Predicted Scuffing Risk to Spur and Helical Gears in Commercial Vehicle Transmissions

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AGMA925–A03 scuffing risk predictions for a series of spur and helical gear sets of transmissions used in commercial vehicles ranging from SAE Class 3 through Class 8.

Introduction

The risk of gear tooth scuffing in commercial vehicle transmissions has gained more attention because of increasing demand for fuel-efficient powertrain systems in which diesel engines run at lower speeds, power density is higher, and lubricants are modified to improve efficiency and compatibility with components of new technologies, such as dual-clutch transmissions.

Accordingly, predicting scuffing risk during the design phase is vital for the successful development of commercial vehicle transmissions. AGMA 925–A03 (Ref. 1) is a comprehensive method for predicting the probability of gear scuffing. This paper presents the AGMA 925–A03 (Ref. 1) scuffing risk predictions for 50 spur and helical gear sets in transmissions used in commercial vehicles ranging from SAE Class 3 through Class 8. Limiting scuffing temperatures using two mineral and three synthetic gear lubricants was determined from FZG scuffing tests. The risk of scuffing was determined for each gear set according to AGMA 925–03 (Ref. 1). The predictions were compared with field and warranty data, and dynamometer test results. The predictor was correct in all cases. High scuffing risk was predicted for gears known to scuff, and low scuffing risk was predicted for all other cases with no history of scuffing. The document correlation between prediction, test results and actual usage instills confidence in the predictor of scuffing risk for gears in commercial vehicle transmissions.

Background

Scuffing failure causes localized damage to gear teeth, resulting in matte and rough finishes of the contacting surfaces, and tooth form changes as well. This type of damage generally occurs in the tooth contact zone where contact pressure and sliding velocity are high, far from the pitch line. The tooth damage can increase vibration and noise, compromise gear load capacity, ultimately leading to catastrophic failures (Ref. 2). As a severe adhesive wear phenomenon, scuffing occurs when the oil film thickness between the tooth contacting surfaces is insufficient to prevent metal-to-metal contact, which in turn causes local welding and subsequent tearing. Metal particles are transferred between the two surfaces or lost from them; they scratch tooth flanks in the sliding direction. Scuffing is not a fatigue phenomenon, as it may occur at the beginning of the operation (Refs. 3–4).

There are several analytical methods for predicting scuffing risk; however, the threshold for determining whether a gear set will scuff remains mostly dependent on empirical results.

The method of evaluating scuffing risk in AGMA 925–A03 (Ref. 1) is a function of oil viscosity and additives; operating bulk temperature of the gear; sliding velocity; surface roughness of gear teeth; gear materials and heat treatments; and surface pressure. The risk of scuffing is defined by comparing the calculated tooth contact temperature with the limiting scuffing temperature; this is done via gear scuffing tests for each gear lubricant.

Contact Temperature Calculation

Flash temperature across the tooth contact path is calculated by Blok's Equation 1.

$$\theta_{fl_{(i)}} = 3162\, K\, \mu_{m(i)} \frac{X_{\Gamma_{(i)}}\, w_n}{\left(b_{H_{(i)}}\right)^{0.5}} \frac{v_{r1_{(i)}} - v_{r2_{(i)}}}{B_{M1}\left(v_{r1_{(i)}}\right)^{0.5} B_{M2}\left(v_{r2_{(i)}}\right)^{0.5}} \tag{1}$$

where

$\theta_{fl_{(i)}}$ is flash temperature, °C;

K is 0.80, a numerical factor for the Hertzian distribution of frictional heat over the instantaneous contact band width;

$\mu_{m(i)}$ is mean coefficient of friction;

$X_{\Gamma_{(i)}}$ is load sharing factor;

w_n is normal unit load, N/mm;

$v_{r1_{(i)}}$, $v_{r2_{(i)}}$ are rolling tangential velocities in m/s of the pinion and gear respectively, m/s

B_{M1}, B_{M2} are 13.6 $N/(\mathrm{mm}^2 K)$, thermal contact coefficient of steel;

$b_{H_{(i)}}$ is the semi-width of the Hertzian contact band, mm;

i is a subscript of line-of-action points

Contact temperature at each line-of-action point is given by Equation 2.

$$\theta_{B_{(i)}} = \theta_M + \theta_{fl_{(i)}} \tag{2}$$

where

$\theta_{B_{(i)}}$ is contact temperature, °C;

θ_M is the tooth temperature, in °C, which is the temperature of the tooth surface before it enters the contact zone. The tooth temperature can be estimated by calculation, testing or experience.

