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Identification and analysis of screw compressor mechanical losses

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Abstract. Screw compressors are compact machines, used for a wide range of applications where gases or vapours are required to be delivered at moderate pressures with high efficiency and reliability. They are most effective when the compressed medium requires power inputs, approximately in the 10 kW - 1-2 MW range. At lower inputs alternatives such as reciprocating and scroll compressors are preferable and at higher inputs turbo-compressors are more suitable.

In industrialised countries, compressors absorb 15-20% of the total electrical power generated. Hence there is a continuing demand to improve their efficiency. This is normally expressed as the specific power consumption, which is the power required to compress unit mass of gas delivered.

There already exist mathematical models to assist in the design of such machines and to estimate their performance, which include the estimation of the dynamic loads acting on the rotors and bearings and these loads determine their mechanical efficiency. However, these models do not estimate the magnitude of the mechanical losses, which are only guesstimated as an additional increment to the power required to compress the gas. Such an approximation does not enable the optimum selection of bearings and lubricating oil to minimise the frictional power losses.

The aim of the study, described in this paper, was to estimate the effect of the individual parameters responsible for mechanical power loss in oil injected screw compressors and is focussed on the losses incurred in the gear box, bearings and shaft seals.

It was found that in the gearbox, meshing, bearing and seal losses all increase both with speed and gear ratio. In the main rotors, it was found that sliding friction losses in the bearings are not significantly affected by the radial load, nor are rolling friction losses significantly dependent on the axial load. However, both axial and radial loads have a significant effect on the total frictional power loss. Lubricant viscosity affects the frictional power losses but the oil level does not.

Notation

A	centre distance	mm
B	bearing width	mm
b	face width	mm
C_1	bath lubrication constant	
C_2	bath lubrication constant	

C_{Sp}	factor of oil spraying	
C_W	variable	
d_m	bearing mean diameter	mm
d_{sh}	shaft diameter	mm
f_t	variable	
F_t	reference circle peripheral force	N
G_{rr}	variable for rolling frictional moment	
G_{sl}	variable for sliding frictional moment	
H	oil level inside the bearing	mm
H_V	factor of power losses in the mesh	
i_{rw}	number of ball rows	
K_A	application factor	
$K_{B\alpha}$	transverse load factor	
$K_{B\beta}$	face factor	
$K_{B\gamma}$	helix angle factor	
K_{ball}	constants for ball bearing	
K_{roll}	constants for roller bearing	
K_V	internal dynamic factor	
M	total frictional moment	N-mm
M_{drag}	frictional moment of drag losses, churning, splashing etc.	N-mm
M_{rr}	rolling frictional moment	N-mm
M_{seal}	frictional moment of seals	N-mm
M_{sl}	sliding frictional moment	N-mm
N	rotational speed	rpm
P	transmitted power	W
P_{loss}	power losses	W
P_S	power losses in seals	W
P_{Z0}	power losses in idle motion	W
P_{ZP}	power losses in mesh, under load	W
R_a	arithmetic mean roughness of meshed teeth	μm
R_S	variable	
T_H	hydraulic moment of power losses	N-m
T_{oil}	operating temperature of oil	^{o}C
U	gear ratio	
V_M	drag loss factor	
$v_{\Sigma c}$	sum of the peripheral speeds in the pitch point	m/s
v_{t0}	constant, 10 m/s	m/s
w_{Bt}	specific load in transverse plane	N/mm
z_1	pinion number of teeth	
z_2	gear number of teeth	

Greek letters

α_w	pitch circle pressure angle	degrees

α_t	transverse plane pressure angle	degrees
β_b	base circle helix angle	degrees
ε_1	partial contact ratio of pinion	
ε_2	partial contact ratio of gear	
ε_α	transverse plane contact ratio	
Φ_{ish}	inlet shear heating reduction factor	
Φ_{rs}	kinematic replenishment/starvation reduction factor	
v	actual operating viscosity of oil or the base oil of the grease	mm^2/s
μ_{sl}	sliding friction coefficient	
v_{40}	ISO VG number of oil	
μ_{mz}	mean coefficient of friction in mesh	
η_{oil}	dynamic viscosity of oil at operating temperature	mPa.s
ρ_{cn}	equivalent radius of meshed profiles curvature in the pitch point in normal plane	mm

