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# DEVELOPMENT, TESTING AND QUALIFICATION OF INNOVATIVE LOW VISCOSITY OIL IN TURBOMACHINERY APPLICATIONS

**Lorenzo Naldi**
Consulting Engineer - Rotordynamic
Baker Hughes, a GE company
Florence, Italy

**Massimo Camatti**
Chief Consulting Engineer
Baker Hughes, a GE company
Florence, Italy

**Manuela Toscanini**
Technical Leader - Industrial Lubricants
Eni SpA- Research & Technological Innovation Department
San Donato Milanese (MI) - Italy

**Marco Lattuada**
R&D Technologist – Industrial Lubricants
Eni SpA- Research & Technological Innovation Department
San Donato Milanese (MI) - Italy

*Lorenzo Naldi is currently a Consulting Engineer at Baker Hughes, a GE company. He joined GE in 2002 as Gas Turbine Design Engineer after an experience in bio-medical industry. He has 15 years of experience in Rotordynamic and vibration data analysis. He covered the role of Engineer Manager of Rotordynamic Team for 7 years.*
*Mr. Naldi received a B.S. degree in Mechanical Engineering from University of Florence in 2000.*

*Manuela Toscanini is Technical Leader for Industrial Lubricants and Technical Assistance at the Research & Technological Innovation Department of Eni. She received a M.S. degree in Chemistry from University of Pavia and joined Eni in 1989 as a researcher in special products development (metalworking fluids, protective oils, engine coolants.). For more than ten years she has been involved in industrial lubricants area.*

*Marco Lattuada received a M.S. degree in Chemical Engineering from University of Milan in 2010. In 2011, he started working in Research & Technological Innovation Department of Eni dealing with automotive and marine lubricants development. Now is currently working as a researcher in industrial lubricants area.*

*Massimo Camatti is currently a Chief Consulting Engineer at Baker Hughes, a GE company. He has 30 years of experience on turbomachinery design since he joined Nuovo Pignone in 1988 as centrifugal compressors design engineer. He covered several roles within GE Oil & Gas engineering dpt., in particular in design and R&D areas, having the opportunity to be involved in most of the technical aspects and disciplines related to turbomachinery systems technology. Massimo received a Master Degree in Mechanical Engineering from University of Florence in 1986.*

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## ABSTRACT

The continued market demand to have more efficient machines, thus achieving energy savings and emission reductions, has motivated manufacturers to look for improvements even in area that have never been explored and evaluated till now.
In this context, the authors focus the research on a high performance and high efficiency lubricant for Turbomachinery applications.
New product has been formulated by a theoretical optimization of power loss reduction limiting the impact on the rotordynamic behavior. The paper reports the entire qualification process of the oil, starting from laboratory tests to certify its chemical properties, then goes through dynamic bearing test rig for the identification of stiffness and damping characteristics to finally present the results of tests on actual machinery (gas turbine, gear box and centrifugal compressor) under real operating conditions in terms of mechanical friction losses, bearing temperatures and rotordynamic behavior.

## INTRODUCTION

Since many years the viscosity of lubricating oil for industrial turbomachinery application is a pre-determined parameter and basically an industry standard set on two reference values: 32cSt and 46CSt at 40 deg. C.
Keeping oil viscosity unchanged, OEMs have been developing their products and technologies achieving significant improvements in terms of machinery performance and power density. Oil Companies at the same time have made available improved quality lubrication products with enhanced properties as far as concern aging resistance and anti-wear characteristics.
The authors' Companies started few years ago (2010) to brainstorm about the opportunity to join their respective competencies and efforts to develop an innovative oil which could contribute to enhance the turbomachinery technology and related industry.
After extensive discussions which evaluated several potential areas of work, the decision was taken to approach the study of a low viscosity oil (15-22 CSt. @ 40 deg. C) having in mind the following potential target benefits:

- Increase the overall machinery efficiency by reducing the mechanical viscous losses and relevant heat generation
- Improve bearings lubrication behavior and minimize pad temperature at very high shaft sliding speed of advanced high-performance turbomachinery
- Reduce the oil flow requirement and optimize design of lube oil systems

Both Companies were conscious of the technical and somehow even psychological challenges of such initiative since, as above said, oil viscosity was, till that time, a fixed parameter never affected by design and engineering activities. On these basis, aiming to build a rigorous and reliable process, it was defined a development plan which combined the theoretical and analytical study with an extensive experimental activity based on incremental approach from component to machinery system level.
This paper provides an overview of the activities performed and the achieved results.

