Fuel Economy Optimization From the Interaction Between Engine Oil and Driving Conditions

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Abstract

Low viscosity engine oils have shown to be an effective solution to the fuel consumption reduction target, however, their potential is closely linked to the vehicle and engine design and to the real driving conditions. In this study the interaction between engine oil and driving conditions of two urban routes and one rural route in Spain and the United Kingdom has been put to test with the aim to evaluate their joint effect over fuel economy of a freight transport vehicle. In a first approximation, six different oil formulations, three of them belonging to the new API CK-4 and FA-4 categories and two with molybdenum-based friction modifier, were tested under stationary conditions with a medium-duty diesel engine. Followed by tests under real driving conditions of a freight transport vehicle, developed by means of computer simulations with an adjusted vehicle model, taking the fuel consumption maps of the six oil formulations, vehicle characteristics and the selected driving cycles as inputs to the model. Results of engine bench tests and simulations with oils of lower HTHS viscosity showed fuel consumption reduction values as expected. However unexpected results were found between the oils with molybdenum-based friction modifier added to their formulation.

Keywords: Low viscosity engine oils, Friction modifier, Fuel economy, Driving

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cycles

1. Introduction

The proven role of anthropogenic greenhouse gases (GHG) in global warming has forced the ground transport industry to find solutions to tackle tailpipe CO_2 emissions [1–5], which are generated by the combustion of fossil fuels in internal combustion engines (ICE) [6]. In the European Union (EU), road freight transport has been growing steadily in the last years [7, 8], closely linked with the use of Heavy Duty Vehicles (HDVs) for goods transport both for domestic and international shipment. In Spain, by instance, on-road commercial vehicles are responsible of moving more than 95% of all goods [9]. In turn, freight demand has been influenced by the growth of e-commerce [10–12], where the final leg in the supply chain, known as last mile delivery, is affected by multiple factors such as delivery times and customer density, making this process very inefficient [13–15] and resulting in fuel consumption increase and the consequent environmental impact [16, 17].

1.1. Engine efficiency and low viscosity engine oils (LVEOs)

Several alternatives have been proposed and evaluated to reduce power trains fuel consumption and CO_2 emissions to levels required by the standards [18–27] however, many of them require big changes in the engine and vehicle design, representing a cost increase for manufactures and consumers [21, 28]. In this regard, LVEOs are an appealing alternative to increase the engine efficiency by reducing mechanical friction losses, with the advantage of not needing engine modifications [29–33]; furthermore, fuel savings can be increased if other lubricants in the vehicle, by instance gearbox and differential, are improved along with the engine oil [30, 34]. Studies that support those findings, also agree that fuel consumption reduction is closely linked to the engine and vehicle characteristics and to the working conditions [29, 30, 33, 35, 36]. Amongst the engine oil viscometric parameters, High

Temperature High Shear (HTHS) viscosity is the one that predicts better fuel economy [37].

1.2. LVEOs and molybdenum-based friction modifier in the HDVs segment

There are two main classification criteria for engine oils, one based on performance and other based just on viscosity. For the first one, there are four main organizations that define the oil classification worldwide, the European Automobile Manufacturers Association (ACEA), American Petroleum Institute (API), Japanese Automotive Standards Organization (JASO) and International Lubricant Standardization and Approval Committee (ILSAC). Most OEMs also define their own engine oil classification. Regarding the oil viscosity, the Society of Automotive Engineers (SAE) classifies the engine oil under the SAE J-300 standard [38]. Following the path of CO_2 emissions regulations and fuel economy requirements for engine oils of HDVs segment, API released the CK-4 and FA-4 categories in December 2016. For the former one, the HTHS viscosity limit was maintained at previous CJ-4 levels (>3.5 cP). While, FA-4 category allows engine oils to be formulated with a reduced HTHS viscosity (2.9 - 3.2 cP) to contribute to fuel economy [39].

Regarding molybdenum-based friction modifiers (Mo FM), they have been widely used in lubricating oils for gasoline engines due to their positive effect on reducing friction during boundary and mixed lubrication [40, 41]. For diesel engines however, the use of Mo FM has been limited due to the negative effect of soot over the Mo FM capabilities to reduce friction [42, 43]. Post injections, by instance, is a common technique to reduce in-cylinder soot, and although research is still needed to completely understand how it helps to reduce soot [44, 45], its effectiveness gives the opportunity to re-evaluate the effect of Mo FM under these new working conditions [46].

