APPLICATION OF DRY GAS SEALS IN SPECIAL PURPOSE STEAM TURBINES

by

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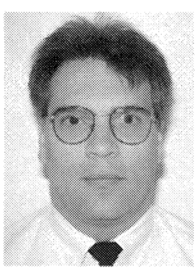

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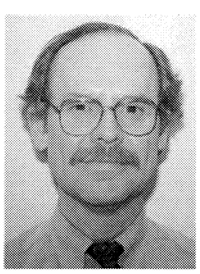

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ABSTRACT

Noncontacting dry gas mechanical face seals are being routinely applied to steam turbines to reduce new construction project costs and improve overall plant operating efficiencies. Historically, most of the applications have been in general purpose turbines. On a recent California refinery expansion project, this innovative seal design was selected for two special purpose steam turbines. Concurrently, a similar turbine for another customer had similar operating conditions and design challenges.

During turbine testing, the seals failed. An extensive root cause failure analysis determined that material face combination and operating procedures were the sources of the failures. With the change in materials and procedure, both turbines and three sets of revised seals passed a series of API tests, including a partial load test run. In the fall of 1996, the two turbines were commissioned and have been in operation since.

INTRODUCTION

More steam turbines are being purchased using dry gas seal technology as the shaft sealing method. The vast majority of these seals are being applied in general purpose steam turbines with relatively small diameter shafts running at slower speeds. On a recent Shell Oil Products Company expansion project in the Martinez, California, refinery, this innovative seal design was selected for two special purpose steam turbines. One turbine drives a recycle compressor in a distillate hydrotreater while the other machine is the driver for a coker wet gas compressor. Both machines are vital for successful continuous plant operations.

Dresser-Rand Steam Turbine Division designed and manufactured two nearly identical multistage steam turbines to fit these process services. The only difference is operating speeds for process reasons. The shafts were sealed using John Crane International Type 28-MST dry gas seals. Turbine and seal selections were made to lower the initial capital investment. Expected side benefits were lower turbine operating cost, higher availability, enhanced operator interface, and a less congested installation. If a conventional labyrinth-type seal system was used, additional cooling water and condensate recovery facilities would have been required. By selecting dry gas seals for the machines, these expensive facilities were not necessary at a capital savings of more than $1.5 million. However, these savings and benefits did not come without some operating risk.

At the time of placing a turbine order, application experience for dry gas seals with this shaft size, operating speed, and sealing pressure was very limited. The new turbines have shaft diameters of 6.5 in and operate at speeds up to 8800 rpm. The turbine inlet pressures are 650 psig with exhaust pressures of 185 psig. At this seal size, pressure, and speed, industrial experience on new machines was being pushed. Up until this point, only a few retrofitted vital turbines had these types of seals operating either at this size, pressure, or speed. None of them had the combination required for this application.

After the turbines and dry gas seals were designed and manufactured, they were tested in the vendors' factories to rigorous internal specifications and the requirements within API 612 [1]. These tests included operating the machines at their design pressures and speeds. Unfortunately, the dry gas seals failed during turbine testing. An extensive root cause failure analysis was undertaken to determine reasons for the failures. The analysis included resonance verification, fracture mechanic analysis, finite element analysis, and a detailed seal design audit. From these data, the root cause for the failures was determined to be a combination of many factors. The more significant factors were seal face material selection and operating procedures.

A new seal design was developed and tested at the seal manufacturer's factory and then in the turbines at the turbine manufacturer's factory. Both the inlet and exhaust seals, along with the turbine's seal housings, were instrumented to monitor surface temperatures during turbine casing warmup, turbine slowroll, and normal operation. The machines ran very well with little-to-no seal leaks and passed a series of API mechanical and rotor response tests, including a partial load test run.

Special operator training classes were conducted to review the history of the seal failures and design enhancements and to highlight the importance of f●llowing the established operating procedures. In the fall of 1996, the two turbines were commissioned and have been in operation since.

BACKGROUND

Two multistage steam turbines purchased as vital compressor drivers were designed for inlet steam pressure of 650 psig and a backpressure of 185 psig. The conventional method sealing turbine shafts for machines with these operating pressures is to use labyrinth rings and a gland seal leakoff system. An ejector is used to pull a slight vacuum on the turbine outboard labyrinth seals. Steam and air flow through a heat exchanger where the steam is condensed and the noncondensables are vented. For these turbines, the gland condenser requires approximately 300 gpm of cooling water to condense 3200 lb/hr of steam.

When the distillate hydrotreater and coker project premises were set, the amount of cooling water required by the gland condensers was not included in the overall cooling water demand. Additionally, condensate collected by the condensers was not included in the water usage permitted by the State of California. Therefore, this condensate must be recovered and returned to the steam system. The equipment required to recover the condensate included a collection tank, pumps, and necessary interconnecting system and header piping. The installed cost of additional cooling water facilities and condensate recovery systems was more than $1.5 million. With an emphasis on low initial capital cost, finding an innovative solution was necessary.

In 1993, Morris, Stroh, and Southcott [2] presented a study on a project to retrofit an existing steam turbine driving a vital compressor from carbon ring seals with gland condensers to dry gas seals. This single stage machine is physically smaller than these new turbines. It had a shaft diameter of 3.5 in and an operating speed of around 10,600 rpm. At the time of the article, the turbine had run for several years with no problems.

