EFFICIENCY IMPROVEMENTS ON LARGE MECHANICAL DRIVE STEAM TURBINES

by

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ABSTRACT

Many of the mechanical drive steam turbines in service in the world today were designed without the demands for optimum efficiency. In order to meet the changing economic requirements that plants face, technological solutions have been developed to improve turbine performance that can be sustained during extended run times. This is in contrast to making conventional component replacements with short-lived or limited gains in performance. Another important consideration is that performance upgrades must be made without adversely impacting the turbine's operating characteristics, especially those during startup.

Turbine efficiency is significantly affected by the performance of shaft and tip seals that control steam leakage within the machine. Conventional seals have a long history of sacrificing operational characteristics by rubbing during unit startup and producing vibration problems that can result in aborted startups. Extended startups are the short-term loss, yet the loss of installed seal clearances has a cumulative financial impact over the operational run cycle.

With the application of retractable packing to prevent rubbing during startups, tip seals not included in the OEM design may be successfully added to provide sustained thermal and operational performance improvements. These features have been successfully implemented on two mechanical drive steam turbines in an ethylene plant in Orange, Texas. Presented will be performance gains realized and experience during startup with these features.

INTRODUCTION

The lower costs for fuel and steam supply of the past led to design practices focusing on reliability. In step with these guidelines, it was common for tip seals to be absent from the design stage, and interstage seal clearances were generous. The combination would sacrifice thermodynamic performance for lower probability of aborted startups due to interstage and tip seal rubs. This represents a fundamental challenge between thermodynamic goals and reliability limitations that the designers face, which will ultimately determine the operating characteristics of the steam turbine.

Process improvements made over time typically lead to a juncture when the turbine becomes the bottleneck and needs a cost-effective upgrade. This became the case for the DuPont, Sabine River Works Ethylene Plant, in Orange, Texas, where turbine output limited plant production 70 percent of the time.

LEAKAGE CONTROL IN STEAM TURBINES

Internal steam leakage is accomplished by an array of steam seals located between the shaft and the diaphragm, at the root of blades, and at the blade tips, as shown in Figure 1. Conventional turbine seals are called labyrinth seals, based on their design to effectively accelerate steam flow and change its direction of flow multiple times.

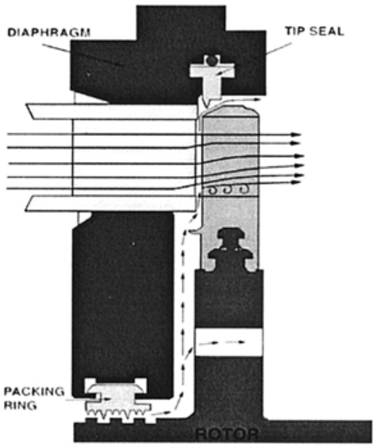

Figure 1. Steam Seals.

Martins Equation equates leakage flows based on the relationship between the various design parameters for labyrinth seals. The number of teeth in a seal, the operating clearance, and the sharpness of the seal teeth have a major impact on the seals' ability to control leakage flows.

The fundamental problem with the use of conventional labyrinth seals in steam turbines is the high susceptibility of the seals to be damaged through rubbing.

Losses due to shaft and tip-seal leakage bypassing the rotating stages result in a reduction in useful work. This reduction of internal steam leakage represents an attractive option for efficiency improvement. Leakage at the shaft seal further deteriorates performance by disrupting the flow that the stationary blading is trying to accelerate and organize (Figure 1). But the challenge to reduce this leakage brings the upgrade designers and implementers back to the mutually exclusive problem of performance versus reliability; as tight clearances are nearly always analogous with seal rubs and aborted starts in multistage turbines. Again the operating characteristics of the machine can suffer.

PACKING RUBS AND ROTORDYNAMIC CONSIDERATIONS

There is an interrelationship between mass unbalance and a rotor's vibration response. While many efforts are taken to deliver turbine rotors that are in perfect balance, this is seldom the case. There is always some component of residual mass unbalance where the center of mass is not concentric with the geometric center of rotation. Nearly all steam turbines operate above their first critical speed, and must pass through this critical where the rotor's natural frequency is stimulated by the speed of the rotor. The vibration of the rotor increases as a result of this excitation, and the amplitude of the vibration is increased with the amount of residual mass unbalance of the rotor.

Rubbing and damage to packing seals in the turbine are promoted by this excitation of the rotor passing through critical. The potential for rubbing increases with the amount of residual mass unbalance and the reduction of seal clearances that are preferable for thermal performance. There will be a predominant sector of the rotor that will rub with the seals. The midpoint of that sector is referred to as the high spot, and below the first critical the high spot is in phase with the mass unbalance.

In a rub condition, the sector of the rotor adjacent to the high spot will contact the packing teeth, and be heated due to friction. As a result, there will be a heat addition to that side of the rotor, yet 180 degrees opposite of this point, the rotor will not be heated. This results in a differential thermal expansion at the outside diameter of the shaft. The side of the rotor with the high spot will incrementally grow in length in accordance with the rate of local heat addition, the specific heat of the material, and its coefficient of thermal expansion. The thermal bow will grow in phase with the rotor high spot, thus increasing the overall unbalance. This, in turn, increases the interference between the rotor high spot and the seal, creating a more drastic asymmetric temperature profile, leading to a more severe thermal bow. The cycle just continues to spiral out of control.