Given the fact that HDVs will continue to be the main freight transport option in the EU for the next years, the novelty of the API oil categories and the need of new studies on Mo FM destined for oils of this segment, it was determined the need to evaluate new oil formulations under controlled and real working conditions. In this regard, the main contribution of the study presented here is the analysis and quantification of the fuel consumption reduction potential obtained from the interaction between the oil formulation and working conditions of a selected engine and vehicle, by means of experimental tests and simulations. To accomplish this objective a medium-duty diesel engine was employed for the engine bench tests, and a simplified vehicle model for the simulations of real driving conditions of a freight transport vehicle. In the first part of the document, screening tests were developed under fired stationary conditions with six oil formulations, four of them belonging to the new API CK-4 and FA-4 categories and two with Mo FM. This stage allowed to determine the engine working zones where the effect of these two oil properties is more evident. In the second part, computer simulations were carried out by means of a vehicle model, where the selected vehicle was tested under real driving conditions of an urban and a rural route in Spain and an urban route in the United Kingdom (UK), along with the chosen oil formulations.

2. Materials and Methodology

The first part of this section explains the methodology and equipment employed to develop the experimental tests under stationary conditions, along with a brief description of the tested oils. The second part refers to the vehicle model used to simulate fuel consumption of the selected vehicle and the methodology employed for this purpose. A description of the three routes selected for the simulations is also presented here.

2.1. Engine bench tests

2.1.1. Test procedure

The bracketing technique used in this study to evaluate each candidate oil formulation consists of testing the selected reference oil before and after the candidate oil. This procedure which includes oil flushing and oil conditioning, guarantees that any changes in the engine and the surrounding environment are taken into account by making a direct comparison of the candidate oil results and the average of the two measurements with the reference oil. Furthermore, attending that any modifications to the engine and its components could have an impact on friction and therefore on fuel consumption, the engine was not disassembled between tests, neither any of its components were modified or replaced.

The steps to test one candidate oil are as follows: the oil deposit is filled with the reference oil (oil flushing) allowing the engine to run at four different speeds and loads (1500 rpm and 211 Nm, 2000 rpm and 141 Nm, 2500 rpm and 70 Nm, 3400 rpm and 94 Nm) for two hours. This step was done with the purpose of removing any of the remaining oil and to ensure the lubrication of all the engine parts. Then, the reference oil was drained and replaced with fresh reference oil followed by a period of oil conditioning running the engine at average speed and torque for an hour. This step, done before every test, allows the oil polymers to reach their final properties after being subjected to shear conditions [47–49]. Working parameters of the engine like torque, throttle position and EGR were recorded in this point to check the correct operation of the engine. Right after the oil conditioning, the reference oil was tested adjusting the engine speed and toque, values defined previously, and leaving the throttle position variable. After completing the test with reference oil, it was drained and replaced with the candidate formulation. The same procedure of oil flushing and conditioning was followed before testing the candidate oil. After completing tests with candidate oil, a final measurement with the reference oil was performed in order to complete the bracketing requisites.

During tests and for every test point, the predefined engine speed and load were reached and after a stabilization time, different engine performance parameters were recorded, including oil and ambient temperature, intake and exhaust pressure, EGR, etc, and averaged for a 30 seconds period. Simultaneously, fifty in-cylinder pressure and crankshaft angle cycles were recorded in order to obtain indicated parameters. Each point had 3 repetitions and the median of the measured averages was set as the definitive value for each parameter.

2.1.2. Experimental setup

The engine used during tests is a 3 liters diesel engine, its main characteristics are shown in Table 1. The experimental setup was located in a climate test chamber which allowed to set and control the ambient pressure (1003 Pa) and temperature (18°C) for the tests. The engine was coupled to a dynamometer capable of absorbing and dissipating the energy developed by the internal combustion engine. The oil temperature was controlled by means of an external system consisting of cooler fluid, a heat exchanger and a water supply network. Flow of water to the heat exchanger was controlled with a PID (proportional-integral-derivative controller) which compared the signal sent by a K-type thermocouple, located in the engine to measure the oil temperature, with a predefined set point and delivered an indication to an electrovalve for it to act consequently, increasing or restricting the flow of water. The intake and exhaust manifold temperatures were also measured. Pressure sensors were located in the lubricating system and in the intake and exhaust manifold. All of these parameters were controlled and registered with an in-house developed software SAMARUC. The engine speed, torque and throttle position were set and controlled by the same software. Piezoelectric sensors were located inside the combustion chamber of each cylinder in order to obtain an accurate reading of the instantaneous pressure value throughout the engine cycles, their signal was recorded with a Yokogawa DL850V data acquisition system, the trigger was sent by the encoder so the sampling frequency was each degree of the crankshaft rotation. Finally, the ETAS INCA software was used to get the engine parameters registered by the engine control unit (ECU), including the engine fuel consumption. The schematic diagram of the experimental setup is shown in Figure 1.

