Newkirk's and Kimball's work of 1924 indicated that we probably had a self-excited whirl problem; however, the onset of whirl would occur at any time with absolutely no way of predicting when whirling would occur. One thing was very clear. Once a machine starts to whirl hysteretically, the onset of whirl becomes easier and easier to initiate. The point at which a machine would suddenly start to whirl after running smoothly for over a year could not be predicted, and anytime thereafter it is more apt to whirl until it finally is sensitized to the point where some disturbance almost as small as a pencil rolling off a table top can initiate the whirling phenomenon.

## SOLUTION TO THE PROBLEM

Critical speed calculations indicated that the root of the problem was that the 4-wheel rotor mode shape had the nodal points almost exactly on the bearing, and that until this critical mode shape was altered to get the nodal points away from the bearings the rotor would continue to be unstable (Figures 2 and 3).

The possible solutions were:

1. Increase the diameter of the rotor shaft enough to achieve the maximum critical frequency increase and machine the diaphragms for larger steam seals.
2. Change from a built-up rotor to a solid-forged rotor as hysteresis whirl has not been observed with solid rotors.
3. Increase the diameter of the rotor shaft to the limit permitted without machining the bore of the dia-

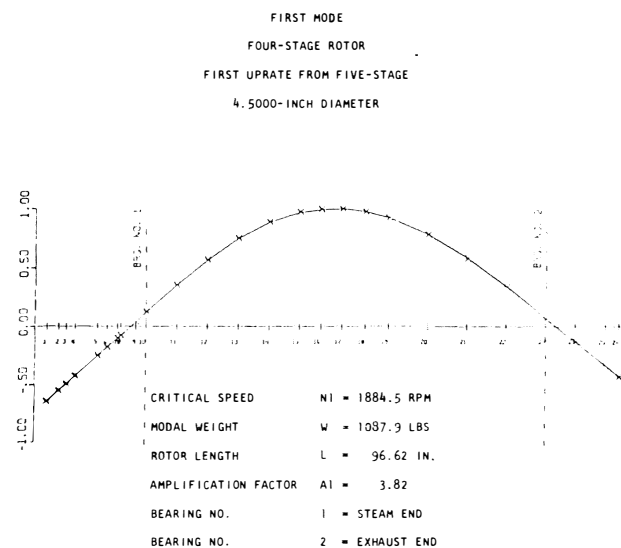


Figure 2. Mode Shape of Original Four-Stage Rotor First Undamped Critical Frequency.

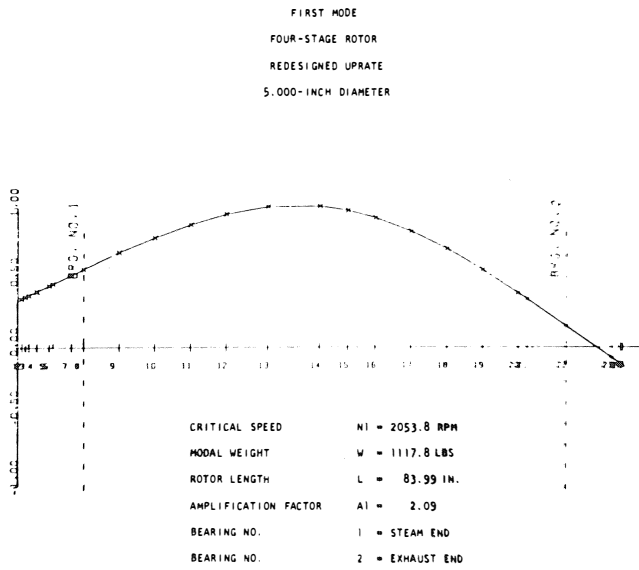


Figure 3. Mode Shape of Revised Four-Stage Rotor First Undamped Critical Frequency.

phragms to measurably increase the critical frequency and install larger ID steam seals with the original seal OD.

For all three cases, reduce the amount of the overhung weight by shortening the coupling end and shorten the governor end of the rotor by removing the overspeed trip bolt and oil pump drive, and modify the trip system to a total electronic overspeed and governing system as shown in Figures 4, 5, and 6.

