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# THE EFFICIENCY OF WORM GEARS LUBRICATED WITH OILS OF MINERAL AND SYNTHETIC BASES

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**Summary**

This paper presents the results of an experimental method for determining the efficiency of worm gears. Research into worm gears is primarily aimed at increasing the load carrying capacity, prolongation of life cycle and/or achieving higher efficiency, which can result in a reduction in exploitation costs. A larger share of sliding movement with respect to sliding-rolling movement of gearing contributes to the quiet operation but, on the other hand, causes a significant power loss. Very often, the load limit is not put by the load carrying capacity of gearing but by the ability to carry away the heat caused by friction. In order to design a worm gearbox that would be closer to the optimal solution from the temperature point of view (the heat), it is necessary to determine the operation losses. Besides the geometry of gearing, which influences the efficiency by forming conditions required for the creation of hydrodynamic lubrication, the applied lubricant also plays a significant role. In the research presented in this paper, mineral and synthetic oils were used for the combination of materials CuSn12/16MnCr5 used for gears operating at variable output load and two different rotational speeds.

*Key words:* *worm gears, efficiency, lubrication*

## 1. Introduction

Given a much higher proportion of sliding relative to rolling movement, worm gears have far more friction between the teeth sides than in the case of cylindrical and cone gears, which results in a significantly lower efficiency [1]. It is known that the efficiency of a worm pair depends on [2-6]:

- Lubricant,
- Sliding speed (input rotation $n_1$),
- Surfaces roughness,
- Load,
- Paired materials,
- Worm profile, and
- Temperature (material and design of gearbox casing, natural or forced cooling).

Because of the nature of worm gears, when dealing with their efficiency, the influence of certain factors cannot be neglected and is considered by almost every researcher. On the other hand, for the sake of simplicity of calculation or conducting of experiment, the influence of some factors is presumed or set to be constant or isolated.

Losses in the teeth of gears are increased by the share of sliding [7], especially for higher transmission ratios. This causes significant heat load of a worm pair. Due to friction and the amount of heat it generates and to limitations imposed by allowable operating temperatures of lubricants, continuous operation often has a limited amount of transmittable power. Worm gears usually operate at the temperatures of up to 70°C (even up to 90°C). Because of that, one of the most important requirements for lubricants involved in transmissions that have to be fulfilled is the stability of viscosity, which is temperature-dependent to a higher or a lesser degree.

The aim of the research is to determine a degree of efficiency as a function of a set of parameters applied through the operating conditions in the test worm gear for two types of lubricants: a synthetic and a mineral oil. The research was carried out for the combination of materials CuSn12 / 16MnCr5 at variable output load and two different rotational speeds. It was carried out on the test stand with a standard gearbox purchased on the market.

