ASPECTS REGARDING THE OPTIMIZATION OF THE USE OF HORIZONTAL CENTRIFUGAL COMPRESSORS

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Abstract: *This paper analyses the possibility of replacing the existing oil seal of axial centrifugal compressors with a dry seal, in order to increase their working performance. Existing dry seal variants have been studied with a view to showing the possible sealing variants of a horizontal centrifugal compressor, as well as the algorithm for calculating the efficiency and power consumed by the compressor.*

Key words*: horizontal centrifugal compressor, dry seal, oil seal*

1. Introduction

Converting the available energy in nature into energy, that finally, in various forms, becomes useful in a particular context of industrial activities, materializes through physical processes, which take place in industrial systems and installations, processes named technological processes.

In relation to work systems, the development of these technological processes implies different process strategies. This implies that the distribution of machines and machinery in a working system, which performs that technological process, corresponds to a logical and technological sequence. In essence, a working system comprises an engine machine (mechanical kinetic energy generators) and an engine driven by the machines (in particular, a generating machine or a working machine). The generating machine is the essential machine of the working system, whose functional and constructive solution is directly influenced by the level of effort variables (forces, moments, pressures) and by motion (linear speeds, angular speeds, flows).

Compressors are generating machines that increase the pressure of the gaseous fluids. Compressors must be driven by an electric motor, internal combustion motor or gas turbine, and they are directly coupled or by means of an intermediate transmission. According to the functional principle (according to the weight of the gas – energy potential energy positioning components, potential pressure energy or kinetic energy) these compressors or gas pumps can be:

- Volume compressors - compression is performed by means of a moving element, the volume of closed gas inside the working machine is reduced (intermittently, volumetric volume). The main part is the potential component of pressure. This category includes piston compressors and rotary compressors or blowers. Volume compressors provide a high compression ratio and, depending on the number of stages, these compressors can have discharge pressures up to 250 MPa. As disadvantages, these compressors make uneven flows rates, they work at low speeds of up to 25 revolutions/minute rpm, and their starting is done without load, and have large size.
- Dynamic compressors – characterized by variation of the component of kinetic energy and partly, of the component of pressure energy by passing the gas through a blister engine. The compression process is followed by a partial transformation of the kinetic energy into potential pressure energy.

Dynamic compressors are:

- ✓ Mono or multi – stage radial centrifugal compressors
- ✓ Mono or multi – stage axial (horizontal) centrifugal compressors.

These compressors have advantages as compared to other types of constructions, they have a constant flow, they are safe to operate and can be integrated into an automated system. Also, the adjutant of the functional parameters is possibly achieved by changing the engine speed – which can be a turbine (preferably in the case of speeds greater than 60 rpm or electric drive type VFD (variable frequency drive). These types of constructions allow the adjusting of the functional parameters and by rolling the gas into the suction line – the suction pressure is changed. For flows higher than 84 000 (85 000)/ m^3/h it is better to use axial compressors than radial compressors, due to the following advantages: higher yields, lower weight, and reduced gauge.

- Jet compressors - perform a mixing of a motor fluid (compressed air or steam) with the pumped fluid, resulting in a medium pressure current. These types of construction have no mobile elements (as in the previous cases), and they are provided with ejectors. These are used as vacuum pumps, and constructively are simpler, robust, cheap and easy to operate. As main drawbacks, the ejectors have a low efficiency, they have a slow start, the motor

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fluid consumption is high, but also the pumped fluid is mixed with the motor fluid (therefore, it is best to use steam as a motor fluid).

2. Types of seals for horizontal centrifugal compressors

a. Mechanical contact seals

Mechanical contact seals with the role of minimizing losses through axial centrifugal compressor shaft seals (Fig. 1) [1]. These seals have a real sealing surface located between the seal components perpendicular to the compressor shaft, unlike conventional radial seals. This has enabled rotating and stationary components to interact during operation, resulting in improved sealing capability with low seal seals.

