ISSN 1330-3651 (Print), ISSN 1848-6339 (Online)
https://doi.org/10.17559/TV-20220207092015
Original scientific paper

The Impact of Lubricant Viscosity and Materials on Power Losses and Efficiency of Worm Gearbox

Aleksandar SKULIĆ*, Saša MILOJEVIĆ, Dejan MARIĆ*, Lozica IVANOVIĆ, Božidar KRSTIĆ, Mladen RADOJKOVIĆ, Blaža STOJANOVIĆ

Abstract: The results of the experimental research of the power losses and efficiency of single-stage worm gearbox, which is designed for such purpose, are presented in this paper. Three worm pairs (coupled gears) ZI type with the same geometric characteristics were used in the research, where worms are made of the same material (hardened and ground steel type 42CrMo4) and worm gears are made of three different types of material (zinc-aluminium alloy ZA12, aluminium alloy A356 and tin bronze CuSn12). The main part of the research is conducted to investigate the application of new materials for the manufacturing of worm gears (alloys ZA12 and A356) and their impact on the power losses and efficiency in comparison to tin bronze CuSn12. The values of efficiency are determined for different operating modes of the gearbox, that is, for different values of input rotational speed and load, where different viscosity lubricants were used. Depending on the operating mode and measured operating temperatures, the calculation of power losses and coefficient of friction for all experimental worm pairs was performed. Based on the obtained results, it can be concluded that the best tribological characteristics has worm pair 42CrMo4/ZA12, which is the recommendation for the use of ZA alloy for manufacturing of worm gears as the alternative for tin bronze and other non-ferrous metals.

Keywords: aluminum; coefficient of friction; efficiency; power losses; worm gearbox lubricant

1 INTRODUCTION

Worm gearboxes belong to a group of hyperboloid gear pairs with non-intersecting shaft axes. Gear pair consists of a worm and worm gearbox. Worm is the most common drive element and is made of higher hardness material (hardened and ground steel for improvement or cementation), while the worm gear is made of softer material, most often of different types of bronze, brass, aluminium alloys, zinc alloys, grey iron, etc. [1-5]. Worm gearboxes are characterised by a line contact of the tooth flanks of meshed gears, which is accompanied by a relatively high sliding friction, which leads to the creation of a large amount of heat and energy losses. The heat generated in this way leads to heating of the gear unit as well as to an increase in the operating temperature of the oil [6-9]. As the temperature increases, the viscosity of the oil decreases, which leads to a decrease in the thickness of the lubricant film between the teeth flanks of meshed gears and greater power losses.

Power losses in gear transmission vary extensively, which depends on many influencing factors, the most important of which are the type of material of meshed gears, geometry of worm pair, gear velocity, type and viscosity of lubricating oil, load, shape of the worm, operating temperature, etc. [10-17]. Based on DIN3996 standard [18], power losses in worm gearbox can be divided into load-dependant losses (mechanical losses) and load-independent losses. The greatest power losses occur due to the fraction in gear tooth mesh and their size can be significantly affected by the selection of appropriate materials and geometric quantities that define worm pairs. The second largest are usually the power losses in bearings, losses due to the contact between the gear and the oil, while the lowest value have the power losses in the shaft seals [19].

In order to reduce power losses and therefore improve the performances of gear transmissions, the selection of the appropriate lubricating oil is of great importance. Accomplished thickness of the oil film has a significant impact on the occurrence of surface fatigue, the durability of contact surfaces, and size of the friction and the efficiency of gear pair.

Mineral-based oils are mainly used for lubrication of gears, while synthetic oils are used in case of high operating temperatures and pressures. Synthetic oils (polyglycol, polyalphaolefin, oil ester, etc.) due to less change in viscosity at higher temperatures and loads allow lower values of the coefficient of friction between the gear teeth compared to mineral oils, which leads to lower power losses [20-25]. Depending on the oil type, the power losses in gear transmissions can be lowered up to 30%. In generally, oils with higher viscosity allow better forming of oil film between the flanks of meshed gears, which is suitable for the creation of hydrodynamic lubrication that implies lower power losses [26, 27].

The research in this paper is based on the investigation of the impact of the lubricant viscosity and types of materials of worm pair on power losses and gear efficiency. The main part of the research is conducted with the aim of the use of new materials for the manufacturing of worm gearboxes (zinc-aluminium alloy ZA12 and aluminium alloy A356) and research of their impact on the efficiency of the gear compared to standard materials.

The use of new materials aims to reduce gear mass, lower values of the friction coefficient in the contact zone of gear teeth, lower scuffing, lower power losses as well as the higher values of the efficiency level.

2 POWER LOSSES AND EFFICIENCY CALCULATION

As is known, the worm gearbox during operation consumes energy to overcome various resistances so that the power at the output shaft P_2 is lower than input power P_1 for the size of the loss P_G, that is:

$$P_2 = P_1 - P_G \tag{1}$$

Basically, total power losses P_G are composed of power losses due to friction in gear mesh and worm gear

P_{GZ}, power losses in bearings P_{GI}, and power losses in shaft seals P_{GD} [28], so that the previous equation can be written as:

$$P_2 = P_1 - \left(P_{GZ} + P_{GL} + P_{GD}\right) \tag{2}$$

Using the mathematical transformation, the Eq. (2) can be written in the following form:

$$P_2 + P_{GZ} = P_1 - \left(P_{GL} + P_{GD}\right) \tag{3}$$

The efficiency of worm pair depends exclusively on the friction power losses in the worm mesh and worm gear P_{GZ} and is determined according to equation:

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

that is, taking into account the Eq. (3) it implies:

$$\eta_z = \frac{P_2}{P_1 - \left(P_{GL} + P_{GD}\right)} = \frac{P_2}{P_1 \cdot \left(1 - \dfrac{P_{GL} + P_{GD}}{P_1}\right)} = \frac{\eta}{1 - \dfrac{P_{GL} + P_{GD}}{P_1}} \tag{5}$$

Eq. (5) represents the dependence of the gear pair efficiency η_z and total efficiency level of gearbox η.

