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Contribution to the assessment of the potential of low viscosity engine oils to reduce ICE fuel consumption and CO_2 emissions

Doctoral Thesis
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DOCTORAL THESIS

CONTRIBUTION TO THE ASSESSMENT OF THE POTENCIAL OF LOW VISCOSITY ENGINE OILS TO REDUCE ICE FUEL CONSUMPTION AND CO_2 EMISSIONS

By

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Resumen

Actualmente la industria de la automoción vive uno de los periodos de cambio más vertiginosos de las últimas décadas, marcado por un creciente interés en reducir los impactos medioambientales negativos generados por el consumo de combustibles fósiles y sus consecuentes emisiones nocivas de dioxido de carbono (CO_2) generados durante el funcionamiento del motor de combustión interna alternativo (MCIA).

Teniendo en cuenta que el proceso de sustitución de la flota actual por una totalmente independiente de los combustibles fósiles puede tomar varias décadas, y ante la urgencia inmediata de reducir las emisiones de CO_2, se puede decir que actualmente es más urgente hacer una optimización de los vehículos con motorizaciones convencionales. Entre las soluciones técnicas que se han desarrollado para mejorar la eficiencia del MCIA destaca la utilización de aceites de baja viscosidad como un método efectivo y de bajo coste de implementación que brinda reducciones del consumo entre el 0.5% y el 5%.

Durante el desarrollo de esta tesis se ha llevado a cabo un plan de ensayos enfocado en determinar valores concretos de ahorro de combustible esperados cuando se utilizan aceites de baja viscosidad en vehículos de trabajo ligero y pesado. El plan de estudios se dividió en tres partes; la primera se centró en el estudio de MCIA de vehículos de trabajo ligero, utilizando un motor Diesel representativo del mercado Europeo y llevando a cabo pruebas comparativas en arrastre, puntos de funcionamiento estacionarios y ciclos transitorios de homologación. La segunda parte del estudio consta de otro ensayo comparativo, esta vez utilizando una flota de vehículos de trabajo pesado. El estudio se realizó con la flota de autobuses urbanos de la ciudad de Valencia, incluyéndose 3 modelos de autobuses, con 2 tipos de motorización diferente. La tercera parte del estudio se centró en el comportamiento del coeficiente de friction en los pares tribológicos del motor haciendo ensayos comparativos con tribómetros especializados; uno de movimiento alternativo para simular las condiciones de la interfaz piston-camisa y un "bola y disco" para simular la lubricación en el sistema de distribución, específicamente en la interfaz leva-taqué.

Los diversos estudios comparativos han servido para analizar como es la respuesta general de la fricción y el consumo de combustible cuando se usan aceites de baja viscosidad, tanto a nivel de motor como para la totalidad del vehículo, encontrando diferencias de par en los ensayos de arrastre, de consumo específico de combustible en los ensayos de motor en estado estacionario y diferencias totales de consumo de combustible en los ensayos en régimen transitorio y en flota, que a su vez han permitido estimar la reducción esperada en la huella de carbono.

Resum

Actualment la indústria de l'automoció viu un dels períodes de canvi més vertiginoses de les últimes dècades, marcat per un creixent interès en reduir els impactes mediambientals negatius generats pel consum de combustibles fòssils i els seus conseqüents emissions nocives de diòxid de carboni (CO_2) generats durant el funcionament del motor de combustió interna alternatiu (MCIA).

Tenint en compte que el procés de substitució de la flota actual per una totalment independent dels combustibles fòssils pot prendre diverses dècades, i davant la urgència immediata de reduir les emissions de CO_2, es pot dir que actualment és més urgent fer una optimització dels vehicles amb motoritzacions convencionals. Entre les solucions tècniques que s'han desenvolupat per millorar l'eficiència del MCIA destaca la utilització d'olis de baixa viscositat com un mètode efectiu i de baix cost d'implementació que brinda reduccions del consum entre el 0.5% i el 5%.

Durant el desenvolupament d'aquesta tesi s'ha dut a terme un pla d'assajos enfocat a determinar valors concrets d'estalvi de combustible esperats quan s'utilitzen olis de baixa viscositat en vehicles de treball lleuger i pesat. El pla d'estudis es va dividir en tres parts; la primera es va centrar en l'estudi de MCIA de vehicles de treball lleuger, utilitzant un motor dièsel representatiu del mercat Europeu i portant a terme proves comparatives en arrossegament, punts de funcionament estacionaris i cicles transitoris d'homologació. la segona part de l'estudi consta d'un altre assaig comparatiu, aquest cop utilitzant una flota de vehicles de treball pesat. L'estudi es va realitzar amb la flota d'autobusos urbans de la ciutat de València, incloent-se 3 models d'autobusos, amb 2 tipus de motorització diferent. La tercera part de l'estudi es va centrar en el comportament del coeficient de friction en els parells tribològics del motor fent assajos comparatius amb tribómetros especialitzats; Un acció reciprocante per simular les condicions del piston camisa i un bola i disc per simular la lubricació en el sistema de distribució.

Els diversos estudis comparatius han servit per analitzar com és la resposta general de la fricció i el consum de combustible quan es fan servir olis de baixa viscositat, tant a nivell de motor com la totalitat del vehicle, trobant diferències de bat a els assajos d'arrossegament, de consum específic de combustible en els assajos de motor en estat estacionari i diferències totals de consum de combustible en els assajos en règim transitori i en flota, que al seu torn han permès calcular la reducció en la petjada de carbono.

Abstract

The automotive industry is currently experiencing one of its most rapidly changing periods in recent decades, driven by a growing interest in reducing the negative environmental impacts caused by fossil fuels consumption and the resulting carbon dioxide (CO_2) emissions generated during the operation of the internal combustion engine (ICE) which have proven to contribute significantly to Global Warming.

Given the fact that a total replacement of the current fleet, dependent of fossil fuels, is unlikely to happen in the immediate future and the urgency to reducing CO_2 emissions from transportation in order to tackle Global Warming, it is possible to say that optimizing current ICE technologies and conventional vehicles and engines is a first order priority. Among the technical solutions developed to improve the efficiency of ICE, low viscosity engine oils (LVEO) have emerged as an effective and low-cost method that provides reductions in fuel consumption between 0.5% and 5%.

During the development of this thesis, a test plan focused on determining fuel consumption reduction when low viscosity oils are used in light duty vehicles (LDV) and heavy duty vehicles (HDV) were carried out. The test plan has been divided in three parts; the first part was focused on the study of light-duty vehicles (LDV) using one diesel engine representative of the European market. During this part three testing modes were used: comparative motored, fired stationary points and transient homologation cycle tests. All test were performed in the engine test bed. The second part of the study consisted of another comparative test, this time using a different engine oils in a HDV fleet. The study was conducted using the urban buses fleet of the city of Valencia, including 3 buses models , with 2 different powertrain technologies. The third part of the study was focused on the friction coefficient behavior within the engine tribological pairs making comparative tests in two specialized tribometers; one of reciprocating action to simulate the lubrication conditions in the piston ring-cylinder liner contact and a "ball-on-disk" tribometer to simulate the lubrication in the distribution system. The various comparative studies have served to analyze how the friction and fuel consumption responded when LVEO were used both in the ICE and the complete vehicle contexts. The fuel consumption benefit found during the test was used to calculate the carbon footprint reduction when LVEO were used.

A todos los lugares y a todas las personas
que han sido mi hogar.

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Nomenclature

Acronyms

$ACEA$ European Automobile Manufacturers' Association.

API American Petroleum Institute.

$ASTM$ American Society of the International Association for Testing and Materials.

DoF Degrees of Freedom.

ECU Electronic Control Unit.

EHL Elasto-Hydrodynamic.

EPA Environmental Protection Agency.

FC Fuel Consumption.

FE Fuel Economy.

FEI Fuel Economy Improvement.

GHG Green House Gases.

HDV Heavy Duty Vehicle.

$HTHS$ High Temperature High Viscosity.

ICE Internal Combustion Engines.

LDV Light Duty Vehicle.

$LVEO$ Low Viscosity Engine Oils.

LVO Low Viscosity Oils.

OCR Oil Control Ring.

SAE Society of Automotive Engineers.

SSI Shear Stress Index.

$ULVEO$ Ultra Low Viscosity Engine Oils.

VI	Viscosity Index.
VII	Viscosity Index Improver.
WOT	Wide Open Throttle.

Greek

α	Metal to metal contact factor.
η	Dynamic viscosity.
η	Efficiency.
η_f	Fuel conversion efficiency.
η_f	Mechanical efficiency.
η_{HTHS}	High temperature high shear dynamic viscosity.
μ	Friction coefficient, mean.
μ_D	Dry friction coefficient.
μ_{EHL}	Elasto-hydrodynamic friction coefficient.
μ_L	Hydrodynamic friction coefficient.
ν	Kinematic viscosity.
ν_{100}	Kinematic viscosity at 100° C.
ν_{40}	Kinematic viscosity at 40° C.
ω	Rotatory speed.
ω_s	Shaft rotatory speed.
ρ	Density.
σ	Standard deviation.
τ	Shear stress.

Latin

$\dot{m}_f$	Fuel flow rate.
A	Oil layer area.
a, b, c, d	Constants for viscosity - temperature equations
b	Distance from load cell to crankshaft axle.
$BSFC$	Break specific fuel consumption.
D	Density at 15° C.
d_n	Piston ring nominal diameter.
F	Force.
F_f	Friction force.

F_t	Tangential force.
$FMEP$	Friction mean effective pressure.
g	Gravity (9.81 m/s^2).
h	Piston ring land width.
h_o	Minimum oil film thickness.
k	Viscometer constant of calibration.
L	Capillary length.
l	Mean hydrostatic head.
m_f	Fuel mass per cycle.
MEP	Mean effective pressure.
N	Engine speed.
N	Normal load.
n_R	Number of crank revolutions per power stroke.
P	Pressure.
P_b	Brake power.
P_f	Friction power.
$P_{i,g}$	Gross indicated Power.
P_i	Indicated Power.
P_i	Indicated pressure.
P_o	Contact pressure.
Q_{HV}	Fuel lower heat value.
r	Capillary radius.
sfc	Specific fuel consumption.
SRR	Slide to roll ratio.
T	Temperature, torque.
t	Time.
T_{10}	Fuel recovery temperature at 10% of distillation.
T_{50}	Fuel recovery temperature at 50% of distillation.
T_{90}	Fuel recovery temperature at 90% of distillation.
T_b	Brake torque.
U	Relative speed.
U_b	Ball linear speed.
U_d	Disc linear speed.
U_e	Oil entrainment speed.

V	Volume.
V_d	Volume displaced per cycle.
W	Work.
$W_{c,ig}$	Gross indicated work per cycle.
$W_{c,in}$	Net indicated work per cycle.
$W_{c,i}$	Indicated work per cycle.
W_p	Pumping work.

Chapter 1

Introduction

In this chapter the motivation for embark on the assessment of the potential of Low Viscosity Engine Oils (LVEO) to reduce the fuel consumption and the CO_2 emissions of the Internal Combustion Engines (ICE) is introduced. First, the context where this thesis has been developed is explained followed by a brief description of the role of the lubricants over the ICE performance. Following on, the purpose and significance that this study attempt to reach are explained. Then, a summary of the subsequent chapters of this thesis is portrayed.

1.1 Background and context

Globally, the second largest energy consumer sector is transportation. This sector consumes nearly 30% of the total energy produced and it has been estimated that it will keep growing in the upcoming future[1]. Within the transportation sector, road transportation is the largest and most important subcategory which consumes about three quarters of this sector's energy. It is expected that road transportation will grow steadily in the next decades, specially in the Light Duty segment, due the economic growth in China and India [2]. The main source of energy used in road transportation is fossil fuels from oil distillation, and their combustion results in harmful emissions including carbon monoxide (CO), unburned hydrocarbons (HC),nitrogen oxides (NO_x) and particulate matter (PM). Fossil fuels combustion also produce harmless emissions as carbon dioxide CO_2 which, on the other hand, has been classified as part of the Green House Gases (GHG), directly related to Global Warming. As said before, this sector is in constant growth and fuel consumption realted to transportation is expected to increase, with a daily consumption 60 million barrels in 2035, around the 61% of the total fuel oil output[3]. Given the disastrous consequences of Global Warming the challenge of the transportation sector is to keep growing but in a sustainable and responsible way. This means that OEMs need to find new technical solutions to reduce the energy consumption of the vehicles whilst protecting the environment and reducing the amount of Green House Gases (GHG) emissions involved in the transportation process. To reduce energy consumption is vital to know where and how the energy is lost during vehicles' operation. In a typical energy break down of a Light Duty Vehicle (LDV) 33% of the energy goes to exhaust gases, mainly as thermal energy, 29% of the energy goes to cooling, that is heat dissipated by the engine structure, 17% of the energy is used to overcome engine and transmission friction, 11% to overcome rolling friction, 5% goes to braking friction losses and 5% is used to overcome air drag[4]. For a Heavy Duty Vehicle (HDV), an averaged energy break down will show 30% of energy going to exhaust losses, 20% to cooling, 12.4% to overcome engine and transmission friction, 13.2% to overcome rolling friction, 7.2% going to braking friction and 13.4% overcoming air drag[5].

Technical solutions to improve vehicles' efficiency can be classified in two main categories: vehicle technologies and powertrain technologies. The former make reference to improvements in aerodynamic performance, reduction of tire's rolling resistance, auxiliary loads, and idle. It also covers mass and weight reduction and intelligent vehicle technologies. The latter makes reference to technologies to improve the efficiency of diesel and gasoline engines as well as technologies for improve efficiency in transmissions and drive axles. Some alternatives to the ICE as the electric, hybrid powertrains and H_2 fuel cells have been proposed to replace it as the road transport preferred powertrain, however, a complete replacement is unlikely to happen during the next decades. This indicates that, as well as focusing in the scientific and logistic developments required for these future technologies implementation, it is important to take immediate actions to reduce CO_2 emissions in the short term. From a realistic approach, fossil fueled vehicles provided with ICE will remain as the majority of total vehicle population in the following decades, and for this reason, research on new ways to improve their efficiency will keep being mandatory.

Being CO_2 a product of fossil fuels' combustion the main goal has to be shifted to increase ICE efficiency[6]. For more than a century, the ICE has been the preferred power source within road transportation sector, mainly due its compact and lighter construction, reliability, safeness, the easy way to be started and the higher efficiency when it is compared to the external combustion engines [7]. Other facts as the use of liquid fuels, as petrol and diesel, allows it to reach higher specific power values and good autonomy, and the wide power range which it could develop have been important to the prevalence of ICE. However, ICE development is far to be completed given the current concerns over its relatively low net efficiency and its contribution to atmospheric pollution, either by incomplete combustion residuals as NO_x, HC, CO and particulate matter or by combustion products, being in this case the CO_2 the principal pollutant [7–9].

1.1.1 Fuel consumption and CO_2 emissions

Among the undesirable emissions from the combustion process of ICE, both spark ignition (SI) and compression ignition (CI), the carbon dioxide (CO_2) outstands for being a direct product of fossil fuels' stoichiometric combustion since they are conformed by hydrocarbons chains. It means that, unlike other emissions as unburnt hydrocarbons (HC), carbon monoxide (CO), oxides of nitrogen (NO_x), oxides of sulphur (SO_x) and solid carbon particulates, which can be diminish by several technical after-treatment solutions as catalitic converters and particle traps, CO_2 is going to be directly proportional to fuel consumption[7].

From the facts presented above a straightforward solution to reduce the carbon dioxide emissions from the ICE would be to improve the overall vehicle efficiency. The breakdown of the energy taken from the fuel during the combustion process is different for each vehicle depending on many variables like vehicle dimensions, vehicle weight, equivalent frontal area, powertrain configuration, vehicle's working cycle and so on. However, in every case the energy losses can be classified into cooling, exhaust, aerodynamic, rolling, auxiliary and mechanical losses. In the same way, for each loss type represents a good starting point to improvement and to reduce the fuel consumption and CO_2 emissions.

1.1.2 Metrics to determine fuel efficiency

Across this document, the results of different studies done by diverse governments, institutions, and research groups related to gains in vehicle efficiency will be presented and discussed. The usual metrics to study vehicle efficiency changes are *fuel economy* (FE) and *fuel consumption* (FC). Even when these terms are related they are not equivalent, hence a fuel economy improvement (FEI) of 1% will differ of a reduction of 1% in fuel consumption (FC).

Fuel economy (FE) is a measure of how far a vehicle can go with a volume unit of fuel, expressed normally in miles per gallon since this parameter is used in countries where the imperial system of units is used. On the other hand, (FC) is the amount of fuel consumed in driving a given distance, usually is measured in liters per 100 kilometer [l/100 km] being this parameter used more often in the European countries to measure fuel efficiency[10].

In mathematical terms FE and FC are reciprocal, and each of the two metrics can be calculated when the other is known. During this document, the original units used by the authors of the respective cited studies will be conserved, nonetheless, the original results will be defined in terms of fuel consumption (FC) and fuel consumption reduction in percentage (%).

1.1.3 Friction losses in ICE

In general terms, friction refers to those forces acting between surfaces in relative motion. In the case of a vehicle in movement the air drag, rolling resistance and energy lost during braking would be included in that definition, however, being this thesis focused specificaly on the ICE friction losses will make reference to mechanical friction forces within the powertrain system, that is, engine and transmissions. As for general losses, the total amount of energy consumption due to friction varies widely depending on the type of vehicle and purpose. Holmberg *et.al.* [4] set the value at 16.5 % (11.5 % within the engine and 5 % in transmission) in average for Light Duty Vehicles. In the case of Heavy Duty Vehicles the energy share of friction takes different values depending on the type of vehicle; 8 % and 4 % for single unit trucks, 6 % and 4 % for semi-trailer trucks, 10 % and 8.5 % for city buses and 6 % and 4 % being all these average values for engine and transmission respectively [5]. In the case of ICE, the friction loss is mainly attributed to the following mechanical losses[8]:

- direct friction losses: it is the power absorbed due to relative motion between different surfaces as in piston ring pack -cylinder liner, camshaft bearings, main crankshaft bearings and the drivetrain system.
- pumping loss: it is the net power spent by the piston on the gases during intake and exhaust strokes.
- power loss to drive the components to charge: This is considered for engines which air supply takes place at higher pressure than atmospheric. For this purpose a mechanically driven compressor (supercharger) or a turbine (turbocharged) could be used; in the case of a supercharged engine, the engine itself supplies power to drive the compressor while in a turbocharged engine the energy to move the turbine is taken from the exhaust gases.
- power loss to drive other auxiliary engine components: it is related to the power used to drive components as pumps (water, oil and fuel), cooling fan, generator, so on.

This thesis work is focused in the reduction of direct friction losses, that means, the reduction of friction due to piston motion, journal bearing friction, and drive train friction. In two of these three main mechanical friction losses engine interfaces, the role of oil viscosity is key in terms of friction and wear reduction since they involve mixed and hydrodynamic lubrication regimes. Holmberg *et.al.* stated that around 70% of the engine friction losses can be modified through lubricant oil viscosity.

1.1.4 The use of LVEO to reduce fuel consumption and CO_2 emissions

In order to reduce engine's friction losses, the use of Low Viscosity Engine Oils emerges as one of the most interesting alternatives. Despite the fact that other sophisticated concepts to overcome vehicle's losses could lead to major overall efficiency improvements, the use of LVEO could be one of the most cost-effective way to improve vehicle's net efficiency as it is demonstrated in [10]. Besides this, it is a well known fact that the CO_2 emissions and fuel consumption reduction targets imposed by the oncoming regulations are tough, and it is more likely to meet them rather combining contributions of the diverse solutions than with a singular disruptive technology.

However, as with other technical solutions, the fuel consumption reduction given by the use of LVEO varies depending on several parameters as the type of powertrain under study, the duty cycle under the test has been made, the baseline oil used to the compare the benefit and the type of test performed, among others. Depending on the study the benefit of using LVEO lays between a 0.5% to 6% and it could be higher if combined with the use of Low Viscosity Oils for transmission and axles.

1.2 Objectives of the thesis

1.2.1 To define the order of magnitude of the fuel consumption savings expected from the use of LVEO

As described in section 1.1.4, there is a wide range of fuel consumption benefit that can be expected from the use of LVEO depending on the viscosity gradient between the oils under comparison, the type of engine and its characteristics, the type of test bench (engine test bed, chassis dynamometer, real world conditions), and the type and vocation of the vehicle. Attending this, different tests will be made, both in test benches and in real world conditions in order to clarify the magnitude of the possible fuel consumption benefits.

1.2.2 To identify the engine operating points where the effect of using LVEO has more impact over the ICE fuel consumption

Given the fact that the ICE works under variable conditions of load and speed, there will be different lubrication regimes within the engine tribo-contacts. As it is going to be explained throughout this thesis, the lubrication regime will define the capabilities of the LVEO to reduce the fuel consumption. In order to find these operating points, different screening tests will take place in engine test bed and tribometers rigs. Then these data could be validated with the fuel consumption reduction found in real fleet tests.

