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.

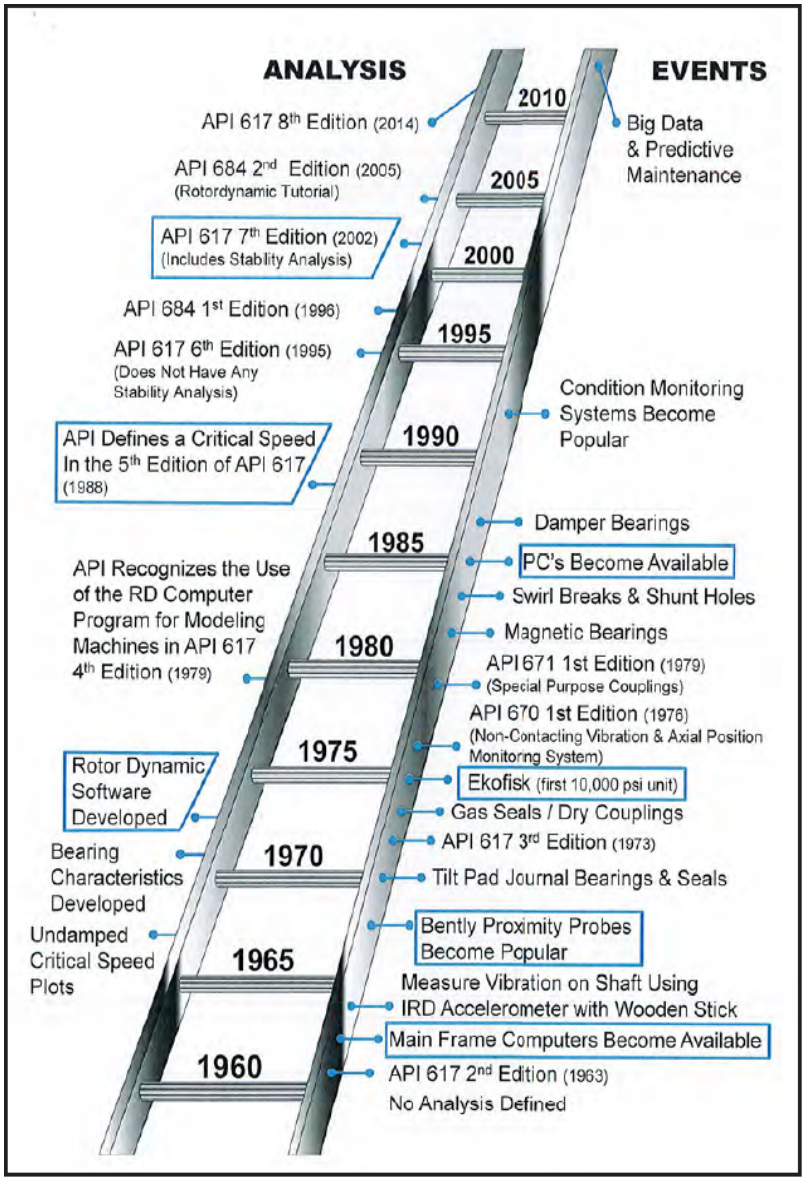
The 1980's were impacted by the personal computer becoming available for use by the general public. It made tools available to individuals and small companies that previously required mainframe computers to run. The use of dry couplings, gas seals, and magnetic bearings became more prevalent and accepted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



For more information:
George H. Donald
Email: gdonald@rotatingmachinery.com
Tel: 484-821-0702

Headquarters
2760 Baglyos Cir.
Bethlehem, PA 18020

Houston Office
16676 Northchase Dr., Ste 400
Houston, TX 77060

rotatingmachinery.com
Tel: 484-821-0702
Parts: rms@rotatingmachinery.com

RMS
Rotating Machinery Services, Inc.