Power losses in bearings P_{GL} are the consequence of friction on the touching surfaces of rolling bodies on the holder and rings. Friction in bearings is composed of rolling friction, sliding friction and lubricant friction. There is very strong and close mutual connection between them, which complicates the detailed analysis of each component [29, 30]. Power losses in bearings depend on the load of the bearings, type and size of bearings, rotational speed and type and viscosity of the lubricant. Power losses due to friction in one bearing P_{GL1} can be determined based on following equation [31]:

$$P_{GL1} = 1{,}05 \cdot 10^{-4} \cdot M \cdot n \tag{6}$$

where:
M - total bearing friction torque, N·mm,
n - shaft rotational speed, min^{-1}.

Total bearing friction torque includes rolling friction torque M_{rr} and sliding friction torque M_{sl}, friction torque of seals M_{seal} and friction torque of drag losses M_{drag}. Equations for their determination are given in the SKF bearing manufacturer's catalogue [31].

On the other hand, power losses in worm shaft seals and worm gear P_{GD} are not load-dependant. The size of these losses is affected by the speed of rotation of the drive and driven shaft, the size of the inner diameter of the seal as well as the pressure exerted between the shaft and the rings [32]. Total power loss in seals is equal to the sum of the losses in all seals of driven shaft. Power losses in one seal P_{GD1} can be determined based on following equation [31]:

$$P_{GD1} = 7{,}9169 \cdot F_{D,\rho} \cdot \left(\frac{d_V}{1000}\right)^2 \cdot \frac{n}{60} \tag{7}$$

where:
$F_{D,\rho}$ - a factor determined by the viscosity of the lubricant,
d_V - shaft diameter in place of the seal, mm,
n - shaft rotational speed, min^{-1}.

The highest losses occur due to the friction in the mesh of worm and worm gear P_{GZ}, which is the consequences of extensive sliding between meshed flank gears. These losses can be determined based on known values of gearbox efficiency η and power losses in bearings P_{GL} and shaft seals P_{GD}, based on the following equation:

$$P_{GZ} = P_1 \cdot (1 - \eta) - \left(P_{GL} + P_{GD}\right) \tag{8}$$

The size of these losses is significantly influenced by the type of worm pair, load, velocity and viscosity of oils that affect the size of the friction in the contact zone of gear teeth. Due to the great number of impact factors that change in the course of time, the defining of friction coefficient value in every point of contact during the meshing of gear teeth flanks is a complex task. The values of the friction coefficient mainly is determined experimentally [33-35]. Friction coefficient of worm pair μ_z can be determined based on known values of efficiency η_z and worm lead angle γ_m according to following equation:

$$\mu_z = \mathrm{tg}\left(\mathrm{arctg}\left(\frac{\mathrm{tg}\gamma_m}{\eta_z}\right) - \gamma_m\right) \tag{9}$$

3 EXPERIMENTAL SETUP

3.1 Test Gearbox and Instrumentation

For the experimental research of efficiency, a single-stage worm gearbox with center distance of 31 mm was used, which was especially designed and manufactured for that purpose. Experimental research are conducted on the device AT200 shown in Fig. 1 with complete installation.

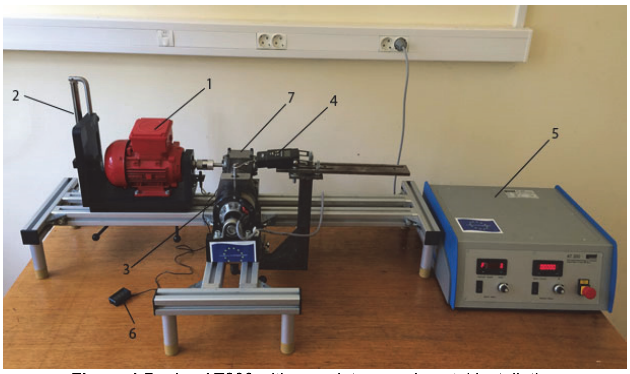

Figure 1 Device AT200 with complete experimental installation

The main parts of the device are electric motor (1) of nominal power 200 W; dynamometer at the input (2), electromagnetic brake (3); dynamometer at the output (4), control unit (5) and thermometer for the measurement of the oil temperature (6). The change of current intensity on control unit un to 0,3 A can be achieved by maximum torque of 10 Nm, which can be taken over by electromagnetic brake.

The connection of worm gearbox (7) with electric motor shafts and electromagnetic brakes is accomplished through claw couplings. Body of the gearbox is a welded construction with openings on the side, which allows easy assembling and disassembling of worm pair, bearings and other elements. The worm is drive element and is placed above worm gear with symmetric position in relation to the bearings. Two annular single-row bearings with radial contact type 6000 RZ are installed on the worm shaft, while the bedding of worm gear shaft was accomplished by two annular single-row bearings type 6001 Rs.

Three worm pairs type ZI with the same geometric characteristic are used for testing, Tab. 1. Worms are made of same material (hardened and ground steel 42CrMo4) and worm gears are made of tin bronze CuSn12, aluminium alloy A356 and zinc-aluminium alloy ZA12, which mechanical and physical properties are shown in the table 2. The main goal of the research is the use of new materials (A356 and ZA12) for construction of worm gears and investigation of their impact on the efficiency and power losses in worm gearbox in comparison to tin bronze CuSn12.

