42nd Turbomachinery
29th Pump SYMPOSIA

Proceedings of the Forty-Second Turbomachinery Symposium
October 1-3, 2013, Houston, Texas

REVAMP / RE-RATE DESIGN CONSIDERATIONS

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ABSTRACT

This tutorial paper describes the mechanical and aerodynamic factors that must be considered when revamping, re-rating or upgrading a centrifugal compressor. The possible motivations for choosing to revamp existing turbomachinery rather than purchasing new equipment are also offered.

INTRODUCTION

As compressor users look to maximize the production of their facilities or to reduce their operating costs, they often face choices whether to purchase new equipment or to upgrade their existing turbomachinery. Given the cost to build an entire new compressor train and its related support systems, it can, at times, be far more cost effective to refurbish existing equipment. However, there are limitations to what can be achieved via the so-called "revamp" or "re-rate." The purpose of this tutorial is to give readers an overview of the various considerations that must be explored before a decision can be made whether to revamp/re-rate existing equipment or to purchase new compressors.

This tutorial is broken into five basic sections. The first section discusses some of the business aspects of the decision between new or revamped equipment. The second section covers the various types of upgrades that can be undertaken. The discussion then moves to three technical sections that focus on various mechanical, aerodynamic and rotordynamic design considerations. Finally, comments are offered regarding the importance of good communication between the end user and the company performing the revamp.

Disclaimer: Compressor original equipment manufacturers (OEMs) and third-party firms that specialize in revamping and/or re-rating compression equipment each have guidelines and/or procedures that they follow when upgrading equipment so the guidelines and suggestions contained in this document might not be applicable in all situations or with all vendors. However, the information contained herein should give the reader a good overview on the subject of compressor revamps or re-rates.

BACKGROUND

Revamping, re-rating or upgrading centrifugal compressors have been common practices in the refining, chemical, petrochemical, gas pipeline, and oil & gas production industries. Given the current economic climate, there is increasing interest in revamping or upgrading existing equipment simply to reduce expenses. Further, the required payback period (the number of years needed for a project to turn profitable) has become increasingly short as competitive pressures increase and concerns arise over political uncertainties at some plant locations. Thus, the need to maximize the capability of existing rotating equipment is important.

Unfortunately, there are limits to what can be accomplished within a given compressor. That is, all compressors are not created equal. Numerous factors will determine the level of performance modifications that can be made. For example, if the goal of a revamp is to increase the operating capacity, one must determine whether the existing flange size on the case can accommodate the higher flow rates without introducing unacceptably high losses. Further, if the inlet flanges can pass the flow, will it be possible to install an impeller that is large enough to accept the incoming flow or will the compressor rotordynamics be acceptable with the new impeller line-up? Moreover, will the Mach numbers, flow range, efficiency, impeller stress levels, etc. be acceptable to the end user? This tutorial will provide attendees with an overview of the factors that compressor designers and analysts review when assessing the "revamp-ability" of an existing compressor. Some of these considerations will seem intuitively obvious, while others might prove thought-provoking.

TERMINOLOGY

The terminology used by OEMs varies somewhat when it comes to revamping, re-rating or upgrading compression equipment. For clarity, in this tutorial, these terms will be defined as follows:

Revamp / Re-rate –These terms will be used interchangeably. That is, there is no functional difference between revamping and re-rating compressor. The process involves changing the performance characteristics (aerodynamic and/or mechanical) of a compressor. The changes might involve increasing or decreasing the flow rate, increasing or decreasing the operating speed, increasing or decreasing the inlet and/or discharge pressure, etc. These will be accomplished by changing at least some, but possibly not all, of the compressor internal components. At the end of the revamp or re-rate, new performance curves are typically supplied and new operating requirements are specified. It might be necessary to have a new operator / service manual.

Upgrade – The term "upgrade" will refer to those instances when the operating requirements of the compressor do not change but changes are made to the components within the machine to improve their mechanical integrity, reliability or longevity. Examples could include replacing old riveted impellers with welded wheels, changing from aluminum labyrinths to polymer seals, changing bearings, or seals or the like. The primary objective of such upgrades is to improve a machine while not necessarily changing its performance characteristics. NOTE: Upgrades are not to be confused with normal preventative maintenance programs during which labyrinth seals, o-rings, bearing pads, etc. are replaced in kind with the same components as originally installed in the compressor.

The choice between a revamp/re-rate or upgrade depends entirely on the end user's needs. For example, the end user might have no interest in changing the performance characteristics of the compressor but might want to extend the operating life of the machine by switching to welded impellers. Similarly, changing from sleeve bearings to a tilt-pad bearing or from a tilt-pad bearing to more advanced damper bearings could improve the rotordynamic characteristics of the machine, with no impact on the aerodynamic performance.

This tutorial will not spend much time on upgrades but instead focuses on the more extensive changes typically associated with revamps/re-rates.

MOTIVATIONS

The primary motivation for conducting re-rates or revamps is to re-purpose the compressor or to modify the compressor's performance characteristics to better match new operating requirements. This might involve increased or decreased flow requirements, increased or decreased head or discharge pressure requirements, or some combination of both while simultaneously decreasing (or at least minimizing the increase in) the horsepower required to drive the compressor or compressor train. Simply put, the existing compressor or compressor train is incapable of meeting the new operating requirements or incapable of meeting the new operating requirements *efficiently*. For whatever reason, the end user has decided against purchasing new equipment and has, instead, decided to revamp the equipment to meet the new operating requirements.

Minimizing cost is another key motivator in revamping or upgrading existing equipment. In many situations, it is possible to re-use the majority of the compressor components including the case, heads, rotor, bearings, seals, and many of the internals. This can result in significant cost savings as the casing is quite often one of the more expensive items. If the casing is re-used, then the foundation, baseplate and existing piping arrangement can also be left in place, again resulting in substantial savings.

In many situations, it is also possible to re-use many of the compressor internal components, such as the impellers, return channels, inlet, discharge volute, and inlet guides. Care must be taken to ensure the components are still structurally sound and that the flow passages are aerodynamically acceptable (i.e., not fouled, eroded or otherwise damaged). The OEM or third-party must thoroughly inspect these components to: (a) ensure they are fit for continued service and (b) acquire the necessary geometry to conduct the necessary analyses (aerodynamic and mechanical) of the parts.

It might also be possible to decrease the long-term operating cost of the compressor or compressor train by installing state-of-the-art aerodynamic components in the older machine. For example, older impellers with simple, less than optimal blading could be replaced by more modern impellers with sophisticated 3-D, inducer-style blades, resulting in higher efficiency and lower operating costs. Likewise, more modern, machined stationary components with their inherently smooth surface finishes and precise dimensions could replace old, rough cast components; again increasing performance and lowering energy costs.

In some cases, replacing a component can increase the overall efficiency of the compressor. In this instance, the evaluating engineer must make an economic analysis based on either dollar per horsepower (HP) of electricity or dollar per pound of steam.

The simplest formula for calculating the value of energy cost savings is payback period:

PAYBACK PERIOD (years) = Project Cost ($)/(HP savings x HP cost)

HP cost for steam varies from plant-to-plant and is stated as dollars per HP-year ($/HP-Yr). A typical cost for steam horsepower is $1,000/HP-Yr. HP Cost for electricity is measured in dollars per kilowatt hour ($/Kw-Hr).

For example, a high-efficiency re-rate option saves 100 HP at an evaluation of $1,000/HP-Yr of steam electricity. The project cost is $200,000 and the payback period is:

PAYBACK PERIOD = $200,000/(100 x 1000) = 2 years

For conversion of electricity cost, multiply horsepower by 0.7457 to convert to kilowatts. Multiply this by the number of operating hours per year, which yields kilowatt-hours.
For an electric cost of 10 cents per kilowatt-hour, the same project payback period is:

PAYBACK PERIOD = $200,000/(100 x 0.7457 x 8760 x 0.10) = 3 years

This formula does not take into account the cost of money. A more accurate method for evaluations would be either internal rate of return or a net present value calculation. Another advantage of the revamp or re-rate when compared to purchasing new equipment is that one can typically minimize the cycle time for the turn-around or, stating this another way, one can minimize the production down time. Clearly, if there is no need to replace piping, build a new foundation or lift a new casing in place, time (and money) are saved.

Some end users also see revamps or re-rates as a way of minimizing risk. They might have good operating experience with the existing compressor and want to maintain the current or similar mechanical components in hopes of avoiding any potential mechanical and rotordynamic issues with a new machine. Similarly, if the aerodynamic components applied in the revamp are from the same OEM product line, the revamp components are perceived as "proven" or "grandfathered" technology. Again, the perceived risk is minimized.

Finally, as suggested above, revamping a compressor does allow the end user to apply new technology to an old machine. Assuming the new components can be made to fit, the end user can have the "latest and greatest" impellers, diffusers, return channels, bearings, seals, materials, etc. and the improved aerodynamic and mechanical characteristics that come with each.

LIMITATIONS OF A COMPRESSOR REVAMP / RE-RATE

When an OEM is consulted regarding the rerate of an existing compressor, there are certain physical and mechanical constraints that can be checked quickly to determine whether or not a revamp is possible. These constraints limit the maximum performance capability of the machine. The physical constraints include the casing internal dimensions (i.e., length and diameter) and the nozzle sizes. The mechanical constraints include casing pressure and temperature ratings as well as any operating speed limits. If any of these constraints are exceeded by the re-rate requirements, the manufacturer will recommend replacing the compressor with a more suitable model.

Horsepower Limits

In many cases, the end user plans to use the available driver. This is acceptable as long as the driver can produce sufficient power to drive the revamped compressor. An assessment must be made on the driver capabilities relative to the new compressor operating conditions. For example, if the compressor speed is dropping considerably, it is possible the driver will not be able to provide the necessary horsepower at the reduced speed. Further, the rotordynamic characteristics of the driver must be assessed at the new required operating speed.

Speed Limits

There are a number of criteria that will limit the design speed for a given model of compressor. These include rotational stresses, Mach number, critical speeds, and (potentially) the driver.

Rotational Stress Limits

Impellers for the API industry are designed with tip speeds of up to 1,000 fps. As a general rule, riveted impeller construction is limited in tip speed to 800 fps; welded impeller construction allows tip speeds up to 1,000 fps.

It is good design practice to limit rotational impeller stresses to 70% of the impeller material yield strength. This is done so that at over-speed test conditions, no material yielding will occur.

Rotational stresses are especially important in sour gas service in which the impeller material yield strength is generally limited to 90 ksi with a reduced hardness requirement of Rockwell C22.

Since the compressor head is proportional to the impeller tip speed, the head requirement for the rerate is limited to using the maximum impeller diameter allowable for the casing and an impeller designed for a tip speed of approximately 1,000 fps.

CASE CONSIDERATIONS

With few exceptions, the casing is the most costly component of a compressor. Therefore, virtually every compressor revamp or re-rate re-uses the casing. A detailed review of the casing geometry and capabilities must be done to ensure that re-using the case is possible.

Casing Pressure and Temperature Ratings

Of primary concern when rerating a compressor, you must recognize that the compressor casing was manufactured with a specific design pressure in mind. It was likely hydro-tested at 1.5 times the design pressure. Normally, the so-called case pressure rating and the hydro-test pressure can be found on the compressor nameplate that is mounted to the casing. The case rating must be compared against the new discharge pressure requirements to ensure that, at the worst set of operating conditions and highest operating speed, the discharge pressure does not exceed the case pressure rating. Now the existing casing design pressure can be exceeded if a detailed stress analysis is done on the casing. If the analysis shows unacceptable safety margins on the casing, another hydro-test is required to certify the case integrity and safety at the new conditions.

Note that the maximum allowable discharge temperature must be reviewed as temperature affects the casing material, the shaft seals and any o-rings used to seal components.

Casing Internal Dimensions

Once it has been established that the case can withstand the new pressure and temperature requirements, the next step is to confirm that the case can accommodate the new aerodynamic and rotordynamic components. Given that compressor cases are typically welded fabrications or castings, the internal dimensions of a given compressor casing are, to a large extent, fixed. These dimensions can limit both the number of stages and the diameter of the impeller that will fit within the casing. A centrifugal compressor stage (consisting of an inlet guide, an impeller, a diffuser, a return bend or crossover and a return channel) requires certain “stage spacing.” The stage spacing is an axial dimension defined as the distance from one impeller disc to the next impeller disk (Figure 1). The stage spacing is heavily dependent on the flow coefficient of the impeller in a stage with higher flow coefficient stages occupying much more axial space than lower flow coefficient stages. Therefore, the number of stages that will fit into an existing casing depends on the flow coefficient of those stages. For a given case length, you can fit more low-flow coefficient stages than high-flow coefficient stages. Further, if you are re-rating a machine to go from lower to higher flow and you require the same amount of pressure rise (which implies you require the same number of stages), it might not be possible to fit the new rotor into the existing casing.

There is also a minimum radial distance or radius ratio required between the impeller exit and the case for the diffuser. If the diffuser is too short, it will not provide the necessary static pressure recovery to ensure an effective stage. A portion of the velocity or dynamic pressure exiting the impeller is converted to static pressure by the diffuser. If insufficient static pressure recovery occurs, the efficiency or pressure ratio of the stage will suffer.

The ratio of diameters from the top of the discharge diffuser to the impeller is called “diffuser ratio”. This ratio is normally in the range of 1.4 to 1.8 for new compressors and those limits are typically applied in revamps / re-rates as well. Thus, you would not design a 40-inch impeller for a casing with an inside diameter of only 48 inches because the resulting radius ratio would be less than a 1.2. You could use such a radius ratio if proper efficiency and head coefficient derations are applied. In short, it might be possible to get the flow through the machine despite the low radius ratios but there is a risk that the performance of the compressor might be unacceptably low.

Aerodynamic Flow Path Integral with Casing

Another constraint might arise when considering re-using a casing. In some situations, portions of the aerodynamic flow path are built integral to the casing. For example, in many horizontally (or axially) split casings, the return bend (or crossover) are cast or machined into the casing (Figure 2). Therefore, when revamping the machine, the designer must either-use the existing return bends or find another way to introduce new return bends inserts (Figure 3). The latter option is only possible if acceptable diffuser radius ratios can be accomplished with the new return bend inserts. If the return bend inserts are not possible, then the integral return bends set the stage spacing for any revamp/re-rate internals since the diffusers must mate up with the existing return bend locations. Therefore, the ability to accommodate the new internals can be limited. In fact, if the goal of the revamp is to achieve a sizeable increase in capacity, it might not be possible to fit the impellers in the spacing available between two return bends, unless it is possible to skip a return bend (Figure 3). NOTE: The return bend locations are typically not an issue with barrel-type compressors because the return bends are either machined into a separate ring that can be replaced or are integral with the return channels or other compressor internals.

Some compressor casings also include portions of the compressor main inlet and/or discharge volute or collector. Designers must determine if the existing geometry can accommodate or be modified to accommodate the new internals. Of course, any modifications could result in the need to re-hydro or re-certify the casing.

The inlet and discharge nozzles are also built integral with the casing. These, too, can limit the amount of flow the casing can handle without incurring large pressure drops due to high pipe velocity. High pipe velocity is also one of the causes of compressor noise.

As a general rule for new centrifugal compressors, the gas velocity in the inlet nozzle is limited to a Mach number (gas velocity divided by inlet sonic velocity) of 0.15 or lower. As examples, an air compressor with a sonic velocity of 1,138 feet per second would be limited to an inlet nozzle velocity of 170 feet per second. A propylene compressor with a sonic velocity of 760 fps would be limited to an inlet nozzle velocity of only 114 feet per second.

In general, minimum inlet losses occur with lower inlet velocities so the pressure loss is minimized and the flow distribution to the first stage impeller is more uniform at lower velocities. However, it is also possible for the flow velocities to be too low which leads to the formation of "dead zones" or "stalled zones" in the inlet. This, in turn, causes higher losses or other aero-mechanical problems for the first stage impeller.

It is possible to violate the 0.15 Mach number rules suggested above. For example, the flow rate for a revamp application might cause the flange Mach number to be higher than 0.15. This does not necessarily mean the revamp / re-rate cannot happen. The higher Mach number will result in additional losses in the inlet section. If these higher inlet losses are accounted for in the performance prediction, the end user might decide that the resulting compressor performance is acceptable. Each OEM has different guidelines regarding maximum flange Mach number. However, as a general rule of thumb, flange Mach numbers above 0.3 are not recommended.

Similar rules can be applied to compressor discharge nozzles. In general, one would expect the Mach number at the exit flange to be equal or less than the inlet flange Mach number. Some suggest a nominal exit flange Mach number of 0.12 to 0.15. As with the inlet, should the Mach number be too low, the flow in the discharge flange could become overly turbulent, causing pressure fluctuations and possible aero-mechanical issues in the discharge pipe. Again, these general rules can be broken for revamps as long as the impact on performance is captured in any performance predictions.

THE RE-USE OF EXISTING INTERNAL COMPONENTS

To maintain the economic attractiveness of a proposed rerate, the re-use of existing compressor components should be maximized. However, there are two important considerations that must be assessed:

(1) Will reusing the component affect the unit's reliability?
(2) Does re-using the component compromise the achievable performance (mechanically or aerodynamically)?

A mechanical reliability evaluation of the existing equipment or components must be done to assess any maintenance- and/or safety-related issues. It is also important to evaluate how any existing components will operate under the new set of operating conditions, i.e., stresses, torques, pressures, and temperatures.

The various options that can be considered are:

- scrapping the existing components
- repairing and re-using the existing component
- re-using the component "as-is."

Scrapping existing components and replacing them with new, reliable components presents the lowest risk but highest cost option. Risk remains when using a repaired component repair but it is less costly than the replacement option. Accepting "as-is" components could involve operating the unit at risk and, quite possibly, at a lower level of performance. The final choice lies with the end user who must discuss these risks with the engineering firm performing the revamp.

It is strongly recommended that a thorough inspection be undertaken on any component that will be re-used "as-is". This is particularly true for any rotating components such as impellers. Non-destructive test methods such as magnetic particle inspection, liquid dye penetrate checks or even modal ring testing can be conducted. Further, if an impeller will be reapplied at a speed higher than its original design, a stress and natural frequency analysis should be conducted. If the stress analysis indicates potential problems, a new impeller over-speed test should be done. Further, if the vane counts of any of the stationary components upstream or downstream of the impellers will change, a high-level natural frequency assessment should be done to ensure no interferences. Finally, prior to rotor reassembly, the impeller should be dynamically balanced.

If a shaft is being re-used, the horsepower transmissibility must be reviewed. In the case where the rerate horsepower is higher than the original, a shaft end stress analysis (including a Goodman diagram) should be provided. Prior to re-use, the shaft should be cleaned and inspected. Indications found in the journal bearing areas can be resurfaced and ground to the specified run-out.

There is one other consideration regarding the shaft. If the revamp is to achieve an increase in capacity, the shaft diameter might limit the capacity of the impeller that can be used. That is, for very high-flow coefficient impellers, it might be desirable to reduce or undersize the shaft to maximize the inlet area and minimize the shroud Mach number in the impeller. The trade-off would be aerodynamics versus rotordynamics but if the smaller shaft size is acceptable rotordynamically, it could make the difference between an aerodynamically successful and unsuccessful re-rate. .

If a rerate involves changing the number of stages or if the differential pressure across the compressor changes, a thrust analysis must also be performed. It might be necessary to resize or re-design the balance piston and balance piston packing to assure proper loading of the compressor thrust bearing. It might also be necessary to change the thrust bearing to ensure sound operation.

Re-Locating Existing Components

With some re-rates, it might be possible to re-use some existing compressor components by simply relocating them within the casing. A common example of a revamp with re-located components is a re-rate for a higher volume flow in which a new, larger flow coefficient first stage is installed and the existing stages are moved back one stage space with the existing last stage being removed. The various stationary components might need to be modified to line up with existing case fits or to ensure that the overall bearing span of the new rotor fits within the available space.

Of course, it will be necessary to assess the performance of the re-located component mechanically and aerodynamically to ensure it still meets standard acceptance criteria, i.e., incidence angle or “loading” guidelines, stress and/or deflection criteria, etc.

AERODYNAMIC CONSIDERATIONS

Assuming the revamp/re-rate is being done for performance reasons, the changes and/or re-use of the aerodynamic components will certainly play an important role in the success or failure of the exercise. However, before discussing the components involved, it is important to review some of the critical aerodynamic parameters that must be considered.

Some, but not all, of the performance parameters that must be assessed are presented in the following section. Those interested in further details are encouraged to review the work of Sorokes (2011) titled “Range v. Efficiency – Striking the Proper Balance” listed in the reference section.

Flow Coefficient

Flow coefficients come in two forms: dimensional and non-dimensional. The most widely-used dimensional flow coefficient relates the impeller’s design volumetric flow rate, Q, to its operating speed, N or Q/N. Non-dimensional flow coefficients in their various forms relate an impeller’s design volumetric flow rate, Q, its operating speed, N, and its exit diameter, D_2. Again, the most widely used (in U.S. customary units) is:

$$\phi = 700.16 \, \frac{Q}{N D_2^{\,3}} \qquad (1)$$

Where: Q = volumetric flow rate in cubic feet per minute
N = speed in rotations per minute (RPM)
D_2 = impeller exit diameter in inches

The flow coefficient can provide designers and end users with insight into an impeller’s configuration, i.e., axial length, basic topology, design style, etc. (Figure 4). As can be deduced from the figure, many more low-flow coefficient impellers could fit into a given case than high-flow coefficient impellers. Therefore, if a re-rate is being made to increase the compressor’s capacity, the impellers will likely get longer axially. If the same number of impellers is required to satisfy the new requirements, it is possible the impellers will not fit in the available space.

Efficiency

Because many revamps are undertaken to reduce the power consumption of the compression system, stage and/or impeller efficiency is a key consideration. The most common efficiency term used by compressor manufacturers and/or users is polytropic efficiency. The equation is given below:

$$\eta_{\mathrm{p}} = \left(\frac{\mathrm{k}-1}{\mathrm{k}}\right)\left(\frac{\ln(\mathrm{Pr})}{\ln(\mathrm{Tr})}\right) \qquad (2)$$

Where: k = ratio of specific heats
Pr = pressure ratio
Tr = temperature ratio

Note that Equation (2) is only valid for a thermally perfect gas. Determination of polytropic efficiency for a real gas is a far more complicated effort.

Another popular expression for efficiency is the isentropic form as given below:

$$\eta_I = \frac{\mathrm{Pr}^{\frac{k-1}{k}} - 1}{\mathrm{Tr} - 1} \qquad (3)$$

Flow Range

Two basic factors limit the overall flow range of a compressor: surge or stall margin and overload capacity. Surge or stall margin limit the compressor’s ability to operate at flow rates lower than design, while overload capacity limits the ability to operate at higher rates.

A tremendous number of factors influence both surge/stall margin and overload capacity including operating speed, gas composition/characteristics and compressor geometry. It is not the intent of this work to discuss all of these in detail but rather to introduce the operating range limits.