Additionally, industrial users continued to retrofit existing turbines with dry gas seals. During 1992 through 1995, more than 220 new applications were sold. The majority of these seals were being applied to general purpose turbines; but a few special purpose applications were being considered. Of all of the seals sold, there were four applications with shaft diameters larger than four inches and sealing pressures greater than 150 psig. Of these seals, three had peripheral speeds greater than 150 fps. This new application had the highest peripheral speed, but not the highest pressure. Within the seal industry, an application factor, which is the product of operating pressure (P) and peripheral speed (V), is used to show industrial experience. As shown in Figure 1, the seal's PV was slightly more than 40,000 psi × fps, while another user's seal had a PV greater than 60,000 psi × fps. At this time, this turbine was also being manufactured.

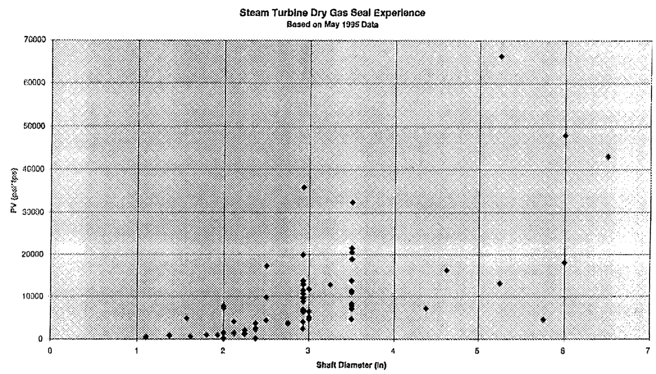


Figure 1. Industry Experience Based on Seal Application Factor (PV).

In the past, an overriding design premise on new construction projects was that the equipment must use proven technology. With $1.5 million at stake and other operating cost savings (Figure 2), the management felt it was worth taking some added risk to purchase these steam turbines with dry gas seals. As a fall back option, the turbines were designed so they could be retrofitted with labyrinths. The contingency plan also included an engineering package for a gland seal system. The hope was that the backup plan would not be needed.

Approximate Cost Comparison Between Steam Turbine Dry Gas And Gland Seal Systems

	Dry Gas Seal System	*Gland Seal System*
Capital Costs		$1 - 2MM
Cooling Water		313 gpm @ $18M / yr
Low Pressure Steam		333 lbs/hr @ 185 psig $15M / yr
Condensate Collected		3200 lbs/hr $32M / yr if routed to drain
Totals		
Capital Cost Difference		$1 - 2MM
Fixed Costs	$0 / yr	$32M - $64M / yr

Figure 2. Seal Cost Savings Justification.

DRY GAS SEAL APPLICATION DESCRIPTION

Generic Steam Turbine Seal Design

Dry gas seals have been developed and used extensively since the early 1980s, primarily on centrifugal compressors, and have only recently been utilized on steam turbines. The dry gas seal design chosen for this application is a single noncontacting dry gas mechanical face seal. The noncontacting operation is achieved by specially designed spiral grooves in the rotating face, which provides a pumping affect and in turn generates a pressure gradient across the seal face. The pressure gradient causes the seal faces to separate and provide a stable operating gap of 0.0001 to 0.0002 in. The seal operates on the process steam without buffer gas or additional support equipment. Additional description of seal operating principles can be found in an article by Shah [3].

Due to the high temperatures, steam turbine seals require secondary sealing materials capable of operating up to 650°F inlet temperature. This temperature is significantly higher than the capabilities of elastomeric materials used in compressor seals. When comparing dry gas seals for compressors to those for steam turbines, other major differences are noticed immediately:

- Flexible graphite gaskets and packing replace static O-rings.
- Collars and bolts compress the packing.
- A split carbon ring assembly is utilized behind the stationary seal face instead of an O-ring.

These and other changes permit the design temperature limits of typical noncontacting dry gas seal to be extended. Material of construction for support hardware is similar to that of dry gas compressor seals (e.g., 410 ss). Stainless steel is used, since it has similar thermal expansion characteristics to that of the casing and shaft. The rotating seal ring material is tungsten carbide, with silicon carbide composite being used for the stationary ring. Tungsten carbide offers thermal stability. Silicon carbide composite is a hard material that can cope with steam contaminants.

Special Design Concerns for Steam Applications

When designing a turbine seal, there are application specific issues that must be addressed.

- Contaminates in the steam can deposit on the face of the seal rings clogging up the spiral grooves, causing it to lose hydrodynamic lift and allow the rings to touch. These deposits may, in themselves, be abrasive. For this application, silicon carbide and tungsten carbide were selected to resist abrasion.
- If free water should enter the seal faces, added heat will be generated from higher shearing forces of water over steam. This could lead to seal face distortion causing contact between the seal faces. This concern is considered low risk, because any significant heat generation would convert the water to steam. The free water would also have a very large expansion ratio as it flashed to steam. This would cause an unstable seal operation. There has been no evidence of instabilities or face bounce observed during testing or operation, so concern over this factor is minimal.
- Free water present during transient heat transfer events (e.g., cold startup) would induce radically higher heat transfer coefficients to the various seal parts. The difference in heat transfer coefficients would cause potentially damaging thermal distortions before a stable steady state condition is reached. These distortions may be large enough to exceed any steam gap that the spiral grooves produce and allow contact between stationary and rotating rings.

