ASIA TURBOMACHINERY & PUMP SYMPOSIUM
23-26 FEBRUARY 2021
SHORT COURSES: 22 FEBRUARY 2021

ADOPTION OF LOW VISCOSITY OIL IN TURBOMACHINERY APPLICATIONS WITH HIGH SPEED GEARS

Sophie Bentz
R&D Engineer – Mechanical components
Power Transmission
Baker Hughes, a GE company
Fougerolles, France

Antoine Lepeule
R&D Engineer – Mechanical components
Power Transmission
Baker Hughes, a GE company
Fougerolles, France

Alessandro Pescioni
Senior Engineer - Advanced Train Integration
Turbomachinery & Process Solutions
Baker Hughes, a GE company
Florence, Italy

Gwenael Perney
Principal Mechanical Engineer
Power Transmission
Baker Hughes, a GE company
Fougerolles, France

Gaspare Maragioglio
PT Engineering Leader
Power Transmission
Baker Hughes, a GE company
Florence, Italy

Angelo Grimaldi
Principal Engineer for Centrifugal Compressor Aerodynamics
Turbomachinery & Process Solutions
Baker Hughes, a GE company
Florence, Italy

Sophie Bentz is currently a R&D Engineer for mechanical components at Baker Hughes, a GE company. She joined Baker Hughes in 2017 as R&D Engineer after experience in the Aerospatiale industry. She's involved in R&D programs, rotor-dynamics and development of new calculation tools for gear and bearings performances. Ms. Bentz received a degree in Mechanical Engineering from ENSEM (Electr. & Mech. Superior Nat. School) of Nancy in 2015.

Gaspare Maragioglio is currently the Power Transmission Engineering Leader at Baker Hughes, a GE company. He has 17 years of experience in rotating equipment He supported R&D programs, manufacturing/test department for full speed full load string test, as well as site commissioning trouble shooting and RCA.
Mr. Maragioglio has a degree in Mechanical Engineering and before joining GE he had a research assignment at University College London. He is currently a member of API613 Task Force and the ATPS Advisor Committee.

Alessandro Pescioni is currently a Senior Engineer - Advanced Train Integration at Baker Hughes, a GE company. He joined GE in 2011 as design Engineer in the Shaft Line Integration team. He has 8 years of experience in projects execution, integrated train rotor-dynamic analysis, vibration analysis, troubleshooting and RCA, support to test bench and R&D programs. Mr. Pescioni received a degree in Energy Engineering from University of Florence.

Antoine Lepeule is currently a R&D Engineer for mechanical components at Baker Hughes, a GE company. He joined GE in 2013 as Gearbox Design Engineer. He's now involved in R&D programs and troubleshooting for all gear issues, including vibration and temperatures problems.
Mr. Lepeule received a degree in Innovation and Mechanical Engineering from University of Technology of Belfort-Montbeliard in 2013.

Gwenael Perney is currently a Principal Mechanical Engineer at Baker Hughes, a GE company. He joined GE in 2002 as Gearbox Design Engineer. He's now Technical Manager specialized in High Speed gears and is deeply involved in troubleshooting for all gear issues, including vibration and temperature problems. His responsibility includes product development and R&D programs.
Mr. Perney received a degree in Engineering Design and Mechanics from the Ecole Polytechnique Universitaire de Lille Politech'Lille.

Angelo Grimaldi is currently Principal Engineer for Centrifugal Compressor Aerodynamics at Baker Hughes, a GE company. He joined GE in 2005 as CFD specialist in the Centrifugal Compressor R&D team. His responsibility includes design, development and analyses of centrifugal compressor aerodynamic components. Dr. Grimaldi graduated in Mechanical Engineering in 2001 from the Polytechnic of Bari; in 2004, he gained the Ph.D. on Numerical Fluid Dynamics from the same University.

ABSTRACT

In turbomachinery applications, the geared transmission equipment is among the most affected by the lube oil characteristics and properties. In a gear system the oil is used in the fluid-dynamic bearings, journal and thrust, as well as to lubricate and especially to cool down the meshing gears. The continued market demand to have more efficient machines, thus achieving energy savings and carbon foot-print reduction has motivated manufacturers to look for improvements even in area that have rarely been explored like the lube oil viscosity in high performance and high efficiency lubricants for Turbomachinery applications.

This paper is providing a comprehensive theoretical analysis of the potential impacts on a high-speed gear when a low viscosity oil is used, together with a validation through experimental data.

The authors used referenced formulas, tools and CFD models to predict the effects of low viscosity oil on scuffing service factor, rotor-dynamics, lube oil flow and power losses where the windage contribution represented one of the most impacted factors.

Finally, a comparative overview is presented to show the gearbox rotor-dynamic behavior and its performances at Full Speed No Load and Full Speed Partial Load with the low viscosity oil versus its behavior with an oil having a more common and diffused viscosity.

INTRODUCTION

A carbon footprint is typically defined as the total emissions caused by an individual, event, organization, or product, expressed as carbon dioxide equivalent.
Due to the complexity of interactions between contributing processes, Wright, Kemp, and Williams (1), proposed to define the carbon footprint as:
"A measure of the total amount of carbon dioxide (CO2) and methane (CH4) emissions of a defined population, system or activity, considering all relevant sources, sinks and storage within the spatial and temporal boundary of the population, system or activity of interest. Calculated as carbon dioxide equivalent using the relevant 100-year global warming potential (GWP100)."

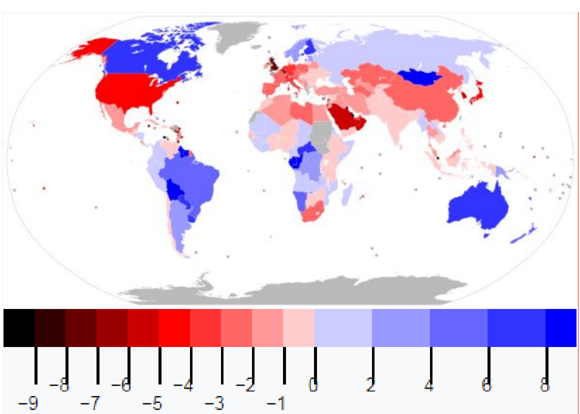

Figure 1: National ecological surplus or deficit

The concept name of the carbon footprint originates from ecological footprint, discussion, which was developed by William E. Rees and Mathis Wackernagel in the 1990s. This accounting approach compares how much people demand compared to what the planet can renew (Figure 1).