Mechanical seals are wet seals, which require the use of oil seals and sealing gaskets, which are complex. Some of the sealing oil can be lost in the compressor, causing oil consumption, but more importantly, the contamination of the compression process. A portion of the sealing oil comes into contact with the process gas. This sealing oil compromised by the process gas has to be treated / recovered or removed. Disposal of contaminated oil requires compliance with environmental protection procedures. On the other hand, the process – seal gas mixture may cause problems in lowering the ignition point of the oil/gas mixture. Decreasing the efficiency of the sealing system leads to lower compressor efficiency, which leads to increased energy consumption (from the motor engine), necessary to ensure the functional parameters required by the gas compression process. Therefore, this paper discusses the possibility of replacing this mechanical seal or wet sealing. Further, dried sealing systems will be studied and presented.

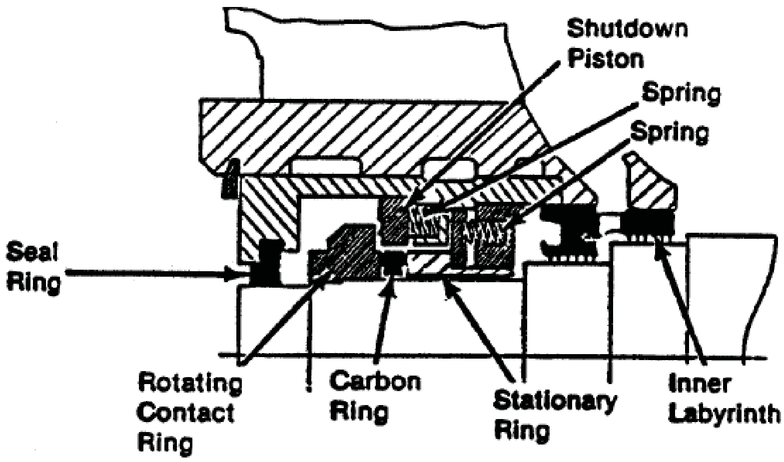

Fig. 1. **Mechanical contact seals [1]**

b. Dry Sealing systems

Dry sealing systems have a number of advantages, necessary for compressing the gases in the catalytic reformer installation:

- Eliminating the contamination of oil pipelines from wet seals and annual savings due to the sealing oil losses and the need to fill it in the tank.
- Savings due to elimination of the system for the transport and regeneration of the oil required for sealing, as well as savings due to the elimination of the nitrogen used for the regeneration of the oil recovered from the wet sealing.
- Eliminating process gas contamination with sealing oil (and, therefore, changing the compressor's functional parameters).
- Increasing the operational safety and reliability of the compressor
- Reducing emissions of pollutants into the atmosphere.

1. *Single Gas Seal*

A single gas seal (figure 2) consists of a single joint seal and a primary seal assembly. Inside the dry gas sealing supply is an internal labyrinth seal that separates the process gas from the sealing gas. A clean and dry gas is injected between the labyrinth seal and the gas sealing gasket, providing the sealing fluid required for sealing. At the outside of the dry gasket is a gas barrier seal from the compressor shaft bearings. The main function of the sealing gasket is to prevent the lubricating oil from migrating into the gasket.

Gas seals are commonly used only in case of low pressure or non-hazardous process gas applications where they can tolerate operation without a spare sealing system. [2].

