Modelling of leakages in rotary twin-screw compressor

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Abstract. The adiabatic and volumetric efficiencies are key performance parameters of any positive displacement machines including twin screw compressors. These efficiencies are affected by clearances and leakages in these machines. The leakage happen because of clearance gap and pressure difference between two pressurized chambers the compression module. In this work, findings from all the numerical and mathematical models are presented for different leakages in the twin-screw compressor. Authors proposed an iterative method to estimate the leakages by taking in account the geometry of the clearance, friction and local resistances. An experimental set up to collaborate the leakage model is presented, which can be used for further investigation of these leakages.

Key words: screw compressor, leakages, clearances, experimental, mathematical model

1. Introduction
Compressed air is used widely in many industries and consumes nearly 10-30\% of the total electricity consumed in the facility and hence the performance improvement of air compressor is important for industries. In comparison to reciprocating compressor, rotary screw compressors are more popular in refrigeration, gas processing and energy industries where applications can vary significantly in terms of flow-rates, pressure, temperature and working fluids. As shown in Figure 1, because of lesser moving components, the rotary screw compressors are simple and compact in design with better reliability [1].

![Figure 1. Screw compressor moving elements: male and female rotors](image)

Due to advancements in the manufacturing processes, it has become possible to maintain close tolerances inside the compression module which results in small internal leakages thereby leading to an
improvement in the performance of the screw compressor [1]. These internal leakages can be classified as the leakages through inter-lobe clearance, the leakages through rotor tip and housing clearance, the leakages through blowhole area and the leakages through end plate clearances. Figure 2 provides an overview of the various leakages in a twin-screw compressor.

![Diagram of leakages in a twin-screw compressor](image)

**Figure 2. Type of leakages in a twin-screw compressor**

### 2. Analytical and numerical models on leakages in screw compressor

The twin-screw compressor performance parameters depend highly on the leakages which in turn depend on the clearances in the compression module. Many researchers focused on the energy efficiency of the twin-screw compressor and studied the effect of these clearances on the performance of compressor [2]. In the past, numerous mathematical models were developed and numerical studies conducted using CFD and simulation tools. Table 1 summarises all the critical findings from developments of the related mathematical models.

| Name of investigator | Leakage Type | Approach | Focus and findings |
|----------------------|--------------|----------|--------------------|
| Fan et al. [3]       | Rotor tip-housing and inter-lobe leakages | Mathematical model to calculate the leakage rates | The mathematical model developed with consideration of both the viscous and inertia forces and compared. The results from the presented model compared with the results from the nozzle model, the viscous model and the experiments (for different gases) at various pressure differences for different clearance. The presented model found in close agreement with the experimental results in comparison to other two models. |
| Prins and Ferreira [4][5] | Rotor tip-housing and inter-lobe leakages | Four different 1D steady state models' comparison and new model development. | Four mathematical models compared with two experimental results available in literatures and based on the analysis new optimized model presented using optimization algorithm and wall shear stress considerations. |
Zaytsev and Infante Ferreira [6] Rotor tip-housing leakages A one-dimensional leakage flow model for two-phase ammonia-water twin screw compressor. The leakage results using one-dimensional (viscous and inviscid) model and with an isentropic nozzle were compared. The convergent nozzle model found predicting the leakages (at upstream pressure of 16bar) 1.6-2 times in comparison of viscous flow model. No much difference found between the results from both the model for the higher downstream pressure.

Fong at el. [7] Interlobe leakages Mathematical Model The model was proposed to calculate the inter-lobe clearances between two rotors. The clearance field was presented by iso-clearance contour diagram (ICCD). This method found good in terms of avoiding the problem of discontinuity and divergence in the optimal programming because of the use of single mathematical model.

Fryc and Vimmr [8][9] Rotor tip-housing leakages Numerical simulation of rotor tip housing leakages using 2D model of a clearance gap. Three different models of turbulence (Baldwin-Lomex, Spalart-Allmaras and k-ε) of different complexity used to find out the leakage flows between two chambers with different pressures. The leakage mass flow rates predicted using described three methods and the results found in the range of 0.034 kg/s to 0.038 kg/s. The leakage rates found consistent for all the three models with fine grids. The k-ε (as dependent on the grid density) needs more computational efforts to get the comparable results obtained by other two methods.

Xiong [10] Interlobe leakages The presented model obtained full interlobe clearance between the two rotor surfaces with high accuracy. The clearance jump exist along the sealing (contact) line and the interlobe clearance distribution along the contact line is quite different in comparison to various clearances generating methods.

