

# Study on nitrogen barrier protection of an air end oil-free compressor bearings in H<sub>2</sub> compression

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**Abstract.** The oil free compressors were specially designed for air compression. The National Research and Development Institute for Gas Turbines COMOTI gained a great deal of experience in producing/designing certified oil injection screw compressors for the natural gas field and for several years it has been focusing its research on the use of "dry" (oil-free) compressors in natural gas compression and more recently in hydrogen compression. Working with an explosive gas, one of an idea was to use a nitrogen barrier in oil bearing sealing, which are open source of gases in the atmosphere for such compressors. Worldwide, on-site nitrogen generators have been developed for a purity range of 95...99.5%, and that nitrogen can be supplied in any environment conditions. The present paper will address nitrogen flow with low percentage of oxygen for bearing sealing at the working pressure, the nitrogen consumption, ideas for H<sub>2</sub> re-injection and the influence over the global efficiency of the process. Due to the Energy Strategy worldwide, and the studies regarding production, transport and storage of hydrogen in natural gas network, COMOTI has involved researches in developing such possibilities and to express a point of view in existing research in the newly created industry.

## 1 Introduction

Based on its experience in the development of screw-type rotary machines with oil injected screw compressor units, COMOTI has begun researching a new type of compressor package with oil-free screw compressor units, as part of a National Research Project. The proposed solutions to the research objectives for these types of compression assemblies with oil-free compressor units will allow COMOTI to demonstrate the functional advantages for the requirements of the beneficiaries, for the parameters of gas type diversity such as pressure, chemical aggression, composition, etc. A compression oil-free unit type CD14D [1] was purchased from the company GHH Ingersol-RAND for the realization of the compression assembly designed by National Research and Development Institute for Gas Turbines COMOTI. The fundamental research of new gas compression solutions has led to innovative solutions in the development of oil-free helical compressors. Within the TURBO 2020 Core

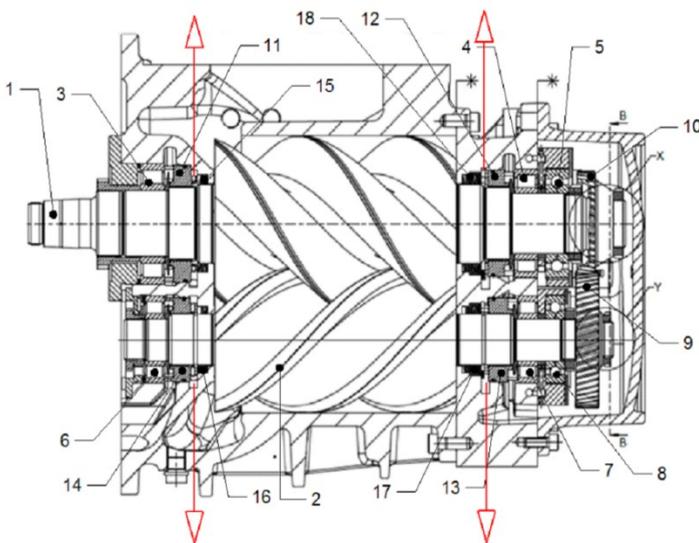
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Program, a fundamental research is conducted, aligning to the general objectives of the institute regarding the development of oil-free rotary machines, which is based on the knowledge attained on oil-injected screw compressors. The operating principle of oil-free compressors is identical to that the oil-injected compressors with the specification that the compression is not isothermal but adiabatically, with a high increasing temperature in the compression process. This is the reason why high-compression ratios between the inlet pressure and output pressure are not possible. The advantage of the oil-free compressors is that the contamination of the fluid is reduced to a minimum. In the next chapter, how to operate the compressor will not be detailed and the focus will be set to its constructive features.

## 2 Constructive features

The oil-free compressor is a volumetric compressor. Rotor 1 and 2 have the outer surface helically profiled after conjugated profiles. The number of lobes of the rotors differs, usually the pair of teeth is  $z1/z2 = 5/6$  or  $4/6$  and it is established by the designer according to the functional parameters required for the compressor. According to figure 1, the rotors are widened with bearings: the driving rotor 1 with the radial bearings 3, 4 and with the bearing in 4 axial points, 5 and the driven rotor, 2 is enlarged with the radial bearings 6, 7 and the axial bearing 8. The lubrication circuit with bearings oil is separated from the gas circulation.



