Experimental Analysis of Twin Screw Compressor's Energetic Efficiency Depending on Volume Ratio

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ABSTRACT

The current paper presents the results of the experimental analysis to assess and optimize the twin-screw compressor's efficiency by varying the volume ratio. The experimental tests are conducted on the compressor's test bench, with a dedicated automation system inside the control console. The control and monitoring software allows parameter recording for subsequent visualization, data curation, and post-processing. The evaluation of screw compressor's performance requires a simultaneous analysis of the

thermodynamic and flow processes, both of which depend on the compressor's geometry. The obtained volumetric and adiabatic efficiencies are good, with values over 0.88 and 0.69, respectively.

Keywords-twin-screw compressor; volume ratio; energetic efficiency optimization; industrial automation

I. INTRODUCTION

Twin-screw compressors belong to the category of positive displacement machines [1], and are very common in industrial applications [2], due to their robustness, reduced dimensions and relatively low maintenance requirements. Twin-screw compressors are commonly manufactured with helically grooved rotors of uniform pitch. The main components include a set of male and female rotors, a set of axial and radial bearings, and a slide valve. All these components are encased in a housing [3]. Oil-flooded twin-screw compressors have proven very reliable for applications in oil and gas industry. Unlike the oil-free version, they have direct contact between rotors. Even though the temperature range during the operation of a compressor is not that large and does not involve very high maximum temperatures, the effect of thermal expansion is significant and the small clearances required between rotors and between rotors and housing should be maintained over all working regime conditions [4, 5]. Screw compressors also generate noise and vibration during their operation [6]. The inlet and discharge ports' shape and size may influence to a great extent the screw compressor's performance, as well as the amplitude of flow oscillations in the discharge chamber. The characteristic pulsations generated in the suction and discharge areas are created by the working fluid and have a large influence on the mechanical noise and on the noise generated due to rotor rattle [5]. There are many approaches for analyzing and improving compressors' performance and efficiency [7, 8], from the helical lobes and rotor mechanical design and minimizing the clearances between rotors and casing to energetic aspects and increasing the efficiency of the driving motor regarding its electric consumption [9-13].

Authors in [14] analyzed and compared two profile types for asymmetrical rotors, demonstrating through CFD analysis that the new methods of generating rotor profiles do not bring a major improvement on screw compressors' energy efficiency, with the differences being at maximum 0.3-0.5%. Authors in [15] analyzed experimentally a screw compressor with rotors on which an anti-friction material was deposed, with the role of both accidental touch protection and clearance reduction. The authors obtained a 3% increase of the air flow vehiculated by the compressor, at constant energy consumption. Authors in [16] simulated with CFD and experimentally verified how the closing time of a compressor's suction port improves the volumetric efficiency. They found that the optimal suction closing angle is 30^o behind the male rotor when the working chamber is at maximum volume. The volumetric efficiency increase reported is 1.5% compared to the conventional closure angle of 15^o behind. Author in [17] reported a 26% performance increase by narrowing the discharge port. The computations, however, were only done for volume ratios of 4 and 8. The paper discusses the experimental works conducted on compressors with intermediate ratios of 2.6, 3.5 and 4.8, but on different discharge pressures. Authors in [18] showed how a gas compression skid with an oil-injected screw compressor can be optimized regarding the energy consumption by adopting a lubrication system equipped with a geared pump with magnetic coupling. The compressor's power consumption estimations, based on the experimental operation of a compressor, is minimized when using a revamped lubrication system.

The literature survey shows that one cannot easily find comparisons between many compressors of the same type with different prebuilt volume ratios. The existent literature tends to present improvements of a single compressor, not addressing if the improvements perpetuate over the entire product series. The current paper deals with an experimental analysis of compressors' efficiency [19-21] by varying the volume ratio. The experimental validations outline the efficiency results. We aim to demonstrate the repeatability of the results over more compressors of the same type, presenting the results from an entire range of products, thus validating the efficiency improvements.

II. COMPRESSOR TEST BENCH

The screw compressor skid is presented in Figure 1. An oil-flooded compressor uses oil for lubricating and sealing the rotors on the air-end. This enables a quick generation of high pressure, as well as delivering compressed air on a single stage. Oil separation from air is carried out in the separator vessel, before the air leaves the system. The air is then recirculated into the air-end for another use [22].