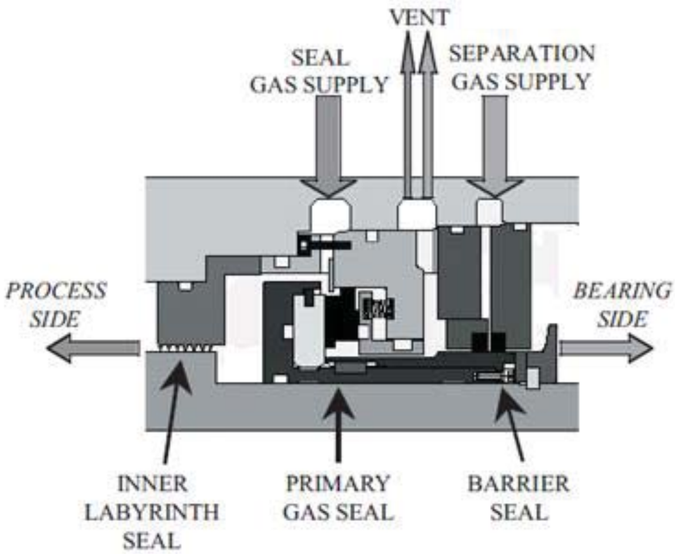

Fig.2. **Dry gas seal [1]**

2. *Tandem Gas Seal* [1]

The tandem-style sealing system is most commonly applied, essentially provided with two single –gas seals – a primary seal and a secondary one – seated in a single sleeve. During normal operation, the primary seal absorbs the total pressure drop on the compressor ventilation or ventilation system and the secondary gasket serves as a reservoir if the primary seal doesn't work. As with single sealing, clean and dry gas is injected between the inner labyrinth seal and the primary gas seal and the small amount of sealing leakage is directed to the primary ventilation system. Even smaller amounts of sealing gas pass through the secondary seal and outside the secondary valve. Most of the flow through secondary ventilation is the injection gas injected into the barrier seal.

Tandem gas seals are common in the process gas industry where it is necessary to operate with a safety seal. [2]. Tandem sealing systems have been applied in almost all types of processes to very large sealing pressures.

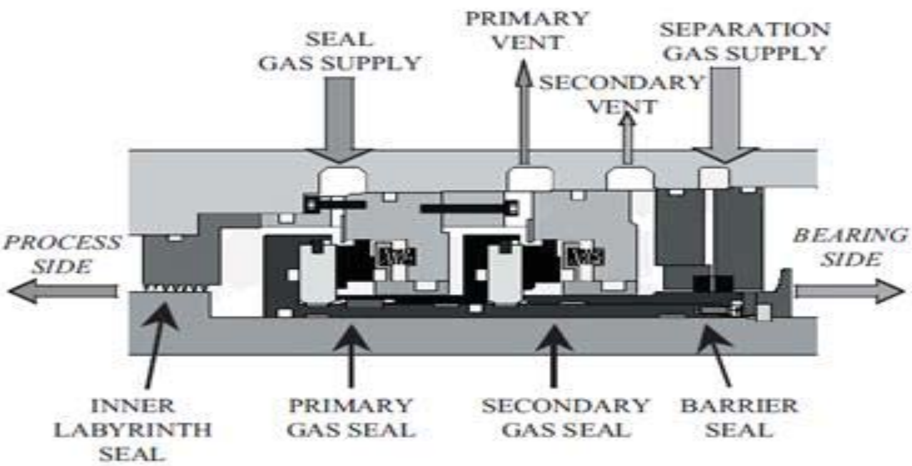

Fig.3. **Tandem gas seal. [1]**

3. *Tandem Gas Seal with Intermediate Labyrinth* [1]

This configuration [1] consists of a labyrinth sealing gasket installed in the primary and secondary seals and applies when leakage of seals in the secondary vent cannot be tolerated. It works in the same way as a gas tandem gas seal in that the primary gas seal absorbs the total as a reserve. [2] The purpose of the intermediate seal of the labyrinth is to prevent leakage from the primary seal.

To achieve this, the intermediate labyrinth is injected with inert gas at a pressure higher than the vent pressure. Gas labyrinth sealing systems are used in the process gas applications where it is necessary to eliminate leakage of sealing gases to the secondary air, which is often vented locally in the atmosphere.

