ANNALS of Faculty Engineering Hunedoara – International Journal of Engineering
Tome XIV [2016] – Fascicule 1 [February]
ISSN: 1584-2665 [print; online]
ISSN: 1584-2673 [CD-Rom; online]
a free-access multidisciplinary publication of the Faculty of Engineering Hunedoara

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# LOAD CAPACITY OF WORM GEARS WITH COMPACT DESIGN

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**ABSTACT**: Worm gearing has very wide use in power transmission in movement and the limitation of the complete machine system depends on it. Load capacity by worm gear transmitters is determinate with their geometry parameters and lubrication. With suited selection of parameters it is possible to have influence on load capacity of worm gear pair and of whole transmitter. This is very important in the transmitters with compact construction when working conditions have limitations working space. In this paper in present one optimization method of geometry parameters with load-carrying capacity and increase of limitations. Here is presented variation of geometry parameters with no change of center distance and materials.
**Keywords**: worm gear transmitter, compact construction, geometry

## 1. INTRODUCTION

Worm gears have very wide use in transmission gear and movement because they have several advantages in comparison to other gear types. Worm gear pair is hyperboloid gear pair which axes are crossing mostly with 90° angle. Small gear is worm and he has shape of spindle. Great gear is worm gear and he has shape that is suitable to the worm.

The basic characteristics of worm gears are:

- Possibility of achieving huge gear ratio of one worm gear pair. If revolution is reduced, gear ratio is in $5 \leq u \leq 70$ (for small power it is possible to achieve gear ratio of up to $u \leq 1000$).
- Considerably smaller internal dynamic forces and sound, by absorbing vibrations.
- Efficiency of worm gearing is relatively high because of huge sliding between sides in contact of gear and worm gear.
- They can be produced as self-stopping transmission gearing, which enables them to have a wide use.

These properties are giving possibility use of worm gear transmitters behind all by transport devices, tool machines, in vehicles for strength transit, and by fine adjust and precise devices for moving transit.

## 2. LOAD CAPACITY OF WORM GEARS

During contact between tooth flanks of worm and worm gear is transmitting normal force $F_N$, which brings to large surface pressures. These pressures during work can bring to pitting devastation of tooth flanks. Beside that between contact flanks there is great rolling which has for result wear and great energy designation. Energy is behind that transforming to heat, which brings to heating of transmitters, failure of their normal work and in critical condition to the jamming.

Root of the worm gear tooth is subject of stress where is dominant deflection and shears strength. DIN 3996 [10] commend verifying of limitation for tooth breakage and for shear. With regard to distance between worm braces can be great, especially by bigger gear ratio, there is dangerous that too big deflection of worm shaft can bring to smaller contact pattern and that can come to contact failure. This can be specially pronounced in period of insufficient flank lubrication, when friction force can expand. In modern constructive solution worm gear is made of bronze or nodular casting and by worm from tin or ground steel.

Table 1 summarizes the potential damage and limitations, which can occur in worm gear pairs. They also give expressions to determine the safety.

