Investigations on the Efficiency of Worm Gear Drives

By Eva-Maria Mautner, Werner Sigmund, Johann-Paul Stemplinger, and Karsten Stahl

Experimental investigations on different worm gears were conducted on several test rigs, taking into consideration the influence of different gear ratios, worm wheel materials, lubricants, and contact pattern on efficiency and load-carrying capacity. Recommendations for an increase in overall worm gearbox efficiencies are presented.

DUE TO A WIDE RANGE OF PROPERTIES, WORM gears are an indispensable element on the current transmission market. Next to a huge gear ratio field in one gear stage of \( i = 5 \) to \( i = 80 \), operation with low noise and vibration is realizable. Furthermore, worm gears provide the opportunity of self-locking, respectively self-braking. Despite these benefits, as a result of greater energy awareness, the efficiency of worm gears is in focus. Because of high sliding velocities, especially at high gear ratios, gearing losses are a main topic of interest. Other gearbox concepts with combined spur and bevel gear sets show smaller gear ratio fields, and therefore the realization of high gear ratios in only one stage is not possible. Consequently, fewer components are necessary for worm gearboxes, which allows savings of assembly and maintenance costs.

The overall efficiency of a gearbox \( \eta \) is generally characterized by the sum of load-dependent gearing, no-load gearing and bearing, load-dependent bearing, sealing, and other losses compared to the driving power (see Equation 1) according to the German Standard \( \text{DIN 3996} \) [1].

\[
\eta = 1 - \frac{P_V}{P_1} = 1 - \frac{P_{VL} + P_{VD} + P_{VX}}{P_1}
\]

Where:
- \( \eta \): overall gearbox efficiency / percent
- \( P_1 \): driving power / W
- \( P_V \): overall power losses / W
- \( P_{VL} \): load-dependent gearing losses / W
- \( P_{VD} \): load-dependent bearing losses / W
- \( P_{VX} \): no-load gearing and bearing losses / W
- \( P_{VL} \): sealing losses / W
- \( P_{VX} \): other losses / W

In the case of oil-lubricated worm gears, the gearing losses \( P_{VL} \) are mainly responsible for high overall power losses \( P_V \) — in particular, at low speeds and high torques. High overall power losses are explained by a high sliding motion rate between worm and worm wheel. In order to reduce gearing losses of worm gears, an ideal combination of geometry and material pairing, as well as lubrication and operating conditions, has to be chosen. Therefore, it is necessary to know to what extent these parameters influence the efficiency of worm gears.

With the aim of evaluating and optimizing worm gears, intensive research and development are conducted at the Gear Research Centre (FZG) of Technische Universität München. Numerous projects have been carried out in recent years and are constantly in progress. Within these research projects, various experimental and theoretical investigations have been executed on worm gears with different sizes, materials, lubricants, and test conditions. The tests are conducted primarily with cylindrical worm gears (flank form Z1) and center distances between \( \alpha = 65 \text{ mm} \) and \( \alpha = 160 \text{ mm} \).

Main research topics of the current project [2] are wear and pitting load-carrying capacity as well as efficiency of large-sized worm gears with center distance \( \alpha = 315 \text{ mm} \). In the scope of this paper, all experimental and theoretical results of this project regarding overall gearbox efficiency are presented in detail. Furthermore, these results are compared to the insights of other research projects. Thereby, the influence of different worm gear geometries, materials, and lubricants, as well as lubrication and operating conditions on efficiency, is considered.

MATERIAL AND METHODS

For the experimental investigations, several worm gear test rigs, designed and constructed by FZG, are available. For this project [2], the largest worm gear test rig for worm gears with center distance \( \alpha = 315 \text{ mm} \) is used. Figure 1 shows a photo and the general principle of this FZG large-sized worm gear test rig.

The essential component of this test rig is the test gearbox. In this gearbox, the investigated worm gear is situated. The test worm wheel is driven by the test worm shaft with a certain input speed \( (n_i) \). The worm wheel is connected to a reverse transfer gearbox with identical geometry (driven by the worm wheel). This connection is realized using a double-joint coupling. In order to load the worm wheel of the test gearbox with a certain output.
torque \( (T_2) \), a hydrostatic torque motor is connected to the reverse transfer gearbox. This hydrostatic torque motor infinitely adjusts the respective load. The bracing cycle of the test rig is closed by a summation gearbox. Consequently, the direct current motor only has to feed in the occurring overall power losses.

In the scope of this research project \([2]\), large-sized worm gears with center distance \( a = 315 \, \text{mm} \) are analyzed regarding their overall gearbox efficiency, as well as wear and pitting behavior at different operating and lubrication conditions. The overall efficiency of the tested worm gearbox is the result of input/output speeds and input/output torques. These are measured continuously by torque-measuring shafts at input and output of the investigated worm gearbox (see Figure 1). The used torque-measuring shafts have a measuring accuracy of 0.5 percent (input), respectively 1.0 percent (output). Wear and pitting load-carrying capacity analysis are executed regularly. Results regarding pitting load-carrying capacity are obtained by the assessment of periodically documented flank pictures of the worm wheel. To measure wear on the investigated worm wheel, a transmission error measuring system (see Figure 1) is used.

The investigated worm gearboxes are cylindrical worm gears with flank form Z1 (right-handed thread). The present large-sized worm gears have a center distance of \( a = 315 \, \text{mm} \) and a gear ratio of \( i = 10.25 \). In Table 1, all important gear data of the investigated test gearboxes is listed.

The investigated worm gears are made of case-hardened steel worms (20MnCr5) and centrifugally casted copper-tin bronze worm wheels with nickel (CuSn12Ni2-C-GZ).

