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Reducing CO2 impact through fuel economy: developing a two-stroke FE system oil – results, next steps

Lubricants

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ABSTRACT

The maritime industry is facing an urgent challenge: how to reduce its environmental impact and to manage the transition towards a sustainable future?

The 2023 IMO revised strategy gives new and more stringent milestones to reduce greenhouse gases emissions from ships. To be in line with IMO's revised GHG strategy, the emission pathway must decrease very rapidly until net-zero in 2050. That means that from 2030 the decrease must accelerate and be about twice as fast as required in 2018. In these past five years all the maritime industry actors agreed to focus more and more on the decarbonization. 2025 is one of the key checkpoints for IMO with, among other things, the adoption of midterm measures.

From 2050, in just 25 years which is about the lifetime of a single vessel, shipping must eliminate its entire contribution to climate change. Various technologies do exist to reduce vessel emissions: ship-based carbon capture, use of carbon neutral or low carbon fuels, optimization of the engine design, wind-assisted propulsion, vessel speed and operation optimization, weather routing, etc. Lubricants, if specifically designed to reduce friction in some engine parts, can also contribute to reduce the fuel consumption, hence to reduce the emissions by an order of magnitude similar to other technologies and for a reasonable cost for the end users.

We already reported about an innovative solution to reduce fuel consumption in a four-stroke engine by the lubricant. Formulating fuel-economy lubricants for a two-stroke engine is a more challenging task and will participate to the decarbonization effort.

Our solution has been successfully tested on a two-stroke marine engine running with liquid diesel oil, with a careful continuous measurement of the fuel consumption.

The engine test assessed a reduction of 1.3% to 2.4% in fuel consumption, depending on engine operating points. Formulation levers and evaluation means will be presented.

Our deep understanding of the behavior of this new lubricant in various friction conditions and of the formulation drivers and constraints will allow us to meet the challenge of designing two-stroke engine fuel economy lubricants for all types of new fuels.

1 INTRODUCTION

The maritime industry, responsible for approximately 3% of global greenhouse gas emissions, is facing an urgent challenge: how to reduce its environmental impact and to manage the transition towards a sustainable future?

The 2023 IMO revised strategy gives new and more stringent milestones to reduce greenhouse gases emissions from ships. To be in line with IMO's revised GHG strategy, emission pathway must decrease very rapidly until net-zero in 2050. That means that from 2030 the decrease must accelerate and be about twice more rapid than required in 2018. In these past five years all the maritime industry actors agreed to focus more and more on the decarbonization. 2025 is one of the key checkpoints for IMO with, among other things, the adoption of midterm measures.

From 2025 to 2050, in just 25 years which is about the lifetime of a single vessel, shipping must eliminate its entire contribution to climate change. Various technologies do exist to reduce vessel emissions: use of carbon neutral or low carbon fuels, ship-based carbon capture, optimization of the engine design, wind assisted propulsion, vessel speed and operation optimization, weather routing.

Lubricant also, if specifically designed to reduce friction in some engine parts, contributes to reduce the energy losses, consequently the fuel consumption, hence the emissions by an order of magnitude similar to other technologies and for a reasonable cost for the end users.

TotalEnergies already reported about an innovative solution to reduce fuel consumption in a four-stroke engine by the lubricant [1]. Formulating fuel-economy lubricants for a two-stroke engine is a more challenging task and will participate to the decarbonization effort.

This paper will discuss the design of the fuel economy marine lubricant, focusing on their ability to reduce friction and wear, its evaluation on a bench engine and will present the results in terms of fuel consumption savings and global performance.

2 LUBRICATION AND FRICTION IN A TWO-STROKE MARINE ENGINE

2.1. The System oil

In a two-stroke marine engine, lubrication plays a crucial role for ensuring the smooth operation of moving components by mitigating friction, minimizing wear, and preventing excessive thermal degradation. The lubrication systems are distinct: one dedicated to the cylinder liner employing cylinder oil, the other dedicated to the crankcase utilizing system oil. The cylinder oil is generally injected into the upper part of the liner to lubricate the piston and provide a strong film between the piston rings and the cylinder liner. The system oil encompasses bearings lubrication, camshaft and crosshead slides lubrication, pistons undercrown cooling and is also functioning as hydraulic fluid for valve actuation in the servo oil system.

