

Performance evaluation of a novel dual vane rotary compressor

A Shouman¹, A El Dein Hussin², A Hamed², M Serag El Din³, N Mahmoud⁴ and A El Baz⁵

¹ Emeritus Prof. of Mechanical Eng., New Mexico State University, Las Cruces, New Mexico.

² Lecturer, Faculty of Engineering, Ain Shams University, Cairo, Egypt

³ Research Student, Faculty of Engineering, Ain Shams University, Cairo, Egypt

⁴ Professor, Faculty of Engineering, Ain Shams University, Cairo, Egypt

⁵ Professor, Faculty of Engineering, The British University in Egypt.

Email: ahmed.elbaz@bue.edu.eg

Abstract: Sliding-vane rotary compressors have considerable advantages in comparison with other positive displacement compressors in terms of volumetric efficiency, and continuity of flow without pulsations. However, sliding-vane rotary compressors have relatively low pressure ratio. This paper presents a novel design of a two-stage sliding-vane rotary compressor in attempt to elevate the pressure ratio of that type of compressors. The basic design features of the novel design are illustrated along with the adopted design geometry after studying the effects of several geometrical factors and their practicality. After several trials, the first prototype of the two-stage sliding-vane rotary compressor has been manufactured and tested experimentally to evaluate its performance. The paper reports the preliminary results of the flow rate, pressure rise and power consumption at different rotating speeds. Comparison of the performance of each stage is also included and the benefits of using the second stage are highlighted. The paper also summarizes the design challenges encountered during its manufacturing in addition to the challenges to reduce inter-stage leakage and the applied solutions.

1. Introduction

Air compressors are widely used in industrial processes such as petrochemicals, water treatment or food industry or as a control fluid for instance in pneumatic systems. Compressed air systems are also considered a main consumer of electricity in several applications. Therefore, air compressors are receiving continuous developments in order to improve their efficiencies and reduce overall plant electricity bill. In many applications, rotary compressors are usually privileged because they can match both pressure level and flow rate requirements within a wide range of electrical power from a few to hundreds of kilowatts. They also have advantages over other positive displacement compressors related to continuity of flow without pulsations. Furthermore, they present a favorable weight to power ratio. Moreover, rotary compressors are characterized by low specific energy consumption per unit mass of fluid inducted. The sliding-vane compressor (SVC) belongs to that family. It comprises a cylindrical rotor with longitudinal slots, within which individual sliding vanes are located. The cylindrical rotor is offset within a cylindrical stator, in which it rotates. As the rotor is running, the



sliding vanes are pushed out towards the cylindrical wall of the stator due to the centrifugal force and forming closed volume cells.

As a result of the eccentricity of the rotor axis the volume of each cell gradually decreases and thus air is compressed, see Fig. 1. The SVC is thus a single stage machine which can be used to compress gases to about 12 bar. To obtain higher pressures, two or more compressors are connected in series. Inter-cooling may be used to minimize compression work and approach isothermal compression work. Cipollone *et al.*, [1] presented a mathematical model for two sliding-vane compressors connected in series with intercooler. They reported that a two-stage compressor with inter-cooling might potentially lead to overall electrical savings up to 9.5%.

To the knowledge of the authors, multi-stage SVC such that the compression is performed inside the same compressor is not commercially available yet. Such design concept may reduce the specific energy cost especially when intercooler is utilized between the stages. Furthermore, that concept may lower the weight to power ratio. Valente and Villante [2] proposed a design for a single rotor with two-stage sliding-vane compressor, Fig. 2. Their method aimed to solve crucial difficulties related to performance matching between the two stages such as maintaining same pressure level at the discharge port opening. However, no physical prototype was produced for testing their proposed design.

The first author of the present paper filed a patent in 1999 [3]. The patent is a two-stage dual-vane compressor such that there is an elliptical inner stator and an elliptical outer stator. Eight sliding vanes are mounted in slots within a cylindrical rotor, such that the vanes can rotate by the rotor between the two stators. Meanwhile, each vane has the ability to slide radially through its slot in the rotor, see Fig. 3. The proposed first stage is located in the outer part, while the second stage is associated with the inner space of the device. Both stages are proposed as dual compressor with path A and B. For each path of A and B, an external pipe connects the outlet of the first stage (from the side) to the inlet of the second stage (perpendicular to the page). The current paper explores the design of a two-stage sliding-vane compressor. First, the basic geometry is introduced with geometric parameters involved. This is followed by details of the manufacturing process of the first prototype of this device. The last part presents the test apparatus and measurements obtained to evaluate the performance of the prototype at different operating conditions.