Design Modifications to Steam Turbines

The high pressure steam end seal housing normally operates at pressures greater than 500 psig. At this pressure, the dry gas seal would have a PV over 110,000 psi $\times$ fps. To keep the seal operating parameters within the range of then current vendor experience (based on PV data), the steam end seal was limited to an operating pressure of 185 psig (corresponding to the exhaust pressure). This was done by adding a bypass line from the inboard side of the seal to the exhaust plenum. Flow is limited by an inboard labyrinth baffle from the first stage pressure (501 psia) to the chamber inboard of the seal. In the piping, a restriction orifice and pressure taps are provided to monitor leakoff flow. A check valve in the line prevents excessive backflow from the low pressure seal in the event of a steam end seal failure. This design is very similar to compressor thrust balancing systems.

INITIAL DRY GAS SEAL AND TURBINE TESTING

Following design and manufacturing, the new seals and turbines were tested.

Seal Vendor Standard Testing Program and Test Rig Description

Initial testing of all seals is done at the seal manufacturer's factory to evaluate the integrity of seals for pressure and speed conditions only. This includes static and dynamic seal testing at the customer's pressure and speeds. The test rig can be retrofitted with heaters, enabling a high temperature test.

The test rig, consisting of a barrel, shaft, and adaptive hardware, are designed to mimic the actual seal cavity in the turbine in an effort to effectively isolate any potential sealing or mounting difficulties. Once the seals are installed, the end bearing plates that support the shaft assembly are positioned. The test rig is then coupled to a gearbox and variable speed motor. A high pressure gas is introduced into the cavity between the two seals. A thermocouple is also added to this cavity. Seal leakage is monitored at the vent ports. A mutually agreed upon test procedure is established among all concerned parties. This is used to evaluate the seal performance at various operating conditions.

Seal Vendor Test Procedure and Test Result

For this project, the test conditions were:

- Maximum pressure: 185 psig
- Maximum continuous speed: 8200 rpm
- Temperature: Heat generated from test
- Gas: Compressed air

In addition to the static pressure integrity tests, the seals were dynamically tested at speeds from slowroll (1000 rpm) to overspeed (9020 rpm). The sealing pressure was maintained at 185 psig, while the leakage rates were measured and recorded. Total test time was approximately two hours. Acceptable seal performance was based on two factors:

- Stable seal leakage and, not so much, the leakage rate
- Visual inspection of the seal faces after test

All six seals (i.e., two sets of turbines seals and one spare set) passed these tests. Post test inspection and evaluation found:

- Seal leakage was slightly higher than expected, but remained very stable for the duration of the test.
- There were no signs of contact on the faces.

Steam Turbine Factory Tests and Test Results

As defined in API 612 [1], mechanical tests should simulate the normal expected operating conditions and should include a four-hour mechanical run test, an unbalance response test, and governor response tests. At the beginning of these routine factory tests, the turbine vendor normally operates the machine with atmospheric backpressure for initial data gathering and then at the design backpressure. When the backpressure was applied to the first turbine, the dry gas seals started leaking. At 30 psig backpressure, steam leakage was so significant that the machine was immediately shut down and inspected. The dry gas seals were found to be damaged.

SEAL FAILURE DESCRIPTION

Upon disassembly, several fractures were found in the rotating seal rings. Rotating and stationary rings had multiple radial cracks, permitting ample area for significant steam leakage. There was rubbing sufficient enough to create a 360 degree notch. Closer examination by microscope showed surface fractures (heat checks) adjacent to fracture surfaces. In the following photograph (Figure 3), surface cracks can be seen as they approach the fractured surface. As is indicated, the fractures often branched and traveled deeper into the material. An area identified as "region of measurement" is where the depth of the initial heat check cracks are found. This is the crack depth prior to fracture.

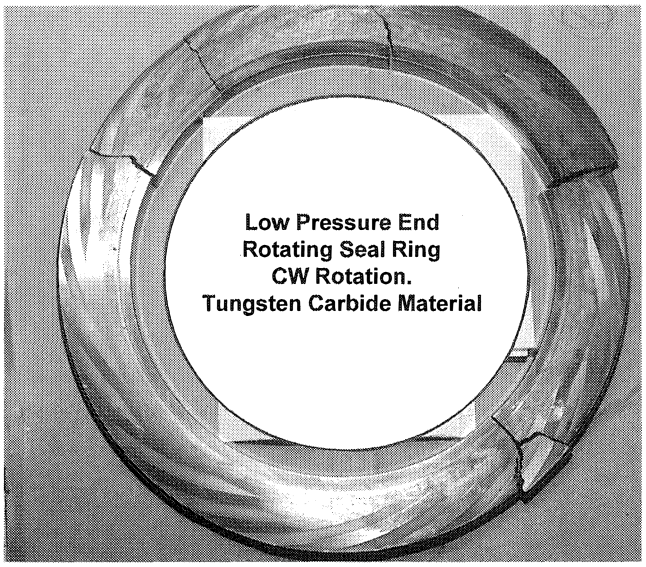


Figure 3. Photograph of Damage to Rotating Ring.