Wang et al.[11] Rotor tip-housing leakages A separated flow model to calculate film thickness in rotor tip clearance (in water-lubricated screw compressor). The rotor tip clearance outflow leakages found zero initially because of low pressure difference between the local and subsequent low pressure cavity. Later small outflow leakages found because of higher RPM (which result in the thinner liquid film because of higher pressure difference between two cavities under the consideration). This small variation (in the outflow) is because of reduced sealing line length (from 500° to 675°).

Table 2 summarises the key findings from the work carried out by researchers using numerical/CFD simulations. The learnings were used to develop commercially available simulation tools which predict various leakages in the twin-screw compressors, however the accuracy of these predictions depend on the quality of meshing the clearances.

| Name of investigator | Approach | Focus of the work | Findings |
|----------------------|----------|------------------|----------|
| Papes et al.[12]     | CFD calculations used for a screw expander leakages | The performance and leakages in the screw expander studied for different pressure ratios and for two different design using CFD. | The internal flow leakages increase with an increase in the pressure ratio. Authors focused on tip and leakage flows for different pressure ratios and two different designs (with and without additional injection ports). The inflow through clearances found increasing because of |
the higher pressure of the preceding chamber during the expansion, while the outflow through the clearances found increasing upon additional working fluid injected.

In variable pitch rotors, the sealing line found longer at the suction end and decreasing towards the discharge end of the rotor. In variable pitch rotors, the sealing line length found 30 mm longer at the suction and 12 mm shorter at the discharge end and total sealing line length found reduced by 11 mm (compared to uniform pitch rotors). In variable profile, the sealing line length found 12 mm longer at the suction with very small difference at the discharge (no overall total gain).

Upon comparing the normalized indicated power, the difference between the CFD and experimental results was observed to be +1.4% at 6000 r/min (-2.7% with base grid) and -2.8% at 8000 r/min (-6.6% with base grid) considering the finest grid for CFD analysis. Similarly, for the normalized flow rate, the difference between the CFD and experimental results was found to be in the range of -4.0% to -5.5% at 6000 r/min (-11.0% with base grid) and -2.9% at 8000 r/min (-8.7% with base grid).

The flow coefficient presented as a function of pressure ratio for different height-diameter ratio (for Re = 20,000 and 700) with stationary boundaries and for different Mach numbers (for Re= 20,000 and h/D = 0.0028) with moving boundaries. The flow coefficient also presented as a function of Reynolds number for different height-diameter ratio for both stationary and moving boundaries.

The largest reduction (from nominal clearance of 180 μm to 80 μm) in the rotor tip housing clearance (because of the deformation due to the temperature) recorded between the male rotor and the high pressure cusp on the discharge side.

### 3. Leakage estimation using analytical models

The twin-screw compressor performance parameters highly depend on the leakages which in turn depends on the clearances in the compression module. Many researchers focused on the energy efficiency of the screw compressor and worked to see the effect of these clearances on the performance of twins-screw compressors.[2]. Numerous mathematical models developed and numerical studies carried out using CFD and simulation tools to predict these leakages. In their recent work, Utri et al. [15],[17] used dimensionless numbers to present fluid flow through rotor tip housing clearances and front end clearances using dimensionless numbers with recommendations to use two different simulations methods to predict the operational behaviour in positive displacement machines, CFD and chamber model. The CFD needs time consuming computations, while the chamber model is more suitable when
many configurations need to be simulated in comparison. The chamber model consider simple isentropic nozzle equations to estimate mass and energy exchange between the chambers under consideration [18].

\[
\dot{m} = \frac{A p_1}{\sqrt{T_1}} \left[ \frac{2 + k \ast \left( R - (r)^{k+1} \right)}{k \ast (k-1)} \right] \quad \text{for } r > \left( \frac{2}{k+1} \right)^{\frac{k}{k-1}}
\]  

(1)

\[
\dot{m} = \frac{A p_1}{\sqrt{T_1}} \left[ \frac{2 + k \ast \left( R - (r)^{k+1} \right)}{k \ast (k-1)} \right] \ast \left( \frac{2}{k+1} \right)^{\frac{k}{k-1}} \quad \text{for } r < \left( \frac{2}{k+1} \right)^{\frac{k}{k-1}}
\]  

(2)

The flow coefficient can be defined as a ratio of experimental leakage flows to theoretical leakage flows,

\[
C_v = \frac{\dot{m}_{\text{exp}}}{\dot{m}_{\text{th}}}
\]  

(3)

These flow coefficients can be presented as a function of dimensionless numbers pressure ratio, Reynolds number, Mach numbers etc.

