A Novel Double Chamber Rotary Sleeve Air Compressor - Part III: Leakage Losses Model

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Abstract. This paper presents comprehensive leakage model of a novel double chamber rotary sleeve air compressor (DCRSC). The compressor mechanism is that of a rotary motion whereby the novelty lies in the usage of two rotating sleeves and a secured vane that has one end fixed to an outer sleeve and the other end to a rotor, respectively. The thermodynamic model and friction model of the DCRSC are respectively presented in Part I and Part II of this paper series. The main goal of this study is to formulate and analyse the internal leakage losses at different rotational speed. Dynamic clearance at the contact regions between the high-pressure and low-pressure chambers were made clear. The variations of the internal leakage rate and volumetric efficiency for three compression approaches namely: adiabatic, polytropic and isothermal compression processes were evaluated. At maximum rotational speed of 1500 rpm, the DCRSC volumetric efficiencies are 53.1%, 69.4% and 98% when air undergoes adiabatic, polytropic and isothermal compression process, respectively. The internal leakage losses can effectively be minimized by selecting the right lubricant and reducing the assembly clearances of the rotating parts that defines the compression chambers.

Keywords: sleeve compressor, double-chamber, leakage losses, modeling; simulation.

Nomenclatures

Subscripts
Nomenclatures

| Subscripts | Description |
|------------|-------------|
| e          | Inner-sleeve offset distance from rotor center, m |
| L          | Length, m |
| L̇          | Channel friction length, m |
| m          | Mass, Kg |
| ṁ          | Mass flow rate, Kg. s⁻¹ |
| N          | Rotational Speed, rev.s⁻¹ |
| P          | Pressure, Pa |
| r          | Radius, m |
| R          | Contact force, N |
| t          | Thickness, m |
| V          | Swept volume, m³ |
| v          | Velocity, m. s⁻¹ |
| V̇          | Volume flow rate, m³.s⁻¹ |
| μf         | Coefficient of friction |
| ω          | Angular velocity, rad. s⁻¹ |
| δ          | Dynamic clearance gap, m |
| φ           | Angle of the discharge groove, rad |

1. Introduction

1.1 Leakage losses model of rotary compressors

Leakage losses can be defined as an undesirable mass loss that occurs in almost all positive displacement type compressors due to the internal leakages of the working fluid during operation [1, 2]. Therefore, it has been acknowledged that internal leakage is a normal occurrence in compressors. It has a great effect on the performance of a rotary compressor. In a rolling piston rotary compressor, the major leakage occurs at the clearance exists between the external surface of the rolling piston and the internal surface of the cylinder known as radial clearance [3]. Volumetric efficiency is decreased due to internal leakage as well and thus, the coefficient of performance and cooling capacity of the compressor are affected. As a result, it is deduced that the investigation of the behaviour of the leakage flow through the clearances is worthy.

