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TOOTH FRICTION LOSS IN SIMPLE PLANETARY GEARS

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Abstract
The efficiency of planetary gears mainly depends on the tooth friction loss. In order to analyze the possibilities to reach the highest efficiency it is necessary to calculate the tooth friction loss. There are some methods for calculation of the tooth friction published in literature, which can be used for the analysis. Using a method developed by Duda to calculate the efficiency of tooth gears and another method suggested by Bartz to calculate the average coefficient of friction between the mating teeth we executed detailed investigation of the parameters influencing the tooth friction loss. Calculations were performed to determine the efficiency of simple planetary gears, evaluating the influence of the tooth profile on the tooth friction loss at different gear ratios. The geometries of tooth profile were changed with different modules and different addendum modifications of the gears.
Key words: *Planetary gear, tooth efficiency, addendum modification, tooth friction loss, tooth geometry*

1. INTRODUCTION

Heavy-duty planetary gears are used in many branches in industry thanks to their well known advantages comparing to common types of gears. Sophisticated compound planetary gearboxes are able to meet more severe requirements; therefore they find applications in power stations, in wind turbines and in hydroelectric power plants. They contain simple planetary gears and are planned to transmit megawatts or even more power. Owing to economy of power transmission, reduction of pollution of environment and also to inhibit the heat generation in gear boxes it is very important to reach the highest efficiency of the constituent elementary planetary drives.

2. EFFICIENCY OF PLANETARY GEARS

The sources of energy losses of a simple planetary gear drives are the followings:

- friction loss between the mating teeth
- friction loss in the bearings

- friction loss at the seals
- energy losses owing to lubricant churning
- energy loss of air-drag.

The main source of energy loss is the tooth friction of gears depending on the arrangements of the meshing gears and the power flow inside the planetary gear drives. Besides these factors the tooth friction loss is influenced by the applied load, the entraining speed, the geometry of gears, the roughness of mating surfaces and the viscosity of lubricant. From these parameters the designer of planetary gear drives can modify the geometry of tooth profile in order to reach a beneficial high efficiency with decreasing the tooth friction loss.

2.1 Tooth friction loss

The efficiency of planetary gears mainly depends on the tooth friction loss. In order to analyze the possibilities to reach the highest efficiency it is necessary to study the parameters determining the tooth friction loss. There are some methods for calculation of the tooth friction published in literature [1, 2, 3, 4, 5], which can be used for the analysis.

The equation developed by Duda [1] takes into consideration of the meshing conditions of tooth wheels.

$$\eta_z = 1 - \mu_m \pi (E_1 + E_2) \left[\frac{1}{z_1} \pm \frac{1}{z_2} \right] \tag{1}$$

where η_z – tooth efficiency, mm – mean coefficient of friction of meshing teeth, E_1, E_2 – parameters taking into consideration of the meshing conditions [1], z – number of teeth on the given gear.

The mean coefficient of tooth friction can be calculated using equation published by Bartz [3].

$$\mu_m = 0.12 \left[\frac{w_t R_a}{\eta_M v r} \right]^{0.25} \tag{2}$$

where $w_t = F_N/b$ – applied normal load/width of gear b, R_a – average surface roughness (CLA). η_M – viscosity at operating temperature, v – entraining speed, r – effective curvature in pitch point.

Using the method developed by Duda (1) to calculate the efficiency of tooth gears and the formula suggested by Bartz (3) to calculate the average coefficient of friction between the mating teeth we executed detailed investigation of the parameters influencing the tooth friction loss.

2.2 Influence of the tooth geometry on the efficiency of gears

Calculations were performed to determine the efficiency of simple planetary gears, evaluating the influence of the tooth profile on the tooth friction loss at different gear ratios. The higher reduction of speed and higher efficiency of this drive belong the operational conditions, where the sun gear *2* drives (input) and the carrier *k* is driven (output), while the ring gear *4* is fixed.

