ROTATING MACHINERY SERVICES, INC.
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THE FINISH LINE

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RMS POWER SOLUTIONS

By Paul Poley

The RMS Power Solutions shop is growing very nicely and is gaining some strength as a full service overhaul provider to the turbo-machinery market. 2010 turned out to be a very successful year and many improvement goals were achieved, but the most significant one was that we were able to have a positive impact on the turn-around time associated to the support we provide for the engineering group. In addition to response time, by bringing more work in house, we made a positive impact to the cost of goods and this will serve as a tool to allow RMS to be more competitive in the market.

Our shop has had as many as 30 turbomachinery rotors in various stages of processing at one time. We have also accomplished many complete rebuilds of various machines, and our complete reworked equipment leaves the facility in world-class order and ready for service.

Our 40,000 pound capacity balance machine is fully functional and has been a major player in supporting the rotor shop work over the past several months since it was commissioned. We have also added one more senior rotor technician to our shop staff and feel prepared to support a much greater level of rotor repair processing.

Planned for the very near future is a climate controlled inspection room. This room will house our inspection equipment and provide a controlled and clean environment for component dimensional inspections to be performed. This room will be managed and maintained by our quality control inspector.

100 WINS - PROUD GRANDPARENTS ANNOUNCEMENT

On Saturday March 6, 2011 at Wilson West Lawn High School Pius X Senior Josh La Bar, Grandson of Don (Bondo) & Caryle Boyer and the son of Kevin and Tracey (Boyer) La Bar, recorded his 100th career wrestling victory. Josh beat Junior Noah Balsone of Bermudian Springs by major decision 13-4. Josh finished 4th at the PIAA District XI AA Wrestling Tournament to qualify to wrestle at the PIAA Southeast Regional Wrestling Tournament at Wilson West Lawn HS.

Josh's started wrestling when he was in 2nd grade and continued participating in the sport throughout his high school career. During high school, he wrestled at the 119-pound weight class. He stayed at this weight for his sophomore year. He moved up to 135 pounds for his junior year and wrestled at 145 for the 2010-2011 season. His final seasonal records were 9th grade 10 wins – 21 losses; 10th grade 28 wins – 11 losses (Finished 6th D11); 11th grade 27 wins – 14 losses (Finished 5th D11) and this season 35 wins – 12 losses (Finished 4th D11). Josh is only the ninth wrestler in Pius X history to be a four-year letter winner and win 100 wrestling matches in a career.

Josh is also a catcher of the 2010 District 11 Class A Baseball team and PIAA State semifinalist team. He was a Split End for the Pius X football team catching 31 passes for 378 yards and 3 TD's. He was named 2nd team All-American League as a receiver and selected by the PA Football News as an honorable mention wide Receiver.

What's Inside

RMS CONFERENCES

SYNGAS Conference
Tulsa, Ok
April 25 - 27
Booth 17

40th TURBOMACHINERY SYMPOSIUM
Houston, TX
September 12 - 15
Booth 740

Please stop by our exhibit!!

TURBOMACHINERY RULES OF THUMB

By Neal Wikert

LUBE CONSOLE TESTING

When buying new, the oil system needs to be run in a shop to test operation and to confirm sound levels and cleanliness. The oil used for testing must be compatible with the (intended) site oil.

Testing must include detailed cleanliness testing with recirculation and cleanliness testing of the screens. The console must be run in a normal operating mode and be performance tested. Tests must be able to confirm design pressure and temperature levels and should also include the following:

1. Leakage test (repair and eliminate all leaks)
2. Proper operating at set point of all relief and pressure limiting valves.
3. Transfers of filters and coolers without the auxiliary pump starting due to low pressure.
4. Successful pump and system performance in both (aux. pump on/off) directions.
5. Successful operation at defined normal flow conditions with only one pump running.
6. Operating checks of all warning, protective, and control devices.
7. Tests of plug leakage across transfer valves.

To conduct testing, screening is needed, as is a recirculation flow meter for measuring the oil flow rate. The (steam turbine) main pump driver needs to be replaced with an electric motor with a suitable motor control. The auxiliary pump driver also needs a suitable motor control, one that provides for automatic startup and shut down.