In terms of calculation of the tooth temperature, it can be estimated by Equation 3.

$$\theta_M = k_{sump}\,\theta_{oil} + 0.56\,\theta_{fl\,max} \qquad (3)$$

where

k_{sump} is 1 for splash lube and 1.2 for spray lube;

θ_{oil} is oil supply or sump temperature in °C;

$\theta_{fl\,max}$ is the maximum flash temperature found over all line–of–contact points *i* (Eq. 1).

The maximum contact temperature is obtained by Equation 4.

$$\theta_{B\,max} = \theta_M + \theta_{fl\,max} \qquad (4)$$

where

$\theta_{B\,max}$ is maximum contact temperature, °C.

When the maximum contact temperature is closer to or higher than the limiting scuffing temperature, scuffing may occur.

Limiting Scuffing Temperature

The limiting scuffing temperature is the tooth contact temperature at which scuffing is likely to occur, given a combination of lubricant and gear material (Ref. 1). It can be obtained from gear scuffing tests, such as FZG tests—an industry standard test used worldwide to rate different lubricants for scuffing resistance. The FZG test method per CEC L–84–02 (Ref. 6) was developed by the Coordinating European Council (CEC) and is extensively used by the automotive and petroleum industries in Europe and throughout the world.

The FZG gear test rig is operated at constant speed for a fixed period of time at successively increasing loads until the failure criteria is reached. The test gears are then examined for scuffing damage to the gear tooth flanks both before and after the prescribed duration at each load stage. The test gear fails when the sum of the scuffed area widths on all teeth exceeds the gear face width (Ref. 3).

In 2004 the Eaton Corporation Vehicle Group developed an extensive project to investigate industry-standard scuffing test methods to and correlate their results with different gear lubricants in transmission tests in dynamometers. The investigation was motivated by longer oil drain intervals and new lubricants being used in Eaton transmissions. Under these conditions the former ASTM D5182–9 (Ref. 3) method did not correlate with known scuffing occurrences.

The FZG method defined by CEC L–84–02 (Ref. 6) and designated as A10/16.6R/120 yielded consistent results and true agreement with dynamometer tests. Eaton Corp. has since adopted this test method as part of its internal procedure for gear lubricant qualification and approval.

The A10/16.6R120 stands for A-type gear geometry, 16.6m/s pitch line velocity, and a sump temperature of 120°C. The A-type gear is designed with longer addendum geometry to generate high sliding velocity, and is manufactured with only a 10 mm face width to increase the contact stress so that the scuffing could be more easily produced than with the 20 mm face width. The 10 mm face width gear test is commonly referred to as the "half-tooth, double-speed test." Figure 1 shows a FZG A-type gear after a test.

Scuffing Risk Prediction

When the maximum contact temperature is close to or above the limiting scuffing temperature for the combination of lubricant and gear material, scuffing is likely to occur. The probability of scuffing can be predicted by comparing the contact temperature with the limiting scuffing temperature. The AGMA 925–A03 probability of scuffing is obtained from a Gaussian distribution of scuffing temperature assuming a 0.15 coefficient of variation. The contact temperature is a random variable, the limiting scuffing temperature is the mean value of the random variable, and the standard deviation of the random variable is assumed to be 0.15 of the scuffing temperature. AGMA 925–A03 (Ref. 1) then proposes a criterion to assess the risk of scuffing based on its calculated probability (Table 1).

Application Examples

In this study, 50 gear sets were evaluated using the AGMA 925–A03 scuffing risk method (Ref. 1). Twenty-four spur gear sets and 26 helical gear sets, whose helix angles were between 12–33°, were evaluated. The gear sets are used in nine different models of commercial vehicle transmissions, with up to 18 forward speeds and transmission torque capacity between 420 Nm–2500 Nm. Their applications include light-duty pick-up trucks, delivery trucks, buses and vocational and heavy-haul trucks.