1. Introduction
Due to the cooling effect of the entrained oil in screw compressors, these machines can attain very high isentropic efficiencies. However, mainly because the flow through them is not axi-symmetric, bearing loads within them are much larger than in turbo-compressors and these, together with other sources of frictional loss, can reduce their overall adiabatic efficiencies by as much as 10%. Currently, although isentropic power input, efficiencies and the magnitude of the bearing loads can be predicted very accurately, [3], there are no established methods to predict frictional losses and these are normally only accounted for assuming them to be some percentage of the total input power. A better understanding of the nature of the sources of loss and an improved ability to predict them, is, therefore essential if these relatively large losses are to be minimised.

In general, it is difficult mechanical losses, but it has been shown [1] that these increase at much lower rate than the compression losses at higher compressor duties. These include losses which happen before the work is transferred to the fluid, in the bearings and seals. In most publications they are assumed to be 7-10% of the shaft power but it has been shown that this value decreases with the increase in compressor size [2].

This study includes the estimation of losses in the gear boxes, usually contained in these machines, with account taken of meshing losses both in idling and under load, as well as losses in the seals, while the model for frictional losses in the bearings includes estimates of the effect of rolling friction, sliding friction, friction in the seals and drag losses. The model also accounts for the type of lubricant used and the effect of its viscosity on frictional losses. A parametric analysis of the type and size of bearings selected and how both axial and radial loads and drive speed act on them to affect the compressor overall performance is also included. However, mechanical losses caused by rotor contact and due to shear stresses within the oil contained in the clearances were not considered.

By including such predictive methods performance estimates should be closer to the experimental results and these can be used to help design machines with lower mechanical power losses.

2. Elements of mechanical power loss in compressor plant
A typical oil-injected air screw compressor package consists of a drive system; where it can be either driven by an electric motor or by an internal combustion engine, a transmission system if required, and the compressor. There are downstream components from compressor as well, like oil separator tank, heat exchangers for oil and for air, and control valves to regulate the discharge pressure. This paper focuses on the mechanical losses of the components which are upstream of the compressor including the compressor as well.

2.1. Drive system

Typically small to medium pressure air screw compressors are driven by an electric motor. The mechanical elements of the electric motor are those which contribute to the mechanical losses inside the electric motor. These elements are bearings and seals. The loading of the bearings and type of lubrication used can be used to calculate the losses inside the electric motor bearing. The diameter of the motor shaft and the speed of rotation define the seal losses which are explained in further sections.

2.2. Transmission system

The transmission system is used to transfer mechanical power from the driver to the driven. The drive in this case can either be an electric motor or an engine while the driven is compressor. The power in small pressure screw compressor system is transmitted by a belt, while for medium to high pressure compressor system, it is transmitted through gear box system.

In a gear box system, the power is transferred through rotational motion at different speeds, torque and direction. A part of power is lost due to friction between the geared elements during mesh and also in other mechanical elements like bearings and seals [4]. The mechanical losses in a transmission system are explained in subsequent sections.

2.3. Compressor

The major contributors to mechanical losses in a screw compressor are bearings and seals. A usual approximation of these losses is if they are considered to be 5-12% of the shaft power. Different types of bearings are used inside of a screw compressor machine to take different loads. Generally, the cylindrical roller bearings are used to carry radial loads while the ball bearings are used to transfer axial loads from the rotors to the housing. In an oil-injected screw compressor machine, the viscosity and level of oil occurring in the bearing housing also affect the frictional losses.

The effects of loads, type of bearing, size of bearing, type and level of oil are analysed and compared in the power loss section.