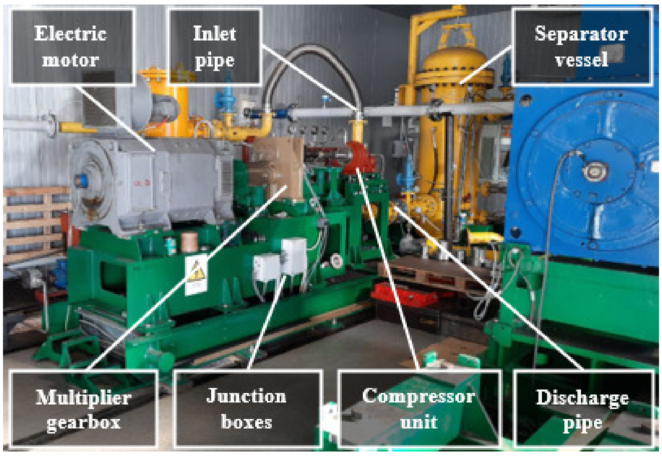

Fig. 1. Test bed with compressor skid.

The compressor is controlled and monitored by a Programmable Logic Controller (PLC) that also has the role of protecting the compression equipment, with warning limits set below the alarm limits in its custom programmed software. The warning limits are set to ensure an enhanced protection by letting the operators know that the parameters exceed the preestablished range of normal operation. These limits precede the alarm thresholds, which lead to the execution of emergency shutdown sequences. The parameters are gathered in the PLC through analogue and digital inputs, and the execution elements are commanded through digital outputs via relays. Normally, the compressors manufactured by the Romanian Research and Development Institute for Gas Turbines COMOTI are provided with automation cabinets. In this case, however, since we are dealing with a fixed facility intended solely for this test bench, the automation elements and the PLC are located inside the control console presented in Figure 2. The important parameters of the compressor (pressures, temperatures, torque, speed) and of the driving electric motor (voltage, current, electric power) are monitored and their real-time values are displayed on the screens (Figure 4).

Fig. 2. Command and control console of the compressors test bench.

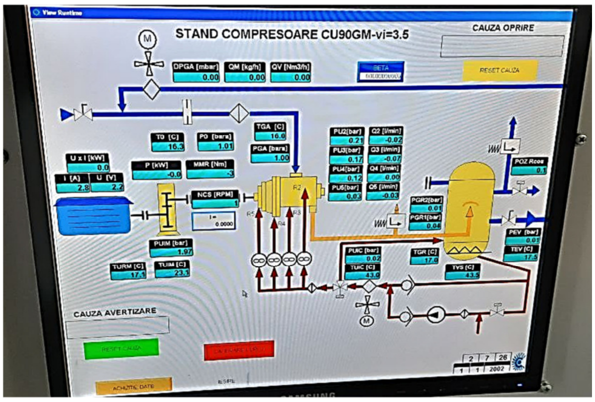

Fig. 3. The left side screen of Figure 2 displaying the parameters monitored by the PLC.

III. THE ANALYTICAL MATHEMATICAL MODEL

The compression capacity of a twin-screw compressor depends on the volume ratio and can be expressed as (1):

$$\Pi_c = V_i^k \tag{1}$$

where Π_c is the air compression ratio, V_i is the volume ratio, and k is the adiabatic coefficient, typically 1.26 ÷ 1.3.

The volume ratio V_i of a twin-screw compressor has a direct effect on the internal compression ratio, P_i. A low V_i compressor exhibits a low compression ratio, while high V_i machines are used in systems where a higher compression ratio is desired. A low volume ratio compressor has a higher gas volume at discharge than a machine with high volume ratio. A lower volume ratio will result in this case, by dividing the suction volume by the discharge volume [3]. The relation between the volume ratio and the compression ratio is expressed as [3]:

$$V_i = p_i^{\frac{1}{k}} \quad \text{or} \quad p_i = V_i^k \tag{2}$$

where V_i is the volume ratio, P_i is the compression ratio, k is the adiabatic coefficient corresponding to air (k = 1.3, it is being used herein).