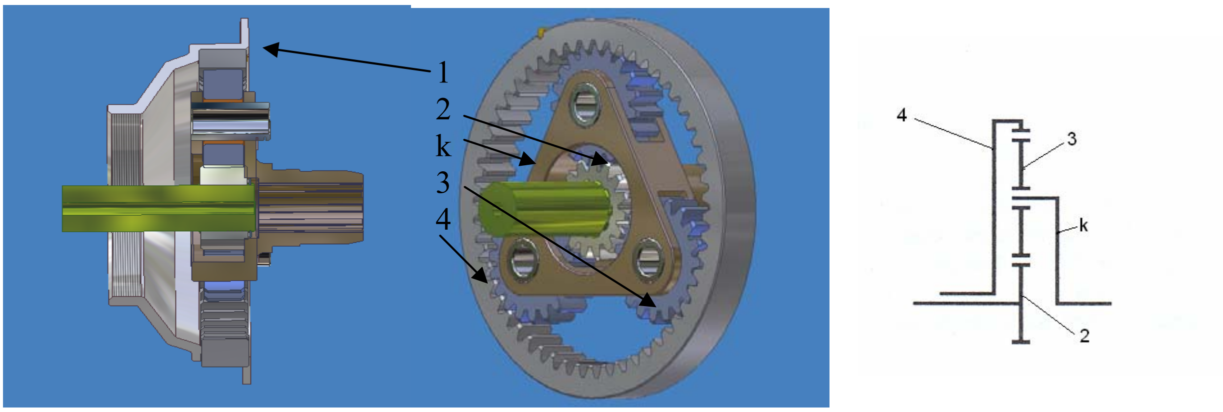

Fig. 1 The main parts of a simple planetary gear
gearbox housing (1); sun gear (2); planetary carrier (k); planet gear (3); ring gear (4)

The calculation were performed for a planetary gear planned to transmit 800 kW power at driving speed of 1440 rpm, at different gear ratios range from $i = 3$ to 10.
Other important parameters are the followings:

σ_F [MPa]	η_M [MPas]	R_{a23} [μm]	R_{a34} [μm]	P_{in} [kW]	n_{in} [1/s]	β [°]	N
500	60	0,63	1,25	800	24	0	3

where σ_F - bending strength of teeth, η_M – viscosity at operating temperature, R_a – average surface roughness (CLA), P_{in}- driving power, n_{in} – driving speed, β – helix angle, N – number of planet gears.

The geometries of tooth profile were changed using different modules and different addendum modifications of the gears. The pressure angle of gear pairs planet gear/ring gear was a constant: $\alpha_{34} = 20°$, while the pressure angle of the gear pair sun gear/planet gear was changed using different center distance parameters: y = 0, 0,5 or 1. ($y = (a_w - a)/m$, a_w – center distance, a – elementary center distance, m – module). In order to reach the same load carrying capacity, the width/diameter ratios were changed.

Beside the efficiency of planetary gears also the masses of gears were calculated in order to take into consideration the influence of the chosen parameters on the weights of planetary gear.

To calculate the mass of the sun gear, planet gears and ring gear, the following simple model was used (where b – width of gear):

$$m_{234} = b \cdot \rho \cdot \pi \cdot \left[r_{w2}^2 + N \cdot r_{w3}^2 + \left(12 \cdot m \cdot r_{w4} + 36 m^2 \right) \right]$$

The calculations were performed for two cases:

Case 1. The center distance a_w = 324 [mm] is constant.

Case 2. The pitch circle radius of the ring gear is constant r_{w4} = 576 [mm].

The result of calculation are presented in (Fig 2-19)

On the diagrams only those results can be seen, where the gears have no undercut or other interferences like too thin top land.

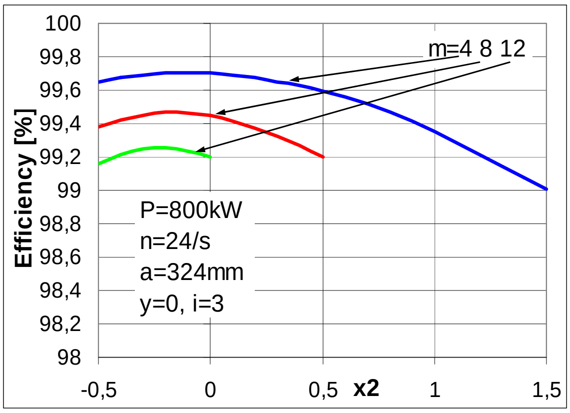

Fig. 2 Efficiency of planetary gears as a function of addendum modification of sun gear. i=3, y=0 Case 1

