"POLITEHNICA" University of Bucharest
DOCTORAL SCHOOL OF ENERGY ENGINEERING

PhD Thesis Title:

Theoretical and Experimental Investigations on Energy Optimization of Screw Compressor Operation

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INTRODUCTION

Screw compressors have become an integral part of many industrial processes [19] due to their high efficiency and reliability in gas and air compression instalations. Their design and operating principles have been extensively studied and refined over the years [3], resulting in state-of-the-art designs [22] that offer improved efficiency and performance. The aim of this thesis is to investigate and evaluate the latest advancements in screw compressor technology and their potential impact on industrial applications.

The thesis will commence by providing an overview of screw compressor technology and its operating principles. This will encompass an in-depth discussion of various types of screw compressors, their components, and modes of operation. It will then analyze the recent advancements in screw compressor technology, including rotor design enhancements.

One of the primary objectives of this thesis is to assess the performance and efficiency of the latest screw compressor models. This will be achieved through a series of experimental tests conducted on various screw compressor models under different operating conditions. The data obtained from these tests will be analyzed to identify trends and determine the overall efficiency of the latest screw compressor models.

In addition to evaluating screw compressor performance, the thesis will also explore potential applications of this technology across different industrial sectors. This will involve discussing the advantages and disadvantages of screw compressors compared to other types of compression equipment, as well as reviewing case studies and real-world examples of screw compressor applications.

This thesis represents a novel contribution in this field within Romania. While screw compressors have been widely used in the industry for many years, there has been a lack of comprehensive study on their design, simulation, and testing within the Romanian technical and scientific context.

By providing a detailed analysis of screw compressor technology and its application in the oil and gas industry, this thesis serves as an inspiration for engineers and students in Romania who are interested in this field. The author has made a significant contribution to the knowledge base on this subject, and their work will undoubtedly inspire others to further explore the potential of screw compressors. In this domain, there are numerous aspects awaiting swift technical and scientific responses.

Furthermore, the author has overcome numerous challenges in testing the compressors used in this study. The oil and natural gas industry presents a complex and demanding environment, and conducting tests on compressors can be both difficult and time-consuming.

Ultimately, the purpose of this thesis is to provide a comprehensive review of the latest advancements in screw compressor technology, their performance and efficiency, as well as their potential applications in various industrial settings. By doing so, this research will contribute to a better understanding of screw compressor technology and its potential impact on industrial processes.

Relevance and Importance of the Research Field

In comparison to other types of compressors, such as piston compressors, the history and technology of screw compressors are relatively new. Alf Lysholm [25], a Swedish engineer, first attempted in 1930 to design a compressor with rotating parts that could operate at higher speeds than piston compressors, enabling it to be directly driven by a turbine. Lysholm employed a combination of rotors in configurations like 3+3 and 4+6 with varying diameters, conducting experiments in 1937, and the results were highly encouraging. The development of the compression concept using screw compressors and its industrial application was further pursued by the company SRM (Swenska Rotor Maskiner)) [5], [21]. The first company to license the technology was HOWDEN from Scotland in 1946, followed by numerous firms across Europe, America, Japan, and Eastern Europe [21]

Initially, the "oil-free" variant was developed, and then in 1957, the "oil-injected" variant for compressing air was introduced, followed by versions for gases and refrigerants in 1961. Ongoing research and development efforts have enabled screw compressors to secure their position in the market, diversifying their size range in a much more explosive manner compared to any other compressor type [3].

CHAPTER 1 Screw Compressor (General Overview)

In pursuit of achieving optimal efficiency conditions (maximum volumetric and adiabatic efficiencies, resulting in minimum specific power consumption for compressing 1 Nm^3 of gas), different aproaches for enhancing the performance of screw compressors are analyzed. Among these, the following are mentioned: clearance between rotors and housing; clearances between rotors; analysis of rotor configurations in terms of deformations (L/D ratio, materials, etc.); control of oil injection for assessing thermal deformations; rotor profiles for minimizing clearances; rotor coatings for reducing clearances/manufacturing costs (significant for tightly toleranced rotors).

On the international stage, industrial use of screw compressors began in the 1970s for oil-free screw compressors, and in the 1980s for oil-injected screw compressors. Compared to piston and centrifugal compressors, screw compressors exhibit several advantageous characteristics:

- High reliability.
- Low maintenance costs.
- Simplified foundation requirements.
- Mounting of the entire installation on an easily maneuverable skid (low transportation costs).
- Low operating costs.
- Low initial cost.

- Low power consumption relative to operating conditions.
- Adaptability to variations such as gas composition, changes in pressure/flow parameters over time.

Ongoing research and technological development have allowed screw compressors to establish themselves in the market, diversifying their size range in a much more explosive manner compared to any other compressor type. Technological progress in screw compressor construction has led to their successful use in replacing piston compressors in the oil industry.

CHAPTER 2 Screw Compressor Operation

2.1. Introduction

The normal operating range of screw compressors extends up to 3.5 bar (abs) for oil-free compressors and up to 16 bar for oil-injected compressors. For pressures exceeding 10 bar, special models are required to ensure the mechanical integrity of the compressor. Through this method, screw compressors have been designed and built to successfully operate at working pressures of 65 bar, and attempts have been made to reach discharge pressures of 80 bar.

"N" profile rotors, developed by City University, have been adopted as the most suitable for high-pressure units, and they have undergone detailed local optimizations to achieve the best performance within the operating range.

2.2. Construction Characteristics

The assembly exhibits the following construction characteristics:

- Rotor profile: "N" Rack Generated type, UK Patent 9610289.2
- Number of lobes: 5 lobes on the driving rotor, 6 lobes on the driven rotor
- Distance between rotor axes: 180 mm
- Rotor length: 412.4 mm
- Length-to-diameter ratio: 1.35
- Helix wrap angle: 300°
- Outer diameter: 256.5 mm for the driving rotor, 202.5 mm for the driven rotor
- Split diameter: 163.63 mm for the driving rotor, 196.36 mm for the driven rotor

- Root diameter: 157.41 mm for the driving rotor, 103.41 mm for the driven rotor
- Theoretical displacement: 12.5651 l/rotation
- Volumetric ratio: vi = 3.5.

2.3. Functional Parameters

a) The nominal functional parameters of the screw compressor assembly are as follows [6]:

- Suction pressure: 1 bar absolute
- Discharge pressure: 16 bar absolute
- Volumetric flow rate: 1860 Nm3/h
- Gas temperature at suction: 15°C
- Gas temperature at discharge: 80-85°C
- Conveyed gas: Natural gas
- Electric motor driving speed: 1475 rpm
- Volumetric efficiency: 0.788
- Adiabatic efficiency: 0.761

CHAPTER 3 Modeling the Operation of the Screw Compressor

3.1. Introduction

During the engagement of the rotors, when the volume created between them and the housing is at its maximum and the passage between the suction port and the volume is closed, the volume starts to decrease, initiating the compression process. When the volume passes the discharge port ("butterfly valve"), the discharge phase begins. The suction, compression, and discharge phases are schematically presented in Figure 3.1. The suction process, where the volume in the suction chamber increases, is shown in blue. The beginning of compression is indicated in green, while intermediate compression stages are shown in yellow and orange. The red portion represents the discharge phase of the compressor.

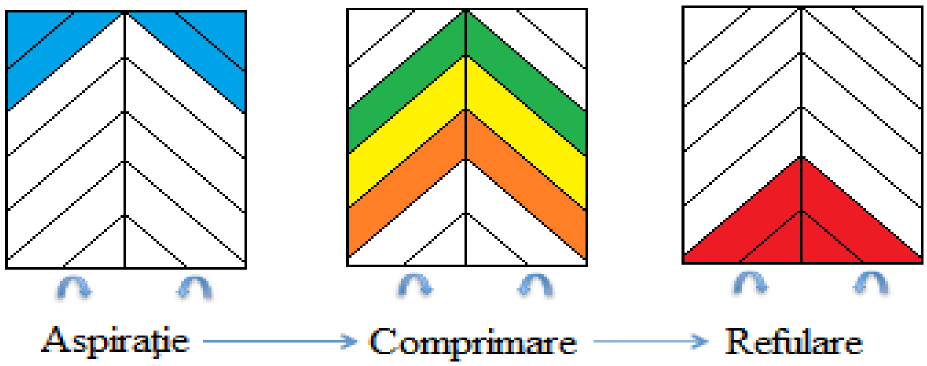


Figure 3.1. Screw Compressor Compression Scheme (author's image)

CFD simulation of oil-injected screw compressors poses several challenges due to the presence of oil within the compressor. Here are some of the challenges that need to be addressed when simulating oil-injected screw compressors using CFD:

- **Oil Injection**: Oil is injected into screw compressors to lubricate and cool the moving parts. Modeling oil injection accurately is crucial to capture the gas leakage reduction between compression chambers. However, predicting the oil injection rate and its behavior within the compressor is challenging.

- **Multiphase Flow**: The presence of oil in the compressor leads to multiphase flow, making simulation using CFD difficult. Oil droplets can coalesce and form a film on the surface, which might deform the mesh and lead to inaccurate simulation results.

- **Turbulence Modeling**: Turbulent flow inside the compressor is another challenge in CFD simulation. The presence of oil droplets can affect turbulence modeling, and different modeling approaches can yield varying simulation results.

- **Computational Resources**: CFD simulation of oil-injected screw compressors is computationally expensive and requires high-performance computing resources to achieve accurate results within a reasonable time frame.

Simulating a single rotation of a screw compressor typically involves modeling the fluid flow inside the compressor over a complete rotation cycle. The simulation captures changes in pressure, temperature, and velocity of the fluid as it moves through the compressor towards the discharge region.

The simulation process begins with creating a detailed 3D model of the screw compressor's geometry, including rotors, housing, and ports. The model is then discretized into small calculation elements, or cells, using meshing software. Mesh resolution is a critical factor that determines the accuracy and computational cost of the simulation.

The simulation software is then used to solve the Navier-Stokes equations governing fluid flow inside the compressor. These equations are numerically solved over a complete rotation cycle, usually 360 degrees, using a time-stepping approach.

The solution considers boundary conditions such as compressor inlet and outlet pressure and temperature, rotation speed, and compressor geometry.

The simulation grid for the screw compressor has a resolution of 1.5 degrees for a 72-degree sector, and the remaining sectors are modeled using cyclic conditions.

To describe the operation of the screw compressor, a complete rotation of the driving rotor is chosen for presentation.

3.2 Calculation Domain, Discretization Domain, and Boundary Conditions

The compressor utilized for this study is an oil-injected compressor with 5/6 lobes. The operational speed of this compressor ranges between 500 and 1500 rotations per minute. The diameter of the driving rotor is 256.692 mm, while that of the driven rotor is 202.796 mm.

