RETROFIT OF A STEAM TURBINE WITH DRY GAS SEALS

by

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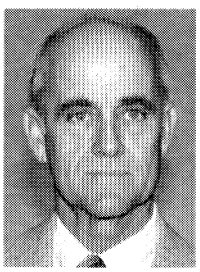

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Prior to joining John Crane, he had over 20 years of experience in a broad range of mechanical equipment applications, design installation, and troubleshooting. This included experience in fluid dynamics/turbomachinery.

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ABSTRACT

The solution to a chronic utility problem, with a critical natural gas compression unit, at the Ammonia plant at Du Pont's Beaumont, Texas facility, is described. The most critical problem, chronic steam leakage from the turbine and subsequent oil system contamination, was corrected by the installation of noncontacting spiral groove seals. In addition, vibration problems caused by coupling lockup and subsequent premature coupling failure were solved by replacing the grease lubricated gear type coupling with a nonlubricated flexible coupling. A feature of the coupling is a keyed hub, designed for hydraulic removal.

The turbine was sensitive to minor upsets and a rotordynamics study was done. The coupling and seal changes were both done at the same time.

The unique features of these designs and measures taken to ensure proper application are addressed along with shop fabrication, field installation, and subsequent operation requirements. A description of project justification of reduced maintenance costs and avoided investments is also included.

INTRODUCTION

In an effort to find a solution to this utility problem, the authors discovered a manufacturer applying existing technology for noncontacting spiral groove dry gas seals to steam turbines (Figure 1). This is the first new development for steam turbines shaft seals in many years and appeared more reliable than current conventional sealing devices. Like the noncontacting spiral groove dry gas seals used in centrifugal compression equipment since the late 1970s, the steam seal uses its process gas (steam) to separate the seal faces by a minute amount, thereby effectively sealing the turbine steam.

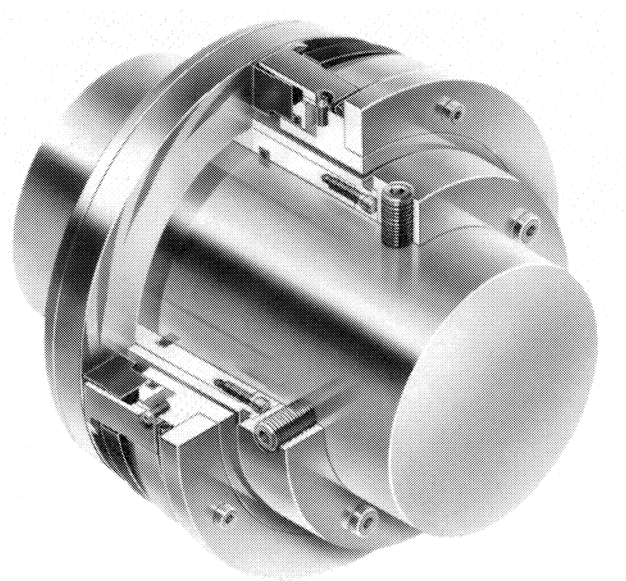

Figure 1. Noncontacting Spiral Groove Steam Seal.

Some of the advantages found in the dry gas seal over the conventional carbon ring bushing are as follows.

- A safer sealing system, reducing potential for injury from leaking steam.
- More reliable, longer life expectancy.
- Reduced steam leakage by 95 percent, improving efficiency of the turbine.
- Reduced bearing oil contamination due to steam leakage, thus decreasing the potential for bearing failure and downtime.
- Eliminated the need for gland condensers, steam extraction and associated piping, reducing maintenance costs.
- Eliminated shaft wear caused by contact with carbon rings.
- Warm up and run in cycles are not required.
- Avoided investment and maintenance of dehydration unit for the oil system.

BACKGROUND

The ammonia plant at Du Pont's petrochemical complex in Beaumont, Texas, was designed and built in the late 1960s, using the latest proven technology of that period. A multistage compressor driven by a medium speed, single stage, noncondensing turbine (Figure 2) is used to boost the pressure of natural gas used as a process feed stock. This compression train is critical to the unit and must be maintained in a high state of reliability. The turbine operates with 550 psig at 610°F steam inlet and discharges into a 200 psig header. The original design used conventional carbon ring seals with a leak off line to a gland condenser, as shown in Figure 3. Typically, carbon ring life is short in high back pressure service. It was not uncommon for the leakage rate to be excessive after running only one week. Every time the plant came down, there was a standing repair order to change the carbon seal rings. These costs are detailed later.

