Reducing rotors clearance - a way to increase the performance of a screw compressor

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Abstract. The screw compressors due to their constructive simplicity and the known advantages in operation is increasingly demanded on the market. Many companies are producing this type of compressor. The influence of leakages – conditioned by rotors clearances – on compressor performance is well known. The literature addresses the problem of coatings, but for dry compressors. Our intention is to apply this technology in the case of oil injected screw compressors (when the profile deviation is large or in the case of rotors with slight wear). Tests were performed to evaluate the performance of a screw compressor with uncoated rotors and performance of compressor assembled with coated rotors (both pairs of rotors have the same profile deviations – the profile accuracy measured is \(-8 \div -10 \mu m\)). The test aim to highlight only the influence of the coating layer – by reducing clearances – on the performance of the compressor, without performing further quality test of the coated layer. The quality of the coating process is the subject of additional research. We intend to use this technology for oil injected screw compressors both for new manufactured rotors (with large profile tolerances) and rotors with slight.

1. Introduction

Within this article, NRDI COMOTI aims to present an innovative method – at least in Romania – to reduce the high costs related to the manufacture of component parts of high-pressure discharge compressors with very tight tolerances, using a modern coating technology.

It’s significant to emphasize that in the open literature, the rotor’s coating process is presented as being applicable only for the dry (oil free) screw compressors types. The arguments specified in open literature and the advantages of applying this type of coating process have led us to consider that this coating process can also be applied to rotors used in oil injected screw compressors in the two aforementioned circumstances. Thus, the possibility to practically verify the stated considerations was offered by a concrete situation: COMOTI ordered the manufacture of rotor pairs to a partner company. From the profile diagrams it resulted that the recorded profile’s deviations fit into the range \(-8 \div -11 \mu m\). The pairs of rotors that will be coated were chosen accordingly to figures 1 and 2, respectively the pairs of rotors without coating accordingly to figures 3 and 4.
By coating, the clearances between male and female rotors were reduced with 20 µm and clearances between rotors and housing were reduced with 10 µm.

Thus, figure 5 shows a sample rotor’s pair of a screw compressor developed by COMOTI, for each rotor that is coated with XILAN 1052 CTG ON LAS, the thickness of the deposited layer fits into the range 0.009 ÷ 0.012 mm. This was a first attempt to test the possibility of coating rotors, checking only the uniformity of the thickness of the deposited layer (see figure 6). Subsequent research will determine the quality of the layer during operation period (checking adhesion, elasticity, oil resistance, resistance to operational stresses, aso).
Figures 7 and 8 suggestively present the pair of rotors both the one with coated rotors and the one with uncoated rotors mounted in the compressor assembly before test and in figure 9 the coated rotors pair after nearly 30 testing hours.

So, it seems that any reduction of the clearance leads to the improvement of the performances. In recent time, a modern machine tool can manufacture helical rotors obtaining a tolerance to the rotor profile around +/- 5 μm, which ultimately leads to a value of less than 15 μm of the clearance between the rotors. With such small gap, collision between the rotors is certitude and therefore the profile and distribution of the clearance along the profile must be generated so as to avoid damage if this collision occurs. It should be mentioned that the manufacture of rotors with tolerances so tight is expensive. A solution would be the acceptance of wider tolerances, which will be adjusted by coating procedures, to comply with the initially prescribed value of clearance between the moving parts.

This article does not intend to analyze the principles and methods of designing the rotor profile / profile tolerances / clearance distribution also, but to analyze solutions to improve the technological process of manufacturing / reconditioning the rotors regarding the following aspects:

- the possibility of reconditioning rotors that have worked in heavy duty and shows wear of profile that leads to the increase of the clearance and the alteration of the performances;
the reduction of the costs related to the manufacture of rotors with tight tolerances, by using the coating procedures.

In order to limit clearances, precise manufacturing of rotors and housings is essential, but even in the case of using precise CNC machines, the ideal case – zero tolerance – is practically unattainable. For example, for small compressors, the recommended clearances are up to 50 μm, values that are difficult to obtain, a solution being the rotor pairing to obtain the optimal clearances. But these operations raise the cost of manufacturing. One solution would be treating of the surface, to reduce clearance and minimize friction. Examples of such solutions would be: hard anodizing, chrome plating, plating or PTFE coating – see figure 10.

