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Analysis of rolling bearing power loss models for twin screw oil injected compressor

S Abdan^{1,2*}, N Stosic¹, A Kovacevic¹, I Smith¹ and N Asati²

¹ Centre for Compressor Technology, City, University of London, London, U.K.

² Kirloskar Pneumatic Company Limited, Pune, India.

E-mail: suraj.abdan@city.ac.uk, n.stosic@city.ac.uk, a.kovacevic@city.ac.uk, i.k.smith@city.ac.uk, neeraj.asati@kpcl.net

*Corresponding author

Abstract. The mechanical losses inside a screw compressor limit the performance of the compressor in terms of efficiency. These losses arise due to relative motion between elements inside the screw compressor. The estimation of mechanical losses predicted in the literature is around 10-15% of the total shaft power. One of the elements which contribute significantly to these losses is rolling element bearings. There are numerous mathematical models available which predict power losses in the rolling bearings.

The objective of this paper is to study different models to predict power loss for rolling bearings and to predict the power losses for the bearings used for oil injected, twin screw compressor. A comparison between different power loss models for different operating conditions of compressor is also presented in this paper and results of analysis are compared with available experimental observations. The analysis helps to determine suitable power loss model for different operating conditions and more realistic predictions of the power losses. This allows designers for more accurate estimation of the performance of screw compressors.

List of symbols

B	bearing width	mm
C_s	bearing basic static load rating	N
C_w	variable	
d_m	bearing mean diameter	mm
F_a	axial load acting on bearing	N
F_r	radial load acting on bearing	N
F_s	bearing static equivalent load	N
F_β	equivalent load depending on magnitude and direction of applied load	N
f_l	factor depending on the bearing design and relative bearing load	
f_i	variable	
G_{rr}	variable for rolling frictional moment	



G_{sl}	variable for sliding frictional moment	
K_{ball}	constants for ball bearing	
K_{roll}	constants for roller bearing	
M	total frictional moment	Nmm
M_f	bearing friction torque due to roller end-flange load	Nmm
M_l	bearing friction torque due to load	Nmm
M_v	bearing friction torque due to lubrication	Nmm
M_{drag}	frictional moment of drag losses, churning, splashing etc.	Nmm
M_{rr}	rolling frictional moment	Nmm
M_{seal}	frictional moment of seals	Nmm
M_{sl}	sliding frictional moment	Nmm
n	rotational speed	rpm
P_{loss}	power losses	W
R_S	variable	
V_M	drag loss factor	
ν_o	lubricant kinematic viscosity	centistokes
Greek letters		
α	contact angle	rad, °
Φ_{ish}	inlet shear heating reduction factor	
Φ_{rs}	kinematic replenishment/starvation reduction factor	
ν	actual operating viscosity of oil or the base oil of the grease	mm ² /s
μ_{sl}	sliding friction coefficient	

1. Introduction

High isentropic efficiencies for low to moderate pressure range, together with purely rotary motion are making screw compressors popular. However, with some sources of frictional loss and relatively higher loads acting on bearings compared to turbo-compressors, the adiabatic efficiencies of these are limited [1]. The frictional losses of the bearings are major contributors to overall mechanical loss and hence, an accurate prediction of it is necessary to understand capacities of machines developed. Although there are some models available to predict the frictional losses of the bearings, the predictions not go well with the experimental observations for certain operating conditions. A better understanding of available models to predict bearings frictional loss along with their comparison for wide range of screw compressor operating conditions is, therefore, essential for more realistic predictions.

The study presented here includes analysis of available frictional loss models for different types of bearings; and the comparison with experimental results. The bearings used in this analysis are selected from one of the oil injected, twin screw air compressors with working pressure of 8.5 bar absolute.

2. Bearing frictional loss models

Bearings are one of the key elements in screw compressors whose main function is to transmit the loads from rotating elements; i.e. screw rotors; to the stationary elements; the housing. During transmission, the rotating elements of the bearings; rollers in case of roller bearing and balls in case of ball bearing; rotate inside the raceway increasing temperature because of friction between them. This temperature is a consequence of frictional losses which is cooled by lubricant continuously circulated through them.