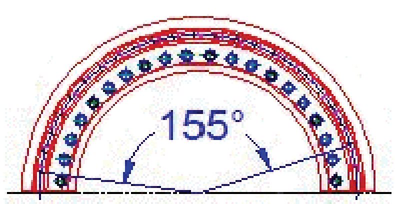
Provision must be made to power up electronic transmitters, as there are few pressure gauges in the system.

Because the main valves are electronically driven, local controllers need to be purchased and programmed as substitutes for the more critical functions that may not exist at the test facility.

STEAM TURBINE ARC OF ADMISSION

By Sydney Gross

Arc of admission pertains to steam turbine nozzles and diaphragms and is the angle through which steam passes to the rotor blades. The figure below shows a typical nozzle that admits steam through a 155° arc. The admission is more commonly referred to as a fraction of a full 360° arc. In this case the admission would be 155/360=0.43 or 43%.



Partial arcs are most commonly used in the first, or control, stage and extraction stage nozzles. However, they can be found throughout all stages of a turbine.

When steam is admitted through a partial arc there are additional losses associated with steam leakage from the ends of the arc and fluid friction in the non-admitting areas that are not found in full arcs. Some of these losses can be mitigated through use of a trough around the rotor blades in the non-admitting areas called a windage ring.

So, why would one design a partial arc admission stage? One prevalent reason is economy. Industrial turbine casings are most commonly designed horizontally split and with a steam chest and valve rack in the top half only. Single valve turbines typically have a steam chest in the bottom half. In order to have a full admission one must get steam to both halves. This would require a more complex casing and valve design, and by complex I mean expensive. The performance loss may not be an issue for the application.

Another reason is control. Whether or not there is steam flow to the bottom half of the casing, the first stage of a multivalve turbine is the control stage and is responsible for varying load conditions. The valves vary the arc of nozzles that see steam, thereby varying the capacity of the machine.

To obtain the design area in a 360° arc, it may be necessary in some cases to shorten the passage of the diaphragm or nozzle to an undesirable height. When nozzle passages approach approximately ½" in height, losses due to the end walls become significant to the point that they surpass partial arc losses with a taller passage. This is the primary reason that partial arcs may be found in diaphragm stages.

In addition to the above concerns, when considering partial arc stage designs one should take into account that the loading on the rotor blade will be greater than full arc by the inverse of the admission fraction. The rotor blades are also subject to full loading and unloading in each revolution, a condition known as partial arc shock loading. These are some of the reasons for the more robust design of control stage blades.

CENTRIFUGAL COMPRESSORS

By Robert Huffman

2011 has picked right up where the second half of 2010 was for RMS in regards to centrifugal compressor work and that is in one word, BUSY! We have grown our capabilities in 2010 and early 2011 by way of addition of industry experienced people in engineering and our shop as well as capital expenditures for shop machines and tools.

The result of this is that RMS has seen a major influx in centrifugal compressor work for services, repairs, overhauls and re-rates as compared previous years before 2010. Some highlights of contracts we have completed or are in the middle of include: complete compressor overhauls for an Elliott® 29MB and Carrier® 18SP450 compressors, rotor overhauls for three I-R® MTA-5124, one Delaval® 4/5C88 and one Delaval® 7CK31 compressors. We additionally have a contract in place for a compressor overhaul of a Delaval® 7CK31 wet gas compressor that will be done inside of a 10-day outage window. Three projects in particular that have presented unique challenges for RMS to showcase our capabilities have been the Carrier® 18SP450 overhaul, one of the MTA-5124 rotors and a new project we have just kicked off for a complete new compressor casing.

1) The Carrier overhaul required a complete machining of the horizontal joint, weld build up of all diaphragm fits, the seal housings to restore the seal cartridge o-ring lands and the bearing case fit bores. The diaphragms have also required restoration of their fits. This machine had been idled for about 2 years and was in extremely poor shape when received. Between the main and spare rotors, components were combined to build up one good rotor for operation as both rotors had impeller vane erosion damage.