Table 1 Basic geometric characteristic of experimental worm pair

Geometric quantities	Symbol	Unit	Value
Worm gears type	-	-	ZI
Number of gear teeth	z_1/z_2	-	1/18
Module	m	mm	2
Gear ratio	i	-	18
Centre distance	a	mm	31
Lead angle	γ_m	°	4,7636
Pressure angle	α_n	°	20
Profile shift	x_2	-	0,5
Diameter of the basic worm circle	d_1	mm	26
Diameter of the basic worm gear circle	d_2	mm	36

Table 2 Mechanical and physical properties of worm gear materials

Properties	Material		
	CuSn12	ZA12	A356
Densiti / kg/m^3	8880	6030	2670
Tensile strength / N/mm^2	260	300	234
Yield strength 0,2 % / N/mm^2	140	210	165
Brinell Hardness	80	94	70-105
Tensile modulus / N/mm^2	88300	80000	72400
Poissons ratio	0,34	0,31	0,33
Elongation / %	12	1 - 2	3,5
Melting area / °C	762 - 928	377 - 432	557 - 613

3.2 The Lubricant

Lubrication of worm pair is done by dipping. Reduktol super oils produced by the oil refinery Modriča are used as lubricants. These are high-quality mineral oils with a high viscosity index, which are alloyed with EP additives based on phosphorus and sulphur. The viscosity of the oil is defined according to the recommendations of the standard ISO 12925-1, which defines the use of oil for closed gears whose viscosity ranges 150 - 1500 mm^2/s [36]. In this regard, oils of viscosity 220 mm^2/s, 460 mm^2/s and 680 mm^2/s were selected, the basic characteristics of which are shown in Tab. 2.

Table 3 Characteristics of Reduktol super oil

Quantity	Unit	Reduktol super		
		220	460	680
Viscosity at 40 °C	mm^2/s	220	460	680
Viscosity at 100 °C	mm^2/s	18,43	28,91	40,8
Viscosity index	-	91	90,3	97
Flow point	°C	−18	−12	−8
Ignition point	°C	260	260	300
Thickness at 15 °C	kg/m^3	896,1	896,1	905

Viscosity is a very important characteristic of the oil, which changes with temperature and pressure. For the different operating temperature values of the oil, ϑ kinematic viscosity can be determined based of known values of oil viscosity under 40 °C and 100 °C according the following equation [34]:

$$\nu = 10^{C} - 0{,}7 \tag{10}$$

where C is constant determined according to the equation:

$$C = 10^{A \cdot \log(\vartheta + 273) + B} \tag{11}$$

In the previous equation A and B are constants that are calculated as follows:

$$A = -13{,}129 \cdot \log\left(\frac{\log(\nu_{40} + 0{,}7)}{\log(\nu_{100} + 0{,}7)}\right) \tag{12}$$

$$B = \log\left(\log\left(\nu_{40} + 0{,}7\right)\right) - 2{,}496 \cdot A \tag{13}$$

During the operation of the gearbox, the operating temperature changes, which leads to a change in the viscosity of the oil. As the operating temperature increases, the viscosity of the oil decreases thus reducing the thickness of the oil film between the meshed flanks of the gear teeth, which leads to increased friction and greater power losses.

4 EXPERIMENTAL RESULT AND DISCUSSION

Values of total efficiency are determined for different operating conditions of the gearbox, that is, for three different values of the rotational speed (1500 min^{-1}, 2000 min^{-1} and 2500 min^{-1}) and five levels of load, where oils of different viscosity, 220 mm^2/s, 460 mm^2/s and 680 mm^2/s, are used. During the testing, the change in current intensity at control unit in the intervals 0.1 - 0.2 A, leads to the change of break force in electromagnetic brake and change in load, that is, output torque T_2. On that occasion, the values of the output torque were in the interval 1.95 - 5.4 Nm, taking into account all experimental worm pairs. At the same time, the values of the input torque T_1 were measured, as well as the values of oil temperature and ambient temperature. The average values of the total degree of utilization η were determined on the basis of the measured values of input T_1 and output torques T_2 according to the equation:

$$\eta = \frac{P_2}{P_1} = \frac{T_2}{T_1 \cdot i} \quad (14)$$

The measurement results of efficiency for worm pair 42CrMo4/CuSn12 for different operating regimens of gearboxes and oil of different viscosity are presented in Fig. 2, Fig. 3 and Fig. 4.

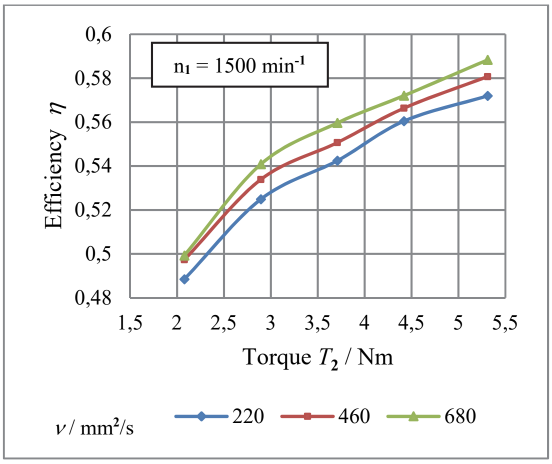

Figure 2 Value of efficiency for input rotational speed 1500 min^{-1} and oils of different viscosity

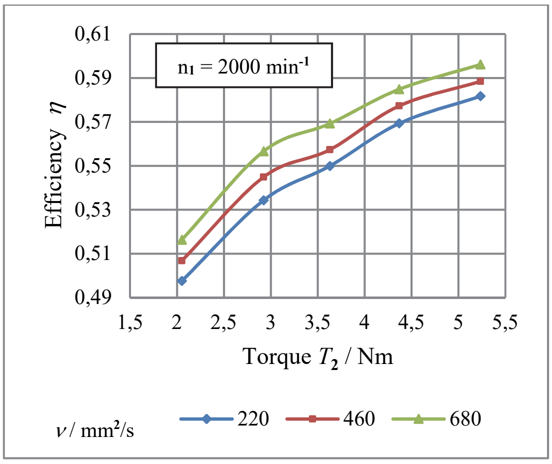

Figure 3 Values of the efficiency for input rotational speed 2000 min^{-1} and oils of different viscosity