## OIL FORMULATION AND LABORATORY TESTING

Since the beginning it was realized that the development of such innovative oil required the evaluation of the relationships among main oil characteristic parameters and the outcome performance expressed in terms of power losses and rotordynamic parameters. Therefore the methodology adopted was a Design of Experiment approach where viscosity, viscosity index, density and specific heat have been varied as input data and bearings power losses, effective stiffness and damping have been selected as output functions.
From a formulation standpoint, a synthetic baseblend with suitable physical-chemical characteristics has been selected according to the model proposed (1) and combined with an innovative new additive system that is able to assure the excellent performances above mentioned.
Among several oil candidates formulated for the scope, two low viscosity oils, viscosity grade (VG) 15 and VG 22, have been developed for the laboratory evaluation.
Based on predicted bearing dynamic coefficients with selected oil viscosities, rotordynamic analysis of several turbomachinery models have been performed showing limited impact with respect to conventional oil viscosity.
The properties of the two oils have been proved in standard laboratory tests according to international standards which are described in detail below:

<u>OXIDATION TEST (method ASTM D 943)</u>
The number of hours needed to reach an acid number of the oil equal to 2.0 mg KOH/g is considered to be the "oxidation lifetime "of the oils. This parameter is an indicator of oil resistance to high temperature conditions and aging. The new oils (ISO VG 15 and ISO VG 22) presented a value higher than 9000 hours that is significantly better than the ones referred to typical good quality turbomachinery oils.

<u>THERMAL STABILITY TEST (Cincinnati - method ASTM D 2070)</u>
This test method evaluates the thermal stability of oils in the presence of copper and steel at 135°C. The specimens have been substituted by GE pad in order to check oil/metal compatibility. No color change and sludge occurred (figure 1).

<u>RPVOT (method ASTM D 2272/B)</u>
This test measures the oxidation stability of oils in the presence of water and copper at 150°C. The number of minutes required to reach a specific drop in gage pressure is the oxidation stability of the test sample. The oils presented outstanding characteristics if compared to typical good quality turbomachinery oils.

<u>OXIDATION TEST (method ASTM D 2893/B)</u>
This test covers the determination of the oxidation characteristics of the oils, at 121°C in presence of air, based on its viscosity increase. Both the oils presented very low values referred to this parameter.

<u>FZG A/8.3/90 (method ISO 14635-1)</u>
This is a widely used test method for the evaluation of the scuffing properties of industrial lubricants.

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ISO VG 15 and ISO VG 22 also demonstrate very good anti-wear and EP properties in FZG test achieving a value equal to 10.

This result indicates the oil suitable for application on typical gear boxes used in turbomachinery trains, that require FZG higher than 7. It is moreover particularly relevant considering that low viscosity could have potentially impacted negatively the anti-scuffing properties.

| | | **ISO VG 15** | **ISO VG 22** | **ISO VG 32** |
|---|---|---|---|---|
| **Property** | **Test Method** | **Results** | **Results** | **Typical Values** |
| | | | | |
| Kinematic Viscosity @ 40°C, mm2/s | ASTM D 445 | 15,19 | 21,82 | 28.8 - 35.2* |
| Kinematic Viscosity @ 100°C, mm2/s | ASTM D 445 | 3,646 | 4,576 | - |
| Viscosity index, min. | ASTM D 2270 | 127 | 127 | 90 min* |
| Flash point, °C, min. | ASTMD 92 | 230 | 226 | 180 min* |
| Bulk fluid dynamic viscosity@cold start-up, temperature for 150'000 mPa.S | ASTM D 2983 | <-40°C | <-40°C | to be reported* |
| Total Acid Number, mgKOH/g | ASTM D 664 | 0,14 | 0,16 | <0,2 |
| RPVOT, min | ASTM D 2272/B | 2500 | 1800 | > 500 |
| Resistance to aging @ 121°C - max. % increase in kinematic viscosity @ 100°C | ASTM D 2893/B | 1,0 | 0,5 | 6 max* |
| Oxidation test @ 95°C - time to reach TAN = 2 mgKOH/g | ASTM D 943 | > 9000 | > 9000 | > 5000 |
| Water separability, min(ml) | ASTM D 1401 | 10,0 | 20 | 30 max |
| Air Release, min | ASTM D D3427 | 1,0 | 2,2 | 4 max |
| Foam suppression - Volume of foam (mL), max after: | ASTM D 892 | | | |
| Seq.I 24°C | | 50/0 | 50/0 | 50/0* |
| Seq.II 93.5°C | | 30/0 | 10/0 | 50/0 |
| Seq. III 24°C | | 50/0 | 50/0 | 50/0 |
| Water separation | ASTM D 2711 | | | |
| % $H_2O$ in oil after 5h test | | 0,1 | 0,05 | 2,0 max * |
| Cuff after centrifuging, ml | | 0,6 | 0,05 | 1,0 max |
| Total free H2O collected during entire test, starting with 90ml H2O, ml | | 85,4 | 88,6 | 80,0 min |
| Rust prevention, 24h | ASTM D 665/B | Pass | Pass | Pass* |
| Copper corrosion prevention, 3 h @ 100°C, | ASTM D 130 | 1A | 1A | 1b max* |
| Scuffing load capacity, FZG visual method, A/8.3/90, fail stage, min. | ISO 14635- 1 | 10 | 10 | 10* |

