ROTATING MACHINERY SERVICES, INC.

VOLUME 5, ISSUE 1

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THE FINISH LINE

PHONE 484-821-0702 FAX 484-821-0710 WWW.ROTATINGMACHINERY.COM

SHOP OPERATIONS UPDATE

By Paul Poley

Our expansion is complete and we are taking full advantage of our 20,000 square feet of shop floor space. We have plenty of overhead crane capacity and versatility with 3 bridges and the ability to handle up to 40,000 pounds. We have plenty of lay down and warehousing space. We recently added a new Schenck 17,500 pound capacity balance machine and are establishing some process specific cells. Our rotor inspection, repair and balance cell is shaping up to be first class. We have areas designated for tear down and assembly, component inspections and light machining.

The RMS Power Solutions shop is committed to being a fully equipped, turbomachinery emergency repair support facility. Throughout 2009 and beyond, we will continue to enhance our capabilities with the addition of equipment and manpower, as required, to meet that goal.

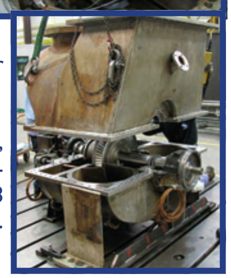
POWER SOLUTIONS

NITRIC ACID EXPANDER RERATE & OVERHAUL

By Sydney Gross

As part of the supply of a nitric acid plant air train, RMS has rerated and overhauled a customer's single stage GE expander from 2600 hp to 4700 hp. The work was accomplished within a 10 day window at a shop local to the Georgia plant.

Starting in the beginning of 2008 with inspections and the customer's dimensional data, RMS was able to redesign the expander which remained in operation until the plant shut down in February of 2009. The redesign included a new flow path, tilt pad bearing retrofit and an electronic overspeed system upgrade. The customer's spare rotor was reworked prior to the shut down with new blades installed and automatic peened by RMS, redesigned thrust collar and a retrofit speed wheel/axial probe target. The analyses that made the redesign possible included aerodynamic, structural, vibration and rotor dynamic lateral critical.



Overhaul and rerate services accomplished during the outage included disassembly, joint milling and line boring, casing hydrotest, bearing housing remachining and reassembly. The expander was installed on a new packaged baseplate with RMS's AXL508 11000 hp axial compressor and a rerated and refurbished Elliott QV7 steam turbine. The train was started in March of 2009 and is part of an acid plant expansion.

RMS BEGINS IMPLEMENTATION OF NEW BUSINESS SYSTEM

By Frank Marrone, Jr.

Over the past two years, RMS has been fortunate to experienced very large growth across our business. This growth has brought about new challenges to the organization needing to maintain an accurate work flow through out all functional areas in the company.

Recognizing this, RMS has spent great deal of time to evaluate and select an ERP business system provider that will help RMS excel into the future with precise order management. This system will touch customers and suppliers providing better communication from the initial inquiry, estimating, planning, order status updates all the way to order shipment and installation. Final implementation is expected to be complete by 3rd quarter of this year. RMS continues to look forward to meet our customer's future demands today.

What's Inside

Up Coming Conferences 2nd Quarter

SynGas 2009, Booth 38,

Tulsa, Ok—April 20-22

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2009 Industrial Maintenance Symposium - GG4

Atlantic City, NJ—May 12 - 13

RULE OF THUMB

By Neal Wikert

Cleaning Components

Pelletized CO_2 cleaning – uses pelletized CO_2 about 1/8" diameter about ¼" long.

Cleaning Disk and Blade attachments: Glass bead clean or Grit blast using 000 aluminum oxide, fresh with no broken or angular grit. Blast w/ great care. Ensure system is clean of coarser grit.

Bently 3300 Proximeter Probe Settings:

Residual Magnetism: 2 gauss max allowable

Gap Voltage: 200 mV / mil (mil = .001 inch)

Recommended Gap: 50 mils

Linear Range: 10 to 90 mils gap

Bearing Alarms – Criteria from a machine with a rated speed of 11,100 rpm

High Vibration – Alarm @ 2.5 mils, trip @ 3.5 mils

Rotor Axial Position – Alarm @ 19 mils, trip @ 20 mils

Journal Bearing RTD's – Alarm @ 225 deg. F., trip @ 250 deg. F.