Bearings play a vital role in the operation of a two-stroke marine engine, as they support critical components and enable smooth movement. Three key bearings require particular attention for lubrication: the crosshead bearing, crank-pin bearing, and main bearing.

The crosshead bearing connects the piston to the connecting rod, transmitting the substantial forces generated by the combustion process without imparting any lateral thrust on the crankshaft.

The crank-pin bearing links the crankshaft to the connecting rod, allowing the conversion of the piston's linear motion into the rotational movement of the crankshaft. It withstands heavy loads and high rotational speeds, thus experiencing significant shear stress.

The main bearing supports the crankshaft, allowing it to rotate seamlessly within the engine. It is one of the most critical bearings in the engine, as it bears the crankshaft's weight, and the forces transmitted from the piston.

For all these main components, a fit-for-purpose system oil can play a key role in controlling and reducing friction losses.

Our research work focuses on the system oil, as we have determined that it will have a more significant impact on the global reduction of fuel consumption compared to cylinder oil.

Moreover, the expertise that we acquired on the four-stroke fuel economy lubricant extends here to the two-stroke system oil, that is operating in the same type of lubrication conditions.

2.2 Friction conditions

Friction aspects of any system, regardless of the contact scale, are described by the Stribeck curve, named after the researcher who studied the most influential parameters on the friction of one surface on another. The Stribeck curve, in Figure 1, illustrates how the friction coefficient changes across different lubrication regimes.

The friction coefficient depends on three terms:

- The viscosity of the fluid between the two surfaces,
- The relative speed between the two surfaces,
- The load applied to the surfaces.

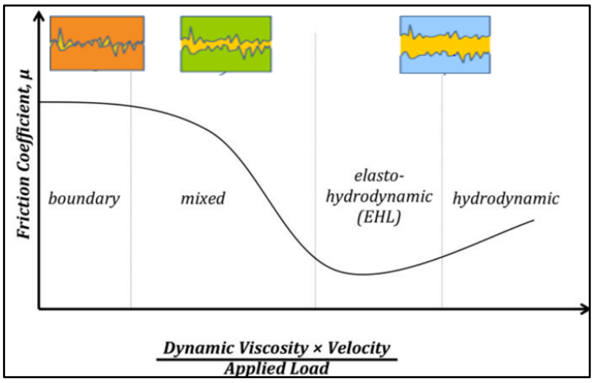

Figure 1. Stribeck curve and lubrication regimes

From left to right: at low speed or low viscosity, the surfaces are very slightly separated and thus in contact by their asperities (boundary lubrication regime), the friction is severe. As we move to the right, we successively pass through the mixed and hydrodynamic / elastohydrodynamic regimes because of speed and viscosity, which will gradually separate the surfaces and thus reduce friction.

The boundary lubrication is the lubrication of surfaces by very thin fluid films, so that the coefficients of friction are affected by both the type of lubricant and the characteristics of the surface. It is associated to a metal-to-metal (asperities-asperities) contact between two sliding surfaces in the engine, which cause the mechanical parts to be submitted to heavily loaded conditions.

In mixed lubrication, some asperities are still in contact, but part of the load is supported by the lubricating film. As the relative velocity begins to increase, the friction falls away sharply, and the friction coefficient decreases until such point as it reaches a minimum.

As the oil film thickness increases further, the system moves into full film lubrication: a complete lubricating film separates the surfaces, considerably reducing friction. Full film conditions consist of two different types of lubrication regime: either hydrodynamic lubrication or elastohydrodynamic lubrication, when the elastic deflection of the contacting surfaces due to the applied load is considered. Friction starts building up again, as the role of the viscosity in the fluid begins to take effect.

In the bearings and crosshead slides zones, the system oil undergoes hydrodynamic lubrication regime under the rated operating conditions. In the camshaft zone, the regime is defined as mixed lubrication.