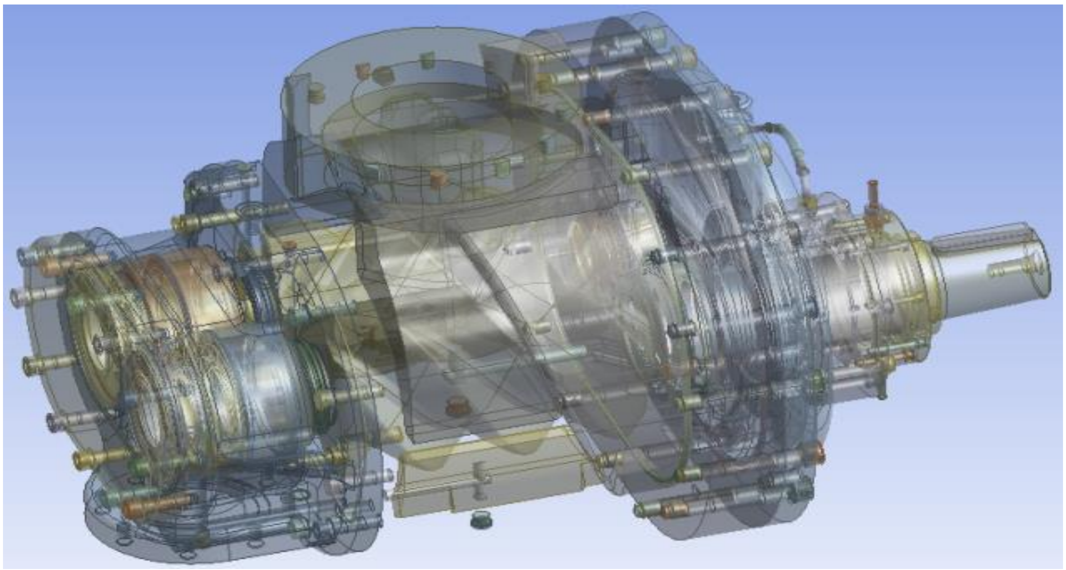
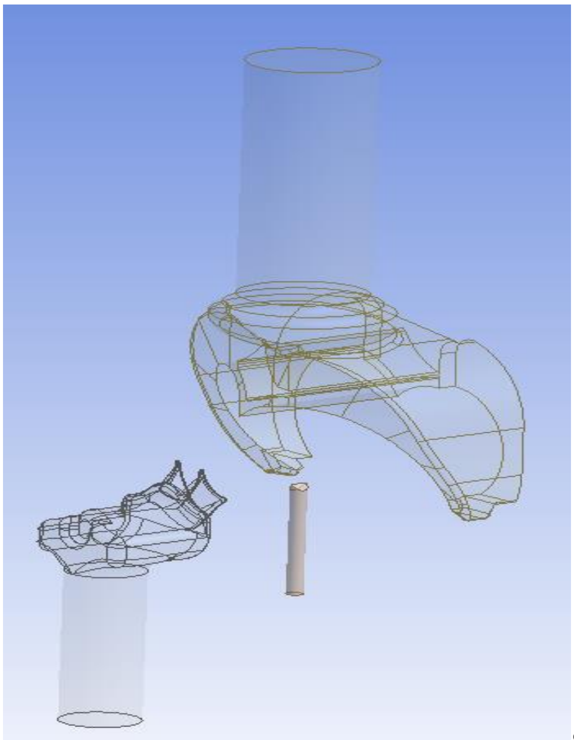


Figure 3.2. Compressor 180 - 3D CAD (author's image)

The rotor length is 413 mm, and the wrap angle is 300 degrees. Clearances between the rotors can vary between 1 micron and 30 microns. The shape of the discharge port was designed for a volumetric ratio of 3.5. Figure 3.2 presents this compressor in a 3D CAD representation.

Using the geometry of the 180 screw compressor, the domain for this study has been defined [10]–[16]. Starting from the 3D CAD model, the domain for the statoric part of the screw compressor was defined, which consists of three subdomains: a) the suction subdomain, b) the oil inlet subdomain, and c) the discharge subdomain. Boolean functions were used to create the walls for domain definition. Figure 3.3 (a) illustrates the three subdomains created using Ansys's DesignModeler software.


a)

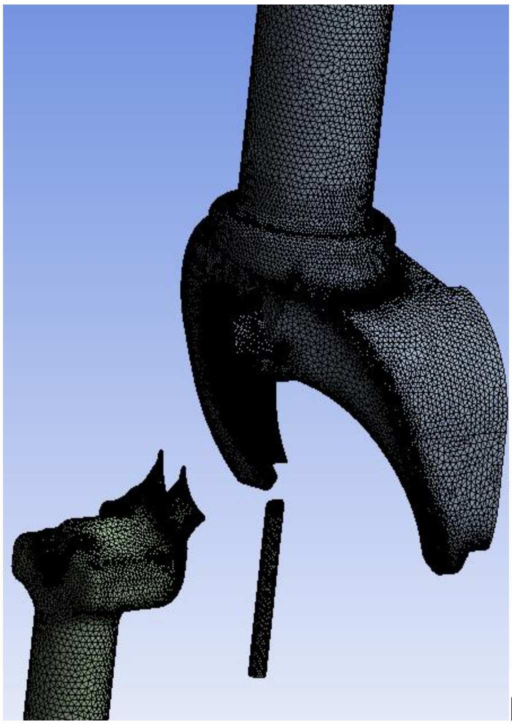

b)

Figure 3.3. a) Stator Domain of the Compressor; b) Calculation Grid for the Stator Domain (author's image)

The grid (Figure 3.3b) for the stator component was generated using Ansys Meshing, employing meshing methods based on maximum element size for both surface and volume.

For the rotor domain, the commercial software TwinMesh, designed for positive displacement rotary machines, was employed to generate the grids. Figure 3.4 displays the 2D grid along with its quality assessment, while Figure 3.5 showcases the 3D grid of the rotor domain.

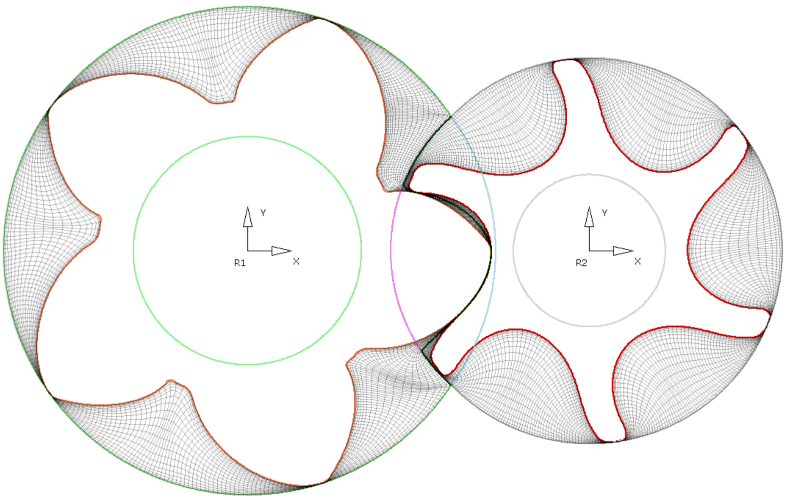


Figure 3.4. 2D Section of the Rotor Domain (author's image)

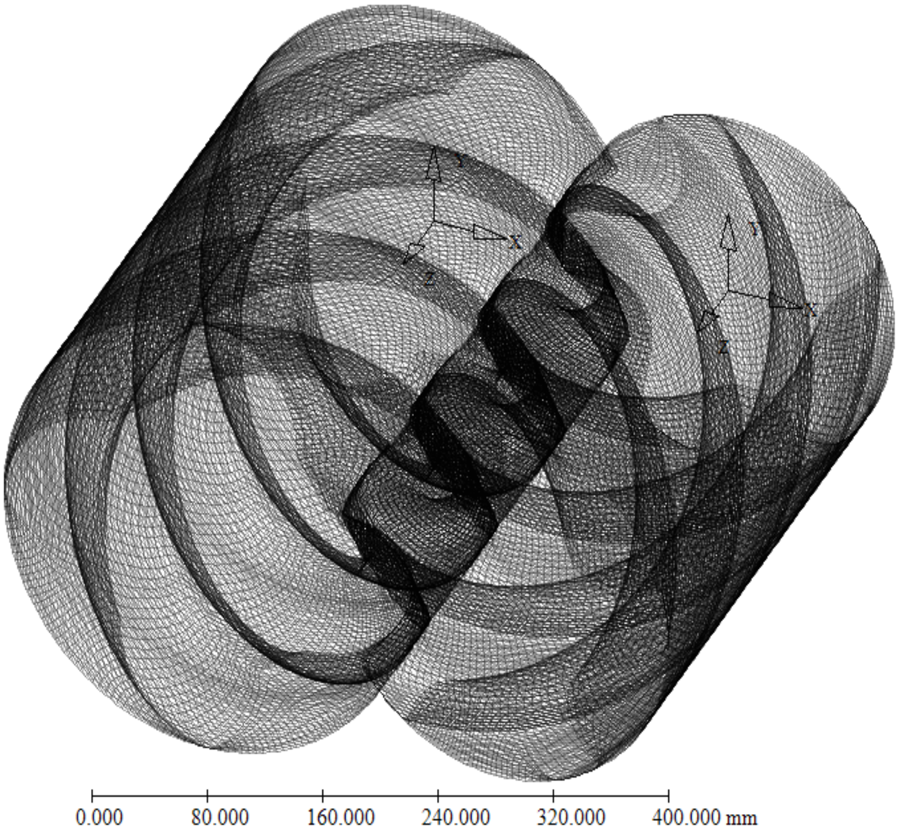
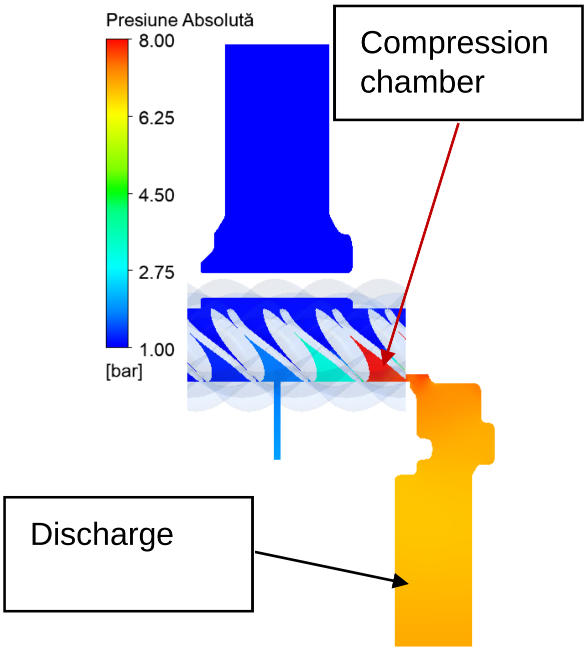


Figure 3.5. 3D Discretized Rotor Flow Domain (author's image)



A 2D section of pressure variation in a screw compressor, parallel to the rotor axis, illustrates the distribution of absolute pressure within the compressor at a specific moment during the compression cycle (Figure 3.6). This section is taken perpendicular to the rotor axis and offers a cross-sectional view of the compressor.

In this section, pressure contours are displayed using different colors to represent various pressure levels. The highest pressure values are located at the discharge port, while the lowest

Figure 3.6. Pressure variation - Pressure 7 bar (author's image)

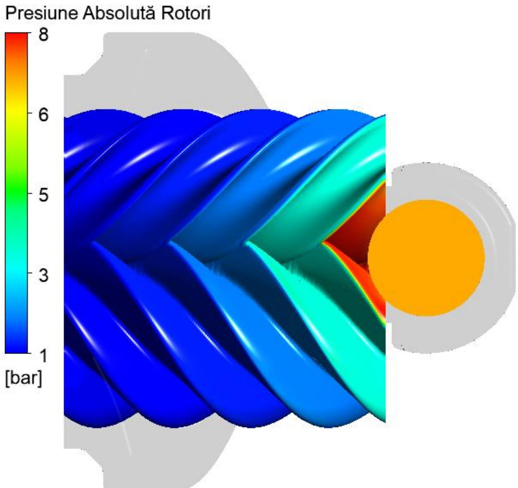


Figure 3.7. Pressure variation - Pressure 7 bar (author's image)

pressure values are found at the suction port.

