A novel double chamber rotary sleeve air compressor part I: design and thermodynamic model

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Abstract. This paper introduces a novel double chamber rotary sleeve compressor (DCRSC) concept. The compression mechanism is basically that of a rotary motion whereby the novelty lies in the usage of a rotating sleeves and a non-sliding vane that has one end fixed to a rotor and the other fixed end to an outer rotating sleeve. The main goal of this paper is to describe the compression processes of air as the working fluid and reveals the variations of pressure, temperature, mass and compression power inside the compressor working chambers at different rotational speed. Friction and leakage model of the proposed compressor are evaluated in Part II and Part III of this paper series. The compressor was theoretically analysed based on three thermodynamic compression approaches namely: adiabatic, polytropic and isothermal approaches. According to the thermodynamic simulation, a 171 cm$^3$.rev$^{-1}$ DCRSC can deliver a 15.4 m$^3$.h$^{-1}$ of compressed air at 9.3 bar, 6.8 bar and 4.9 bar when air undergoes adiabatic, polytropic and isothermal compression processes, respectively. The DCRSC compression mechanism shows a potential capabilities of a new compressor concept and to be well suited for the air conditioning and compressed air systems articles must contain an abstract.

Keywords: Rotary air compressor; double chamber; design; thermodynamic model; simulation.

Nomenclatures

| Subscript | Description |
|-----------|-------------|
| A | Area, m$^2$ |
| e | Inner sleeve offset distance from rotor center, m |
| k | Index of adiabatic process |
| L | Length, m |
| m | Mass, Kg |
| m | Mass flow rate, Kg. S$^{-1}$ |
| N | Rotational Speed, rev. s$^{-1}$ |
| n | Index of polytropic process |
| P | Pressure, Pa |
| PW | Power, W |
| r | Radius, m |
| Rg | Gas constant, J. Kg$^{-1}$. K$^{-1}$ |
| T | Temperature, K |
| b | End plate bearing |
| c | Compression |
| d | Discharge |
| ic | Inner chamber |
| is | Inner sleeve |
| in | Inner surface of the inner sleeve |
| oc | Outer chamber |
| os | Outer sleeve |
| ou | Outer surface of the inner sleeve |
| r | Rotor |
| s | Suction |
| tc | Both chambers |
1. Introduction

The term rotary describes a class of compressors that operate on the positive-displacement principle and employ rotary motion to transfer energy, that is, to compress gas [1]. Positive-displacement rotary compressors can be classified into four main types: rotary vane compressor, scroll compressor, screw compressor and rolling piston compressor. They are commonly used in the vehicle air-conditioning units. Compared to other rotary type compressors, scroll compressors have displayed a higher performance and a smoother operation, yet categorized as complex structure and difficult to manufacture [1, 2].

Literature of past studies highlighted the advantages of rotary compressor such as its small size, lower vibration level, light weight and its simple structure [3, 4]. In general, rotary compressors are able to perform well in both automotive and residential applications. However, the rotary vane compressor introduced in Chang’s paper still has lower efficiency when compared to the scroll compressor. Friction and internal leakage losses are the critical factors affecting the overall efficiency of rotary compressors. The sliding vane in most of the existing rotary vane compressor usually possesses high friction at high relative velocity which in return limits the compressor’s performance [5].

Due to the low relative pressure ratio of the sliding-vane rotary compressors, a study done by Shouman’s team introduced a novel design of a two-stage sliding-vane rotary compressor in order to elevate the pressure ratio of the compressor [5]. In addition, Gurnule’s team highlighted the importance of the intercooling for air compressor has been highlighted in order to achieve efficient operating processes [6]. The design constraints and the operational conditions was also discussed in [7]. This paper optimized the performance of a rolling piston compressor with taking geometrical configuration, thermodynamic effects, valve dynamics, flow and mechanical considerations into account. A new multi-chamber rotary compressor (MCRC) was designed by [8]. In this paper, mathematical models which include geometrical, thermodynamics, mass flow and discharge valve are formulated to evaluate the performance of MCRC. A geometrical analysis for conical rotary compressor was introduced by the use rapid prototyping modelling [9]. A study used a multi-objective optimization technique for the design of compressors [10]. The model illustrates the effect the coefficient of performance, refrigerating capacity, motor input power, friction power, adiabatic work, discharge valve loss, suction valve loss, overall size of the compressor, and machine cost by utilizing nine objective functions. A comprehensive literature review was conducted for the geometrical parameters of scroll compressor [11]. Another research introduced a model analysis of a novel compressor with a dual chamber for high-efficiency systems [12]. Furthermore, a study developed a new thermodynamic model which can include open system loss due to leakages by predicting leakage losses via the minimum radial clearance in a wide set of operating conditions for the sliding-vane compressor [13]. Therefore, a concept of a non-sliding vane rotary compressor (double chamber rotary sleeve compressor, DCRSC) to minimize the friction and leakage losses was introduced [14].