Engine characteristic	Values
Cylinder bore	102 [mm]
Stroke	96 [mm]
Engine displacement	3 [l]
Cylinders	4
Valves	2 per cylinder
Max. effective power [kW]	111.3 @ 3600 [rev/min]
Max. effective torque [Nm]	350 @ 2000 [rev/min]
Emissions control	EGR
Turbocharger	Variable geometry (VGT)
Emissions standard	Euro VI

Table 1: Engine main characteristics

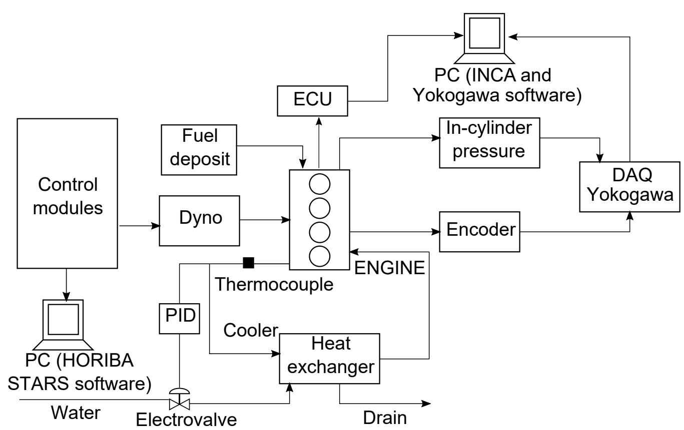

Figure 1: Schematic diagram of the experimental setup

2.1.3. Test points

Based on the engine working map, stationary test points were chosen to cover different working zones from low load to maximum load and four different engine speeds, from 1500 to 3000 rpm. A total of 18 test points were defined, (see Table 2), where the last one corresponds to idle conditions. During tests, every point comprises three repetitions of 30 seconds each one.

Point	Engine speed [rpm]	Torque [Nm]	Point	Engine speed [rpm]	Torque [Nm]
01	1500	70	10	2500	141
02	1500	141	11	2500	211
03	1500	211	12	2500	Max
04	1500	Max	13	3000	94
05	2000	70	14	3000	164
06	2000	141	15	3000	235
07	2000	211	16	3000	Max
08	2000	Max	17	3000	70
09	2500	70	18	700	0

Table 2: Points definition for the stationary tests

2.1.4. Oil formulations

Oils chosen for the test include one SAE 10W30 with 3.67 cP HTHS viscosity used as reference oil and five candidate formulations of different SAE grades. In order to make the analysis of results clearer, lubricating oils were designated with letters and numbers, R for the reference oil and C for the candidates. The main characteristics of the engine oil formulations are presented in Table 3. Formulation C2 and C5 meet the requirements of the new API FA-4 category, C5 however, was formulated with a HTHS viscosity lower than the specified by API for this category (Section 1.2). Reference oil and C4 belong to the API CK-4 category. In order to evaluate the effect of Mo FM, this additive was added to formulations C3 and C4. Except for C1, none of the formulations tested in this study are commercially available.

2.2. Vehicle model

In the present work the impact of the oil formulation in real driving conditions is assessed by modelling. To this aim, a simple but comprehensive

Oil designation	R	C1	C2	C3	C4	C5
SAE grade	10W30	10W40	10W30	5W30	5W30	5W20
Kv @ 40°C [cSt]	75.23	86.71	56.45	70.60	70.15	43.72
Kv @ 100°C [cSt]	12.05	13.34	9.607	11.85	12.08	7.992
HTHS @ 150°C [cP]	3.67	3.85	3.17	3.61	3.61	2.7
SAPS level	Medium	Medium	Medium	Low	Medium	Medium
API base oil group	III	III	III	III + IV	III	III
API category	CK-4	CI-4	FA-4	CJ-4	CK-4	FA-4
Friction modifier	[-]	[-]	[-]	Mo	Mo	[-]

Table 3: Rheological characteristics of the oils

model has been developed. The model represents vehicle dynamics and powertrain behaviour. From a vehicle point of view, the model considers longitudinal dynamics[50], including the aerodynamic drag:

$$F_a = \frac{1}{2}\rho A C_d v^2 \tag{1}$$

where ρ is the air density, A represents the vehicle frontal area, C_d is the drag coefficient and v the vehicle speed; the rolling resistance [51]:

$$F_r = c_r m g \cos\beta \tag{2}$$

where c_r the road friction coefficient (typically around $0.010 - 0.015$ [52]), m is an equivalent vehicle mass including the inertia of the powertrain elements, g is the gravity and β the road gradient; the effect of the road slope is considered through:

$$F_g = m g \sin\beta \tag{3}$$

and the vehicle inertia is:

$$F_i = m\dot{v} \tag{4}$$

The driving force is:

$$F_t = \frac{T_w}{r_w} \tag{5}$$

with T_w the driving torque at the wheels and r_w the wheel radius. Finally, the effect of the brakes is modelled as:

$$F_b = u_b \widehat{F}_b \tag{6}$$

where u_b is the brake actuation (from 0 representing no braking to 1 representing full braking) and $\widehat{F}_b$ the maximum braking force. Considering all the previous forces, the following ordinary differential equation can be used to represent the longitudinal dynamics of the vehicle:

$$\dot{v} = \frac{F_t - F_a - F_r - F_g - F_b}{m} \tag{7}$$

For the present work, a backward quasi-static approach has been used to model the powertrain[53]: equation 7 is solved for F_t assuming that the vehicle is able to follow a predefined speed trajectory along the route (see section 2.2.1). Then, equation 5 can be used to obtain the driving torque at the wheels (T_w) from the driving force (F_t). On the one hand, the transmission is modelled as an ideal set of gears [54] that allows to compute the engine torque and rotating speed from the corresponding wheel parameters. Finally, the engine is represented by an operating map [55] extracted from the engine bench tests (Section 2.1). In particular, fuel consumption is tabulated as a function of engine speed and torque for every oil tested; in Figure 2 it is shown the fuel consumption map for reference oil R.

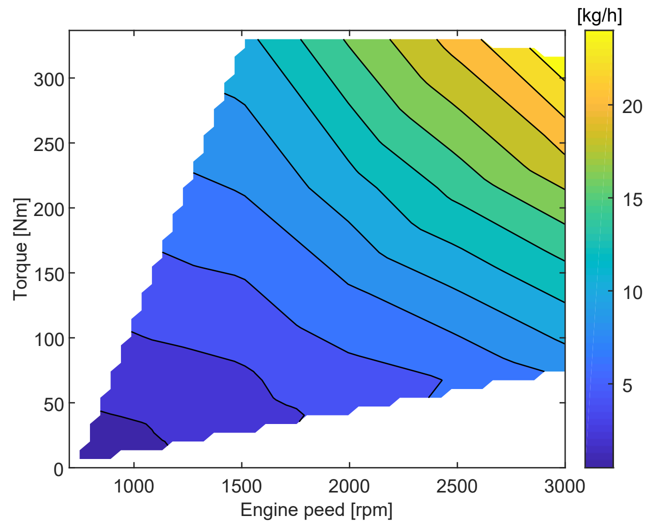

Figure 2: Fuel consumption map with reference oil used in the vehicle model

The main specifications of the freight transport vehicle used in the model are presented in Table 4.

<table>
<tr><td>Mass [kg]</td><td>m</td><td>2465</td><td rowspan="5">Gear ratios</td><td>1st</td><td>4.225</td></tr>
<tr><td>Wheel size</td><td></td><td>195/80 R15 LT</td><td>2nd</td><td>2.341</td></tr>
<tr><td>Frontal area [m^2]</td><td>A</td><td>4.30</td><td>3rd</td><td>1.458</td></tr>
<tr><td>Drag coefficient [-]</td><td>Cd</td><td>0.3</td><td>4th</td><td>1.000</td></tr>
<tr><td>Road friction coefficient [-]</td><td>Cr</td><td>0.01</td><td>5th</td><td>0.796</td></tr>
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Table 4: Vehicle main specifications

2.2.1. Selected routes

Three representative driving cycles of a medium-duty vehicle destined for freight transportation were chosen for the simulations with the vehicle model. Two urban routes were measured, one in the city of Valencia, Spain and the other measured in the market town Romsey, UK. The third driving cycle consists of a rural route between two municipalities, Canals and Quesa, both located in the Valencian community (Spain). Elevation and time-vehicle speed profiles of the cycles are shown in Figure 3, where it can be seen that the urban cycle in Spain had a total duration of 2039 s (10.4 km), whilst the one measured in UK comprises 596 s (2.1 km). The rural cycle had a duration of 1763 s (26.6 km). If the cycles are compared, besides the higher speeds reached by the vehicle during the rural cycle, it is important to note the amount of stops in the urban cycles, specially in the one measured in Spain, which is considerably higher than that of the rural cycle as a consequence of urban traffic, some of these stops are idle periods of about 40 s. Regarding the elevation profile, route covered in the rural cycle starts at about 220 meters above sea level (m.a.s.l.) and the urban cycle in UK at about 30 m.a.s.l; both of them consist of multiple slopes but also of fairly flat areas. Flatness of the urban road in Spain and the diverse topography of the urban cycle in UK and the rural one, have a consequent impact over the power needed from the engine and thus over fuel consumption.