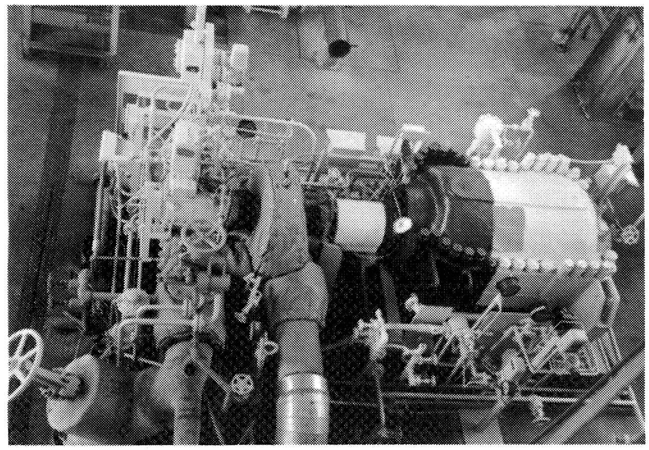

Figure 2. Natural Gas Compression Unit.

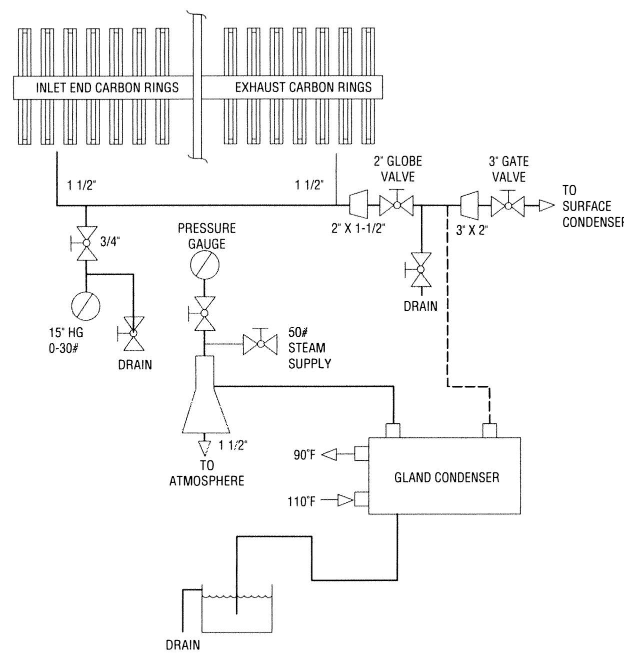

Figure 3. Schematic of Leakage System to Gland Condenser.

The design also included a grease lubricated gear coupling, key fit to the turbine and compressor. This combination of accessories produced a long string of chronic maintenance problems that began in 1968 and extended to early 1993.

These included:

- Safety hazard from steam leakage causing potential personal injury.
- Steam seal leakage that produced water that contaminated the oil system.
- Shaft damage from carbon rings rubbing on the shaft.
- Vibration problems from poorly fitted carbon rings.

- Reduced life for the gland seal condenser from excessive moisture impingement.
- Lockup and wear of the coupling, due to centrifuging of the lubrication causing premature failure.
- Damage to the compressor seals on account of water contamination of the lube/seal oil console.

Applying today's standards and technology would have prevented many of the basic design errors that led to these problems.

PROJECT CONCEPT

Business needs required extended runs (formerly two years extended to three years) and improved reliability. This dry gas seal and dry coupling retrofit project was initiated because it was less costly than the other options. Greater investment was avoided, because the following equipment costs were not needed:

- Larger gland condenser
- Dehydration unit to clean moisture from oil
- Continual seal oil replacement

DESCRIPTION OF STEAM SEALS

Spiral groove steam seals have been in commercial operation for about five years in low duty, single stage, noncondensing turbines. They are similar to conventional mechanical seals, except for wider sealing faces and spiral grooves in the mating ring, as shown in Figure 4. The seal consists of a tungsten carbide mating ring (face) that runs on an extremely thin steam film between itself and the carbon ring.

