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Validation of Leakage Model for an Oil-free Twin-Screw Compressor

Hitesh H Patel[1], Vikas J Lakhera[1] and, Ahmed Kovacevic[2]

[1]Department of Mechanical Engineering, Institute of Technology, Nirma University, Ahmedabad-382481, INDIA.
[2]Centre for Compressor Technology, City University of London, London EC1V0HB

ABSTRACT

The performance of a twin-screw compressor is influenced by leakages, which are normally calculated using isentropic nozzle equations with flow coefficients. The changes in the clearances during operation necessarily require detailed studies for accurate leakage estimations. In this research, the flow coefficient correlations are derived for various shapes of leakage gaps using previous experimental results and regression analysis. They are integrated into SCORG (Screw Compressor Rotor Grid Generator) software to estimate leakages and performance of an oil-free twin-screw compressor. The results obtained from the proposed flow coefficients are firstly compared with the original inbuilt flow coefficients and experimental results of the same compressor for the designed assembly clearances. The clearances are then adjusted to approximate the operating clearances, and they are used for further investigation of the same compressor profile (different sizes) at various pressure and speed conditions. The findings indicate that the flow of the compressor using the new flow coefficients is greater than the flow estimation using the original flow coefficients, up to ~41%, ~31% and ~24% for three different sizes of compressors, each larger than the previous one. The total power consumption for all models in all sizes and operational circumstances is within 1% of the calculation with original clearances.

Keywords: Flow coefficient, leakage, twin-screw compressor, performance, SCORG™

1. INTRODUCTION

Twin-screw compressor clearance gaps (Figure 1) are essential for their performance and reliability. Advanced manufacturing processes allow to manufacture rotors with very close tolerances up to 5 microns [1] which in turn enables improvement in performance. It is obvious that reducing the clearances in the twin-screw compressor results in the reduction of leakages and increase in the volumetric and adiabatic efficiencies. However manufacturing and assembly errors, rotor and casing deformation because of the thermal effects and rotor deflection during the operation becomes very important consideration for the safe operation of the compressor [2][3]. The change in clearances during the operation influence the leakage and overall performance estimations, and that is the reason it is important to understand the operating clearances and their effect on the performance.

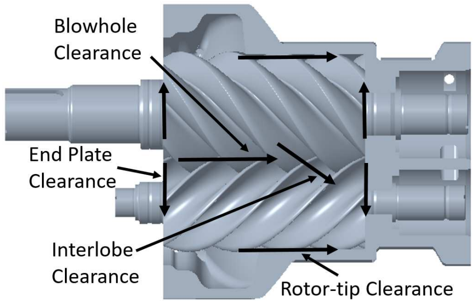


Figure 1: Leakage clearances in a screw compressor

2. LITERATURE REVIEW

The twin-screw compressor performance depends on various leakages, which occur through the clearances between the rotors and between the rotor and the casing. Fujiwara et al. [4][5] presented a computer program to estimate the performance of oil-free and oil-flooded twin-screw compressors using the isentropic convergent nozzle equations for all types of leakage except the oil leakage through the tip housing clearance leakage. Fleming et al. [6] defined and analyzed six different types of leakages and presented a computer program to calculate these leakages.. Prins and Ferreira [7] studied four different models (quasi one dimensional) and used results from the experimental work of Ishii et al. [8] and Peveling [9] to validate these models. The authors concluded that none of the models could accurately estimate leakage. The wall shear stress model found a better fit with the experimental data [10].

Several experimental studies have been conducted by researchers to investigate the effect of various parameters on the performance of the twin-screw compressor [11]. An experimental investigation was carried out to simulate leakages in positive displacement machines and the flow coefficient correlations were derived to enhance the leakage prediction accuracy [12].

