APPLICATION OF DRY GAS SEALS ON A HIGH PRESSURE HYDROGEN RECYCLE COMPRESSOR

by

Donald R. Carter

Craft Area Supervisor

Marathon Petroleum Company

Illinois Refining Division

Robinson, Illinois

Donald R. Carter is Craft Area Supervisor for Marathon Petroleum Company in Robinson, Illinois. He is responsible for electrical, instrument, and machine shops. He has worked for Marathon since 1974.

Mr. Carter previously worked for P.M.C./Beta Corporation and Shell Oil Company.

He has extensive seminar experience, and has served on various professional committees and is a Senior Member of ISA.

ABSTRACT

Maintenance and operation of high pressure seal oil systems on centrifugal compressors has been a major problem faced by refinery operators. This problem is intensified when sealing gases like hydrogen and mixtures of hydrocarbons. High pressures, low vilcosity, and solubility in oil pose unique problems in sealing with conventional wet seal oil systems. A breakthrough in sealing high pressure hydrogen with a gas seal design, which incorporates a natural breakdown of pressure across two seals without external pressure or flow controls is described. The retrofit has resulted in increased safety, low maintenance, and ease of operation; thus eliminating seal oil leakage and contamination.

The recycle gas compressor that was retrofitted operated on 94 percent hydrogen at a discharge pressure of 12,896 kpa (1870 psig), a discharge temperature of 71°C (160°F) and an operating speed of 10,250 rpm.

The problems that existed with the conventional seal oil system, the economics of the retrofit, the applicable dry seal design, process problems encountered during the retrofit and startup of the compressor and the successful resolution of these problems are covered. Operating data on the test bench and in field service are described and discussed.

INTRODUCTION

Maintenance and operation of high pressure seal oil systems on centrifugal compressors has been a major problem faced by Marathon. This problem is intensified when sealing gases such as hydrogen with mixtures of hydrocarbons. High pressures, low viscosity, and solubility in oil pose unique problems in sealing with conventional wet seal oil systems.

A breakthrough is described for sealing high pressure hydrogen with a dry gas seal design, which incorporates a planned breakdown of pressure across two seals without external pressure or flow controls. This dry gas seal retrofit has resulted in increased safety, low maintenance, and ease of operation by eliminating seal oil leakage and contamination.

The recycle gas compressor retrofited operated on an average of 90 percent hydrogen at a discharge pressure of 2050 psig, a discharge temperature of 165°F, and an operating speed of 10,250 rpm.

HISTORY

The need for finding a better "mousetrap" for the 20-year old barrel high pressure hydrogen compressor, became apparent as the problems of operating the existing oil (non-contacting seal) system were reviewed. The problems were as follows:

- Oil usage
- Length of time before seal replacement
- Process problems (seal contamination due to chlorides)
- Large delta P between suction and discharge ends
- Operator training
- Startup problems
- Instrumentation requirements.

There was an oil usage problem with the compressor. Immediately following an overhaul, oil usage would begin at approximately three 55-gallon drums per day. As the seal became dirty, oil usage would increase to approximately ten drums per day. Finally, as the oil usage continued to increase, the seal support system became overloaded, and eventually the usage would reach as high as 30 drums per day (Figure 1). At this point, the unit would be shut down to replace the seals.

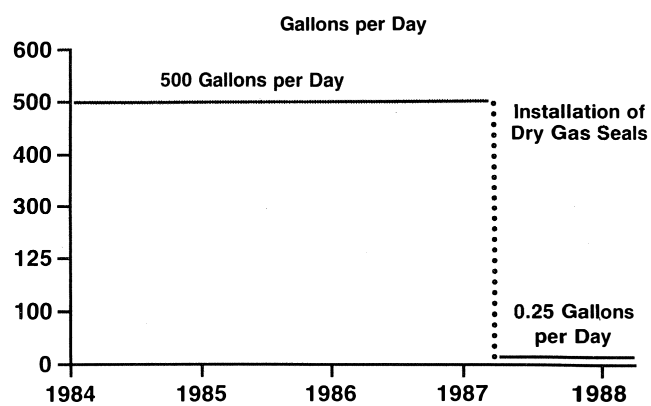

Figure 1. Average Oil Usage.