Figure 3.7 provides a 3D view of pressure during the start of the discharge process, when the discharge port begins to open for the first time.

Pressure pulsations on the discharge pipe are observed due to the cyclic nature of the screw compressor's operation, generating pressure waves that travel through the discharge pipe. These waves can cause fluctuations in flow rate, which can impact compressor efficiency, vibration levels, and the fatigue life of components.

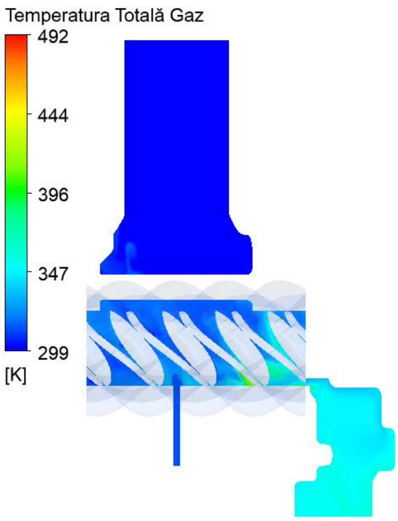


Figure 3.9. Temperature variation - Pressure 7 bar (author's image)

In Figure 3.8, temperature changes are depicted as the gas goes through the compression process. The temperature evolution is closely related to pressure variation and can be influenced by factors such as gas properties, compressor geometry, compression ratio, and operating conditions.

During the suction stage of the compression cycle, the gas enters the compressor chamber at a relatively low temperature. As the gas is compressed, its temperature increases due to the adiabatic compression process, where temperature rises as pressure increases.

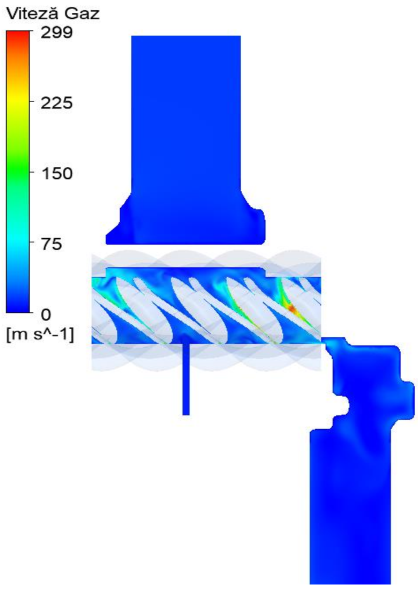


Figure 3.9. Gas velocity variation - Pressure 7 bar (author's image)

The temperature increase is influenced by gas properties, such as heat capacity ratio and specific heat, as well as compressor geometry, such as screw profile and rotor clearances.

Figure 3.9 illustrates the velocities of leaks occurring in a 2D plane between the compressor chambers, leading to oil loss and reduced compressor efficiency. The oil and gas velocities associated with these leaks can depend on various factors, including the size and location of the leak, pressure difference between the gas and oil systems, and the gas flow rate through the compressor.

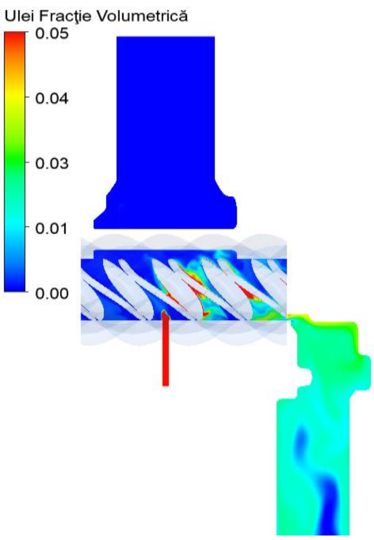


Figure 3.10. Oil fraction variation - Pressure 7 bar (author's image)

In Figure 3.10, the volumetric fraction of oil in the screw compressor is presented in a 2D section. In a screw compressor, oil is typically injected into the compressor chamber through one or more injection ports.

The distribution of the oil fraction from the injection port in a screw compressor can be influenced by various factors, including the location, size, and number of injection ports, as well as operating conditions and compressor geometry. This visualization provides insights into how the oil is distributed within the compressor chamber and its interaction with the gas during the compression process.

3.3. Torque Diagram:

The torque generated by a screw compressor is an important parameter that reflects the power required to drive the compressor rotors. To create a torque diagram, simulation results can be used to calculate the forces acting on the compressor rotors and convert them into torque values. The torque diagram provides information about the torque required to rotate the compressor under various operating conditions, such as different gas flow rates and compressor speeds (Figure 3.11).

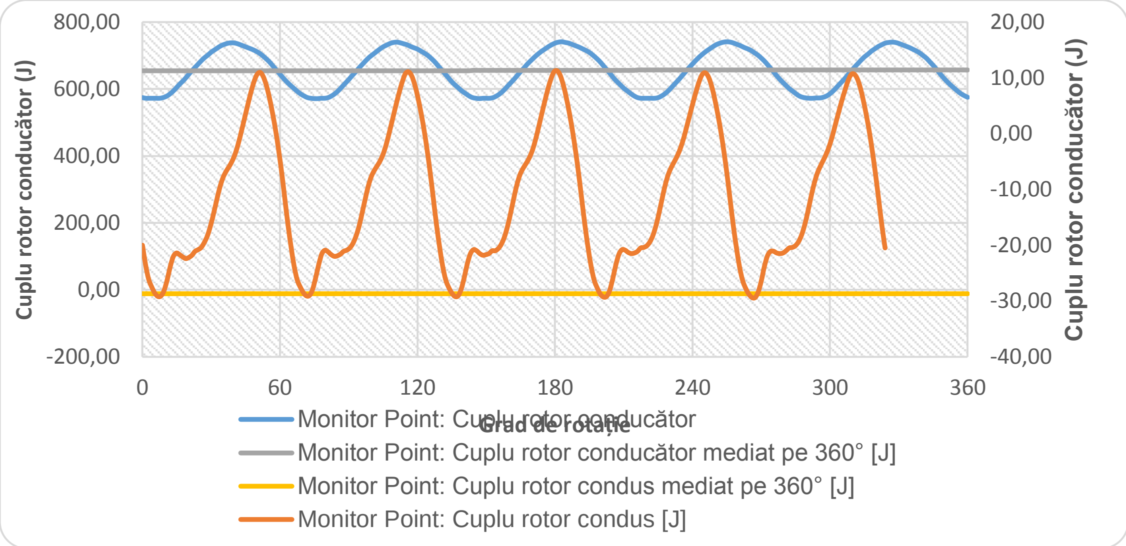


Figure 3.11. Torque Variation at 7 bar (author's diagram)

3.4. Gas Flow Diagram:

The gas flow rate through a screw compressor is another critical parameter that reflects its performance. To obtain the gas flow diagram, simulation results can be used to calculate the gas flow rate under various operating conditions, such as different compressor speeds and suction pressures. The gas flow diagram provides information about the compressor's capacity to deliver gas under different operating conditions (Figure 3.12). This diagram is essential for assessing the compressor's ability to handle varying gas demand and optimizing its operational efficiency.

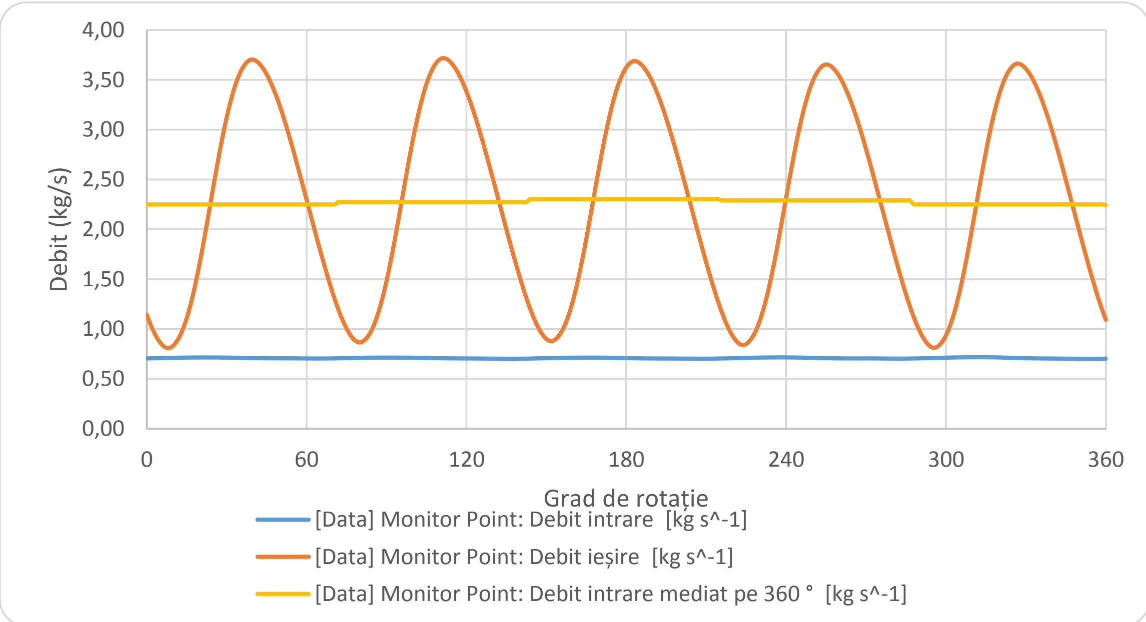


Figure 3.12. Compressor Flow Rate at 7 bar (author's diagram)

3.5. Volumetric Efficiency Diagram:

Figure 3.13 presents the volumetric efficiency of a screw compressor. This efficiency reflects the compressor's ability to deliver a certain gas flow rate for a given displacement volume. To obtain the volumetric efficiency diagram, simulation results can be used to calculate the volumetric efficiency under various operating conditions, such as different discharge pressures and compressor speeds. The volumetric efficiency diagram provides information about the compressor's efficiency in delivering gas under different operating conditions (Figure 3.13). This diagram is crucial for understanding how effectively the compressor can meet gas delivery demands while considering its internal dynamics and interactions with various parameters.

