Worm Gear Efficiency Estimation and Optimization

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This paper outlines the comparison of efficiencies for worm gearboxes with a center distance ranging from 28–150 mm that have single reduction from 5 to 100:1. Efficiencies are calculated using several standards (AGMA, ISO, DIN, BS) or by methods defined in other bibliographic references. It also deals with the measurement of torque and temperature on a test rig—required for the calibration of an analytical model to predict worm gearbox efficiency and temperature. And finally, there are examples of experimental activity (wear and friction measurements on a block-on-ring tribometer and the measurements of dynamic viscosity) regarding the effort of improving the efficiency for worm gear drivers by adding nanoparticles of fullerene shape to standard PEG lubricant.

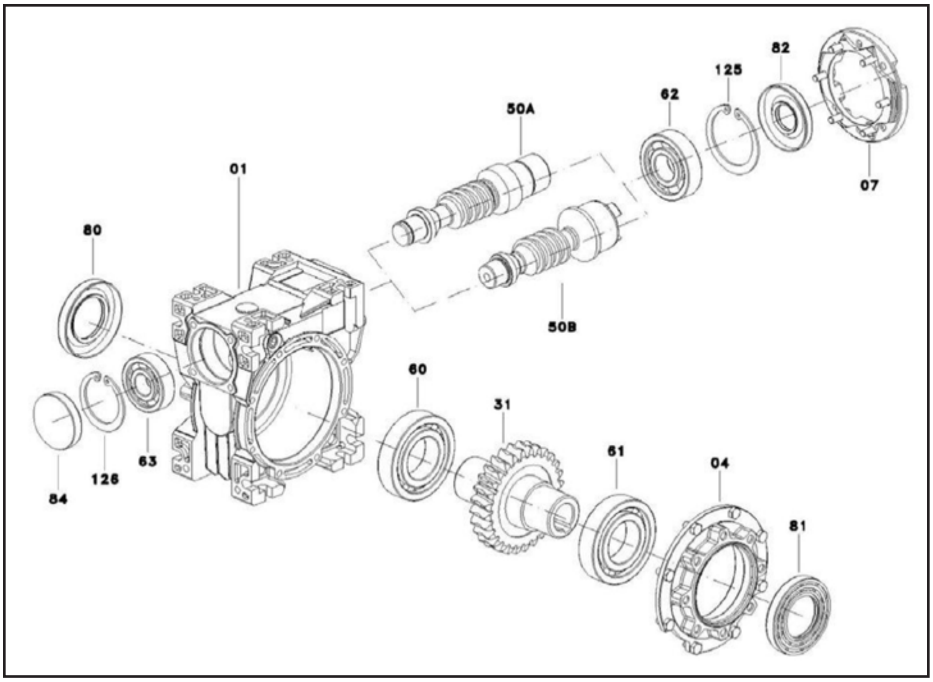

Figure 1 Typical worm gearbox components (Ref. 1)—aluminum housing; case-hardened steel worm with *ZI* profile; bronze wormgear molded on a cast iron shaft; four ball bearings with Seeger rings and oil seals; the lubricant is a synthetic PEG oil with viscosity grade 320 per AGMA 9005-E02 (Ref. 2)—tab B.5.

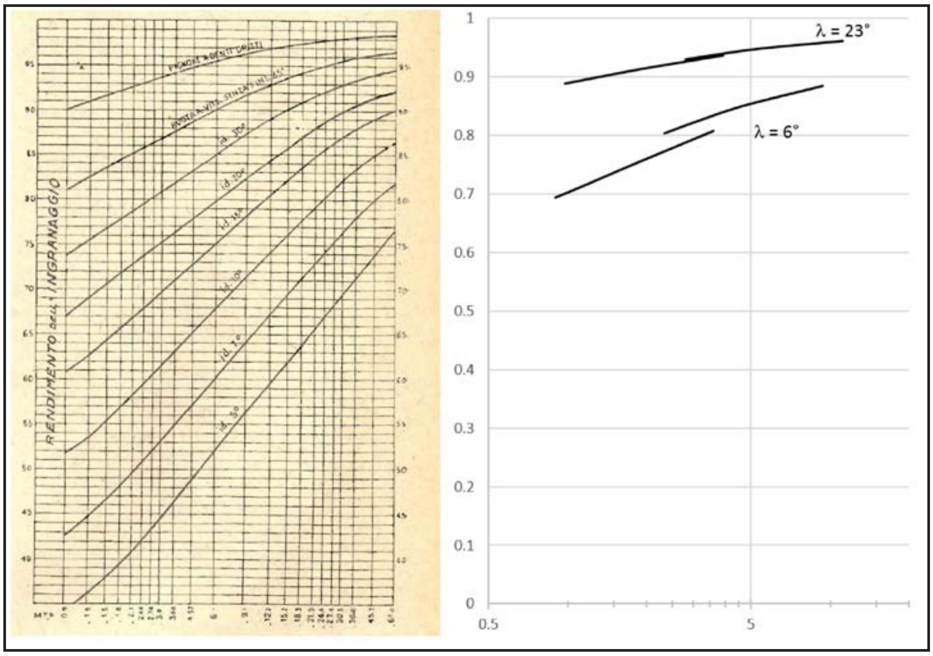

Figure 2 Worm gearbox efficiency vs. sliding velocity (m/s) for different lead angles: a) historical data (Ref. 4); b) AGMA 6034-B92 (Ref. 5) on current worm speed reducer (Ref. 1).

Introduction

The worm gearbox is widely used for high reduction ratios, because it can provide both "small cost" and "small size" by allowing a single reduction unit to be used for many applications.