2) The MTA-5124 rotor was received with one damaged vane in the first stage impeller. This rotor is a riveted vane construction on all impellers. The single damaged vane was removed from the rotor, the vane was weld repaired and re-riveted back into the impeller. All impellers were mag particle inspected and no other damage was indicated. The impeller was re-mounted on to the rotor and the rotor was final balanced with the addition of balance weights.

3) The new compressor casing order represents a milestone in RMS's history as it will be the first complete new casing designed and manufactured by RMS. This will be a horizontally split compressor casing for a refrigeration service machine. The casing will be a drop in replacement and will utilize the existing spare rotor. The customer also has some of the diaphragms as spares that will also be used, with RMS designing and manufacturing the balance of the required stationary components. RMS has reverse engineered the existing bearings so that this design can be utilized for the new compressor casing, where by enabling the customers stock of existing spares to be utilized. A new shaft end seal design will be employed to resolve current reliability issues.

A key shop machine tool addition in 2010 was a 40,000 lb capacity Schenck Balance Machine. This size of balance machine allowed us to handle the Delaval 4/5C88 and MTA-5124 rotors which are each in the 20,000 lb rotor weight range. We continue to improve our rotor handling capabilities for cleaning, indicating and in-process handling to improve our productivity to enable us to offer quick turnarounds to our customers.

Having just started work at RMS in January of 2011, I have been impressed with RMS's ability to quickly react to customer needs and the willingness to bring on the people and tools needed to have a first class compressor service and re-rate capability.

SUPPORTING OUR CUSTOMERS

By Glenn Gaddis

The first quarter of 2011 has been a good for us here at RMS. We have received several orders for new equipment and many repair orders. We are also currently supporting five (5) Turnarounds. Several of the turnarounds are being supported on site while others are being supported here in Bethlehem on a 24 hour manufacturing basis.

We received an order to provide a new, first of its kind, 10mw Gen Set package mounted on a base plate complete with lube system and complete control system for a client in the Middle East.

We also received an order to make a new Compressor Casing for a refinery here in the states. This casing will be fabricated with cast nozzles and will include a new lube oil system.

Support for our customers is first priority, and we welcome any challenge brought to us.

REMAINING LIFE ASSESSMENT

By William Sullivan, PE

In the last newsletter we discussed the remaining life assessment of turbomachinery rotors. In order to obtain a good estimate of the rotor blades and disk lives, the rotor blade and disk operating temperatures are required. The rotor blade airfoils can conservatively be estimated to be at the relative total temperature of the flowpath gas through the blades. The blade root and disk attachments can be estimated from a combination of the heat flow from the blades to the disk and from the blade root and disk exposed surfaces into the disk cavities. This leaves heat flow from the disk to the rotor shaft and cavities.

A simple schematic of a single stage power turbine or expander can be seen in Figure 1.