Several past studies identified various leakage paths in the rolling piston compressor such as [3, 4]. The leakage flow is modelled as the flow of gas and is being driven by pressure difference. It flows through a convergent-divergent nozzle with ideal gas properties considered. The upstream pressure and the constant flow area are approximated as an average of the discharge and suction pressures. A question about the nature of the leakage fluid was raised by [5]. It proposed two sensitive cases for the investigation of leakage flow in the rolling piston compressor with the assumption that the upper bound is entirely compressible refrigerant filling the clearance while the lower bound is presumed to be a mixture of incompressible lubricating oil and the refrigerant. A derivation of the flow is carried through a convergent-divergent nozzle. Then, the leakage flow measurement is observed to be between the upper bound and the lower bound. A claim suggested by study [6] which stated that the most appropriate model to predict realistic leakage flow is the liquid oil-refrigerant mixture leakage flow model. Studies such as [7, 8] conducted an investigation on the leakage losses of a rolling piston compressor with the major focus on the leakage flows through rolling piston end face clearances and the radial clearances by taking dynamic clearances into account. Furthermore, viscous effect of the leakage flow analysis was included in studies [9-11].
Leakage loss is considered as one of the important factors affecting the performance of the rotary compressor. Leakage losses occur between the touching components whereby two different pressure chambers assist the flow of the fluid to travel from the high-pressure cell to the lower one. For instant, studies [3, 4, 7, 8, 12] have discovered that refrigerant internal leakages in the rolling piston type occur at the axial and radial clearances of the compressor. Similarly to the rolling piston compressor, the scroll compressors internal leakages occur at the radial and axial clearances between both scrolls [13]. There is a certain size of leakage losses which could be tolerated for an optimized efficiency. Even though, zero leakage losses are usually impossible to achieve. However, internal leakages can be reduced effectively by introducing a certain design whereby the relative velocities between the components are decreased which concluded to less dynamic clearances during the rotation of the compressor [14, 15]. In the rolling piston compressor, studies [7, 8] have deduced that the size of the end face clearance of the rolling piston significantly influences the amount of fluid leakages during the compression operation and subsequently the volumetric efficiency of the compressor is affected.

In the sliding vane rotary compressor, studies [16-21] have concluded that the losses due to internal leakage happen between the high and low pressure. This process occurs through the radial clearance between vane tip, the stationary cylinder and the axial clearances between the end plate bearings and the vane end faces. With high operating speed, the vane radial clearance was found to be reduced while the leakages located at the vane end faces are not highly affected by the increase of the rotational speed. A two-phase leakage mathematical model for gas-oil flow was presented in [22] to predict the gas leakage rate of a single screw refrigeration compressor.

Studies [2, 7, 8, 11, 23, 24] present set of calculations for the internal leakage flow in rolling piston compressor with the adoption of wall frictional resistance located at the leakage port. The leakages in these studies have been modelled and analysed based on the frictional Fanno flow at the leakage port instead of a nozzle model. However, it is assumed that the fluid is a pure refrigerant and the lubricating oil effects are ignored. Other Studies have made the internal leakage flow under the assumption that it is a single phase pure refrigerant type despite the fact that the fluid is a mix of oil-refrigerant [3, 25]. In the leakage analysis, one isentropic refrigerant was considered when flowing through an ideal nozzle. The functional formula obtained from Computational Fluid Dynamics calculation was developed to calculate the leakage flow through the radial clearance [26]. Furthermore, an estimation of leakage through radial clearance during compression process of a rolling piston rotary compressor was presented in [27]. Another research found that Leakage increases during the compression process and decreases when discharge process starts in a rotary swing compressor [28]. A model has been demonstrated for the leakage of an air synchronal rotary compressor [29]. The leakage flows through the radial clearance which is considered as an isentropic. It then flows through a convergent nozzle while adiabatic frictional (Fanno) flows through the straight channel. A study constructed a mathematical model of working process of detailed leakage path in a water-lubricated twin-screw compressor [30]. In this paper, the effects of each leakage path on performance including volumetric efficiency were calculated and discussed at different rotating speed under the three thermodynamic theories stated in Part I of this paper series.

2. Mathematical leakage losses model of the DCRSC
As shown in Figure 1, The DCRSC has three radial and four axial contact regions where the high pressurized gas leaks from the high to the low-pressure chamber, the two radial contacts are between the inner surface of the inner sleeve and the rotor and between the outer surface of the inner sleeve and the outer sleeve, and one radial contact between the inner sleeve tip and the side wall of the vane, two axial contacts between the vane two ends and the two end plate bearings, two axial contacts between the inner sleeve ends and the sleeve path on the end plate bearings [31].
Yanagisawa and Shimizu [7, 8] have stated that the decrease of volumetric efficiency is small when the axial clearance is below 10 micrometers. Therefore, the two axial clearances between inner sleeve end and the two end plate bearings are expected to be smaller than 10 micrometers due to the inner sleeve pathway acts as sealant with assembly clearance less than 10 micrometer. Therefore, in this analysis, the effects of the leakage flow at these two axial clearances are neglected.