**Table 1.** Safety of worm gear pair for different limitations

| Limitations | Safety |
|---|---|
| Pitting is actually the destruction of tooth surface as an effect of huge surface pressures and dynamic stress wear. We can see the difference between the initial and advance stage of pitting. The initial stage of pitting is the effect of the first phase of working of transmission gears. Medium and highly loaded worm gear pairs can be attacked by advance pitting. This pitting has progressive character, so that destroyed surfaces. | $S_H = \frac{\sigma_{Hkr}}{\sigma_{Hm}} \geq S_{H\min} = 1$<br>$\sigma_{Hkr}$ - limiting value of contact stress;<br>$\sigma_{Hm}$ - mean contact stress. |
| Wear: such damage may appear on the tooth flanks of bronze worm wheels. During the period of wearing mining continually wearing material tooth width is smaller. In the first phase, wearing has a positive effect because it leads to rubbing of material and accommodation between shapes of tooth surfaces and later stopping any further wear. However, in the cases of huge intensity rubbing, wearing resistance can be a criterion of working period. Wearing basically depends on working criteria. | $S_W = \frac{\delta_{W\lim n}}{\delta_{Wn}} \geq S_{W\lim} = 1{,}1$<br>$\delta_{Wlimn}$ - permissible wear;<br>$\delta_{Wn}$ - abrasive wear in the normal section. |
| Tooth breakage: Worm gear pairs have dangerous working loads only in tooth root of worm gear. In the root tooth of a worm gear, there is a complicated load, and the dominant stresses are shear and bending. Worm gear tooth breakage is very rare. Most often causes are striking overloads when loads appear that are larger than the static strength of material. Wear is also a considerable effect that causes tooth breakage, and wearing diminishes its cross-section. | $S_F = \frac{\tau_{Fkr}}{\tau_F} \geq S_{F\lim} = 1{,}1$<br>$\tau_{Fkr}$ - permissible shear stress;<br>$\tau_F$ - shear stress. |
| Working temperature: When designing a gearbox, one should also consider the heat generated inside the gearbox. Thermal safety has great importance for a correct design to ensure gearbox function within the permitted temperature range of oil. Thermal design/safety tends to be one of the limiting factors when designing transmissions. | $S_T = \frac{\vartheta_{S\lim}}{\vartheta_S} \geq S_{S\lim} = 1{,}1$<br>$\vartheta_{Slim}$ - boundary value of oil temperature;<br>$\vartheta_S$ - oil temperature. |
| Worm shaft deflection: Worm shaft is loaded by radial, axial and tangential force, which leads to considerable bending of shaft. Considering relatively large distance between bearings. Excessive deformation under load modifying contact pattern between worm and worm wheel. | $S_\delta = \frac{\delta_{\lim}}{\delta_m} \geq S_{\delta\min} = 1$<br>$\delta_m$ - worm shaft deflection;<br>$\delta_{lim}$ - permissible worm shaft deflection. |

There is mutually depends between some damages by worm gear pair. Pitting development can be stopped with expand wear of flanks. If there is wear critical, than pitting has secondary importance. If pitting is critical, than wear is not primary for calculation.

There is and mutually depends between wear and tooth breakage. Wear is making worm gear tooth thinner, which should be used in calculation of limitations tooth breakage.

Analysis of worm gear load capacity is made for gear drives with characteristics given in Table 2. Calculation of load capacity for worm gear according to [10] is done through 5 criteria (table 1).

The calculation was performed for various numbers of input speed and gear ratio. For each calculation the maximum output torque $T_{2max}$ was obtained for the required safety. Calculation results of load capacity are shown in Figures 1 and 2.