ROOT CAUSE ANALYSIS OF FAILED SEALS

Potential Explanations

There were several plausible causes identified using a root cause analysis. Each cause had to be investigated. The most likely causes included:

- Assembled waviness of rotating ring
- Material combinations—especially face materials
- Rotating ring or stationary ring distortions due to severe thermal gradients during startup
- Thermal distortions of support structure and subsequent interference
- Improper tracking of split carbon secondary seal behind stationary ring
- Rotating ring resonance
- Mechanical loads higher than film can support due to:
  - Improper working height of stationary ring
  - Springs
  - Pressure distortion
  - Debris restricting movement
  - Centrifugal effects from speed

Several different analytical tools and component tests were employed to zero in on the actual cause for the failures.

High Temperature Seal Tests

High temperature tests were performed at the seal vendor's factory to study the impact of thermal distortions on the seal faces. The test rig, as previously described, was outfitted with electric cartridge heaters. Two consecutive tests at 540°F and 440°F were conducted to approximate operating temperatures. In both cases, it took over 2½ hours to reach stable test temperatures. Once at test temperature, it was maintained for a 1½ hours. Post test inspection showed no evidence of face contact, not even a scratch.

Fracture Mechanics

Fracture mechanics analysis was performed on the fracture surface. The heat check damage was initially believed to be caused by rubbing contact between the two seal faces. Various dimensional measurements and features are shown in Figure 4. The fracture generated surface contains features that can help predict what caused the damage to the material. As an example, the fracture generated surface shows a rough surface where it is believed that the crack propagation velocity reached a very high magnitude. Using linear elastic fracture mechanics (LEFM), the stress levels in the part at the time of failure can be predicted. LEFM provides a way to estimate actual stress near the failure location, if the crack geometry can be characterized and fracture toughness of the material is known. For tungsten carbide, the fracture toughness value is 10,000 psi$\sqrt{\text{in}}$. Using this, the magnitude of stress at the time of final fracture is approximately equal to 11,300 psi. This stress includes centrifugal, along with thermal, stress generated after rubbing. The hoop stress due to centrifugal loading (from finite element calculation) is two- to three-fold higher than the predicted stress at the time of fracture. This would indicate that the ring should have broken at much lower speeds. It was then realized that local stress induced by rubbing will be compressive in nature, thereby allowing the notched ring to operate at higher speeds than initially expected.

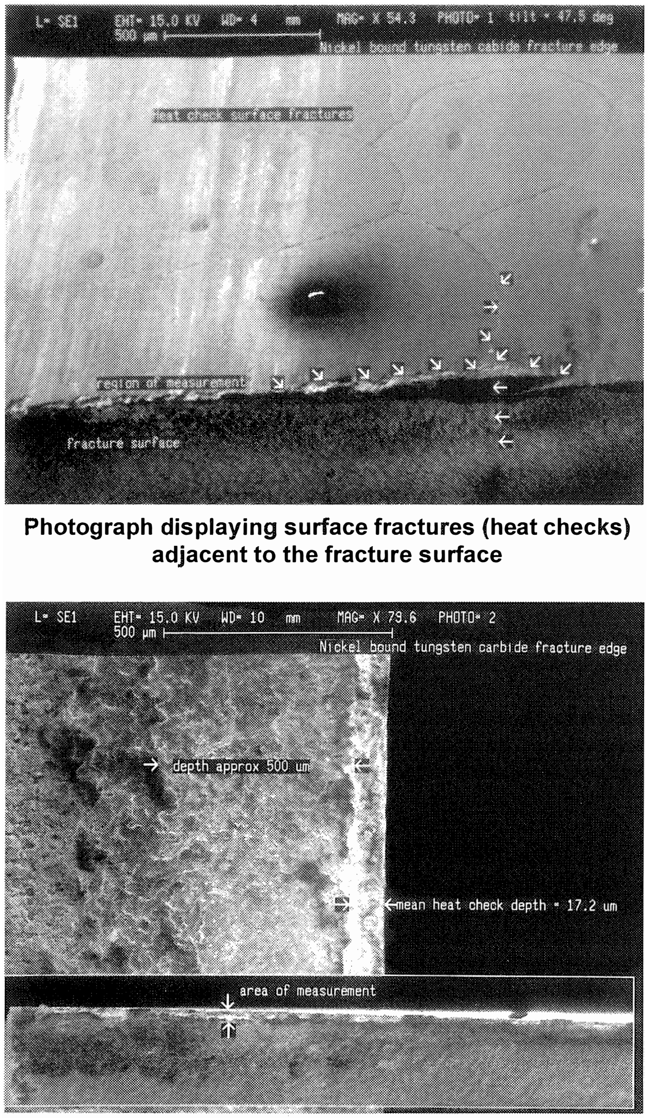
Based on the fracture mechanics analysis, it was concluded that the notch was created by rubbing contact between rotating and stationary seal faces during initial operation. This notch lowered the rotating ring's allowable material strength. As the turbine speed increased, the stress in the ring increased, initiating a crack at the notch. This crack continued until a total ring failure.