The normal life expectancy of the unit catalyst should be approximately 18 months. The compressor records show an average of greater than one seal change-out between catalyst exchanges.

The other problems, as mentioned, also contributed to the high operating costs of the compressor. Constant work with the

OEM did not improve the performance of the seals. Something had to be done.

FINDING A SOLUTION

Looking for a better sealing method for this machine had proved fruitless. No one with an improved sealing system could be located.

The concept of dry running seals was introduced. At this time, no one could be found with experience with dry running seals in a hydrogen service at 2,000 psig. Two dry seal manufacturers were approached about this compressor. John Crane took on the task to design a cartridge seal for the service that would fit the machine without modification.

With safety of the operation being the number one priority, alarms and backup instrumentation were required to be incorporated into the design. After several months of testing and design engineering, the manufacturer submitted a proposal.

The proposed seal design can be seen in Figure 2. A triple seal was used so the sealing pressure, approximately 1800 psig, could be broken down over Seal A at a ratio of approximately 50 percent. Seal B would then handle the remaining pressure. Seal C would then remain as backup in case one of the other seals developed a problem.

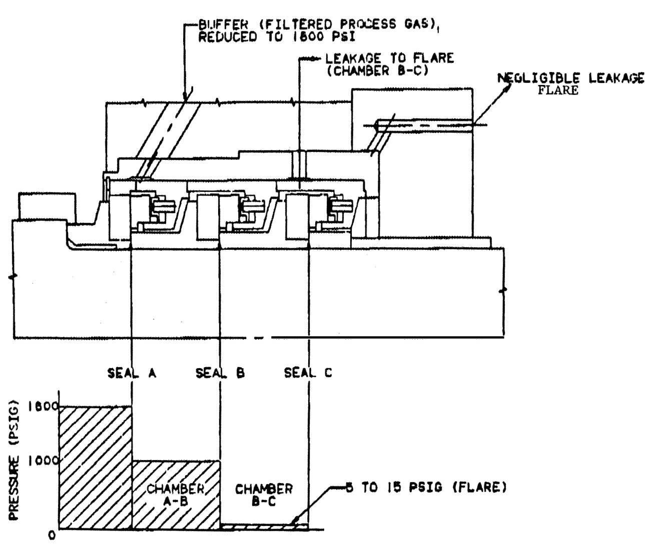

Figure 2. Type T28AT Triple Seal (Marathon-Elliott Discharge Shown).

The breakdown of the pressure is a function of tuning in the seal faces and not of external control. The faces operate in conjunction with the other; thus, if seal A were to fail, Seal B would become the breakdown seal, and Seal C would become the sealing seal.

The seals were to be monitored with an instrumented panel, allowing the user monitor and alarm/shutdown the following conditions of the seals:

- Buffer gas flow/pressure
- Buffer gas filter D/P
- Interstage pressure Seal A
- Interstage pressure Seal B
- Seal leakage A and B
- Seal leakage to flare.

These conditions met the standards for equipment protection.

PRINCIPLE OF OPERATION

Basic Seal Design and Principles:

Spiral groove seals work with the viscosity of the fluid to complement the pumping function. The two sealing surfaces are so tightly controlled that the boundary layers of the sealed viscous fluid overlap. The smooth sealing surfaces drag the sealed fluid circumferential in the direction of rotation across the grooved surfaces. The spiral grooves are logarithmic or a similarly controlled function which tends to deflect the fluid in a radial motion. The intensity of the "pumping" strongly depends on the viscosity and the distance between the two sealing surfaces.

