Journal of Applied Fluid Mechanics, Vol. 14, No. 6, pp. 1881-1891, 2021.
Available online at www.jafmonline.net, ISSN 1735-3572, EISSN 1735-3645.
https://doi.org/10.47176/jafm.14.06.32557

Prediction of Interlobe Leakages in Screw Compressors

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(Received January 10, 2021; accepted June 6, 2021)

ABSTRACT

The clearance gaps in twin-screw compressors are critical for their performance and reliable operation, as the leakage flows through these clearances influence the volumetric and adiabatic efficiencies. The amount of leakage flows depends on the clearance size and shape as well as various geometric and operating parameters. Usually, the isentropic nozzle equations along with appropriate flow coefficients are used for more accurate estimation of the leakage flow rates through the clearance gaps. Hence, a proper understanding of the flow coefficients and their relationship with the dimensionless parameters (such as pressure ratio, Reynolds number, and aspect ratios) is critical for an accurate prediction of the leakage flows. In the present study, considering the interlobe clearance gap in screw compressor in terms of rectangular openings, the interlobe leakage flow rates are estimated for various opening sizes and pressure conditions using isentropic nozzle equations and an iterative method. The flow coefficients are determined by comparing the experimental values obtained using a specialized test rig and the flow rates obtained from the analytical methods. The dimensionless parameters are varied to see their individual effect on the leakage mass flow rates and on the flow coefficients. The mean deviation from the experimental results when using an analytical iterative procedure (-8.5%) is substantially lower in comparison to the mean deviation (+26.8%) using the isentropic nozzle equations. The study validates that the iterative method can be preferred (for an interlobe leakage flow rate prediction) over the isentropic nozzle equation method.

Keywords: Interlobe clearance; Nozzle equations; Flow coefficients; Leakage flow; Compressor.

NOMENCLATURE

A	clearance leakage area
h	rectangular clearance height
k	specific heat ratio
$\dot{m}$	leakage mass flow rate
p	perimeter
P_1	upstream pressure
P_2	downstream pressure
R	gas constant
r	ratio of downstream pressure (P_2) to upstream pressure (P_1)
Re	Reynolds Number
u	mean flow velocity
w	rectangular clearance width
ε	ratio of upstream pressure (P_1) to downstream pressure (P_2)
ρ_2	downstream density
μ_v	dynamic viscosity of gas
ξ	local resistance coefficient at a gap entry
λ	friction factor
Σ	form factor
Δ	convergence factor
AR	Aspect Ratio (height to width)
CFD	Computational Fluid Dynamics
CAV	Critical Arc Venturi
DAS	Data Acquisition System
FC	Flow Coefficient
ISO	International Organization for Standardization
Exp/exp	for experimental data
Ana/ana	for analytical data

1. INTRODUCTION

Compressed air is often considered as the "fourth utility" as it has application in almost all types of industries. The size of the compressed air system can vary from 5 hp to 50,000 hp and in many facilities, the compressed air consumes more electricity than any other equipment used in the industry (U.S. Department of Energy, 2003). Due to a lesser number of moving components as compared to other compressors, the rotary screw compressors are simple and compact in design as well as possess better reliability. In the last few decades, improved rotor profiles and casing designs have led to fewer internal leakages along with cost effective and more accurate manufacturing processes due to which the reciprocating compressors have been replaced by screw compressors (Stošic *et al.* 2010).

2. INTERLOBE CLEARANCE LEAKAGES IN TWIN-SCREW COMPRESSOR

The twin-screw compressors are designed to have clearances for safe operation and these clearances result in leakages. The major leakage paths (Fig. 1) include the rotor tip housing clearances, triangular blowhole clearances, end plate clearances and interlobe clearances (Buckney *et al.* 2011).

The leakages within the screw compressor are complex phenomenon and each leakage path has a different effect on the performance of a twin-screw compressor. That is the reason why an accurate prediction of these leakages using experimental and numerical studies is important (Patel *et al.* 2020). The rotor tip housing leakages, end plate leakages and blowhole leakages occur from one enclosed chamber to another adjacent chamber and result in a reduction in the indicated efficiency. The interlobe leakage occurs from one enclosed chamber to the suction chamber and results in reduction of the volumetric and indicated efficiencies (Fleming and Tang 1995). An interlobe clearance is the minimum gap between two matching rotors needed to allow deflection due to gas loading, thermal distortion of rotors, bearings and housing, manufacturing errors and oil film thickness. Interlobe clearance is an important factor in the design of rotary screw compressors, and it is critical to have a uniform clearance that modern manufacturing operations can achieve (Tang and Fleming 1993) (Rane, 2015). Figure 2 shows an interlobe distance length and its replication across the rotor length.