Finite Element Analysis

Finite element analysis (FEA) was used to judge the impact on various pressure and temperature loads on the seal face distortion. The analysis looked to the relative position of the seal faces and the thickness of the gas films for two different stationary ring materials. The heat transfer model is shown in Figure 5. The most significant region of heat transfer is Region 1, where under startup conditions, condensing heat transfer will cause high temperature gradients and distortion through the parts. This figure shows temperatures and heat transfer coefficients in the initial conditions. The relative face position for these conditions is illustrated in Figure 6 for both silicon carbide and carbon stationary rings. As shown, the carbon rings are more compliant to the rotating rings. Transient response of the relative gap (average across the ring faces) of the ring geometry is given for both silicon carbide and carbon (Figure 7). Thermal distortions of support hardware and subsequent interference were found not to be a problem in this analysis.

Transient Seal Film Analysis

Data from this FEA were used to calculate seal film stiffness and gap. From the transient model, during the first few seconds of operation, there were apparently substantially large thermal rotations of the seal faces. These rotations caused a large convergence operating angle. This angle produces an excessive face gap in the groove dam area, which in turn dramatically lowers the hydrodynamic film stiffness of the seal. If the film stiffness is reduced enough, the seal can no longer produce a stable gap and face contact is usually imminent.



Photograph displaying surface fractures (heat checks) adjacent to the fracture surface

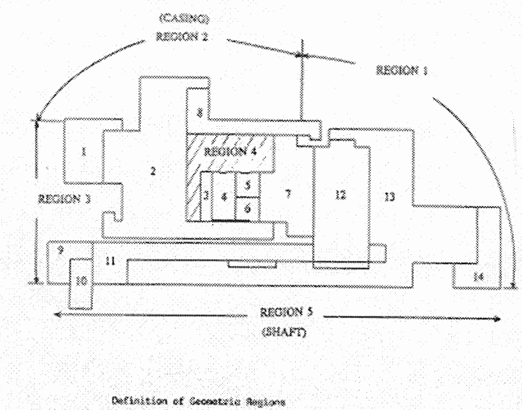
Photograph displaying initial depth of surface crack and subsequent depth of flaw prior to complete fracture

Figure 4. Fracture Mechanic Dimensional Measurements and Features.

Since this phenomenon is a function of materials properties, seal geometry changes have little effect. An alternative face material is required. Since the only other proven stationary ring material for this application was carbon, it was analyzed as a pair with tungsten carbide rotating ring. The results were dramatically different. The carbon/tungsten carbide combination distorted far less than the silicon composite within the first few seconds of operation. This reduction in thermal rotation has a compounding effect on film stiffness by almost an order of magnitude. A comparison of film stiffness is shown in Figure 8.

Resonance of Seal Parts

Test and analysis for rotating ring natural frequencies were examined and there were no low order modes that were coincident with groove passing frequencies, nor were there modes that were coincident with low order excitation frequencies. However, there were higher order interferences for 18× running speed (original number of grooves was 18). The magnitude of excitation forces for this mode is very small. Therefore, vibration was ruled out as a cause of ring contact and subsequent damage. The number of



THERMAL/PRESSURE LOADINGS

REGION	TEMP. (°F)	PRES. (psig)	H (Btu/h-ft²°F)
1	300	30	1000
2 (Casing)	200	Na	Na
3	100	0	3
4	300	30	500
5 (Shaft)	150	Na	Na

Figure 5. Heat Transfer Model.

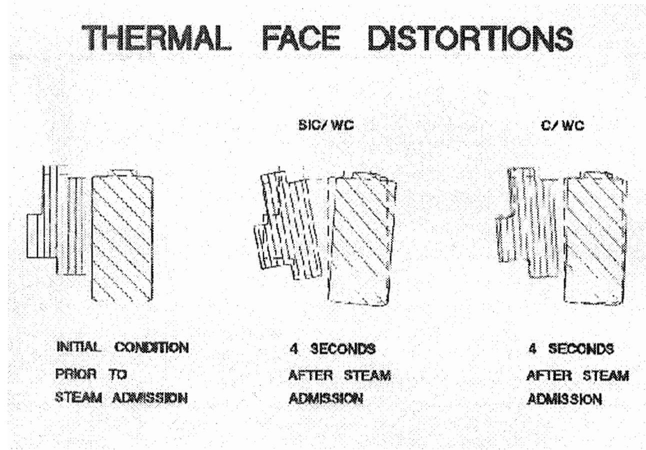


Figure 6. Relative Face Positions for Silicon Carbide and Carbon Face Materials.

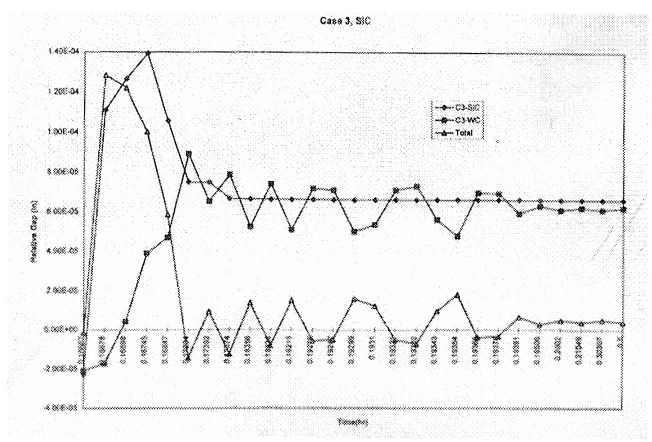


Figure 7. Transient Response of Relative Seal Gap for Silicone Carbide and Carbon.