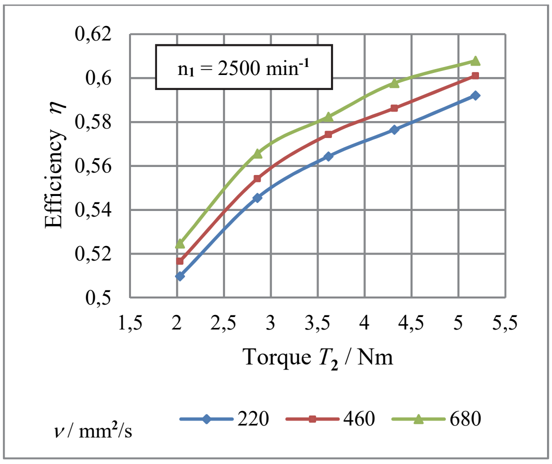

Figure 4 Values of the efficiency for input rotational speed 2500 min^{-1} and oils of different viscosity

The analysis of obtained diagrams points out that the efficiency increases with the increase of load, that is with greater values of output torque T_2. Furthermore, greater values of efficiency are obtained using oils of greater viscosity and at greater rotational speeds (extensive velocities) at all levels of load. Namely, with the increase of the viscosity of oil the value of efficiency also increase by 1 - 2%. On that occasion, taking into account different operating regimes of the gearboxes and oils of different viscosity, the values of efficiency were in the interval η = 048 - 0.61. The maximum value was recorded for oil of viscosity 680 mm^2/s and highest level of load, and minimal for oil of viscosity 220 mm^2/s and lowest level of gearboxes load. Application of oils with higher viscosity deteriorates heat dissipation and increases resistance due to difficult shaking and squeezing of oil from the gap between the teeth, which is especially noticeable at slightly lower load levels. In general, oils with higher viscosity allow the formation of hydrodynamic lubrication, which leads to better formation of the oil film between the meshed flanks of the gear teeth and higher values of the efficiency.

Fig. 5, Fig. 6 and Fig. 7 give a comparative presentation of efficiency of the gearboxes for materials of worm gear CuSn12, A356 and ZA12 for the rotational speed n_1 = 2000 min^{-1} and oils of different viscosity, where the dependence between the degree of utilization and the output torque T_2 is established.

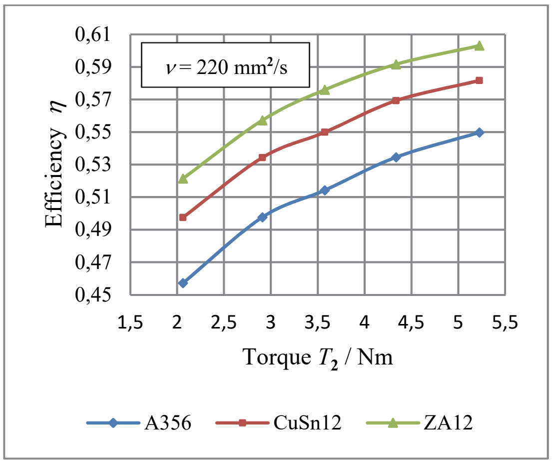

Figure 5 Values of the efficiency of the oil of viscosity 220 mm^2/s, rotational speed 2000 min^{-1} and different materials of worm gears

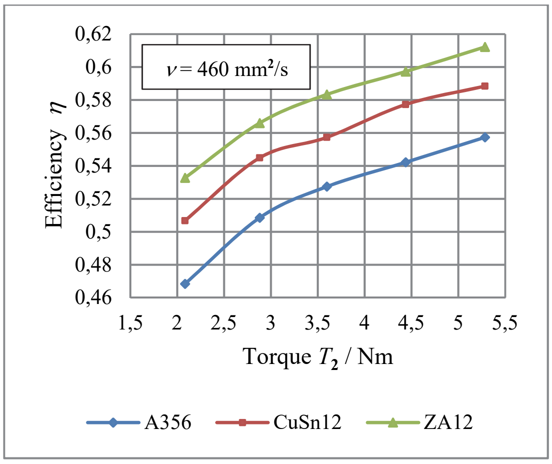

Figure 6 Values of the efficiency of the oil of viscosity 460 mm^2/s rotational speed 2000 min^{-1} and different materials of worm gears

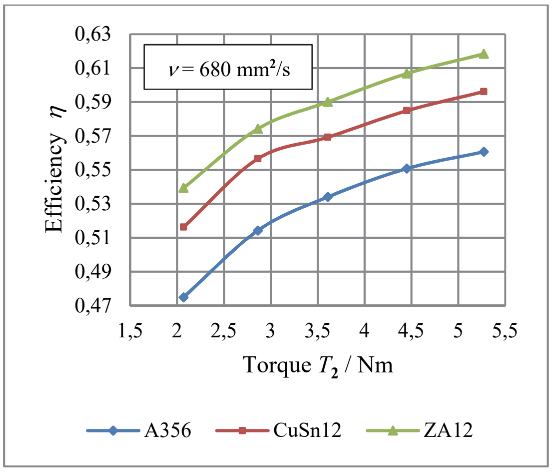

Figure 7 Values of the efficiency of the oil of viscosity 680 mm²/s, rotational speed 2000 min^{-1} and different materials of worm gears

From the previous figures, it can be seen that the values of the efficiency increase with increasing load, taking into account all experimental worm pairs. In addition, higher values of the degree of efficiency are noticeable when lubricating with greater viscosity oil. Comparing the obtained results, it is concluded that the highest values of the efficiency were recorded in the worm pair with a worm gear made of zinc-aluminium alloy (ZA12), which ranged from η = 0.52 - 0.62. In the case of a worm pair with a worm gear made of tin bronze CuSn12, the values of the efficiency were lower in average by 3 - 4% compared to the previous worm pair. The lowest values of the efficiency were recorded in the worm pair with a worm gear made of aluminium alloy A356, which moved in the interval η = 0.45 - 0.56. A slightly larger difference in the values of the efficiency is noticeable at higher values of the output torque T_2.