In addition to the characteristics of the materials, the characteristics of the lubricant will influence these regimes:

To remain in the hydrodynamic regime and achieve low friction, the oil viscosity must be precisely adjusted: not too high to prevent excessive friction resistance, not too low to prevent the wear risk. Therefore, controlling the viscosity of the lubricant is important and of the first order. It must be controlled under all the operating condition: at various temperatures, at various relative speeds of the parts (shear), at various pressures. We'll rather talk about controlling the rheological profile of the lubricant. The chemical composition of the lubricant is also important, yet of secondary order in the ability to reduce friction.

The control of the rheological profile is also essential to manage the mixed lubrication regime and to avoid shifting towards the boundary regime, with potential oil film rupture. As the conditions are severe, it may be necessary to add to the oil specific chemical components which have the ability to create surface films (tribofilms) to lower the friction coefficient.

Rheology and chemistry sciences, applied to the lubricant formulation, give the solution to target the optimized friction coefficient at each contact area in the engine. A reduction of energy losses due to friction and wear will result, as well as a fuel consumption saving.

2.3 Rheology and chemistry levers to design the fuel economy lubricant

The formulation levers to optimize the lubricant composition have already been described [1] for the innovative TPEO lubricant.

In order to optimize the viscosity behaviour of the lubricant in terms of reducing global friction losses, specific rheologically active additives are considered in the formula: they are commonly called "viscosity modifiers" (VM) and provide a non-newtonian (rheofluidifying) character to the lubricant. This means that the viscosity of the lubricant depends not only on temperature and local contact pressure, but also on the shear rate (indicating the existence of mechanical constraints in the friction zone).

These viscosity modifiers have an impact on the relation of the viscosity versus temperature and consequently on the Viscosity Index (that is why they are also called “VI improvers”).

Moreover, the use of those additives provides a huge advantage: when shear rate is high in the lubricated system, the viscosity of the lubricant is lowered in the contact, which decreases at any local point the potential friction losses and thus increases the engine efficiency. However, this also leads to a decrease of the oil film thickness which potentially promotes the risk of disrupting the oil film and causing wear, if the mixed/boundary lubrication regime is reached or if excessive shearing of the polymer occurs.

This advantage is characterized by the measurement of the High Temperature High Shear (HTHS) viscosity. The HTHS dynamic viscosity is measured at 150°C under high shear rate of 10^6 s^{-1}. It indicates how the viscosity of the lubricant is modified in a high loaded contact. HTHS viscosity of SAE 30 grade is typically about 3.5 mPa.s and above. Higher HTHS viscosity means higher potential friction losses. Therefore, HTHS reduction has been targeted for the new fuel economy lubricant. Typically, introduction of viscosity modifiers allows to get HTHS viscosity reduced to about 2.5 mPa.s.

Thickening efficiency of VMs depends on their chemical structure. The higher the thickening power, the less polymer is needed to achieve a given viscosity. As a result, this also enables to choose low viscosity base stocks to design the finished lubricant. However, it should be noted that the shear stability of a VM evolves inversely to its thickening power [2, 3]. Adjust all these parameters, in the right way, will enhance the Fuel Economy potential of a finished lubricant.

Our mapping of an exhaustive series of VM polymers led us to identify one offering the best compromise between thickening power and temporary shear. Its impact on the global performance of the lubricant also has been evaluated on various laboratory tests and bench tests. It is described in the following pages.

When high mechanical constraints are reached, the requirements for the lubricant must focus on its capability to provide a wear and friction protective film in addition to its ability to provide an adequate oil film thickness. The protective film can be provided by friction modifiers commonly used to adjust friction characteristics in mixed / boundary lubrication. Boundary layer between contact surfaces is formed by physicochemical processes, which can decrease the number of direct interactions of metal asperities. Most chemically surface-active anti wear additives and friction modifiers contain sulfur, phosphorus, or molybdenum. These additives can be inorganic or organic. They form metal salt films with low shear strength at the interface, which are effective at high loads, temperatures and sliding velocities.