The volume ratio can be calculated by:

$$V_i = \frac{V_s}{V_d} \tag{3}$$

where V_i is the internal volume ratio, V_s [m^3] the internal suction volume, and V_d [m^3] the internal discharge volume.

Alternatively, the compression ratio is calculated by:

$$p_i = \frac{p_d}{p_s} \tag{4}$$

where p_i is the internal compression ratio, p_d [bar] the internal discharge pressure, and p_d [bar] the internal suction pressure.

Depending on the manufacturer [23], different volume ratio machines can be available. Usually, the volume ratio of a twin-screw compressor lies within the range from 2.2 to 5.0 [3]. Compressors' efficiency can be assessed by calculating the volumetric, aerodynamic, and mechanical efficiencies. In each case the comparison is made to a theoretical optimum efficiency [24]. Adiabatic efficiency is a measure of the compression process efficiency with respect to the power consumption of the process. The theoretical power used in the compression of gases is defined by:

$$P_t = p_1 V_1 \frac{k}{k-1}\left[\left(\frac{p_2}{p_1}\right)^{\frac{k-1}{k}} - 1\right] \text{ [W]} \tag{5}$$

where P_t [W] represents the theoretical power, p_1 [bar] is the suction pressure, V_1 [m^3] is the suction gas volume, p_2 [bar] is the discharge pressure, and k is the isentropic coefficient of the gas.

The compressor's adiabatic efficiency is given by:

$$\eta_{ad} = \frac{P_a}{P_t} \tag{6}$$

where η_{ad} is the adiabatic efficiency, P_a [W] the absorbed power, and P_t [W] the theoretical power.

The absorbed power is calculated with (7), within the software, relying on the values given by a torque transducer and a tachometer measuring the rotational speed:

$$P_a = T \cdot \omega \Rightarrow P_a = T \cdot N \cdot \frac{\pi}{30} \tag{7}$$

where P_a [W] is the absorbed power, T [N·m] is the torque, ω [rad/s] is the angular velocity, and N [rpm] is the rotational speed.

IV. EXPERIMENTAL TESTS

The compressors were tested at the optimal pressure P_{optim} for each volume ratio V_i. The optimal pressures are provided in Table I. It is worth mentioning that bar is the standard unit for pressure when dealing with compressors, either bara (bar absolute) or barg (bar gauge), depending on the measuring reference.

TABLE I. OPTIMAL PRESSURES FOR COMPRESSORS' V_I

V_i	P_{optim}
2.6	6.1 bara
3.5	10 bara
4.8	13.5 bara

For a screw compressor with a volume ratio of V_i = 3.5, experimental tests were conducted, covering the entire operating range at different pressures and working speeds. Figure 4 illustrates the change in volumetric efficiency with respect to speed for different pressures. The volumetric efficiencies of the tested compressors are higher at lower speeds than the estimated values and the minimum efficiency obtained is 0.84, which denotes a significant improvement over 0.65. A decrease in the volumetric efficiency is observed with discharge pressure increase, due to the occurrence of more losses between the discharge chambers. A lower measured power of the screw compressors than the theoretical power calculated using the computation program provided by the original manufacturer [25] was recorded during testing. Possible reasons for this are the mechanical losses occurring due to the frictions between the mechanical parts, vibrations due to misalignments, bearings wear out, or damages of the rotor profile. All these reduce the compressor's efficiency. On the other hand, internal leakages are prone to take place, due to the clearances between the rotors and the compressor housing. During the compressor operation, a share of the pressurized air can leak back into the suction side, reducing the volumetric efficiency. This can lead to a reduction in compressor efficiency, and therefore the measured powers are lower than the calculated powers.