Today roughly 80% of Oil and Gas related greenhouse gases are emitted during the end use of petroleum-based fuels, gas and other products, compared to just 20% during their production. Nevertheless, several Oil and Gas Companies unveiled plans to reduce carbon footprint of their energy product and services.
Targets are differently set based of different process and equipment portfolio, but these targets are in most of the cases ambitious, typically looking for a reduction of around 30% within the next 20 years.
The scope includes emissions covered directly from site operations such as extraction, transportation and processing of raw materials and transportation of products.
The decision comes at a time when Oil and Gas companies are facing increasing shareholder pressure to address carbon emissions.
In the Turbomachinery industry we are all called to optimize the energy efficiency of the rotating equipment to reduce the energy consumption of the production facilities around the world and therefore to alter the current trajectory of energy-related carbon emissions.
One area to explore in turbomachinery equipment, to increase the efficiency, is relevant to the mechanical viscous losses and a way to reduce these losses is the use of low viscosity oils. In this paper the authors are specifically addressing opportunities and challenges in the geared systems, due to the use of an oil having a viscosity of 15CSt. @ 40 deg. C.
The target is to validate theoretical calculations and analyses through Full Speed Full Load (FSFL) test data, and therefore demonstrate that this is a viable solution to:

- Increase the overall equipment efficiency;
- Improve bearings and gear-mesh lubrication dynamics;
- Reduce the oil flow consumption;

USE OF LOW VISCOSITY OIL – DESIGN CONSIDERATIONS FOR HIGH SPEED GEARS

Gear surface distress

Design considerations

Rating methods for high speed gears (AGMA 6011-J14 (2) – API 613 5th (3)) are concerned with two main failure modes in gear teeth which are the surface pitting and root bending fatigue failure of the tooth material for a known/given number of cycles. Another design consideration in addition to bending and pitting with high capacity high-speed gears is surface distress phenomena referred to as scuffing (sometimes wrongly referred to as scoring). Both AGMA 6011-J14 (2) & API 613 5th edition (3) have referenced paragraphs entitled "Scuffing" which mandates scuffing as a design factor for AGMA & API gear unit rating.

Scuffing definition

When gears are subject to highly loaded conditions with high sliding velocities the lubricant film may not adequately separate the surface. As the gear teeth engage and disengage, a welding and tearing of the tooth surface from one flank to another occurs (transfer of metal from one tooth to another). The scuffed area appears to have a rough or matte texture. The damage typically occurs in the addendum, dedendum, or both, away from the operating pitchline in narrow or broad bands that are oriented in the direction of sliding. Scuffing may occur in localized patches and not on the whole contact surface. Scuffing is not a fatigue phenomenon and may occur instantly, many times during early stages of operation. The risk of scuffing damage varies with the material of the gear, lubricant being used, viscosity of the lubricant, surface roughness of the tooth flanks, sliding velocity of the mating gear set under load and geometry of the gear teeth. Changes in any or all of these factors can affect the scuffing risk.

Background – Risk evaluation analytical method

There are several analytical methods for scuffing risk evaluation. However, the threshold for determining whether a gear set will scuff remains mostly dependent on empirical results. Those known methods are not real standards but technical publications which are subject to improvement dictated by experience. Here below few known methods:

- AGMA 217.01 – "Information sheet Gear Scoring Design Guide for Aerospace Spur and Helical Power Gears" (4)
    - Document "Withdrawn" but still a recognized & referenced method for some gear OEM's since based on very large experience on the field.
- AGMA 925-A03 – "Effect of lubrication on gear surface distress" (5)
    - Information sheet means it is not an AGMA standard; still subject to further development
    - It is an enhancement of Annex A of AGMA 2101-C95.
- ISO/TS 6336-20 – "Calculation of scuffing load capacity – Flash temperature method (6)
    - "TS" means it contains calculation methods that are still subject to further development.
    - Method almost identical to AGMA 925-A03.
- AGMA 6011-J14 – Annex B – "A simplified method for verifying scuffing resistance" (2)

As they are called by API and AGMA gear rating methods, two methods are selected for the comparative analysis; AGMA 925-A03 (5) and AGMA 6011-J14– Annex B (2).

AGMA 925-A03 (5) is based on a Gaussian probability law (3 levels of severity) and AGMA 6011 J-14 – Annex B (2) is based on load function to be compared to a geometric function essentially dictated by load; pitch line velocity and oil viscosity.

Oil requirements

The key functions provided by the lubricant are to minimize the friction and wear between surfaces in relative motion, also to remove heat generated by the mechanical action of the system. In order to accomplish these tasks, the lubricant must have sufficient viscosity to separate the mating surfaces as much as possible have the appropriate chemical (additive) system to minimize thermal and oxidative degradation and finally provide anti-scuff performance.

AGMA 9005-E08 standard (7) provides the end user, original equipment builder, gear manufacturer and lubricant supplier with guidelines for minimum performance characteristics for lubricants suitable for use in general power transmission applications. These guidelines cover both open and enclosed gearing which have been designed and rated in accordance with applicable AGMA standards (standards called as well by API).

Low viscosity oils are not covered by AGMA 9005-E08 (7) guidelines however a previous paper "DEVELOPMENT, TESTING AND QUALIFICATION OF INNOVATIVE LOW VISCOSITY OIL IN TURBOMACHINERY APPLICATIONS" (8) presented at ATPS 2018 shows an extensive analysis of the low viscosity oil ISO VG 15 on a known turbomachinery system. Data presented show positive results in terms of fatigue of different oil parameters such as anti-scuff properties. This offers a good base to consider the use low viscosity oils for gearboxes.

Bearings performances and rotor dynamics

Bearings performances are critical for a gearbox. Due to mesh forces; and potential load cases (power/speed combination not necessarily constant) the forces withstood by radial bearings are quite significant and high in comparison to other rotating equipment. Oil viscosity and more generally, oil parameters (specific heat, density, additives) play a key role in the bearing's performances. Indeed, oil needs to have adequate viscosity to generate the hydrodynamic "film", move the generated heat so it can be transferred and flush the bearings to keep them clean.
Viscosity is basically the measure of fluids resistance to shear. It has to be high enough to get sufficient film thickness and low enough to limit power losses. Viscosity choice is therefore a balance between those two main important needs. One of the objectives of this paper is to analyze the impacts on friction bearings for gearboxes using low viscosity oils.

In addition to bearing performance (oil consumption / power losses / operating temperatures); bearings hydrodynamic parameters (stiffness and damping capabilities) are critical to rotor dynamics (lateral and/or torsional analysis). To make a gearbox operate successfully within its speed/power range for which it is designed for (including transient conditions such as start-up – coast down or barring conditions); viscosity plays a key role. Indeed, stiffness and damping capabilities of the oil film are mainly driven by the oil viscosity. In this paper rotor dynamics aspects were not considered due to the good behavior of the system with the use of VG15. Nevertheless, this could be deeply analyzed in a separate study.

Gear mesh power losses

Gear mesh mechanical power losses are composed of load-dependent losses and no-load dependent losses. Oil viscosity is involved in both and is therefore a key factor for gear mesh efficiency.
The load -dependent losses are the results of the oil shear stresses in between teeth generated by gear mesh contact along line of action and are called friction losses. The lower the viscosity is, the lower the shear resistance and the friction losses are.
The no-load dependent losses are mainly due to the aerodynamic drag forces (pressure and viscous) that act on rotating gear and are called windage losses. The lower the viscosity is, the lower the drag forces and the windage losses are.
To be noted that in high speed gearbox applications the windage losses are the most important one and make the friction losses negligible.