*ANSI AGMA 9005-E02

*Table 1: OTE GT 15 – OTE GT 22 characteristics*

| | | **OTE GT 22** | **Cincinnati Machine P-68** |
|---|---|---|---|
| **Property** | **Test Method** | **Results** | **Minimum performance requirements** |
| Thermal Stability@168h, 135°C | Cincinnati Milacron - ASTM D 2070 | | |
| Kinematic Viscosity, increase % | ASTM D 445 | 2,327 | 5 max |
| NN change, % | ASTM D 664 | -46,67 | ± 50 |
| Sludge, mg/100 ml | - | 6,7 | 25 max |
| Pad apparance, merit | Visual | no discolor | no discolor |

*Table 2: OTE GT 22 Cincinnati thermal stability test*

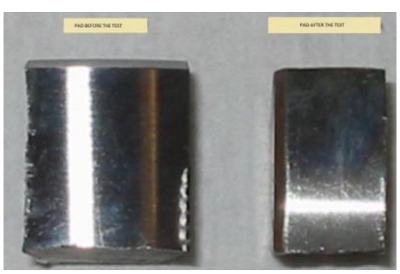

*Figure 1: OTE GT 22 Cincinnati pad*

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**Journal Bearing Dynamic Test**

The choice to select low viscosity lubricant, if from one side is an inevitable choice to reduce viscous losses, on the other hand it introduces a series of uncertainties about the impact it might have on the dynamics of turbomachine. After completing all the laboratory tests to certify the performance of the lubricant, the validation process for rotating machines has begun. The first step was to characterize from a dynamic standpoint a tilting pad journal bearing with the two oils, ISO VG 15 & 22. The tests were performed at the bearing test rig, widely used for the dynamic characterization of all the bearings adopted by authors' Company, where the main measured parameters are oil flow and temperature, stiffness and damping, metal temperatures and power loss.
In the figure 2 the scheme of the test rig arrangement.

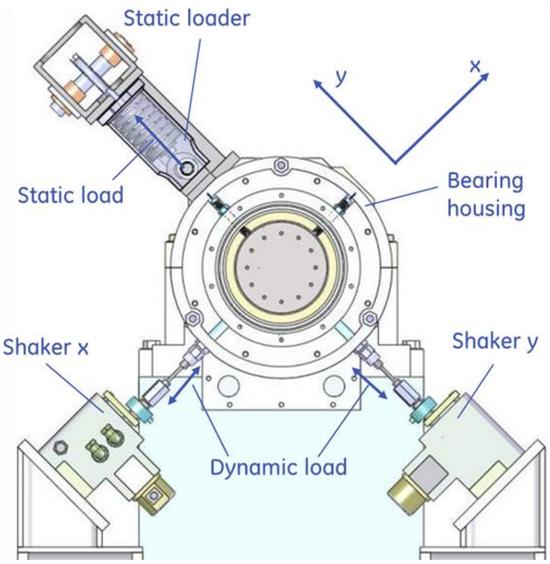


*Figure2: schematic view of the dynamic bearing test rig*

- D =110 mm
- Number of pads= 5 (Load Between Pads)
- L/D =0.4
- Offset = 60%
- Pad Clearance (avg) = 0.345mm
- Preload (avg) =0.435
- Pad arc= 54 deg
- Pad thickness (avg) = 19.61 mm
- Pivot single radius

*Figure 3: Journal bearing used in the dynamic test rig*

In figure 3, geometrical characteristics of the five pads, direct lubrication, bearing adopted for test are provided
All test was performed under the same conditions of clearance, load, speed, flow, and oil temperature.
The figures 4,5,6 and 7 show power loss, dynamic coefficients and bearing temperature for the three-different oil viscosity versus bearing sliding speed.