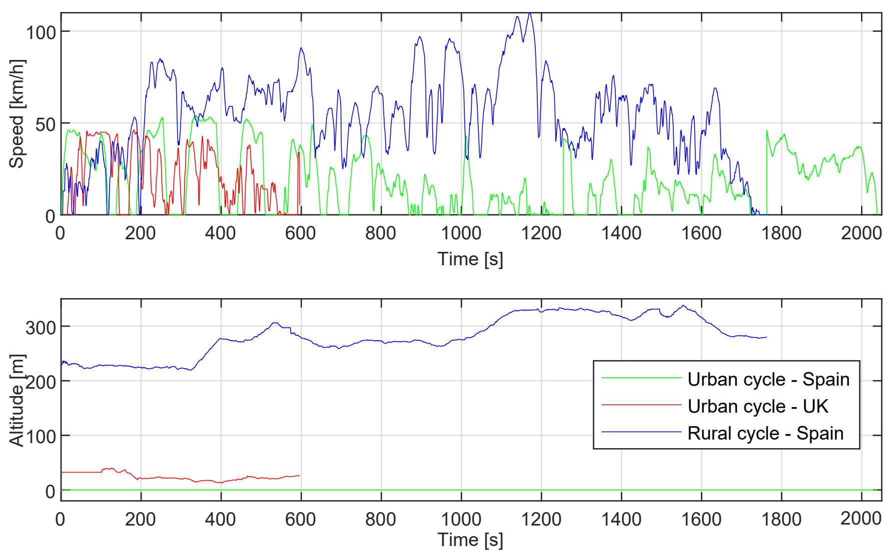

Figure 3: Time-vehicle speed profile (up) and elevation profile (down) for the urban and rural cycles

3. Results and discussion

3.1. Impact of oil formulation over fuel consumption under stationary conditions

In this section the most significant results of tests under stationary conditions are shown and discussed. Fuel consumption comparisons between candidate and reference oil are shown as percentage difference, in this way, results with negative values indicate a decrease in fuel consumption with the candidate oil, while positive values indicate an increase in fuel consumed by the engine. Standard error of the percentage difference in fuel consumption was calculated for every test point and oil formulation, showing deviations in the interval ± 0.0003 and ± 0.02.

3.1.1. Fuel consumption reduction with lower HTHS viscosity

Figure 4 presents fuel consumption differences obtained comparing candidate oils with the reference. Contribution to fuel economy due to the use of a less viscous oil can be clearly seen with candidate oil C2 and specially with C5. For the first one, fuel consumption reduction reached 2.7% at idle conditions, where power from the combustion process is used only to overcome friction mechanical losses, consisting of pumping work, engine auxiliaries drive and friction due to the relative motion of the engine components. This performance was maintained at around 1.2% at low load and the entire speed range, but was reduced with the load increase, which is in accordance with findings in previous studies [56, 57]. For the second oil, a maximum 5.3% fuel consumption reduction was found at a specific point (2000 rpm and 70 Nm), whereas at low load and the entire speed range, fuel consumption reduction was kept at around 4%. Results of these two candidate oils show the contribution of a lower HTHS viscosity to fuel economy during hydrodynamic lubrication, that is when the oil film is thick enough to separate the moving parts restricting the direct contact between them, and the contribution of the additives of the API FA-4 category during mixed and boundary lubrication, therefore extending the benefit zone of LVEOs.

The increase of fuel consumption with an oil of higher HTHS viscosity can be observed with candidate oil C1 of 3.85 cP HTHS viscosity. Overall, fuel consumption increase occurs throughout the map, being more pronounced at low-medium load and medium engine speed. In this zone, the high oil viscosity and the engine speed promotes the appearance of hydrodynamic lubrication, nonetheless, this relatively thick oil film increases the amount of energy needed to shear the oil, due to the increased viscous friction, and therefore the fuel consumed by the engine [58]. This behavior is reduced when the engine load increases and the oil film gets thinner. During idle conditions there is no effect over fuel consumption, situation that could be explained from the similar rheological characteristics between candidate and reference oil.