SCUFFING RISK EVALUATION

Scuffing typology

The existing normative references present how a scuffed gear looks like (ANSI/AGMA 1010-E95 (10) & ISO 10825:1995 (11). Here below an example showing the surface distress (Figure 2):

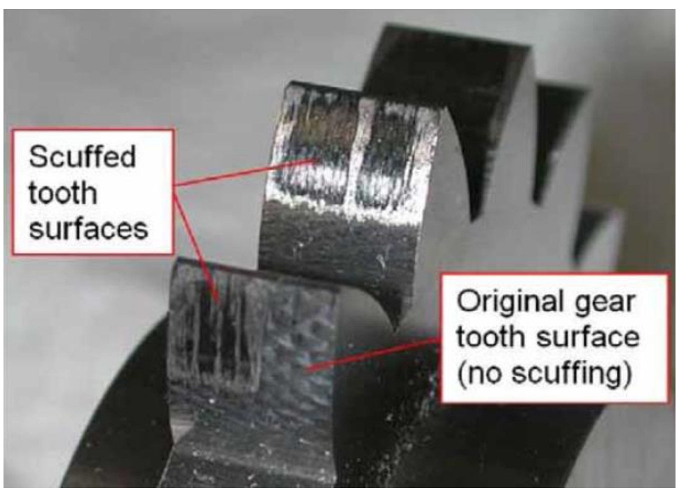

Figure 2: Scuffed Tooth surfaces (12)

Analytical calculations

AGMA 925-A03 (5) – Risk evaluation

Scuffing risk is calculated from a Gaussian distribution of scuffing temperature about the mean value. The evaluation of the scuffing risk is based on the probability of scuffing as per following Table 1:

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

Table 1 : Scuffing risk per AGMA 925-A03 (5)

Several parameters can play a role in reducing the scuffing risk such as load (linear load), sliding velocity and tooth surface finish. Those three parameters were evaluated for the mitigation of the risk.

The gearbox used for the Gas Turbine - Electric Generator test is an API 613 (3) unit. It is designed for VG32 oil use and the tooth flanks surface roughness is Ra 0.4 µm (API 613 (3) limit). For this design baseline, the scuffing risk is “Low” as per AGMA 925-A03 (Table 3). Having a slight increase of the tooth flanks surface roughness (Ra 0.5 µm) significantly impacts the risk of scuffing (18%, “Mod”) and even worst with Ra of 0.6 µm (45%, “High”) (Figure 3).

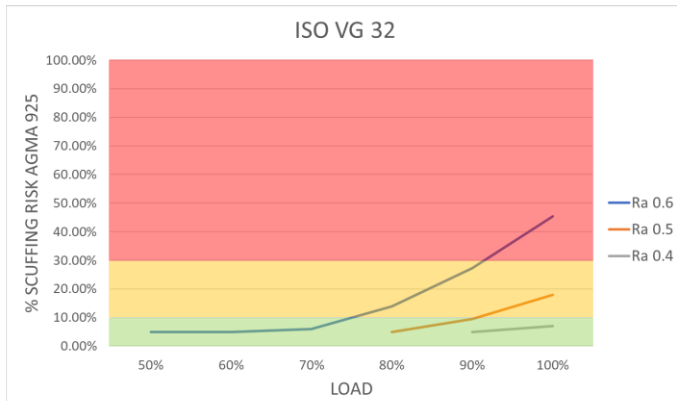

Figure 3: VG32 Scuffing risk graph according to load and surface roughness

A brutal change to VG15 without any considerations leads to increase the scuffing risk up to 40%, “High” at full load. To have a “Low” risk, only 70% of the transmitted power must be considered (Figure 4).
To have a “Low” risk for 100% of the transmitted power the surface roughness needs to be significantly improved; i.e. Ra 0.1 µm or better. This almost perfect surface quality is not easily achievable with conventional finishing processes (such as form grinding) but would require other manufacturing methods such as super finishing.

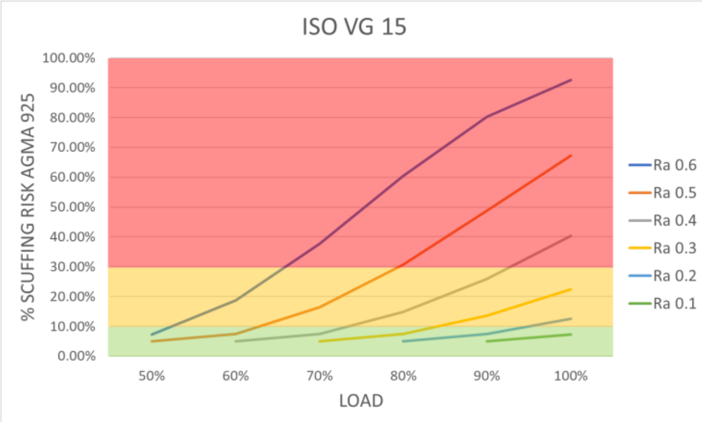

Figure 4: VG15 Scuffing risk graph according load and surface roughness

In order to better understand the influence of the load on the scuffing risk, a sensitivity versus the effective face width of the existing gearset has been carried out. Having VG 15 oil and standard roughness of Ra 0.4 µm, the scuffing risk is "Low" if we increase the effective face width by 33%. With Ra 0.3 µm, +19% and finally with Ra 0.2 µm, +5% (Figure 5). A roughness of 0.3 µm is reachable by conventional finishing processes but we still need to increase the face width by 20% which is a quite significant change.

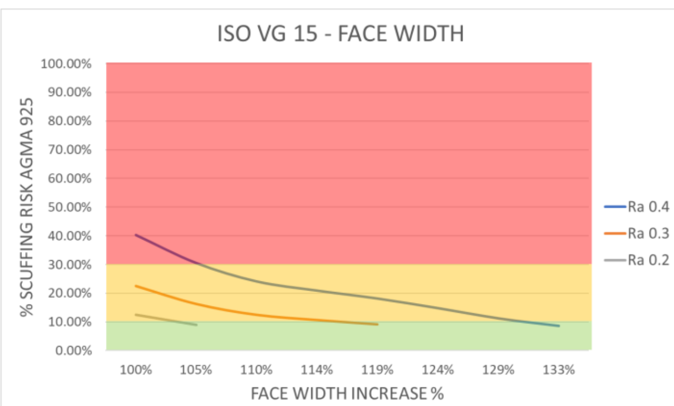

Figure 5 : VG15 Scuffing risk according face width and surface roughness

Another parameter which could play a role on the scuffing risk mitigation is the sliding velocity at engagement and disengagement at the gear mesh. When playing on the addendum modification (or profile shift coefficient) it increases or minimizes the scuffing risk and it is possible to find an equilibrium (see Figure 6).