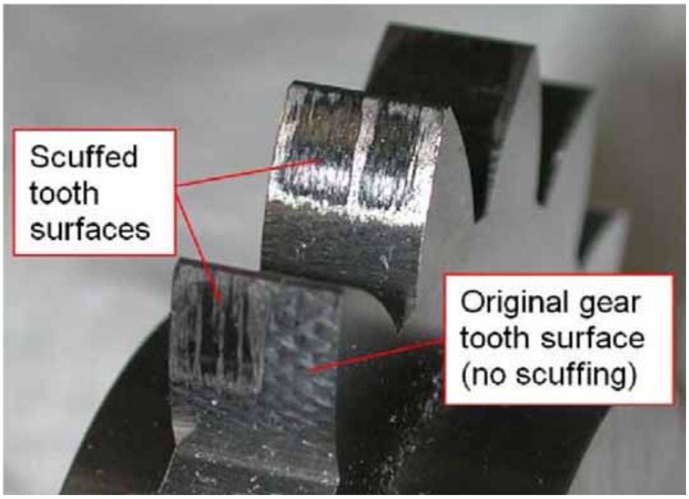

Figure 1 FZG A-type gear after test.

Table 1 Scuffing risk per AGMA 925–A03

Probability of scuffing	Scuffing risk
< 10%	Low
10% to 30%	Moderate
> 30%	High

Figures 2 and 3 show basic geometric and operating parameters of the gear sets, such as module, pressure angle, contact ratio, pitch line velocity and sliding velocity.

All gears were of case-hardened steel with a surface hardness of 58–63HRc. Among the gear sets, three tooth flank finish processes were used—1) shaving: surface roughness Ra = 1.25 mm; 2) honing: Ra = 0.40 mm; and 3) CBN grinding: Ra = 0.63 mm.

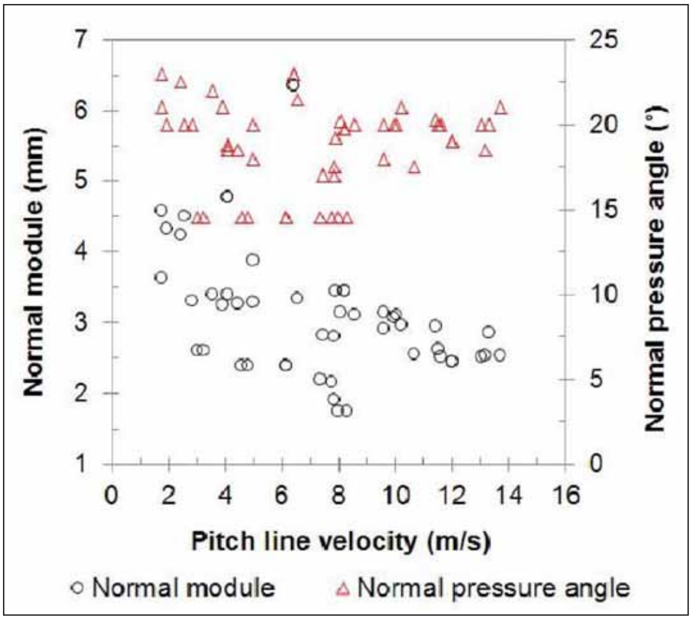

Figure 2 Characteristics of studied gear sets.

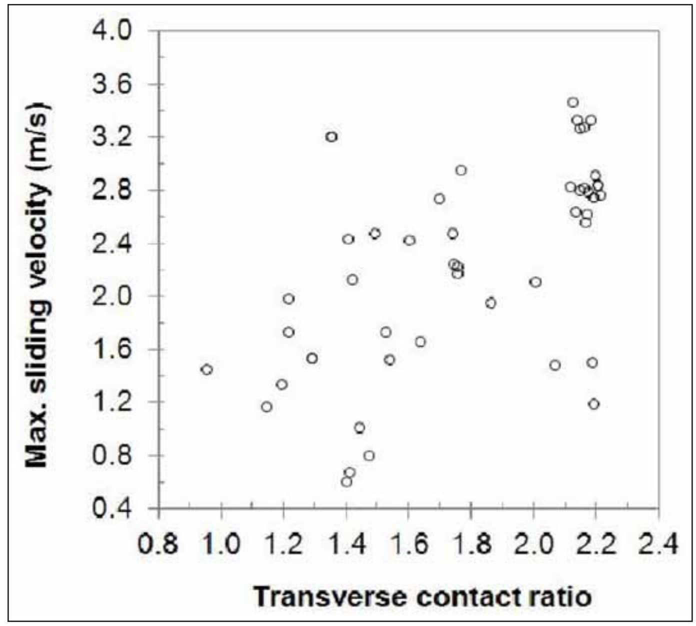

Figure 3 Sliding velocity and transverse contact ratio of gear sets.