1.2.3 To measure the friction coefficient variation of engine tribo-contacts using laboratory equipment and the correlation with the previous findings on fuel consumption and CO_2 emissions

Another objective of this thesis is to identify the friction coefficient behavior of some of the lubricated pairs as the piston ring pack - cylinder liner, the drive train and the journal bearings, in order to find correlations with the fuel consumption benefits found during the engine benches and the real world tests.

1.2.4 To determine the cost-benefit balance of using LVEO

The final objective of the thesis is to guide the final user when deciding about either or not to use LVEO. To make it possible, a cost-benefit analysis will be made over the results of a real-world test focused on large Heavy Duty Vehicles (HDV) fleets.

1.3 About this thesis

With the objective of guaranteeing the correct comprehension of the work developed, the methodology used, the results analysis, and conclusions, this thesis has been divided in 5 different chapters.

In **chapter 1**, the motivation for undertaking the assessment of LVEO effect over ICE of Light Duty Vehicles and Heavy Duty Vehicles' fuel consumption and CO_2 emissions is portrayed. Complementary, the context, the background and objectives of this thesis are stated.

In **chapter 2**, a specific description of tribology applied to ICE environment and the correspondence with the overall engine losses and fuel consumption is made. The core of the chapter is set in the fundamental role of the lubricant to reduce and control these losses. Following up, the lubrication regime concept is introduced as well as its dependence on the load, the relative speed and the lubricant dynamic viscosity, which relationship is determined by the Sommerfeld number. Then, some of the alternatives to enhance engine efficiency depending on the prevailing lubrication regime of the engine are stated. Finally the mechanism of LVEO as a friction damper is explained as well as the constant trend in decreasing the viscosity of engine lubricant oils in the Light Duty Vehicles (LDV) segment and why this has not been directly adopted to the commercial vehicles in an easy manner.

Through **chapter 3** the experimental tools and the methodologies applied are described such as the engine test beds used to perform motored and fired tests (stationary and transient) with real engines and the specific test rigs to measure friction coefficients in tribo-contacts similar to the ones found in the engine as the Cameron Plint Machine and the Ball on disc WAM machine to emulate piston ring - cylinder liner interaction and valve train contacts respectively. In addition, the laboratory instruments used to determine the characteristics of the lubricants and the fuels used during the tests are depicted as well as the ASTM methods which define them.

In **chapter 4** the results and analysis of the different tests performed are shown. During the chapter the effects of LVEO use in friction coefficient, fuel consumption, break specific fuel consumption (BSFC) and CO_2 emissions are shown in the following order; engine tests results (torque differences between engine oils under motored conditions, BSFC differences under fired stationary conditions and total fuel consumption differences in transient conditions), the fleet test fuel consumption and CO_2 emissions differences after completing 2000000 km of operation and finally the results of friction coefficient variation due the use of LVEO in the specific tribometers: the Cameron-Plint machine TE77 and WAM machine.

The conclusions extracted from the test results and the discussion sections are stated in **chapter 5**. In addition a set of future works are proposed from the expertise and knowledge acquired during the thesis development.

References

[1] ATABANI, A. E., I. A. BADRUDDIN, S. MEKHILEF, and A. S. SILITONGA. "A review on global fuel economy standards, labels and technologies in the transportation sector". *Renewable and Sustainable Energy Reviews* 15.9 (Dec. 2011), pp. 4586–4610.

[2] INTERNATIONAL ENERGY AGENCY. "CO2 Emissions From Fuel Combustion Highlights". *IEA Statistics* (2013), p. 158.

[3] HE, Y., J. RIOS, M. CHOWDHURY, P. PISU, and P. BHAVSAR. "Forward powertrain energy management modeling for assessing benefits of integrating predictive traffic data into plug-in-hybrid electric vehicles". *Transportation Research Part D: Transport and Environment* 17.3 (2012), pp. 201–207.

[4] HOLMBERG, K., P. ANDERSSON, and A. ERDEMIR. "Global energy consumption due to friction in passenger cars". *Tribology International* 47 (Mar. 2012), pp. 221–234.

[5] HOLMBERG, K., P. ANDERSSON, N. O. NYLUND, K. MÄKELÄ, and A. ERDEMIR. "Global energy consumption due to friction in trucks and buses". *Tribology International* 78 (Oct. 2014), pp. 94–114.

[6] BERGGREN, C. and T. MAGNUSSON. "Reducing automotive emissions-The potentials of combustion engine technologies and the power of policy". *Energy Policy* 41 (Feb. 2012), pp. 636–643.

[7] PAYRI, F. and J. M. DESANTES. *Motores de Combustión Interna Alternativos*. Barcelona: Reverté, 2005.

[8] GANESAN, V. "Engine Emissions and Their Control". *Internal Combustion Engines*. Second Edition. Chennai: McGraw-Hill, 2004, pp. 495–538.

[9] HEYWOOD, J. B. *Internal combustion engine fundamentals*. Ed. by A. DUFFY and J. M. MORRISS. First Edition. McGraw-Hill, 1988, p. 750.

[10] NATIONAL RESEARCH COUNCIL. *Technologies and Approaches to Reducing the Fuel Consumption of Medium- and Heavy-Duty Vehicles*. Tech. rep. Washington D.C.: National Academy of Sciences, 2010.

Chapter 2

ICE friction losses and tribology fundamentals

The first step to address the research question of this thesis is the understanding of the fundamental concepts of physics that lie down underneath the efficiency improvements gained through the use of LVEO. Tribology emerges then, as the essential science to explain how LVEO could contribute to improve ICE efficiency, since it embraces the three key players in the CO_2 emissions and fuel consumption reduction; friction, lubrication and wear. This chapter undertakes the explanation of these concepts from the ICE perspective starting with a general summary of ICE energy balance, ICE losses, a specific review of the mechanical friction losses, the ICE efficiency definition and the role of lubrication and lubricants in the ICE operation. The later is explained through the most used way to define lubricated pairs: the Stribeck curve and the lubrication regimes, among which the hydrodynamic lubrication regime will be distinguished as the most suitable to reduce fuel consumption and CO_2 emissions when LVEO are used. Finally, a brief literature review of the use of LVEO on ICE both in LDV and HDV is made.

2.1 Introduction

As it was explained in the chapter 1, improving the vehicle efficiency in order to reduce fuel consumption and CO_2 emission of vehicles powered by ICE is the main challenge for the automotive OEMs given the rising concerns about Global Warming , alongside with the projected increasing demand of fuels and the fact that supply of fossil fuels has became more challenging during the last decade.

In general terms, efficiency could be defined as the ratio of the useful work performed by a machine or a process, to the total energy expended or heat taken in. In automotive terms the useful work would be the one which is available at the vehicle wheels and the total energy expended, the energy content of the used fuel which is typically given by the fuel's heating value. For vehicles driven by fossil fuel powertrains, the available power at the wheels often does not exceed a 40% of the total energy contained in the fuel as can be seen in table 2.1. This energy available at the wheels to move the vehicle is then used to overcome air drag, rolling resistance and braking and the respective energy breakdown will depend strongly on the type of vehicle and its duty cycle characteristics (*e.g.* for buses the air drag present a wide variation between the intercity operation and urban operation with values of 20% and 2% respectively).

Energy share	Single unit truck	Semi trailer truck	City bus	Coach	Passenger car
Exhaust + cooling	55	49	51	50	62
Engine friction + Transmission	12	10	18.5	10	17
Energy at wheels	33	41	30.5	40	21
Total [%]	100	100	100	100	100

Table 2.1: Energy distribution for different vehicles powered by fossil fuels

The fact that vehicle efficiency is diminished by several factors as aerodynamics and the thermodynamic processes within the ICE, opens the door for at least an equal number of possible work fronts when it comes to make improvements. As cited by several authors, the solutions to increase vehicle efficiency are countless and they are normally gathered as follows [1–16] :

- *ICE solutions:* Are those intrinsically related to losses occurring within the limits of the ICE, *i.e.*, exhaust losses, cooling losses, mechanical losses, so on. Some of the current solutions under development would be: combustion optimization, dual fuel engines, engine fluid preheating, engines with variable compression ratio systems, double stage turbo-compressors to recovery wasted heat, advanced valve management to reduce pumping losses, downsizing and turbocharging petrol engines, GDI systems, the use of low viscosity lubricants and the electrification of engine's accessories among others.

- *Non ICE:* All the technical solutions which address losses occurring outside the limits of the ICE, *i.e.*, air drag, rolling resistance, energy loss during braking, among others. Some of the proposed solutions are: vehicle mass reduction (either body interiors and structural mass), automatic tire pressure systems, new wheel materials to reduce rolling resistance, and the use of alternative power drives as fuel cells, fully electric vehicles, and hybridization.

- *Traffic psychology and behavior:* From a complete different perspective, some studies have been focused on how driving behavior could be changed to make it more energy efficient such as moderating acceleration, deceleration, and top speeds since the energy impact of those behaviors are most directly reflected with in-vehicle energy feedback. Claimed improvements range from 5% to 20% depending on the vehicle's technology and their duty cycle[17]. Moreover, it has been proved that reducing the legal speed limits in highways could represent a significative fuel consumption reduction in the long term[18].

The suitability of each solution depends on a variety of factors including the effective CO_2 emissions reduction, cost of implementation, penetration rate of the technical solution in the market, the political willingness to support and embrace it, among others. It is well known that low or zero carbon vehicles, powered with fuel cells or full electric engines, can reduce significantly the carbon footprint in transportation, however, the penetration rate and the cost of implementation is considerable and a complete market replacement is not expected in the following decades. Even when some of the mentioned solutions could not achieve CO_2 reductions as big as fuel cell and electric driven vehicles, they could be implemented in a short period of time, with wide market penetration and less associated costs; alternative forms of combustion, GDI engines, the use of aerodynamics fairings and stop-start systems can be adapted to current state-of-art vehicles reducing effectively the carbon footprint and closing the gap to overcome the CO_2 emissions reduction goals. Among these straightforward solutions, tribologic ones standout due their advantageous cost-benefit ratio.

Tribology is the science that studies the friction forces, wear and lubrication of interacting surfaces in relative motion [19]. In every contact where friction is present, valuable energy will be dissipated in form of heat and in the worst case scenario in permanent deformation of the surfaces, which implies material wastage and in the long term, loss of mechanical performance and fatal failures. It is unnecessary to state that ICE can not be designed without considering the effects of friction, hence the importance of ICE tribology, specially when it comes to reduce the adverse effects mentioned above which can be directly translated into increased fuel consumption, reduced engine performance and durability, maintenance interval reduction among other undesirable consequences. This friction and wear control can be done by means of a correct lubrication, which connotes the use of a lubricant between the two surfaces in relative motion.

Tribology solutions in the ICE context include, but not are limited to: low friction coatings, surface finishing methods (topography and texturing) and fuel economy lubricants, formulated with lower viscosity and with antiwear and friction modifiers additives focused on reduce friction when surfaces in relative motion are in contact.

2.2 ICE work, losses and efficiency

Accordingly to the first law of thermodynamics, energy can not be either created or destroyed, hence an energy balance can be made of inputs and outputs of the ICE as a system. Taking the vehicle as a system, it is possible to address those processes where energy is lost, hence not available to produce work at the engine outlet. Homberg *et.al.* claims that engine losses may reach up to 29% to 73% and 6% to 11.5% of mechanical losses of the fuel energy depending on the type of vehicle and duty cycle[4].

In this section the definition of some concepts required to understand the ICE efficiency is made. The definitions of efficiency used in this document are the ones used previously by Heywood[20].

2.2.1 Indicated work per cycle

The indicated work per cycle is a measure of the total work done by the combustion gases at the expansion stroke, when the piston head is pushed downwards, providing rotatory movement of the axle by means of the connecting rod. The cylinder pressure and corresponding cylinder volume can be plotted on a $P-v$ diagram as shown in figure 2.1(a).

From here, the indicated work per cycle $\mathrm{W}_{c,i}$ (per cylinder) is obtained integrating the $P-v$ diagram equation 2.1.

$$W_{c,i} = \oint p dV \tag{2.1}$$

When the ICE under study are two stroke engines the use of the equation 2.1 can be made without any further appreciations. However, in four stroke engines some ambiguity is introduced so it is of common use to use to related definitions:

- *Gross indicated work per cycle:* $\mathrm{W}_{c,ig}$. Work delivered to the piston over the compression and expansion strokes only.

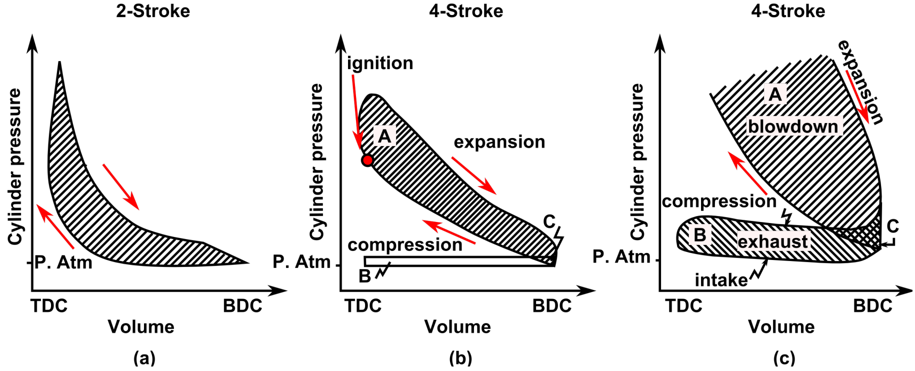

Figure 2.1: P−v diagrams for (a) 2-stroke ICE, (b) 4-stroke engine and (c) 4-stroke engine with pumping loop in detail.

- *Net indicated work per cycle:* $W_{c,in}$. Work delivered to the piston over the entire four stroke cycle.

$W_{c,in}$ takes into account the work transfer between the piston and the cylinder gases during the inlet and exhaust strokes and is called the pumping work W_p. This work will be transferred to the cylinder gases in a naturally aspirated ICE, however, in turbocharged engines this work will be transferred from the cylinder gases to piston.

2.2.2 Indicated power

From the definition of $W_{c,i}$ the indicated power (per cylinder and cycle) is defined by the equation 2.2

$$P_i = \frac{W_{c,i} N}{n_R} \tag{2.2}$$

where n_R is the number of crank revolutions for each power stroke (2 for four stroke engines and 1 for two stroke engines). As in Heywood, here the indicated work and power will make reference to the gross definition. This indicates primarily the sum of the useful work available at the shaft and the work required to overcome engine losses.

2.2.3 Brake torque and power

The brake torque T_b and brake power P_b are the value of these parameters measured at the crankshaft. Normally these values are measured with a dynamometer brake. The torque available at the shaft can be calculated by the equation 2.3

$$T_b = Fb \tag{2.3}$$

Where F is the force measured with a force cell and b is the distance between the load cell and the crankshaft rotation axle. In the same way, the brake power is defined by the equation 2.4

$$P_b = 2\pi NT \tag{2.4}$$

Where N is the crankshaft rotational speed. The primarily difference among these two parameters is that torque is the measure of the engine ability to perform work and power is the rate at which work is done. Brake power P_b, is the usable power delivered at some engine load.

2.2.4 Mean effective pressure

Given that torque as described in equation 2.3 is a measure that depends strongly on engine's size, it is not useful to make comparisons among other engines. A more accurate engine performance comparative base can be obtained by dividing the work per cycle by the cylinder volume displaced per cycle. This parameter is called as mean effective pressure (MEP), and is described by the equation 2.5:

$$mep = \frac{Pn_R}{V_d N} \tag{2.5}$$

And it also could be described in terms of torque by the expression:

$$mep = 2\pi \frac{n_R T[Nm]}{V_d[dm^3]} \tag{2.6}$$

The maximum brake mean effective pressure for engine design is well established, and is essentially constant over a wide range of engine sizes.

2.2.5 ICE efficiency

The effective efficiency of an internal combustion engine is defined by (2.7) and represents the ratio between the energy at the input, which is the energy content of the fuel and the available energy in the crankshaft.

$$\eta_f = \frac{W_c}{m_f Q_{HV}} = \frac{\frac{P n_R}{N}}{\frac{\dot{m}_f nR}{N} Q_{HV}} = \frac{P}{\dot{m}_f Q_{HV}} \tag{2.7}$$

where W_c is the work effectively available in the crankshaft and Q_{HV} is the lower heating value of the fuel and m_f is the mass of fuel inducted per cycle. This efficiency term is also known as fuel conversion efficiency. It has to be mentioned that the energy supplied by the fuel is not totally converted in thermal energy in the combustion process (sometimes referred as the fuel conversion factor with values around 92% to 97% due incomplete combustion[21] and volumetric efficiency which is the ratio between the actual amount of air filling the combustion chamber and the theoretic maximum amount of air). As these efficiencies affect the combustion process and the indicated power available at the piston head they are not going to be discussed in this document.

2.2.6 ICE mechanical efficiency

As seen in section 2.2.1 part or the gross indicated work per cycle $W_{c,ig}$ is used to perform the inlet and exhaust strokes. An additional portion is used to overcome the friction of different engine components and to drive engine accessories. All these power requirements are usually grouped together as friction power P_f. Hence:

$$P_{ig} = P_b + P_f \tag{2.8}$$

As said before P_f takes into account the engine mechanical losses that can be subdivided in:

- direct friction losses: is the power absorbed due to relative motion between different surfaces as in piston ring pack -cylinder liner, cam shaft bearings, main bearings and drive train system.
- pumping loss: is the net power spent by the piston on the gases during intake and exhaust strokes.

- power loss to drive the components to charge: This is considered for engines which air supply takes place at higher pressure than atmospheric. For this purpose a mechanically driven compressor (supercharger) or a turbine (turbocharged) could be used; in the case of a supercharged engine, the engine itself supplies power to drive the compressor while in a turbocharged engine the energy to move the turbine is taken from the exhaust gases.

- power loss to drive other auxiliary engine components: is related to the power used to drive components as pumps (water, oil and fuel), cooling fan, generator, so on.

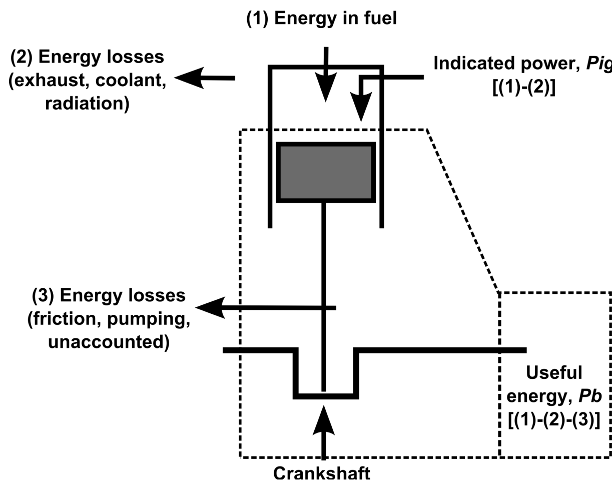

Figure 2.2: Energy flow in an ICE and mechanical efficiency

The mechanical efficiency can be described then by the ratio of useful power delivered in the shaft P_b and the indicated power P_{ig} by the expression 2.9

$$\eta_m = \frac{P_b}{P_{ig}} = 1 - \frac{P_f}{P_{ig}} \tag{2.9}$$

A diagram of the energy flow in an ICE can be seen in figure 2.2. In general, mechanical efficiency of engines varies from 65% to 85%[22] and it depends strongly on engine speed: as the engine is throttled, the mechanical efficiency decreases.

2.2.7 Specific fuel consumption and brake specific fuel consumption (BSFC)

The specific fuel consumption is an important parameter that reflects how good the engine performance is, defined by the fuel flow rate per unit power output, as stated in equation 2.10. It defines how efficiently an engine is using the fuel supplied to produce work.

$$sfc = \frac{\dot{m}_f}{P} \tag{2.10}$$

Brake specific fuel consumption (BSFC) and indicated fuel consumption (ISFC), are the specific fuel consumption measurements based on brake and indicated power.

2.3 Mechanical losses in Internal Combustion Engines (ICE)

Power losses related to friction are inherent between surfaces in relative motion, both sliding and rotatory. As seen in the equation (2.8) friction power losses are given by the difference between the indicated power available at the piston head P_{ig} and the power at the crankshaft P_b.

2.3.1 Friction losses in ICE

The engine friction losses contributors can be seen in figure 2.3. These friction contributors are going to be explained in detail in the following sections.

