ROTATING MACHINERY SERVICES, INC.

VOLUME 5, ISSUE 4

October - December 2009

THE FINISH LINE

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

RMS POWER SOLUTIONS IMPROVEMENTS

By Paul Poley

The shop operation has again made improvements and notched up our level of capabilities. We have added a new welding machine and Bader belt polishing machine. We have designed and built several tooling mandrels for balancing turbine discs and through the use of indexing technology built into our Schenck balance equipment, we are achieving some very low residual, close tolerance balancing results.

In order to evaluate the distortion of large gas casings, we have developed the capabilities to perform run out inspections. Our milling machine has been a real asset and quite a few small machine jobs have been processed on it with good results.

In development are several new improvements both in operational procedures and tooling capabilities. All of the current improvements at RMS Power have been implemented in order to support and strengthen our core competency. That core competency being the ability to execute high quality and fast delivery rotor repairs.

As a result of our efforts in developing a world-class rotor shop, we currently have 13 rotors in various stages of inspection and repair processing.

Thanks to all our customers for the business and we are looking forward to working with you next year. Happy Holidays!

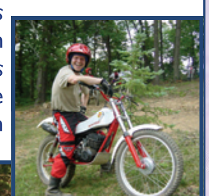
API 687 SEMINAR AT RMS

By Robert Klova

During the week of Nov 2, RMS hosted the API 687 Rotor Repair Course at our Bethlehem, Pennsylvania facility. It is the first time the course was given in the Northeast. The API Rotor Repair Course, based on API standard 687, covers the minimum requirements for the inspection and repair of rotating equipment rotors, bearings and couplings used in the petroleum, chemical and gas industry services. The week long course was presented by Cliff Cook, Chevron Fellow, former rotating equipment specialist with Texaco and chair of API 687. RMS engineers Dave Linden and Tony Rubino contributed to the writing of the API 687 standard.

"DIRT DON'T HURT…"

By Kathy Ehasz



As Bob Klova puts it, "*When I tell people that I race dirt bikes, everyone thinks 'Motocross', and probably thinks 'At your age?!'*" But the sport Bob has been involved in since he was 14 is called Observed Trials. He explains, "*It's slow, actually, and involves some precision - probably the kind of thing you'd expect an engineering type to do!*" The sport has a small, but very committed following in the United States, but is much more popular in Europe, especially England and Spain.

In simple terms, you are judged for how well, or even if, you can ride through sections full of obstacles such as hills, rocks and logs. *"It's fairly modest on our local level, but the top national guys do some wild stuff. Seeing a motorcycle clear an 8 foot high ledge or literally ride up a waterfall is pretty amazing."*

"More in keeping with my age and abilities, I compete mostly in vintage events in my old Montesa. I also like the aspect of restoring and maintaining old equipment, which is part of it."

What's Inside

HAPPY NEW YEAR

ROTATING MACHINERY SERVICES, INC.

&

RMS POWER SOLUTIONS, LLC.

WOULD LIKE TO WISH OUR CUSTOMERS & SUPPLIERS A VERY HAPPY & PROSPEROUS NEW YEAR!

RULE OF THUMB - MATERIALS TURBOMACHINERY

By Neal Wikert

AISI 310 SS: 25% Chrome – 20 % Nickel grade is known to embrittle in high temperature service and is rarely selected for turbine applications. Above 800 deg. F. the precipitation of carbides at grain boundaries reduces corrosion resistance by promoting inter-granular attack. This phenomenon is known as "**sensitization**". 300 series stainless steels are also subject to stress corrosion cracking, particularly in the presence of chlorides.

Incoloy 800: 30% Nickel, 20% Chrome material that offers good corrosion resistance, has high strength and resists oxidation, carburization and other harmful effects from high temperature exposure. The chromium imparts the resistance to oxidation and corrosion. The nickel maintains an austenitic structure so that the alloy is ductile. The nickel also contributes resistance to scaling, general corrosion and stress-corrosion cracking.

