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Efficiency Investigation on a Helical Gear Transmission

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Gears are extensively used in many applications, such as: automotive, drive trains, industrial gearboxes and machine tools. They are designed to transmit power and rotational motion from the input shaft to the output shaft. In the design process of a gearbox, factors such as: load capacity, size, lifetime or manufacturing cost are often taken into consideration. However, precise measures of efficiency are frequently forgotten issues in the design process. Such shortfalls relate to oil churning, windage, oil squeezing during gear mesh, or the friction processes at the level of the gear pairs, seals and bearings. This paper presents a calculation method used to determine the efficiency of helical gear transmissions. The method was validated by experimental measurements. Finally, influence factors on the gearbox efficiency, such as transmitted load and operating speed are presented.

Keywords: *efficiency, friction, helical gear, losses*

1. Introduction

Gears are extensively used in many applications, such as automotive, drive trains, industrial gearboxes or machine tools. They are designed to transmit power and rotational motion from the input shaft to the output shaft. In this process, a part of the power is unavoidably lost due to either friction in the system, or oil churning.

Prime movers as internal combustion engines were in the past the main focus of efficiency improvement efforts. As most of the potential efficiency improvements of these engines have been understood and implemented, the focus to improve the efficiency shifted towards remaining elements of the drive train, such as, for instance, the gear transmission. Due to the continuous increase of petroleum price, the fuel economy became nowadays an important demand. Furthermore, national regulations and the environmental pressures became stricter in terms of emission-related regulations, such as those concerning gases and particles released in the environment. For this reason, any improvement in gear transmission efficiency becomes of great importance and will, potentially, significantly reduce the fuel consumption and air pollution.

Besides these main reasons, improving the efficiency of gear transmissions, comes with a few other benefits. In this respect, as several gear failures, such as scoring or contact fatigue failures are influenced by the heat generation inside the gearbox, more efficient gears would generate less heat, and consequently, their performance in terms of failure resistance would increase.

The potential sources of power loss inside a gearbox are shown in Figure 1. To calculate these losses, it is necessary to know the loads in every involved machine element. A literature review [1-5] indicates that the power loss sources are divided into load-dependent and load-independent components. Load-dependent power loses are represented by the tooth friction and the bearing friction, while the load-independent power loses refer to the friction in the contact-seals and the loses due to oil churning, windage and oil squeezing during gear mesh.

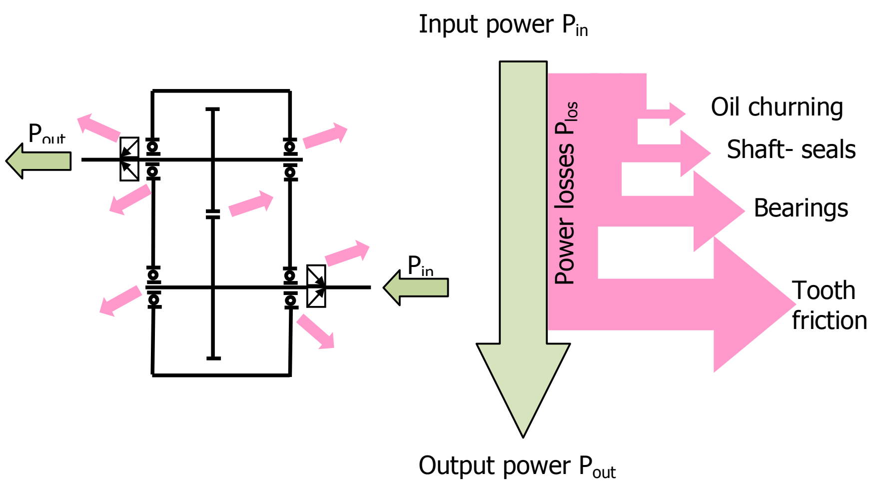

Figure 1. Power losses and power flow in a gearbox

The gearbox efficiency η is evaluated as:

$$\eta = \frac{P_{out}}{P_{in}} = \frac{P_{in} - P_V}{P_{in}}, \quad (1)$$

where:
P_{out} - the gearbox output power
P_{in} - the gearbox input power;
P_V - the gearbox power loss.