Rane et al. [13] improved the leakage estimation accuracy by enhancing and refining the meshing in the flow region near the blowhole and the interlobe areas. The study also described a test case showing the effect of mesh refinement on performance prediction. As it is difficult to estimate the clearance changes during the operation, Buckney [3] developed and presented a procedure to enable a chamber model to predict the performance with consideration of rotor and casing expansion because of the thermal effects. Authors developed a simple way to estimate an operating clearance as a tool for determining the operating envelope of a certain screw compressor application. Utri et al. [14][15]

investigated two-dimensional fluid flow through the end clearance and rotor tip housing clearance of twin-screw compressors. The authors used dimensionless numbers and varied them systematically to check their individual effect on the endplate leakage rate. Sun et al. [16][17] performed a PIV test to obtain the velocity field around the tip gap and used SCORG™ software to predict the leakage flow under same operating conditions.

The leakage mass flow rate through the clearance gaps are determined using isentropic nozzle equations, which uses flow coefficients to take into account actual flow conditions. As a result, a thorough knowledge of the flow coefficients is essential for accurate leakage flow prediction. In the present study, the flow coefficients correlation in terms of pressure ratio, aspect ratio and leakage area are derived and are used in SCORG™ software to predict the performance of an oil-free twin-screw compressor of N35 Profile. The results are compared with the results derived using SCORG™ original performance predictions and the experimental results of same oil-free twin-screw compressor [18][19].

3. METHODOLOGY

SCORG™ is a specialized software used for design and analysis of screw compressors, expanders, pumps, and motors. It facilitates the evaluation of the performance and operation of these machines with the use of CFD and Thermodynamic Multi Chamber Models. SCORG™ consists of four modules, which enable the user to do the following: 1) import male and female rotor profiles as point files, 2) compute the change in working chamber volume over time and identify leakage areas as well as input and output port areas for a specific machine, 3) conduct thermodynamic calculation of a machine's performance using a chamber model, 4) generate a numerical grid of moving rotor domains and stationary inlet and outlet domains, which are then loaded into commercial CFD solvers for analysis [20].

3.1The flow coefficient correlations and performance prediction models

The performance prediction outputs from the SCORG™ thermodynamic module include flow, power, specific power, adiabatic efficiency, volumetric efficiency, and so on. This module estimates the leakage flow based on the orifice flow calculation for theoretical Couette flow multiplied by the appropriate flow coefficient. Internal leakages have a significant impact on screw compressor performance, which makes leakage calculation an important part of the thermodynamic model. In SCORG™, the leakage flow between the working chamber and neighbouring chambers is assumed to be orifice flow and the mass flow rate between the chambers is defines by continuity equation as:

$$\dot{m} = \delta * A * v \qquad (1)$$

where velocity $v = \Phi * \sqrt{2 * \frac{\gamma}{\gamma - 1} * \left(\frac{p_2}{\delta_2} - \frac{p_1}{\delta_1}\right)}$ (2)

In equation 2, the flow coefficient Φ is derived using the standard equation recommended by ISO1567-1 as under:

$$\Phi = 0.5959 + 0.312 * \beta^{2.1} - 0.184 * \beta^{8} + 0.0029 * \beta^{2.5} * \left(\frac{10^6 * \beta}{Re_D}\right)^{0.75} \qquad (3)$$

In this study, Model 1 refers to the original inbuilt flow coefficients (Φ) calculated by equation 3 to estimate all the leakage rates (which includes the blowhole clearance leakages, rotor tip housing clearance leakages, interlobe and axial clearance leakages), while the Model 2 and Model 3 use the flow coefficient (Φ) correlations derived based on the experimental results from the previous study [12] and multivariable regression analysis. In the previous work, an experimental study was conducted to simulate leakages through circular and rectangular clearances of varying sizes under various pressure conditions [12].

