

# Case Study regarding the test of the new screw compressor with high delivery pressure - 45 bara - on the test bench (with air)

Marian NIȚULESCU<sup>\*1,a</sup>, Cristian SLUJITORU<sup>2</sup>, Valentin PETRESCU<sup>2</sup>

\*Corresponding author

<sup>\*1</sup>National Research and Development Institute for Gas Turbines COMOTI  
220D, Iuliu Maniu Avenue Bucharest 061126, Romania  
marian.nitulescu@comoti.ro

<sup>2</sup>National Research and Development Institute for Gas Turbines COMOTI  
220D, Iuliu Maniu Avenue Bucharest 061126, Romania  
cristian.slujitoru@comoti.ro, valentin.petrescu@comoti.ro

DOI: 10.13111/2066-8201.2012.4.4.14

**Abstract:** Screw compressors, which are part of positive displacement compressors, can find an increasing use in gas industry and beyond due to their advantages. Maximal parameters of these devices, done by Comoti under GHH-Rand license stop at a maximum flow of 3000 Nm<sup>3</sup>/hour and 26 bara discharge pressure. The needs of potential beneficiaries have imposed the necessity to design and manufacture a new family of screw compressors and oil injection able to develop a discharge pressure of 45 bara with flow rates up to 5000 Nm<sup>3</sup>/hour. This paper presents the design and experimental test phases – for the first screw compressor with discharge of pressure up to 45 bara – CU90 HP. Continue research and development have allowed to this type of screw-compressor – to find a market position, diversifying its size range in the last decade in a manner more explosive, if we refer to other types of compressors.

**Key Words:** screw compressors, oil injected, high pressure, testing

## 1. COMPRESSION PROCESS

Screw compressor is part of the volumetric compressor family (Fig. 1).

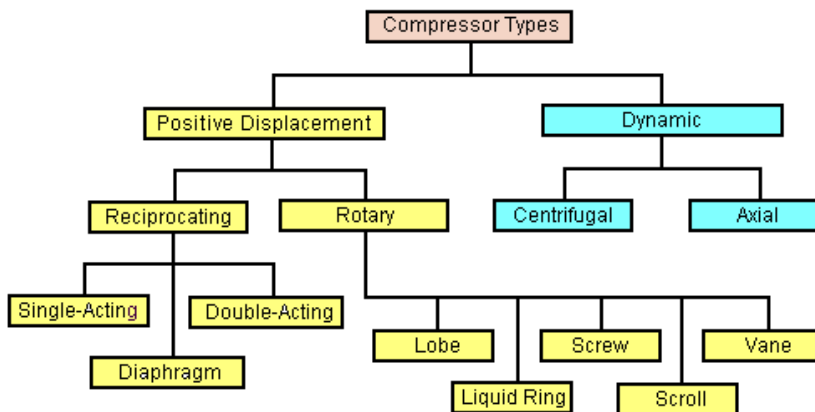


Fig. 1 - Compressors classification

<sup>a</sup> Case study coordinator

Screw compressor is made of two pairs of rotors, driver and drive - male and female - first, male, with convex lobes and the other one, female, with concave lobes. During meshing, the two rotors intertwined to form a continuous line from suction to discharge.

The space between the rotor teeth, rotor and housing becomes increasingly smaller from suction to discharge, thus compressing the gas, Fig. 2.

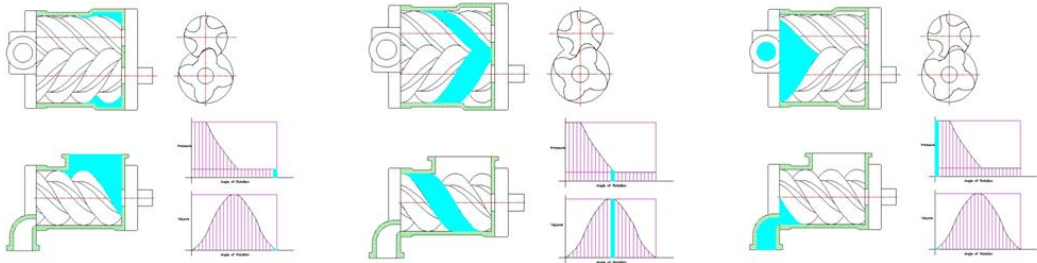


Fig. 2 - Gas compressing

Actuating the rotors is made either directly or through a pair of gears (different speeds so different flows). By lubricating the rotor, actuating the rotors - between male and female - is allowed, oil injection eliminates the need for timing gears, used in the oil-free compressor (dry).