A number of operational and non-operational factors affect the bearing friction. The friction between interacting surfaces which are in relative motion always changes. Hence, it is difficult to predict exact frictional loss values. However, there are certain models available to predict losses which estimate loss values close to experimental observations. Based on available experimental results, two models; Harris and SKF; are selected for analysis and their predictions are compared with available experimental observations.

According to Harris model, the frictional losses of the ball bearings are classified into two categories, namely; due to load and due to lubricant viscosity, while that for the roller bearing an additional frictional loss because of sliding between roller ends and ring flange is taken into account. In SKF model, the total frictional loss is calculated as a sum of losses due to rolling, sliding, seal and drag frictional moment.

2.1 Harris model

As explained in previous section, for ball bearings, total frictional torque is divided into two categories, load dependent and load independent [2].

2.1.1. *Frictional torque due to load.* The load dependent frictional torque is given by following equation

$$M_l = f_l F_\beta d_m \quad (1)$$

The factor f_l which is dependent on bearing design and load is expressed as follows

$$f_l = z \left(\frac{F_s}{C_s} \right)^y \quad (2)$$

where, F_s and C_s are the static equivalent load and the basic static load rating respectively. The factors z and y are defined based on type of ball bearing and nominal contact angle between balls and raceways. The equivalent load, F_β , depends on magnitude and direction of the applied load.

$$\begin{aligned} F_\beta &= \max(0.9F_a \cot \alpha - 0.1F_r, F_r) \dots\dots\dots \text{ball bearings} \\ &= 3F_a - 0.1F_r \quad \dots\dots\dots \text{deep-groove ball bearings} \\ &= F_a \quad \dots\dots\dots \text{thrust bearings} \\ &= \max(0.8F_a \cot \alpha, F_r) \quad \dots\dots\dots \text{cylindrical roller bearings} \end{aligned} \quad (3)$$

2.1.2. *Frictional torque due to lubricant viscosity.* For the lubricated bearings, when lubricant occupies the free space between rolling elements and raceways, it gives resistance to the motion generating frictional loss. This resistance is a function of type of lubricant and its thermo-physical properties, lubricant level, speed and temperature of operation.

$$\begin{aligned} M_v &= 10^{-7} f_o (v_o n)^{2/3} d_m^3 \quad v_o n \geq 2000 \\ &= 160 \times 10^{-7} f_o d_m^3 \quad v_o n < 2000 \end{aligned} \quad (4)$$

f_o is a factor depending of type of bearing and method of lubrication while, v_o is kinematic viscosity of lubricant in centistokes.

2.1.3. *Frictional torque due to roller end-ring flange sliding.* Few designs of cylindrical roller bearings have flanges on either inner ring or outer ring. When these bearings are subjected to axial load in addition to radial load, they carry thrust load and frictional torque between roller end-ring flange sliding as given below

$$M_f = f_f F_a d_m \quad (5)$$

The value of factor f_f depends on type of cylindrical roller bearing, lubricant and lubrication method.

2.1.4. Total frictional torque. An estimate of total friction torque is given as sum of all frictional moments which can be expressed as follows

$$\begin{aligned} M &= M_l + M_v \quad \text{.....ball bearings} \\ &= M_l + M_v + M_f \quad \text{.....roller bearings} \end{aligned} \quad (6)$$

2.2 SKF model

A model developed by SKF, calculates frictional moments based on advanced computational methods [3]. The total frictional moment in a rolling bearing consists of four elements; the rolling frictional moment, sliding frictional moment, frictional moment of the seals and frictional moment caused by the drag losses, churning and splashing. It is defined as follows:

$$M = M_{rr} + M_{sl} + M_{seal} + M_{drag} \quad (7)$$

The rolling frictional moment is calculated by using the following equation:

$$M_{rr} = \phi_{ish} \phi_{rs} G_{rr} (vn)^{0.6} \quad (8)$$

The equation indicates that the rolling frictional moment is a function of the inlet shear heating reduction factor, kinematic reduction factor caused by replenishment/starvation, viscosity and speed, together with several other geometrical and bearing load variables. The sliding frictional moment is the function of sliding friction coefficient and can be expressed as follows:

$$M_{sl} = G_{sl} \mu_{sl} \quad (9)$$

The effect of the full-film lubrication and mixed lubrication conditions can be approximated by evaluation of the sliding friction coefficient. The drag losses caused by rotation of the bearings inside the oil bath can influence total frictional moment. Not only the bearing speed, but also the oil viscosity, oil level, size and shape of the oil reservoir affect the drag loss. The drag loss in the oil bath of the ball bearings can be calculated as follows:

$$M_{drag} = 0.4V_M K_{ball} d_m^5 n^2 + 1.093 \times 10^{-7} n^2 d_m^3 \left(\frac{nd_m^2 f_t}{\nu} \right)^{-1.379} R_s \quad (10)$$