Inconel 600: Inconel 600 can be used in the as-welded condition. There is no requirement for a PWHT to get mechanical properties. However, a stress relief would be required if the part is subject to stress corrosion cracking or for dimensional stability. Inconel 600 is non magnetic at room temperature.

S. S. Type 347: The material will generally need to be stress relieved to accomplish dimensional stability. For the scroll application, the entire scroll will be stress relieved and this will eliminate the need for the intermediate stress relief.

Babbitt: Is ASTM B23 grade (or Alloy) 2. This is tin based with some copper and antimony. For journal bearings in high-speed industrial-sized machinery, we use 7 mils thick babbitt on bronze back pads (for increased fatigue strength) and 25 mils on copper thrust pads. We normally use 1/16th inch babbitt thickness on steel pads (journal and thrust)

Inconel 718: Has some limitations from a corrosion-cracking standpoint above 400 degrees F. in certain environments namely salts and chlorides. Inconel 718 for use as pins in steam turbines should be ordered to AMS 5663H, which calls for a RT minimum yield of 150 ksi, 185 tensile and 12% elongation, 15% R of A, and 1200 deg. F properties of yield 125, tensile 145, elongation of 12%, and a R of A of 15%.

A516 Grade 70: Material has been successfully used in expansion joint piping up to 1300 degrees F.

S.S. 403: The difference between 403 and 410 stainless steel is the amount of Silicon. 403 SS (.5 max) 410 SS (1.0 max)

The silicon content is what forms the delta ferrite in the steel. At higher silicon content, the delta ferrite becomes extensive and networked. The delta ferrite is what causes stringers in the material. 403 Stainless steel has a propensity to embrittle at temperatures above 900 degrees F.

COOPER RF2B BEARINGS UPGRADE

By Tony Rubino

A mid-western utility company compressor station utilizes a Cooper RF2B-24 two stage compressor in a gas storage application. The compressor provides inlet pressure boost (supercharging) to 2 or more downstream reciprocating compressors. The RF2B compressor operates seasonally, usually only during winter months. During a 2008 inspection, it was discovered that the compressor end journal bearing had catastrophically failed. The majority of the babbitt had separated from the steel-backing ring in what originally appeared to be a simple failure of the bond from the babbitt to the steel backing ring. Further investigation revealed that failure of the impeller end bearing had been a recurring event at this facility since 1993. At one time, the failure had been severe enough to damage the journal area of the rotor.

In September 2008, RMS provided a new bearing with anti-whirl features, larger clearance and increased oil flow. The bearing was not instrumented with either RTDs or thermocouples due to time constraints. A spare bearing was not ordered at that time pending validation of the new design. Subsequently, the new bearing was installed and appeared to have met design goals by reducing bearing operating temperature based on oil drain temperature measurements. However, compressor operation was still limited due to having to estimate bearing babbitt temperatures.

The bearing was returned to RMS in June of 2009 for inspection and installation of bearing temperature instrumentation. Installation of bearing temperature detectors is expected to provide increased operational control and compressor output. At inspection, the bearing looked as good as new with no signs of damage. RMS reworked the bearing and bearing housing to install two RTD's as well as fluid seal fittings. RMS also provided site installation drawings to address the necessary site rework to the compressor casing. The bearing has been installed and begun operation. Preliminary data indicates the bearing is operating 20°F cooler than anticipated and that operational restrictions will most likely be removed as more trend data becomes available.