Based on inspections of seal clearances during outages, it has been generally noted that machines that have tip seals typically incur more severe rubs at the tip seals than the shaft seals. It is logical to ask why this occurs, since the two seals are located on the same rotor and the rotor vibrates by the same displacement for the same stage location. The answer to this puzzle has to do with when the seals actually rub.

A conventional packing ring is held in a location close to the rotor by a system of springs, shown in Figure 2. At low rotor speeds, the packing rings are only held in place by the force of the springs. Therefore, if the rotor contacts the shaft seal, the packing ring can move away from the shaft a small amount. The seal teeth still become damaged, and the localized rubbing still produces the bowed rotor condition explained above. However, the tip seals are not able to move away from the shaft, and typically result in very high vibration levels that can trip the unit on vibration.

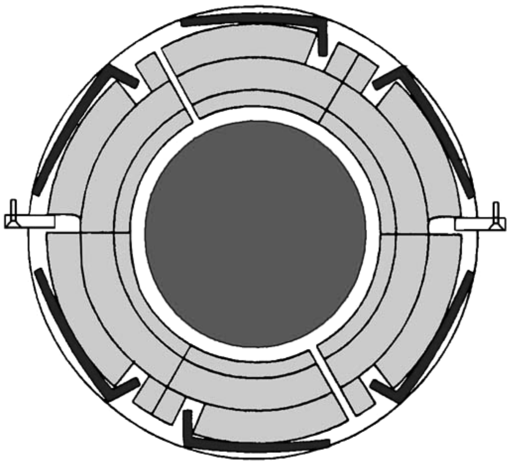

Figure 2. Conventional Packing Springs.

Units that come offline from a thermal bow induced vibration trip encounter startup delays that affect plant performance. After such an event, the unit will typically need to be run on turning gear for some time in order to reduce the eccentricity. Once the eccentricity levels become acceptable to roll the machine, low speed holds at a fixed rpm are needed to smooth out the rotor condition and allow another attempt for full speed operation. It is not uncommon for units to encounter several such trips and restarts after running into seal rubs.

Unlike shaft seal damage, tip seal damage in extreme cases can create reliability problems for the machine. Along with tip seal rubs there can be significant damage not only to the seal tooth but also to the corresponding rotating surface that it rubs on. Deep cuts into the bucket covers, and into the blade tenons that pass through the cover, can result in losing portions of bucket covers. Covers that come loose will always cause a step change in vibration, and may be enough to trip the machine offline. Sometimes the machine can make it to the next outage with the application of a trim balance-shot. However, the loss of a bucket cover will increase the potential for that group to fail from blade vibration.

Another factor is the ability to maintain the sharp tooth profile of the labyrinth seals. When the seal teeth become rubbed, they lose their sharp edge and the resultant coefficient of discharge increases, resulting in a significant reduction in leakage control with only a small increase in effective clearance.

The potential for seal rubs, vibration trouble, and extended startup times are all reasons why machines may be designed with generous shaft and tip seal clearances, or the omission of tip seals altogether.

RETRACTABLE PACKING CONSIDERATIONS

Advanced sealing technology for mechanical drive steam turbines must take into account design considerations so that the turbine's operating characteristics are not adversely affected. The goal of the designer is to make use of these technologies to improve turbine performance without compromising unit reliability and operating characteristics. Design considerations for retractable packing and tip seals will be discussed individually.

Design seal clearances for conventional packing represent a compromise between leakage control and the operating characteristics of the turbine. Often times neither objective is satisfactorily accomplished. For example, it is very common for an overhauled machine returning from an outage to encounter seal rubs, delay startup, and then run with excessively large clearances until the next shutdown. The preference would naturally be to have a technology that satisfies both leakage control and operating characteristics.

Retractable packing is a technology that was developed to specifically dismantle the mutually exclusive relationship between seal performance and acceptable operating characteristics. The purpose of retractable packing is to avoid rubs during turbine startup that are typically caused by rotor critical vibration amplitude, and prevent the resultant differential thermal expansion that causes a bowed rotor condition. In the simplest of terms, retractable packing is designed to maintain a generous clearance until the rotor has passed through its critical speed/s and then close to the desired operating clearance.

To understand the operation of retractable packing, we must first review the forces at work on conventional packing rings. Traditional labyrinth seals are held in a desired radial location by the use of springs behind the packing ring segments. With little or no steam flow through the packing, the conventional spring provides the only force locating the packing ring to its hook position.

As throttle flow increases, a force will be directed on the back of the packing ring body. The packing ring will be shuttled downstream by the flow of steam, until the steam joint of the packing or downstream neck comes to rest against the downstream hook of the diaphragm. Upstream pressure, P1, will pressurize the area behind the packing ring, and exert a force on the back of the segments, as shown in Figure 3. The force against the back of the packing segments will be directly proportional to the product of the upstream pressure and the projected area of the packing body, whose force will increase with throttle flow and be directed in a radial direction toward the rotor.

In addition, the pressure drop across a given packing ring increases with throttle flow as well. Steam will pass through the labyrinth produced by the packing teeth and the rotor lands, with pressure decreasing from the P_1 upstream pressure, down to the P_2 downstream pressure. Assuming a linear pressure drop through the packing, a median pressure one-half the difference between P_1 and P_2 will be exerted against the body of the packing, producing a radial force away from the shaft.

