Rotating Machinery Services, Inc.

SPECIAL EDITION

The Finish Line

VOLUME 13 ISSUE 1 December 2017

RMS NEWS

RMS EXPANDS CAPABILITIES WITH AQUISITION

By Chot Smith

In late May 2017 RMS completed the acquisition of Houston based Mepco Inc. The RMS/Mepco Service Center now gives RMS a substantial presence on the Gulf Coast enabling us to expand our customer base and to offer options to better serve our existing southern customers. We recently completed our first major turn-around project from a refinery in northern Texas. Prior to our Houston acquisition we would not have been awarded this project due to logistics. We removed a 3013 I-R style Axial Compressor used as the main air blower for the refineries catalyst cracking process, shipped it to our Houston facility and refurbished the unit with the customer's spare rotor and components. The project was scheduled for a 10 day duration and was accomplished in 9 days using a combination of our field and shop personnel. Our customer stated that they were extremely pleased with our performance and delivery improvement and look forward to utilizing our Houston facility for future projects.

Inside Report

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RMS NEWS

2017 PRT Roundtable

By Don Shafer

Rotating Machinery Services, Inc. was pleased to present our FIFTH! Power Recovery Train Roundtable on May 2nd and 3rd of 2017. Our postponement from the original dates in 2016 due to attendee travel restrictions and unplanned outages, did not impact attendance and the majority of attendees were able to adjust their schedules and attend on the new dates.

During the presentations over the two days the presenters provided information on all aspects of Power Recovery Train design, reliability operation and lessons learned. Some of the key topics of discussion were FCC Expander Design, Axial Compressor Design, Expander Deposition and Erosion, Online Monitoring, Structural Analysis, Rotor Dynamics and Field Service / Outage Planning. A tour of the RMS shop was also given on the last day to highlight our shop capabilities and all the new capabilities and upgrades. The majority of attendees were PRT end users / machinery engineers who during the discussions shared valuable first-hand experience on field problems and issues that affect them at their sites. This type of interaction with the end users helps to improve RMS's ability to provide more reliable PRT equipment and makes us all better machinery designers and operators.

Once again the RMS Team would like to thank all of the attendees, presenters and RMS staff for making our fifth PRT Roundtable a huge success. We would like to make everyone aware that based on the continued requests and positive responses from the attendees, RMS plans to continue to hold the PRT Roundtables annually. Look for the 2018 dates in future newsletters.

A Lesson in Safety – Learning From Each Other

By John Bartos

For those of us who have worked our entire careers in the machinery, petrochemicals, oil & gas industries, an emphasis on safety has become second nature. Whether we're in the shop, in the field, or at site there are constant reminders in the form of posters, policies and procedures. Almost all participants in these industries undergo some kind of formal training centered on safety on a regular basis. These are all excellent tools that exist to serve as a continual reminder of the need to remain diligent and to always conduct ourselves in the safest manner possible.

The ever-improving safety records within these industries are a testimony to the positive impact that this emphasis on training and culture has had. However, there are always areas for improvement.

One very effective tool that I've found in driving a safety-oriented culture within the workplace is the practice of sharing personal experiences. This initiative was introduced at RMS earlier this year. The premise is really quite simple. At the start of a meeting, one of the participants either volunteers or is "volunteered" to share with the group something from their own experiences that is a valuable safety lesson for the rest of the group. Like anything new, this practice took a little while to get used to. However, once we got through the first few sessions the idea began to catch on. Suddenly, I had people sending me emails and stopping me in the halls to volunteer their safety story for the next upcoming meeting. I watched the teams transform into safety advocates eager and willing to share their experiences with others so that they could learn, and avoid a similar situation.

RMS NEWS

A Lesson in Safety – Learning From Each Other *con't.*

By John Bartos

Nurturing a safety culture is not something that can be driven only from the top. It must be embraced at all levels of the organization. Encouraging teams to share their safety thoughts with others enables them to be a leader of that cultural movement.

Cooper RF2B - 24 Refurbished and Seals Upgraded

by Tom Keating, PE

A Midwest utility company had its Cooper RF2B-24 two stage centrifugal compressor refurbished and the seals upgraded by RMS. The compressor is used in booster service primarily in the winter months, with outage availability in the summer to late fall. RMS assisted this customer with a bearing upgrade in September 2008, which addressed repeated bearing failures. In September 2015, RMS renewed the rotor to the customer's specifications and made recommendations for upgrades and repair for a future outage. This led to an award to RMS in June 2016 to refurbish the entire compressor at its Bethlehem shop and to design, manufacture and install an upgraded seal package.

The customer had been experiencing high oil leakage into the gas path for many years. As of late, the loss had been as high as 50 gallons per day. This amount of oil caused problems for the downstream equipment with coking deposits and fouling. The source of the lost oil was suspected to be the process seal, a bushing type, which could allow a large amount of seal oil to pass on to the process side on a regular basis. RMS suggested a face type mechanical oil seal which could potentially reduce the leakage by ten times and the customer agreed.

It was noted that the compressor casing had been in place for about fifty years with only routine maintenance, so RMS did a complete teardown, blast clean and paint job. The compressor's o-rings, gaskets, labyrinth seals and fasteners were replaced. The suggested rotor repairs were performed: a new main shaft was provided, and impellers were re-bored and treated with an anti-corrosion coating. As a part of the seal upgrade, a Rotordynamic analysis was performed which confirmed the rotor's stability and predicted long term reliable service at design conditions.

RMS precision balanced the rotor to API 687 standards and fully assembled the compressor and then readied it for its trip back to the utility. The compressor was reinstalled by RMS Field Service personnel who had supervised its removal from site, ensuring that the compressor was exactly and efficiently positioned as before.

The unit is currently running providing pressure boost for the winter months. The customer graciously remarked that the compressor vibration readings were the lowest they had ever seen and that they appreciate everything RMS did and its attention to detail. High praise indeed, but RMS approaches all customers and projects with the same philosophy – "Quality from start to finish".