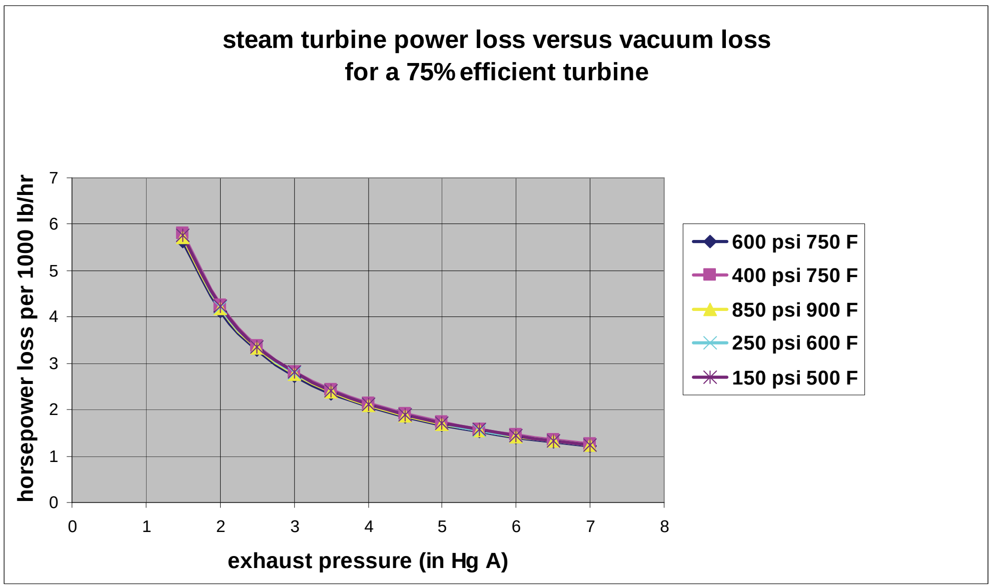
THE COST OF VACUUM LOSS FOR A CONDENSING STEAM TURBINE

By Sydney Gross

I was asked recently by a customer to quantify the penalty on horsepower for a given loss of vacuum. The machine was a six stage condensing impulse design that operated between 600 psig and 25 inches of Mercury vacuum. I was familiar with rule of thumb for such instances but had never verified it for myself. What I found was interesting and, with a little work, practical for application.

The first question I needed to answer for myself was the effect of the inlet conditions. I started by taking a survey of the inlet conditions of approximately 55 surplus condensing machines and using a representative sample of five. I then calculated the gas horsepower for each of these at several exhaust pressures assuming a nominal 75% efficiency. The difference in horsepower between increments of ½ inch Hg was plotted and is shown below.

Regardless of the inlet steam conditions, the results were nearly identical. The inlet conditions had little to do with the penalty of vacuum loss.



In order to put this into a useful format, the results were charted and are shown below. It should be noted that these figures do not take into account any change in turbine efficiency resulting from operating off-design. Expect less accuracy at larger differences in vacuum. If you know the isentropic efficiency of the turbine, you can ratio with the 75% used for these calculations.

STEAM TURBINE HORSEPOWER LOSS CHART

FOR LOSS OF VACUUM

	28.9												
1.0		28.4											
5.6	1.5		27.9										
9.7	4.1	2.0		27.4									
13.0	7.4	3.3	2.5		26.9			"HGV					
15.8	10.1	6.0	2.7	3.0		26.4		"HGA					
18.1	12.5	8.4	5.1	2.3	3.5		25.9						
20.2	14.5	10.4	7.1	4.4	2.1	4.0		25.4					
22.0	16.4	12.3	9.0	6.2	3.9	1.8	4.5		24.9				
23.7	18.0	13.9	10.6	7.9	5.6	3.5	1.7	5.0		24.4			
25.2	19.6	15.4	12.2	9.4	7.1	5.0	3.2	1.5	5.5		23.9		
26.6	21.0	16.8	13.6	10.8	8.5	6.4	4.6	2.9	1.4	6.0		23.4	
27.9	22.3	18.1	14.9	12.1	9.8	7.7	5.9	4.2	2.7	1.3	6.5		22.9
29.1	23.5	19.4	16.1	13.4	11.0	8.9	7.1	5.4	3.9	2.5	1.2	7.0	