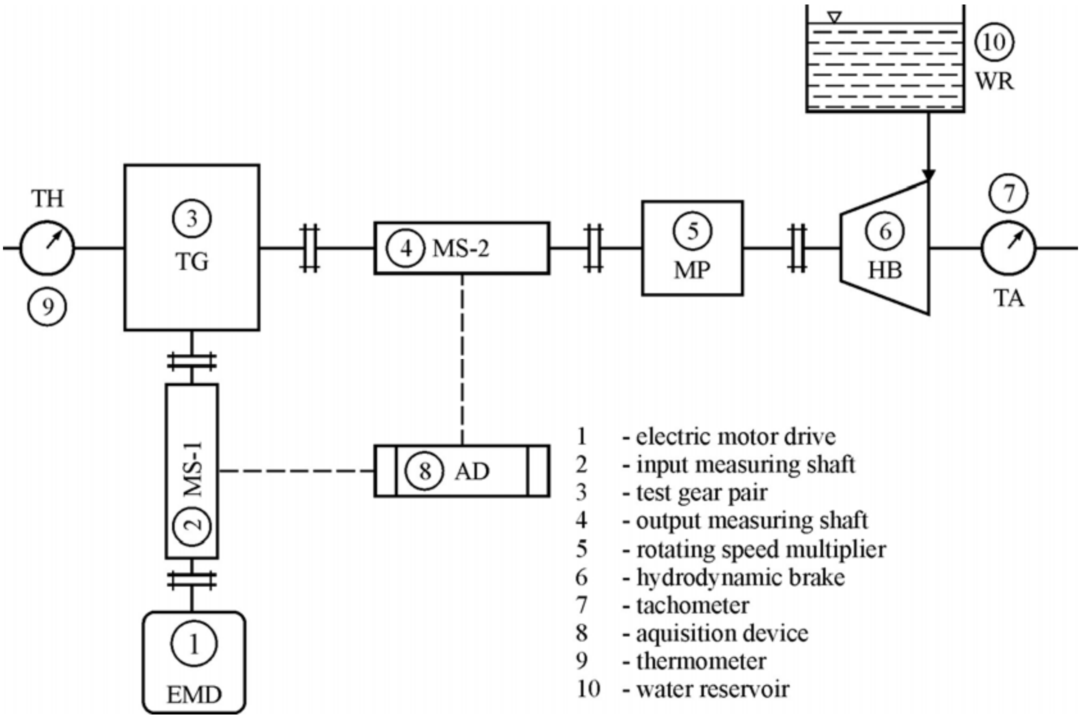


**Fig. 1** Schematic of the open-power test stand for worm gear.

## 2. Experimental testing of efficiency

### 2.1 The test stand working principle

Figure 1 shows a schematic of the test stand [1-2]. The test stand is of an open-power type where the output power is completely transformed into heat by means of hydrodynamic breaking.

The test stand is built with an electric motor drive (1) controlled by a frequency converter to enable the variation in rotational speed. The gearbox containing the test gear pair (3) is connected to the motor via the input measuring shaft (2). Torque at the output of the test gear is measured with the output measuring shaft (4) that is connected to an additional gearbox (5). The function of this gearbox is to multiply the number of revolutions at the input of the hydraulic brake (6) which is providing load through the fluid friction between a series of static and rotating tiny plates and water that the brake contains. It is necessary because the performance of the brake greatly depends on the rotational speed of the shaft. Rotational speed is measured with a tachometer (7). The hydrodynamic load is maintained manually. Both measuring shafts and the thermometer (9) are connected to the measuring amplifier (8) allowing continuous data acquisition and monitoring via a personal computer.

Measuring of the efficiency of the gear pair is indirect as it is based on the measurement of the input ($T_1$) and output ($T_2$) torques (metering shafts (2) and (4)) and the known transmission ratio of the gear pair which is taken as a constant value:

$$\eta = \frac{T_2}{\mathrm{u} \cdot T_1} \tag{1}$$

## 2.2 Test gear pair

The gearbox is a standard product with an axial distance of 90 mm (Figure 2). The housing is finned and cooled by natural air convection.

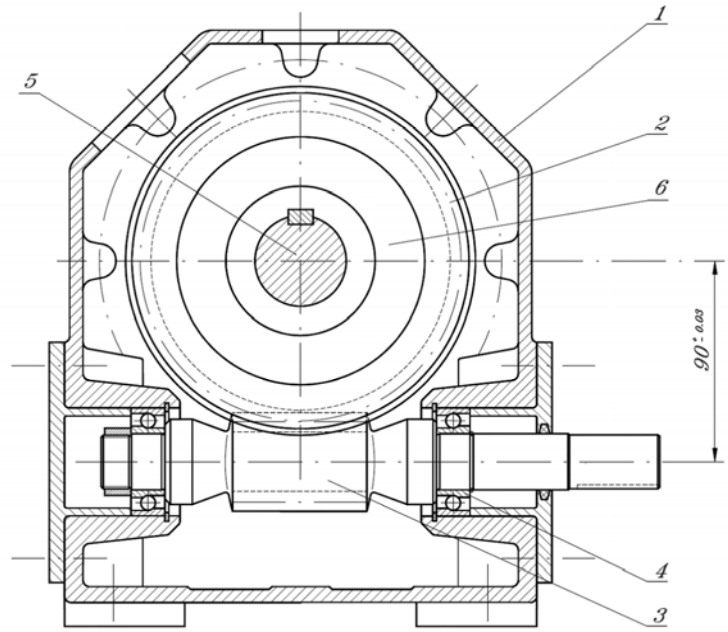


**Fig. 2** Cross-section of the test gearbox : 1 - Housing, 2 - Worm wheel, 3 - Worm, 4 - Rolling bearing, 5 – Output shaft, 6 - Worm wheel hub