TECH TALK

COUPLING BUCKLING ANALYSIS

By Christopher Sykora

Recently, RMS was contracted to perform a Root Cause of Failure Analysis (RCFA) for a major oil refinery in the Southwest US. The high speed coupling (flexible disc type) on their coker compressor train had a bad habit of failing during startup, multiple times in recent years. The coupling had been designed to meet the margin requirements of API 671 during maximum predicted, peak torque delivered from the induction motor/gearbox. However, the coupling still had repeated failures that resulted in a permanent deformation of the flexible disc web into the shape of a spiraling wave and eventually, shearing of the metal & complete separation on one side. A picture of the failed coupling is shown in **Figure 1**. Due to the shape of deformation, a buckling induced failure mode was suspected.

Buckling is a failure mode that is quite different from the more typical tensile overload or cyclic fatigue scenarios. Buckling is a sudden, non-linear increase in lateral deformation of a structure when compressive loading exceeds a critical level. Since buckling is an instability failure, it is largely dependent upon the stiffness of the structure, rather than its material strength. The structure can typically still support some load after buckling, but capability is significantly reduced and as in this case it can quickly lead to shear/bending failure. The large deformations associated with buckling might also be a problem (even if the metal hasn't failed) if they interfere with other equipment (i.e. thin walled casing deforming into a rub situation). Some classic examples of buckling failures are columns in compression and thin walled cylinders experiencing external pressure or axial compression. **Figure 2** shows some examples of these.

For this RCFA, the couplings were inspected for cracks and the material tensile strength and ductility were determined. None of these steps produced any evidence of material defects that could have been the cause of the failure. So a finite element analysis (FEA) was used to investigate the buckling capability of the as-built coupling. This served as an independent check on the coupling manufacturers stated design limits. The analysis geometry was based on a detailed Faro arm inspection of the as-built coupling with the minimum inspected thicknesses. A steady stress analysis with maximum speed, torque, & misalignment bending was performed first and it verified that the steady stress in the coupling was less than the tensile strength measured for the 4340 steel material.

The buckling capability of the coupling was predicted in the FEA by doing a nonlinear structural analysis with the torque load incremented in many small steps from zero to high load. While this analysis technique increases computational time due to the many small steps, it is the most accurate method of predicting the buckling. When the coupling structure reaches the load at which buckling begins, there is a significant increase in the slope of the graph of out of plane deformation vs applied load. **Figure 3** shows the resulting graph from the analysis and the associated spiral pattern buckling deformation. The RMS analysis predicted buckling load capability very similar to the design rating specified by the coupling manufacturer. Subsequently, the manufacturer performed a static load test (also incremented in small steps) on a spare coupling that would further validate their original design predictions. Buckling occurred in the test at a slightly higher load than in the prediction since the coupling web thickness was closer to nominal than the minimum thicknesses used in the analysis.

This advanced analysis was a good opportunity for RMS to further exercise its broad analytical capabilities. Ultimately it was determined that the electric motor must have been producing a higher peak torque than was previously understood. Therefore, RMS recommended a higher torque rating value for the next coupling purchased. The higher rated coupling was installed and survived a unit startup test without any damage at all. It was disassembled, inspected, reinstalled, and the compressor train has been operating successfully ever since.

TECH TALK

COUPLING BUCKLING ANALYSIS con't.

By Christopher Sykora

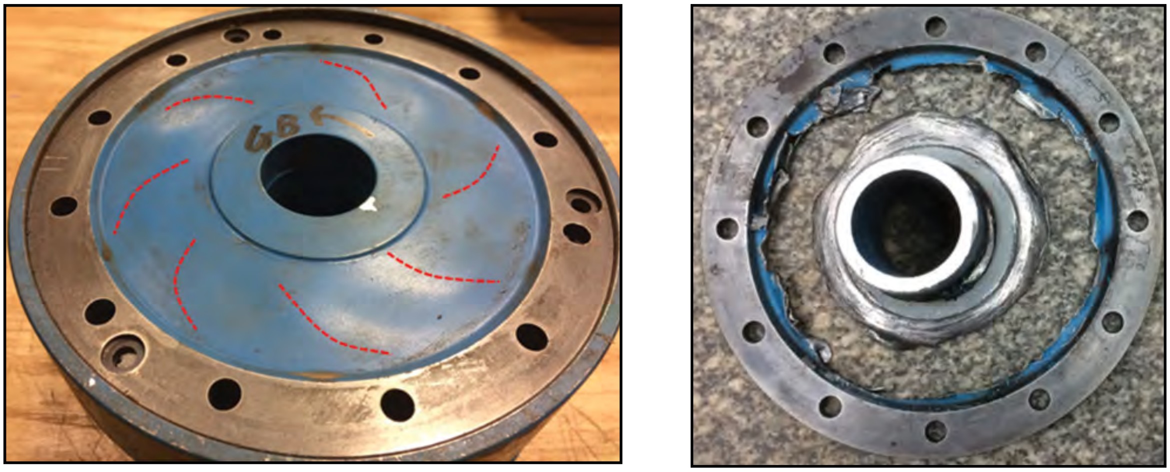


Figure 1 – Spiral Buckled Coupling (Left) & Completely Failed Other End (Right)