The experimental investigations are carried out at various input speeds \( n_1 \) and output torques \( T_2 \). This serves to evaluate wear and pitting load capacity as well as efficiency of worm gears of this size at different operating conditions. The gained results are used to verify current calculation methods for worm gears in the German Standard DIN 3996 \([1]\). The conducted test program in the scope of this project \([2]\) is shown in Table 2. Altogether, four tests are executed at the flanks (fore and rear flanks) of two worm gears.

**Figure 1:** FZG worm gear test rig for large-sized worm gears with center distance \( a = 315 \, \text{mm} \)

**Table 1:** Gear data of investigated test gearboxes

<table>
<thead>
<tr>
<th>symbol</th>
<th>value</th>
<th>unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>center distance ( a )</td>
<td>315 mm</td>
<td></td>
</tr>
<tr>
<td>gear ratio ( i = z_2/z_1 )</td>
<td>41/4 = 10.25</td>
<td></td>
</tr>
<tr>
<td>axial module ( m_x )</td>
<td>12.5 mm</td>
<td></td>
</tr>
<tr>
<td>lead angle ( \gamma )</td>
<td>21.8 °</td>
<td></td>
</tr>
<tr>
<td>profile shift coefficient ( x_2 )</td>
<td>-0.3</td>
<td></td>
</tr>
<tr>
<td>reference ( d_{n1} ), worm</td>
<td>125 mm</td>
<td></td>
</tr>
<tr>
<td>reference ( d_{n2} ), wheel</td>
<td>505 mm</td>
<td></td>
</tr>
<tr>
<td>face width ( b_{3R} )</td>
<td>90 mm</td>
<td></td>
</tr>
<tr>
<td>nominal torque ( T_{2N} ) for ( n_1 = 1200 , \text{min}^{-1} )</td>
<td>11 000 Nm</td>
<td></td>
</tr>
<tr>
<td>for ( n_1 = 300 , \text{min}^{-1} )</td>
<td>22 800 Nm</td>
<td></td>
</tr>
</tbody>
</table>

\( ^1 \) according to \([4]\) for CuSn12Ni2-C-GZ

**Table 2:** Test program on large-sized worm gears (CuSn12Ni2-C-GZ worm wheel with 20MnCr5 worm shaft, \( a = 315 \, \text{mm} \), \( i = 10.25 \))

<table>
<thead>
<tr>
<th>test</th>
<th>input speed ( n_1 ) / ( \text{min}^{-1} )</th>
<th>output torque ( T_2 ) / kNm</th>
<th>output power ( P_2 ) / kW</th>
<th>lubricants for running-in</th>
<th>main test lubricant</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>1200</td>
<td>14.7</td>
<td>180</td>
<td>Klübersynth GH 6-460 (polyglycol)</td>
<td>Klübersynth GH 6-460 (polyglycol)</td>
</tr>
<tr>
<td>2</td>
<td>300</td>
<td>45</td>
<td>45</td>
<td>Klübersynth GH 6-460 (polyglycol)</td>
<td></td>
</tr>
<tr>
<td>3</td>
<td>1200</td>
<td>180</td>
<td>45</td>
<td>Klübersynth GEZ 6-220 (polyglycol)</td>
<td></td>
</tr>
<tr>
<td>4</td>
<td>300</td>
<td>45</td>
<td>45</td>
<td>Renolin CLP 460 (mineral oil)</td>
<td></td>
</tr>
</tbody>
</table>

**Running-in**
To smooth tooth flanks and enlarge the contact pattern, a running-in is carried out at the beginning of each test. The running-in process is executed at a low input speed of \( n_1 = 300 \, \text{min}^{-1} \) in order to cause a high wear rate and therefore a fast enlargement of contact pattern. Meanwhile, the output torque is increased step-by-step \( (T_2 = 2 \ldots 14.7 \, \text{kNm}) \). During running-in, periodic wear/efficiency measurements are executed at an input speed of \( n_1 = 300 \, \text{min}^{-1} \) and an output torque of \( T_2 = 10 \, \text{kNm} \) (measurement condition 1).

The influence of different lubricants on wear load-carrying capacity and efficiency during running-in is evaluated by testing.
diverse oils (see Table 2). All tests are conducted with injection lubrication ($\theta_{\text{Oil}} = 80^\circ\text{C}$). The synthetic oil Klüber synth GH 6-460 (polyglycol, ISO VG 460) is used for the running-in of tests 1 and 2. A polyglycol (Klüber synth GEZ 6-220) is also used for test 3. However, the viscosity class is hereby lower (ISO VG 220). Furthermore, in test 4, a different oil type is utilized. In contrast to tests 1, 2, and 3, running-in is carried out with the mineral oil Renolin CLP 460 (ISO VG 460).

The running-in process of each test is finished by reaching an almost complete contact pattern (contact area $f_t \approx 90$–100 percent). In the present project with large-sized worm gears, running-in lasts approximately between 300 and 550 hours.

**Main Test Run**

After the running-in process, the main test run takes place. For the main tests, two different input speeds are examined to estimate wear behavior and its influence on pitting development. The input speed varies between $n_1 = 1200 \text{ min}^{-1}$ (tests 1 and 3) and $n_1 = 300 \text{ min}^{-1}$ (tests 2 and 4). During all tests, the worm wheel is loaded with an output torque of $T_2 = 14.7 \text{ kNm}$ ($n_1 = 1200 \text{ min}^{-1}$; measurement condition 2a; $n_1 = 300 \text{ min}^{-1}$; 2b). This corresponds to 135 percent ($n_1 = 1200 \text{ min}^{-1}$), respectively 65 percent ($n_1 = 300 \text{ min}^{-1}$) of the nominal torque $T_{2\text{N}}$ according to [4].