The correlation for flow coefficient (Φ) (for circular clearance) in terms of pressure ratio (PR) and leakage areas (A) gives the relation as:

$$\Phi = [(0.00706 * \mathrm{PR}) + (0.0021 * \mathrm{A}) + (0.89664)] \qquad (4)$$

A similar multivariable regression analysis procedure is used to develop a correlation for flow coefficient (Φ) of rectangular clearance in terms of ratio of pressure difference to the upstream pressure ($(p_2 - p_1)/p_2$) and the ratio of the perimeter to the clearance height ($\mathrm{P/h}$). The correlation for flow coefficient (Φ) of rectangular clearances gives the relation as:

$$\Phi = [(-0.15879 * (p_2 - p_1)/p_2) - (0.00003 * \mathrm{P/h}) + (0.91464)] \qquad (5)$$

A correlation for flow coefficient (Φ) of rectangular clearance also derived in terms of ratio of pressure difference to the downstream pressure ($(p_2 - p_1)/p_1$) and the ratio of perimeter to the clearance height ($\mathrm{P/h}$). The correlation for flow coefficient (Φ) of rectangular clearance also gives the relation as:

$$\Phi = [(-0.00835 * (p_2 - p_1)/p_1) + (0.00001 * \mathrm{P/h}) + (0.83891)] \qquad (6)$$

The correlation derived in equation 4 is used in both Models 2 and 3, to calculate the leakages through the blowhole areas using thermodynamic module of the SCORG™. The correlations using equations 5 and 6 are incorporated in Model 2 and 3 respectively, to calculate leakages through the interlobe gaps, rotor tip housing gaps and axial gaps using thermodynamic module of the SCORG™. The difference between Models 2 and 3 is that Model 2 considers the ratio of pressure difference to upstream pressure, whereas Model 3 considers the ratio of pressure difference to downstream pressure. This work presents an investigation in which the thermodynamic performances of oil-free twin-screw compressor of different sizes are compared under various operating conditions using the original performance prediction model (Model 1) and suggested performance prediction models (Model 2 and Model 3). Except for the flow coefficient equations, all three models (Model 1, Model 2, and Model 3) use identical equations to estimate leakages in the thermodynamic performance prediction of a twin-screw compressor.

3.2 Performance prediction using "Cold Clearance" and "Operating Clearance"

The experimental results from this testing are compared with the performance prediction results using all three models

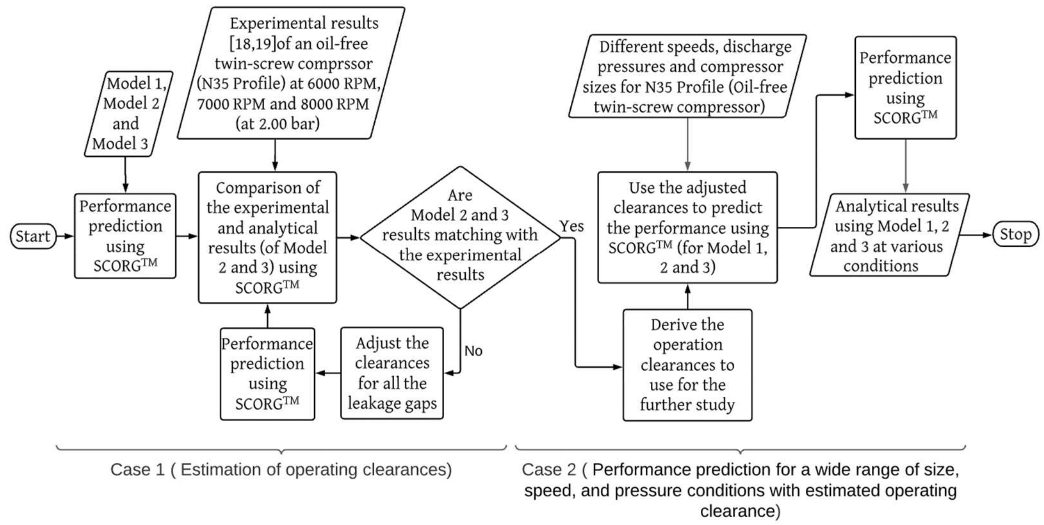


Figure 2: An overview of methodology used in the present study

considered in this case study. To begin (Case 1), the performances are estimated using the design clearances (called also 'Cold' clearances) and the results are compared to the experimental findings. Because the experimental results take into account the change in clearances caused by thermal deformation of the rotors and casing during operation, the design clearances need to be modified to the operating clearances. The iterations are carried out by changing the clearance values in SCROG™ to get the results (using Model 2 and 3) which match the experimental results (Figure 2). The purpose of this exercise is to approximate the operating clearances during the running condition of compressor (using N35 profile) and use the same adjusted clearance for wide range of discharge pressures, rotating speeds and compressors sizes.