**Fig. 1.** Oil-free compressor structure

Sealing boxes are used to seal the active part of the bearing rotors. Thus, the driving rotor is sealed by the boxes 15 and 18 and the rotor driven by the boxes 16 and 17. A characteristic of this compressor is the fact that between the discs with gaskets, “O”-ring, labyrinths and sealing boxes, there are channels that correspond to the outside, according to the arrows represented on the drawing, which can be connected to the suction of the compressor or to the atmosphere (basket pipe). This is essential for the operation of the gas compressor with explosive potential. In this hypothesis, if gas leaks occur from the active area of the rotors through the sealing boxes, the gases are collected in the afore mentioned channels and can be discharged to the chimney or to the inlet of the compressor where the pressure is lower. In order to protect the compressor rotors which are covered on the surface by anti-corrosion and

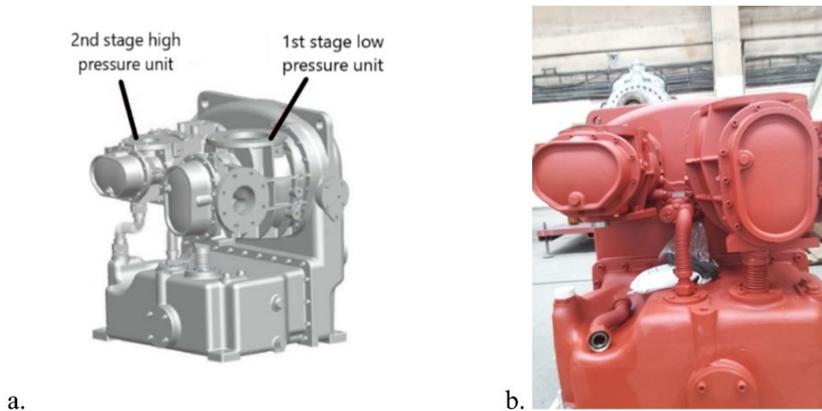
anti-thermal treatments, the pair of gears 9 and 10 synchronizes the movement of the rotors. The clearance between the sidewalls of the wheel teeth is smaller than the clearance between the sidewalls of the rotors.

The actual compression process differs from the theoretical one in that there are internal gas leaks through the gaps between the compressor working members, pressure losses due to the gas discharge through the discharge port, and losses due to friction.

The compression process must be related to the thermodynamic process with variable mass. If this cavity is a compression cavity, the change of mass is done on account of the existence of leaks through the gaps between them and the neighbouring cavities. If the cavity is discharged, the change of mass occurs both on account of gas leakage through the gaps and on the seed of pushing the gas through the discharging port.

### 3 COMOTI Oil free compressor research

One of research branches of the INCDT COMOTI on the development of oil-free rotary machines is based on the high experience in oil-injected screw compressor packages for natural gas compression. For the purpose of developing a new research branch, an oil-free compression unit type CD14D1 [4, 5] was purchased from GHH RAND, Germany (Figure 2).



**Fig. 2.** a. Oil-free compression unit type CD14D1; b. The purchased unit from GHH-Rand

The following table shows the different compressor stages and their application in the respective compressor blocks [4], and the used ones are bolded:

**Table 1.** Assignment of compressor stages to compressor blocks

Type		Single-stage blocks	Double-stage blocks	Booster
1st stage (low pressure)	CDA80	-	CD8D	-
	CDA82	CD9S	CD14D1	-
	<b>CDA96</b>	CD14S	<b>CD14D1</b>	-
	CDA128	CD23S CD26S	CD26D	-
	CDA160	CD42S	CD42D1	-
	CDA208	CD72S1	-	-
2nd stage (high pressure)	CDB54	-	CD8D	CD8HD
	<b>CDB68</b>	-	<b>CD14D1</b>	CD14HD1
	CDB90	-	CD26D	CD26HD
	CDB114	-	CD42D1	-

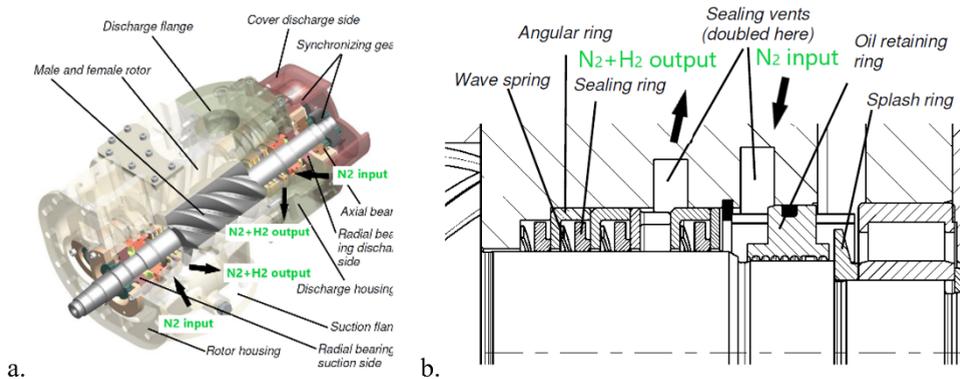
The cells with bolded frame in the left in Table 1 are the units composing the double stage CD14D1 compressor. In Table 2 [4] are the nominal parameters of the compressor stages. In the paper, the values from the columns with normal frame were used.