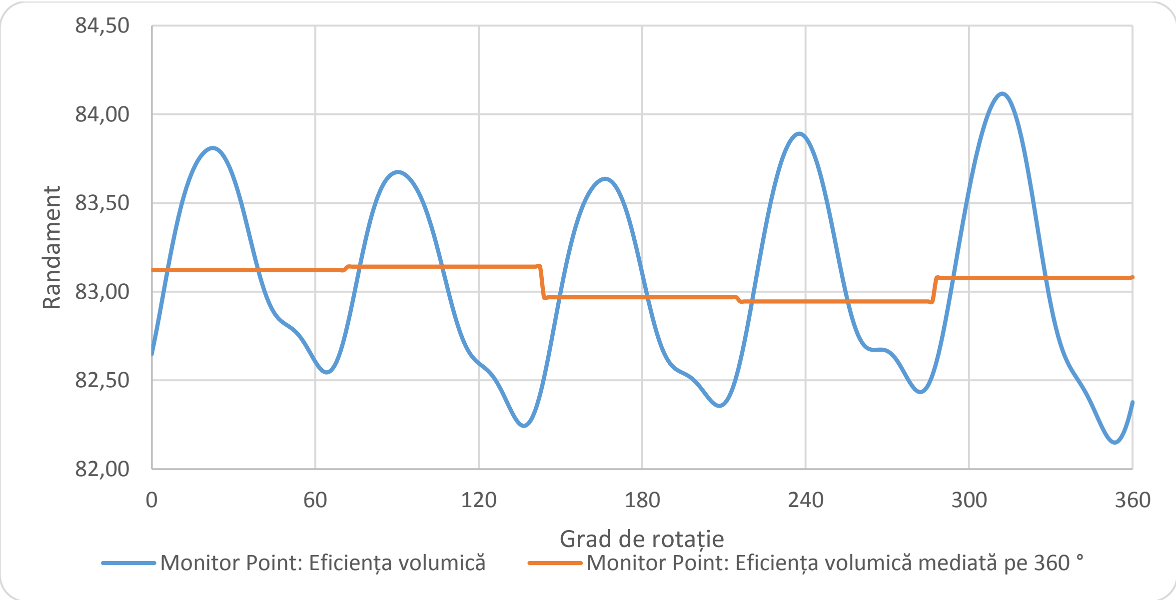


Figure 3.13. Volumetric Efficiency at 9 bar (Author's Diagram)

3.6. Diagram of Pressures in the Compression Chamber:

The pressure within the four adjacent compression chambers of a screw compressor, as depicted in Figure 3.14, reflects the pressure profile of the compressor after the discharge chamber has closed. To obtain the pressure diagram, simulation results can be used to calculate the pressure in each of the four adjacent compression chambers under different operating conditions. The pressure diagram provides information about the pressure distribution within the compressor and can be utilized to optimize the compressor's geometry and enhance its performance.

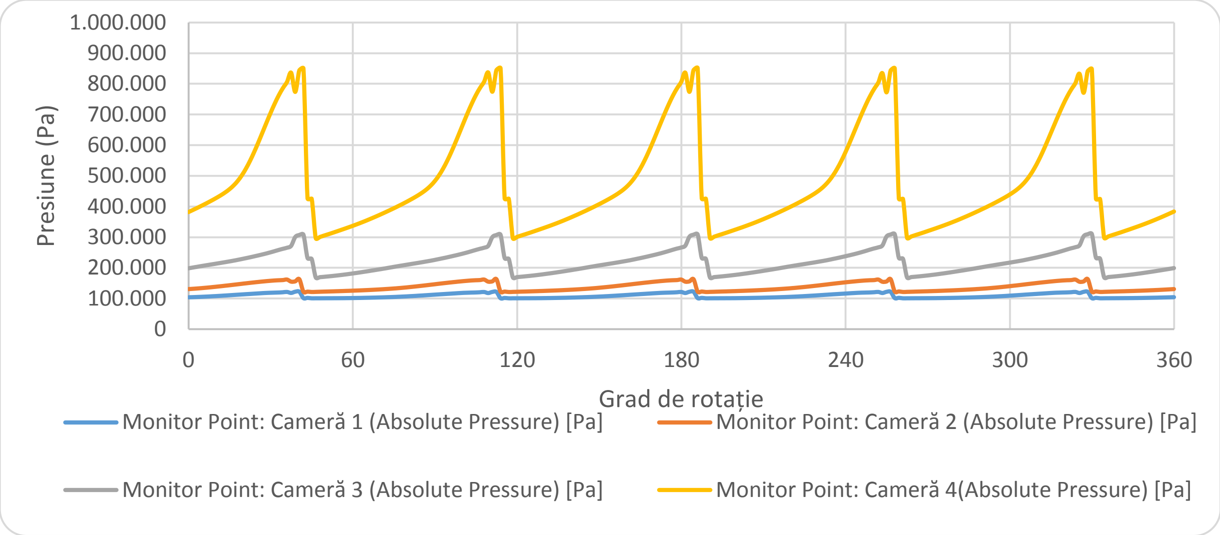


Figure 3.14. Pressure Variation at 7 bar (Author's Diagram)

3.7. Absorbed Power Diagram

The power absorbed (Figure 3.15) by a screw compressor reflects the energy consumption required to rotate the compressor and deliver gas at a specific flow rate and pressure. The power absorption diagram provides information about the energy consumption of the compressor and can be used to optimize its energy efficiency.

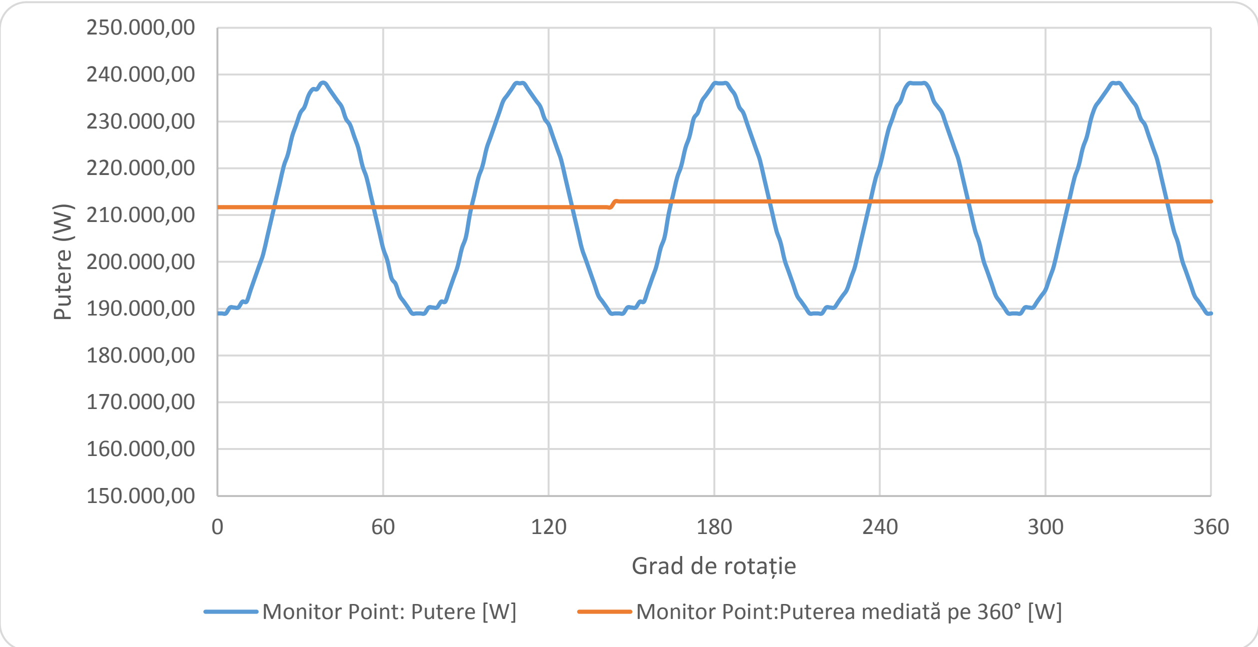


Figure 3.15. Power Variation at 7 bar (author's diagram)

3.8. Conclusions

In short, simulating the gas flow in a screw compressor can provide valuable insights into its performance and efficiency [23], [24], [26]–[28]]. By using simulation results to generate diagrams for torque, gas flow rate, volumetric efficiency, pressure in the 4 adjacent compression chambers, power absorption, maximum pressure, and temperature, we can optimize the design and operating parameters of the compressor to enhance its performance and efficiency.

In the case where the compressor completes a full 365-degree rotation, the compression chambers open and close five times, resulting in a sinusoidal variation of the gas flow, power, pressures, and torque of the compressor.

In the design of screw compressors, each of the following parameters plays a crucial role in their performance and efficiency:

- Suction pressure: It's the pressure of the gas entering the compressor, influencing its ability to handle gas efficiently. Too high suction pressure can overload the compressor, while too low pressure can reduce its efficiency.
- Discharge pressure: It's the pressure of the gas at the outlet, affecting the compressor's ability to deliver gas at the desired pressure. Too high discharge pressure can overload the compressor, while too low pressure can reduce efficiency.
- Volumetric flow rate: It's the amount of gas per unit time that the compressor can compress, influencing its capacity to handle application flow rate requirements.
- Suction gas temperature: It's the temperature of the gas entering the compressor and can affect the compressor's lifespan and efficiency. Too high suction temperature can lead to overheating, while too low temperature can cause condensation inside the compressor.
- Discharge gas temperature: It's the temperature of the gas at the outlet and can affect the compressor's performance and lifespan. Too high temperature can lead to overheating and damage lubricating oil, while too low temperature can impact efficiency.
- Compressed gas: It's the gas being compressed and must be compatible with the compressor design. Physical and chemical properties of the conveyed gas can affect the compressor's lifespan and performance.
- Electric motor drive speed: It's the speed at which the electric motor driving the compressor operates and can affect volumetric flow rate, efficiency, and power needed to drive the compressor.
- Volumetric efficiency: It's the ratio of the volume of gas sucked to the volume of gas expelled and influences compressor efficiency. The higher the volumetric efficiency, the more efficient the compressor.
- Adiabatic efficiency: It's the ratio of adiabatic power performed by the compressor to the theoretical energy required to compress the gas. Adiabatic efficiency measures the compressor's ability to convert electrical energy into

mechanical energy and is influenced by the compressor's design and the properties of the compressed gas.

In general, an efficient screw compressor design must consider all these parameters and strike a balance between them to achieve optimal performance and reduced operating costs.

CHAPTER 4 Design and Experimentation of the Screw Compressor

4.1. Development of the Technological and Instrumentation diagram

This diagram is used to select the equipment and instrumentation necessary for creating the testing stand for the screw compressor, their interconnections, and their placement within the assembly. Additionally, the technological diagram defines the role of the equipment during the compressor's operation, the limits imposed for each measuring instrument, as well as the safeguards defined for emergency shutdown of the assembly in case of a malfunction or exceeding a specified parameter value.

The operational diagram of the screw compressor stand forms the basis of the 3D assembly drawing, where the necessary equipment for stand operation and their interconnections are defined. Following this diagram, the testing procedure for the screw compressor and the stand's automation system are developed.

The screw compressor (4) is placed on a frame and is driven by a main direct current electric motor of 500 kW (57), a speed multiplier (58), and a torque transducer (59). The screw compressor stand allows testing the compressor over a speed range from 500 rpm to 3000 rpm, thus enabling the determination of functional parameters across a wide speed range.

Next to the frame, an oil injection pipe is located. It consists of four branches entering the compressor: mechanical seal, speed multiplier, injection between rotors, and discharge bearings.

The lubrication circuit for the mechanical seal, speed multiplier, injection between rotors, and discharge bearings consists of a flow restrictor (44, 43, 42, 41), a flow transducer (48, 47, 46, 45) to restrict the oil flow. At its end, a pressure gauge (56, 55, 54, 53) and a pressure transducer (52, 51, 50, 49) measure the oil pressure right at the compressor inlet.

The pressure in the injection line is provided by a gearwheel pump driven by an alternating current motor controlled by a frequency converter (25). The oil pump ensures an oil flow rate between 48 and 233 l/min within speeds of 250-1500 rpm. The oil entering the pump passes through a 25 μm filter (24).

The oil pump is placed in parallel on the injection line so that it can be stopped in case the compressor needs to operate in a self-lubricating mode. In order to operate in parallel on the injection line, two check valves (26 and 23) are installed to prevent reverse oil flow.