It is very difficult to measure the interlobe clearance of a rotor pair because of complexity, Wu and Hoang proposed a simplified method to estimate this interlobe clearance (Hoàng and Wu 2020). A mathematical model was presented by Yuan *et al.* (1992) to calculate the rotor tip housing and interlobe leakages with consideration of both the viscous and inertia forces and compared with the experimental results. Prins and Ferreira presented (Prins and Ferreira 1998b) (Prins and Ferreira 1998a) four mathematical models, compared the same with two experimental results available in literature and presented a new optimized model using an algorithm and wall shear stress considerations. Li *et al.* (2007) used Fluent CFD package to simulate the leakages

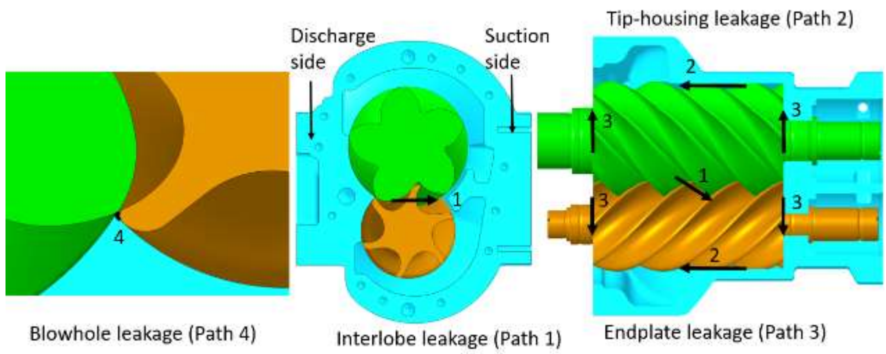


Fig. 1. A cross-section of screw compressor showing the location of various types of leakages.

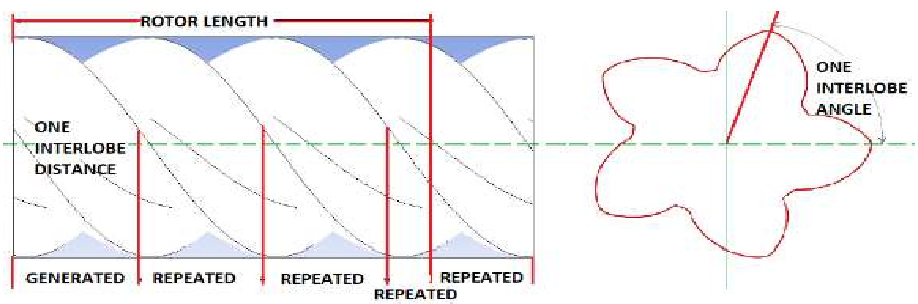


Fig. 2. Interlobe clearance in a screw compressor (Rane, 2015)**.**

and found the interlobe leakages lower in comparison to all other leakages. Fong *et al.* (2001) proposed a mathematical model calculating the interlobe clearances between two rotors. The clearance field was presented by an iso-clearance contour diagram (ICCD). This method avoided the problem of discontinuity and divergence in the optimal programming because of the use of a single mathematical model. Xiong (2006) described a mathematical model that demonstrated a clearance jump along the sealing (contact) line and an interlobe clearance distribution along the contact line that differed significantly from other clearance generation methods. Rane *et al.* (2015) presented a procedure of analytical grid generation for the interlobe clearances and explained the influence of mesh refinement in this clearance on the performance prediction in a test case. (Seshaiah *et al.* 2005, 2006, 2007) conducted an experimental study on air and helium explaining the effect of interlobe clearance on PV (Pressure-Volume) diagram and volumetric efficiency at different RPM for a fixed injected oil temperature and inlet conditions. The volumetric efficiency of helium was found to decrease more severely with an increase in interlobe clearances at lower rpm than air (due to lower molecular weight). Buckney *et al.* (2017) predicted the performance with revised clearances, tested for two different compressors at 5 bar pressure ratio and compared with the original clearance design. The revised design with new clearances predicted a 0.5% drop in flow while in reality the drop with the new design was found to be 1.7%.

In the past, numerous mathematical models were developed and various studies (using CFD and simulation tools) were carried out to predict the mass flows through clearance gaps. Utri *et al.* (2018) and Utri and Brümmer 2018a,b) used dimensionless numbers to present fluid flow through rotor tip housing clearances and front end clearances. The flow coefficients were determined by comparing CFD simulation results with the analytical and experimental results available in the literature (Sachs 2002). The study recommended using two different simulation 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 considers simple isentropic nozzle equations (Eqs. 1 and 2) to estimate mass and energy exchange between the chambers under consideration (Saint *et al.* 1839). The Saint-Venant equation was modified and used to calculate the leakage rate through the labyrinth seals. The model included a new form of flow coefficient that was calculated using experimental data and defined in terms of Reynolds number and radial clearance (Joachimiak 2020)(Joachimiak and Krzyślak 2019).

$$\dot{m}_{ana} = \frac{A\,p_1}{\sqrt{T_1}}\sqrt{\frac{2*k*\left((r)^{\frac{2}{k}}-(r)^{\frac{k+1}{k}}\right)}{R\,(k-1)}} \quad (1)$$

$$for\ r > \left(\frac{2}{k+1}\right)^{\frac{k}{k-1}}$$

$$\dot{m}_{ana} = \frac{A\,p_1}{\sqrt{T_1}}\sqrt{\frac{2*k}{R\,(k-1)}} * \left(\frac{2}{k+1}\right)^{\frac{k}{k-1}} \quad (2)$$

$$for\ r < \left(\frac{2}{k+1}\right)^{\frac{k}{k-1}}$$

The flow coefficient (FC) can be defined as a ratio of experimental leakage flows to analytical leakage flows,

$$FC = \frac{\dot{m}_{exp}}{\dot{m}_{ana}} \quad (3)$$

Utri and Brümmer 2018a and Utri *et al.* (2018) used CFD-simulation (using ANSYS CFX) to derive the mass flow rate and compared the same with the experimental data from the work of Kauder and Sachs (2002) to estimate the flow coefficient (FC).