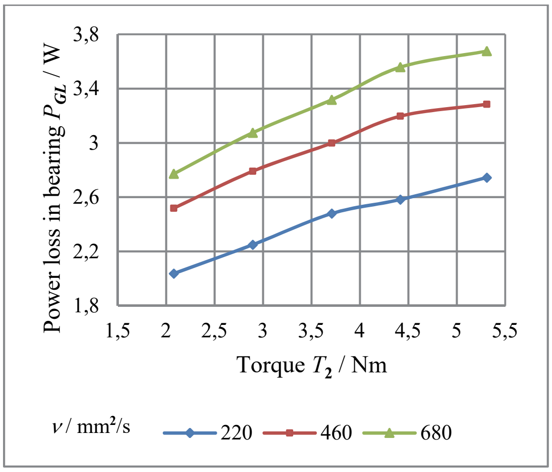

Figure 8 Impact of oil viscosity on power losses in rolling bearings (worm pair 42CrMo4/CuSn12)

In order to determine the power losses in bearings, the values of the radial and axial forces of the bearings were previously determined for different modes of operation of the gearbox that is for different load levels and rotational speeds. Change in operating regime of gearbox implies the change in operating temperature, which ranged in the interval ϑ = 40 - 63 °C taking into account all experimental worm pairs and oils of different viscosity. Kinematic viscosity of oil at operating temperatures is determined based on the expression 10. After that, power losses in bearings and shaft seals are determined based on mathematical calculation of bearing manufacturer SKF [31].

Results of the calculation of the power losses in bearings for input rotational speed n_1 = 1500 min^{-1} and oils of different viscosity depending on the output torque T_2 are shown on Fig. 8.

It can be seen from the diagram that the power losses in the bearings increase with the load increasing and with oil viscosity increasing. Namely, at greater loads, there is an increase in axial and radial forces in the bearings, which leads to greater energy losses. On the other hand, with the increase of the viscosity of the lubricant, the hydrodynamic resistances of the lubricant flow increase, which results in an increase in the power losses in the bearings. The size of these losses depends on the amount of oil, pressure, operating temperature as well as the internal geometry of the bearing.

The power losses in gear mesh is affected by many factors, out of which the type of material of worm pair has very important impact. Values of power losses in gear mesh P_{GZ} are determined according to Eq. (8) based on experimentally measured values of the efficiency of the gearbox η and calculated values of power losses in bearings P_{GL} and shaft seals P_{GD}.

On that occasion, the dependence of power losses and output torque was established, whose average values were in the interval T_2 = 2 - 5.3 Nm, taking into account all experimental worm pairs. The results of the calculation of power losses in the gear mesh for the input rotational speed of 2000 min^{-1} and oils of different viscosities are shown in Fig. 9, Fig. 10 and Fig. 11.

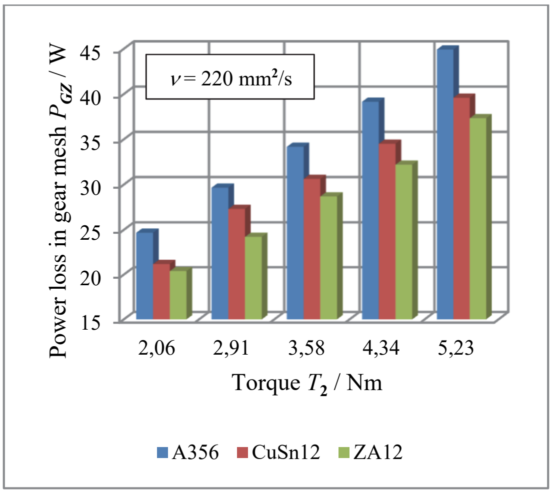

Figure 9 The impact of material on power losses in gear mesh for rotational speed 2000 min^{-1} and oil viscosity 220 mm²/s

Based on obtained diagrams, it can be clearly observed that power losses in gear mesh increase as the load increases. Namely, at greater loads due to the increase of operating temperature the lubricant viscosity decrease, whereby the thickness of oil film decreases between meshed flanks of gear teeth leading to the increase of power losses. On the other hand, these losses are reduced when lubricating with oils of greater viscosity. Such a trend of changing power losses was observed in all experimental worm pairs. In this case, also, the worm pair with a worm gear made of ZA12 alloy has the best characteristics in terms of the lowest power losses, while the greatest power losses occur in the worm pair with a worm gear made of A356 alloy.

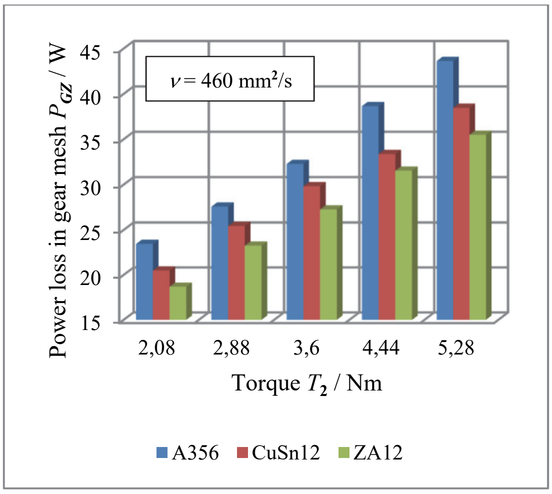

Figure 10 The impact of material on power losses in gear mesh for rotational speed 2000 min^{-1}and oil viscosity 460 mm^2/s

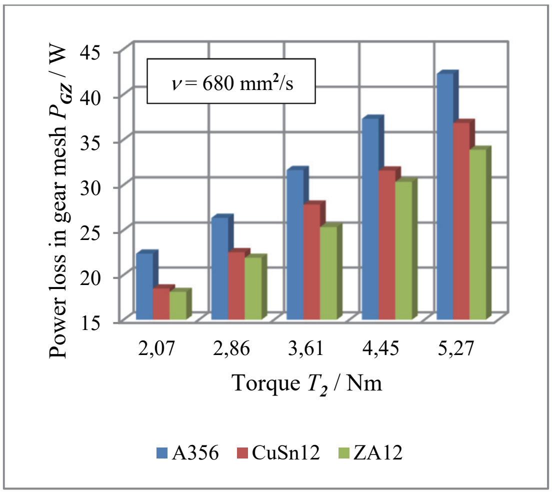

Figure 11 The impact of material on power losses in gear mesh for rotational speed 2000 min^{-1}and oil viscosity 680 mm^2/s