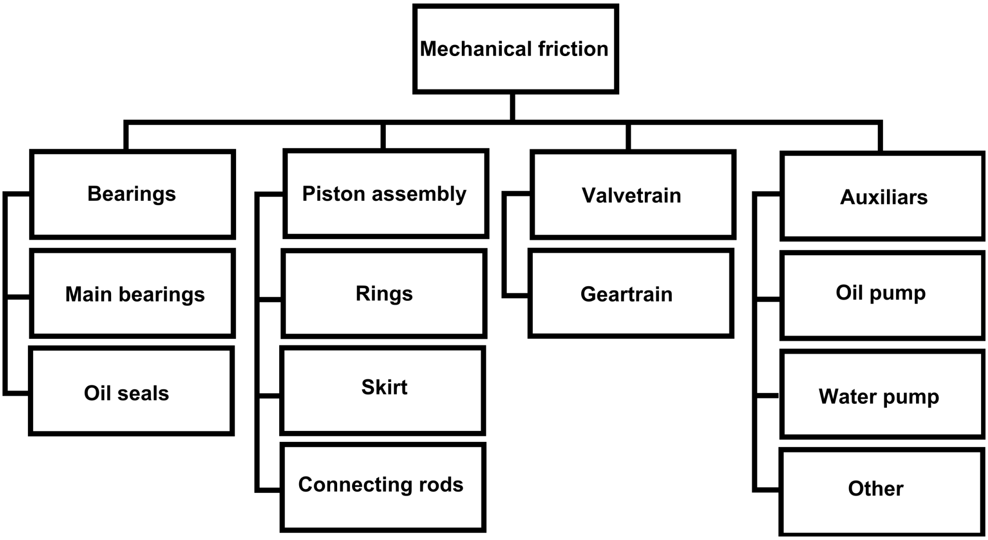

Figure 2.3: Mechanical losses in ICE (Adapted from [23])

Piston assembly

The main task of the piston is to convert thermal energy into mechanical work. Furthermore, the piston rings seal the combustion chamber from the crankcase and transfers heat to the coolant. The piston skirt acts as a load-carrying surface, which keeps the piston properly aligned within the cylinder bore.[24]

The piston assembly is the main contributor to friction losses in the ICE [25] (see figure 2.4). The friction losses of this assembly come from the interaction of the cylinder liner, compression rings, oil control ring and piston skirt. Additionally, the piston throughout the combustion cycle exhibits significant secondary motions: lateral movement across the cylinder and rotation around the gudgeon pin causes the piston to tilt in the bore [26]. In the following paragraphs the different friction contributors are going to be explained in detail.

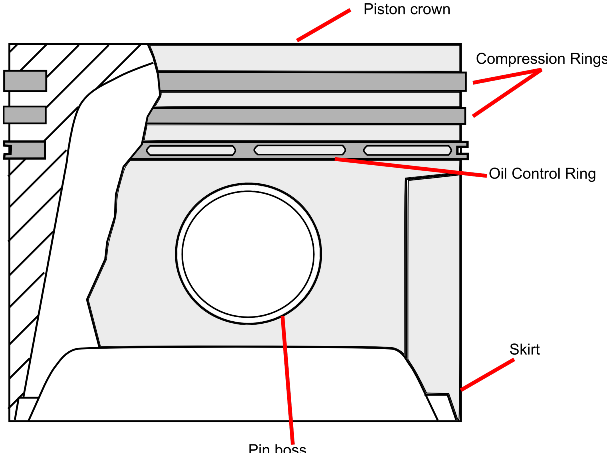

Figure 2.4: Piston assembly components.

Ring pack power loss: The main purpose of the piston rings is to prevent gas leakage from the combustion chamber to the crankcase. If the ring is performing well the pressure of the oil film between the ring face and the liner will prevent blow-by as well. In addition to the task of seal off the combustion chamber from the crankcase of the compression rings a mechanism to distribute the lubricant oil evenly onto the liner is needed. This is made by the Oil Control Ring (OCR). The oil control ring is perforated by slots in the peripheral direction which provides a way to excess oil to leave the ring pack area. The oil control ring may have a coil spring inserted as pre-tension of the ring is not sufficient in all instances.

The forces acting on the piston ring can be seen in the figure 2.5 and they are:

- Gas force above and below the ring.
- Viscous traction and lubricant pressure force within the film.
- Piston ring tension.
- Reaction from the contacting flank between the ring and the piston ring groove.

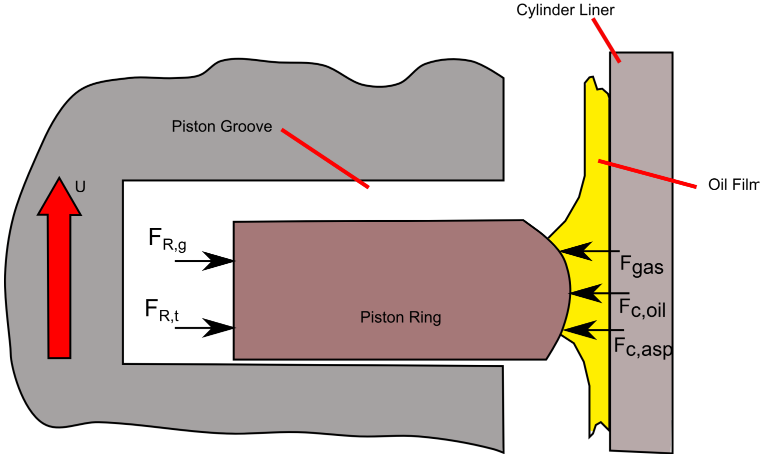

Figure 2.5: Forces acting on a single piston ring.

friction losses are higher at mid-stroke due to high speed [27–29]

Piston skirt power loss: The friction between the piston skirt and the cylinder liner is controlled by the diameter clearance , the piston skirt design, the tilting action of the piston, the surfaces characteristics and the lubricant conditions. Under normal operating conditions the lubrication conditions for the piston skirt and cylinder liner surfaces mirror the lubrication conditions of the ring pack, with lower contact pressures.

Engine bearings

The crankshaft and the connecting rods convert the reciprocating motion of the pistons into rotating motion. The journal bearings that accomplish the transmission of load and keep the crankshaft in place are designed to run under hydrodynamic conditions which is a special kind of lubrication where there are no contact between moving parts, this topic will be explained in section 2.4.1. h_o is the minimum oil film thickness between the shaft and the bearing. If h_o is sufficiently thick to completely separate the running surfaces the interface would be at hydrodynamic conditions. As ICE stop frequently during operation, the shaft and the bearing would be in contact at the starting moment.

In terms of bearings power loss, it would be assumed that, aside from starting moments, all the power loss comes from shearing of the lubricant viscous friction.

Valvetrain

Typically, an ICE's valvetrain group comprises the use of valves, valve springs, valve spring retainers, valve keys, rocker arms, piston rods, lifters/tappets, and a camshaft [30].

The camshaft is the rotating component that transforms it into linear valve motion in order to control the fluid flow into and out of the combustion chamber. There are several configurations of the engine's valvetrain, however, the most commonly used are: poppet, sleeve and rotatory valvetrains.

Among them, the poppet valve system is the preferred in the automotive industry. The cam profile of the diverse cams placed in the camshaft, drives the valve opening crank angle, closing crankangle, duration and valve lift. Several variations of the poppet valve system have been proposed and used, and some of them are depicted in figure 2.7.

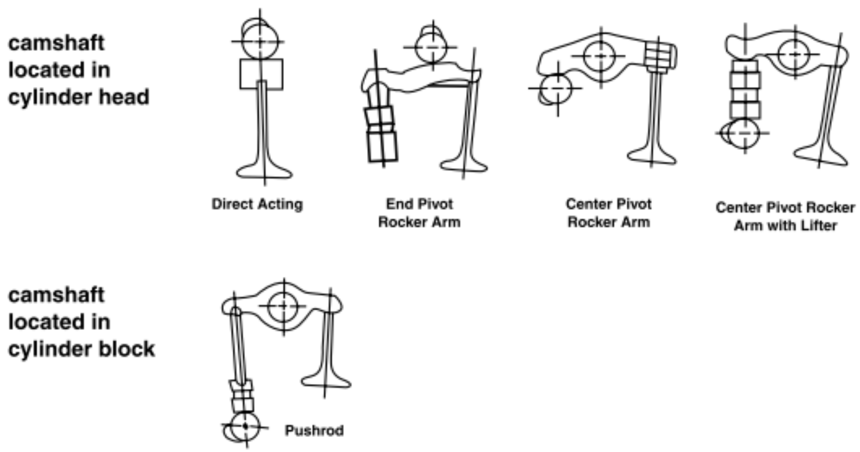

Figure 2.6: Different configurations of the valvetrain system (1)Direct acting, (2) End pivot rocker arm, (3) Center pivot rocker arm, (4) Center pivot rocker arm with lifter (5) Pushrod.

A comparison of friction values for the differente valvetrain configurations was done by Tung *et.al.* [8]. The best configuration in terms of low friction is roller follower (see figure **??**) since rolling friction is less by and order of magnitude than sliding friction. Other factors relevant to friction control as mass, least number of components and low number of friction interfaces should be considered.

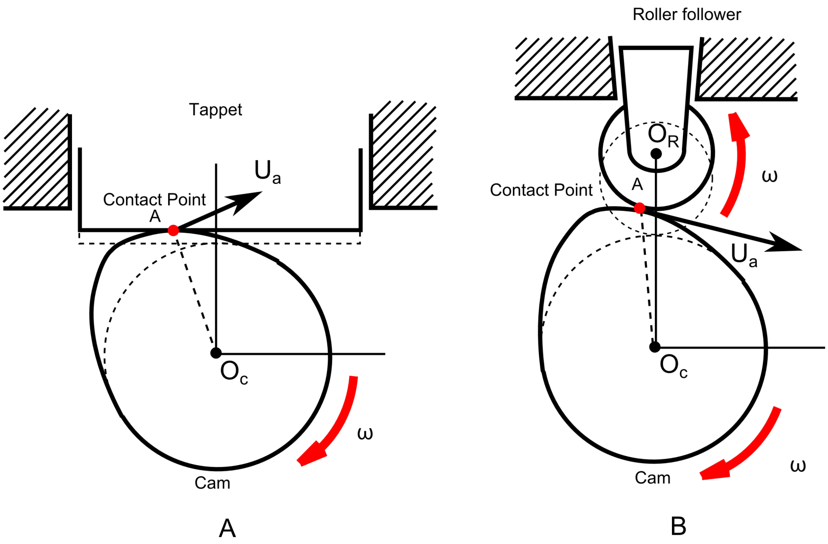

Figure 2.7: (a)Cam-tappet and (b)Cam-roller follower valvetrain configurations.

2.3.2 Friction losses distribution and design considerations

In the table 2.2 the distribution of total mechanical friction given by several authors is shown. It is well established that the major part of friction losses comes from the piston assembly, followed by the engine bearings and the valvetrain. It has to be stated that this is normally true for most of automotive engines at normal operation speeds, however at lower speeds, the valvetrain friction could take the second position in friction losses after the piston assembly [31].

Authors	Piston Assembly	Bearings	Valvetrain	Auxiliars
Wakuri (1995)	30% - 40%	35% - 50%	5% - 10%	-
Taraza (2000)	40% - 50%	20% - 30%	7% - 15%	20% - 25%
Taylor (2000)	42% - 46%	39% - 49%	5% - 19%	-
Comfort (2003)	40% - 50%	30% - 30%	7% - 15%	20%
Tung (2004)	57%	35%	8%	-
Holmberg (2012)	38% - 68%	20% - 44%	3% - 34%	10%
James (2012)	40% - 55%	20% - 30%	7% - 15%	-
Holmberg (2014)	45% - 55%	20% - 40%	7% - 15%	10%

Table 2.2: Friction losses distribution by several authors

Some design considerations which should be taken into account since affect directly the engine friction losses are:

- *Stroke-bore ratio:* lower stroke-bore ratios led to lower FMEP mainly due less frictional effective area.
- *Engine size:* smaller engines have smaller surfaces, hence lower FMEP.
- *Piston rings tension:* reducing the nominal contact pressure of the rings reduces the friction with the cylinder wall. However, as counter effect, this could increase blow-by resulting in oil consumption and pollutant emissions increase.
- *Journal bearings:* The FMEP can be reduced by lowering the diametrical clearance of journal bearings.
- *Engine speed:*Friction increases rapidly with increasing speed. At higher speeds mechanical efficiency starts deteriorating rapidly, this is one of the reasons for restricting engine speed.
- *Engine load:*Operation points with high load values increase the pressure at the back of the piston rings which results in increased values of friction. In contrast, higher pressures and frictions lead to higher temperature values within the combustion chamber, including the lubricant oil which will suffer a viscosity decrease that could reduce the friction losses if the lubricant layer is thick enough to prevent contact of moving parts. In diesel and modern Gasoline Direct Injection (GDI) engines frictional losses related to load tend to be constant since they do not depend on a throttle position as in the case of regular gasoline engines.

2.4 Lubrication in ICE

Lubrication is essential to reduce friction and wear of engine components. A close up on lubrication functionality is discussed on the following section, then a discussion over the specific requirements of the ICE is made.

2.4.1 General lubrication theory

To correctly understand the lubrication mechanism, consider two surfaces in contact with each other. In order to produce relative movement between surfaces a tangential force should be applied, at the same time, the weight of the upper surface is acting perpendicularly to the movement and is called normal load W. the ratio between the friction force and the normal load is known as the friction coefficient as it can be seen in equation 2.11

$$\mu = \frac{F_f}{W} \tag{2.11}$$

The resistance between moving surfaces can be diminished with the introduction of a layer of a substance such as oil or grease in between [19, 22, 32]. The substance used is the lubricant and its lower shear strength resistance interposed between the surfaces is the reason why the friction is reduced when the movement between parts takes place. This layer prevents the direct contact between surfaces in spite of its small thickness range of 1-100 μm. The lubricant can be gaseous, liquid or solid and the suitability of each option will depend strongly on the contact stress. If the contact has low relative speed and high loads a solid lubricant will be more appropriate. The most used model of surfaces in relative motion lubricated with any substance is the Stribeck model which describes the friction coefficient μ in term of the Sommerfeld number, defined by the specific characteristics of the contact: relative speed $[U]$, supported load $[W]$ and the dynamic viscosity of the used lubricant $[\eta]$ as it is shown in the equation 2.12

$$Sommerfeld = \frac{\eta U}{W} \tag{2.12}$$

Stribeck found this relation in the early 20^{th} century while studying the limits of lubrication in journal bearings and plot the results in the commonly named Stribeck curve (see figure 2.8). The friction coefficient μ commonly ranges from 0.001 to much higher levels of 0.2 or more [19, 25]. Besides the definition given by the equation 2.11 the friction coefficient can be described by means of the equation 2.13 where μ_D is the metal to metal dry friction coefficient, μ_L the hydrodynamic friction coefficient and α is the metal to metal factor which varies from 0 (complete surfaces separation) to 1 (intense contact between surfaces).

$$\mu = \alpha\mu_D + (1-\alpha)\mu_L \tag{2.13}$$

From the Stribeck curve and the variation of the friction coefficient μ three major lubrication regimes can be described

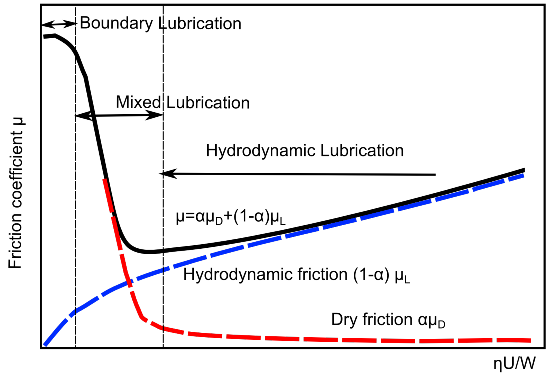

Figure 2.8: Stribeck curve.

Hydrodynamic lubrication

This lubrication regime is characterized by a full separation of the surfaces in relative motion by the action of the lubricant layer as it can be seen in the figure 2.9(a). There are two conditions that must be fulfilled: the surfaces must be in relative motion with enough velocity to generate sufficient pressure to generate the lubricant film and it has to be some angle different to zero between the surfaces. This lubrication regime is often referred as ideal since low friction coefficients are obtained and wear is nonexistent ($\alpha = 0$). A high viscous lubricant (η $\uparrow$), lower loads ($W \downarrow$) and high relative speeds ($U \uparrow$) will favor the appearance of this lubrication regime.

Boundary lubrication

On the other corner of the hydrodynamic regime is the boundary lubrication regime where the surfaces in relative motion are in contact, hence $\alpha = 1$ as is shown in figure 2.10. Since the lubricant layer has been broken and metal to metal contact is taking place the contact will be dominated by the surfaces characteristics as rugosity, elasticity module, hardness, and the chemistry at a molecular level of the lubricant additive layers covering the metal surfaces. If the reader is interested in the additives used in the oil and the chemistry involved please see further information in Heinemann[33]. Lubricant viscosity does not affect the friction coefficient in this lubrication regime.

Mixed lubrication

This regime is the result either of increased entrainment speed, oil viscosity increase or load decrease from boundary conditions which will lead to lubricant film formation. Even when this film is very thin it can support some of the load applied to the contact. As a result a sharp drop in the friction coefficient is expected and it can be seen in 2.8. The friction coefficient will reach its minimum in this regime where the load in the interface is completely support by the lubricant film. If any of the variables continue to favor the appearance hydrodynamic lubrication the friction coefficient will increase due the enlargement of lubricant film [34].

Elastohydrodynamic lubrication

This type of lubrication can be defined as a special form of hydrodynamic lubrication where the changes in viscosity due to extreme pressure and the elastic deformation of the bodies in the interface is fundamental. Hydrostatic lubrication is obtained by introducing the lubricant into the loaded area at a pressure high enough to separate the surfaces with a relatively thick lubricant film, making unnecessary the relative motion unlike hydrodynamic lubrication. The lubricant films are very thin, ranging from 0.1 to 1 [μm]. Generally this regime operates between non-conformal contacts, but it can also occurs between conformal interfaces as highly loaded journal and pad bearings.

2.4.2 Lubrication regimes of the main lubricated interfaces of ICE

As it was discussed before, depending on the load [W], relative speed [U] and lubricant viscosity [η] each tribo-contact interface will be located at one of the three lubrication regimes explained in section 2.4.1. Regarding the ICE, as described in section 2.2.5, the principal focuses of friction are: the piston assembly, the journal bearings and the valvetrain. However, it would be of special interest to know the specific lubrication regime of each of these engine interfaces in order to adopt the best approach to reduce the related losses.

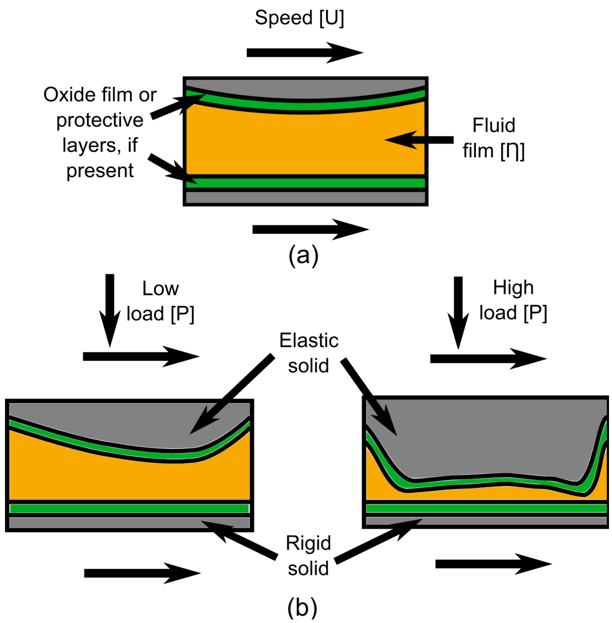

Figure 2.9: Regimes of hydrodynamic lubrication (a)fully hydrodynamic regime (b) elastohydrodynamic lubrication. (Adapted from [34])

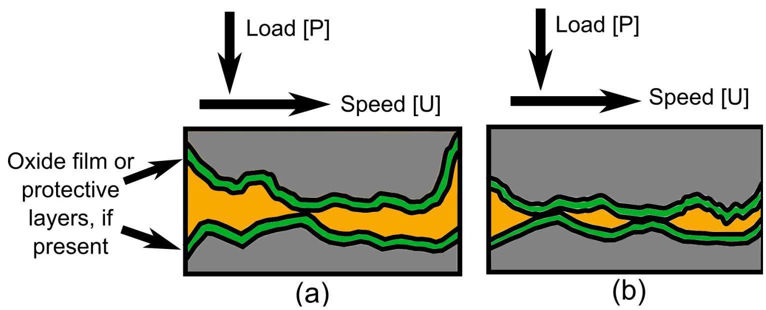

Figure 2.10: (a) Boundary lubrication and (b) mixed lubrication.