Zakharenko (1965) and Sakun (1970) developed a 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.* (2018a); Kotlov *et al.* (2018b) used the same mathematical model (an iterative method) to find out the gas leakages from high-pressure cavities to the low-pressure cavities in the vane compressor and dry claw compressor. Kotlov *et al.* (2018b) used the following equation to calculate the leakage flow rates through the gap at each fixed angle.

$$\dot{m}_{ana} = FC * A * \sqrt{\frac{\rho_2 * p_2 * (\varepsilon^2-1)}{ln\,\varepsilon^2 + \xi + (\lambda * \sum)}} \quad (4)$$

In this equation, downstream density $\rho_2 = \frac{p_2}{R*T_1}$ and $\varepsilon = \frac{p_1}{p_2}$

In Eq. (4), ξ is the total coefficient of local resistance at the entrance and exit of the gap, which can be calculated as under.

$$\xi = \frac{(2*\Delta P_1)}{\rho * u^2} \quad (5)$$

where ρ is fluid density, and ΔP_1 is pressure drop which is approximated by:

$$\Delta P_1 = \Delta P_{fr} = \lambda * \frac{1}{D_h}\frac{(\rho * u^2)}{2} \quad (6)$$

where ΔP_{fr} is frictional pressure drop and λ is friction factor which can be calculated as:

$$\lambda = 189.2 * \mathrm{Re}^{-1.127} \text{ for Re} < 1200 \quad \text{or}$$

$$\lambda = 3.6 * \mathrm{Re}^{-0.566} \text{ for Re} \geq 1200 \quad (7)$$

In Eq. (6), D_h is the hydraulic diameter of the rectangular gap, which can be calculated as:

$$D_h = 2 * h \quad (8)$$

where h is leakage gap height.

In Eq. (4), $\sum$ is form factor, which can be calculated as:

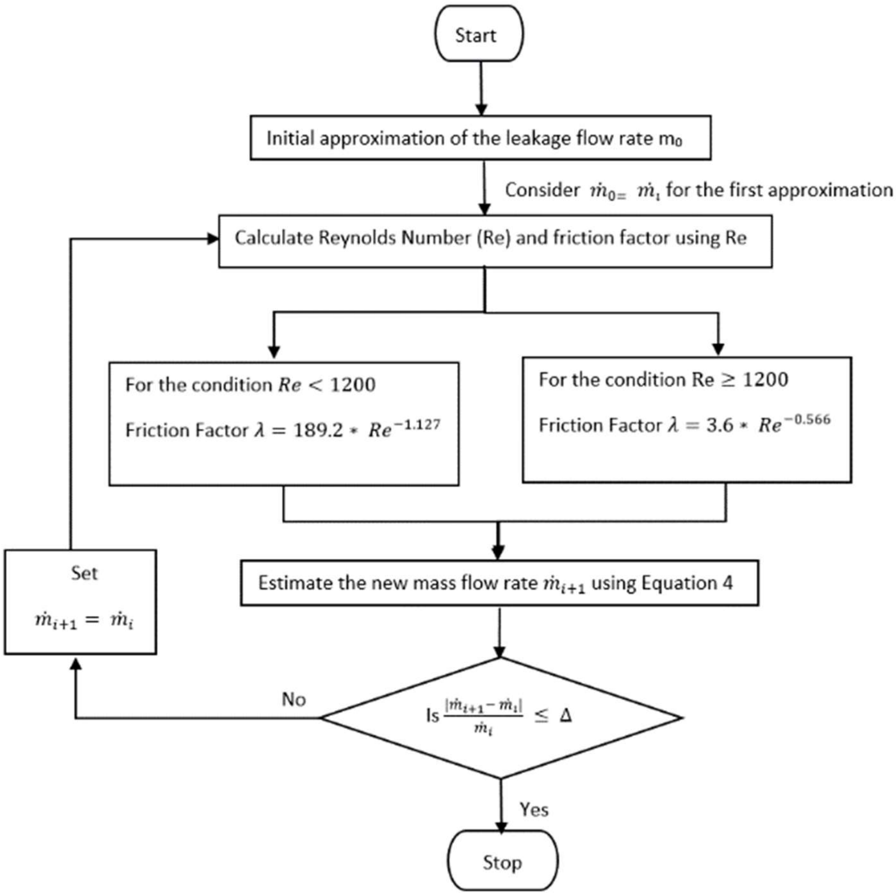


Fig. 3. An iterative method for the leakage mass flow rate estimation.

$$\Sigma = \frac{b}{2*h}\left(1 + \frac{h}{L}\right) \tag{9}$$

where b is clearance width and L is the clearance length.