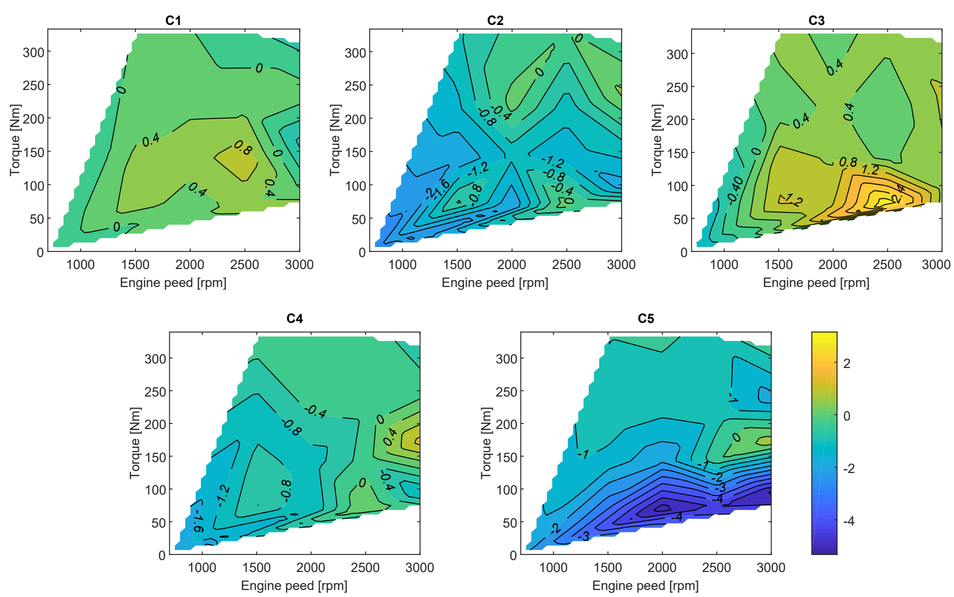

Figure 4: Fuel consumption differences between candidate and reference oils [%]

3.1.2. Fuel consumption reduction with Mo FM

Two oil formulations were selected for this purpose, candidate oil C3 and C4 with Mo FM added to their formulation. These candidate oils share the same HTHS viscosity value of 3.61 cP, but belong to different API categories, CJ-4 and CK-4 respectively.

In Figure 4, the effect Mo FM over fuel consumption showed opposite results for each candidate oil. On one hand, the use of this additive with C4 gave positive results in terms of fuel consumption reduction, mainly in the zone ranging from idle to low engine speed and low-medium load, where the effect of the Mo FM was expected to be more evident by reducing the direct contact between the engine parts during boundary and mixed lubrication. With candidate oil C3 on the other hand, although there is a 1.2% fuel consumption reduction at idle conditions, and 0.4% at low load and low engine speed, the increase of fuel consumed by the engine reached 3% in zones where hydrodynamic lubrication prevails. Another research work [58], however, found that the positive effect of FM (the FM compounds are not shown in the article) seemed to be extended to hydrodynamic lubrication conditions, reducing shear forces in the lubricant film. Therefore, the performance of the Mo FM presented in this study suggests a poor synergy between the base oil and the additive.

Results shown in this section comprise the first step of the study presented in this document, where different oil formulations were tested to determine their potential to reduce fuel consumption under stationary conditions, along with the identification of the working zones where higher fuel economy can be achieved.

3.2. Impact of oil formulation over fuel consumption under real driving conditions

In order to have a more detailed view of the urban and rural cycle's data points, bivariate histograms were plotted, from which it is possible to determine the areas where there are the greatest amount of points and therefore where the vehicle has worked longer. Figure 5, 6 and 7 are bivariate histograms of the three cycles used in this analysis, where the colored scale represents the points distribution. For the urban cycle in Spain, by instance, the vehicle has worked mainly from 0 to 800 rpm and low load (Figure 5), while for the rest of the cycle, points are distributed along the engine speed range (up to 2200 rpm) and

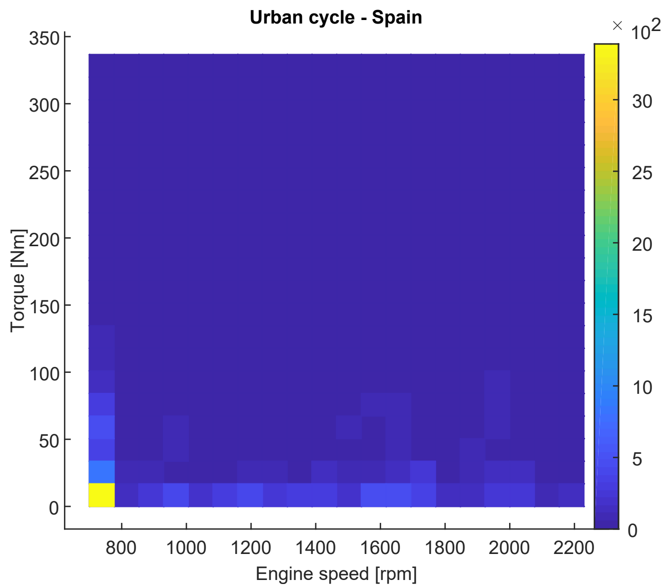

Figure 5: Bivariate histogram of the urban cycle measured in Spain

low load. On the other hand, the urban cycle measured in UK shows a more scattered points distribution both in speed range and load. Nonetheless, from Figure 6 two zones stand out where the points density is greater, one at low engine speed and the other between 1600 and 1800 rpm. In the rural cycle, most of the points are distributed between 1500 and 2500 rpm and low-medium load, which represents common driving conditions found in rural roads. There is also a considerable concentration of points at high engine speed (from 2900 rpm) and low load.