**Table 2.** Nominal limitations of using CD-D (according to clearance classes).

Parameter	1st stage		2nd stage	
	close	open	close	open
Maximum suction temperature [°C]	46	50	51	60
Maximum discharge temperature [°C]	227	250	227	270
Maximum discharge pressure [bar absolute]	3.96	4.17	8.63	12.7

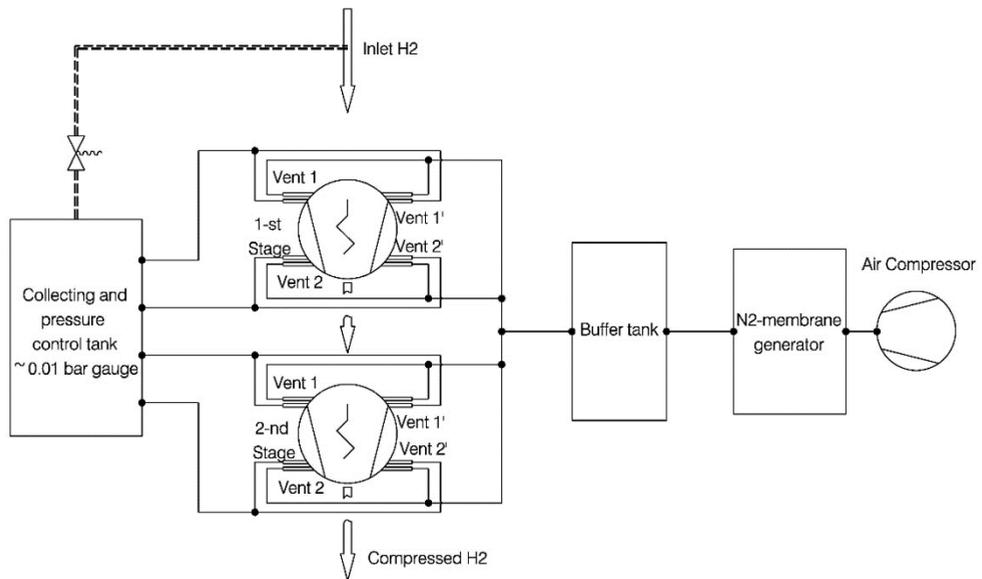
Sealing is an essential factor in the construction of oil free gas compressors. A nitrogen seal is required to work in potentially explosive environments. The contact between the compressed gas and bearings' oil is not allowed. According to [5], it is recommended for compressors working with neutral gas. The functional parameters of the oil-free compressor also play an important role in choosing the constructive variants of the compression unit components.

The compressor stages in the CD series are equipped with an efficient sealing system. The oil-lubricated bearing area is sealed toward the compressor chamber while the majority of the incoming oil quantity is initially rejected and discharged by a co-rotational splash ring. The oil-retaining ring, which consists of a combination of labyrinth seal ring and re-feeding thread, is attached. The compression chamber is sealed toward the bearing area by a sequence of floating metal seal rings that are placed with minimal clearance on the coated rotor shaft. The oil-side and compression-side sealing areas are separated by an intermediate space that is ventilated outward via a drill hole, thus approximating atmospheric pressure. This intermediate space and associated drill hole are referred to as a sealing vent, see figure 3. Apart from the suction-side sealing vents of the low-pressure compressor stages, a compression medium leakage flow unavoidably escapes through all sealing vents. This leakage flow can be up to 2% of the suction volumetric flow rate depending on the operating conditions and compressor type. However, it is already calculated out of the guaranteed delivery quantity of the compressor module. Some types of CDB compressor stages have a double sealing vents arrangement. In other words, there are 2 sealing vents arranged in succession in the sealing area that are separated by an additional sealing ring. For certain applications, this double arrangement can be used to create a sealing medium.



**Fig. 3.** Sealing system of a CDB68 unit. a. Section through CDB68; b. CDB68 male rotor discharge side drawing view.

For applications in which compression medium loss should be avoided as far as possible, it is possible for the leakage loss from the sealing vents to return to the compressor module suction area, see the figure 4 below.