The purpose of the spiral grooves is to pump a very small flow of steam across the faces of the seal. The small flow of steam causes a separation between the carbon ring and the metal ring, so there is no contact between the two faces. The molecules of steam form a stiff fluid layer between the two faces. This lack of contact allows the dry gas seals to operate for a number of years with no measurable amount of wear.

The flow of steam across the seal face is normally less than 0.4 lb/hr. This is considerably less than the up to 300 lb/hr steam leakage in carbon ring seals.

The steam seal that was installed has the configuration as shown in Figure 5.

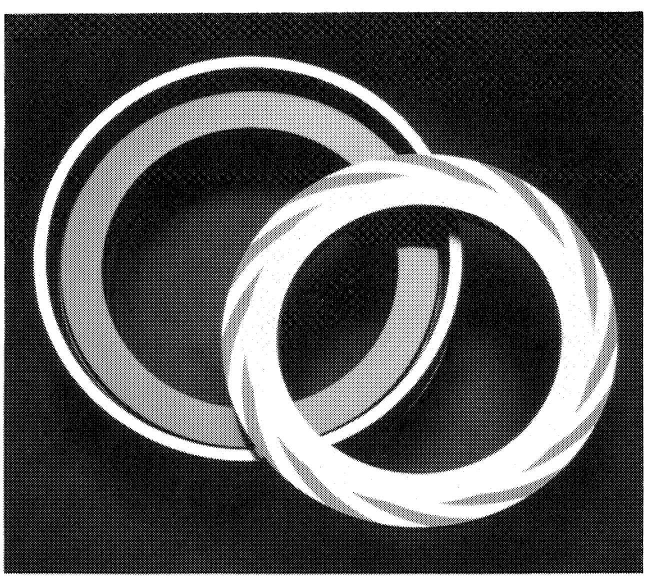

Figure 4. Mating Ring Showing Spiral Grooves.

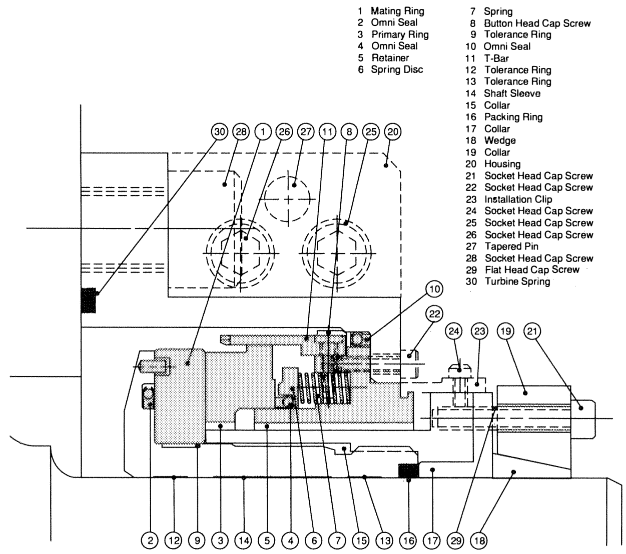

Figure 5. Steam Seal Cross-Sectional View.

ECONOMIC JUSTIFICATION

A work sheet was used to calculate the economic benefits of the dry gas seal. The tabulation is shown in Table 1. Following is an explanation of how the work sheet was used and what the different values mean. Calculations and backup documentation are included for reference.

Cost of seal—The cost of the seal was taken from the quote provided from the manufacturer. OEM price for split carbon rings are approximately $500/set. Split glands can run as high as $5000/each.

Leakage rate/side—The carbon ring leakage rate is based on the back pressure of the unit and OEM information. A steam leakage rate of 200 lbs/hr/seal was used. The noncontacting, spiral groove seal leakage was based on 0.4 lbs/hr/seal (from manufacturer).

Cost of steam—Each plant should be able to obtain a unit cost for steam from their accounting department. The authors calculated the total steam costs from the unit steam cost times total leakage. The cost will vary depending on the availability of fuel and water.