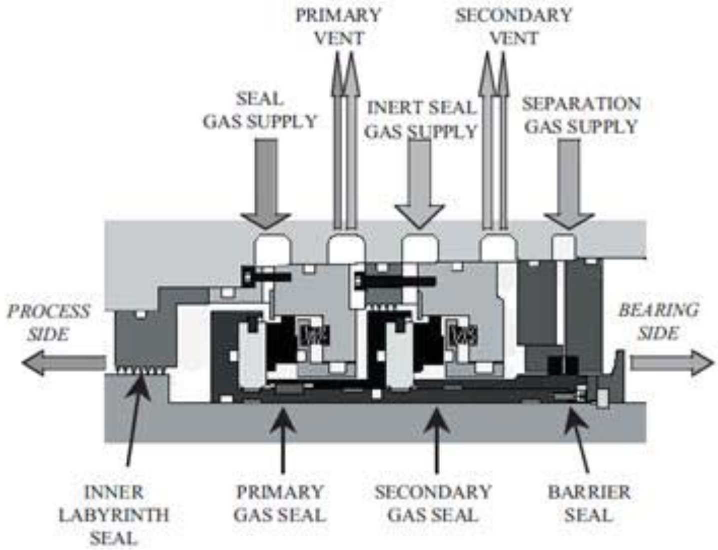

Fig.4. **Tandem gas seal with intermediate labyrinth [1]**

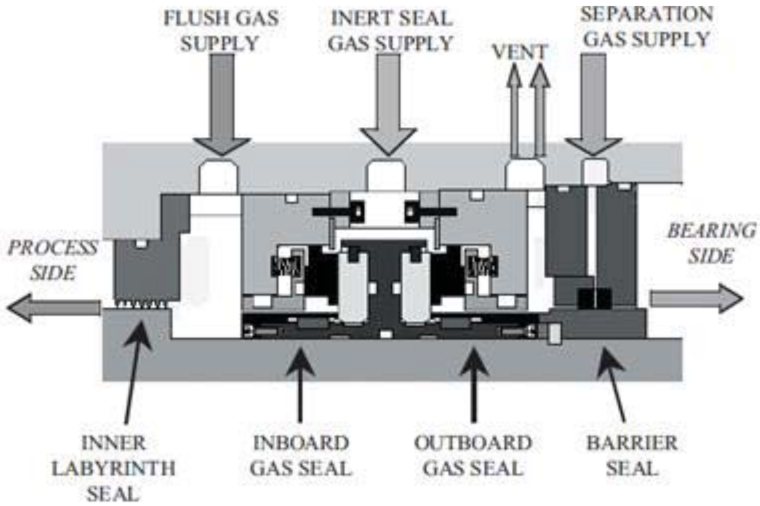

Fig.5 . **Double opposed gas seal [1]**

4. *Double Opposed Gas Seal*

Double gas sealing is similar to tandem gas sealing, except that the two seals work in parallel in a back-to-back configuration. The dry double seal is made by injecting with an inert gas sealant (e.g. nitrogen), which is injected between the inboard gas seal and the outboard gas seal. To ensure that the process gas in the compressor does not come in contact and eventually contaminate the interior seal, normally a clean and dry washing gas is injected into the inner labyrinth seal. The washing gas is properly treated upstream of the injection point and is provided at a slightly higher pressure than the sealing pressure.

This ensures a positive flow of clean and dry scrubbing gas inside the labyrinth and in the compressor side, which reduces the potential for compressor side, which reduces the potential for contamination of gas sealing, increasing the reliability of gas sealing. Double gas opposed gas seals are used in the process gas applications where it is necessary to eliminate all process gas emissions.

3. Efficiency and power consumed by the horizontal centrifugal compressor [4], [5], [7]

The required power to the actual compressor shaft is:

$$P_e = \frac{P_i}{\eta_m} \tag{1}$$

where η_m is the mechanical efficiency of the centrifugal compressors (the loss of shaft friction in the bearing).

$$\eta_m = (0{,}97...0{,}99).$$

P_i – is internal power, consumed by a *n*-stage compressor. This is calculated with the formula:

$$P_i = n\, l_i\, q_m = \frac{nl\, q_m}{\eta_i} \tag{2}$$

l_i – is the mechanical work or the internal mechanical work consumed in a real stage of a centrifugal compressor for the Δp pressure increase. This mechanical work considers all the joints inside the compressor, except for mechanical losses through bearing in the bearings.