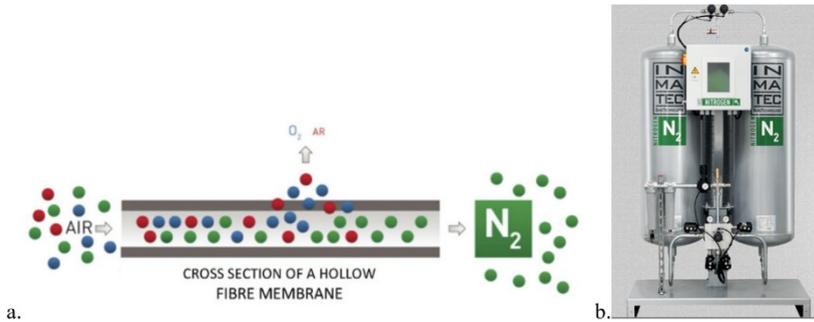
**Fig. 4.** Connection diagram adapted for H<sub>2</sub> compression

As a rule, these applications require keeping leakages of the medium to an absolute minimum. The medium must also be protected against ambient air contamination. The compressor modules from the acquired compressor (see the figure 2 above) are approved for the compression of nitrogen. Given that a minimal difference in the pressure between the vents and the oil system cannot be averted (and is actually intentional for the purpose of sealing), it is impossible to prevent a (minimal) amount of nitrogen from escaping into the oil tank and from there into the atmosphere. Moreover, the CD compressor stages are not tested for 100% leak-free performance when used with gas. Consequently, minimal direct leaks from the compression room into the atmosphere cannot be ruled out completely either. Figure 4 from above depicts how the vents should be connected to make the grade of the special requirements of hydrogen operation. To achieve this, the vents of the low and high-pressure stage are guided to a collecting control tank. The container is set to a pressure that is just slightly higher than the ambient pressure. Leaked gas H<sub>2</sub> is guided to the suction side vents of the low-pressure stage from this container. The slight excess pressure warrants that no oil is sucked-in by this vent on the one hand, and that no air can penetrate the process through the suction side vents, on the other hand. The remaining leaked gas H<sub>2</sub> is guided to the suction side of the low-pressure stage. The necessity of separation between H<sub>2</sub> and N<sub>2</sub> and other impurities, before H<sub>2</sub> re-injection in the suction pipe, will be not addressed in this paper.

#### 4 N<sub>2</sub> production and O<sub>2</sub> % calculation for ATEX proof

The proven INMATEC membrane technology, see the figure 5.a below, allows the continual supply of nitrogen with the purity between 95% to 99.999%. The capacity of these generators is between 0.40 Nm<sup>3</sup>/h and 10,000 Nm<sup>3</sup>/h. In figure 5.b it is given an example of such

installation [6]. In the present paper, an installation with N2 purity generation of 99.5% has been considered.



**Fig. 5.** The INMATEC membrane technology; a. Membrane separation; b. N2 generator

In the membrane process the water vapour, oxygen, nitrogen, noble gases and carbon dioxide diffuse through the hollow fibre membrane at different speeds depending on their molecular structure. Oxygen and nitrogen are separated in the membrane based on their different diffusion grade. The pressure and quality determine the purity of the product [6].

From the data given by an INGERSOL-RAND software for oil-free applications with CD14D compressor unit (see the table 1 from above), there is calculated for nitrogen application at the maximum pressure, the maximum flow in  $m^3/h$  at the given inlet conditions. In figure 6 below, the printed data is shown.