2. Analytical determination of the gearbox power losses

As mentioned above, the total power loss in a gearbox is represented by the sum of all loses in each gearbox element. According to [6], the total power loss of a gearbox, P_v,can be calculated by using the equation:

$$P_V = P_{Vz} + P_{Vz0} + P_{VB} + P_{VS} + P_{VX} , \tag{2}$$

where:

P_{Vz} - gear power loss, due to the friction between the gear flanks;

P_{Vz0} - no-load gear power loss, due to windage, oil churning and oil squeezing during gear mesh;

P_{VB} - bearing losses;

P_{VS} - seal losses due to the interaction between the shafts and the seals;

P_{VX} - auxiliary losses from other gearbox components, such as: pumps, fans, heating, clutches or control system.

2.1. Gear power loss P_{Vz}

The overall average gear power loss, along the path of contact, at a gear mesh, can be expressed, according to [6] as:

$$P_{Vz} = H_V \cdot P_{in} \cdot \mu_m , \tag{3}$$

where:

H_V - power loss factor;

P_{in} - input power;

μ_m - average coefficient of friction.

Even if different formulations are presented in the literature [7], [8], for the calculus of the power loss factor H_V, [5] and [6] show that the relation of Ohlendorf [9], initially employed only for spur gears, applied with a deviation under 10%, may be used for helical gears too.

Therefore, in this approach, the power loss factor will be calculated with Ohlendorf's relation:

$$H_V = \frac{1+u}{u} \cdot \frac{\pi}{z_1 \cdot \cos\beta_b} \cdot \left(1 - \varepsilon_\alpha + \varepsilon_1^2 + \varepsilon_2^2\right), \tag{4}$$

where:

u- gear ratio;

z_1 - pinion number of teeth;

β_b - helix angle on the base circle;

ε_α - profile contact ratio;

ε_1 - addendum transverse contact ratio of pinion;

ε_2 - addendum transverse contact ratio of driven gear.

The average coefficient of friction μ_m depends on the applied load, the sliding velocity, the lubricant properties, and the gear surface roughness. For its calculation, Niemann's [6] formulation is widespread accepted:

$$\mu_m = 0{,}048 \cdot \left(\frac{F_{bt} / l_{min}}{v_{\Sigma C} \cdot \rho_{redC}} \right)^{0{,}2} \cdot \eta_{oil}^{-0{,}05} \cdot R_a^{0{,}25} \cdot X_L , \tag{5}$$

where:

F_{bt} - circumferentiual force at base circle [N];

l_{min}- minim contact length of the gear pair (equal to face width in case of spur gears [mm];

$v_{\Sigma C}$ - sum velocity at operating pitch circle [m/s];

$$v_{\Sigma C} = 2 \cdot v_t \cdot sin\alpha_{wt} , \tag{6}$$

ρ_{redC} - reduced radius of curvature at pitch point [mm];

$$\rho_{redC} = 0{,}5 \cdot d_{w1} \cdot sin\,\alpha_{wt} \cdot \frac{u}{(u+1) \cdot cos\,\beta_b} , \tag{7}$$

η_{oil} - dynamic oil viscosity at oil temperature [mPas];

R_a - arithmetic mean roughness [µm];

$$R_a = 0{,}5 \cdot (R_{a1} + R_{a2}) , \tag{8}$$

X_L- lubricant correction factor (unit-less), which, for oil, is calculated as:

$$X_L = \left(\frac{F_{bt}}{b} \right)^{-0{,}0651} , \tag{9}$$

b- face width [mm]

v_t- pitch line speed [m/s]

α_{wt}- working transverse pressure angle;

d_{w1}- pinion operating pitch diameter.