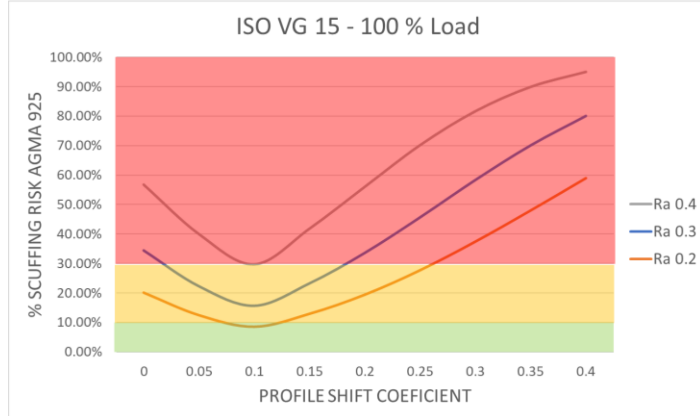

Figure 6: VG15 Scuffing risk graph versus profile shift coefficient and surface roughness

With VG15 and tooth flanks surface roughness of Ra 0.4 µm the scuffing risk is 40%, "High". Optimizing the teeth geometry by playing on the hob shift constant; for maximum scuffing safety the risk is reduced to 30% (still "High"). With Ra 0.3 µm it is reduced to 18 %, "Mod".

Analyzing the influence of the roughness, the load and the tooth geometry we can demonstrate that the roughness remains a key factor in scuffing risk mitigation. By the way, the teeth geometry represents a strong contributor which is worth to considering at the design phase.

AGMA 6011-J14 (2) – Annex B - Risk evaluation

AGMA 6011-J14 (2) suggests a simplified scuffing criterion that is suitable for general high-speed design work. From the values of tooth loading, pitch line velocity, and viscosity of the lubricant, a condensed load function, F (load) is formed, to assure scuffing resistance shall be less than (or equal to) the geometric function, F (geometric). The geometric function is based on gear characteristics such as number of teeth of the pinion and gear, center distance and gearset ratio. As long as the value of the load function, F (load), does not exceed that of the geometric function, F (geometric), there is adequate safety against scuffing.

$$F_{(\text{load})} \le F_{(\text{geometric})}$$

Load function, F (load):

$$F_{(\text{load})} = \left[\frac{w'}{C_w}\right]\left[v'\right]^{0.25}\left[\frac{46}{\nu_{40}}\right]^{0.22}$$

where

w' is specific tooth load on the operating pitch circle, N/mm;
v' is pitch line velocity, m/s;
ν_{40} is viscosity of lubricant at 40° C, mm^2/s (cSt);
C_w = 1.10 (conservative value);
C_w = 1.15 (nominal value);
C_w = 1.20 (maximum value).

Geometric function, F (geometric):

$$F_{(\text{geometric})} = \frac{(50 + z_1 + z_2)(a)^{0.5}}{A}[C_u]$$

where

z_1 is number of teeth of the pinion;

z_2 is number of teeth of the gear;

a is center distance, mm:

A is taken from Table 2

C_u is taken from Table 2

α	A	C_u at $1 \le u < 3$	C_u at $3 \le u \le 10$
15	350	$95 + 28.6\ (3 - u)$	$130 - 10\ [112.5 - (13 - u)^2]^{0.5}$
17.5	300	$90 + 30\ (3 - u)$	$120 - 10\ [90 - (12 - u)^2]^{0.5}$
20	300	$100 + 33.3\ (3 - u)$	$130 - 10\ [109 - (13 - u)^2]^{0.5}$
22.5	250	$95 + 28.5\ (3 - u)$	$130 - 10\ [112.5 - (13 - u)^2]^{0.5}$
25	250	$105 + 31.4\ (3 - u)$	$140 - 10\ [133.5 - (14 - u)^2]^{0.5}$
NOTE: α is normal pressure angle, degrees; u is gear ratio (z_2/z_1).			

Table 2: Values A and Cu as per AGMA 6011-J14 (2)

The exercise was done for 100% load case at nominal speed with respectively VG32 & VG15 load (Table 3 & 4).

F (geometric)		
z1	29	
z2	151	
a	812.8	mm
A	300	
u	5.2	
Cu	60.5	
F (geometry)	1322.9	

Table 3: Geometric function for tested gearbox

	F (LOAD)			
	VG32		VG15	
w'	986.39	N/mm	986.39	N/mm
v' (nominal)	107.1	m/s	107.1	m/s
Viscosity @ 40°C	32	mm²/s	15.23	mm²/s
Cw	1.1		1.1	
F (load)	3124.5		3678.9	

Table 4: Load function for tested gearbox with VG15 & VG32

AGMA 6011-J14 (2) method is called "simplified" which is as a matter of fact a conservative method mainly driven by the load, the pitch line velocity and geometry of the tooth. The surface roughness is not considered. The results demonstrate that the method shows a scuffing risk not acceptable even when using VG 32 while AGMA 925-A03 (5) show satisfactory results well proven by testing data (no visible damages caused by scuffing on the teeth). To summarize, AGMA 6011-J14 (2) method is not the right one to follow to confirm if the use of VG15 is acceptable or not.

Nevertheless, we could state that if AGMA 6011-J14 (2) criterion is respected scuffing risk could be well under control.

Conclusions

For the scuffing risk evaluation, the AGMA 6011-J14 (2) method is too conservative due to the simplified approach not based on the surface roughness which is clearly a key factor. Despite AGMA 925-A03 (5) not being a recognized standard yet it is suggested to use it as a guideline for scuffing risk evaluation.

Prior performing the endurance test on the gearbox and following AGMA 925-A03 (5) it was decided to limit the transmitted power to 70%. Indeed @ 70% load using VG15 the risk remains as low as @ 100% using VG32. The objective was to not reach the breaking point and damage the gears. Visual inspections were performed to validate this approach with satisfactory observations.

Nevertheless, to better evaluate the method a destructive endurance test should be performed playing on several parameters (surface roughness, load, sliding velocity, oil parameters…). This would be a good opportunity to improve the AGMA 925-A03 (5) method based on the empirical approach.

OIL REQUIREMENTS

It was demonstrated in Naldi et. al. (8) paper at ATPS2018 that VG15 oil kept its characteristics stable after an endurance test of almost 8000 hours (low acidity value in Figure 7, typical RPVOT trend, no wear metals detected, no varnish formation).
Based on this previous paper it is agreed/considered for the present document that VG15 performance over time remained stable for a gear application. This offers good perspective to not impact Mean Time Between Maintenance for oil (MTBM).

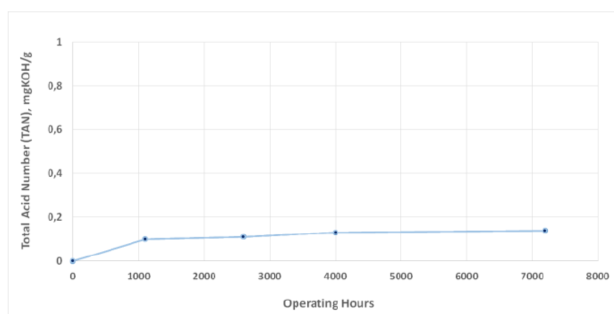

Figure 7: Total Acid Number evolution from Naldi and al. (8) endurance test

BEARING PERFORMANCE AND ROTOR-DYNAMIC

A rotor dynamic analysis of the gearbox was done with both types of oil (ISO VG32 & ISO VG15). Bearing coefficients were calculated with generic parameters for VG 32 and ENI parameters for VG15. Those calculations both predicted a limited impact in terms of critical displacement, level of stability and rotor vibrations.