3. Power loss in transmission system

3.1. Gearbox

As explained in the previous section, a gear box is one of the types of mechanical power transmission devices. Power losses in the gear box consist of the power losses in a gear mesh under load, the power losses during idle motion, the power losses in the gearbox bearings and seals. The power losses in mesh, under load, for a single gear pair in a cylindrical gear drive are obtained by the following equation [4].

$$P_{ZP} = \mu_{mZ} P H_V \tag{1}$$

The mean coefficient of friction in the mesh is approximately equal to the coefficient of friction at the gear pitch point, and is given by:

$$\mu_{mZ} \approx \mu_c = 0.12 \times \sqrt[4]{\frac{w_{Bt} R_a}{\eta_{oil} v_{\Sigma C} \rho_{cn}}} \tag{2}$$

Specific load in the transverse plane w_{Bt} is calculated by following equation

$$w_{Bt} = K_A K_V K_{B\alpha} K_{B\beta} K_{B\gamma} \frac{F_t}{b} \tag{3}$$

The sum of peripheral speeds in the pitch point is equal to

$$v_{\Sigma c} = 2v \tan \alpha_w \cos \alpha_t \tag{4}$$

The equivalent radius of meshed profiles curvature at pitch point in the normal plane is given by:

$$\rho_{cn} = a\frac{u}{(u+1)^2}\frac{\sin\alpha_w}{\cos\beta_b} \tag{5}$$

The dynamic viscosity of oil at operating oil temperature is determined by the following equation:

$$\eta_{oil} = 0.1282\times\eta_{50}^{\frac{174}{\upsilon_{oil}+95}-0.198\frac{298}{e^{\upsilon_{oil}+95}}} \tag{6}$$

where, η_{50} is the dynamic viscosity of oil at operating temperature of 50°C and υ_{oil} is the operating temperature of oil (in °C).

The power loss factor of the gears in mesh is determined by the following equation:

$$H_V = \pi\left(\frac{1}{z_1}+\frac{1}{z_2}\right)\frac{1}{\cos\beta_b}\left(1+\varepsilon_1^2+\varepsilon_2^2-\varepsilon_\alpha\right) \tag{7}$$

It is also necessary to calculate the power loss when the gears are not under load but are still in motion during the compressor unloading conditions. The power loss of idle motion can be expressed as follows:

$$P_{Z0} = \sum_{i=1}^{k} T_{Hi}\frac{\pi n_i}{30} \tag{8}$$

where the hydraulic moment of power loss is given as:

$$T_H = C_{Sp}C_1 e^{C_2(v/v_{t0})} \tag{9}$$

The oil spraying factor C_{Sp} and, C_1, C_2 factors for gear width and depth of their position in the oil bath are considered while calculating the power loss.

3.2. Power losses in seals

The power loss in shaft seals is function of the shaft speed only. Hence the variation of input power has no effect on power loss in seals. The power losses in the shaft seals can be calculated using following equation [4]:

$$P_{Si} = \left[145 - 1.6T_{oil} + 350\log\log\left(v_{40} + 0.8\right)\right]d_{sh}^2 n.10^{-7} \tag{10}$$

Figure 1 shows the effect of input power upon the power losses in seals, power losses during the idle motion and power losses during the gear mesh under load. The helical gears are presented in Figure 1 have a gear ratio of 2.23 with a centre distance 125 mm, helix angle 12°, pressure angle 20° and a module of 2.5.

The power losses in idle motion are mainly function of peripheral speed, position of the gears in oil and the direction of rotation of mating gears. Since the input power does not affect any of the parameters mentioned above, the power loss in idle motion for various input power remains constant. Similarly, the power losses in seals are function of the shaft diameter, shaft speed and oil temperature. Hence, the input power does not influence the power losses in the seals.

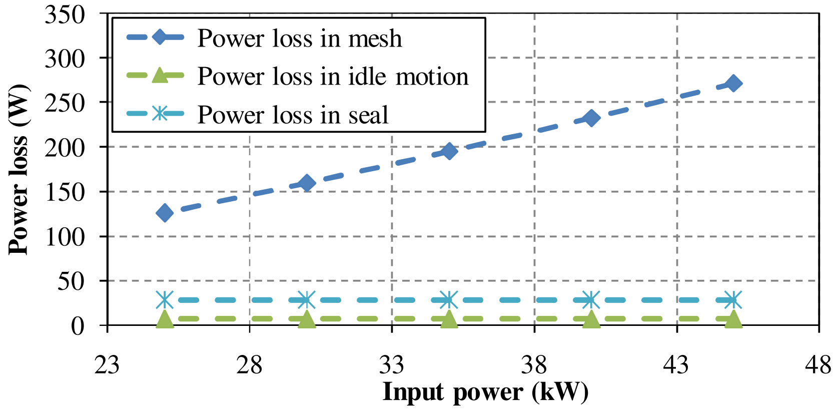


Figure 1. Power loss v/s input power.