The shear stress (T) of a gas can be written as:

$$T = U\,dV/dY$$
$$U \underline{\Delta} \text{ Viscosity}$$
$$V \underline{\Delta} \text{ Local Velocity}$$
$$Y \underline{\Delta} \text{ Height of Gap}$$

The viscosity of a fluid is that property which determines the amount of resistance to a shear force. Viscosity is due primarily to the interaction between fluid molecules. Consider two parallel plates at a small distance "Y" apart, the space between plates being filled with a fluid, and the upper plate is moving with a velocity "V." The fluid in contact with the upper plate will adhere to it and will move with a velocity "V," and the fluid in contact with the fixed plate will have velocitY zero. Experiments show that the force varies with the area of the plate, with viscosity "U," with velocity "V" and inversely with the distance "Y."

Based on these assumptions, the dry gas seal utilizes spiral grooves starting at the outer edges of the seal face extending across part of the sealing surface to gather in gas and compress it into a smaller chamber, causing a separation between the two faces. The faces are rotated in a direction opposite that of an impeller (Figure 3). Thus, the face separation is dependent on parameters such as viscosity of the gases, speed, depth, and shape of the spiral grooves. The manufacturer used these elements to design the pressure breakdown and minimize seal leakage.

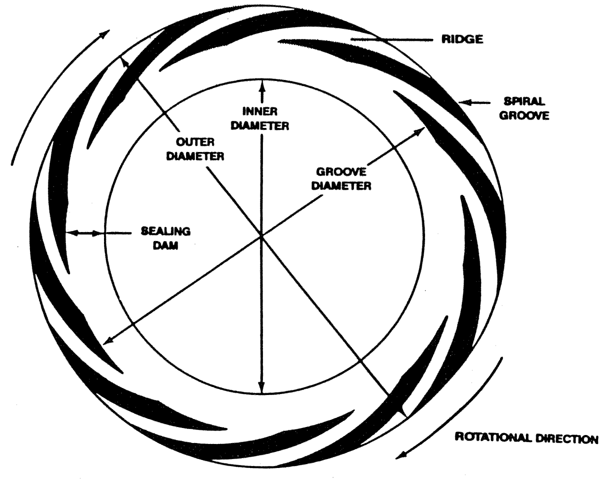

Figure 3. Spiral Groove Pattern.

COST JUSTIFICATION

The task now was to justify to management that this was a viable economic solution to the oil usage and seal replacement problems. The initial cost of the retrofit was approximately $130,000.

Five years of maintenance history/cost was used for justification of the seal replacement.

Based upon the cost of these problems, the total cost of operating with oil seals was calculated to be approximately $1.7 MM. The calculations are shown as follows:

Oil Usage

10 drums per day × 365 days = 3,650
drums × 55 gallons × $2.19 per gallon = $ 439,642.50

Seal Replacement

The cost = labor and material
Cost per seal replacement = 43,258 ×
2 times per year = $ 86,516.00

Unit Downtime

To replace seals average 4 days charge rate = $ 492,800.00

Manhours to Handle the Lube Oil

72 hours per week × 52 weeks = 3,744 ×
$25 per hour = $ 93,600.00
(see Figure 3)

Operators Salary While the Unit is Down

4 operators per shift × 3 shifts = $ 4,800.00

Treatment of Contaminated Lube Oil Drained to the Oily Sewer:

550 gallons per day × 365 = 200,750 gallons/
year × .26 = $ 52,020.00

Horsepower Required to Operate and Maintain the Seal Oil System

Seal oil pumps: One 75 HP Electric
One 75 HP Steam Turbine
Vacuum Pump: 7.5 HP Electric
Transfer Pump: 2.5 HP Electric
Oil Shear HP: 75 HP
Total = 235 − 75 spare = 160
160 HP × 746 ÷ 1000 = 119.3 KWH ×
$0.041 × 24 × 365 = $ 42,847.79

Hydrogen Loss to Flare:

Leakage rate = 1,251 SCFM (see Figure 2)
Minutes in a day = 1,440
Days per year = 365
Cost in 1,000 cubic feet = $.735
Fuel Value
1,251 × 1,440 × 365 − 1,000 × $.735 = $ 483,281.00

Total of the Above:
Represents one year cost to operate the oil bushing seals: $1,695,507.20

Table 1. Oil Usage

Average per day	10 drums
Operator: add oil, move drums, pull samples, etc.	2 hr/shift
Lab Tech: pick up sample and run.	½ shift
Materials: handler whse. receive/delivery pick up empty drums	8 hrs/week
Labor: clean/wash drum ready for shipment.	2 hours per day
Purchasing/Accounting: process requisitions, paper/payment	2 hrs/week
Total Manpower to handle oil:	72 hrs per week

(The above are based on the average usage of 10 drums per day.)