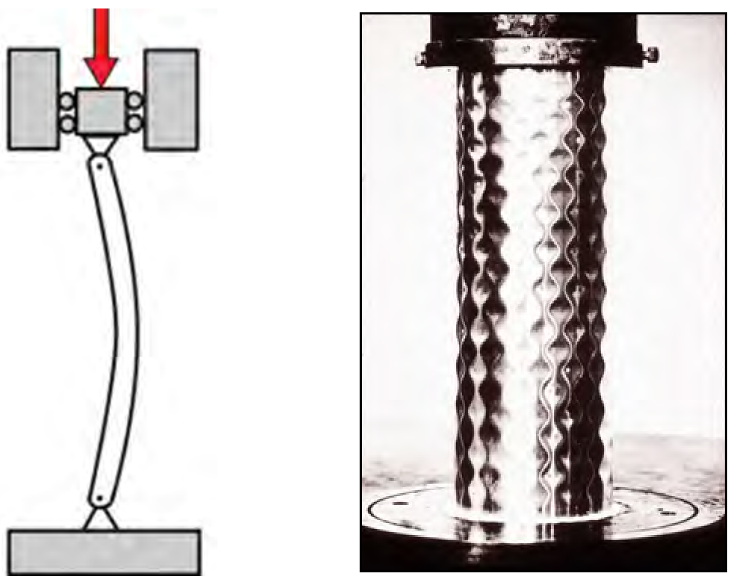


Figure 2 – Column Buckling (Left) & Shell Buckling (Right)

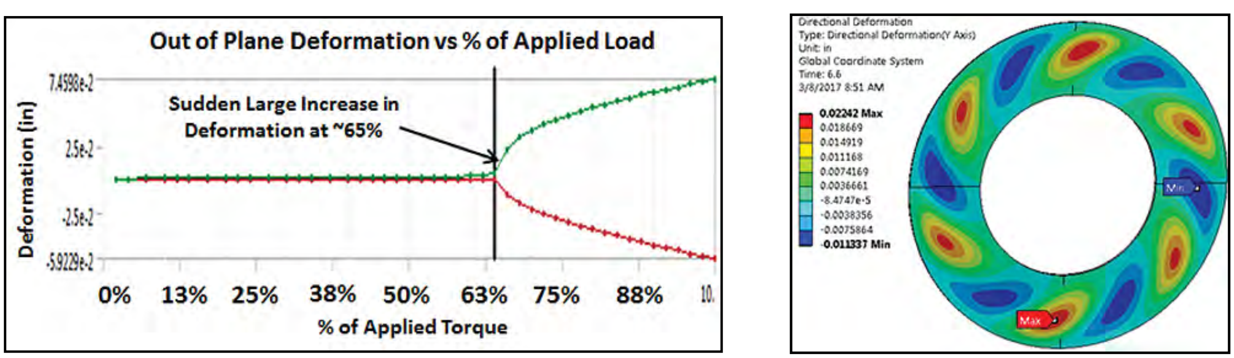


Figure 3 – Deformation Graph (Left) & Buckling Shape (Right)

TECH TALK

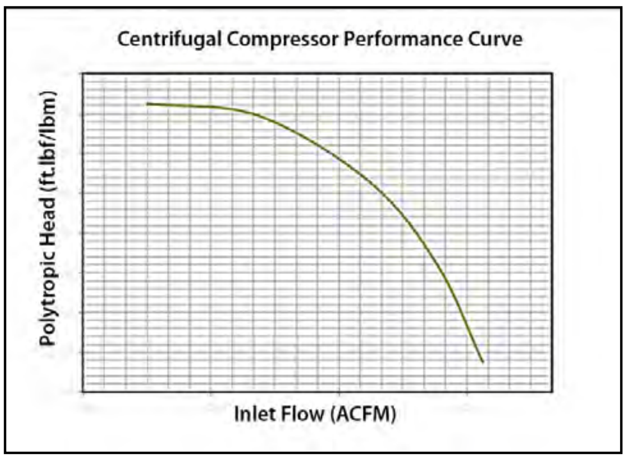
CENTRIFUGAL COMPRESSORS 101

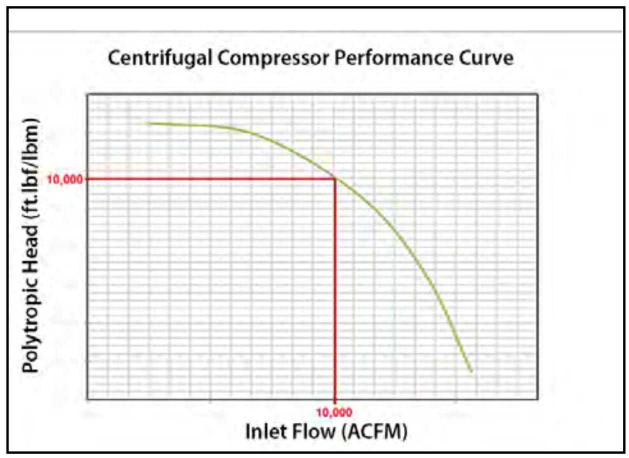
By Steven Kaulius

STAGE PERFORMANCE, EFFECT OF MOLECULAR WEIGHT AND STAGE STACKING

Centrifugal compressors are machines that ingest volume and produce head. As George Donald likes to say "all the machine knows is flow and head". A performance curve for a centrifugal compressor has an operating range of volume flow and produces head that varies inversely with flow. The amount of flow and head are dependent on the physical design of the impeller in the case of one stage. This curve characteristic is also true for a multistage unit whereas the entire flowpath determines the curve.

So since we know this about the performance, let's do an experiment where we put different gases through a single stage of a centrifugal compressor and see what happens. These machines can be designed for flows from about 800 ACFM to about 300,000 ACFM. We will look at a stage that is designed to handle 10,000 ACFM on the inlet and create 10,000 ft (ft-lbforce/lbmass) of head. We will run different gases through the stage and see what happens at the discharge.





Curve for theoretical stage

The analysis assumes 80% polytropic efficiency and 100 degrees F gas at 14.7 PSIA.