Piston assembly

The piston assembly, is located in the heart of the energy transformation from chemical to mechanical energy from the fuel combustion in the combustion chamber to a rotatory movement of the crankshaft. The piston assembly is shown in figure 2.11. One of the important factors while studying piston assembly tribology is that total friction is the summation of friction between cylinder/liner and different parts of the piston like the piston ring pack and the piston skirt. The lubrication of this interface is very complex since it depends on several factors which change constantly like, lubricant viscosity, lubricant temperature, entrainment speed, etc. It can be observed that whenever piston changes its direction, it becomes stationary for a very short period of time. From several studies it has been determined that lubrication in piston assembly varies from boundary-mixed in the semi stationary points to hydrodynamic at mid-stroke points where entrainment speed is high enough to form a lubricant layer which prevents the contact between the cylinder liner and piston ring pack and skirt [8, 27, 35–37].

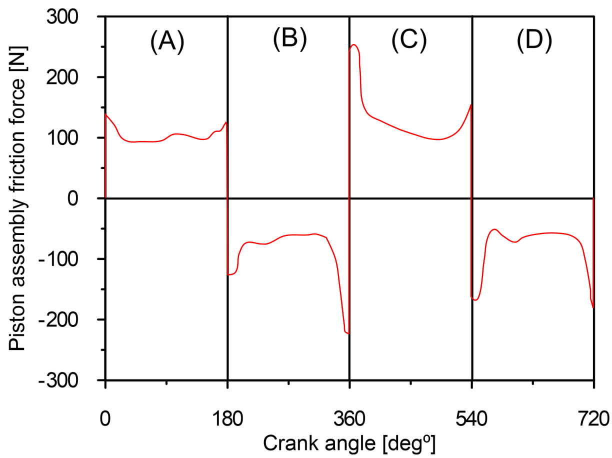

Figure 2.11: Schematic piston assembly instantaneous friction force. (Adapted from [38]).

Under any operating conditions the most challenging lubrication conditions can be found at the start of the power stroke because of the presence of boundary lubrication due to peak combustion pressure[39], however, as it was exposed in section 2.3.1 the frictional losses are higher at mid-stroke than they are at piston dead centers due the high speed. Therefore, tackling engine friction would imply the friction reduction at the mid-stroke typically under the hydrodynamic lubrication regime.

Engine bearings

Bearings are provided with lubricant in order to reduce the friction coefficient, control the operation temperature and evacuate possible wear particles. Accordingly to their purpose, both crankcase and connecting rod journal bearings will be working, in absence of failure, under the hydrodynamic lubrication regime given the fact that high rotational speeds (up to 6000 [min^{-1}]) of the crankcase and the associated components enables the appearance of enough pressure in the lubricant film to sustain the crankcase while operating[19, 20, 22, 25, 34, 36]. However, at starting conditions when oil has not been pumped from the sump yet, these bearings work briefly under boundary and mixed lubrication regimes. Once this initial moment has passed, the wedging action of the crankshaft climbing towards the surface of the bearings makes possible the oil film formation between surfaces. In the figure 2.12 the pressure distribution on the oil film within the interface under hydrodynamic regime is depicted. The tribology of journal bearings is complicated since lubricant supplies, thermal effects, dynamic loading and elasticity must to be taken into account [8].

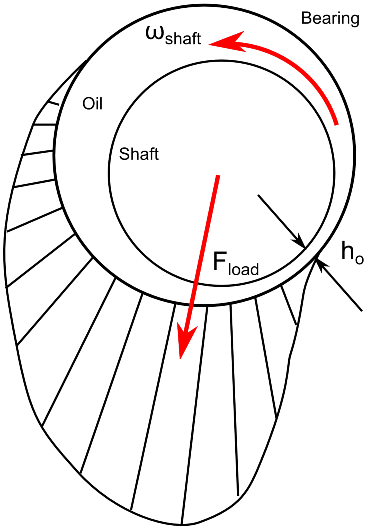

Figure 2.12: Lubricant pressure distribution in a dynamically loaded journal bearing. Adapted from [23]

Valvetrain

This interface is normally associated with boundary, elastohydrodynamic and mixed lubricating regimes. The specific lubrication regime will depend on several factors as the valvetrain mechanical solution, oil temperature and engine operating speed and load among others.

In this engine subsystem the friction between the cam and tappet is the most important contributor to overall mechanical losses. Research has shown that this particular interface works under elastohydrodynamic lubrication regime [36].

In figure 2.13 the three main friction sources of ICE's are related with the principal lubricant regimes over the Stribeck curve seen in section 2.4.1.

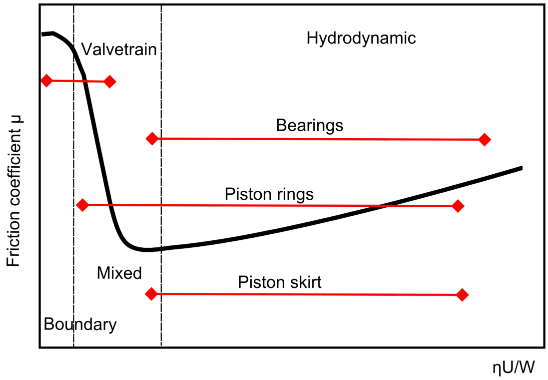

Figure 2.13: Engine main tribo-contacts and their respective lubricant regimes during operation.

2.4.3 Engine oil characteristics

Different to other type of lubricants, engine oil should work in a highly complex environment, under multiple lubrication regimes: *i.e.* normally works under high temperatures, it deals with the presence of other hydrocarbons and works in an environment which favors chemical reactivity. For these reasons, engine oil should be consider rather a design element than a simple engine consumable. Engine oil results crucial not only reducing the inner friction but to other vital areas as preventing wear, transmitting forces, neutralizing undesired products, avoiding corrosion, sealing the piston assembly to prevent undesired blow-by and cooling the engine while operating.

Nowadays, along to the previous mentioned requirements, modern engine oils should fulfill as well some special and somehow contradictory goals as the ability to reduce fuel consumption, reduced oil consumption, reduction in harmful exhaust emissions, extended oil drain intervals (ODI), improved durability and present low volatility, among others.

Engine oils composition

Typically an engine oil consists of a base oil and additives or agents. The composition normally is around 60-85% base stock and 15-40% additives[19]. The rheological properties of the oil will depend on the characteristics of the base stock which in most cases is chemically inert. For automotive applications there are two main types of base stock: the mineral bases, extracted from oil, used when temperature requirements are moderate like gear oils, bearings, and ICE. On the other hand, the synthetic bases, artificially developed to substitute mineral oils for specific applications like high performance machinery both at extreme low or high temperatures. The base stocks normally are classified in Groups depending on the type of base and the processes required to obtain them. Nowadays, 5 different groups have been established, being the first three refined directly from petroleum crude oil. Group IV are fully synthetic (polyalphaoleofin) oils and Group V is reserved for all the base stocks not included in the previous four groups. In extend, Group I contains oils from the most simple solvent-refined process, therefore, they are the cheapest in the market. Group II bases are manufactured by hydrocracking. This oils have better antioxidation properties and a clearer color. Group III bases, as Group II are hydrocraked, this time under more sevear pressure and heat, resulting in a purer oil. Given the complexity of the hydrocracking, sometimes this oils are also described as synthetic. Group IV base oils are the polyalphaolefins (PAOs). These base oils come from a synthesizing process. They are great for extreme temperature (cold and high heat). Group V include other type of base stocks as silicone, glycol, biolubes, so on[40].

Some of the main type of additives and their respective functions are: detergents, which neutralize the acids and inhibit varnish formation in the crankcase, ashless dispersants, which disperse the soot and other oxidation products. Antioxidants, that inhibit oil oxidation and thickening. High pressure additives, to prevent wear. Corrosion protection, to prevent undesired corrosion. Viscosity index improvers, to reduce the viscosity drop due temperature increase. Antifoaming agents, which prevent foam formation. Friction modifiers, to reduce friction when the lubricant layer is not thick enough to separate surface in relative motion[21].

Viscosity

The viscosity of a fluid is its resistance to flow, or in other terms, the force required to move a given layer of fluid past another layer at certain speed and certain separation[41]. This fluid flowing over a stationary surface does not move as a whole at the same speed. There is a continuous variation from zero at the surface to a maximum at the farthest distance from it (see figure 2.14).

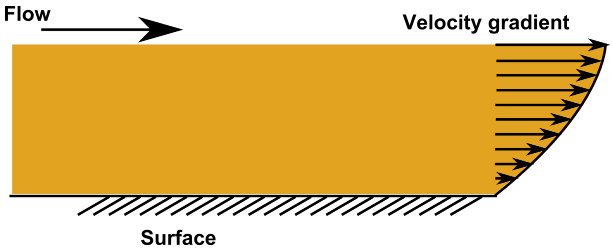

Figure 2.14: Flow of a liquid over a surface.

The viscosity gradient of the oil produces a shear rate ($\frac{U}{h}$ in [s^{-1}]) due the displacement of the one layer over another. The force (F in [N]) acting upon the oil layer with area (A in [m]) causing it to flow is called shear stress (τ in [Pa]) which can be expressed by the equation 2.14

$$\tau = \eta \frac{U}{h} \tag{2.14}$$

The viscosity unit in the CGS system is the Poise (P), defined as the force in dynes required to move a 1 cm^2 layer of fluid parallel to another static layer of 1 cm^2 at a distance of 1 cm within the fluid at a speed of 1 $\frac{cm}{s}$. This is the definition of dynamic vicosity η. The dynamic viscosity is more commonly expressed, particularly in ASTM standards, as centipoise (cP). In the International system, the unit of dynamic viscosity is the Pascal-second (Pa s), which is identical to 1 kg m^{-1} s^{-1}, being 1 cP = 1 mPas.

The kinematic viscosity (v) is the ratio of the viscous force to the inertial force or fluid density ρ, given by the equation 2.15.

$$v = \frac{\eta}{\rho} \tag{2.15}$$

The physical unit for kinematic viscosity is the Stokes (St), named after George Stokes, and as in the case of engine oil, it is commonly expressed in terms of centistokes (cSt). The International units of kinematic viscosity are m^2s^{-1}[42].

Viscosity and temperature relationship

The viscosity of lubricant oils varies extremely with the engine operating temperature. With increasing temperature, the viscosity could fall rapidly; in some cases the fall could be around 80% with a temperature increase of 25%[19]. Exists several viscosity-temperature equations, both empirical and derived from theoretical models and the choice to use one or another relies in the application. Reynolds presented an early equation with a very limited range of application 2.16 where a and b are constants and T the temperature in absolute Kelvin [K]. Other equations as Slotte (equation 2.17), Walter (equation2.18) and Vogel (equation 2.19) are more precise and more accurate, very useful in engineering calculations (a, b, c, d are constants and ν is the kinematic viscosity [m^2s^{-1}]).

$$\eta = be^{-aT} \tag{2.16}$$

$$\eta = \frac{a}{(b+T)^e} \tag{2.17}$$

$$(\nu + a) = bd^{\frac{1}{T^c}} \tag{2.18}$$

$$\eta = ae^{\frac{b}{(T-c)}} \tag{2.19}$$

Viscosity index (VI)

The Viscosity Index is an empirical parameter that establishes how sensitive is certain engine oil viscosity to temperature changes. By definition the viscosity index is and inverse measure of the decline in oil viscosity with temperature, this means that high VI values indicates less decline of viscosity as temperature rises. A wider explanation on how VI was defined and the calculation formula can be found in ASTM D2270[43].

Viscosity modifiers

To prevent an excessive viscosity decline at high temperatures often find at engine operation, viscosity modifiers have been included in oils formulations to extend the applicability of the regular base stocks. Viscosity modifiers increase the VI because they are more soluble in the base stock at high temperatures than at low temperatures.

Viscosity-shear relationship and Shear Stability Index (SSI)

It is common to think about liquid lubricants as Newtonian fluids with their viscosity changes being proportional to a shear rate, as defined in equation 2.14. This is usually true for mineral base stock oils up to relative high shear rates 10^5 s^{-1} - 10^6 s^{-1}, but at higher shear rates the lubricants starts to behave as Non-Newtonian fluids, making the viscosity dependent on shear rate. To the particular case of engine lubricants, the Non-Newtonian behavior is also pseudoplastic, which is associated with thinning of the fluid as the shear rate increases; during this process of shearing in polymeric fluids, long molecules which are randomly oriented tend to align given a reduction in apparent viscosity. Most of modern engine lubricants are mineral or synthetic base stocks containing polymeric VI improvers that stick out the pseudoplastic behavior of the oil. A typical polymer solution gives a behavior as plotted in 2.15; viscosity remains steady up to a critical shear rate, after which it falls linearly to reach a second stable zone. Shear rates in lubricant applications range from low values to the order of 10^6 s^{-1}.

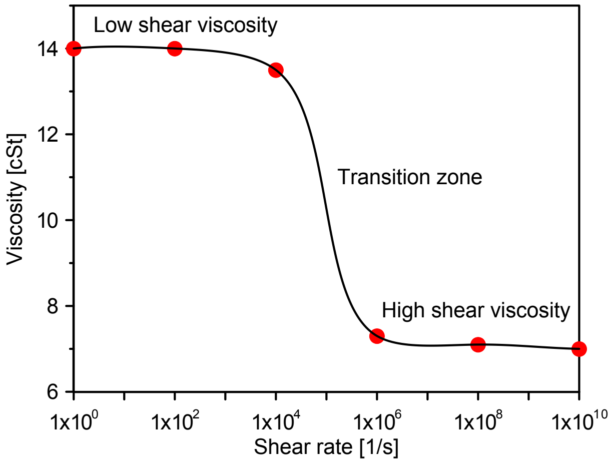

Figure 2.15: Viscosity dependence on shear rate

Depending on the shear rate applied to the lubricant, the temporary viscous loss could become permanent due high temperature o high energy situations making the polymer undergo into a permanent change by ruptures of the molecular chains. This results into a permanent viscosity loss of the lubricant which will be determined by the molecular weight and structure of the polymer and therefore, a measure of the performance of a certain oil can be given by the Shear Stability Index (SSI). This parameter can be measured by the ASTM D6278 and ASTM D7109[44, 45].

2.4.4 ICE lubrication systems

As it was explained in section 2.2.5 there are several interfaces in the ICE where part of the work transferred from the combustion gases to the piston is dissipated in form of friction. This friction not only leads to efficiency reduction but ultimately to wear in components in relative motion which in the worst case scenario could end up in ICE failure. As it was shown at the beginning of section 2.4 the use of a lubricant is essential to counter back the undesired effects of mechanical friction. The lubrication system in ICE has been set up to provide sufficient quantity of cool oil to give sufficient lubricant oil to all the moving parts of the engine. There are three types of lubrication systems regarding the ICE:

- mist lubrication system
- dry sump lubrication system
- wet sump lubrication system

Mist lubrication systems are typically used in two stroke engines where there is not enough space to place a crankcase. In dry sump lubrication systems the oil supply is carried in an external tank and it is dragged to the engine by means of an oil pump. Since the oil has to be returned to the tank a scavenging pump is used to remove the oil from cylinders and bearings and fed it back to the supply. Since these two types of lubrication systems are not commonly used in passenger cars and commercial vehicles their description will be outside of these thesis scope, if the reader is interested some authors like Ganesan make a good description[22]. Finally, in wet sump lubrication systems (see figure 2.16) a fully flooded sump houses the lubricant, from there the oil pump suck out the oil and transports it through the lines to the consumers. Since the oil volume circulated over time increases with the engine speed, a valve limits the pressure to values around 0.5 MPa, depending on the engine model, in order to prevent damages in the oil filter and seals. The oil passes trough an oil cooler before the main line diverge in several to reach the different lubricated mechanisms of the engine one of these branches goes to crankshaft and connecting rod bearings and through bores in the later to piston pins. Another branch goes to camshaft, tappets and rollers. Some other systems like the turbocharger and the injection pump could be lubricated from the main system as well. Once the different engine parts have been lubricated the unpressurized oil flows to the sump.

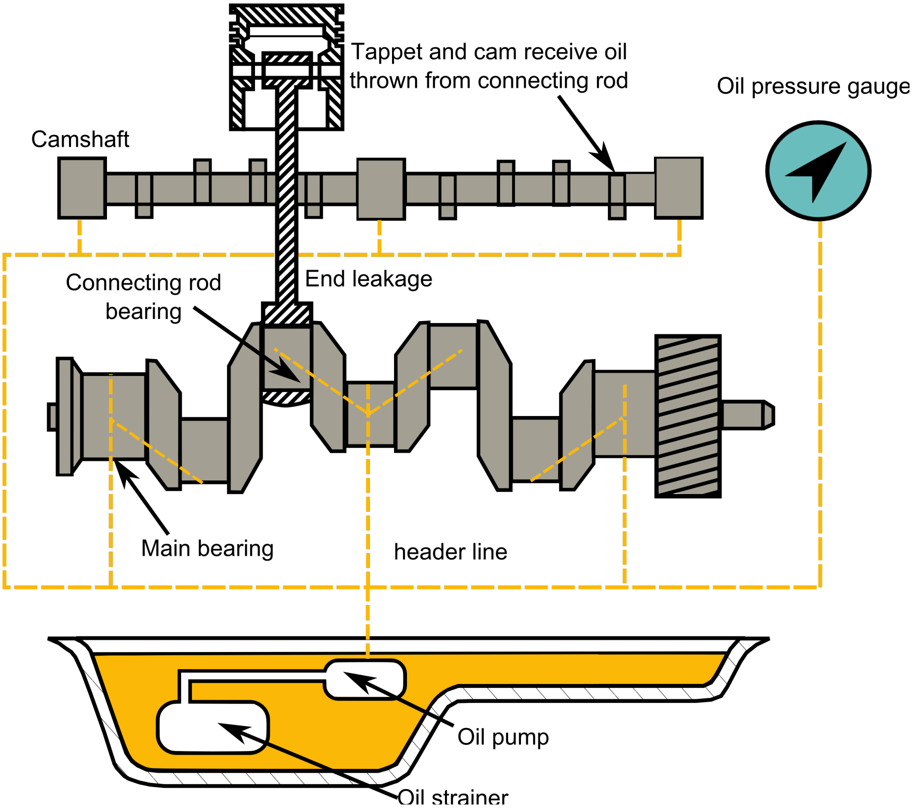

Figure 2.16: Wet sump lubrication with pressure feed system.

2.5 Low viscosity engine oils and their effect on CO_2 emissions and fuel consumption

From the friction and lubrication in ICE facts exposed in 2.3.1 and 2.4 respectively it is clear that a possible way to reduce overall engine mechanical losses is to reduce engine friction, which means to reduce the friction coefficient at the interfaces as bearings, piston assembly and valvetrain. As it was shown in figure 2.13, given the complexity of an ICE all types of lubrication regimes are present during engine's operation. Hence, from a tribological point of view, friction reduction has two possible approaches:

- To reduce friction coefficient at boundary and mixed lubrication regimes through the use new materials or the use of friction modifier additive packages.

- To reduce friction coefficient at hydrodynamic lubrication regime by means of reducing oil's viscosity.

As Fenske *et.al.* exposed, solutions that reduce surface asperities will reduce the friction coefficient at boundary and mixed lubrication regimes whereas solutions focused on reducing viscous losses will reduce the friction coefficient ar hydrodynamic and mixed lubrication regimes. While friction reduction at boundary and mixed regimes can be done regardless the viscosity of the oil, the opposite is not always true, since lubricant viscosity reductions can lead to increased friction coefficients if the resulting lubricant layer is not thick enough to prevent contact. That is, lowering boundary friction while keeping viscosity the same reduces total friction, however, lowering viscosity while keeping boundary friction constant does little at high loads and low boundary friction enables the use of LVEO[13].

Holmberg *et.al.* have summarized the lubrication regimes of the different engine tribo-contacts, as can be seen in tables 2.3 and 2.4.

Friction [%]	Contact	Regime
45 (45-55)	Piston Assembly	Several
30 (20-40)	Bearings	Hydrodynamic
15 (7-15)	Valvetrain	Mixed
10	Pumping and hydraulic	Viscous Losses

Table 2.3: Distribution of engine friction losses for the different tribo-contacts and their respective lubrication regimes

Friction [%]	Lubrication regime
40	Hydrodynamic
38	EHL
11	Mixed
11	Boundary

Table 2.4: Distribution of friction losses for the piston assembly and its respective lubrication regimes

As exposed in section 2.1, the use of LVEO presents a great cost-benefit ratio, which means that a considerable fuel savings can be achieved with low technical effort compared with other solutions.

2.5.1 Towards the use of LVEO

The use of LVEO has been a way to reduce vehicles' fuel consumption driven by the public environmental concern and the quest for energy independence. However, the penetration level of this solution varies depending on the market and the type of vehicle, *i.e.* the SAE viscosity grade has been changing in passenger cars during the past decades towards lower viscosity grades as can be seen in figure 2.17, however, this has been not the same for the Heavy Duty segment where concerns about possible wear appearance have delayed their use.