PROGR.: T1996	GHH RAND Schraubenkompressoren GmbH	PAGE : 1/	Oil viscosity group	cSt	46
STATUS: 20090720	Design Program for oil-free	DATE : 15.07.2021	Oil volume flow to compressor *)	L/min	40.6
VERS. : 2.2.22	Screw Compressors	S/N : 8300/2/0	Oil inlet temperature	°C	50.0
			Temperature oil out (mean value) *)	°C	64.1
					10.7
					122.0
					147.4
					°F
CODEWORD : Double stage	PREPARED: Iulian Vladuca		*) jacket cooling included		
JOB-/PROJECT-NO. : Nitrogen compression	CHECKED :		Medium jacket cooling	Oil	
Maximum flow,			STAGE DATA	CD066	CD068W
AMBIENT DATA			Male rotor speed	1/min	12396
Ambient air pressure abs.	bar	1.000	14.50	psia	19572
Ambient temperature	°C	20.0	68.0	°F	
Rel. humidity	%	60.0			
SET DATA			Male rotor tip speed	m/s	94.2
Compressor type / Gear set / i		CD14D / 20 / 4.167	Suction pressure abs.	bar	1.000
Motor speed	1/min	2975	Suction temperature	°C	15.0
Minimum speed for proper lubrication	1/min	2456	Rel. humidity	%	0.0
Medium to be compressed		Nitrogen	Discharge pressure abs.	bar	3.657
			Discharge temperature	°C	174.6
Suction pressure abs.	bar	1.000	14.50	psia	
Suction temperature	°C	15.0	68.0	°F	
Rel. humidity	%	0.0			
Discharge pressure abs.	bar	11.400	165.34	psia	
Discharge temperature	°C	222.4	432.3	°F	
Rel. humidity	%	0.0			
Intercooling to	°C	50.0	122.0	°F	
Volume flow (related to suct. cond.)	m <sup>3</sup> /h	1543.2	908.3	cfm	
Mass flow (wet)	kg/h	1804.6	66.31	lbs/min	
Mass flow (dry)	kg/h	1804.6	66.31	lbs/min	
Volume flow (rel. to std.temp. & pr.)	m <sup>3</sup> /h	1443.7	849.8	cfm	
Volume flow (related to 30" Hg, 60 F)	m <sup>3</sup> /h	1522.0	895.8	scfm	
Power consumption	kW	197.4	264.7	HP	
Thermal output oilcooler *)	kW	17.4	16.5	BTU/s	
Condensating water in intercooler	kg/h	0.0	0.00	lbs/min	
Thermal output intercooler	kW	65.6	62.2	BTU/s	
Condensating water in aftercooler	kg/h	0.0	0.00	lbs/min	
Thermal output aftercooler	kW	0.0	0.0	BTU/s	
Pressure loss before block	bar	0.000	0.00	psi	
Pressure loss between LP- and HP-stage	bar	0.200	2.90	psi	
Pressure loss after block	bar	0.200	2.90	psi	
OILPUMP		R25/20FL-M6-S0			
Volume flow oilpump	L/min	55.5	14.7	gpm	
Shaft power oilpump	kW	0.83	1.11	HP	

**Fig. 6.** The calculation data for N2 fluid at 0% humidity.

From figure 6, at an inlet pressure of 1 bar (abs) for the first stage and discharge pressure of 11.4 bar (abs), the N2 flow is 1543.2  $m^3/h$  at 20°C. It is assumed that the flow of 1543.2  $m^3/h$  at 20°C is a maximum flow for the conditions of 12.5 m/s fluid flow in the inlet pipe, limited at 15.0 ÷ 20.0 m/s [1]. The mass flow is 1804.6 kg/h ~ 0.5 kg/s.

In the H<sub>2</sub> compression, it is assumed equally limited flow, but the mass flow will be decreased significantly because the molar mass of H<sub>2</sub> is  $H_2 = 2$ . The N<sub>2</sub>/H<sub>2</sub> molar mass ratio is 14, and one can estimate the mass flow of the H<sub>2</sub> to be:

$$\dot{M}_{H_2} = \frac{\dot{M}_{N_2}}{14} = 0.0357 \text{ kg/s} \quad (1)$$

The leakage flow of the H<sub>2</sub> escaping from the vents is about 2% of the total flow in the compressor. The Leakage flow will be:

$$\dot{M}_{leakage H_2} = \dot{M}_{H_2} \cdot 0.02 = 0.000714 \text{ kg/s} = 2.57 \text{ kg/h} \quad (2)$$

From the ATEX point of view, the lower flammable limit for O<sub>2</sub> concentration in H<sub>2</sub> and pure O<sub>2</sub> mixture is about 4.65% in pure oxygen [7], and in air about 4.0% [7, 8]. In the literature, flammability limits and burning velocities of gaseous mixtures containing hydrogen have been measured to reach temperatures up to 250°C and pressures up to 4 MPa [9]. The methodology of EN 60079 (SR EN IEC 60079-0:2018) [10] provides for an estimate of the hypothetical volume over which the mean concentration of the flammable gas will be 0.25 times the lower explosive limit. Because the oxygen is mixed with nitrogen, one can consider an air mixture with the concentration of 0.05% oxygen, in conformity with nitrogen delivered by an INMATEC membrane technology [6].

The nitrogen flow for sealing from the figure 3, should be at the same hydrogen leakage flow, that is  $\dot{M}_{leakage H_2} = 0.000714 \text{ kg/s}$ . To calculate the volume of the oxygen in the N<sub>2</sub>/O<sub>2</sub> mixture, we should calculate the density of the mixture. We consider the volume of a kmol = 22.4 m<sup>3</sup>. The density of the mixture is:

$$\rho_{mixture} = M_{mixture} \text{ (kg/kmol)} / 22.4 \text{ m}^3 / \text{kmol} \quad (3)$$

assuming that the  $M_{mixture}$  is the mass of a kmol.