Thermodynamic model is therefore critical to describe the compression process of working fluid in a compressor operation and reveals the variations of pressure, temperature and mass inside the compressor working chamber. Figure 1 shows the configuration of the DCRSC. To determine the compressor’s capacity, mechanical design and driver size of DCRSC in different heat transfer scenarios, thermodynamic model is predicted based on three thermodynamic compression approximations namely: adiabatic, polytropic and isothermal compression processes.
2. DCRSC configuration and compression mechanism

Figure 1 shows the main components of the proposed double chamber rotary sleeve air compressor. These components are the inner sleeve, the outer sleeve, the rotor (driven by a shaft), the vane and the two end plate bearings at both ends of the compressor. The inner sleeve is assembled in an eccentric position to the rotor, while the outer sleeve is assembled in a concentric position to the rotor. One end of the vane is fixed to the rotor and the other end to the outer sleeve, respectively. While rotating, the vane rotates and pushes the inner sleeve to rotate along in an eccentric path, thus creating the two suction and compression chambers, simultaneously. As shown in Figure 1a, the compressed gas is discharged for an angle of $\phi_{ic,d} = 130˚$ or $(95˚+35˚)$, $\phi_{oc,d} = 116˚$ or $(90˚+26˚)$ for the inner chamber and outer chamber, respectively.

The working cycle for a complete revolution is shown in Figure 1. An electrical motor drives the rotor through a common shaft. During the rotation, the rotor revolves the fixed vane, which in turn pushes the inner sleeve to rotate along its eccentric path and drives the outer centric outer sleeve. The inner sleeve is an active component that continuously defines the volume of the two working compressions chambers.

![Figure 1](image1.png)

**Figure 1.** Main components of the DCRSC, (a) Plane view, (b) Cross-sectional view.
Figure 2. Compression cycle of the DCRSC. The abbreviation means LP = low pressure, MP = medium pressure, HP = high pressure, oc = outer chamber, ic = inner chamber, c = compression and, s = suction.

3. Thermodynamics model of DCRSC

The DCRSC was analysed using the thermodynamic laws where three compression processes for a reversible adiabatic, polytropic and isotropic compression processes are assumed [15, 16]. Thus, the analysis of double chamber air compressor here follows the same approaches. In double chamber air compressor, a compression cell is a space defined by the motions of the inner sleeve, the vane, the outer sleeve and the rotor. The DCRSC were designed to remove heat as much as possible from the thermodynamic system. However, during the theoretical simulation three heat transfer approximations were assumed.

First compression approach is a reversible adiabatic compression process (Constant entropy). This process is applicable if there is no enough time to remove the heat during the compression cycle. it is an ideal cycle for most positive displacement compressors where rotational speed is high (900-1200 rpm) [15, 16]. In this approach, few assumptions have been made as follow: No external or internal heat transfer between the surroundings, inner chamber, outer chamber occur during the compression process, air as an ideal dry gas and heat due to friction was neglected.

Second compression approach is a polytropic compression process. The polytropic approach is a modification of the adiabatic compression approach and closely describe the actual compression process inside the compressor where gas characteristics change during compression with some heat removed from the thermodynamic system [15, 16]. In this approach, few assumptions have been made as follow: Some heat is removed from the compressor by cooling, air as an ideal dry gas and heat due to friction was neglected. For an ideal dry air, the polytropic index (n = 1.2) is assumed to be the average of the adiabatic index (k = 1.4) and the isothermal index (n = 1).

Third compression approach is an isothermal compression process (constant temperature). This process is applicable if there is enough cooling to remove all the heat generated inside the compressor and this usually happen in slow rotational speed compressors (200-600rpm) [15, 16]. In this approach, few assumptions have been made as follow: compressed air temperature is constant due to the continues cooling during compression cycle, air as an ideal dry gas and heat due to friction was neglected.
The geometrical configuration of the double chamber rotary sleeve compressor is shown in Figure 3. The suction pressure of the inner, \(P_{\text{ic,s}}\) and outer, \(P_{\text{oc,s}}\) chambers are assumed constant under steady state condition. The cell volume of the inner and the outer chambers at the end of the suction or at the beginning of the compression are \(V_{\text{ic,s}}\) and \(V_{\text{oc,s}}\), respectively. The cell volume and chamber pressure are expressed as a function of the rotational angle, \(\beta\).