At lower load levels, the difference in power losses is smaller, taking into account all materials, while with a further increase in load, this difference becomes somewhat larger. It is evident that the worm pair with the worm gear made of ZA12 alloy has a better ability to maintain the oil film between the meshed gear teeth, especially at greater loads (output torques), compared to the other two alloys that resulted in lower power losses

The change in power losses is closely related to the change in the coefficient of friction in the contact zone of the flanks of the meshed gears. In this regard, for experimental worm pairs, the values of the coefficient of friction according to Eq. (9) were calculated based on the results obtained by experimental research.

The results of the calculation of the friction coefficient for the input rotational speed of 2000 min^{-1} and oils of different viscosities are shown in Fig. 12, Fig. 13 and Fig. 14.

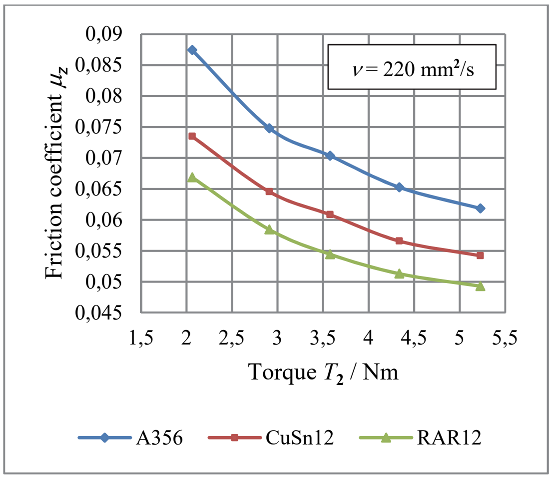

Figure 12 Impact of material on friction coefficient of worm pair for rotational speed 2000 min^{-1} and oil viscosity 220 mm^2/s

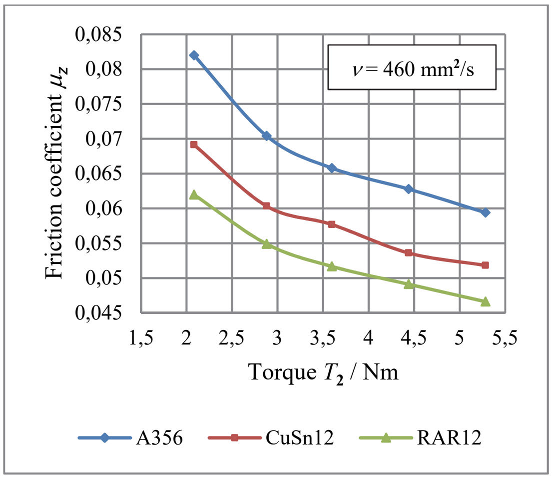

Figure 13 Impact of material on friction coefficient of worm pair for rotational speed 2000 min^{-1} and oil viscosity 460 mm^2/s

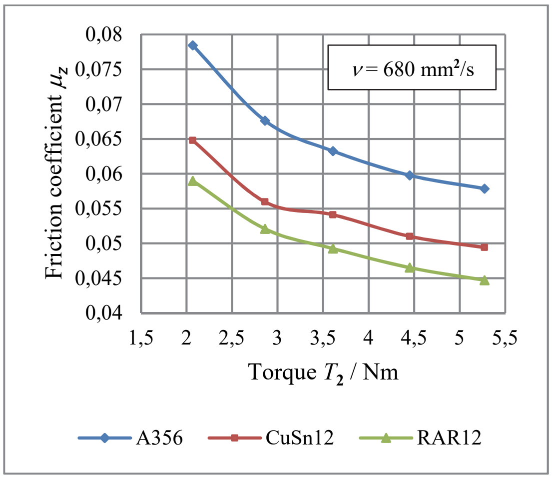

Figure 14 Impact of material on friction coefficient of worm pair for rotational speed 2000 min^{-1} and oil viscosity 680 mm^2/s

Based on the obtained diagrams it can be concluded that the values of friction coefficient decrease with the increase of oil viscosity and increase of load. Such a course of change is significantly influenced, along with working conditions, by the operating temperature of the oil, power losses in the gear mesh and the type of material of worm pair. The greatest values of friction coefficient occurs in the worm pair mesh with worm gear made of alloy A356, which ranged in interval μ_z = 0.088 - 0.057, taking into account all three lubricants. Lower values of friction coefficient are noticeable at worm pair with worm maid of tin bronze CuSn12, which ranged in interval μ_z = 0.074 - 0.049, while the best lubricating conditions were achieved at worm gear made of alloy ZA12 where lowest friction occurred (μ_z = 0.067 - 0.044). Lower values of friction coefficient in the contact zone of gear teeth imply lower power losses as well as the greater level of worm pair efficiency and thus in the gearbox as a whole.

5 CONCLUSION

Compared to other types of teeth gearboxes, worm gearboxes have lower values of the efficiency due to the extremely high sliding friction between meshed flanks of gear teeth, which causes significant energy losses. These losses depend on many impact factors. Lubricant viscosities input rotational speed, load (output torque) and type of worm gear material were considered as influential factors in this paper.

Based on the obtained results, it can be concluded that the values of the efficiency increase from 1 - 2% with the increase of lubricant viscosity. The greatest values of the efficiency are accomplished at highest rotational speed (extensive gear speed) and load by utilizing oils with greater viscosity (680 mm^2/s). On the other hand, oils with greater viscosity, due to higher hydraulic resistances of the lubricant flow, lead to increased power losses in the bearings. However, these resistances decrease with the increase of the operating temperature, which leads to decrease in oil viscosity. In general, oils with greater viscosity lead to lower power losses in the gear mesh and lower values of the friction coefficient due to better formation of the oil film, which resulted in greater values of the efficiency.