While screw compressors without oil injection can be used at a compression ratio of 3.5 ... 4, screw compressors with oil injection can achieve a compression ratio of 10 ... 15, max. 20, in one stage. Oil is injected during compression, thus making the isotherm compression process to be close to ideal, leading to high performance (energy consumption by 10 ... 20% lower than the oil free compressor). Other advantages related to the injection of oil:

- ✚ Cooling gas through the injection of oil increases the compression ratio
- ✚ Oil injected increases the volumetric compression ratio
- ✚ By injecting the oil between the male and female rotors, it is not necessary to adjust the tolerance between the two rotors, system that is used with oil-free screw compressors.

## 2. OIL INJECTED SCREW COMPRESSORS WITH HIGH DISCHARGE PRESSURE

From the beginning we want to emphasize that long-term strategy of COMOTI is to build high in efficiency, high returns and original technological and constructive solutions, applied in engines construction, aviation systems and equipment - the Institute has a tradition of over 20 years in this field - by creating and developing efficient products and technologies, with a wide application in industry.

The quality of the human resources, together with an appropriate management strategy, allowed COMOTI products to be a constant presence in various sectors represented by an increasing number of beneficiaries of the car industry, mining, oil and gas, petrochemical industry, defense, energy industry, building materials, metallurgy and environmental protection.

Trying to meet the demands / estimates on the evolution of the operating parameters of the gas stations INCDT-COMOTI develops a wide range of screw compressors with oil injection, with custom operational parameters.

So far, we have realized in partnership with GHH-Rand, compressor provider - screw compressors with oil injection with max flow 3000 Nm<sup>3</sup>/hour and discharge pressure of max. 30 barg. We emphasize that good working relationships we had with GHH-Rand company resulted in allowing us to manufacture under license - since April 2010 - the CU screw compressors family.

The requests / signals from potential beneficiaries required to solve the need to design a family of screw compressors, capable of developing the discharge pressure of 45 bar, in a first phase, followed by the next phase with the discharge pressure of 75 bar. We also proposed increasing the flow displaced, by achieving the value of 5000 Nm<sup>3</sup>/hour.

### 3. DESIGN OF THE NEW SCREW COMPRESSOR

The experience accumulated in the partnership with GHH-Rand gave us confidence in addressing the challenge of achieving a family of screw compressors with oil injection, high discharge pressure (at this stage 45 bara).

To design and implement a completely new screw compressor, in the first phase we validated the method for designing a compressor housings derived from the licensed-type screw compressor CU 90. In this way was developed the screw compressor with oil injection CU90 HP, with the following parameters:

- ✚ Suction pressure - 5 bara
- ✚ Discharge pressure - 45 bara
- ✚ Flow rate - 1000 Nm<sup>3</sup>/h
- ✚ Working fluid - natural gas

We kept the following parts - rotors pair, bearings, parts inside, etc. And we redesigned the balancing piston and compressor housings (to withstand the test pressure of 76 bar according to API 619/ISO 10440). The material for the housing was also changed, from iron - GGG25 - to steel WCC of A 216 type. Hydrostatic pressure test was made - for housing - were the compressor assembly is subjected to experimental testing according to a schedule (tests were made with the working fluid air on the compressor test stand used for licensed CU type). Also we replaced the single seal with double seal, having in mind to reduce the leakage to atmosphere. The screw compressor of CU 90 HP type made by Comoti is presented in Fig. 3:

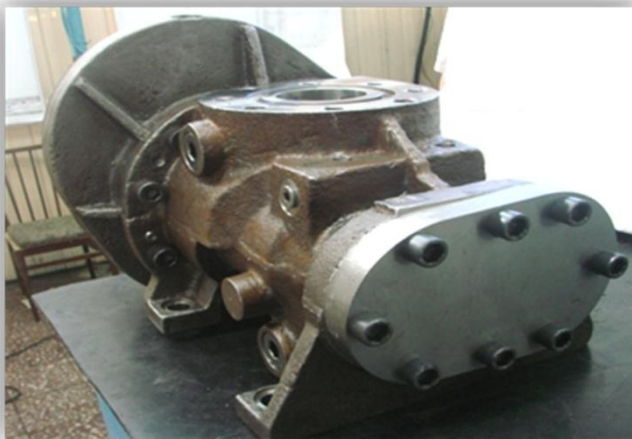


Fig. 3 - Screw compressor of CU 90 HP type - made by Comoti

## 4. EXPERIMENTAL PHASE

### 4.1. TEST BENCH

Screw compressors in the HP family - max. 45 bar - were tested on the test bench for screw compressors, existent in Comoti (used to test the licensed CU screw compressors). The experimental program for CU 90 HP compressor must take into account the technical limitations of some parts from the stand (maximum heat flux provided by the oil cooler is 55 kW, maximum power is 60 kW for the drive motor), mainly, the tests performed can be treated as partial mechanical tests (see fig. 4 and 5).

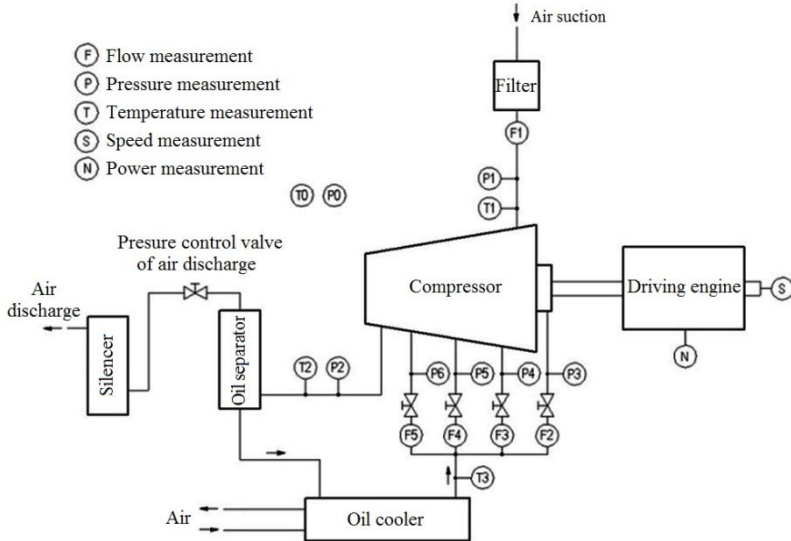


Fig. 4 - Simplified diagram of the experimental stand



Fig. 5 - Test bench for CU 90 HP

### 4.2 TESTS RESULTS

The main constrains for the COMOTI screw compressor CU 90 HP test bench were:

- maximum heat flux of the oil cooler - 55 kW;
- maximum power of the electric motor drive - 60 kW.

Technical conditions:

- suction pressure - atmospheric pressure;
- suction temperature - atmospheric temperature (100...200C – during the test);
- discharge pressure - 10; 15; 20; 25; 30; 35; 40 bar (30 minutes between stage, 15 minutes per stage);
- electric motor speed - 1300...1050 (depending on the power);
- multiplication ratio between electric driver and male rotor -  $2,25 \times 1,167 = 2,62575$ ;
- speed of the male rotor for the screw compressor - 3413...2757 rpm;
- volumetric ratio -  $V_i = 4,8$ ;

We established that the discharge pressure stages are 10, 15, 20, 25, 30, 35, 40 bar, values achieved through a manually operated control valve which is positioned at discharge, after air/oil separator (See Fig. 4). To stabilize the parameters, we stayed at each value about 15 minutes.