![Figure 3. Geometrical view of the inner and outer chambers of DCRSC.](image)

The area of the inner compression chamber can be formulated as follows,

\[
A_{\text{ic}} = A_{\text{abc}} - A_{\text{od}} - A_{\text{bc}} - A_{\text{vc,ic}}
\]  

(1)

Where,

\[
A_{\text{abc}} = \pi r_{\text{is,in}}^2 \left[ \frac{\alpha}{2\pi} - \frac{\beta}{2\pi} \right] A_{\text{od}} = \pi r_i^2 \left[ \frac{\pi - \beta}{2\pi} \right] A_{\text{bc}} = \frac{1}{2} e r_{\text{is,in}} \sin[\pi - \alpha] A_{\text{vc,ic}} = t_v (g - r_p).
\]

\[
g = e \cos \beta + \left( r_{\text{is,in}}^2 - e^2 \sin^2 \beta \right)^{1/2} \alpha = \pi - \beta - \theta \beta = \sin^{-1} \left( \frac{e}{r_{\text{is,in}}} \sin \beta \right) \beta = [0 \rightarrow 2\pi].
\]

The swept volume of the inner chamber can be defined by,

\[
V_{\text{ic}} = L_c \times A_{\text{ic}}
\]  

(2)

The area of the outer compression chamber is formulated by,

\[
A_{\text{oc}} = A_{\text{os}} - A_{\text{is,ou}} - A_{\text{vc,oc}} - A_{\text{cikh}}
\]  

(3)

Where,

\[
A_{\text{os}} = \pi r_{\text{os,in}}^2 A_{\text{is,ou}} = \pi r_{\text{is,ou}}^2 A_{\text{vc,oc}}(\beta) = t_v (r_{\text{os}} - g).
\]

The outer suction chamber area, \(A_{\text{cikh}}\), can be formulated by,

\[
A_{\text{cikh}} = A_{\text{ikh}} - A_{\text{cik}} - A_{\text{bchc}}
\]  

(4)

Where,

\[
A_{\text{bchc}} = \pi r_{\text{os,in}}^2 \left[ \frac{\alpha}{2\pi} - \frac{\beta}{2\pi} \right] A_{\text{bchc}} = \pi r_{\text{is,in}} \sin[\pi - \alpha].
\]

The volume of the compressed fluid in the outer chamber at any time of rotation is defined by,

\[
V_{\text{oc}} = L_c \times A_{\text{oc}}
\]  

(5)

Thus, the total displaced volume of the compressor is obtained by,

\[
V_{\text{te}} = V_{\text{ic,s}} + V_{\text{oc,s}}
\]  

(6)

For a reversible adiabatic compression process \((k=1.4)\) the air pressure, temperature, work and power can be expressed for each compression chamber following an adiabatic process of \(PV^k=\text{constant}\), giving the expressions,

\[
P_{\text{ic,ad}} = P_{\text{ic,s}} \left( \frac{V_{\text{ic,s}}}{V_{\text{ic}}} \right)^k, \quad P_{\text{oc,ad}} = P_{\text{oc,s}} \left( \frac{V_{\text{oc,s}}}{V_{\text{oc}}} \right)^k
\]  

(7)

\[
T_{\text{ic,ad}} = T_{\text{ic,s}} \left( \frac{P_{\text{ic,ad}}}{P_{\text{ic,s}}} \right)^{\frac{k-1}{k}}, \quad T_{\text{oc,ad}} = T_{\text{oc,s}} \left( \frac{P_{\text{oc,ad}}}{P_{\text{oc,s}}} \right)^{\frac{k-1}{k}}
\]  

(8)

\[
W_{\text{ic,ad}} = \left[ \left( \frac{k}{k-1} \right) \left( m_{\text{ic,s}} R_{g} (T_{\text{ic,d}} - T_{\text{ic,s}}) + m_{\text{oc,s}} R_{g} (T_{\text{oc,d}} - T_{\text{oc,s}}) \right) \right]
\]  

(9)
In case of polytropic compression process (n=1.2):

\[
\eta_{c,ad} = \left( \frac{P_{ic}}{P_{oc}} \right)^{\gamma/\gamma-1} \times \left( \frac{T_{oc}}{T_{ic}} \right)^{n/\gamma-1}
\]

(10)

In case of isothermal process, where compressed fluid temperature remains constant during compression:

\[
P_{ic,iso} = P_{ic} \left( \frac{V_{ic}}{V_{oc}} \right)^{n/\gamma}
\]