![Figure 10](image)

Figure 10. Coating of the male / female rotors’ surfaces

In the article [1] it is stated that compressor efficiency is limited by geometry. Air leaks through the gaps between the rotors, between the housing and rotor tips, and at the output ends of the rotors. And the higher is the pressure, the more likely air is to escape back to the low-pressure side of the compressor. All that leakage is lost efficiency and lost power. In fact, pumping efficiency is related to the clearances between surfaces that push air. The answer seems easy: the tighter the clearance the better. But that isn’t easily attained. Because metal surfaces in these screw-type compressors are a three-dimensional spiral slope, machining them to precise dimensions is difficult. Manufacturers usually match each male-female set manually to get the best fit.

In the article [2] it is stated that screw compressor efficiency is improved by reducing internal leakage and this can be affected by minimizing the clearance between the rotors and the casing. The effect on performance of three factors which influence the working clearance was analyzed. These are: the interlobe clearance, adjusted to the rotor temperature change, rotor contact on the lobe flat side and displacement of the discharge bearing centers.

In the article [3] it is stated that designing twin-screw compressors to safely operate at higher-than-normal temperatures poses a challenge as the compressor must accommodate larger peak thermal distortions while maintaining efficiency at nominal operating conditions. This paper presents a case study of an oil-injected compressor tested at elevated discharge temperatures with original and revised clearances. A procedure is presented to use boundary conditions derived from a chamber model to approximate component temperature distributions that are then used to predict possible thermal distortions and the resulting effect on clearance gaps. The original and revised clearance designs are evaluated and performance penalties incurred due to the modifications are discussed.
2. Tests

2.1. General conditions

The configuration/instrumentation layout of the test bench is shown in figure 11. The parameters to be measured for the estimated performance of the compressor will be: compressor suction/discharge temperature and pressure; speed of the male rotor; power of the electric motor driver; pressure/temperature of oil after the oil cooler; differential pressure, pressure and temperature across the differential devices (designed according to ISO 5167-2); in/out temperature for water, in the oil cooler; oil flow rates, for each oil point injection (seal, suction bearings, injection, discharge bearings); pressure of oil at each oil injection point; atmospheric pressure, atmospheric temperature and the relative humidity.

In figure 12 there is a photo of the test bench, with the compressor installed on it.

![Figure 11. Test bench configuration](image1)

![Figure 12. Compressor on test bench](image2)

2.2. Measured parameters. Measuring equipment. Precisions

The equipment, precision and measurement methods are in accordance with the recommendations of ISO 1217, ASME PTC9, relating to the determination of the performance of a compressor during the test, delivered flow or required power.

We emphasize once again that for licensed compressors, the performance test - with air - is done for a point of operation indicated by the licensing company. Depending on the type of compressor, respectively volumetric ratio Vi, the licensor indicates the specific test parameters:

- suction pressure / suction temperature;
- discharge pressure;
- flow;
- speed.
- During the test, you measure:
  - the gas flow rate;
  - the power;
  - discharge temperature (controlled by oil flow).

For compressors developed by COMOTI, the test conditions follow the recommendations in Annex E-ISO 1217- concerning the acceptance test for volumetric compressors driven by variable speed electric motors.
At the same time, ISO 1217 recommends that the measuring equipment that may affect test results be calibrated at the specified range to ensure the accuracy/reliability of the measured information. Measurements of parameters used in performance calculations are made with instruments / devices that meet the following conditions (for the COMOTI stand):

- for pressure – pressure transducer, signal 4...20mA, line precision ±0,2%;
- for temperature – thermocouple, signal 4...20mA, line precision ±0,5%;
- for oil flow - flow meter, signal 4...20mA, line precision ±0,2%
- for gas flow - differential pressure transducer, signal 4...20mA, instrument precision ±0,1%, line precision ±1%.

The flow rate is measured at the suction of the compressor accordingly to ISO 5167-2, the flow meter being mounted on the pipe according to the installation conditions. For calculating the corrections and displaying the flow in Nm³, the acquisition system picks up atmospheric pressure and ambient temperature signals (with transducers mounted in the vicinity of the suction port).

- for power – torque converter mounted on the drive shaft of the compressor, type HBM-T40 precision class 0,05%;
- for speed - speed transducer, signal 4...20mA, line precision ±0,2%

During the tests, the vibrations are measured with VIBER X5—see figure 13 and thermal imprint with Fluke—see figure 14.

![Figure 13. Vibration analysis](image1)

![Figure 14. Thermal image](image2)

When testing a new compressor developed by the COMOTI, the tests in the stand are detailed, aiming at the acquisition of data allowing the development of the operating diagrams (previously mentioned: flow/speed and power/speed chart for a given suction pressure, a given volume ratio Vi and different discharge pressures, in operating range).