The terms “stability” or “aerodynamic stability” are frequently used to refer to a compressor’s surge or stall margin. This is not to be confused with “rotordynamic stability,” which assesses the mechanical aspects of the compressor. “Aerodynamic stability” is related to the quality of the aerodynamic flowfield. Typically, a very well-behaved aerodynamic flowfield will result in higher “aerodynamic stability.” That is, it is possible to reduce the flow rate further until the flow path goes aerodynamically unstable.

“Aerodynamic stability” is typically expressed as a percentage:

$$\text{Aerodynamic Stability} \quad 100 - \frac{\phi_{des} - \phi_{surge/stall}}{\phi_{des}} = \% \qquad (4)$$

Where: ϕ_{des} = flow coefficient at design
$\phi_{surge/stall}$ = flow coefficient at surge / stall

"Aerodynamic stability" is specified along a constant speed line and reflects the flow range from design to surge/stall (Figure 5).

"Overload capacity" and "choke margin" are terms used to quantify a compressor's ability to operate at higher-than-design flows. As seen in Figure 5, these parameters indicate how much the flow rate may be increased before reaching the maximum useable flow rate. "Overload capacity" is a bit more difficult to define than surge margin since it is heavily dependent on the supplier's (or user's) interpretation of what constitutes "overload" or "choke." Still, operation in overload can be as or more detrimental than operation in surge. Sorokes et. al. (2006) described the consequences of overload operation.

Rise-To-Surge

"Rise-to-surge" relates how much more head or pressure, typically expressed as a percentage, a compressor generates at the surge/stall line as compared to the head or pressure level at design (Figure 5). Rise-to-surge can help determine a compressor's or compressor section's controllability, assuming the control system is sensitive to the discharge pressure and/or pressure ratio. That is, if the control system determines where the compressor is operating based on the discharge pressure or on the overall pressure ratio, it is advantageous to have greater rise-to-surge because the greater slope in the pressure or head curve will allow a more precise assessment of the compressor flow rate. Conversely, if the compressor has a very low rise-to-surge, it is more difficult to know precisely where the unit is flow-wise.

Machine Mach Number, U_2/A_0

The machine Mach number, also called the tip Mach number relates the impeller's physical tangential velocity to the sonic velocity of the gas at the inlet of a given impeller. The parameter is defined as U_2/A_0 where:

$$U_2 = \frac{N \pi D_2}{720} \qquad (5)$$

Where: N = operating speed (in RPM)
D_2 = impeller exit diameter (in inches)

$$A_0 = \sqrt{k z R T g_c} \qquad (6)$$

Where: k = ratio of specific heats
z = gas compressibility
R = gas constant (1,545/mole weight)
T = inlet temperature in °R
g_c = gravitational constant

The machine Mach number is NOT based on the velocity of the gas exiting the impeller and must not be confused with the shroud inlet relative Mach number discussed below. Rather, machine Mach number gives the OEM and the end user an indication of the overall flow range one might expect for an impeller. Typically, as the machine Mach number gets higher, the overall flow range gets narrower, i.e., less flow range from the choke point to minimum stable flow or surge (Figure 6). Therefore, the parameter can be used to assess potential changes in range of individual stages within the machine and, by inference, the changes in the overall compressor range following a revamp. Again, if the new conditions will cause the impellers to operate at a higher U_2/A_0, the compressor range will likely get narrower. Conversely, if the U_2/A_0 drops, the compressor range should improve… provided the stages are properly matched aerodynamically (Sorokes, 2011).

The allowable U_2/A_0 is heavily dependent on the details of the impeller designs in question. Modern centrifugal impellers that have been designed for high Mach number applications can operate effectively at U_2/A_0s of 1.2 or higher. However, if a revamp uses older impellers that were not specifically design for high Mach number conditions, the U_2/A_0 limit will be much lower, i.e., 1.0 or lower.. The consequences of applying a low Mach number impeller at high Mach numbers will be reduced surge/stall margin and reduced overload capacity, i.e., a much narrower flow map. Therefore, the end user must receive assurances from the OEM that the re-purposed impeller is capable of operating effectively at the new U_2/A_0.

Inlet Relative Mach Number

Inlet relative Mach number or more specifically, shroud inlet relative Mach number is the highest Mach number and/or highest velocity within a centrifugal stage. It occurs along the impeller cover or shroud at or very near the blade leading edge. Unlike machine Mach number, shroud inlet relative Mach number is a gas velocity. It is the inlet relative velocity at the shroud divided by the inlet sonic velocity to that stage. Like machine Mach number, it gives the designer and end user insight into the flow range of an impeller, especially toward the overload or choke side of the flow map, e.g., the higher the inlet relative Mach number, the lower the flow range to choke. As with machine Mach number, it is important to consider how the inlet relative Mach number changes for the revamp conditions. Again, older impeller designs were likely not designed to operate much above a shroud inlet relative of 0.85 or lower. Should they be applied at higher inlet relative Mach number, the consequence will again be shorter surge/stall margin and overload capacity. The impellers designed to operate in today's more sophisticated process market compressors can work effectively up to shroud inlet relative Mach number of 0.94 or higher.

Impeller Relative Velocity Ratio or the like

Digging a bit deeper into the impeller performance characteristics, one parameter that should be considered when reviewing revamp selections is the impeller relative velocity ratio or some similar relationship. The velocity ratio compares the relative velocity at the inlet of the impeller to the relative velocity at the exit. The ratio gives the designer (or the user) an indication of the level of diffusion in the impeller. Typically, as the relative velocity ratio increases, the impeller's aerodynamic stability decreases. Increasing the operating speed of the impeller will also increase the relative velocity ratio. Therefore, it is important to review this parameter when an existing impeller is going to be applied at new set of operating conditions. It is possible the new conditions could result in a reduction of the impeller operating range.

The following discussions focus on the aerodynamic considerations that must be taken into account.

Impellers

Given their importance in the performance of a centrifugal compressor, the initial focus of any revamp effort is on the impellers. No amount of changes to stationary components can overcome the shortcomings related to poorly performing impellers. It is therefore critical to assess how the impellers will perform at the new operating conditions. The end user and OEM must reflect on the changes in critical parameters such as machine Mach number, inlet relative Mach number, relative velocity ratio, leading edge incidence levels, and impeller exit flow angle to ensure they fall within acceptable ranges and to gain a better understanding on how the revamp conditions will affect the overall compressor performance map.

There are also some considerations regarding the impeller manufacturing techniques. Quite often during a revamp, the impeller fabrication technique could be changed. For example, the original equipment might contain riveted impellers, old cast impellers, three-piece welded impellers, or the like. The end user might choose to replace impellers built with older techniques with impellers built using more modern 5-axis machined, electron-beam welded, single-piece milled, or other methods. Therefore, it is important to understand the potential consequences of these changes in manufacturing techniques. NOTE: Many of these observations also apply to the so-called compressor upgrades mentioned earlier. That is, an end user might order an upgrade expecting no performance improvements or changes only to discover that the upgraded internals do change the compressor's performance.

Riveted to Welded or Brazed

Quite often, riveted impellers are converted to welded or brazed because of the higher strength and fatigue resistance of the welded or brazed part relative to the riveted version. The resulting performance changes will depend heavily upon the style of riveted impeller being replaced and on any changes in blade shape made in the replacement design. The details of the replacement design will also depend on the intent of the change, i.e., is it an upgrade with no intended change in performance or is it a revamp/re-rate in which the impellers are being re-designed for higher performance.

In general, there are two styles of riveted wheels: "through blade" rivet and "Z"-bladed rivet, where the rivets pass through a flange at the top and bottom of the blade (Figure 7). In the latter, the portion of the impeller flow passage blocked by the flange is typically similar to the portion that will be blocked by the weld fillet for an upgrade where no aerodynamic enhancements are made to the impeller; the performance of the welded impeller will be essentially the same as the "Z"-bladed riveted wheel. If the "Z"-blade riveted impeller is replaced by a brazed or electron-beam welded impeller (i.e., with no corner fillets), the replacement impeller might have more overload capacity and possibly less surge margin than the original because of the reduction in passage blockage. In short, without the flange blockage, the replacement impeller has a larger flow area so it has higher flow capacity.

One other consideration on "Z"-bladed impellers is that most of the "Z"-blades were built from sheet metal that approximately 0.090 inches thick. The blades in the welded impeller are normally thicker, i.e., 0.125 inches or greater. Therefore, it is possible the welded impeller will have a smaller passage area than the riveted impeller *unless* design changes are made to account for the increased thickness.

The reverse could be true when converting a "through blade" riveted impeller to a welded equivalent. That is, the "through-blade" riveted impeller has no blockage at the junction between the blade and the hub or shroud (Figure 8). The welded impeller will have some amount of blockage in the corners due to the weld fillet. The result will be a reduction in the passage area and a reduction in the impeller capacity. It is possible to correct for the fillet blockage described above but the end user needs to be assured by the supplier that such corrections have been applied.

Three-Piece Welded to 5-Axis Machined

Many older compressors contain impellers that were fabricated as three pieces. That is, the blades were welded to both the disk and cover such that there were four weld fillets in each impeller passage. Quite often during revamps and/or upgrades, these impellers are replaced by impellers that were 5-axis machined and either the cover or disk welded or brazed in place. In this case, the weld fillet blockage is typically replaced by a machining fillet so one might not expect a change in performance. However, the large amount of heat input to the three-piece welded impeller often caused distortions in the flow passage. While these distortions were typically very small (approximately 0.100" or less), the more accurate flow passages achieved with 5-axis milling can result in improved efficiency and capacity.

Improved Blading

Many advances have been made in the design of centrifugal impeller blading. These were made possible by more advanced manufacturing techniques and more sophisticated design / analysis tools. As a result, the efficiency and flow range of more modern centrifugal impellers exceed those of earlier designs. The three-dimensional and/or sculpted blade shapes being developed today are vastly superior to the old two-dimensional shapes commonly used in the 1960s and earlier. As a result, the performance levels of new stages have continued to increase, though the trend has "flattened" since 2000 (Sorokes and Kuzdzal, 2011). It is possible to achieve significant increases in efficiency and throughput during a revamp by applying more modern impellers; the incremental increase dependent on the vintage of the impellers being replaced.

Stage Spacing

The stage spacing for impellers typically varies in direct proportion to the flow coefficient, i.e., the larger the flow coefficient, the longer the axial stage spacing. Very high-flow coefficient impellers typically have wide flow passages and long axial inducer sections that increase the overall axial length of the impeller and their associated stationary components. Conversely, low-flow coefficient stages have narrow passages that typically do not include an axial inducer (Figure 4). As mentioned in the Case section, the change in impeller flow coefficients from the existing to the new rotor configuration will determine whether or not there is sufficient axial space within an existing case to fit the revamp rotor.

It might be possible to develop an impeller that is axially shorter and that will pass the required flow. However, squeezing the impeller into the available spacing will likely require tighter turns along the impeller cover and not allow a long axial inducer. This can negatively impact the performance of high-flow coefficient stages, severely impacting peak efficiency, pressure ratio and range. For example, installing a non-inducer style impeller with a flow coefficient of 0.08 or greater could yield an efficiency loss of five points or more. This would be much akin to installing a 1950's vintage impeller in a modern application. It might work but performance is compromised.

Modifying Impellers/Inlet Guides for Reduced Stage Space

One method that can be used in lower flow coefficient (i.e., flow coefficient ≤ 0.05) is to use the impeller cover to form a portion of the upstream inlet guide. In this situation, the impeller cover is shaped to form the shroud surface of the inlet guide (Figure 9). The flow exiting the return channel passes directly onto the cover of the impeller. While this configuration does allow a large impeller to fit into shorter stage spacing, the consequences are shorter surge/stall margin, lower efficiency and lower head rise or pressure ratio. These consequences are primarily due to the extra swirl caused by exposing the flow to the whirling cover and the non-uniform inlet conditions that also result. Still, this arrangement has proven effective and is a viable option as long as the end user understands the potential performance effects.

Inlet Guide Changes

Another key component often changed during a revamp is the inlet guide. Guidevanes can be used to adjust the capacity of the downstream impellers (Sorokes, 2011). If one puts curvature in the vanes or orients the vanes other than in a purely radial direction, the exit flow will have a tangential velocity component called "pre-whirl" or "pre-swirl." Depending upon the direction of rotation of the compressor shaft (*not* indicated in Figure 10), the "pre-whirl" can be either "against" the direction of rotation or "with" the direction of rotation; hence the names "against IGV" and "with IGV." The "pre-whirl" causes a change in the inlet velocity field or inlet velocity triangle into on the downstream impeller. Therefore, by changing the inlet guide in front of a given impeller, it is possible to adjust the flow map (Figure 11). Again, adding "with pre-whirl" shifts the map to lower flow rates while adding "against pre-whirl" moves the map to higher flow rates. Therefore, it is possible to adjust the location of the peak efficiency with "pre-whirl" inlet guide vanes. Consequently, it might be possible to increase or decrease the capacity of a revamped compressor simply by changing the guidevanes. In short, it might be possible to achieve the desired change in capacity without changing the rotor. The advantages are structurally and rotordynamically obvious, i.e., re-use of rotor, bearings, seals, reduced cost, and reduced turn-around time.

Several factors limit the amount of shift that can be effectively achieved. First, the additional turning of the flow can result in additional losses in the guide vane, reducing the overall efficiency of the stage. Second, if the turning in the inlet guide vanes becomes too severe, the inlet guide will behave more like a throttle valve, which results in pressure loss and further efficiency degradation. Third, the "pre-whirl" causes a change in the inlet relative gas velocity. While potentially advantageous for "with" rotation because "with" rotation decreases the inlet relative velocity, this can be a problem for "against" rotation because the inlet relative velocity and inlet relative Mach number increase. Fourth, to achieve reasonable turning, the vane count in the inlet guide must increase, causing more wetted surface and higher friction losses.

Diffusers

The diffuser often becomes a critical consideration during a revamp. This is particularly true when attempting to install higher diameter impellers in an existing casing. As noted earlier, the diffuser radius ratio is a critical parameter in achieving acceptable diffuser performance, especially in the case of vaneless diffusers. If the vaneless diffuser is too short, diffuser static pressure recovery will suffer and the efficiency and volume reduction of the machine will decrease. The resulting impact on the overall compressor could make a revamp unworkable.

It is possible to offset the consequences of low vaneless diffuser radius ratios by switching to a form of vaned diffuser such as a low solidity vaned diffuser (LSD). LSDs can provide the necessary static pressure recovery in a shorter radial space. There might be a small reduction in the overall flow range (surge to overload) but this reduction might be an acceptable compromise if it allows for re-use of an existing casing.
As noted in the Case section, when attempting to re-use some axially- or horizontally-split cases, one must contend with fixed return bend locations. In these situations, it is quite often necessary to adjust the diffuser design to "meet up with" the return bend locations. For example, it might be necessary to "over-pinch" the diffuser to squeeze the flow path into the exiting bend. This over-pinching reduces the diffuser static pressure recovery and the stage efficiency. Again, it might be the difference between re-using and not re-using the casing. The end user will have to weigh the benefits of re-using the case against the reduced performance because of the compromised diffuser sizing.

Diffuser passages can be altered in other ways to meet existing return bends. As shown in Figure 12, it is possible to slope the diffuser and the downstream return channel to re-use the existing bend. Care must be taken not to "over-slope" the diffuser and return channel because this can induce flow separation and premature stall. However, again, if this concept makes or breaks a revamp/re-rate, this option must be considered.

Vaned Diffusers

It is possible to re-use existing vaned diffusers. Care must be taken to review the resulting incidence angles on the diffuser as the result of the changes in upstream impellers and/or the change in impeller exit conditions (if existing impellers are being re-used). The supplier must assure the end user that the incidence angles fall within mutually acceptable guidelines. In the case of a channel or passage diffuser, the OEM must also confirm that the diffuser throat will not be choked at the new operating conditions.

Diffuser vanes can also be adjusted to achieve a slight improvement in surge margin in a compressor, though guidevane changes are much more effective for this purpose.

Other Stationary Components

Return Channels

Two issues related to return channels deserve attention during revamps / re-rates. First, return channels can be re-used provided there is sufficient area to accommodate the new flow conditions and the incidence on the return channel vanes is within ±5° of the return channel vane angle at the new required flow condition. If the return channel is undersized or if the incidence angle is highly negative (i.e., -10 or more negative) at the new operating conditions, the return channel losses will increase nearly exponentially and significantly reduce the overload capacity. In fact, the performance will appear like that of a compressor with a choked impeller.

The return channel bulb can also be re-sized to allow for re-use of existing return bends. For example, if higher flow coefficient stages are being retro-fitted into an existing casing, the diffuser width and return channel width must both be increased. There is only so much width in an existing bend so the only way to increase the flow passage on both sides of the bend is to decrease the width of the bulb (Figure 13). The narrower bulb can impact the surge/stall margin of the stage so this must be anticipated when developing the prediction for the revamped compressor. Again (ad nauseam), the compromised performance might be acceptable when weighed against the need for a new casing.

Inlets/Volutes/Collectors

When revamping a compressor, the primary concern with inlets, volutes and collectors is that those components do not cause a flow restriction. The result will be excess losses that will compromise the compressor performance. Again, the OEM can account for these losses in the performance prediction and, if the impact is acceptable to the end user, overloaded inlets, volutes and collectors can be used.

Note: there are some concerns with severely under-loaded inlets, volutes and collectors. Significant over-sizing of these components can lead to flow separation, vortices and other untoward flow phenomenon that can reduce the useable operating range of the compressor. Again, if these effects are properly anticipated and accounted for in the performance maps generated, it is possible to use the over-sized components.

ROTORDYNAMIC CONSIDERATIONS FOR REVAMPS AND UPGRADES

Motivations for a Lateral Analysis

For safety's sake there are many reasons to make a lateral analysis to the latest edition of API 617 for centrifugal compressors that are being revamped or upgraded and to write a lateral report to that edition for delivery to the purchaser. Some of them are described below.

If there is no original lateral report or if there is one and it is not written to the latest edition of API 617

There might be no report because for the 1st and 2nd Editions issued in 1958 and 1963 there were no reports required. Then for the 3rd Edition of 1973 a rotor response had to be done only if specified or it could just be a critical speed map. If done to the 4th Edition of API 617, issued in 1979, a response analysis and a critical speed map were required. The response analysis in the 4th Edition was probably was for just average clearance at the bearings. For the 5th Edition, issued in 1988, the minimum and maximum clearances were required and the report requirements were extensive. That edition had a new requirement that the amplitudes at close clearance locations, except floating-ring seals, should be no more than 75% of the minimum running clearances. The 6th Edition, issued in 1995, was pretty much the same as the 5th Edition with regard to rotordynamics. Finally for the 7th Edition, issued in 2002, there was a requirement for a stability analysis and requirements on the value of the log dec. If a screening criteria (which included a stability analysis) was not passed then a second stability analysis with the inclusion of the labyrinths was required. But vendors had been doing stability analyses since around 1973 with increasing sophistication as the years passed.

If there is a change to the rotor, or aerodynamics or aerodynamic flow path

Changes to the rotor or aerodynamics or aerodynamic flow path might involve more impellers on an existing shaft or there could be a decrease to the impeller bore. Either case could result in a lower first and second critical speed. A lower second critical speed would mean that the margin of the second critical speed from the maximum continuous speed should be re-evaluated. A lower first critical speed could mean a higher amplification factor of the first critical speed, implying that the amplitudes seen at the mid-span while running through the first critical speed vs. the running clearances of the labyrinths are more in question.

Another possible change to the rotor or aerodynamics or aerodynamic flow path might involve impellers with higher impeller bore. This then could mean a higher first critical speed, which would mean that the margin of the first critical speed from the minimum governor (operating) speed could be in question. Since the bearing span could not change for a revamp and if the first critical speed was pushed into or too close to the operating speed range then undercutting the shaft between the impellers or wherever there is space to do it is an option that could be pursued. There are limits to how much the shaft should be undercut; certainly the undercut diameter should be no less than the bore of the drive-end coupling.

If the speed range is being changed, especially to a higher maximum continuous speed

Sometimes the owner will come back with a request to increase the maximum continuous speed to get more throughput, or the inlet pressure is falling and the user wants to maintain the same discharge pressure by increasing the speed, or might just want to raise the discharge pressure. There are other considerations, such as will the impellers handle the higher stresses or will the case handle the higher pressure, but a significant consideration is will the rotor dynamic acceptability be affected. The margin of the second critical speed to the maximum continuous speed will be less.

It may seem that this is just a matter of checking the critical speed locations in the original report, if one exists, and if the margin is wide assume that some change won't hurt. But would it meet the requirement of the 7th Edition that the log dec > 0.1? If the bearings are the fixed geometry type, the log dec and thus the stability, are highly dependent on speed and can decrease rapidly with speed.

If the coupling size is being increased

The coupling size may have to be increased if there is more horsepower to be absorbed by the compressor. This usually would mean more half-weight on the compressor, which for small- to medium-size compressors could mean a significantly lower second critical speed. For very large compressors this might not be a problem, as usually the second critical speed is mostly affected by the bearing span and the impeller bore. The use of a titanium spacer or even titanium in the body of the coupling would lower the weight and might get it back near the original half-weight.

If the coupling is being changed from a reduced moment style to a marine style coupling

It would not usually be the case to go from a reduced moment to a marine-style coupling. Reduced moment style means that the cg of the half-coupling is significantly inboard of the shaft end. Marine-style coupling means that the cg of the half-coupling is close to or off of the shaft end. The further the cg is from the bearing center-line the lower the criticals which are affected by the overhang of the compressor. A marine-style coupling could mean a significantly lower second critical speed and the margin that is left would have to be evaluated

If there are no bearing probes

If there are no bearing probes, a lateral analysis is recommended no matter how well the compressor is running. If there are no bearing probes then there is no measure of the amplitudes at the bearings; it could be they are near the bearing clearances and bearings could be wearing out prematurely from high synchronous amplitudes (the operators might think that this is normal). And if there are no bearing probes then there is no way of knowing if the vibration has been at typical alarm and shutdown levels.

If there are no bearing probes there could be significant subsynchronous amplitudes at the first natural frequency while the machine is running at the maximum continuous speed and labyrinths have been degraded or wiped (and there has been significant performance loss). Then when new impellers and new labyrinths are put into the machine, the promised performance might not last very long if mitigation efforts were not made because the lateral analysis was not done.

If there are fixed geometry bearings

It is not unusual to consider revamps and upgrades for compressor that were built in the 1960s or earlier and find that fixed geometry bearings are still being used. Fixed geometry bearings are known to have a propensity for speed-dependent oil whirl. This has been known to happen even if the machine is considered to have a stiff shaft, i.e., the first critical speed on rigid supports is well above the operating speed range. A case history is shown by DeSantiago and Memmott (2007) and is discussed later in this tutorial.