(11)

\[
P_{oc,iso} = P_{oc} \left( \frac{V_{oc}}{V_{oc}} \right)^{n/\gamma}
\]

(12)

\[
W_{c,iso} = W_{c} \left( \frac{V_{ic}}{V_{oc}} \right)^{n/\gamma}
\]

(13)

\[
\eta_{c,iso} = \left( \frac{n-1}{n} \ln \left( \frac{P_{ic,d}/P_{ic}}{P_{oc,d}/P_{oc}} \right) \right) \times \left( \frac{n-1}{n} \ln \left( T_{ic,d}/T_{ic} \right) \right)
\]

(14)

The mass and volume flow rates are obtained by:

\[
\hat{m} = \dot{m} \cdot N
\]

(18)

\[
\dot{V} = \dot{V} \cdot N
\]

(19)

From the calculated work in one compression cycle, the theoretical compression power of the DCRSC

\[
P_{cW} = W_{c} \times \frac{N}{60}
\]

(20)

4. Main results and discussion

The mathematical model has been used to theoretically predict the compressor performance with the aid of computing effort. The theoretical analysis can be served as the tool for parametric studies and optimization for a better compressor design. The input temperature has been set at 284 K which is lower than the room temperature to match the input temperature of the experimental results which will be reported in future publications. Table 1 illustrate the various parameters used for the double chamber rotary sleeve air compressor:
Table 1. Mathematical model parameters and dimensions of DCRSC.

| Model specification                  | Values                                    |
|--------------------------------------|-------------------------------------------|
| Inner chamber volume, $V_{ic,s}$     | 67.4 cm$^3$.rev$^{-1}$ or 67.4E-6 m$^3$.rev$^{-1}$ |
| Outer chamber volume, $V_{oc,s}$     | 103.6 cm$^3$.rev$^{-1}$ or 103.6E-6 m$^3$.rev$^{-1}$ |
| Total swept volume, $V_{tc}$         | 171 cm$^3$.rev$^{-1}$ or 171E-6 m$^3$.rev$^{-1}$ |
| Rotational speed, $N$                | 100-1500 rev.min$^{-1}$ or 1.67-25 rev.s$^{-1}$ |
| Working fluid                        | Air                                       |
| Suction pressure, $P_s$              | 1 bar/10$^5$ Pa                           |
| Suction temperature, $T_s$           | 284 K                                    |
| Discharge temperatures, $T_{d,ad}$, $T_{d,poly}$, and $T_{d,isoth}$ | 534.30, 390.87, 284.15 K |
| Discharge angle of the inner chamber, $\phi_{ic,d}$ | 130$^o$                   |
| Discharge angle of the outer chamber, $\phi_{oc,d}$ | 116$^o$                   |
| Discharge pressures, $P_{d,ad}$, $P_{d,poly}$, and $P_{d,isoth}$ | 9.3, 6.8, 4.9 bar or 9.3E5, 6.8E5, 4.9E5 Pa |

DCRSC Main dimensions

| Specification                  | Values                  |
|--------------------------------|-------------------------|
| Rotor radius, $r_r$            | 0.03 m                   |
| Inner sleeve radius, $r_{is}$  | 0.0355 m                |
| Inner sleeve thickness, $t_{is}$ | 0.004 m              |
| Outer sleeve radius, $r_{os}$  | 0.045 m                 |
| Vane radial length, $L_v$      | 0.025 m                 |
| Vane thickness, $t_v$          | 0.005 m                 |
| Chambers axial length, $L_c$   | 0.066 m                 |
| Inner sleeve offset distance from rotor centre, $e$ | 0.0045 m |
| Assembly clearance, $\delta$  | 20E-6 m                 |

4.1 Thermodynamics analysis: Swept volume, air pressure and temperature

The chambers volume and compressed gas thermodynamic characteristics during the compression cycle are of importance to predict thermodynamic and mechanical performance of the DCRSC. The simulation was made based on the compressor dimension stated in Table 1. Figure 4 shows the swept volume of the inner and outer chambers during two complete rotational cycle. The inner chamber has a swept volume of 67.4 cm$^3$.rev$^{-1}$ and the outer chamber has a swept volume of 103.6 cm$^3$.rev$^{-1}$. As calculated by Equation 6, the total swept volume of the proposed DCRSC compressor is 171 cm$^3$.rev$^{-1}$, which is commonly used for bus and train air-conditioning systems. The outer chamber is leading the outer chamber by 180 degrees. The inner chamber maximum cell volume decreases from 67.4 cm$^3$.rev$^{-1}$ to 0 for 180 to 540 degrees of rotational angle. The outer chamber maximum cell volume decreases from 103.6 cm$^3$.rev$^{-1}$ to 0 for 0 to 360 degrees of rotational angle.