The power loss is also a function of gear ratio. Figure 2 shows the effect of gear ratio on power losses. As the gear ratio increases, the power loss during mesh under load increases. The reason behind this is due to increase in gear ratio, the contact ratio between mating gears increases which causes increase in the power loss. However, the power losses in seals and power losses in idle motion do not change, as explained in the earlier section. The same helical gear specification as mentioned above with the power input of 24 kW is used for analysis of Figure 2 and Figure 3.

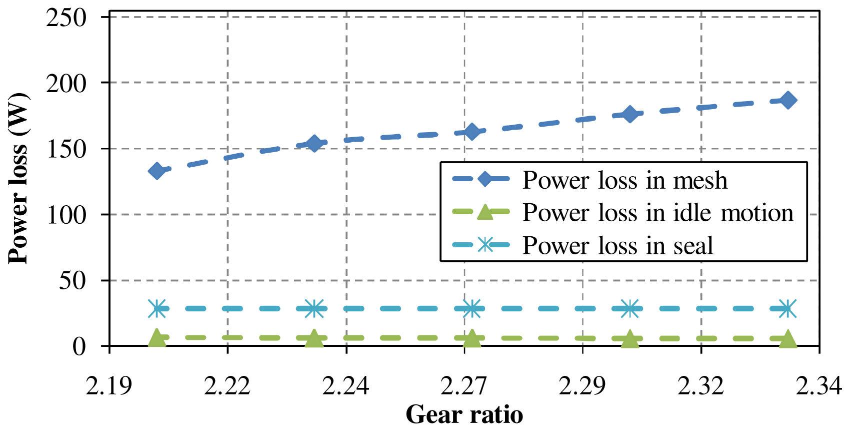


Figure 2. Power loss v/s gear ratio.

Figure 3 shows that the transmission efficiency decreases with the increase of gear ratio. As gear ratio increases, the efficiency drops because of the increased power losses for the gears in mesh under load.

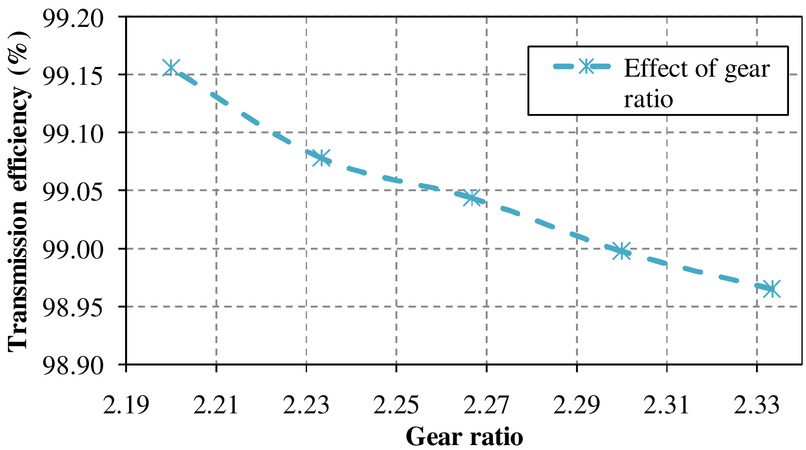


Figure 3. Transmission efficiency v/s gear ratio.

The parametric study shows that total power loss increases with the increase of input speed, increase in gear ratio and increase in input power. The major contributor to the total power loss is the power loss in mesh under load, while the power loss in seals and bearings also contribute by considerable amount.

3.3. Types of bearing frictional loss

A number of operational and non-operational factors affect the bearing friction. The friction between interacting surfaces which are in relative motion inside the bearings always changes. SKF model for calculating the frictional moment [5] presents how the friction changes with the bearing speed, as shown in Figure 4.

