# Case Study of ISO/TS 6336-22 Micropitting Calculation

Robin Olson, Mark Michaud, Jonathan Keller

## Introduction

Micropitting is a form of Hertzian fatigue damage that occurs on gear teeth. It appears as ultrafine cracks on the surface of the flank, with the resulting loss of material looking like grey staining. Although the cause of micropitting is not fully understood, it appears to be caused by cyclic stresses and plastic deformation on the asperity scale. In addition, sliding between gear teeth causes traction forces that subject asperities to shear stress. Micropitting is influenced by a number of factors including loads, speeds, temperatures, gear tooth macro- and micro-geometry, flank surface finish, heat treat, and lubricant properties.

Micropitting predominantly occurs on case-hardened gear teeth. Figure 1 illustrates the appearance of micropitting.

**Figure 1 Micropitting on a carburized gear (From ANSI/AGMA 1010-F14 [1]).**

Multiple papers have been written about micropitting, its description, and its causes (Refs. 1–2). Micropitting can lead to significant surface damage, macropitting, and catastrophic failure. Alternatively, it may appear in patches and arrest its growth as tribological conditions improve during run-in. If one is designing gearing for critical applications, it is desirable to be able to calculate the risk of micropitting in an effort to avoid it.

The presence or absence of micropitting is not easy to determine with an analytical model because micropitting occurs on the asperity level, The engineer needs to determine what percentage of the asperities will come into contact through the lubricant film thickness, the asperity plasticity, the number of cycles the asperities experience as they travel through the contact zone, the fatigue limit of the asperities, and the pressure applied to the asperities. In 3-dimensional calculations, this is dependent on loads, local tooth geometry, roughness along the direction of tooth motion, lubricant selection, and the metallurgy of the gear. As a result, there is no comprehensive model to predict micropitting risk.

ISO/TS 6336-22 (*Calculation of load capacity of spur and helical gears — Part 22: Calculation of micropitting load capacity*) is the ISO technical specification containing a proposal for a calculation of risk of micropitting in gear sets (Ref. 3). This document was originally published in 2010 as ISO/TR 14179-1 and added to the ISO 6336 suite of documents in 2018. It was developed based on testing and observations of many gear sets with normal modules between 3 millimeter (mm) and 11 mm, and pitch line velocities between 8 meters-per-second (m/s) and 60 m/s. The analytical calculation in ISO/TS 6336-22 focuses on film thickness as a determinant for when micropitting will occur. This paper uses ISO/TS 6336-22 to calculate the risk of micropitting for gear sets in three different operating conditions and compares that to field experience. The simplified computation in Method B is utilized in order to simulate how the typical gear engineer will use the method. For these examples, micropitting is not predicted to occur and this points out some limitations in the method.

## Overview of the ISO/TS 6336-22 Calculation

ISO/TS 6336-22 contains a calculation of the micropitting load capacity of external gear sets that is based on testing. It assumes that micropitting occurs when the minimum specific film thickness of a gear set in application falls below a permissible value for specific film thickness. The ratio of the minimum specific lubricant film thickness to the permissible specific lubricant film thickness is the safety factor against micropitting. "Specific film thickness" is also called "lambda ratio" in some industries and is expressed as the ratio of the film thickness to the arithmetic mean roughness.

In other sections of ISO 6336, safety factors are used to calculate the risk of macropitting and bending fatigue. Advice about the acceptable minimum

value of the factor can be found in a general rating calculation, and app 6336-22 does not contain advice for a minimum safety factor; instead, it provides the following guidance:

"In other sections of ISO 6336, safety factors are used to calculate the risk of macropitting and bending fatigue. Advice about the acceptable minimum value of the factor can be found in a general rating calculation, an application rating specification, or a user specification for equipment design. ISO/TS 6336-22 does not contain advice for a minimum safety factor. Instead, it provides this guidance:"

***Minimum specific film thickness.*** The calculations for minimum specific film thickness are performed at multiple contact points in the tooth mesh region, with the minimum selected as the lowest value in the results array. This allows for the prediction of both the risk of micropitting and the region on the tooth flank that will experience damage.

In the document, the minimum specific lubricant film thickness can be determined using two different methods. Method A allows the engineer to calculate the value with a gear computing program that models the complete contact area of the mesh. The results appear as a map of pressures and film thicknesses across the face of the pinion and gear flanks.

Method B starts with the assumption that the minimum specific film thickness will be on the tooth flank in the region of negative sliding. The lubricant film thickness is calculated with a modified Dowson/Higginson analysis along the line of action. It deviates from the norm, though, with the addition of a local sliding parameter. This parameter accounts for the influence of sliding on temperature, which affects film thickness. This changes the pressure-viscosity coefficient and dynamic viscosity, thus adjusting the film thickness in the regions of negative-specific sliding.

$$\lambda_{GF,Y} = h_Y / Ra \quad (1)$$

$$h_Y = 1600 \cdot \rho_{n,Y} \cdot G_M^{0.6} \cdot U_Y^{0.7} \cdot W_Y^{-0.13} \cdot S_{GF,Y}^{0.22} \quad (2)$$

Where

$\lambda_{GF,}$ is the local specific film thickness

$h_Y$ is the local lubricant film thickness

$Ra$ is the effective arithmetic mean roughness value (averaged between pinion and gear roughnesses), µm

Y indicates the local contact point along the line of action

$\rho_{n,Y}$ is the normal radius of relative curvature at point Y along the path of contact, mm