Average repair cost—Repair cost for the steam turbine was comprised of the cost of labor and the parts required to repair the turbine. This information was also available from the accounting department. Preventive maintenance costs and costs for repairing gland condensers was also factored in. The average repair cost for the dry seal was based on replacement of the foil packing, screws, springs, split carbon ring secondary seal, the primary ring, and associated labor for disassembly, clean up and reassembly. An estimated cost of $2300/seal was used. The labor involved for the disassembly and assembly of the steam turbine was assumed to be equal. The dry gas seal eliminated preventive maintenance costs associated with draining the water out of the oil reservoir, and the need for gland condensers.

Expected seal life—the expected seal life for carbon rings was found to be one to two years. Life expectancy depended on the operating conditions and how the carbon rings were "broken in."

The expected seal life of a noncontacting, spiral groove steam seal is five years. There are seals that have been operating for over six years without change out.

Operating hours/yr—operating hours per year for the steam turbine were 7,560. This value took into consideration operation at full load, operation in slow roll condition, and operation in standby mode. This value is identical for both the carbon ring seal and the dry gas seal.

Table 1. Cost Justification — Dry Steam Seal Vs Carbon Ring Seal.

COST COMPARISON
Carbon Ring Seal vs Type 28ST Seal

	CARBON RING SEAL	TYPE 28ST SEAL
Cost of Seal	$719	$13,960*
Leakage Rate/Side	200 lbs/hr	0.4 lbs/hr
Cost of Steam	$0.006/lb	$0.006/lb
Average Cost/Repair Labor: Parts:	$4,320 $2,277	$4,000 $4,800
Preventive Maintenance Cost/Yr	$1,500	0
Maintenance Cost/Yr on Auxiliary Seal Equipment (Gland Condenser)	$1,120**	0
Expected Seal Life	2 Years	5 Years
Operating Hrs/Yr	7,560	7,560
Operating Costs/Year A) Labor on Repairs B) Parts on Repairs C) Preventive Maintenance D) Maintenance, Auxiliary Equipment E) Steam Losses	 $2,160 $1,139 $1,500 $1,120 $18,144	 $800 $960 0 0 $36
Total Operating Cost/Year	$24,063	$1,796
Payback Period		8 Months

AVOIDED COSTS:
- Retubing condenser - Approx. $1,200/Yr
- Purchase of dehydration unit - $14,370

* Initial project including modifications, 2 seals and one time charge for housings.

** $720 - Plant labor, $400 - Contract labor for cleaning, hydroblasting and inspection.

Operating costs/yr—operating costs, including parts, labor, and steam loss, were in real life. The cost for repair of a steam turbine would be applied when it occurs, but for this case, the cost was split over the life of the sealing device. The formulas were used to calculate the three components of operating costs:

$$\text{Cost of labor/yr} = \frac{\text{Cost of labor}}{\text{Expected seal life}} \quad (1)$$

$$\text{Cost of Parts/yr} = \frac{\text{Cost of parts}}{\text{Expected seal life}} \quad (2)$$

$$\text{Steam losses/yr} = (\text{Leakage rate/side})(\text{No of sides})(\text{Cost of steam})(\text{Operation hrs/yr}) \quad (3)$$

These costs were calculated for carbon rings and the dry gas seal and totaled separately.

Payback period— The payback period was calculated by adding together the cost for the noncontacting spiral groove steam seal and the operating costs/yr for the seal, and divided by the operating costs/yr for the carbon ring seal. The payback period was 7.54 months.

ROTORDYNAMIC CONSIDERATIONS

Past rotordynamic performance of this turbine was stable during normal operation, but showed instability when soloed for the purpose of setting the overspeed trip mechanism. The coupling hub had to be removed in order to stabilize the turbine during those occasions. The engineers wanted to improve on this, and make sure the additional weight added to the shaft from the gas seals and lighter coupling did not make the unit completely unstable.

The turbine rotor consists of one wheel with one row of buckets. Steam is introduced by way of a partial arc nozzle block. The arc of admission is from 4:30 to 7:45 o'clock, and rotation is counterclockwise when viewed in the direction of steam flow, toward the coupling end of the machine. Because of the partial arc admission, bearing loading is increased by steam forces when under load and the turbine is stable. If improvements can be made to achieve stability during solo operation, normal operation will be no problem.