The following diagrams show the comparative predicted response levels of the gearbox with the two different oils at probe location A & B (Figure 8, 9 & 10). A 10% increase for amplitude is expected and still within API limit. This is consistent with a loss of viscosity directly impacting the damping.

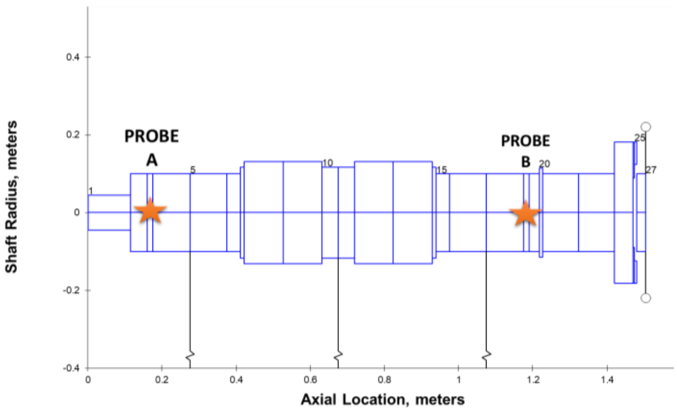

Figure 8: Probe A & B location

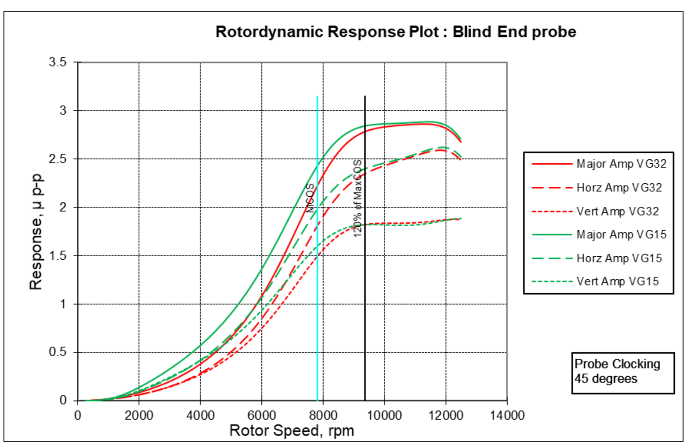

Figure 9: Probe A rotordynamic response plot

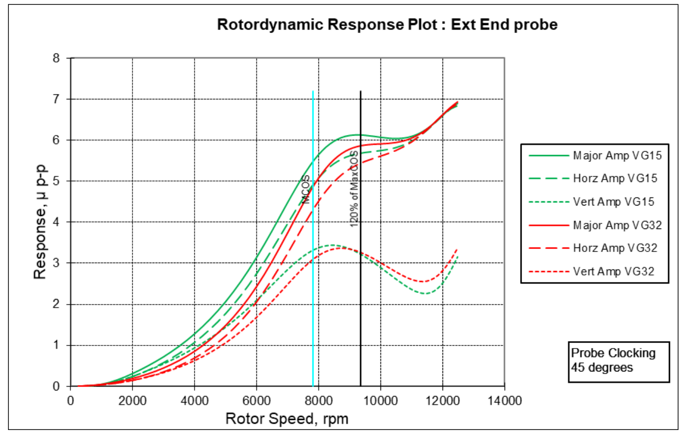

Figure 10: Probe B rotordynamic response plot

Based on direct shafts displacements, in opposition to expectations, amplitudes recorded were lower with VG15 on probe A (Figure 11 & 12) and B (Figure 13 & 14). For this particular case no negative impacts were observed using VG15. Direct values measured are summarize in table 5:

Test	Oil type	HS probe A Direct X vibration p-p, µm	HS probe A Direct Y vibration p-p, µm	HS probe B Direct X vibration p-p, µm	HS probe B Direct Y vibration p-p, µm
1	VG32	32	32	31	26
2	VG15	27	27	25	25