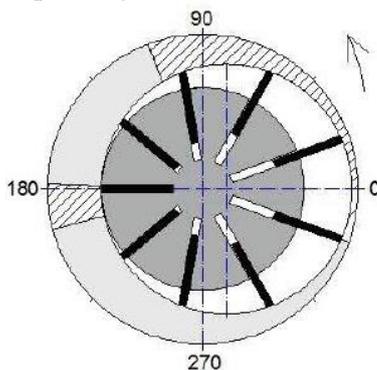


Fig.1: Typical single-stage SVC, [1].

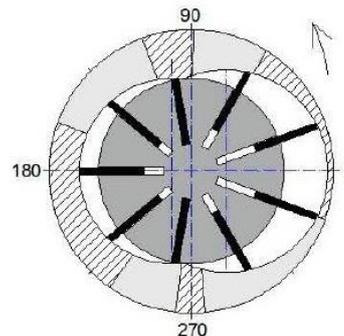


Fig.2: Conceptual design of two-stage SVC, [2].

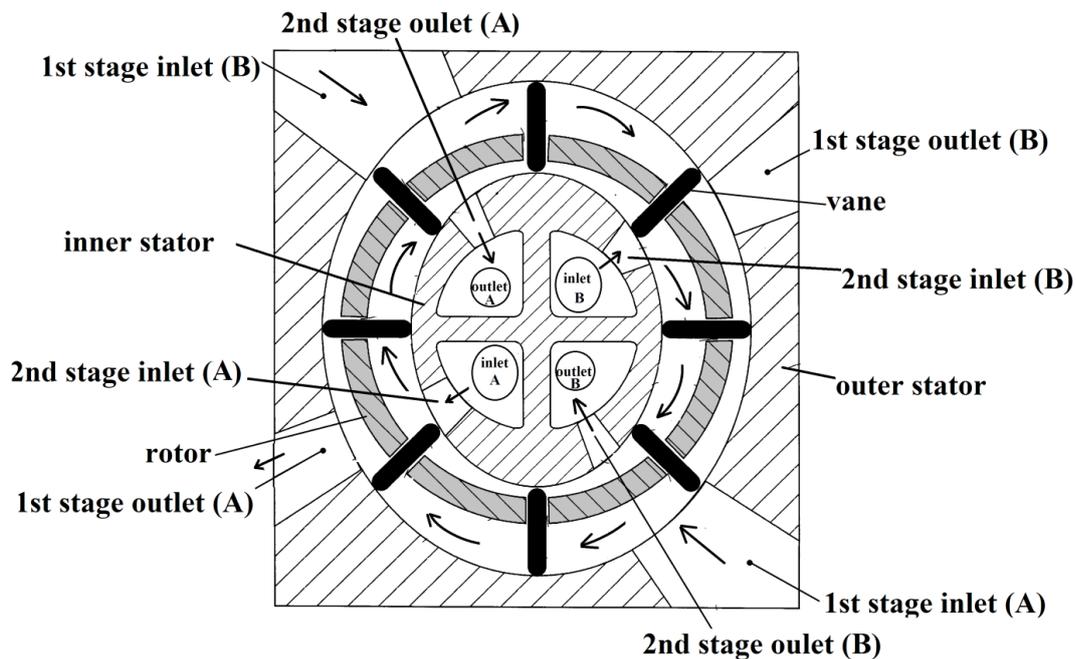


Fig. 3: Patent of a two-stage dual vane compressor, [3].