Results of compression are shown in the table below:

Gas	Molecular Weight	Discharge Pressure (PSIA)	Pressure Ratio	Discharge Temp (°F)	Mass Flow (LB/MIN)	Discharge Density (LB/FT³)	Discharge Flow (ACFM)
Hydrogen	2.016	15.04	1.023	104.67	49.3	.005	9860
Methane	16.04	17.62	1.199	129.61	393.2	.0448	8777
Ethylene	28.05	20.13	1.369	141.1*	690.1	.0881	7833
Air	28.96	20.17	1.372	166.84*	709.0	.0869	8159
Propane	44.09	24.25	1.650	138.94	1093.7	.1695	6452

TECH TALK

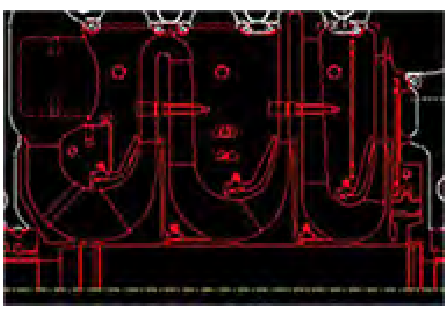
CENTRIFUGAL COMPRESSORS 101 con't.

By Steven Kaulius

STAGE PERFORMANCE, EFFECT OF MOLECULAR WEIGHT AND STAGE STACKING (con't.)

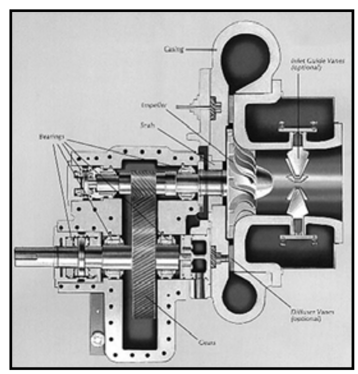
Some observations:

- You can see that as the molecular weight goes up, the discharge density goes up and the volume flow after the stage goes down. The implications of this for multistage machines are critical. Since the subsequent stage in the compressor needs to be operating near its design flow, getting this correct is imperative. This is the reason aerodynamic designers are always concerned about the gas analysis.
- The other interesting thing to note (and the reason for its inclusion here) is the difference in the discharge temperature and volume flow of Ethylene and Air. Even though these gases have very similar molecular weights, the discharge density/volume flow is different. This could change the designs of subsequent stages and the performance of the entire machine. The difference in discharge temperature is related to the ratio of specific heat (Cp/Cv) of the gas, Air = 1.402, Ethylene = 1.231
- Lower MW machines may have several stages of the same design since volume flow does not change that much from stage-to-stage.
- Changing the volume flow of a low MW machine usually requires more scope since the stages are all of similar flow designs.
- Changing the volume flow of higher MW machines may allow reusing some stages by simply moving existing stages and adding new higher/lower flow designs ahead or behind them. See typical multistage air compressor below.



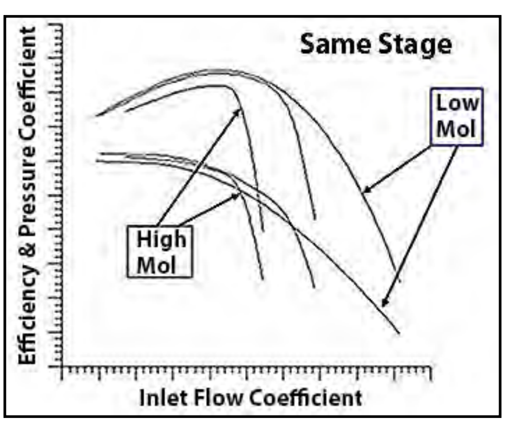
Typical multistage air compressor

Single stage of a multistage compressor



A single stage compressor

One more complication, the molecular weight of the gas affects the stage performance.