Worm gearboxes are a strategic product for many Italian companies, constituting up to 50% of sales. For this reason, the paper presents some examples of factors affecting worm gearbox efficiency and investigates means of optimization, particularly on medium-sized worm gearboxes (with center distance between 28 and 150 mm).

Efficiency Calculation According to Standards

Current Italian worm gearbox catalogues often show data calculated according to the old BS 721 (Ref. 3). Many companies are now evaluating which one of the latest standards they should adopt. For this reason, the Authors made a massive efficiency calculation on several size and ratio

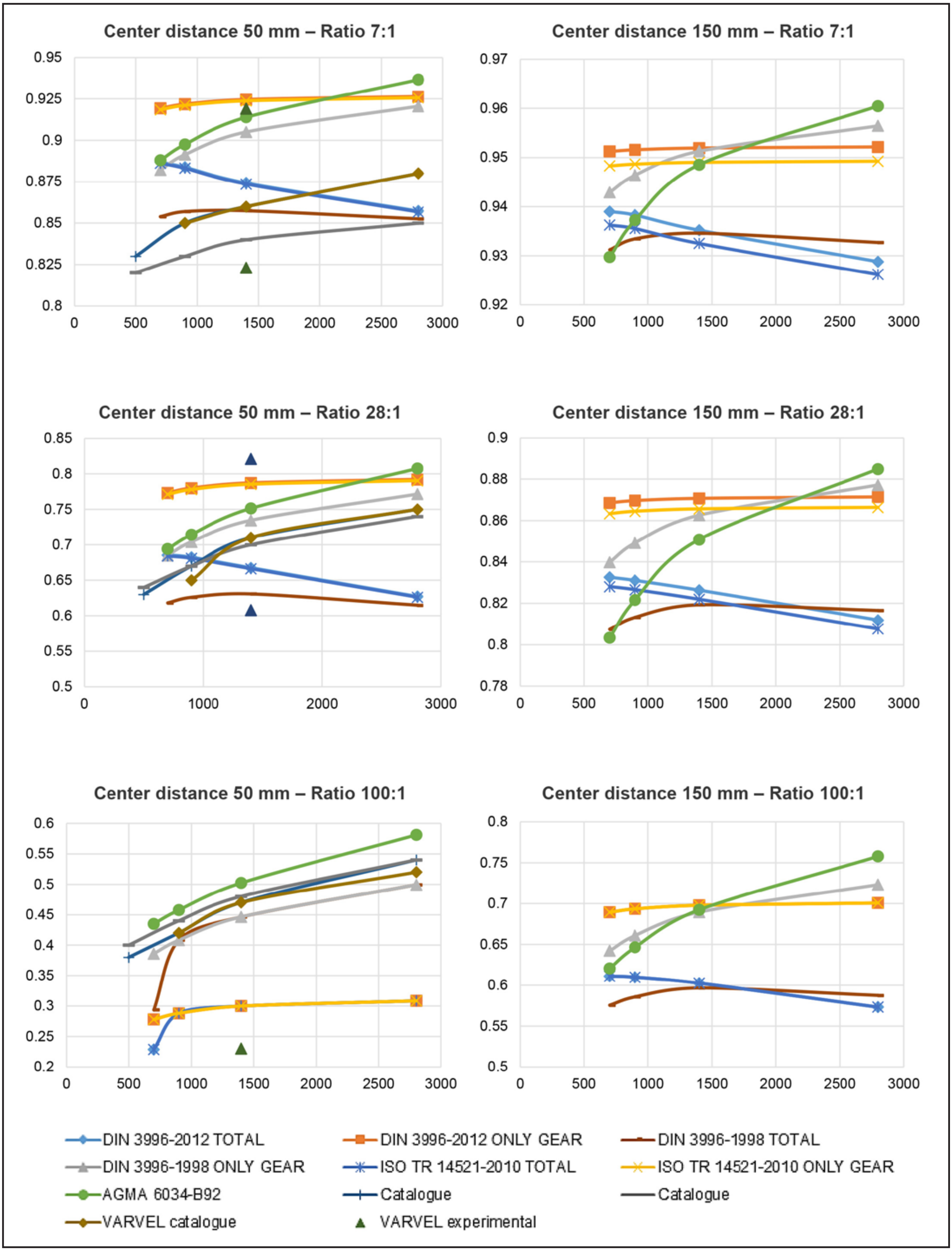

Figure 3 **Worm gearbox efficiency according to different standards and catalogues—*η* vs. *n* (rpm).**

configuration worm gearboxes, according to many standards, and also comparing results with values from catalogues of some companies.

The first studies of this kind in Italy date back to a century ago (1920s) as reported in (Ref. 4). A comparison between historical data (Fig. 2b) and the results obtained by the authors (Fig. 2a)

Figure 3 shows results applying the nominal output torque and according to DIN 3996-1998 (Ref. 6), DIN 3996-2012 (Ref. 7), ISO/TR 14521 (Ref. 8), and AGMA 6034-B92 (Ref. 5). When it is possible, the plot shows both meshing and total efficiency.

There are not any results about smaller gearboxes ($a = 28$ mm) because they are out of the range of ISO and DIN standards.

Figure 3 also shows efficiency from competitor catalogues. Note that often in the catalogues there is an indication that the efficiency settles after the running-in period, but the indication of time ranges from "a few hours" to "200 – 800 h."

In the plot there are also some points experimentally detected in the tests described below.

For the calculation of the friction coefficient, the authors used the method C for the DIN and ISO standards, and the oil temperature does not influence the efficiency.