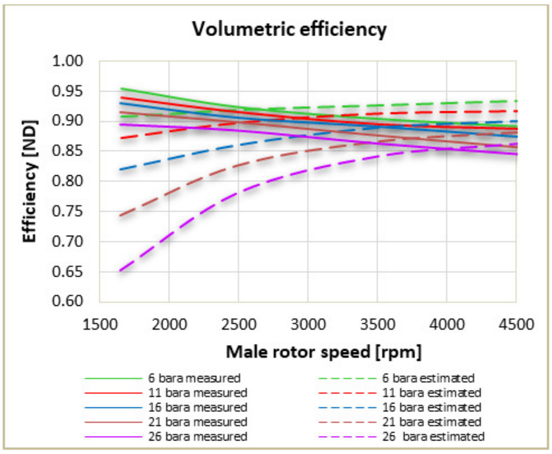

Fig. 4. Volumetric efficiency of CU90 compressors.

In Figure 5, the variation of the consumed power at various pressures is shown as a function of speed. The data obtained show a decrease of the power consumption compared to the data obtained from the theoretical prediction program [25], this decrease being more significant at high discharge pressures. At the discharge pressure of 6 bar the differences are small, up to 2 kW, while at a discharge pressure of 26 bar, the difference is significant (25 kW).

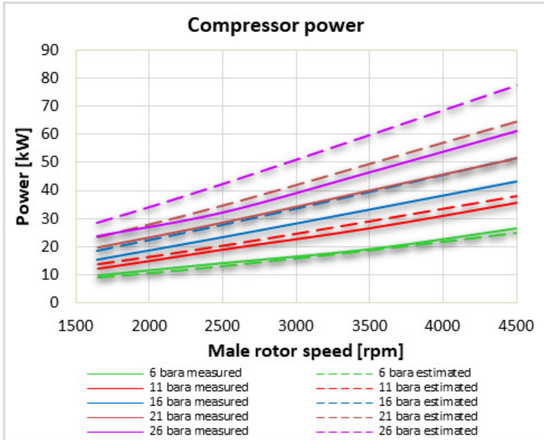

Fig. 5. Power of CU90 compressors.

The specific power is maintained quasi-constant regardless of speed. The specific power on each volume ratio did not record significantly high variations between the tested compressors with the same volume ratio. The specific power increases with the discharge pressure, considering that the volumetric and the adiabatic efficiencies are similar. The losses are generally of mechanical and thermodynamic nature, due to the friction between rotors, the friction within the multiplicator gearbox, oil and gas leakages between the compressor chambers, and heat exchange between the gas and the compressor casing (making the process to be non-adiabatic). Figure 6 presents an undesirable situation when the rotors touch the housing during operation, leading to friction and potential mechanical gripping of the compressor. In this case, critical parameters monitored by the PLC (measured absorbed electric power, vibrations, or speed) exceed the alarm thresholds, triggering the execution of the automatic emergency shutdown sequence programmed.

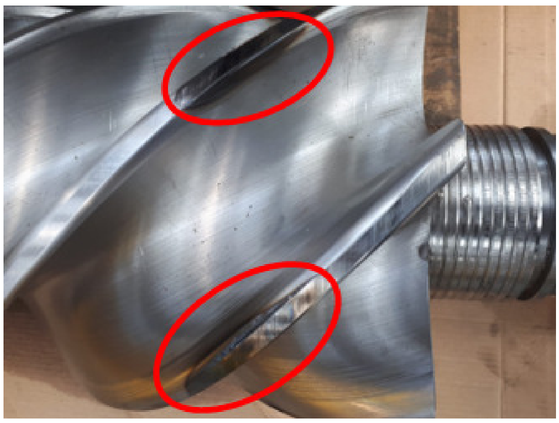

Fig. 6. Traces of rotors friction with the housing.

Figure 7 shows the variation of adiabatic efficiencies with respect to speed for various working pressures. The graph shows that the highest efficiencies are achieved when the compression ratio is close to the optimal value of 10 bar.

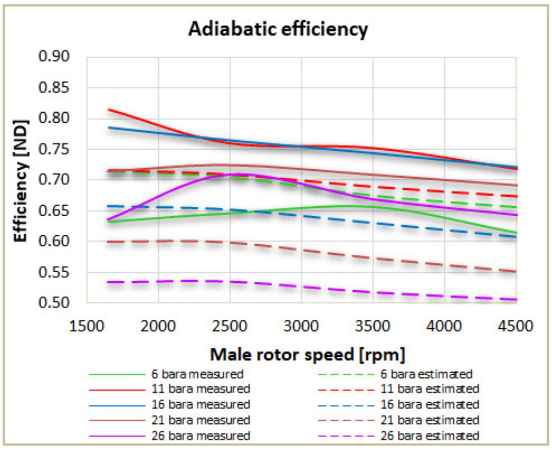

Fig. 7. Adiabatic efficiency of CU90 compressors.