Our selection of the Case No. 3 rotor diameter allowed us to use the diaphragms without machining and to install different seals that had a larger ID but with the same size OD as before. The seal material was also changed to Ni-Resist which was more forgiving and would have less chance to set off vibrations in the event of a touch or rub.

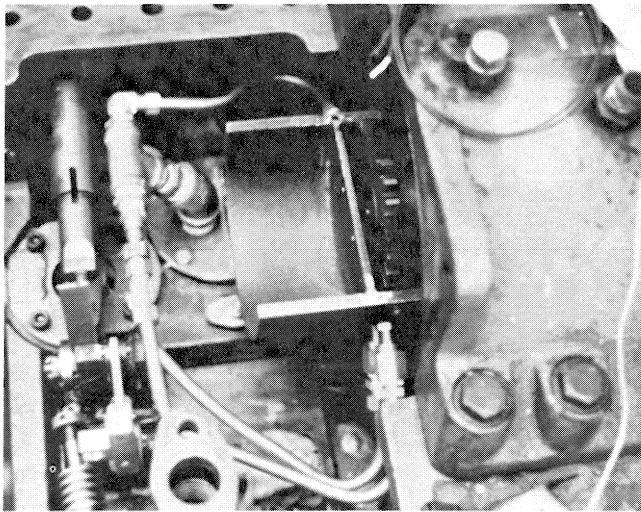


Figure 4. View of Steam End Bearing Housing Showing Shortened Rotor, Probe Mounting Bracket and Toothed Speed Wheel.

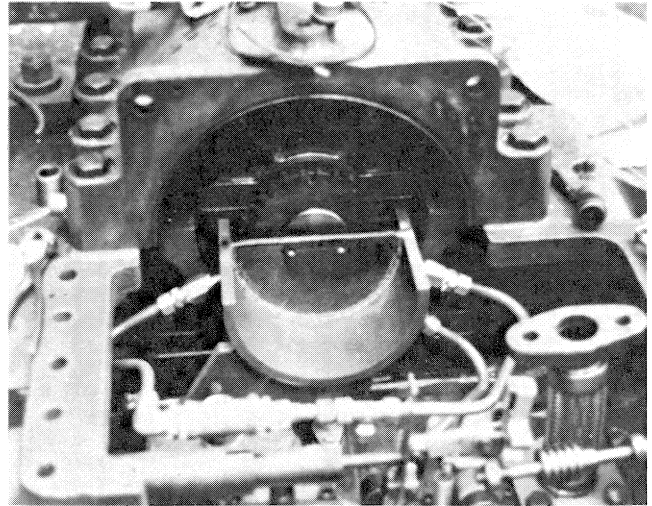


Figure 5. Probe Mounting Bracket with Magnetic Speed Pickup Probes Installed.

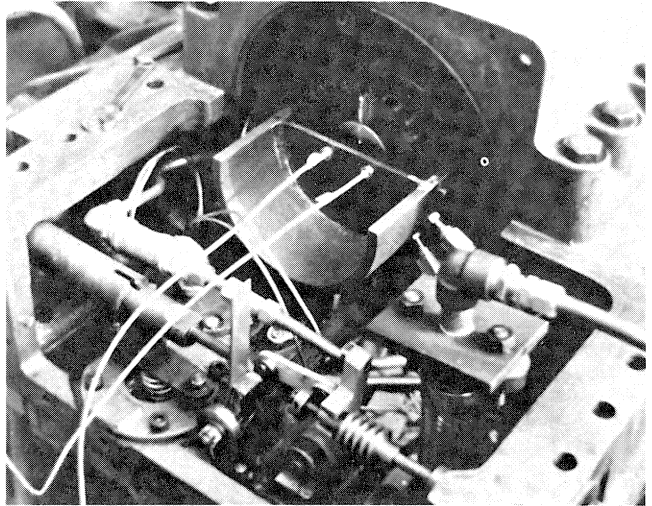


Figure 6. Probe Mounting Bracket with Axial Position Proximity Probes Installed.