For roller bearing, the same can be estimated as follows

$$M_{drag} = 4V_M K_{roll} C_w B d_m^4 n^2 + 1.093 \times 10^{-7} n^2 d_m^3 \left(\frac{nd_m^2 f_t}{\nu} \right)^{-1.379} R_s \quad (11)$$

Once total frictional moments could be calculated, the power loss from the bearing can be defined using following expression [4]

$$P_{loss} = 1.05 \times 10^{-4} M n \quad (12)$$

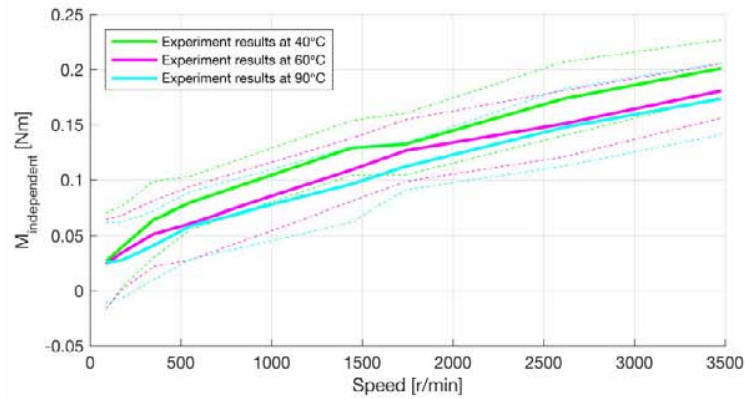
3 Programme results of models and discussion

A comparison between Harris model, SKF model programme results [5] and available experimental results is presented in this section. The analysis helps for better prediction of bearing frictional losses by selecting appropriate model for particular operating condition. Firstly, the frictional losses of roller bearings are analysed followed by the same with ball bearings. The analysis is focused on cylindrical roller bearings and angular contact ball bearings as these types of bearings are most commonly used in twin screw oil injected compressors.

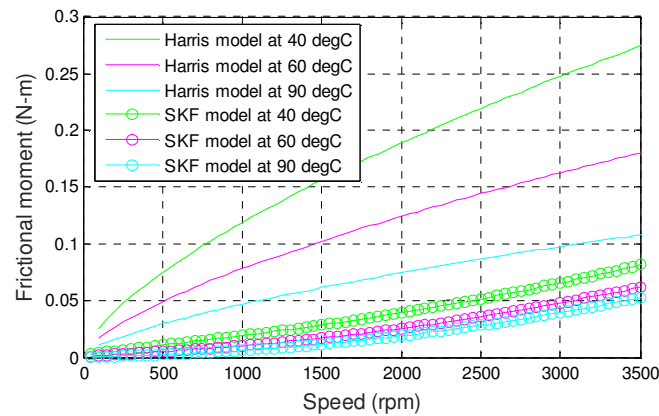
3.1 Roller bearings

Minghui Tu presents experimental results for cylindrical bearing, NJ 406 [6]. The parameters varied during the testing are radial loads (up to 3750 N), operating speed (up to 3500 rpm), lubricant level, lubrication types and methods.

3.1.1 Load independent losses. The load independent losses are the losses because of lubrication and lubricant properties.



(a)



(b)

Figure 1. Load independent friction torque from the experimental data and output of existing models at different oil temperature (PAO std oil and 103 mm oil level).

The results shown in Figure 1(a) are experimental results for oil operating viscosity of 18.6 cSt at 90°C while those in Figure 1(b) are from Harris and SKF model. As can be seen from above figures, the frictional torque estimated by Harris model is much closer to the experimental values compared to SKF model estimation. The analysis is done for PAO oil operating at oil temperatures 40°C, 60°C and 90°C.