During the test, the data acquisition system recorded and stored in the computer hard disk the operating parameters. At the end of each test threshold, pressure step, the control operator made a screenshot of the parameters. The parameters values at 15, 30, 35 and 40 bar discharge pressure can be identified in Fig. 6.

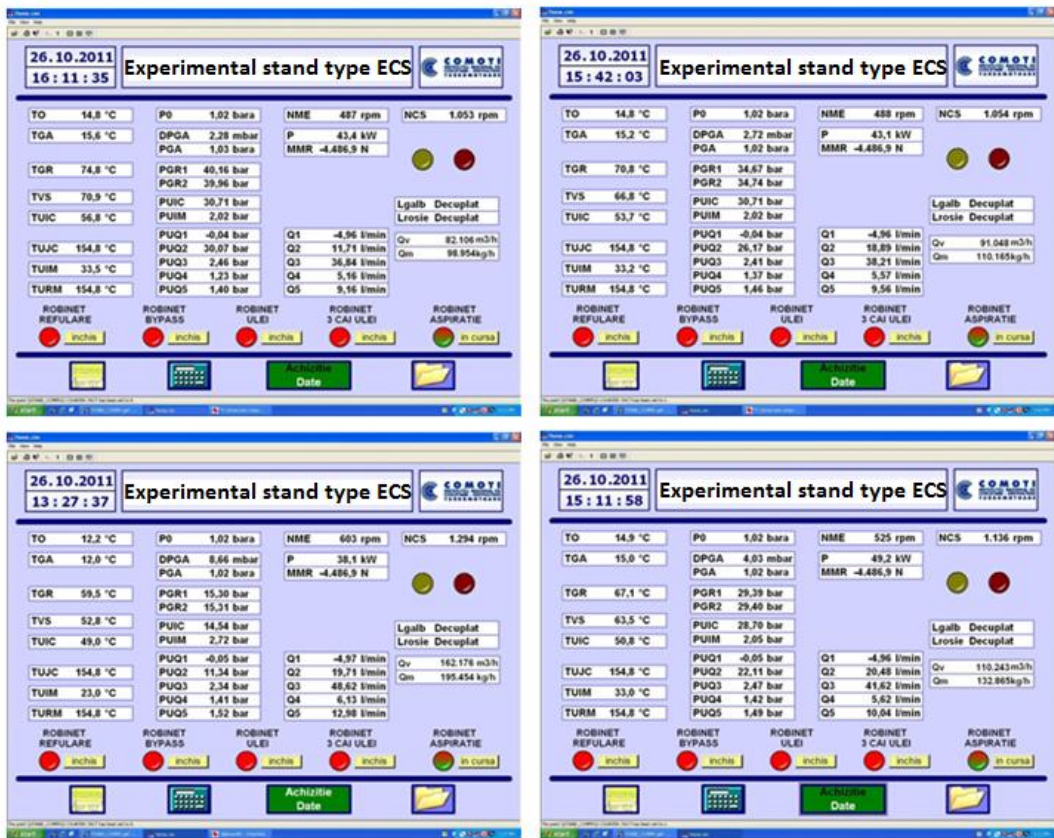


Fig. 6 - Screen shots of the operator panel for CU 90 HP

The motor drive power shall not be exceeded at the discharge pressure of 35 and 40 bar, the compressor suction flow was reduced (by lowering the engine speed drive). During the test the temperatures of the compressor surfaces were automatically measured with Fluke monitor. Fig. 7 gives information on the temperatures value during the compression process.



Fig.7- Temperatures of the compressor surfaces

### 4.3 INTERPRETATION OF THE EXPERIMENTAL TEST

We showed that, due to the conditionings imposed by the parameters of the equipment for the test stand of CU 90 HP the resulting tests performed can be assimilated as partial mechanical tests.

We considered that although, during tests there were limitations due to equipment performance, by processing the acquired data we got useful information necessary to finalize the experimental program. Thus, based on data acquired, both the volumetric efficiency and adiabatic efficiency were calculated.