*Figure 4: Power losses comparison*

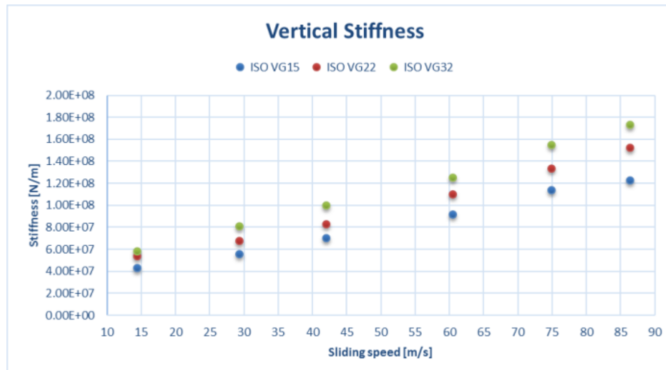


*Figure 5: Direct Stiffness coefficient*

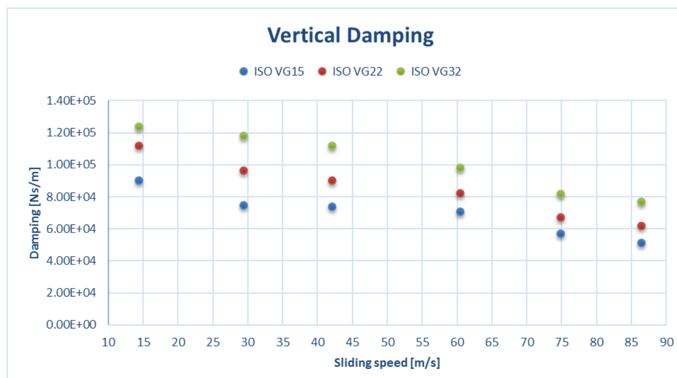


*Figure 6: Direct damping coefficient*

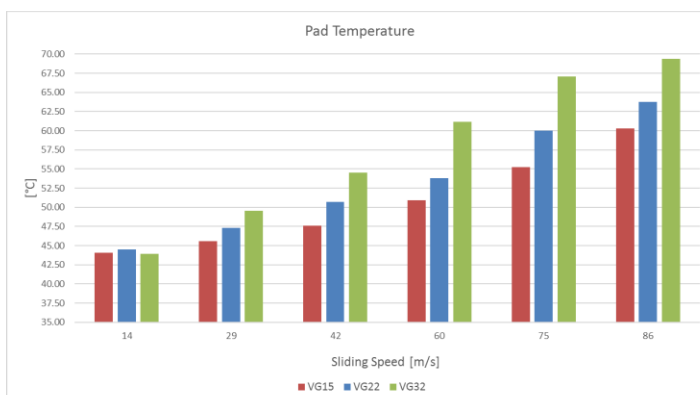


*Figure 7: bearing metal temperature with 3 different oils*

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The positive outcome of the tests convinced the authors to proceed with a further validation step, aiming to achieve a higher Technology Readiness Level from laboratory/ test rig experience to Full-scale machine tested in a relevant environment.
After careful assessment of the available options, the choice fell on a refinery plant in Italy with a steam turbine directly connected to a centrifugal compressor. In order to ensure a thorough evaluation of oil performance, in particular mechanical losses, additional instrumentation were installed, compatible with the availability of a fully operational and in-service machine.

Data were collected by using a measuring system described as follows:

- all pipes coming from the oil tank and feeding each bearing (journal and thrust) were instrumented with pressure transducer, thermo-resistances and ultrasonic liquid flowmeter
- thermo-resistances were added at the outlet oil side

In addition, standard vibration instrumentation and pad thermocouples were available to monitor rotordynamic behavior and metal temperature

The estimated viscous power losses were calculated through the measured parameters according to the following relationship:

$$P_{Loss} = Q \cdot C_p \cdot \Delta T \ [W]$$

Where:

$Q \ [\frac{kg}{s}]$ is the average value of oil flow

$C_P \ [\frac{J}{kg\,°C}]$ is the heat capacity at the related temperature

$\Delta T = T_{OUT} - T_{IN} [°C]$ is the temperature difference between oil bearing inlet and outlet.

Test was performed in different step, starting with the "original" lubricant (ISO VG 48) and doing partial dilution with ISO VG15 oil, with the goal to map performance vs different viscosity grade.