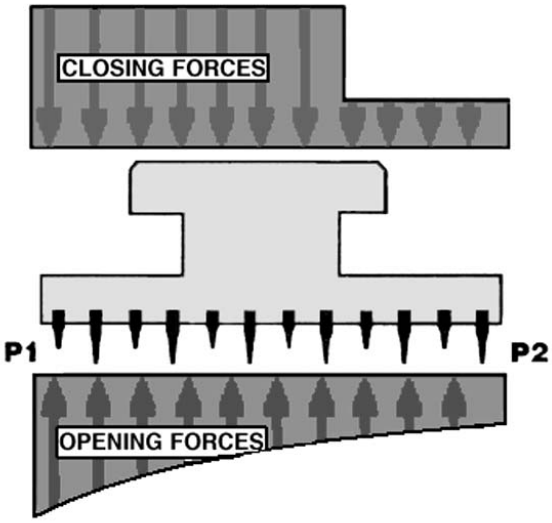

Figure 3. Opening/Closing Forces Due to Steam Pressure.

The difference between the pressures acting radially inward and outward will increase with throttle flow, resulting in a net force projected against the back of the packing segments toward the rotor. The operation of retractable packing is made possible by this pressure differential and resultant force unbalance.

As shown in Figure 4, application of retractable packing replaces the springs behind the segments with springs inserted into the ends of the segments, approximately parallel to the circumferential tangent. While the conventional springs work to hold the packing closed, the retractable springs work to keep it open; and a complex design methodology calculates spring variables to produce ring closures at the desired throttle flow rates. The challenge is not in closing the retractable packing rings, but in keeping them open long enough to avoid rubbing contact with the shaft during startup.

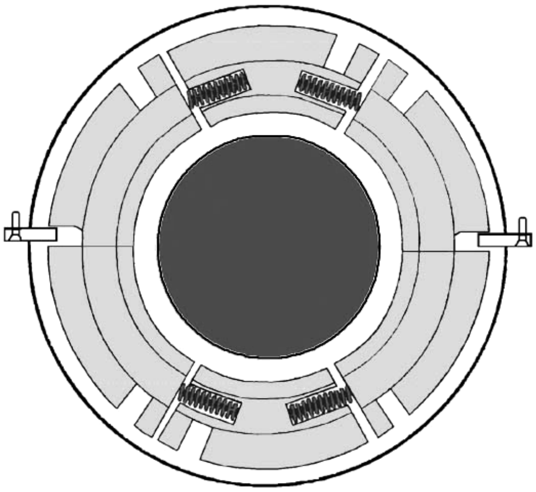

Figure 4. Retractable Packing Spring Locations.

There are many variables that are considered in the application of the springs used in retractable packing. The material must be selected to resist high temperature creep phenomena, as well as resist stress corrosion cracking for the expected duration of the operational run. Free lengths, wire diameters, and coil diameters must be manipulated to produce a spring where peak stresses are managed to acceptable values while providing adequate travel without coil-bind. All these variables are taken into consideration in the design of the retractable packing.

In summary, retractable packing is designed such that it remains open at a larger radial clearance with respect to the rotor until a point in time after the turbine rotor has passed through its first critical speed, shown in Figure 5. The packing closes due to an increase in throttle flow and the resultant increase in differential pressure. Upon completion of closure, the radial clearance between the packing teeth and rotor is reduced to the design operating clearance.

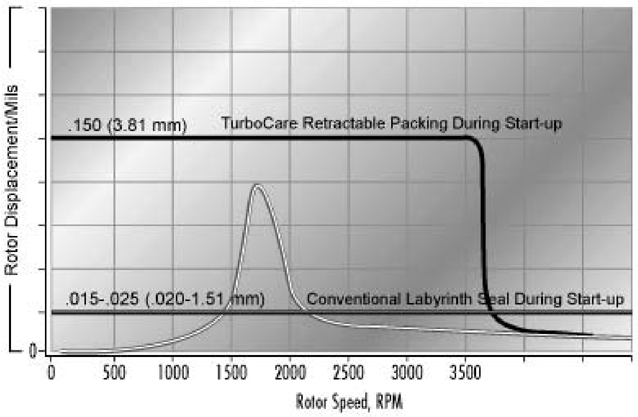

Figure 5. Retractable Packing Closes above the Critical Speed of the Rotor.

The application of retractable packing has different design considerations for mechanical drive and power generation turbines. In a generator drive steam turbine, the rotor passes through its first critical before the turbine starts to be loaded. At this point in time the differential pressure and throttle flow are very low. Therefore, the retractable packing can be designed to close earlier in the startup sequence for a power-generation application. In the case of mechanical drive steam turbines, the turbine starts up under load and the packing must be designed to remain retracted or open until later in the startup sequence, well after rotor first critical. With mechanical drive steam turbines, the allowable window available to close the retractable packing is shortened. This must be taken into account in the design of the packing.

With retractable packing keeping the seal teeth away from the rotor during startup, as shown in Figure 6, the compromise of thermal performance in exchange for a desirable operational characteristic is eliminated. Seal clearances may now be reduced below that of the original design, shown in Figure 7, resulting in a performance increase through improved leakage control.

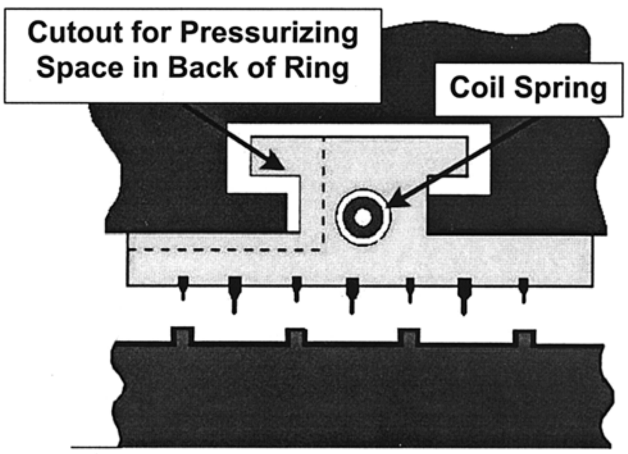

Figure 6. Retractable Packing Remains Away from the Rotor During Startup.