$$M_{mixture} = 0.995 \cdot 28 \text{ (molar mass of the } N_2) + 0.005 \cdot 32 \text{ (molar mass of } O_2) = 28.02 \text{ kg/kmol} \quad (4)$$

$$\rho_{mixture} = 1.2509 \text{ kg/m}^3 \quad (5)$$

The mass flow of the oxygen in the N<sub>2</sub>/O<sub>2</sub> mixture flow of 0.000714 kg/s is:

$$\dot{M}_{oxygen} = 0.005 \cdot 0.000714 = 3.57 \cdot 10^{-6} \text{ kg/s} \quad (6)$$

The volume flow of the oxygen in the N<sub>2</sub>/O<sub>2</sub> mixture is:

$$\dot{V}_{oxygen} = \frac{\dot{M}_{oxygen}}{\rho_{mixture}} = \frac{3.57 \cdot 10^{-6}}{1.2509} = 2.854 \cdot 10^{-6} \text{ m}^3/\text{s} \quad (7)$$

The density of the H<sub>2</sub> in normal conditions (20°C and atmospheric conditions) is  $\rho_{H_2} = 0.08375 \text{ kg/m}^3$ .

The volume flow of the hydrogen will be:

$$\dot{V}_{hydrogen} = \frac{\dot{M}_{leakage H_2}}{\rho_{H_2}} = 0.0084059 \text{ m}^3/\text{s} \quad (8)$$

The O<sub>2</sub>/H<sub>2</sub> volume ratio is:

$$R_{O_2/H_2} = \frac{V_{oxygen}}{V_{hydrogen}} = \frac{2.854 \cdot 10^{-6}}{8.406 \cdot 10^{-3}} = 0.00034 \quad (9)$$

The result is 0.034% concentration of O<sub>2</sub> in the H<sub>2</sub> flow, which is a negligible quantity of oxygen in the H<sub>2</sub>/O<sub>2</sub> mixture, to create an explosion area, much lower than 4% low limit for air or pure oxygen atmosphere.

The volume of N<sub>2</sub> consumed in the process will be calculated same as O<sub>2</sub> volume, and from equation 6, we will use the 0.995 concentration of N<sub>2</sub>. We will have:

$$\dot{M}_{nitrogen} = 0.995 \cdot 0.000714 = 7.1043 \cdot 10^{-4} \text{ kg/s} \quad (10)$$

The volume flow of the nitrogen in the N<sub>2</sub>/O<sub>2</sub> mixture is:

$$\dot{V}_{nitrogen} = \frac{\dot{M}_{nitrogen}}{\rho_{mixture}} = 5.679 \cdot 10^{-4} \text{ m}^3/\text{s} \sim 2.04 \text{ m}^3/\text{hour} \quad (11)$$

The volume is calculated at 20°C and 1.0 bar pressure. The normal volume will be:

$$\dot{V}_{nitrogen} \sim 1.9 \text{ Nm}^3/\text{hour} \quad (12)$$

The volume is calculated at 0°C and 1.0 bar pressure [11]. The generator IMT-PN 1150 can be chosen from the catalogue [12], with a capacity of 2.6 Nm<sup>3</sup>/h. The power consumption of the installation is just 150 W. The specific mechanical work for compression in two stages with intercooling is [13, 14]:

$$W_{compression,specific,H2} = \frac{R}{H_2} \cdot \left\{ \frac{n_{ST1}}{n_{ST1}-1} T_{inlet\_ST1} \cdot \left[ (\pi_{c\_ST1})^{\frac{n_{ST1}-1}{n_{ST1}}} - 1 \right] + \frac{n_{ST2}}{n_{ST2}-1} \cdot T_{inlet\_ST2} \cdot \left[ (\pi_{c\_ST2})^{\frac{n_{ST2}-1}{n_{ST2}}} - 1 \right] \right\} \quad [J/kg] \quad (13)$$

where  $\pi_{cST1}$  is the compression ratio in the first stage and this calculated according to the values from figure 6:

$$\pi_{cST1} = \frac{p_{ev,comp,ST1}(abs)}{p_{inlet,comp,ST1}(abs)} = \frac{3.457}{1} = 3.457 \quad (14)$$

and  $\pi_{cST2}$  is the compression ratio in the second stage and this calculated according to the values from figure 6:

$$\pi_{cST2} = \frac{p_{ev,comp,ST2}(abs)}{p_{inlet,comp,ST2}(abs)} = \frac{11.6}{3.457} = 3.355 \quad (15)$$