Later in Case 2, performances are estimated using SCORG™ for an oil-free twin-screw compressor (of N35 profile) with adjusted clearance (called 'Operating' clearances) using the original prediction model (Model 1) and compared the same with prediction results from the proposed Model 2. The performances are predicted for different sizes (Size 1, Size 2 and Size 3) of oil free screw compressors (of N35 profile). Figure 2 outlines the methodology followed in the present study .The main rotor outer diameters of Size 1, Size 2 and Size 3 are 127.32 mm, 140.05 mm and 152.78 mm respectively. Similarly, the axial distances of Size 1, Size 2 and Size 3 are 93.00 mm, 102.30 mm and 111.60 mm respectively. The Rotor length and male rotor helix angle for all the sizes are 204 mm and 285° respectively. The performances are estimated for different rotational speeds (6000 rpm, 7000 rpm, 8000 rpm and 9000 rpm) and different discharge pressures (2.00 bar, 2.50 bar, 3.00 bar and 3.50 bar) using all three models. The Size 2 and Size 3 are 1.10% and 1.20% of the Size 1.0. The approximated adjusted clearances are used for the Size 1 compressor performance estimation, while Size 2 and Size 3 clearances are scaled up in accordance with the compressor sizes.

4. RESULTS AND DISCUSSION

The flow results for all models (with design and adjusted clearances) are compared in Case 1, while the flow results for Models 1 and 2 for various speeds, discharge pressures, and sizes are compared in Case 2 with adjusted operating clearances. Models 2 and 3 produce similar results, so only Model 2 results are presented in comparison to Model 1 (for Case 2).

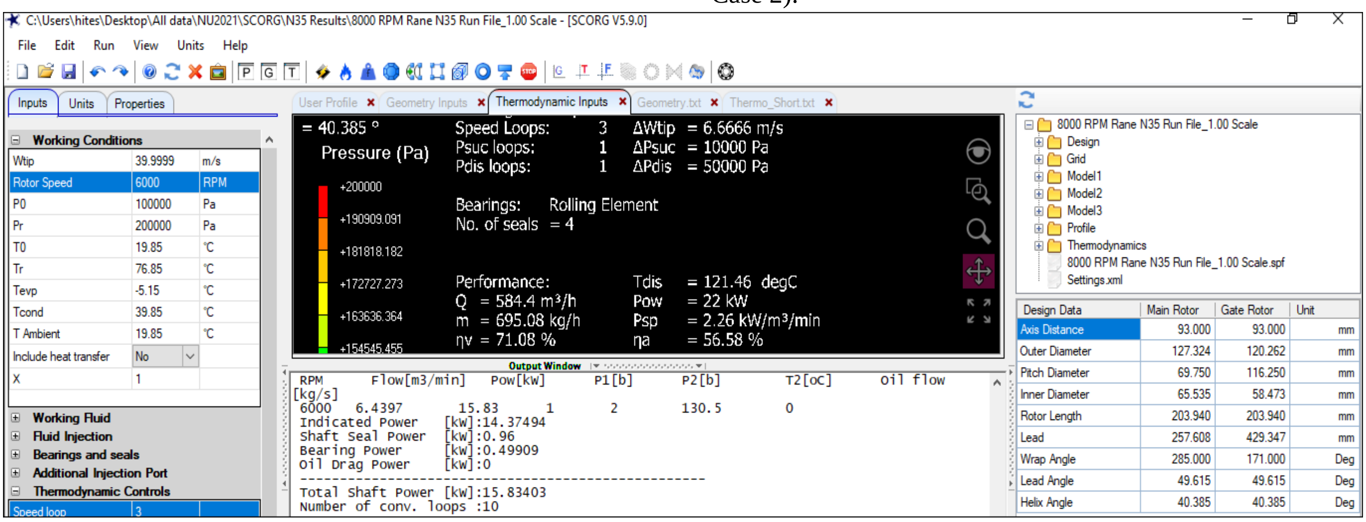


Figure 3: SCORG™ window showing performance results and inputs (6000 RPM, 2 bar)