When the oil film thickness is significantly reduced, adding friction modifiers helps to decrease friction losses by preventing metal-to-metal contact. In such conditions of very thin oil film, friction modifiers positively impact fuel economy, although to a lesser extent than rheology active additives.

We also mapped an exhaustive series of organic and inorganic friction modifiers and identified the component with the highest positive impact on the global performance of the lubricant.

3 PROOF OF CONCEPT FOR A NEW PRODUCT

3.1 Evaluation of the fuel economy potential on a bearing test bench

A preliminary measurement campaign was conducted on a bearing test bench at IST Prüftechnik GmbH (Aachen). Its objective was to measure the friction torque of different system oil experimental formulations, to anticipate their fuel economy potential.

The IST bearing test bench consists of an assembly of a cylindrical shaft and a connecting rod from a MAN MD08 heavy-duty engine, which is servo-controlled and instrumented. The shaft is driven by an electric motor and is connected to the connecting rod via a plain bearing (connecting rod bearings). A hydraulic jack applies a periodic load to the connecting rod, replicating the load cycle seen by a connecting rod bearing in a heavy-duty engine. The assembly is instrumented with torque, speed, bearing temperature, oil temperature, pressure, and flow sensors. The contact voltage across the bearing can also be recorded. The oil bath temperature is regulated.

The operating conditions aim to replicate those of a two-stroke MAN B&W K98MC-C marine engine. The conditions were adapted to maintain the same contact pressure and linear speed, considering the scale change of the parts. The measurement protocol was defined to maintain very stable test conditions.

The test campaign aims to study the impact of the rheology profile and the chemical composition of experimental system oils on the measured friction torque. The tested candidates contain various type and amount of viscosity modifier polymers, and various type and amount of friction modifiers. Their resulting SAE grade are SAE 20 or SAE 30.

The results in Figure 2 show that:

- The repeatability of the test bench is good (repetition of formula n°1), it has been calculated at an acceptable value (5%),
- The friction torque can be considerably reduced if the VM polymer and the friction modifier are well chosen. In other terms, if the kinematic viscosity and the viscosity under shear are well adjusted, the friction torque can be reduced by 20%.
- The impact of the VM polymer is strong, the impact of the friction modifier is weaker on this bench test (hydrodynamic lubrication).

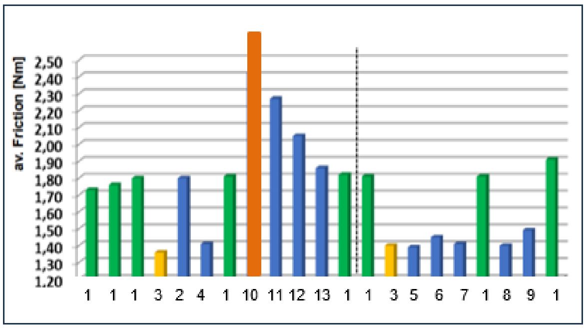

Figure 2. Measured friction torque

Green bars: measured torque for the reference conventional system oil (n°1, SAE 20), repeated 8 times.
Orange bar: measured torque for a conventional system oil (n° 10, SAE 30)
All the other bars: measured torque for experimental formula.

This bench test results helped us select the best candidate to continue developing our fuel economy system oil. Experimental formula n° 13 (Figure 2) was the optimal starting point for the development: SAE grade is SAE 30, as usually specified for a system oil. Friction torque reduction is approximately 20% compared to a conventional system oil.

3.2 Specific properties of the fuel economy new product

The development of the new fuel economy product went then through the choice of the detergents, and the other additives usually present in system oils. The new product must comply with all the requirements of a two-stroke marine engine.

The system oils play several roles in addition to that of bearing oil, including that of hydraulic fluid for certain engine subsystems (valve actuators, etc.). The incorporation of VM polymers can potentially affect certain properties (notably interfacial) of the oil, and thus its performance in certain laboratory tests of a hydraulic fluid specification. It was therefore agreed, before final evaluation on an engine, to characterize the hydraulic specific performance of the fuel economy newly developed product.