The mechanical mass consumed (internal) is calculated with the formula:

$$l_i = l_0 \frac{q_m + q_p}{q_m} + l_f \tag{3}$$

q_p - is the mass flow rate of loss through leakage;
l_f – is the massive mechanical work consumed to overcome the friction between the gas and the rotor;
l_o – is the mechanical work required to achieve the pressure increase Δp, which takes into account the deviation of the gas flow at the inlet to the rotor, the friction from the interior of the gas and the walls of the channels through which it flows;
q_m – mass flow;
η_i –the internal yield of the centrifugal compressor stage (all losses considered) is defined by the ration:

$$\eta_i = \frac{l}{l_i} \tag{4}$$

l – is the theoretical mass mechanic (not all frictions and losses are taken into consideration).

This theoretical mass mechanic is determined by the actual process of gas in the compressor:

a. isothermal, when there is no gas leakage through sealing and no friction between the rotor and the gas;
b. adiabatic, when heat does not change with the environment;
c. polytropic, when there is no gas leakage through sealing and no friction between the rotor and the gas. [4]

Taking into account these aspects, the appropriate internal yields can be defined:

- adiabatic internal yield:

$$\eta_{i,ad} = \frac{l_{ad}}{l_i} \tag{5}$$

- internal isothermal yield:

$$\eta_{i,iz} = \frac{l_{iz}}{l_i} \tag{6}$$

- polytropic internal yield:

$$\eta_{i,p} = \frac{l_p}{l_i} \tag{7}$$

where l_{ad}, l_{iz} şi l_p represents the mechanical work required for adiabatic compression, isothermal, polytropic respectively.

Gasodynamic yield is a characteristic of the quality of gas dynamics processes of the compressor stage (similarly, hydraulic efficiency of the hydrodynamic generators) and is defined by the ratio:

$$\eta_{gd} = \frac{l}{l_0} \tag{8}$$

4. The study of dry sealing at the catalytic reformer at the horizontal centrifugal compressor from the catalytic reformer installation of a petroleum refinery

This paper has studied the possibility of replacing the existing oil seal of the axial centrifugal compressor with a dry seal.

The compressor for this study is driven by a 35 bar backpressure turbine with the output power of 1980 kN and speed of 6670 rpm.

The compressor operating parameters are shown in the table 1. The schematic of the compressor integration in the catalytic reformer installation is shown in the figure 6.

Table 1 **Operating parameters for axial centrifugal compressor**

Operating parameters	Value
Discharge flow in normal conditions	550000 – 112760 Nm^3/h
The suction temperature	38 ^{0}C
Discharge temperature	69,5 – 74 ^{0}C
Suction pressure	9,50 bar
Discharge pressure	12,00 bar
Differential pressure	2,50 – 3,50 bar
Compression ratio	1,34 – 1,40
Gravimetric flow max	33650 kg/h
The power absorbed by the compressor shaft	1500 – 1800 kW
Speed of the rotor shaft	4700 – 5800 rot/min
The average steam consumption of the turbine	16 – 18000 kg/h
Min/Normal/Max steam temperature	290 / 310 / 350 ^{0}C
Inlet steam pressure – min/normal/max steam	30 / 35 / 42 bar

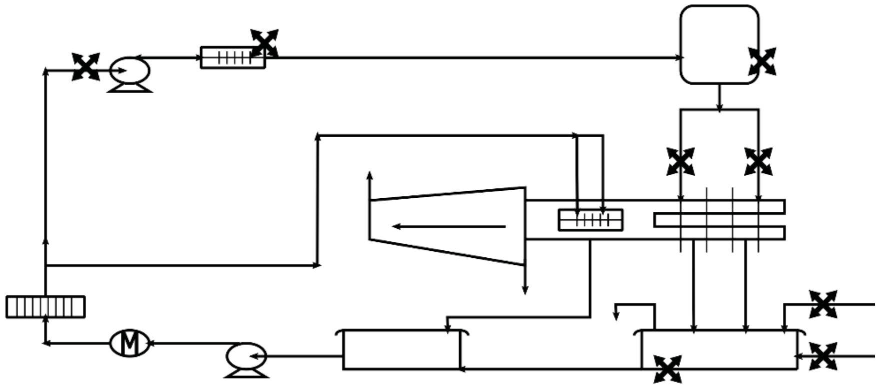

Fig. 6. **The horizontal centrifugal compressor in the RC installation [3]**

The gases conveyed by the axial centrifugal compressor have the composition shown in table 2.