2.1 Leakage at the radial clearances

At the radial clearance, the internal leakage losses are affected by the change of the clearance size during motion [32]. The clearances between the rotating components will dynamically vary due to the variations of force loads during rotation. The amount of leakage flow is sensitive to the cross-sectional area of the leakage channel [24]. It is, therefore, important to study the variation of the radial clearance during rotation.

The dynamic radial clearances between the inner sleeve and rotor, $\delta_{is,r}$, and between the inner sleeve and outer sleeve, $\delta_{is,os}$, are formulated by considering the motion of the eccentric components [24], giving the dynamic clearance ($\mu$m) expressions:

$$\begin{align*}
\delta_{is,r} &= \delta_{0,r} + (r_{is,in} - r_r) - e_{c,r} \\
\delta_{is,os} &= \delta_{0,os} + (r_{os} - r_{is,out}) - e_{c,os}
\end{align*}$$

(1)

As described by [24] the distance between the journal centers of two eccentric components, $e_{c,r}$ and $e_{c,os}$, has the geometrical relation:

$$e_c = \begin{cases} e_{c,r} = \sqrt{\psi_a^2 + \psi_b^2} \\
{e_{c,os} = \sqrt{\psi_c^2 + \psi_d^2}}
\end{cases}$$

(2)

Where

$$\begin{align*}
\psi_a &= \left[ \delta_{b,r} e_r \sin(\Phi_r + \beta) - \delta_{b,is} e_{is,in} \sin(\Phi_{is,in} + \beta) \right], \\
\psi_b &= \left[ \delta_{b,r} e_r \cos(\Phi_r + \beta) - \delta_{b,is} e_{is,in} \cos(\Phi_{is,in} + \beta) - r_{is,in} - r_r \right], \\
\psi_c &= \left[ \delta_{b,os} e_{os} \sin(\Phi_{os} + \beta) - \delta_{b,is} e_{is,out} \sin(\Phi_{is,out} + \beta) \right], \\
\psi_d &= \left[ \delta_{b,os} e_{os} \cos(\Phi_{os} + \beta) - \delta_{b,is} e_{is,out} \cos(\Phi_{is,out} + \beta) + r_{is,out} - r_{os} \right].
\end{align*}$$

In this analysis, the characteristics of the eccentricity position of the inner sleeve to each of the outer sleeve and rotor are calculated based on dynamically loaded finite length bearings [33]. The contact forces acting on the inner sleeve were formulated in Part II of this paper series [34]. The radial
and angular velocities of the eccentric components are respectively given by:

\[
\varepsilon = \begin{bmatrix}
\varepsilon_x = \frac{R_{r, is} (\delta_{b, r} / r_r)^2}{6 \mu_f L_c r_r} M_1^\varepsilon \\
\varepsilon_{is, in} = \frac{R_{r, is} (\delta_{b, is} / r_{is, in})^2}{6 \mu_f L_c r_{is, in}} M_1^\varepsilon \\
\varepsilon_{os} = \frac{R_{os, is} (\delta_{b, os} / r_{os})^2}{6 \mu_f L_c r_{os}} M_2^\varepsilon \\
\varepsilon_{is, ou} = \frac{R_{os, is} (\delta_{b, is} / r_{is, in})^2}{6 \mu_f L_c r_{is, in}} M_2^\varepsilon
\end{bmatrix}
\]