$G_M$ is the material parameter

$U_Y$ is the local velocity parameter

$W_Y$ is the local load parameter

$S_{GF,Y}$ is the local sliding parameter

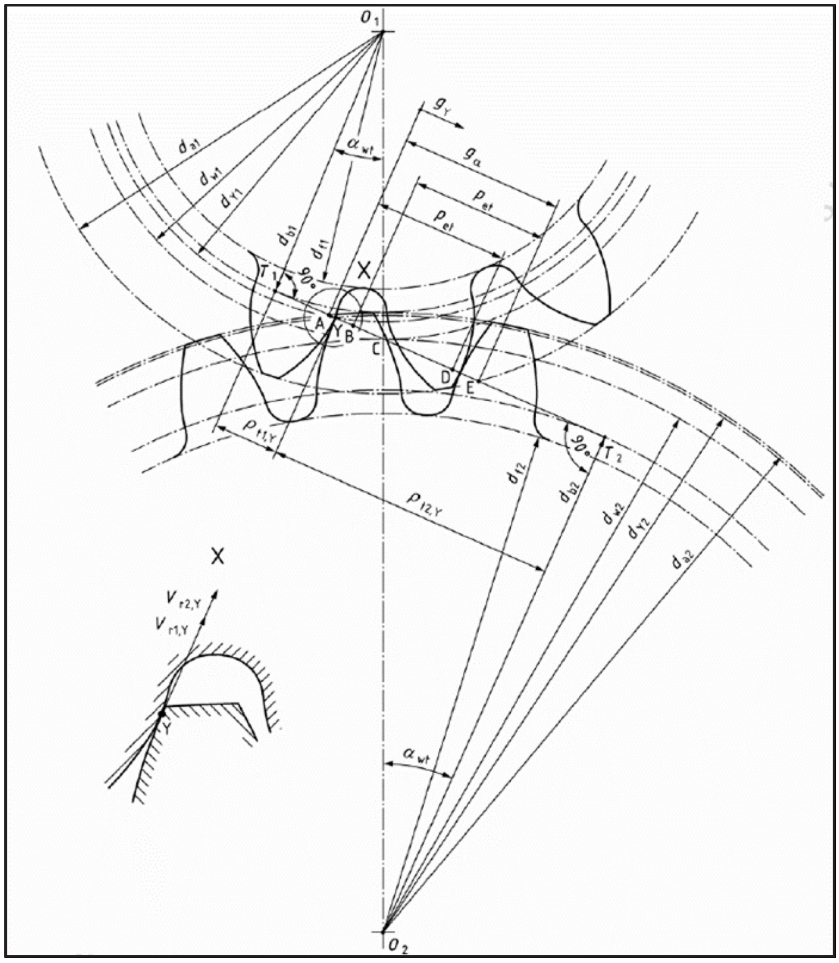


**Figure 2 Points along the line of contact as illustrated in ISO/TS 6336-22.**

The contact points along the line of action are determined with the familiar calculations for the lower point of active profile, lower point of single tooth contact, pitch point, upper point of single tooth contact, and upper point of active profile. Figure 2 shows this in a gear mesh. ISO/TS 6336-22 also considers mid- points between the lower and upper points of active profile and single tooth contacts.

***Permissible specific film thickness.*** The permissible specific film thickness can be determined using several different procedures. All procedures require some level of experimental investigations, whether they be conducted with the actual gear set or with representative gear sets. Ideally, testing is conducted with the actual gear sets, lubrication, and inlet temperatures that match the operating conditions of the gearing. Method A recommends that this testing be conducted until micropitting begins to occur. The permissible specific film thickness is then calculated per the Method A calculation for minimum- specific film thickness using the conditions of the final load stage.

Method B uses two different options to set the permissible specific film thickness. One option is to conduct studies with gearing that is similar in geometry, quality, and material of the gearing being designed. Using controlled tests, the gearing is run until the micropitting failure limit is reached. The critical specific film thickness for the test gearing is then calculated using the data from the failure stage. This is the permissible specific film thickness.

If comparative testing cannot be performed (due to cost, timeline, test stand availability, etc.), the permissible specific

film thickness can be generally determined from a simplified set of curves based on the lubricant's performance in FVA-FZG micropitting tests (Ref. 4) and its viscosity. These curves are derived from mineral oils and are shown (Fig. 3).

ISO/TS 6336-22 also contains alternative curves that can be used to determine a value of the permissible specific film thickness based on the results of FVA-FZG micropitting testing of mineral oils at temperatures of 60°C, 90°C, and 120°C. These curves account for the additives in the lubricant by accounting for its "quality." High-quality lubricants are specifically formulated with base stocks, additives, and thickeners to prevent micropitting. These are used when the costs of failure are high and maintenance is challenging. Mid-quality lubricants have some micropitting-preventing additives and are used in industrial gear lubrication when reliability is important and maintenance is scheduled. Low-quality lubricants have not been adjusted to prevent micropitting and are used in basic applications. In application, the choice of lubricant can be recommended by the gear manufacturer or specified by the equipment owner. It can result in micropitting if the lubricant is not formulated to minimize the phenomenon.

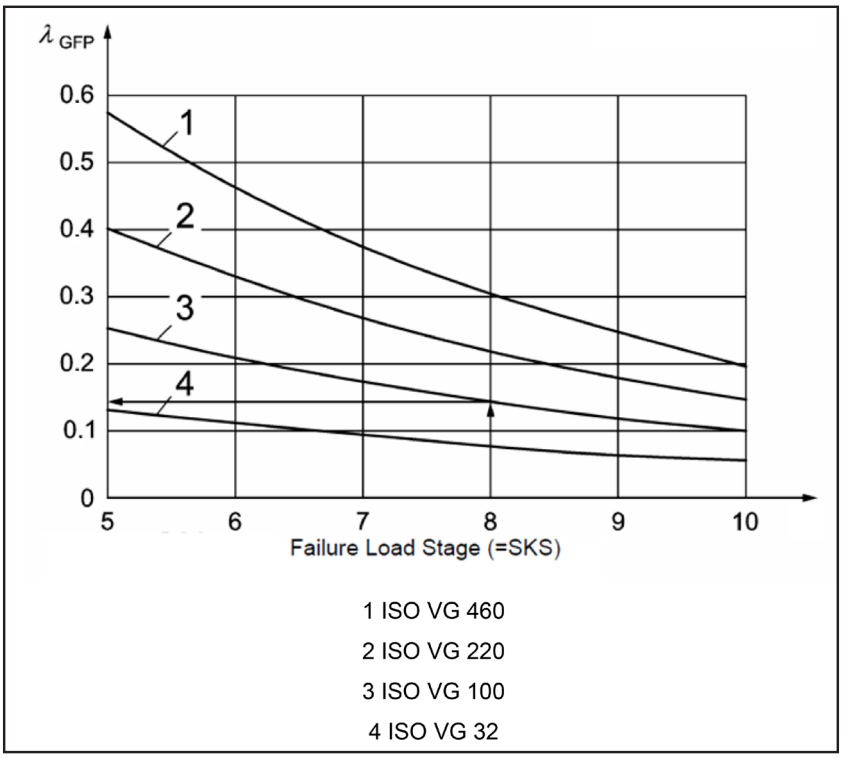


**Figure 3 ISO/TS 6336-22 Figure A.1 - Minimum permissible specific film thickness for mineral oils as a function of nominal lubricant viscosity and failure load stage in FVA-FZG micropitting tests.**

Clearly, testing is a prominent theme in the determination of the permissible-specific film thickness! ISO/TS 6336-22 is careful to point out that testing should be carefully conducted and well-documented; testing variability is a risk to the precision and/or accuracy of the results. Practically speaking, three to five tests are conducted with comparable load stage results and either the average or (more conservatively) the minimum value is used for the calculation.

In summary, the uncertainty in the permissible specific film thickness value increases as one moves further from Method A. Figure 4 illustrates the options for methods in ISO/TS 6336-22.

## Case Study – Using the Calculation

Case studies using ISO/TS 6336-22 or its predecessor document, ISO/TR 15144-1, have been performed in previous papers (Refs. 5–6). Many of those focus on using the Method A calculations for calculating tooth pressure and the specific film thickness.