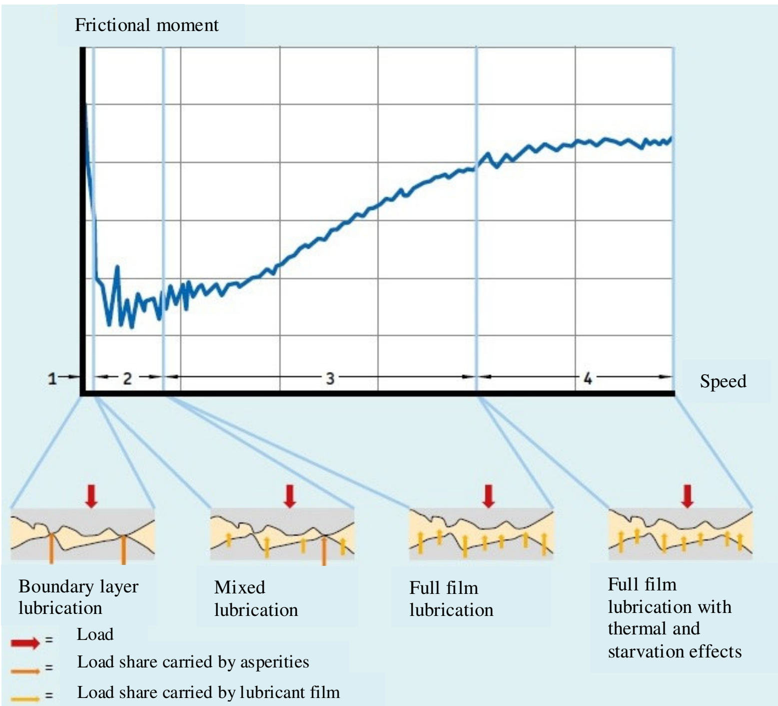


Figure 4. Bearing frictional moment v/s speed. [5]

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:

$$M = M_{rr} + M_{sl} + M_{seal} + M_{drag} \tag{11}$$

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

$$M_{rr} = \phi_{ish}\phi_{rs}G_{rr}\left(\nu n\right)^{0.6} \tag{12}$$

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.

A reverse flow of lubricant just behind the rolling element shears the lubricant which generates heat and consequently reduces oil viscosity. This results in reduction of the lubricant film thickness and rolling friction. The inlet shear heating factor takes into account this phenomenon. The kinematic replenishment/starvation factor estimates reduction of the lubricant film thickness because the oil starvation. The continuous rolling displaces lubricant from the racer ways not allowing sufficient time to replenish the lubricant at higher speeds. The influence of the rolling frictional moment depends upon the bearing mean diameter and its radial and axial loads.

The sliding frictional moment is a function of sliding friction coefficient and sliding frictional moment and it can be expressed as follows:

$$M_{sl} = G_{sl}\mu_{sl} \tag{13}$$

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:

$$M_{drag} = 0.4V_M K_{ball} d_m{}^5 n^2 + 1.093\times10^{-7} n^2 d_m{}^3 \left(\frac{n d_m{}^2 f_t}{\nu}\right)^{-1.379} R_s \tag{14}$$

For roller bearing, the same can be estimated as follows

$$M_{drag} = 4V_M K_{roll} C_w B d_m{}^4 n^2 + 1.093\times10^{-7} n^2 d_m{}^3 \left(\frac{n d_m{}^2 f_t}{\nu}\right)^{-1.379} R_s \tag{15}$$

The bearing frictional power loss can be calculated using the following equation [6]

$$P_{loss} = 1.05\times10^{-4} M\ n \tag{16}$$

The mathematical model of compressor mechanical losses is developed by using MATLAB software [7], and a parametric analysis of the bearing frictional losses is carried out. The results of this parametric analysis are presented in the next section, where the effects of shaft speed, bearing loads, lubricant viscosity, its level and the size of bearing are analyzed.

An oil-injected air screw compressor with the input power of 145 kW and rotor bearing loads calculated for 8.5 bar absolute discharge pressure is considered as a calculation case for the bearing frictional power loss. The radial loads of the rotors at the suction as well as at the discharge side are carried by cylindrical roller bearings, while the axial loads of the rotors on discharge side are taken by angular contact ball bearings.

3.3.1. Effect of speed. Figure 5 shows the effect of speed upon the bearing frictional power loss for the selected compressor set of bearings. The frictional power loss is directly proportional to the shaft speed; and hence as the speed increases, the frictional power loss also increases. The frictional power loss increases with the increase in the speed for all bearings; but for a given speed, it is noticed that different bearings have different frictional power loss.