The adiabatic efficiency of a twin-screw compressor decreases with the increase of speed and discharge pressure. This occurs because at higher speeds and pressures, the compression process becomes more difficult to control. At higher speeds, there is less time for gas compression, which can lead to higher losses. Moreover, higher gas speeds and pressures lead to a more turbulent gas flow and can also cause energy losses due to friction and heat transfer. At higher discharge pressures, the gas is compressed to a higher final temperature, which increases heat transfer. The temperature increase is proportional to the compression ratio, so at higher pressures, the temperature rises and energetic losses due to heat transfer can become significant, leading to a decrease in the adiabatic efficiency.

Hereinafter, we aim to show the changes in the specific power with respect to speed at various discharge pressures. A considerable improvement of the specific power consumed by the compressor can be noticed, especially at higher discharge pressures. The specific power of a screw compressor varies with compressor speed and discharge pressure. Typically, the specific power decreases with the increase of the compressor speed and with the decrease of the discharge pressure. At lower compressor speeds, the specific power is higher due to increased leakage losses and reduced efficiency. As compressor speed increases, leakage losses decrease and efficiency improves, resulting in a lower specific power. However, at higher speeds, internal losses increase due to friction and other factors, leading to a gradual increase of the specific power. Overall, the specific power of a screw compressor can be optimized by adjusting the compressor speed, the discharge pressure, and other operating parameters to obtain the optimal desired balance between efficiency and energy consumption. In the graph in Figure 8, one can observe that the specific power is quasi-constant regardless of speed. The compressor power represents the consumption in kW absorbed at a certain operating condition. The specific power represents the energy consumption to compress 1 m^3 of gas (air) in 1 min.

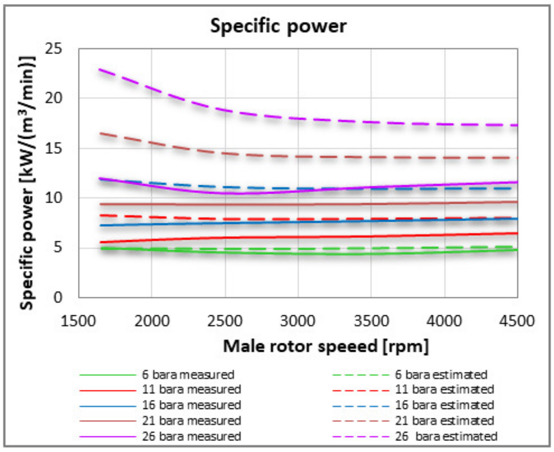

Fig. 8. Specific power of CU90 compressors.

V. RESULTS AND DISCUSSION

For the series manufacturing, a sample of 30 different compressors was considered for testing. The experimental parameters for these compressors are presented below, as 3 sets of 10 compressors designed with the same volume ratio (V_i = 2.6, V_i = 3.5, V_i = 4.8). Figure 9 shows the specific power derived experimentally for the tested compressors. It can be seen that the experiments show the specific power increase when the volume ratio is higher.

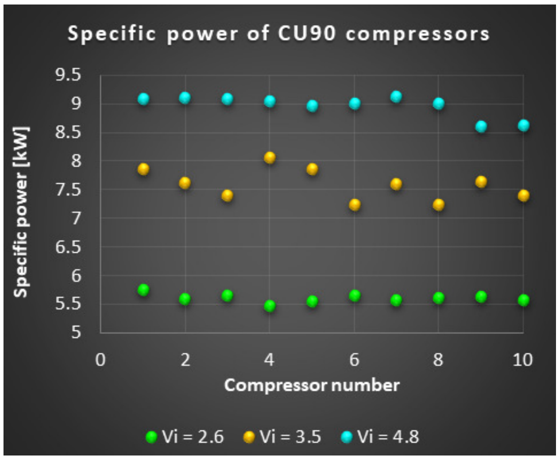

Fig. 9. Specific power for 30 CU90 compressors with different V_i.