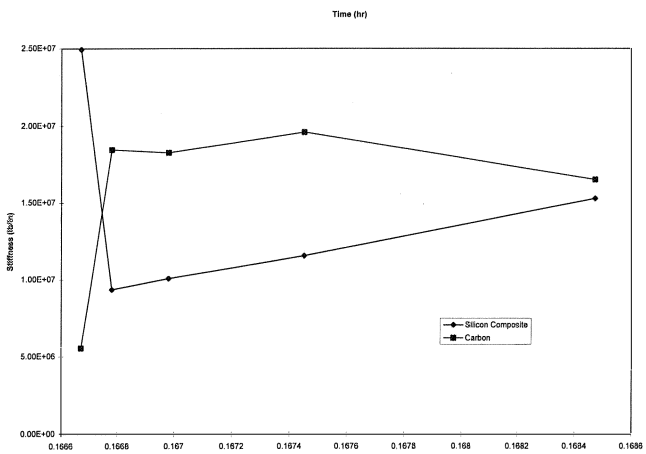


Figure 8. Comparison of Film Stiffness.

grooves was changed from 18 to 22 to accommodate low speed lift off, which eliminated any concern about resonance.

ROOT CAUSE ANALYSIS CONCLUSIONS

The strength of the tungsten carbide rotating ring was reduced, due to rub induced "heat check" cracks caused by contacting with the stationary ring. At design speeds, the centrifugal stresses exceeded resultant material strengths, resulting in a failure of the rotating ring.

Transient thermal analysis shows significant geometric distortion of silicon carbon stationary ring, under conditions of aggressive condensing heat transfer during the first minute of startup. Analysis of film performance did not predict conclusively that the rings would rub. Film characteristics did trend, however, to conditions that make rubbing more likely to occur. If the gas film was less stiff than design, the probability of seal face contact during operating upsets or geometry variation (i.e., waviness) is increased. FEA analysis with carbon stationary ring material had a much lower tendency for geometric distortion during startup.

Based on the results of this analysis, changing the stationary ring material to carbon and modifying the turbine startup sequence has the highest probability of providing reliable operation.

DRY GAS SEAL DESIGN CHANGES

After a thorough review of the data, a number of necessary design changes became apparent. The changes can be broken down into categories.

- *Materials*—The stationary seal face material was changed from silicon composite to carbon. Carbon would reduce the transient thermal distortions.

- *Groove Geometry*—Grooves were modified and the number increased to improve film stiffness at lower speeds and remove any doubt of seal part resonance.

- *Hardware Modification*—The least significant changes were the hardware design changes. This group included upgrades not only to enhance the seal's performance, but to reduce handling and other associated assembly difficulties. The changes in this group are composed of the following:

  - Coating to reduce frictional drag of the split carbon ring assembly
  - Improve flatness on gasket behind rotating ring
  - Counter bore holes for cap screws to reduce windage
  - Increase relief behind rotating face to eliminate potential for hardware distortions affecting seal face.

The final seal design is shown in Figure 9.

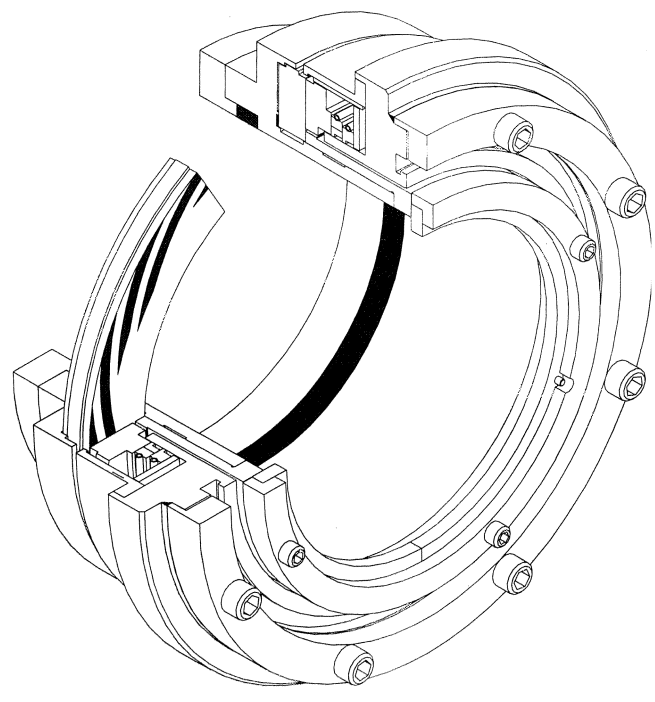


Figure 9. Cross Section of Final Seal Design.

SUCCESSFUL TURBINE TESTING

Once the design revision had been made, the seals and the turbine had to be rigorously tested to demonstrate that the seals could meet the design objective of four years' continuous operations with unlimited starts and stops. These tests included the following parts:

- Mechanical testing per API 612 [1] for each rotor
- Unbalanced response test on one rotor
- Operation of the governor systems and its control program
- Operation of the overspeed trip system
- Partial load test

One of the most important objectives of the test protocol was to determine the temperature operating limits of the dry gas seals and turbines. All three sets of seals had to be tested.