The flash temperature was calculated for each gear set using Equation 1 for 25 line-of-action points, as recommended by AGMA 925–A03 (Ref. 1).

The coefficient-of-friction (COF) was calculated with the formulae in AGMA 925–A03 (Ref. 1) for an approximation of the COF by a constant. The COF was calculated with the average surface roughness of the pinion and gear teeth, and was assumed constant along the line-of-action. The maximum COF value was limited to 0.11. The effect of the COF on the maximum contact temperature predicted by the analytical model was investigated for one of the gear sets. The COF was arbitrarily changed from 0.03 to 0.11, and the contact temperature calculated.

The results showed that small variations on the COF may not significantly affect contact temperature results. For example, a 10% variation of a 0.10 COF caused the maximum contact temperature to vary only about 6% (Fig. 4). However, more pronounced COF changes—e.g., changing finish processes and obtaining much smoother surfaces—can significantly affect maximum contact temperature.

Another uncertainty when calculating the contact temperature of gear sets used in commercial vehicle transmissions was the load condition for the calculation; i.e.—torque and speed, or power. Vehicles powered with combustion engines run under a certain power regime, which is a function of the characteristic engine power curve. In typical engine power curves the point of maximum power does not match the point of maximum torque. Typically, the maximum torque is at a lower engine speed than the maximum power point. In order to find the critical load

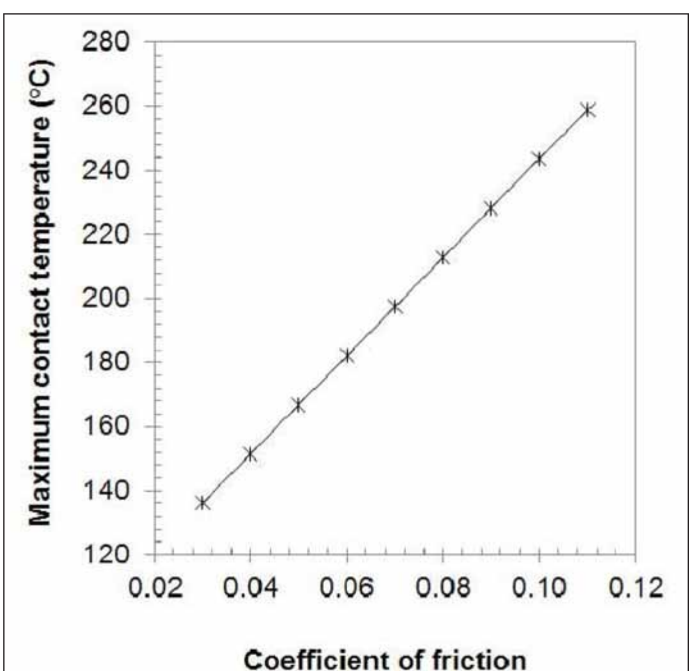

Figure 4 Effect of coefficient of friction on contact temperature results.

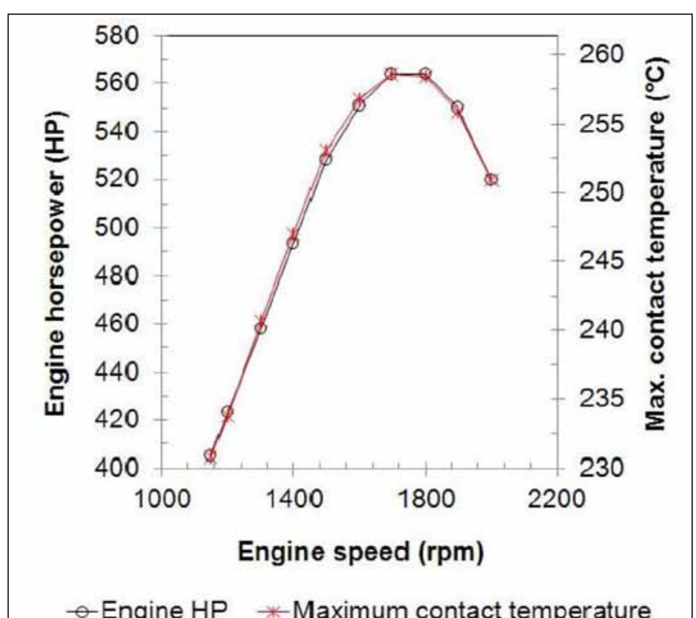

Figure 5 Effect of engine power on contact temperature results.