Zakharenko [19] and Sakun [20] developed mathematical model taking into consideration various factors such as the shape and geometrical dimensions of the gap, frictional forces in the gap, before and after flow parameters and local inlet and outlet pressure losses. Kotlov et al.[21][22][23] used the same mathematical model to find out the gas leakages from high-pressure cavities to the low-pressure cavities in their recent study related to vane compressor, dry claw compressor and the screw machines. The leakage flow rates through the gap at each fixed angle were calculated by Kotlov et al. using the following equation:

\[
\dot{m} = C \ast A \ast \frac{\rho_2 \ast p_2 \ast (\varepsilon^2 - 1)}{\sqrt{\ln \varepsilon^2 + \xi + (\lambda \ast \Sigma)}}
\]  

(4)

where conventional density \( \rho_2 = \frac{p_2}{R \ast T_1} \) and \( \varepsilon = \frac{p_1}{p_2} \)

As the above equation (Equation 4) cannot be solved directly because the friction factor \( \lambda \) depends on the Reynolds number (which in turns requires the mass flow rate to be known).

\[
R_{\text{eq}} = \frac{4m_{\text{th}}}{\mu \ast \dot{v}}
\]  

(5)

Hence in order to solve Equation 4, the method of successive approximation can be used. The initial approximation of the mass flow rate (\( \dot{m}_1 \)) at the critical discharge area can be determined by using Equations 1 and 2.

Further this value of \( \dot{m}_1 \) can be used iteratively to find out the new values of Reynolds number, friction factor etc. until the following condition is fulfilled.

\[
\frac{[\dot{m}_{i+1} - \dot{m}_i]}{\dot{m}_i} \leq \Delta
\]  

(6)

where \( \Delta \) is the pre-set accuracy of the calculation.

This iterative method can be used to estimate the leakages through cross sections of any shape like circular, triangular, rectangular, etc. in the screw compressors and other positive displacement machines.
4. Experimental study

Though many mathematical model developments and simulation works carried out in the area of leakage predictions, experimental investigation of these leakages is still an open area of research. An experimental set up (Figure 3) which can be used to simulate various leakages in any positive displacement machines includes a compressor (to get the compressed air), CAV nozzle set up, receiver and control valves for the regulations and a DAQ system with instrumentation to capture the operating parameters required to calculate the flow.

![Experimental Test Rig](image)

Figure 3. Photograph of the experimental test rig used for the study

This test set up can be used to simulate the leakages of a twin-screw compressor and the results from the experimental work can be compared with analytical results to derive flow. These flow coefficients can be used along with basic equations for better prediction of the leakages irrespective of geometry and pressure conditions.

5. Conclusion

The phenomenon of leakages through various clearance gaps in the screw compressor and its accurate prediction impact the performance of compressors. The mathematical and simulation models can be used to predict the leakages and the effect on the performance of screw compressor. The mathematical model presented in present study takes into account, the shape of a gap and its geometric dimensions, the parameters before and after the gap, the friction and the local resistance at entry and exit. An iterative model is suggested to calculate the leakage rate through any cross section of different geometries.

Further experimental investigation using presented test rig can be used to collaborate and improve the leakage prediction models in screw compressors.

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Nomenclature

- $m$: Leakage mass flow rate (kg/sec)
- $C$: Coefficient of discharge (dimensionless)
\( A \): Clearance leakage area (\( m^2 \))  
\( T_1 \): The upstream temperature (K)  
\( p_1 \): The upstream pressure (Pa)  
\( p_2 \): The downstream pressure (Pa)  
\( R \): Gas constant of oil gas mixture (J/kg-K)  
\( k \): Specific heat ratio (dimensionless)  
\( r \): Ratio of pressure, \( p_2 \) (Downstream) to \( p_1 \) (Upstream)  
\( C_v \): Flow coefficient  
\( \rho_2 \): Conventional density (kg/m\(^3\))  
\( \varepsilon \): Ratio of pressure, \( p_1 \) (Upstream) to \( p_2 \) (Downstream)  
\( \xi \): Total coefficient of local resistances at the entrance and exit of the gap (Dimensionless)  
\( \lambda \): Friction factor (Dimensionless)  
\( \Sigma \): Form factor (Dimensionless)  
\( R_e \): Reynolds Number (Dimensionless)  
\( P \): Perimeter (m)  
\( \Delta \): Accuracy

**Subscript**

0: Initial  
1: Upstream  
2: Downstream  
exp: Experimental  
anly: Analytical

**Abbreviations**

CFD: Computational Fluid Dynamics  
CAV: Critical Arc Venturi  
DAS: Data Acquisition System  
SCORG: Screw Compressor Rotor Grid Generator

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