Tabel 2 **The volumetric composition of recirculated gases**

Component	Composition, procent vol,
Hydrogen	80,24%
Methane	5,80%
Ethane	5,83%
Propane	4,43%
Butane	2,42%
Pentane	1,25%
Hexane	0,03%

Let's consider the case of a compressor fitted with an oil seal, where a leakage loss of a mass flow q_p will occur, losses which will lead to an increase of the power consumed by the compressor P_{em}, so it will increase the energy consumption (from the motor machine) (figure 7).

Figure 7 shows the variation curves of the functional parameters of the axial horizontal centrifugal compressor.

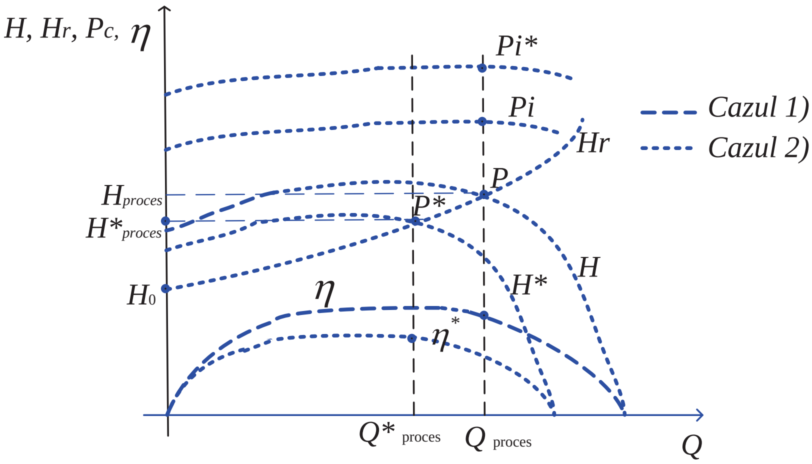

Fig. 7 **The curves of the functional parameters of the axial horizontal centrifugal compressor**

Case 1) is the case of the axial horizontal centrifugal compressor with the internal characteristic ă $H=H(Q)$ and the operating point P. the functional parameters made in the installation are ($Q_{process}$, $H_{process}$). The compressor yield is η , and the power consumed according to the formula (2), is P_i

Case 2) is the case where the gas leakage occurs in the compressor. It observes, in addition to a decrease in flow and load and a decrease in efficiency and an increase in power consume by the compressor.

5. Conclusions

The study has emerged from the need to increase the working performance of the axial centrifugal compressor from the catalytic reformer installation in an oil refinery. This paper proposes replacing the existing sealing of the centrifugal compressor with a dry mechanical seal; this is a technical solution that allows the elimination of deficiencies in operation and the obtaining of technical and environmental benefits.

References

1. Stahley J.S.
Dry gas seal Handbook, PennWell, 2005.

2. ***
API617- Axial and Centrifugal Compressors and *Expandder*-compressors, Eight Edition, September 2014

3. Suciu Gh., Bohiltea I., Platon Al.
Ingineria prelucrării hidrocarburilor, vol. III, Editura Tehnică. 1986

4. Berzănescu A.
Calculul și construcția utilajului petrolier, Editura de Stat Didactică şi Pedagogică, 1961

5. ***
http://turbolab.tamu.edu/proc/turboproc/T11/T11113-120.pdf

6. ***
www.slideshare.net/kstnhdk54/dry-gas-seal-systems

7. ***
www.slideshare.net/draghunathan/compressors-training-sessions