2.2. No- load gear power loss P_{Vz0}

The load independent gear losses, due to windage, oil churning and oil squeezing during gear mesh, are influenced, among others, by the circumferential speed of the gears, the internal gearbox design, the operating temperature, the oil viscosity and the design of the rotating wheels. [6] proposes the following expression for the calculation of the power losses in case of gearboxes with splash lubrication:

$$P_{Vz0} = 76{,}92 \cdot 10^{-6} \cdot b \cdot h \cdot v^{1{,}5} \ [kW], \tag{10}$$

where:

b- gear face width [mm];

h- oil immersion depth of the gear wheel [mm];

v- peripherical speed of the gear wheel [m/s].

2.3. Bearing losses P_{VB}

The losses in bearings are the result of the rolling and sliding friction which occurs between the different parts of the bearing. Therefore, an accurate prediction of losses has to take into account all these heat generation sources. The relative new approach of SKF [10] considers four different losses to predict the torque loss in a bearing:

$$M_{VB} = M_{rr} + M_{sl} + M_{seal} + M_{drag} , \qquad (11)$$

where:

M_{VB} - bearing torque loss;

M_{rr}- rolling friction torque;

M_{sl} - sliding friction torque;

M_{seal} - frictional torque of seals;

M_{drag} - frictional torque of drag losses, churning, splashing etc..

The conversion from torque losses to power losses can be done using the relation:

$$P = \frac{M \cdot n}{9{,}55} , \qquad (12)$$

where:

P- power [kW];

M- torque [Nmm];

n- rotational speed [rpm].

The rolling and the sliding friction torques M_{rr} and M_{sl} are given by:

$$M_{rr} = \varphi_{ish} \cdot \varphi_{rs} \cdot G_{rr} \cdot (\nu \cdot n)^{0,6} \text{ [Nmm]}, \qquad (13)$$

$$M_{sl} = G_{sl} \cdot \mu_{sl} \text{ [Nmm]}, \qquad (14)$$

where:

φ_{ish}- inlet shear heating reduction factor (describes the influence of the lubricating film thickness on the rolling friction);

φ_{rs}- kinematic replenishment/starvation reduction factor (considers the lubricant displacement in the contact zone due to over rolling, which generates a lower rolling friction torque);

G_{rr} and G_{sl}- variables depending on bearing type, bearing mean diameter ($d_m=0,5(d+D)$), the radial force F_r and the axial force F_a;

ν- kinematic viscosity of the lubricant [mm^2/s];

n- rotational speed [rpm]

μ_{sl}- sliding friction coefficient.

The frictional torque in bearing seals is given by:

$$M_{seal} = K_{s1} \cdot d_s^{\beta} + K_{s2} \text{ [Nmm]}, \qquad (15)$$

where:

K_{S1} and K_{S2}- constants depending on the seal type, respective bearing type and size;

d_S- seal counterface diameter [mm];

β- exponent depending on seal type and bearing type.

The drag losses occur when a bearing is rotating in an oil bath, while being influenced by the bearing type, speed, oil level and oil viscosity. For a ball bearing, the drag losses are expresses by:

$$M_{drag} = V_m \cdot K_{ball} \cdot d_m^5 \cdot n^2 \ \ [Nmm], \qquad (16)$$

where:

V_m- variable depending on the oil level;

K_{ball}- bearing type related constant;

d_m- bearing mean diameter [mm];

n- rotational speed [rpm]

The Bearing Calculator, accessible on skf.com/bearingcalculator, allows an easy and fast calculation of the bearing losses for any type of standard bearing.