Looking at the plots in Figure 3, note that:

- For the same input speed and ratio, the efficiency changes as a function of size (center distance), especially because there is the friction coefficient variation as a function of absolute speed in the reference diameter, which changes with the gearbox size
- DIN 3996-2012 and ISO/TR 14521 have similar plots
- AGMA 6034-B92 is quite similar to the old version of DIN 3996
- AGMA 6034-B92 has only the mesh efficiency

Experimental Definition of Temperature and Efficiency; Calibration of the Model for Their Prediction

Varvel SpA R&D and sales departments usually work with *KISSsys* (software that allows users to model drivetrains, based on *KISSsoft*, the popular program for sizing, optimizing, and recalculating designs

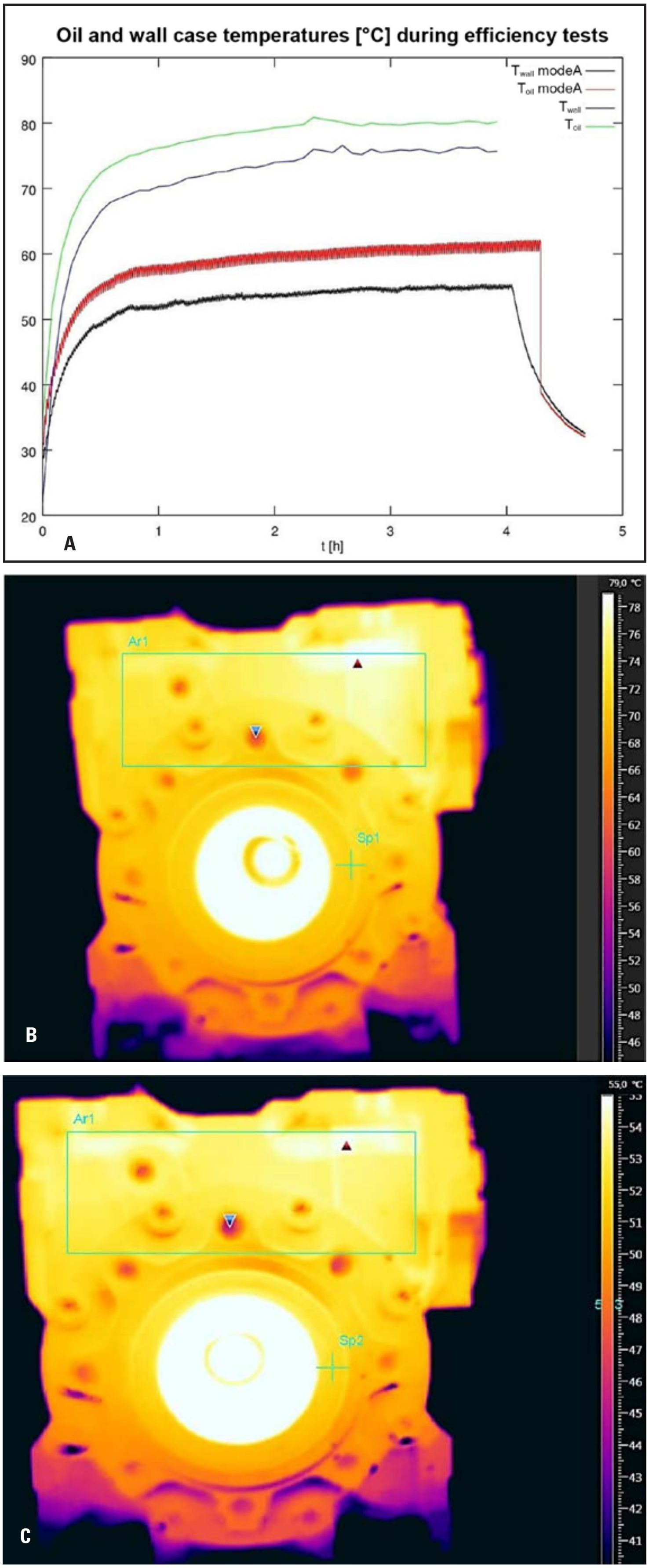

Figure 4 **Comparison between T_{oil} and T_{wall} for continuous vs. alternating operating modes — a) temperature trend during efficiency tests; b) infrared frame after 4h in continuous operating mode; c) infrared frame after 4h in alternating mode.**

Table 1 Experimental results of efficiency η and temperature tests compared with calculated values

Ratio	Input rotation	Output Torque measured [Nm]	Input Power measured [KW]	Input Power KISSsys [KW]	η measured [%]	η KISSsys [%]	T_{wall} measured [°C]	T_{oil} measured [°C]	T_{oil} KISSsys [°C]	T_{wall} KISSsys [°C]
7	CW	68.07± 0.18	1.559± 0.005	1.6	91.9± 0.3	89.1	75.2± 0.4	79.2± 0.5	73.3	68.9
7	CCW	72.8± 0.2	1.863± 0.006	1.7	82.3± 0.2	89.2	78.8± 0.5	85.1± 0.3	75.1	70.6
28	CW	75± 0.4	0.481± 0.007	0.53	82.1± 0.2	73.9	74.5± 1.5	78.8± 1.4	71.4	67.1
28	CCW	76.6± 0.4	0.663± 0.009	0.542	60.8± 0.9	74.0	78.4± 1.1	82.1± 0.7	70.9	66.6

for machine components such as gears, shafts and bearings (Ref. 9)), to help customers with the right gearbox choice from the catalogue for their application. With a single model they can calculate lifetime or safeties for gearboxes of the same type with different size and ratio.

When a customer is looking for a worm gearbox, often he enquires about the efficiency and housing temperature for its application — especially in non-steady operating conditions. And so the model has been enhanced with this feature.

But to do this, initially, an accurate experimental campaign was conducted in order to define worm gearboxes' performance in terms of efficiency and temperature. Shown (in this presentation) are some results obtained on gearboxes with center distance 50 mm and ratios 28:1 and 7:1; these data will be used to calibrate the model for efficiency prediction.