**Table 1** Data sheet of the test worm gear pair

| Parameter | | A- profile of worm |
|---|---|---|
| Centre distance | $a$ | 90 mm |
| Number of worm teeth | $z_1$ | 2 |
| Number of worm wheel teeth | $z_2$ | 36 |
| Axial module | $m$ | 4 mm |
| Factor of profile shift | $x_2$ | 0 mm |
| Pitch radius of worm | $r_{m1}$ | 18 mm |
| Outside radius of worm | $r_{a1}$ | 22 mm |
| Throat radius of worm wheel | $r_{a2}$ | 76 mm |
| Outside radius of worm wheel | $r_{A2}$ | 78 mm |
| Pressure angle | $\alpha_N$ | 20° |
| Length of the worm | $b_1$ | 60 mm |
| Width of the worm wheel | $b_2$ | 32 mm |

All gearbox bearings are rolling bearings. Gearbox housing is provided with openings for monitoring procedures and other necessary adaptations. The worm gear pair has the teeth installed by the manufacturer with special attention devoted to accuracy. The profile of the worm gear in axial cross-section is a straight line (A-profile, Figure 3). Data for the pair are given in Table 1.

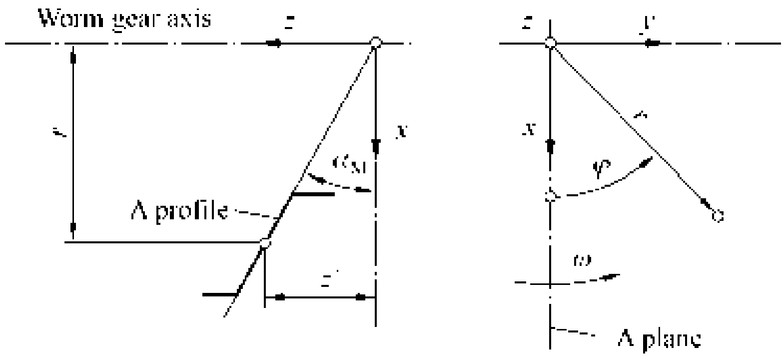


**Fig. 3** Worm gear profile (ZA - Archimedes spiral worm) where $r$, $z'$ and $\varphi$ are the coordinates of the contact point in the cylindrical coordinate system

## 2.3 Applied materials and lubricant

The worm gear material is 15MnCr5 steel (steel for cementing) with the following chemical composition: 0.16% C, 1.15% Mn, 0.95% Cr, while the worm wheel material is tin bronze alloy with 12% Sn, i.e. CuSn12 containing 88% Cu, 12% Sn (permitted additives 0.1% Sb, 0.2% Fe, 0.01% Al, 0.5% Ni).

Generally, industrial worm gears are used with unalloyed mineral oils or mineral-based oils and lubricants of synthetic origin which, in some cases, provide incomparably better results than mineral oils regarding viscosity-temperature dependence, seizure torque-temperature dependence and pitting resistance-temperature dependence [8,9]. For the purpose of this research, two types of lubricants were used (Table 2): oil of mineral origin and oil of synthetic origin.

