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# Theoretical Investigation of the Effect of Changes in Some Geometrical Parameters from the Point of View of the Energetic Behavior of the Roller Gearing Gearbox

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In the article, the energetic analysis of the propulsion unit is presented by examining the geometrical parameters determining the friction phenomena at the ball-track connection. The novelty of the topic stems from the fact that the propulsion unit under investigation is a Hungarian invention, the energy analysis of which has not yet been dealt with in any form. To carry out the investigation, a mathematical model had to be created, which was built on the basis of analytical principles. With the help of the model, the energetic behavior of the propulsion unit can be analyzed at a theoretical level. To construct the mathematical model, the principles, research results, and models used in the case of ball bearings and ball screw drives were used, transformed, and applied in a suitable way.
After the parameter tests, the analysis extended to the entire gear model, so the losses outside the ball-track connection, e.g., bearing friction, seal friction, oil friction, air friction, etc. losses, were also implemented in the mathematical model. By exploring the full efficiency field of the gearbox, it is possible to build a mathematical model that can be used to implement the propulsion unit model in an energy optimization algorithm for a given vehicle, engine-propulsion unit-vehicle combination.

## 1. Introduction

Nowadays, one of the most significant research directions in sustainable road transport is the wide expansion of the development of electric vehicles. Thanks to a Hungarian patent, it became possible to research a gear with rolling elements suitable for compensating the difference between the speed of electric motors and vehicle wheels (Bogár, 2015). When examined from a geometrical point of view, the shaft arrangement and gear ratio of the rolling element gear unit show great variability, so it is suitable for the implementation of almost any construction (gear ratio, geometrical arrangement).
In the article, the energy analysis of the roller gearing gearbox is presented. In technical life, especially in the case of propulsion units used in the vehicle industry, the energetic behavior of propulsion units is of great importance, which typically means the discovery of propulsion unit loss sources and the determination of their magnitude (Concli, 2016). This is particularly important for electric vehicles, as these losses have a significant impact on the vehicle's range (Németh and Fischer, 2021).
The total friction loss in the linear ball gear is the sliding and rolling friction loss in the ball-track connection and the friction loss in the return path, which occurs between the balls and the return path (Houpert, 2002). The friction loss in the system can be described well with some empirical relationships. These relationships take into account the most important parameters that develop during driving, such as the load, rotational speed, lubricant, surface roughness, friction coefficient, etc. (Olaru et al. 2005). For the relationships describing the phenomenon, the friction forces and torques acting on the driving and driven discs must be accurately estimated at the various working points (Polák, 2021).
In the last chapter, a geometric parameter is presented. This parameter is the diameter of the ball ($d_w$), the effect of which is examined on the efficiency field of the gear unit by changing it (Szalai et al., 2022). This parameter is interesting because increasing the size of the ball is favorable in terms of strength but not energetically (De Guido et al., 2019).

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## 2. Presentation of roller gearing gearbox

The essence of the operation of the roller gearing gearbox is that ball(s) are used as transmission elements to transfer the movement and torque between the driving and driven wheels, as shown in Figure 1 (Polák, 2021). The ball enters the drive at a specific point and rolls into the converging grooves on the surface of the two discs. The groove transfers the drive from the driven wheel to the drive wheel through the wall (Lakatos, 2010). As the two discs rotate together, the ball travels along the grooves at their intersection point, and at the end of the grooves, the ball leaves the connection. Balls that are out of contact are guided back to the entry point on a leading path (Polák, 2021).

Knowing the operation of the propulsion unit, it can be seen that the examination and analysis of the ball-track relationship are crucial from the point of view of the examination of the behavior of the propulsion unit, which is why this has already been done before.

A physical and mathematical model of the drive is prepared in parallel with the examination of the ball-track relationship, with the help of which the drive can be modeled and the energetic behavior of the drive can be estimated in the pre-design phase (Polák, 2021).

