Estimation of radial shaft seal, oil drag and windage loss in twin screw oil injected compressor

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Abstract. The objective of this paper is to study the mechanical power loss arising from radial shaft seal, oil drag, and windage within the twin-screw oil-injected screw compressor. The previous papers in the series presented the evaluation of power loss due to the gears and bearings where it was found that the calculated total power loss was underestimated when compared to the experimental results for different sizes of compressors and different operating conditions. This indicates that few additional contributing elements were missing and needed due consideration to enhance the prediction. Literature review of the basic fluid dynamics and transmission gears confirms that oil drag loss and windage loss can also contribute to the power loss considerably. This paper presents a review of a few methods for the prediction of the radial shaft seal, oil drag, and windage loss. Predictions with the use of these methods and their comparison with the experimental results are analysed in this paper. Experimental results of three different sizes of the compressors tested in laboratory conditions were taken as reference for the comparison at different operating speeds and pressure ratios. The analysis helps to understand the contribution of each element of the power loss which can help a designer to optimally design a screw compressor.

Keywords: Oil drag, Radial shaft seal, Windage loss, Screw Compressor

Nomenclature

| Symbol | Description | Unit |
| --- | --- | --- |
| B | width of rotor tip | m |
| Bg | body forces | N |
| d | diameter | m |
| F | gear width/rotor length | m |
| M | gear module/ratio of pitch to number of lobes | m |
| n | rotational speed | rpm |
| Pdrag | power loss in drag | W |
1. Introduction

With the increasing demand for more energy-efficient machines, one of the ideas in the screw compressor to focus on is to reduce the power loss. To do this, one first needs to understand the elements of the mechanical power loss in the screw compressor and their contribution to different sizes of the compressors at different operating conditions. Although a few methods are available in the literature to independently predict the individual element’s power loss, a comprehensive model suitable for the screw compressor application is unavailable. Realisation of major contributors of power loss elements in the screw compressor can help a designer to optimally design the screw compressor as per the need.

A Sankey diagram representing elements of power loss in a screw compressor package is shown in Figure 1. Only a part of the total input power to the screw compressor package is available for the compression where the remaining part is lost in overcoming prime mover loss, transmission loss, and bearing, seal, and oil drag loss within the bare compressor.

![Sankey diagram of power loss in screw compressor package](image)

Referring to previous studies in this series on bearing power loss estimation, the study presented here includes power loss estimation of radial shaft seals, oil drag, and windage loss. A comparison of the SKF and Harris model for the estimation of bearing power loss has been presented in Abdan et al. (2019) [1]. Frölich et al. (2104) [2] and Engelke et al. (2011) [3] have presented semi-analytical and
experimental studies on the radial shaft seal power loss. A rotating Couette flow model along with a

correlation proposed by Cicchitti et al. (1959) [4], for calculation of the two-phase flow properties is

used for the prediction of the oil drag loss. Figure 2 shows a general arrangement of the shaft seal

position and oil injection for an oil-flooded, twin-screw compressor block (Image: Kirloskar

Pneumatic Company Limited).

Dawson (1984) [5] presented an approximation for the calculation of windage loss in power

transmission gears which is referred to and adopted with a modification for prediction of windage loss

in screw compressors. The experimental results of three different sizes were taken as a reference for

comparison and results of the proposed method were compared with the experimental results.

2. Elements of mechanical loss

2.1 Bearing

Rolling element bearings are preferred over journal bearings because of their low starting friction,

intermittent operation and are more adaptable for handling misalignment between shafts.

The Harris model for bearing power loss estimation shows better agreement with the experimental observations

as presented by Tu (2016) [6]. According to the Harris model, the frictional loss of the ball bearings is classified

into two categories, one due to load and another due to lubricant. For roller bearings, an additional loss because of sliding

between roller ends and ring flange is taken into account as given in Haris et al. (2006) [7]. For ball bearings,

total frictional torque is divided into two categories, load-dependent, and load-independent. The details of the model

and approximations are presented in Abdan et al. (2019) [1] with verification.

2.2 Shaft seal loss

The radial shaft seal is an integral part of the screw compressors whose function is to prevent oil leakage out of

the compressor. Since, the lip of the stationary seal tightly sits on the rotating shaft, thus creating friction and ultimately loss of power. Fröhlich et al. (2104) [2] presents a semi-analytical approach for the calculation of contact temperature and an empirical approach for the calculation of friction. The experimental measurements of the influence of elastomeric lubricant combination on the operating performance of radial shaft seals are presented by Engelke et al. (2011) [3] with an algorithm to estimate the seal frictional loss.
The method to calculate the radial shaft seal power loss is the combination of iterative and semi-analytical approaches. The value of friction coefficient, radial force on the seal and seal-shaft contact surface width is assumed and this assumption is validated by Engelke et al. (2011) [3]. The shaft diameter, shaft surface roughness, rotating speed and oil temperature are used as input parameters. In the first iteration, the friction torque value is calculated for the assumed temperature while for the next iteration, the seal surface temperature is raised by 16.5 K for 1 W/mm² sealing contact area as presented in Engelke et al. (2011) [3]. Once the friction torque for raised surface temperature is calculated, the difference in the last two time steps for friction torque is calculated and compared with the convergence criteria. A flow chart for the calculation of shaft seal power loss is given in Figure 3.