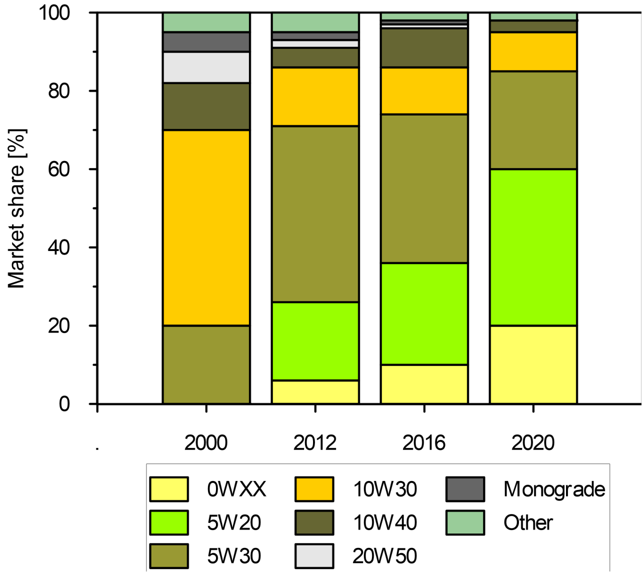

Figure 2.17: SAE viscosity grades share for different decades for Light Duty Vehicles (LDV) in the United States.

Base stock, VI and SSI impact on fuel consumption

Accordingly with the factors exposed in sections 2.4.3 and 2.4.3, the physical properties of an engine oil are given by the nature of its base stock. The effect of the base stock viscosity index (VI) can be measured blending oils to meet specific SAE grade requirements using different base stocks. van Dam *et.al.* did this, blending 4 different oils which met SAE 10W40 specifications, using a different base each time (groups I to IV) and running a Volvo D12D FE test using a SAE 15W30 oil as baseline.

The results showed that fuel consumption was higher for the oils belonging to groups II, III and IV, being the latter the most extreme case with almost 5% increase at some point of the Volvo D12D test [46]. This counter intuitive behavior (is usually accepted that synthetic oils should enhance fuel consumption reduction) is explained later in a subsequent study where the effect of viscosity index improvers (VII) was assessed. The lubricants with high VI base oil require only a very small amount of VII, then, they do not experience a high HTHS viscosity drop when exposed to shear. The authors state that the best way to optimize the formulation of an engine oil is to select the base stock to meet a kinematic viscosity at 100° C target for the fresh oil, being the Groups III and IV capable to be selected for low SAE grades without any concern about the oil being sheared out of grade during use[47].

In the same study, the effect of the shear stability index (SSI) was tested. The results turned out to be very similar to those found during the VII studies where, if two oils were formulated to meet a fresh kinematic viscosity at 100° C value, one with low SSI and other with high SSI, the former will present better fuel consumption benefits after shearing due to the viscosity loss (see figure 2.18). Once again, it remarks the importance of design the lubricant to meet certain characteristics after completing some shearing cycles, when it reaches its operative form.

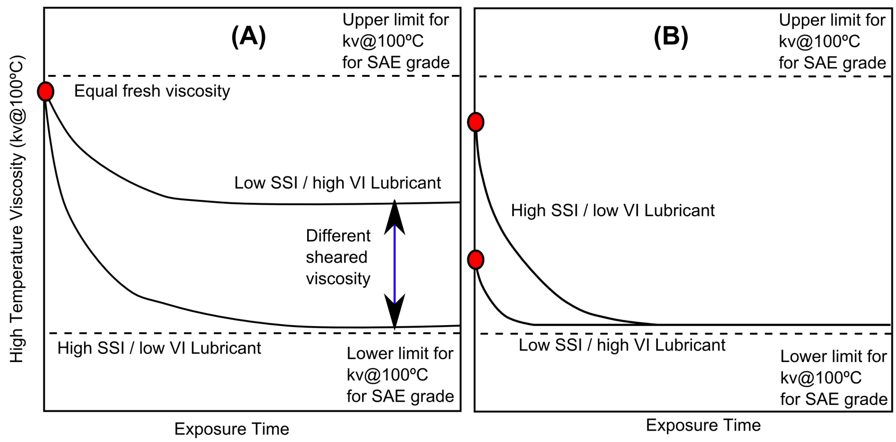

Figure 2.18: Viscosity vs. exposure time for oils formulated with base stocks with different viscosity index (VI) and stability shear index (SSI). Depending on how the oil is formulated, high VI and low SSI can be a fuel consumption disadvantage (a) or an advantage (b). (Adapted from [47])

HTHS viscosity

As presented in sections 2.4.3 and 2.4.3 the viscosity of engine oils is sensible to temperature and share rate changes. Given the engine operation requirements, with high shear rates where the apparent viscosity declines temporary, ASTM established a High Temperature High Shear viscosity (HTHS), more representative to the conditions in the engine. Tipically HTHS viscosity values are taken at 150° C and 10^{-6} s^{-1} of shear rate by several ASTM procedures which can be found in SAE J300[48, 49].

Several authors have studied the relationship of the HTHS viscosity with the fuel consumption benefits of a specific oil. Kaneko *et.al.* found that reducing HTHS viscosity of engine oils a reduction up to 3.5% could be found during the FTP-75 cycle. However, the reduction was not linear, being the optimum HTHS viscosity value around 2.6 cP, the bottoom limit given to a SAE XXW20 grade (see figure 2.19). Souza de Carvalho *et.al.* have found a relationship between HTHS viscosity and specific fuel consumption: increasing lubricant HTHS viscosity increased the specific fuel consumption linearly. The Willans' lines showed that operating the engine with a high viscosity monograde lubricant produces higher friction power than an operation with low viscosity multigrade oil for various points of operation of a passenger car diesel engine [50].

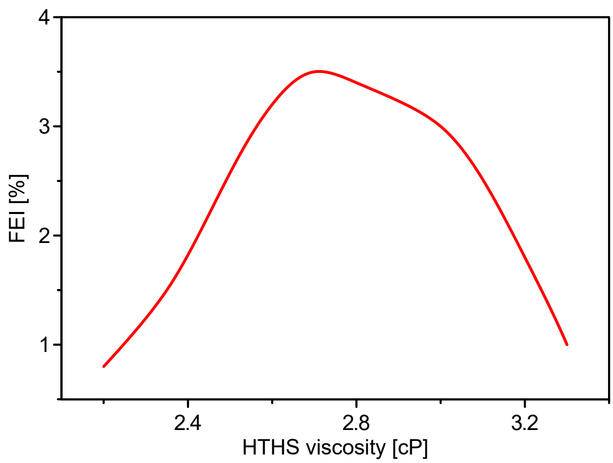

Figure 2.19: Fuel Economy improvements and HTHS viscosity relationship for several oil formulations, tested accordingly to CEC L-54-96 standard engine fuel economy test. (Adapted from Kaneko et. al. [51])

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In the same way, Van Dam *et.al.* studied the influence of the oil viscosity in a deeply, finding correlations of each type of viscosity with the resulting fuel economy improvement in the Volvo D12D test. The viscosity types under study were: SAE grade (as seen in figure 2.20), kinematic viscosity at 40° C, kinematic viscosity at 100° C and HTHS viscosity. The results showed that the parameter with greatest correlation was the HTHS viscosity after 90 cycles of operation (R^2=0.92), followed by kinematic viscosity after 90 cycles (R^2=0.91)[47].

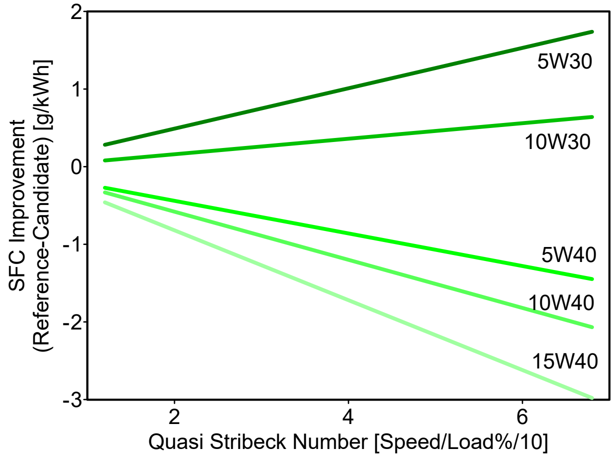

Figure 2.20: Fuel economy improvements on Volvo D12D test for several oil formulations

In a recent study, Maroto-Centeno *et.al.* studied the effects of the three main drivers of an engine fuel consumption reduction potential: the HTHS viscosity η_{HTHS}, the boundary friction coefficient μ_D and the EHL friction coefficient μ_{EHL}. They found that the parameter with more incidence over later fuel economy improvement values on a CEC L-54-96 test was the HTHS viscosity[52]. Figure 2.21 presents the correlation claimed by the authors.

2.5.2 Oil standards regarding fuel consumption

As for other engine components, the engine oil characteristics are controlled and standardized by specific standards that guarantee the correct performance of the oil under the severe conditions within the engine. Oil attributes like, pour point, seal compatibility, volatility, deposits control, viscosity under several conditions and fuel economy are defined by the standards. Depending on the type of vehicle (Light or Heavy Duty), the type of service and the engine manufacturer origin, different standards will be required. Typically there are two main organisms that verify that oils meet the requirements. The EOLCS (Engine Oil Licensing and Certification System) which include API, SAE and ILSAC committees with oil standards for vehicles to be used in North America and Japan, and ACEA (European Automobile Manufacturers Association) which represents the 15 Europe-based car, van, truck, and bus makers. While in the EOLCS countries, Light Duty and

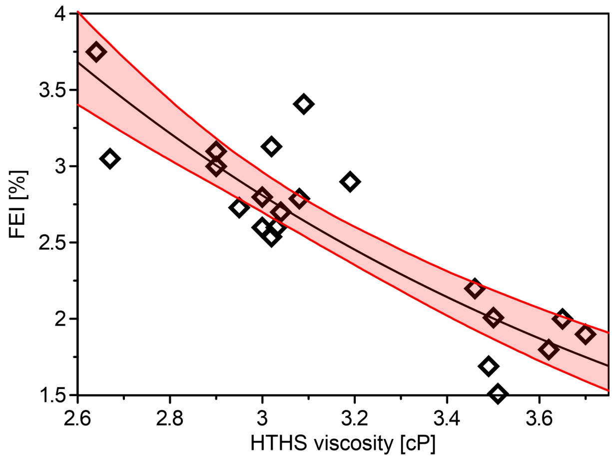

Figure 2.21: Fuel Economy improvements and HTHS viscosity relationship for several oil formulations, tested accordingly to CEC L-54-96 standard engine fuel economy test. (Adapted from Maroto et. al. [52])

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Heavy Duty engine oil standards are handed by different associations (ILSAC and API respectively), in the European market only ACEA hand down these standards. In terms of viscosity however, both standards adopted the SAE J300 definitions for oil viscosity as it is going to be explained in the following section.

SAE J300

The rheological performance of a given lubricant oil is given by its viscosity grade, determined by tests completed in four different viscometers. Each viscometer tests the oil performance at different operation conditions as low temperature dynamic viscosity (ASTM D5293), the low temperature pumping dynamic viscosity (ASTM D4684), the kinematic viscosity at 100° C (ASTM D445) and the dynamic viscosity at high temperature and high shear (ASTM D4683, ASTM D4741, ASTM D5481, CEC L-36-90). The failure criteria of these test are stipulated in SAE J300 and will define the SAE viscosity grade of a given engine oil[49]. From the four viscosity measurements mentioned above, the High Temperature High Shear viscosity (HTHS) appeared in 1992 SAE J300 version to define the rheological behavior at engine operative condition both in terms of temperature and shear stress. A direct correlation of this parameter with fuel economy was found, however, concerns about the oil film thickness ability to prevent wear came up. This classification has been upgraded several times, the most recent on January 2015, marking a milestone in terms of low viscosity lubricants: grades 16, 12, and 8 which present values as low as 1.7 $mPas$ in HTHS viscosity. The driver to introduce new categories is fuel economy as it was exposed on [53].

The complete SAE J300 can be seen in table 2.5[49] .

SAE Viscosity Grade	Low-Temperature (° C) Cranking Viscosity $mPas_{max}$	Low-Temperature (° C) Pumping Viscosity $mPas_{max}$	Low-Shear-Rate Kinematic Viscosity (mm/s) at 100° C Min	Low-Shear-Rate Kinematic Viscosity (mm/s) at 100° C Max	High-Shear Rate-Viscosity ($mPas$) at 150° C Min
0W	6200 at -35	60000 at -40	3.8	-	-
5W	6600 at -30	60000 at -35	3.8	-	-
10W	7000 at -25	60000 at -30	4.1	-	-
15W	7000 at -20	60000 at -25	5.6	-	-
20W	9500 at -15	60000 at -20	5.6	-	-
25W	13000 at -10	60000 at -15	9.3	-	-
8	-	-	4.0	<6.1	1.7
12	-	-	5.0	<7.1	2.0
16	-	-	6.1	<8.2	2.3
20	-	-	6.9	<9.3	2.6
30	-	-	9.3	<12.5	2.9
40	-	-	12.5	<16.3	3.5 (0W-40, 5W-40, and 10W-40 grades)
40	-	-	12.5	<16.3	3.7 (15W-40, 20W-40, 25W-40, 40 grades)
50	-	-	16.3	<21.9	3.7
60	-	-	21.9	<26.1	3.7

Table 2.5: SAE J300-SAE viscosity grades for engine oils

ILSAC GF-5 and GF-6

The ILSAC (International Lubricants Standardization and Approval Committee) is formed by AAMA (American Automobile Manufacturers Association) and JAMA (Japan Automobile Manufacturers Association) to define the parameters and specifications required by their vehicles' engine oils. The current standard is the

GF-5 which presented a remarkable intention to improve fuel economy compared with the GF-4 standards. At the moment GF-6 is being developed and it will take effect probably on January 2018. Once again, it would be of paramount interest to increase again engine oils fuel economy, in such a way that GF-6 will include two subcategories, GF-6A and GF-6B, one with compatibility with oils with SAE viscosity grade 20 as bottom limit and the other category focused on the new SAE LVEO and ULVEO.

PC-11

This second engine oil specification, is for Heavy Duty engine oils. Further segmentation of this engine oil category is made based on backward compatibility. Two subcategories are based on a difference in High Temperature High Shear viscosity (HTHS) rates. The subcategory PC-11A, will be compatible with the current HDEO category (API CJ-4)and have an HTHS of 3.5 minimum; the second subcategory, PC-11B, is for new lower viscosity engine oils with an HTHS of 2.9-3.2. Probably the first license date for PC-11 will be in December 2016.

ACEA standards

There are ACEA specifications for passenges car motor oils (the A/B class) for catalyst compatible motor oils (the C class) and for Heavy Duty diesel engine oils (the E class). The classes are further divided into categories to meet the requirements of different engines. The A/B class's A1/B1 and A5/B5 oils have lower HTHS viscosities, which means that they provide better fuel economy but they may not provide adequate protection in engines that are not designed for them. ACEA A3/B3 and A3/B4 on the other hand require oils with higher HTHS viscosities that may not provide as good fuel economy as an A1/B1 or A5/B5 oil but may offer better engine protection in certain engine designs. The categories within the C class are devided along SAPS limits and along HTHS viscosities. C1 and C4 are low-SAPS oils, while C2 and C3 are mid-SAPS oils. On the other hand C1 and C2 oils have lower HTHS viscosities, while C3 and C4 oils have higher HTHS viscosities.

2.5.3 CO_2 emissions and fuel consumption vehicles' legislation

As it was mentioned before, the use of LVEO has been addressed as one effective solution to reduce the fuel consumption in the Light Duty Vehicles segment.

Japan legislation

Regarding the Light Duty Vehicles segment (defined by Japanese authorities as vehicles up to 10 passengers and freight vehicles with a gross vehicle weight of 2.5 tons or less), the fuel efficiency parameter has been the fuel efficiency [km/l] and limits has been set since the last decades of the past century. The two most recent updates to this segment have taken place first in 2007 to take effect in 2015 with fuel economy improvements of 23.5% for passenger cars, 7.2% for small buses (more than 10 passengers and up to 3.5 tons) and 12.6% for small freight vehicles compare to actual values measured in japans fleet in 2004[54]. The most recent update took place during 2015 to take effect in 2020 and will represent an improvement for the passenger cars segment of 19.6%[55]. On the other hand in 2006 Japan was the first country to set fuel economy standards for Heavy Duty Vehicles up to 20 tons to take effect in 2015[46]. The legislation set a fuel efficiency parameter called "Weighted Harmonic Average" which is obtained by dividing the total fuel consumption value by the shipped volume. The required improvements for 2015 were about 12.2% compared with vehicles of the same category shipped in 2002.

United States legislation

US has been controlling fuel economy and Green House Gases (GHG) for Light Duty Vehicles since the 1970's by means of the Corporate Average Fuel Economy (CAFE) standards for passenger cars and light trucks. At that time the prime goal was to reduce fuel consumption by a half (13.6 mpg in 1974 to 27.5 mpg for passenger cars by 1985). Even when the United States was a pioneer when these fuel economy and Green House Gases limits were implemented, during the following decades the nation stood behind other developed countries. However, in 2003 some of the limits were updated for light trucks in order to reach a fuel economy of 22.2 mpg by 2007 and, in 2009 given the public concerns about Global Warming and the back-on-the-day rising oil prices, the National Highway Traffic and Safety Administration (NHTSA) boosted the fuel economy enhancing program for the United States fleet setting targets as ambitious as decrease CO_2 emissions at a rate of 4.2% per year and increase fuel economy by 3.7% per year from 2012 to 2016. Nowadays, these limits have been taken further and for the oncoming years (2017-2025) an annual reduction of 4.6 % per year in the CO_2 emissions, and a 4.25% increase per year in miles-per-gallon fuel economy are expected. Regarding the Heavy Duty Vehicles segment, the first GHG and fuel economy legislation not appeared until 2011, to cover models year 2014-2018, being mandatory for all vehicles from 2016.

European Union legislation

Starting in the 1990's decade, the EU introduced voluntary limits to reduce the CO_2 emissions average of the European fleet. In 2009 the limits turned mandatory and new goals were adopted in order to limit the CO_2 emissions to 130 gCO_2/km by 2015, 95 gCO_2/km by 2020 and 68-78 gCO_2/km by 2025. This legislation only covers the Light Duty Vehicles segment, however, there is a current effort to include Heavy Duty Vehicles in oncoming legislation updates. It has to be mentioned that EU only measures CO_2 an no other GHG.

2.6 Previous research on LVEO effects over fuel consumption in Light Duty (LDV) and Heavy Duty Vehicles (HDV)

2.6.1 Research on Light Duty Vehicles (LDV)

Manni *et.al.* studied engine oil bulk properties' effects over engine emissions, both in engine bed and chassis dynamometer performing ECE15 and EUCD cycles. One of the main results showed the important role of oil's viscosity to reduce fuel consumption, particularly in urban running conditions, using Low Viscosity Engine Oils with the correspondent decrease in carbon dioxide emissions. The engine tests kept in the standard configuration of its electronic control unit (ECU) pointed out an ambiguous effect of rheology on exhaust emissions: low viscosity oils gave higher HC and CO emissions but guaranteed lower NO_x emissions[56]. Tseregounis *et.al.* compared the fuel consumption performance of two different engine oils, a SAE 5W20 and a SAE 5W30, in a chassis dynamometer test under the EPA FTP test. The use of the LVEO resulted in 1.5% gains in fuel economy over a the SAE 5W30 oil. No significant gains in fuel economy were observed during the cold transient portion of the FTP test [57]. Mufti *et.al.* demonstrated that the solely influence of the viscosity over piston ring pack-cylinder liner interface should be taken carefully: using the indicated mean effective pressure (IMEP) to determine the piston assembly friction losses they found that, the friction benefits of using a SAE 0W20 oil with no additive package over a SAE 5W30 were vanished after the engine's temperature reached high temperatures (80° C). Friction curves of the piston assembly showed how the piston assembly lubrication regime was prevalent under mixed and boundary conditions.

Smith studied the interaction of different friction modifiers additives with different LVEO in the piston ring pack - cylinder liner interface [58]. The main goal was to find an engine oil formulation capable to maintain the friction reduction gained during the hydrodynamic regime by lowering the engine oil dynamic viscosity using friction modifiers.

Fontaras *et.al.* carried out a series of measurements to investigate the effect of LVEO on fuel consumption and emissions profile of diesel engines. Two passenger cars (Euro 3 and Euro 4 tehnology) and a test bench engine were used in the experiments. Fuel consumption benefits around 3% were found during NEDC chassis dynamometer testing and further gains were found at test cell engine measurements in specific engine working zones, however, the authors state that being the NEDC a non representative cycle of real world conditions, a real expectation on fuel consumption reduction would be around 1.5% when LVEO are used in passenger cars [59].