The radial bearings are a pressure dam construction with the lower half relieved on the outside ends to increase unit loading. Design diametral bearing clearance is 7.0 to 8.0 mils and dam depth is 10 to 15 mils. The dam depth in use was found to be 12 mils. Published literature suggests that for a bearing with these clearances, generally, a dam depth of 7.0 to 8.0 mils would be optimum. For this particular case, a 15 mil dam depth was optimum.

As previously mentioned, there was no problem with stability when the turbine was under load. To understand why, the engieers needed to sort out how the bearing loads changed. Gravity loading on each bearing was 130 lb down. The 15 mil deep pressure dam in the bearing developed 150 lb of additional loading and moved into a stable region of operation without steam loading effects included. Combining this with the 425 lb per bearing of additional load from partial arc admitted at 1200 hp, it is easy to see why there is no stability problem when the turbine is under load.

Logarithm decrements (log dec) for the turbine with the original bearings found no additional cross coupling excitation. Steam loading effects were + 0.043 with coupling hub installed and + 0.070 with the hub removed. Both were positive indicating stable operation. Since the turbine was unstable with the coupling hub installed, there must have been some additional cross coupling excitation in the turbine beside that produced by the bearing alone.

Exploring further, it took 3700 lb/in of cross coupling excitation at the rotor midspan to reduce the log dec to zero with the coupling hub mounted. This level was called the mid span cross coupling excitation. With the coupling hub removed, a log dec of + 0.027 indicated stable operation as observed in the field. The mid span cross coupling threshold with the coupling hub removed was 6000 lb/in, 60 percent more than when the coupling hub was mounted.

By changing to a dam depth of 15 mils, the log dec increased to a stable + 0.051 with an original destabilizing of 3700 lb/in of mid span excitation, all with the coupling hub installed. The mid span cross coupling threshold with the coupling hub in place was raised to 8700 lb/in, a 135 percent increase.

The response of the rotor has always been acceptable and remained so after the dam modification, so no additional changes were made. The new bearings with the 15 mil deep dam have been installed along with the new gas seals and nonlubricated coupling. The turbine was returned to service and solo operation with the coupling hub mounted was then stable.

As a final note on the preceding stability analysis, the reader might find it informative to go back over the information with the intent of evaluating the value of the term log dec in such analyses. The term midspan cross coupling threshold seems more informative in comparing design changes for stability improvements.

INSTALLATION OF SEALS

The turbine case assembly was removed and sent to the OEM shop for repairs and installation of dry steam seals. When the machine was opened for inspection, the following deficiencies were found:

- Several oil passages in the case were about 50 percent blocked by buildup of debris.
- Varnish deposits were found on parts caused by oil breakdown.
- Numerous carbon rings in the original gland seal packing boxes were broken. (Figure 6.)
- The carbon ring boxes were eroded on the down stream side of the sealing area.
- Bluing of the horizontal split casing showed little contact in the gland face area.
- The trailing edges of the nozzle block vanes were very thin from erosion with some sections missing.

These deficiencies came as no great surprise, since this was the first time this machine had been removed and case inspected since installation in 1963.

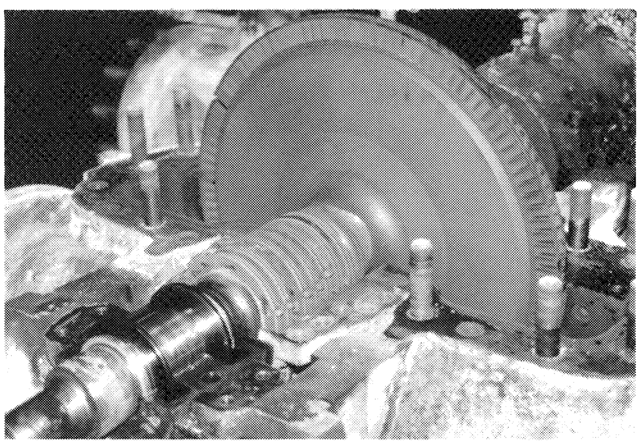

Figure 6. Turbine with Original Carbon Rings.

MACHINING AND REPAIR WORK:

In order to return the turbine to original design specifications the following work was performed, although no special machining was required to adapt these seals for this installation.