Equation 4 required friction factor λ, which depends on the Reynolds number (which in turn requires the mass flow rate to be known).

$$Re_0 = \frac{4\dot{m}_0}{\mu_v\, p} \tag{10}$$

where Re_0 is an initial Reynold number used for the iterative method.

The method of successive approximation can be used to solve Eq. 4 and the initial approximation of the mass flow rate ($\dot{m}_1$) at the critical discharge area can be determined by using Eqs. 1 and 2. This initial mass flow rate ($\dot{m}_1$) can be used to find out the Reynolds number, friction factor etc. until the following condition is fulfilled.

$$\frac{|\dot{m}_{i+1} - \dot{m}_i|}{\dot{m}_i} \leq \Delta \tag{11}$$

Here Δ is the pre-set accuracy of the calculation.

Figure 3 presents the flow chart to calculate the leakage mass flow rate using an iterative method.

In the present study, considering the interlobe clearance gap in the screw compressor in terms of rectangular openings, the interlobe leakage flow rates are estimated for various opening sizes and pressure conditions using the isentropic nozzle equations and iterative method (Fig. 3). The flow coefficients are determined by comparing the experimental values obtained using a specialized test rig and the flow rates obtained from the analytical methods.

Experiments and analytical techniques were used to analyse blowhole leakages in the twin-screw compressor. The same approach is used in this analysis to estimate the interlobe leakages in the twin-screw compressor. The setup is slightly different from that used in the previous analysis. In place of the circular clearances used in previous work, rectangular clearances (representing the unwrapped interlobe clearance) are used (Patel and Lakhera 2021a) (Patel and Lakhera 2021b).

3. EXPERIMENTAL INVESTIGATION

The leakages happen through the clearance gaps because of the pressure difference and the clearance gaps. In the present work, the upstream pressure (P_1) varies and the downstream pressure (P_2) is maintained at atmospheric pressure to obtain the pressure ratio and simulate interlobe leakage conditions. Table 1 shows the geometry details of rectangular clearances of all three plates.

Table 1 Geometry details of rectangular clearances

Plate Number	Width -w (mm)	Height – h (mm)	Areas (mm2)	AR ratio
P1	40	0.180	7.2	0.0044
P2	45	0.200	9.0	0.0045
P3	44	0.250	11.0	0.0058

3.1 Test Rig Description

A test setup to simulate and determine the interlobe leakages experimentally is shown in Fig. 4 and Fig. 5. Two sets of flow measurements were carried out for each combination of clearance size and pressure difference to calculate the leakage. In the first case (no leakage condition), the flows were measured at a specific compressor discharge pressure condition using CAV nozzle without opening the leakage line. In the second case (leakage condition), the flows were measured at a compressor discharge (using CAV nozzle) by keeping the leakage line open and allowing leakage though the rectangular clearances. The difference between two values of measured flow (from both cases) gives a leakage flow rate through the clearance at a specific pressure ratio.

A twin-screw compressor (22 kW constant speed) was used to generate compressed air at 10 bar pressure and the compressed air flow rate (in both with and without leakage conditions) was measured with the help of CAV (Critical Arc Venturi) nozzle as per ISO 1217 (ISO, 2009). A globe valve was installed to control the upstream pressure (before the leakage clearance in spool assembly). The compressor discharge pressure, compressor discharge temperature, CAV nozzle pressure and temperature, barometric pressure, upstream pressure and downstream pressure were recorded using various sensors connected to the DAS. A ball valve after the tee connection was installed to isolate/connect the leakage line with the main stream. The spool assembly with a sandwiched plate (having a rectangular slit on the plate) was installed in the leakage line after the globe valve to simulate the leakages.

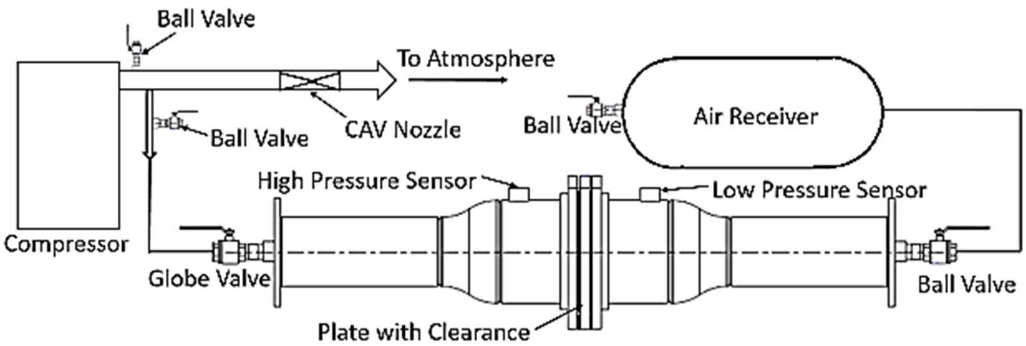


Fig. 4. A schematic diagram of the experimental facility.