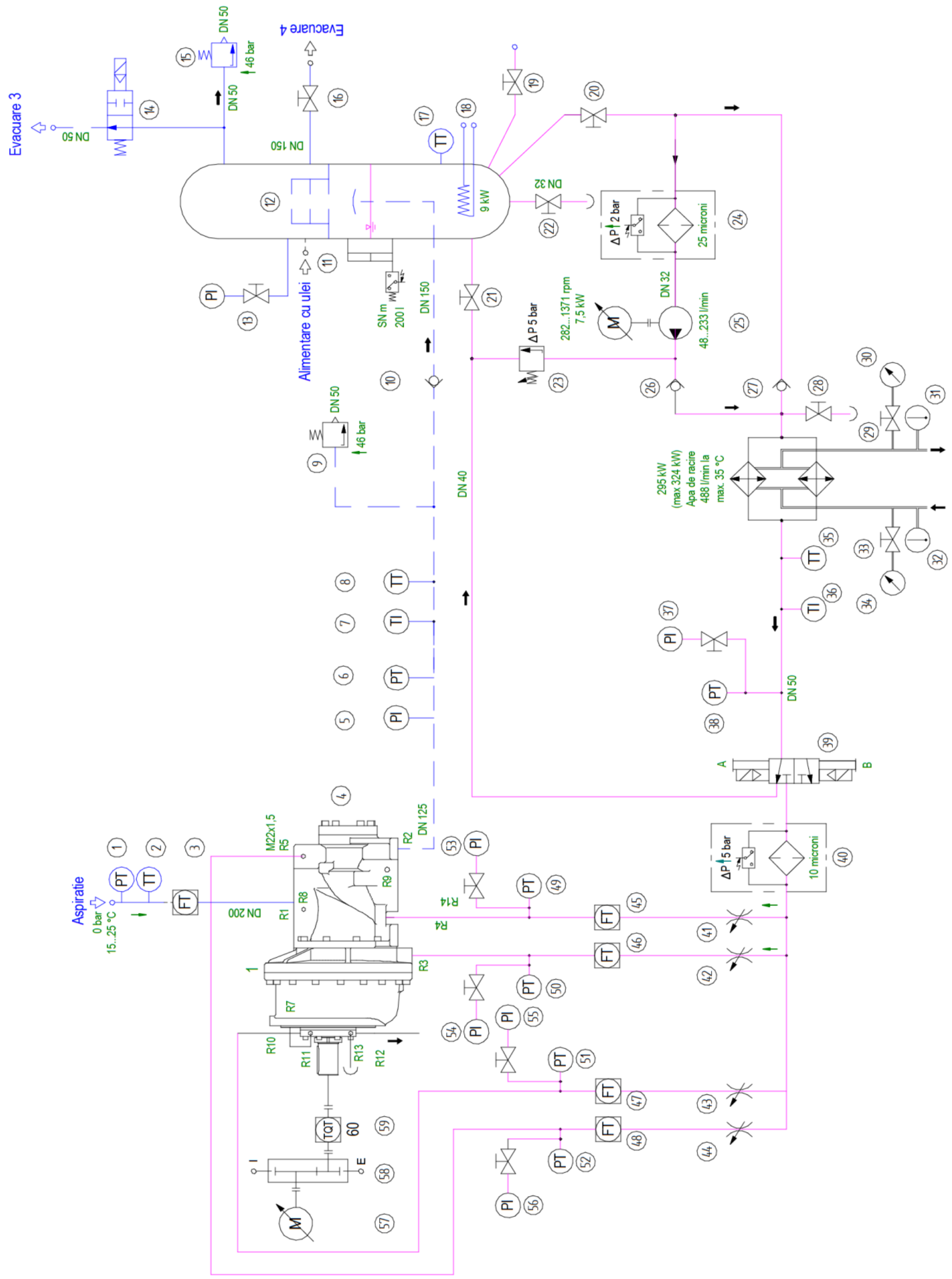


Figure 4.1. Technological and Instrumentation Diagram of the Screw Compressor Testing Stand (author's diagram)

To protect the lubrication system, a safety valve (23) is installed on the pump and discharges into its suction. The safety valve opens when the pressure differential between the pump suction and discharge reaches 5 bar.

This oil flow rate is sufficient for the entire operating range of the screw compressor.

At the pump outlet, a water-oil heat exchanger with a capacity of 320 kW is located. To adjust the amount of heat dissipated through the cooler on the inlet and outlet, two valves (62 and 63) have been installed to regulate the water flow into the heat exchanger, along with two pressure gauges (30 and 34) and thermometers (31 and 32) to monitor pressure losses across the cooler and the temperature of the water.

At the outlet of the cooler, there are a pressure transducer (38), a pressure gauge (37), a temperature sensor (35), and a thermometer (36) to monitor the oil pressure and temperature in the injection manifold.

After the oil cooler, a three-way valve (39) is installed, allowing oil recirculation during standstill. This operation is used to raise the oil temperature to 35°C before starting, which is the recommended minimum temperature for oil when using self-lubrication.

Oil injection into the screw compressor can be carried out through self-lubrication or with the help of the oil pump. Self-lubrication injection can only be performed when the pressure drop across the entire lubrication system is less than 2 bar, and the pressure in the separator vessel is above 4.5 bar.

On the lubrication system, after the three-way valve, a 10 µm oil filter is installed, which is changed when a pressure drop greater than 2.5 bar is detected.

On the discharge line from the compressor, a pressure transducer (6), a pressure gauge (5), a temperature sensor (8), and a thermometer (7) are installed to monitor the pressure and temperature of the gas-oil mixture. To prevent reverse flow, a check valve (10) and a safety valve are installed to prevent pressure from exceeding the 45 bar limit.

The gas-oil mixture enters the separator vessel (12), where the oil is separated from the gas through multiple separation stages. The vessel is equipped with a pressure gauge (13) to visualize the air pressure after separation and a temperature sensor (17) to monitor the oil temperature in the vessel. During each startup, the oil temperature is checked with the temperature sensor (17), and if it's below 35°C, the heating resistor (18) is turned on.

The separator vessel is equipped with a level indicator (11) that is inspected at each startup to avoid testing compressors below a certain minimum level.

In case of an emergency shutdown of the unit, the pneumatically operated valve (14) opens to depressurize the separator vessel.

4.2. Design Model of the Test Compressor

Based on the technological and instrumentation diagram, the screw compressor and the testing stand were modeled. The presented screw compressor is an INCDT COMOTI design, illustrated in a 2D drawing where its components can be observed.

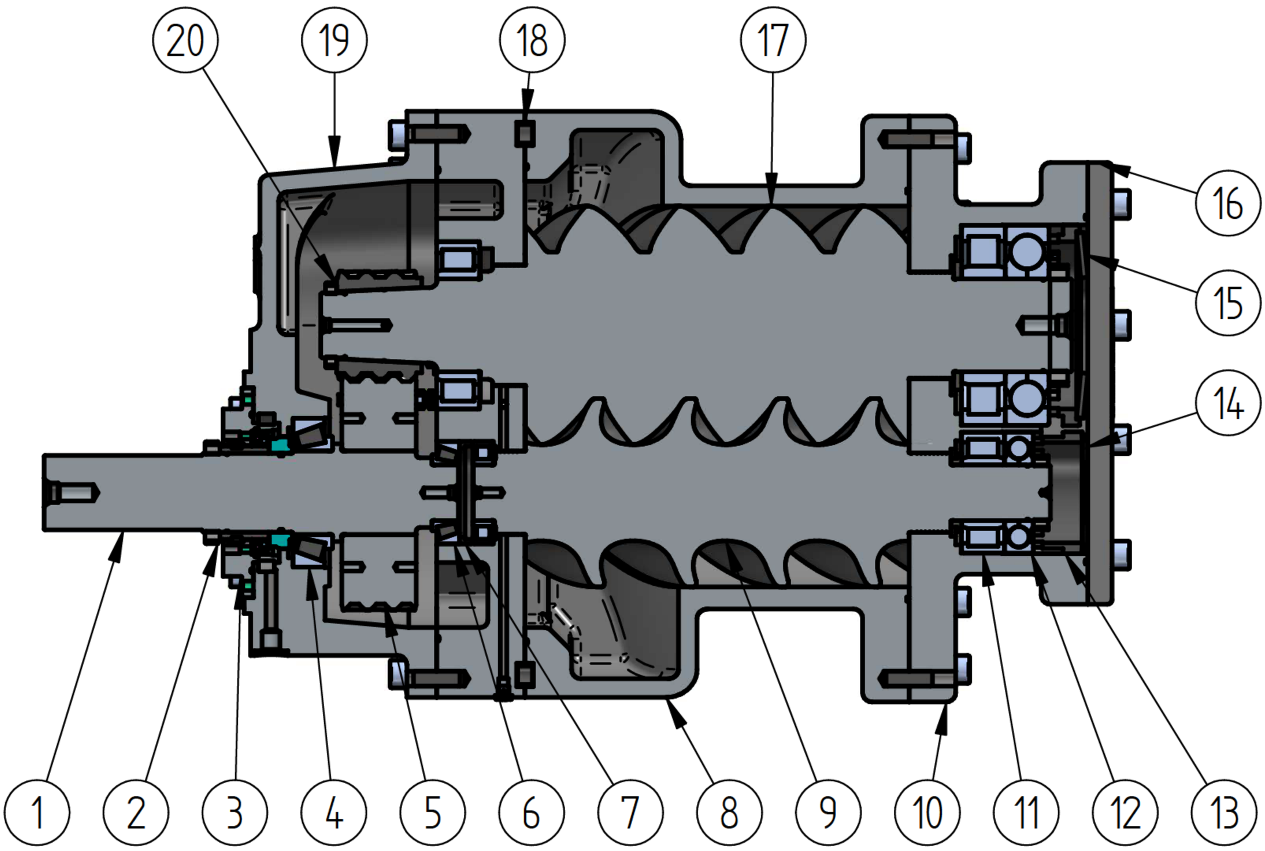


Figure 4.2. Component Parts of the Screw Compressor (author's image)

Table 4.1. Screw Compressor Composition

1	Drive Shaft	11	Roller Bearing
2	Locking Nut	12	Ball Bearing
3	Mechanical Seal	13	Spacer Ring
4	Tapered Roller Bearing	14	Disc B
5	Drive Pinion	15	Locking Nut
6	Tapered Roller Bearing	16	Rear Cap
7	Spacer	17	Male rotor
8	Rotor Casing	18	Centering Bush

9	Female rotor	19	Multiplyer Casing
10	Discharge Casing	20	Multiplying Pinion

In Figure 4.2 and Table 4.1, the main component elements of the screw compressor are presented. The gear pair can be changed to achieve multiple multiplication ratios between 0.6119 and 2.4839. In order to achieve better performance during operation, axial clearances at the discharge must be between 0.050 mm and 0.070 mm.

4.3 Energy Efficiency of the Test Compressor

Testing a screw compressor at different discharge pressures and speeds is a crucial step in determining its efficiency. In this chapter, we will discuss the testing process of a screw compressor at different pressures and speeds and what factors are considered during the experimentation process.

The first discharge pressure at which the compressor is tested is 7 bar. At this pressure, the compressor's performance is evaluated, and measurements are taken for flow rate, energy consumption, and efficiency. These values are essential in understanding how the compressor operates.