The first set of dry gas seals was instrumented with thermocouples on the outside diameter of the stationary seal rings and in the turbine seal housings. With these devices, the temperature of the seals and the turbine casing could be monitored during initial warmup. Based on the transient FEA results, it was critical not to slowroll the turbine until the seal faces had reached a minimum operating temperature of 300ºF. At this temperature, the faces were predicted to be parallel to each other and in their normal position. Once the temperature is reached, it had to be maintained for an additional 30 minutes to stabilize the faces. Only after this warmup period could the machine be slowrolled. As can be seen in the graph (Figure 10), it takes nearly one hour to warmup the faces completely.

As a precaution, and to check if there was any low speed face contact with the new seal design, the initial test was to slowroll the turbine at 1000 rpm for 15 minutes and then inspect the seals. The seal faces were found to be in excellent condition. The next test was to warmup the machine and complete the following API Mechanical Run Test [1].

API 612 Mechanical Run Test [1]

- Warmup turbine with 175 psig exhaust end pressure for minimum of one hour to make sure the seals are above 300ºF.

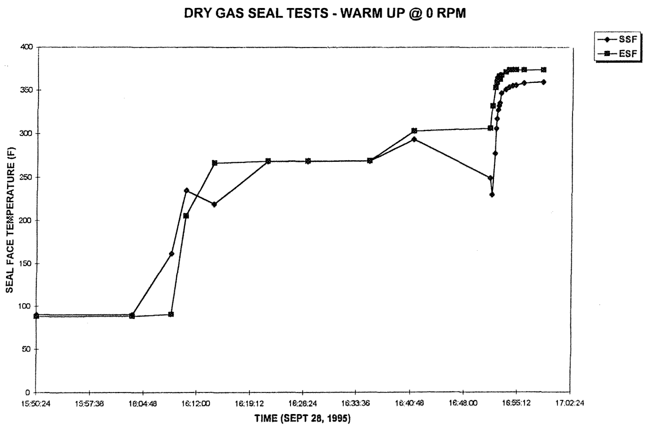


Figure 10. Seal Face Temperature During Warmup.

- Slowroll the turbine at 1000 rpm for one hour while monitoring seal face temperatures.
- Accelerate the turbine to 8772 rpm in 500 rpm intervals. Hold each speed for a minimum of five minutes. Record seal temperature, vibration, and bearing temperature data at each speed level.
- Accelerate to one percent below overspeed trip speed and operate for 15 minutes.
- Accelerate to 110 percent of trip speed and hold for one minute.
- Decelerate turbine speed to maximum continuous operating speed.
- Trip the turbine.
- Bring the turbine to slowroll speed for five minutes.
- Accelerate to minimum governor speed of 5848 rpm using the governor controls. Ramp up to 8772 rpm for a four hour run. During this run, change the governor speed by 10 percent rpm (decreasing and increasing) every 30 minutes. Change exhaust temperature from 400°F to 600°F during this period.
- At the conclusion of the four-hour run, trip the turbine from maximum governor speed to monitor the machine's mechanical performance.
- Accelerate the turbine to maximum governor and operate for five minutes. Record bearing data.
- Trip turbine and allow to come to a complete stop.
- Cool overnight.
- Warmup turbine with 175 psig exhaust end pressure for the determined time.
- Slowroll the turbine at 1000 rpm for one hour.
- Accelerate to minimum governor speed of 5848 rpm using the governor controls. Record seal temperature, vibration, and bearing temperature data.
- Run turbine for one hour or until bearing and seal temperatures are stable.
- Accelerate to one percent below overspeed trip speed and operate for 15 minutes.
- Conduct overspeed trip tests.
- Trip turbine.

This procedure was completed on the first rotor. As can be seen in the graph (Figure 11), the seal face temperature changes with speed. The exhaust end seal ran hotter than the steam end seal. The turbine and seal ran very well with only minor leakage and no vibration. The mechanical behavior of the machine did not change throughout the four-hour mechanical run.

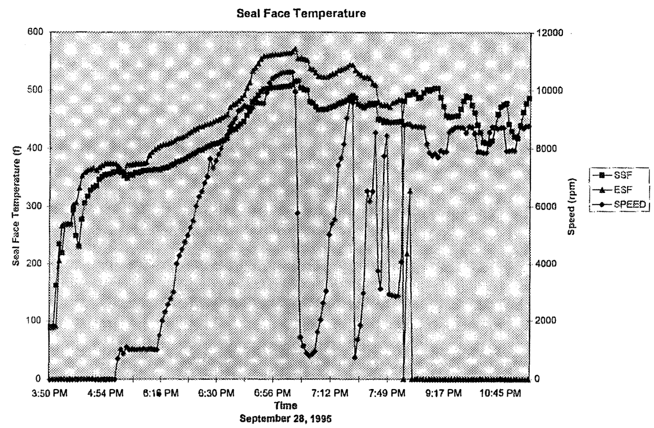


Figure 11. Seal Face Temperature During Startup and Mechanical Testing.