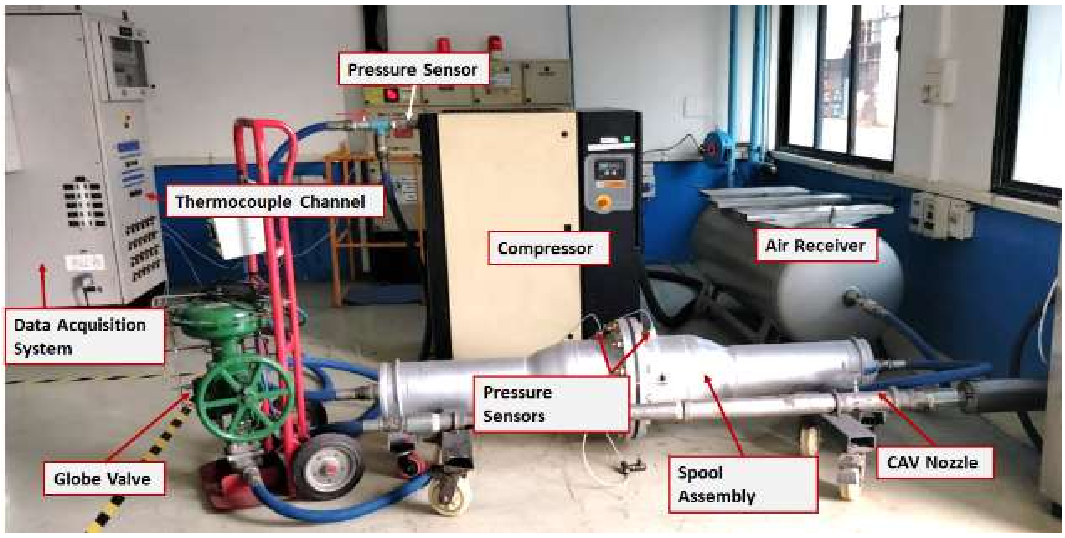


Fig. 5. Photograph of the experimental test rig used for the study.

3.2 Test Methodology to Measure the Interlobe Leakage

The present study involves simulation of the interlobe leakage flow using an experimental test set up. The test methodology (Fig. 6) for estimating the leakage involves plates (with rectangular clearances) and a specific pressure difference across the plates. Measuring the flow using a CAV nozzle gives an estimation of the leakage mass flow-rate taking place across the plate for the specific pressure difference. Initially, the total flow without any leakage was considered in the experiment by bypassing the leakage line (Case 1) wherein all the pressurized air at 10 bar pressure from the compressor flowed through the CAV nozzle. In the next step (Case 2), the flow through the leakage line was allowed for the specific combination of the rectangular slit and pressure difference. The difference between the measured values of the compressed air flow in Case 2 through the CAV nozzle, which was lower in comparison to the compressed air flow measured in Case 1, provided an experimental estimation of the leakages

The details of the testing in the two cases are included as per the following:

Case 1: Flow measurement - No leakage condition

In this case of zero leakage, the leakage line was isolated from the discharge line by closing the ball-valve installed before the plate. The compressed air flow at 10 bar was measured in the discharge line as per ISO 1217 with the help of a CAV nozzle.

Case 2: Flow measurement – Leakage condition

In this case, the compressed air leakage was allowed to pass through the plate (having a rectangular slit) installed in the leakage line by keeping the ball valve fully open. The ball valves (installed on the package discharge) and the globe valve (installed before the CAV nozzle) were used to maintain the desired pressure difference across the plate having rectangular clearance.

In the present study, the leakage mass flow-rates were measured for three plates having different cross-section areas (refer to Table 1) with upstream pressure varying and downstream pressure kept at atmospheric pressure.

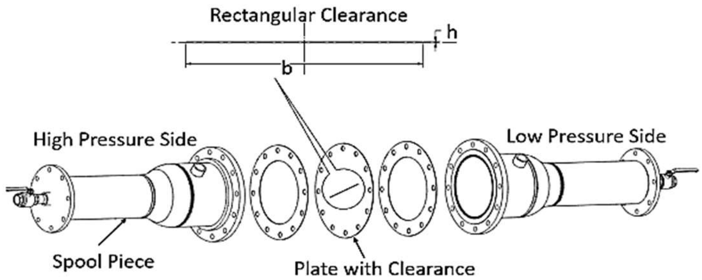


Fig. 6. Spool assembly used in testing.

Table 2 Details of the equipment/ instruments used in the experimentation (Patel & Lakhera, 2021b)

Equipment/ Instrument	Model	Make	Uncertainty	Specification
Twin-screw compressor	RS22-7-AC	Ingersoll Rand	---	Oil flooded twin-screw compressor 22kW, 7 bar, 140 CFM
Data Acquisition System (DAS)	34980A	Agilent	---	140 channel universal input and connectivity to lap top for real time monitoring and capturing the readings
Pressure transducer	245 series	Viatran	± 0.1%	Measuring range- 0 to 300 psig
T type thermocouple	TQSS-18U-6	Omega	0.5° C	Measuring range -200 to 350°C
Barometric pressure gauge	CPG2400	Mensor	0.03%	Measuring range 8-17psia
CAV pressure gauge	CPG2400	Mensor	0.03%	Measuring range 0-100 psig
Globe valve with actuator	GX+FIELD VUE DVC6200	Emerson	± 0.5%	Medium: Air or Natural Gas Independent linearity Typical Value: ±0.50% of output span