Table 5: Measured direct vibration with VG32 & VG15

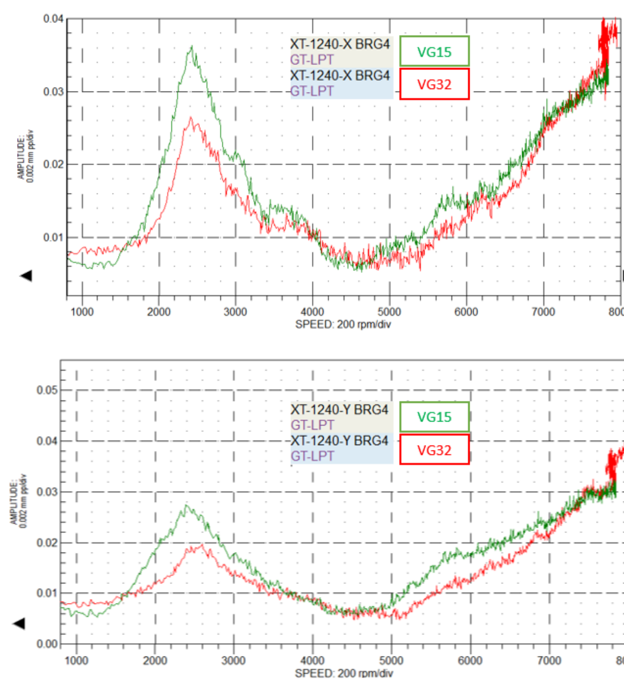

Figure 11: Probe A Bode Plot X & Y comparison VG32 and VG15

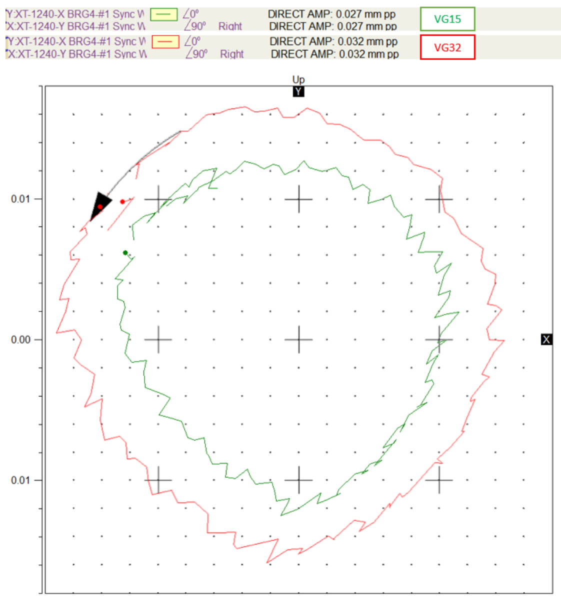

Figure 12: Probe A orbit comparison VG32 and VG15

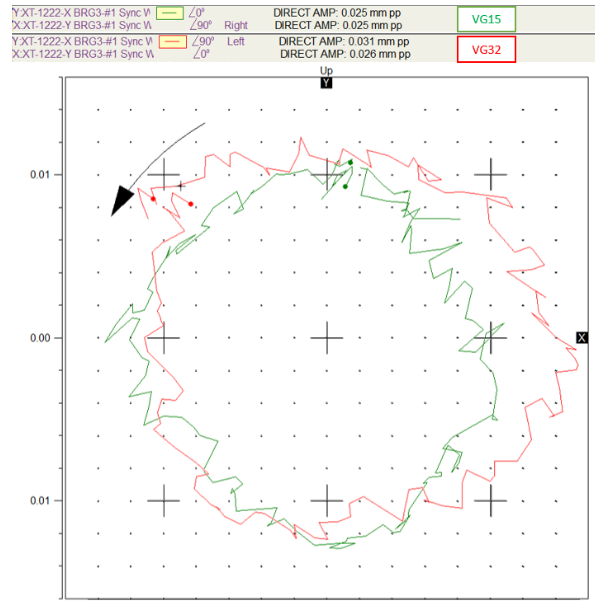

Figure 14: Probe B orbit comparison VG32 and VG15

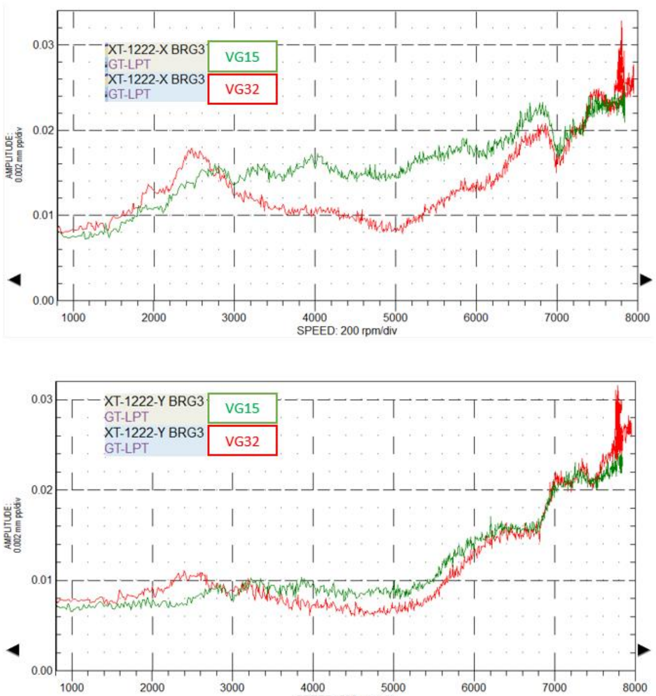

Figure 13: Probe B Bode Plot X &Y comparison VG32 and VG15

GEAR MESH POWER LOSSES

As said earlier in this paper, in high speed gearbox applications the windage losses are the most important one and make the friction losses negligible. A lower viscosity oil is expected to generate lower windage losses due to drag forces (pressure and viscous) being lower.

To evaluate the sensitivity of oil viscosity on windage losses a CFD analysis was carried out. The CFD model was developed within the ANSYS CFX Software, the computational domain is separated in three components: two rotating domains for each gear wheel and a stationary domain for the housing (see Figure 15). Several runs where performed to simulate oil and air mixture; all methods showed similar comparable results: oil viscosity has very limited impact on windage losses. This can be explained by computed Reynolds number being greater than 10^5, hence flow is fully turbulent and drag dependency from viscosity is low.

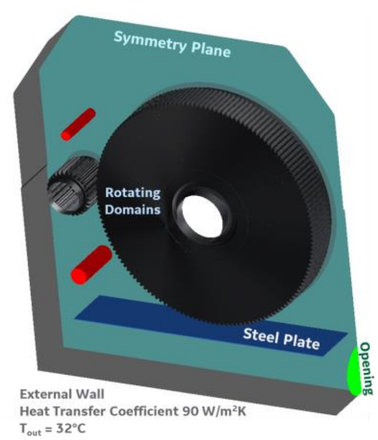

Figure 15: CFD symmetric boundary condition model

EXPERIMENTAL SET UP AND PROCEDURE

Experimental tests were carried out at BH facilities on two turbomachinery trains in loaded conditions, with Gas Turbine driven Centrifugal Compressor or Electric Generator.
These packages were equipped with BH Power Transmission parallel shafts gearboxes. Both the gearboxes use fixed profile bearings (HSS & LSS) and double combined thrust bearing on LSS. Mesh is lubricated by oil injection with calibrated spray nozzles. Gearbox data are listed in Table 6 below.

Rated power, kW	20500
Module, mm	8
Effective width, mm	420
Pitch line velocity, m/s	107
Ratio (Z1/Z2)	5.2 (29/151)
Teeth surface roughness, µm	Ra 0.4
Material	Case hardened
Teeth sliding velocity, m/s	20

Table 6: Gearbox data

The gearbox ran 8000 hours with VG 32 under 70% load and 2800 hours with VG 15 under 70% load (Table 7). The number of testing hours are enough to consider a fatigue condition according AGMA 2001 -D04 (9).

Test conditions	Oil	Power,MW	Hours of operation	Fatigue cycle number
1	VG32	15	8000	3.10^9
2	VG15	15	2800	10^9

Table 7: Test conditions

Gearbox was widely instrumented (Figure 16) with temperature sensors, pressure gauges for each bearing, proximity sensor for shafts displacements, temperature probes for oil inlet and outlet enabling detailed monitoring, data post-processing and comparison of ISO VG 32 and ISO VG 15 configurations. Torque-meter was used for measuring total power losses.

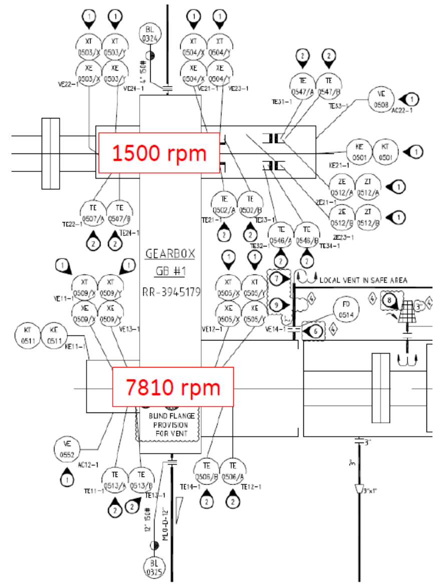

Figure 16: Gearbox instrumentation scheme

Both lube oil tanks have been filled with 10000 liters of ISO VG15 oil after an accurate cleaning to avoid contaminations.
For all critical parameters; analytical approaches were compared with testing results with appropriate assessments.

GEARBOX PERFORMANCES.