\[
\phi = \begin{bmatrix}
\phi_x = \frac{60}{N} \left( \frac{R_{r, is} (\delta_{b, r} / r_r)^2}{6 \mu_f L_c r_r \varepsilon_r} M_1^\phi + \frac{\omega_{is}}{2} - \beta \right) \\
\phi_{is, in} = \frac{60}{N} \left( \frac{R_{r, is} (\delta_{b, is} / r_{is, in})^2}{6 \mu_f L_c r_{is, in} \varepsilon_{is, in}} M_1^\phi + \frac{\omega_{is}}{2} - \beta \right) \\
\phi_{os} = \frac{60}{N} \left( \frac{R_{os, is} (\delta_{b, os} / r_{os})^2}{6 \mu_f L_c r_{os} \varepsilon_{os}} M_2^\phi + \frac{\omega_{os}}{2} - \beta \right) \\
\phi_{is, ou} = \frac{60}{N} \left( \frac{R_{os, is} (\delta_{b, is} / r_{is, in})^2}{6 \mu_f L_c r_{is, in} \varepsilon_{is, in}} M_2^\phi + \frac{\omega_{is}}{2} - \beta \right)
\end{bmatrix}
\]

Where \( M_{1,2}^\varepsilon \) and \( M_{1,2}^\phi \) are mobility components which normally have variable values starting from 0 to 1 for one rotating cycle. In this study, each of mobility of the components was simply taken as a constant having a value of one.

The channel leakage path across the radial clearances between the inner sleeve and rotor and between the inner and outer sleeves are modelled using the flow channel as shown in Figure 2. In this analysis, the flow channel is assumed to be an ideal straight channel where the fluid will flow from the high compression chamber to the low suction chamber [24].

![Figure 2. Flow channel for leakage path [21]](image)

The channel friction length at the radial contacts of the rotor, \( L_{fr} \), and outer sleeve, \( L_{fo} \), can be calculated from the expression [33]:

\[
\text{friction length} = \frac{L_{fr}}{L_{fo}} = \frac{L_{fr}}{L_{fo}}
\]
The leakage mass flow rate at the radial clearances can be calculated by the relation [8]:

$$L_f(\beta) = \left\{ \begin{array}{l}
L_{f,r}(\beta) = \frac{2\pi \delta_{i,s} r \gamma_f \nu_{f,ri}}{(r_{i,s} \gamma_f - r_f) \left(1 - (r_{i,s} \gamma_f - r_f)^2\right)} \\
L_{f,os}(\beta) = \frac{2\pi \delta_{o,s} r \gamma_f \nu_{f,osi}}{(r_{o,s} \gamma_f - r_{osi}) \left(1 - (r_{o,s} \gamma_f - r_{osi})^2\right)}
\end{array} \right. \quad (5)$$

The leakage mass flow rate at the radial clearances can be calculated by the relation [8]:

$$\hat{m}_{\text{Leakage,radial}} = \hat{m}_{\text{is,r}} + \hat{m}_{\text{is,os}} \quad (6)$$

Where,

$$\hat{m}_{\text{is,r}} = \delta_{i,s} r \nu_{f,i} \left(\frac{P_{e,r}}{R_g T_{e,r}}\right), \quad \hat{m}_{\text{is,os}} = \delta_{i,s} r \nu_{f,os} \left(\frac{P_{e,os}}{R_g T_{e,os}}\right)$$

In which the pressures of $P_{e,r}$, $P_{e,os}$ and the temperatures of $T_{e,r}$, $T_{e,os}$ at the leakage channels, are respectively assumed to be equal to the pressures and temperatures of the refrigerant inside the compression chamber $P_{ic}$, $P_{oc}$, $T_{ic}$, $T_{oc}$. The velocity at the channel exit is given by,

$$v_e = \left\{ \begin{array}{l}
v_{e,r} = \frac{2k}{k-1} \left(\frac{P_{e,s}}{\rho_{e,s}} - \frac{P_{ic}}{\rho_{ic}}\right) \\
v_{e,os} = \frac{2k}{k-1} \left(\frac{P_{e,os}}{\rho_{e,os}} - \frac{P_{oc}}{\rho_{oc}}\right)
\end{array} \right. \quad (7)$$

Where

$$\rho_{ic} = \left(\frac{P_{ic}}{R_g T_{ic}}\right), \quad \rho_{oc} = \left(\frac{P_{oc}}{R_g T_{oc}}\right)$$

Where the fluid is considered compressible and the refrigerant flows adiabatically in the infinitesimal cross-sectional control volume. The inlet of the channel is assumed in a stagnation state where the flow expands at the exit of the channel.