3.3 Uncertainty Analysis

The total air flow uncertainty at the confidence level of 95% calculated as per the following equation:

$$U_{95} = \sqrt{B_w{}^2 + (t_{95}\, S_w)^2} \quad (12)$$

where,

B_W is a systematic error in the flow measurement S_W is an experimental standard deviation in the flow measurement, and t_{95} is a Student's statistical parameter at the 95% confidence level

The flow measurement standard deviation (B_w) and systematic error (S_w) can be derived using the following equations:

$$B_w = \dot{m}\sqrt{\left(\frac{B_{P1}}{P_1}\right)^2 + \left(\frac{-B_{T1}}{2T_1}\right)^2 + \left(\frac{B_C}{C}\right)^2 + \left(\frac{2B_d}{d}\right)^2} \quad (13)$$

where,

B_C is a systematic error for discharge coefficient
B_d is a systematic error for throat diameter
B_{P1} is a systematic error for the upstream pressure
B_{T1} is a systematic error for the upstream temperature
C is a discharge coefficient
d is a throat diameter in meters
$\dot{m}$ is a mass flow rate in kg/sec

$$S_w = \dot{m}\sqrt{\left(\frac{S_{P1}}{P_1}\right)^2 + \left(\frac{-S_{T1}}{2T_1}\right)^2 + \left(\frac{S_C}{C}\right)^2 + \left(\frac{2S_d}{d}\right)^2} \quad (14)$$

S_C is an experimental standard deviation for a discharge coefficient
S_d is an experimental standard deviation for a throat diameter
S_{P1} is an experimental standard deviation for the upstream pressure
S_{T1} is an experimental standard deviation for the upstream temperature

The uncertainty analysis for the measurement involved in the experiment was carried out as per the procedure described in ISO 5168:2005 (ISO, 2005). Based on the measured parameters, the uncertainty in the total airflow ($\dot{m}$) with a confidence level of 95% was found to be ±1.29%. The details of the equipment / instruments used in the experimentation along with the uncertainty involved are elaborated in previously published work (Patel and Lakhera 2021b).

4. RESULTS AND DISCUSSION

The experimental results from the study are compared with the analytical results (isentropic nozzle equations and an iterative method used by Kotlov *et al*. (Kotlov *et al.* 2018a; Kotlov *et al.* 2018b) and are presented in the form of dimensionless numbers like the flow-coefficients, the Reynolds numbers and the pressure ratios. The flow coefficients derived for all the three plates (having rectangular clearances) with different pressure conditions and presented in this section.

Figure 7 shows the experimental and analytical results of leakage mass flow rates (kg/sec) and the flow coefficient (FC) as a function of pressure ratio (ε) for Plate 1 (Clearance Area = 7.2 mm2). The analytical leakage mass flow rates (using both isentropic nozzle equations and iterative method) are compared with experimental mass flow rates. The isentropic nozzle equations were discovered to over-predict leakage mass flow rates for all pressure conditions, whereas the iterative method was discovered to under-predict leakage mass flow rates. The leakage mass flow rates were found higher for the lower value of pressure ratio because of the higher pressure difference across the plates and found reduced with increase in the pressure ratio (reducing the pressure difference across the plate).

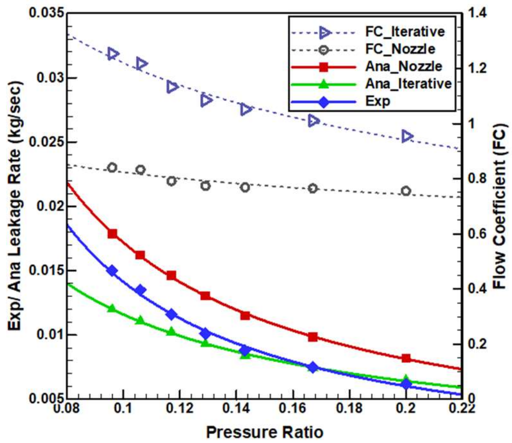


Fig. 7. Experimental and analytical results of leakage mass flow rates (kg/sec) and the flow coefficient (FC) as a function of pressure ratio (ε) for Plate 1 (Clearance Area = 7.2 mm2).

The flow coefficient using isentropic nozzle equations was found in the range of ~0.75 to ~0.85 while the flow coefficient with iterative method was found in the range of ~1.00 to ~1.25. The isentropic nozzle equations take frictionless flow into account and estimate leakage mass flow rates. The iterative approach, on the other hand, takes into account the local resistance, geometry of the clearance, and friction factor in addition to pressure ratio and clearance areas and results in under predicting leakages as compared to isentropic nozzle equation results and experimental results.