All test runs are lubricated with the polyglycol Klübersynth GH 6-460 (ISO VG 460). The type of lubrication is injection lubrication at an injection temperature of $\theta_{\text{Oil}} = 80^\circ\text{C}$ and an injection amount of 25 l/min.

To analyze wear behavior of this kind of large-sized worm gear, regular measurements are carried out during the main test run with the transmission measuring system (see Figure 1). Hereby, the output torque is $T_2 = 10 \text{ kNm}$ and the input speed is $n_1 = 300 \text{ min}^{-1}$ (measurement condition 3). Each wear measurement during test runs lasts exactly one hour.

**Measurement Conditions**

Table 3 gives an overview of all measurement conditions for the documentation of overall gearbox efficiency during running-in and the main test run.

**RESULTS**

In the following section, the results of all executed tests on large-sized worm gears in the course of this research project [2] are introduced and explained in detail. At the same time, the influence of various factors on overall gearbox efficiency is described in particular. These influencing factors concern operating and lubrication conditions, geometry, and material.

**Influence of Contact Pattern and Roughness**

During the experimental investigations, the influence of varying sizes of contact pattern on efficiency is considered in detail as well. For the determination of the contact area $f_t$, all worm wheel flanks are photographed regularly. The evaluation of the contact pattern development of each flank is performed with a specially developed, color-based MatLab program.

At the beginning of the experimental investigations, the contact pattern of each test is adjusted. All initial contact patterns are shown in Table 4.

Each contact area $f_t$ corresponds to the mean contact area of all worm wheel flanks of one test. On average, both tests on the fore flank (tests 1 and 4) show contact areas of 65 to 75 percent. On the rear flank (tests 2 and 3), an average contact area of 60 percent is documented. During the running-in process at $n_1 = 300 \text{ min}^{-1}$ and with gradually enhanced torque $T_2$, a higher wear rate is present. This causes an enlargement of contact. At the end of running-in, tests 1 and 4 show a contact area of approximately $f_t = 90$ percent. In spite of an extended running-in process, test 3 only reaches $f_t = 76$ percent. A small contact pattern of $f_t = 68$ percent is purposely reached for test 2 in order to evaluate wear and pitting behavior of incomplete contact patterns.

In the case of test 1, which is exemplarily shown in Figure 2, first efficiency measurements (measurement condition 1) are carried out after 0.3 million load cycles. At this time, an average contact area of $f_t = 86$ percent is already given. This leads to an initial overall gearbox efficiency of 93.6 percent.

In the further course of the running-in process of test 1, an enlargement of contact pattern from $f_t = 86$ percent to $f_t = 93$ percent after 0.5 million load cycles is realized. As with the contact pattern, an increase in overall gearbox efficiency from 93.6 per-

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**Table 3: Measurement conditions for efficiency evaluation**

<table>
<thead>
<tr>
<th>Test</th>
<th>Test 1</th>
<th>Test 2</th>
<th>Test 3</th>
<th>Test 4</th>
</tr>
</thead>
<tbody>
<tr>
<td>Contact pattern</td>
<td>$f_t = 65%$</td>
<td>$f_t = 60%$</td>
<td>$f_t = 60%$</td>
<td>$f_t = 75%$</td>
</tr>
</tbody>
</table>

**Table 4: Initial load contact patterns ($T_2 = 10 \text{ kNm}$) of conducted tests**

<table>
<thead>
<tr>
<th>Test</th>
<th>Test 1</th>
<th>Test 2</th>
<th>Test 3</th>
<th>Test 4</th>
</tr>
</thead>
<tbody>
<tr>
<td>Contact pattern</td>
<td>$f_t = 65%$</td>
<td>$f_t = 60%$</td>
<td>$f_t = 60%$</td>
<td>$f_t = 75%$</td>
</tr>
</tbody>
</table>
cent to 95.5 percent is documented. After an additional 0.4 million load cycles, an overall gearbox efficiency of $\eta = 96.0$ percent is measured ($f_r = 96$ percent). Consequently, by the contact area enlargement of 10 percent, an efficiency improvement of $\Delta \eta = 2.4$ percent is measured. This conclusion corresponds to the results made by [5] and [6] (see Figure 3). With an increasing contact area, the overall gearbox efficiency improves during the running-in of worm gears.

Besides the enlargement of the contact area, running-in is accompanied by a smoothing of gear flanks. The roughness of the worm wheel flanks adjusts to the roughness of the worm flanks. It is therefore essential to have low worm flank roughness in order to reduce the coefficient of friction and therefore power losses.

The influence of arithmetic mean roughness $Ra$ is considered as well in the German Standard DIN 3996 [1]. In Figure 4, a schematic, mathematical course of the degree of losses $P_V/P_1$ is shown for different roughness values. Therefore, a standard worm gear with center distance $\alpha = 100$ mm and $i = 20.5$ (CuSn12Ni2-C-GZ/16MnCr5, injection lubrication, polyglycol) is used.

The degree of losses is the ratio of overall power losses $P_V$ to driving power $P_1$ — the higher the degree of losses, the lower the overall gearbox efficiency (see Equation 1).

The figure illustrates the reduction of the degree of losses with lower arithmetic mean roughness values. The share of gearing losses $P_{VZ}$ decreases due to lower coefficients of friction. For the other losses, there is no dependence of the arithmetic mean roughness $Ra$. The sealing losses $P_{VD}$ contributes the smallest share of degree of losses.