2. Two-Stage dual vane compressor

The proposed design of the patent was altered to avoid internal leakage, which may occur between the second stage and the first stage via the gap between the vane and its slot, see Fig. 3. This was one of the challenges encountered during the compressor manufacturing. The new design of the two-stage dual vane compressor consists of two concentric circular stators with a circular rotor mounted eccentrically inside, see Fig (4). The circular stators and rotor are easier to manufacture and must provide better contact angle with the rotating vanes. This is an essential requirement to reduce leakage between compressor chambers formed between successive vanes. Separate vanes are employed for each stage. Rotor slots, which house the vanes, are CNC machined along the outer and inner diameter of the rotor. A working cell exists between an adjacent pair of vanes, the rotor, stator surfaces and end plates. The volume of the cell changes as the rotor rotates. The value of the pressure at the end of the compression process is predetermined by suction pressure and the compression ratio. Fig. 5 shows the first prototype of the compressor with its cover removed. Each vane is pushed towards its stator surface to keep them contact using helical springs inserted inside the vane slots. Suction and discharge ports of the first stage are grooved in the outer stator while those for the second stage are grooved in the inner stator. The ports locations are selected such that the maximum cell volume position corresponds to end of suction port closing angle while the minimum volume position corresponds to the discharge port closing. The discharge port opening angle should correspond to the position where the cell pressure reaches a value just above the nominal discharge pressure. Fig. 6 and Fig. 7 show the port positions for the two stages. Discharge from the first stage is transferred to the second stage through an external pipe connected to the inner stator side. No inter-cooling is considered in the current work which only aims at proofing the new design concept. In future stages of the current work, air/water intercooler shall be employed between the two stages. The main dimensions of the prototype developed in the current work are listed in table.1

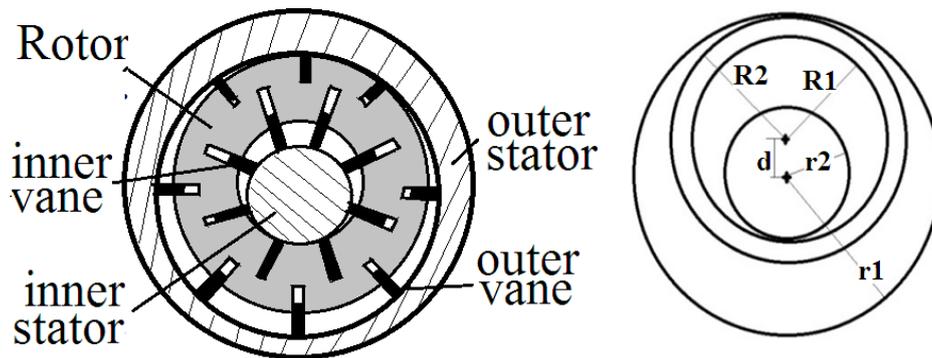


Fig. 4: Basic dimensions of the new design of the two-stage SVC.



Fig. 5: The first prototype of the dual vane, two-stage SVC.

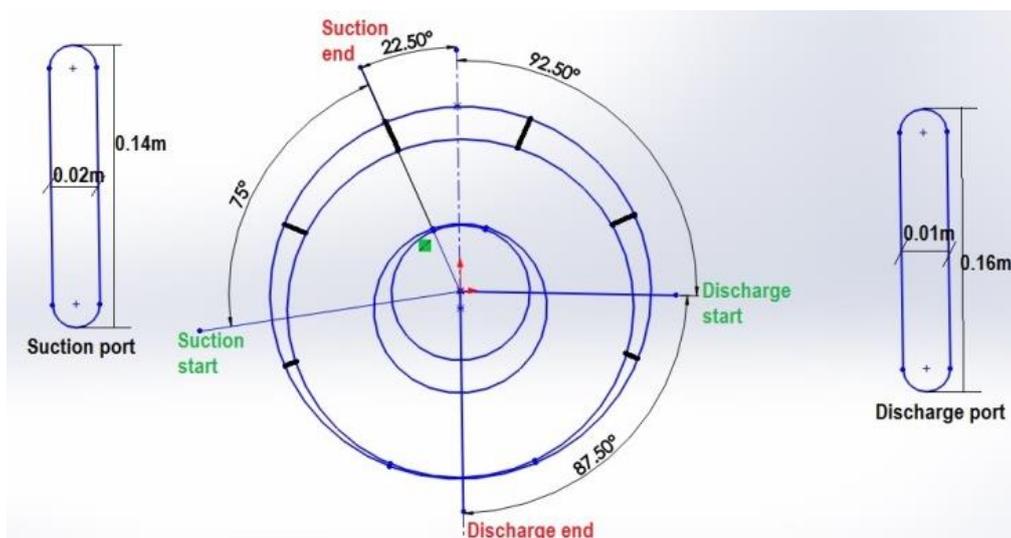


Fig. 6: Locations of suction and discharge ports of the first stage.