The scope of our tests is to compare the performance of the two compressors:

- with coated rotors;
- with uncoated rotors.

We keep the other parts – housings – unchanged to determine only the influence of the coating on the performance (keep constant axial clearances), and made tests in the same conditions (similar suction pressure, discharge pressure, discharge temperature, male rotor speed).
2.3. Recorded parameters. Diagrams

We specify that the equipment, precision and measurement methods are in accordance with the recommendations of ISO 1217, ASME PTC9, relating to the determination of the performance of a compressor during the test, delivered flow or required power. Tests were performed to evaluate the performance of a screw compressor with uncoated rotors and performance of compressor assembled with coated rotors (both pairs of rotors have the same profile deviations – the profile accuracy measured is -8 … -10 µm). The test aim to highlight only the influence of the coating layer – by reducing clearances – on the performance of the compressor. As a procedure, we kept the discharge pressure constant and gradually decreased the speed (keeping it constant until the discharge temperature stabilized) – see figures 15, 16.

![Operating test conditions for coated rotors at 16 bar abs.](image1)

Figure 15. Operating test conditions for coated rotors at 16 bar abs.

![Operating test conditions for uncoated rotors at 16 bar abs.](image2)

Figure 16. Operating test conditions for uncoated rotors at 16 bar abs.

We attach print-screen with a sample of recorded parameters for male rotor speed at 6000 rpm – see figure 17 and table 1 (the notations of the measured parameters are the same for the print-screen and for the table 1).

![Sample of print-screen for recorded data at 11 bar abs. – uncoated rotors](image3)

Figure 17. Sample of print-screen for recorded data at 11 bar abs. – uncoated rotors
Table 1. Sample of recorded parameters for uncoated (UC) and coated (C) rotor’s pair at 11 and 16 bar abs.

| Rotor’s pair UC or C-p | Differential pressure suction diaphragm DPGA (bar) | Momentum MMR (Nm) | Driver speed NCS (RPM) | Ambient pressure P0 (bar abs) | Suction pressure PGA (bar abs) | Discharge pressure PGR1 (bar) |Lube oil pressure PUIC (bar) | Suction temperature TGA (°C) | Discharge temperature TGR (°C) | Ambient temperature TO (°C) | Oil injection temperature TUIC(°C) | Mass flow rate QM (kg/h) | Volumetric flow rate QV (Nm3/h) | Power consumption P_MMR (kW) |
|------------------------|---------------------------------------------------|------------------|-----------------------|-------------------------------|-----------------------------|-----------------------------|------------------------|---------------------------|---------------------------|--------------------------|-----------------------------|-----------------------------|--------------------------|--------------------------|
| UC-11                  | 43,69                                             | 152,88           | 2843,28               | 1,01                          | 9,49                        | 6,73                        | 16,4                   | 70,95                     | 19,2                      | 49,95                    | 435,57                      | 361,77                     | 47,12                    |
| UC-16                  | 41,53                                             | 194,13           | 2944,69               | 1,01                          | 9,49                        | 14,21                       | 12,21                  | 16,7                      | 71,6                      | 18,3                     | 49,85                       | 426,5                       | 59,86                    |
| C-11                   | 45,47                                             | 158,5            | 2954,53               | 1,01                          | 9,49                        | 7,48                        | 13,75                  | 68,4                      | 16,5                      | 50,25                    | 448,64                      | 372,62                     | 49,04                    |
| C-16                   | 44,34                                             | 192,25           | 2981,25               | 1,01                          | 9,49                        | 14,08                       | 11,01                  | 9,5                       | 70,85                     | 11,05                    | 49,95                       | 446,12                      | 60,02                    |

So, applying the recommendations about computation of tests result specific diagrams were plotted. For an easy understanding, we synthesized the test results, presenting, in a diagram, the measured and corrected values at a certain speed/discharge pressure (Pref) – for coated rotor/uncoated rotor – see figures 18+21.

![Figure 18. Flow rate for Pref =16 bar abs.](image1)

![Figure 19. Power for Pref =16 bar abs.](image2)

![Figure 20. Flow rate for Pref =11 bar abs.](image3)

![Figure 21. Power for Pref =11 bar abs.](image4)

Starting from recorded parameters also we achieve following diagrams – figures 22 and 23 – that show – in percentages – the rapport between flow rate (Vcr - volumetric flow rate coated rotors, Vur - volumetric flow rate uncoated rotors) and power (Pcr - power consumption for coated rotors, Pur - power consumption for uncoated rotors) at different speed, for coated and uncoated rotors.
It is also suggestive to show in the tables 2, 3 and 4 the values of the volumetric efficiency, for compressor with coated and uncoated rotor, and the increase of volumetric efficiency -%-.