At the time the proposal was made to utilize Case 3 which also included removal of several inches from both ends of the shaft, we investigated the use of and selected a diaphragm-type coupling with a center dropout spool in order to remove or replace the hubs without removing the rotor from the compressor or from the turbine. A diaphragm coupling with a backup gear unit or tooth coupling mounted concentrically with the diaphragms was designed and manufactured. The backup drive is sometimes called "come-home drive" in the marine industry (Figure 7). However, the turbine manufacturer had to alter the bearing housings in order to allow the hub of the coupling to extend back into the bearing housing with an oversized oil labyrinth sealing the bearing housing on the hub. This meant that the shaft keys and keyways had to be sealed with elastomeric compound to prevent oil from the bearing housing from being expelled through the coupling.

The mode shapes of the different designs were reviewed before actual modifications began. Although the largest diame-

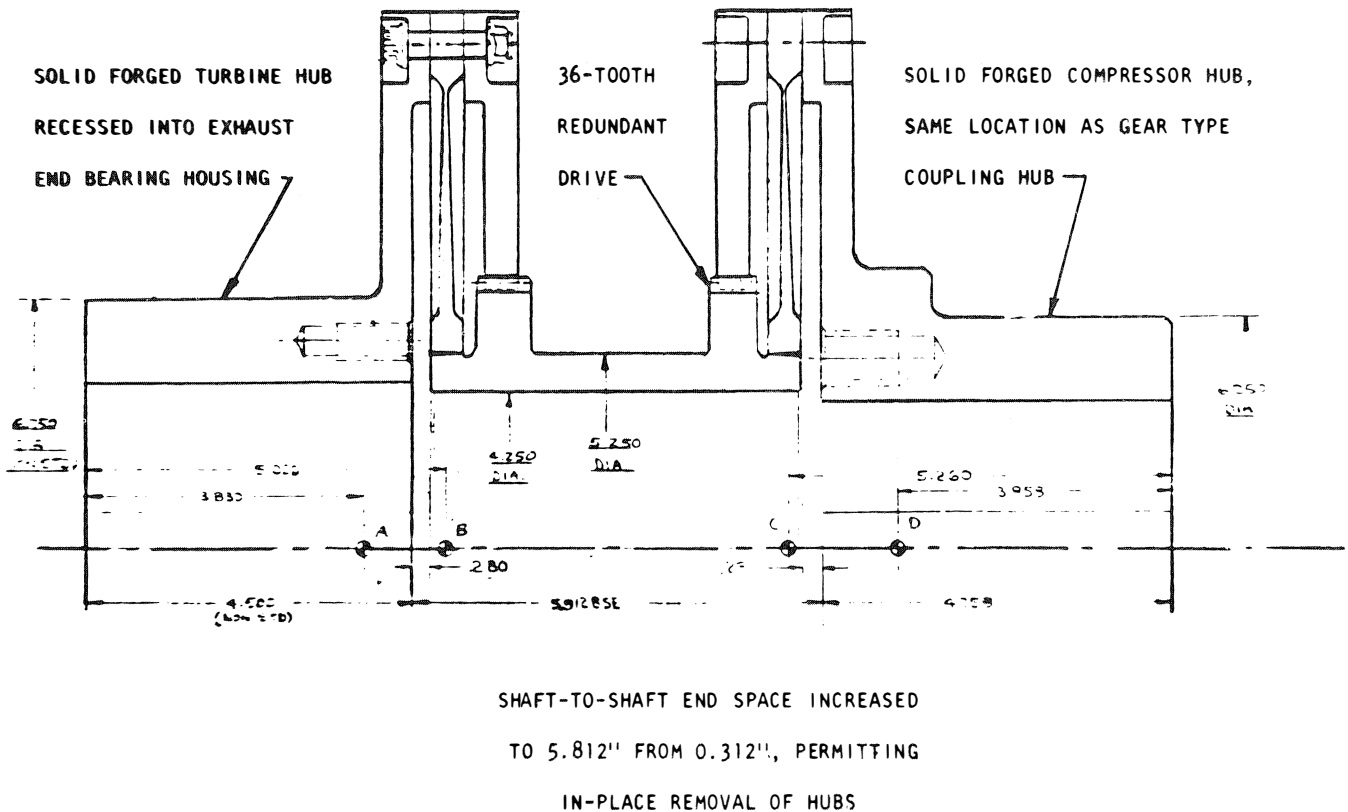


Figure 7. Contoured Diaphragm Special Flexible Coupling with Redundant Drive.