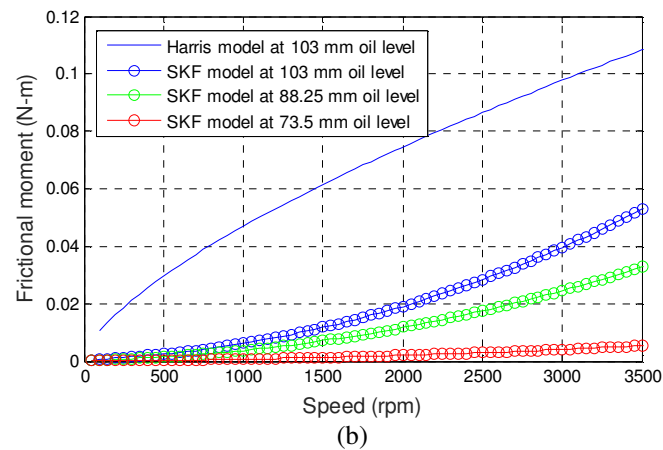
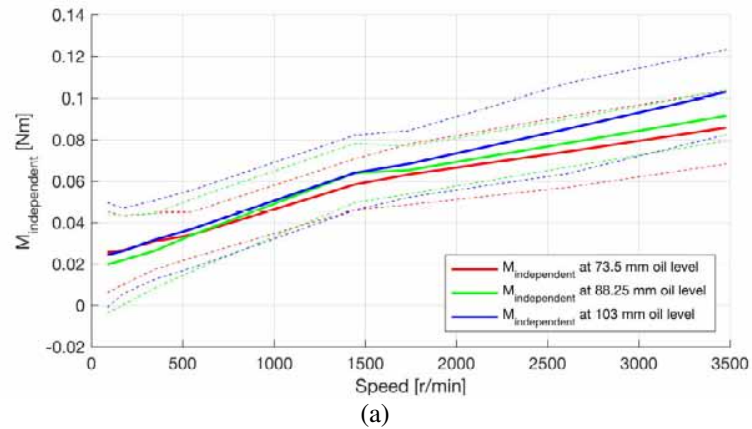


Figure 2. Load independent friction torque from experimental data and output of existing models for different oil levels.

Higher levels of lubricant impose higher resistance to the motion. Hence, the effect of increase of oil levels is as shown Figure 2, where, the load independent frictional torque increases marginally with oil level. The oil with 18.6 cSt viscosity at 90°C is used for analysis. The Harris model does not take into account the effect of lubricant level in frictional moment estimation while SKF model does show increase in the frictional torque with the increase in lubricant level. Even though the experimental characteristics match with SKF model, the Harris model predictions for magnitude of load independent frictional torque matches reasonably well with experimental results.

3.1.2 Load dependent losses. The load dependent friction torque is divided into rolling friction and sliding friction according to SKF model. Similar to load independent frictional loss analysis as presented in previous section, this section presents estimation of load dependent frictional torque between Harris model, SKF model and the comparison with experimental results.