Bearings performances

Bearings characteristics are as below table 8:

	HS bearings	LS bearings
Type	Pressure dam	Pressure dam
Test load, MPa	1.8	1.1
Inlet temperature, °C	54°C	54°C
Design load, MPa	2.6	1.7
Journal velocity, m/s	82	23

Table 8: Tested gearbox bearing features

The calculation tool used to predict bearings performances is Lufkin internal tool based on extensive testing data.

Metal temperature

Predictions are slightly higher than the measured temperature which is conservative and a good practice for a design phase.
As shown in Figure 17, for HS bearings a reduction of 4°C was predicted while 3°C reduction was measured. For LS bearings a reduction of 7°C was predicted while 6°C reduction was measured.
The calculation tool was not originally developed for low viscosity oils but testing data showed good correlation with temperature reduction prediction.

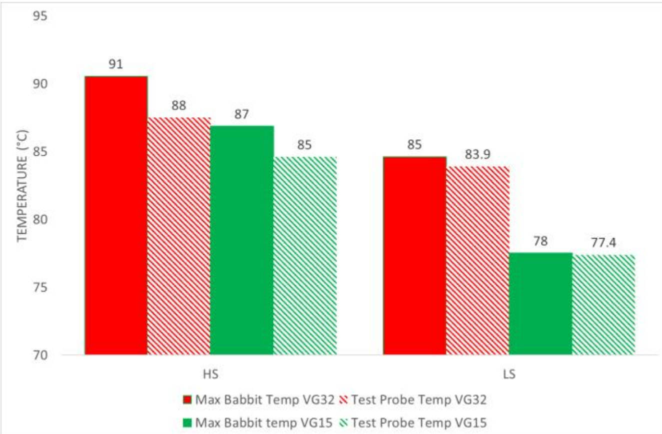

Figure 17: HS & LS bearing metal temperature predicted and measured

Oil film thickness

As shown in Figure 18, the calculation tool predicted a significant reduction of the oil film thickness with the use of ISO VG 15 (-15% HS bearings/ -23% LS bearings).

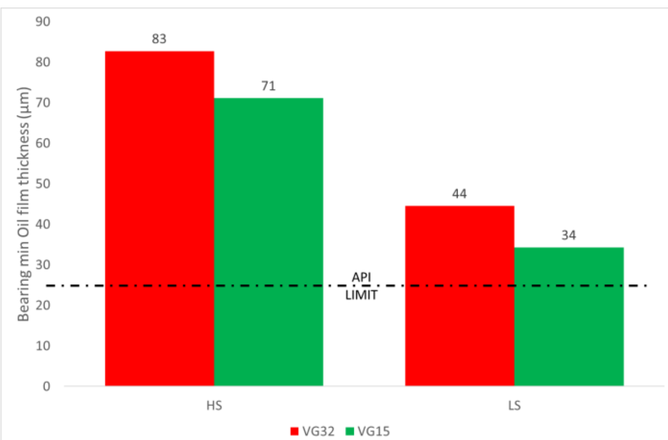

Figure 18: HS & LS bearings minimum oil film thickness predicted

Despite significant reduction; from analytical approach all bearings oil film thicknesses remain acceptable versus API limit 25µm.

It is really complex to characterize the effective residual film thickness in the bearings (not directly measurable). A deep analysis is required separately from this study to validate the expectations/predictions.

Based on the results obtained following endurance tests; we can only verify how we affect all parameters/behavior directly dependent to it (rotor dynamics, metal temperature, ….).

Based on shaft displacement (Table 5) and bearing metal temperature (Figure 17) measurement during VG15 and VG32 the minimum oil film thickness should not be as reduced as predicted.

Oil flow consumption and power losses (efficiency)

Due to time and environment constraints, the test configuration did not allow to precisely isolate each consumer (bearings drain temperature and oil flows, teeth bulk temperature…). However, we were able to collect enough data for an overall effect (Figure 19 & 20).

Observations on testing results:

- VG32 calculations are in line with experimentation (within 5% standard error).

- By introducing VG15 some discrepancies appeared. Oil flow and power losses are both underestimated.
  Mesh flow is well under control because calibrated by spray nozzles (mass flow dependant to the density).
  Saying that and based on oil viscosity sensitivity analysis on windage losses (CFD analysis) the excess of flow and power losses measured cannot be only attributed to the mesh but most likely to HS bearings which are the main consumers (60% of the total losses).

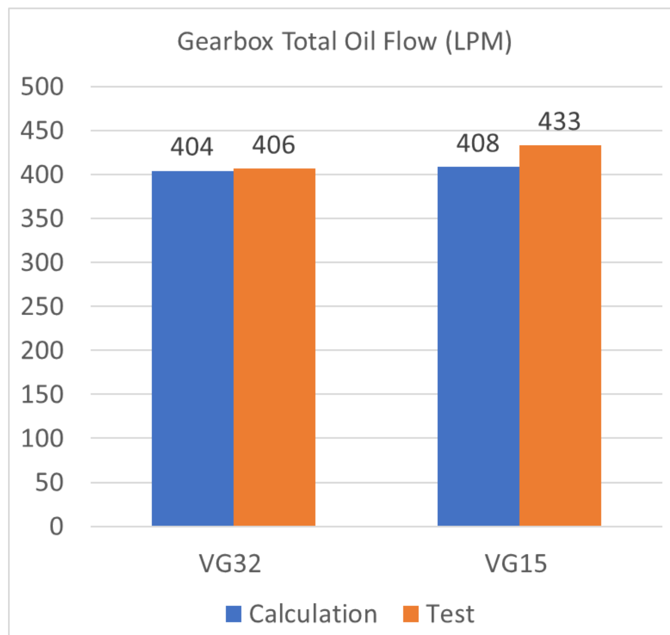

Figure 19 : Gearbox total oil flow predicted and measured with VG15 & VG32

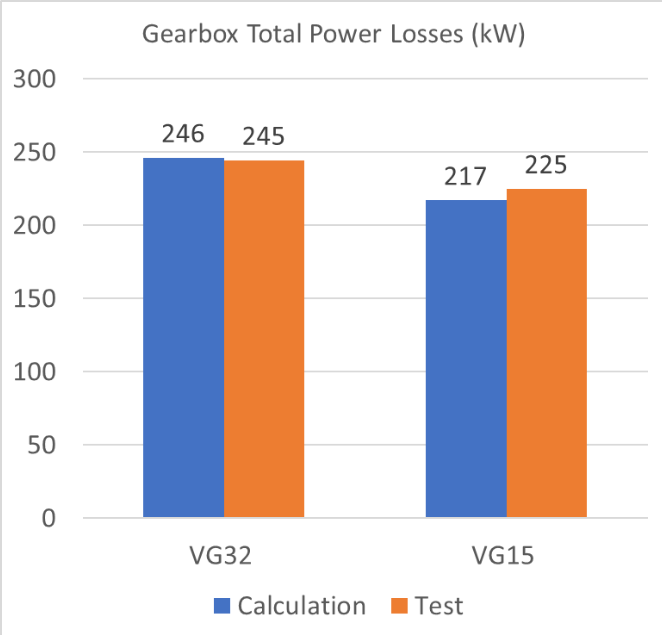

Figure 20 : Gearbox total power losses predicted and measured with VG15 & VG32

Using VG15 shows an oil flow increase (~ +6%) and a reduction of losses (~ -9%). Trends were well predicted by calculation tools but not the amplitudes (~ 6% / 4% of error).
Now, it could be interesting to quantify from where the benefits come from, with more detailed and dedicated tests for each consumer to optimize calculations tools (Air/Oil mixture density with VG15).