There are two assumptions in the model, one is the radial force per unit circumferential length and the second one is the coefficient of friction. The experimental measurements have indicated the values of the radial force per unit circumferential length and the coefficient of friction for combinations of the seal material and the seal-shaft material respectively by Engelke et al. (2011) [3].

2.3 Oil drag loss
The oil flow in rotor tip-housing clearance can be equated to rotating Couette flow where flow is dominated by viscous effects and inertial effects are negligible. In a Cylindrical coordinate system, this is compared to the flow in annular space. A two-dimensional Navier-Stokes equation that can be solved exactly by analytical techniques with a number of significant assumptions is as follows.

\[ \rho \frac{\partial u_\theta}{\partial t} + u_\theta \frac{\partial u_\theta}{\partial r} + \frac{u_\theta}{r} \frac{\partial u_\theta}{\partial r} + u_r u_\theta = - \frac{1}{r} \frac{\partial p}{\partial \theta} + B_\theta + \mu \left( - \frac{1}{r} \frac{\partial u_\theta}{\partial r} + \frac{u_\theta}{r^2} \frac{1}{r^2} \frac{\partial^2 u_\theta}{\partial \theta^2} + \frac{2}{r^2} \frac{\partial u_\theta}{\partial \theta} \right) \] (1)

While assuming the flow is steady, body forces are negligible, flow is incompressible and pressure gradient is negligible, using continuity equation, the above equation reduces to the following form

\[ r^2 \frac{\partial^2 u_\theta}{\partial r^2} + r \frac{\partial u_\theta}{\partial r} - u_\theta = 0 \] (2)

This is a second-order, ordinary differential equation. After applying the following boundary conditions, at \( r = r_i \) (on rotot tip), \( u_\theta = V_i \), and at \( r = r_o \) (on housing inner surface) and \( u_\theta = 0 \), the equation reduces to

\[ u_\theta = \frac{V_i r_i}{r_o - r_i} \left( \frac{r_o^2}{r_i} - r \right) \] (3)

This is the velocity profile in the clearances. The shear stress can be calculated by differentiating the velocity.

\[ \tau_i = -\frac{2\mu V_i r_i^2}{(r_o^2 - r_i^2) r_i} \] (4)

The product of shear stress, area, and speed gives the power loss through drag.

\[ P_{\text{drag}} = \tau Br_i \omega \] (5)

2.4 Windage loss
The force created by the friction between the air and the object is referred to as windage. The higher the relative velocity between the rotors and the air, the higher is the frictional force and ultimately the loss. An equation for approximating the windage loss in gears is given by Dawson (1984) [5]. The windage power loss is a function of the rotational speed of the gear (\( n \)), gear width (\( F \)), tooth module (\( M \)), the amount of oil mist present inside the casing, and the diameter of the gears.
The same equation has been used to predict the windage loss through the compression chamber of the screw compressor. The factor $\phi$ represents the oil mixture function while the factor $\lambda$ represents a gearbox space function. However, the predictions of the model did not match in good agreement with the experimental results. Hence, the equation proposed by Dawson (1984) [5] is modified to make it suitable for the screw compressor application. The modified form of the equation is as below:

$$P_{\text{windage, n}} = n^2 \left(0.16 d^{3.9} + d^{2.9} F^{0.75} M^{1.15}\right) \times 10^{-20} \phi \lambda \quad (6)$$

The results of the equation for different sizes of the compressor are presented in further sections.

3. Results and Discussion

Three sizes of the twin-screw, oil-injected compressors used for 15-30 kW (size 1), 37-55 kW (size 2), and 75-160 kW (size 3) with “N” rotor profile, 4/5 lobe combination, and L/D=1.55 were considered for the calculation. These compressors were tested in laboratory conditions at different pressure ratios for the measurement of shaft power and airflow.

Harris model is used for the calculation of power loss through the anti-friction bearings. The results and verification of the calculation for a different set of bearings were presented by Abdan et al. (2019) [1]. With the use of methods mentioned in the previous section for calculation of power loss through shaft seal, oil drag, and windage loss, the total power loss through the above three sizes of screw compressors was estimated.

