Analysis of a high-pressure screw compressor performances

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Abstract. Oil flooded screw compressors with discharge pressures of up to 26 bar have been available, commercially for some years. However, demands for higher pressure operation led to the design, construction and testing of a compressor operating with discharge pressures of up to 45 bar. To maximize the compressor efficiency, extensive performance simulations were carried out. Since 1980 oil flooded screw compressors have been used in many process gas compression applications and in the last year the higher pressure and larger capacity was realized. From 2010 to 2015 INCDT COMOTI manufactured screw compressors under GHH- Rand license (CU type, with max. delivery pressure 26 bar). Experience gain in this process gives us the power to develop the new type of screw compressors - based on license - with high delivery pressure - max. 45 bar. This paper will present details about the tests with a new high pressure screw compressor test in a closed loop test configuration. The machine was then tested in a closed loop test rig and, as is shown, the measured performance agrees well with that predicted by the numerical model. The parameters will be analysed starting from CFD modelling of the process and finally comparative analysis with actual parameters. The demonstrated advantages of this type of compressor, like high reliability, simple foundations, low operational cost, low initial cost, suitability for process fluctuation - gas pressure, gas composition - led to a significant demand for such compressors.

1. Introduction

In development strategy of INCDT-COMOTI, from 2016 to 2020, an important achievement was the development of a new range of screw compressors with high discharge pressures, up to 80 bar [1], for natural gas, in collaboration with the company GHH-Rand, for both internal market and export.

The good working relationships we developed with the GHH-RAND company ended by allowing us to manufacture under license - since April 2010 - the CU screw compressors family, also the replays from our potential customers require the need to design a new family of screw compressors, with oil injection, capable of developing a maximum discharge pressure of 45 bar [2, 3]. So, we develop the screw compressor type CU128 GM, with nominal suction pressure 4.5 bar, nominal discharge pressure 45 bar, and maximum compressing power 250 kW.

The paper is logically divided, following the developing stages for a new product, in our case a screw compressor with maximum discharge pressure of 45 bar (this compressor represents a
development project of the licensed compressor type CU128G manufactured by GHH-Rand, for which the maximum discharge pressure is 26 bar). Thus, are drafted three main sections:

- Screw compressor flow simulation with CFD method [4];
- Configuration of test stand for screws compressors with maximum discharge pressure of 45 bar, suction pressure 4.5 bar, stand achieved in close-loop version [5];
- Testing of the compressor CU128 GM type on INCDT-COMOTI test stand. Data acquisition. Comparing theoretical parameters (computed in the first section of this paper) with the measured parameters. Conclusions.

2. Screw compressor flow simulation with CFD method

The CFD techniques in engineering domains became a usual approach in the last time. In our study we use a commercial CFD code, Ansys CFX, to predict the efficiency of a screw compressor [4, 6]. For this numerical investigation a CU type screw compressor class was studied [7, 8]. The fluid domain was divided in three subdomains, one rotating for the male and female rotors and two stators for suction side and pressure side. It is well known that the biggest challenge of a volumetric machine with positive displacement numerical analysis is represented by the rotors domains mesh generation. The rotors mesh was done with the TwinMesh software [4]. In the figure 1 is represented a 2D mesh section of our case. The URANS method with SST turbulence model with the BC (figure 2) was used to simulate the flow through the screw compressor [9, 10].

The CFD techniques in engineering domains became common. In our study we use a commercial CFD code, Ansys CFX, to predict the efficiency of a screw compressor. For this numerical investigation a CU screw compressor class was studied, 5 lobes for the male (300° wrap angle) and 7 lobes for the female rotor. The rotors have a length of 266.7 mm with axis distance of 128 mm and rotation speed of 4500 rev/min of the male. The gap between the rotors is 42 μm, between rotors and casing the gap is 65 μm.

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In the figure 1 is represented a 2D mesh section of our case. In the rotors-casing clearance we have 20 elements and between rotors we have 40. The number of elements for the case, rotors domains and stator parts, is over 4.5 mil elements.
As results the velocity magnitude, torque, mass flow and absolute pressure variation by rotors position resulting from the numerical simulations are presented.

Figure 2. The 3D mesh for: a) stationary fluid domains (1.9 elements); b) rotating fluid domains (2.6 elements).