The polytropic coefficient for each stage was calculated using the formulas [15]:

$$n_{ST1} = \frac{1}{1 - \frac{\ln\left(\frac{T_{output\_ST1}}{T_{inlet\_ST1}}\right)}{\ln(\pi_{cST1})}} = 1.662 \quad (16)$$

for the first stage, and:

$$n_{ST2} = \frac{1}{1 - \frac{\ln\left(\frac{T_{output\_ST2}}{T_{inlet\_ST2}}\right)}{\ln(\pi_{cST2})}} = 1.546 \quad (17)$$

for the second stage. For  $T_{asp, ST1} = 0^\circ\text{C} = 293.16 \text{ K}$ ,  $T_{ev, ST1} = 174.6^\circ\text{C} = 447.76 \text{ K}$ ,  $T_{asp, ST2} = 50^\circ\text{C} = 323.16 \text{ K}$ ,  $T_{ev, ST2} = 222.4^\circ\text{C} = 495.56 \text{ K}$ ,  $H_2=2$ ,  $R = 8314.3 \text{ J}/(\text{KmolK})$ , for ideal gases, results from the equation (13) the specific mechanical work for H<sub>2</sub> compression:

$$W_{compression,specific,H2} = 3.851 \cdot 10^6 \text{ J/kg} \quad (18)$$

The power consumed is given by the equation:

$$P_{compression,H2} = \dot{M}_{H2} \cdot \frac{W_{compression,specific,H2}}{\eta_{compression}} / 1000 \text{ [kW]} \quad (19)$$

assuming that compression efficiency  $\eta_{compression} = 75\%$ , and the total compression power will be:

$$P_{compression,H2} = 183.3 \text{ kW} \quad (20)$$

From figure 6, 42% in kW is the cooling power need for cooling the oil and gas intercooling. Also, 0.4% from the shaft power is used for oil injection. The total power consumption is:

$$P_{total} = P_{compression,H2} + P_{cooling} + P_{oil injection} = 261.1 \text{ kW} \quad (21)$$

From the total power used for H<sub>2</sub> compression, the power consumed for N<sub>2</sub> production, of 150 W = 0.15 kW, about 0.057% is negligible, and the total efficiency of the compressing process it is not influenced.

## 5 CONCLUSIONS

The sealing of the shafts of the compressors with neutral gases is a relatively new idea, research studies have been conducted successfully beginning with 1990s. Using neutral gases as a barrier sealing is also improving the protection against harmful or potentially explosive emissions, or the formation of inflammable or explosive gas/gas or gas/liquid mixtures. Many research studies and industrial achievements have been performed and obtained in in the field of turbomachines.

One of the ideas from the actual paper was to create the sealing neutral gas, in our case nitrogen, directly from the environmental air, through a membrane technology, see the figure 5, such technology being a new but mature one. While using traditional gas supply methods such as liquid or bottled hydrogen, users pay higher costs because of the rental fees, refill, delivery order processing charges, as well as environmental fees. Generating nitrogen in-house is a cost-effective and reliable alternative to the other methods of nitrogen supply [16].

The present work does not take into consideration the warm gases in the H<sub>2</sub> compression that can be observed in figure 6 to be about 222.4°C in stage 1 and 174.6°C in the 2nd stage of compression. The limit of the hydrogen autoignition at atmospheric conditions is between 500÷577°C. Studies have been performed for different pressures and temperatures and the increase of the temperature can slowly modify the lower limit of H<sub>2</sub>/O<sub>2</sub> concentration. The given value R<sub>O<sub>2</sub>/H<sub>2</sub></sub>=0.034% from the equation 9, for the concentration of O<sub>2</sub> in the H<sub>2</sub> flow showed that the concentration of O<sub>2</sub> in the N<sub>2</sub>/O<sub>2</sub> mixture for barrier sealing is negligible and the method proposed can be used in safe conditions conditions. The power consumption for N<sub>2</sub> generation compared with the power need for H<sub>2</sub> compression also give a negligible value, with insignificant over the global efficiency.

The presented solution with nitrogen barrier for bearing sealing of oil-free compressors for their use in potentially explosive environments is reliable, and can be implemented in the existing transport gas grid, thanks to the ability of the gas grid to integrate admixtures of hydrogen. With a view to 2050, Hydrogen Europe considers that through a hydrogen and gas integration, a significant cost-effective decarbonisation can be achieved. Furthermore, the gas infrastructure can transport larger volumes over longer distances at a fraction of the costs when compared to the electricity grid. With the switch to renewable and low-carbon

hydrogen, among other sustainable gases, gas grids can be adapted and used for a cost-effective decarbonisation across Europe and worldwide. In addition, the conversion of gas grids to hydrogen is the fastest way to fully decarbonise as it can be achieved via central and decentralised actions [17].