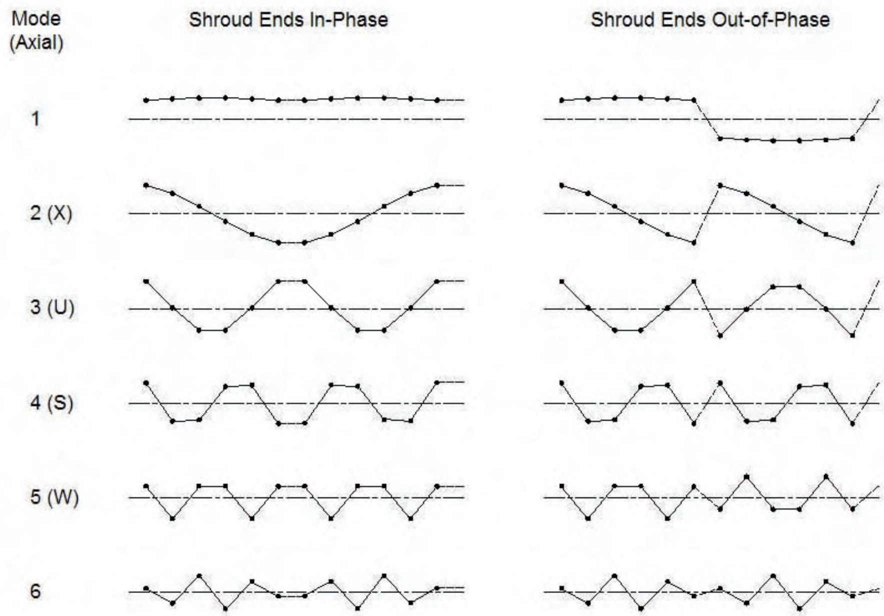
locate initial pressure in column and final pressure in row, the intersection is the loss in HP for 1,000 lb/hr flow

example: from 2.0"HGA (27.9"HGV) to 4.0"HGA (25.9"HGV), lose 10.4 HP per 1000 lb/hr steam

MODE SHAPES OF PACKETED BLADES

By William Sullivan

The last installment of the interference diagram series described the mode shapes possible with a single packet of six blades. This installment will describe how these mode shapes arrange themselves around a bladed disk with ten packets of six blades each. For simplicity (and visibility) we will look at axial modes only. The concepts described for axial modes apply to tangential modes as well.



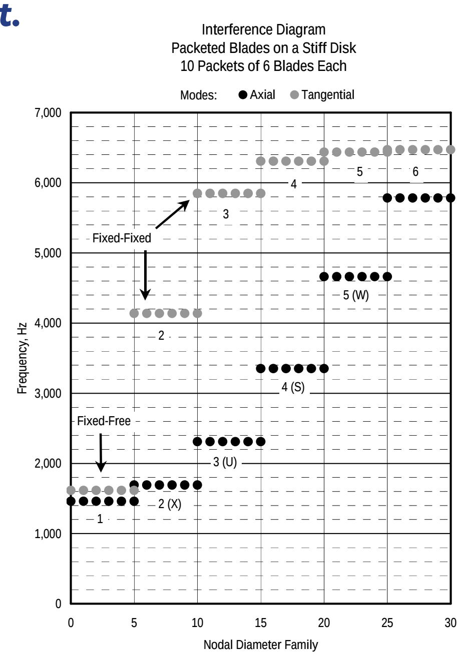
In the axial blade modes of vibration, most of the blade motion is perpendicular to the plane of the disk. This makes defining the nodes (locations of no displacement) relatively easy. The nodes will be the locations where a line running from the point of maximum deflection from blade to blade crosses the center plane of the disk. These are called "0 (zero) crossings". For a single packet of six blades there can be up to five nodes (0 crossings) within the packet. For any two adjacent packets of blades, there can be no more than one node from one packet to the next packet. Therefore, for ten packets of six blades each arranged around a disk rim, there can be a total of up to 60 nodes. These nodes represent nodal radii. For interference diagrams of bladed disks, we work in nodal diameters. Therefore, the maximum of 60 nodal radii described above represents 30 nodal diameters (1/2 the number of blades). There always will be an even number of nodes and, therefore, a whole number of nodal diameters around a bladed disk because the end of a line going from maximum deflection to maximum deflection from blade to blade must come back to the original maximum deflection. In other words, the nodes must occur in pairs.