ter rotor was most stable, the pros and cons of going to the solid forged versus large rotor and remachining all the diaphragms were weighed against how far we would get toward a stable machine with the mid-diameter fix. The conclusions were that we would be 97% of the way to having a perfectly stable machine versus 98-99% with the solid forged rotor or the maximum diameter built-up rotor with rebored diaphragms.

Damper bearings were also investigated as a possible solution to the problem as well as tilting pad or pivoted shoe bearings. The investigation indicated that, with this particular machine, damper bearings would have no effect on correcting the problem since the nodal points were nearly at the centers of the bearings. Tilting shoe bearings were selected.

Discussions of others during the Fourth Turbomachinery Symposium in 1975 informed us that one of the compressors in the North Sea had a very similar problem. K. J. Smith [5] gave a presentation at the 1975 Turbomachinery Symposium concerning another unit that nearly paralleled our turbine experiences. The problems with their compressor were nearly the same as our turbine problem and the corrective methods similar. The compressor was corrected by enlarging the diameter of the shaft and moving the bearings closer together. It was impossible for us to move the bearings closer together, so we made the shaft larger and reduced the overhang weight by shortening the rotor at both ends.

The first modified rotor was installed in the original machine that was field modified with the diaphragm coupling and started up. Vibration data were recorded on tape during the startup. The unit appeared to be extremely stable but, with the very low amount of residual unbalance in the rotor, we could see a certain amount of what we termed "inch worm" activity on the spectrum analyzer at the first critical speed of

the machine (Figures 8 and 9). This activity is present on all units; however, we do not see it as a vibration problem but as activity only, especially when at design speed the total amount of rotor vibration peak-to-peak double-amplitude is 3/10 of a mil. We continued with the modification of rotors to the larger shaft diameter, replacing the coupling with the diaphragm type, shortening the shaft, and installing the electronic overspeed trip system.

At the present time, we have modified, large-diameter, shortened rotors installed in most of the units. One machine remains to be modified to date. All modified units have had the shaft-driven main oil pump drive removed and discarded. A replacement main oil pump was installed on the lube oil console and the overspeed trip bolt removed; and, at the position of the overspeed trip bolt, a toothed wheel for the speed pick-up probes installed. The toothed wheels have 30 teeth with a large tooth space to give a good spike for speed indication at high speed. This toothed wheel is also used for the speed governor as well as for the overspeed trip. There are four magnetic pickup probes looking at the toothed wheel plus two axial position eddy current proximity probes mounted on the same bracket that look axially at the tooth wheel but not at the teeth. One looks at the smooth portion of the tooth disc; the other probe looks at a different diameter circle, and has a keyphasor hole drilled in it. In this manner we can use the axial position probes to look at axial position with dual voting for alarm and shut-down while one can be used to measure axial vibration, and the other used as a keyphasor in conjunction with the probes that are monitoring the turbine and compressor shaft radial bearings.

We have also installed five solenoids in the hydraulic dump line from the trip throttle valve arranged in such a way