Figure 4. Chambers displaced volume of the DCRSC for two rotational cycles.
Figure 5. (a) Chambers air pressure (b) Chambers air temperature, for two complete revolution, (Conditions: $T_S = 284.15$ K, $P_S = 1$ bar).

Figure 5 (a) and (b) show the air pressure and temperature, respectively, for two complete working cycles. The pressure versus volume (P-V) and the temperature versus volume (T-V) diagrams of the air are shown in Figure 6 (a) and (b), respectively. The area under the P-V diagram represent the compression work during one complete cycle. For the same output pressure, the required compression work of the outer chamber is higher than the compression work needed on the inner chamber due to the larger cell volume on the outer chamber. The pressure and temperature at the compressor inlet were assumed to be 1 bar and 284.15 K, respectively.

Figure 6. (a) air pressure versus cell volume, and (b) air temperature versus cell volume for two complete rotational cycle.

From Figure 6 (a), the pressure ratio of each of the inner and outer chambers is 1:9.3, 1:6.8 and 1:4.9 for adiabatic, polytropic and isothermal compression process, respectively. From Figure 6 (b), the discharged air temperature from the inner and outer chambers are 534.30 k and 390.87 k for adiabatic and polytropic compression process, respectively. The air temperature during an isothermal compression process remain constant at 284.15 k. The air start to discharge from the inner-chamber after 230° rotational from the start of the inner-chamber compression, while the air start to discharge from the outer-chamber after 244° rotational from the start of the outer-chamber compression.

4.2 Compression power, mass flow rate and volume flow rate
The variation of rotational speed on compression power at different adiabatic, polytropic and isothermal compression processes are presented in Figure 7. With reference to the Figure 7, the
compressor at 1500 rpm requires a 1.34 KW of hydraulic power to compress the air from 1 bar to 9.3 bar under an adiabatic compression conditions; 0.97 KW of hydraulic power to compress the air from 1 bar to 6.8 bar under the a polytropic compression conditions 0.68 KW of hydraulic power to compress the air from 1 bar to 4.9 bar under an isothermal compression conditions.

![Figure 7](image_url)

**Figure 7.** Total compression power and pressure ratio for adiabatic, polytropic and isothermal compression process at different rotational speed.

The mass and volume flow rates at different rotational speed are shown in Figure 8. With reference to Figure 8 (a) and (b), the compressor delivers a 0.35-5.2 g.s\(^{-1}\) of mass flow rate or a 1-15.4 m\(^3\).h\(^{-1}\) of volume flow rate at a perspective rotational speed of 100-1500 rpm.

![Figure 8](image_url)

**Figure 8.** (a) Mass flow rate and pressure ratio, and (b) Volume flow rate and pressure ratio, at different rotational speed.

### 5. Conclusion

In this paper, three thermodynamic compression processes namely adiabatic, polytropic and isothermal has been evaluated for a novel double chamber rotary sleeve compressor. The proposed compressor uniqueness lies in the concept of having a non-sliding vane that is secured between a rotating concentric rotor and outer sleeve, the secured vane revolves an inner sleeve through an eccentric pathway which creates two compression chambers. The thermodynamic analysis of the DCRSC has been evaluated where air as the working fluid. A general conclusion of the thermodynamic analysis of the DCRSC can be summarized as follow: The geometrical model shows the DCRSC has a total swept volume of 171 cm\(^3\).rev\(^{-1}\). For a reversible adiabatic compression process: DCRSC will require a minimum compression power of 1.34 KW to deliver a 15.4 m\(^3\).h\(^{-1}\) of air at 9.3 bar. For a polytropic compression process: DCRSC will require a minimum compression power of 0.97 KW to deliver a 15.4 m\(^3\).h\(^{-1}\) of air at 6.8 bar. For an isothermal compression process: DCRSC will
require a minimum compression power of 0.68 KW to deliver a 15.4 m$^3$.h$^{-1}$ of air at 4.9 bar. The thermodynamic analysis shows the new compressor concept can be utilized as compressor and might be well suited for air-condition and compressed air systems. Further thermodynamic analysis with different working fluid and operating condition is recommended with considering the fixed pressure ratio for all of the compression process theories. For further performance evaluation, Part II and Part III of these paper series describe the friction and leakage models of the DCRSC, respectively.

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