To ensure the safety of the presence of VM polymer and friction modifier in the system oil, the new product was compared to a conventional system oil on a set of laboratory tests representative of a hydraulic oil.

The selected tests were those applicable to a hydraulic oil and whose results could be sensitive to the presence of the polymer (and/or interfacial properties). As defined in the specifications for hydraulic fluids and system oils for two-stroke marine engines, these performance tests are listed in Table 1, with their results obtained for the new product and for the conventional system oil. Table 1 also reports the physico-chemical characteristics of the oils.

Table 1. Characteristics of the conventional system oil and the new fuel economy product

Characteristics	Conventional S.O	New Fuel Economy S.O
KV 100 (ISO 3104, mm^2/s)	10.99	11.04
KV40 (ISO 3104, mm^2/s)	91.3	59.6
VI (ISO 2909)	105	180
HTHS (ASTM D5481, mPa.s)	3.25	2.60
BN (ASTM D2896, mg KOH/g)	5.6	5.6

Performances	Conventional System Oil	New Fuel Economy System Oil
Deaeration (50°C, ASTM D3427 – ISO 9120, min)	17	17.3
Demulsibility (82°C, ASTM D1401 – ISO 6614, cotation (min))	0-30-50 (60)	40-40-0 (20)
Foaming (ASTMD892 – ISO 6247, seq 1, seq 2, seq 3, ml)	0/0 0/0 30/0	10/0 0/0 90/0
Filterability (5 µ, NF E48-690)	FI = 0.98	n.d, but almost the entire volume was filtered in 2 hours (280 cm^3/300 cm^3)
Kurt Orbahn Shear stability test (30 cycles, ASTM D6278, ISO 20844, ΔKV 100 %)	-0.4	-3.9
KRL shear stability test (20h, CEC L45-A-99, ISO 26422, ΔKV 100 %)	-0.3	-49.2

Among the tested performances: the deaeration is not affected, the foaming is slightly affected, the demulsibility is improved, possibly due to lower polarity of the new formula. The near failure in the filterability result indicates that the filter size (5 µm) is too fine for this amount of polymer. CIMAC recommends filtration below 10 µm for fluids used as hydraulic fluids.

The shear resistance in the Kurt Orbahn test is also slightly affected but the more severe KRL shear resistance test is affected (the fluid is sheared 1,740,000 times during the test instead of 30 times for the Kurt Orbahn). For comparison, the system oil reservoir is pumped approximately 112,500 times before being (statistically) completely renewed. The operating conditions of this test (for which no specification - other than "Report" - is generally defined in hydraulic standards) therefore correspond to more severe uses of hydraulic oil (notably off-road) than the marine application.

To conclude, the results demonstrate here that the presence of a VM polymer in the new system oil does not significantly affect the performance of the oil as a hydraulic fluid. Depending on the specifications defined for a marine system oil, the criteria of filterability and shear resistance could be used to "calibrate" the maximum percentage of polymer in the formula.

The other performances (detergency, high temperature stability, oxidation resistance, wear resistance) have also been evaluated on the new product, confirming that they are not degraded compared to a conventional system oil.

3.3 Evaluation of the fuel consumption on a two-stroke engine

As a final evaluation to prove the fuel economy performance of the new product, an engine test had to be carried out.

It was conducted at the Maritime University of Shanghai (SMU) under the supervision of Total Energies. The objective of the test on the two-stroke engine was to measure fuel consumption at various operating points. The two-stroke engine is a MAN 6S35ME-B9 engine coupled to a generator.

3.3.1 Engine test procedure

For the test, Marine Diesel Oil (MDO) was chosen as the fuel. Compared to conventional marine fuels (HFO, VLSFO) its quality allows for better stability during the engine operation and, ultimately, a better repeatability of fuel consumption measurements. The MDO consumption was measured by a Coriolis flowmeter. Many other engine parameters were measured to monitor the proper operation of the engine test.

The test procedure was defined to obtain a reliable measurement of fuel consumption under stabilized engine conditions of a large vessel. Three engine loads representative of the vessel's operating points were chosen: 80%, 60%, 40%. On the MAN 6S35ME-B9 engine, the output powers are then 2856 kw at 80% load, 2140 kW at 60% load, 1428 kW at 40% load.