**Table 2** Basic features of applied lubricants

| No | Oil type | Viscosity, cSt at 50°Ct | Viscosity, cSt at 100°Ct | Density ρ, g/cm$^3$ | Ignition temp., °C | Viscosity index |
|---|---|---|---|---|---|---|
| 1 | Mineral oil | 100 | 19.3 | 0.897 | 247 | 100 |
| 2 | Synthetic oil | 94 | 23.2 | 1.03 | 283 | 140 |

## 2.4 Worm gearing efficiency

Direct measurement of gearing efficiency is not possible on this type of test stand. The reasons for this lie in the fact that, in addition to losses that occur in the gear meshing $P_{gz}$, there are losses in the bearings $P_{gl}$ and load losses $P_{go}$. However, based on the torque measurement on the input and the output shaft of the gearbox, accompanied by the empirically-based estimation of other losses, it is possible to determine the overall efficiency of gears according to the following formula:

$$\eta = \frac{P_2}{P_1}. \tag{2}$$

Power at the input shaft $P_1$ of the worm gear, taking into account all of these losses, can be expressed as

$$P_1 = P_2 + P_G = P_2 + P_{go} + P_{gl} + P_{gz} \tag{3}$$

where the values used represent the following:

- $P_1$ – power at the input shaft
- $P_2$ – power at the output shaft
- $P_G$ – overall loss
- $P_{go}$ – load losses (sealing and oil churning)
- $P_{gl}$ – losses in the bearings
- $P_{gz}$ – losses in the gear meshing.

Efficiency of the gearing is calculated as:

$$\eta_z = \frac{P_2}{P_2 + P_{gz}} . \tag{4}$$

Rearrangement of equation (3) into $P_2 + P_{gz} = P_1 - (P_{go} + P_{gl})$ and substitution in the denominator of equation (4), after the extraction of $P_1$ and the substitution of the numerator with equation (2), result in the formulae for the calculation of the gearing efficiency:

$$\eta_z = \frac{P_2}{P_1 - (P_{go} + P_{gl})} = \frac{P_1}{P_1} \cdot \frac{P_2 / P_1}{\left(1 - \dfrac{P_{go} + P_{gl}}{P_1}\right)} = \frac{\eta}{\left(1 - \dfrac{P_{go} + P_{gl}}{P_1}\right)} . \tag{5}$$

Equation (5) shows the dependence of the efficiency of the gearing on the overall level of efficiency of the gearbox. With the known values of the relation $(P_{go}+P_{gl})/P_1$ and the overall efficiency $\eta$, which is calculated from the measured torques, one can determine the value of $\eta_z$. According to [4], the dependence of losses in the bearings is given by equation $P_{gl}=0.02 \cdot P_1$. Power $P_{go}$, which can be measured at $T_2=0$, has a very modest effect on the power losses in bearings $P_{gl}$.

## 3. Results of the experiment

Before the start of the actual testing, a trial run (running-in) is performed. The trial run is done gradually according to [1], i.e. the load is increased in several phases before it reaches the nominal value. During the trial run, base oil mixed with Petroleum is used as a lubricant. The goal of mixing the oil with additional material is to accelerate the wear and precipitate the initial meshing of the pair and thus to reduce the running time of the trial period. After the trial run, the gearbox is cleaned up and the lubricant is replaced. Gears were loaded in steps, followed by continuous monitoring of the oil bath temperature and the metering of overall level of efficiency. The periods of applying individual loads were one hour per load in order to avoid the influence of adjustment due to wear. The indicator used as the finish mark was the temperature stabilization of the lubricating oil for a given load. During the experiment, the following measurements were carried out:

- Measuring of the degree of efficiency and the temperature depending on the output load of the gear for two rotational speeds at the input: 1500 $\min^{-1}$ and 710 $\min^{-1}$ and with mineral oil as a lubricant.

– Measuring of the degree of efficiency and the temperature depending on the output load of the gear for two rotational speeds at the input: 1500 $min^{-1}$ and 710 $min^{-1}$ and with synthetic oil as a lubricant.

Figures 4 and 5 show the dependence of the overall degree of efficiency on the gear output load with the use of synthetic and mineral oil for two rotational speeds at the input: 1500 $min^{-1}$ and 710 $min^{-1}$. Figures show the beginning of growth in the overall level of efficiency with an increase in the load and a subsequent decline with further growth in the load. The reason for this progress should be sought in the lubricant. At higher loads, amid huge pressure, there is a break in the molecular film of applied lubricant. This leads to an increase in the friction coefficient resulting in a decreased level of overall efficiency.