2.4. Seal losses P_{VS};

The power loss of the radial seals is the result of the friction between the lip of the seal and the rotating shaft. Several computational models are presented in the literature, but due of its simplicity, we ilustrate here the approach of Kettler [11]:

$$P_{VS} = 7{,}69 \cdot 10^{-6} \cdot d_{sh}^2 \cdot n \ \ [kW], \qquad (17)$$

where:

d_{sh}- shaft diameter [mm];

n- shaft rotational speed [rpm].

3. Materials and methods

The aim of this project was to compare the analytically determined efficiency of a helical gearbox, with experimental measurements. The experiment was performed on the test rig shown in Figure 2. Technical data of the gearbox are presented in Table 1.

As can be seen in Figure 2, for loading the gearbox a gear pump was used. For variation of the transmitted torque, a spherical valve has been placed on the pressure pipe of the gear pump.

The measurement of the gearbox transmitted load was done with two torque flanges (type T 1 0FS, made by HBM, Germany). One was mounted in the front of the input shaft, and the other behind the output shaft. These devices allowed measuring both the torque, and the speed. Three MP 60 modules (two for torque and one for speed), were used for data conversion.

In the first stage of the experiment, the gearbox was operated at the maximum input speed (n_1= 1.500 [RPM]), six different input torques (M_{t1}= 6, 8, 10, 12, 14, 16 [Nm]) being in turn used, while, the output torque measurement was performed for each operating condition corresponding to the different input torques. By means of Eq. (12), the corresponding input and output powers P_{in} and P_{out} respectively have been determined and the experimental efficiency was calculated with Eq. (1). In the meantime, the analytical efficiency of the gearbox was computed, using the approach presented in the previous chapter.

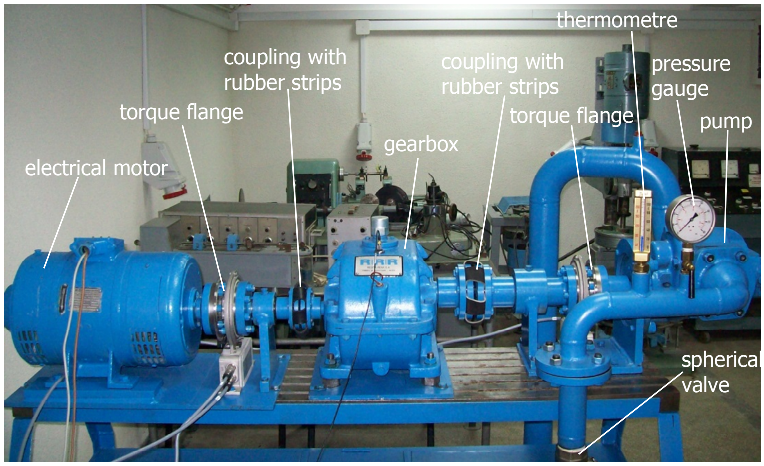

Figure 2. General view of the gearbox test rig [12]

Table 1. Technical data of the gearbox

Main geometrical data of the gears	
Centre distance	A= 125 [mm]
Module	m_n= 4 [mm]
Teeth number	z_1/ z_2= 17/ 43
Helix angle	β= 11°
Tooth width	b= 40 [mm]
Normal pressure angle	α_0= 20 [°]
Addendum modification coefficients x_1/ x_2	0,39/ 0,34
Reference diameters d_1/ d_2	69,27/ 175.22 [mm]
Performance data	
Maximum power	P= 2,5 [kW]
Input speeds	n_1= 1000, 1100, 1200, 1300, 1400, 1500 [rpm]
Input torque	M_{t1}= 6, 8, 10, 12, 14, 16 [Nm]
Lubrication	
Oil splash lubrication	
Oil viscosity	220 [mm^2/s]@ 40°C
Oil immersion depth of the gear	30 [mm]
Bearings	
Input shaft:	2x 6207
Output shaft:	6211 and 6307
Seals	
Input shaft:	Radial shaft seal Ø32
Output shaft:	Radial shaft seal Ø55

In the second stage of the experiment, the gearbox was operated in turn at six input speeds (n_1= 1.000, 1.100, 1.200, 1.300, 1.400 1.500 [RPM]), while the input torque was maintained constant (M_{t1}= 16 [Nm]). Similarly to the first stage of the experiment, the gearbox efficiencies were also determined, both analytically and experimentally, for the six operating conditions.