Thrust Bearing RTD's – Alarm @ 235 deg. F., trip @ 250 deg. F.

Labyrinth Seal May be subjected to a pressure differential of 4-6 times that of a carbon ring seal with no shaft speed limitation. A stepped labyrinth can reduce leakage by up to 30 % over a straight labyrinth.

Lube Oil Systems

Drain Header Slope: ½" per foot minimum 45 deg. maximum

STEAM TURBINE ROTATION

By Sydney Gross

When specifying a turbine or certain turbine components, it is essential to identify the rotation. By this, it is meant the direction that the rotor spins in the casing. There are only two choices but a mistake will most definitely be costly. While this may sound like common-sense, anyone involved in turbomachinery, from the design and manufacture to the end user, can cite examples where a misinterpretation has resulted in an embarrassing discovery. To avoid being the face with egg, one should not only understand the conventional identification but also employ steps to carry through a clear understanding to all parties, from design to maintenance, what is the correct machine rotation and part orientation.

To begin with, there is a convention for identifying steam turbine rotation. The turbine is designated either clockwise (CW) or counter-clockwise (CCW) as viewed in the direction of steam flow. Alternate ways to describe the same orientation are; looking at the inlet or high pressure end or as viewed from the governor end. The CW or CCW is the simple part. It's the orientation of the viewer that usually results in trouble. Often, the orientation is assumed and not specified. But beware, it is not always standard. One should verify the orientation and/or identify it in the description when specifying at every step.

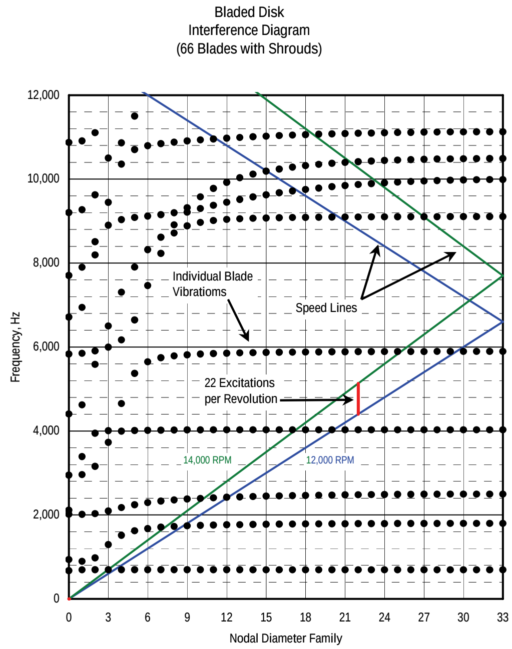
Although the above discussion pertains to the machine as a whole, equal attention must be given to individual components that are rotation sensitive. When designing, specifying or purchasing items such as bearings, always verify the rotation and ensure that it is specified on the drawing and/or documentation. Be sure that you understand the direction of view on parts drawings and models. It may not seem possible but even rotor assemblies, nozzles and diaphragms have been built with the wrong rotation.

You may have been meticulous in observing the rotation during design and manufacture but once the part gets to the field it is possible to install it backwards? It is always good practice to mark the rotation of the turbine on the parts with an arrow. Bearings and rotation sensitive seals should be marked. If possible, the parts should be designed such that they can only be assembled one way. The exterior of the turbine should always be marked to ensure that the rotor is not turned in the wrong direction and that parts are assembled correctly. It is equally important when disassembling equipment to mark the orientation on the components so that they go back together the same way.

Remember, identify the rotation and orientation, verify rotation throughout design and manufacture, mark parts with an arrow and make it so they can only be assembled one way. If you remain vigilant, you may not have any stories to share.

NODAL DIAMETERS and the INTERFERENCE DIAGRAM

By William Sullivan



The interference diagram has become the preferred tool for displaying the interactions of rotor blade natural frequencies with periodic flowpath excitations for bladed disks with shrouds or tie-wires (typical of both compressor impellers and steam turbine stages). On the other hand, it also seems to be one of the least understood tools in the dynamic response arsenal. A typical interference diagram is shown on the right.