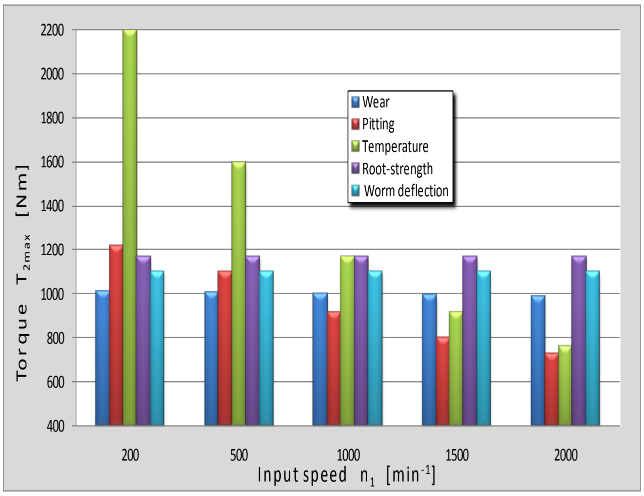


**Figure 1**. Worm gear load capacity for difference values of input speed $n_1$

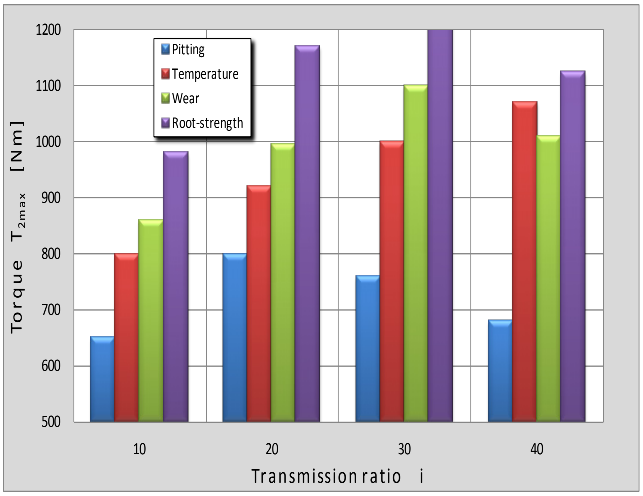


**Figure 2**. Worm gear load capacity for difference values of transmission ratio i

Balanced load capacity is calculated for the selected gear drive (a = 100 mm, i = 20.5) with different input speed $n_1$ = 200 ... 2000 min$^{-1}$ (Figure 1). From figure can be observed that with increasing input speed slightly declining wear load capacity. When increasing the input speed of 10x, pitting load capacity is reduced by 60%. The biggest change is at load capacity with respect to thermal stability, as expected, because at higher input speeds the growth of energy losses.

Load capacity in this case is limited by pitting safety for $n_1$ > 750 min$^{-1}$. For $n_1$ < 750 min$^{-1}$ load capacity case is limited by wear safety.

Load capacity for variation of gear ratio i for constant values of center distance and input speed (a = 100 mm; $n_1$ = 1500 min$^{-1}$) is shown in Figure 2. Load capacity in this case is limited by pitting safety. Other criteria have balanced values. The maximum load capacity by all criteria obtained for the area of transmission ratio i = 20-30.

**Table 2**. Transmission worm gear data

| Geometrical size | Values | Working size | Values |
|---|---|---|---|
| Central distance a [mm] | 100 | Application factor $K_A$ | 1 |
| Transmission ratio i | 20.5 | Output torque $T_2$ [Nm] | 700 (1050) |
| Module m [mm] | 4 | Working life L [h] | 10000 |
| Diameter quotient q | 9 | Wheel material | CuSn12-C-GZ |
| Number of teeth $z_1$/ $z_2$ | 2/41 | Worm material | 16MnCr5 |
| Pitch circle $d_{m1}$/ $d_{m2}$ mm] | 36/164 | Input speed $n_1$ [min$^{-1}$] | 1500 (200 – 2000) |
| Distance worm shaft bearing $l_{11}/l_{12}$ [mm] | 100/100 (85/85) | Synthetic oil (Polyglykol) | $\nu_{40}$ = 220 mm²/s; $\nu_{100}$ = 41 mm²/s |

## 3. COMPACT DESIGN OF WORM GEAR TRANSMITTERS

Geometry of worm gear pairs has great influence on their load-carrying capacity. There can be changed some geometry values and basic parameters of transmitters are connected for dimensions (center distance, housing dimensions), gear ratio, number of rotations, used materials and lubrications stays unchanged. This is very important in the transmitters with compact construction when working conditions have limitations working space and working and manufacturing conditions does not allow change of stated parameters.

For analyze is taken standard transmitter according to table 2. In table 3 are given calculated limitations for output torque $T_{2max}$ = 700 Nm. If the output torque expand for 50% on $T_{2max}$ = 1050 Nm, than some limitations does not satisfied. There is question, is it possible with geometry changes to expand load-carrying capacity of transmitter.

For values of geometry parameters from table 2 according to DIN 3996 is calculated wear limitation (for criteria bounding value of backlash and bounding of addendum width of worm gear), pitting limitation, tooth breakage (for new one and after 10000 hours wear tooth), worm shaft deflection and working temperature. Limitations values are given in table 3 are nominal values. It can be seen that values are unsatisfied for wear criteria bounding value of backlash (0.681), pitting limitation (0.887), worm shaft deflection (0.720), and for working temperature (0.910). Only limitation for tooth breakage is bigger than limitation bounds.

Optimization parameters are tooth width, distance of worm shaft bearings, lead angle and lubrication way.