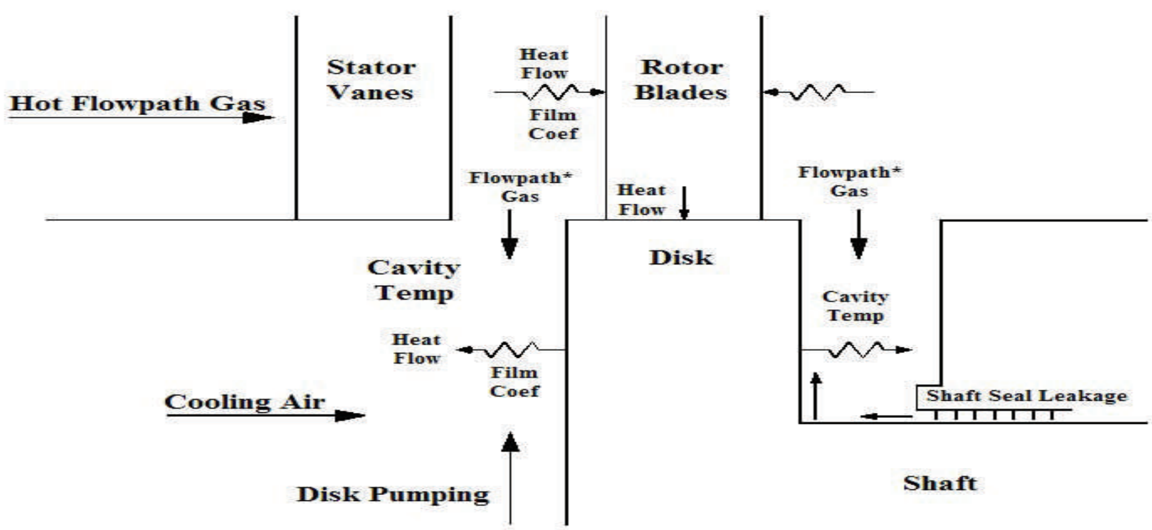
The calculation heat flow from the disk to the cavity (or from the cavity to the disk) is not particularly difficult provided all of the flows and temperatures are defined. However, due to the complex thermal interactions and interdependencies in the disk cooling system, there can be a degree uncertainty associated with the calculated disk temperatures that requires experienced judgment in the analysis. Of course, accurate measurements of actual disk or cavity temperatures will greatly reduce the uncertainty.

One of the more interesting aspects of the disk cavity thermal system is the effect of disk pumping. A disk rotating in a fluid is going to “pump” the fluid. That is, as a disk starts to spin, a film of fluid will start to flow along the disk due to viscous drag. However, in this case, instead of the fluid slowing the disk, the disk accelerates the fluid. At the same time, as the viscous drag is pulling the fluid in a circular direction (with the disk surface) inertial (mass) forces are adding a radial component to the fluid flow.

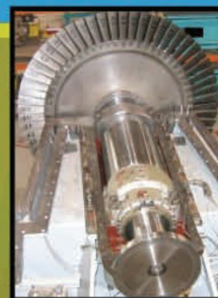
As the disk pumping action pulls fluid from the base of the disk, the pressure at the base starts to decrease and this causes fluid from the slightly higher pressure area away from the disk surface to flow to the disk base and, ultimately, up the disk face. In a system with no disk rim sealing, like that shown in Figure 1, the pumped flow enters the flowpath. If there is no source of fluid in the cavity to make up for the lost flow, the required flow is drawn in from the flowpath. Therefore, with no (or worn) seals, hot gas flow will enter the disk cavity at the same time cooler gas is leaving the cavity.

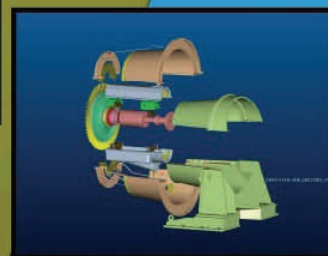
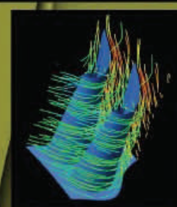
If fluid from some other source (cooling flow, for example) enters the cavity, that flow will also enter the “pumped” system and flow from the flowpath to the cavity will decrease. If the added flow equals or exceeds the disk-pumping requirement, no flowpath gas will enter the cavity.

The disk pumping rate, cavity fluid temperatures and disk surfaces temperatures all affect the heat flow from the disk to (or from) the cavity and all are necessary to estimate the disk film heat transfer coefficients used in the analysis. Therefore, the calculation of the disk temperature profile is an iterative process with each step yielding a better estimate (until the step-to-step temperatures no longer change).



* Leakage from Flowpath (Disk Pumping - Cooling Air). If the Cooling Air Flow Rate exceeds the Disk Pumping Flow Rate then there is No Leakage from the Flowpath.