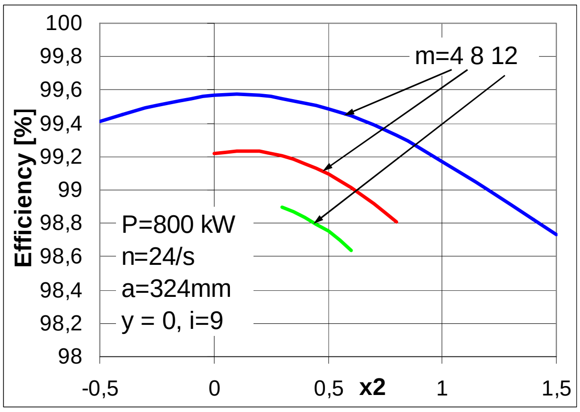

Fig. 3 Efficiency of planetary gears as a function of addendum modification of sun gear. i=9, y=0 Case 1

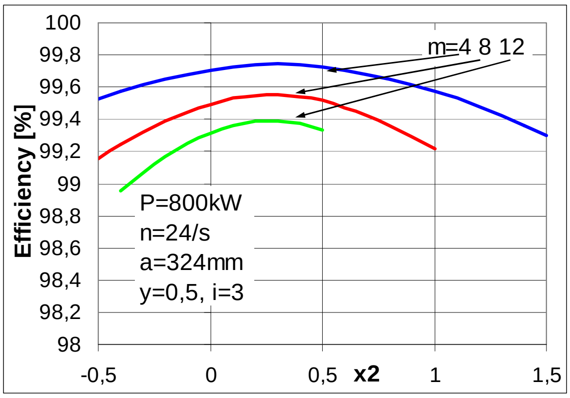

Fig. 4 Efficiency of planetary gears as a function of addendum modification of sun gear. i=3, y=0,5 Case 1

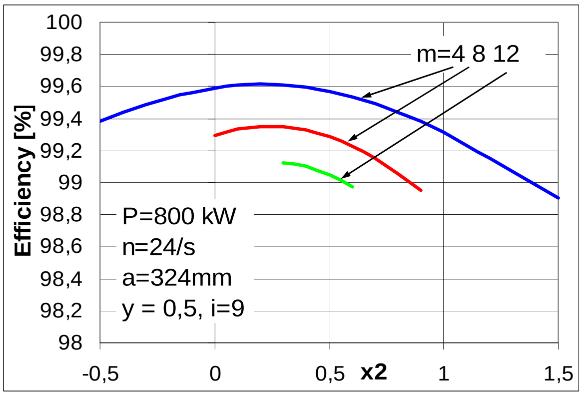

Fig. 5 Efficiency of planetary gears as a function of addendum modification of sun gear. i=9, y=0,5 Case 1

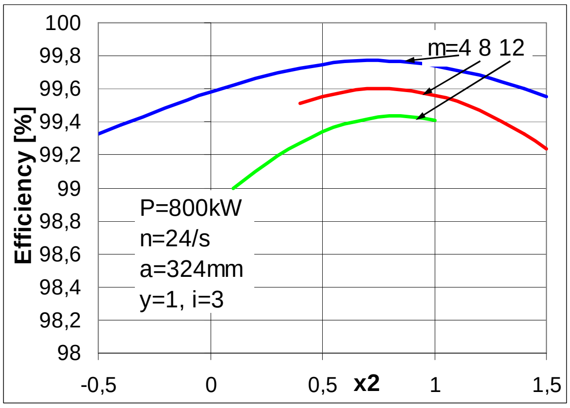

Fig. 6 Efficiency of planetary gears as a function of addendum modification of sun gear. i=3, y=1