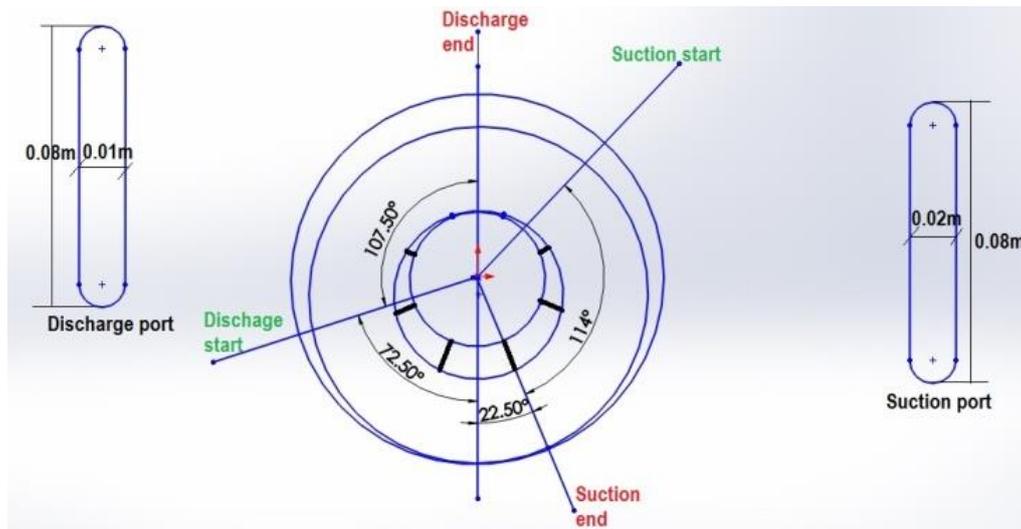


Fig. 7: Locations of suction and discharge ports of the second stage.

Table 1: Dimensions of prototype

	r_1	R_2	L	No of vanes	Eccentricity, d	Vane thickness
First stage	110 mm	100 mm	60 mm	8	10 mm	8 mm
Second stage	50 mm	60 mm	60 mm	8	10 mm	8 mm

The outer stator has an inner radius, r_1 , of 110 mm while the rotor outer radius, R_2 , is 100 mm. The eccentricity between outer stator and rotor, d , is 10 mm. The same eccentricity is maintained between the inner stator and rotor. The thickness of each of the compressor rotor and stator, L , is 60 mm. The values of these parameters were selected after parametric studying of the effects of each factor on the compressor performance and the manufacturing constrains such as the maximum and minimum eccentricity and the length and thickness of the vane to bear stresses and keep contact with its stator.

The theoretical flow rate of the compressor can be calculated using the maximum cell volume matching with the end of suction port. The cell total angle is 45° for 8 vanes. The theoretical discharge is obtained as:

$$Q_{th} = V|_{\theta_{suc}} \cdot z \cdot \frac{S}{60} \quad (1)$$

Where, $V|_{\theta_{suc}}$, is the maximum cell volume, z is the number of cells, and S is the rotational speed in rpm. The volumetric efficiency, η_{vol} , of the compressor is the ratio between the measured flow rate, \dot{Q}_{act} , at suction to the theoretical flow rate, \dot{Q}_{th} , at the corresponding speed of rotation:

$$\eta_{vol} = \frac{\dot{Q}_{act}}{\dot{Q}_{th}} \quad (2)$$

Based on the prototype dimensions and position of ports, the theoretical compression ratio for first stage is 2 while for second stage is 1.5, assuming isentropic compression.

3. Cell Thermodynamics

The compressor cell acts as a variable volume that interacted with adjacent cells and with the rest of the compressor parts through the suction and discharge ports. A one-dimensional model is used to evaluate the compressor geometry, also to calculate volumetric flow rate which will be delivered by

the compressor, cell volume as per rotational angle to predict the discharge angle for each stage and indicated power so a compatible electric motor can be selected correctly.

The mass flow rate, \dot{m}_{in} , can be calculated as:

$$\dot{m}_{in} = \rho_{suc} \cdot \dot{Q}_{th} \quad (3)$$

where, ρ_{suc} , is the air density at suction, and is calculated using the equation of state.