Next to the size of contact area, the position of contact pattern can influence efficiency. According to Niemann and Winter [5], contact patterns positioned at inlet side cause higher power losses than contact patterns at outlet side. Wakuri [6] measured about 6 percent lower efficiencies for worm gears with contact patterns positioned at inlet side than with contact patterns at outlet side. This is explained by higher coefficients of friction at inlet side.

The coefficient of friction as well as the build-up of lubricating film of worm gears depends on sum velocity $V_S$ [5]. To evaluate friction behavior, the ratio of sliding velocity to sum velocity $\nu_s/V_S$ is used according to Niemann [5] and Wilkesmann [7]. With a decreasing ratio, the coefficient of friction decreases as well [5]. The distribution of sum velocities and sliding velocities can be calculated for all kinds of worm gear geometries with the software SNESYS [8] (in particular, with the included program SNETRA). Next to this, SNETRA [8] delivers a course of contact lines, contact pattern under load and no-load, as well as distribution of equivalent radius of curvature, Hertzian stresses, and lubricant film thickness. It is also possible to simulate complete and incomplete contact patterns. To simulate present investigations on the large-sized worm gear with center distance $\alpha = 315$ mm in SNETRA, the contact patterns after running-in are used. Figure 5 shows the results regarding the amount of sum velocities (left) and sliding velocities (right) achieved for test 1. Higher coefficients of friction at inlet side than those at outlet side are explained by lower sum velocities at this side of the worm wheel (see Figure 5). In the area where low
sum velocities occur, low lubricant film thicknesses arise. Thus, worm gears with contact patterns at outlet side show better efficiencies. The distribution of lubricant film thickness is illustrated in Figure 6.

**Influence of Input Speed**

Sum and sliding velocity mainly depend on current input/output speed. In Figure 7, a typical course of measured overall gearbox efficiency $\eta$ for the test gearbox during the main run of test 1 at the input speed of $n_1 = 1200 \text{ min}^{-1}$ (measurement condition 2a) is illustrated.

Thereby, the worm gear is loaded with an output torque of $T_2 = 14.7 \text{ kNm}$ at an input speed of $n_1 = 1200 \text{ min}^{-1}$ (measurement condition 2a). It is lubricated with the polyglycol Klübersynth GH 6–460. During the measurement, a constant overall gearbox efficiency of approximately $\eta = 96$ percent is documented.

Further measurements during test 1 result in similar values between 94.6 percent and 96 percent. The test at the same operating conditions (test 3 according to Table 2) shows a comparable course of overall gearbox efficiency of up to $\eta = 94.6$ percent during the main test run.

For the lower input speed of $n_1 = 300 \text{ min}^{-1}$ with same output torque of $T_2 = 14.7 \text{ kNm}$ (tests 2 and 4, measurement condition 2b), slightly lower efficiencies of $\eta = 94.0–94.5$ percent are achieved than for test 1 ($\eta = 94.6–96.0$ percent, measurement condition 2a: $n_1 = 1200 \text{ min}^{-1}$). This is explained by lower sum velocities and an unfavorable lubrication film formation. Lower sum velocities lead to an increase in coefficient of friction and therefore to a decrease in efficiency.

The high influence of input speed on worm gearbox efficiency is considered as well in DIN 3996 [1]. Figure 8 illustrates a schematic, mathematical course of the degree of losses $P_V/P_1$ via the input speed $n_1$ of a standard worm gear (CuSn12Ni2-C-GZ/16MnCr5, $a = 100 \text{ mm}$, $i = 20.5$, injection lubrication, polyglycol) according to DIN 3996 [1].

Hereby, lower speeds lead to higher degrees of losses. The share of gearing losses $P_{VZ}$ increases due to higher coefficients of friction with lower input speeds. According to DIN 3996, the shares of bearing losses $P_{VL}$ and no-load losses $P_{V0}$ decrease with a slower rotating worm. This is explained by lower input power $P_1$, which leads to a higher degree of losses $P_V/P_1$ (see Equation 1).

Consequently, large-sized worm gears have a similar behavior regarding the influence of input speed on overall gearbox efficiency.

**Influence of Output Torque**

Next to speed, the present output torque $T_2$ influences the overall efficiency of worm gears [5]. Usually, lower loads lead to lower efficiency values [5]. In the scope of this project [2] with large-sized worm gears, the same behavior is documented. With a lower load and the same input speed during the main test measurements of tests 2 and 4 (measurement condition 3: $T_2 = 10 \text{ kNm}$, $n_1$...
300 min\(^{-1}\), slightly lower efficiencies (\(\Delta = 0.1–0.7\) percent) toward the results with \(T_2 = 14.7\) kNm and \(n_1 = 300\) min\(^{-1}\) (measurement condition 2b) are recorded. Figure 9 shows a schematic, mathematical course of the degree of losses \(P_{V}/P_1\) via the output torque \(T_2\) for a standard worm gear in conformity with Figure 4 and Figure 8.

In general, the degree of losses \(P_{V}/P_1\) increases with a decreasing output torque \(T_2\). The absolute values of the no-load losses \(P_{V0}\) and the sealing losses \(P_{VD}\) are not influenced by the output torque. Because of a decreasing driving power \(P_1\), the shares of \(P_{V0}\) and \(P_{VD}\), however, are becoming bigger. With respect to the load-dependent gearing losses \(P_{VZ}\), a decrease of its share of the degree of losses results in lower torque. Load-dependent bearing losses \(P_{VL}\) have an almost constant share of the degree of losses.

An exemplary efficiency field for worm gears at different operating conditions is shown in Figure 10. The results are taken from former and actual experimental investigations on steel-bronze worm gears, with center distances \(a = 65, 100, 160,\) and \(315\) mm, executed at FZG by [9], [10], and [2]. All tests are injection-lubricated with polyglycol (ISO VG 220 or 460).