If there are fixed geometry bearings, a damped eigenvalue analysis should be done to see if the fixed geometry bearings are susceptible to oil-whirl. Consideration should be given to have funds available for the contract to replace them with tilt pad bearings. API 671 8th Edition will require a stability analysis for any compressor with fixed geometry bearings, no matter if it is a stiff shaft or not.

If there are oil-film ring casing end seals

There are known cases where compressors with oil-film ring casing end seals had unacceptable values of subsynchronous vibration at the first natural frequency while the compressor was running at the maximum continuous speed. This has happened even when the compressor had tilt pad bearings. Damped eigenvalue analysis for compressors with oil-film ring seals has shown that there can be a speed-dependent eigenvalue which tracks half-frequency and eventually turns into oil-whip, i.e., locks on to the first natural frequency and is unstable.

Several papers, Memmott (1990, 1992, 2004, have shown that this problem can be eliminated by using tilting pad oil-seals. At higher speeds, ring style oil-film seals may have a significant amount of cross-coupled stiffness. Tilt pad oil seals to do not have a significant amount of cross-coupled stiffness, as tilt pads themselves do not have significant cross-coupled stiffness, and they control the sweet oil ring that they sit in, lower the eccentricity of the ring and thus lower the cross-coupled stiffness of the ring. A case history of an upgrade(s) from oil-film ring seals to tilt pad seals is discussed later in the tutorial. API 671 8th Edition will require a stability analysis for any compressor with oil film ring seals, no matter if it is a stiff shaft or not.

If oil film casing end seals are being changed-out to dry gas seals

There is a significant amount of damping available from oil-film seals, ring style or tilt pad style. This can have a good effect, except not enough for oil-film ring seals at higher speeds. A change to dry gas seals eliminates this damping. API 617 8th Edition says that dry gas seals can be assumed to have no stiffness or damping. In turn this elimination of damping can increase the likelihood of instability. This has been shown by Kocur et al. (1987) and Memmott (2007). The later paper will be given as a case history in this tutorial.

If the type of seal is being changed at the balance piston or division wall.

There have been examples where the change-out of the type of sealing at the balance piston or division wall has affected the rotor dynamic stability of the compressor. There is one case where the change from toothed labyrinth to honeycomb style at the balance piston had unintended consequences to the stability of the system Kocur and Hayles (2004). Another case history, which is discussed later in the tutorial, is for a revamp of a compressor from three to six stages, where a toothed labyrinth at the division wall seal was changed to a hole pattern-type and a low frequency instability occurred. This was resolved by a change in the geometry of the hole pattern seal (Memmott, 2012).

If the tilt pad bearing style or geometry is changed

Many times, older style line contact tilt pad bearings have been changed out to ball and socket or elliptical pivot bearings to better handle misalignment. In the case of the ball and socket bearings, there can be a reduced spread in the bearing clearance range because of the use of shims. Also, there may be room to increase the length of the pads, which results in a more favorable relationship of damping to stiffness for the bearings. In any case the lateral rotor dynamics are going to change and the effects of these changes should be quantified.

If damper bearings are being introduced

The original bearings could have been tilt pad bearings without squeeze-film dampers. The vendor might make a recommendation that squeeze-film dampers be applied in series with the tilt pad bearings. This is going to be a completely different rotor dynamic system. To perform due diligence, a rotor dynamic analysis and report with the squeeze-film dampers is needed.

Exceptions to doing a lateral report

If it is only a coupling change-out to an existing train of equipment that is known to be running well, then a lateral analysis might not be needed if the half-weight and center of gravity of the new coupling is close to that of the existing coupling.

Considerations for torsionals

If the torsional stiffness and inertia of the new coupling is close to that of the existing coupling then a torsional analysis would probably not be needed. However, if the driver is being changed, then a new torsional report is needed.

Lateral Rotor Dynamic Programs - History

A short history of key lateral rotor dynamic programs and their use follows:

Undamped Critical Speed

In the early years, before 1965, the only program available to perform lateral rotor dynamics, was an undamped critical speed program, the theory of which is in Myklestad (1944) and Prohl (1945). The first rigid bearing forward critical speed was calculated; this had to have some separation margin from the minimum governor speed. The rigid bearing second critical speed was calculated. In order to have a 20% margin of the actual second critical speed above the maximum continuous speed, the vendor wanted more margin for the rigid bearing second critical speed. This was to account for the actual second critical being lower than the critical on rigid supports because of the softness of the actual bearing.

Tilt Pad Bearings

In 1964 Jorgen Lund (1964) introduced a method for analyzing tilt pad bearings in which he applied the pad assembly method to assemble synchronous stiffness and damping coefficients. Around that time, tilt pad bearings were first used in centrifugal compressors. By the mid- to late - 1960s, most new centrifugal compressors were being built with tilt pad bearings.

Rotor Response

In 1965, the manual (Lund, 1965) came out for Lund's rotor response program. The analysis was by the transfer matrix method. This program is commonly used today. The theory for the program is shown in Lund and Orcutt (1967).

In 1974, Lund (1974) produced one of the most influential papers written in the field of rotordynamics. It was on the calculation of damped eigenvalues and log decs. He did it by the transfer matrix method, just as he had done for the rotor response program. The program manual is given by Smalley, et. al. (1974). From 1973 and on, stability analyses were common. The paper by Memmott (2003) discusses the history of the usage of the Lund Programs.

In the 1970s, other authors followed Lund in writing programs for the analysis of stability (Bansal & Kirk (1975), Barrett et. al. (1976) and Rouch and Kao (1979). The first uses the transfer matrix method and the third one uses the finite element method.

Toothed Labyrinth Programs

By the 1980s, it was well recognized that the toothed labyrinth seals had significant effects on the stability of centrifugal compressors. Papers and programs were written on the analysis for the stiffness and damping coefficients of toothed labyrinth seals. (See the papers by Benckert and Wachter (1980), Iwatsubo et al (1982), Kirk (1985, 1986), and Childs and Scharrer (1986)).

Honeycomb and Hole Pattern Seal Program

A program at Texas A&M was devoted to calculating the frequency dependent stiffness and damping coefficients of honeycomb and hole pattern seals (Kleynans and Childs, 1996, and Childs and Wade , 2003). This is much used in the centrifugal compressor industry.

Additional Material on Lateral Analysis Programs

It is critical in the lateral analysis of centrifugal compressors for revamps and upgrades and for new equipment to have a set of proven core programs that are versatile and efficient in conducting the lateral analysis of centrifugal compressors. Below we describe the lateral analysis programs currently used by this vendor. Most of the case histories were analyzed with these programs and this is to document what was used.

Undamped Critical Frequency Program

The undamped critical frequency program used by the author's company has some features believed to be uncommon but extremely useful. Besides the forward synchronous critical speeds, which are what are shown in the critical speed map required by API 617, it can also calculate the backward synchronous critical speeds, the planar (which means that the rotor is at rest) critical speeds, and critical frequencies for the shaft running at a fixed speed. The undamped critical speed map in the API report does not tell the whole story; it only shows the forward synchronous critical speeds. Critical frequency maps are made for the forward, backward and planar speeds when the software program described below is run. There are internal rules for the separation margins from the first and second forward synchronous rigid bearing critical speeds and the fourth backward synchronous soft bearing critical speed.

If there is some asymmetry in the journal bearing or oil-film seal coefficients, or a large overhung mass, then it is not unusual for a backward critical speed to show up in the response results. The backward critical frequencies also show up in the eigenvalue results. The planar option is used for the modeling of a modal ring test of a rotor. The fixed speed option has been used to make a simple redesign of pinions, where the fourth mode of the pinion had coincided with eight times the running speed and vibrations were unacceptable (Memmott, 2005, 2006). Those papers showed that it is accurate enough to use the planar option in this case, but not accurate to use the synchronous option.

Bearing Programs

For the bulk of the analysis of tilt pad bearings, the author's company uses the tilt pad bearing program (Nicholas et al, 1979). This program uses the finite element and pad assembly methods. It is used extensively in the compressor industry. The author's company has found it to be fast, efficient, accurate, and suitable for a production environment. Before the joint venture, one of the author's companies exclusively used the Lund tilt pad bearing program (Lund, 1964). For some unique geometry of tilt pad bearings, the author's company uses the THPAD program from the ROMAC program from the University of Virginia (Branagan and Barrett, 1988-1999). For fixed geometry bearings, such as pressure dam bearings, the author's company uses the program of Nicholas (1980, 1981, 1985)

Response, Damped Eigenvalue and Oil-film Ring Seal Programs

The response and damped eigenvalue programs used by the vendor's company are the programs of Lund, as described above. For the eigenvalue program a sub-routine has been added to connect the dots, i.e., connect the frequencies as they vary with speed. Also, the eigenvalue results of log dec and frequency are used to predict the location and amplification factors of the response critical speeds (Kirk, 1980). This gives a numerical judgment of the location and severity of the response critical speeds rather than trying to estimate the severity by looking at the intersections of the dynamic bearing stiffness with the undamped critical speeds on the critical speed map. See the results shown in Memmott (2003). The oil-film ring seal program is the one by Kirk (Kirk & Miller, 1979 and Kirk, 1986). This program had been integrated with the response program years ago.

Toothed Laby, and Hole Pattern or Honeycomb Programs

The toothed laby program used is the one by Kirk and the honeycomb and hole pattern seal used is the one by TXAM, as described previously.

Lateral Analysis Procedures for Revamps and Upgrades (and New Compressors)

It helps greatly in the lateral analysis of revamps and upgrades (and of new compressors) to have a software tool which combines the various analyses discussed above and below to produce a detailed lateral analysis of the revamped compressor. It should do the undamped critical speed analysis, the journal bearing analysis – for both tilt pad and fixed geometry bearings – the unbalance response analyses and the stability analysis in a seamless fashion. The program should be able to model squeeze-film dampers and oil-film seals.
It should be able to import the aerodynamic properties from the performance results and then include the effects of toothed labyrinths and hole pattern or honeycomb seals in the rotordynamic calculations, so that a API 617 Level II analysis can be done. It should do all of this in a short time, so that there is no headache in making adjustments to rotor models, journal bearings, oil seals, and annular gas seals in order to produce an acceptable rotor dynamic design for the purchaser.
It should have built-in rules to check the results of these calculations against the API 617 7th Edition specifications and the vendor's internal design guidelines for centrifugal compressors. The results should show the amplification factors of the critical speeds, whether or not the required separation margins are met and the log decs, and check them versus the API and internal requirements. Then, one should be able to make a few simple choices and enter in some descriptive information and then push a button to create a report that shows that the API requirements are met. Such a software tool is discussed by Ramesh (2002). See Figure 14, the flow chart from Ramesh and Memmott (2007).

Stability History

Much of the rest of the tutorial on rotor dynamics is about stability. Stability is a very important subject with respect to revamps and upgrades, as the compressor must be stable. First we will give a short history of the subject.

In the 1970s several vendors endured a series of instability incidents on the test stand or in the field, such as those at Kaybob (Fowlie and Miles, 1975 and Smith, 1975) and Ekofisk (Booth, 1975). In the next paragraph, we discuss how the author's company solved its instability problems.

During the 1970s, the author's company started using several parts to combat stability problems in the field and then applied them to original equipment and upgrades. Tilt pad seals were first used by this vendor in 1972 (Figures 32 and 34 show sketches of a tilt pad seal). Damper bearings were first used by this vendor in 1973 (Figure 19 shows a sketch of the damper bearing type). Shunt holes were first applied by this vendor in 1974 (Figure 20 shows a sketch of the shunt hole system). See Memmott (1990, 1992) for a discussion of these parts and the early history of their usage.

In 1981 a paper by Wachel and von Nimitz (1981) presented an empirical formula for representing the aerodynamic excitation in a centrifugal compressor. It was called the Wachel formula and was derived from the Alford equation (Alford, 1965). Some vendors may have applied versions of the Alford formula to estimate the cross-coupling in compressors. The Wachel formula was based on stability analyses of a collection of compressors in mostly gas injection service that had been unstable. The formula was used extensively. In 2000, Memmott (2000) issued a formula that was a modification of the Alford and Wachel formulas and this formula is being used in the API 617 7th Edition today as part of screening criteria for compressors that may need more in-depth analyses.

Also introduced for stability screening criteria were two different plots, both of which had several data points and lines, based on experience, dividing the plots into regions of higher or lower concern.

The first that was published in 1983 by Kirk and Donald (1983) had a plot of discharge pressure times differential pressure across the case vs. flexibility ratio. The flexibility ratio was defined as the ratio of the maximum continuous speed to the first forward undamped critical speed on rigid supports. The second that was published in 1984 by Fulton (1984) had a plot of flexibility ratio as defined above vs. average gas density for the compressor, the average of the inlet and discharge densities. It was developed for compressors with oil seals, which was the way compressors were built at that time. Then in the same year Fulton published his plot again (Fulton, 1984b) and on the Kirk-Donald plot he added a worst-case line below that given on the original Kirk-Donald plot.

Honeycomb and hole pattern seals at the division wall or balance piston were first applied as damping devices in the mid-1990s. Honeycomb seals had been used years before that in some syn gas compressors before rotordynamic benefits were recognized. One of the programs that covered the stiffness and damping coefficients was the one from Texas A&M mentioned earlier. There were some hiccups along the road in the use of such seals, as it was proven that they needed to be de-swirled (shunt hole systems were used to do this). See the papers by Memmott (1994), Gelin et al (1996), and Camatti et al (2003). A summary of the experience with the use of honeycomb and hole pattern seal is given by Memmott (2011). Papers were written by Nielsen and Myllerup (1998), Nielsen et al (1998), and Moore and Hill (2000) analyzing and describing the application of swirl brakes, which are used to deswirl toothed labys and hole pattern and honeycomb seals.

API 617 7th edition – July 2002 to ? – Stability Criteria

The big change in the 7th Edition was that a section was added with stability requirements – approximately thirty years after instability problems were first encountered with centrifugal compressors. There were some changes made to the response requirements, but they mostly were fine nuances in how the response analysis was to be done. The stability criteria are divided into Level I and Level II.

Level I Stability Criteria

The Level I spec is a screening tool and the Level II specification requirements had to be met only if the Level I spec indicated that it had to be checked. The Level I screening criteria was developed from the practices of several vendors and purchasers. It included a modified Alford-Wachel number as in Memmott (2000) and a modified version of the Fulton plot, as described by Memmott (2002). Both levels require analytical calculations of the log dec. The Level I analytical model includes the same rotor model, journal bearings, squeeze-film dampers, bearing supports, and oil-film casing end seals as for the rotor response. The Level I analysis does not include toothed labyrinths, nor does it include damper seals, such as honeycomb and hole pattern seals.

Level II Stability Criteria

The toothed labyrinths and, if used, damper seals, such as honeycomb and hole pattern seals, are typically included in the Level II analysis. The final log dec from the Level II analysis is supposed to be greater than 0.1 to meet the API requirement, unless a lower value is negotiated between the vendor and the purchaser. Some vendors, purchasers and consultants have a higher lower limit for the log dec.

Speed Dependency of the Log Dec with Fixed Geometry Bearings or with Oil-film Ring Seals

As mentioned earlier, with fixed geometry bearings or with oil-film ring seals there can be strong speed dependency of the log dec. This is best examined by a calculation of the log dec for a range of speeds from low-speed to well above the maximum continuous speed. Any tendency for half-frequency whirl should show up. The sensitivity of the log dec to speed will not show with a calculation only at the maximum continuous speed, so the recommendation in these cases is to do more than what is required by API. This is shown in DeSantiago & Memmott (2007), Memmott (2004), Marshall et al (1993), Memmott (2008), and Memmott and Buckvich (2008). There will be such plots in the case histories.

Use of Experience Plots – With Dry Gas, Toothed Laby, or Very Low Pressure Oil-film Ring Casing End Seals

There are more experience plots than the ones of Kirk-Donald, Fulton and the API plot. Many data points are added to the API plot and to a modified version of the Kirk-Donald plot in papers by Memmott (2002, 2010, and 2011). Also in those papers are plots where bearing span/impeller bore replaces flexibility ratio. It was seen that the bearing span/impeller bore does better than flexibility ratio. The flexibility ratio is just a measure of how fast the compressor is running relative to its first critical speed and does not give the appropriate amount of concern if the compressor is running slow, yet the bearing span is long and the rotor is flexible. Flexibility ratio is a good measure if the bearings are fixed geometry bearings or there are oil-film ring seals, both of which can be fixed by the use of tilt pads.

These plots are used to assess the revamped or upgraded compressor (or new compressor) to see how it fits on the plots. If it stands by itself or if it does not have the attributes of nearby compressors, such as squeeze-film dampers, hole pattern or honeycomb seals, shunt holes and swirl brakes, then there is a concern and an extensive stability analysis is needed. Most all of the high density experience or high discharge pressure times case differential pressure shown by these plots in those papers is with shunt holes at the division wall or balance piston or both swirl brakes and shunt holes at the division wall or balance piston.

The conclusions from the preponderance of the data in Memmott (2011) is that for compressors with high density or high discharge pressure times case differential pressure one would be advised to use a hole pattern or honeycomb seal, that those seals have greatly extended the experience of compressors into the high density regions, and that this can be done with or without squeeze-film dampers.

Other authors Camatti et al (2003), Bidaut et al (2009), and Bidaut and Baumann (2010) have presented similar type experience plots, all of flexibility ratio vs. average gas density. The use of such plots are probably vendor specific, as the suitability of the compressors likely depends on the specific design practices of the individual vendor. For critical applications, it would provide assurance to the purchaser if they could find nearby experience of the vendor with compressors from that vendor with the same attributes on those plots.

Use of Experience Plots – With Tilt Pad Oil-film Seals

Beginning in 1992, Memmott (1992, 1998-1999, and 2004) published case histories and experience plots that showed when using tilting pad oil-film seals, the critical speed with the effect of the tilt pad seals should be used in the flexibility ratio calculation, instead of the first undamped rigid bearing critical with just the journal bearings. The inclusion of the tilt pad seals raises the first critical speed and effectively shortens the bearing span. Memmott (2004) will be discussed as a case history. See Figure 38 from Memmott (2004) for experience plots without and with the tilt pad seal effect.

Case Histories

Several case histories are discussed that demonstrate some of the issues noted above. The first will be with respect to rotor response and the rest cover stability.

Case History 1 – A Large Compressor- Load On vs. Load Between the Pads [Ramesh & Memmott, 2007]

This is a very large centrifugal compressor; it showed high vibrations on the mechanical test in late 1992. The vibrations exceeded the test limit of 1.79 mils (5th edition), and were mostly at one times running speed. A study of the response report showed that the vibration was predicted to be much higher in the horizontal direction than in the vertical direction. This was due to the softness of the five-pad load on the pad bearings in the horizontal direction for this large compressor with large clearance at the bearings. The response report showed that the compressor met the lateral rotor dynamic requirements of the 5^{th} Edition.

A study with 5-pad load between the pads bearings showed that the predicted response to unbalance was much lower, since there was much more stiffness in the horizontal direction than with the load on the pad bearing. The bearings were clocked so that they were loaded between the pads and even at trip speed there was only .76 mils and the compressor shipped.

See Figures 15 and 16, Bode plots for load on vs. load between the pads and Figures 17 and 18, the critical speed maps for load on and load between the pads. Figure 15 did show a sufficient margin of the, horizontal speed above the maximum continuous speed. However, the vibration is much higher with the load on vs. with the load between the pads. Both are with the same amount of unbalance.

Case History 2 – A High-pressure CO_2 Compressor [Memmott(1990, 2010)]

This is a 10-stage back-to-back compressor with a final discharge pressure of 146 barA (2,115 psia). The casing end seals are labyrinth seals. Toothed labyrinths and no swirl brakes are used at the impeller eyes and division wall. It was unstable with non-damper bearings and no shunt holes at the division wall when started in the field in 1974. A field upgrade was made to apply squeeze-film dampers and shunt holes at the division wall. It was stable with those parts. Multiple other such compressors were built for the same application and they were stable with damper bearings and shunt holes at division wall. See Figure 19 for sketches of squeeze-film damper bearings. For this compressor, the o-ring centered type was applied. This OEM has applied damper bearings in more than 800 compressors, with half-centered by o-rings and the other half centered by a mechanical spring. See Figure 20 for the shunt holes and Figure 21 for the stability analysis.

Case History 3 – High-pressure Hydrogen Recycle Compressor [Memmott, 2007]

This is a 185 barA (2,667 psia) discharge hydrogen recycle compressor in a refinery. The original installation was in 1971 and had two impellers, oil-film ring seals, taper land sleeve type bearings, stationary toothed labyrinth at balance piston, and no swirl brakes (it was years later before swirl brakes were invented). In 2001, gas seals were installed because of problems with oil seals and to reduce maintenance costs. Also, there was a change to tilt pad non-damper bearings. Then there were vibration problems showing the first critical frequency and they were related to surge. In 2003, a surge control system was installed. In 2005, at the request of the client, the compressor was replaced with a stub out compressor with three new, more efficient impellers and higher impeller bore. Tilt pad damper bearings were used and a hole pattern balance piston seal with swirl brakes was also used. The compressor has been running well.

See Figure 22 for the log dec vs. speed for various bearing and seal configurations with the old shaft and Figure 23 for the critical frequency vs. speed for the old shaft with oil seals and with gas seals with the original bearings. Figures 22 and 23 show the half-frequency whirl with the original sleeve bearings and oil seals and the log dec with the original sleeve bearings and the gas seals. This is the type of analysis that should be done with sleeve bearings or ring-type oil-film seals. See Figure 24 for the log dec vs. cross-coupled stiffness for the new configuration with tilt pad damper bearings vs. tilt pad non-damper bearings.

Case History 4- Classical Sleeve Bearing Instability in a "Stiff Shaft" Overhung Compressor [DeSantiago & Memmott(2007)]

This was an aerodynamic revamp of a rigid rotor overhung compressor. The flexibility ratio was approximately 0.4, i.e., the rigid bearing first was approximately 2.5 times the maximum continuous speed. After the revamp there was an oil whirl problem. On the impeller end there was a three-lobe sleeve bearing combined with a ring-type oil-film seal, and on the coupling end there was a two-pad sleeve bearing and the compressor was unstable (oil-whirl). It was stabilized by installing tilt pad bearings on both ends and on the impeller end the tilt pads were fitted into the oil seal (somewhat like a tilt pad seal).

See Figure 25 for the instability with the sleeve bearings after the revamp; figure 26 for the analysis of the instability problem; and figure 27 for the clean spectrum plot after the upgrade to tilt pad bearings.

Case History 5 –Rotordynamics and the Journal Bearing Selection for the Revamp of a Wet Gas Centrifugal Compressor [Memmott and Buckvich, 2008]

Three wet gas centrifugal compressors at a refinery were revamped. Two low-pressure barrel-type compressors had been manufactured by this OEM. One higher pressure horizontally split compressor was manufactured by another OEM. The gas from the low-pressure compressors is combined and cooled before it enters the high-pressure compressor. Each compressor was driven by a steam turbine, which also were revamped. New impellers were manufactured for each compressor.