Figure 3. The Ansys CFX BC

The URANS method with SST turbulence model with the BC (figure 3) was use to simulate the flow through the screw compressor. For the suction and pressure side we choose the opening condition with 3 bars pressure at the inlet and 30 bars at the outlet. In figure 3 is represented the boundary conditions for Ansys Cfx case. To simulate the oil injection, we use at the fluid models the multiphase with heat transfer and homogeneous model. At the oil boundary conditions, the pressure is 9.8 bars and the temperature 50ºC.

The aim of this study is to compare the numerical data with the date from the experimental tests. so, in Figure 4 is represented the absolute pressure contours on the screw compressor rotors. At the beginning in the screw compressor chamber the pressure is 3 bar because is connected with the suction side but after closing the compression starts through the chamber volume decreasing.

Figure 4. 2D cross section mesh.
Figure 5 shows absolute pressure and temperature in a cross plane for 72° rotation angle and in Figure 6 is represented the mass flow at inlet and outlet variation.

This study shows the CFD analyses results for an oil injected screw compressor, compressing air from 3 bars to 30 bars. The meshes for the rotating domains were done with a special meshing software and the unsteady simulations were performed with ANSYS CFX version 17.2 with multiphase, homogenous model and SST turbulence model.

Figure 5. Pressure (a) and temperature (b) on cross section at rotation angle 72°.

Figure 6. Mass flow inlet and outlet variation per one revolution.
3. Configuration of close-loop test stand for screws compressors

To achieve the configuration of test stand for screw compressor with maximum discharge pressure of 45 bar – for close loop version – were taken into account the existing equipment in the test stand, buying the new needed equipment and reconfiguring routes, in order to carry out the tests under the following conditions:

- rated suction pressure: maximum 4.5 bar
- rated discharge pressure: maximum 45 bar
- maximum compressing power: 250 kW

The stand configuration and data acquisition is achieved accordingly to the requirements of following standards [11, 12, 13, 14, 15]:

- API 619 Rotary type Positive displacement compressors for petroleum, petrochemical and natural gas industry;
- ISO 1217 Displacement compressors. Acceptance test (Annex C);
- PTC 9 Performance test code for displacement compressors, Vacuum pumps and blowers;
- ISO 5167-2 Measurement of fluid by means of pressure differential devices inserted in circular cross-section conduits running full.

The configuration/instrumentation layout of the test bench is shown in figure 7.

Figure 7. Configuration/Instrumentation layout of the test bench.
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 [16]; 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.

All measurements were obtained from electrically generated signals derived directly, respectively, from the speed, pressure, temperature and flow rates measurements. All the installation parameters (4-20 mA analogical, 0-24 Vcc digital) are supervised by a Programmable Logic Controller. The values of parameters are recorded at a certain rate of time in computer memory (25 μs).

The compressor process evolution was analysed through parameter values data acquisition [7]. Also, the parameters can be viewed in real time on the operating panel, which contains a series of screens that allow an operator to monitor the parameters.

In the figure 8 there are photos of the test bench, with the compressor installed on it.

Figure 8. Test bench.

We can see: the electric motor driven; the multiplier; the compressor unit; the oil cooler/oil filter; the oil pipeline; the air pipeline suction and discharge; reservoir; pressure regulators; instrumentations for pressures, temperatures, flows, torque, rotation; air/oil separator etc.

During the course of the test, the temperatures of the compressor surfaces were measured automatically with a Fluke monitor - Figure 9.
4. **Testing of the compressor CU128 GM type on INCDT-COMOTI test stand.**

The first tests were done by configuring the gas routes for suction pressure of 1 bar and discharge pressure of 13.5 bar (accordingly to the reception test regarding the GHH Rand schedule for testing of constructive version for CU128 G, with a compression ratio of 4.8). According to test reception, estimated parameters are as follows (for suction pressure - atmospheric pressure - and speed of 4500 rpm):

- discharge pressure: 13.5 bar
- volume flow related to suction condition: 847 m$^3$/h (1004 kg/h).