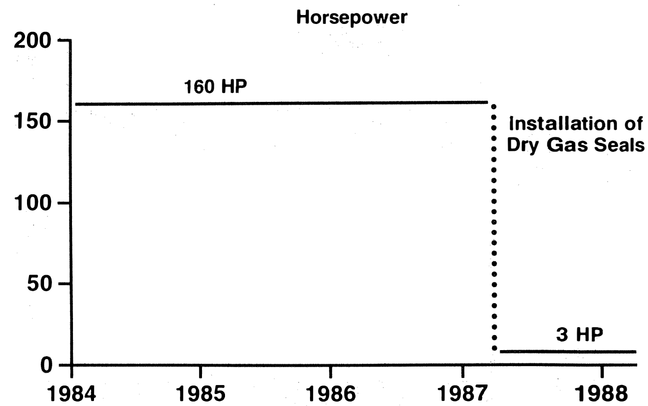

Figure 4. Parasitic and Support Horsepower Loss.

R = 766.8
Sg = 0.0695
K = 1.14
P_1 = P + 14.7 = 1,650 + 14.7 = 1,664.7 psia
Delta P/P_1 = 1,648/1,644.7 = 0.989
d_2 = 1.049
B = 0.750 + 1.049 = 0.715
Y = 0.65
C = 0.71 turbulent flow assumed
T = 460 + 130 = 590°F

$$P_1 = \frac{(2.70)\ (1{,}664)\ (0.0695)}{590} = 0.53$$

$$W = (0.525)\ (0.65)\ (0.125)^2\ (0.71)\ [(1{,}648)\ (0.53)]$$

W = 0.11 #/Sec or 396 #/M or 198 Moles/M
PV = NRT

$$V = \frac{(198)\ (10.72)\ (520)}{14.7}$$

V = 75,083 ft^3/M or 1,251 SCFM

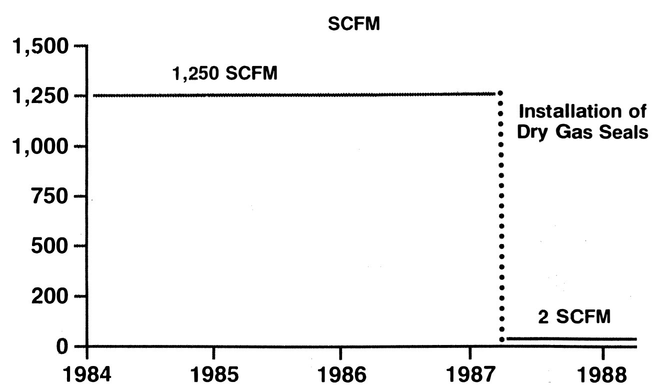

Figure 5. Hydrogen Loss to Flare.

ADVANTAGES OF THE DRY RUNNING SEAL

- Elimination of the oil system
- Simple operation
- Reduced hydrogen loss
- Elimination of the horsepower to shear the oil in the seals
- Elimination of unit shutdown to replace the oil seals

- Longer life
- Safety of seals.

The mechanical aspects of the retrofit were now taken into consideration. Removing the oil seals would possibly have an effect on the stability of the rotor. The oil support in the seal cavity would no longer be there for damping purposes. The gas seals were designed to be approximately the same weight as the old seals.

After reviewing the parameters that affect the stability of a rotor and cause subsynchronous vibrations, the user elected to proceed with the installation of the dry gas seals.

RETROFIT

As the unit neared its scheduled turnaround, the oil usage was averaging 18-20 barrels/day. The unit was shut down and readied for maintenance. The machine was torn down, cleaned, and inspected. All worn parts were renewed. The old seal sleeves were removed from the rotor; it was cleaned, inspected, and balanced.