The aerodynamic internals were completely changed out to enhance aerodynamic performance. The goal of this revamp was to increase the throughput flow of the compressor system. This was done using higher flow DATUM® impellers and internals in both casings. The overall efficiency of the system also increased as a result of the new internals.

All compressors had dry gas casing end seals. The low-pressure compressors kept their tilt pad bearings. The high-pressure compressor bearings were changed out from pressure dam to tilt pad bearings (Figure 28). There was no shop testing of the revamped compressors.

Figure 29 shows the stability results with the original pressure dam bearings vs. with tilt pad bearings. Log dec is plotted vs. speed for both and frequency is plotted vs. speed for the pressure dam bearing. With the pressure dam bearings: instability is predicted at MCOS as the log dec < 0 and when NC1 = half the speed of the rotor it goes unstable. Center pivot bearings were used for the tilt pads. After careful application of the rotordynamic considerations during the revamp build, this wet gas compressor has run well with vibration levels typically at 0.5 mils or less with a maximum vibration of about one mil upon start-up.

Case History 6 – Oil-Film Ring Seals vs. Tilt Pad Seals – [Memmott (1990) and Marshall et al (1993)]

This was a five-stage straight-through natural gas compressor on a platform in the North Sea. The discharge pressure was 133 barA (1,935 psia). It was unstable with triple breakdown ring seals and non-damper bearings and no shunts at the balance piston in field. It was upgraded to double breakdown tilt pad seals, damper bearings and shunt holes at balance piston. Then it was stable, and the preponderance of the analytical evidence was that the most effect was from the tilt pad seals. All the labyrinths were toothed labys on the stator and there were no swirl brakes.

Figure 30 shows the triple breakdown ring seal and a waterfall plot showing the instability with the first natural frequency showing up. Figure 31 shows the calculated log dec vs. speed for the triple breakdown ring seal vs. with the double breakdown tilt pad seal. The log dec of the compressor with the ring seal drops very rapidly with speed and is almost zero at the maximum continuous speed (is sensitive to speed). The triple breakdown seal acts as a very long sleeve bearing. The log dec of the compressor with the tilt pads seals does not drop rapidly with speed and is near 0.5 at the maximum continuous speed (Figure 32). The tilt pads of the seal are inside the outer ring and control the rotor dynamics. This is a Level I analysis and does not include the labyrinths. There is no need to do a Level II analysis, and in fact no need to plot the log dec vs. aero excitation as in the Level I analysis to diagnose the problem and find the solution.

Case History 7 – No Load Oil-Film Ring Seals vs. Tilt Pad Seals – [Memmott, 2004]

This is a 2,986 PSIA (206 barA) discharge hydrogen recycle compressor in a West Coast refinery. The compressor was unstable and the instability was sensitive to speed. The compressor was built by another OEM, not the author's company. The compressor had no-load oil-film ring seals, non-damper tilt pad bearings, no swirl brakes, no hole pattern or honeycomb seal/labyrinths, and no shunt holes. The author's company recommended an upgrade to tilt pad seals. This was all that was done and the compressor was stable.

Figure 33 shows spectrum plots with the instability with the no-load oil-film ring seals and the stability with the tilt pad seals. Figure 34 shows the calculated log dec vs. speed for the no-load ring seals vs. with the tilt pad seals. The log dec of the compressor with the no-load ring seals drops very rapidly with speed and is zero or negative at the maximum continuous speed. The log dec of the compressor with the tilt pads seals does not drop rapidly with speed and is above 0.5 at the maximum continuous speed. The tilt pads of the seal are inside the outer ring and control the rotor dynamics. This is a Level I analysis and does not include the labyrinths. There is no need to do a Level II analysis and in fact no need to plot the log dec vs. aero excitation as in the Level I analysis to diagnose the problem and find the solution.

Case History 8 – A 314 bar (4540 psia) discharge natural gas injection compressor [Memmott(2004)]

This case history is about a compressor installation at the North Slope. We will discuss the high-pressure compressors, which have 301 barA (4,372 PSIA) discharge. The high-pressure compressors are of the back-to-back design. They had had tilt pad bearings without dampers and a honeycomb seal at the second section inlet since the first build.

They were revamped three times, the last time in 2003. A new case was built with new impellers for 35% higher flow. Each time at the OEM's facility full pressure and full density tests were conducted at close to full load. The second revamp involved a hydrocarbon test.

Tilt pad seals were applied during the first full-load test because of instability seen with oil-film ring seals. There were high levels of subsynchronous at 5,400 cpm with the ring seals. The oil-film ring seals were replaced with tilt pad seals and the first natural frequency was now 7,300 cpm and the compressor was stable. The compressor shipped. Shunt holes were installed at the division wall in the field as an upgrade after the initial shipment. A hole pattern seal was used at the division wall for the last revamp. The full-load test for the last revamp was completed in one day.

Figure 35 shows the original double breakdown ring seals and the final as first shipped and still in the machine double breakdown tilt pad seals. Figure 36 shows the hole pattern division wall seal that was used for the last revamp. Figure 37 shows the Level I and Level II stability analyses for the last revamp. The hole pattern seal at the division wall provides a very high log dec. Figure 38 shows plots of flexibility ratio vs. average gas density. The one on the left is with the rigid bearing first critical speed in the flex ratio, as in the API plot, and also shows the Fulton typical and worst-case line. The one on the right used the first critical speed with the tilt pad seal effects. It has a different region A and B then the API plot. The tilt pad seal raises the first critical speed and lowers the flexibility ratio. Now every data point is below or on the Fulton Typical line. Figure 39 shows the spectrum plot from the full-pressure test conducted in the OEM's facility for the last revamp. The spectrum is clean.

Case History 9 – Another High-pressure CO_2 Compressor [Memmott, 1990, 2002 and 2010]

This is a six-stage in-line compressor with a final discharge pressure of 168 barA (2,400 psia). The casing end seals are labyrinth seals. Toothed labyrinths and no swirl brakes are used at the impeller eyes and balance piston. It was unstable with non-damper bearings and no shunt holes at the balance piston when started in the field in 1986. A field upgrade was made to apply squeeze-film dampers and shunt holes at the balance. It was stable with those parts.

See Figure 40 for the API Level II stability analysis. The benefits of the damper bearings and possibly the shunt holes are evident. It shows minimum and maximum clearance should be used in the stability analysis. It shows that some other factor than the labyrinth analysis is needed in the Level II analysis and the QM equal the modal sum of the API aerodynamic cross-couplings is needed.

See Figure 41 for a collection of experience with damper bearings on the API plot. A few of these compressors did not start out with damper bearings. Case History 9 is number 2 on this plot. Case History 2 is number 1 on this plot. Case History 2 was just below the Fulton typical line, as was shown in Memmott (2002). Using the Fulton typical line in this case might not raise enough concern. See Figure 42 for a collection of experience with damper bearings on the plot of bearing span/impeller bore vs. average gas density. It also has Regions A and B like the API plot. Case History 9 (point 2) jumps up and is definitely in a region of concern. It just has a flexible shaft but is running slow. API takes care of the problem by having a cut off average gas density above which the criteria for doing a Level II analysis is more conservative than it is in Region A

Case History 10 – Undamped and Damped Analysis of Low Frequency Instability in a Revamp [Memmott, 2012]

The compressor is driven by a Constant Speed Induction Motor. It shipped in 1992 with three impellers and damper bearings and dry gas seals. The MW of the gas is 29 and the discharge is 82.9 bara (1202 psia). It was revamped in 2004 to six impellers. For the revamp, a toothed laby at the balance piston had been replaced by a straight hole pattern seal with shunt holes and swirl brakes. Also, swirl brakes had been added at the impeller eyes. After the revamp, instability showed at approximately 6% of running speed. This was believed to be due to a negative stiffness at the hole pattern balance piston seal. Convergence was introduced at the balance piston and it was stable.

The paper (Memmott, 2012) shows an interesting application of undamped critical frequency analysis to show that the low frequency seen is still the first bending natural frequency. The undamped critical frequency program was run at a fixed speed, the compressor maximum continuous speed, and a series of negative direct stiffness were entered at the balance piston hole pattern seal location. Figure 43 shows NC1(undamped)/Nmc vs. Negative Stiffness at the Balance Piston. Figure 44 shows the NC1(undamped) Mode Shape for Balance Piston Stiff = 0 and NC1/Nmc = 6 %.

STRETCHING THE RULES

Throughout the tutorial, mention has been made of possibly violating "tried and true" rules that are typically applied in new equipment applications for revamps / re-rates. Some of the rules that might be broken include: diffuser radius ratio; incidence angles; flange Mach number; impeller relative velocity ratio; inlet, volute or collector sizing; stage spacing; and many others. While each OEM has very specific rules regarding each of these, it is possible to stretch or violate each as long as the end user understands the potential impact of each violation. Typical consequences include reduced efficiency / increased horsepower, reduced operating range, and possibly higher vibration levels. However, it is possible these consequences are more than offset by the cost savings derived via the re-use of the components from an existing machine and avoiding changing the case, piping and other auxiliaries at the end user's facility. In other words, the upfront savings and short "time to money" of the revamp might outweigh the long-term costs of the reduced compressor performance.

COMMUNICATIONS

The above discussion regarding stretching the rules provides a good segue into a key prerequisite to the success of a revamp (or any compressor proposal for that matter) – communication! It is absolutely critical that the OEM and end user or the company performing the re-rate and the end user have an open, honest dialog on the entire matter. The end user *must* provide all details regarding the newly required operating conditions and the motivation for the revamp. Further, if the OEM is not performing the revamp, the end user must provide all available details on the compressor geometry, the control system, the current performance map, and any available field performance data. The supplier performing the revamp must provide the end user with a detailed understanding of any design rules that were compromised to fit new internals in the casing and the potential consequences of these. They also must provide the end user with a detailed performance prediction for the machine after the revamp. It is also possible that a performance guarantee will be proposed. Again, this must be negotiated between the end user and the company doing the revamp (OEM or otherwise). However, the guarantees are typically not as tight as for a new compressor. For example, the typical API guarantee on new equipment is ±4% on horsepower but for revamps, the tolerance can be ±6% or higher.

Part of the discussion must also center on determining how success will be defined. That is, what will determine if the revamp was successful? Some of the possible options include:

- Using the end users metrics for throughput and energy consumption relative to expectations. This implies that both organizations have confidence in the accuracy of the end user's measurements before and after the revamp.
- Conducting a field test on the revamped compressor.

Here, the two organizations must agree on the instrumentation, flow measurements, etc. that will be used to establish the compressor performance. They also must map out how many test points will be taken, the flow conditions for each point, any variation in speed that will be investigated, etc. Quite often, either the end user or the revamping organization will require that the revamping company have field engineers on-site to oversee the testing.

- Perform an in-house test at the OEM or revamp company's facilities. This will require that the compressor casing be shipped back to the OEM or revamp company's facilities. However, if the revamp is considered to be high risk and absolute assurances are required that the revamped compressor will perform as predicted, the in-house test might warrant the extra expense associated with shipping the case. Again, the OEM and the revamping company will have to agree on the test procedure, instrumentation, etc.

The importance of "wide-open," honest communication cannot be over-stated. It is essential for developing the trust that must exist between the end user and the company performing the revamp.

OEMs v. OTHERS

This tutorial has, until now, avoided the discussion of whom the end user should engage to actually revamp the compressor. As a general rule, the OEM is going to have the most knowledge of the existing compressor, i.e., case and component dimensions; design philosophy of components; performance characteristics; operating history with similar units; potential pitfalls associated with stretching the rules; and such. The OEM also has a vested interest in keeping the equipment at the end user's facility *and* in keeping competitors or a third-party vendor from having access to the compressor, albeit older technology. Therefore, the end user would be wise to approach the OEM first when considering a revamp. Should the OEM show no interest or possibly no longer exist, the end user might consider approaching a different OEM who specializes in centrifugal compressors for similar services. The end user might also engage any number of companies that specialize in revamping / re-rating compressors regardless of who the OEM happens to be. Regardless, if the enlisted company is not the original OEM, new considerations arise. Primary among these is that the "non-OEM" must find some way to obtain the geometry of the existing machine. As it is highly doubtful that the OEM would be willing or able to provide these, the "non-OEM" must measure the existing components and possibly reverse engineer the aerodynamic flow path and mechanical components.

With the advent of coordinate measuring system (e.g., ROMER arms, light scanning systems, laser scanning systems), the "non-OEM" can gather extensive amounts of geometric data that can then be drawn into a computer-aided drafting (CAD) system so that reverse-engineered drawings of the existing components can be developed. The "non-OEM" can then determine if the new internals will fit or whether some compromises on standard design guidelines (i.e., diffuser radius ratios, stage spacing) must be made. The "non-OEM" might even use these reverse-engineered models to conduct aerodynamic and/or stress analyses to gain a better understanding of the existing machine before developing the revamp internals. For example, the "non-OEM" could use the case measurements to conduct a finite element analysis (FEA) to ensure the casing can withstand the new operating conditions. Likewise, a computational fluid dynamics (CFD) could be completed on an inlet or discharge volute that is integral with the casing to determine if it will have any impact on the compressor performance. In doing these types of analyses the "non-OEM" can give the end user greater confidence that the re-rate internals will perform as expected. In short, it is possible for a "non-OEM" to measure sufficient geometry and conduct any necessary analyses to overcome the lack of drawing of the existing compressor components. The remaining risk to the end user, then, is whether the "non-OEM" has sufficient insight into or expertise on the potential pitfalls that might arise during the revamp of the compressor (i.e., violating the OEM's design guidelines). The end users only recourse in this regard is to ask for demonstrated experience and success in revamping similar compressors for similar services and operating conditions.

The final decision will often come down to cost v. risk. The "non-OEM" might quote a lower price than the OEM, but the OEM has more tribal knowledge about and experience with the product. The end user ultimately must make the choice between cost and risk; unfortunately, no universally accepted guidelines exist for making that choice.

CONCLUSIONS

The tutorial provides an overview of the concerns that must be addressed when considering the revamp / re-rate of a centrifugal compressor. The motivations, cost benefits and technical considerations have been presented, along with commentary on the potential aerodynamic, mechanical, rotordynamic, and cost trade-offs that might result from the compromises that might be made. Finally, the tutorial emphasizes the need for effective, open, honest dialogue between the end user and the company performing the revamp / rerate.

As suggested in the introduction, the contents of this tutorial are somewhat biased by the experiences of the authors' company and it is possible that other OEMs, having different experiences would have differing opinions on some of the guidelines and comments offered. Still, the hope is that the reader gleans some value from the information contained herein.

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ACKNOWEDGEMENTS

The authors thank the staff and management at DRESSER-RAND for its support in the writing of this tutorial and for allowing us to publish this document.

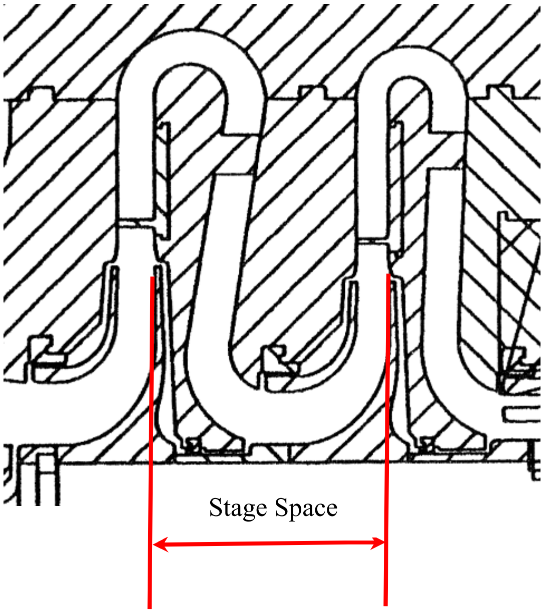


Figure 1 – Stage spacing

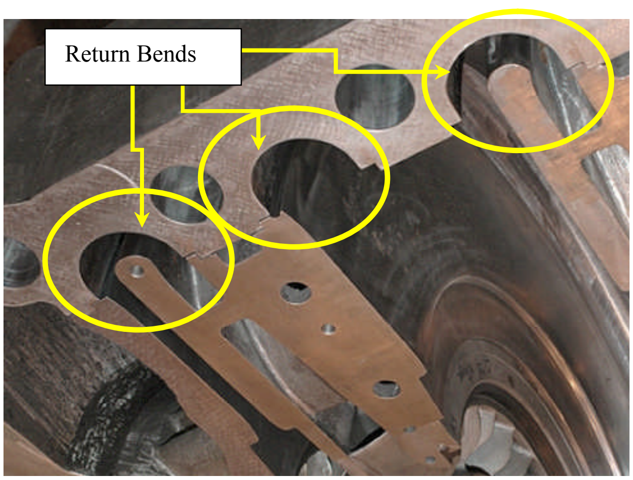


Figure 2 – Return bend in case

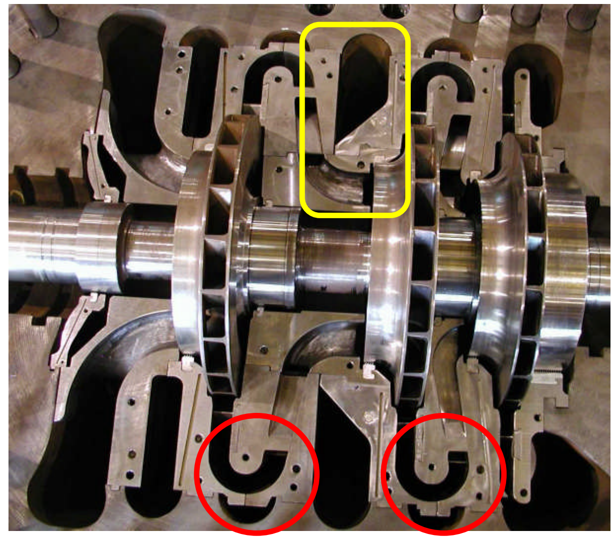


Figure 3 – New pieces to form return bend (red) – Bridge across return bend (yellow)

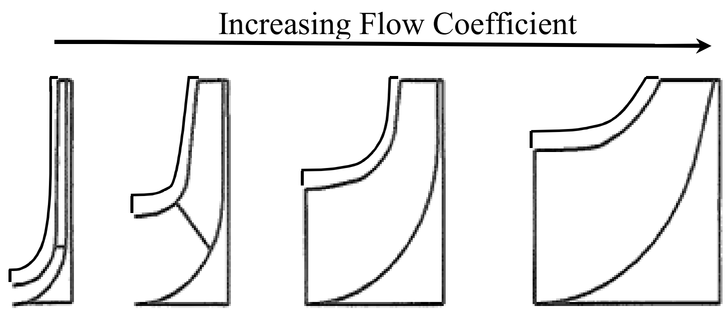


Figure 4 – Change in impeller topology with flow coefficient

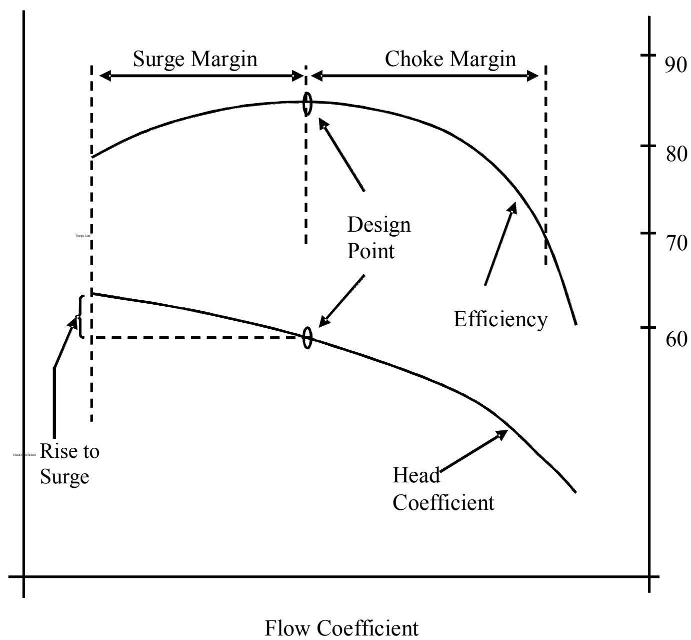


Figure 5 – Common compressor (or stage) performance map

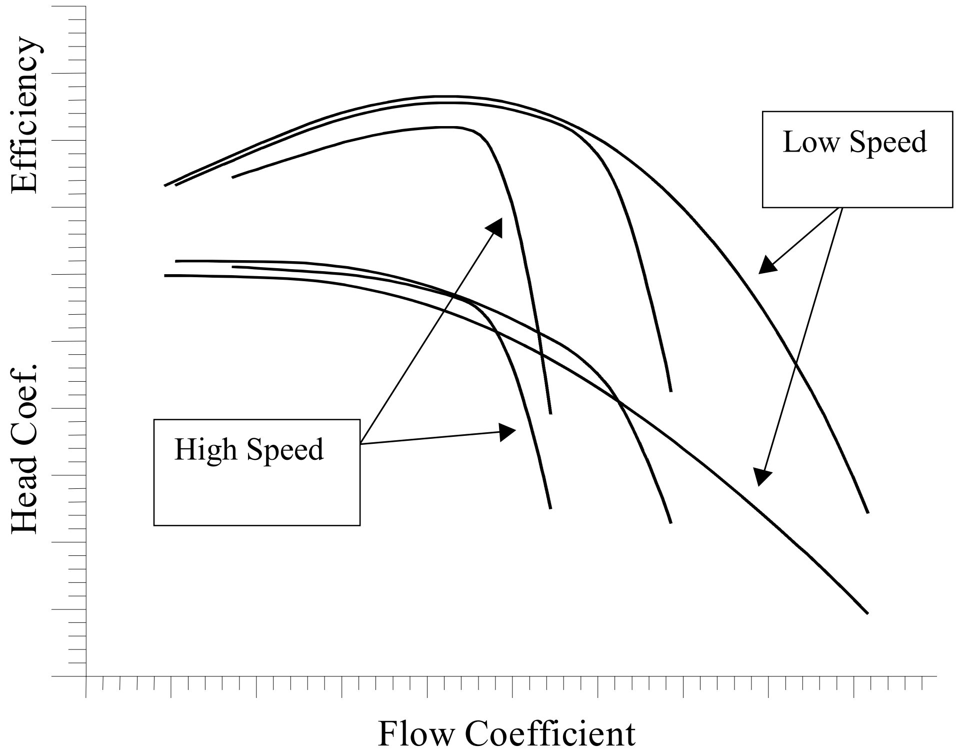


Figure 6 – Impact of changing speed on stage performance

Figure 7 – "Z"-blade for riveted impeller

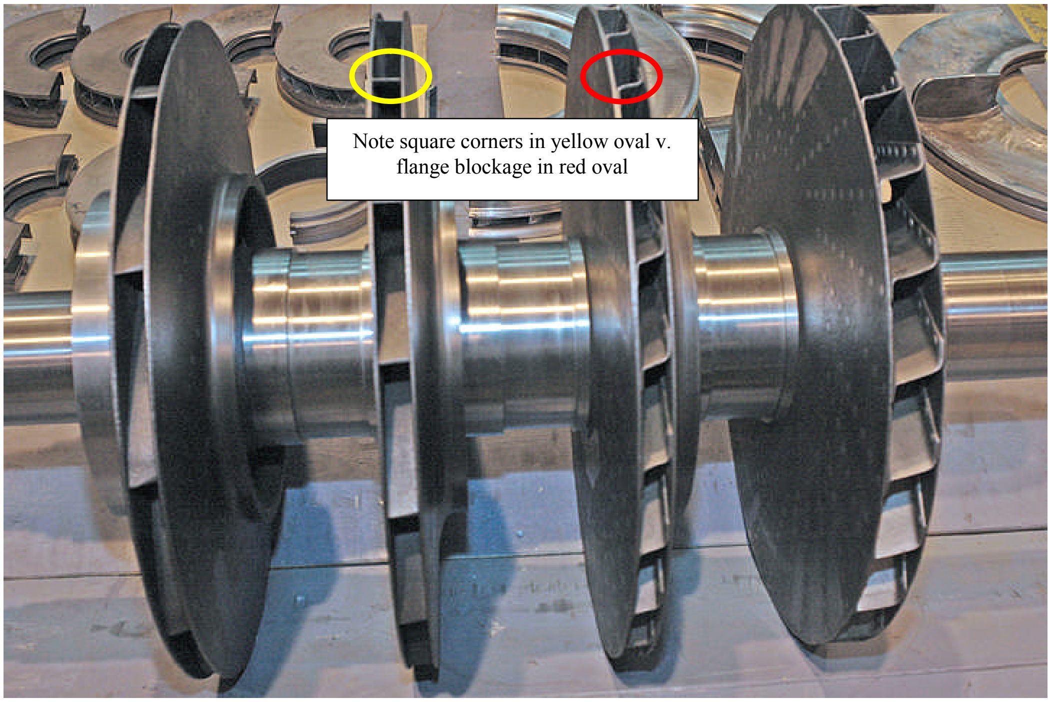


Figure 8 – Through-blade rivet v. “Z”-blade riveted – note no corner radius or fillet blockage in through-blade impeller (yellow)