By applying the isentropic compression law, the pressure of discharge cell, p_{dis} , is calculated:

$$p_{dis} = p_{suc} \left(\frac{V|_{\theta_{in}}}{V|_{\theta_{out}}} \right)^k \quad (4)$$

where, k , is the air specific heat ratio and, p_{suc} , is the suction pressure. The discharge pressure can also be calculated as function of rotation angle, [4-6]:

$$\left(\frac{p_{dis}}{p_{suc}} \right)^{1/k} = \frac{2}{1 + \cos \theta - dr \times \sin \theta^2} \quad (5)$$

where, dr , is the relative eccentricity d/r_1 . The cell volume, V_{θ} , can be calculated as function of rotation angle, [6]:

$$V_{\theta} = R \times d \times L \left\{ \beta + 2 \times \sin \frac{\beta}{2} \times \cos \theta + \frac{1}{2} d \times r \times \sin \beta \times \cos 2\theta - \frac{1}{2} d \times r \times \beta \right\} \quad (6)$$

where, $\beta = \frac{2\pi}{z}$

Hence, the indicated power required, P_{ind} , can be obtained as, [5]:

$$P_{ind} = z \omega \int p dV \quad (7)$$

where z is the number of compressor vanes and ω is the rotational speed in rad/s.

3. Design challenges

3.1 Leakage:

Leakage potential in this design is located at two main locations; first, the contact point between end plates and rotors where high compressed air in second stage tends to return back to first stage. The front plate was designed to be in continuous contact with rotor to prevent leakage where friction is also taken into account. Second, the contact point between second stage vane blades and inner stator. The vane blade tip of the second stage was machined with an inclination allowing the vane to be in continuous contact with the outer surface of second stage inner stator and helical springs were installed inside rotor slots between vanes and rotor; by applying these solutions leakage was eliminated between adjacent cells.

3.2 Friction and vane dynamics

Friction occurs at several locations in the two-stage SVC; between shaft and stators, side wall end plates, rotor and blades of the two stages. The friction power consumed in the two-stage compressor is expected to be higher than that in conventional single stage SVC. Many forces act on the vane blade during rotation especially during compression phase. The blade in first stage has sliding motion combined with rotation motion. The rotational motion generates a centrifugal force which forces the vane outside the rotor beside spring force. During compression phase, a pressure difference acts on the blade between adjacent cells generating a pressure force. In the second stage, the same motions and forces mentioned in first stage act on the blades of the second stage except springs force which act in opposite direction where they force blades to slide inward to overcome centrifugal forces in order to keep blades in continuous contact with second stage stator to prevent leakage between adjacent cells. Oil lubrication was used to minimize friction forces between moving parts and fixed parts. Oil was injected to inlet air by gravity and recovered from second stage outlet through an oil separator.

4. Experimental test setup

The prototype of the two-stage dual vane compressor was tested at the fluid machines laboratory of the Faculty of Engineering, Ain Shams University. The compressor was driven by a 5 kW 3-phase induction motor using a V-belt, Fig. 8. The electric motor was driven by a frequency inverter through which rotational speed was changed. The rotational speed was kept below 1000 rpm in the initial testing. Compressor suction is equipped with oil injection system. An oil separator at compressor discharge was used to separate oil and return it to the oil reservoir again, Fig. 9. Air discharge was connected to a storage tank from which air can be consumed at the required pressure and flow rate via a needle valve, through which the compressor was tested at two discharge pressure; 4 bar and 5 bar absolute. At the inlet and outlet of each stage, pressure transducers and temperature sensors, RTD type were used to measure pressures and temperatures, respectively. Air flow rate was measured using sharp edged orifice plate, in compliance with ISO 5167. The orifice plate was installed in the suction line to the compressor. A torque meter was used to measure torque and rotational speed of the compressor. Two sets of experiments were carried out on the compressor. In one set, the single stage was active only while the second stage was not activated by discharging the outlet of the first stage to the tank directly. In the second set of tests both stages were activated.