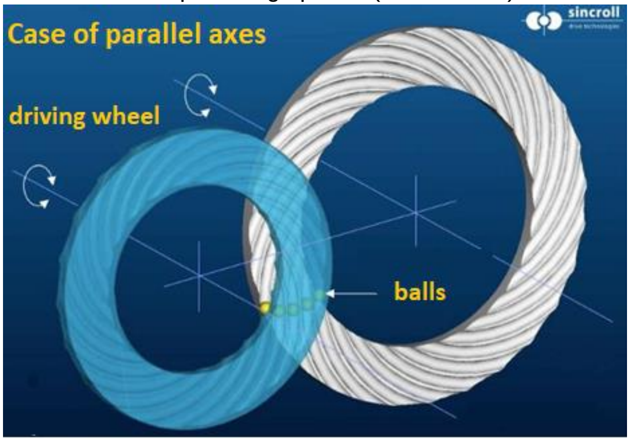


*Figure 1: The 3D model of the roller gearing gearbox with parallel axis arrangement, which is the basis of the physical model*

## 3. Losses in the gearbox

In order to model the gear unit, it is necessary to know the sources of losses in the gear unit, so they are presented in the following (Polák, 2021).

It is important to note here that the losses were not determined in power, but in torque.

The sources of loss are the following:

- The moment of the rolling friction loss of the ball and the track, Nm $T_{fl}$,
- Loss torque of the bearing friction, Nm $T_{bf}$,
- Loss torque of oil turbulence, Nm $T_{ot}$,
- Loss torque of air turbulence, Nm $T_{at}$,
- Torque of the seal friction loss, Nm $T_{sf}$,

Knowing the sources of loss, the loss torque of the roller gearing gearbox at all working points of its operating range can be determined with the following relationship:

$$T_{\sum l} = T_{fl} + T_{bf} + T_{ot} + T_{at} + T_{sf} \tag{1}$$

It can be seen from Eq(1) that the total loss torque ($T_{\Sigma l}$) at certain operating points of the gear unit can be determined as the sum of the loss sources (Polák, 2021).

## 4. Construction of the roller gearing gearbox model

When constructing the model, it can be clearly seen that some of the loss sources are load-dependent, and some are speed-dependent, as shown in Figure 2. A quasi-stationary model is used, the solution of which is done using an iteration method. Based on this, the torque balance can be written with the following Eq(2), where it can be seen that the torque exerted by the engine ($T_{mot}$) must cover the torque demand arising from the traction force ($T_{tf}$) and the sum of the internal loss torque of the gear unit ($T_{\Sigma l}$):

$$T_{mot} = T_{\Sigma l} + T_{tf} \tag{2}$$

Since the losses of the gear unit depend on the torque ($T_{mot}$) and speed ($\omega$) of the motor, it is necessary to provide an equation describing the motor torque in an implicit form of Eq(3):

$$T_{mot} = T_{\Sigma l}(T_{mot}, \omega) + T_{tf} \tag{3}$$

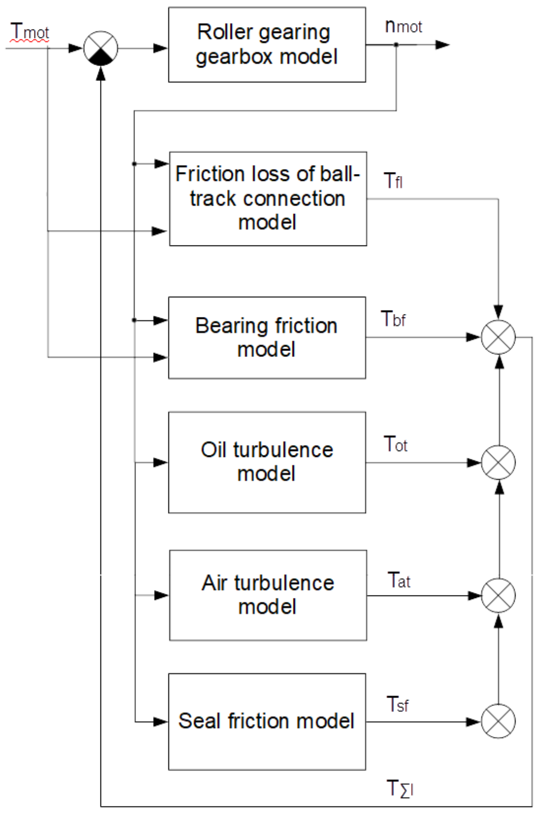


*Figure 2: Construction of the propulsion unit model*

## 5. Presentation of the operation of the created mathematical model through a case study

In order to demonstrate the operation of the model, as a first step, the most important geometric, lubrication and operational parameters of the examined roller gearing gearbox are determined (Caporusso et al., 2022).