The type of material of meshed gears has a great influence on power losses and the efficiency. The highest values of the efficiency were achieved in worm pair with worm gear made of alloy ZA12, which were up to 4% higher than worm pair made of tin bronze CuSn12 and up to 10% compared to alloy A356. It is evident that the worm pair 42CrMo4/ZA12 has better tribological characteristics and better ability to maintain the oil film between the meshed gears compared to the other two worm pairs, which resulted in lower power losses and friction coefficient values.

Among other things, ZA alloys are much lighter than tin bronze, have a lower casting temperature and have greater hardness. An important aspect that makes these alloys attractive is the cost reduction of 25 - 50% compared to tin bronze and from 40 - 75% compared to aluminium. Therefore, ZA alloys can be very successfully used as an alternative to tin bronze and other non-ferrous metals out of which the worm gears are made.

Acknowledgements

This work has been performed as a part of activities within the project TR 35021, supported by the Republic of Serbia, Ministry of Education, Science and Technological Development, whose financial help is gratefully acknowledged.

6 REFERENCES

[1] Jelaska, D. T. (2012). *Gears and gear drives*. John Wiley & Sons. https://doi.org/10.1002/9781118392393

[2] Radzevich, S. P. (2016). *Dudley's handbook of practical gear design and manufacture*. CRC press. https://doi.org/10.1201/9781315368122

[3] Fontanari, V., Benedetti, M., Straffelini, G., Girardi, C., & Giordanino, L. (2013). Tribological behavior of the bronze-steel pair for worm gearing. *Wear*, *302*(1), 1520-1527. https://doi.org/10.1016/j.wear.2013.01.058

[4] Fontanari, V., Benedetti, M., Girardi, C., & Giordanino, L. (2016). Investigation of the lubricated wear behavior of ductile cast iron and quenched and tempered alloy steel for possible use in worm gearing. *Wear*, *350*, 68-73. https://doi.org/10.1016/j.wear.2016.01.006

[5] Benedetti M., Fontanari V., Torresani E., Girardi C., & Giordanino L. (2017). Investigation of lubricated rolling sliding behaviour of WC/C, WC/C-CrN, DLC based coatings and plasma nitriding of steel for possible use in worm gearing. *Wear*, *378-379*, 106-113. https://doi.org/10.1016/j.wear.2017.02.029

[6] Paschold, C., Sedlmair, M., Lohner, T., & Stahl, K. (2020). Efficiency and heat balance calculation of worm gears. *Forsch Ingenieurwes*, *84*, 115-125. https://doi.org/10.1007/s10010-019-00390-1

[7] Bhat, J. S. & Sonawane, B. U. (2019). Analysis of wear particles in lubricating oil of worm gearbox using ferrography. *Manufacturing Technol Today*, *18*(5), 34-40.

[8] Bhat, J. S. & Sonawane, B. U. (2018). Evaluation of performance deterioration of the used equipment oil in sliding pair. *International Journal of Research in Advent Technology*, *6*(8), 2061-2066.

[9] Chougale, A. B., Bhat, J. S., Bhapkar, U. S., & Sonawane, B. U. (2021). Optimization of process parameters by Hybrid Taguchi-Grey Relational Analysis for thermal behaviours of lubricant oil of worm gearbox. *Materials Today: Proceedings*. https://doi.org/10.1016/j.matpr.2021.03.158

[10] Michaelis, K., H¨ohn, B. R., & Hinterstoißer, M. (2011). Influence factors on gearbox power loss. *Industrial Lubrication and Tribology*, *63*(1), 46-55. https://doi.org/10.1108/00368791111101830

[11] Sabiniak, H. G. (2017). Testing worm gears with cooperating elements made of different materials. *Proceedings of the Institution of Mechanical Engineers*, *Part J*: *Journal of Engineering Tribology*, *231*(3), 341-346. https://doi.org/10.1177/1350650116656982

[12] Riečičiarová, E. & Oravcová, J. (2018). The Effect of Materials on the Efficiency of the Spiroid Gearing. *In Materials Science Forum. Trans Tech Publications Ltd*, *919*, 362-369. https://doi.org/10.4028/www.scientific.net/MSF.919.362

[13] Deng, X., Wang, S., Hammi, Y., Qian, L., & Liu, Y. (2020). A combined experimental and computational study of lubrication mechanism of high precision reducer adopting a worm gear drive with complicated space surface contact. *Tribology International*, *146*, 106261. https://doi.org/10.1016/j.triboint.2020.106261

[14] Chothani, H. G. & Maniya, K. D. (2020). Experimental investigation of churning power loss of single start worm gear drive through optimization technique. *Materials Today: Proceedings*, *28*, 2031-2038. https://doi.org/10.1016/j.matpr.2019.12.365

[15] Skoczylas, L. & Wydrzyński, D. (2017). Operational tests of worm gearbox with ZK2 concave profile. *Eksploatacja i Niezawodnosc-Maintenance and Reliability*, *19*(4), 565-570. https://doi.org/10.17531/ein.2017.4.10

[16] Heilemann, J. (2005). Tragfähigkeit und Wirkungsgrad bei unterschiedlichen Schnecken-Zahnflankenformen unter