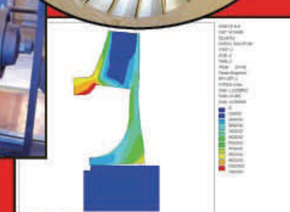









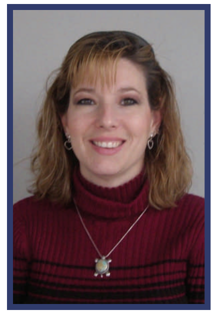


RMS WELCOMES TO OUR TEAM…

By Kathy Ehasz

ROBERT HUFFMAN
PRODUCT MANAGER— CENTRIFUGAL COMPRESSORS

Bob has twenty two years of turbomachinery engineering experience spanning steam turbines, turbochargers and centrifugal compressors. Experience in design engineering, application engineering, testing, field support, project management and sale for custom engineered solutions for turbomachinery applications. Also experienced in mechanical drive and generator drive turbine applications, air and process gas compression and lean burn emissions projects for turbochargers on large gas and diesel engines. Previously held positions at Elliott Turbocharger Group, GE Conmec and GE Oil & Gas.



CAROL HAMM
PROJECT MANAGER

Carol has twenty-two years of Pump & Turbomachinery contracts, customer service & project management experience. Her experience includes various products from Steam Turbines, Expanders, Axial and Centrifugal Compressors.

Previously held positions at Ingersoll Rand and GE Conmec.

LINDA AMATO
ADMINISTRATIVE SUPPORT

Linda has over twenty years experience in the Turbomachinery Industry in the areas of order entry process, accounts receivable support for engineering, marketing, purchasing and contract / legal functions. Previously held position as Ingersoll Rand and GE Conmec.

JOHN A. INNES
MARKETING SPECIALIST

John has forty-two years of turbomachinery experience in areas of cost analysis, parts and repairs pricing, marketing, sales order entry, stock and inventory control, and accounting. Instrumental in installation and transition to an enterprise computer system. Previously held positions with Ingersoll-Rand, Kearfott, and Conmec.

THOMAS STRAUB
ASSEMBLER

Tom joins RMS Power Solutions with forty years of experience in machine shop and assembly of rotating machinery. Previous experience with Ingersoll-Rand, Cooper Energy and GE/Conmec.

MIKE YOUNG
ASSEMBLER

Mike joins RMS Power Solutions with 30 years experience in machine shop and assembly of rotating machinery. Previous held positions with Ingersoll-Rand, GE Conmec and Dresser-Rand.

SUPERIOR CUSTOMER SERVICE: CONTINUOUS IMPROVEMENT

By Robert Klova, PE

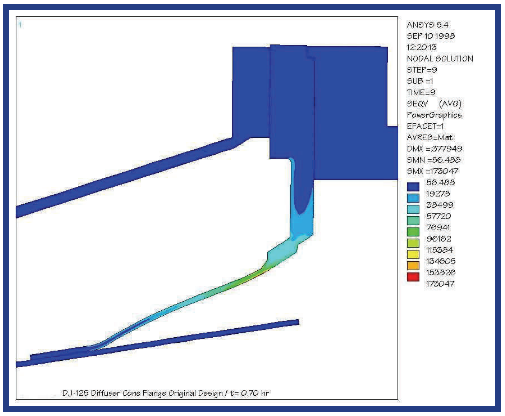
Inlet Annulus Being Installed in the Field

In 1998, RMS was contracted by a major US natural gas company to correct a chronic cracking problem on their Dresser-Clark DJ125 power turbine inlet ducts. The component, called an Inlet Annulus, directs flow from a GE LM1500 gas generator into the DJ125 aero derivative power turbine. Our customer experienced cracking at the Annulus mounting flange which required weld repair on a yearly basis.

To determine the cause of the cracking, RMS performed a finite element analysis (FEA) to determine peak stresses, which, it was determined, are caused by high transient thermal loading during start-up. These thermal stresses are caused by the rapid heating of flow-washed surfaces, being constrained by the relatively cool Annulus mounting flange. As shown in the stress contour plot (on page 7), peak stresses were calculated to be 175,000 psi. Since the part is manufactured from AISI 304 stainless steel, which has a yield strength at temperature of under 30,000 psi, cracking at the peak stress location was predicted to occur with very few thermal cycles (start-up and shut-down). *(Con't next page)*