The test is composed of three steps: engine warm-up, test sequence which will be run twice, engine stop.

The new system oil candidate was evaluated comparatively to a reference lubricant, tested as well on the engine. The reference was tested after the candidate. In between each test, the engine was required to be perfectly emptied and flushed. The marine cylinder oil was provided by SMU and has been the same all along the test campaign, and for all cylinders.

In accordance with the requested procedure, each product was tested for 22.5 hours.

3.3.2 Monitoring of the engine parameters

The crankshaft bearing temperature was measured at 36-42°C for both the new oil and for the reference, which is the usual temperature. This consistent temperature during the tests indicates that the bearings operated normally without lubrication failure.

Oil pressure was measured during each test and no difference was observed.

The processing of raw data provided by SMU allowed visualization of some other parameters: Intake air pressure, Scavenging air pressure, Scavenging air temperature, Jacket cooling outlet temperature, Cylinder exhaust temperature, …

3.3.3 Fuel Economy Results

From the raw data files giving the fuel oil consumption in g/h, it is necessary to compute the specific fuel oil consumption (SFOC) to be able to compare all the data and to assess a Fuel Economy performance:

- Specific Fuel Oil Consumption (SFOC, g/kWh):

$$\frac{\text{Mass of Fuel consumed per hour [g/h]}}{\text{Power developed [kW]}}$$

(Mass of Fuel consumed must not include unburnt fuel).

- Fuel Economy compared to a reference oil:

$$\frac{\mathrm{SFOC_{REF}} - \mathrm{SFOC_{CAND}}}{\mathrm{SFOC_{REF}}}$$

SFOC and Fuel Economy must be calculated for each load level (80%, 60% & 40%). As the test is composed of two sequences, results must be calculated by sequence and by test. The test result is the average of sequence 1 result & sequence 2 result.

The results are plotted below to calculate the fuel consumption gain per load, after eliminating the outlier points, arbitrarily defined by an SFOC > 200 g/kWh or < 160 g/kWh.

Figure 3 gives SFOC for the reference and the new oil (all measured points, including transition phases, except outliers). Figure 4 shows the averages made on locally close points.

At first glance, it can be observed that the points corresponding to the new oil are always below those corresponding to the reference (at all load points, including transition phases).

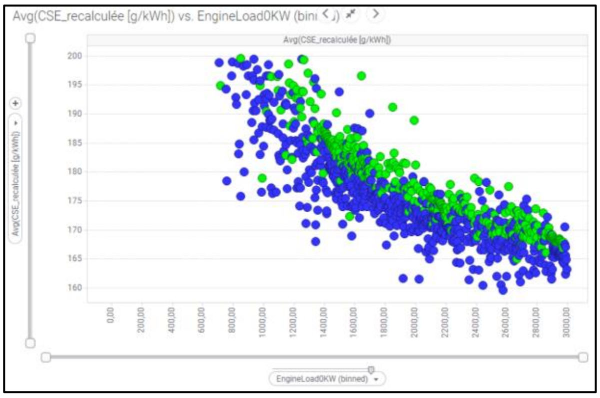

Figure 3: SFOC for the tested oil (in **blue**) and the reference (in **green**) – all measured points.

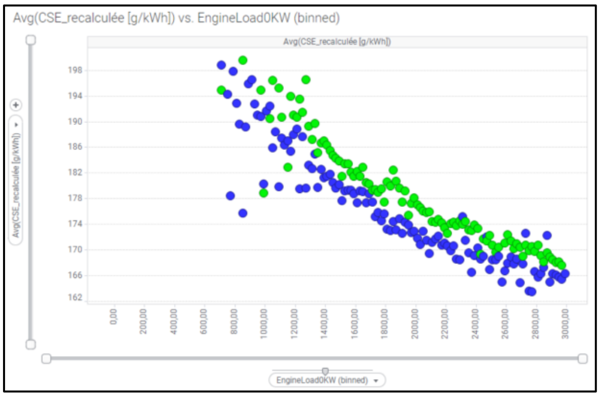

Figure 4: Average SFOC for the tested oil (in **blue**) and for the reference (in **green**).