Figure 10 shows the experimental volumetric efficiency for the 30 tested compressors. We can notice that volumetric efficiency increases when the volume ratio is lower. Compressors with an intermediate volume ratio of V_i = 3.5 show a rather unpredictable efficiency, some of them being situated both above the values for V_i = 2.6 and below for V_i = 4.8 in terms of volumetric efficiency. Figure 11 shows the adiabatic efficiency resulted from the bench tests with the 30 CU90 compressors. The compressors were tested at the same speed of 25 m/s at the tip of the male rotor. The somehow random distribution of the adiabatic efficiency is caused by the variable clearances between the rotors and between rotors and housing. The clearances between rotors range from 1 to 84 μm.

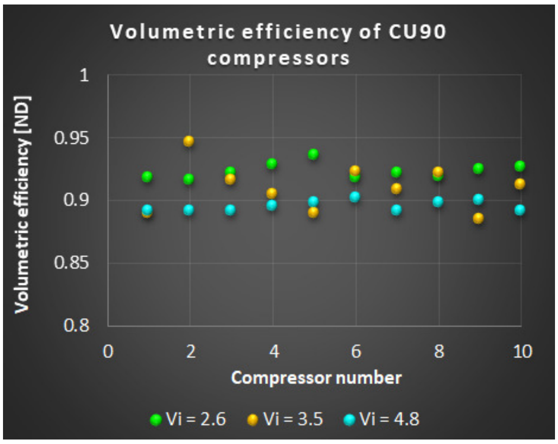

Fig. 10. Volumetric efficiency of 30 CU90 compressors with different V_i.

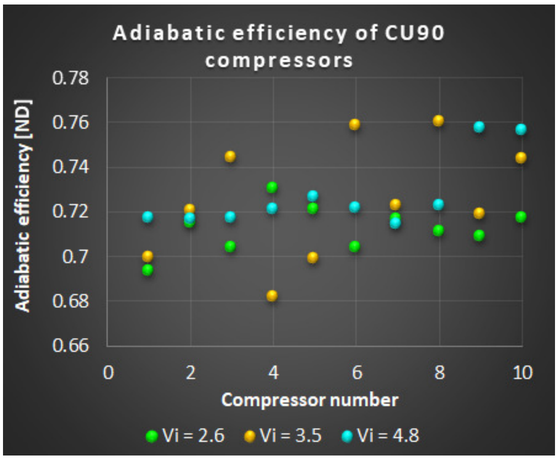

Fig. 11. Adiabatic efficiency of 30 CU90 compressors with different V_i.

VI. CONCLUSIONS

The current paper aimed to demonstrate the repeatability of the energy efficiency improvements over an entire series of compressors of the same type, presenting the results from a whole range of products. The findings have been validated experimentally. The differences between the tested compressors are due to the different internal clearances between the parts. The graphs illustrating the experimental comparisons present a sample of 30 compressors of the same type.

Achieving high volumetric efficiency at low speeds in a screw compressor can be done from multiple causes. One possible explanation is that at low speeds, the compressor can operate in a range where there are fewer leaks and more time for the gas to be compressed in the compression chambers. This may lead to higher volumetric efficiencies since more gas is compressed per unit time. Another possible explanation is that the low speed can maintain an improved stability and operation efficiency due to lower internal losses, such as friction and pressure drops between compressor components. Additionally, the compression process may be more adiabatic at low speeds, meaning less heat transfer between the working fluid and compressor walls, resulting in less energy losses and a more efficient compression process.

As a general conclusion, the volumetric efficiencies obtained in our compressors are good, with values over 0.88. The adiabatic efficiencies obtained are also good, with values over 0.69. Nevertheless, it is worth mentioning that high volumetric efficiency at low speeds is related to the specific design and operating conditions of the twin-screw compressor, as well as to the compressed gas. Therefore, further analysis and experiments are needed to be carried out in order to fully understand the reasons behind the observed behavior.

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