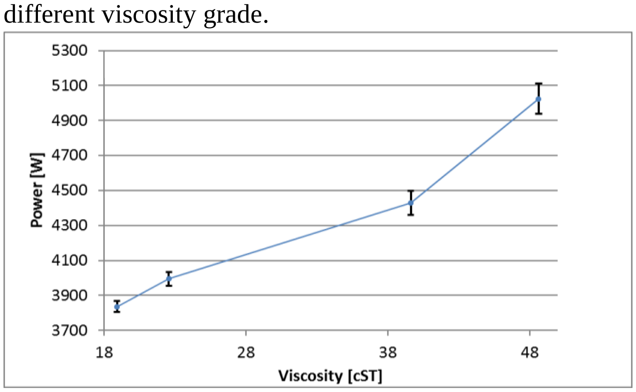


*Figure 8 – Absorbed power variation against oil's viscosity (referred to 40°C)*

The test outcomes were aligned with the results obtained at the bearing test rig, confirming the expected relationship between viscosity reduction and mechanical losses saving, and leaving the turbomachinery's rotordynamic behavior unchanged. The bearing metal temperatures, although constantly monitored (a reduction of few degrees °C was appreciated) were not systematically recorded and therefore a careful evaluation of thermal behavior was postponed to successive tests.

**Endurance test**

All tests done have shown very positive results, however the capability of the new VG15 oil to maintain its characteristic and performance over time required to be validated by a dedicated test.
To cover this topic, it was selected a turbogenerator to perform a long run with new oil. VG 32 oil charge was replaced by low viscosity oil and the machine run has been monitored at least for 8'000 hours. (new low viscosity oil is still working and experiencing additional running hours).
Periodically oil samples have been taken and tested to check oil conditions.
The outcome of the laboratory analysis indicates that physical-chemical characteristics are stable maintaining high-performance level (see Figures 9&10).
No wear metals have been detected, the RPVOT shows a typical trend and the neutralization properties of the oil are attested by very low acidity value (TAN).
Indicative evidences of no varnish formation have been obtained by membrane colorimetry patch test (internal method).

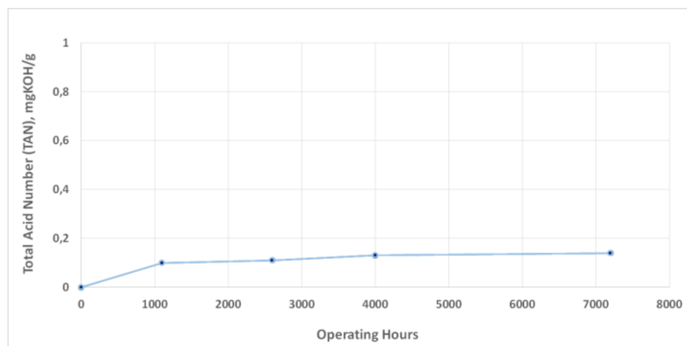


*Figure 9: Low viscosity oil - Total Acid Number*

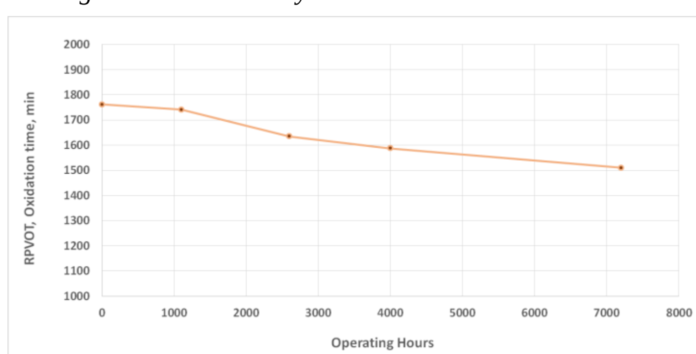


*Figure 10: Low viscosity oil - RPVOT*

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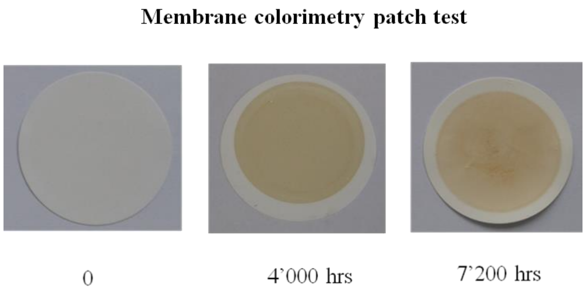


*Figure 11: Low viscosity oil – Patch Test*

**High speed Centrifugal Compressor and Gas Turbine NovaLT™16 Test**

After the positive conclusion of test on steam turbine-compressor train, the project review board decided to continue the validation with ISO VG15 oil on gear box, high speed/high performance centrifugal compressor and latest generation of two shaft gas turbines to check the following:

- anti-scuffing phenomena on gear box teeth
- rotordynamic stability on centrifugal compressor
- thermal behavior of gas generator "hot" bearing

The tests were organized at the authors' facility on two different trains - using Centrifugal Compressor Test Vehicle and the prototype of the new Gas Turbine NovaLT™16. Both configurations were equipped with parallel axes speed increase gear box

**Experimental set up and procedure**

All test machines were widely instrumented, enabling detailed monitoring, data post-processing and comparison of ISO VG 32 and ISO VG 15 configurations.