ROTOR VIBRATION  
5.000-INCH DIAMETER SHAFT

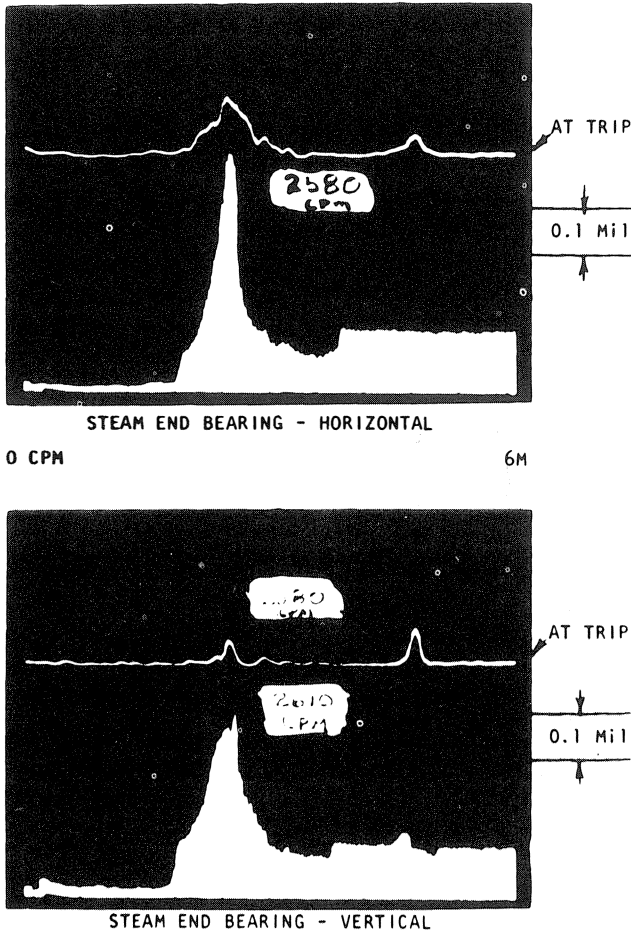


Figure 8. Photos of Real Time Analyzer Scope Face (Peak Hold) During Solo-Overspeed Check.

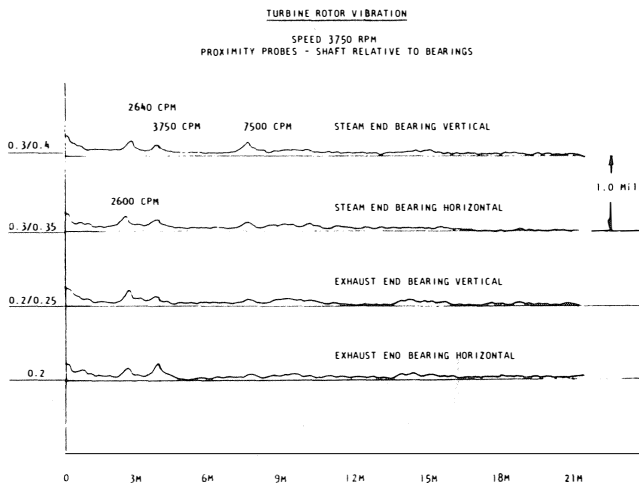


Figure 9. Plots of Frequency VS. Amplitude of Turbine Rotor Vibration, Coupled to Compressor Operating at Reduced Speed.

that there are two in parallel and two in series and one cross-linked so that it is absolutely impossible for a single solenoid failure either opening or closing or coil burn-out to ever cause the machine to trip off the line unless the trip is actuated by a trip signal.

One problem still persists that has nothing to do with the instability or vibration of the machines and that is, after the machines are running satisfactorily for some period of time, the steam conditions are such that salting of the bushing on the trip throttle valve can actually prevent the trip throttle valve from closing in the event that a trip is called for. A water washing mechanism has been installed to wash this bushing. This is not the same as water washing the turbine. This is only to wash the salts built up from around the bushing of the trip throttle valve.

The binding or the sticking of the throttle valve stem can also be relieved by requesting that the operators at least once a week start to close, via the handwheel, the trip throttle valve until the machine speed just starts to fall off, and at that point reopen the valve fully. This would break loose any solids or sediment around the stem and allow it to trip in the event that a trip is called for.

HOW TO RECOGNIZE THE PROBLEM

One of the first things to recognize in a whirling problem of this kind is that the machine will tend to vibrate at a fixed frequency as you vary the running frequency of the machine. In other words, the frequency of vibration remains constant regardless of the speed. The best way to diagnose a problem of this type is to use the field analysis tools that are now available such as spectrum analyzers and noncontacting vibration probes.