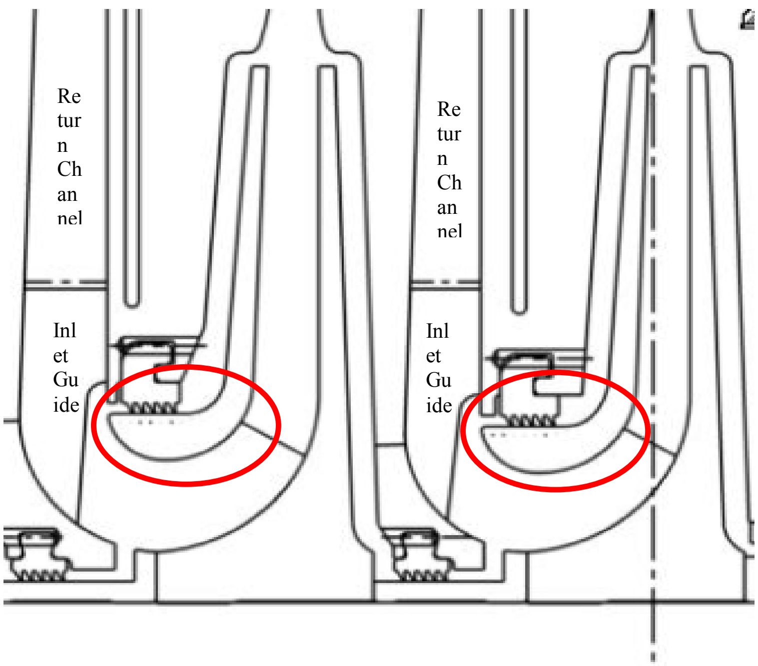


Figure 9 – Impeller covers form inlet guide shroud

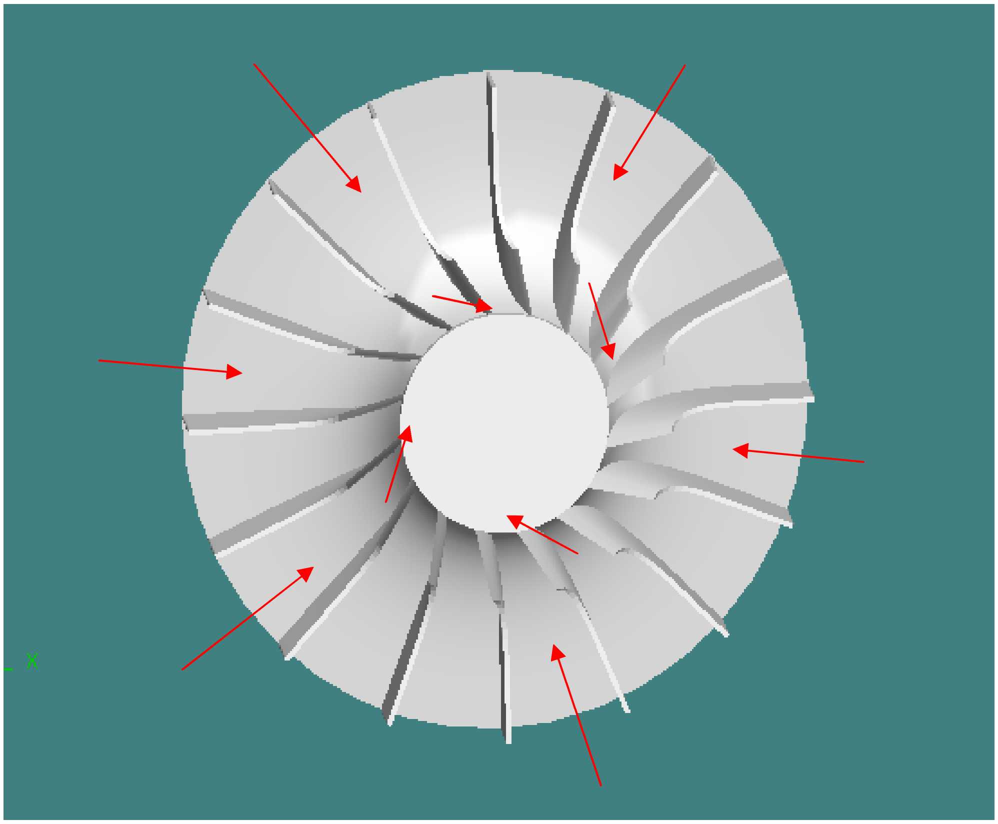


Figure 10 – Guidevane introduces tangential pre-whirl to impeller – “With” or “Against” depends on direction of impeller rotation

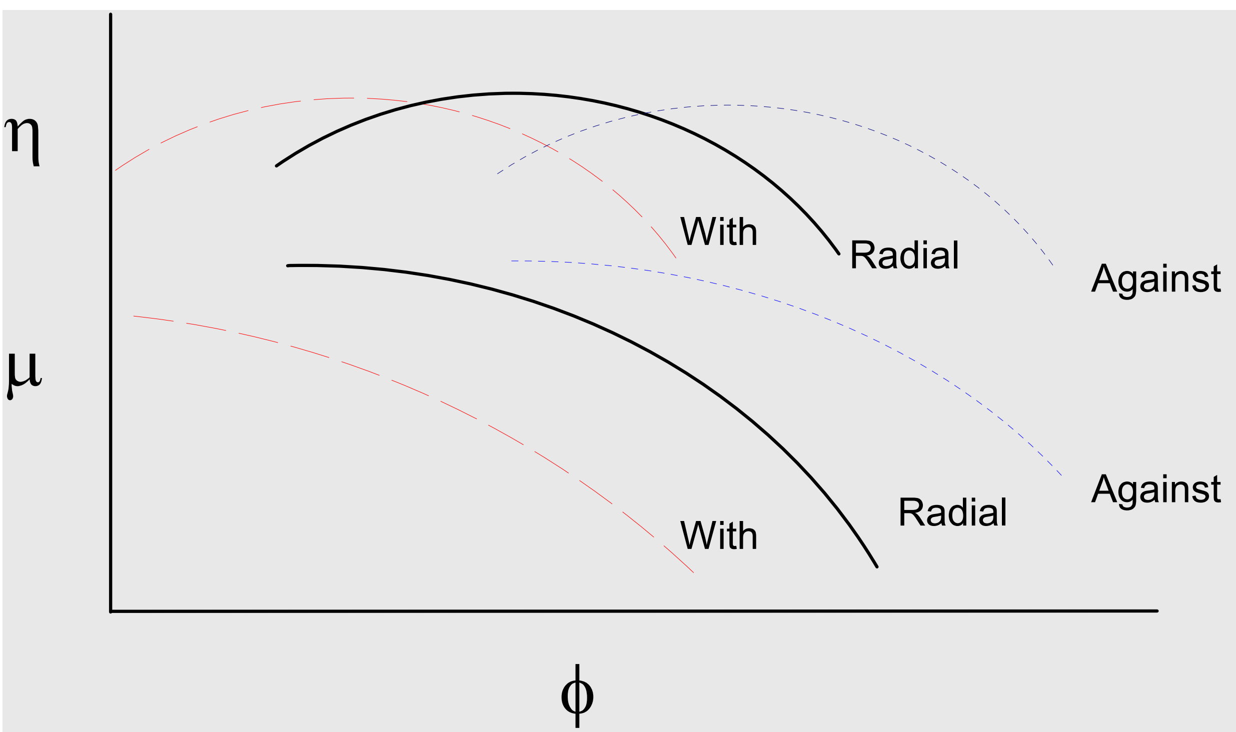


Figure 11 – Impact of adding prewhirl from inlet guide on stage performance

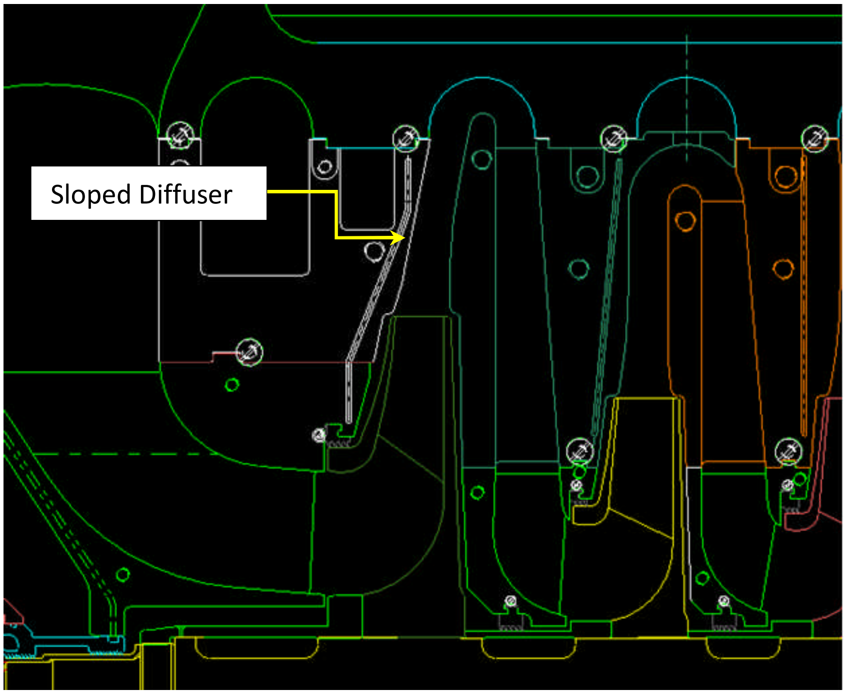


Figure 12 – Sloped diffuser to match existing return bends (also note insert piece in second stage to avoid use of existing bend)

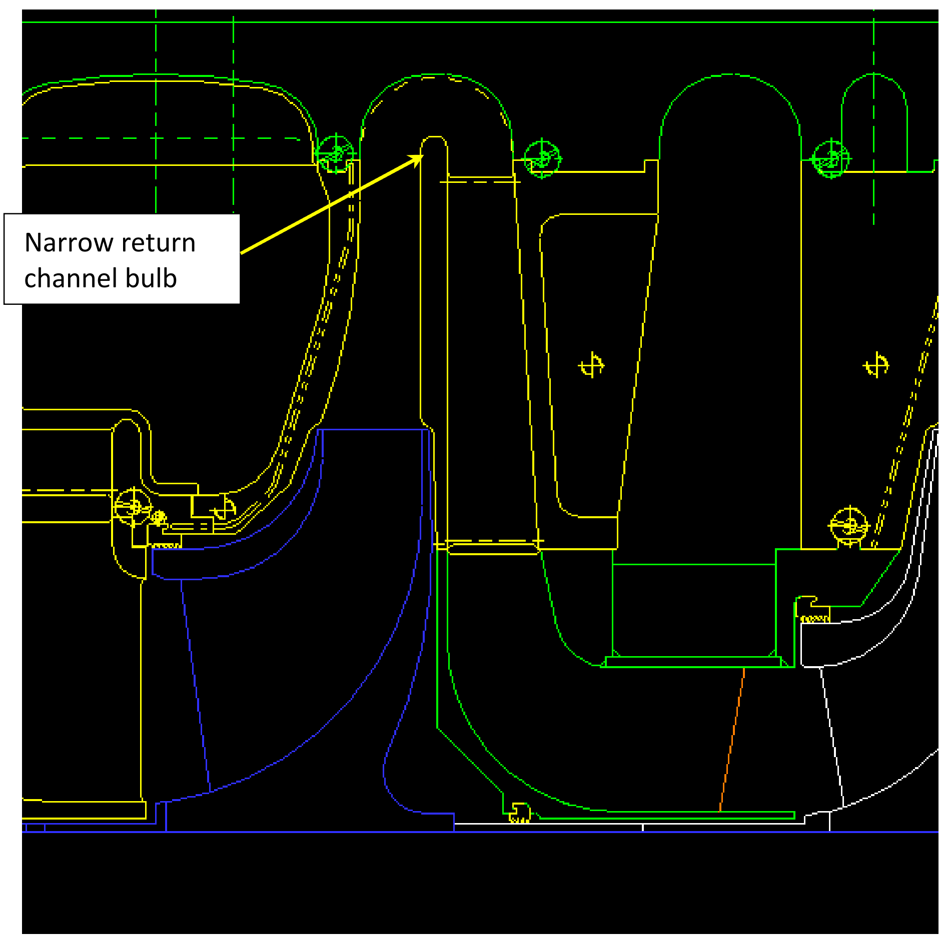


Figure 13 – Return channel with narrow bulb to fit existing return bend (also provides another example of bridging across an existing return bend)

Rotor dynamic analyses procedure:

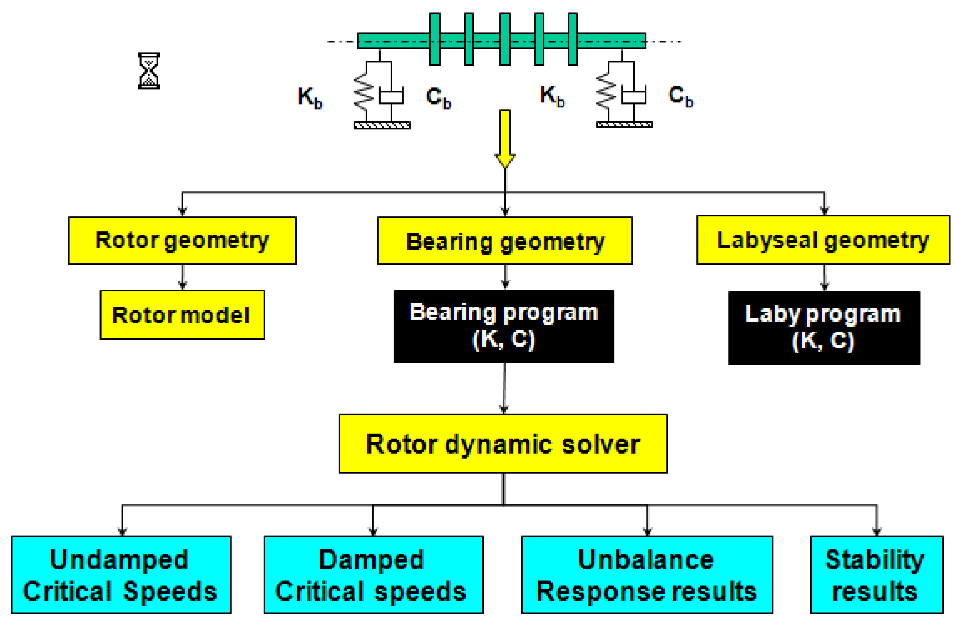


Figure 14 – Rotor dynamic Procedure Flow Chart

Plot of Amplitude vs. Speed from the the Lund Rotor Response Program - 5 Pad - Load on the Pad - A Large Compressor

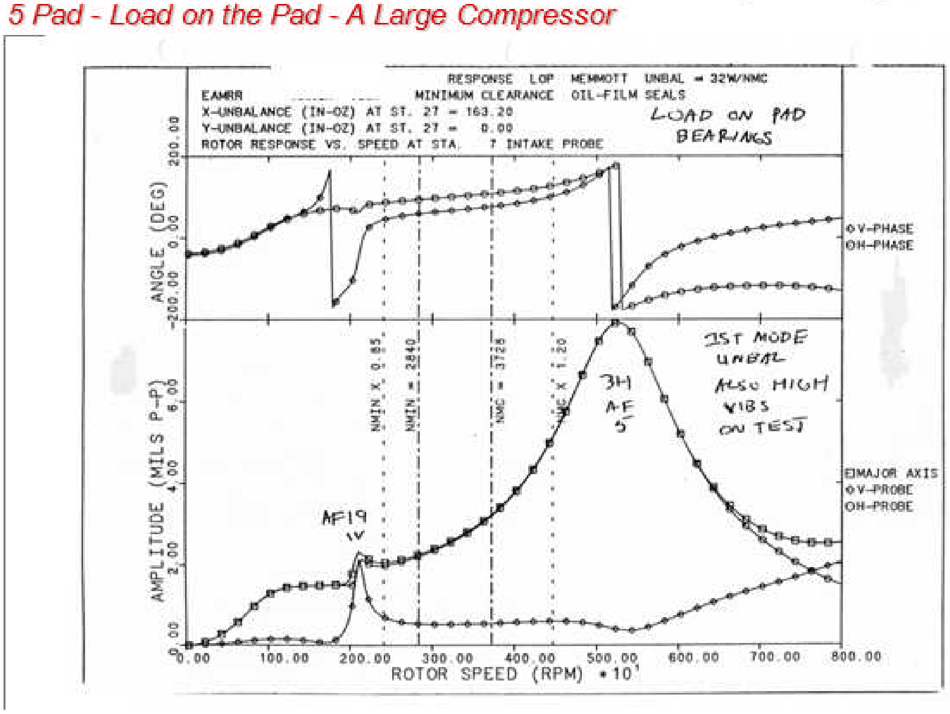


Figure 15 – Case History 1 – Bode Plot Load On Pad Bearings

Plot of Amplitude vs. Speed from the the Lund Rotor Response Program -
5 Pad - Load between the Pads - A Large Compressor