Figure 5 (1st graph) shows the average SFOC specifically for the selected loads. At 40% load, the average SFOC has been calculated between 172 and 188 g/kWh. At 60% load, the measures are between 168 and 178 g/kWh, at 80% load between 160 and 170 g/kWh.

The second graph in Figure 5 shows the averaged SFOC for each sequence and the repeatability between the two sequences, for each load and for both the tested oil and the reference.

Usually, at TotalEnergies Research Center the repeatability and reliability of the test bench (four-stroke) are evaluated via the "global standard deviation" on a number of repeated tests being at least six. The measured consumption gain for a candidate is thus compared to the "critical gain" involving the global standard deviation, a Student t-factor, and the number of tests performed on the candidate and on the reference (evaluated twice, before/after the candidate). This allows evaluating whether the measured gain is statistically significant.

Given the size of the two-stroke engine at SMU, its consumptions, and the cost of the test, it was not conceivable to repeat the tests multiple times.

The only possible statistical analysis is therefore the one done on the two sequences. The maximum value of the difference between the two sequences is 0.67% (at 40% load, on the candidate), with most differences being equal to or less than 0.32%. If the measured fuel oil consumption gain is higher than this calculated difference, it will be considered significant.

The third graph shows the averaged SFOC measured in the engine test, for each load and for both tested oil and reference. It also gives the SFOC difference between tested oil and reference in absolute and in relative terms.

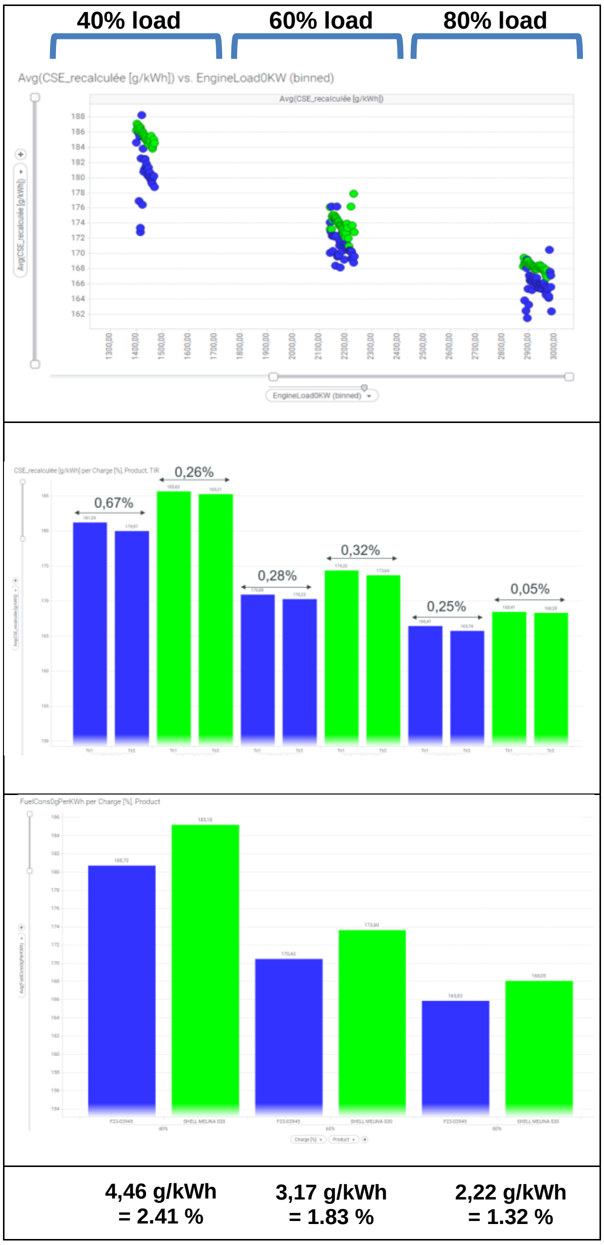

Figure 5: Average SFOC for the tested oil (in **blue**) and for the reference (in **green**), per load :
- With the repeatability between sequence 1 and sequence 2,
- With the numerical values of specific consumption gain in absolute and relative terms, having taken the average of the two sequences.