2.2 Leakage at the axial clearances

The internal leakage flow across the clearances between the fixed vane and the end plate bearings can be formulated using the above method and assuming no variation on the axial clearance during rotation. For the vane end face clearances, the friction length is almost equal to vane thickness. The internal leakage mass flow rate at the axial clearances can be calculated from the expression [24]:

$$\hat{m}_{\text{Leakage,axial}} = \hat{m}_{v,b,ic} + \hat{m}_{v,b,oc} \quad (8)$$

Where,

$$\hat{m}_{v,b,ic} = \left[ \delta_{v,b} \nu_{v,b} \left(\frac{P_{ic}}{R_g T_{ic}}\right) \right]_{v,b,ic} + \left[ \delta_{v,b} \nu_{v,b} \left(\frac{P_{ic}}{R_g T_{ic}}\right) \right]_{v,b,ic}$$

$$\hat{m}_{v,b,oc} = \left[ \delta_{v,b} \nu_{v,b,os} \left(\frac{P_{oc}}{R_g T_{oc}}\right) \right]_{v,b,ic} + \left[ \delta_{v,b} \nu_{v,b,os} \left(\frac{P_{oc}}{R_g T_{oc}}\right) \right]_{v,b,ic}$$

The total mass leakage rate and volumetric efficiency can be formulated using the following equations:

$$\hat{m}_{\text{Leakage,ic}} = \hat{m}_{\text{Leakage,radial}} + \hat{m}_{\text{Leakage,axial}} \quad (9)$$
Where
\[ \eta_{voi} = \frac{(\dot{m}_{ic} + \dot{m}_{oc}) - \dot{m}_{leakage,ic}}{(\dot{m}_{ic} + \dot{m}_{oc})} \times 100 \] (10)

3. Results and discussion
The operating specifications and main dimensions of DCRSC being studied are presented in Part I of this paper series. Figure shows the variation of the radial dynamic clearances between the inner sleeve and the rotor, \( \delta_{is,r} \), and between the inner sleeve and the outer sleeve, \( \delta_{is,os} \), at maximum rotational speed of 1500 rpm under adiabatic, polytropic and isothermal compression process. Since the inner sleeve creates two radial clearances for the two inner and outer chambers, the two radial clearances change in a way that the increase of the clearance between the inner sleeve and the rotor will eventually decrease the clearance between the inner sleeve and the outer sleeve. Taking the assembly clearances, \( \delta_0 \) as 20 \( \mu m \), the variation of the pressure and force loads during the rotation will create a dynamic clearance.

From Figure, it is seen that the dynamic clearance between the inner sleeve and the rotor has a range of 12-23 \( \mu m \), while the clearance between the inner sleeve and the outer sleeve will vary within the range of 19-21 \( \mu m \). Variation of the dynamic radial clearance at the inner chambers, at inner-sleeve and rotor radial clearance, has larger dynamic clearance range compared to the radial clearance between the inner sleeve and outer sleeve due to the small cell volume of the inner chamber; also, it is expected that wear will severely occur at the radial surface between the inner sleeve and the rotor. The overall dynamic clearances are not significantly affected by the different compression processes which means the leakage clearances at the flow channels will most likely be the same when air undergoes adiabatic, polytropic and isothermal compression. However, the leakage mass flow rate will mostly be affected on the air pressure, temperature and rotational speed.

**Figure 3.** Dynamic clearances on the internal leakage locations for two complete rotational cycle, conditions:.
and rotor contact.