Table 1: Comparison of estimated results with experimental results (Design clearances, 2 bar)

No	Speed (RPM)	Exp. Results Flow (m^3/min)	Exp. Results Power (kW)	Model 1 Results Flow (m3/min)	Model 1 Results Power (kW)	Model 2 Results Flow (m3/min)	Model 2 Results Power (kW)	Model 3 Results Flow (m3/min)	Model 3 Results Power (kW)
1	6000	7.05	15.94	5.98	15.77	6.67	15.83	6.75	15.87
2	7000	8.73	19.79	7.64	19.55	8.32	19.69	8.40	19.75
3	8000	10.36	23.63	9.30	23.30	9.98	23.48	10.0	23.54

Table 2. Comparison of estimated results with experimental results (Adjusted clearances, 2 bar)

No	Speed (RPM)	Exp. Results Flow (m^3/min)	Exp. Results Power (kW)	Model 1 Results Flow (m3/min)	Model 1 Results Power (kW)	Model 2 Results Flow (m3/min)	Model 2 Results Power (kW)	Model 3 Results Flow (m3/min)	Model 3 Results Power (kW)
1	6000	7.05	15.94	6.44	15.83	7.05	15.90	7.11	15.94
2	7000	8.73	19.79	8.11	19.63	8.70	19.79	8.76	19.84
3	8000	10.36	23.63	9.74	23.47	10.36	23.57	10.43	23.63

4.1 Case 1 - Comparison of performance prediction results with experimental results

Performance predictions are made with the design clearances using the original flow coefficient model (Model 1) and the suggested flow coefficient models (Models 2 and 3), and the results are compared to the experimental data as shown in Table 1. Figure 3 shows the SCORG^{TM} window showing the input parameters and results for one of the performance run (6000 RPM, 2 Bar). The results show that Model 1 under predicts the flow and power up to 15.13% (at 6000 RPM) and 1.40% (at 8000 RPM) respectively in comparison to the experimental results. Model 2 and Model 3 under predict the flow up to 5.32% and 4.27% at 6000 RPM in comparison to the experimental results. The Model 2 and Model 3 power estimations are lower in comparison to the experimental results up to 0.80%. Although flow coefficients are used to consider real conditions, the difference between the performances are because of the difference between design clearances (uses in the prediction) and operating clearances (actual during the operation).

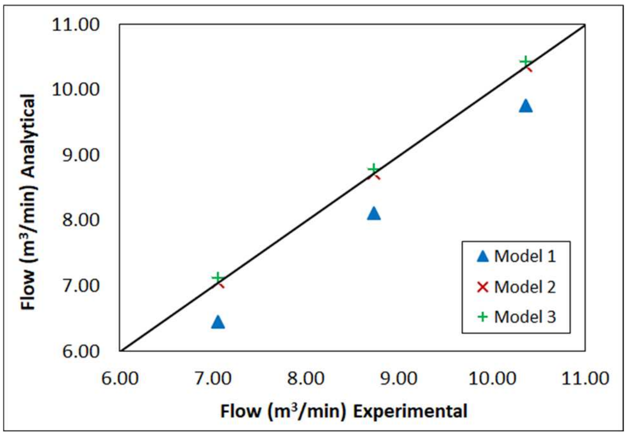


Figure 4: The comparison of experimental and analytical flow (using all the three models).