### 5.1 Presentation of the most important parameters of the tested roller gearing gearbox

The most important geometric, lubrication, and operational parameters of the ball gear selected for testing are listed in Table 1. It is important to note here that there are parameters in the table that cannot be used directly, as they are constantly changing during the operation of the gear.
This parameter is the radius of the grooves on the surface of the discs mounted on the axles, as shown in Figure 3. The change in the radius is significant because it determines the location of the ball from the center of rotation of the shaft ($r_i$), and as it changes, it leads to a pulsation of torque and traction force, as shown in Figure 3.
Using the motor torque and the radius, the force ($F_{di}$) on the ball in the connection and the groove wall can be determined using the following relationship Eq(4):

$$F_{di} = \frac{T_{mot}}{r_i} \tag{4}$$

*Table 1: The parameters used for testing the gearbox*

| No. | Parameters | Values |
|---|---|---|
| 1. | ball diameter | $d_w$ = 6 mm |
| 2. | average radius of the tracks of drive disk in the x-y plane | $r_{m1}$ = 47.5 mm |
| 3. | average radius of the tracks of driven disk in the x-y plane: | $r_{m2}$ = 65 mm |
| 4. | total number of balls: | $Z$ = 86 pcs |
| 5. | number of connected balls: | $z$ = 5 pcs |
| 6. | contact angle: | $\alpha$ = 20° |
| 7. | surface roughness of the ball: | $R_g$ = 0.6 µm |
| 8. | Surface roughness of the track of the drive and driven disk: | $R_t$ = 1.6 µm |
| 9. | coefficient of friction between balls: | $\mu_b$ = 0.1 |
| 10. | lubricant viscosity: | $\eta_0$ = 0.1 Pas |
| 11. | For the parameter *Y* without dimension, it is assumed to be 0.34 for both the drive disc and the driven disc. | $Y$=0.34 |
| 12. | the curvature parameters f1 and f2 for the drive disk and the driven disk are 0,53. | $f_1$, $f_2$=0.53 |
| 13. | The drive motor torque was varied between 0… 100 Nm. | $T_{mot}$=0… 100 Nm |
| 14. | driving disc small diameter | $r_{min}$= 25 mm |
| 15. | driving disc large diameter | $r_{max}$= 61 mm |
| 16. | driven disc small diameter | $r_{min}$= 128.5 mm |
| 17. | driven disc large diameter | $r_{max}$= 215 mm |

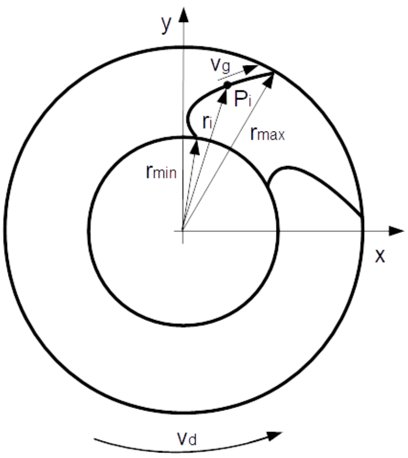


*Figure 3: The change in the distance of the connected ball from the center of rotation of the shaft during rotation*

In order to exclude torque and force pulsation, the radius was determined as a constant as the arithmetic mean of the minimum ($r_{min}$) and maximum ($r_{max}$) radius Eq(5).

$$r_i = \frac{r_{min} + r_{max}}{2} \tag{5}$$

## 6. The efficiency of the selected roller gearing gearbox based on the model

After defining the individual loss sources and the model, the energetic behavior of the drive can be specified in the entire operating range with the following general relationship Eq(6):

$$\eta = \frac{P_{mot_i} - P_{\Sigma l_i}}{P_{mot_i}} \tag{6}$$

However, since the model calculates with the loss torques for the given test, they can also be used to determine the efficiency of the gear over the entire operating range with the following relationship Eq(7):

$$\eta = \frac{T_{mot_i} - T_{\sum l_i}}{T_{mot_i}} \tag{7}$$

The above equation Eq(7) can be used to determine the efficiency of the gearbox at given operating points. When a surface is fitted to the points, the gear efficiency field is formed over the full operating range, as shown in Figure 4.

The test range of the roller gearing gearbox was chosen in such a way that the traction demand and wheel speed of a vehicle weighing about 500 kg and traveling at a maximum speed of about 70 km/h were taken into account (Twheel=0… 100 Nm, nwheel=0… 600 rpm).