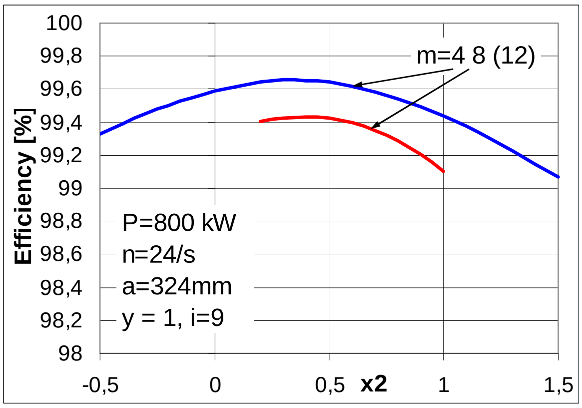

Fig. 7 Efficiency of planetary gears as a function of addendum modification of sun gear. i=9, y=1

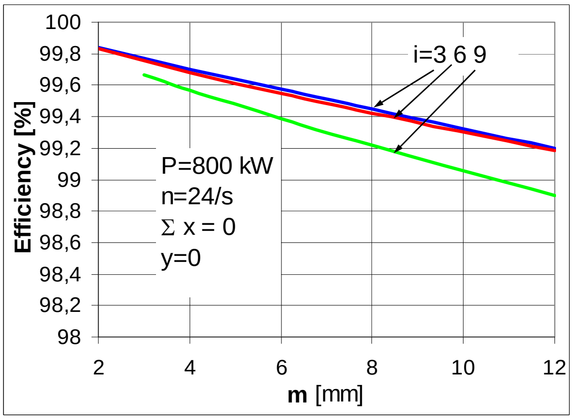

Fig. 8 Efficiency of planetary gears as a function of module

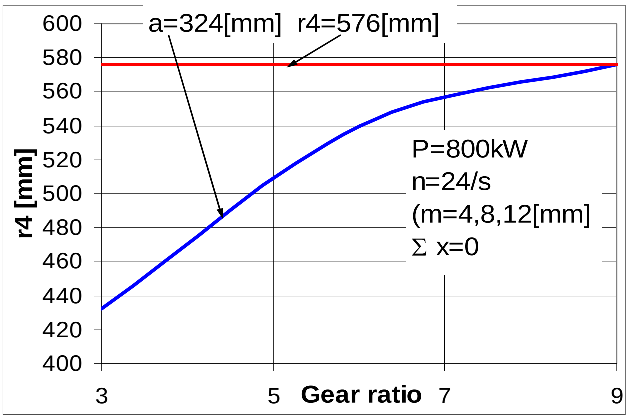

Fig. 9 The radius of ring gears as a function of gear ratio. Case 1-2

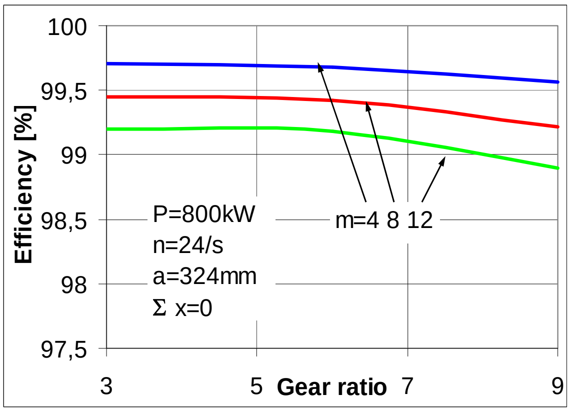

Fig. 10 Efficiency of planetary gears as a function of gear ratio. Case 1

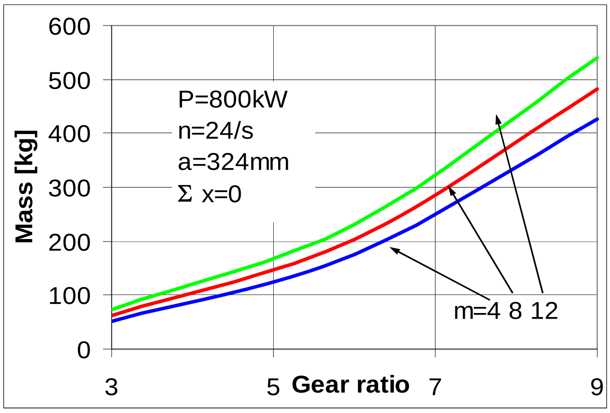

Fig. 11 Mass of planetary gears as a function of gear ratio. Case 1