The figure emphasizes the described influence of speed and load on overall efficiency of steel-bronze worm gears. Depending on the combination of input speed \(n_1\) and output torque \(T_2\), overall efficiencies between \(\eta = 84\) percent and \(\eta = 95\) percent are reached. For a worm gear with center distance \(a = 100\) mm and ratio \(i = 10.3\), for example, Rank [9] measured an overall gearbox efficiency of \(\eta = 94.7\) percent at the operating conditions \(n_1 = 2900\) min\(^{-1}\) with \(T_2 = 1180\) Nm.

**Influence of Oil Viscosity**
To evaluate the influence of oil viscosity, efficiency results at the end of running-in of tests 1 and 2 (polyglycol ISO VG 460) are compared to those of test 3 (polyglycol ISO VG 220) (see Table 2). All three measurements are conducted at the same input speed \(n_1 = 300\) min\(^{-1}\) and output torque \(T_2 = 10\) kNm (measurement condition 1). According to Niemann [5], oils with higher viscosity lead to higher no-load losses but lower power losses due to better lubricant film formation. In the present investigations, slightly higher overall gearbox efficiency (\(\Delta \approx 1–2\) percent) can also be measured with higher oil viscosity.

In the present German Standard DIN 3996 [1], the influence of different oil viscosities on the efficiency of worm gears is currently not taken into account.

**Influence of Oil Type**
Next to oil viscosity, oil type influences worm gear efficiency significantly. In general, synthetic lubricants (e.g., polyglycols, polyal-phaolefins, or ester oils) lead to lower coefficients of tooth friction in comparison to mineral oils [5]. Low power losses and therefore better efficiencies come along with low coefficients of friction [5]. The influence of the oil type is more significant with higher sliding velocities.

In the present tests, the impact of different oil types is clear. Therefore, efficiency values measured at the end of running-in of test 3 and test 4 are compared. Test 3 is lubricated with the polyglycol GEZ 6–220 during running-in, whereas for the running-in of test 4, the mineral oil CLP 460 is used. Both tests are conducted at a lubricant injection temperature of \(T_{Oil} = 80\) °C. This leads to similar kinematic viscosities for GEZ 6–220 (test 3) and CLP 460 (test 4) according to Figure 11. In addition, comparable dynamic viscosities also result according to the German Standard DIN 3996 [1]. An influence of different oil viscosities can therefore be excluded.
With the mineral oil CLP 460, an overall gearbox efficiency of only $\eta = 91$ percent is measured at an input speed of $n_1 = 300 \text{ min}^{-1}$ and an output torque of $T_2 = 10 \text{kNm}$ (measurement condition 1) compared to $\eta = 93.7$ percent with the lubricant GEZ 6-220. Consequently, the reduction of efficiency is explained by the use of mineral oil.

The influence of oil type on efficiency is also considered in the German Standard DIN 3996 [1]. Worm gears lubricated with mineral oils show higher degrees of losses than worm gears lubricated with polyglycol. Figure 12 illustrates the schematic, mathematical influence of oil type on degree of losses for a standard worm gear (CuSn12Ni2-C-GZ/16MnCr5, $\alpha = 100 \text{ mm}$, $i = 10.33$, injection lubrication, polyglycol, $n_1 = 1500 \text{ min}^{-1}$, $T_2 = 680 \text{ Nm}$).

The share of gearing losses $P_{\text{VZ}}$ in the degree of losses is especially higher. This is explained by higher coefficients of friction for gears lubricated with mineral oils. The other shares in the degree of losses are not influenced by the oil type, pursuant to DIN 3996 [1].

All described test results from this project [2] regarding overall worm gearbox efficiency are summarized in Table 5 for the end of running-in and in Table 6 for the results during the main test run.

### Table 5: Overall gearbox efficiency at the end of running-in

<table>
<thead>
<tr>
<th>Test</th>
<th>Input speed $n_1$ / min$^{-1}$</th>
<th>Output torque $T_2$ / kNm</th>
<th>Test lubricant</th>
<th>Overall gearbox efficiency $\eta$ / %</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>1200 (2a)</td>
<td>14.7</td>
<td>GH 6-460</td>
<td>94.6–96.0</td>
</tr>
<tr>
<td>2</td>
<td>300 (2b)</td>
<td>94.5</td>
<td></td>
<td></td>
</tr>
<tr>
<td>3</td>
<td>1200 (2a)</td>
<td>93.5–94.6</td>
<td></td>
<td></td>
</tr>
<tr>
<td>4</td>
<td>300 (2b)</td>
<td>94.0</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

### Table 6: Overall gearbox efficiency during the main test run

<table>
<thead>
<tr>
<th>Test</th>
<th>Input speed $n_1$ / min$^{-1}$</th>
<th>Output torque $T_2$ / kNm</th>
<th>Test lubricant</th>
<th>Overall gearbox efficiency $\eta$ / %</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>1200 (2a)</td>
<td>10</td>
<td>GH 6-460</td>
<td>94.6–95.6</td>
</tr>
<tr>
<td>2</td>
<td>300 (2b)</td>
<td>94.4</td>
<td></td>
<td></td>
</tr>
<tr>
<td>3</td>
<td>1200 (2a)</td>
<td>93.3–94.6</td>
<td></td>
<td></td>
</tr>
<tr>
<td>4</td>
<td>300 (2b)</td>
<td>93.5</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

With the mineral oil CLP 460, an overall gearbox efficiency of only $\eta = 91$ percent is measured at an input speed of $n_1 = 300 \text{ min}^{-1}$ and an output torque of $T_2 = 10 \text{kNm}$ (measurement condition 1) compared to $\eta = 93.7$ percent with the lubricant GEZ 6–220. Consequently, the reduction of efficiency is explained by the use of mineral oil.