Table 1 lists the anti-friction bearings used in the compressors with their location. The radial and axial loads generated during compression were taken from DISCO (Design Integration for Screw Compressors), a software package developed through continuous research and development at City, University of London, UK.

| Location                | Size 1         | Size 2         | Size 3         |
|-------------------------|----------------|----------------|----------------|
| Male-Radial-Suction     | NU 205 ECP     | NU 203 ECP     | NU 205 ECP     |
| Female-Radial-Suction   | NU 203 ECP     | NU 2304 ECP    | NU 1009 ECP    |
| Male-Radial-Discharge   | NU 205 ECP     | NU 2207 ECP    | NU 211 ECP     |
| Female-Radial-Discharge | NU 205 ECP     | NU 2207 ECP    | NU 211 ECP     |
| Male-Axial-Discharge    | 7305 BEP       | 7407 BEP       | 7311 BECBP     |
| Female-Axial-Discharge  | 7205 BEP       | 7207 BEP       | 7211 BEP       |

The radial shaft seals of sizes 25x32x4 mm, 32x42x4 mm, and 60x80x8 mm were used for sizes 1, 2, and 3 respectively. It is assumed that the friction coefficient between seal and shaft is 0.35, the average radial force on seal lip is 145.9 N/m based on experimental measurements presented by Engelke et al. (2011) [3] while the shaft surface roughness is 0.8 micrometers. Vogel model is used for the calculation of temperature-dependent viscosity as presented by Knežević et al. (2006) [8].

A correlation proposed by Cicchitti et al. (1959) [4] was used for the evaluation of the two-phase thermo-physical properties of the air-oil mixture. A constant oil-air mass ratio of 4 was assumed with an oil density of 950 kg/m³ and viscosity of 68 cSt at 40°C. The air mass flow rate varies with respect to the rotor shaft speed and operating pressure ratio which is obtained from DISCO. From the total peripheral area of the rotor housing, the cusp area was deducted using the Cosine law of trigonometry. Similar to the Couette flow model in a Cylindrical coordinate system used for calculation of drag loss in radialclearances between the rotor and the housing, a Couette flow model in the Cartesian coordinate system is used for the calculation of drag loss in the axial clearances.

The experimental results were obtained for three sizes, sizes 1, 2, and 3, at different pressure ratios (6.5-12.5) and rotor tip speeds (10-40 m/s) at laboratory conditions. The compressor inlet and outlet temperatures were measured by Platinum Resistance Thermometers with errors within ±0.50°C. Oil
temperatures were measured by K-type thermocouples with errors within ± 1.0°C. All pressures were measured with transducers with errors within ± 0.6%. The compressor speed was measured by a shaft encoder with ± 2.7% error. The compressor torque was measured by the torque meter with a strain gauge transducer with an accuracy of ± 0.25%. A comparison between the experimental results, DISCO estimation, and calculation from the above-mentioned methods is plotted in Figure 4.

![Figure 4: Comparison of total power loss in compressor size 1 (a), 2 (b) and 3 (c)](image)

Although total shaft power was measured using the torque meter at laboratory conditions, the experimental total mechanical power loss is calculated from the measured shaft power by deducting the indicated power as predicted by the DISCO programme. Figure 4 clearly shows that the prediction for total power loss with the use of the proposed method matches closely with the experimental results for different sizes of screw compressors at a pressure ratio of 6.5. However, for a pressure ratio of 12.5, the proposed model slightly under-predicts the mechanical loss as compared to the experimental measurements as can be seen in Figure 4 (b) and (c).

Upon quantification of the results, the effect of shaft speeds and pressure ratios on the individual elements of the power loss is presented in Figure 5 and Figure 6.
Effect clearly shows that the anti-friction bearings are the main contributors to the power loss followed by oil drag loss, windage, and radial shaft seals in decreasing order of their contribution. However, with the increase in the size of the compressor, windage loss becomes more dominant and can contribute substantially to the power loss.

4. Conclusions
Different elements contributing to the power loss within the air screw compressor with oil injection and twin screw rotor configuration have been studied. The elements are anti-friction bearings, shaft seal, oil drag, and windage. The details of different power loss prediction models for anti-friction
bearings and their comparison with available experimental results have been presented in the previous work along with a recommendation to use the Harris model.

A combination of semi-analytical approach and iterative procedure proposed by Frölich et al. (2014) [2] and Engelke et al. (2011) [3] is used for the prediction of power loss through radial shaft seal. For the prediction of power loss through oil drag in radial and axial clearance gaps, a rotating Couette flow model is used with suitable boundary conditions. A modified Dawson’s correlation is used for the prediction of windage loss. With the use of the above-mentioned models, total power loss prediction for three different sizes of the compressor, 15-30 kW, 37-55 kW, and 75-160 kW at different speeds and operating pressures found to be in good agreement with the experimental data as indicated in Figure 4.

A parametric analysis shows that anti-friction bearings are the biggest contributors to the power loss followed by oil drag loss, windage loss, and the radial shaft seal. The percentage contribution of anti-friction bearing reduces while that of oil drag loss, windage, and radial shaft seal loss increase with respect to the tip speed as shown in Figure 5. The trend reverses when the pressure ratio is increased for the same tip speed as seen in Figure 6.

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