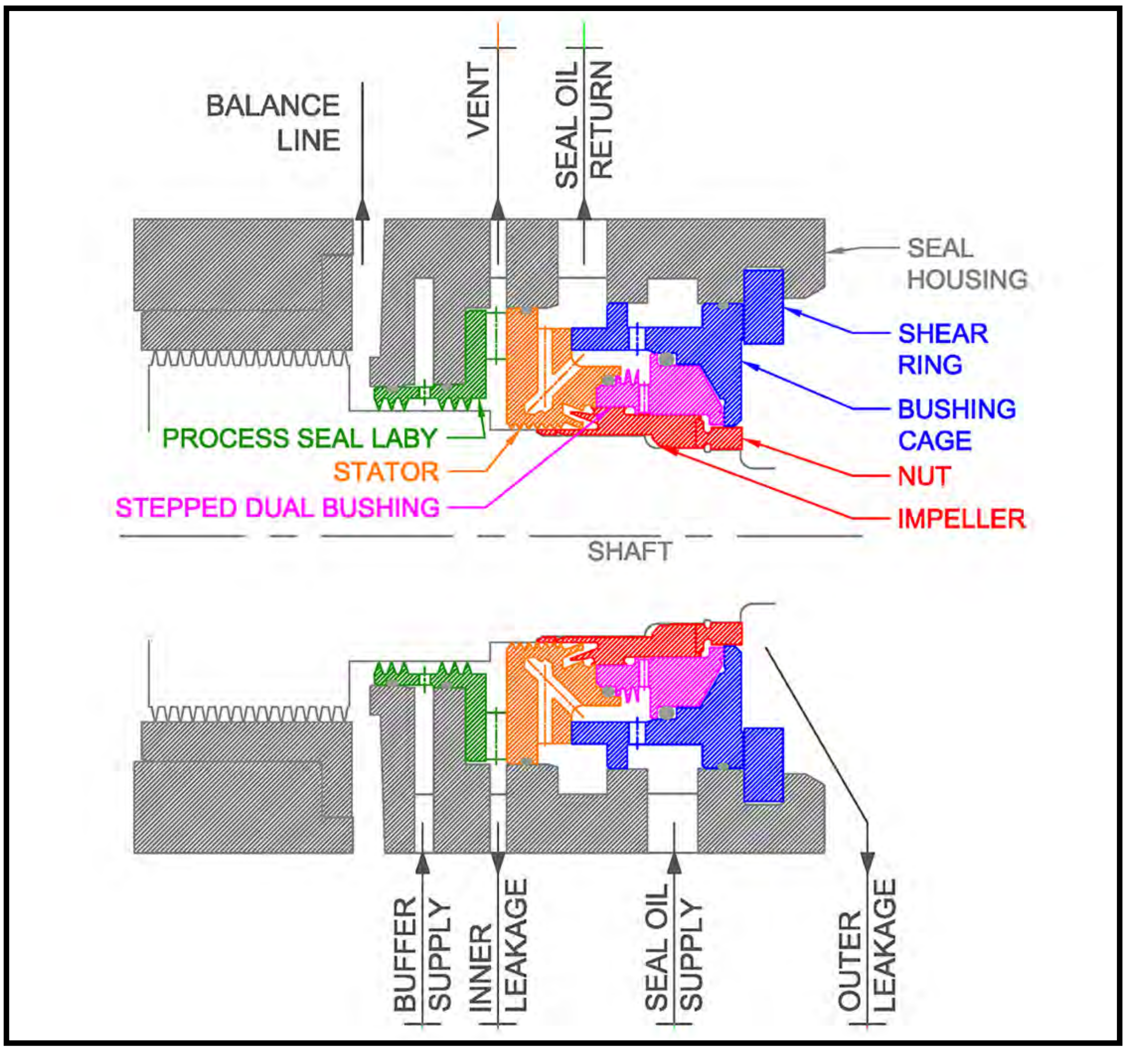
TECH TALK

AC COMPRESSOR TRAPPED BUSHING SEALS

By Rob King and John Decker

LEVEL CONTROL

The seals low leakage rate is predicated on the dynamic function of the seal's design features and clean oil supplied to the seal at a defined differential pressure relative to the gas pressure adjacent to the seal. This is accomplished by the use of an overhead tank with a level control scheme to maintain 5 feet oil level above the compressor centerline with gas reference on top of the oil from the discharge end, inner seal drain and vent annulus in multistage applications. This gas pressure is for all intents and purposes the same pressure as the inlet seal due to equalizing line function. (In single stage overhung units the sensing point is the inner leakage/ vent annulus.) This level control scheme essentially tracks process pressure as the controller and its' associated level control valve react to varying gas pressures while maintaining the level set point. The level control logic also provides outputs for level alarm, aux pump start and compressor trip, as well as the signal to the level control valve. When the trip level is reached, the output to the level control valve is vented and the level control valve goes to its fail close mode on loss of signal. The result is the remaining oil in the tank is shunted to the seals for coast down purposes. The level control scheme versus DP control offers improved tracking to process changes and will absorb most process upsets do to the capacitance offered by the volume of gas above the oil in the head tank without creating potential instability of 'hard' DP system.



TECH TALK

AC COMPRESSOR TRAPPED BUSHING SEALS con't.

By Rob King and John Decker

SEAL OIL SYSTEMS

Seal oil systems may be combined with the compressor lube oil system or separated if needed due to process concerns of contaminating lube oil or higher process gas sealing requirements. In the combined system, the reservoir is shared while separate pumps, filters, and coolers may be required for the higher pressure seal oil requirement than the lube oil pump circuit can accommodate. In separate systems having unique seal system and a unique lube oil system, a barrier seal/ baffle is used between the bearing housing and seal cavity to keep the oil from the individual systems from migrating between systems. "Traps" (sometimes called drain pots) are an important part of the seal oil system which captures the inboard sour oil leakage. They are gas tight with regard to the drain function and are typically snap acting to open/close upon reaching level set point, to dump the oil either back to the reservoir, via a degassing tank or to an alternate location depending on the concern for contaminates to the seal oil integrity. In most installations there is an oil gas mixture that goes to the trap. This mixture enters the trap on the side of the vessel with the gas exiting the top of the vessel through a demisting element and a monitored /controlled orifice assembly and finally back to the compressor inlet, flare line or alternate location. The flow of the gas from the trap back to the inlet or flare can also be used in a scheme to control the buffer gas being supplied to the process seal labyrinth inboard of the trapped bushing seal. The process labyrinth seal and trapped bushing seal are separated by the drain and vent annulus of the seal housing/cavity. By measuring the seal flow to the process labyrinth and the flow downstream of the trap, the gas flow from the trap can be controlled to be 50% of the total flow supplied to the process labyrinth. The 50% of flow not going toward the trap is then going toward the process. Calculations can be made to size the orifice and control elements appropriately to provide sufficient gas velocities to process labyrinth to keep process debris from back diffusing toward the seal, using this flow control logic. The use of the buffer system keeps the trapped bushing seal clean and also keeps the gas – oil interface in the overhead tank clean from contaminates. The buffer gas is also not subject to the same criteria of seal gas for a dry gas seal, and needs to be considered in the overall process gas stream make up.

As with most all lube oil and seal oil systems, cleanliness is of critical importance to reliability. Careful attention to cleanliness details must be observed during all seal installations and systems maintenance to avoid contamination and their inherent problems. It is recommended to functionally test the seal system including the alarm and trip set points, during the course of major outages to ensure the system functions as originally intended. This exercise also provides operators and equipment engineers the opportunity to refresh their knowledge of the systems.

The trapped bushing seal can still offer the reliability needed and the RMS engineering team has the resources to support these efforts.

Here is a powerful yet simple rule. Always give people more than they expect to get – Nelson Boswell

TECH HISTORY

THE PROGRESSION OF ROTOR DYNAMICS

By George H. Donald

In 1965, I started at Ingersoll-Rand Co. in Phillipsburg, NJ as a Design Engineer and have been in the business ever since. I was a member of the API Subcommittee for Mechanical Equipment for over 20 years, beginning in 1968.

One must start this journey by knowing that in the early 1960's, engineers were using slide rules to do calculations. Electronic calculators were a miracle invention and a blessing. No more wondering where the decimal point was. (Slide rule people can appreciate this).

When I started my journey in the mid 1960's most all machines, turbines and compressors, were built with sleeve bearings. At that time, the vibration on the test stand was measured using a hand held IRD accelerometer with a wooden stick attached to it which was pressed onto the shaft for measurement.