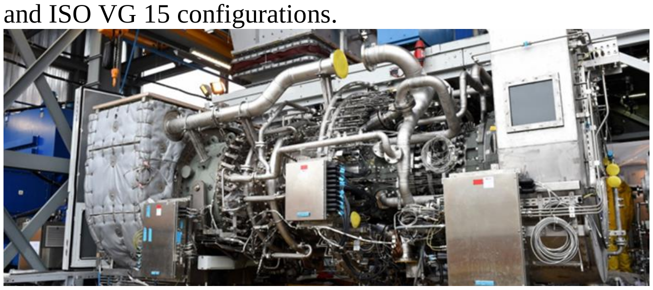

*Figure 12: Nova LT16*

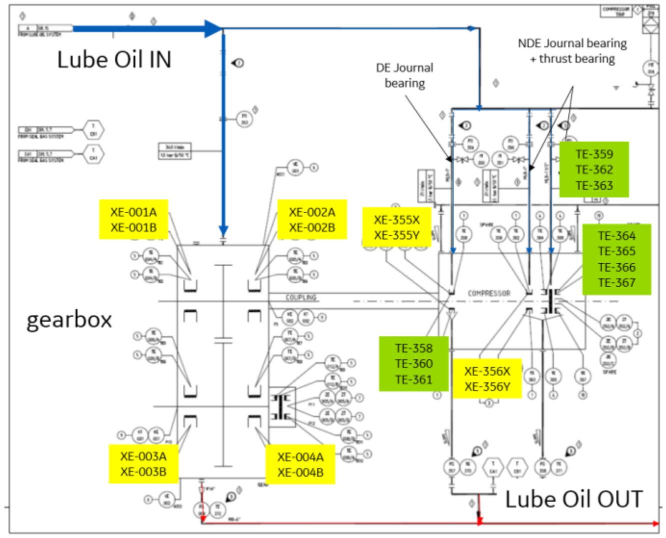


*Figure 13: Compressor and gear box instrumentation scheme*

Both centrifugal compressor and gas turbine trains were tested on a wide envelope of operating conditions with full characterization of all main parameters.
Both lube oil tanks have been filled with 10000 liters of ISO VG15 oil after an accurate cleaning to avoid contaminations.

**Results**

In this section, a summary of the main results obtained from the comparison between the use of a low viscosity oil and one with standard grade are presented.
As expected, and already observed in previous tests, a mechanical losses reduction is observed as shown in Figure 14 and 15

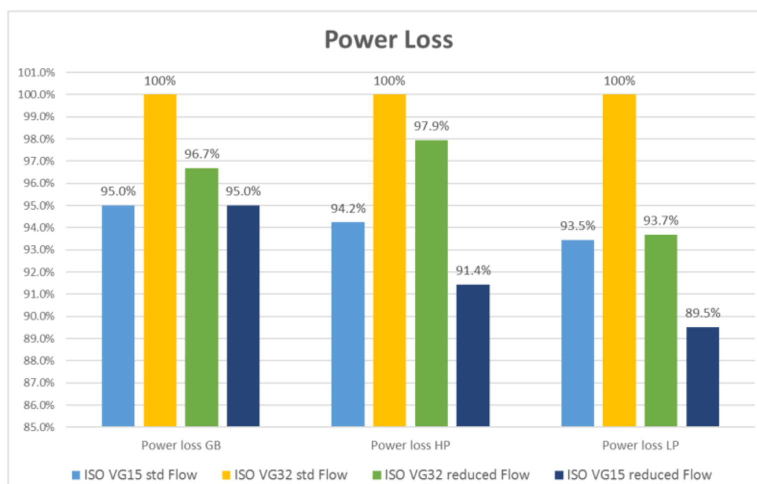


*Figure 14: mechanical power loss NovaLT16 Gas Turbine*

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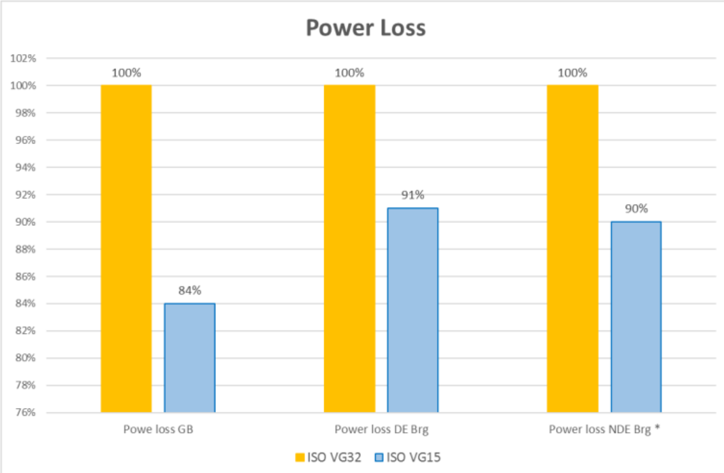