Kaneko *et.al.* presented a study where a diesel engine oil was specifically designed to achieve 2% of fuel consumption improvements compared with a SAE 5W30 oil. Taking advantage of the European fuel consumption cycle (NEDC) an oil with similar HTHS but lower kinematic viscosity at 40° C was formulated. An engine bench test was performed to verify the improvements[51]. Park *et.al.* conducted a study of the relationship of oil's viscosity and friciton modifiers with engine friction. A motored diesel engine provided with a torque meter in the coupling was used. The pistons were drilled to reduce pumping loss during the compression stroke and several parts of the engine were removed if they did not affect engine friction related to engine oil and engine oil temperature was controlled. 5 different engine oil formulations were used, however, it should be mentioned that the viscosity effect was only studied indirectly varying the temperature from 30° C to 140° C, the different formulations corresponded to different degrees of fricton modifier additives. The study found a significant engine friction reduction as temperature risen, the authors claimed correlations of those friction tests with previous fuel economy tests, concluding that a 11% friction reduction led to 1% fuel economy differences[60].

Another good example of LVEO effects over fuel consumption in small ICE was presented by Singh *et.al.* testing the fuel consumption diferences of a motorcycle when two different engine oils were used, a SAE 20W40 oil as baseline and a SAE 5W30 as candidate. The test was made on a chassis dynamometer and the results showed fuel economy improvements around 4% depending on the fuel used [61].

One interesting study was made by Guinter *et.al.* to show how the homologation cycles used to certificate the fuel consumption benefits of certain engine oil can not be correlated with real-world behavior. In the study the sequence VID was used. Proposing a more realistic engine oil temperature and ageing values, the authors enhanced the ability of the sequence to find significance in the fuel consumption differences given by the use of LVEO. The study main significance relies in the fact that under sequence VID SAE 5W20 oil gave similar fuel consumption numbers to a SAE 0W16, one of the new SAE J300 classifications introduced specifically to satisfy the OEMs callings to reduce the engine oil viscosity[62]. Another study which involved new SAE J300 LVEO was presented by Manni *et.al.* where the effect of the new SAE grades on fuel economy was tested in a chassis dynamometer throught two common cycles: the NEDC and ARTEMIS[63]. Oils used had an HTHS viscosity range decreasing from 2.9 to 2.0 cP, in -0.3 cP steps. The study results showed how the fuel consumption presented an inicial downward trend as HTHS decreased. However, both for NEDC and ARTEMIS cycles, the oil with lower HTHS (2.0 cP) presented a negative effect compared with the previous oils as fuel consumption resulted to be higher[64]. In a recent study, Taylor *et.al.* [31] have studied the influence of ultra low viscosity engine oils (ULVEO), formulated to meet the SAE 8 grade specifications (at that moment yet to be included in the SAE J300), over engine friction mean effective pressure (FMEP), using as a SAE 0W20 as baseline in a motored rig test over a wide range of temperatures. For all conditions the SAE 8 oil reduced the FMEP values.

Fleet tests are rather difficult to find in the literature since they are usually expensive in terms of time and logistics. Most of the information available comes from tests run by the oils manufacturers as the study presented by Castrol, involving around 2100 vehicles of a variety of OEMs, each of them completing a total mileage of 2000 km using SAE 15W40 engine oil as baseline and a SAE 5W30 as candidate oil. The average fuel consumption benefits reported were around 5.54%, however, no information about the test methodology is available[65].

2.6.2 Research on Heavy Duty Vehicles (HDV)

As in the case of passenger cars segment, the first momentum of fuel saving automotive technologies came after the 1970's oil crisis. Among other devices as lower rpm/higher torque engines, transmissions and rear axle rear ratios redesign, improved aerodynamics and the use of radial tires, the use of fuel saving lubricants emerge as an effective way to reduce fuel consumption. In the following section some previous works addressing the LVEO use and fuel consumption reduction correlation are exposed.

Using the Volvo D12D test Van Dam *et.al.* tested the fuel consumption effects of using LVEO in a Heavy Duty engine. Using a SAE 15W30 as a baseline oil, they calculated a Fuel Economy Improvement (FEI) index expressed in percentage. The outcome of the study showed how engine oils with higher SAE grades in the second number (XXW40) presented negative FEI values, being this more noticeable at hilly conditions. On the other hand, engine oils with lower viscosity presented positive FEI of 0.6% (5W30 with low HTHS on flat conditions) [46].

In further studies Van Dam *et.al.* studied the correlation of some engine oil bulk properties like cold crank simulation viscosity (CCS), kinematic viscosity, high temperature high shear viscosity (HTHS) both for fresh and used oil(after the 90 cycles Bosch Shear Test[66]). The study showed that HTHS viscosity of oils aged by the 90 cycles procedure was the property with better correlation with FEI (R^2=0.92) [47].

In another extensive study, Carden *et.al.* carried out engine bed tests on a Heavy Duty engine to adress the effects of one LVEO (HTHS 2.71cP) and two ULVEO (HTHS 2.26cP and 1.82cP) over fuel consumption and wear using the radiotracing Thin Layer Activation method. Controlling test temperatures (coolant to 90° C, fuel to 40° C, engine oil to 90° C and 130° C and cell air to 20° C), and measuring fuel consumption with a gravimetric balance, European Stationary Cycle tests (ESC) were run. Results showed that a 23% HTHS viscosity reduction from the baseline value gave 0.6% of improvement in break specific fuel consumption (BSFC) during the ESC cycle. In the case of the oil with even lower viscosity a 36% HTHS viscosity reduction gave an improvement around 0.9%. Additional tests were undertaken to measure the effects of the same oils over engine friction by means of motored and fired tests. The impact of the ULVO was the greatest at minimum load conditions, however, the effect of the reduced viscosity was lower and less clear at full load. Regarding the wear studies, it was demonstrated that the possible risks associated with engine oil with reduced viscosity can be addressed with proper additive package formualtion[67].

Fleet testing studies to measure the effects of LVEO on HDV fuel consumption are harder to find than engine and chassis dynamometer tests given the amount of background noise which implies testing under uncontrolled conditions. Some of the first studies presented by Hetrick *et.al.* involved a reduced number of vehicles (two matched pairs of tank trucks with two stroke diesel engines) conducted over a 190 km route. As baseline oils, truck used SAE 40 for the engine and SAE 80W90 for transmission and rear axles and as candidate low viscosity counterparts, a SAE 30 was used for engine and transmission. Results over the 2 day test exhibited a fuel economy benefit of 3.4%[68]. In a subsequent study, Keller *et.al.* used a match couple of 4 stroke diesel engine trucks completing a 4 days test over a highway route to assess the fuel consumption benefits of several oil formulations against a mineral SAE 30, SAE 80W90 and SAE 85W90 oils for engine, transmision and rear axle respectively. The data showed a fuel economy benefit of 1.4% for a synthetic SAE 30 oil and 2% for a SAE 5W20 oil. From these results, subsequent ambitious fleet tests were made extending both the number of vehicles involved and the test duration. In one of those tests, 15 vehicles with 9 baseline vehicles (5 using regular SAE 30 and 4 units using SAE 15W40) and 6 vehicles using the synthetic SAE 30 were tested during 16 months, resulting in fuel economy benefits of 4.4% between the synthetic oil for the former and 3.1% with the latter. On a subsequent test, a total number of 22 vehicles with 6 units using synthetic SAE 30 and 16 on SAE 40were tested for 16 months as well. In this case the synthetic oil exhibited a fuel economy benefit of 2.7%[69].

Browne *et.al.* presented a study where two significantly different buses were tested over a two week period in a test track using two commercial engine oils; a SAE 15W40 as baseline (HTHS 3.97cP) and a SAE 5W30 as candidate (HTHS 3.55 cP). During the test, the buses run the First Millbrook Fuel Economy Test (FMFET)[70] composed by a rural and an urban section. Results showed an overall fuel consumption benefit of 1.5% being more prominent on the urban part of the cycle for both buses[71].

References

[1] CALWELL, C. *California State Fuel-Efficient Tire Report, Volume II, California Energy Commission*. Tech. rep. California Energy Commision, 2003.

[2] HM TREASURY. *The King Review of Low Carbon Cars: Part II - recommendations for action*. Tech. rep. March. London, 2008, p. 114.

[3] JOCSAK, J. "The Effects of Surface Finish on Piston Ring-pack Performance in Advanced Reciprocating Engine Systems". Master Thesis. Massachusetts Institute of Technology, 2005.

[4] HOLMBERG, K., P. ANDERSSON, and A. ERDEMIR. "Global energy consumption due to friction in passenger cars". *Tribology International* 47 (Mar. 2012), pp. 221–234.

[5] TAYLOR, A. M. K. P. "Science review of internal combustion engines". *Energy Policy* 36.12 (2008), pp. 4657–4667.

[6] BAKER, H., R. CORNWELL, E. KOEHLER, and J. PATTERSON. *Review of Low Carbon Technologies for Heavy Goods Vehicles*. Tech. rep. June. Ricardo, 2010, pp. 1–206.

[7] ALAGUMALAI, A. "Internal combustion engines: Progress and prospects". *Renewable and Sustainable Energy Reviews* 38 (2014), pp. 561–571.

[8] TUNG, S. C. and M. L. MCMILLAN. "Automotive tribology overview of current advances and challenges for the future". *Tribology International* 37.7 (July 2004), pp. 517–536.

[9] ECONOMY, V. F. *Assessment of Fuel Economy Technologies for Light-Duty Vehicles*. National Academies Press, National Research Council, 2011.

[10] WIESENTHAL, T., G. LEDUC, J. KÖHLER, W. SCHADE, and B. SCHADE. *Research of the EU automotive industry into low-carbon vehicles and the role of public intervention*. Tech. rep. October 2015. JRC- IPTS European Commission, 2010.

[11] HOLMBERG, K., P. ANDERSSON, N. O. NYLUND, K. MÄKELÄ, and A. ERDEMIR. "Global energy consumption due to friction in trucks and buses". *Tribology International* 78 (Oct. 2014), pp. 94–114.

[12] BERGGREN, C. and T. MAGNUSSON. "Reducing automotive emissions-The potentials of combustion engine technologies and the power of policy". *Energy Policy* 41 (Feb. 2012), pp. 636–643.

[13] FENSKE, G., L. AJAYI, R. ERCK, and N. DEMAS. "Lubricants - Pathway to Improving Fuel Efficiency of Legacy Fleet Vehicles". *DEER 2011*. Detroit, MI: U.S. Department of Energy, 2011.

[14] HU, J. et al. “Integrated optimal eco-driving on rolling terrain for hybrid electric vehicle with vehicle-infrastructure communication”. *Transportation Research Part C: Emerging Technologies* 68 (2016), pp. 228–244.

[15] MANSFIELD, L. R., F. GUROS, D. M. TRUXILLO, and J. MACARTHUR. “Individual and contextual variables enhance transfer for a workplace eco-driving intervention”. *Transportation Research Part F: Traffic Psychology and Behaviour* 37 (2016), pp. 138–143.

[16] JEFFREYS, I., G. GRAVES, and M. ROTH. “Evaluation of eco-driving training for vehicle fuel use and emission reduction: A case study in Australia”. *Transportation Research Part D: Transport and Environment* (2016).

[17] STILLWATER, T. and K. S. KURANI. “Drivers discuss ecodriving feedback: Goal setting, framing, and anchoring motivate new behaviors”. *Transportation Research Part F: Traffic Psychology and Behaviour* 19 (2013), pp. 85–96.

[18] HOSSEINLOU, M. H., S. A. KHEYRABADI, and A. ZOLFAGHARI. “Determining optimal speed limits in traffic networks”. *IATSS Research* 39.1 (2015), pp. 36–41.

[19] STACHOWIAK, G. W. and A. W. BATCHELOR. *Engineering Tribology*. Third Edition. Oxford: Elsevier Butterworth - Heinemann, 2005.

[20] HEYWOOD, J. B. *Internal combustion engine fundamentals*. Ed. by A. DUFFY and J. M. MORRISS. First Edition. McGraw-Hill, 1988, p. 750.

[21] BINDER, K. B., K. MOLLENHAUER, and H. TSCHÖKE. *Handbook of Diesel Engines*. Ed. by K. MOLLENHAUER, H. TSCHOEKE, and O. VON GUERICKE. First Edition. Berlin: Springer, 2010, pp. 61–75.

[22] GANESAN, V. “Engine Emissions and Their Control”. *Internal Combustion Engines*. Second Edition. Chennai: McGraw-Hill, 2004, pp. 495–538.

[23] JAMES, C. J. “Analysis of Parasitic Losses in Heavy Duty Diesel Engines by”. Master Thesis. Massachusetts Institute of Technology, 2012.

[24] ANDERSSON, P. *Piston ring tribology A literature survey VTT Industrial Systems*. Tech. rep. VTT Industrial Systems, Helsinki University, Internal Combustion Engine Laboratory, 2002, pp. 1–108.

[25] PAYRI, F. and J. M. DESANTES. *Motores de Combustión Interna Alternativos*. Barcelona: Reverté, 2005.

[26] MUFTI, R. A. “Total and Component Fricition in a Motored and Firing Engine”. PhD thesis. The University of Leeds, 2004, p. 276.

[27] NAGAR, P. and S. MIERS. "Friction between Piston and Cylinder of an IC Engine: a Review". *SAE Technical Paper* 2011-01-1405 (2011), pp. 4–11.

[28] SMEDLEY, G. "Piston ring design for reduced friction in modern internal combustion engines". Master thesis. Massachusetts Institute of Technology, 2004.

[29] SPENCER, A. "A Simulation Tool for Optimising Combustion Engine Cylinder Liner Surface Texture". Doctoral Thesis. Lulea Tekniska Universitet, 2013.

[30] KAPOOR, A., S. C. TUNG, S. E. SCHAWARTZ, M. PRIEST, and R. S. DWYER-JOYCE. "Automotive Tribology". *Modern tribology handbook*. Ed. by B. BHUSHAN. First Edition. Boca Ratón: CRC Press, 2001. Chap. 32.

[31] TAYLOR, O. P., R. PEARSON, R. STONE, P. CARDEN, and H. BALLARD. "Tribological Behavior of Low Viscosity Lubricants in the Piston to Bore Zone of a Modern Spark Ignition Engine". *SAE Technical Paper* 2014-01-2859 (2014).

[32] OED. *Oxford English Dictionary*. 2008.

[33] HEINEMANN, H. *Lubricant Additives-Chemistry and Applications*. Ed. by L. R. RUDNICK. Second Edition. Boca Raton: CRC Press, 2009, p. 777.

[34] FUENTES, E. and K. KANASE. "Relevance of Lubrication Regimes in the Study of Bearing Wear". *Critical Component in Heavy Duty Engines*. Ed. by P. LAKSHMINARAYANAN and N. S. NAYAK. First Edition. Singapore: John Wiley & Sons, INC., 2011. Chap. 12, pp. 199–252.

[35] WAKURI, Y., M. SOEJIMA, Y. EJIMA, T. HAMATAKE, and E. AL. "Studies on Friction Characteristics of Reciprocating Engines". *SAE Technical Papers* SAE-952471.412 (1995), SAE 952471.

[36] TARAZA, D., N. HENEIN, and W. BRYZIK. "Friction Losses in Multi-Cylinder Diesel Engines". *SAE Technical Paper 2000-01-0921* 724 (2000).

[37] LIVANOS, G. and N. P. KYRTATOS. "A Model of the Friction Losses in Diesel Engines". *SAE Technical Paper 2006-01-0888* 724 (2006), p. 2006.

[38] SHERRINGTON, I. "Measuremetn techniques for piston-ring tribology". *Tribology and Dynamics of Engine and Powertrain*. Ed. by H. RAHNEJAT. First Edition. Oxford: Woodhead Publishing, 2010. Chap. 11.

[39] PRIEST, M. "Reducing Automobile Engine Friction Through Tribology Tribology". *Internal CO2 Seminar*. Leeds, 2006.

[40] NORIA CORPORATION. *Understanding the Differences in Base Oil Groups.* 2012.

[41] HAYCOCK, R. F., A. J. CAINES, and J. HILLIE. *Automotive lubricants reference book*. First Edition. Warrendale: Society of Automotive Engineers, Inc., 2004.

[42] MARTYR, A. and M. PLINT. *Engine Testing*. Fourth Edition. Oxford: Butterworth-Heinemann, 2007, pp. 170–196.

[43] ASTM INTERNATIONAL. *ASTM D2270-10e1, Standard Practice for Calculating Viscosity Index From Kinematic Viscosity at 40 and 100° C*. Tech. rep. West Conshohocken, PA: ASTM International, 2010.

[44] ASTM INTERNATIONAL. *ASTM D6278-12e1, Standard Test Method for Shear Stability of Polymer Containing Fluids Using a European Diesel Injector Apparatus.* Tech. rep. West Conshohocken, PA: ASTM International, 2012.

[45] ASTM INTERNATIONAL. *ASTM D7109-12, Standard Test Method for Shear Stability of Polymer Containing Fluids Using a European Diesel Injector Apparatus at 30 and 90 Cycles.* Tech. rep. West Conshohocken, PA: ASTM International, 2012.

[46] DAM, W. VAN, P. KLEIJWEGT, M. TORREMAN, and G. PARSONS. "The Lubricant Contribution to Improved Fuel Economy". *SAE Technical Paper 2009-01-2856* (2009).

[47] DAM, W. VAN et al. "The Impact of Lubricant Viscosity and Additive Chemistry on Fuel Economy in Heavy Duty Diesel Engines". *SAE Technical Paper 2011-01-2124* (2011).

[48] MORTIER, R., M. FOX, and S. ORSZULIK. *Chemistry and Technology of Lubricants (Google eBook)*. Ed. by R. M. MORTIER, M. FOX, and S. ORSZULIK. Third Edition. Vol. 53. Springer, 2011, p. 574.

[49] SAE INTERNATIONAL. *SAE J300. Engine Oil Viscosity Classification*. Tech. rep. Warrendale: SAE International, 2015, pp. 1–9.

[50] SOUZA DE CARVALHO, M. J., P. RUDOLF SEIDL, C. R. PEREIRA BELCHIOR, and J. RICARDO SODR. "Lubricant viscosity and viscosity improver additive effects on diesel fuel economy". *Tribology International* 43.12 (2010), pp. 2298–2302.

[51] KANEKO, T., M. YAMASHITA, M. OHORI, and M. MURAKAMI. "Development of Low Sulfated Ash and Fuel Economy Diesel Engine Oil". *SAE Techinical Paper* 2009-01-1845 (2009).

[52] MAROTO-CENTENO, J. A., T. PÉREZ-GUTIÉRREZ, L. FERNÁNDEZ-RUÍZ-MORÓN, and M. QUESADA-PÉREZ. "Prediction of fuel economy performance of engine lubricants based on laboratory bench tests". *Tribology International* 94 (2016), pp. 67–70.

[53] COVITCH, M. J. et al. "Extending SAE J300 to Viscosity Grades below SAE 20". *SAE Technical Paper 2010-01-2286* (2010).

[54] THE AUTOMOBILE EVALUATION STANDARDS SUBCOMMITTEE ENERGY EFFICIENCY STANDARDS SUBCOMMITTEE OF THE ADVISORY COMMITTEE FOR NATURAL RESOURSES AND ENERGY and THE AUTOMOBILE FUEL EFFICIENCY STANDARDS SUBCOMMITTEE AUTOMOBILE TRANSPORT SECTION LAND TRANSPORT DIVISION OF THE COUNCIL FOR TRANSPORT POLICY. *Concerning revisions of evaluation standards for manufacturers with regard to improvement of automobile energy consumption efficiency*. Tech. rep. March. 2007.

[55] THE AUTOMOBILE EVALUATION STANDARDS SUBCOMMITTEE ENERGY EFFICIENCY STANDARDS SUBCOMMITTEE OF THE ADVISORY COMMITTEE FOR NATURAL RESOURSES AND and POLICY THE AUTOMOBILE FUEL EFFICIENCY STANDARDS SUBCOMMITTEE AUTOMOBILE TRANSPORT SECTION LAND TRANSPORT DIVISION OF THE COUNCIL FOR TRANSPORT. *Final Report of Joint Meeting between the Automobile Evaluation Standards Subcommittee , Energy Efficiency Standards Subcommittee of the Advisory Committee for Natural Resources and Energy and the Automobile Fuel Efficiency Standards Subcommittee , Automo*. Tech. rep. December. 2011.

[56] MANNI, M., C. GOMMELLINI, and G. SABBIONI. "Effect of Physical Characteristics of Lubricating Oils on Emissions, Fuel Economy and Oil Consumption in a Light Duty Diesel Engine." *SAE Technical Paper* 952552 (1995).