4. Results and discussion

The results of the gearbox efficiency obtained by analytical determination, and those obtained by experimental measurement for the 12 working conditions, are comparatively shown in Tables 2 and 3.

Table 2. Efficiency assessment under variable input torque

n_1= 1.500 [RPM]						
M_{t1} [Nm]	Analytical approach			Experimental measurements		
	P_{in} [kW]	P_V [kW]	h [%]	M_{t2} [Nm]	P_{out} [kW]	h [%]
6	0,94	0,140	85,14	12,80	0,79	84,34
8	1,26	0,142	88,70	17,80	1,11	87,96
10	1,57	0,144	90,83	23,00	1,43	90,93
12	1,88	0,146	92,25	28,10	1,74	92,58
14	2,20	0,148	93,27	33,20	2,06	93,75
16	2,51	0,150	94,03	37,90	2,35	93,65

Table 3. Efficiency assessment under variable input speed

M_{t1}= 16 [Nm]						
n_1 [RPM]	Analytical determination			Experimental measurements		
	P_{in} [kW]	P_V [kW]	h [%]	M_{t2} [Nm]	P_{out} [kW]	h [%]
1.000	1,68	0,145	91,35	24,70	1,53	91,55
1.100	1,84	0,146	92,08	27,30	1,70	91,99
1.200	2,01	0,147	92,69	29,90	1,86	92,35
1.300	2,18	0,148	93,20	32,70	2,03	93,23
1.400	2,35	0,149	93,65	35,50	2,20	93,98
1.500	2,51	0,150	94,03	37,90	2,35	93,65

For a better understanding of the way the input speed and the input torque are influencing the gearbox efficiency, Figures 3 and 4 provide the corresponding variation graphs clarifications.

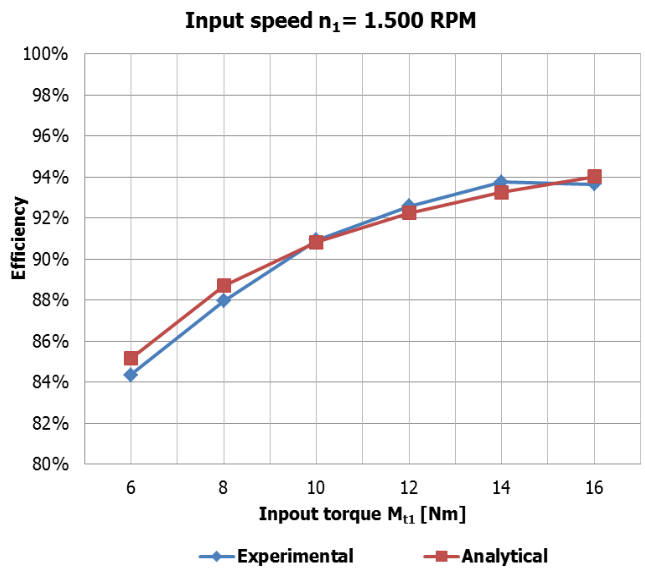

Figure 3. Influence of the input torque on the gearbox efficiency at constant input speed