The gearboxes were test-rigged with an input speed of 1,400 rpm and output torques listed in the catalogue for continuous operating mode. Each housing was monitored with an infrared camera (Figs. 4b and 4c) while a thermocouple inserted from the top acquired the inner oil temperature. Table 1 summarizes results for both clockwise and counter-clockwise rotations of the input shaft; it is clear how performances of the CCW are worse than that of the CW. Changing the rotation, the axial reaction force moves from one bearing to the other in the input shaft. Because the two bearings have different sizes, the power loss is different.

Although the total efficiency calculation is already in the software (as seen previously), the authors improved the possibility of better controlling each parameter of the efficiency calculation in the model, adding all power losses — according to many standards or references — chosen by the user:

- Bearing P_{VL}
- Churning P_{VZP}
- Meshing P_{V0}
- Seals P_{VD}

The equations used in the model are similar to those in ISO/TR 14179 (Refs. 10 – 11) for cylindrical and bevel gears:

$$T_{wall} = (\varepsilon_{thermal})(T_{oil}) \text{ is the definition of the } \varepsilon_{thermal} \text{ (not present in any standards)} \quad (1)$$

$$P_V = P_{VZP} + P_{V0} + P_{VLP} + P_{VD} \text{ total power losses} \quad (2)$$

$$P_V = k(A_{ca})(T_{wall} - T_\infty) \text{ total heat flow} \quad (3)$$

Unlike the thermal capacity calculation for cylindrical and bevel gearboxes, here the oil temperature has no influence on Equation 2 factors, so Equation 1 is not needed and the T_{wall} is easily calculated from Equation 3.

Nevertheless, the factor $\varepsilon_{thermal}$, experimentally measured, was included in the model: $\varepsilon_{thermal}$ medium values T_{wall}/T_{oil} is 0.94 (from Table 1).

The calculation model validation required two steps.

Step one: after entering the input speed and output torque of each experimental case, choose the right equations so the model calculates the same input power measured in the test rig. The better mix of standards is this:

- ***Bearing*** P_{VL}: formulas from SKF catalogue (Ref. 12) (where there are new formulas compared to old catalogue editions)
- ***Churning*** P_{VZP}: ISO/TR 14521 (Ref. 8); (the other choice was ISO 14179)
- ***Meshing*** P_{V0}: ISO/TR 14521 (the same as DIN 3996:2012 (Ref. 7))
- ***Seals*** P_{VD}: ISO/TR 14179-2 (Ref.11) based on a German proposal

Note that the analytical model does not show a true efficiency variation when changing the rotation direction, as shown before in the test rig.

Step two: set the value of the global thermal coefficient k to obtain (in the model) the same housing temperature measured by the infrared camera.

$k = 31\ W/(m^2 °C)$: similar to vales in Table 7 of ISO/TR 14179-1 (Ref. 10) — based on an American proposal.

The authors already calculated, using ISO/TR 14521, the same (within an acceptable range of ±15%) oil temperature measured by the thermocouple (Table 1), without any other calculation. Therefore Step 1 and Step 2 are independent, and T_{wall} is calculated from this value of T_{oil} from Equation 1.

So the model, validated for ratios 7:1 and 28:1, is quite ready to calculate efficiency (power losses) for all other ratios or load cases (torque and speed), and to

Table 2 Alternate operating modes and DOE parameters *A* and *B*, ΔT_{wall} measured values, $\Delta T_{wall}M$ medium between values with the same *A*, $\Delta T_{wall}C$ calculated from Equation 4

Operating Mode	Start Frequency *A* [min⁻¹]	Operating percentage *B* [%]	$\Delta T_{wall} + \sigma$ [°C]	$\Delta T_{wall}M + \sigma M$ [°C]	$\Delta T_{wall}C$ Equation 4 [°C]
G	30	25	29.6 ± 0.8		
H	60	25	49.9 ± 0.5	**42.4 ± 0.6**	**40.75**
I	180	25	47.75 ± 0.54		
C	30	50	17.5 ± 0.9		
A	60	50	20.4 ± 0.6	**19.5 ± 0.7**	**22.75**
B	180	50	20.69 ± 0.57		
E	30	75	5.4 ± 1.2		
D	60	75	7.4 ± 0.9	**7 ± 1**	**4.75**
F	180	75	6.6 ± 1		

calculate the housing temperature in continuous operating mode — as required. Perhaps, however, it should be modified a bit to allow for different gearbox sizes.

But note that the model does not yet enable calculation of efficiency and housing temperature for alternate operating mode of worm gearboxes. In order to define an empirical law to estimate such performances from the theoretical evaluation, further tests were conducted on a worm gear speed reducer with center distance 50 mm and ratio 7:1, with start frequencies "A" and operating percentages "B" summarized in Table 2, and with clockwise input shaft rotation. A and B values were chosen by a DOE 3^2 approach, which, with an optimized number of tests, allows one to define the variables with statistical effectiveness on temperature, together with an eventual, combined effect.

Figure 4 shows a comparison of temperature trends for continuous and alternate operating mode; in both ways housing and oil temperatures stabilize after almost 3 hours, but for the alternate one, lower values are reached. All the alternate modes show the same behavior. Table 2 shows ΔT_{wall}, the housing temperature difference, between alternate and continuous operating mode. From the statistical analysis of the DOE results pattern, only B effects seem to be significant (P value lower than 1), so it's possible to record linear Equation 4 for ΔT_{wall} housing:

$$\Delta T_{wall} = 58.75 - 0.72B \tag{4}$$

Figure 5 compares measured values (from Table 2 with the same A) with calculated values of ΔT_{wall} from Equation 4. The experimental trend shows a good correspondence with the theoretical one, also considering the relative σ, and confirms the strength of DOE approach. Regardless, this is just a first step in dealing with an empirical law, and further tests must be done to make it more accurate and applicable to other sizes of worm gearboxes.