Con't. SUPERIOR CUSTOMER SERVICE: CONTINUOUS IMPROVEMENT

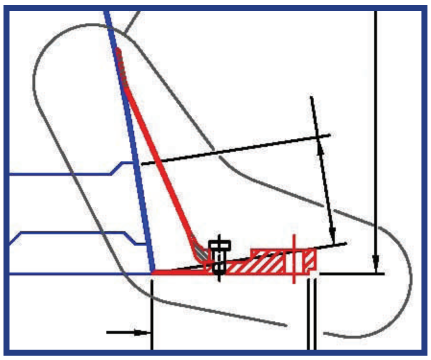
By Robert Klova, PE



Original Design Showing Extreme Stresses in the Area Where Cracking Occurred

RMS's solution to the problem was to separate the hot flow-washed section of the annulus from the much cooler mounting flange. Axial expansion pins were employed to maintain concentricity of the duct. By doing so, thermal stresses were reduced to 20,000 psi. RMS provided three new Inlet Annuluses to our customer with the redesigned geometry. The design has been very successful, and completely eliminated the need for weld repair since the new ducts were placed in service.

After a number of years of operation, however, cracking began to occur in around the expansion pins that align the hot duct section to the mounting flange. This has been managed by stop-drilling these cracks when they are found. RMS recently recommended a redesign of the mounting flange to our customer, to eliminate this cracking, since their turbines are now expected to continue in operation for another five to ten years.

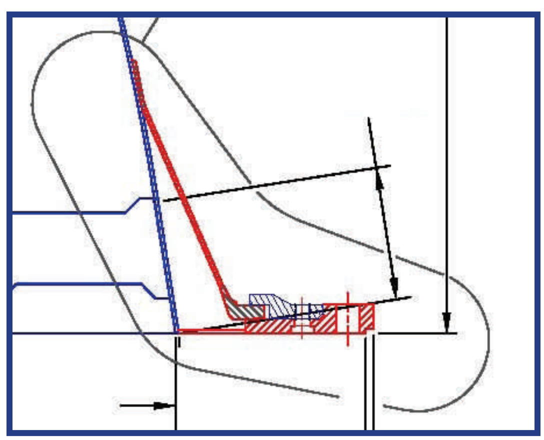


First Redesign Showing Expansion Pin

Expansion Pin During Routine Inspection

The new design eliminates the expansion pins entirely and controls the position of the duct using a pilot fit that is loose when cold and tightens during operation. Since the duct has no critical clearances to other components, concentricity while shut-down is not important. The design challenge for RMS was fit the new arrangement in the existing flange location, and also to allow the flange, which is relatively inexpensive, to be retrofitted to their existing Annuluses. Our final solution is shown in the following figure. This is one example of RMS' unique engineering and manufacturing capabilities helping our customers operate their older equipment reliably for much longer than their original design lives.

Flange Cracking Around Expansion Pin



Redesigned Flange Eliminating Expansion Pins

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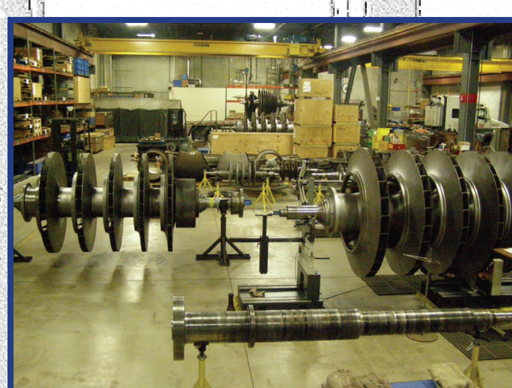

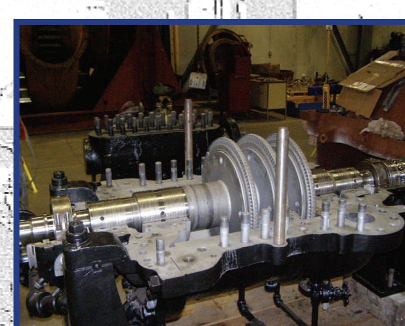

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