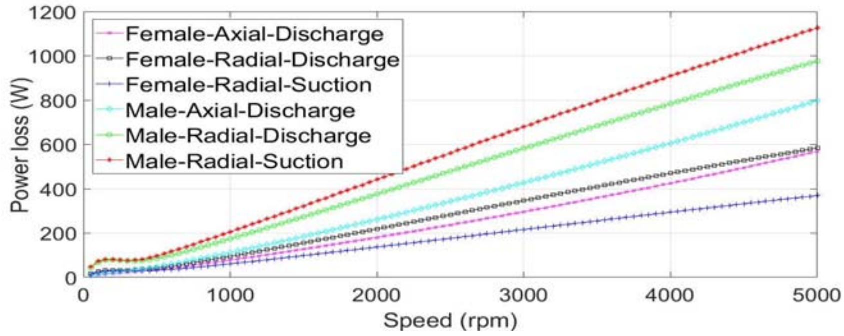
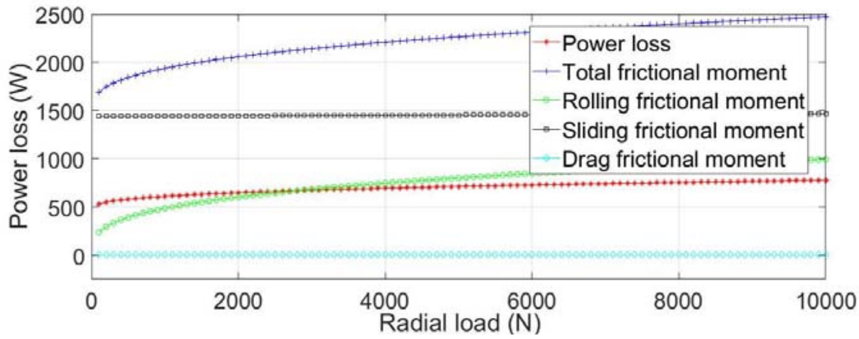


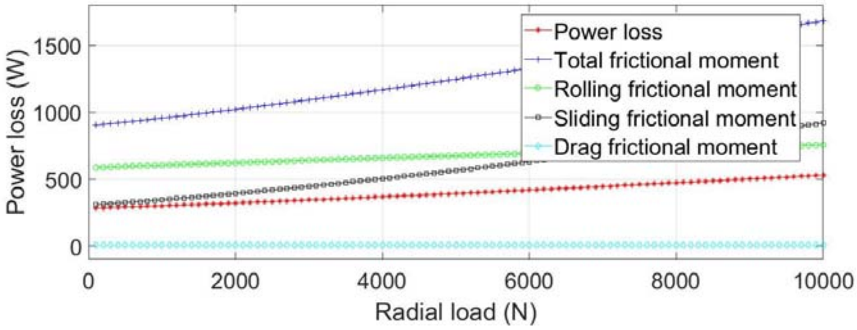
Figure 5. Bearing power loss v/s shaft speed.

For a given speed, the highest frictional power loss is noticed for the male radial suction bearing which is a cylindrical roller bearing. This can be attributed to the fact that it is the biggest bearing in the bearing set. The bigger the size means the larger the contact area in the bearings which increases the frictional power loss. The female suction radial bearing is the smallest bearing, and hence it causes the lowest frictional power loss.

3.3.2. Effect of bearing loads. As previously discussed, the bearing frictional loss consists of the components which represent the rolling, sliding and drag losses. The effect of radial loads upon these components along with total frictional moment and power loss is shown in Figure 6 where (a) shows effect of the radial load on roller bearing (male radial suction) and (b) shows effect of the radial load on ball bearing (male axial discharge).



(a) Effect of radial load on roller bearing.