Figures 6 and 7 show the relation between the lubricating oil temperature and the load on the output side with the use of mineral and synthetic oil for two rotational speeds at the input: 1500 $min^{-1}$ and 710 $min^{-1}$.

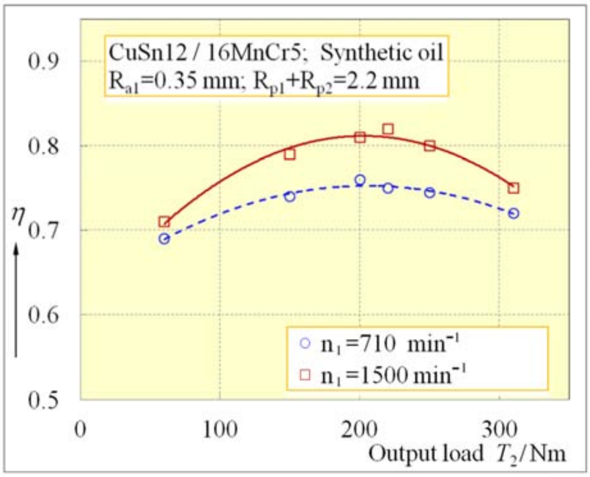


Fig. 4 Overall efficiency depending on the output load; lubrication with synthetic oil

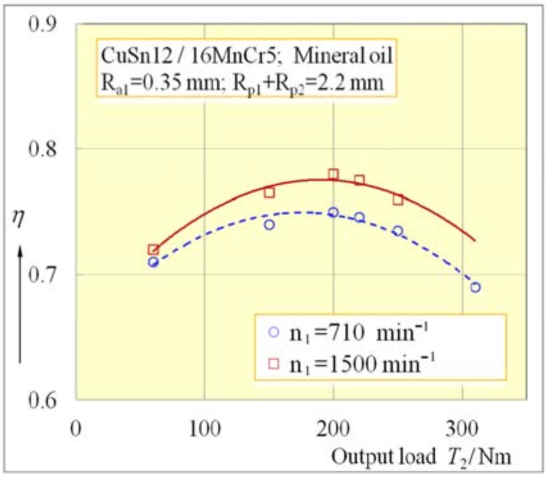


Fig. 5 Overall efficiency depending on the output load; lubrication with mineral oil

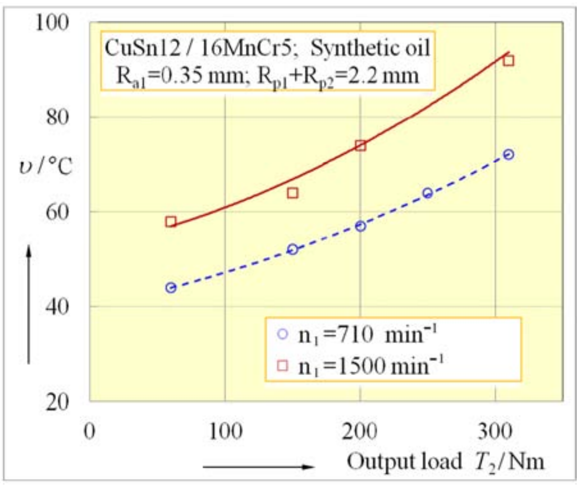


Fig. 6 Oil temperature depending on the output load; lubrication with synthetic oil