In addition to the normal API 612 [1] testing, the turbine was given a partial load test. The purpose of this procedure was to determine if there would be any impact on the seals while the machine was under load. A summary of the procedure is as follows:

Load Test Run

- Couple the turbine to a water brake.
- Warmup turbine with 175 psig exhaust end pressure using the agreed to procedure.
- Slowroll the turbine at 1000 rpm for one hour.
- Accelerate the turbine to maximum speed of the brake in 500 rpm intervals. Hold each speed for a minimum of five minutes. Record seal temperature, vibration, and bearing temperature data at each speed level.
- Run turbine for one hour or until bearing and seal temperatures are stable.
- Trip turbine.
- Slowroll for five minutes and restart using governor controls.
- Run for an additional 30 minutes.
- Trip turbine and allow to cool.
- Tear down the turbine and inspect the dry gas seals.

During this test, the turbine consumed approximately 800 bhp. The turbine shaft vibration increased from a solo level of 0.3 mils to 0.6 mils. For the first 20 minutes of operation, the exhaust end seal had a very minor leak. After that, it ran completely dry. For the last 10 minutes of the test, the speed was lowered to the damped critical speed. The machine ran well at 6500 rpm with only a minor increase in vibration (i.e., 0.6 mils to 0.8 mils).

OPERATING EXPERIENCE

Procedure Development

Following successful factory testing of all three rotors, both turbines were shipped to the jobsite and installed in their respective process units. From the data gathered during factory testing, it is critical that the plant operators understand the importance of following proven procedures. These procedures were developed based on the seal face and turbine casing temperature data gathered during the testing of the first turbine rotor. The operating procedures are summarized as follows:

- *Turbine Warmup and Operating Procedure for Cold Startup*
  - Inlet and exhaust steam must be superheated and free of water before opening turbine isolation valves.

• Turbine and dry gas seals must be warmed by blowing steam through casing drains for a minimum of one hour before slowrolling the turbine.

• From a cold start, slowroll for a minimum of one hour. "Do not operate at speeds less than 1000 rpm."

• Increase the turbine speed to minimum governor using the electronic governor system.

• *Turbine Warmup and Operating Procedure for Hot Restartup*—If turbine trips and is down for more than five minutes, hot restart procedure is:

• Remove water from turbine casing and inlet steam line. Slowroll turbine for half the amount of time the turbine was down. For example, if the turbine were down for 20 minutes, slowroll at 1000 rpm for 10 minutes before increasing speed. "Do not operate at speeds less than 1000 rpm."

• If the turbine is down for less than five minutes, immediate restart is acceptable.

Operator Training

A formal operator training program was developed to highlight their role in achieving high equipment reliability. This class included the reason for selecting the dry gas seals, the economics of the selection, the problems during factory testing, and the engineering analysis and changes to achieve success. The operators were shown the test data that were the bases for the above procedure. During commissioning of the turbines and compressors, the compressor trains were started several times with several different operators so they could have hands-on experience operating these machines. Each time, the written operating procedure was followed.

Plant Startups

In October of 1996, the distillate hydrotreater was started up. The turbine dry gas seals operated with very little leakage. Using a mirror, minor wisps of steam could be seen leaking from each seal. This small amount of leakage was considered to be normal. The amount of leakage did not change with load; however, there were some minor changes with speed. Over several days, the amount of leakage did not significantly change. The hydrotreater had to be shut down several times for other plant problems. Each time, the recycle compressor was successfully restarted with no turbine dry gas seal issues. A photograph of the installed turbine is shown in Figure 12.

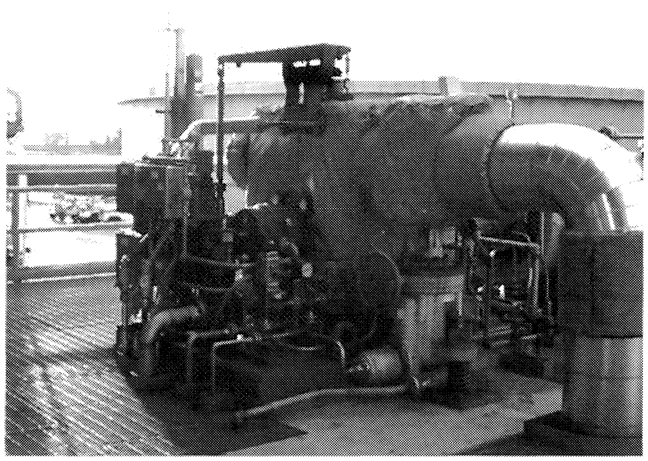


Figure 12. Photograph of Distillate Hydrotreater Steam Turbine While in Operation.

In December of 1996, the coker compressor was started up. The dry gas seals in this turbine responded in a similar manner to the distillate hydrotreater seals. The coker was shutdown in late December for other plant issues and restarted in early January 1997. With each startup, the machine ran very well with minor seal leakage.

CONCLUSION

This dry gas seal implementation to special purpose steam turbines in vital process services is an example of the application of new and innovative technology. When applying new and innovative technology, cost, benefits, and risks need to be carefully evaluated by the user and the equipment manufacturers. The new hardware then needs to be rigorously tested to prove its reliability. If a problem develops, fact-based data must be collected so a complete root cause failure analysis can be done. When evaluating data, use all of the analytical tools necessary to fully understand the probable causes for the problem and determine what mitigation is necessary. Even then, the root cause may not be found with certainty. Make the necessary design change and test the results.

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