Of particular interest is a paper by Pinnekamp and Heider (Ref. 7) that contains ISO/TR 15144-1 calculations with practical examples from industry. The FVA-FZG software *RIKOR* was used to determine the specific film thickness per Method A. Method B was used for the permissible specific film thickness. The resulting safety factors ranged between 1.0 and just over 3.0. Micropitting was observed on examples with safety factors over 2.0. The authors created a zoned diagram to predict the risk of

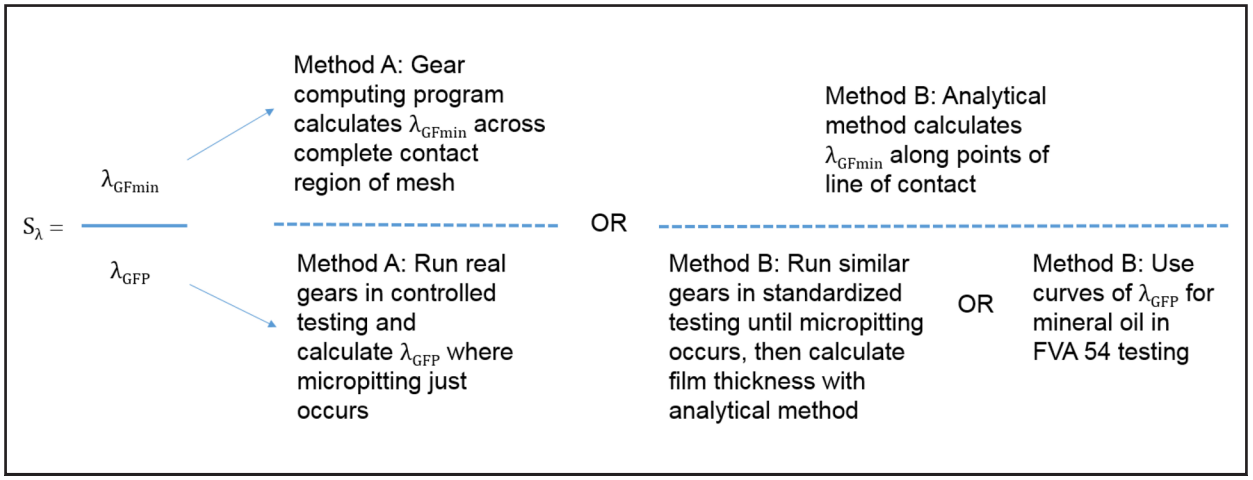


**Figure 4 Options for calculation of the micropitting safety factor.**

micropitting based on quality of the calculation, knowledge of operating conditions, and calculated safety factor.

This paper tests the behavior of the calculation using the analytical calculations in Method B to calculate specific film thicknesses. In two of the cases, it was not practical to calculate the permissible specific film thickness with comparative testing. The simplified curves based on viscosity and failure load stage of the lubricant were used for these. As much as possible, the paper simulates the path that the typical gear engineer would take to evaluate an existing gear set that experienced micropitting in operation or to assess a new design for the risk of micropitting.

The input data for each calculation consists of the gear geometry and arrangement, the lubrication, and the gear loads. The results are presented as the minimum specific film thickness, the permissible specific film thickness, and the safety factor that is calculated. Pictures of micropitting damage are also included.

***Case 1 – high-speed gear set.*** The first case is a speed increasing gear set from a centrifugal compressor. Micropitting was found on the pinion on the dedendum extending through the pitch line to the addendum, favoring the drive end. Macropitting was also present. Micropitting was also found on the gear around the pitch line. The gear set had run for approximately 120,000 hours ($54.6 \times 10^9$ cycles) (Figs. 5–7).

ISO/TS 6336-22 applies to gearing with pitch line velocities between 8 m/s and 60 m/s. This example has a pitch line velocity of 88 m/s, which exceeds the limits of the calculation. We will apply the analysis to see how it works in this case. The input data for this calculation is shown in Table 1.

ISO/TS 6336-22 calculates the minimum specific film thickness as 2.117, meaning that the minimum film thickness of the lubricant was much larger than the average roughness value of the tooth surfaces. This minimum occurred at the lower point of the active profile, which corresponds with the actual location of micropitting in the zone closest to the compressor (Fig. 8).

The permissible specific film thickness was calculated using the lubricant viscosity and failure load stage. The value determined from ISO/TS 6336-22, Figure A.1 was 0.157. This is consistent with the expectations of the performance of an ISO viscosity grade

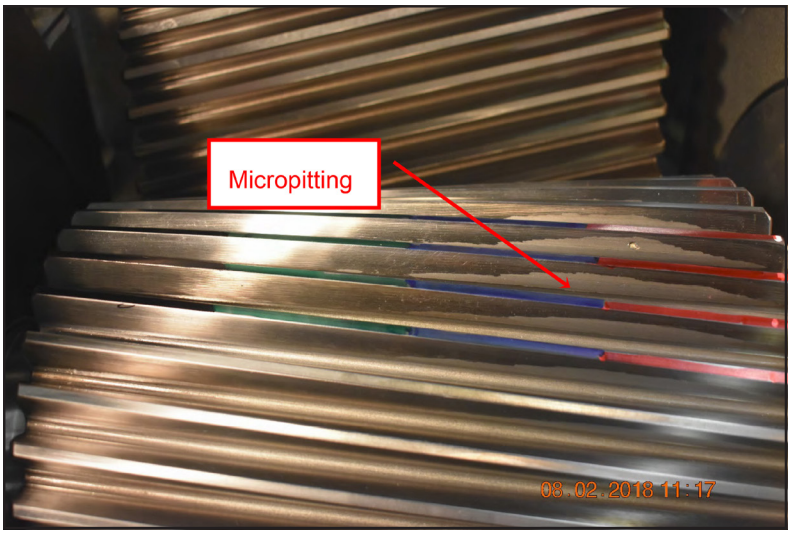


**Figure 5** Micropitting on the flank of the pinion teeth. The red zone is closest to the compressor (drive end). The black zone is the motor side. (Photo courtesy of Artec Machine Systems).

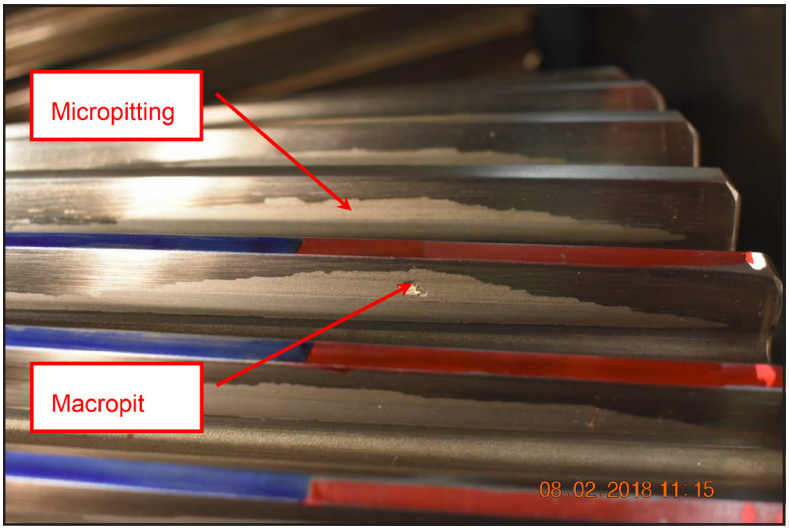


**Figure 6** Close-up of micropitting, which begins 100 mm into the face and extends to 270 mm. There is a single macropit on one tooth. (Photo courtesy of Artec Machine Systems).