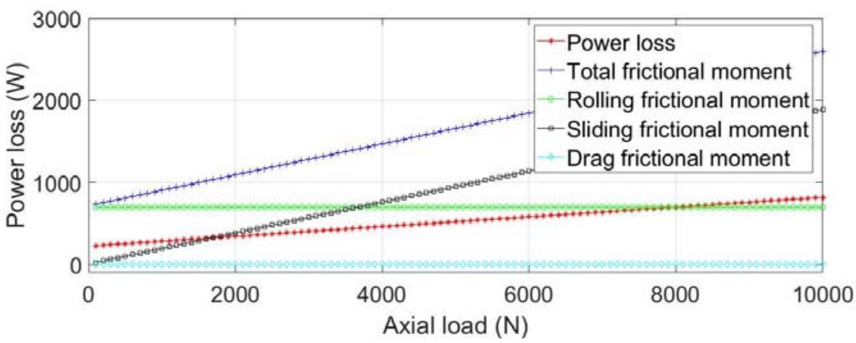
(b) Effect of radial load on ball bearing.

Figure 6. Effect of radial load on bearing loss.

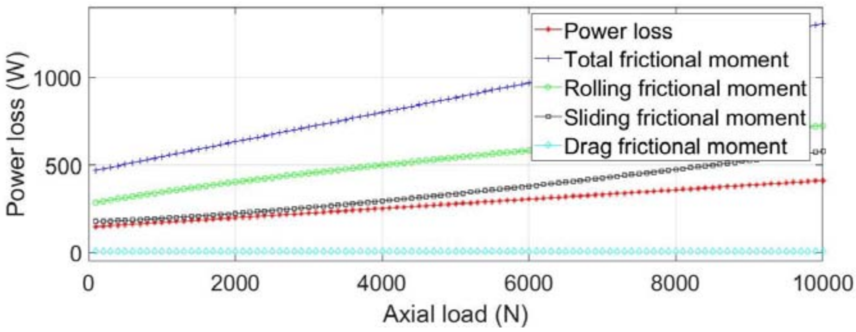
For the roller bearing at the male rotor suction, the effect of the radial load is significant in the rolling frictional component, while the change in the sliding frictional componentis negligible and can be ignored. The frictional drag component is constant and the radial load has no effect upon. Since the total frictional moment is added for all components, the total frictional moment as well as power loss of the radial bearings increases substantially with the increase of the radial load.

In case of the axial ball bearing at the male rotor discharge, along with the rolling frictional component, the sliding frictional element also increases with the increase of the radial load. The rolling and sliding variables which are dependent upon the geometry and loads are G_{rr} and G_{sl} . The roller bearings are considered to take purely radial load and no axial load; while the ball bearings are considered to take the axial load and some part of the radial load. When the axial load is zero, the effect of radial load upon the sliding variable G_{sl} is negligible, and this is the reason of the nearly constant sliding frictional loss of the roller bearing.

3.3.3. Effect of axial load on bearing frictional loss. Figure 7 (a) shows the effect of axial load on roller bearing (male radial suction), and Figure 7 (b) shows the effect on ball bearing (male axial discharge). As the axial load on roller bearing increases, only sliding component of frictional loss increases, while the rolling and drag component remains constant. In the case of ball bearing, the rolling component along with the sliding component of the frictional loss increases with the increase of the axial load.



(a) Effect of axial load on roller bearing.



(b) Effect of axial load on ball bearing.

Figure 7. Effect of axial load on bearing loss.

The effect of axial load is not significant for the rolling frictional loss, while the effect of radial load is not significant for the sliding frictional loss, as can be seen in the Figure 6 and Figure 7.

3.3.4. Effect of viscosity. The effect of viscosity upon the bearing frictional power loss is presented in Figure 8. The frictional power loss is the largest for the roller bearing at the male radial suction and it is lowest for the female radial suction. Similar explanation is applicable here, that with the bigger bearing, the contact area is larger and hence the viscosity effect is higher.

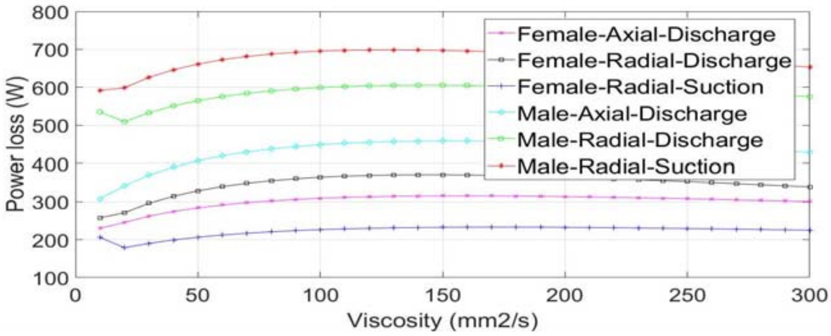


Figure 8. Frictional power loss v/s Viscosity.