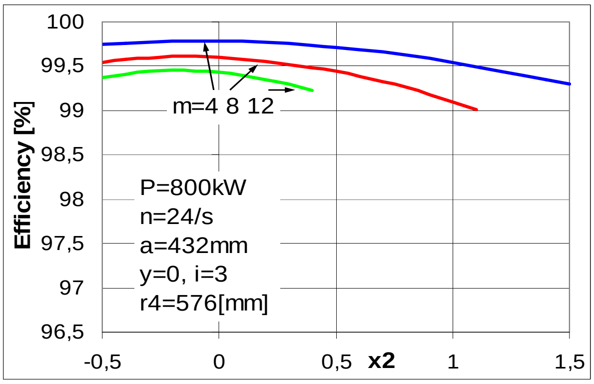

Fig. 12 Efficiency of planetary gears as a function of addendum modification of sun gear. i=3, y=0 Case 2

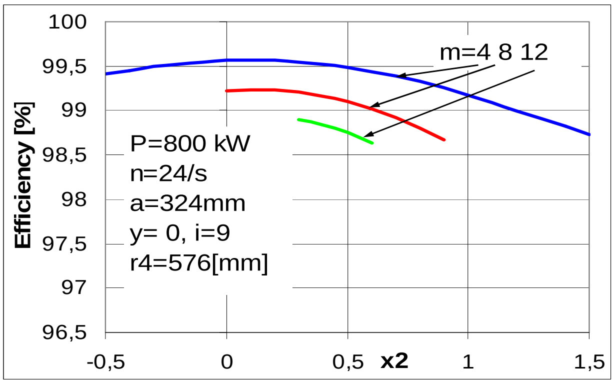

Fig. 13 Efficiency of planetary gears as a function of addendum modification of sun gear. i=9, y=0 Case 2

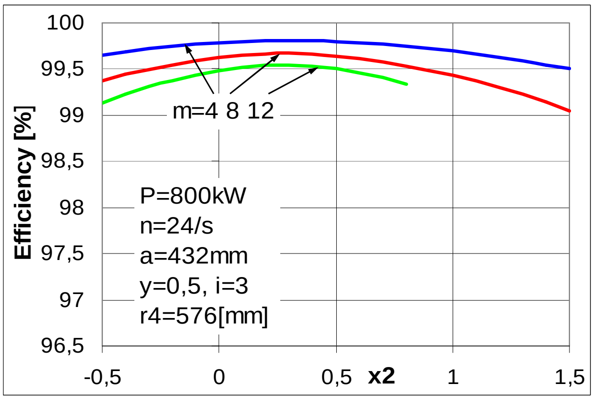

Fig. 14 Efficiency of planetary gears as a function of addendum modification of sun gear. i=9, y=0,5 Case 2

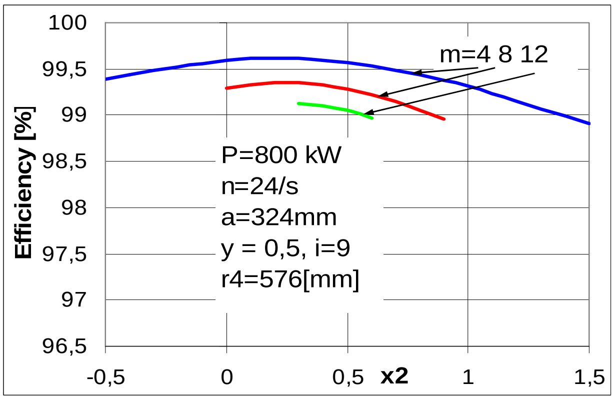

Fig. 15 Efficiency of planetary gears as a function of addendum modification of sun gear. i=9, y=0,5 Case 2