### 3.1. Variation of tooth width

Tooth width $s_{mx}$ in axial pitch circle of worm (Figure 2) is:

$$s_{mx} = p_x \cdot s_{mx}^* = \pi \cdot m_x \cdot s_{mx}^*$$

Normal space width $e_{mx}$ on the reference cylinder

$$e_{mx} = m_x \cdot \pi \left(1 - s_{mx}^*\right)$$

Tooth width factor $s_{mx}^*$ define mass of worm tooth width $s_{m1}$, from which can be found worm gear tooth width $s_{m2}$. This factor can be taken individually but as referent values is taken $s_{mx}^* = 0,5$ [9].

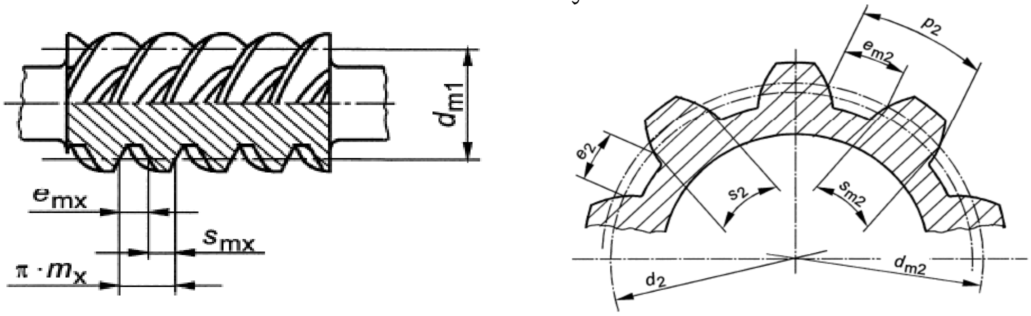


**Figure 3**. Worm and worm gear tooth width

Worm tooth width $s_{m1}$ and worm gear tooth width $s_{m2}$ can be determent as:

$$s_{m1} = s_{mx} = s_{mx}^* \cdot \pi \cdot m_x$$

$$s_{m2} = \pi \cdot m_x \cdot \left(1 - s_{mx}^*\right)$$

In that respect that for worm thread brakeage does not exist calculation, determination of worm thread width limit can be only with probe. In certain cases, when gear breakage limit unsatisfying tooth width factor $s_{mx}^*$ can be smaller.

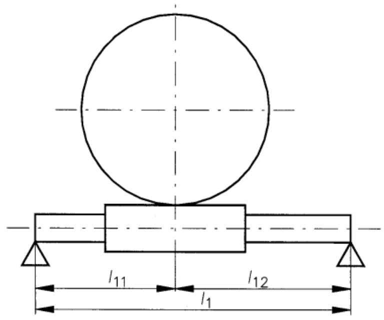


**Figure 4**. Worm shaft deflection

### 3.2. Variation of worm shaft bearing distance

Worm shaft is loaded with radial force $F_{rm}$ and transversal force $F_{tm1}$, which come to greatly worm shaft deflection. Because of worm shaft deflection it comes to shaft bending which can make problems in contact.

The greatest influence on shaft defection has distance between bearings of worm shaft. This value has greatest importance for worm shaft deflection limitation.

**Table 3**. Worm gear limitations according to [10] with variation of the parameters

| Limitations | sign | nom. val. $T_2$ | | Bounding | $l_{11}/l_{12}$ | $s_{mx}^*$ | $\alpha_0$ | | | | Injection lubrication |
|---|---|---|---|---|---|---|---|---|---|---|---|
| | | 700 | 1050 | | 85/85 | 0,4 | 15° | 20° | 25° | 30° | |
| Wear (addendum width) | $S_W$ | 5.649 | 1.914 | 1.1 | 2.270 | 2.270 | 1.660 | 2.270 | 2.740 | 2.579 | 3.838 |
| Wear (backlash) | $S_W$ | 2.011 | 0.681 | 1.1 | 0.808 | 0.808 | 0.481 | 0.808 | 1.288 | 1.859 | 2.078 |
| Pitting | $S_H$ | 1.086 | 0.887 | 1.0 | 0.887 | 0.887 | 0.862 | 0.887 | 0.915 | 0.945 | 0.887 |
| Worm deflection | $S_\delta$ | 1.080 | 0.720 | 1.0 | 1.005 | 1.005 | 1.211 | 1.005 | 0.839 | 0.705 | 1.002 |
| Root of tooth (new) | $S_F$ | 2.096 | 1.397 | 1.1 | 1.397 | 1.575 | 1.441 | 1.575 | 1.718 | 1.873 | 1.575 |
| Root of tooth (with wear) | $S_F$ | 1.969 | 1.148 | 1.1 | 1.187 | 1.365 | 1.088 | 1.365 | 1.586 | 1.781 | 1.493 |
| Thermical stability | $S_T$ | 1.099 | 0.910 | 1.1 | 1.030 | 1.030 | 1.030 | 1.030 | 1.030 | 1.030 | 2.132 |