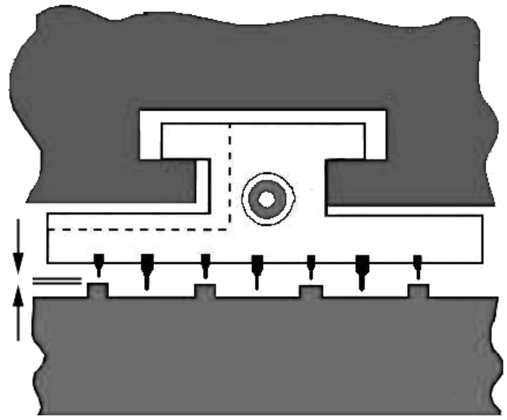

Figure 7. Design or Smaller Clearance above Critical.

The second major benefit with the use of retractable packing is the prevention of a bowed rotor condition from packing rubs. This clears the way for the installation of tip seals into machines that were originally designed without them. For machines already having tip seals, significant clearance reductions can be provided for improved leakage control. In most cases, the gain from tip seals will be equivalent to that gained by the reduction of shaft seal clearances.

Retractable packing produces performance gains over an extended period of time because the shaft seals are kept away from the rotor during startup. This is far different from conventional applications where packing is often rubbed on the first startup after a shutdown. Performance takes a sharp reduction from the initial startup and the associated losses are experienced through the entire operational run.

In addition to a thermal performance increase, the application of retractable packing generally improves the operational characteristics of the steam turbine allowing faster and smoother starts. As mentioned earlier, residual mass unbalance along with tight clearances contribute to packing rubs during startups with conventional seals. With retractable packing, the seal teeth are retracted from their design running position by 0.125 to 0.150 inch. The extra distance away from the shaft eliminates the rubbing damage during startup, and generally results in faster startups allowing the plant to reduce nonproductive turbine roll time.

The benefits of retractable packing outlined above allow the addition of tip seals, or the reduction in existing tip seal clearances below that of the OEM design, because of the elimination of packing rubs during startup. The application of blade tip seals or spill strips also requires certain design considerations. The geometry of the tip seal design must be able to fit into the existing turbine components with minimal modifications. Axial and radial clearances must be carefully considered so that tip seal rubs are not incurred due to rotordynamic effects or differential thermal expansion between stationary and rotating components. Because the tip seals are installed at a fixed radial clearance with respect to the rotor, material must be selected that will be tolerant in the event of a rub. When properly applied, significant increases in thermal performance may be obtained without sacrificing operational characteristics.

OPTMIZING GAINS FROM RETRACTABLE PACKING

Retractable packing represents a significant and proven advance over conventional packing and seals in steam turbines. With the reduction or preservation of OEM specified clearances and the potential addition of tip seals, there are several factors that should be considered in order to maximize the gain from retractable packing.

Diaphragm alignment is a key component to achieving optimal performance from retractable packing, especially when reducing clearances below that of the OEM design values. If the alignment is in error, then there will be an increased risk of the packing rubbing upon the completion of closure. While vibration will not be a significant problem, the packing can still incur damage depending on the severity of the alignment error.

In an effort to improve alignment accuracy and maximize the gain from retractable packing, the opening inspection of the machine should be used to collect valuable wear data. It should be noted that opening horizontal joint readings represent only a small portion of the packing wear picture. If packing damage occurred at the horizontal joint alone, then horizontal joint readings would give a good indication of the wear picture. Yet inspection experience shows that individual tooth heights must be taken around the circumference of the ring, in order to determine where the rubbing damage is predominant. This information can be utilized by the alignment specialist to make corrections to the generic alignment instruction for the machine, or can refine the present alignment methodology.

Diaphragms and packing holders should be checked for concentricity. Field experience indicates that machines can be prone to distortion of the packing holders, where there will be a difference between the horizontal to vertical hook diameters. For example, look at what happens if a holder has a horizontal hook diameter that is only 0.040 inch larger than its vertical hook. If the closing horizontal joint readings are used to fit the packing to a radial clearance of 0.025 inch, then the effective vertical clearance will only be 0.005 inch radially, and is certain to rub.

RETRACTABLE BRUSH SEALS

While retractable packing is a successful technology of its own, it still has some limitations. The product is designed to provide protection from packing rubs during startup, at which it is very successful. The forces acting to close the packing rings are also in effect when the machine is at full-load. At full-load, that net force acting to push the ring toward the rotor can exceed several thousand pounds for one ring, virtually making the ring immobile. Any rotor contact under full-load conditions will result in severe damage to the seal teeth. In addition, the effect of residual mass unbalance will still create some component of rotor displacement when operating at load. And while packing clearances can be reduced below that of the original design in some cases, alignment tolerances and vibration levels pose a practical limitation on clearance reduction.