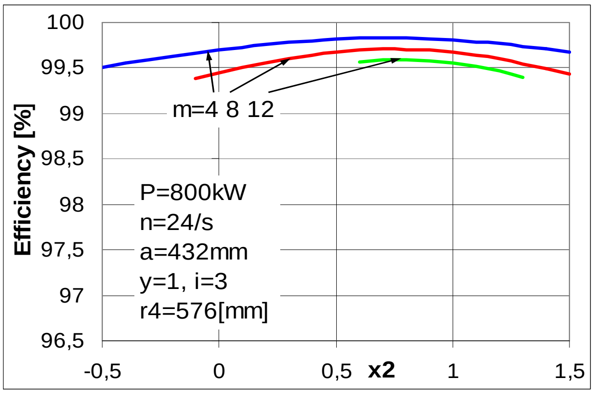

Fig. 16 Efficiency of planetary gears as a function of addendum modification of sun gear. i=3, y=1 Case 2

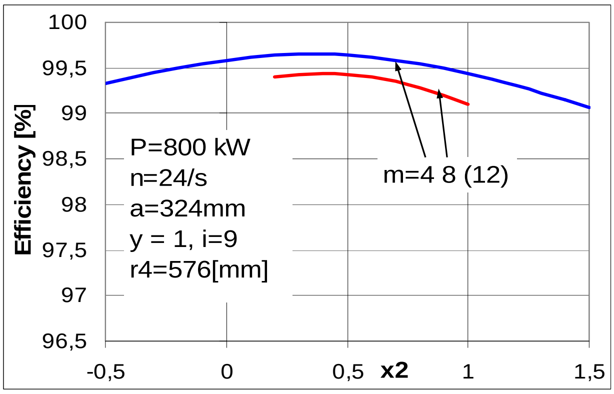

Fig. 17 Efficiency of planetary gears as a function of addendum modification of sun gear. i=9, y=1 Case 2

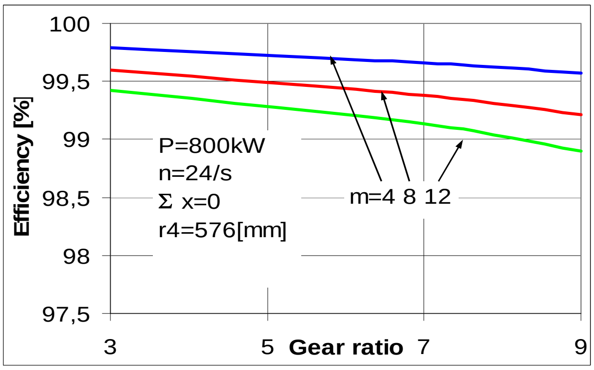

Fig. 18 Efficiency of planetary gears as a function of gear ratio. Case 2

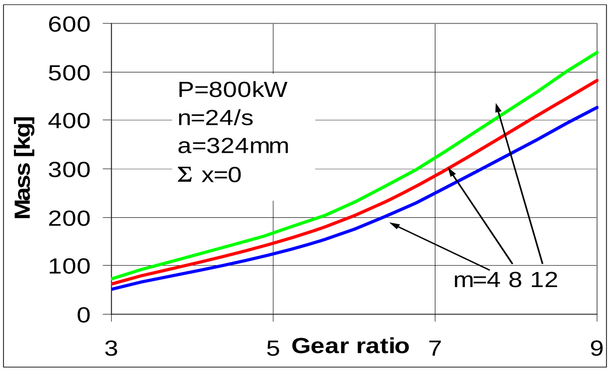

Fig. 19 Mass of planetary gears as a function of gear ratio. Case 2

3. CONCLUSIONS

Comparing the results of the calculation presented in Fig 2-19 the following can be stated:

- With variation of addendum modification of sun gear *2* the efficiency of investigated planetary gear can be modified and reach its maximum at an optimum value of x_2.
- Increasing the pressure angle α_{w23} above 20°, the efficiency of planetary gear enhance a little, while the optimum value of addendum modification x_2 of sun gear *2* also increases.
- Increasing the gear ratio of planetary gear drives decreases their efficiency.
- Increasing the module decreases the efficiency of planetary gear drive.
- The main differences between the construction case1 and case2 that the diameter of ring gear is smaller at lower gear ratios if the center distance is constant (Case1).
- The investigated planetary gear drives has a very high efficiency in the optimum regions (above 90% even at high gear ratio i = 10).

4. REFERENCES

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