Results obtained after completing simulations with the simplified vehicle model are presented in Figure 8, where the accumulated fuel consumption [g] has been represented over the cycle duration for each oil formulation and driving cycle. Boxes inside each graph are the accumulated fuel consumption at the end of the cycle, where major differences between oils can be seen, specially with C5. Looking at the trend of the urban cycle in Spain (Figure 8.a) and in the UK (Figure 8.b), it can be seen that the vehicle has worked mainly under low load and low speed conditions that lead to low fuel consumption, whereas in the

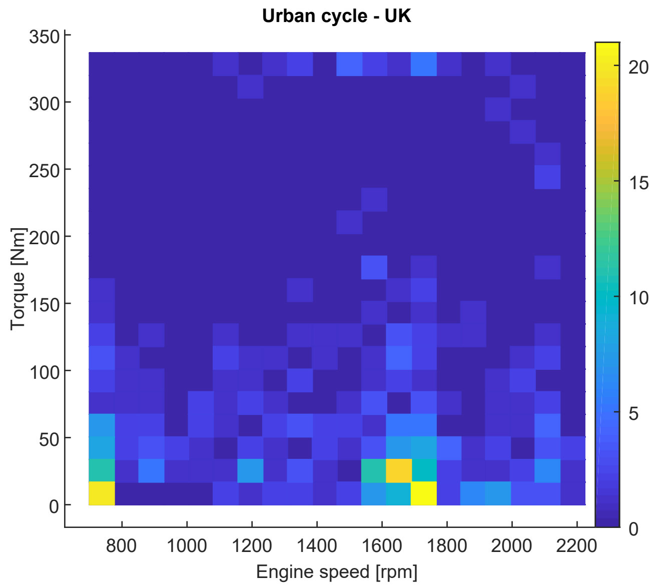

Figure 6: Bivariate histogram of the urban cycle measured in UK

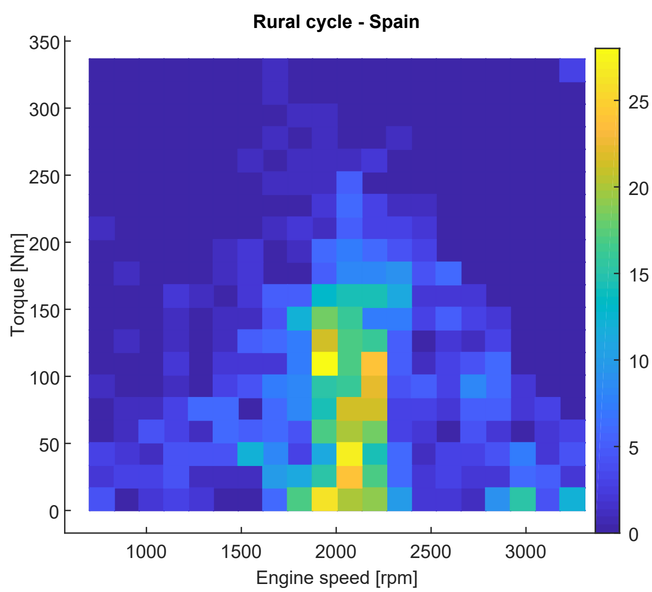

Figure 7: Bivariate histogram of the rural cycle measured in Spain

rural cycle (Figure 8.c) the slope of the lines indicate higher fuel consumption as a consequence of higher velocities and loads.