The number of nodes around a disk for a given shroud mode shape will depend on the mode being excited and the phase relationships of the packets around the disk. That is, adjacent packets of the same mode shape can be in-phase or out-of-phase. The packet-to-packet phase relationship determines whether or not there is a node between the packets. The figure above shows the packet-to-packet phase relationships possible with packets of six blades arranged around a disk. Although two packets are shown for each case, the nodes (0 crossings) are tabulated below the figure for one packet only. Two cases are shown for each mode: shroud ends in-phase and shroud ends out-of-phase. In any grouping of separately packeted blades around a disk, all of the shroud ends can be in-phase (exceptions occur with odd numbers of packets), all can be out-of-phase (exceptions occur with odd numbers of packets), or there can be any combination between these extremes (as long as the result is a even number of nodes).

Mode	Number of Nodes (0-Crossings) in				Number of Nodal Diameters in 10 Packets of 6 Blades Each	
	1 Packet of 6 Blades		10 Packets of 6 Blades Each			
	Ends In-Phase	Ends Out-of-Phase	Ends In-Phase	Ends Out-of-Phase	Ends In-Phase	Ends Out-of-Phase
1	0	1	0	10	0	5
2 (X)	1	2	10	20	5	10
3 (U)	2	3	20	30	10	15
4 (S)	3	4	30	40	15	20
5 (W)	4	5	40	50	20	25
6	5	6	50	60	25	30

MODE SHAPES OF PACKETED BLADES *Con't.*

The interference diagram to the right shows how the mode shapes described above are arranged for ten packets of six blades each. This diagram shows only the first order (0 nodal circle) modes of vibration on a stiff disk for clarity. (The tangential modes are shown along with the axial modes described above because they do not reduce the clarity and it is important to show that the tangential modes exhibit the same behavior as the axial modes, although the nodes are not as evident.) The numbers near each group of modes identifies the shroud (not the blade) mode shapes. Note that the modes are grouped together by frequency and possible nodal diameters.



For example, a second axial shroud mode of vibration, the "X" mode, in this arrangement will always have from 5 to 10 nodal diameters. This behavior often can be used to move vibration modes away from possible sources of vibration or to determine what nozzle count to use to avoid exciting a particular mode. This is one of the chief uses of the interference diagram and why it is such a powerful tool. It shows not only what configurations could cause a blade excitation, it shows what configurations <u>will not</u> cause a blade excitation.

FCC EXPANDER FLOW PATH EROSION - EXPANDER EROSION

By Dave Linden

Despite the use of various separation technologies, all FCC expanders ingest a significant amount of catalyst throughout thier operating period. The quantity as well as the size of the particles determines both the erosion and deposition characteristics that are experience in the expander. The following Figures #1 & 2 show two different expander installations.