Equation 4 was not added to the model because it required more validations, but it is starting to help R&D and sales departments to better answer customer requests, giving them indications that are more precise about the real temperature they should expect on worm gearboxes.

Besides having a strong computational tool to predict and define gearbox behavior in terms of efficiency and temperature, the R&D department frequently needs to answer customer requests for improved efficiency for the worm gearboxes. In order to fulfill this target, while also avoiding strong geometrical modifications, the authors studied the effects of adding on fullerene-shaped WS_2 nanoparticles to standard PEG lubricant.

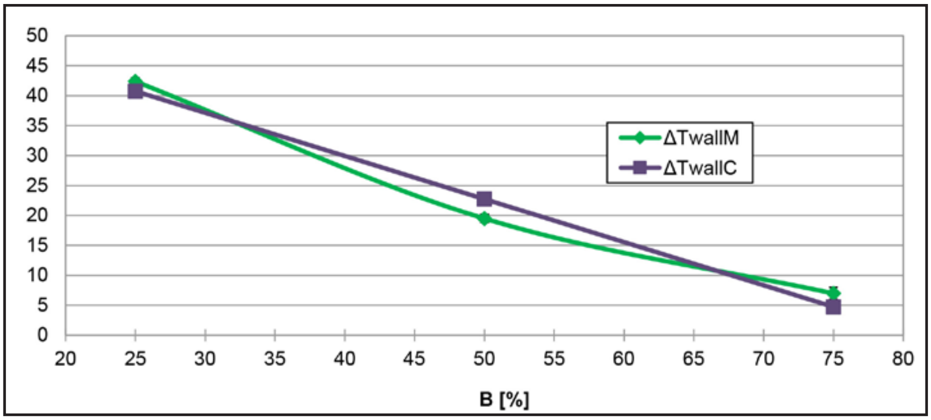

Figure 5 Difference of temperature between housing (wall) and oil for worm gearbox with center distance 50 mm and ratio 7:1, rotation CW — comparison between measured ($\Delta T_{wall}M$) and calculated ($\Delta T_{wall}C$) values.

Table 3 Rheological properties of the three lubricants tested

Rheological Properties	PEG	ESTER	PEG + WS_2
Viscosity 40°C [mm^2/s]	320	295	320
Viscosity Index	230	174	-

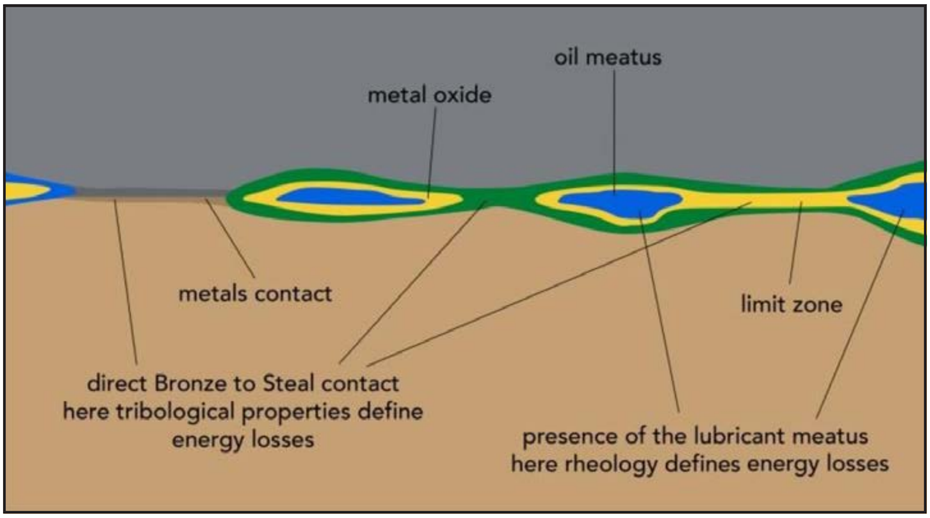

Figure 6 EHL mixed lubrication, influence of tribology between contact surfaces and rheology of lubricant.

Commercial PEG rheological properties are listed in Table 3; the additive is made of WS_2 nanoparticles of fullerene shape (diameter 13 Nm) added on the lubricant for a 4% WT. These particles are only supposed to improve tribological properties of the lubricant by flaking and going to attach to the sliding counterpart, reducing wear and facilitating the reciprocal motion.

WS_2 fullerene nanoparticles should not influence the rheology of the PEG lubricant — which is quite essential in order to guarantee constant performance of the worm gearboxes — while the oil temperature increases during operations. Such a high value of the PEG lubricant viscosity index, 230, is the expression of this aspect and guarantees that the viscosity does not drop too much with the temperature, constantly providing the right combination of controlled power losses due to shear stress and hydrodynamic lift between the teeth coupling.