The next discharge pressure at which the compressor is tested is 9 bar. At this pressure, the compressor's performance is evaluated again, and any changes in flow rate, energy consumption, and efficiency are noted. These data are compared with the results obtained at 7 bar, and any discrepancies are analyzed.

The third discharge pressure at which the compressor is tested is 10 bar. At this pressure, the compressor's performance is evaluated again, and any changes in flow rate, energy consumption, and efficiency are recorded. These results are compared with the previous two tests.

Finally, the compressor is tested at 11.5 bar. This is the maximum testing pressure for the compressor, and its performance at this pressure is critical. Any changes in flow rate, energy consumption, and efficiency are noted.

In addition to testing the compressor at different discharge pressures, it is also tested at different speeds. The compressor is tested at 1150, 1680, 2220, 2790, and 3344 rpm. At each speed, the compressor's performance is evaluated, and any changes in flow rate, energy consumption, and efficiency are recorded. These results are analyzed, and any necessary adjustments are identified.

In conclusion, testing a screw compressor at various discharge pressures and speeds is essential to ensure its operation, efficiency, and reliability.

Table 4.2. Data Obtained from Compressor Testing

Male Rotor speed [rpm]	Discharge Pressure [barg]	Measured Volumetric Flow [Nm^3/h]	Calculated Volumetric Efficiency	Calculated Adiabatic Efficiency	Measured Power [kW]	Measured Specific Power [kW/(m^3/min)]
1116	5.98	655.01	0.7787	0.6164	69.80	6.3940
1680	5.98	1056.51	0.8342	0.6380	108.78	6.1775
2219	6.09	1488.82	0.8900	0.7062	140.27	5.6529
2787	5.96	1855.34	0.8833	0.6889	176.46	5.7066
3345	6.04	2189.37	0.8684	0.6593	219.66	6.0198
1110	8.04	648.10	0.7749	0.6316	78.93	7.3071
1674	8.06	1041.22	0.8252	0.6529	122.89	7.0815
2232	8.04	1493.48	0.8878	0.7148	160.73	6.4573
2790	8.04	1838.88	0.8745	0.7006	201.90	6.5877
3358	7.98	2187.79	0.8645	0.6706	249.67	6.8472
1122	9.06	648.63	0.7668	0.6249	89.66	8.2938
1674	8.94	1035.70	0.8209	0.6497	136.56	7.9114
2232	9	1460.54	0.8682	0.7036	178.57	7.3358
2793	9.02	1815.72	0.8625	0.6888	227.07	7.5035
3361	9	2171.10	0.8571	0.6710	278.33	7.6919
1116	10.59	641.19	0.7623	0.6125	99.25	9.2874
1677	10.54	1025.52	0.8113	0.6343	152.85	8.9428
2232	10.46	1450.39	0.8622	0.6932	196.92	8.1462
2793	10.46	1812.20	0.8608	0.6840	249.38	8.2567

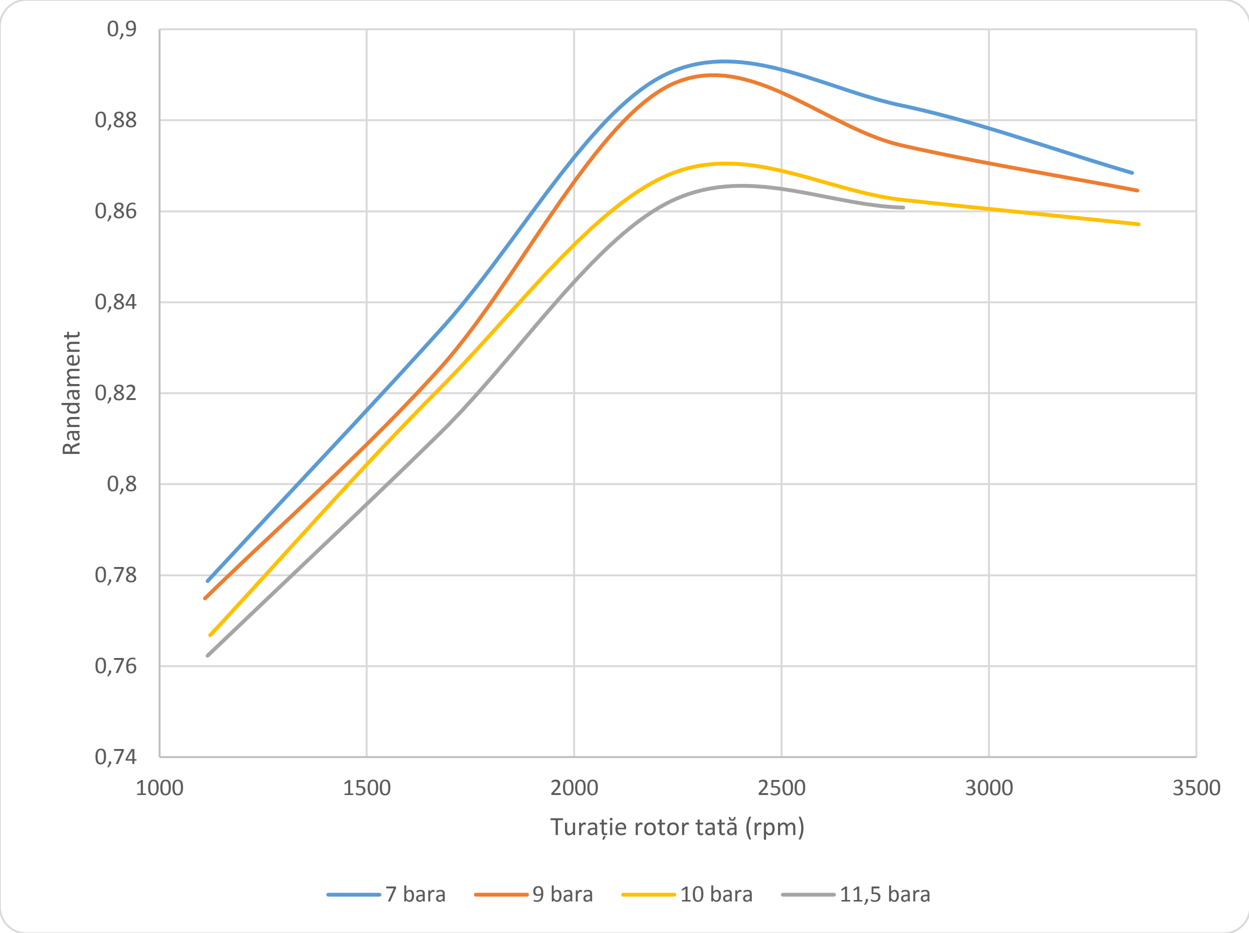


Figure 4.3. Volumetric Efficiency (author's graph)

In Figure 4.3, it can be observed that the volumetric efficiency of the screw compressor starts at around 0.77 at low speed and increases to nearly 0.9 at intermediate speed (between 2200-2500 rpm) before decreasing to around 0.86 at high speed. This behavior is typical for many screw compressors and can be explained by the compression process.

At low speeds, the compressor might not completely fill the compression chamber due to leakage, leading to lower volumetric efficiency. As the speed increases, the compressor can fill the compression chamber more effectively, resulting in higher volumetric efficiency. However, as the speed continues to rise, the compressor will experience higher discharge temperatures and pressure ratios, leading to a reduction in volumetric efficiency.

From the volumetric efficiency graph, an optimal speed value for the screw compressor can be observed, where the volumetric efficiency is at its maximum. In this case, it can be seen that the optimal volumetric efficiency range lies between 2200 and 2500 rpm.

This suggests that, to maximize the volumetric efficiency of the compressor, it would be ideal to choose a speed within this range. If the speed is too low or too high, the volumetric efficiency could significantly decrease.

For instance, if the speed is too low, the compressor might not be able to deliver enough air or gas at the desired pressure, leading to a decrease in efficiency. On the other hand, if the speed is too high, the compressor could consume more energy than necessary to compress the air or gas, which could reduce efficiency.

In conclusion, to achieve the best compromise between performance and energy consumption, it's important to consider the optimal volumetric efficiency range and adjust the compressor speed accordingly.

Additionally, it's noticeable that there are four curves for different discharge pressures of 7, 9, 10, and 11.5 bar. Discharge pressure is an important factor that affects the volumetric efficiency of a screw compressor. Higher discharge pressures can lead to higher compression ratios and gas densities, which can have a negative impact on volumetric efficiency. Therefore, it's common to observe a decrease in volumetric efficiency as discharge pressure increases for screw compressors.

In summary, the volumetric efficiency of a screw compressor generally follows a trend of increasing from low speeds to an intermediate speed before decreasing at high speeds, with the efficiency drop becoming more pronounced at higher discharge pressures.

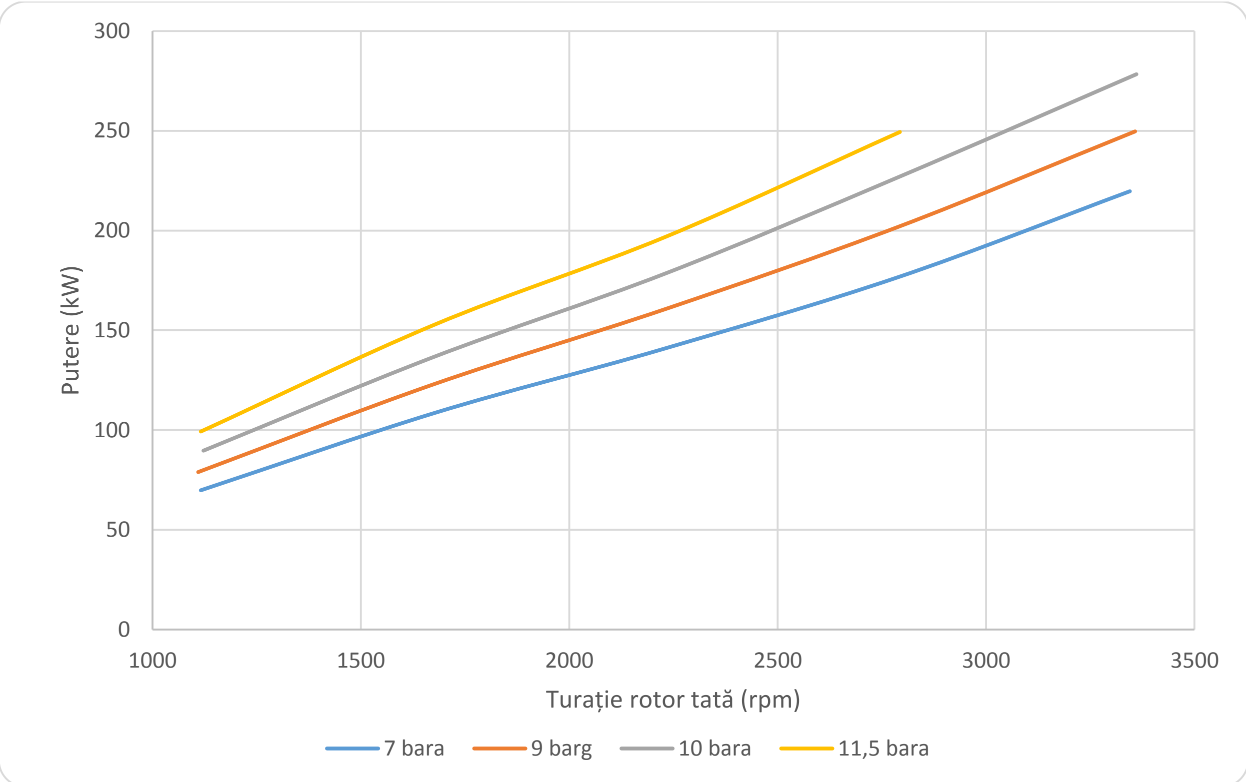


Figure 4.4. Power (author's graph)

The power required by a screw compressor, as shown in Figure 4.4, depends not only on the compressor speed but also on the discharge pressure. As the discharge pressure increases, the power required to compress the gas or air also increases.