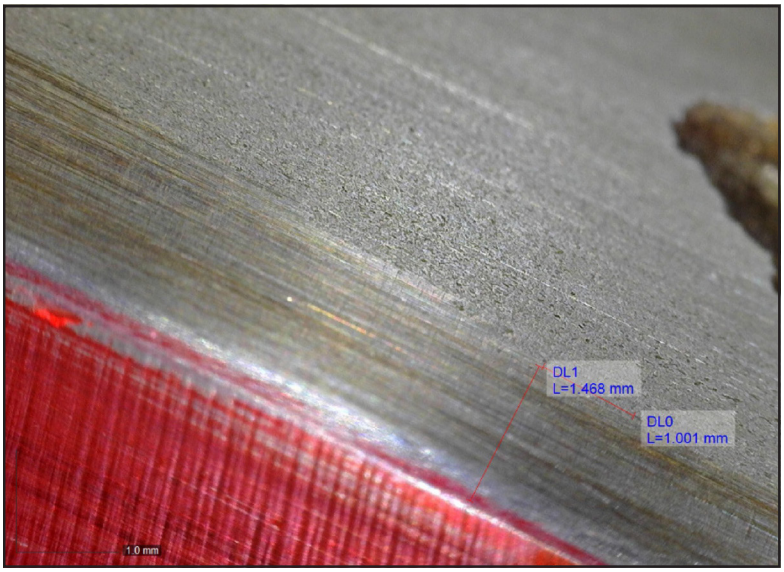


**Figure 7** Close-up of the micropitting in the region just above the macropit. (Photo courtesy of Artec Machine Systems).

(vg) 32 mineral oil in an FVA-FZG micropitting test rig. By comparing the values of the minimum specific film thickness and the permissible value, a safety factor of 13.484 was calculated.

This safety factor is very high and our first concern is that the high pitch line velocity is outside the regime where the calculation model has been tested. In order to test whether the result is more reasonable within the limits of 8 m/s to 60 m/s, we assumed constant torque and brought the pinion speed down to 5,000 rpm; the pitch line velocity became 58 m/s. This resulted in a minimum specific film thickness of 1.68 and a safety factor of 10.701. We cannot claim that being outside the limits of the calculation is the only reason for the high safety factor.

Safety factors of this magnitude normally indicate that this gear set is safe from micropitting, which is not the case in operation. The large value occurs as the specific film thickness becomes much larger than the permissible specific film thickness of the lubricant. With a specific film thickness that is over 2.0, the film thickness is much larger than the average roughness value of the surfaces. Theoretically, the tooth surfaces are not coming into contact and we are in a regime of full elastohydrodynamic lubrication (EHL) lubricant film (Ref. 8). This can be verified by observing the non-micropitted tooth flanks in Figures 5 and 6, which look almost new. Micropitting has occurred without apparent tooth flank or asperity contact! In this regime, film thickness is not an accurate indicator of micropitting risk. Micropitting may have been caused by hydraulic forces due to contact pressure, lubricant viscosity, or shear forces.

***Case 2 – wind turbine gear set.*** Case 2 is a gear set from a 1.5-MW wind turbine at the National Wind Technology Center on the National Renewable Energy Laboratory's (NREL's) Flatirons Campus in Colorado. The turbine began operating in May 2009. It was physically inspected in 2012 with a borescope with no signs of damage to the bearings or gear teeth. Further inspections revealed damage, though, such as:

- A 2015 borescope inspection of the gearing and bearings showed damage on the sun pinion teeth in the form of hard contact lines on several teeth, abrasive wear marks on one tooth, and evidence of micropitting on the flank of one tooth and in the region of the start of active profile (SAP) of several teeth. Normal wear was observed on the planet gears and bearings.
- A 2017 inspection noted excessive black sludge in the lubricant filter sediment collector. Tests of lubricant samples pointed out particle counts that were higher than acceptable, reduced lubricant performance, and warning levels of water in the lubricant.
- The gear drive was removed from service in December 2017. A third borescope inspection was conducted on the gear drive at the repair shop before tear down in April 2018. Micropitting was found in the start of active profile (SAP) of all of the sun pinion teeth. Micropitting and some abrasion were also found higher on the flanks of the sun pinion teeth. The gear set had been installed for 8.5 years. It had produced approximately 6-million kilowatt-hours of energy in 14,170 hours of grid operation time. This represents approximately 8% of its minimum design life. Micropitting was noted after 6.5 years of operation. Figure 9 contains pictures of the micropitting and additional details about the gear drive condition are reported in (Ref. 9).

Additional details about the gear drive condition are reported (Ref. 9).

As mentioned in Case 1, ISO/TS 6336-22 applies to gearing with pitch line velocities between 8 m/s and 60 m/s. This example has a pitch line velocity of 3.0 m/s, which is below the limits in the calculation. In order to review the results of the method with slow speed applications, though, we will apply the analysis to see how it correlates to field experience.

**Table 1 Input data for Case 1**

| Dimension | Units | Pinion | Gear |
|---|---|---|---|
| Number of teeth | - | 37 | 163 |
| Ratio | - | 4.405 | |
| Center distance | mm | 600 | |
| Normal module | mm | 5.90 | |
| Face width | mm | 280 | |
| Outside diameter | mm | 236.30 | 987.29 |
| Pressure angle | degrees | 20.00 | |
| Helix angle | degrees | 10.00 | |
| Addendum modification coefficient | - | 0.2407 | -0.0876 |
| Surface roughness | µm | 0.41 | 0.40 |
| ISO accuracy grade | - | 4 | 4 |
| Material surface hardness | HRC | 58-64 | 58-64 |
| Pinion speed | rpm | 7,582.0 | |
| Pinion torque | N-m | 12,209.3 | |
| $K_A$ $K_v$ $K_{H\alpha}$ $K_{H\beta}$ product | - | 1.4170 | |
| Lubricant | - | Mobil Teresstic AC 32 | |
| Inlet lubricant temperature | °C | 54 | |