Both tribology and rheology aspects are involved in the contact area on an EHL mixed regime, such as the one associated with the high Hertzian pressure experienced by worm gearboxes (Fig. 6). Friction

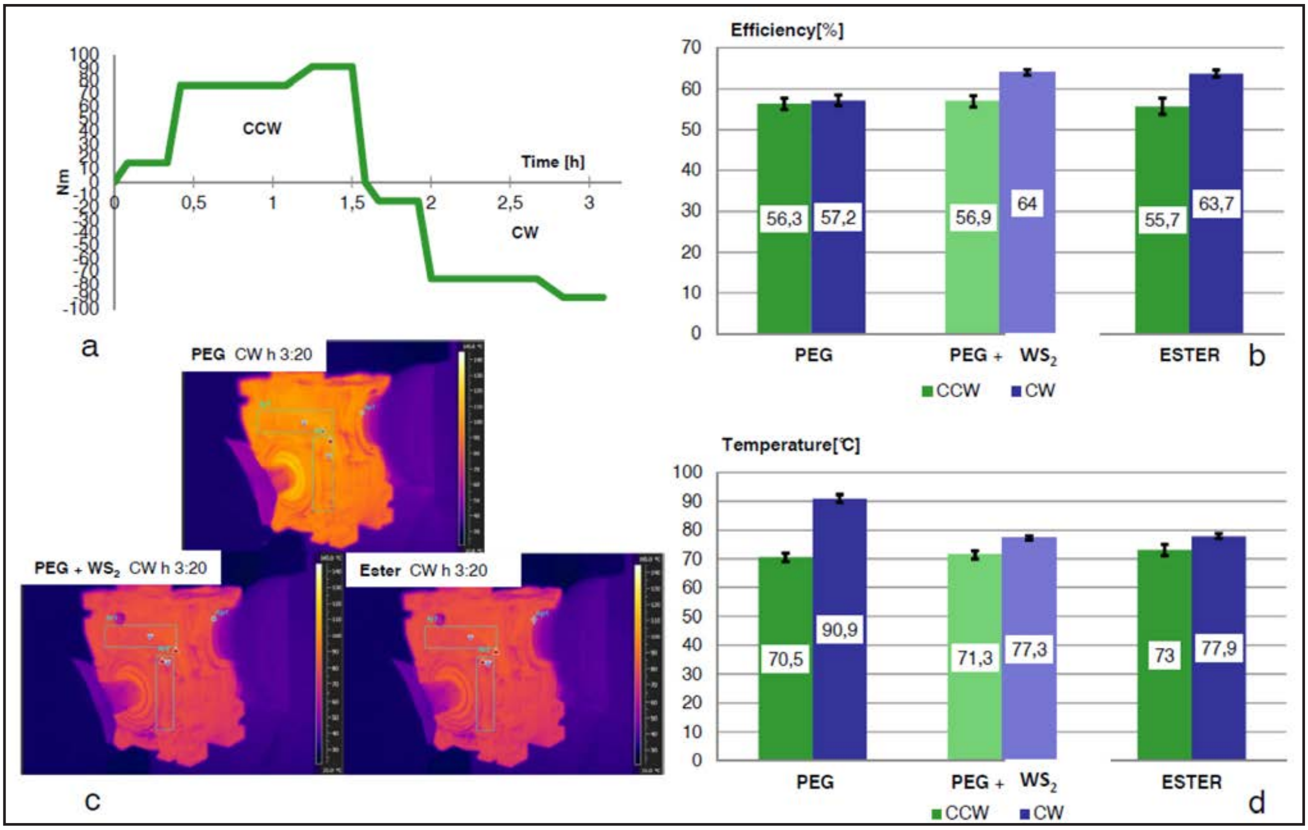

Figure 7 Efficiency and temperature tests on worm gearbox with center distance 50 mm and ratio 1:49 with PEG, PEG + WS_2, and ester lubricants — a) output torque trend during test; b) efficiency results; c) infrared picture of the gearboxes at the end of the efficiency tests; and d) temperature results, meaning maximum *T* value acquired.

and wear are related to the limit zone, where the metallic counterparts come in direct contact. Here the fullerene WS_2 nanoparticles flake and facilitate the reciprocal sliding, attaching themselves to the counterparts. Instead, rheology is related to the contact areas where the meatus is still present; here the energy losses are related to the shear stress of the lubricant, which needs to be won while sliding.

Efficiency tests were run on a worm gearbox with center distance 50 mm and ratio 49:1, after a proper running-in, with 1,400 rpm as input speed and an output torque following the trend in Figure 7a.

Rotation of the input shaft switched from counterclockwise to clockwise without cooling; the temperature of the housing was collected with an infrared camera during all tests (Fig. 7c).

Figures 7b and 7d show the comparison of efficiencies and temperatures of RT50 PEG lubricated with and without WS_2 nanoparticles. The filler brings an increase of almost 12% between the counterclockwise efficiency and the clockwise one, which is not present with the only PEG. This improved efficiency is also clear in the temperature confrontation; for the WS_2 nanoparticles the temperature reached during the clockwise phase is comparable to the counterclockwise one, while for the only PEG it's almost 20° C higher. A similar behavior was observed for the same gearbox, while using an ester-based lubricant (Table 3) with a viscosity index lower than the PEG one — i.e., 174.

This third lubricant showed a positive effect on gearbox efficiency, but even the small increase in temperature it goes through means a decrease of the cinematic viscosity to 55 mm^2/s from the 320 mm^2/s at 25° C. As explained above, a decrease in viscosity is good to limit energy losses in the EHL mixed regime contact, but it should not run too low, in order to avoid seizing of the gearbox system. To exclude any unexpected influence of the WS_2 nanoparticles on the rheology of PEG, further tribological and rheological testing on the three lubricants was conducted.

Figure 8a shows the effect of oil temperature on the dynamic viscosity of the PEG — with and without fullerene particles. Measurements were run by a rotational viscometer according to the ISO 2555 (Ref. 13) standard. The measured parameter is the dynamic viscosity, which is related to the cinematic one by the fluid density.

Values of dynamic viscosity are comparable, as well as temperature trends; this first result excludes the effects of WS_2 on rheology.