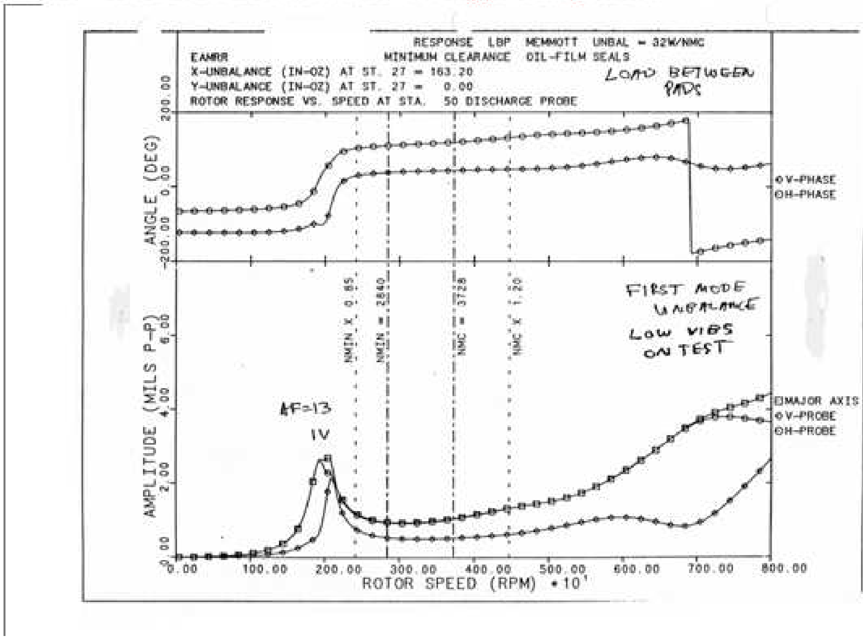


Figure 16 – Case History 1 – Bode Plot Load Between the Pads Bearings

Plot of Critical Speed Map
5 Pad - Load on the Pad - A Large Compressor

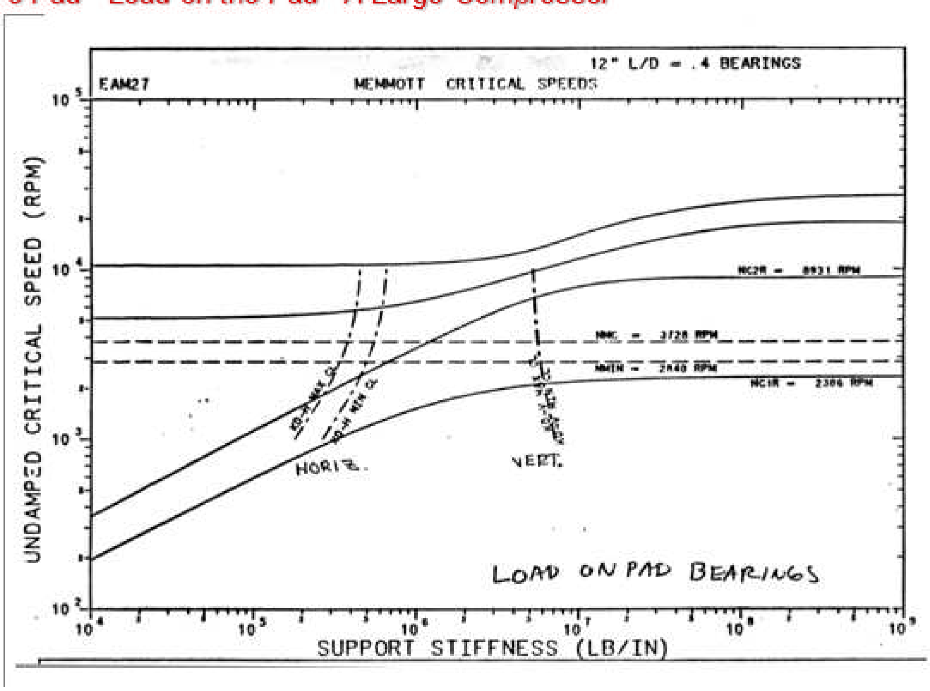


Figure 17 – Case History 1 – Critical Speed Map Load On Pad Bearings

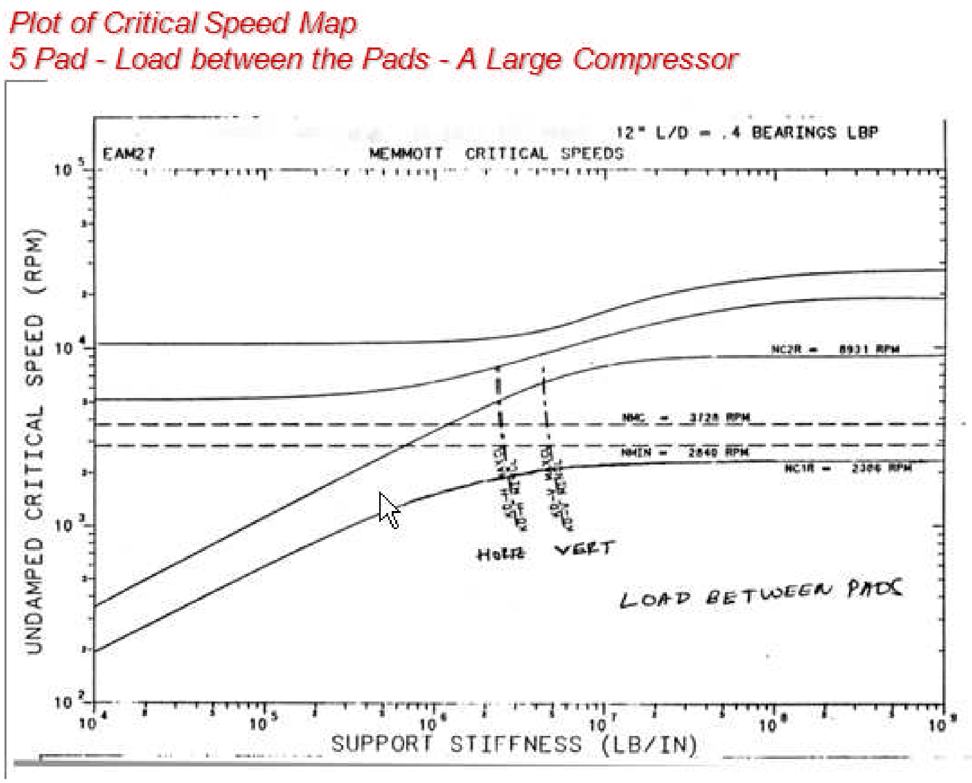


Figure 18 – Case History 1 – Critical Speed Map Load Between the Pads Bearings

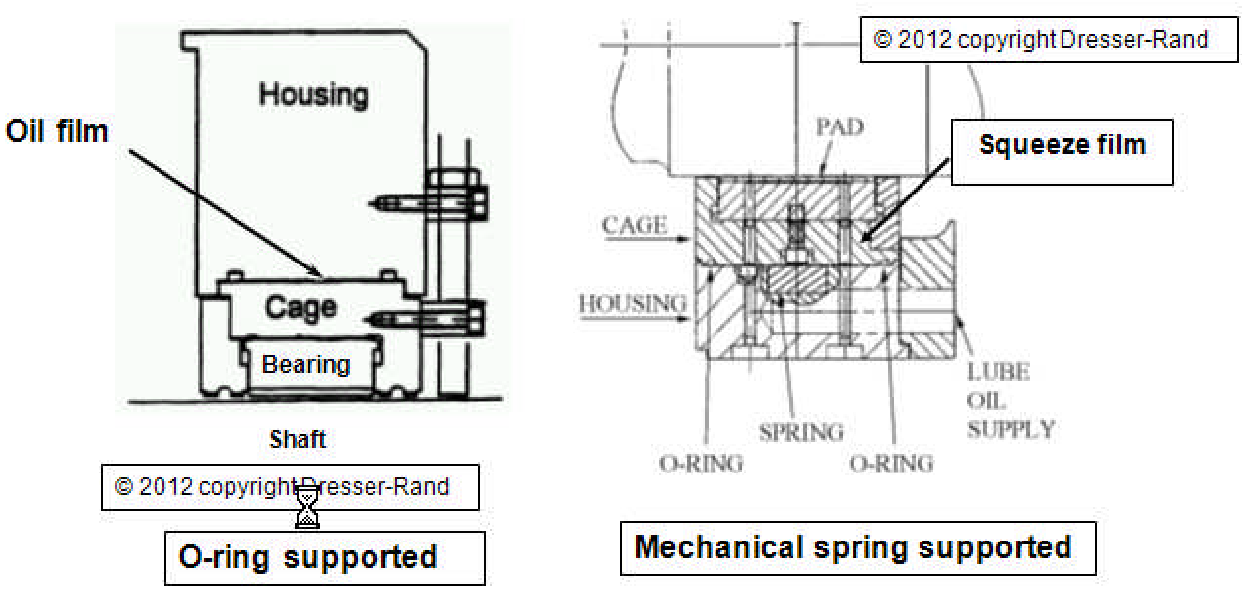


Figure 19 – Damper Bearings – For Case History 2 the o-ring centered was used

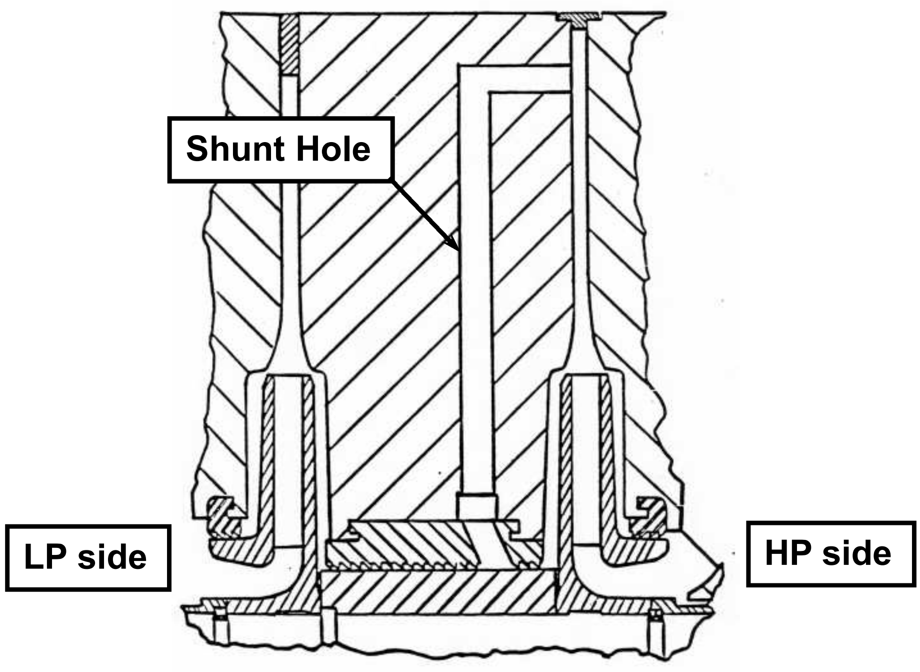


Figure 20 – Case History 2 – Shunt Hole System

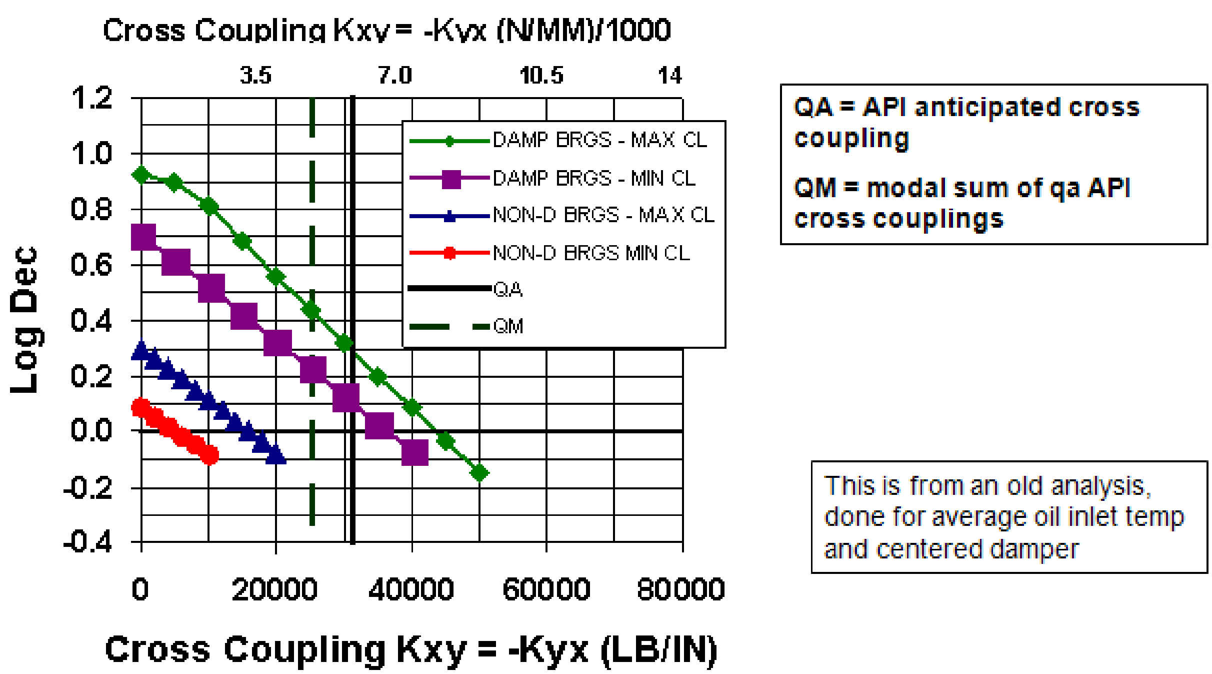


Figure 21 – Case History 2 – Log Dec vs. Cross-Coupled Stiffness at the Midspan Damper Bearings vs. Non-Damper Bearings

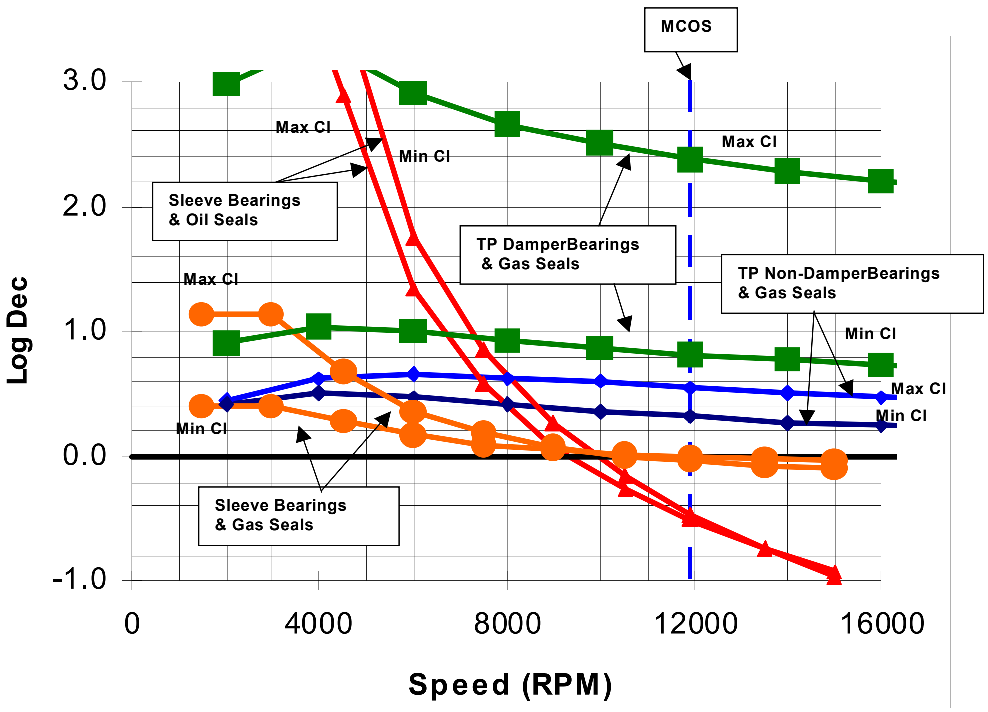


Figure 22 – Case History 3 – Log Dec Vs. Speed Original Rotor

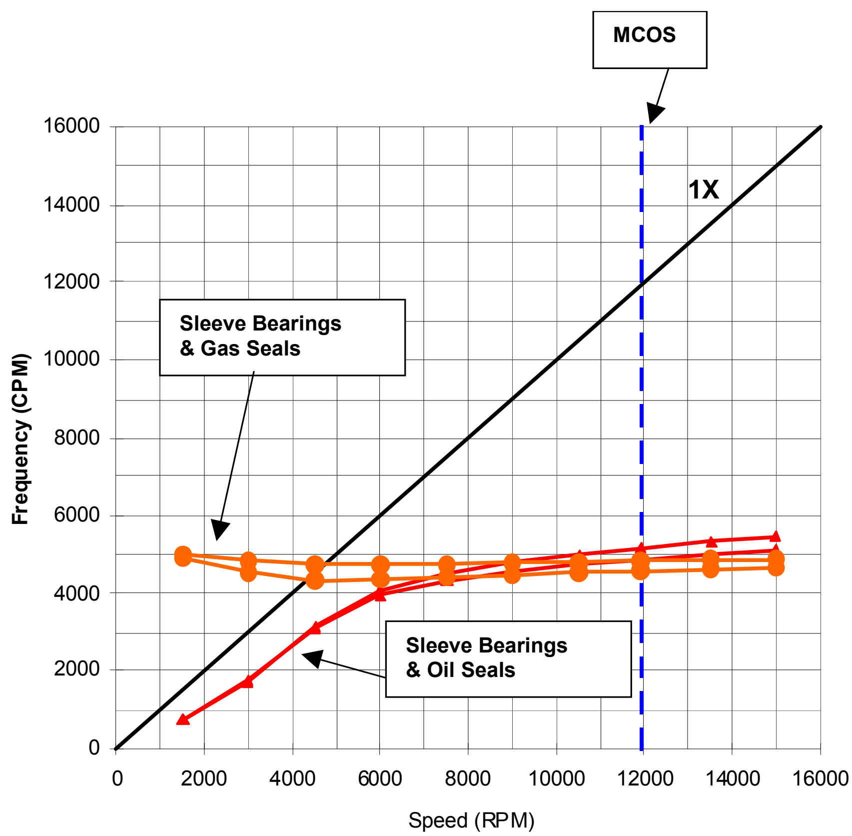


Figure 23 – Case History 3 – Frequency Vs. Speed Original Rotor with Sleeve Bearings – Gas Seals vs. Oil Seals

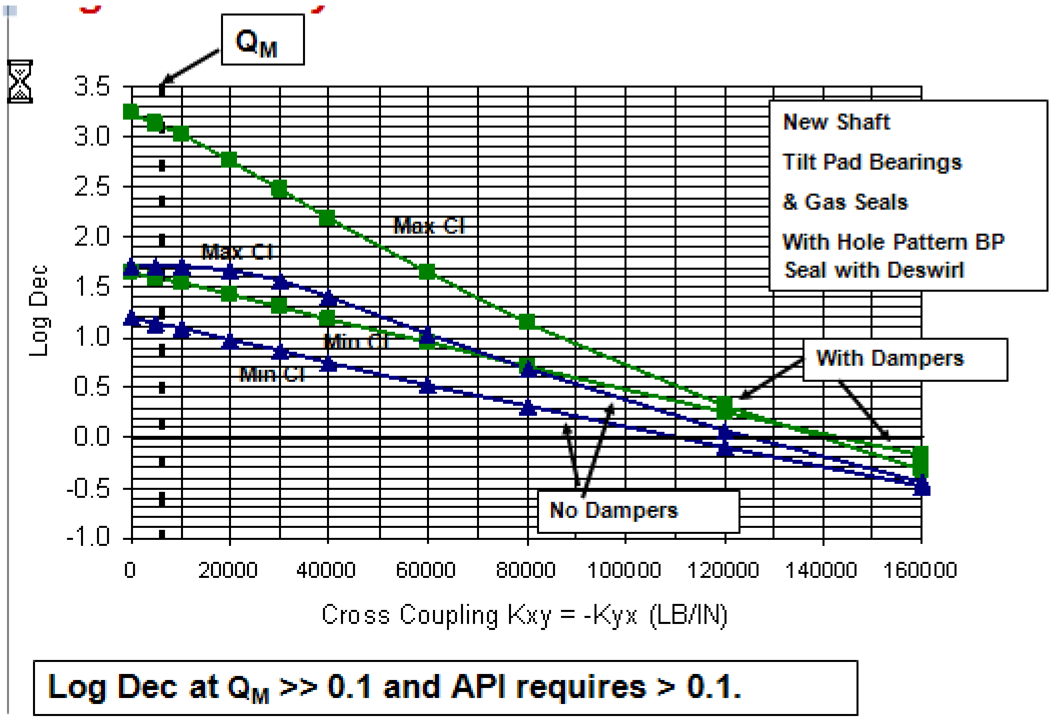


Figure 24 – Case History 3 – Log Dec vs. Cross-Coupled Stiffness at Midspan New Rotor with Tilt Pad Bearings and Gas Seals and Hole Pattern at Bal Piston Damper Bearings vs. Non-Damper Bearings

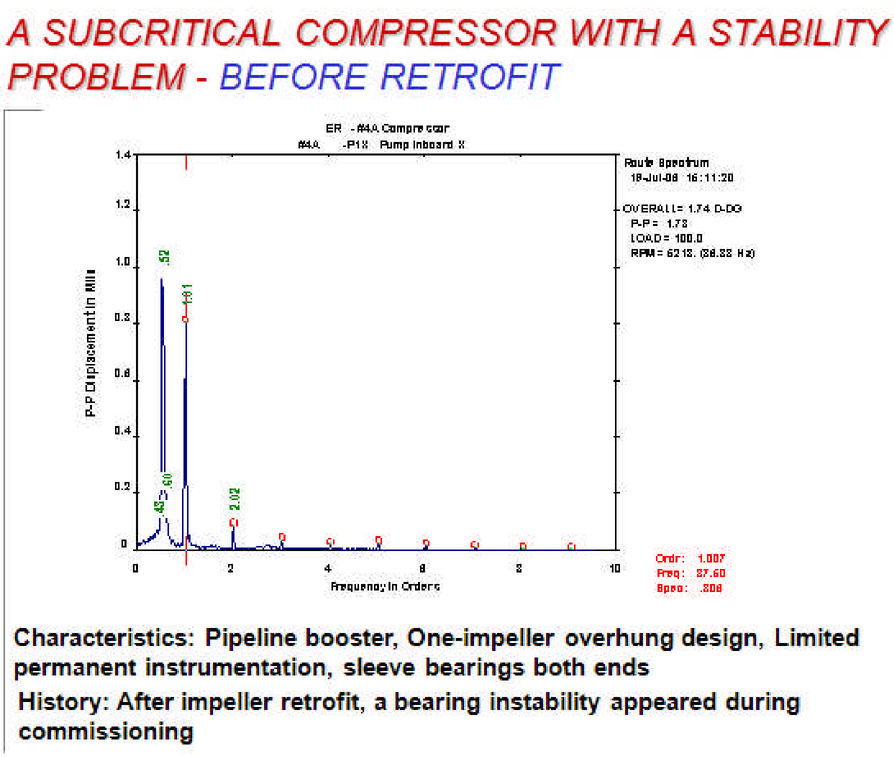


Figure 25 – Case History 4 – Instability after Impeller Retrofit Sleeve bearings both ends, impeller end bearing combined with oil seal

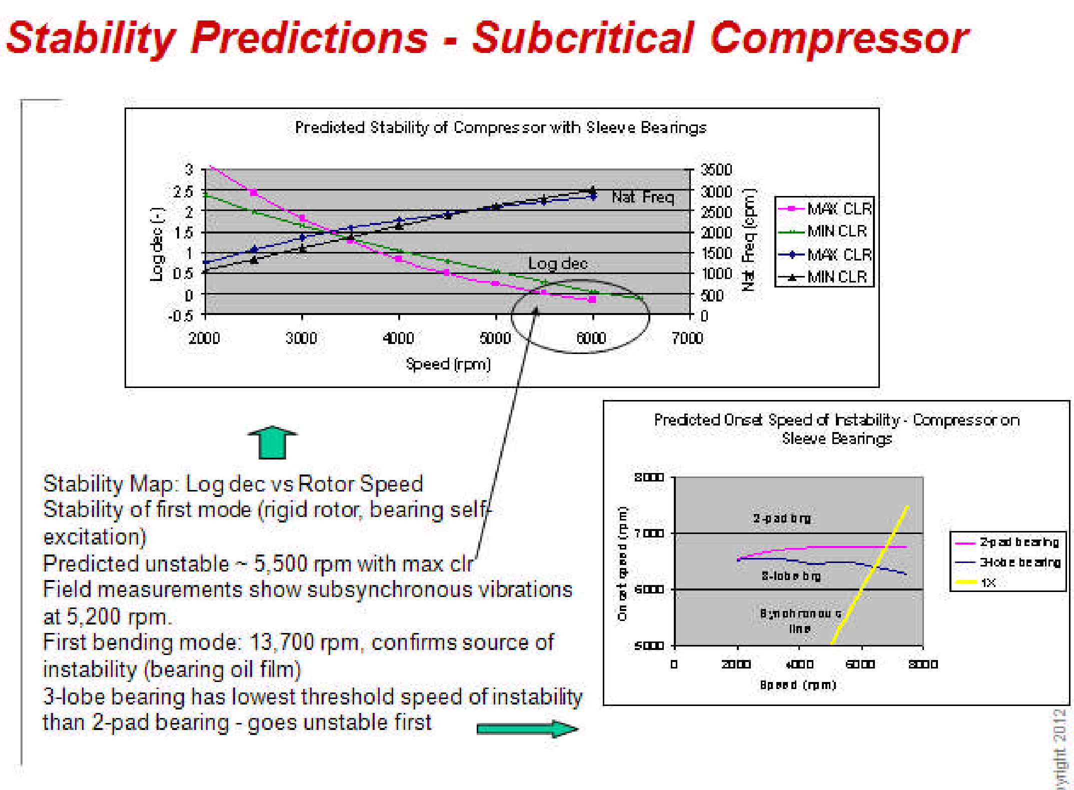


Figure 26 – Case History 4 – Stability Predictions Sleeve bearings both ends, impeller end bearing combined with oil seal