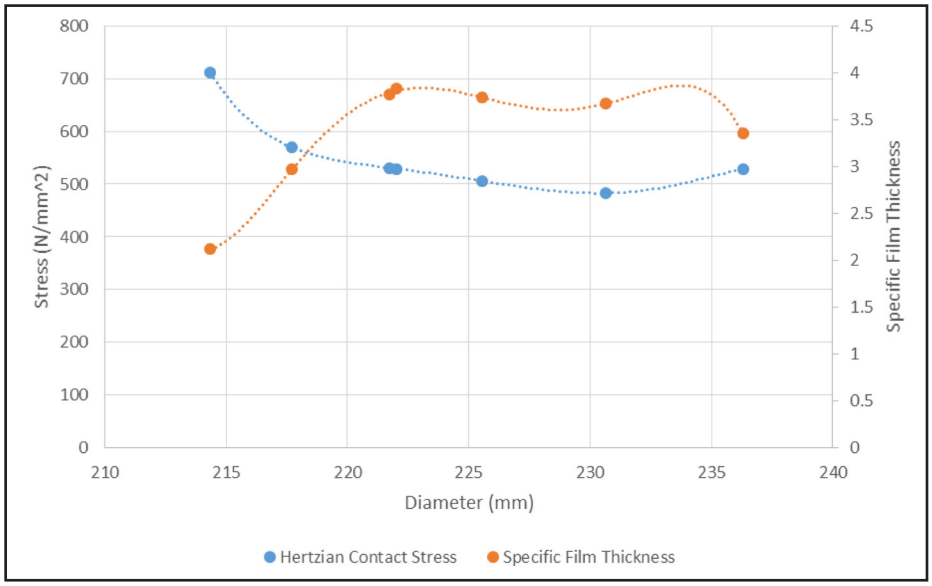


**Figure 8 Graph of contact stress and specific film thickness along the line of action.**

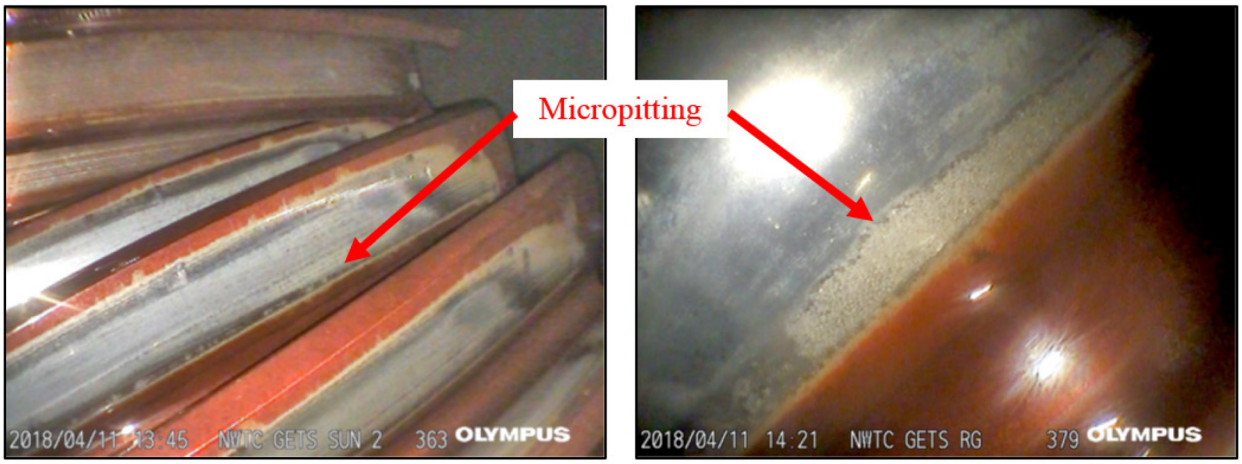


**Figure 9 Micropitting on the sun pinion teeth. (Photos by Scott Eatherton, Wind Driven, NREL 61193 and 61194).**

ISO/TS 6336-22 calculates the minimum specific film thickness as 1.589, indicating that the film thickness is much larger than the roughness of the teeth. The specific film thickness was determined at the lower point of the active profile of the pinion. This agrees with the location of the start of micropitting in the SAP of the flanks of the sun pinion.

The permissible specific film thickness was calculated using the simplified graphs in Method B, relying on the lubricant viscosity and failure load stage. The value determined from ISO/TS 6336-22, Figure A.1 was 0.239. This is consistent with the expectations of the performance of an ISO vg 320 synthetic lubricant in an FVA-FZG micropitting test rig. By comparing the values of the minimum specified film thickness and the permissible value, a safety factor of 6.635 was calculated.

It was noted that the gear drive sump temperature averages 50°C, but can get as high as 70°C. To test the range of sump temperatures, the calculation was run again using 70°C as the lubricant inlet temperature. The thinner film thickness caused the curve of the specific film thickness over the contact points in the line of action to flatten, decreasing the safety factor to 3.614 (Fig. 10).

This safety factor is lower than what was calculated in the first case, but still higher than what is typically seen in a mechanical rating of a gear set. Without knowing that the gear set has experienced micropitting, it would be easy to assume that a safety factor over 1.0 means that damage will not occur.

**Table 2 Input data for Case 2**

| Dimension | Units | Pinion | Gear |
|---|---|---|---|
| Number of teeth | - | 28 | 88 |
| Ratio | - | 4.6987 | |
| Center distance | mm | 487.51 | |
| Normal module | mm | 8.0609 | |
| Face width | mm | 200.9 | |
| Outside diameter | mm | 247.955 | 759.079 |
| Pressure angle | degrees | 22.5 | |
| Helix angle | degrees | 17.584 | |
| Addendum modification coefficient | - | 0.0804 | 0.0804 |
| Surface roughness | 1.1m | 0.22 | 0.55 |
| ISO accuracy grade | - | 6 | 6 |
| Material surface hardness | HRC | 59-63 | 58-62 |
| Pinion speed (nominal) | rpm | 254.17 | |
| Pinion torque (nominal) | N-m | 20,880 | |
| KA Kv KHa K-03 product | - | 1.478 | |
| Lubricant | - | Castrol Optigear A320 | |
| Inlet lubricant temperature | °C | 50 | |