- Majority of sealing faces were refinished.
- Case horizontal split was ground.
- Inlet and discharge faces were machined.
- Trip and throttle valve faces and valve seats were machined.
- Eroded nozzle block was corrected by weld overlay.
- Other deficiencies were corrected as needed.

A progressive balance procedure was done on the spare rotor before seals were installed. Once the seals were installed on the shaft (Figure 7), no further balance checks could be made because some parts of the cartridge were stationary in service. For this reason, it was recommended that the rotor be progressively balanced to minimize effects of removing and reinstalling rotor parts.

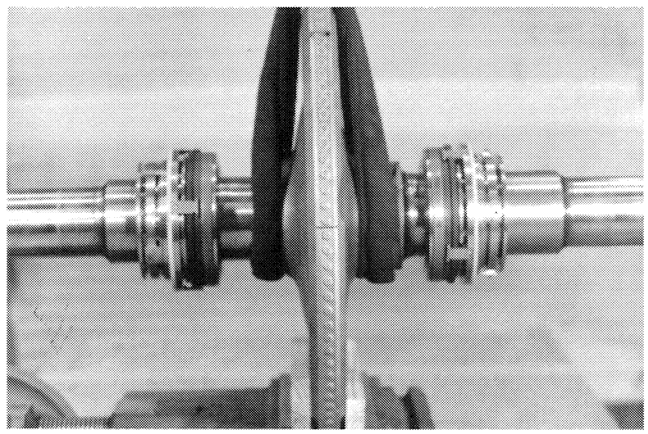

Figure 7. Turbine Rotor with Dry Steam Seals Installed.

INSTALLATION

- To mount the seal cartridges on the shaft, all rotating flinger rings, thrust disc, coupling hub, and worn gear were removed.
- The bare rotor was balanced, thrust disc and the remaining accessories were added and balanced with the assembly.
- The rotor was installed in the lower case and the running clearance of the nozzle block was established at 0.54 in and the thrust running clearance of 0.011 in was set.
- The upper case was set and the seal gland faces were checked relative to the shaft at that time. They were 0.004 in TIR. The seal manufacturer required they be within 0.020 in of being square with the shaft, so no further corrections were made. This tolerance came about because the axial springs that adjust for this variation were mounted in the stationary part of the seal. The seal was able to accommodate + or -0.100 in axial changes as well.
- The split seal housings were installed and swept with an indicator. The housings were within 0.003 in TIR. the seal manufacturer asked for this to be within 0.005 in TIR.
- The upper case was then removed. All accessories were removed from the rotor and the seals were installed on the rotor. The rotor was set in the case, the seals located and secured (Figure 8), and the turbine assembly was completed.

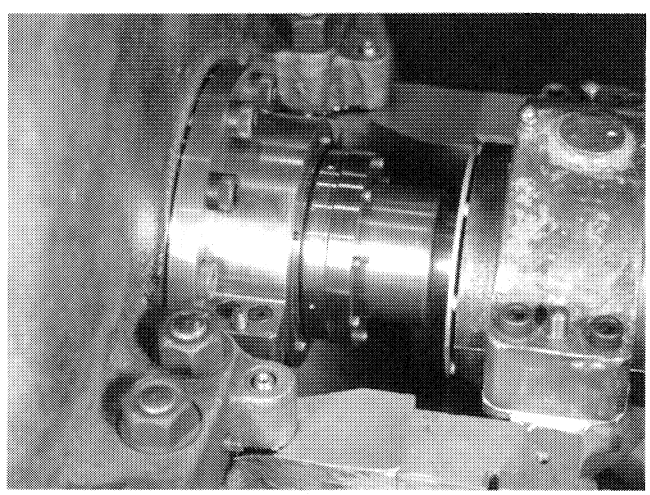

Figure 8. Completed Installation of Dry Steam Seal.

SUMMARY

The machine was returned to the plant after the new dry coupling was installed. The machine was reinstalled on schedule, and the unit was started up. The seals have operated virtually trouble free since plant startup. There was no way to measuring seal leakage, except to place a flag (tubing with a rag attached to it) in the area of the seal and detect no movement, collection of condensate, or other indications of leakage.

The authors are presently looking at the possibility of two other applications in the ammonia unit and at other locations in the company.