Following the compressor test, recorded data through data acquisition system enabled the drawing of diagrams with pressure suction / discharge pressure as parameters in figure 10 and flow rate/discharge pressure as parameters in figure 11.
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The experimental tests, with recirculation configuration, were carried out under the following conditions:

- the suction pressure was adjusted in (1 ... 4.5) bar range correlating it with the discharge pressure, respecting the optimal compressor compression ratio (air, $K = 1.4$ so $\pi_c = 8.99$).

The following test regimes have been defined:

A) 4500 rpm for male rotor
   - suction pressure 2 bar discharge pressure 20 bar;
   - suction pressure 3 bar discharge pressure 30 bar;
   - suction pressure 4 bar discharge pressure 40 bar.

B) 4000 rpm for male rotor
   - suction pressure 4.5 bar discharge pressure 45 bar.

The regimes were thus chosen at suction pressures of 4 and 4.5 bar - as it does not exceed 300 kW maximum drive power. Finally, due to the fact that CFD simulation requires a long running time for a regime, was chosen as the test version with 3 bar for suction pressure and 30 bar for discharge pressure.

In figure 12 shows the data obtained through the experimental testing of the chosen regime.

Through the analyse of the temperature variation from the suction side and the pressure side of the screw compressor, a 53.8 °C temperature decreasing can be observed at the outlet route (air/oil separator, pressure regulators, air cooler, pipelines). Because of that, it wasn’t needed to start the air cooler fan.

Table 1. The comparison between the GHH estimated values and the measured values from COMOTI’s test bench is presented below (SS 1 bar and PS 13.5 bar).

| Section | Suction pressure (bar) | Discharge pressure (bar) | Male speed RPM | Suction temperature °C | Theoretical flow kg/h | Real flow kg/h | Estimated power kW | Real power kW |
|---------|------------------------|--------------------------|----------------|------------------------|-----------------------|---------------|-------------------|--------------|
| Subsubsection 1 | 1 | 13.5 | 4500 | 20 | 1108 | 974 | 121 | - |
| Subsubsection 1 | 1 | 13.5 | 4500 | 21 | - | 1004 | - | 129 |
Figure 12. Test bench functional diagram.

Table 2. The comparison between the GHH estimated values and the measured values from COMOTI’s test bench is presented below (SS 3 bar and PS 30 bar).

|                    | Suction pressure (bar) | Discharge pressure (bar) | Male speed rpm | Suction temperature °C | Theoretical flow kg/h | Real flow kg/h | Estimated power kW | Real power kW |
|--------------------|------------------------|--------------------------|----------------|------------------------|-----------------------|----------------|--------------------|---------------|
| Estimate of GHH    | 3                      | 30                       | 4500           | 40                     | 3113                  | 2670           | 279                | -             |
| Test measurements  | 3                      | 31.2                     | 4500           | 36.9                   | -                     | 2615           | -                  | 289           |

The diagram from the figure 10 was done by using the parameters from PLC.

Figure 13. Suction pressure, discharge pressure and torque.

The comparison between the numerical data, obtained with CFD method, and experimental data will lead to an improvement of the design/testing procedure of the new screw compressor developed by NRDI Gas Turbine COMOTI.
5. Conclusion

The compressors are energy-consumer and a goal in the designing and the manufacturing of compressor is to reduce that energy consumption. Due to the specific process, the screw compressors can be designed so that the energy consumption is minimal for each process.

The oil injection used in technological processes for the minimization of frictions and to favour the two wounded wheel gear influences the thermodynamic process. The used oil flow may be varying between some limits which do not affect too much the compressor functioning.

The study presented in this paper shows results of a CFD analysis of an oil injected screw compressor which compresses air from 3 to 30 bar. The meshes for the rotating domains were generated by use of a special meshing software Twin-Mash and the unsteady simulations were performed with ANSYS CFX version 17.2 with multiphase homogenous flow model and embedded SST turbulence model. The aim was to compare the numerical data and the experimental test results.

Because the numerical investigation was not completed in time, finally we compared the test results with GHH Rand estimated schedule. We can see that the difference between estimated performance and measured performance-on the test bench- are small.

Finally, we believe that comparison between the numerical data, obtained with CFD method, and experimental data, will lead to an improvement of the design/testing procedure of the new screw compressor developed by NRDI Gas Turbine COMOTI.

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