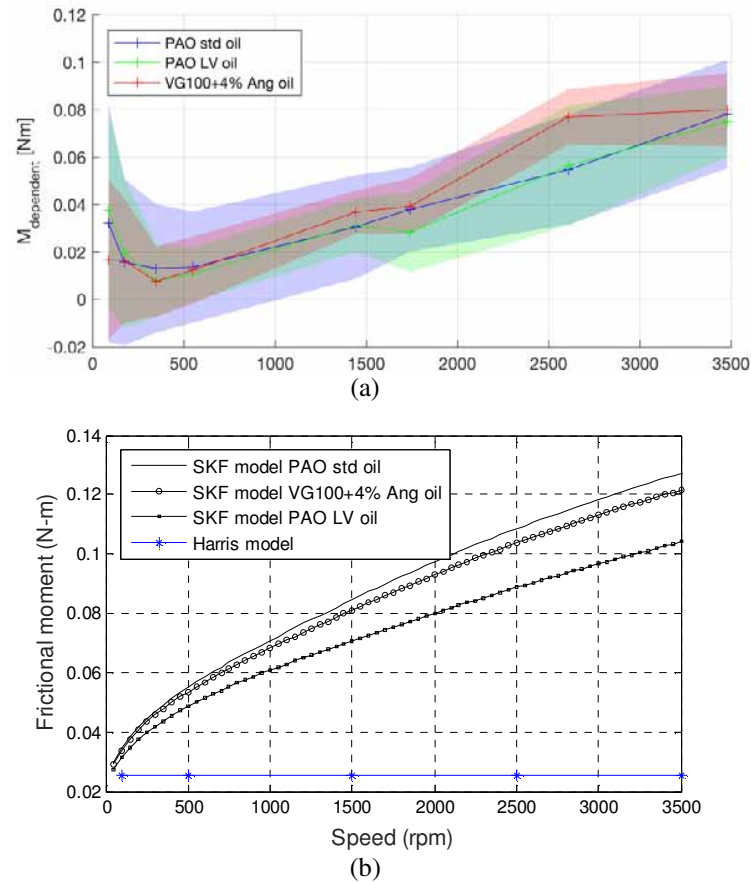


Figure 3. Load dependent friction torque from experimental data and output of existing models at different oil types (103 mm oil level, LS5 and 90°C).

The effect of type of lubricant on load dependent frictional moment is shown in Figure 3. The respective oil viscosities of PAO std, PAO LV and VG100+4% Ang oil are 18.6 cSt, 12.5 cSt and 17 cSt at 90°C with radial load of 1405 N. Since, the Harris model does not consider effect of lubricant viscosity and speed of rotation; it remains constant for load dependent frictional moment loss. However, increase in lubricant viscosity increases frictional torque in SKF model as could be seen in Figure 3(b). This behaviour is similar to the experimental observations as shown in Figure 3(a).

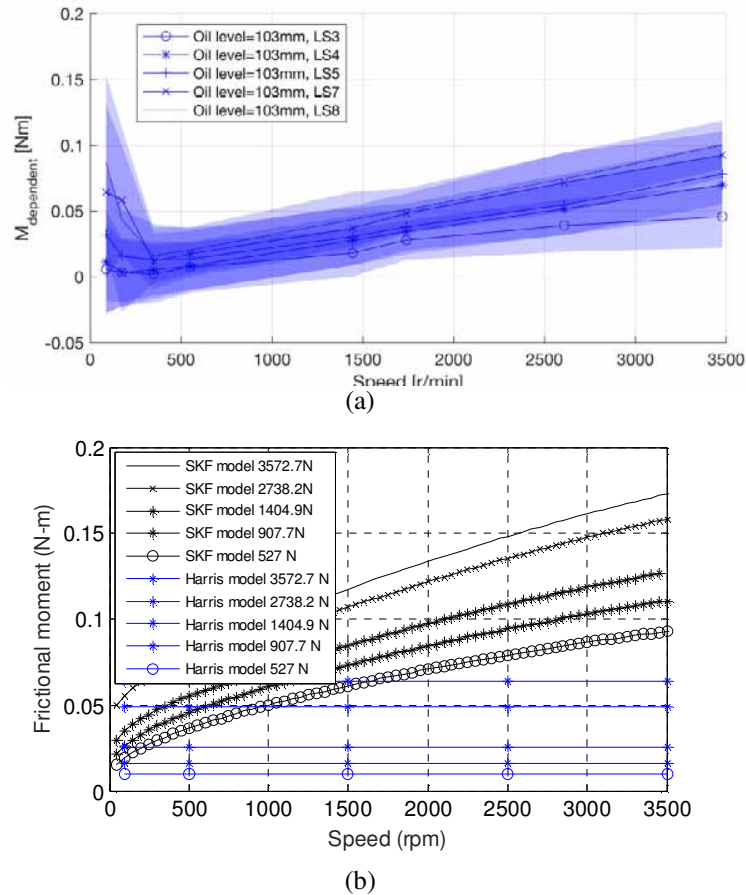


Figure 4. Load dependent friction torque from experimental data and output of existing models at different load (103 mm oil level, PAO std oil and 90°C)

Increase in load on bearing and speed increases frictional torque as shown in Figure 4(a). This characteristic matches with SKF model prediction similar to previous load dependent analysis. The Harris model also shows increase in frictional torque with increase in load but does not change with speed as presented in Figure 4(b).