The influence of oil type on efficiency is also considered in the German Standard DIN 3996 [1]. Worm gears lubricated with mineral oils show higher degrees of losses than worm gears lubricated with polyglycol.

### Influence of Pitting

During all conducted tests according to Table 2, pitting on the worm wheel flanks occurs. Rank [9] and [11] determines — in his examinations on bronze worm wheels paired with case-hardened worms, mainly with center distance $\alpha = 100 \text{ mm}$ — a characteristic of pitting development. According to Rank, the lifetime of bronze worm wheels is divided into three stages. This typical three-phase course is illustrated in Figure 13 (CuSn12Ni-C-GZ/16MnCr5, $\alpha = 100 \text{ mm}$, $i = 10.33$, injection lubrication, polyglycol, $n_1 = 1500 \text{ min}^{-1}$, $T_2 = 1180 \text{ Nm}$).

In general, the pitting development is characterized by different specific values. $A_{\text{Pm}}$ is the average of the pitting area of all worm wheel flanks. $A_{\text{Pmax}}$ describes the maximum pitting area of all worm wheel flanks. The value $A_{\text{P10}}$ is the mean pitting area of 10 percent of the most damaged worm wheel teeth. $A_{\text{P10}}$ represents the decisive parameter for the description and calculation of pitting development.

At the beginning of the service life of a worm gear, first pits occur in the area of highest worm wheel flank pressures. This section is called the no pittings stage (phase I) [11]. After reaching $A_{\text{P10}} = 2$ percent, pits grow approximately in a linear way. During this period — called the pitting growth stage (phase II) — pitting area increases up to a maximum value ($A_{\text{P10,max}}$). After reaching this maximum, pitting area decreases again, due to high wear during the wear stage (phase III).

In accordance with the examinations of Rank [9] and [11], the present large-sized, bronze worm wheels show the typical three-phase course. An exemplary course of pitting development...
damage documented during present examinations [2] is shown in Figure 14 for test 1.

Table 7 illustrates the corresponding photos of pitting damage on worm wheel flanks after different load cycles $N_L$.

After $N_L = 2.8$ million load cycles, a pitting area of $A_{P10} = 2$ percent and consequently the end of phase I is reached. Subsequently, pitting area increases further during the pitting growth stage (phase II) until a maximum of $A_{P10,max} = 46$ percent is reached. As with the examinations executed by Rank, a decrease of pitting area in phase III is documented.

Despite an increasing pitting area of up to $A_{P10} = 46$ percent (phase II), no significant change in overall gearbox efficiency is documented, as it is shown for test 1 in Figure 15.

In Figure 15, the results of all efficiency measurements of test 1 in accordance to Table 3 are illustrated. During running-in, periodic wear/efficiency measurements (measurement condition 1) are executed at an input speed of $n_1 = 300$ min$^{-1}$ and an output torque of $T_2 = 10$ kNm. For the main test, the input speed is $n_1 = 1200$ min$^{-1}$ with an output torque of $T_2 = 14.7$ kNm (measurement condition 2a). To analyze wear behavior of this kind of large-sized worm gear, regular measurements are carried out during the main test run with the transmission measuring system (see Figure 1). Hereby, the output torque is $T_2 = 10$ kNm and the input speed is $n_1 = 300$ min$^{-1}$ (measurement condition 3). Each wear measurement during test runs lasts exactly one hour.

The same behavior regarding pitting development, as well as overall gearbox efficiency, is documented for the other conducted tests 2, 3, and 4. In tests 2 and 4, a different input speed of $n_1 = 300$ min$^{-1}$ (measurement condition 2b) is examined to estimate wear behavior and its influence on pitting development. The results for overall gearbox efficiency, in dependence of the load cycles of the worm wheel, are shown in Figure 16 (test 2), Figure 17 (test 3), and Figure 18 (test 4).

At the same time, material is removed from the worm wheel flanks due to wear. The intensity of wear varies during the different pitting phases. As well as for pitting, no influence of wear on overall gearbox efficiency is documented during all conducted experimental investigations.

Efficiency Calculation with SNDIN [8]

Next to the described simulation with SNESYS [8] (previously discussed), the included program SNDIN allows a load-carrying capacity and efficiency calculation according to DIN 3996 [1] with SNDIN. For the present large-sized worm gear, a calcula-
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Influence of Center Distance and Lead Angle

To improve gearbox efficiency, a right choice of worm gear geometry is important. Niemann and Winter [5] particularly show a correlation between the ratio of diameter to center distance \( d_{m1}/a \), the lead angle \( \gamma_m \), the coefficient of friction \( \mu_z \), and the gearing efficiency \( \eta_z \). Higher gearing efficiency \( \eta_z \) is accompanied by a decreasing ratio of diameter to center distance \( d_{m1}/a \) [5]. This, however, results in lower safety factors for pitting and deflection. With an increasing lead angle \( \gamma_m \), the coefficient of friction gets lower, which leads to higher gearing efficiencies (see Figure 19) [12].

Influence of Gear Ratio

Furthermore, there is an influence of varying gear ratio \( i \) on overall efficiency of worm gears. A decrease in gear ratio \( i \) (multi-start worm) can be accompanied with an improvement of efficiency [5]. According to Niemann and Winter [5], overall gearbox efficiencies of \( \eta_z = 96 \) percent (gearing efficiency \( \eta_z = 98 \) percent) are documented with small ratios \( i = 5 \) at an input speed of \( n_1 = 1500 \text{ min}^{-1} \) (lubrication with mineral oil).