3.3.5. Effect of oil level. One of the factors that influences the drag losses is oil level. The oil level is defined here as a level of oil above the inner diameter of the outer bearing racer. As can be seen from Figure 9, the drag loss does not change significantly with the oil level. The drag loss is higher for the bigger bearing compared to the smaller bearing.

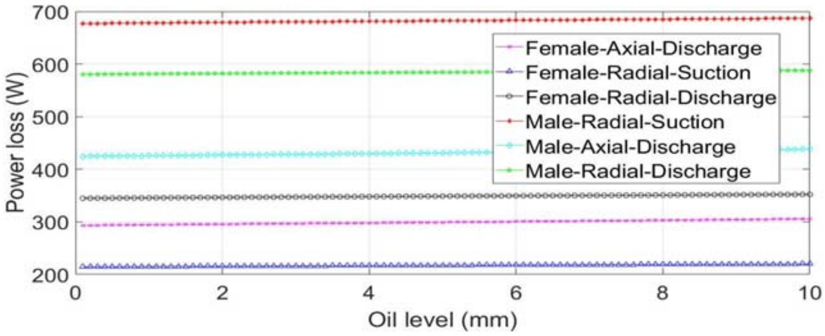


Figure 9. Effect of oil level on bearing drag frictional loss.

3.3.6. Effect of size of bearing. An analysis is carried out to see the effect of the bearing size upon the frictional power loss. The analysis is carried out for roller as well as for ball bearings. For this analysis, the radial and axial loads are considered to be similar for different bearing sizes, and the calculated values of frictional losses are presented in Table 1 and Table 2. It is noticed that the frictional power loss increases with the bearing size. This behaviour is observed for the roller as well as for the ball bearings.

Table 1. Effect of radial bearing size on power loss.

Bearing	Size (all in mm)	Mrr (N-mm)	Msl (N-mm)	Mdrag (N-mm)	Ploss (W)
NU 2212 ECP	60x110x28	754.37	1343.64	8.35	663.50
NU 2213 ECP	65x120x31	897.69	1449.96	9.60	742.54
NU 2214 ECP	70x125x31	991.78	1520.36	11.02	794.79

Table 2. Effect of axial bearing size on power loss.

Bearing	Size (all in mm)	Mrr (N-mm)	Msl (N-mm)	Mdrag (N-mm)	Ploss (W)
7410 BEP	50x130x27	745.423	666.517	5.31	446.434
7411 BEP	55x140x33	854.805	709.578	5.72	494.582
7412 BEP	60x150x35	971.374	761.412	5.843	547.668

The bigger size of the bearing makes larger contact surfaces and hence it contributes more towards the frictional power loss. From Table 1 and Table 2, it is observed that the rolling element of frictional power loss is affected more than the sliding and drag components. For 5 mm increase in the shaft diameter or bearing bore diameter, the increase in rolling frictional moment is about 10-20%; while for the sliding frictional element, it is 3-8%, and for the drag frictional element it is 2-14%.

4. Conclusions

A discussion on the effect of individual parameters upon the mechanical power losses is presented in this paper. It is noticed that in the gearbox system, the power losses of the gear engagement under load and under idle motion, as well as the power losses in the bearings and seals increase with the gear ratio and input speed. Consequently the gearbox efficiency decreases. The power losses during the idle motion are only functions of the peripheral speed, oil bath depth of gears and direction of rotation, while the power losses in seals are functions of the shaft diameter, shaft speed and oil temperature.

The power losses in the bearings are affected by several parameters. The frictional losses rise with the shaft speed and also with the bearing size. The frictional power loss increases with the increase of the lubricant viscosity. The sliding frictional component is not significantly changed by radial load, while the rolling frictional element is not significantly influenced by the axial load. However, both loads, the radial, as well as the axial, individually have significant effect upon the total frictional power loss. The increased oil level does not significantly affect the drag frictional losses in the bearings.

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