Nevertheless, it is important to note that the bearings were designed and optimized for VG 32 use. In case of use of VG15; based on analytical predictions we can expect some additional benefits by reducing the clearance.
This could be a contributor to consider in discrepancies we observed.

VG15 design consideration perspective

In the aim to reduce carbon footprint and improve the efficiency for gearboxes the first step was to focus on the global effect caused using low viscosity oil. The results obtained are really encouraging without any obvious criticalities and quite significant improvements. Nevertheless, deep analysis still needs to be performed to better understand and characterize the observations on bearings and gear mesh.

Gear teeth status inspections

A visual inspections of gear teeth has been done after 2800 hours with VG15 at 70 % load. As shown in Figure 19 & 20 no surface distresses were found on flanks after the test.

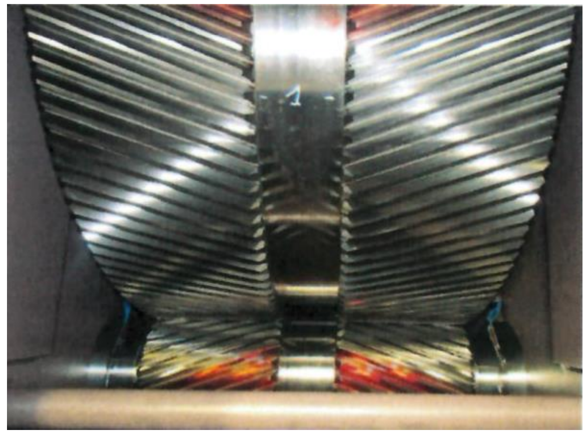

Figure 21: Teeth visual inspection before VG15 test

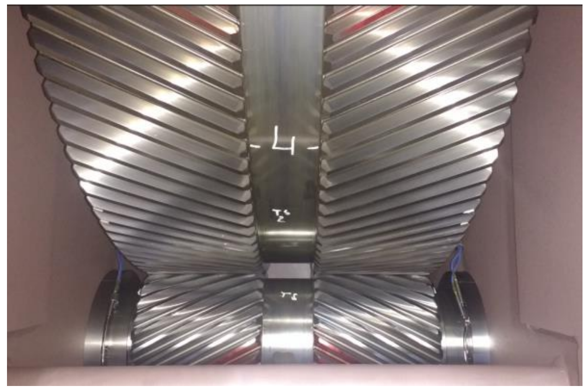

Figure 22: Teeth visual inspection after VG15 test

CONCLUSIONS

The studies and analyses carried out by the authors, together with the experimental validation through test data are demonstrating that it is possible to use low viscosity oils in the geared transmission equipment, but of course this option will require special considerations and specific design features to not jeopardize equipment reliability and performances.

Phenomena related to gear teeth surface distresses can be properly addressed using appropriate Scuffing assessment rules, and adopting the best design compromise on the following parameters:

- Tooth geometry;
- Tooth surface roughness;
- Extension of the tooth contact pattern;

The rotor-dynamics do not look compromised by the adoption of low viscosity oils, moreover the available software's and practices, largely used in the industry, like for instance that one relevant to the minimum oil film thickness, remain valid tools to be used during the design phase.

The gear mesh power losses do not change significantly with the low viscosity oil, thus not providing a limiting factor in their adoption. The gear mesh flow is controlled by the spray-nozzles size and lube oil pressure, so it doesn't represent a challenge.

About the bearing's performances, the authors noted a good correlation between the theoretical and experimental data in terms of operating temperature, with a general improvement with the VG15.
Similar story about the bearings power losses, which have been observed to be expectedly smaller with the VG15 oil, even if the dependence of this gain by the bearing flow is strongly suggesting to carry out specific bearings test, on dedicated test rig, to farther characterize these components

NOMENCLATURE

α	= Normal pressure angle	(°)
a	= Center distance	(mm)
A	= Geometric factor	
Cu	= Ratio facto	
Cw	= Load factor	
u	= Gear ratio	
v'	= Pitch line velocity	(m/s)
w'	= Specific tooth load on the operating pitch line	(N/mm)
$\upsilon 40$	= Viscosity of lubricant at 40°C	(Cst)
z_1	= Number of teeth of the pinion	
z_2	= Number of teeth of the gear	

ACRONYMS

CFD	= Computational Fluid Dynamics
CH4	= Methane
CO2	= Carbon dioxyde
FSFL	= Full Speed Full Load
GWP100	= 100-year global warming potential
HS	= High Speed
HSS	= High Speed Shaft
LS	= Low Speed
LSS	= Low Speed Shaft
LPM	= Liter per minute
MTBM	= Mean Time Between Maintenance
RPVOT	= Rotating Pressure Vessel Oxidation Test
TAN	= Total Acid Number
VG	= Viscosity Grade

REFERENCES

(1) Wright, Kemp, and Williams, 2011, Carbon footprinting: Towards a universally accepted definition.
(2) AGMA 6011-J14, 2014, Specification for High Speed Helical Gear Units.
(3) API standard 613, fifth edition, 2013, Special Purpose Gear Units for Petroleum, Chemical and Gas Industry Services.
(4) AGMA 217.01, 1965, Information sheet Gear Scoring Design Guide for Aerospace Spur and Helical Power Gears.
(5) AGMA 925-A03, 2013, Effect of lubrication on gear surface distress
(6) ISO/TS 6336-20, 2017, Calculation of scuffing load capacity – Flash temperature method
(7) AGMA 9005-E08, 2008, Industrial Gear Lubrication.
(8) Naldi, L., Toscanini, M., Camatti, M. and Lattuada, 2018, Development, testing and qualification of innovative low viscosity oil in turbomachinery applications, @ Asia Turbomachinery & Pump Symposium 2018. doi:1969.1/172515
(9) AGMA 2001-D04, 2004, Fundamental Rating Factors and Calculation Methods for Involute Spur and Helical Gear Teeth
(10) ANSI/AGMA 1010-E95, 1995, Appearance of Gear Teeth - Terminology of Wear and Failure
(11) ISO 10825, 1995, Gears -- wear and damage to gear teeth – terminology
(12) Wink, 2012, Predicted Scuffing Risk to Spur and Helical Gears in Commercial Vehicle Transmissions

ACKNOWLEDGEMENTS

The authors would like to acknowledge the important work done by T. Shoup and M. Libraschi from Baker Hughes, a GE company for supporting the entire validation phase.
The authors would also like to thank Baker Hughes, a GE company and ENI for permission to publish this work.