This retrofit would be accomplished without the aid of the compressor manufacturer. All dimensions for the new seals were gathered from a prior seal change.

The new dry seal cartridges were placed on the shaft to ensure a correct fit. All other dimensions were double checked, and all the porting holes were rechecked. Next, the rotor was placed back into the bundle with dummy bearings. All labyrinth clearances were corrected including the balance piston clearance. With this accomplished, the bundle was reinstalled into the barrel. The radial bearing bottoms were installed. With the dimensions acquired from the rotor fit, the rotor was located, and the thrust bearing set. The T dimension was taken and recorded to ensure proper rotor location. The bearings were removed and the seals installed. All seal porting was again checked. This triple seal would require several ports to enable us to monitor interstage pressures and leakages. The rotor was then placed back on its bearings, proper clearances achieved, all temperature and vibration probes installed, and the machine was buttoned up.

Two tests were performed while the compressor was in a rest position. The tests were accomplished with nitrogen. The nitrogen was introduced into the belly drain of the compressor. The compressor was pressured statically up to 2,000 psig. The first test consisted of dial indicators placed on the barrel to shaft to measure the case extrusion under pressure and temperature. This test was performed to ensure the O-ring fits, and critical dimensions of the seal would remain in tolerance, as the barrel extruded under pressure. The test revealed the case extrusion to be approximately 28-30 thousandths. The amount of growth did not affect the performance of the seals.

A second test was performed to measure the static leakage of the seals and compare it to the test stand leakage. This comparative test would reveal any damage that could occur during installation. Examples are cutting an O-ring, seals not properly seated, etc. After satisfactory results were achieved, the machine was realigned using a laser alignment tool and coupled up.

The new seal operation procedure which consists of (1) ensuring the buffer gas line is open, and (2) the flare line for leakage is open was reviewed with the operators.

Finally, the machine was ready for operation.

STARTUP

Show time: 2:00 a.m. Sunday morning. The unit was pressured up to 250 psig. The lubrication oil pumps were started, and the oil temperature adjusted to 95°F. The product (nitrogen, ammonia, and some moisture) that the seals would experience during startup would not be the same as the hydrogen product that the machine would operate with during normal conditions.

The possibility of using nitrogen as a buffer to the seals during startup had been discussed. The system had been designed so that nitrogen could be hooked up and used during startup until stable unit conditions were achieved, then the source could be switched on the run back to the discharge slip stream. The decision was not to use the nitrogen. The seal buffer line was opened up to the discharge of the machine, and the machine was started.

The unit came up to speed and the interstage pressures stabilized. Normal startup procedure was underway.

The operators immediately got a high differential alarm on the buffer gas filter, the filter was switched, and the dirty filter replaced. The 10 micron filter was examined and found to be full of moisture. With this taken care of, the operators now had an opportunity to observe the vibration readings on the compressor. The questions regarding possible problems with the rotordynamics were answered. The initial readings showed a reduction in overall vibration. Compared to past performance history, an overall average reduction of better than five tenths of one mil were realized. Further indepth analysis proved the initial observations to be correct; this improvement in vibration was an unexpected benefit. The operators could now direct their attention to monitoring the performance of the seals and machine as the unit heated up and came up to pressure.

After about twelve hours of dry out operation, the unit had to be shut down because of a leak on one of the reactors. Again, the compressor was restarted without incident. The confidence with the seal modification was growing.

INTRODUCTION OF MOISTURE

After the unit was on line, and with the seals performing well for about 8 months, the users began to observe the interstage pressures start to move. The investigation revealed condensing liquid which is entrained in the process as the ambient temperature dropped to around freezing (Figures 6 and 7). The solution to this problem was relatively easy to resolve. A small amount of steam tracing and insulation around the lines and filters kept the temperatures elevated. With the moisture problem solved, the seals returned to normal operating conditions.