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## References

1. <https://oem-solutions.ingersollrand.com/en-us/products/integrated-compression/oil-free-double-stage-airends-cdd>, [Accessed 22.06.2021]
2. ISO 10440-1:2007, Petroleum, petrochemical and natural gas industries -- Rotary-type positive-displacement compressors -- Part 1: Process compressors
3. I. Vladuca, R. Stanciuc, R. Bimbasa, S.G. Tomescu, *Waste-Energy (Heat) Recovery System from the Gases Compressed by an Oil-Free Screw Compressor*, E3S Web Conf., **112** (2019), <https://doi.org/10.1051/e3sconf/201911201012>
4. GHH Rand Installation manual for CD stages and CD blocks 23152176A, (2009)
5. X.Peng, Z.Xing, *Experimental Study of oil injection and its effect on performance of twin screw compressors*, International Compressor Engineering Conference, (2000)
6. [https://www.inmatec.de/en/nitrogen-technology.html?gclid=EA1aIqobChMI5o6Pus\\_i8QIVDdN3Ch0HXgGVEAAYASAAEgKOWyD\\_BwE](https://www.inmatec.de/en/nitrogen-technology.html?gclid=EA1aIqobChMI5o6Pus_i8QIVDdN3Ch0HXgGVEAAYASAAEgKOWyD_BwE), [Accessed 08.07.2021]
7. H. F. Coward, G. W. Johnes, Limits of Flammability of gases and Vapors, **503**, (1952), Bureau of Mines, Alaska. Available on: <https://shepherd.caltech.edu/EDL/PublicResources/flammability/USBM-503.pdf>, [Accessed 15.07.2021]
8. J. Joongoo and K. Sung-joong, Recent Progress in Hydrogen Flammability Prediction for the Safe Energy Systems, *Energies* 2020, **13**, 6263, (2020), <https://doi.org/10.3390/en13236263>
9. Yu. N. Shebeko, S. G. Tsarichenko, A. Ya. Korolchenko, A. V. Trunev, V. Yu. Navzenya, S. N. Papkov, And A. A. Zaitzev, Burning Velocities and Flammability Limits of Gaseous Mixtures at Elevated Temperatures and Pressures, **102**, 4, (1995), 427-437, [https://doi.org/10.1016/0010-2180\(95\)00002-N](https://doi.org/10.1016/0010-2180(95)00002-N)
10. IEC 60079:2021 SER, Explosive atmospheres
11. IUPAC. Compendium of Chemical Terminology, 2nd ed. (the "Gold Book"). Compiled by A. D. McNaught and A. Wilkinson. Blackwell Scientific Publications, Oxford (1997). Online version (2019) created by S. J. Chalk. ISBN 0-9678550-9-8. <https://doi.org/10.1351/goldbook>
12. [https://www.caps.com.au/wp-content/uploads/product-download/2019\\_IMT\\_Flyer\\_PN\\_OnGo\\_EN.pdf](https://www.caps.com.au/wp-content/uploads/product-download/2019_IMT_Flyer_PN_OnGo_EN.pdf), [Accessed 26.07.2021]
13. P. A. O'Neill, *Industrial Compressors: Theory and equipment*, (Butterworth-Heinemann, Oxford, 1993)
14. N. Băran, M. Marinescu, V. Radcenco, A. Dobrovocescu, A. Chisacof, M. Grigoriu, P. Răducanu, G. Popescu, I. Ganea, T. Duicu, S. Dimitriu, C. Papadopol, V. Bădescu, T. Blusalis, N. Boriaru, V. Apostol., E. Vasilescu, D. Stanciu, D. Isvoranu, R. Dănescu, C. Dinu, M.C ostea, O. Mălăncioiu, C. Mladin, O. Crăciunescu, *Termodinamică tehnică –teorie și aplicații*, volumul I, (Editura Matrix Rom, București, 1998)
15. A. L. Allen, *Efficiency and performance measurements of a pdc inc. Single Stage diaphragm hydrogen compressor*, A Thesis, Humboldt State University, (2009). Available on: <https://scholarworks.calstate.edu/downloads/xd07gw01s> [Accessed 29.07.2021]
16. [https://www.n-psi.com/nitrogen-generators-adwords?gclid=EA1aIqobChMIvdnxwNnk8QIVitd3Ch2XEAnvEAAAYASAAEgIUfD\\_BwE](https://www.n-psi.com/nitrogen-generators-adwords?gclid=EA1aIqobChMIvdnxwNnk8QIVitd3Ch2XEAnvEAAAYASAAEgIUfD_BwE), [Accessed 15.07.2021]
17. [https://ec.europa.eu/info/sites/default/files/hydrogen\\_europe\\_-\\_vision\\_on\\_the\\_role\\_of\\_hydrogen\\_and\\_gas\\_infrastructure.pdf](https://ec.europa.eu/info/sites/default/files/hydrogen_europe_-_vision_on_the_role_of_hydrogen_and_gas_infrastructure.pdf), [Accessed 28.07.2021]