Table 2 Gear lubricants

Gear lubricants	1	2	3	4	5
Oil base	Synthetic	Mineral	Synthetic	Mineral	Synthetic
Viscosity index (ASTM D227)	151	194	146	164	150
Kinematic viscosity @ 40°C (mm2/s)	135.0	33.8	132.0	51.0	40.5
Kinematic viscosity @ 100°C (mm2/s)	18.2	7.6	17.5	9.1	7.4
Scuffing failure load stage of FZG test*	3	6	9	7	6

* FZG A10-16.6R120

condition for scuffing, the maximum contact temperature was calculated to 10 points along the engine power curve (Fig. 5).

The maximum contact temperature followed the engine power curve; i.e., as power increases, so, too, the contact temperature. The maximum value of contact temperature was obtained at maximum engine power. Based on the results, all calculations were made at maximum engine power rather than at peak torque.

Five different gear lubricants were used in this study—two mineral and three synthetic lubricants. Table 2 shows the basic characteristic of the gear lubricants and their load stage (the torque at that stage of the test) at which scuffing occurred.

The limiting scuffing temperature was then calculated for the FZGA-type gear (A10/16.6R120) at the scuffing failure load using the AGMA 925–A03 method, which was implemented into a spreadsheet.

With the scuffing temperature defined for each one of the five gear lubricants in Table 2, and the maximum contact temperature calculated for each of the 50 gear sets, the scuffing risk was determined by comparing contact temperature and scuffing temperature. Two approaches were taken for this: 1) the ratio of contact temperature and scuffing temperature was calculated and named "scuffing ratio"; ratios close or above one usually indicate that scuffing is likely to occur; and 2) the risk of scuffing was assessed as low, moderate or high, based on the probability of scuffing defined in AGMA 925–A03 (Ref. 1).

Among the 50 gear sets studied, four showed scuffing failures in both dynamometer tests and in the field (Fig. 6).

Figures 7 and 8 show the scuffing ratio and probability of scuffing, respectively, for all 50 gear sets. The four scuffed gear sets are identified in red and with an asterisk in front of the results in Figures 7 and 8.

Scuffing ratios closer to, and above, one, were predicted for all four gear sets in which scuffing failures were known (Fig. 7). However, looking at the scuffing ratio only can lead one to think that scuffing could occur on gear set Number 5, which has a scuffing ratio of around 0.9.

This pointed out the advantage of using AGMA 925–A03 probability of scuffing (Fig. 8). The results clearly show a high risk of scuffing—greater than 30%—to all four gear sets at which scuffing was observed.

Looking at gear set Number 5, its probability of scuffing is around 20%, which is classified as moderate risk in AGMA 925–A03 (Ref. 1). Thus, a clearer assessment can be done using the probability of scuffing recommended in the AGMA standard.

Figure 6 Scuffed gear set from dynamometer test.