In our last newsletter, we discussed aliasing, which is the phenomenon that allows so much information to be displayed on one chart. In the next few newsletters, we will try to explain the diagram itself. That is, what it is and what it can tell us about the risk of exciting rotor blade natural frequencies. A good way to start would be to describe the X-axis (abscissa). To describe the X-axis, however, we first should describe what we mean by nodal diameters and nodal diameter families.

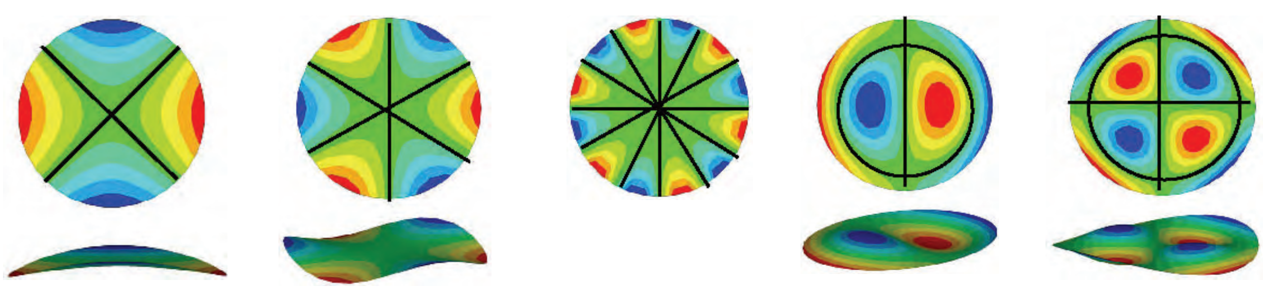
Nodal diameters can best be described by visualizing a flat round plate. If we were to suspend the plate in air, by a string, and tap it sharply with a hammer, we would excite all of modes of vibration. This can easily be simulated on a computer using the modal option in a finite element analysis (FEA) program. In this case, ANSYS was used to generate the following plots.

Plot A show a two-nodal diameter pattern on the circular plate. The upper view of Plot A shows a full view of the plate. In this, and all of the remaining full view plots, the red portions can be thought of as moving out of the paper (toward you) and the blue portions can be thought of as moving into the paper (away from you). Between the peaks, the plate displacement must pass zero. Zero displacement is marked by a line on all of the full view plots. For a continuous flat plate, these nodal lines will be diameters. Since places of zero displacement are called nodes, these diameters are called nodal diameters. Just below the full view plot is a side view plot. In this side view plot, one can see why the two-nodal diameter mode is often called the "potato chip" mode. The side view plots were added to aid in the visualization of the mode shapes. Plot B shows a full view and edge view of a three-nodal diameter mode.

Plot C shows a full view of a six-nodal diameter mode. You should see a pattern emerging at this point. The number of red (or blue) portions, or peaks, around the edge of the plate indicates the number of nodal diameters in the mode shape. Counting the peaks around the edge makes identifying the number of nodal diameters much easier than trying to count the diameters directly, especially with many nodal diameters. In fact with the complex geometry and modes encountered with actual bladed disks, counting diameters often is impossible.

Plot D shows a one-nodal diameter mode with one nodal circle. Note that there are two sets of peaks. One set of peaks is on the edge of the plate while the other set of peaks is inside the plate. As you move across the plate from the far left to the far right of the full view, you will encounter a peak moving out of the paper, then a peak into the paper, then a peak out of the paper and finally a peak moving into the paper. As explained above, between each peak the plate displacement must pass through zero. If we mark the zero displacement with lines, we will see that now, along with nodal diameters, we have nodal circles. Therefore, this pair of plots represents a one-nodal diameter, one-nodal circle mode.

PLOT A **PLOT B** **PLOT C** **PLOT D** **PLOT E**



Plot E shows a two-nodal diameter, one-nodal circle flat plate mode. This is the potato chip mode shown in Plot A with a nodal circle.