Figure 8 and 9 show the experimental and analytical results of leakage mass flow rates (kg/sec) and the flow coefficient (FC) as a function of pressure ratio (ε) for Plate 2 (Clearance Area 9.0 mm2) and Plate 3 (Clearance Area 11.0 mm2) respectively. In both cases (Plates 2 and 3), isentropic nozzle equations were found to overestimate leakage mass flow rates, while iterative methods were found to underestimate leakage mass flow rates (similar to Plate 1). The changes in the flow coefficients were also comparable to those observed in Plate 1.

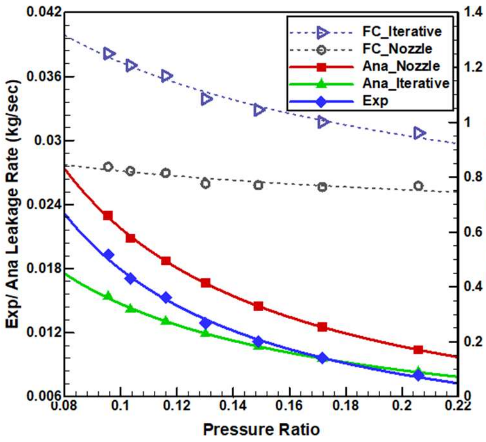


Fig. 8. Experimental and analytical results of leakage mass flow rates (kg/sec) and the flow coefficient (FC) as a function of pressure ratio (ε) for Plate 2 (Clearance Area = 9.0 mm2).

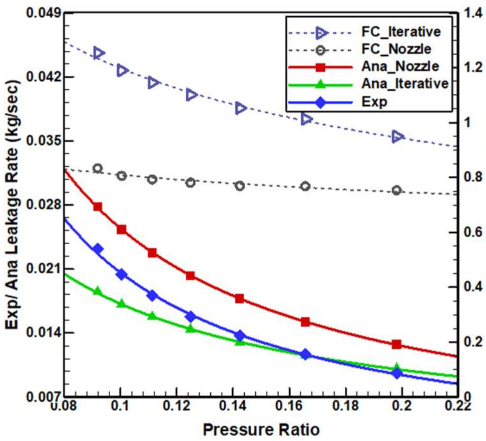


Fig. 9. Experimental and analytical results of leakage mass flow rates (kg/sec) and the flow coefficient (FC) as a function of pressure ratio (ε) for Plate 3 (Clearance Area = 11.0 mm2).

Figure 10 depicts the flow coefficient variance as a function of pressure ratio for various clearances using the isentropic nozzle equations. The flow coefficient value was found to be a near match for all three plates at a particular pressure ratio, indicating that the flow coefficient is not affected by the leakage area (clearance height) for the specified pressure ratio range. The isentropic nozzle equations assume frictionless flow, while in fact, leakages are influenced by friction, that's why flow coefficients are lower in the range of 0.75 to 0.85. Because of the low velocity and increased friction, the flow coefficient decreases with increasing pressure ratio (at reduced pressure difference).

Figure 11 shows the variation of flow coefficient as a function of pressure ratio for different clearances using the iterative method. The flow coefficient values found unchanged for all the plates at a given specific pressure ratio, which indicates that the clearance height does not have much influence on the flow coefficient. The flow coefficients using an iterative method found in the range of ~1.00 to ~1.25. The iterative method underestimates the leakage mass flow rate because it considers shape, friction forces, and local resistances. Because of the reduced velocity caused by the increased value of friction, the flow coefficient is lower for higher pressure ratios (similar to isentropic nozzle equation results).

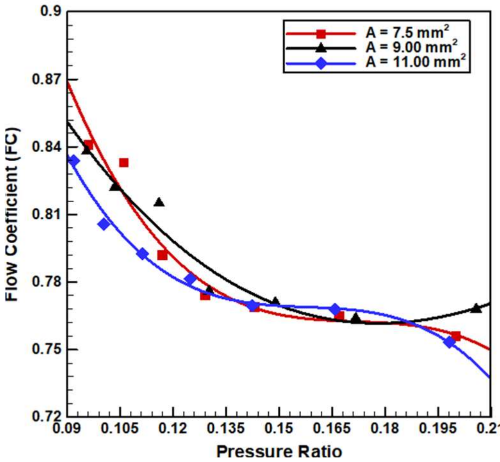


Fig. 10. Variation of Flow coefficient (FC) with pressure ratio (ε) for different clearances considering isentropic nozzle equations.