Assuming a constant rotational speed for the screw compressor, power curves can be generated for different discharge pressures. In this case, Figure 4.4 represents power curves for a screw compressor operating at discharge pressures of 7, 9, 10, and 11.5 bar.

The curves start at a low power value and then rise steeply. These power curves can be useful in understanding the power requirements of a screw compressor at different discharge pressures and can be used to optimize the compressor's performance for a specific application. For instance, a compressor operating at a higher discharge pressure will require more power, which can increase operating costs. By analyzing the power curves, it's possible to optimize the compressor's performance to reduce operating costs while still meeting application requirements.

Experimentally, it can be observed that the power consumed for compressing gas can be characterized within a pressure range if the power consumed at a certain speed is known, using a formula.

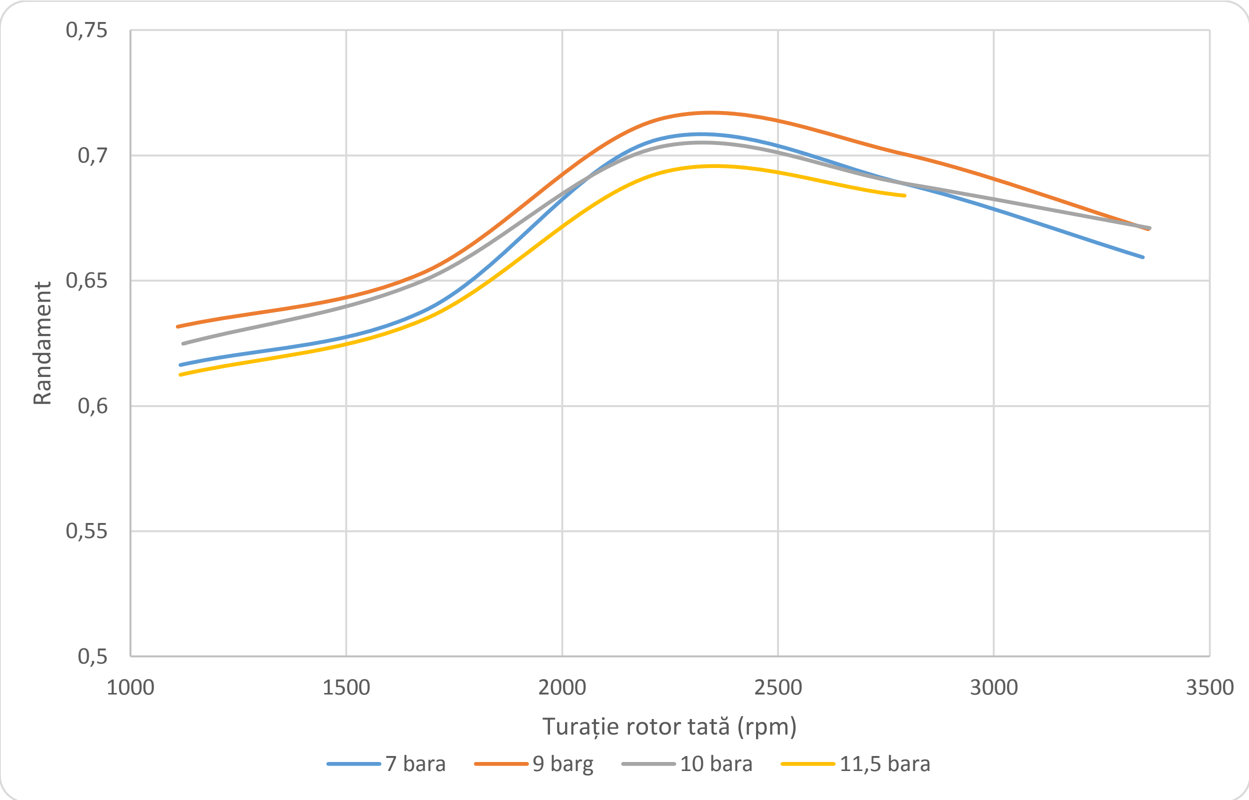


Figure 4.5. Adiabatic Efficiency (author's graph)

The adiabatic efficiency of a screw compressor is a measure of how efficiently the compression of gas or air occurs without losing energy in the form of heat. This efficiency depends on the compressor's design as well as operating conditions, including the discharge pressure. Adiabatic efficiency curves for a screw compressor at different discharge pressures are presented below:

As observed in the curves, the adiabatic efficiency of the screw compressor decreases as the discharge pressure increases. This is because at higher discharge pressures, the compressor needs to compress the gas or air to a higher pressure, which increases the amount of heat generated during the compression process. This results in a decrease in efficiency as some energy is lost in the form of heat.

At discharge pressures of 9 and 10 bar, the adiabatic efficiency of the compressor is relatively high, with values approaching 72%. However, at 7 and 11.5 bar, the efficiency decreases significantly, with values dropping below 70%.

The adiabatic efficiency (Figure 4.5) starts at a lower value between 0.61-0.63 at low speeds, reaches a maximum value in the 2200-2400 rpm range, and then decreases as the speed increases.

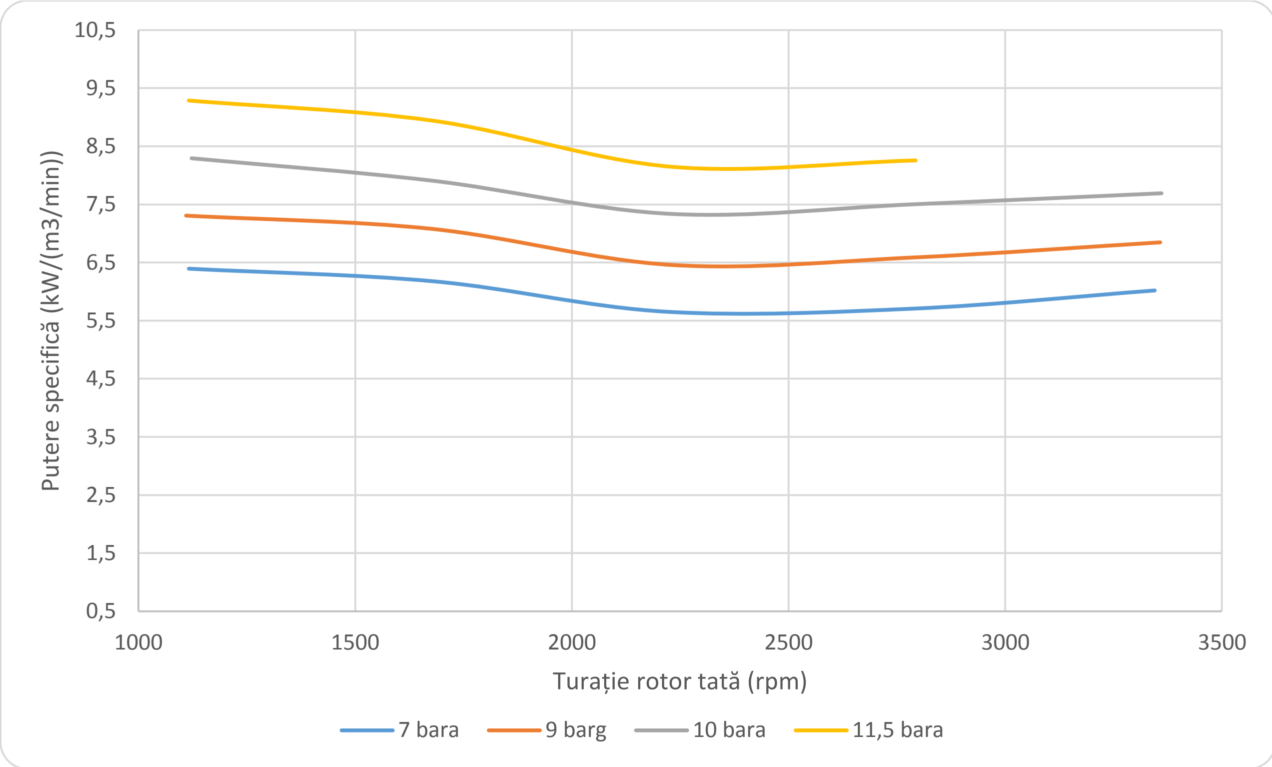


Figure 4.6. Specific Power (author's graph)

Specific power (Figure 4.6) is an important performance measure for screw compressors. It's a measure of the compressor's energy efficiency, indicating how much power is required to compress a certain quantity of gas or air.

Specific power is defined as the power absorbed by the compressor (in kW) divided by the mass flow rate of the compressed gas or air (in kg/s). A lower specific power indicates higher energy efficiency, meaning less energy is needed to compress the same quantity of gas or air.

The significance of specific power for screw compressors can be explained by the fact that compressing gas or air is an energy-intensive process, and the energy costs associated with operating a compressor can be a significant expense for industrial processes relying on compressed air. Therefore, optimizing the energy efficiency of a compressor can lead to significant cost savings over time.

Screw compressors with lower specific power generally have lower operating costs and a smaller environmental footprint, as they require less energy to achieve the same level of compression as compressors with higher specific power. Additionally, compressors with lower specific power can have longer lifespans and require less maintenance due to their more efficient operation.

From the specific power graph, it can be observed that there is an optimal speed value for the screw compressor, where the specific power is minimized for each pressure range. In this case, it can be seen that the optimal range lies between 2200 and 2500 rpm.

CHAPTER 5 Validation of CFD Simulation with Experimental Data

The comparison should encompass:

a. Comparing the corrected consumed power (or specific power consumption or efficiency, depending on warranty conditions) with the theoretical power consumption (or specific power consumption or guaranteed efficiency).

b. Comparing the corrected volumetric flow rate with the theoretical volumetric flow rate at specific pressure rise (or compression ratio).

For the comparison, the following aspects should be taken into account:

Measurement uncertainties according to ISO 1217:1996;

Errors due to imprecise methods used to correlate test results with general operating conditions.

The above-mentioned errors should be combined to determine the total test uncertainty. This, along with the manufacturing tolerances, should be clearly presented in the comparison.

CFD simulation is a widely used technique for predicting the performance of screw compressors. CFD simulations can provide valuable insights into flow patterns, pressure distributions, and energy consumption within the compressor, aiding designers in optimizing system performance and efficiency. However, to ensure the accuracy and reliability of CFD simulations, it's important to compare simulation results with experimental data and strive for a difference of less than 5% between the two.

There are several reasons why comparing CFD simulation with experimental data is crucial. Firstly, it helps validate the accuracy and reliability of the simulation. CFD simulations rely on complex mathematical models to describe the gas or air flow through the compressor, and these models may contain assumptions and simplifications that can impact result accuracy. By comparing the simulation with experimental data, designers can ensure that the simulation provides accurate predictions of compressor performance.

Secondly, comparing CFD simulation with experimental data can help identify areas where the simulation may be inaccurate or incomplete. For instance, if the simulation predicts significantly higher energy consumption or flow rates than experimental data, it may indicate that the simulation does not account for certain physical phenomena or that the assumptions made in the simulation are incorrect. By identifying these discrepancies, designers can refine the simulation to improve its accuracy.