Generally, as the number of nodal diameters increases on a flat plate the natural frequency increases as well. Furthermore, the natural frequency of a mode containing a given number of nodal diameters will increase as the number of nodal circles increases. A fully bladed disk with shrouds or tie-wires will have much more complicated mode shapes than a flat circular plate. However, the same basic principles regarding the frequencies and the number of nodal diameters and the complexity of the modes apply.

ROTATING MACHINERY SERVICES, INC.

ENGINEERING EXCELLENCE

PROBLEM: FCC EXPANDER DISK EROSIVE WEAR

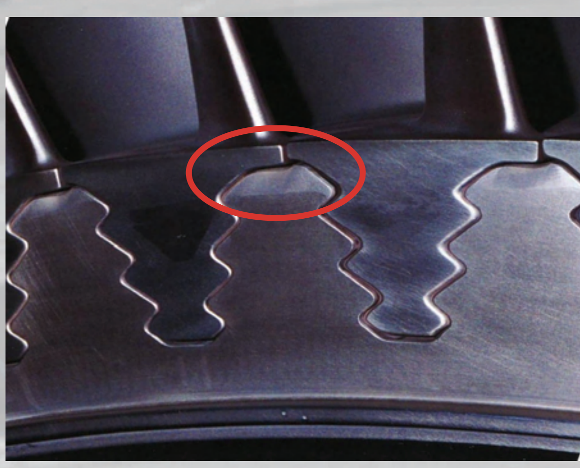
- **Significant erosive wear due to catalyst channeling between rotor blade platforms & in disk cavity region**
- **Increased corrosion can also occur**

Top of Tenon Wear

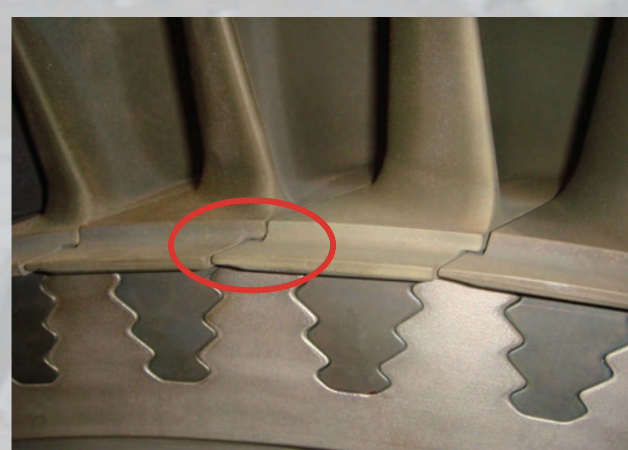
Disk Face Erosion

RMS SOLUTION: UNIQUE LOW EROSION EXPANDER BLADE DESIGN

- **RMS *One of a Kind* overlapping platform design upgrade**
- **CFD aided state of the art aerodynamic design**
- **Blade seal lip reduces amount of flue gas & catalyst entering disk cavity**
- **Platform shields rotor disk from harmful effects of catalyst laden flue gas**
- **Reduces erosive effects of catalyst & flue gas**
- **Low stress design increases structural margins & life of rotor disk & blades**



IR/DR Blade Platform Design



RMS Blade Platform Design

FCC EXPANDER FLOW PATH EROSION

By David Linden

The root cause of expander flow path erosion and deposition is catalyst carry over from the FCC regenerator. FCC catalyst is an alumina–silica based powder that aids in cracking heavy oil molecules into the more commercially desirable lighter end products such as gasoline, heating oil, jet fuel, etc.

FCC regenerators are constructed with cyclone separators to separate and collect as much of the process catalyst as possible and return it to the FCC cycle for reuse. Despite these particle separators being more than 98% efficient, typical catalyst carryover to the downstream power recovery system can be in excess of 400 ppm. For a 90 KBPD FCCU, this can translate into more than 300 lbs/hr of catalyst being lost and potentially carried over to the PR Expander and/or the atmosphere.