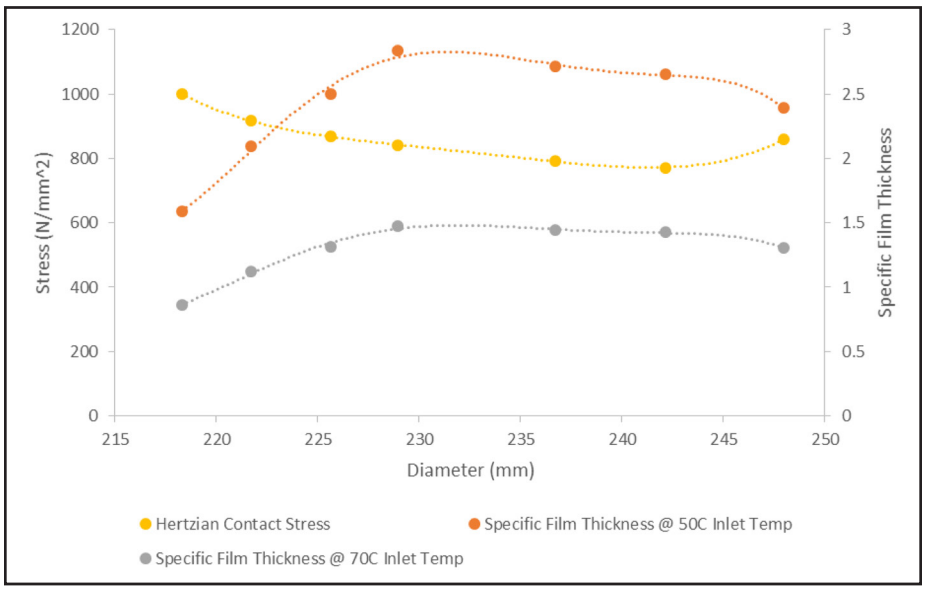


**Figure 10 Specific film thickness at the points along the line of action with inlet temperatures of 50°C and 70°C.**

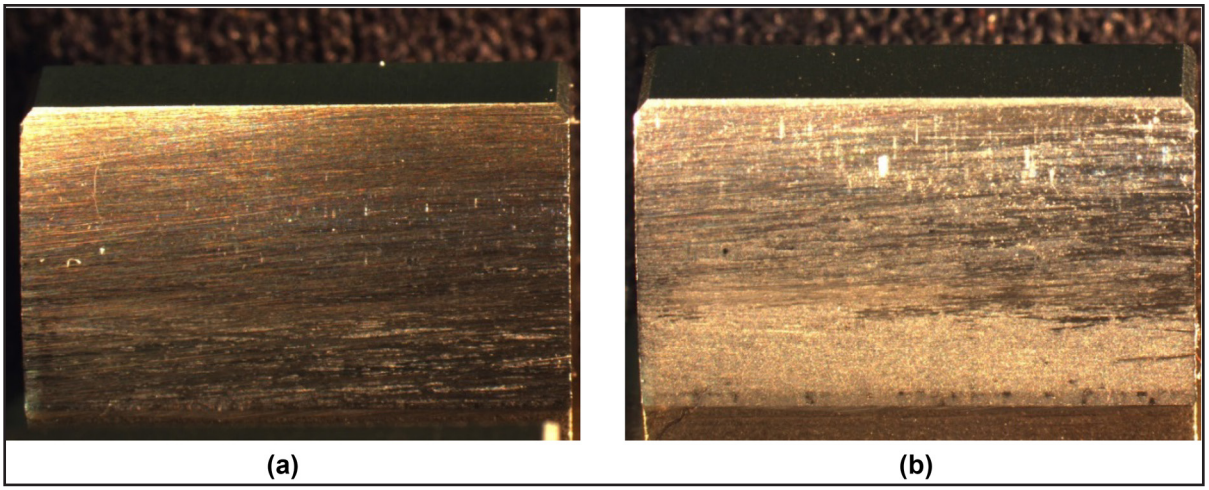


**Figure 11 Micropitting images of test pinion after (a) Stage 6 and (b) Stage 10.**

**Table 3 Input data for Case 3**

| Dimension | Units | Pinion | Gear |
|---|---|---|---|
| Number of teeth | - | 20 | 30 |
| Ratio | - | 1.50 | |
| Center distance | mm | 91.50 | |
| Normal module | mm | 3.629 | |
| Face width | mm | 13.97 | |
| Outside diameter | mm | 82.042 | 116.716 |
| Pressure angle | degrees | 20.00 | |
| Helix angle | degrees | 0 | |
| Addendum modification coefficient | - | 0.2533 | -0.0296 |
| Surface roughness | Idm | 0.34 | 0.22 |
| ISO accuracy grade | - | 4 | 5 |
| Material surface hardness | HRC | 59-61 | 59-61 |
| Pinion speed | rpm | 2.250 | |
| Pinion torque | N-m | 265 | |
| KA Kv KHot KHp product | - | 1.0826 | |
| Lubricant | - | Various | |
| Inlet lubricant temperature | °C | 80 | |

**Table 4 Lubricant data for Case 3**

| Lubricant Designation | M-460-EP | M-220-EP | M-220-RO | M-068-EP |
|---|---|---|---|---|
| Base Lubricant Type | Mineral | Mineral | Mineral | Mineral |
| Additive Type | EP | EP | R&O | EP |
| Kinematic Viscosity @40°C (cSt) | 427.9 | 211.9 | 215.5 | 68.2 |
| Kinematic Viscosity ©40°C (cSt) | 30.6 | 18.7 | 19.0 | 8.5 |

**Table 5 Input data for FZG "C" gear set**

| Dimension | Units | Pinion | Gear |
|---|---|---|---|
| Number of teeth | - | 16 | 24 |
| Ratio | - | 1.50 | |
| Center distance | mm | 91.50 | |
| Normal module | mm | 4.5 | |
| Face width | mm | 14 | |
| Outside diameter | mm | 82.45 | 118.35 |
| Pressure angle | degrees . | 20.00 | |
| Helix angle | degrees | 0 | |
| Addendum modification coefficient | - | 0.1817 | 0.1716 |
| Surface roughness | !dm | 0.5 | 0.5 |
| Pinion speed | rpm | 2,250 | |
| Pinion torque | N-m | 265.1 | |
| Ka Kv KHa KI-Ip product | - | 1.05 | |
| Inlet lubricant temperature | °C | 90 | |

As with Case 1, this case has a high value for the specific film thickness and the tooth flanks and asperities should not be in contact. The undamaged sections of the tooth flanks are witness to this. Yet micropitting occurred on the flanks in the area of highest sliding.

As in the previous example, studies were not done with similar gearing to determine the permissible specific film thickness. Micropitting test results per Method A were not available. As a result, the calculation used the general permissible specific film thickness curves from FVA-FZG testing of mineral oils per ISO/TS 6336-22, Annex A. If the lubricant quality and operating temperature are taken into effect, the permissible specific film thickness is calculated to be 0.319 assuming a high quality lubricant.

Whether the minimum specific film thickness is calculated with an inlet lubricant temperature of 50°C or 70°C, the safety factor will still be greater than 2.50.

Wind turbines are considered to be critical applications due to the difficulty of repairing a gear drive that is high above the ground in close quarters. To seriously apply ISO/TS 6336-22 to a wind turbine gear drive, Method A should be used to determine the lubricant film thickness in the entire contact zone. A more representative way of determining the permissible specific film thickness should also be assessed.

***Case 3 – AGMA tribology test gear set.*** The third case is the gearing used in the AGMA Tribology Test Program (Ref. 10). This program was a project of the AGMA Helical Gear Rating Committee to explore the effect of lubricants on gear

life with tests that mimic actual operation. Several AGMA member companies collaborated to design and manufacture gearing that is similar to FZG "C" gears, but more representative of industrial gears. They have finer pitch, different tooth counts, and incorporate tip relief and profile modifications to remove interference. They also have axial crowning to increase compressive stress near the center of the face width. One hundred gear sets were manufactured and run in a four-square FVA-FZG test rig with the sump temperature set to 80°C. The tests were stopped every 24 hours for inspection with a stereomicroscope set to 10x magnification. The test was terminated if one of three events happened:

- Macropitting damage that exceeded 1% of the total surface area of all pinion or gear teeth.
- Macropitting damage that exceeded 4% of the total surface area of a single tooth.
- 400 hours of running time without damage.

Note that the presence of micropitting didn't stop the test—it was noted in the results.

The gear sets were run with five different mineral lubricants from three viscosity grades (68, 220, and 640) and two additive packages (EP and R&O). The torque was set to 265 Newton-meter (N-m) and the pinion speed was 2,250 rpm. The pitch line velocity of this gear set is 8.624 m/s, which is at the lower limit of the speed range of ISO/TS 6336-22.

Unfortunately, detailed pictures of the gear teeth after testing have not been retained. The Tribology Test Program report describes micropitting on most gear teeth. Houser included some pictures of micropitted pinion teeth his 2015 AGMA Fall Technical Meeting paper (Ref. 11).

The gear sets in this example are relatively close to the FZG "C" gears. It's possible to calculate the permissible specific film thickness with the geometry of the FZG "C" gears and the failure load stage of each lubricant. ISO/TR 6336-31 (Ref. 12) illustrates this calculation in Example 1. The FZG "C" gears are known to begin micropitting at contact point A. The nominal torque for failure load stage 10 is 265.1 N-m. Method A calculations of the nominal Hertzian contact stress at point "A" come to 1,476 N/mm$^2$. This stress was entered into the analytical model to find the permissible specific film thickness. The input data for the FZG "C" gears is shown in Table 5.