### 3.3. Variation of pressure angle $\alpha_0$

Variation of pressure angle is changed and geometry of worm gear tooth. The greatest influence has change of crossing section of worm gear tooth. With bigger of pressure angle greatest is width of root and with of addendum is smaller. That practically means that with bigger pressure angle are bigger tooth breakage and wear limitation.

### 3.4. Variation of lubrication way

Change the lubrication ways has a significant impact on the performance of the gear. By changing splash lubrication with pressure lubrication leads to an increase in load capacity at all criteria.

For more precise see out on load-carrying capacity, it is calculated all important limitations for variation of these values (tooth width, distance of worm shaft bearing, pressure angle and lubrication way). Calculated limitations are given in table 3. On figure 5 graphically are shown variation of limitations for changes of worm shaft bearing, tooth width and lubrication way.

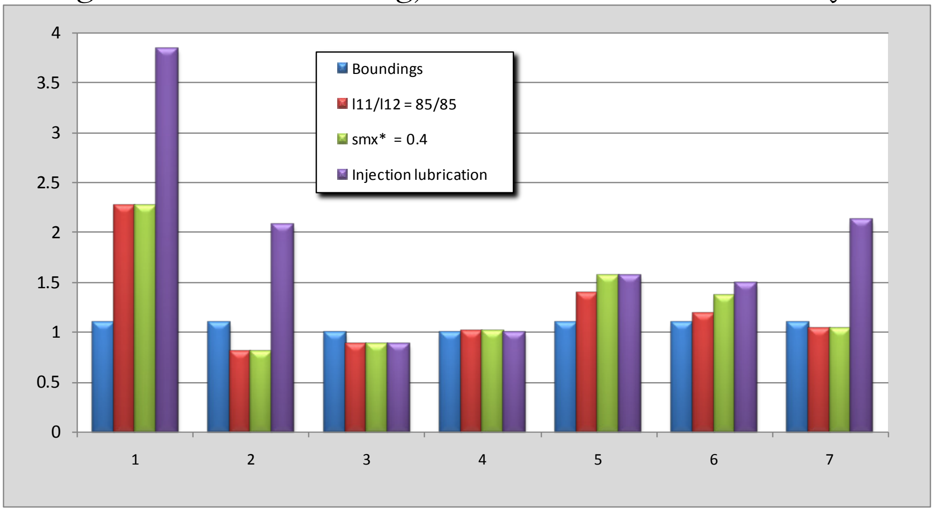


**Figure 5**. Limitations for different values of parameters

## 4. CONCLUSION

Based on the analysis above we can conclude the following:

≡ By reducing the distance of worm shaft bearing, reduces the risk of bending the worm shaft, and comes to increase of wear safety.

≡ Decrease of worm tooth width $s^*_{mx}$ comes to significant increase of tooth breakage limitation of worm gear. With it, no change in the other limitations.

≡ Change of pressure angle make wear limitation and tooth breakage limitation. To a lesser degree increases the pitting limitation. But at the same time increases the load of worm shaft

≡ By changing the way of lubrication (instead of splash lubrication apply pressure lubrication) with the other aforementioned changes load capacity satisfies on all the criteria. The exception is the pitting limitation, where the pitting safety slightly deviates from the limit values. On this way it is possible to increase load capacity up to 50%.

**Note**

This paper is based on the paper presented at The 12th International Conference on Accomplishments in Electrical and Mechanical Engineering and Information Technology – DEMI 2015, organized by the University of Banja Luka, Faculty of Mechanical Engineering and Faculty of Electrical Engineering, in Banja Luka, BOSNIA & HERZEGOVINA (29th – 30th of May, 2015), referred here as[11].

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