With the invention of mainframe computers, the analysis consisted of making a geometric model of the rotor and then making a plot of critical speed vs. support flexibility. This is the undamped critical speed map still in use today. The location and acceptability of the actual critical speed was estimated from this plot based on a support stiffness estimate from similar machines that were built and tested.

Then in the 1970's, work was done to determine the characteristics of bearings. This then led to the development of computer programs that used these stiffness and damping values to predict rotor response to unbalance. A result of this effort was the general application of tilt pad bearings to machines. Also, API 617 4th Edition recognized this analysis as a tool for modeling units.

Now we had a way to model machines and Donald E. Bently came up with his invention to measure shaft vibration. This opened a whole new "can of worms", so to speak, for the industry. What did all this mean? Now, what is the acceptance criteria for vibration? The analysis tools were soon to follow. Then we had this non-synchronous stuff to address. This was a whole new world.

Another item to note in the 1970's was the use of centrifugal compressors in higher pressure applications. The most significant of these was the Elliott® compressors for Ekofisk, designed for 10,000 psi discharge pressure. This was new territory for the industry and a great deal was learned from these units. This was the start of understanding the destabilizing effects of labyrinth seals and impellers, etc. It was the beginning of things such as shunt holes and swirl brakes.

For the next 20 years as units were built and tested for higher pressures, we learned and became more knowledgeable as to how to make units more stable. The analytical tools continued to improve and things like the space shuttle engines and the work done at the Texas A&M Turbo Laboratories helped. It wasn't until 2002 that API 617 7th Edition recognized that the tools existed to predict stability in these compressors and included a section covering that.

The 1980's were impacted by the personal computer becoming available for use by the general public. It made tools available to individuals and small companies that previously required mainframe computers to run. The use of dry couplings, gas seals, and magnetic bearings became more prevalent and accepted by the industry during this time as well.

After all this history of events and the development of the analytical tools, API finally defined what a critical speed is in the 617 5th Edition; (a peak response with an amplification factor greater than or equal to 2.5). Stability analysis still seems to be in the area of development. I am somewhat humbled to see that the concept of looking at

TECH HISTORY

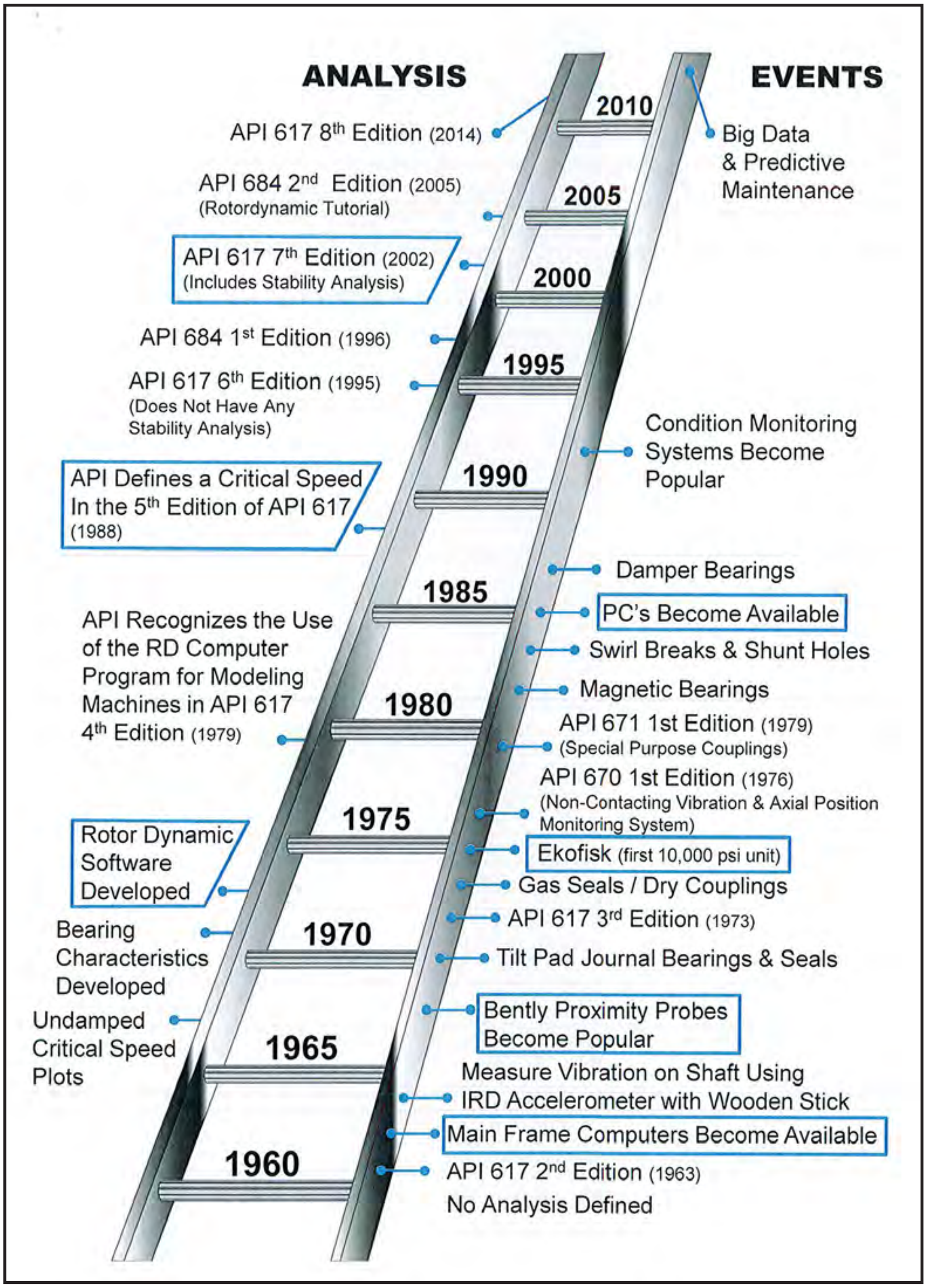
THE PROGRESSION OF ROTOR DYNAMICS con't.