3.2 Ball bearings

Similar to the analysis of roller bearings, this section presents analysis of model results of total frictional torque for ball bearings and comparison with available experimental results. Mircea Gradu presented an experimental evaluation of frictional torque for ball bearings used for transmission shaft [7].

The bearing used in this analysis is 7305 which is an angular contact ball bearing. Two cases are analysed by Gradu with different transmission torques. For case 1, where transfer shaft is transmitting a torque of 100 Nm, the radial loads acting on floating bearing is 3500 N and axial load is 1000 N and for fixed bearing the radial load is 4250 N and axial load is 3500 N. For torque transmission of 200 Nm, the floating bearing experiences radial load of 7000 N with axial load of 2000 N while fixed bearing experiences radial load of 8500 N and axial load of 7000 N. Total frictional torque is summation of individual frictional torques of floating and fixed bearing. The oil viscosity considered for calculation is 38 mm²/s.

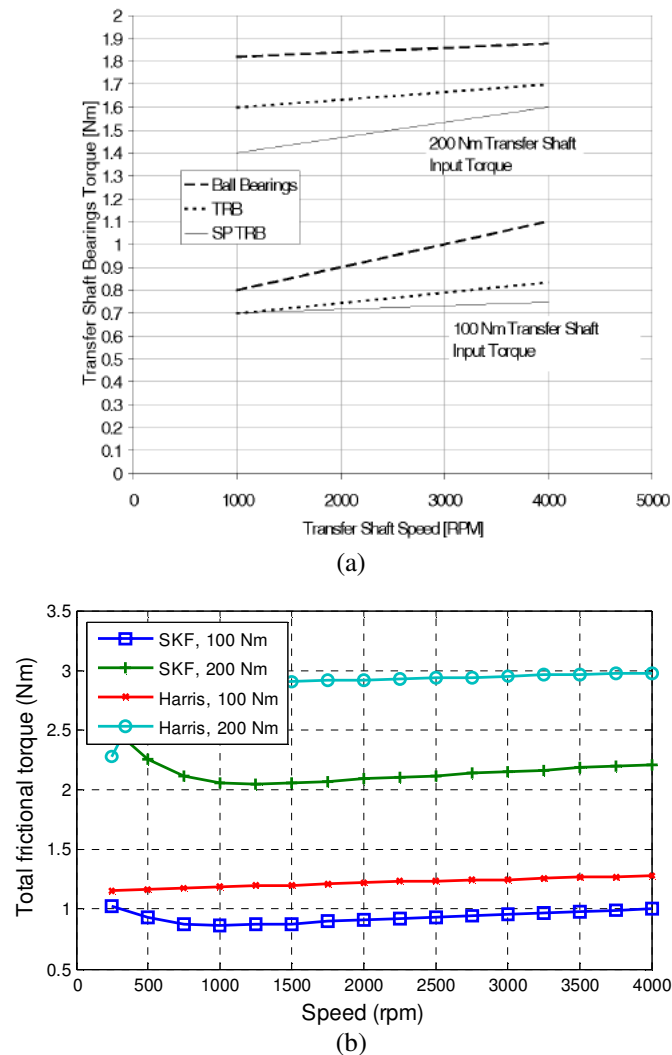


Figure 5. Frictional torque of ball bearing from experimental data and output of existing models at different loads

The analysis and comparison shown in Figure 5 confirms that SKF model results match well within limits with experimental results for ball bearings compared to Harris model.

4 Conclusions

The results of frictional torque for roller bearings and ball bearings from Harris model and SKF model programme are analysed and compared with available experimental results. For roller bearings, the Harris model shows better estimation for load independent frictional torque. However, for load dependent frictional case, even though SKF model results show slightly overestimated numbers, the power loss characteristics are matching with experimental observations. It is therefore necessary, for more precise prediction, some changes in factors defined for roller bearing load dependent frictional torque analysis.

In case of ball bearings, the SKF model results are in good agreement with experimental results compared to the Harris model which overestimates total frictional torque. Overall, the combination of

Harris model and SKF model for roller as well as ball bearing and load independent and dependent case should be used for accurate prediction of total frictional loss of bearings for screw compressor.

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