The effect of fullerene nanoparticles was better understood by performing wear tests for the three oils using a block-on-ring (BOD) tribometer (Fig. 14a). In the BOD configuration a sample block rests on a rotating 100Cr6 ring (HRC 62) half-plunged in a small tank of sample lubricant. While rotating, the ring picks up the oil that goes to lubricate the contact area, with the same operating principle of lubrication in worm gearboxes. The tested contact geometry is comparable to the one of worm-wheel teeth; in this way one can observe wear resistance of the bronze counterpart in a much shorter time than on the driver itself. Bronze blocks (HRB 130) are obtained directly from a worm wheel trough-wire EDM.

The ISO/TR 14521 (Ref. 8) wear intensity J_{OT} was used to evaluate the amount of wear on the bronze block. In Equation

5 the wear depth P of each sample was measured by a 3-D profilometer (Fig. 9). In Equation 6 the wear path length L is calculated from BOD test parameters (Fig. 14b). Values of normal loads on the bronze block correspond to Hertzian pressure equivalent to the values experienced by the teeth at the output torques that were used at the test rig during the efficiency tests.

$$J_{OT} = \frac{P}{L}\left[\frac{mmWear}{mmPath}\right] \tag{5}$$

$$L = (t)(\omega)(R_{ring})\,[mm] \tag{6}$$

In Figure 8b the J_{OT} values measured for each lubricant are compared. PEG plus WS_2 showed the lowest value of J_{OT} — proving the effects on bronze wear as described above. Ester lubricant had the worst performance in terms of wear — probably because in BOD tests the viscosity decreases while the temperature rises; thus the bronze wears out against the harder 100Cr6 roughness.

This result explains the efficiency performances (Fig. 7) in terms of two different causes for the same effect. Ester lubricant improves efficiency, reducing energy shear losses in the meatus zone of the EHL contact area due to its rheological properties. PEG plus WS_2 fullerene nanoparticles act upon the metal contact zone, thereby reducing losses due to bronze wear and friction. In other words, the additive has effect only on the tribology of the system, as expected. This one is definitely the best solution for improved efficiency requests of final customers, guaranteeing constant temperature performances in terms of hydrodynamic lift, while introducing a solution focused for the metal contact area. It is also a cheaper solution because total cost of the gearbox increases by just 2%.

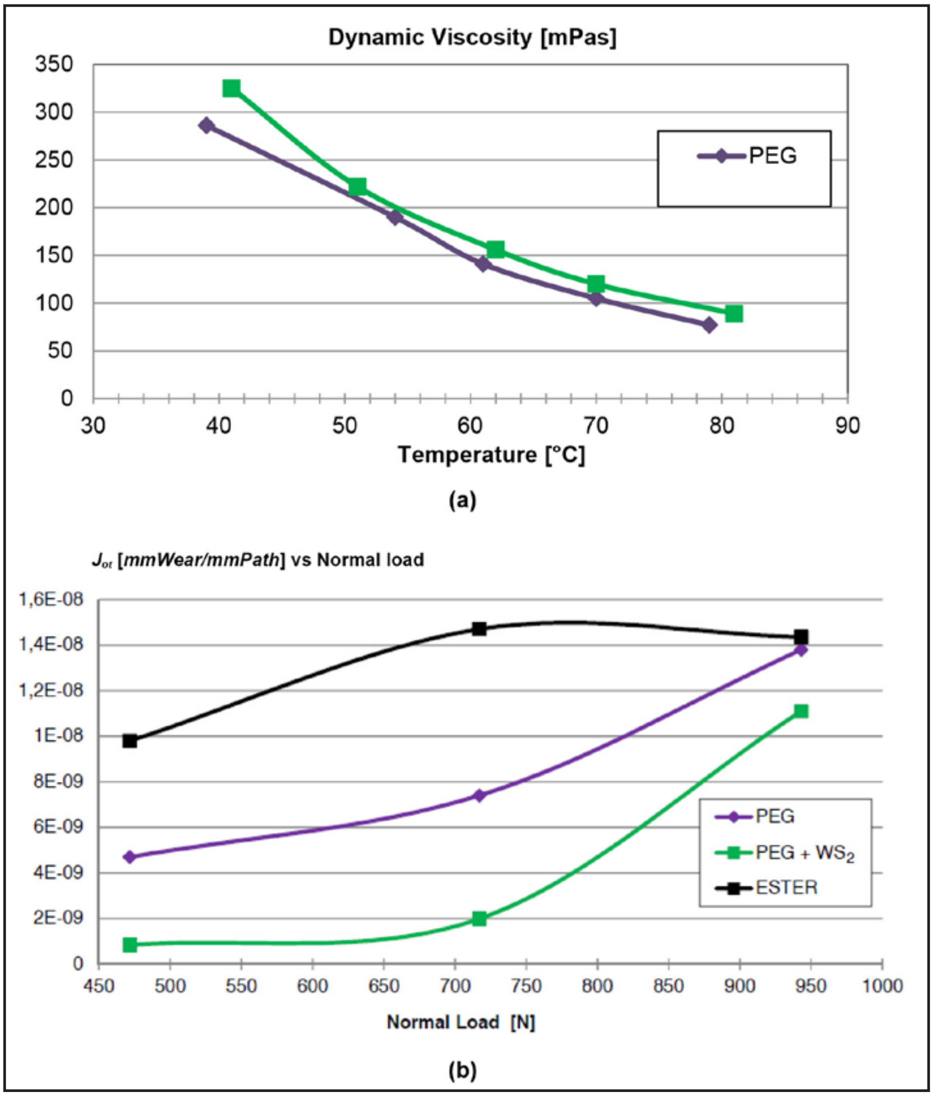

Figure 8 **(a) Dynamic viscosity vs. lubricant temperature for PEG with and without fullerene WS_2 nanoparticles, ISO 2555; (b) wear coefficient J_{OT} vs. normal load after BOD tests on bronze with three different lubricants.**

Conclusions

Although wormgear speed reducers are less efficient than other gearboxes with equal ratio and size (e.g., orthogonal ones), they are still very popular — especially for occasional or intermittent service per day — because they are simple and inexpensive. The improvement of their efficiency over the years, along with the continuous evolution of the standards, bears this out.