*Figure 15: mechanical power loss Compressor test vehicle*

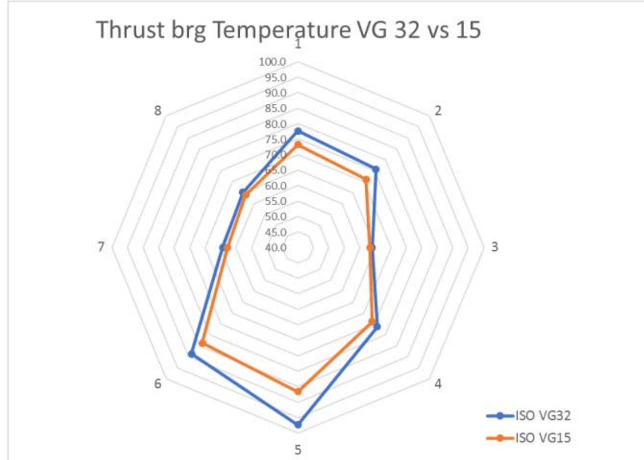


*Figure 17: Thrust bearing temperature*

The two experiences have highlighted that mechanical power loss reduction varies in a range from 5% to 15%; the authors deem that this variability is related to the type of bearing and associated oil flow.

Additional advantage of the VG15 oil is an average reduction of 5-6 °C of metal bearing temperatures up to 10 ° C for the hottest bearing. A sensitivity has also been made to verify that even under low flow conditions similar pattern is kept.

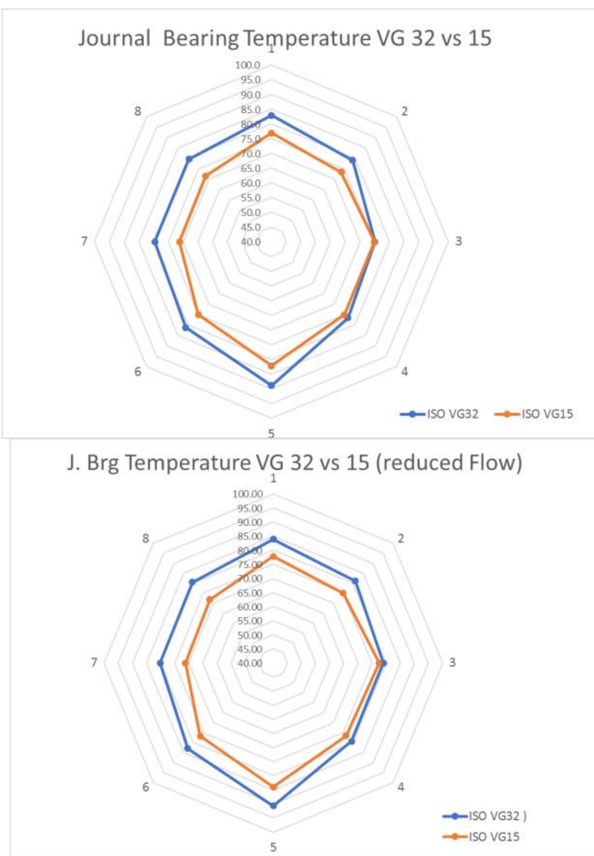


*Figure 16: Journal bearing temperature*

While the measured direct relationship between oil viscosity and mechanical losses reduction was easily expectable, not so obvious was the impacts on the machine rotordynamic behavior.

Bearing dynamic tests had already shown that ISO VG 15 oil present a reduction of stiffness and damping in the range of 20-30%, which is an effect comparable to an enlargement of assembly bearing clearances from typical minimum to maximum values. On this basis, rotordynamic analysis of the two test machines was repeated, predicting a limited impact in terms of critical displacement, level of stability and rotor vibrations.