[57] TSEREGOUNIS, S. and M. L. MCMILLAN. "Fuel Economy Gains with Modern Technology, SAE 5W-20 Engine Oils in a GM Engine as Measured in the EPA FTP Test." *SAE Technical Paper* 2001-01-1900 (2001).

[58] SMITH, O. M. E. "In-Cylinder Fuel and Lubricant Effects on Gasoline Engine Friction". Ph.D. Thesis. The University of Leeds, 2007.

[59] FONTARAS, G., E. VOUITSIS, and Z. SAMARAS. "Experimental Evaluation of the Fuel Consumption and Emissions Reduction Potential of Low Viscosity Lubricants". *SAE Technical Paper 2009-01-1803* (2009).

[60] PARK, S., Y. CHO, K. SUNG, and N. HAN. "The Effect of Viscosity and Friction Modifier on Fuel Economy and the Relationship Between Fuel Economy and Friction". *SAE Technical Paper* 2009-01-2662 (2009).

[61] SINGH, D., M. SHUKLA, S. SINGH, Y. SINGH SR, and N. SINGH. "Fuel Economy of SI Vehicle Fueled by Ethanol/Gasoline Blends - Role of Engine Lubricant Viscosity". *SAE Technical Paper 2013-01-0051* (2013).

[62] GUINTHER, G. and J. STYER. "Correlation of the Sequence VID Laboratory Fuel Economy Test to Real World Fuel Economy Improvements". *SAE Technical Paper 2013-01-0297* (2013).

[63] ANDRÉ, M. *Real-world driving cycles for measuring cars pollutant emissions Part A: The ARTEMIS European Driving Cycles*. Tech. rep. June. Institut National de Recherche sur les Transports et Leur Sucurite, 2004, p. 45.

[64] MANNI, M. and S. FLORIO. "An Experimental Evaluation of the Impact of Ultra Low Viscosity Engine Oils on Fuel Economy and CO2 Emissions". *SAE Technical Paper 2013-01-2566* (2013).

[65] CALWELL, C. "Fuel Savings Possibilities from Low Viscosity Synthetic Motor Oils". *International Energy Agency*. Paris, 2005.

[66] ASTM INTERNATIONAL. *ASTM D7109-12, Standard Test Method for Shear Stability of Polymer Containing Fluids Using a European Diesel Injector Apparatus at 30 and 90 Cycles.* Tech. rep. West Conshohocken, PA: ASTM International, 2012.

[67] CARDEN, P., C. PISANI, J. ANDERSSON, and I. FIELD. "The Effect of Low Viscosity Oil on the Wear, Friction and Fuel Consumption of a Heavy Duty Truck Engine". *SAE Technical Paper* 2013-01-0331 (2013).

[68] HETRICK, S., J. KELLER, and H. LOWTHER. "Performance Advantages of Synthesized Commercial Engine Oils." *SAE Technical Paper 780183* (1978).

[69] KELLER, J. A., N. S. KOTUSZENKO, and R. R. MCCOY. "Fuel Economy Benefits of Synthetic Lubricants in Heavy-Duty Trucks". *SAE Technical Paper 801345* (1980).

[70] MILLBROOK PROVING GROUND. *FirstGroup Millbrook Fuel Economy Test*. 2016.

[71] BROWNE, D. et al. "Novel Fuel Efficiency Engine Lubricants for Urban Transit Applications". *SAE Technical Paper* 2014-01-2793 (2014).

Chapter 3

Experimental tools and techniques

This chapter comprises the explanation of all the tests made during the thesis development including both, equipment and instruments and tests methodology. In the first part, the tests performed in the engine test bench are described. To run these tests, a representative multi-cylinder commercial engine was mounted on a bench capable to absorb the produced power (dynamometer). Three different engine bed tests were run during this thesis development; a motored test, which primary objective was to determine the differences in friction power when oils with different viscosities were used, a fired stationary test where Break Specific Fuel Consumption (BSFC) of 12 different engine's operating points were compared for two different commercial engine oils, and finally a transient test, where the effects of the oil viscosity on total fuel consumption were measured directly when the engine ran under the New European Driving Cycle test (NEDC). In the second part of the chapter a field test set with an urban buses fleet to assess the effect of the LVEO on the fuel consumption is described. The description includes the vehicles and oil characteristics, fuel consumption measurement methods, fuel characteristics, and statistical principles applied in the analysis. This part of the thesis is of special interest since field testing normally is the most meaningful despite the difficulties to obtain worthwhile results. Finally, at the end of the chapter, laboratory tests to measure rheologic properties of engine oils are described alongside with specific laboratory rigs to test friction behavior and lubrication regimes of tribo-contacts similar to those found during engine operation.

3.1 Introduction

The effect of LVEO on friction, fuel consumption and CO_2 emissions can only be determined by controlled tests where one or more oil formulations performances are compared against a control group which uses a reference oil, usually with higher viscosity. The complexity of this task could range from a single comparative test, where the oil viscosity would be the only factor affecting the response, for example friction coefficient or lubricant film thickness and using the hypothesis testing as statistical inference technique to find meaningful differences among the different levels of the oil viscosity factor to a complex multivariable environment where oil viscosity effect over fuel consumption would be mask by several noise factors as ambient temperature, vehicle load, air drag, rolling resistance, fuel quality and heat value, and so on. The first case would be typically a laboratory test, where noise factors can be controlled (*e.g.* a friction measurement of two different engine oils with a tribometer where oil temperature, load, entrainment speed, room temperature and humidity among others are under control of the researcher), and the second case would be typically found on a test under "real-world" conditions where two oils are tested in different vehicles where ambient factors cannot be controlled and will affect the experiment outcome[1].

The normal equipment and methods to test engine oil performance related to friction reduction, fuel consumption and CO_2 emissions are; tribometers, engine test cells, chassis dynamometers, and track testing under real world ambient conditions. General in testing, repeatability is a critical issue and this is specially true in the case when engine oil effect over fuel consumption is measured, since the expected changes are very low and do not exceed in most cases 5%[2–13] making this task very challenging. For this reason engine dynamometer tests has been favored since they could reach high levels of accuracy, like the European M111e CEC L-54-T-96 fuel economy test[14] and the sequence VID ASTM D7589[10, 12, 15–19] which have been the reference for the passenger car oil industry. Regarding the Commercial vehicles, even when there are no standards on fuel economy tests for oils, some OEM's have been using the World Harmonized Transient Cycle (WHTC)[20] to measure differences in fuel consumption given by the use of different engine oil formulations. The positive aspect of engine cell testing is that oil and coolant temperatures can be controlled, if transient conditions are required it will be driven by a closed loop control system (varying the torque and engine speed to reach the exact required operation conditions) and measuring fuel consumption and CO_2 emissions with precise instruments.

The next level of testing would be the chassis dynamometer tests, where the complete vehicle is brought under testing. This kind of test is the preferred option when it comes to emissions legislation. Good examples of these tests are the

New European Driving Cycle(NEDC) in Europe, the JC08 cycle for Japan and the Corporate Average Fuel Economy (CAFE) cycle used in the United States for the passenger car segment and FIGE cycle (developed by the FIGE institute in Aachen, Germany) in Europe, the JE05 test in Japan and Federal Test Procedure (FTP) in the United States for Heavy Duty Vehicles. These type of tests give useful information of how changes in the engine affect other powertrain systems and allows as well to test Low Viscosity Oils (LVO) in other systems as the axles and transmissions. Despite these advantages, specific trained pilots are needed in order to complete one test and it is harder to find claimed differences of fuel consumption under 1%.

The main drawback however, for engine testing and chassis dyno testing would lie behind the fact that both, at stationary conditions and transient conditions, most of the time do not reproduce the "real-world" demands which vehicles have to deal with during normal operation. This inconvenient has been widely describe for other vehicle emissions like NO_x, CO, PM and HC in several studies among them Demunynck *et.al.*[21] and Sileghem *et.al.* [22] have address the problem in a very deep way showing that for the majority of time, vehicles which have approved certification tests, exceed the limit values at real conditions operation. Another handicap of all the tests mentioned above is the fact that none of them are easily extrapolable to the end user experience.

In the same way, Tietge *et.al.* have found a systematic increasing gap between the CO_2 emissions proclaimed by OEM's and governments and "real-world" CO_2 emissions and fuel consumption in the passenger car segment. Regarding the NEDC cycle, the gap has increase from 10% in 2002 to 35% in 2014 and it is expected to grow to reach 49% in 2020 (see figure 3.1)[23]. In addition the study summarizes the possible causes of the increasing gap in:

- *road load determination:* Aerodynamic drag and rolling resistance coefficients are determined through a series of coast-down tests on an outside track prior the laboratory tests. Within this procedure exist a number of tolerances and flexibilities including; tire selection and preparation, selection of the test track, ambient test conditions, and pre-conditioning of the vehicle, among others.

- *Chassis dynamometer testing:* the chassis dynamometer permits a vehicle to be "driven" while remaining stationary (by placing the vehicle on rollers) and simulates road load. under the EU regulation,there are a number of "loopholes" that can potentially be exploited by vehicle manufacturers during chassis dynamometer testing; include break-in periods for the test vehicle, tolerances regarding laboratory instruments, the state of charge of the vehicle's battery, special test driving techniques, and use of preseries parts that are not representative of production vehicles. The analysis indicates that vehicle manufacturers have found ways to optimize chassis dynamometer type-approval testing over time, which at the same time made it less representative of average "real-world" driving conditions.

- *Other parameters:* operating equipment such as air conditioning systems and entertainment systems increases fuel consumption during "real-world" driving. However, these devices are usually switched off or are not fully taken into account during the type-approval emissions test, leading to unrealistically low CO_2 emission values.

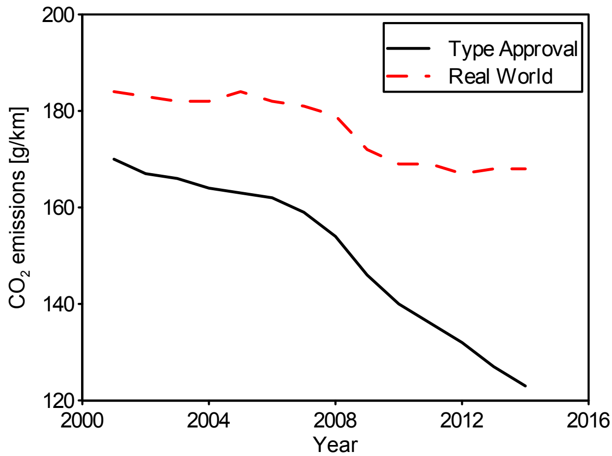

Figure 3.1: 2001-2014 real-world vs. type-approval CO_2 emissions. (Adapted from [23]).

In order to serve this lack of impact of the certification tests, the fuel efficiency SAE J1321 test procedure [24] where two identical vehicles (one control and one test vehicle) cover a certain distance on an oval circuit was set. This procedure measures on-road fuel consumption utilizing a similarly equipped, unchanging control vehicle operated in tandem with a test vehicle to provide reference fuel consumption data. This procedure has become the *de facto* test for both carrier and manufacturer fuel economy evaluations, largely due to its ability to use real-world vehicles and routes. The specification requires both careful control of potential operational variables and numerous replications to validate the difference statistically. The procedure is claimed to provide precision within ±1%. EPA modified the SAE J1321 test procedure to require use of a test track environment, and each test segment incurs only one acceleration and deceleration. It measures fuel consumption and requires that average speed be controlled to 55 mph to 62 mph (90 km/h - 100 km/h) preferred, with 65 mph (105 km/h) as maximum[25, 26].

Despite the fact that fuel consumption results on this test are direct applicable to real world application, the conditions of the test itself make those result irrelevant if the vehicle vocation is other than highway freight. For the same reason, fuel consumption differences from changes in viscosity of engine oils would be hard to determine, since at the speeds that the test is performed, other vehicle losses as air drag and rolling resistance have more impact over fuel consumption than the oil itself[27].

Making a decision from all the possible tests describe above to assess the effect of the viscosity on fuel consumption would be a compromise between resemblance to reality and noise factors affecting the experiment outcome. The noise factors will be directly related to the outcome variability, being the latter greater if the noise factors are many or impact in a heavy way the outcome variable, *e.g.* in a fleet test under "real-world" conditions, the effect of the engine oil viscosity over the fuel consumption which is expected to not exceed 5% would be easily masked by other factors as the use of air conditioning system (AC) during summer, the impact of the driver, the traffic, the vehicle load and the air drag just to mention a few. The presence of these factors will hinder the mean fuel consumption comparison to perform the hypothesis test and the only possible way to void the noise effect is to increase the number of measurements in order to reduce variability.

3.2 General description of comparative tests' methodologies

During this thesis development, all levels of experimentation to find friction and fuel consumption differences due the use of LVEO were covered, *i.e.* tribometers, engine (both motored and fired), and real-world testing. For the real worl testing, the choose of a buses fleet was intentional: among many others, one interesting and specific type of HDV is the urban bus, which energy shares are about 4% of the transportation sector. Some interesting characteristics of this type of vehicles were pointed out by Holmberg *et.al.*; they rely on diesel fuel due the extended use of ICE, they have a repetitive duty cycle which leads to homogeneous energy consumption, and they are usually part of fleets which makes easier to influence decision-making in order to implement methods or policies to enhance their energy efficiency[28].

The reason to perform comparative tests between regular and LVEO at different levels was to cover all the spectra of the phenomenon, from the most precise but unrelated to the engine conditions point of view (as the case for Cameron Plint Machine (see section 3.5.1) to a real fleet test where a considerable sample of urban public buses was studied for a long period in order to find statistical fuel consumption differences which could be effortlessly include in any end user cost benefit balance (see section 3.4).

3.3 Engine bed tests

As mentioned in the introduction of this chapter, engine bench test are one of the most reliable methods when it comes to measure fuel consumption differences when using engine oils with different viscosity. Three different and complementary test methods were planned in order to depict the changes of the ICE operation when a LVEO is used. These three methods were:

- Motored test: measurement of friction power differences.
- Break Fuel Specific Consumption (BFSC) Screening test, under stationary conditions: identification of the engine map zones where the effect of the use of LVEO could contribute to fuel consumption reduction.
- Dynamic tests under the New European Driving Cycle (NEDC): correlation of the identified zones found during screening test with close to "real-world" engine behavior. Assessment of other factors affecting engine oil influence over fuel consumption as coolant and engine oil temperature.

As exposed by MAHLE[29], when motored, the ICE is driven by the dynamometer, being the ignition and fuel supply cut to prevent combustion. Coolant and oil temperatures are maintained at operating temperature by means of external conditioning units. The friction can be then determined from the power consumed by the electric motor.

General disadvantages of motored tests are:

- in the absence of gas pressure from combustion, the loads are very low.
- the operational and component temperatures are significantly lower than in a fired engine. Clearances in several systems (like piston assembly) will be significantly different distorting results of friction coefficient, hence, friction mean effective pressures.

- normally the friction losses distribution will be different than the one of a fired engine, as the case of valvetrain friction losses over total mechanical losses.

Regarding the dynamic test, several dynamic cycles could have been selected to study the effect of LVEO over fuel consumption, moreover, when the results depends strongly on the kind of cycle as it was described by Cui *et.al.*[30] (see figure 3.2 and the already mentioned differences between the NEDC cycle and the real world emissions and fuel consumption data. However, given the fact that NEDC is still the type approval test, and that NEDC has been the preferred cycle to test many technical solutions, prototypes, fuels, engine oil additives among others[31–43] it seemed suitable to measure the fuel consumption effects under engine transitory under this cycle.

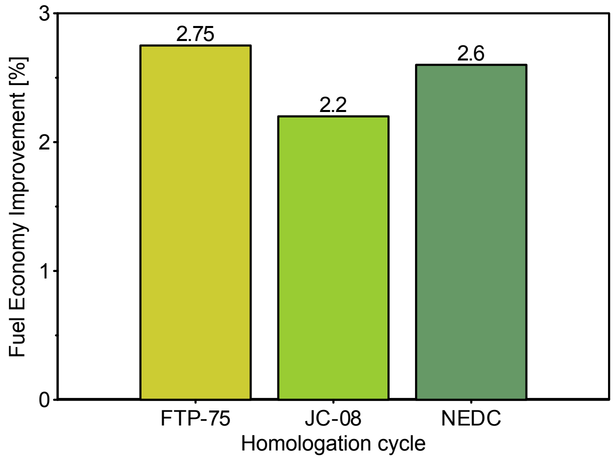

Figure 3.2: Driving cycle effect on fuel economy improvements for a 0W20 oil using RL191 as baseline oil (Adapted from [30])

3.3.1 Experimental Setup

A high pressure direct injection, 4 cylinder, 1.6 l, turbo diesel engine, which meets Euro 5 regulations for Light Duty Vehicles was employed. The engine specifications and the lubricants main characteristics are shown in table 3.1 and table 3.2.

The engine was coupled with a Schenck-Pegasus dynamometer controlling online engine torque and speed. The control software used was a CMT "in-house" development named SAMARUC. In order to register engine's parameters, Engine Control Unit (ECU) was totally opened and the engine setting maps could be

Engine	
Displacement	1560 cc
Cylinders	4 in line
Valves	2 Valves per cylinder
Max power [kW]	82@3600 [min^{-1}]
Max torque [Nm]	280@1750 [min^{-1}]
Turbo	Variable geometry
Emissions control	EGR, particle trap

Table 3.1: Engine main characteristics

calibrated with the ETAS INCA Software. The engine test bed was equipped with a series of temperature, pressure and air mass flow sensors in order to control the engine precisely. Fuel consumption was measured by means of a fuel gravimetric system, the AVL 733S Dynamic fuel meter. It consists of a measuring vessel filled with fuel suspended on a balance system. Fuel consumption values were then obtained by calculating the vessel's time related weight loss. As the response time of this system was too long for the dynamic study, a calibration of the fuel consumption signal provided by the ECU was performed in steady state. This ECU signal was used as a secondary fuel consumption measurement.

In this engine setup an external circuit to control coolant temperatures was set. However, the set up had not an external circuit to control oil temperatures. Oil temperatures in this case were controlled varying the coolant flux in the engine intercooler, having reasonable results for the most of the test performed with the setup.

3.3.2 Oils

The oils used during this test were all commercial common oils, two of them labeled as "Fuel Economy" oils, with typical 5W20 and 5W30 SAE grades used in the Spanish automotive market. As baseline oil one 15W40 was used. All the used oils met the engine OEM quality standard requirements. The complete description of used oils can be seen in the table 3.2.

3.3.3 Friction and fuel consumption test procedures

As it was mentioned before, the goal of this study is to assess the effect of lubricant viscosity on fuel consumption in Light Duty Vehicle engines. To do so, an initial motored test focused on determining the real potential of the LVEO to reduce the engine friction when the engine works on different engine speeds was conducted.

SAE Viscosity grade	5W20	5W30	15W40
Base oil	API G-III	API G-III	API G-I
CCS viscosity [cP]	4519@-30° C	5120@-30° C	4878@-20 ° C
kv@40° C [cSt]	45	53	107
kv@100° C [cSt]	9.0	9.7	14.6
HTHS@150° C [cP]	2.8	2.9	3.7

Table 3.2: Engine oil characteristics

This test intended to measure the torque differences required by the dynamometer to reach several engine speeds, being this a clear indicator of possible changes in mechanical losses. Then a screening over the engine's functional map was made by means of a stationary fired test. The purpose of this second test is to report the engine operating points where potential fuel consumption reduction due LVEO use are more noticeable. In this stationary fired test, BSFC obtained for each point with every oil is used as a comparison parameter. Finally, a transient cycle test was performed in order to address the effect of LVEO when the engine works under real driving conditions. In this final test, the comparison was made taking into account the overall fuel mass consumed.

3.3.4 Motored test

This procedure consists in measure the required torque used by the dynamometer to motor the engine at certain speed. One objection to this method is the fact that in absence of combustion the entirely variables which affect the engine's performance are misplaced (i.e. temperature profiles, air in cylinder pressure, parts strain, etc.). To get a more accurate approximation to the engine's operating conditions, motored tests should be performed after the engine has been working under fired conditions and controlling coolant and oil temperatures [44].

Although it does not simulate the engine's working conditions due its unfired nature it has been widely used as an indicator of the engine frictional behavior. For this test in particular, torque measurements were taken for seven engine speeds ranging from 1000 min^{-1} to 4000 min^{-1}, every 500 min^{-1}.