Whirling can be a function of friction of shrunk-on sleeves, shrink fits under the wheels, and friction forces in the coupling teeth that increase as the load transmitted across the coupling increases to the point that whirling is initiated. Also as the load is increased, the coupling may tend to stiffen laterally and can change the critical frequency of the coupled system.

It has also been stated that a sensitivite rotor can develop a memory; and once it starts to whirl, it remembers, after which it will get more erratic, become less predictable, more sensitive, and will whirl more often. This is exactly the condition that we observed over the three-year problem period with these machines.

Let's assume a hypothetical emergency. When a rotor that has exhibited the tendency to whirl is removed from the machine, there is no possibility of getting an immediate replacement, and the unit must remain out of service for some number of months. Being a built-up rotor, it is possible to quickly unstack and to increase the interference fits of the wheels to shaft by metallizing or plating either the ID of the wheels or the OD of the shaft and restack it even though there is absolutely no evidence of any fretting or movement of the wheels on the shaft. This increase in shrink may alter the rotor just enough to enable it to run for another year or until a replacement redesigned rotor can be manufactured.

Design analysis tools that are available are computer programs and design audits. Use these tools during design stages because they do not require the opening of the machine for the modifications. They are much faster and more economical. Listen to the consultants who have developed the programs. They learned on the problems of others that may have been of a similar nature to yours, and they have used them to develop their programs. Also, work with the manufacturer and the consultants as a team and not as a committee.

There are also damped lateral critical speed computer programs and response to unbalance programs used to determine the mode shape of the rotor and its stability. These should be used during the design audit to identify possible problems. When you determine the mode shape and the nodal points of the proposed design, you will be a step closer to having a stable rotor. If the mode shape indicates the nodal points are away from the bearings, you probably have a stable design. If the design audit indicates a problem, it is much more economical to correct it then than to wait until after installation when the machine trips on high vibration and tries to tear loose from the foundation. The trial and error method, shutting down a unit, etc., is by far the most costly program for problem solving.

Damped lateral critical speed programs are also available that include the cross-coupling necessary to destabilize the rotor. Cross-couplings may be termed as aerodynamic cross-coupling or mechanical cross-coupling. What is meant by aerodynamic cross-coupling is that variations in the small radial clearances between the rotor and stator as it rotates or spins in the cylinder act with the steam or fluid on the rotor as a source of excitation. It is stated that variations of .006 to .015 in. will tend to excite a sensitive rotor. This has been called aerodynamic cross-coupling and can initiate whirling.

#### CONCLUSION

It is possible for turbomachinery to appear to be basically stable for some period of time and suddenly start to exhibit characteristics of instability. The symptoms can be misleading without accurate frequency versus amplitude plots that are properly interpreted. Similar vibrations can be initiated by different causes, and similar machines may exhibit different characteristics.

Long bearing spans and high speed are major contributors.

The calculated critical frequency will be somewhat less than the actual whirl frequency.

Increasing the shaft diameter, thereby increasing the critical frequency of an unstable rotor bearing system, will probably result in a satisfactory rotor.

Cross-coupling, aerodynamic or mechanical, should be considered as a source of excitation.

Bearing damping will not contribute to suppressing whirl if the nodes are at the bearings.

It is hoped that passing our experiences on to other users will help to provide them with more direct courses of action for optimizing corrective measures.

#### REFERENCES

1. Newkirk, B. L., "Shaft Whirling," General Electric Review, vol. 27, 1924, p. 169.
2. Kimball, A. L., Jr., "Internal Friction Theory of Shaft Whirling," General Electric Review, vol. 27, 1924, p. 244.
3. Gunter, E. J., Jr., "The Influence of Internal Friction on the Stability of High Speed Rotors," Trans. ASME No. 69-VIBR.-2.
4. Alford, J. S., "Protecting Turbomachinery from Self-Excited Rotor Whirl," Trans. ASME Journal of Engineering for Power, Oct. 1965, p. 333.
5. Smith, K. J., "An Operational History of Fractional Frequency Whirl," Proceedings of the 4th Turbomachinery Symposium, Texas A&M University, 1975, p. 115.