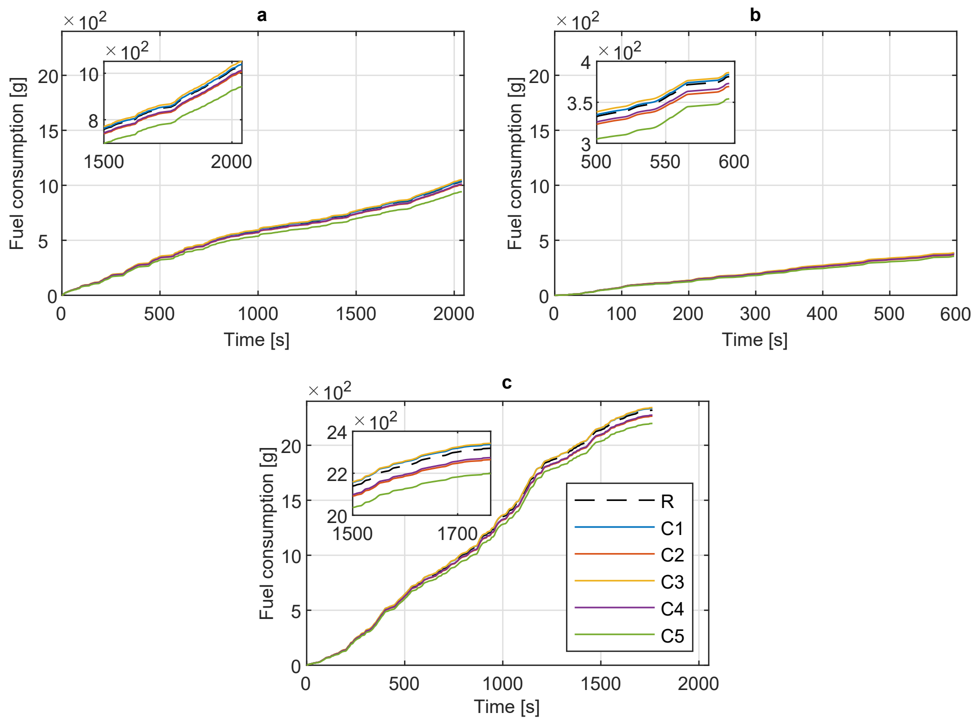

Figure 8: Accumulated fuel consumption for the urban cycle in Spain (a), urban cycle in UK (b) and rural cycle (c)

Comparisons between the reference and candidate oil are summarized in Table 5 as absolute and percentage difference of the total fuel consumed by the vehicle with each formulation and driving cycle. Given that the stationary fuel consumption maps were used in the vehicle model, the correlation between results is as expected; depending on the oil that is being tested, the same tendency to increase or decrease fuel consumption appears. From the results of the urban cycle in Spain, it is remarkable the fuel consumption reduction obtained with candidate oil C5 reaching a 8.84% fuel economy. Situation that can be explained from Figure 5 where most of the data lie within the zone of high fuel consumption reduction given by this oil formulation. It has to be noted that the topographic characteristics of the city of Valencia provide favorable conditions to obtain the best fuel economy potential of a reduced HTHS viscosity. With an uneven topography, (urban cycle in UK and the rural cycle) the effect of the LVEO is reduced as a result of higher load conditions. Candidate oils C2 of 3.17 HTHS viscosity and C4 with 3.61 cP and Mo FM, also gave fuel consumption reduction for the three cycles. Results of these three oil formulations demonstrate the greater effect that has a lower HTHS viscosity (C2 and C5) on the reduction of total fuel consumption, than that of Mo FM.

For the rural cycle, the same tendency to reduce fuel consumption with candidate oils C2, C4 and C5, was observed despite of the lower percentage differences between candidate and reference oil, due to the presence of a considerable amount of data points at medium-high load, where fuel economy given by these formulations decrease. Furthermore, as explained in ,

		Urban cycle - Spain		Urban cycle - UK		Rural cycle	
Oils		Absolute difference [g]	Percentage difference [%]	Absolute difference [g]	Percentage difference [%]	Absolute difference [g]	Percentage difference [%]
R	C1	4.77	0.46	2.56	0.67	18.26	0.79
	C2	28.12	-2.72	12.00	-3.15	54.06	-2.33
	C3	18.03	1.75	5.05	1.32	23.80	1.03
	C4	21.43	-2.07	8.30	-2.18	44.44	-1.92
	C5	91.32	-8.84	27.07	-7.10	120.25	-5.19

Table 5: Results of fuel consumption under real driving conditions.

4. Conclusions

(1) Stationary tests allowed to determine the engine working zones where fuel economy can be obtained and the effect of two oil properties, HTHS viscosity and Mo FM. For the former, results are in accordance with the lubrication theory; oil C1 of higher viscosity, gave fuel consumption

increase during hydrodynamic lubrication, whereas C2 and C5, presented fuel consumption reduction. Regarding Mo FM, its positive effect is clearly seen at low speed and low-medium load with C4. With C3, the addition of Mo FM resulted in fuel consumption increase.

(2) Interestingly, fuel consumption increase with C3 occurred at hydrodynamic conditions, whereas at low speed and low load the positive effect of the FM is present. Based on previous experience of the authors, this situation suggests a poor synergy between the base oil and additive, however further research is planned to clearly understand this phenomenon.

(3) Comparing results of oils C2 and C5 of low HTHS viscosity, with results of C4 with Mo FM, it was observed the greater effect that has a lower HTHS viscosity over fuel economy, both for engine bench tests and simulations of real driving conditions.

(4) Finding an optimum synergy between vehicle/engine, driving conditions and oil has demonstrated to be fundamental to improve fuel economy and thus reduce CO_2 emissions.

(5) Results obtained with C2 and C5 after simulations of real driving conditions demonstrate the potential of new oils (complying with the new API FA-4 category) to reduce fuel consumption of a freight transport vehicle under both urban and rural conditions.

(6) Methodology proposed in this study can be widely used by companies with medium-duty and HDV fleets to find the optimum synergy between vehicle, oil and common or predefined driving conditions of the fleet, to obtain maximum benefits of fuel economy and CO_2 emissions reduction.

(7) Engine wear due to degradation and contamination of the oil formulations tested here are out of the scope of the present study, however additional studies to cover this lack are part of the proposed future work.

Acknowledgments

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