By George H. Donald

compressors on a rough basis, as presented in a paper that Dr. Kirk and I wrote in 1983, is still used today.

Today, Rotor Dynamics is a very exact science. The combination of computer power and our many years of experience allow us at RMS to optimize your rotor and bearing systems for maximum performance and reliability.

And finally, I have highlighted the seven items on the ladder that I believe made step changes to the centrifugal compressor business in the past 50 years.



SPOTLIGHT ON:

MEET OUR NEW STAFF MEMBERS

John Bartos - Chief Operating Officer

John began his career in 1986 as a Design Engineer for both the Turbomachinery & Engineered Pump Divisions of Ingersoll-Rand; Phillipsburg, NJ. In 1988 John left I-R to join CONMEC, Inc. as a Senior Design Engineer/Centrifugal Compressors, then was promoted to Supervisor/Centrifugal Compressor Design, then to Product Manager/Centrifugal Compressors.

In 1994 John left CONMEC to join A-C Compressor in Appleton, WI as Director of Engineering. In 1996 John was promoted to VP/Centrifugal Compressor Business. He was then recruited back by Ingersoll-Rand, Mayfield, KY to become the Manager, Advanced Engineering for Centrifugal Compression.

In September of 2000 John was recruited to the Cooper Turbocompressor Division of Cameron as Division VP/Engineering. From 2001 through March of 2017 John has held a number of senior management positions with Cameron, and subsequently, Schlumberger. Among those John was Corporate VP/Enabling Technology, leading both the pre-merger and post-merger (Schlumberger acquired part of Cameron) integration planning efforts with both Cameron & Schlumberger engineering teams. Most recently, John was Corporate VP/3MT (Materials, Modeling and Mechanical Technologies), which was a "bridged" organization combining the talents of Schlumberger and Cameron engineering personnel following the acquisition.

George Donald - Technical Director for Centrifugal Compressor Development

After graduating from Drexel University with a BSME degree, George went to work for Ingersoll-Rand in 1965 and has spent his entire career working on Turbomachinery. In 1987 after years with I-R, George left when the company merged with Dresser Industries and became Dresser-Rand. This merger led to the formation of CONMEC, Inc. later in 1987, and George was one of the founding partners.

After the acquisition of CONMEC by GE, George negotiated a return to Dresser-Rand in 2006 and led a core Centrifugal Compressor engineering team back to D-R and set up their Bethlehem engineering office. He remained with this D-R office until just prior to joining RMS.

George was a member of the API subcommittee on mechanical equipment for nearly 20 years, and he contributed to the generation of many of the API specifications still in use today. He was also an active participant for years in the Texas A&M Turbomachinery Symposium held annually in Houston, where he participated as an author, short course speaker, and panel member on numerous occasions.

Steve Kaulius - Director of Centrifugal Compressor New Business Development

With a BSME degree from New Jersey Institute of Technology and a MBA from Lehigh University, Steve has spent 37 years working with engineered equipment…. 27 of those in turbomachinery. He began his career as a Compressor Design Engineer for the Ingersoll-Rand Turbo Division. After 4 years at I-R, Steve left to accept a position as Applications Engineer with Pennfield Industries, followed by 7 years with CECO Filters as Sales Manager.

SPOTLIGHT ON:

MEET OUR NEW STAFF MEMBERS (con't.)

In 1994 Steve returned to the turbomachinery industry accepting positions with CONMEC, Inc., first as Senior Product Specialist/Centrifugal Compressors, then under Dover Corporation's ownership he was promoted to Manager of Sales Support (Applications Engineering). After the GE acquisition of the CONMEC business from Dover, Steve became a Project Development Manager working with the GE sales team and clients globally on turbomachinery revamp projects. In 2006 Steve left GE to accept a number of Project Development and Market Development positions with Dresser-Rand in the Bethlehem office.

With extensive experience and knowledge with both Centrifugal & Axial Compressors, Steam & Gas Turbines, Expanders, Motors & Gears, and Auxilliary & Control Systems, we here at RMS are very pleased to have Steve on our team!

Jeremy Simpson - Senior Project Manager

Jeremy comes to us with over 18 years of experience in the aeroderivative industry having worked for Continental Express Maintenance, ExpressJet System Operation and most recently GE Oil & Gas as a Lead Commercial Manager- Global Field Service, Repairs and Parts. His diverse background will be invaluable in meeting the growing demands of our new RMS Houston facility.

Jeremy graduated from LeTourneau University with a Master of Business Administration.

CAREER OPPORTUNITIES

We are growing and always looking for talented individuals for the following positions in our Houston, Texas and Bethlehem PA offices.

- Project Management
- Engineering
- CAD Designers
- Applications Engineers
- Scheduler / Master Scheduler
- Assemblers
- Machinists
- Mechanics

Please send your resumes to HR@rotatingmachinery.com

SPOTLIGHT ON:

STAFF PROMOTIONS

Chot Smith - Director of Business Development - Operations & Field Service

In this new role, Chot will have a greater focus on leading and growing RMS' vital Field Service business. Chot will continue to lead RMS' manufacturing efforts through the strategic selection of machine tools and major equipment in our Bethlehem and Houston Service Centers. Chot has been a key contributor to the growth of RMS in setting up a first-class machine shop in Bethlehem as well as establishing RMS as a well-respected source of field service.

Tom Edwards - Manager of Manufacturing, Quality, Health, Safety & Environmental

Tom will be responsible for management of the day-to-day operations of the Bethlehem shop and QHSE activities across the RMS enterprise. Since joining RMS, Tom has established the quality processes and standards for the business. He has built a trusted reputation with RMS' customers that will continue and expand with his new responsibilities.