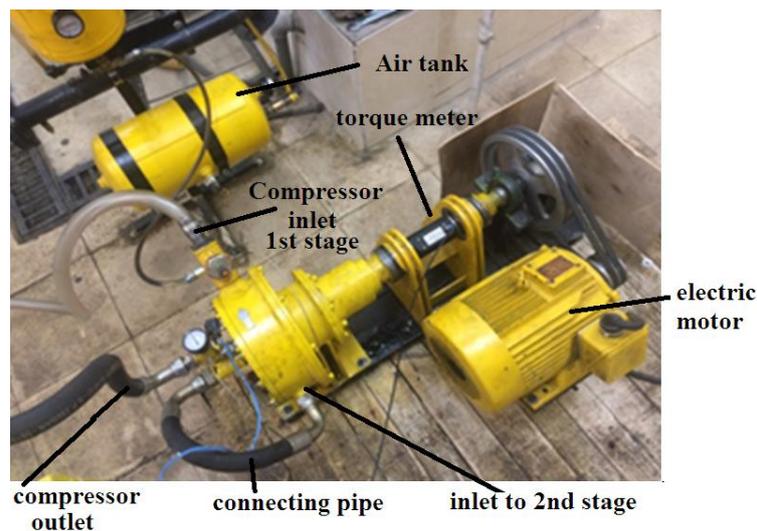


Fig. 8: Photographic illustration of the experiment test rig.

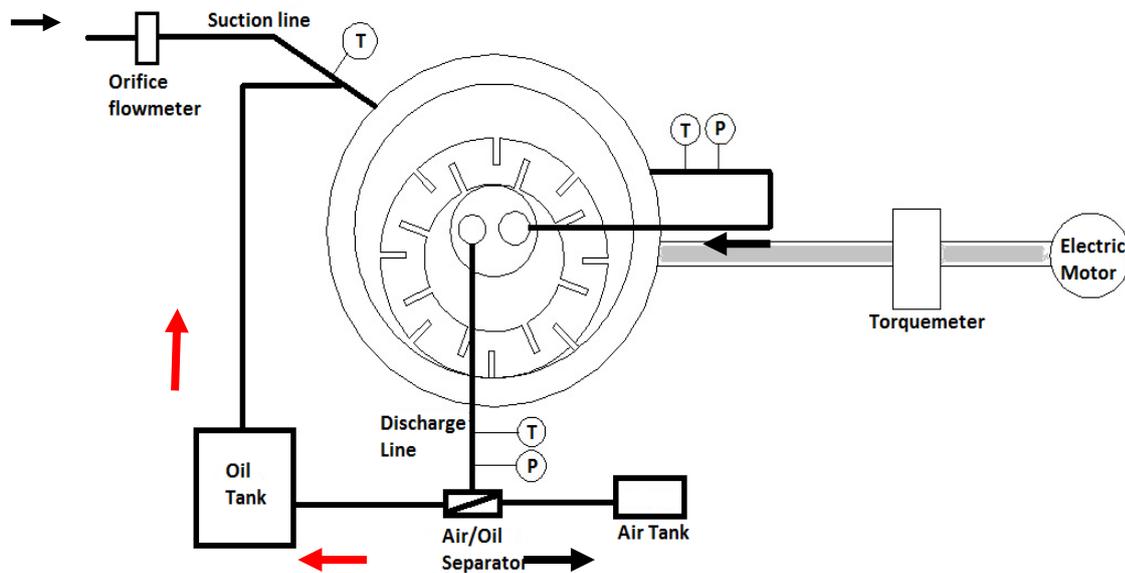


Fig. 9: Schematic of the experiment test rig showing locations of pressure sensor P and temperature sensors T.

5. Results and Discussion

Figure 10, shows the measured variation of inlet air volume flow rate versus rotational speed, at a discharge pressure of 4 bara, in the two cases; single stage active and both stages active. The flow rate increases linearly with rotational speed which is typical for SVC. The actual flow rate is less than the theoretical flow rate at each operating speed, which indicates relatively low volumetric efficiency. The calculated average volumetric efficiency for single stage operation was 76% while being 66% for two stage operation. When the two stages were activated, the flow rate was less than that of the single stage, at the same rotational speed and discharge pressure. The same behavior is shown in Fig. 11, when the discharge pressure was 5 bara. This may be related to a mismatch between the chamber volume at second stage during suction and the flow rate delivered from first stage. Another explanation for the reduced flow rate for two stage operation could be the internal leakage from second stage to first stage.