These relative differences being higher than 0.67%, **they represent a significant Fuel Economy result for the new system oil which is:**

- **1.3 % at 80 % load**
- **1.8 % at 60 % load**
- **2.4 % at 40 % load**

3.3.4 Analysis of the collected lubricant samples

For a final checking, it was interesting to control the behavior of the oils during the engine test. For this purpose, a few samples have been collected for analysis:

- Candidate at start of test
- Candidate at end of test
- Reference at start of test
- Reference at end of test

Table 2 shows the results of the analysis.

Table 2. Analysis of the collected samples during the engine test

Oil sample	New oil start of test	New oil end of test	Ref oil start of test	Ref oil end of test
KV40 (ISO 3104, mm²/s)	58,19	56,41	99,53	98,60
KV100 (ISO 3104, mm²/s)	10,16	9,97	11,62	11,53
VI	163	164	104	104
Insolubles %	0,0	0,0	0,0	0,0
Water cont %mass	0,00	0,00	0,04	0,04
BN (ASTM D2896, mg KOH/g)	5,7	5,7	7,4	7,4
WEAR ELEMENTS				
Iron (Fe)	5	5	27	27
Chromium (Cr)	0	0	0	0
Molybdenum (Mo)	0	0	0	0
Copper (Cu)	2	2	11	11
Lead (Pb)	0	0	1	1
Silver (Ag)	0	0	0	0
Tin (Sn)	0	0	1	1
Aluminium (Al)	1	1	1	1
CONTAMINANTS				
Nickel (Ni)	0	0	2	2
Vanadium (V)	0	0	1	1
Silicon (Si-T)	19	18	9	10
Boron (B)	2	1	4	4
Sodium (Na)	3	2	3	3
Magnesium (Mg)	13	10	12	12
OTHER METALS				
Phosphorus (P), Zinc (Zn), Calcium (Ca)	No change		No change	

For the candidate samples, no change was observed: the viscosity remains constant (10.16 to 9.97 mm²/s, at 100°C), no increase in wear metals, no change in other elements contents.

For the reference samples, the analysis results demonstrate also that there was no change in the product during the test.

In conclusion, the engine tests did not impact the characteristics of the new system oil. The formula technology keeps its performance along the time.

4 CONCLUSIONS

The design study of the innovative fuel economy system oil included several steps: implementation of the expertise about the formulation levers selected for marine Fuel Economy lubricants in a four-stroke engine, validation of the FE potential of several candidates by measuring friction torque on the IST bearing test bench, checking of the safety of the formula for the presence of polymer when used as a hydraulic fluid, and last but not least evaluation on a real two-stroke bench engine.

The campaign of engine tests on this new marine two-stroke system oil compared to a conventional reference system oil has been successfully carried out at Shanghai Maritime University to assess its Fuel Economy potential on the MAN 6S35ME-B9 engine.

The processing of the raw data collected during the engine tests has been done by TotalEnergies.

The results of this proof of concept are significant and show a Fuel Economy of 1.3% to 2.4% in fuel consumption, depending on engine operating loads, thanks to the use of the new formula of system oil. This Fuel Economy figure is consistent and of the same order of magnitude as the one obtained for our TPEO lubricant, as the lubrication regimes are close. This figure also is of the same order of magnitude as the economy obtained by other ways (partial load optimization, engine design, …) for a limited cost.

The novelty provides improved fuel economy without reducing the effective life of the lubricant and impacting the cleanliness and the durability performance of the engine.

Our deep understanding of the behavior of this new lubricant in various friction conditions, of the formulation drivers, and of the engine constraints may allow us to meet the challenge of designing two-stroke engine fuel economy lubricants for all types of new fuels.

By exploring innovative lubrication solutions, this study seeks to contribute to the development of more sustainable maritime operations, aligning with global efforts to reduce carbon footprints.

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