**Table 6 Results of ISO/TS 6336-22 calculations for AGMA Tribology Test gearing**

| Case Number | Lubricant | Torque N-m | $\lambda_{GFmin}$ | $\lambda_{GFP}$ | $S_\lambda$ |
|---|---|---|---|---|---|
| 1 | M-460-EP | 265 | 0.331 | 0.185 | 1.79 |
| 2 | M-220-EP | 265 | 0.226 | 0.129 | 1.75 |
| 3 | M-220-RO | 265 | 0.237 | 0.136 | 1.75 |
| 4 | M-068-EP | 265 | 0.123 | 0.073 | 1.69 |
| 5 | M-220-EP | 300 | 0.201 | 0.129 | 1.56 |
| 6 | M-220-EP | 400 | 0.147 | 0.129 | 1.14 |

The calculation results for the four different lubricants are shown in Table 6. After running all calculations at 265 N-m, additional load cases of 300 N-m and 400 N-m were calculated to see if increasing the load would push the safety factor below 1.0.

Micropitting was found in the dedendum of most gearing during the AGMA Tribology tests. The minimum specific film thickness is below 1.0 in all cases, indicating that gears are running in a boundary lubrication regime. In boundary lubrication, the properties of the lubricant and the roughness of the tooth flank surfaces can contribute to surface distress, such as micropitting. The safety factors against micropitting remain above 1.0, though.

This is not an unusual result from ISO/TS 6336-22 and it has been reported in other papers. Both Pinnekamp (Ref. 7) and Sagraloff (Ref. 13) present graphs of safety factors calculated for various examples as functions of the quality of the calculations and knowledge of the operating conditions; these graphs are similar to Figure 12. Assuming that the calculations for this case are average, the results fit into the grey space in the chart.

As the load increased, the shape of the film thickness curve along the points of the line of action remains consistent. This can be seen in Figure 13. Note that the contact stresses for each case are not shown in the figure in order to make the graph more readable. The prominent "bump" in the curves is caused by the local sliding factor in the film thickness equation, which rises sharply to a value of 1.0 at the pitch point and then drops off quickly.

## Conclusion

ISO/TS 6336-22 is the only published technical document for predicting the occurrence of micropitting on gear tooth flanks. It should not be applied without a careful study of the calculations in the document and an understanding of the

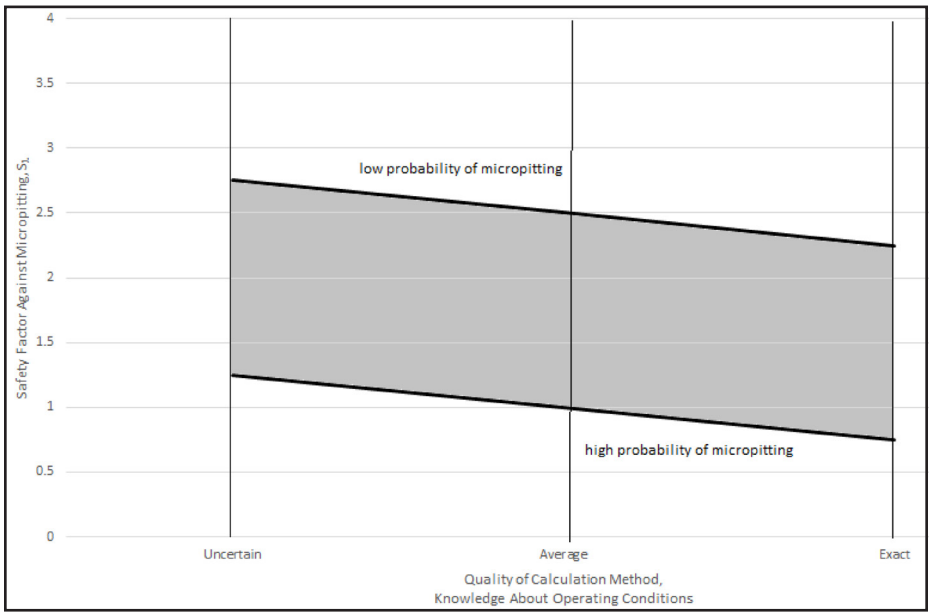


**Figure 12 Probability of micropitting as a function of calculation method and application knowledge.**

gear set application. This paper uses ISO/TS 6336-22 to calculate the risk of micropitting for three cases: a high-speed centrifugal compressor example, a low-speed wind turbine example, and the AGMA Tribology Test Program gearing. From the results, one can observe limitations in the method.

- When the specific film thickness is much larger than unity, micropitting cannot be predicted by film thickness and surface roughness alone. This was seen in the Cases 1 and 2. The science behind micropitting is still being researched and developed. As more is understood about the causes and predictability of micropitting, ISO/TS 6336-22 should be updated. Until then, it would be good to include in the document a discussion of the applicability of the method when the minimum specific film thickness is above 1.0.
- The permissible specific film thickness may not be representative of the lubrication used in service if testing with real gears cannot be done. ISO/TS 6336-22 contains Method B to calculate this value based on standardized testing, but this will become less certain as the geometry of the gear set differs from the geometry of the test gearing. The formulation of the lubricant used in the application also may differ from the reference curves in Annexes B and C. ISO/TS 6336-22 contains warnings about these limits and the engineer should be aware of them.
- High safety factors do not indicate low risk of micropitting. Users of ISO/TS 6336-22 should review guidance to select the minimum safety factor based on the critical nature of the application, the accuracy of the gear measurements, the availability of test data, and the uncertainty of operating conditions. If the application is critical, Method A should be used for the calculation rather than Method B.

Future work with the cases in this paper would further explore the behavior of the ISO/TS 6336-22 specification by calculating the safety factor using Method A to arrive at the minimum- specific film thickness. Ideally, full roughness profiles would also be used for the calculations so that the impact of roughness variation in the contact zone and of different types of roughness measurements (*Ra*, *Rq*, *Rmr*, etc.) can be determined. These results would be compared to Method B and field experience.

There are limits to the accuracy of ISO/TS 6336-22, both in the specific film thickness calculation and the determination of permissible specific film thickness. It is suggested that the minimum safety factor against micropitting be selected based on experience with similar gear sets. If this is not available, the gear designer must decide if micropitting can be tolerated in the application. If not, micropitting can best be avoided by modifying tooth microgeometry to reduce overall contact stresses, reducing surface roughness (through superfinishing or by other means), and/or using high-quality lubricants.