Single stage brush seals have been successfully used as alternative gas seals for a number of years in aircraft engines (Flower, 1990; Mahler and Boyes, 1995) and can offer as much as a 17× reduction in leakage flow as compared with a four-tooth knife-edge seal of 0.51 mm (0.013 inch) clearance shown in Figure 8. Utilizing this technology in land-based turbines by combining the conventional, knife-edged labyrinth seal with the brush seal offers many distinct advantages, while retaining the labyrinth seal as a backup. These advantages include the following:

- Improved leakage control through the preservation of very small clearances as compared to knife-edged labyrinth seals.
- Improved turbine operating characteristics and reliability due to brush seal compliance.
- Sustained performance improvement as a result of extended seal life.

A cross section of a typical brush seal with the corresponding nomenclature is shown in Figure 9. As can be seen in the side view of Figure 10, the bristles are canted at an angle, along the circumference of the seal, in the direction of shaft rotation. This allows the brush pack to move radially to accommodate rotor transients, and is referred to as compliance. Haynes® 25 alloy is commonly used for the bristle material. The combination of wire diameter and material provide an optimum balance between wear resistance, flexibility for compliance, and leakage control.

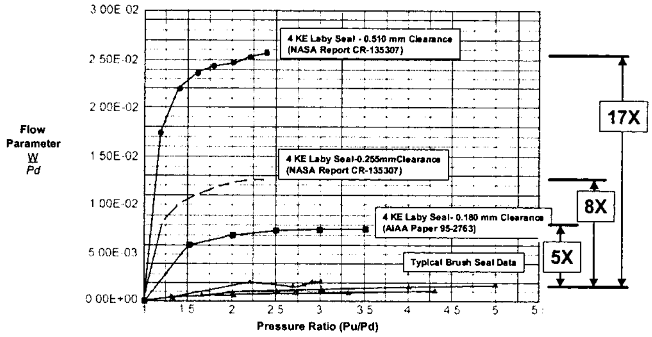

Figure 8. Brush Seals Are Capable of Dramatic Reductions in Leakage as Compared to Conventional Knife-Edge Labyrinth Seals.

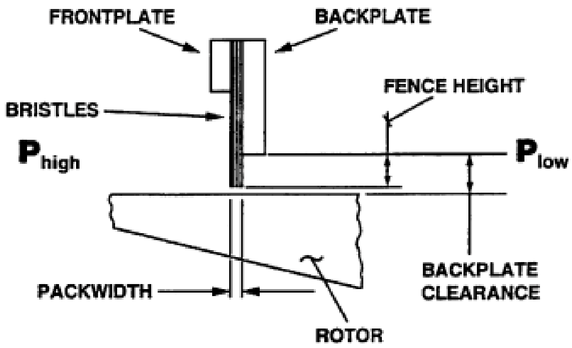

Figure 9. Section Through Typical Brush Seal.

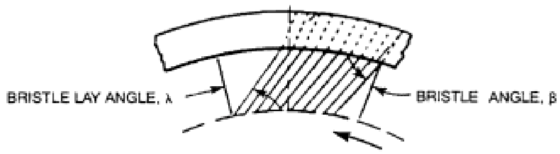

Figure 10. Brush Seal, Side View.

The frontplate and backplate are typically manufactured from 300 or 400 series stainless steel. The backplate provides support for the bristle pack in the axial direction. And the frontplate is used to control radial stiffness of the seal in conjunction with the bristle lay angle. The backplate clearance is the radial distance between the inside diameter of the backplate and the rotor. This clearance is chosen to prevent contact of the backplate to the rotor during transient operation. The fence height is the radial distance between the inside diameters of the backplate and bristle pack. Fence height is chosen to minimize leakage flow, and prevent bristle blow-over from high differential pressure across the seal. If the fence height were too large, then a high-pressure differential across the seal could cause the bristles to deflect downstream, and would compromise leakage control.

It should be noted that if a seal is used outside its design pressure differential, it does not fail. The blow-over phenomenon merely results in a temporary reduction in sealing effectiveness. The full sealing integrity will be reestablished once the pressure differential is reduced below the design limit. In the case of low-pressure differentials, a small fence height is not needed for leakage control, and therefore a larger height can be used to keep the plate further away from the rotor.

As shown earlier by previously published performance data (refer to Figure 8), brush seals provide a significant reduction in flow relative to conventional knife-edge labyrinth seals. In addition to available performance data, the manufacturer conducted its own testing as part of the design validation process. A portion of those results is shown in Figure 11; showing a graphical comparison of leakage control between a typical brush seal and a seven knife-edge (KE) labyrinth seal.

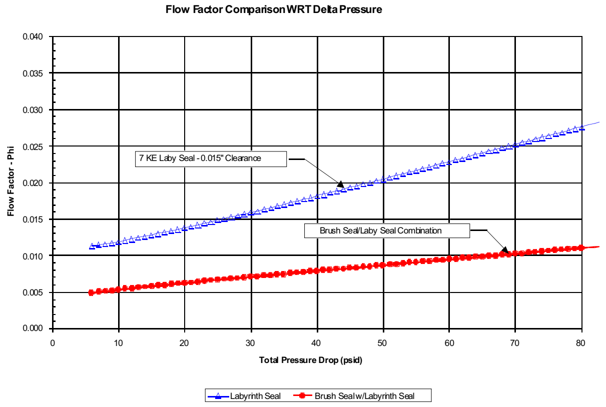

Figure 11. Flow Comparison of 7-KE Labyrinth Seal at 0.015 Inch Clearance, to a Brush Seal/Labyrinth Seal Combination.