The relations utilized are:

$$\eta_v = \text{volumetric efficiency} = \frac{Q_v^{\text{measured}}}{Q_v^{\text{theoretic}}} \quad (1)$$

$$Q_v^{\text{measured}} = \text{measured volumic flow rate} \left[ \frac{m^3}{h} \right] \quad (2)$$

$$Q_v^{\text{theoretic}} = \text{theoretic volumic flow rate} \left[ \frac{m^3}{h} \right] \quad (3)$$

$$Q_v^{\text{theoretic}} = 1,164 \times 10^{-3} \left[ \frac{m^3}{h} \right] \text{ for CU 90 HP} \quad (4)$$

$$P_t = \text{theoretical power for compression} = \frac{k}{k-1} RT_a \left( \pi_c^{\frac{k-1}{k}} - 1 \right) G \text{ [kW]} \quad (5)$$

$$k = \text{adiabatic coefficient (} k=1,4 \text{ for air)} \quad (6)$$

$$R = \text{gas constant} \left( R = 287,14 \left[ \frac{J}{kg^\circ K} \right] \text{ for air} \right) \quad (7)$$

$$T_a = \text{suction temperature} [^\circ K] \quad (8)$$

$$\pi_c = \text{compression ratio} \quad (9)$$

$$G = \left[ \frac{kg}{s} \right] \quad (10)$$

$$P_a = \text{absorbed power} = m_g C_p \Delta T_g + m_u C_u \Delta T_u \text{ [kW]} \quad (11)$$

$$m_g = \text{gas flow rate} \left[ \frac{kg}{s} \right] \quad (12)$$

$$C_p = \text{specific heat for gas} \left[ \frac{kJ}{kgK} \right] \quad (13)$$

$$\Delta T_g = t_g^{out} - t_g^{in} [^{\circ}K] \quad (14)$$

$$m_u = \text{total oil flow rate} \left[ \frac{kg}{s} \right] \quad (15)$$

$$C_u = \text{specific heat for oil} \left[ \frac{kJ}{kgK} \right] \quad (16)$$

$$\Delta T_u = t_u^{out} - t_u^{in} [^{\circ}K] \quad (17)$$

$$\text{Absorbed power} = \frac{\text{theoretical power}}{\eta_{ad}} \Rightarrow \eta_{ad} = \frac{P_t}{P_a} \quad (18)$$

To calculate the adiabatic efficiency, we didn't use the information regarding the power absorbed by the electric motor - we measured manually the voltage U and the intensity I - taking into consideration that the value for the absorbed power measured this way can lead to a large error.

Utilizing the data automatically acquired - by the data acquisition system - we realized the Table 1 and Table 2.

Table 1 Volumetric efficiency CU90 HP

	Male speed/discharge pressure	$\eta_v$
1	3400 / 15,3	0,68
2	2983 / 29,39	0,53
3	2767 / 34,67	0,47
4	2767 / 40,16	0,42

Table 2 Adiabatic efficiency CU90 HP

	Male speed/discharge pressure	$\eta_{ad}$
1	3400 / 15,3	0,63
2	2983 / 29,39	0,63
3	2767 / 34,67	0,40
4	2767 / 40,16	0,40

Given the conditions in which the experiments were conducted, the technical constraints that had to be taken into account while comparing with information from technical literature and business prospects, we can say that the values of these parameters are as expected and in acceptable limits.

Fig. 8, 9 show the evolution of the gas temperatures and delivery flow-measured at the suction. These diagrams were realized using the dates from the data acquisition system, automatically provided by the program.