Thirdly, comparing CFD simulation with experimental data can aid in optimizing compressor performance and efficiency. By accurately predicting flow patterns, pressure distribution, and energy consumption within the compressor, designers can identify areas for improvement to increase efficiency and reduce energy consumption. For example, the simulation may reveal that certain areas of the compressor experience significant pressure drops, indicating the need for geometry modifications or operational adjustments to reduce pressure drop and enhance efficiency.

To ensure the accuracy and stability of CFD simulation solutions, it is generally recommended to achieve a difference of less than 5% between simulation and experimental data. This level of precision can be challenging to achieve, as it requires meticulous attention to detail both in simulation setup and experimental configuration. However, through careful calibration of the simulation and ensuring that experimental data are collected under controlled and representative conditions, designers can attain a high level of accuracy in their predictions.

In conclusion, comparing CFD simulation with experimental data is a critical step in designing and optimizing screw compressors. By validating simulation accuracy, identifying areas for improvement, and optimizing compressor performance and efficiency, engineers can ensure that their projects meet desired specifications and operate reliably in real-world applications. By aiming for a difference of less than 5% between simulation and experimental data, designers can achieve a high level of precision in their predictions and ensure that their projects are optimized for maximum performance and efficiency.

To simulate the performance of a screw compressor at two different discharge pressures of 7 and 9 bar and a speed of 3344 rpm, the first step is to define the compressor's geometry and operating conditions. This involves creating a three-dimensional model of the compressor in a CFD software package and defining input and output conditions, as well as the rotation speed.

To obtain accurate results for a screw compressor, it's important to use a high speed for the main rotor. High rotation speed generates significant centrifugal forces that create a high-pressure gradient between the suction and discharge parts of the compressor, which is essential for the compression process.

The table below presents the measured data.

Table 4.3 Measured and Simulated Data

	Measured values		Calculated values		Error (%)	
	7 bara	9 bara	7 bara	9 bara	7 bara	9 bara
Power [kW]	219,66	249,67	212.4381	255.6390	3,4	2,3
Flow rate [Kg/s]	0,7328	0,7322	0,7062	0,7002	3,7	4,6

The conclusion is that the experimental results have confirmed the simulation data with an error of less than 5%. This indicates that the simulation model is accurate in predicting the behavior of the studied system, and the experimental data obtained are generally consistent with these predictions. It's important to note that an error of less than 5% is relatively small and suggests that the simulation model is very close to reality. However, we need to be aware that there's a possibility that this error might increase on a larger scale or under extreme conditions, so we should consider other factors in evaluating the performance of the simulation model. In general, validating experimental data with simulation data is an important step in developing a precise and useful model to understand and predict the behavior of complex systems.

In this chapter of the thesis, tests were conducted on two screw compressors at various speeds and pressures, aiming to evaluate their performance under varied operating conditions. As a result of the tests, performance maps were developed for each compressor, illustrating the relationship between discharge pressure, power, and volumetric airflow at the respective speeds.

Additionally, we calculated the energy efficiency of the compressors using specific formulas and experimental data obtained during the tests. These calculations were performed to determine the ratio between consumed power and developed power of the compressor, providing a measure of their energy performance.

The results obtained in this chapter are highly significant in evaluating the performance of screw compressors and can be utilized for future design improvements. Furthermore, this information is valuable for engineers, students, and researchers working in the field of compressors, seeking a better understanding of their operational characteristics and performance.

The experimental results obtained have validated the experimental model used. By comparing the experimental data with those obtained through computerized simulation, we were able to assess the accuracy of the model and identify any potential discrepancies.

GENERAL CONCLUSIONS

In this thesis, the author conducted a thorough research on the design solutions of screw compressors, investigating the current state of the field and addressing issues related to their performance and efficiency.

Various design solutions of screw compressors were evaluated, analyzing the advantages and disadvantages of each option. Additionally, the materials used in the construction of these compressors were examined, with a focus on their performance and costs. In this context, the need to optimize energy consumption was also considered.

Furthermore, the current state of the art was examined, investigating the latest innovations and trends in the industry. Current models of screw compressors were evaluated, along with their pros and cons. The use of these equipment in various industries, including natural gas, construction, refrigeration, and other domains, was also investigated.

A detailed research into the design solutions of screw compressors was conducted, addressing the issues of performance and efficiency of these equipment and proposing solutions for their improvement. This research can serve as a foundation for the future development of screw compressors and the optimization of their use in various industries.

Two simulations of the screw compressor were performed in the course of the research: one preliminary simulation without oil injection and one with oil injection. From these simulations, it was observed that oil injection significantly increased the mass flow rate at the inlet. This improvement is attributed to the fact that oil injection enhances the sealing between the screw lobes, reducing pressure losses and increasing the airflow.

A multiphase simulation of the screw compressor was also conducted to gain a better understanding of its behavior and optimize its performance. A multiphase simulation involves modeling the interaction between oil, gas, rotors, and housing. Such a simulation is challenging due to the complex interactions involved, and it requires substantial computational power to achieve accurate results.

The simulation was conducted for two different pressure points, 7 bar and 9 bar, to provide a comprehensive insight into the compressor's behavior under different operating conditions. Using the simulation results, diagrams were generated for flow rate, pressure, torque, power, volumetric and adiabatic efficiency. These diagrams offer an overview of the compressor's performance and aid in predicting it under various operating conditions.

By examining these diagrams, the thesis author was able to identify the influence of design parameters on the compressor's performance. For example, a direct relationship was observed between the speed of the electric motor and volumetric flow rate, but an inverse relationship between the motor speed and adiabatic efficiency. From the simulation performed by the doctoral thesis author, it was noted that the operation of the screw compressor has a pulsating nature due to the periodic opening and closing of the volumes.

However, a challenge can arise in measuring this pulsating behavior with existing equipment. Most flow and pressure measurement equipment used in the industry is designed to measure average flow and pressure in steady-state conditions. For screw compressors, the periodic variations are substantial and can be difficult to accurately measure using this equipment.

The doctoral thesis author played a significant role in creating the technological and instrumentation diagram for the operation of the screw compressor. Careful selection of necessary instruments and equipment was required to measure and monitor the operating parameters of the compressor, such as pressure, temperature, flow rate, and power. Furthermore, the author designed and modeled the screw compressor in a 3D format, enabling more precise and detailed simulations of its behavior under different operating conditions.

Designing and modeling the screw compressor in 3D was a complex and labor-intensive task, involving a detailed analysis of all components and their assembly. The author had to consider aspects such as component geometry, dimensions, materials used, tolerances, and interactions between different components. By creating an accurate 3D model of the compressor, the author was able to enhance the design and provide a clearer representation of its components, aiding in a better understanding of its operation and behavior.

Overall, special attention was given to designing and creating all components necessary for the efficient and reliable operation of the screw compressor, ensuring they are made from high-quality materials and designed to offer optimal performance during operation.

The thesis author tested two types of screw compressors, with rotor distances of 90 mm and 180 mm. These tests were conducted across the entire range of speeds and pressures. For the compressor with a rotor distance of 90 mm, the tests were compared with GHH prediction software, and for the compressor with a rotor distance of 180 mm, the tests were compared with simulation results. Compressors are often used in industrial applications that require a wide range of flow rates. To achieve this, most compressors are equipped with an internal speed multiplier. This multiplier allows variation in the rotor speed to achieve the desired flow rate at a fixed speed of the electric motor.

The speed multiplication ratio can vary between 0.61 and 2.48. This wide range of multiplication ratios allows the compressor to be used at different flow rates without the need for a frequency converter, which could be more costly and complex in terms of installation and maintenance.

By using the speed multiplier, the compressor can operate at an optimal speed, thereby improving its energy efficiency and resulting in reduced operating costs. Additionally, this multiplier allows the compressor to be adapted to the specific requirements of the application, based on the need for compressed air flow. The theoretical model was validated for the two simulated points (7 bar and 9 bar) by comparing the simulation results with the experimentally measured values. The simulation results were very close to the measured ones, and the error between them was less than 5%, demonstrating that the theoretical model is highly accurate and can be confidently used to predict the performance of screw compressors under various operating conditions.

This validation of the theoretical model is crucial as it shows that the simulation results align with experimental values, enabling the use of the theoretical model to make predictions about the performance of screw compressors under different operating conditions. In conclusion, this doctoral thesis provided valuable information about screw compressors and demonstrated the utility of mathematical modeling, CFD simulation, and experimental testing in their development. These research findings can be useful in the future for enhancing the performance of screw compressors and reducing energy consumption in various applications.

ORIGINAL CONTRIBUTIONS

The author of this work describes the results of theoretical and experimental research conducted on a screw compressor to enhance energy efficiency [2], [9], [20]. A novel testing method on an experimental stand for this compressor is introduced [7]. This compressor was designed and manufactured by INCDT COMOTI in Romania and is set to be installed in several natural gas compression equipment in operation at PETROM, replacing GHH Germany's CF 180 compressors. This type of compressor is a novelty in the Romanian market, with a wide range of flow rates (between 8 and 43 m3/min) and pressures (between 3 and 16 barg), making it ideal for industrial applications where air or natural gas compression is desired.

The thesis presents technical and scientific contributions in the field of screw compressors:

1. It synthesizes the current state of research and development in the field of screw compressors, providing valuable information for engineers, students, and researchers.

2. A screw compressor was designed, fabricated, and tested. During the compressor's production, various technical challenges arose related to fitting within the imposed dimensions and connections.

3. A testing stand for screw compressors was designed, built, and used to validate functional performance at various operating pressures and speeds. Optimal oil flow rates for operation under these conditions were determined [8], [29]. Operating limits, operation, and protection were also imposed for operation.

4. A mathematical model was created for the pre-sizing of the compressor and performance estimation.

5. Simulations of dynamic two-phase flow within the compressor were performed to achieve improved performance [1], [4], [17], [18]. These simulations were carried out at different discharge pressures.

6. During the design process, special attention was given to the construction of the compressor's housing. Typically, the rotor housing and discharge housing are made from a single piece, but it was observed during the operation of screw compressors that most defects occur around the discharge port due to the entry of solid particles into the compressor [8]. In most cases, changing the housing is necessary. To reduce repair costs, the housing was divided into two parts: the rotor housing and the bearing housing.

7. Characteristic curves of flow-rate vs. speed at constant pressure and power vs. speed at constant pressure were plotted based on experimental results, defining the operating range of the compressor from minimum to maximum speed and pressure.

8. The functional diagram of the compressor was defined, which is used to develop the operating philosophy and control software for the PLC. This software includes protective elements for compressor operation parameters to prevent potential malfunctions or human errors. The testing stand is shut down by depressurizing the separator vessel before stopping the compressor.

PERSPECTIVES FOR FURTHER DEVELOPMENT

Screw compressors are positive displacement compressors that can operate over a wide range of pressures. They have a predefined volumetric ratio, thus an optimal compression ratio depending on the tested gas. If the operating pressure differs from the optimal pressure, phenomena such as under-compression or over-compression can occur. To verify and minimize these phenomena, a CFD calculation should be developed to highlight pressure variation in the compression chamber and calculate mass losses between the compression chambers.

Another development direction involves determining the viability of using a Teflon coating or other materials on the rotors to address defects that arise during operation and to enhance energy efficiency by applying layers of varying thicknesses.

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