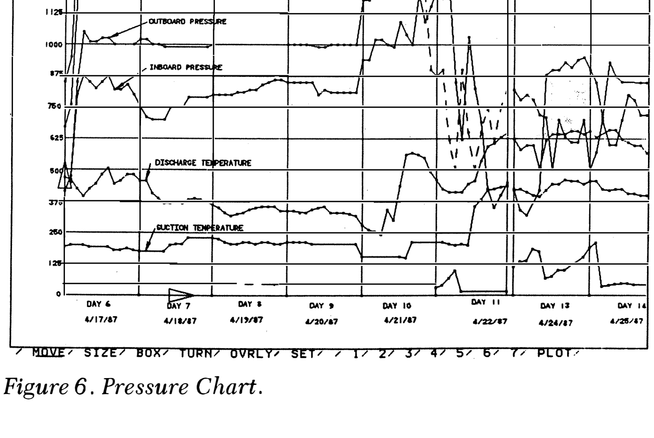

Figure 6. Pressure Chart.

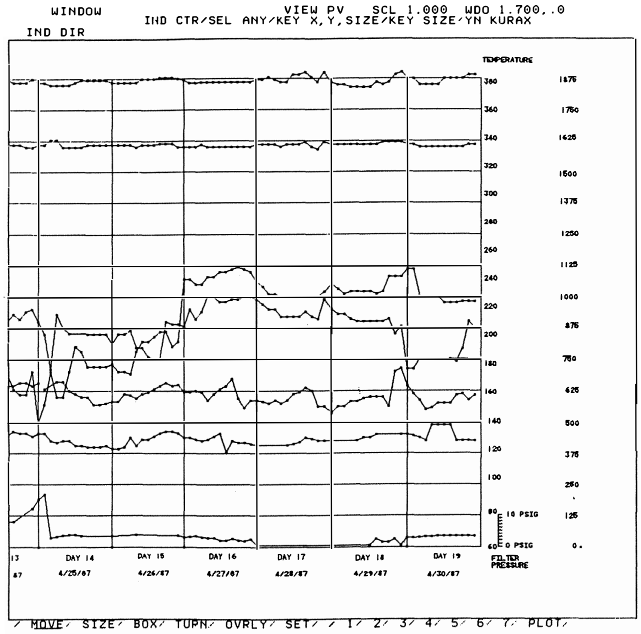

Figure 7. Pressure Chart (continued).

One point should be made here. The ability of the seal to handle the problem with moisture increased confidence in the system's ability to handle situations other than ideal without catastrophic failure.

CONCLUSION

The compressor has been on stream a little over one year. The performance of the seals has been flawless. The unit is scheduled for a September turnaround. Plans are to remove the seals and examine them. This eighteen-month run will be a record for the compressor in its 20-year history. The examination of the seals after their service of one and one-half years will help establish future planned seal replacement schedule.

This was the users' first experience with dry running seals. The documented savings and the ease of operation has made this design one of the most satisfying experiences during the author's 22 years in the rotating and reciprocating equipment field. Since this installation, the company has installed four other single gas seals in two other split case machines in wet gas service, with results almost as gratifying as the described herein (Figure 8).

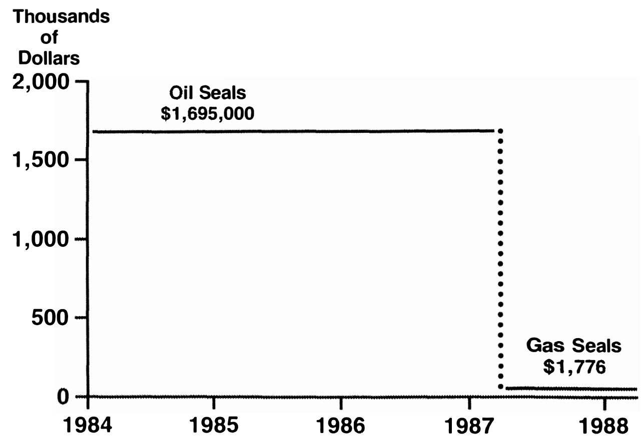

Figure 8. Operating Expense Oil Seals vs *Gas Seals.*