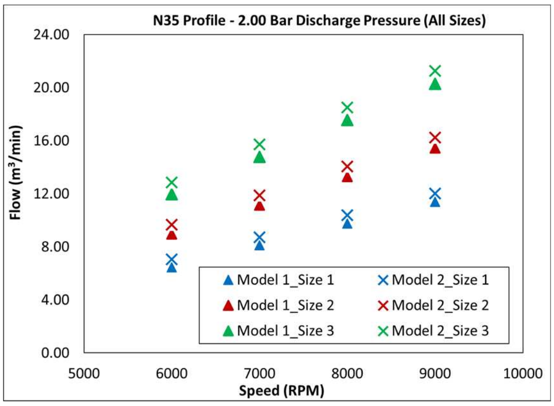
As the performance of Model 2 and Model 3 are closer to the experimental results (compared to results of Model 1), clearances are adjusted to match the Model 2 and Model 3 results with the experiment results to approximate the operating clearance condition. Interlobe clearance (0.160 mm), radial clearance (0.180 mm), and axial (end plate) clearance (0.120 mm) are all modified to 0.144 mm, 0.162 mm, and 0.108 mm, respectively after couple of iterations. Now, the performance predictions are made with the adjusted clearances using the original flow coefficient model (Model 1) and the suggested flow coefficient models (Models 2 and 3), and the results are compared to the experimental data as shown in Table 2. Figure 4 shows the comparison of prediction flow with respect to the experimental flow (for adjusted clearances) using all three models.

The Model 1 under predicts the flow up to 8.67% (at 6000 RPM), while Model 2 and Model 3 predicts the flow within ± 1% (Figure 4). The flow prediction using Model 2 and Model 3 are closer to the experimental results in comparison to the flow prediction using Model 1, as clearances are adjusted to the operating clearances. The power prediction using all the three models are within ± 1% in comparison to the experimental power. The Model 1 over predicts the specific power up to 5.4% while Model 2 and Model 3 under predicts the specific power up to 1% The improvement in the specific power using Model 2 and Model 3 is because of better flow prediction with almost same power consumption. The flow predictions using Model 2 and Model 3 are higher up to 9.40% and 10.40% respectively in comparison to the flow prediction using Model 1. Similarly, the specific power consumption prediction using Model 2 and Model 3 are lower up to 8.22% and 8.81% respectively in comparison to the specific power prediction using Model 1.

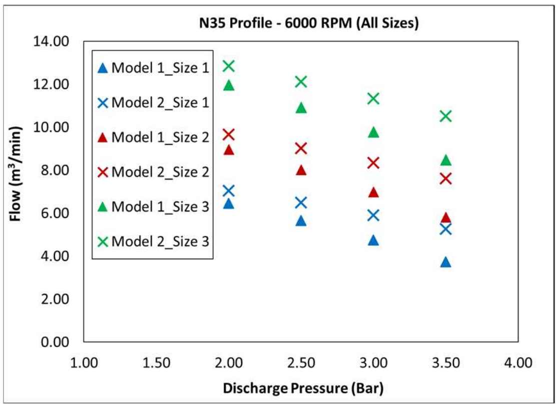
4.2 Case 2- Performance prediction for three different sizes of oil free twin-screw compressors

In Case 2 (for varying size, speed, and pressure conditions), performance predictions are made using Models 1 and 2, and the results for flow, power, specific power, adiabatic efficiency, and volumetric efficiency are compared. Model 3 is not used for prediction in Case 2 because the results of Models 2 and 3 are nearly identical (as observed in Case 1).

Figure 5(a) shows the comparison of prediction results of the flow using Models 1 and 2 (for all the compressor sizes) in terms of the rotational speed for 2.00 bar discharge pressure. The results show that the Model 2 predicts the higher flow in comparison to the original model for all the speed and size combinations. The similar prediction trend (for both the models and sizes) observed for the discharge pressure of 2.50, 3.00 and 3.50 bar also. Figure 5(b) shows the comparison of flow results using both the models (for all the compressor sizes) in terms of the discharge pressure for 6000 RPM of compressor speed. The results show that Model 2 predicts the higher flow in comparison to Model 1, which is applicable to other rotational speeds as well.