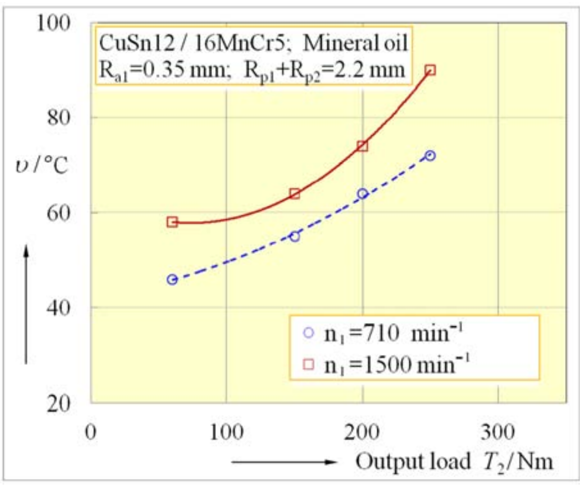


Fig. 7 Oil temperature depending on the output load; lubrication with mineral oil

Figures 8 and 9 show a comparison between the results of the degree of efficiency depending on the load output for mineral oil and those for synthetic oil and for two input rotational speeds: 1500 $min^{-1}$ and 710 $min^{-1}$.

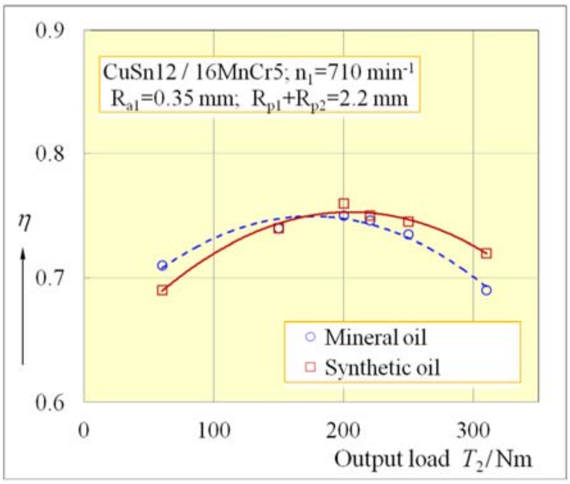


Fig. 8 Overall efficiency depending on the output load for input rotational speed $n_1$=710 $min^{-1}$

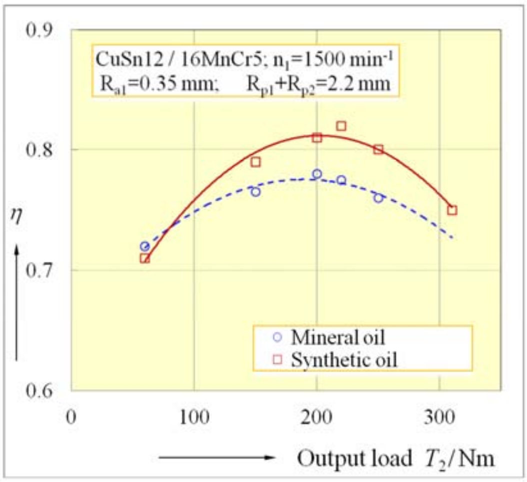


Fig. 9 Overall efficiency depending on the output load for input rotational speed $n_1$=1500 $min^{-1}$

## 4. Conclusion

On the basis of the results of conducted experiments it can be concluded that synthetic oils provide a higher level of efficiency compared to mineral oils. Mineral oils have a higher coefficient of friction compared to synthetic oils in the same operating conditions: oil temperature, sliding speed and load capacity of the gears.

The research results of overall efficiency also show a difference in the temperature load of the gear lubricated with mineral oil and synthetic oil. Synthetic oil has a much larger load capacity with respect to the temperature limit. The biggest difference in load capacity was obtained for a temperature of 70°C and the sliding speed of 1.41 m/s. In these conditions, the gear load, when using synthetic oil, increased by 40%. Less heat in the worm gear lubricated with synthetic oil is a result of an increased hydrodynamic lubrication. It occurs by reason of lower dependence of viscosity on the change in pressure and temperature. Bearing in mind the fact that worm gears operate in the boundary and the mixed friction area, the influence of lubricant adhesion is also important. Synthetic oils have far greater adhesion than mineral oils, which results in a significantly better performance.

For low sliding speeds we obtained the expected lower values of the degree of efficiency as a result of deterioration in the hydrodynamic conditions.

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