The flow factor (phi) in Figure 11 is directly proportional to the leakage flow and is defined in Equation (1):

$$Phi = \left[m \times \left(T_{avg}\right)^{\wedge 1/2}\right] / \left(Pu \times Di\right) \quad (1)$$

where:
m = Air leakage flow rate, lbm/s
T_{avg} = Average upstream air temperature, °F
Pu = Upstream air pressure, psia
Di = Outside diameter of the seal rotor, inch

As can be seen in Figure 11, the brush seal/labyrinth seal combination offers a 60 percent reduction in leakage over the conventional, knife-edge labyrinth seal.

BRUSH SEAL APPLICATION CONSIDERATIONS

Advanced sealing technology for mechanical drive steam turbines must take into account design considerations so that the turbine's operating characteristics are not adversely affected. The goal of the designer is to make use of these technologies to improve turbine performance without compromising unit reliability and operating characteristics.

As mentioned previously, one of the key attributes of the brush seal is its compliance, which allows it to move radially to accommodate rotor transients. Its ability to sustain significant leakage reduction is a function of its wear resistance during transient operation. The magnitude and rate of bristle wear is influenced by a number of factors including, but not limited to, rotor surface finish, surface velocity, magnitude of radial interference, and the length of operating time with interference.

The brush seal's wear resistance is determined by two important brush seal design characteristics:

- Wire bristle selection and
- Bristle stiffness.

These criteria are evaluated and chosen in order to obtain a favorable balance between sealing effectiveness and wear resistance. The stiffness of the seal must be chosen such that it maintains reduced leakage while optimizing component life. Figure 12 shows the relationship between brush seal wear and time as a function of seal stiffness. In the graph, A is the lowest stiffness value and D is the highest. Generally speaking, bristle wear increases with seal stiffness.

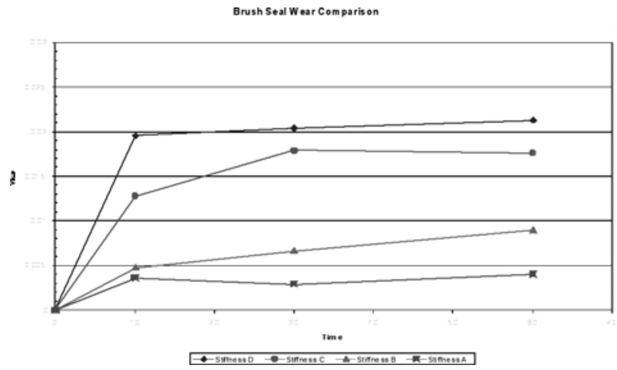

Figure 12. Brush Seal Wear for Various Stiffness Values.

The manufacturer's testing has demonstrated the effects of running brush seals with varying degrees of interference and stiffness as a function of time. For a typical brush seal design, a 0.010 inch radial shaft excursion for a continuous run time of 60 minutes resulted in only 2 to 3 mils of bristle wear. The number and severity of radial transient cycles of the rotor, along with operating conditions, drive the life expectancy of a particular brush seal design.

The cross sectional geometry of the brush seal must also be designed to accommodate the existing labyrinth seal and turbine shaft geometries, as shown in Figure 13. In the case of stepped labyrinth teeth with corresponding grooves or lands on the shaft, axial clearance considerations must be taken into account when applying the brush seal technology.

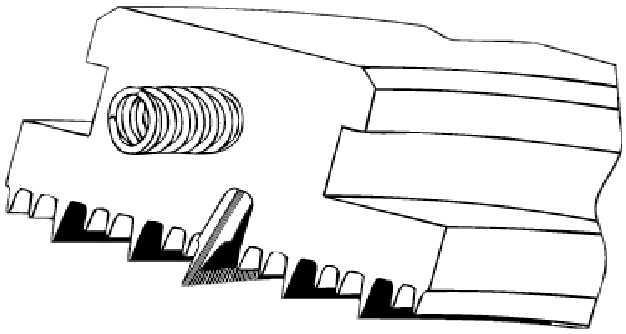

Figure 13. Retractable Brush Seal Geometry.

Utilizing retractable brush seals, where a brush seal is applied in conjunction with retractable packing, reduces unnecessary wear on the brush by avoiding contact or interference with the shaft during turbine startup. This is especially important with respect to rotordynamics, which is discussed in detail below. The fact that the brush seal is designed to operate at zero clearance with respect to the rotor must be accounted for in the design process. Wear of the brush seal during startup transient operation must be avoided in order to maintain optimum leakage control and improved turbine performance.

The application of blade tip seals or spill strips also requires certain design considerations. The geometry of the tip seal design must be able to fit into the existing turbine components with minimal modifications. Axial and radial clearances must be carefully considered so that tip seal rubs are not incurred due to rotordynamic effects or differential thermal expansion between stationary and rotating components. Because the tip seals are installed at a fixed radial clearance with respect to the rotor, material must be selected that will be tolerant in the event of a rub.

Finally, thermodynamic design considerations include maximum differential pressure and operating temperature. Both play a role in the design of these advanced sealing technologies and must be accounted for in the design process so that performance and operational reliability are optimized.

BRUSH SEAL ROTORDYNAMICS AND ROTOR BOW

The manufacturer has spent a considerable amount of effort studying the effects of seal rubs on turbine rotors as previously outlined. In the worst case, a brush seal can generate an asymmetric temperature profile around the shaft such that one side of the shaft is hotter than the other. This is referred to as "thermal bow." It is very important to design the brush seal to avoid thermal bow and not adversely affect unit operational characteristics. This is accomplished by utilizing retractable brush seals with maximum bristle flexibility or compliance for the application.