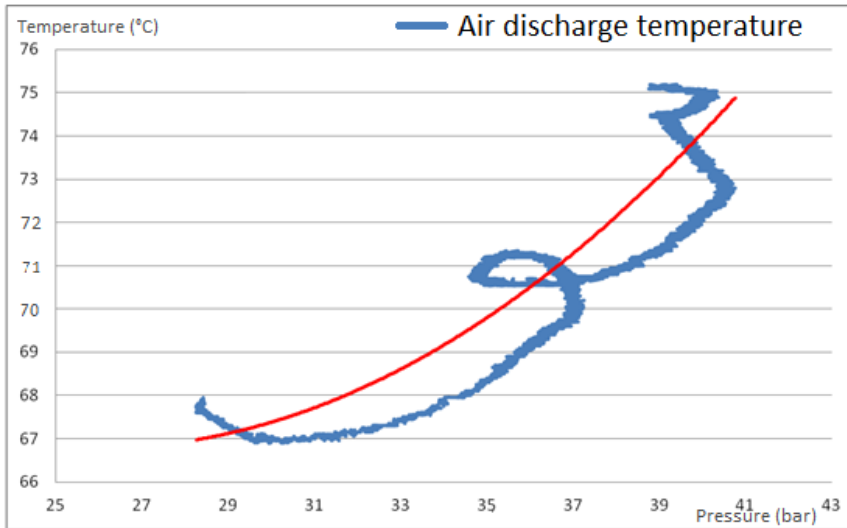


Fig. 8 Temperature variation according to discharge pressure

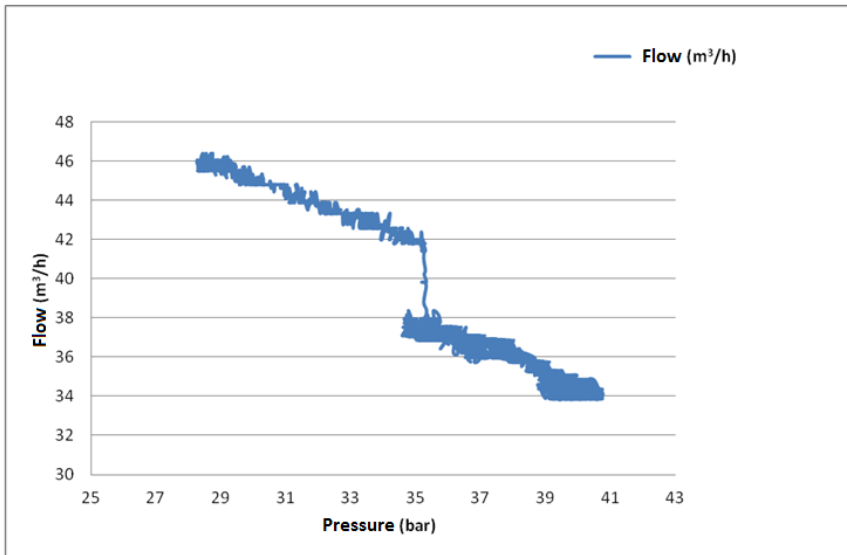


Fig. 9 Volumetric flow according to discharge pressure

## 5. CONCLUSIONS

The design of an oil injected screw compressor, with high discharge pressure - 45 bar has passed the design and an experimental phase. The compressor has undergone specific test production.

Test samples, on the stand, took place under the constraints generated by its technical possibilities (maximum heat flux of oil cooler, maximum power of the drive motor for the testing CU 90 HP).

The results can be considered normal for the testing stage performed. The completion of the tests stand for the compressors in the HP family will allow the carrying out of the experimental air tests and also the performance gas tests, in a pumping station.



The scope of this test was to validate the method for designing compressor housings-derived from the licensed type screw compressor CU 90 - and to verify how the compressors work when the compressor ratio is very high, 40 in this case.

The information's obtained was very useful for us when we decide to design a new type of screw compressor, with high delivery pressure (45 bara discharge pressure, pressure ratio 10, flow 5000 Nm<sup>3</sup>/hour at 1500 rpm).

## REFERENCES

- [1] \*\*\* API Standard 619 – December 2004, *Rotary-Type Positive-Displacement Compressors for Petroleum, Petrochemical, and Natural Gas Industries*, Fourth Edition – BINDER INCLUDED.
- [2] \*\*\* SR EN ISO 10440-1:2008, *Industria petrolieră, petrochimică și gaze naturale. Compressoare volumetrice de tip rotativ. Partea 1: Compressoare de proces.*
- [3] \*\*\* SR EN ISO 1217:1996, *Displacement compressors – Acceptance tests.*