Ryan Montero - Aerodynamicist II

Ryan has quickly grown into an invaluable member of our engineering team, taking on the challenges of performance prediction and rerate design for our assortment of product lines. His ability to master a variety of performance software, including CFD analysis, along with bringing us the capabilities to generate custom computer code when required, has been truly helpful.

Kathy Ehasz - Director of Human Resources

Kathy joined Rotating Machinery Services, Inc. in 2004 and has served in the roles of Senior Office Administrator, Manager of Engineering Support/Marketing and Manager of Human Resources. Kathy has always shown initiative in the performance of her duties, even going above and beyond what is expected of her. She has played an important part in the growth of RMS, Inc. Kathy will be responsible for the smooth and professional operation of the company's human resources department and will provide consultation to management on strategic staffing plans, compensation, benefits, training and development, budget and labor relations at all the RMS Facilities.

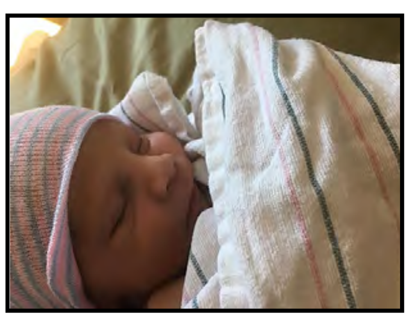
Gabrielle Koltisko - HR Assistant

Gabby has been with RMS since October of 2014. In this timeframe, Gabby has performed exceptionally as the Receptionist and the Engineering Administrative Assistant. She has been very valuable to her coworkers and continues to grow and take on more responsibilities. She is excited to take this next step in her career.

RMS Family News

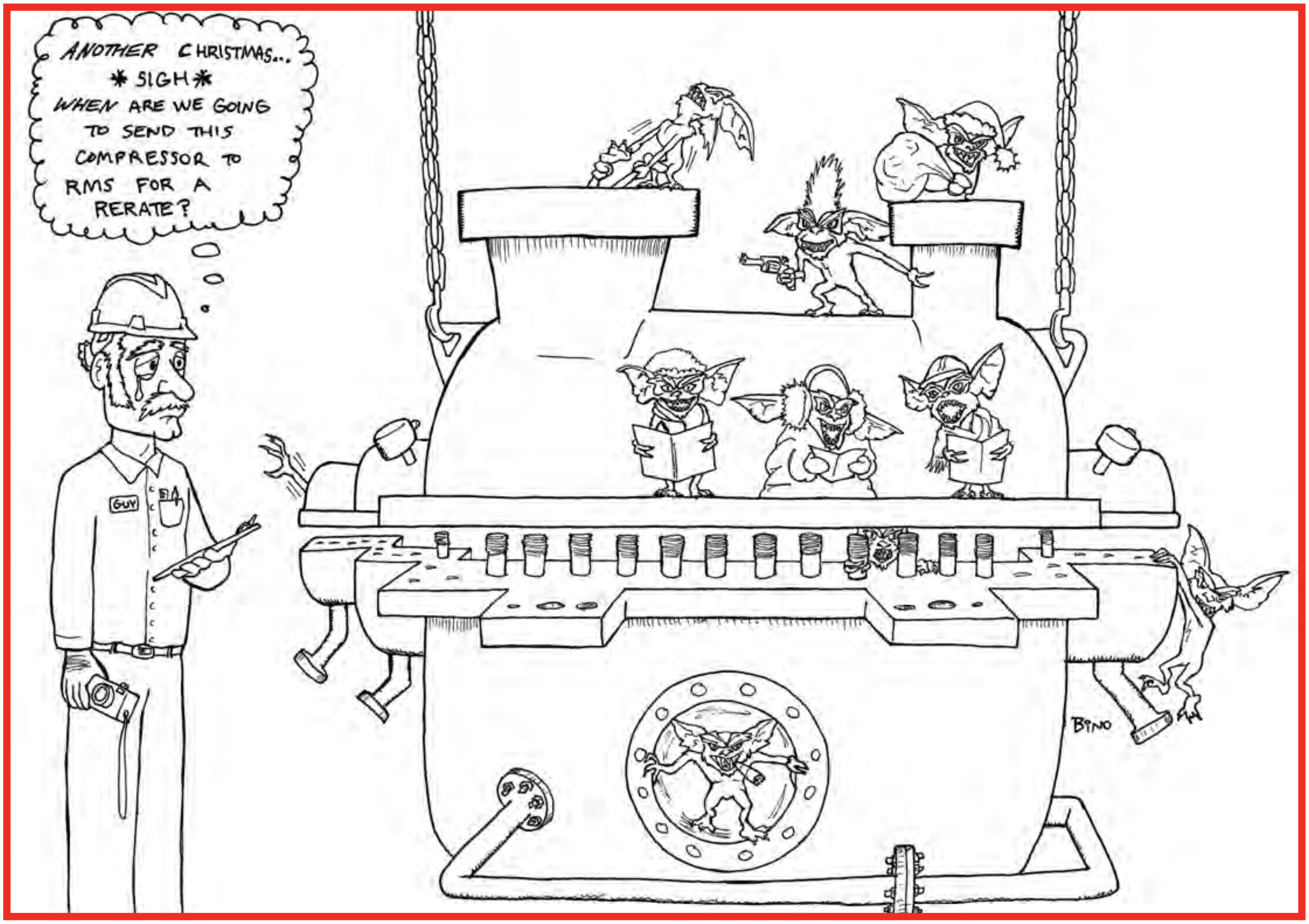
Kristen and Chris Sykora were blessed with the birth of this handsome lad. Benjamin Christopher Sykora was born on March 22, 2017 a healthy 7lbs 10oz baby boy!

We're sorry about reporting so late, but maybe Chris will share some 1 year old birthday pics. A belated congratulations to Kristen and Chris!



TURBO TOONS

By Marc Rubino



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From all of us at RMS and we hope you have a

Prosperous New Year!

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