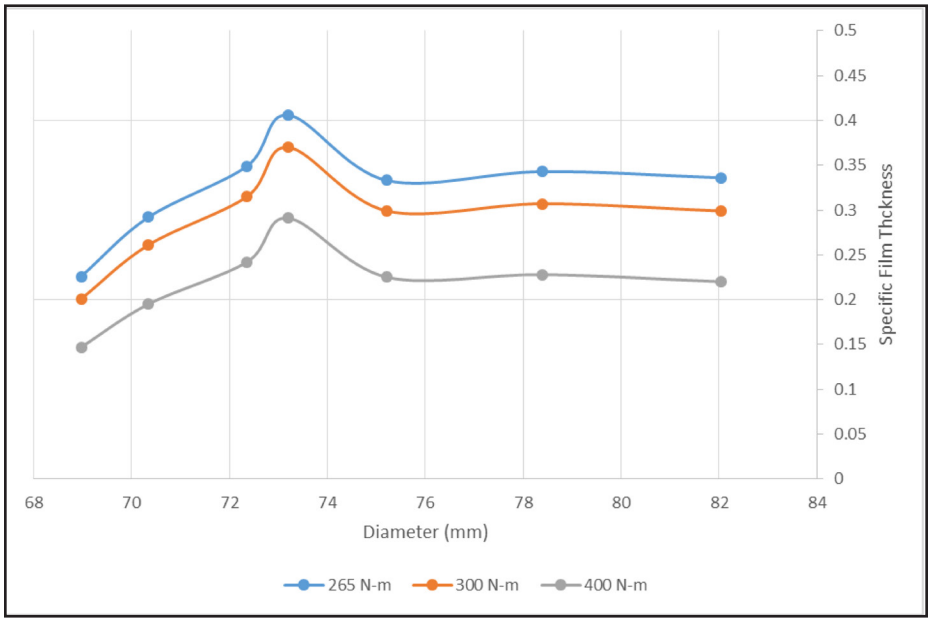


**Figure 13 Specific film thickness at the diameters along the path of contact for M-220-EP at various loads.**

**For more information.**
Questions or comments regarding this paper? Contact Robin Olson at *Robin.Olson@rexnord.com.*

***Acknowledgment.*** This work was authored in part by the National Renewable Energy Laboratory, operated by Alliance for Sustainable Energy, LLC, for the U.S. Department of Energy (DOE) under Contract No. DE-AC36-08GO28308. Funding provided by the U.S. Department of Energy Office of Energy Efficiency and Renewable Energy Wind Energy Technologies Office. The views expressed in the article do not necessarily represent the views of the DOE or the U.S. Government. The U.S. Government retains and the publisher, by accepting the article for publication, acknowledges that the U.S. Government retains a nonexclusive, paid-up, irrevocable, worldwide license to publish or reproduce the published form of this work, or allow others to do so, for U.S. Government purposes.

"The Alliance for Sustainable Energy, LLC (Alliance) the manager and operator of the National Renewable Energy Laboratory (NREL) Employees of Alliance for Sustainable Energy, LLC, under Contract No. DE-AC36-08GO28308 with the U.S. Dept. of Energy, have authored this work. The United States Government retains and the publisher, by accepting the article for publication, acknowledges that the United States Government retains a non-exclusive, paid-up, irrevocable, worldwide license to publish or reproduce the published form of this work, or allow others to do so, for United States Government purposes."

## References


1. ANSI/AGMA. 2014, "Appearance of Gear Teeth - Terminology of Wear and Failure," 1010-F14.
2. Errichello, R.L. 2012, "Morphology of Micropitting," ***Gear Technology***, November/December, pp. 74–81.
3. ISO/TC 60/SC 2/WG 6. 2018, "Calculation of Load Capacity of Spur And Helical Gears — Part 22: Calculation of Micropitting Load Capacity," ISO/TS 6336-22.
4. FVA-Information Sheet 54/7. "Test Procedure for the Investigation of the Micropitting Capacity of Gear Lubricants," 1993.
5. U. Kissling, 2011. "The Application of the First International Calculation Method for Micropitting," *AGMA Fall Technical Meeting*, Paper 11FTM12.
6. Pinnekamp, B., T. Weiss, and G. Steinberger. 2011, "Micropitting - A Serious Damage? Testing, Standards, and Practical Experience," *AGMA Fall Technical Meeting*, Paper 11FTM15.
7. Pinnekamp, B. and M. Heider. 2015, "Calculating the Risk of Micropitting Using ISO Technical Report 15144-1:2014 – Validation with Practical Applications," *AGMA Fall Technical Meeting*, Paper 15FTM26.
8. Dudley, D. 1981, "Characteristics of Regimes of Gear Lubrication," *International Symposium on Gearing 2039 and Power Transmissions*, Tokyo, Japan.
9. Keller, J., Michaud, M., and Lambert, S., 2020, "Report of the Condition of a General Electric Transportation Systems Gearbox," NREL/TP-5000-76004, National Renewable Energy Laboratory, Golden, CO.
10. Bradley, W.A. "AGMA Tribology Test Report," AGMA Foundation Report, 2008.
11. Houser, D.R. and Shon S. 2015, "An Experimental Evaluation of the Procedures of the ISO/TR 15144 Technical Report for the Prediction of Micropitting," AGMA Fall Technical Meeting, Paper, 15FTM25.
12. ISO/TC 60/SC 2/WG 6, 2018. "Calculation of Load Capacity of Spur and Helical Gears — Part 31: Calculation Examples of Micropitting Load Capacity," ISO/TR 6336-31.
13. Sagraloff, N., M. Hein, T. Tobie and K. Stahl. 2017, "Practical use of micropitting test results according to FVA 54/7 for calculation of micropitting load capacity acc. to ISO/TR 15144-1," *VDI International Conference on Gears*.



**Robin Olson** is the Director of Applications Engineering for the Material Handling Vertical at Rexnord here. She is a member of the AGMA Helical Gear Rating Committee, Chairperson of the AGMA 925 subcommittee, and is honored to act as US delegate to ISO Working Groups 6 (Gear calculations). Previously, Robin has also been a member of the AGMA Computer Programming, Enclosed Drives, and Marine Drive committees. Robin holds a Bachelor of Science in Physics from the University of Wisconsin—LaCrosse and a Master of Science in physics from the University of Wisconsin—Madison.

**Mark Michaud** recently celebrated his 40th year with REM Surface Engineering. Mark is the inventor and pioneer of REM's chemically accelerated finishing technology; currently serving as a Technical Fellow. Mark Michaud is a leading scientist, technical mentor and continues to play a crucial role in the company's future. Mark has authored numerous patents and technical papers and served a term on the AGMA Board of Directors. Mark continues to serve as vice-chair of the AGMA Aerospace Committee, as a member of the AGMA Wind Turbine Committee and as a shadow delegate on the ISO 614-4 Wind Turbine Committee. He graduated with a Bachelor's degree in Chemistry from Reed College and an MBA from the University of Hartford.

**Jonathan Keller** is the team leader for drivetrain technology. Jon leads projects ranging from the development and verification of new drivetrains with improved performance, power density, and efficiency and to in-depth research investigations of failure modes and improvement in reliability of existing drivetrains often conducted in the dynamometers or field turbines at the NREL Flatirons Campus. Examples of these projects are the Drivetrain Reliability Collaborative and the NREL-led Next Generation Drivetrain. Prior to joining NREL, Keller worked for the U.S. Army at Redstone Arsenal, Alabama for 10 years, where he developed condition monitoring systems for Army rotorcraft to reduce the cost and maintenance burdens while increasing availability and safety.