Excessively high values of gear ratio can result in self-locking worm gears (with driving worm \( \eta_z < 50 \) percent). Figure 20 illustrates this influence of different gear ratios \( i \) on overall gearbox efficiency according to [13].

In the scope of Weisel’s investigations [10], he carried out test runs at the same large-sized test rig (see Figure 1). These worm gears (CuSn12Ni2-C-GZ/16MnCr5, \( \alpha = 315 \text{ mm} \), \( \gamma_m = 14.04^\circ \)) have a higher gear ratio of \( i = 20.5 \) compared to the tests conducted in [2] \( i = 10.25 \). The main test runs are executed at \( T_2 = 15.75 \text{ kNm} \) and \( n_1 = 1500 \text{ min}^{-1} \). Despite the difference in lead angle \( \gamma_m \) and input speed, similar sliding velocities are resulting ([2]: \( v_{gm} = 8.5 \text{ m/s} \); [10]: \( v_{gm} = 8.7 \text{ m/s} \)). Furthermore, similar mean flank stresses (\( \sigma_{HM} = 343 \text{ N/mm}^2 \))

FURTHER INFLUENCING FACTORS ON WORM GEAR EFFICIENCY

Next to the influencing factors on overall worm gearbox efficiency, described earlier, by mainly using the results of the project on large-sized worm gears [2], further variables can result in an optimization of efficiency.

<table>
<thead>
<tr>
<th>Input speed ( n_1 ) / min(^{-1})</th>
<th>Output torque ( T_2 ) / kNm</th>
<th>Overall gearbox efficiency</th>
</tr>
</thead>
<tbody>
<tr>
<td>300</td>
<td>10</td>
<td>94.1</td>
</tr>
<tr>
<td></td>
<td>14.7</td>
<td>92.1</td>
</tr>
<tr>
<td>1200</td>
<td>14.7</td>
<td>-</td>
</tr>
<tr>
<td></td>
<td>95.1</td>
<td>-</td>
</tr>
</tbody>
</table>

Table 8: Efficiency calculation with SNDIN [8] according to DIN 3996 [1]

In SONDIN was undertaken with all used lubricants and investigated operating conditions. The calculated overall gearbox efficiency results are shown in Table 8.

For the input speed of \( n_1 = 300 \text{ min}^{-1} \), no influence of both output torques on calculated efficiency is recognizable. The effect of different oil viscosities is not considered yet in DIN 3996 [1].

Altogether, a good correspondence between the calculated values and the experimentally gained overall gearbox efficiencies is determined. Thereby, the measured efficiencies are slightly better than the calculated values. In total, efficiencies of up to 96 percent can be gained in these worm gearboxes.
according to DIN 3996 [1] are calculated. In comparison with test 1 (measurement condition 2a) previously described, lower overall gearbox efficiencies of \( \eta = 93 \text{ percent (} \Delta = 2–3 \text{ percent) are documented by Weisel [10] during these main test runs. As a consequence, this fact is explained, for example, by the different gear ratios as well as the used lubricant, which has a lower viscosity (polyglycol, ISO VG 220) compared to test 1 (polyglycol, ISO VG 460).

### Influence of Profile Flank Form

According to Niemann and Winter [5], higher efficiencies are reached by using concave profile flank forms (ZH respectively ZC). This is explained by a better osculation and a steeper course of the contact lines, which causes a better lubricant film and fewer power losses. Hellemann [14] shows gearing efficiencies for worm gears with profile flank form ZC and an improved variant of it. He indicates overall gearbox efficiencies of up to 97 percent.

Furthermore, globoid worm gears (globoid worm with globoid worm wheel) show similar efficiencies as cylindrical worm gears with ZI profile flank form [5]. Helical worm gear units (globoid worm with spur wheel) are theoretically similar to ZI worm gears regarding power losses [5]. However, the manufacturing of globoid worms is more complicated.

### Influence of Material

Due to a high sliding motion rate, usually hard worm shafts are paired with soft worm wheels. With regard to efficiency, worm gears with wheels consisting of Cu-Sn-bronze and worms consisting of case-hardened steel tend to be better than wheels made of cast iron, Al-bronze, or special brass [5]. This can be explained by better sliding properties of bronze worm wheels [15]. Figure 21 illustrates this influence of materials on the degree of losses for worm wheels made of bronze (CuSn12Ni) and cast iron (GJS-400) according to DIN 3996 [1] (\( \alpha = 100 \text{ mm, } i = 20.5, \text{ dip lubrication, polyglycol} \)). The figure emphasizes the higher shares of gearing losses \( P_{VZ} \) in the degree of losses for cast iron worm wheels due to higher coefficients of friction.

### Influence of Type of Lubrication

By using oil, different types of lubrication are realizable. Next to injection lubrication, dip lubrication is a typical lubrication type for worm gears. These different types of lubrication affect overall gearbox efficiency. Figure 22 shows the theoretical degree of losses depending on the type of lubrication for a standard worm gear (CuSn12Ni2-C-GZ/16MnCr5, \( \alpha = 100 \text{ mm, } i = 20.5, \text{ polyglycol, } n_1 = 1500 \text{ min}^{-1}, T_2 = 680 \text{ Nm} \)) according to DIN 3996 [1]. With injection lubrication, the degree of losses is slightly higher than with dip lubrication. This is explained by a higher share of gearing losses \( P_{VZ} \). An influence of lubrication type on the other losses has not yet been taken into account in DIN 3996 [1]. Especially with regard to no-load losses \( P_{V0} \), the type of lubrication can have considerable impact. In order to realize injection lubrication, an oil pump is necessary. This leads to further losses, which are not considered in the comparison in Figure 22.