There are two significant ways to generate heat with a brush seal. One is eccentricity. In this case, the seal center is offset from the shaft center. Over the course of one revolution, the entire shaft circumference passes through this region of high interference. So, the seal will become hotter at the high interference location, while the rotor is expected to have a uniform temperature distribution along the circumference. This does not contribute to thermal bow.

The second way that the brush seal can generate heat is rotor unbalance. With an unbalanced rotor, there is one point on the shaft circumference that will have the greatest interference with the seal, and another point 180 degrees opposite that will have the least interference with the seal. Similar to the first case, the hottest region will be near the point of highest interference. But, in the case of unbalance, the peak interference follows the rotor, thus creating the asymmetric temperature profile along the rotor circumference. It is this asymmetric profile that causes thermal bow.

In both the above cases, the degree to which the temperatures rise is strongly dependent on the amount of unbalance, eccentricity, and seal stiffness. A soft seal (as typically installed) can run with rather large amounts of unbalance (in terms of 4W/N) and eccentricity while maintaining a tolerable temperature profile. But, in the case of an excessively stiff seal, the residual unbalance can lead to elevated temperature gradients along the rotor circumference.

ROTORDYNAMIC TEST DESCRIPTION

As mentioned earlier, the stability of a rotor experiencing thermal bow is dependent on rotordynamics. Rotordynamic testing with an excessively stiff brush seal was performed on an existing test rotor whose configuration is shown in Figure 14. This rotor and seal assembly had been used in the past for many hours of brush seal testing. For this test, two pairs of brush seals were installed, one pair on each side of the disk, about 9 inches from the rotor midspan. The rotor is very flexible, with an amplification factor of about 12.

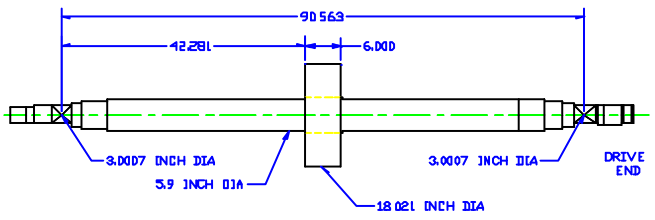

Figure 14. Flexible Rotor for Brush Seal Testing.

The testing consisted of three runs:

1. Run rotor without any seals installed to obtain reference data on residual rotor unbalance;

2. Run rotor with typical brush seals (of design stiffness) and 0.020 inch radial eccentricity;

3. Run rotor with excessively stiff seals at same eccentricity.

For each of the runs with brush seals installed, the goal was to run up to full speed (8000 rpm) and hold until reaching equilibrium.

The higher stiffness brush seals were produced from previously used test seals by intentionally crimping the front plate into the bristle pack, thereby producing a seal with highly restricted compliance. A brush seal stiffness guage was designed and used to quantify the relative stiffness of the test seals.

ROTORDYNAMIC TEST RESULTS

Figure 15 shows very clearly the impact of running a rotor with extremely stiff brush seals. As mentioned previously, Run 1 is for reference purposes with no seals installed. Maximum amplitude at the first critical is about 1 mil peak-to-peak. Running speed amplitude is about 0.5 mils peak-to-peak. Run 2 was an unrelated intermediate run and is not displayed in this report. Run 3 demonstrates the expected behavior of an unbalanced rotor running with eccentric brush seals. Critical speed amplitude is about 3.25 mils peak-to-peak and running speed amplitude is about 0.75 mils peak-to-peak, just slightly larger than without the seals.

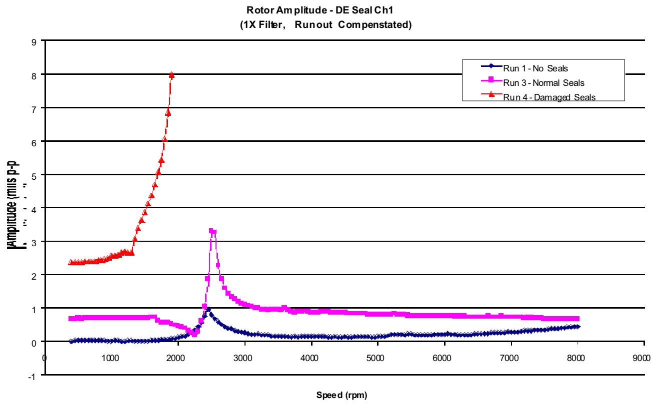

Figure 15. Influence of Brush Seal Stiffness on Vibration.

Finally, the Run 4 attempt is shown to behave as expected for a rotor running below critical with a severe thermal bow. The unbalance resulting from the thermal bow runs away very quickly. Although it is not shown in the data, the amplitude proceeded well past 100 mils peak-to-peak as all probes were wiped out and the seals were severely damaged from the excessive rotor motion.

The photograph in Figure 16 clearly shows the heavily worn area where the thermal bow high spot rubbed against the seals. The remaining circumference of the rub area looks like a typical brush seal rub location. Notice also the rub mark to the right of the brush seals. This is from the proximity probe rubbing against the rotor high spot. Again, this is not uniform around the circumference of the rotor.