Figure 4. Effects of rotational speed on (a) the radial clearances between the inner sleeve and the outer sleeve, (b) the radial clearances between the inner sleeve and the rotor for adiabatic compression process.

Error! Reference source not found. shows the variation of the leakage mass flow rate at each dynamic clearances and total leakage rates at the two compression chambers. As shown in Figure 5a, 5c and 5e, the leakage rate at the radial contacts is significantly larger than the leakage rate at the axial contacts due to the larger variation of the dynamic clearance. To the mass leakage at the radial clearances, it is seen that the leakage mass flow rate at the axial clearances (at the vane ends) is small and has less significant effects on the volumetric efficiency. With reference to Figure 5b, 5d and 5f, the adiabatic compression process will have larger leakage rate due to the high pressure and temperature when compared to polytropic and isothermal compression process. Despite the different compression approach, the maximum leakage rate occurs at the end of compression stage where the gas pressure is at its peak for each of the inner and outer chamber.
Figure 5. Leakage mass flow rate (a) at the radial clearances for adiabatic compression, (b) at the compressor chambers for adiabatic compression, (c) at the radial clearances for polytropic compression, (d) at the compressor chambers for polytropic compression, (e) at the radial clearances for isothermal compression, (f) at the compressor chambers for isothermal compression.

Figure 6 shows the total internal leakage mass flow rate of the DCRSC at different rotational speed for adiabatic, polytropic and isothermal compression process. As shown in Figure 6, the current design of the DCRSC shows the increase of the rotational speed has small effects on the gas leakage rate for the same compression process. Therefore, the type of compression process or gas discharge pressure and temperature will have large impact on the leakage rate more than the rotational speed of the DCRSC. For the isothermal compression process, the leakage rate is too small since the air temperature is constant and pressure ratio is smaller when compared to adiabatic [31].
Figure 6. Rotational speed effects on the total leakage mass flow rate of the DCRSC for adiabatic, polytropic and isothermal compression processes.

Figure 7 shows the volumetric efficiencies of the DCRSC at a range of 100-1500 rpm for adiabatic, polytropic and isothermal compression process. From Figure 7, the DCRSC has a volumetric efficiency of 53.1%, 69.4% and 98% at maximum rotational speed of 1500 rpm when air is compressed under adiabatic, polytropic and isothermal process. Compare to polytropic and isothermal compression, the adiabatic compression process will lead to lower volumetric efficiency and that is mainly due to the high-pressure ratio and high air temperature which increases the internal leakage at Dynamic clearance or leakage channels. Figure 7 indicates the increase of rotational speed will increase the volumetric efficiency. Therefore, it can be concluded that the type of compression process (pressure and temperature) will have large impact on the compressor volumetric efficiency.

Figure 7. Volumetric efficiency of the DCRSC at different rotational speed for adiabatic, polytropic and isothermal compression processes.

4. Conclusion
In this paper, DCRSC leakage losses model were formulated and analyzed. The thermodynamic and friction loss model of the DCRSC are presented in Part I and Part II of this paper series. The effects of dynamic clearances at the radial and axial contacts were taken into consideration on the leakage model. The leakage losses of the DCRSC compressor largely occurs at the radial contact regions of each of the inner and outer compression chamber. The leakage model was evaluated under three
compression conditional theories namely, adiabatic, polytropic and isothermal compression process. The DCRSC volumetric efficiencies at maximum rotational speed of 1500 rpm are 53.1%, 69.4% and 98% when air undergoes adiabatic, polytropic and isothermal compression process, respectively. As compared to polytropic and isothermal process, the adiabatic compression process has large leakage losses, thus, low volumetric efficiency. The reduction of clearance at the contact region reduces the leakage mass flow rate and increase the friction. Therefore, optimization study of the assembly clearance, material selection and lubrication is highly recommended to maximize the mechanical and volumetric efficiency.

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