### Influence of Type of Lubricant

Usually, worm gears are lubricated with oil. In addition to oil lubrication, grease can also be used for the lubrication of worm gears in certain applications. Here, however, it is important to note that they are mostly only applicable at very low circumferential velocities (\( < 8 \text{ m/s} \)) [5]. Monz [12] investigates worm gears lubricated with different greases. He points out the possibility of increasing efficiency of worm gears by using greases — especially greases of NLGI class 2 — instead of the corresponding base oils. This is explained by a reduction of no-load losses \( P_{V0} \), in particular, at very low speeds and small worm gear sizes in comparison to oil lubrication. However, grease-lubricated worm gears show higher wear due to less favorable heat dissipation and therefore higher thermal loads, opposed to oil. Further potentials of grease-lubricated worm gears with regard to thermal performance, wear behavior, and efficiency are currently being examined in [15].

### CONCLUSION

Worm gears are still a fundamental component in the field of high transmission gears (\( i = 1 \) to 80). Overall worm gearbox efficiencies of up to \( \eta = 96 \text{ percent} \) are realizable. This was confirmed at FZG in the scope of experimental and theoretical investigations on large-sized worm gears ((CuSn12Ni2-C-GZ/20MnCr5, \( \alpha = 315 \text{ mm, } i = 10.25 \)) [2]. In some cases, even higher efficiency values than \( \eta = 96 \text{ percent} \) are achievable.

With regard to an optimization of worm gear efficiency, different influencing factors were investigated during the experimental tests. In
this paper, these factors are described in detail and are compared to further experimental and theoretical results. The most important influencing factors on worm gear efficiency are:

Geometry:
- Higher gearing efficiency is accompanied by a decreasing ratio of diameter \( d_{m1} \) to center distance \( a \).
- An increasing lead angle \( \gamma_m \) leads to higher gearing efficiencies.
- A decrease in gear ratio \( i \) can result in an improvement of efficiency.

Material pairing:
- The pairing of copper-tin bronze worm wheel with case-hardened worm tends to have higher efficiency than wheels made of cast iron, aluminum bronze, or special brass, because of lower gearing losses.

Operating conditions:
- High input speeds \( n_1 \) and high output torques \( T_2 \) usually lead to an increase in overall gearbox efficiency.

Lubricant:
- Oils with high viscosity result in higher no-load losses but lower load power losses due to better lubricant film formation in gears and bearings.
- With synthetic oil types (especially polyglycols), higher efficiency values are reached in comparison to mineral oils (for the pairing of bronze worm wheel with case-hardened worm).
- According to DIN 3996 [1], slightly higher losses are calculated with the lubrication-type injection lubrication than with dip lubrication. This is explained by higher gearing losses. The influence of lubrication type on the other losses is actually not considered.
- Next to oil, grease can be used for the lubrication of worm gears at special operating conditions. Especially at very low input speeds, grease lubrication (e.g., greases of NLGI 2) often has a reducing effect on no-load losses in comparison to oil.

Contact pattern:
- An enlargement of contact pattern mostly correlates with an increase in efficiency.
- Contact patterns positioned at inlet side cause higher power losses than contact patterns at outlet side due to higher coefficients of friction at inlet side.

Roughness:
- Low worm flank roughness can reduce the coefficient of friction and therefore power losses.

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REFERENCES

Dr.-Ing. Johann-Paul Steemplinger has been responsible for testing automatic gearboxes at AUDI AG since 2015. Until December 2015, he worked as a research group manager at FZG, Gear Research Centre, Technische Universität München in Germany and was responsible for the group’s gearbox efficiency, worm, bevel, and hypoid gears. In 2012, he was responsible for gearbox efficiency as a team leader at FZG. Steemplinger holds a M.Sc. in mechanical engineering of Technische Universität München and finished his doctoral thesis in the field of tribology of gear contacts at FZG in 2013. He has authored over 20 publications including four patents in the field of powertrain and transmission focusing on tribology and lubrication of gears. He received the Oskar Karl Forster and the Prof. Dr. Wilhelm Wittmangesch receipt of the 2008.

Prof. Dr.-Ing. Karsten Stahl studied mechanical engineering at the Technische Universität München before serving as research associate at the Gear Research Centre (FZG) at the Technical University Munich. In 2001, he received his Ph.D. in mechanical engineering and started as gear development engineer at the BMW group in Dingolfing, subsequently named head of Prototyping, Gear Technology & Methods in 2003. From 2006–2009, he changed to the BMW/MINI plant in Oxford, U.K., first as group leader, and in 2007, as department leader for validation driving dynamics and powertrain. In 2009, Stahl returned to Munich and was responsible for predevelopment and innovation management within BMW Driving Dynamics and Powertrain. In 2011, Stahl was named head of the Institute for Machine Elements and the Gear Research Centre (FZG) at the Technische Universität München.

ABOUT THE AUTHORS: Eva-Maria Mautner is currently a research associate and Ph.D. student at the Gear Research Centre (FZG) at the Technische Universität München (TUM) in Germany, specializing in efficiency and load-carrying capacity of worm gears with regard to geometry, lubrication, and operating conditions. She studied mechanical engineering at the Technische Universität München. She is currently working on her Ph.D. thesis on “Wear and pitting load carrying capacity of large-sized worm gears.”

Werner Sigmund studied mechanical engineering at the Technische Universität München (TUM) in Germany, specializing in efficiency and load-carrying capacity of worm gears with regard to geometry, lubrication, and operating conditions. She studied mechanical engineering at the Technische Universität München. She is currently working on her Ph.D. thesis on “Wear and pitting load carrying capacity of large-sized worm gears.”

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