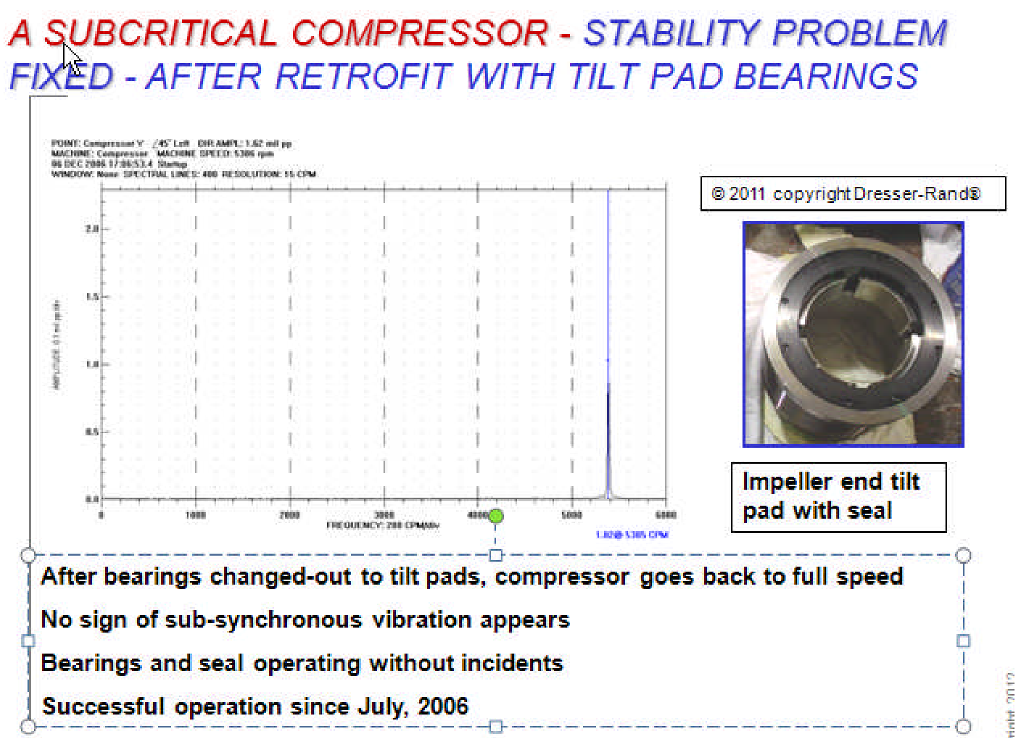


Figure 27 – Case History 4 – Stability Predictions Tilt Pad Bearings both ends, impeller end bearing combined with oil seal

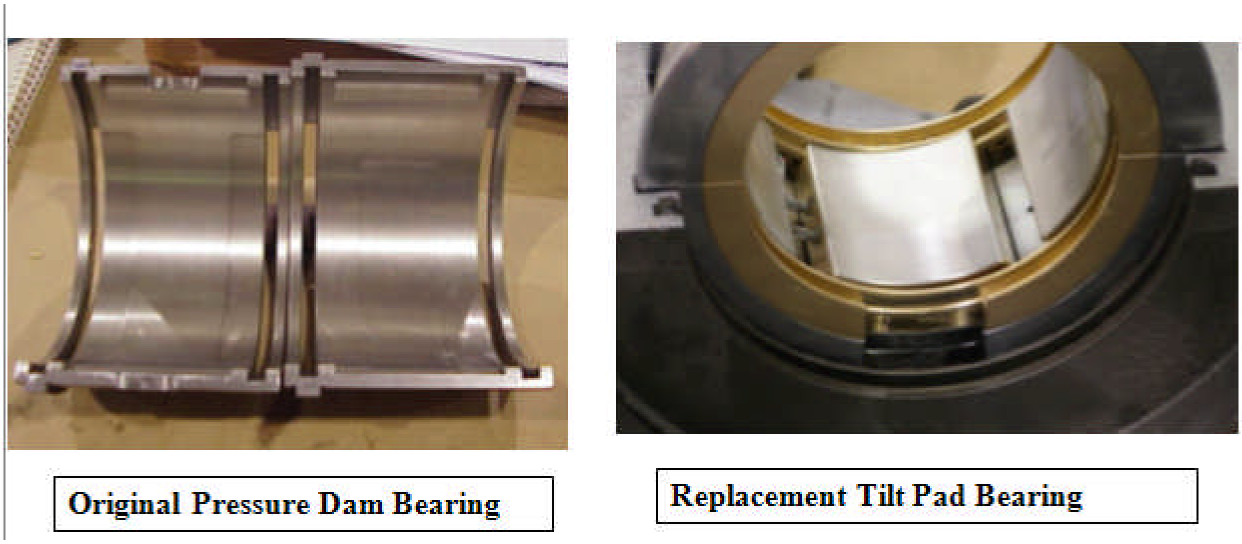
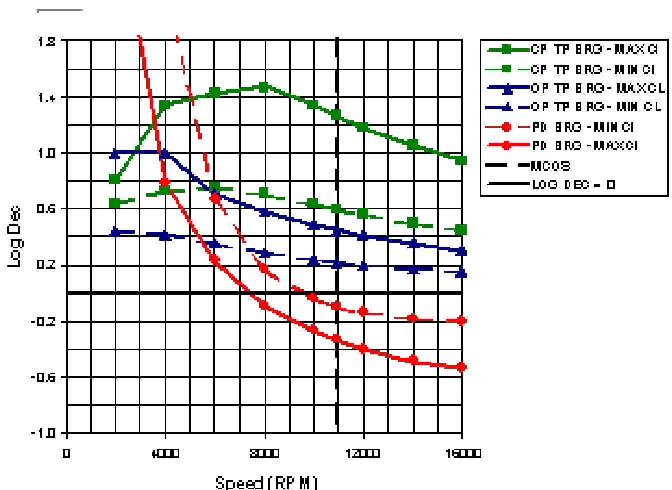


Figure 28 – Case History 5 Original Pressure Dam Bearing vs. Replacement Tilt Pad Bearing



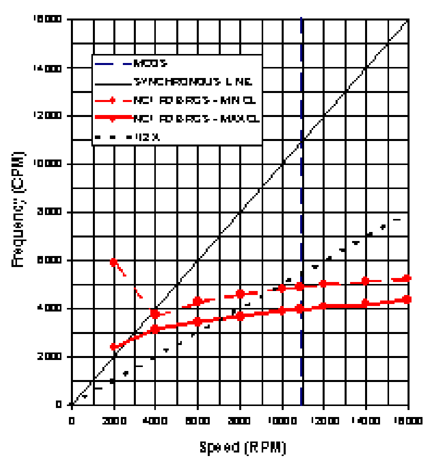


Figure 7 – Log Dec of NC1 *vs.* Speed
Tilt Pad Bearings *vs.* Pressure Dam Bearings

Figure 8 – First Critical Frequency *vs.* Speed
With Pressure Dam Bearings

With pressure dam bearings: Instability predicted at MCOS, log dec < 0 & when NC1 = half the speed of the rotor it goes unstable

Figure 29 – Case History 5 Log Dec vs. Speed Tilt Pad Vs. Pressure Dam Bearings Nc1 vs. speed with Pressure Dam Bearings

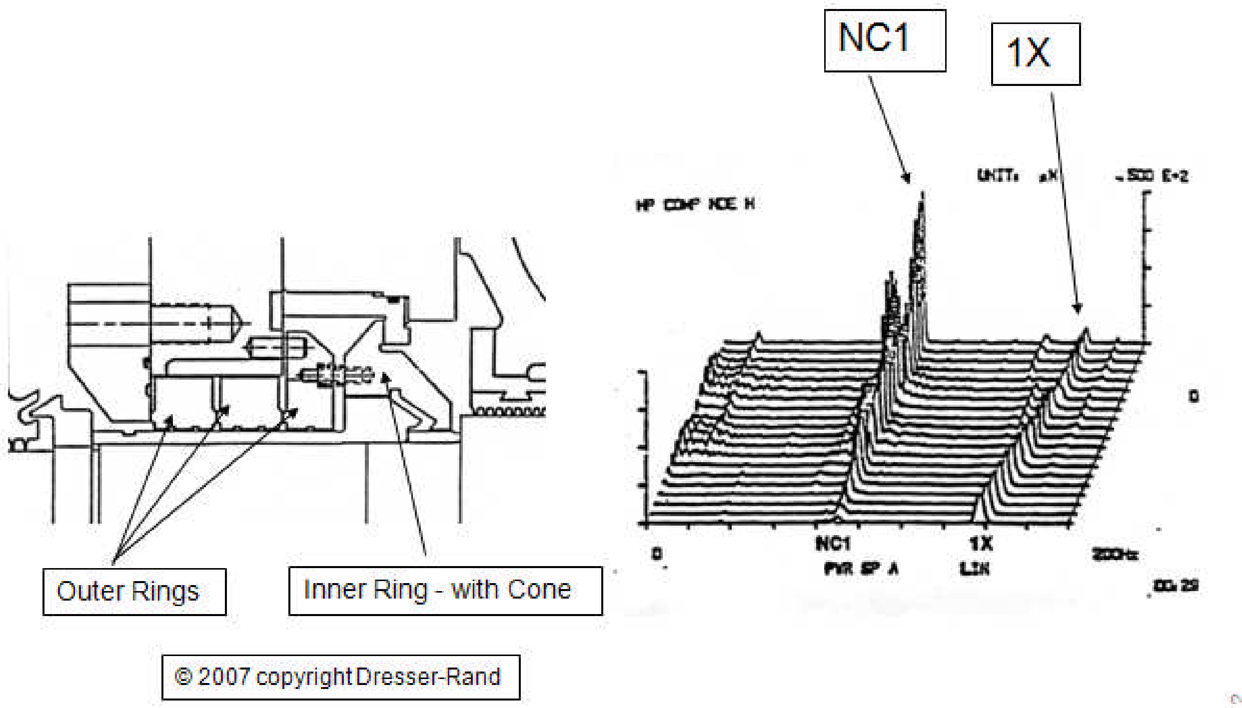


Figure 30 – Case History 6 Trip Breakdown Ring Seal with Instability

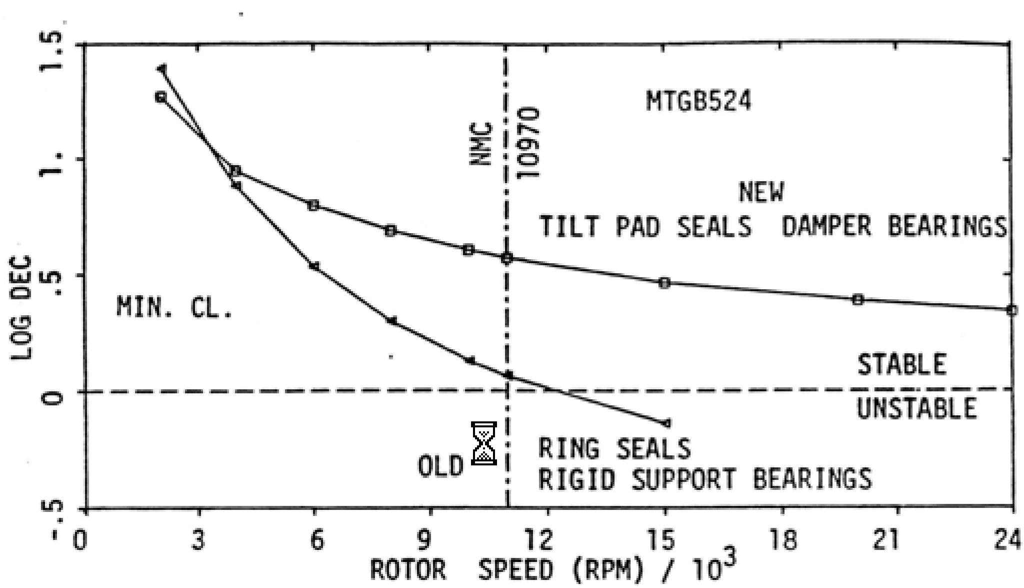


Figure 31 – Case History 6 Log Dec vs. Speed – With Oil-film Ring Seal vs. With Tilt pad Seal

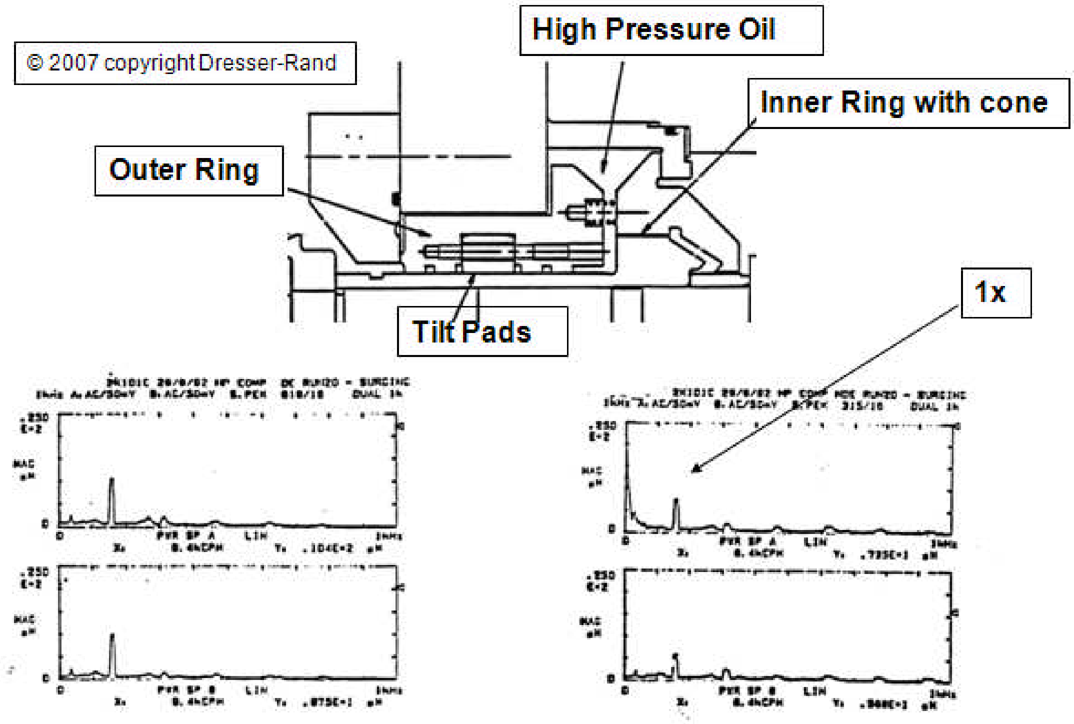


Figure 32 – Case History 6 Double Breakdown Tilt Pad Seal + Spectrum Plots Showing Stability

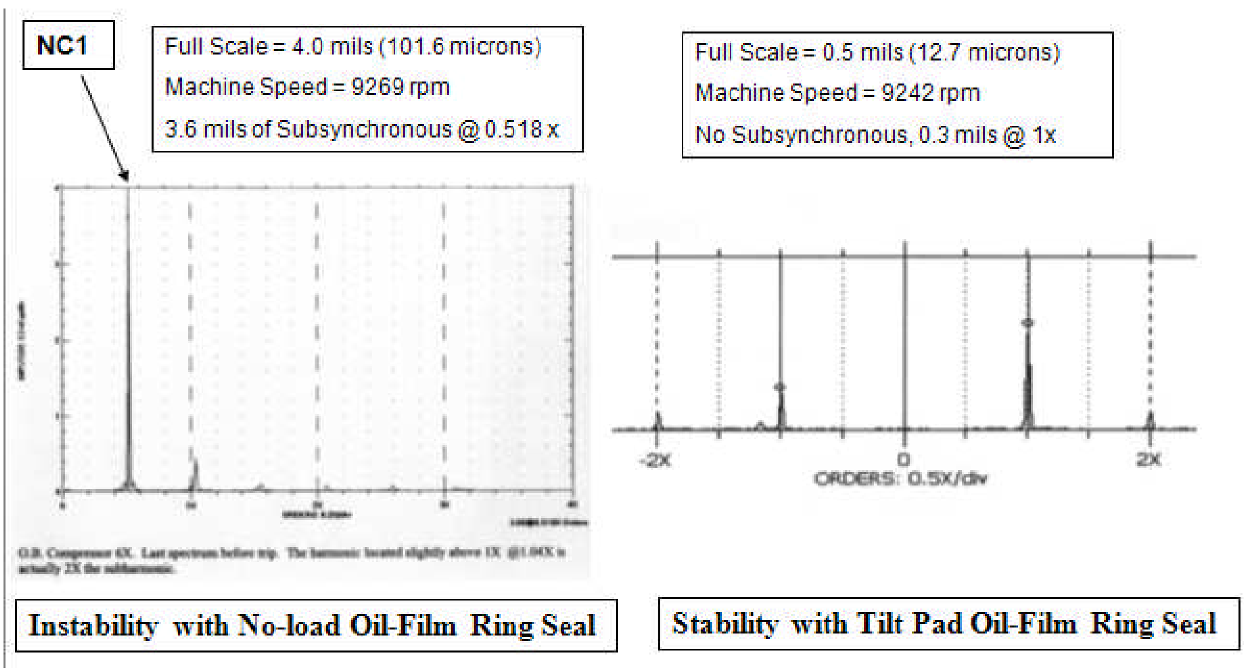


Figure 33 – Case History 7 Instability with No-load Oil-film Ring Seal – Stability with Tilt Pad Oil-Film Ring seal

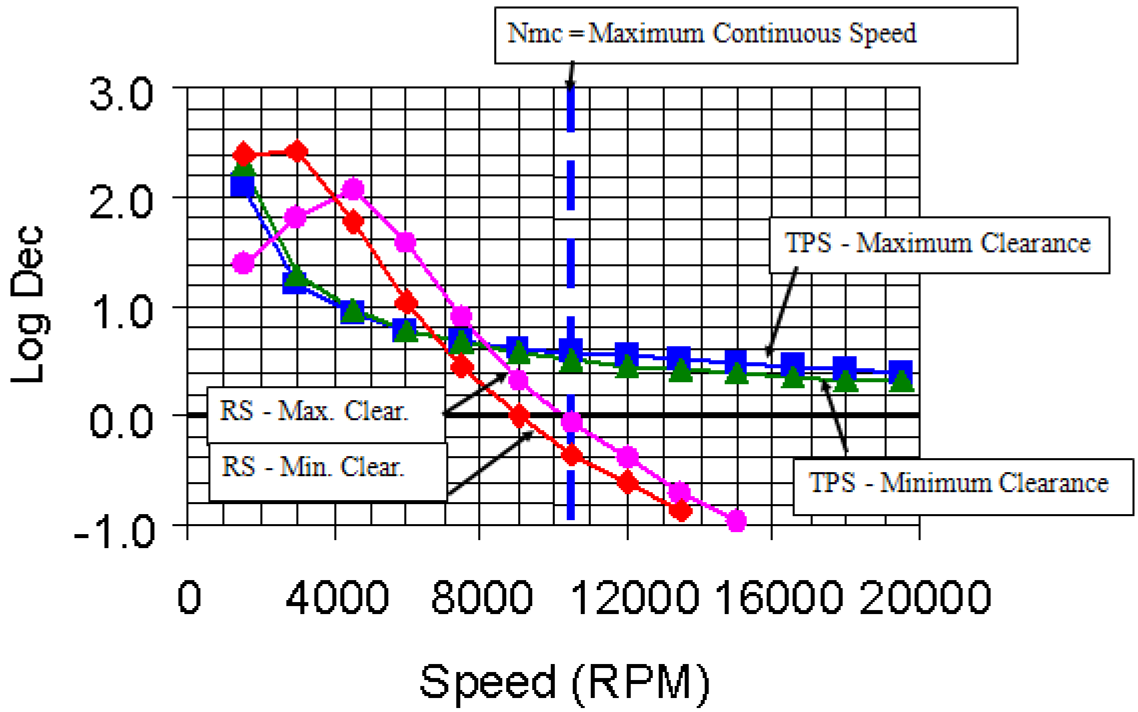


Figure 34 – Case History 7 Log Dec vs. Speed – With No-Load Oil-film Ring Seals vs. With Tilt pad Seals

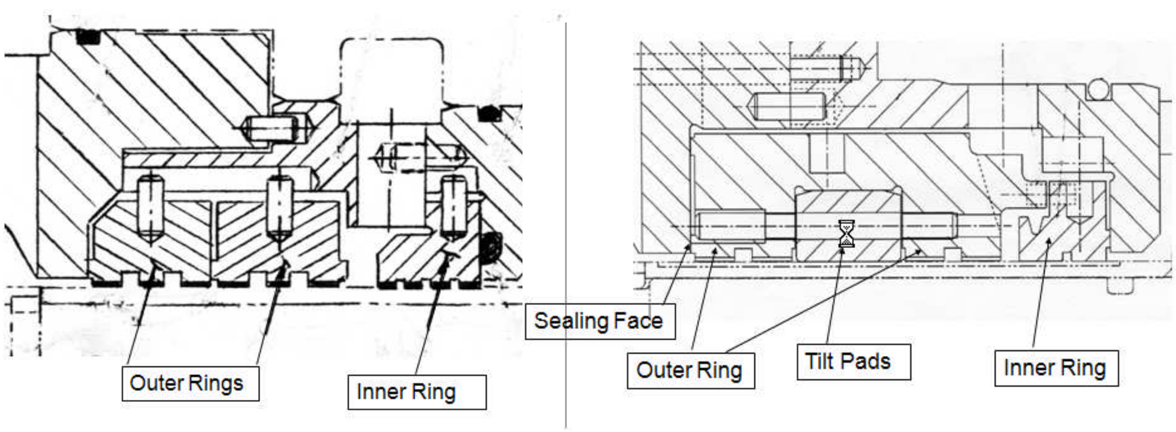


Figure 35 – Case History 8 Double Breakdown Oil-ring seal vs. Double Breakdown Tilt Pad Seal

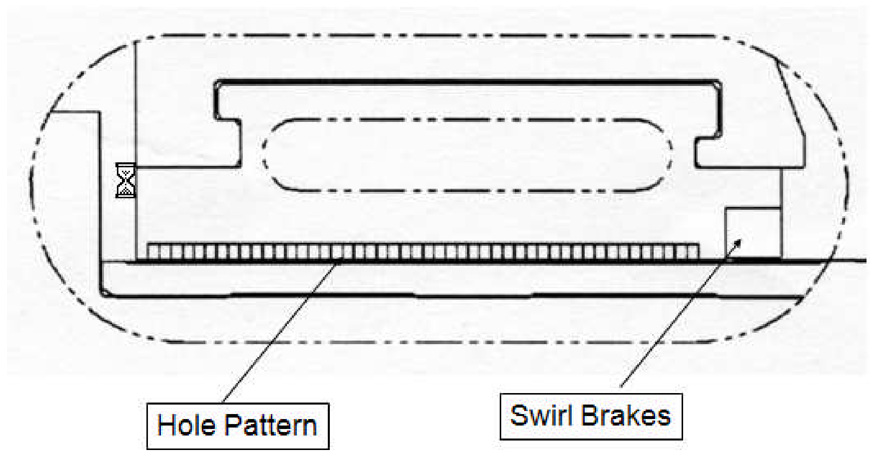


Figure 36 – Case History 8 Hole Pattern Division Wall Seal with swirl brakes (also has shunt holes)