Figure 12, shows the measured variation of mechanical power input to the compressor with rotational speed for single stage operation and two stage operation and a discharge pressure of 5 bara. The power input increases nearly linearly with rotational speed. The measured power consumption for single stage operation is approximately 8% higher than two stage operation for the same discharge pressure and different rotational speeds. The power consumption at 750 rpm is 5.2 kW for two stage operation while being 5.5 kW for single stage operation. This difference is not significant since similar friction losses are present in both cases. However, a slight drop of power consumption is expected for two stage operation. For single stage operation, higher resisting forces are expected to act on compressor vanes due to the relatively higher size of vanes. The two stage operation, part of the pressure rise occurs in the second stage which has smaller vanes and resisting forces on smaller vanes may contribute to reduced power consumption.

The estimated efficiency, calculated as the ratio between output fluid power and input mechanical power, is approximately 20%. This low efficiency is attributed to high friction forces resulting from friction between rotating parts and end plates and between rotor vanes and stators. The end plates are forced against the rotating parts by springs to minimize leakage in this initial design which causes continuous rubbing between two rows of vanes with these end plates. We have doubts that these contact surfaces have any lubricating oil to reduce frictional forces.

Another reason for the increased compressor power consumption could be attributed to high oil level inside the compressor. The high oil level imposes high torque on compressor vanes as well. Therefore, the observed low efficiency can be improved if detailed design tips of the compressor are improved to minimize these losses. However, the present results can be considered as a good proof of concept of the dual vane, two-stage SVC. In the future upgrade other methods shall be applied to reduce oil level. Also, a sight glass shall be added to monitor oil level.

6. Conclusions

The current research evaluates the performance of a prototype of a two-stage, dual vane SVC with conventional oil injection at suction line and without intercooler between the two stages. Volumetric flow rate showed similar characteristics of two stage, dual vane design and single vane design. The volumetric efficiency is low due to internal leakage which can be reduced if better design of end plate is adopted. Overall compressor friction losses in the two-stage design is contributing to the high mechanical power requirements observed in the experiment and hence low compressor efficiency. Despite the several weak points observed in the first prototype performance, the present results can be considered a good proof of concept of dual vane, two stage sliding vane compressor design.

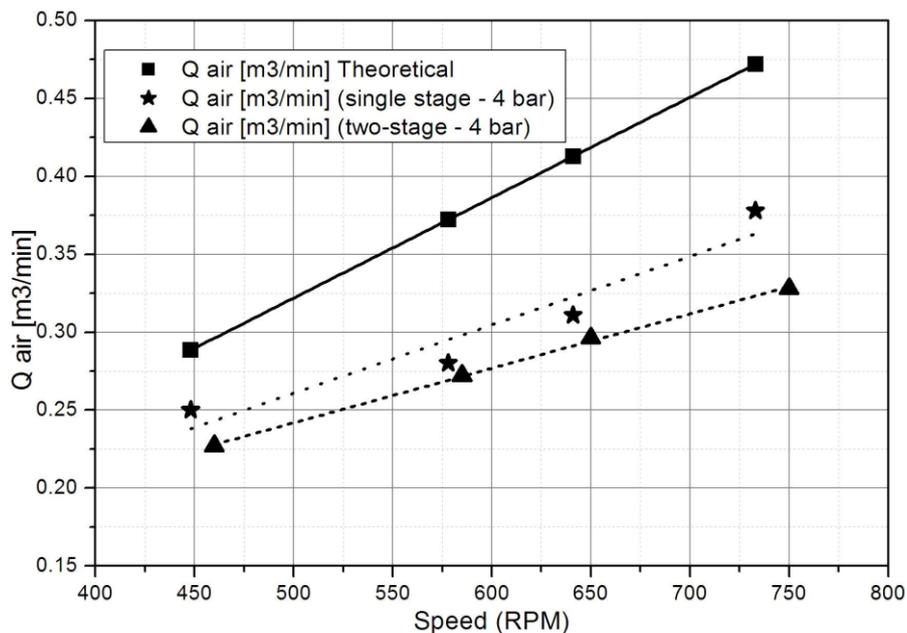


Fig. 10: Measured variation of inlet air flow rate with rpm for a discharge pressure of 4 bara.