Berücksichtigung der Oberflächenhärte und Härtetiefe. *Dissertation*, Technische Universität München.
[17]Vasić, M. P., Stojanović, B., & Blagojević, M. (2020). Fault analysis of gearboxes in open pit mine. *Applied Engineering Letters*, 5(2), 50-61. https://doi.org/10.18485/aeletters.2020.5.2.3
[18]DIN 3996. (2012). Tragfähigkeitsberechnung von ZylinderSchneckengetrieben mit sich rechtwinklig kreuzenden Achsen.
[19]Magyar, B. & Sauer, B. (2015). Calculation of the Efficiency of Worm Gear Drives. *Power Transmission Engineering*, *9*(4), 52-56.
[20]Mautner, E. M., Sigmund, W., Stemplinger, J. P., & Stahl, K. (2016). Efficiency of worm gearboxes. *Proceedings of the Institution of Mechanical Engineers, Part C: Journal of Mechanical Engineering Science*, *230*(16), 2952-2956. https://doi.org/10.1177/0954406215602286
[21]Muminović, A., Repčić, N., & Žeželj, D. (2014). The efficiency of worm gears lubricated with oils of mineral and synthetic bases. *Transactions of FAMENA*, *37*(4), 65-72.
[22]Martins, R., Cardoso, N., & Seabra, J. (2008). Gear power loss performance of biodegradable low-toxicity ester-based oils. *Proceedings of the Institution of Mechanical Engineers, Part J: Journal of Engineering Tribology*, *222*(3), 431-440. https://doi.org/10.1243/13506501JET345
[23]Höhn, B. R., Michaelis, K., & Hinterstoißer, M. (2009). Optimization of gearbox efficiency. *Goriva i maziva*, *48*(4), 462-480.
[24]Ziegltrum, A., Lohner, T., & Stahl, K. (2017). TEHL simulation on the influence of lubricants on load-dependent gear losses, *Tribology International*, *113*, 252-261. https://doi.org/10.1016/j.triboint.2016.12.018
[25]Marques, P., Fernandes, C., Martins, R., & Seabra, J. (2014). Efficiency of a gearbox lubricated with wind turbine gear oils. *Tribology International*, *71*, 7-16. https://doi.org/10.1016/j.triboint.2013.10.017
[26]Skulić, A., Stojanović, B., Radosavljević, S., & Velickovic, S. (2019). Experimental determination of worm gearing efficiency. *Applied engineering letters*, *4*(4), 115-119. https://doi.org/10.18485/aeletters.2019.4.4.2
[27]Radosavljević, S. Z., Stojanović, B. Ž., & Skulić, A. D. (2018). Determination of power losses in worm gear reducer. *In IOPC onference Series: Materials Science and Engineering*, *393*(1), 012050.
[28]Miltenovic, V. (2009). *Mechanical elements-form, calculation, implementation*. Faculty of Mechanical Engineering, Niš.
[29]Krsmanović, V. & Mitrović, R. (2015). *Klizni i kotrljajni ležaji*. Univerzitet u Beogradu, Mašinski fakultet.
[30]Fernandes, C. M., Marques, P. M., Martins, R. C., & Seabra, J. H. (2015). Gearbox power loss. Part I: Losses in rolling bearings. *Tribology International*, *88*, 298-308. https://doi.org/10.1016/j.triboint.2014.11.017
[31] SKF: Hauptkatalog, Das Wälzlager-Handbuch für Studenten Neuwertig, Januar, (2014).
[32]Chernets, M., Shil'ko, S., & Kornienko, A. (2021). Calculated assessment of contact strength, wear and resource of metal-polymer gears made of dispersion-reinforced composites. *Applied Engineering Letters*, *6*(2), 54-61. https://doi.org/10.18485/aeletters.2021.6.2.2
[33]Magyar, B., Sauer, B., & Horák, P. (2012). Tribological Investigation of K Type Worm Gear Drives. *Acta Polytechnica Hungarica*, *9*(6), 233-252.
[34]Miltenović, Đ. (2017). *Istraživanje termičke stabilnosti i habanja pužnih prenosnika*. Doktorska disertacija. Univerzitet u Banjoj Luci, Mašinski fakultet.
[35]Monz, A. (2012). *Tragfähigkeit und Wirkungsgrad von Schneckengetrieben bei Schmierung mit konsistenten Getriebefetten*. Technische Universität München.
[36]ISO 12925-1. (2018). Lubricants, industrial oils and related products (class L)-Family C (gears), Specifications for lubricants for enclosed gear systems, Part 1.

Contact information:

Aleksandar SKULIĆ, PhD student
(Corresponding author)
University of Kragujevac,
Faculty of Engineering,
Department for Mechanical Constructions and Mechanization,
Kragujevac 34000, Sestre Janjić 6, Serbia
E-mail: aleksandarskulic@gmail.com

Saša MILOJEVIĆ, expert advisor
University of Kragujevac,
Faculty of Engineering,
Department for Motor Vehicles and Engines,
Kragujevac 34000, Sestre Janjić 6, Serbia
E-mail: sasa.milojevic@kg.ac.rs

Dejan MARIĆ, PhD
(Corresponding author)
University of Slavonski Brod,
Mechanical Engineering Faculty in Slavonski Brod,
Trg Ivane Brlić Mažuranić 2, HR-35000 Slavonski Brod, Croatia
E-mail: dmaric@unisb.hr

Lozica IVANOVIĆ, prof.
University of Kragujevac,
Faculty of Engineering,
Department for Mechanical Constructions and Mechanization,
Kragujevac 34000, Sestre Janjić 6, Serbia
E-mail: lozica@kg.ac.rs

Božidar KRSTIĆ, prof.
University of Kragujevac,
Faculty of Engineering,
Department for Motor Vehicles and Engines,
Kragujevac 34000, Sestre Janjić 6, Serbia
E-mail: bkrstic@kg.ac.rs

Mladen RADOJKOVIĆ, PhD, Assistant professor
University of Priština,
Faculty of Technical Science,
Department for Mechanical Constructions and Mechanization,
Kosovska Mitrovica 38220, Kneza Miloša 7, Serbia
E-mail: mladen.radojkovic@pr.ac.rs

Blaža STOJANOVIĆ, prof.
University of Kragujevac,
Faculty of Engineering,
Department for Mechanical Constructions and Mechanization,
Kragujevac 34000, Sestre Janjić 6, Serbia
E-mail: blaza@kg.ac.rs