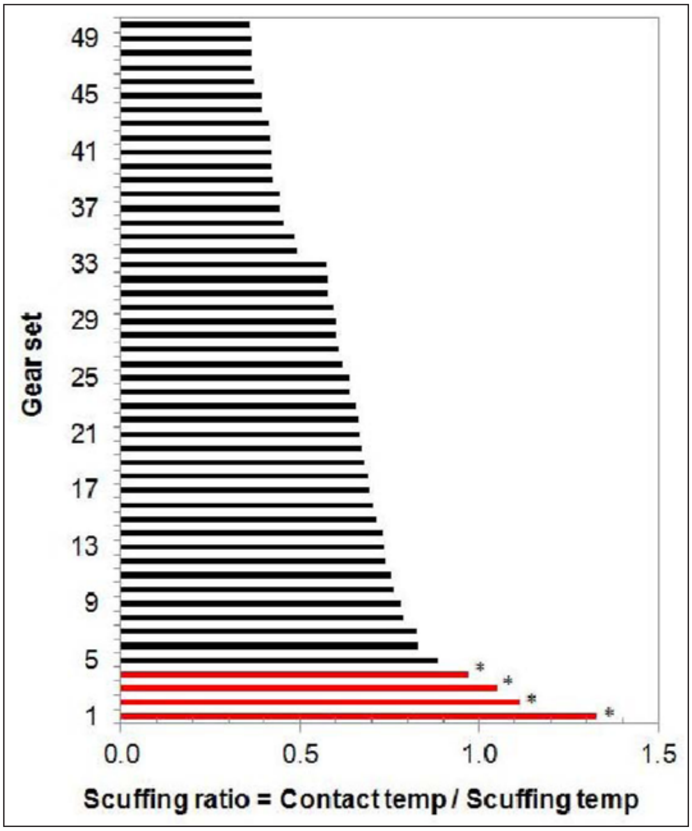

Figure 7 Scuffing ratio results (scuffed gear set).

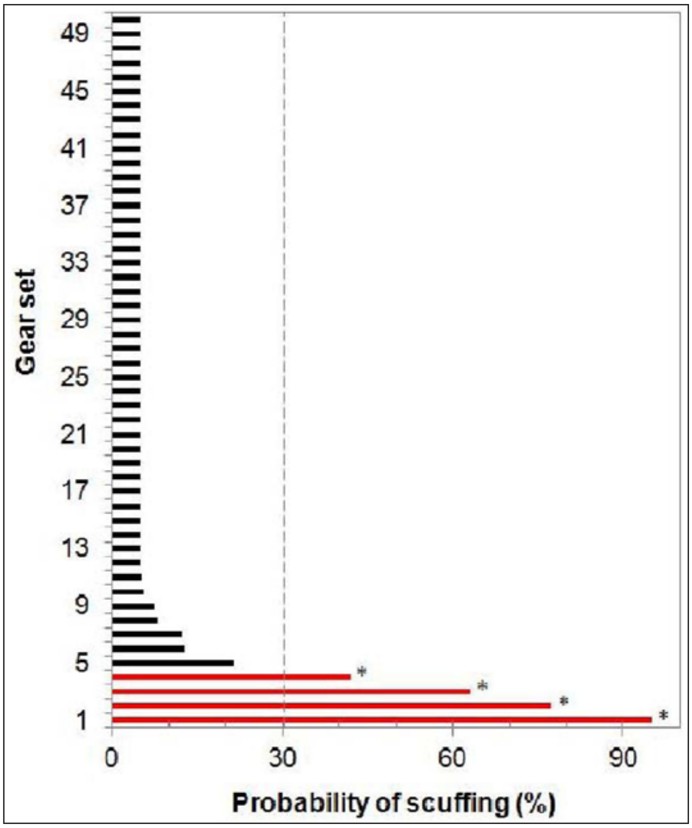

Figure 8 **Probability of scuffing: > 30% = high risk (*scuffed gear set).**

Discussions

The AGMA 920–A03 standard was used to predict the scuffing risk of 50 spur and helical gear sets in nine transmission models, which are applied to commercial vehicles ranging from SAE Class 3 through Class 8. Scuffing temperatures of two mineral and three synthetic gear lubricants were defined, based on FZG test results. Scuffing risks were assessed based on the probability of scuffing according to AGMA 925–A03. The results indicated that the AGMA 925–A03 is indeed effective and consistent in predicting scuffing risk, as evidenced by the good match to dynamometer and field results for the known scuffed gears, and also for the gears with no history of scuffing failures. The agreement between prediction, test results and actual usage provides confidence in the predictor of scuffing risk of gears in commercial vehicle transmissions.

Acknowledgments. The author thanks the Eaton Corporation Vehicle Group for support on the development of this paper.

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CORRECTION

A Technical Paper in the October *Gear Technology*—"**Pitting Resistance of Worm Gears: An Advanced Model for Contact Pattern of Any Size, Position or Flank Type,**" by Dr. Karsten Stahl, Dr. Bernd-Robert Höhn, Dr. Joerg Hermes and Alexander Monz, was incorrectly attributed. Attribution should have read:

This article appeared previously—and exclusively—(in German) in the April 2012 issue of Antriebstechnik.