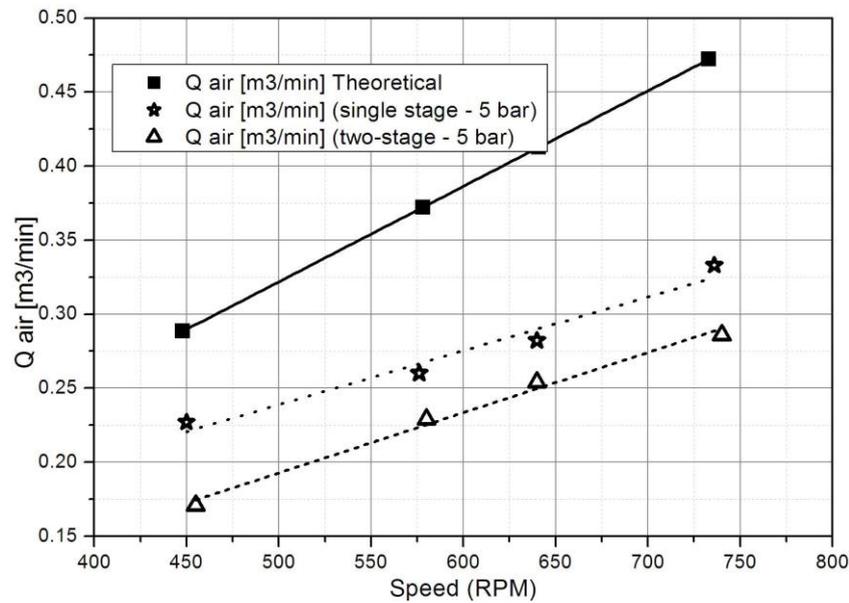


Fig. 11: Measured variation of inlet air flow rate with rpm for a discharge pressure of 5 bara.

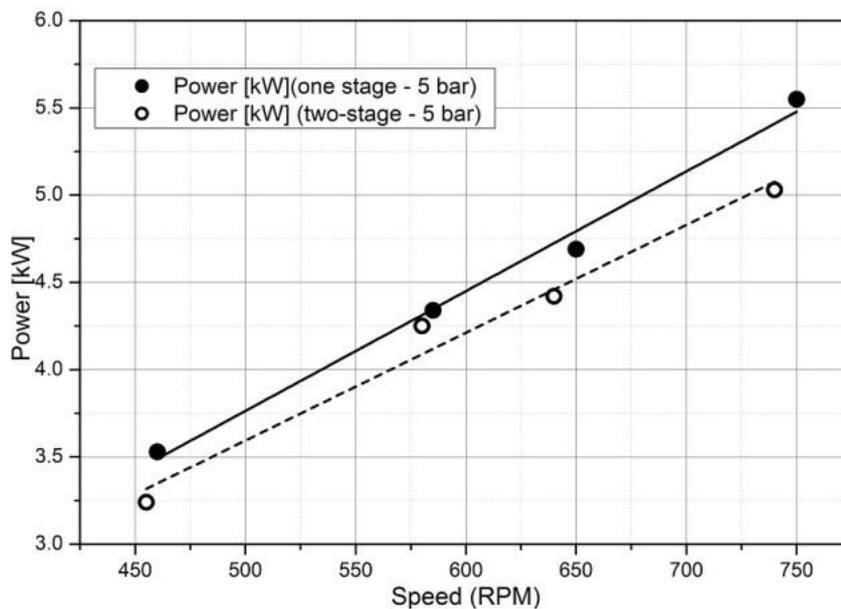


Fig. 12: Comparison between measured variation of mechanical power with rpm for single stage and two stage operation. Discharge pressure = 5 bara.

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Nomenclature

d	Eccentricity (mm)	β	$2\pi/z$
dr	Relative eccentricity= d/r_1	η_{vol}	Volumetric efficiency (%)
L	Compressor radial length (mm)	ρ_{suc}	Air density at suction port (kg/m^3)
m_{in}	Mass flow rate at suction (kg/s)	ω	Rotational speed (rad/s)
p	Pressure (bar)		
P_{ind}	Indicated power (kW)		
Q	Volumetric flow rate (m^3/s)		
r_1	First stage stator inner radius (mm)		
r_2	Second stage outer radius (mm)		
R_1	Rotor inner radius (mm)		
R_2	Rotor outer radius (mm)		
S	Compressor shaft speed (RPM)		
V	Volume (m^3)		
V_θ	Cell volume		
z	Number of vanes		

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