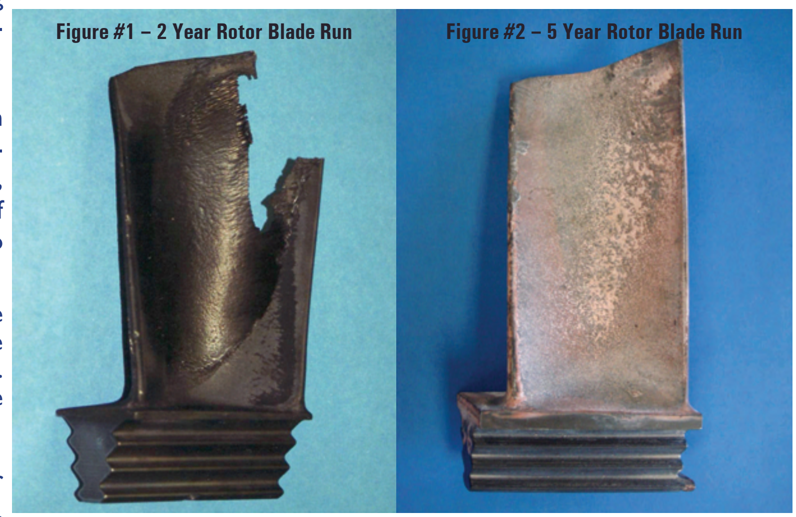


Figure #1 – 2 Year Rotor Blade Run

Figure #2 – 5 Year Rotor Blade Run

Figure #1 shows a rotor blade that operated with above normal catalyst loadings. The noted wear occurred in less than two years of operation. In contrast, the blade shown in Figure #2 operated in excess of five years of service and has experienced virtually no erosive wear.

<u>Catalyst Quantity</u> – Field testing and operational experience has shown that expander blades will experience erosive wear when the particle loadings exceed 150 ppm by weight. Rapid and excessive wear occurs when the loadings are above 300 ppm.

The new separation technologies of today are far more efficient than the early third stage separators. Expander catalyst loadings of less than 50 ppm have been achieved and 5+ year expander rotor runs with blading being removed in the condition of Figure #2 have become common.

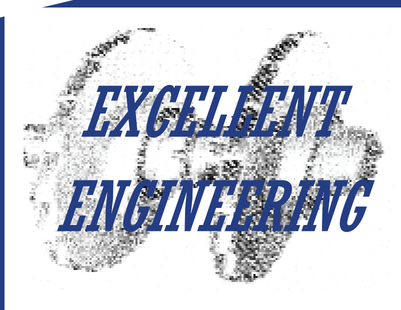
<u>Particle Distribution</u> – As previously discussed, the particle size distribution is just as important from an erosion standpoint as is the quantity of catalyst. Larger catalyst particles (> 20 microns) are far more erosive than the fine particles. At one time, a rule of thumb was that if the separator could remove all of the 10-micron and larger particles, the expander could operate with minimal erosion over a four year operating campaign. Experience has shown, that even the small catalyst particles (1 to 10 micron) can be erosive if there are sufficient quantities. The erosion patterns caused by large catalyst particles, differs greatly from that caused by small catalyst particles.

Rotating Machinery Services, Inc.

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

PRSRT STD
US POSTAGE PAID
EASTON, PA
PERMIT NO. 65

RETURN SERVICE REQUESTED



PRODUCT LINES:

AXIAL COMPRESSORS • CENTRIFUGAL COMPRESSORS • EXPANDERS • GAS TURBINES • POWER TURBINES • STEAM TURBINES

RMS COMMUNITY INVOLVEMENT

By Kathy Ehasz

Rotating Machinery Services, Inc. strongly believes in the philosophy of "Giving with a Purpose." We seek to enhance quality of life in the communities, in which we operate and to serve humanity by supporting and inspiring involvement with causes that make a profound social impact.

We support organizations both financially and through volunteerism, by helping to build awareness of their mission and goals and by inspiring our employees, customers and associates to become involved in their good work.

RMS participated in helping to provide some holiday cheer to those less fortunate this past Thanksgiving. This past year has brought tough economic times to many people. The food banks and shelters have been depleted of food, clothing and necessities that many were left without.

RMS and neighboring companies participated in a food drive with Second Harvest. A challenge to collect 500 pounds of food was set and achieved. The final total of food collected was 619 pounds.

Organizations RMS has supported in 2009 are:

National Multiple Sclerosis Society • American Diabetes Association • Action for Animals • Susan G. Komen Breast Cancer Foundation • United German Hungarians • American Heart Association • American Cancer Society • Special Olympics

If you would like to receive our newsletter via email, please contact Kathy Ehasz at 484-821-0702 or kehasz@rotatingmachinery.com.