The following bode diagrams show the comparative vibration levels of the Power Turbine rotor with the two different oils

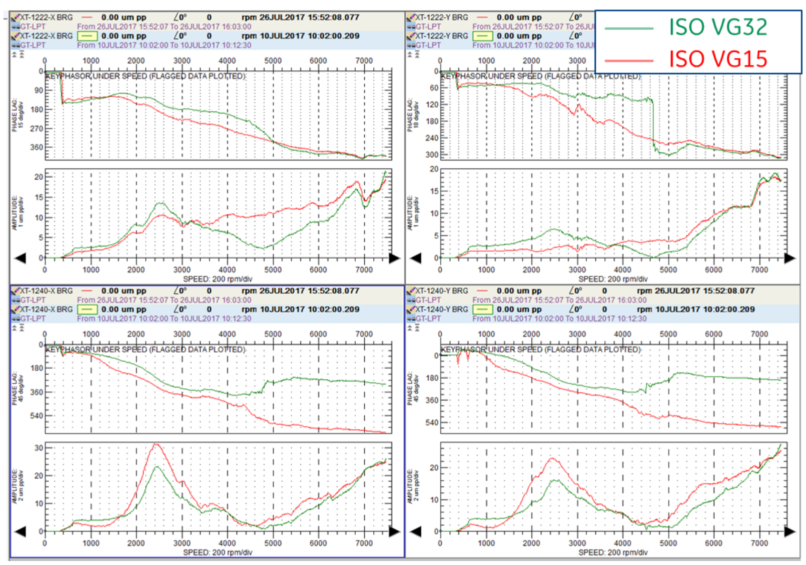


*Figure 18: LT16 PT Bode diagram*

The synchronous frequency and amplification factor are easily derived by the comparison of the startup Bode Diagram, therefore plots in fig. 18 are representative of a substantially similar behavior as expected.

Additionally, in order to assess the stability at the operating speed, in particular for centrifugal compressor, a different methodology has been adopted: Operational Modal Analysis is

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an identification method, originally developed for analysis of structure dynamics but nowadays adopted also for rotordynamic assessment. The method is able to detect natural frequencies by the analysis of vibration row data measured at steady state condition.
Figure 19 depicts the OMA results of the centrifugal compressor in the most severe operating conditions; the first mode has a limited shift with lower viscosity oil and an increase of logarithmic decrement, well in line with the dynamic coefficient identified on bearing test rig.

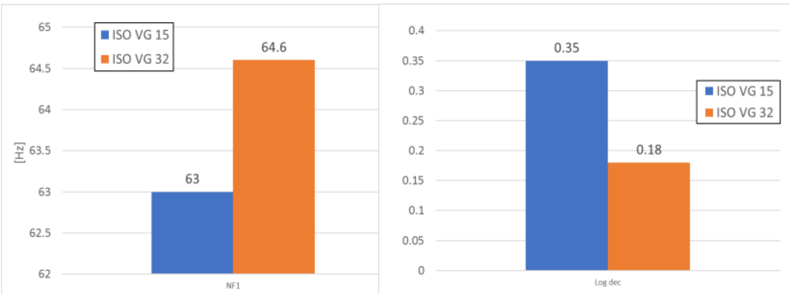


*Figure 19: 1°Natural frequency and relevant log dec assessment by OMA*

## CONCLUSIONS

Starting from the idea that lubrication oil is actually an important element of a turbomachinery system and therefore it might contribute to provide more efficient products, a large activity has been carried out to identify the formulation of a new high-performance lubricant to maximize energy saving.
The new product, characterized by a 15cSt viscosity grade, has been subjected to a wide range of tests to evaluate its characteristics and impact on turbomachinery performance: validation was initiated with laboratory tests and proceeded by running representative models of the turbomachinery families under typical operating conditions.
The entire validation process has been completed successfully and the main outcome can be summarized as follows:

- Low viscosity oil has been tested (still running) for more than 1 year keeping stable physical-chemical characteristics and high-performance behavior
- Proven reduction of mechanical viscous losses of lubricated rotating parts in a range of 5-15%, with respect to a standard ISO VG 32 oil;
- Smooth running behaviour and operating parameters of machinery and related auxiliary equipment in line with those recorded with a standard ISO VG 32 oil

The VG15 oil has been approved for deployment on actual installations (TRL9) by authors' Companies.

## NOMENCLATURE

| | | |
|---|---|---|
| P | = power loss | (W) |
| *Q* | = flow | (Kg/s) |
| *Cp* | = heat capacity | (J/(Kg*s)) |
| *T* | = Temperature | (°C) |

## ACRONYMS

RPVOT= Rotating Pressure Vessel Oxidation Test
FZG=Forschungsstelle fur Zahnrader und Getriebebau
OMA= Operational Modal Analysis
VG = Viscous grade
GT = Gas Turbine
CC = Centrifugal Compressor

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