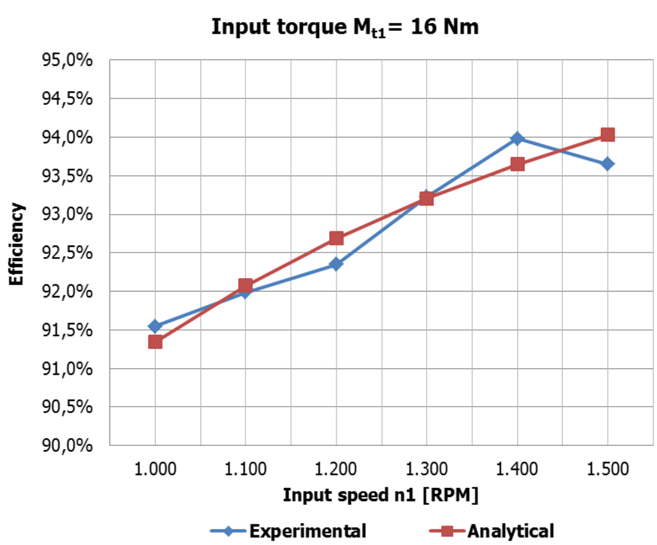

Figure 4. Influence of the input speed on the gearbox efficiency at constant input torque

Comparing the results of the gearbox efficiency acquired by analytical approach and by experimental measurements, one can observe the values are very similar: a maximum calculated deviation between the two approaches is represented by 0,4%.

First, by increasing the input torque from 6 to 16 Nm, while maintaining the input speed at 1.500 rpm, the gearbox efficiency increases too, from about 85% to nearly 94% (see Figure 3). Second, by increasing the input speed from 1.000 to 1.500 rpm, while maintaining the input torque at 16 Nm, the gearbox efficiency increases from nearly 91% to almost 94% (see Figure 4).

For the investigated operating conditions, it can be determined that, while maintaining the input speed at 1.500 rpm, by doubling the input torque from 8 to 16 Nm, the gearbox efficiency is improving by about 5%. Similarly, while maintaining the input torque at 16 Nm, by growing the input speed by 50% (from 1.000 to 1.500 rpm), the gearbox efficiency is correspondingly increased by about 2,5%.

The lower values of the efficiency assessment may be a result of the corresponding torque and speed operating conditions, in which the load independent losses have a significant contribution in the total amount of power loss. Such loses refer to the windage, oil churning and oil squeezing during gear mesh, the no-load bearing losses and the seal losses, which have a constant sum during the process. In addition, in the working domains where the transmitted power has higher values, the contribution of the no-load losses becomes negligible in the total amount of power loss, even if the load dependent power losses are increasing (due to tooth and bearing friction).

Finally, the efficiencies obtained here are far below the values indicated by the literature in the field, according to which, for one-step gearboxes, the power loss does not exceed 2%, and, consequentially the efficiency should not be below 98%. The efficiency values obtained here can be explained by the fact that the maximum power that has been transmitted by the investigated gearbox during the testing was $P_{in} \approx 2,5$ kW. This power is well below the maximum power that can be transmitted by the gearbox ($P_{max} = 5$ kW). This finding leads us to the conclusion that, in order to obtain a maximum efficiency, the gearboxes must be used at the power at which they were dimensioned

5. Conclusions

This paper presents analytical and experimental results concerning the efficiency investigation of a gear transmission. Improving gear transmission efficiency is motivated by several reasons, such as energy efficacy and environmental pressures.

We first describe an analytical model of the gearbox power losses based on mathematical equations previously proposed in the literature. Second, we perform experimental measurements concerning the efficiency of a helical gearbox. Using a test rig, we collected data in 12 distinct working conditions, which were different in terms of input speed and the input torques. We showed that by increasing the input torque while maintaining the input speed constant, the gearbox efficiency increases too. Similarly, the gearbox efficiency increases with increasing the input speed, while maintaining the input torque constant. Third, we performed a comparison between the results of the gearbox efficiency acquired by applying the analytical approach, and those acquired by experimental measurements. The outcome indicated similar values for the two approaches.

Our efficiency results are lower than previously reported in the literature, possibly due to the low transmitting power by the used gearbox, relative to its maximum possible power. Finally, we recommend gearboxes to be used at the power at which they were dimensioned, in order to obtain a maximum efficiency.

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