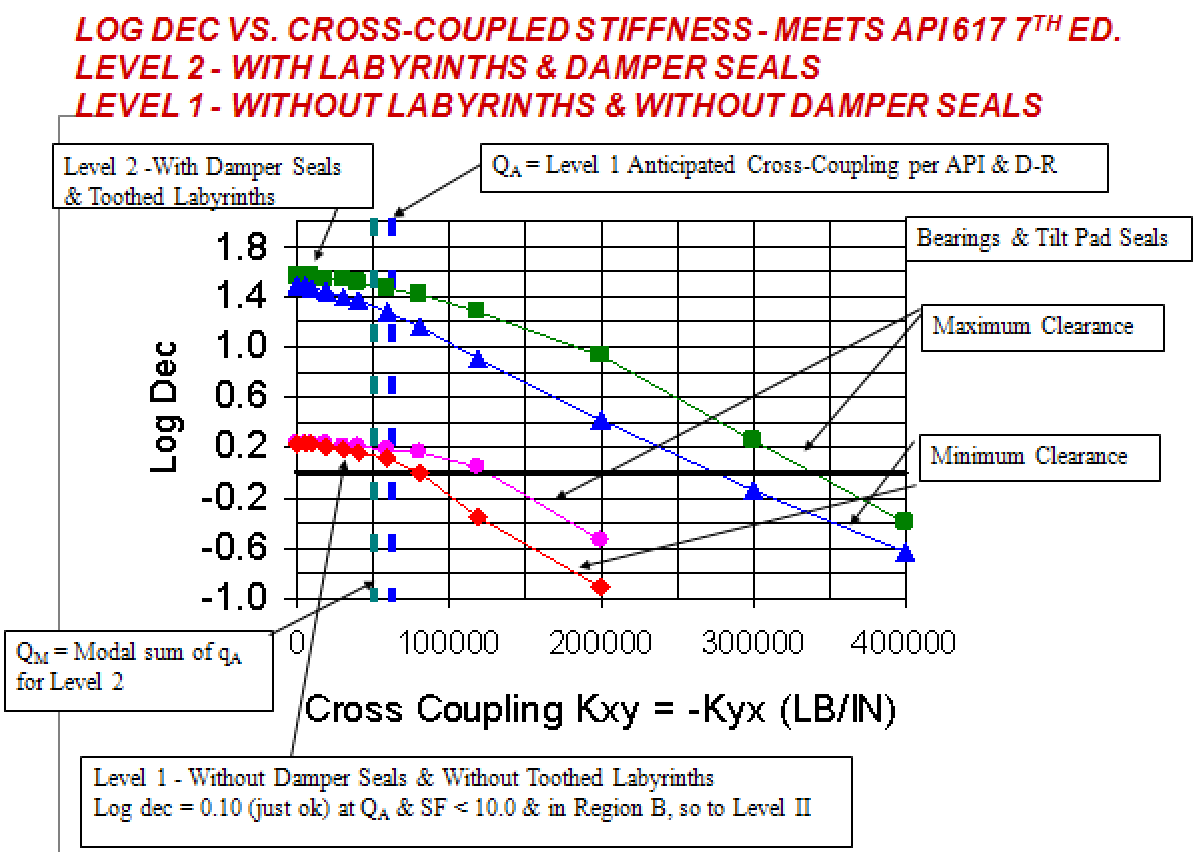


Figure 37 – Case History 8 Level I and Level II API Stability Analysis for the Last Revamp

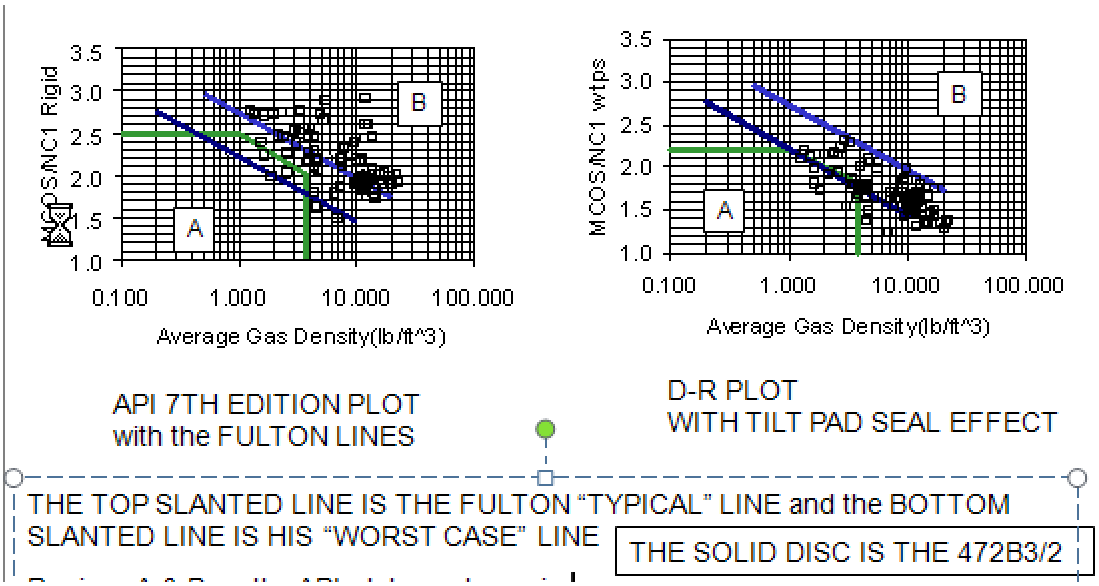


Figure 38 – Case History 8 Experience plots, Flexibilty ratio vs. Average Gas Density on the left flex ratio with rigid bearing Nc1, on the right with tilt pad seal effect

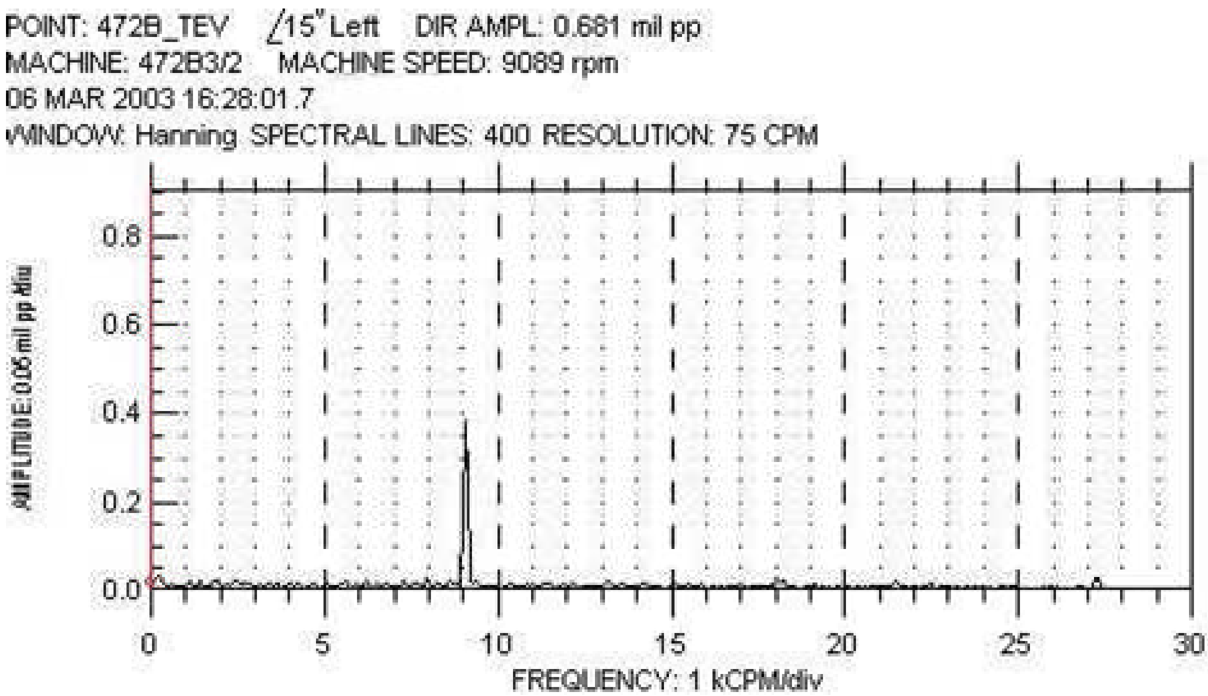


Figure 39 – Case History 8Spectrum Plot At Full Pressure – From 2003 Revamp

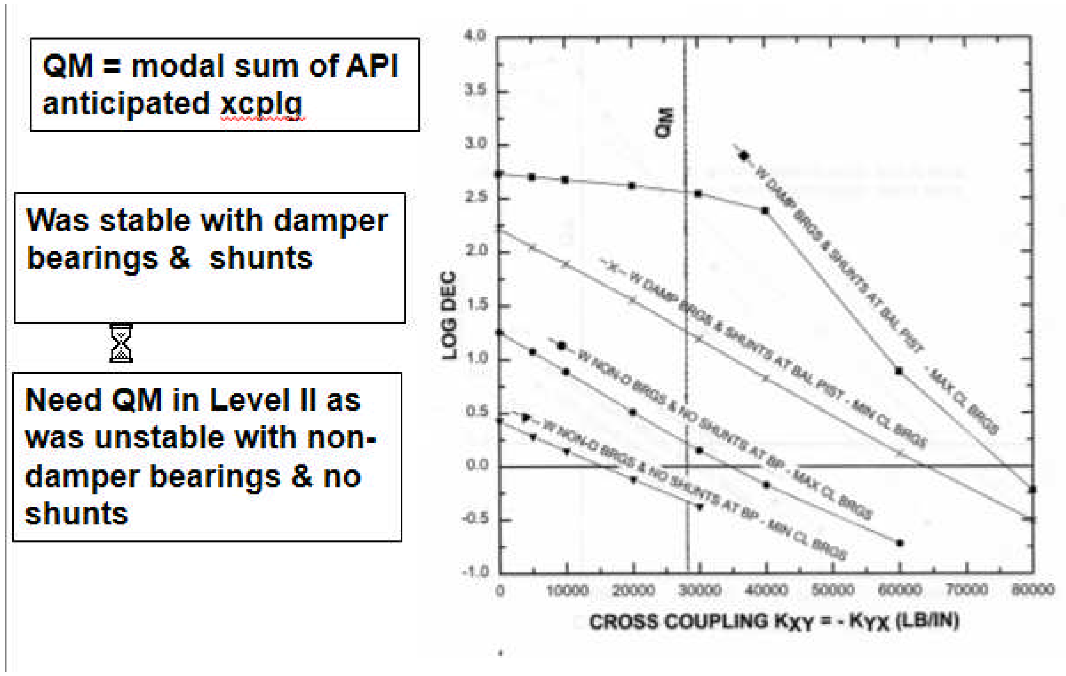


Figure 40 – Case History 9 – Log Dec vs. Cross-Coupled Stiffness at the Midspan Level II Analysis Non-Damper Bearings & No Shunts vs. Damper Bearings and with Shunts

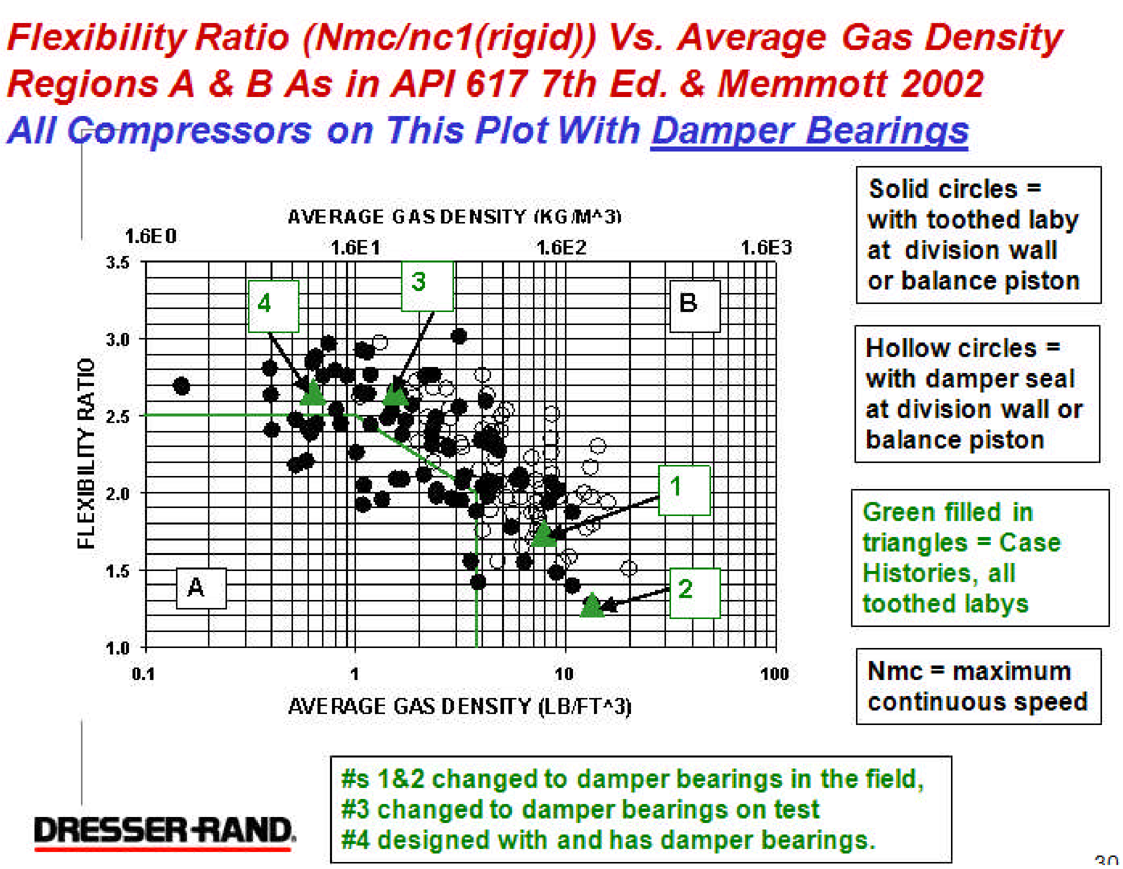


Figure 41 – Case History 9 Bearing Span/Impeller Bore vs. Average Gas Density Final Configuration of all data points with damper bearings - Number 2 is Case History 9

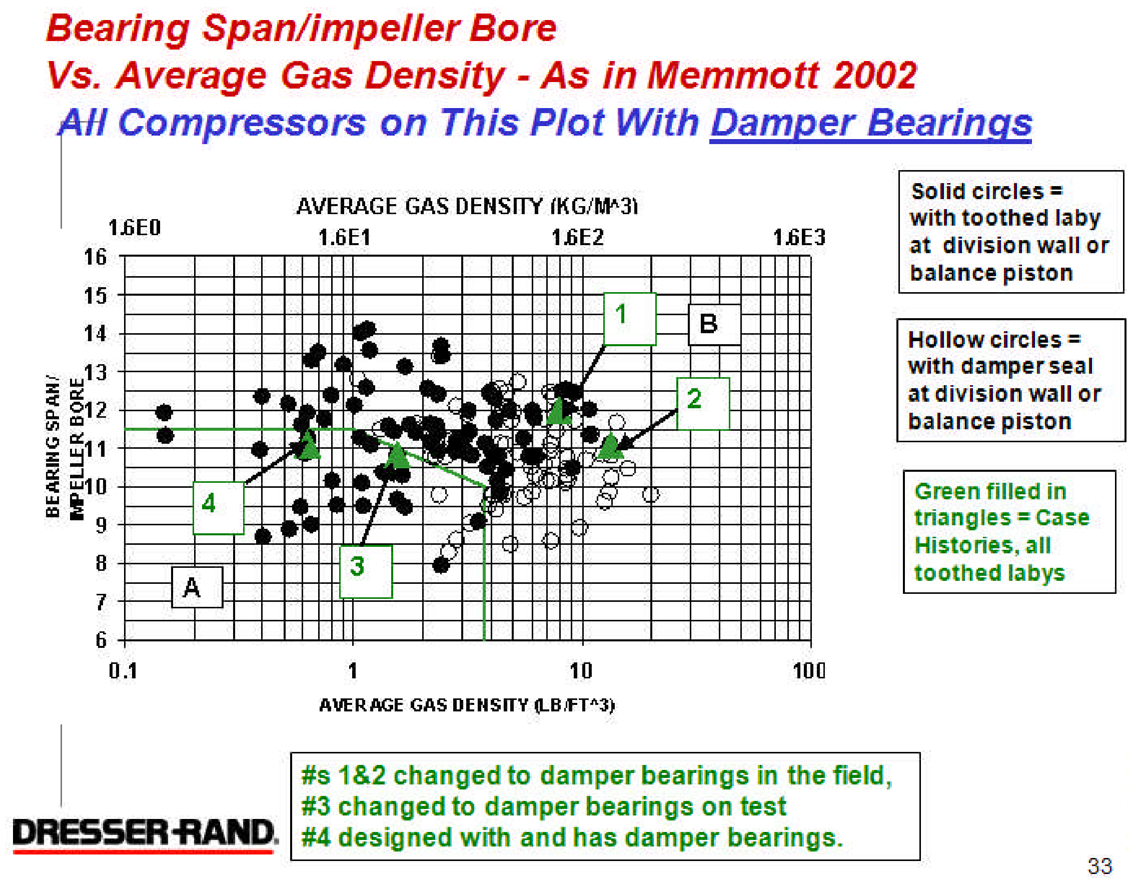


Figure 42 – Case History 9 API plot - Flexibility ratio vs. Average Gas Density Final Configuration of all data points with damper bearings - Number 2 is Case History 9

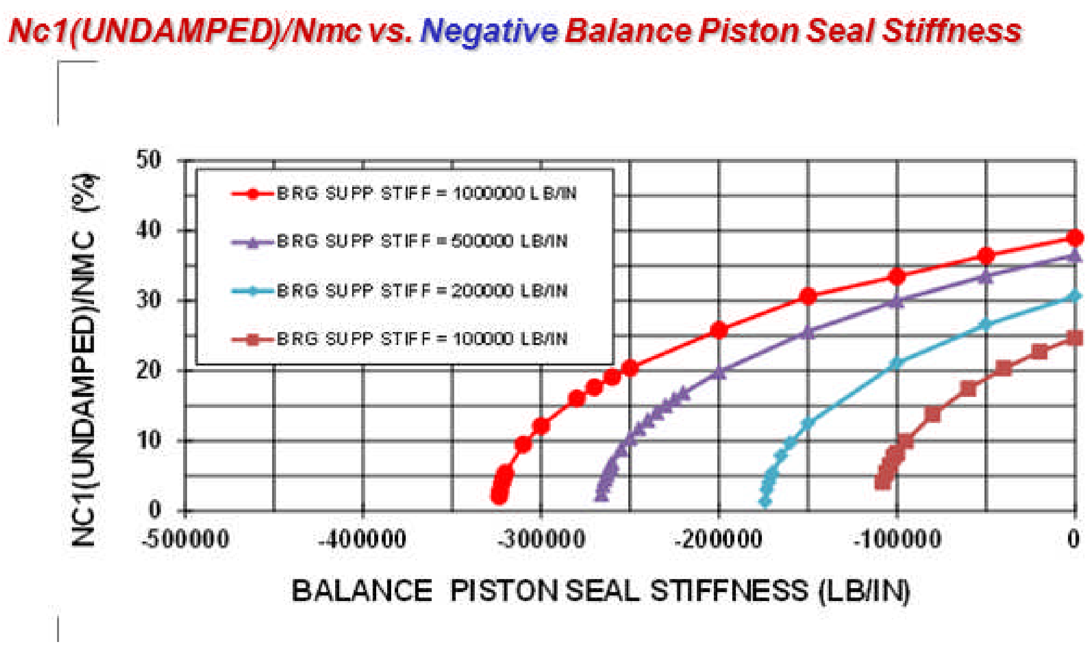


Figure 43 – Case History 10 NC1(undamped)/Nmc vs. Negative Stiffness at the Balance Piston

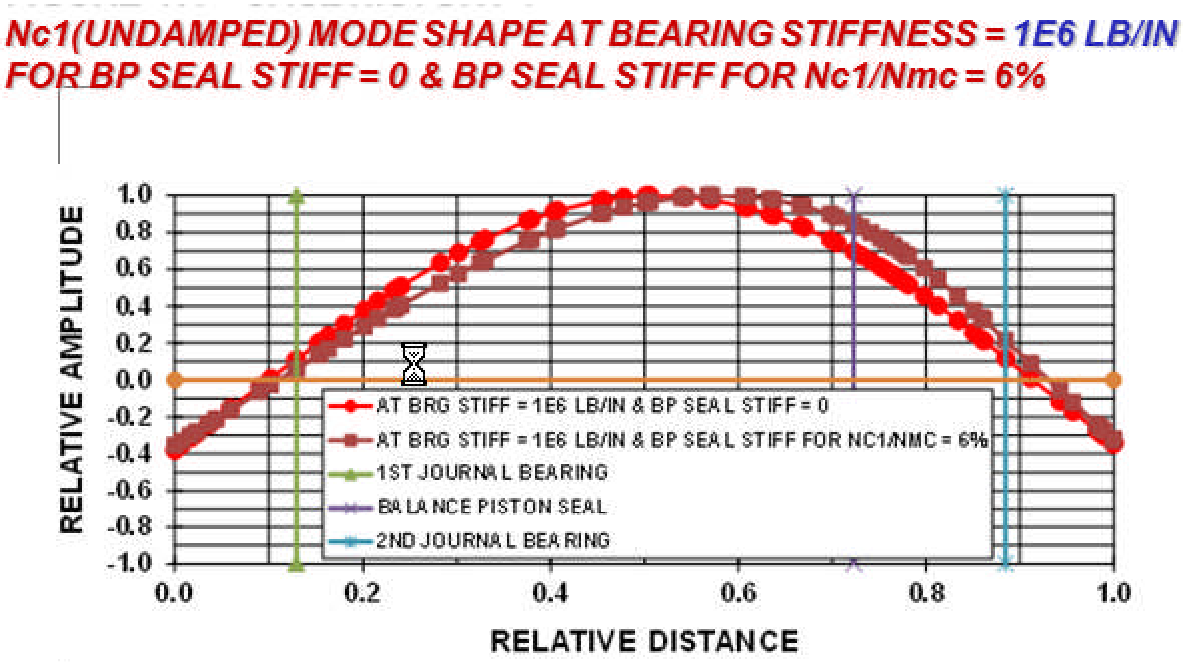


Figure 44 – Case History 10 NC1(undamped) Mode Shape for Balance Piston Stiff = 0 and NC1/Nmc = 6 %