Practically, concluding, analyzing the power and flow diagrams it can be stated that, by coating the rotors, an increase of the flow is obtained, in the conditions of a practically constant power. So, the specific consumption kWh / 1000Nm3 decreases (with approx.3%).

### Table 2. Volumetric efficiency for uncoated rotors

| Speed (RPM) | Pressure at 6 bar abs | Pressure at 11 bar abs | Pressure at 16 bar abs |
|-------------|-----------------------|------------------------|------------------------|
| 2000        | 0.806314              | 0.776078               | 0.740404               |
| 3000        | 0.80887               | 0.796326               | 0.776858               |
| 4000        | 0.826                 | 0.802018               | 0.781326               |
| 5000        | 0.830231              | 0.792316               | 0.777829               |
| 6000        | 0.818061              | 0.789565               | 0.773123               |

### Table 3. Volumetric efficiency for coated rotors

| Speed (RPM) | Pressure at 6 bar abs | Pressure at 11 bar abs | Pressure at 16 bar abs |
|-------------|-----------------------|------------------------|------------------------|
| 2000        | 0.813112              | 0.806042               | 0.770096               |
| 3000        | 0.831638              | 0.81652                | 0.793172               |
| 4000        | 0.844572              | 0.822248               | 0.800414               |
| 5000        | 0.825293              | 0.817781               | 0.809175               |
| 6000        | 0.817807              | 0.813257               | 0.808689               |

### Table 4. Volumetric efficiency increase from uncoated rotors to coated rotors

| Speed (RPM) | Pressure at 6 bar abs | Pressure at 11 bar abs | Pressure at 16 bar abs |
|-------------|-----------------------|------------------------|------------------------|
| 2000        | 0.843057 %            | 3.860977 %             | 4.010283 %             |
| 3000        | 2.814755 %            | 2.535852 %             | 2.100056 %             |
| 4000        | 2.248338 %            | 2.522376 %             | 2.443014 %             |
| 5000        | -0.59475 %            | 3.217659 %             | 4.029867 %             |
| 6000        | -0.031102 %           | 3.000666 %             | 4.600234 %             |
3. Conclusions
Following the tests results we can draw the following conclusions:

- tests were suited to evaluate the performance of a screw compressor with uncoated rotors and performance of compressor assembled with coated rotors (both pairs of rotors have the same profile deviations – the profile accuracy measured is \(-8 \pm 10\) µm);
- applying this technology are reduced the costs related to the manufacture of rotors with tight tolerances, (by using the coating procedures); also, by coating, the price of the compressor unit rises with nearly 2% (for tested type compressor);
- the test show that, for coated rotors the compressor performance has improved;
- for discharge pressure 11 bara and 16 bara, the average flow increase is approx. 3% (the power remaining practically constant);
- applying this technology, the specific consumption in kWh / 1000Nm3 decreases (if we compare the parameters of two compressors – with/without coating layer), with nearly 3%;
- we specify that quality of the coating process is the subject of additional research. But after approximately 30 hours of tests, a visual analysis of coating looks well, without detachments or other traces (see figure 9).

4. References

[1] Bruce Nesbitt: *Conformable coating narrows compressor air gaps*. https://www.machinedesign.com/archive/conformable-coating-narrows-compressor-air-gaps-boosts-efficiency

[2] Nikola Stosic, Ian K. Smith, Ahmed Kovacevic: *Improving screw compressor performance* https://www.researchgate.net/publication/289206577_Improving_screw_compressor_performance

[3] Buckney D, Kovacevic A, Stosic N 2016 *Design and evaluation of rotor clearances for oil injected screw compressors*, proceedings of the institution of Mechanical Engineers, Part E: Journal of Process Mechanical Engineering 231 26-37

[4] ISO10440 Rotary-type Positive Displacement Compressors for Petroleum, Petrochemical and Natural Gas Industries (API 619)

[5] ISO 1217 – Displacement compressors – Acceptance tests (PTC 9)

[6] ISO 5167-2 – Measurement of fluid flow by means of pressure differential devices in circular cross-section conduits running full – Part 2 Orifice plates