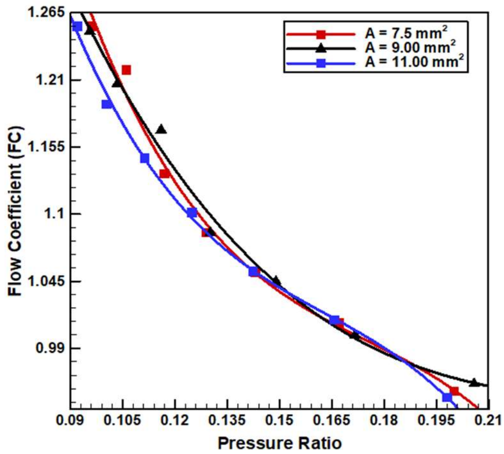
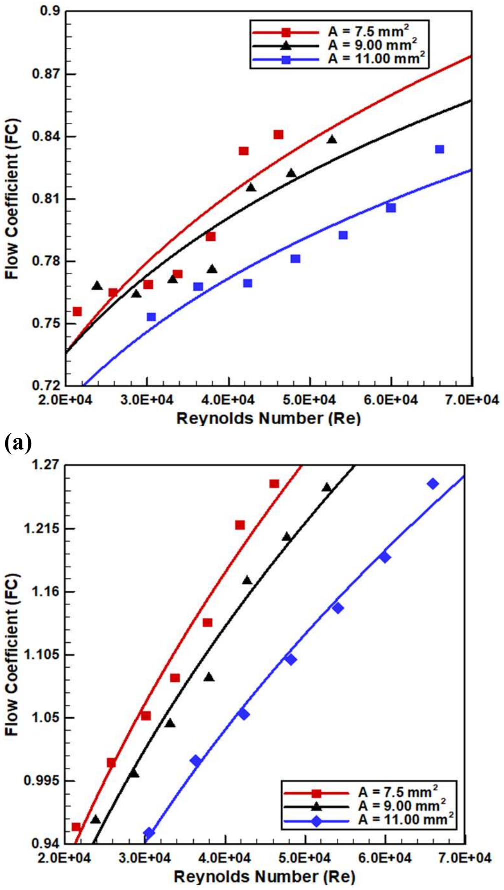


Fig. 11. Variation of Flow coefficient (FC) with pressure ratio (ε) for different clearances using the iterative method.

Figure 12 shows the variation of flow coefficient as a function of Reynolds number at different clearances using both the analytical models. The flow coefficient found increasing with an increase in the Reynolds number for all the plates (in the case of both the models). The flow coefficient found changed with the change in clearance area for the same value of Reynolds number, which indicates a significant influence of clearance height on the flow coefficients. The higher value of flow coefficient for lower clearance area at the same Reynold number is due to lower friction value. The iterative method provides a more rapid determination of the change in flow coefficient as a function of the Reynolds number value.



(a)

(b)

Fig. 12. Variation of Flow coefficient (FC) with Reynolds number at different clearances (a) isentropic nozzle equation method (b) iterative method.

Figure 13 shows a comparison between the experimental results and the analytical results considering the nozzle equation and the iterative procedure. It indicates that the iterative method results (mean deviation is -8.5%) are closer to the experimental results in comparison to the results using an isentropic nozzle method (mean deviation is +26.8%).

5. CONCLUSIONS

The adiabatic and volumetric efficiencies are influenced by the interlobe clearance, which is critical for safe and reliable operation of a twin-screw compressor. An unwrapped interlobe clearance is approximated as rectangular clearance in this study, and the interlobe leakages are simulated using an experimental setup and compared with the analytical results to derive the flow coefficients. Using isentropic nozzle equations, the flow coefficients are found to be in the range of 0.75 to 0.85, compared to 1.00 to 1.25 using the iterative process. When predicting leakage mass flow rates, the isentropic nozzle equation considers frictionless flow, while the iterative approach considers local resistance, geometry of the clearance, and friction factor (in addition to pressure ratio and clearance areas). This results in an iterative method under predicting the leakages when compared to isentropic nozzle equation results and experimental results.

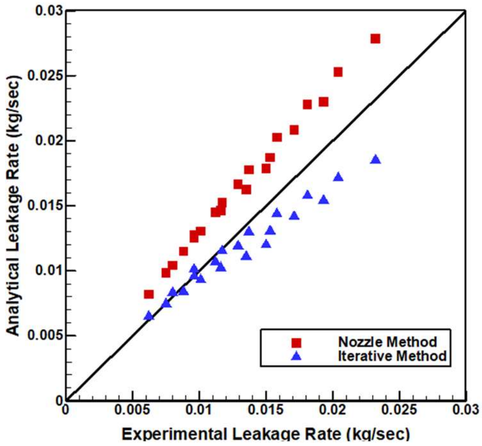


Fig. 13. A comparison between the experimental results and analytical results considering the nozzle equation and the iterative procedure.

Because of the low velocity and increased friction, the flow coefficient decreases with an increase in the pressure ratio (at reduced pressure difference). The flow coefficient values do not change drastically across all plates at a given specific pressure ratio in either model, but they do change dramatically with pressure ratio change for a specific clearance area. For the same Reynolds number, the flow coefficient changes as the clearance area changes, indicating that clearance height has a significant influence on the flow coefficients. At the same Reynold number, the lower friction value accounts for the higher flow coefficient for lower clearance area. In the case of the iterative process, the difference in the flow coefficient with the change in the value of the Reynolds number is observed more rapidly.

In comparison to the results of the isentropic nozzle equation, an analytical iterative procedure yields more accurate results (specifically at higher pressure ratios). When using the iterative procedure, the mean deviation from the experimental results (-8.5%) is significantly lower than the mean deviation (+26.8%) when using the isentropic nozzle equations. The study indicates that the iterative approach is preferable to the isentropic nozzle equation method for predicting interlobe leakage flow.

ACKNOWLEDGEMENTS

The authors acknowledge the support received from Ingersoll-Rand (India) Limited and Ingersoll-Rand USA for conducting the experimental work.

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