The low efficiency of these gearboxes can even be an advantage because it avoids the introduction of the brake when irreversibility is required. Because of this, it is important to provide a reliable value of the efficiency. If specifically requested, it can still be increased without any geometric changes, but only while acting on the lubricant additive composition.

Acknowledgements. The authors would like to thank KISSsoft AG for the support and the development; Varvel SpA for the tests; the AGMA WG06a committee for the ideas; Tec Star Srl for Fullerene nanoparticles; Dr. Alessio Passalacqua for DOE; and the reviewers for their comments that helped to improve the paper.

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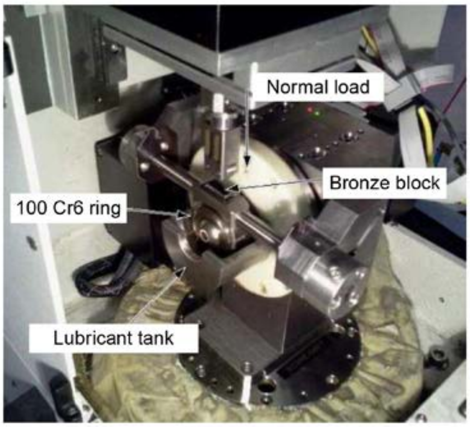
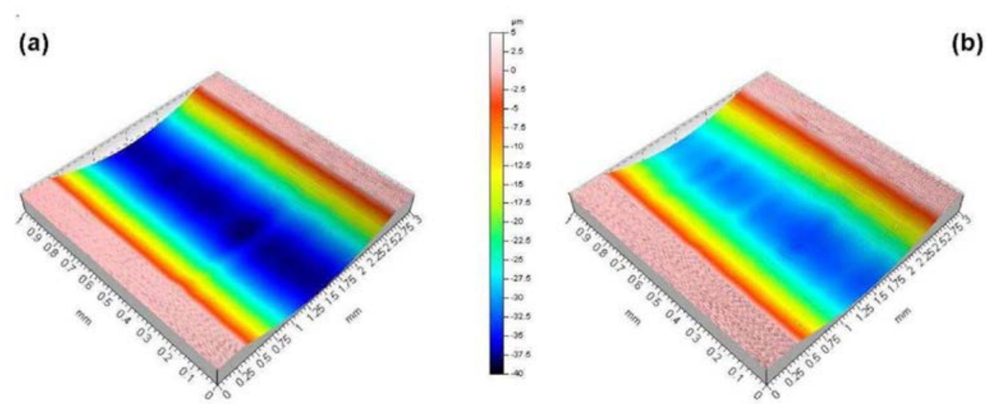


A

BOD parameter	Values
Normal load	471.5 N 717 N 943 N
Ring Speed ω	867 rpm
Period T	30 minutes
Ring radius Ruing	17.49 mm
Ambient temperature	25°C

B



C

Figure 9 Tribological BOD tests on bronze block with PEG, PEG+ WS_2, and ester lubricants. (a) BOD tribometer configuration; (b) test parameters; (c) 3-D maps of wear tracks on bronze block — *a* with ester, *b* with PEG + WS_2 lubricant.

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Fabiola Bisanti has a master's degree in industrial engineering — specializing in production and logistics. In 2011 she worked at Caterpillar Global Paving in collaboration to implement a health & safety management system — in accordance with OHSAS 18001 — to align standards work production with safety law. In 2012 she completed a post-master's degree in vibro-acoustic engineering and began work with Bologna, Italy-based Varvel SpA as technical collaborator for an R&D project focused on development of a diagnostics method for detecting gear and gearbox defects via vibration analysis and test bench. Since 2013 Bisanti has worked in Varvel's R&D department, her principal responsibility being the diagnostic and quality control of Varvel's products via vibration and acoustic signals analysis.

Elena Ferramola has a master's degree in physics, specializing in condensed matter. Her collaboration with Bologna, Italy-based Varvel SpA began in 2009 with a 2-year R&D project intended to define alternative solutions to traditional lubricants. In so doing she developed an expertise in polymers for metal replacement, and tribological and mechanical solutions for gear materials issues. Ferramola's collaboration continues today in research and development, where she focuses on with new products feasibility studies and continues her research for innovative tribological solutions.

Giampaolo Giacomozzi has since 2012 held the position of Research & Development manager at Varvel SpA in Bologna, Italy. A 1995 graduate in mechanical engineering at the University of Bologna, for the first five years he worked in the packaging industry as a mechanical designer in the technical department at CAM SpA. He next (2001) worked as a Quality Assurance manager and head of the Prevention and Protection service for Bottazzi SpA, before moving on to Varvel SpA in 2002 where — until 2005 — he was Quality and Environment manager and head of the Prevention and Protection Service. In 2005 Giacomozzi was named assistant technical manager — and from 2007–2012 — technical manager.

Massimiliano Turci is a consultant in gear technology and the design of cam mechanisms. Upon earning his master's degree in mechanical engineering at Studiorum University of Bologna, Italy he began his career as a CAD expert. He then developed X-Camme — a CAM design software used in the packaging and beverage machinery industry. He is a member of the Italian KISSsoft staff for training and engineering — especially for industrial gearboxes. His professional experience is now primarily in gearbox model calculations. Turci is a member of the AGMA worm gear committee.