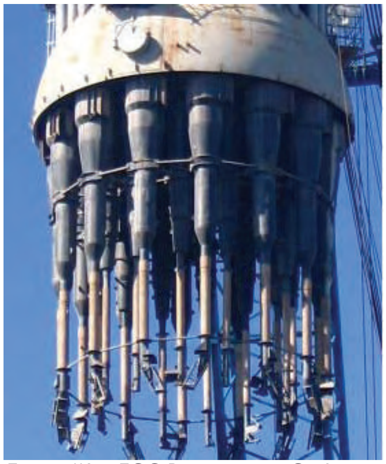
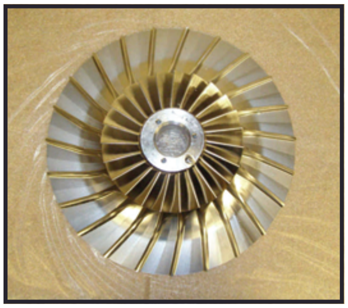


Figure #1 – FCC Regenerator Cyclones

Catalyst particles carried over from the regenerator can range from sub micron to over 80 microns in size, with an average particle size of approximately 40 microns. The alumina-silica (sand) catalyst particles are very erosive when carried over in the process flue gas. During the initial application of power recovery expanders, the flue gas was sent directly to the expander with the full catalyst carryover. The operating experience was not good and expander blades were being worn out in as short as three months of operation. This operating experience highlighted the need for additional particle separation capabilities and hence the third stage separator (TSS) was developed. Our next newsletter will explore the various TSS designs.

CENTRIFUGAL COMPRESSOR RERATE

By Matt Konek



A Midwest manufacturer and distributor of industrial gases is relocating an idle air separation plant and acquired a vintage 1965 surplus 3-stage centrifugal compressor for nitrogen recycle service. However, the original compressor design did not provide the required discharge pressure.

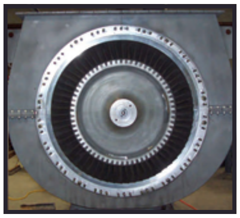
RMS performed a rerate study and determined the compressor was a suitable candidate to meet the uprated conditions. Detailed analytical work included aerodynamic, lateral rotor dynamic, vibration, and shaft stress analyses.

The rerated operating conditions were achieved with minimal modification to the compressor. Flowpath components in the first and second stages were not modified. The uprate in discharge pressure was accomplished by only redesigning the third stage, which included a new impeller, diffuser and a reworked pinion. The surplus machine was completely overhauled by RMS with new bearings and seals.

A final documentation package that includes the final performance analysis with compressor curves, assembly drawings and installation instructions is included in the work scope. The new components will be installed in the spring of 2009.

REBUILT EXPANDER ROTOR/SUPPORT/EXHAUST ASSEMBLY

By Kurt Diekroeger



RMS developed a procedure that would allow for an obsolete, 20 year old expander exhaust casing and bearing support to be a overhauled to provide a drop in spare exhaust casing and rotor, bearing & seal sssembly (RBSA) for a unit that was installed in 2000.

The expander design required major disassembly/re-assembly for swap out of the rotor, which is typically the life-limiting item. Our customer owned an idle expander exhaust casing and a bearing support that could be reconditioned/modified and assembled with the current spare parts to be a drop in replacement. Utilization of a drop in replacement rotor, bearing support, and exhaust casing assembly will reduce rotor replacement time by 4 days.

This project involved a number of challenges including modifications of the casing and bearing housings to match the operating unit.

Rotating Machinery Services, Inc.

2760 Baglyos Circle
Bethlehem, PA 18020
Phone: 484-821-0702 / Fax: 484-821-0710
www.RotatingMachinery.com

Congratulations

RMS

~ 11 YEARS ~

ACCIDENT FREE

Our commitment is to our Worker and Worker Safety. We follow the ZERO INJURY safety philosophy – that no employee should ever be injured on the job.

Available Now / Quick Delivery •
Limited Quantities / Subject to Prior Sale •
Non-Emergency Breakdown Pricing •
Significant Savings

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For Inquires, Please contact:
Ken Kanopka
Regional Sales Manager - Northeast
Main Office: 484-821-0702 ext. 332
Cell: 484-896-8438

RMS WELCOMES...

KENAN KANOPKA
Regional Sales Manager—Northeast

Kenan Kanopka has joined our staff as Regional Sales Manager – Northeast. He brings over fifteen years experience in aftermarket parts, service and repairs to rotating equipment serving the steel, power generation, chemical, cement, and oil and gas industries.