(a)



(b)

Figure 5: The flow comparison for both the models (a) in terms of rotational speed (at 2 bar) and (b) in terms of discharge pressure (at 6000 RPM)

The results show that Model 2 predicts lower specific power in comparison to Model 1 for all the speed, discharge pressure, and size conditions, which is obvious because Model 2 offers more flow with nearly the same power. The higher flow using Model indicates that in the proposed model the flow coefficients are lower in comparison to the Model 1, which results in lower leakages, which in turns give higher discharge flow. The Model 2 predicts higher flow for all the pressure, speed, and size conditions (in the range of 4.81% to 40.95%) in comparison to the flow prediction using the Model 1. For the same size and same discharge pressure, the difference between the prediction results (Model 2 in comparison to the Model 1) decreases with increase in the rotational speed, which is because of lower leakages at higher rotational speed of the compressor. Similarly, for the same size and same rotational speed, the difference in the prediction results increases with the increase in the discharge pressure, which is because of higher leakages presences at increased pressure. The similar trend observed for all the sizes of the compressors. The difference in the results decreases with the increase in the size of compressor for the same discharge pressure and rotational speed, and the reason is that the package discharge flow increases significantly more in comparison to the increase in the leakage flow for increased size of the compressor.

4.3 Deviation in results for all models

Figure 6 shows the comparison of flow prediction of Model 2 with respect to the Model 1 for all the pressure, speed, and size conditions. Model 2 predicts the higher flow and lower specific power for all the results in comparison to the Model 1. In comparison to the Model 1 flow results, the Model 2 flow results for Size 1, Size 2, and Size 3 are higher in the ranges of 5.57 % to 40.95 %, 5.14 % to 30.91 %, and 4.81 % to 23.99 %, respectively.

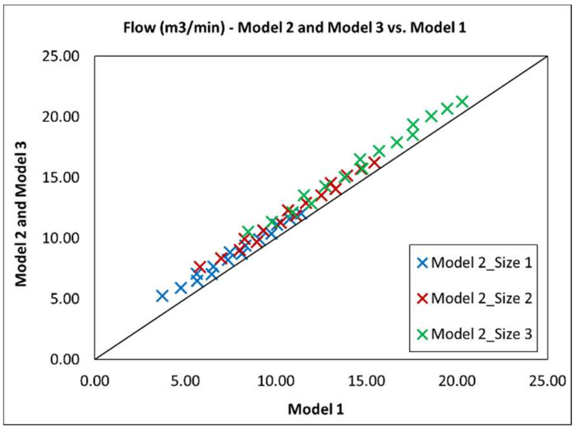


Figure 6: The flow comparison of Model 2 in comparison to the Model 1 (for all discharge pressure, speed and size)

5. CONCLUSIONS

The flow coefficient correlations from Models 2 and 3 are derived and incorporated in the SCORG^{TM} software to estimate the performance of an oil-free twin-screw compressor. The results are compared to the experimental results from the previous work and the results obtained using the original flow coefficients in Model 1. Following are the key findings from the study:

- The Model 1, Model 2 and Model 3 found under predicting the flow up to 15.13%, 5.32% and 4.27% in comparison to the experimental results with design (cold) clearances. The adjusted operating clearances were derived by matching the Model 2 and Model 3 results with the experimental results. The Model 1 is found to under predict the flow for up to 8.67% when calculated with the operating clearances.
- The performance predictions extended to various speeds, discharge pressures and sizes of the compressor and the Model 2 flow results were higher in the range of 4.81% to 40.95 in comparison to the flow results using Model 1. The difference between the prediction results of Model 2in comparison to Model 1 are decreasing with the

increase in the rotational speed for the same discharge pressure. Similarly, the difference in the prediction results increases with the increase with the increase in the discharge pressure for the same speed. Models 2 and 3 predict similar results.

- The power consumption for all the results using all the models and different cases was found in the range of ±1% .

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NOMENCLATURE

A	Leakage area	[m^2]
h	Clearance height	[m]
$\dot{m}$	Leakage flow rate	[kg/sec]
P	Perimeter	[m]
p	Pressure	[bar]
PR	Pressure ratio	[-]
Re	Reynolds Number	[-]
v	Velocity	[m/sec]
Φ	Flow coefficient	[-]
γ	Specific heat ratio	[-]
β	Diameter ratio	[-]
δ	Density	[kg/m^3]
SCORG	Screw Compressor Rotor Grid Generator	
CFD	Computational Fluid Dynamics	

SUBSCRIPT

1	Downstream
2	Upstream

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