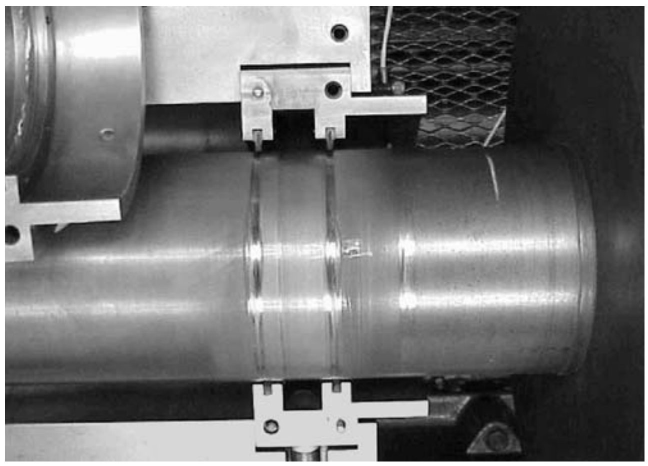

Figure 16. Photographic Evidence of Rubbing Damage Possible from Excessively Stiff Brush Seals, as Proven in Testing.

The testing performed at the manufacturer's Houston facility clearly demonstrates the impact of excessively stiff brush seals on rotor behavior. In the case of this test rotor, the runaway nature of the thermal bow is indicative of a rotor running below the first critical speed. If it were possible to engage the seals after reaching operating speed, the thermal bow would not have run away, but would have produced significant rotor vibration for reasons described above. The results of these tests validate two important brush seal design considerations:

- Optimum bristle compliance or stiffness must be achieved, and
- Retractable brush seals must be utilized to prevent contact between the rotor and the brush while the rotor passes through its first critical speed.

HISTORY

The Orange, Texas, ethylene unit became interested in the potential application of retractable packing and tip seals in late 1996. Through industry contacts within a major petrochemical company, it was learned that they had recently installed these upgrades on one of their large steam turbines, in their Norco facility, during a recent turnaround when they completely removed the turbine and refurbished it. Details of this refurbishment were presented in a technical lecture given in the Twenty-Sixth Turbomachinery Symposium (1997). By the latter part of the year, they had been in service for several months with very favorable results. The user then contacted the manufacturer of the seals, to further evaluate the applicability of retractable packings and tip seals to their steam turbines. Through the manufacturer, a number of additional users relating successes of the technology were contacted for information. It was determined that the retractable seal technology could be used to improve efficiency of the steam turbine, which was determined to be the plant constraint for approximately 70 percent of the time.

Initially, there were several concerns that needed to be addressed before the decision could be made to retrofit the turbine. Primarily, there were concerns about miscalculations of pressure differentials across internal stages, which could prevent a packing ring from closing, thus reducing overall efficiency of the turbine. To evaluate this concern, calculations were run to determine how much of an efficiency loss would occur as a result of increased shaft clearances. The losses associated with this type of failure were found to be nearly the same as that to be gained from the addition of the tip seal. Since it appeared that there was little risk of adversely affecting the overall machine performance, it was determined to be worth the risk of installing the technology. There were also concerns about reliability in the event a tip seal created an internal rub and resultant increase in vibration. After contacting various users in the utility industry, it was discovered that there were very few reliability problems associated with tip seal rubs. This technology had been previously applied in the utility industry for many years, and the associated data were readily available. With these concerns addressed, plans were made to proceed with the preparation for installation.

A manufacturer, who was not the turbine OEM, was performing the design of the retractable packing and tip seals. Consequently, the user worked with the OEM to obtain the information crucial to the proper design of the packings and seals. Specifically, all of the inner-stage pressure drops, turbine blade outside diameters, and shroud-to-diaphragm clearances had to be obtained. The OEM was very helpful in providing this critical information. In some locations within the turbine, the shrouds were tapered. This created a problem trying to locate and size the tip seals in the diaphragms. To address this problem, the tip seals installed on the tapered shrouds were designed with the ability to be shimmed in the axial direction, thus providing a method to adjust the seal-to-shroud clearance.

During the installation phase of the upgrades, additional problems were encountered. Due to a prior repair of the turbine case, the spare diaphragm for the ninth stage would not fit into the turbine. The diaphragm removed from the turbine was quickly overhauled and modified to receive the tip seal. Also, the fourth stage tip seal did not have enough axial clearance and the diaphragm had to be moved upstream to open the clearance. These problems were overcome and the retractable packing and tip seals were installed in 1997.

The successful results of the technology upgrades were manifest by increased efficiency and a decrease in turbine slow roll times. The compressor flow rates increased by 9000 lb/hr, 2.7 percent, while steam flow through the turbine decreased by 6000 lb/hr, 2.2 percent. Summing the total efficiency improvement associated to the upgrades totals approximately 5 percent. Additionally, slow roll times originally varied anywhere from 30 minutes as an absolute best to having to shut the turbine down and let it sit for 24 hours to cool down. Since the installation of retractable packing and tip seals, slow roll generally takes 10 to 20 minutes and occasionally there is little indication of any thermal bow whatsoever.

Based on the results from the upgrades made in 1997, the same upgrade was made in 1999 to the sister turbine of the one mentioned in this case study. Additionally, brush seals have been added to the turbine in this case study, with very promising results. While the amount of improvement gained from the brush seals cannot be isolated due to a large number of upgrades made to the turbine and driven compressors, the overall expected gains from the project were exceeded, and a 14 percent increase in plant capacity was realized.

CONCLUSION

The results outlined in this paper verify that significant, cost-effective performance gains can be produced through the careful application of new technologies. Furthermore, the traditional sacrifice of performance in exchange for increased reliability and desirable operating characteristics no longer limits the upgrade options for owners of mechanical drive steam turbines. Many times the application of retractable brush seals and the addition of tip seals can provide performance that exceeds original design.

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