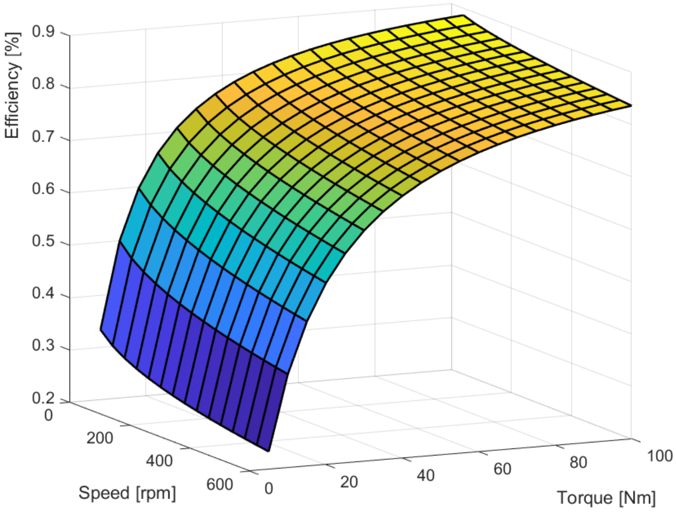


*Figure 4: The roller gearing gearbox efficiency field is in the entire operating range ($T_{wheel}$=0… 100 Nm, $n_{wheel}$=0… 600 rpm)*

## 7. Examination of some geometrical parameters of the roller gearing gearbox

Examining the gear geometric parameters, two geometric parameters seem interesting and important: one is the ball diameter ($d_w$) and the other is the average groove radius in the x-y plane ($r_x$). These are the parameters that typically determine the efficiency of the gear in its operating range. The parameter vector of the roller gearing gearbox can be written with the following relation Eq(8):

$$\boldsymbol{p} = [d_w, \quad r_x] \tag{8}$$

In this article, the effect of the diameter of the ball was investigated by examining the change in the efficiency of the propulsion unit in the entire operating range with four different diameter balls, as shown in Figure 5.

The ball diameters were 2 mm, 4 mm, 8 mm, 12 mm.

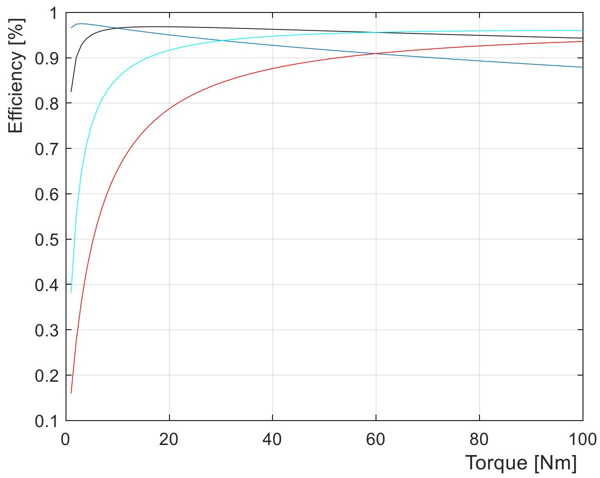


*Figure 5: The effect of a change in ball size on the efficiency of the roller gearing gearbox: $n_{wheel}$= 600 rpm, $T_{wheel}$= 1… 100 Nm (blue line: $d_w$=2mm, black line: $d_w$=4mm, cyan line: $d_w$=8mm, red line: $d_w$=12mm)*

It is clear from the obtained efficiency curves that the optimal ball size is not easy to determine since the gear unit built with a small diameter ($d_w$= 2 mm) ball has a good efficiency in the initial stage, but it continues to deteriorate as the load increases. This is probably due to the load transfer on a small surface and is due to the resulting overload.

As the ball size increases ($d_w$= 4mm, 8mm), the efficiency of the gear is worse in the initial low-load stage, but it improves significantly in the higher-load ranges, so this means that the energetically optimal behavior (deformation, lubricating film) develops in the higher-load ranges. By further increasing the ball size ($d_w$= 12 mm), the maximum efficiency does not increase, but the losses increase in the initial stage.

Based on his examination, the tendency of the optimal ball size, whether the ball size should be as small or as large, cannot be determined, but must always be chosen according to the specific load conditions (Lakatos and Titrik, 2015).

## 8. Conclusions

The mathematical model presented in this paper opens up the possibility of modeling the energetic behavior of the roller gearing gearbox during the design phase.

By setting up the equation of motion of the roller gearing gearbox, the propulsion unit can be tested in stationary (at work point) and nonstationary (acceleration, braking) states.

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