3.3.5 Stationary fired test

The test under stationary fired condition took place in order to address the relative impact and possible fuel consumption benefit of LVEO in specific stationary points of the engine's map. The stationary test offers a significant control level over the engine's variables (*i.e.* temperatures, engine's speed, among others), making easier to address the effect of any particular change of these variables in engine's performance. A 12 point screening on the engine's working map was planned to identify the working zones with more potential for fuel consumption reduction. The method employed consisted in compare the final torque output for each point at "iso-consumption" conditions for the two levels of viscosity given by the different oils and having 5W30 SAE grade oil as the baseline. Each single point measurement has involved a three time repetition, every one of them being the average of engine's fuel consumption values on a 30 s period. To complete the test under "iso-consumption" conditions an initial round of measurements was made with 5W30 SAE grade oil and using as inputs for each points the values given in table 3.3.

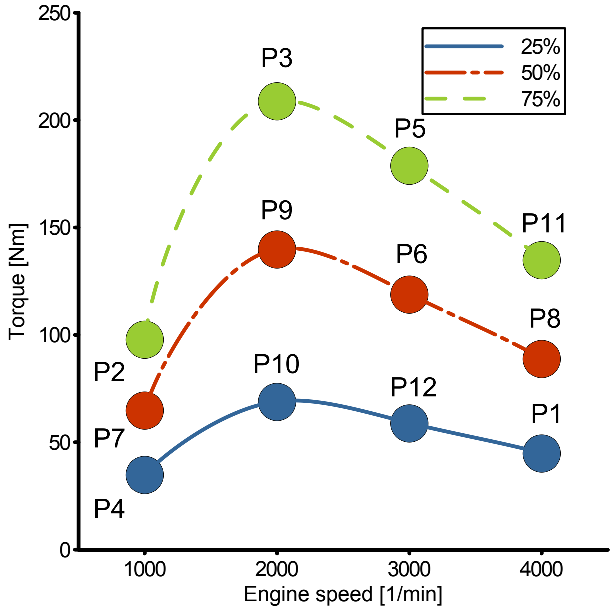

Figure 3.3: Engine operation points for the screening test. The load percentage [%] is based on the maximum torque at the given engine speeds [min^{-1}]

Output parameters as fuel flow rate, EGR %, GVT %, manifold inlet air pressure, and SOI were registered for each of the 12 points. After flushing oil A and replacing it with the candidate oil (oil B or C), the 12 points were measured again fixing this time, engine speed and fuel flow rate measured with oil A as inputs. Values of EGR%, GVT %, manifold air pressure and SOI were controlled to assure similar combustion conditions.

Finally, resulting torque registered for each point and oil was used as the main source to compare the BSFC differences leaded by the use of the different lubricant

Point	Load [%]	Speed [min^{-1}]
1	25	4000
2	75	1000
3	75	2000
4	25	1000
5	75	3000
6	50	3000
7	50	1000
8	50	4000
9	50	2000
10	25	2000
11	75	4000
12	25	3000

Table 3.3: Stationary fired test points.

oils. This approximation could give more precise results than the "iso-power" like test, where the engine could deliver the same power working in different points of its functional map.

3.3.6 New European Driving Cycle (NEDC) test

The NEDC test was planned mainly to bear out the gain on efficiency leaded by the LVEO when the engine operates on transient conditions. This approach gives the closest approximation of real "on road" benefit in fuel consumption that could be reached by the use of LVEO, being this value the most important for OEM and end-users.

Also known as the MGEV-A, this cycle was used in the European Union to test vehicles emissions and fuel economy behavior. Originally developed to be performed on a chassis dynamometer, the cycle emulates the typical driving conditions of a Light Duty Vehicles in Europe, with vehicle velocity profiles for both urban and extra urban driving conditions, with a total duration of 1200 s. At the beginning of the test, the room, the engine coolant and oil temperature should be between 20° C and 30° C. The first part of the cycle is known as UDC (Urban Driving Cycle) consisting of four ECE-15 segments of 200 s. In the other hand the last 400 s of the driving cycles simulates highway conditions, and is known as the EUDC (Extra Urban Driving Cycle), where the vehicle can reach 120 km/h. The NEDC could be simulated as well on an engine test bed, controlling the engine's speed and load. As these values were used to perform the NEDC it could be said that the comparison between the two oils is made under "iso-power" conditions.

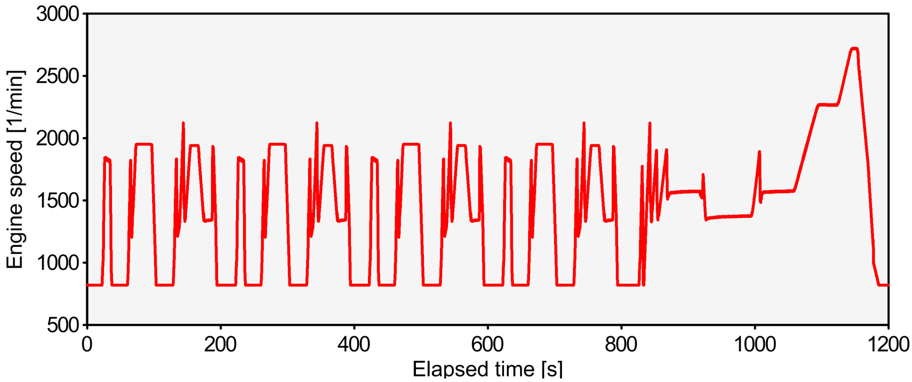

Figure 3.4: Adapted NEDC cycle for an engine test controlled by pedal position [%] and engine speed [min^{-1}]

3.4 Fleet test

The test objective was to compare the fuel consumption of a representative group of urban buses using different engine oils which viscosity ranged from SAE 15W40 and SAE 10W40 Low SAPS to SAE 5W30 and SAE 5W30 Low SAPS (see figure 3.5).

To accomplish the test goals, a comparative long term test using 39 buses from 3 different models of the Valencia public transport fleet (EMT-Valencia) was perfomed. The fleet was divided in two groups one of them using market-standard SAE grade oils as baseline and the other using LVEO. The daily fuel consumption was recorded in a daily basis until the buses reached a mileage equal to two oil drain intervals (ODI) or 60000 km. This long period of time was established given the great number of other variables during real service which were affecting fuel consumption besides the oil formulation as the environmental conditions (*e.g.* pressure, weather, season of the year), route conditions (*e.g.* route grade of slope, average velocity, so on), driving behavior and specific bus operation conditions variables (urban traffic, number of passengers, vehicle weight, rolling resistance, type of engine, so on), masking the effect of oil viscosity over bus fuel consumption expected to be as low as 1% [3]. Every worked day by a bus was counted as a test repetition in order to establish a fuel consumption value statistically significant for each case. Having a large amount of data were crucial as Browne *et.al.* describe it in their "SAE J1321-like" experiment "*By increasing the number of experimental measurements or test runs, and thus utilising the power of n, a reduction in the variability around the mean fuel consumption can be achieved. This approach does*

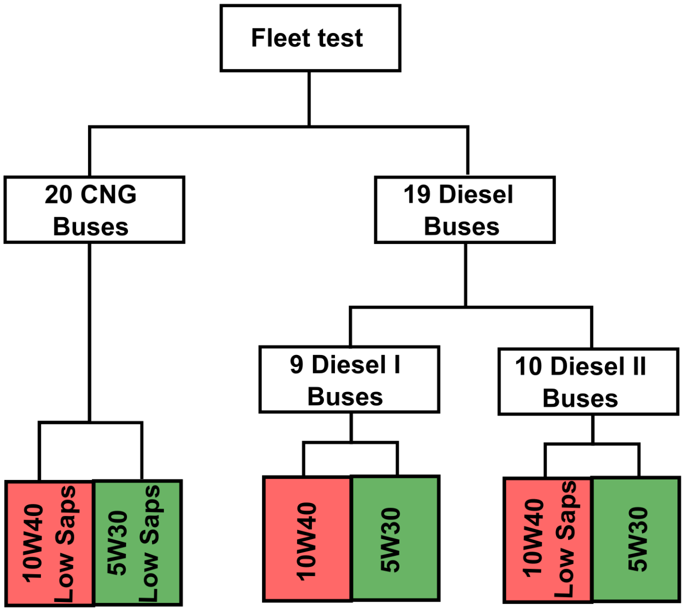

Figure 3.5: Fleet test overlook.

not change the mean fuel consumption values, rather it decreases the standard error of the means and reduces the width of their confidence intervals"[12].

The test characteristics are explained in the following sections.

3.4.1 Test vehicles

39 buses of 3 different models were used to assess the effect of LVEO over their fuel consumption. Two of this bus models use a diesel powertrain, and the other one use a CNG powertrain meeting Euro emissions standards EURO IV, EURO V and EEV respectively [45]. From now on, Diesel buses meeting EURO IV emission standards will be address as Diesel I buses, in the same way Diesel buses meeting EURO V emissions standards will be address as Diesel II buses. All CNG buses belong to the same model and meet EEV emission standards and they will be referred simply as CNG. The 39 vehicles were distributed in the three models as follows; 9 Diesel I buses, 10 Diesel II buses and 20 CNG buses. The vehicle characteristics per model can be seen in figure 3.7 and table 3.4.

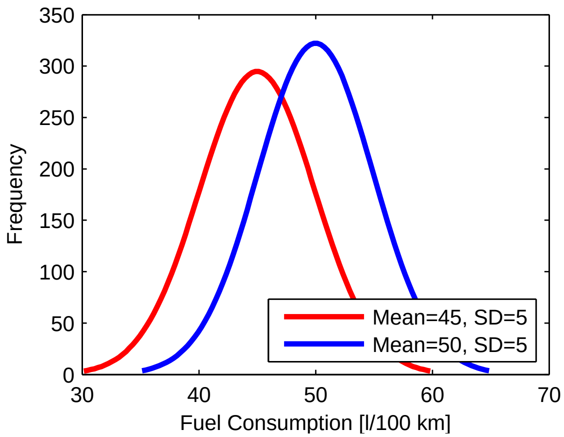

Figure 3.6: Test fuel consumption measurements for two groups using two levels for each factor (e. g. oil viscosity), with a $\mu_1 = 45l/100km$ *and* $\mu_2 = 50l/100km$ *and* $\sigma_1 = \sigma_2 = 5$ *and 1000 measurements*

Bus model	Diesel I	Diesel II	CNG
Model year	2008	2010	2007
Length/width/height [m]	17.94/2.55/3	11.95/2.55/3	12/2.5/3.3
Engine displacement [cm^3]	11967	7200	11967
Cylinders	6	6	6
Emissions certification level	Euro IV	Euro V	EEV
BMEP [bar]	16.8	19.55	9.24
Max. effective torque [Nm]	1600@1100 [min^{-1}]	1100@1100 [min^{-1}]	880@1000 [min^{-1}]
Max. effective power [kW]	220@2200 [min^{-1}]	210@2200 [min^{-1}]	180@2200 [min^{-1}]
Thermal load [$\frac{W}{mm^2}$]	2.85	3.97	2.33
Turbo-charging	Turbo+intercooler	Turbo+intercooler	Turbo+intercooler
Valvetrain configuration	OHV Roller follower	OHV Cam follower	OHV cam Follower
EGR	No	No	-

Table 3.4: Buses characteristics

3.4.2 Oils

As the main goal of test was to evaluate the effect of LVEO over fuel consumption and CO_2 emissions it was critical to establish one parameter to choose correctly the different oils to test. van Dam *et.al.* [3, 7] have demonstrated that for Heavy Duty Vehicles, the two most relevant oil rheological characteristics regarding fuel consumption are the kinematic viscosity at 100° C (kv@100° C) measured under ASTM D-445, and High Temperature High Shear Viscosity at 150° C (HTHS@150°

Figure 3.7: Test vehicles.

C) measured under ASTM D4683, CEC L-36-A-90 (ASTM D 4741), or ASTM D 5481. In order to evaluate the difference in fuel consumption terms of the different oils the test design include the use of one common oil as reference and one candidate oil with lower values of kinematic viscosity (kv@100° C) and HTHS. Due the different oil standards required by the bus models OEM's it was not possible to use the same LVEO and reference oil in all the models; additionally being some buses still in guarantee period only approved commercial oils were used as candidates. The main characteristics of engine and differential oils can be seen on table 3.5.

Oil	15W40	10W40 Low SAPS	5W30	5W30 Low SAPS	80W90	75W90
Used as	Ref	Ref	Cand	Cand	Ref	Cand
Buses	Diesel I	Diesel II + CNG	Diesel I + Diesel II	CNG	Diesel I	Diesel I
Base oil	API G-I	API G-III	API G-III + G-IV	API G-III + G-IV	-	-
kV@40° C [cSt]	108	96	71	68	131	102
kV@100° C [cSt]	14.5	14.4	11.75	11.7	14.3	15
HTHS@150° C [cP]	4.082	3.853	3.594	3.577	-	-
VI	>141	>145	>158	<169	105	154

Table 3.5: Characteristics of engine and differential oils used during the fleet test.

It has to be remembered that although SAE J300 standard sets the lower limit for HTHS dynamic viscosity at 2.9 cP for SAE 30 grade oils, ACEA specifications limit this value to 3.5 cP, which is very closed to the SAE J300 lower limit of SAE 40 oils being this the main reason for baseline and candidate oil HTHS viscosity values proximity.

3.4.3 Routes

As mentioned before, given the fact that a real fleet under normal operation was used for this test, it was not possible to randomize some factors affecting the fuel consumption outcome. One of those was the route of the buses. Responding to users demand, type of route and street specifications, normally the bus operator fix all buses of a single model to a single route, *e.g.* all the articulated buses operate in routes characterized to have open wide roads with heavy passenger transfers. The three models used in the test were assigned into 4 different routes which main characteristics can be seen in table 3.6.

Route	Buses	Length [km]	Avg. speed [km/h]	Bus stops	Type
10	12 CNG	17.5	11.1	66	Urban
62	8 CNG	18.7	15.1	61	Urban; Extra-urban
70	10 Diesel II	17.3	12.1	59	Urban
90	9 Diesel I	12.3	13.5	36	Urban

Table 3.6: Routes characteristics.

Insomuch as GNC worked in two separate lines this effect was included in the ANOVA analysis. Regarding the CNG routes and the distinction found in the table of urban and extra-urban, it has been made since one of the routes includes almost 6 km (around 40% of the distance) in open roads with lower traffic density and higher average speed, from Valencia to reach the near town of Benimamet.

3.4.4 Fuel consumption measurements

A daily basis calculation of buses fuel consumption was made by means of mileage performed and liters of fuel consumed. Covered distance was measured via GPS, on the other hand fuel consumed was measured by refueling both diesel and CNG buses. The diesel fuel dispenser (Tokheim quantium 110) was able to send the refueling data directly to the Computerized Maintenance Management System (CMMS) in liters. For CNG consumption measurement, a different approach was done. Since the dispenser were not able to provide a single measure per bus, due the CNG refueling facility was erected in such a way that all the CNG fleet had to be connected at the same time for refueling. The fuel has to be taken directly from the distribution line, then a compressor rise up the pressure to 200 bar and the buses start the refueling. The fuel flows to buses tanks due the pressure differential until the pressure in the tank reach the 200 bar. As the final pressure and the bus CNG tank volume are known, these values were used alongside the initial pressure in the buses tank at the beginning of the refueling to estimate the amount of CNG refueled. All natural gas consumption values listed in this document are referred as Nm^3 (normalized cubic meters), that is at 1 atm (101.325 kPa) pressure and 0° C. Buses fuel tank pressure was read from a mechanical pressure gauge placed by default by the OEM. This device has an accuracy of 0.5% and a thermal deviation of 0.4% of the read pressure by every 10° C.

3.4.5 Oil sampling to control HTHS behavior

Alongside the fuel consumption studies conducted for this thesis, an assessment of the effect of using LVEO in HDV was made. As this parallel is study out of the scope of this document, the studied parameters, experimental techniques and instrumentation and results will not be mentioned, if the reader is interested in the conclusions of that study they could be found in Macián *et.al.* [46]. However, the engine oil samples from certain buses were selected in order to study the behavior of the HTHS viscosity, given this parameter importance over vehicles fuel consumption as seen in section 2.5.1.

In this task, the oil samples of 2 buses from each Diesel model, and 4 buses of the CNG, with a sampling period of 6000 km. As expected, half of the sampling corresponded to LVEO and the other half to baseline oils. The sampling period was set on bus mileage, each sample taken once the bus reached 6000 km from the last sampling to sum up a total of 10 samples every ODI per bus.

Model	Diesel I	Diesel II	CNG
Buses	2	2	4
ODI	2	2	2
15W40 samples	10	-	-
5W30 samples	10	10	-
10W40 Low Saps samples	-	10	20
5W30 Low Saps samples	-	-	20

Table 3.7: HTHS sampling program during the fuel consumption test.

3.5 Tribometers, laboratory tests and equipment

The use of tribometers to study engine oils, surface treatments, coatings, additive performance among other tribology factors in all the possible lubrication regimes of an ICE is a common place both for researchers and OEM's. In order to complete the engine and fleet tests, two tribometers were used to mimic certain ICE tribocontacts; the Cameron Plint Machine TE77 reciprocating rig, widely used to reproduce the piston ring/cylinder liner contact [47–55] and the WAM Ball-on-Disc machine, which is normally used to reproduce valvetrain and transmission gears tribocontacts[56–60] were used alongside some of the engine oils used in the fleet test.

3.5.1 Cameron Plint Machine TE77

The Cameron Plint TE77 is a reciprocating test rig, which could use piston rings and cylinder liner specimens from real engine parts in order to mimic the contact inside the combustion chamber of the piston assembly of an internal combustion engine. The machine comprises an upper holder where the piston ring is mounted, this holder moves against a fixed specimen of the cylinder liner placed in the bottom holder which is fixed in an oil bath to ensure oil-flooded conditions when required. The test rig allows changing the normal force from 0 N to 250 N applied directly over the upper holder. An electric motor and an eccentric cam produce the reciprocating movement enhancing the possibility to control the linear speed through the motor frequency and the stroke length. The stroke length was fixed at 8 mm, the maximum value permitted by the rig, and the minimum and maximum frequencies were 1 Hz and 7 Hz respectively. A piezoelectric transducer measured the friction force along the reciprocating direction.

The measurements were focused on the oil control ring (OCR) which is the one that works under oil-flooded conditions and the responsible for a major part of the losses of the piston ring pack. Oils used during the measurements were fresh.

Two different tests were performed with this machine: one screening test, and one with the most realistic operating conditions reachable with the Plint TE77. During the former test, several load and entrainment speed values were used, to see the behavior of friction, similarly to the test in Spencer 2013. In the latter, load was fixed to the required value for achieve the Nominal Contact Pressure values given in the table 3.11. However, a 10 N load was used instead to assure repeatability. During both tests, oil temperature was controlled in order to maintain the viscosity steady during the tests. An oil washing procedure comprising 60 minutes at 250 N and 1 Hz was made with the oil to be tested.

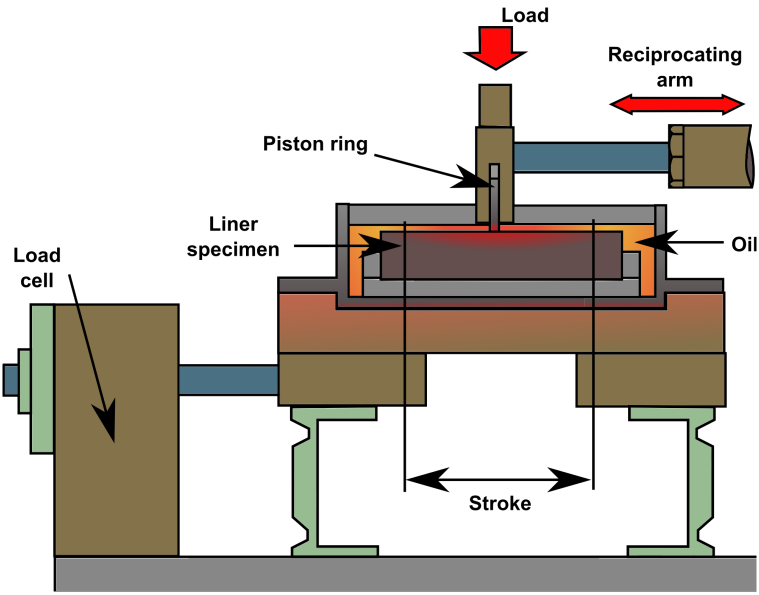

Figure 3.8: TE77 test configuration

Test points

Test points for the two tests are described in the table 3.8.

Test	Screening	Operation conditions
Points	9	7
Repetitions	3	3
Oils	2 levels	2 levels
Load [N]	3 levels [20,70,150]	1 level P_o [I]
Speed [Hz]	3 levels [1,3,7]	7 levels [1-7]

Table 3.8: TE77 tests characteristics

During the Cameron-Plint test, a common method used in engine oils comparative test known as bracketing was used (see figure 3.9). This method is in line with industry tests such as the M111e CEC L-54-T-96 procedure, the Daimler OM501LA WHTC and the Sequence VID ASTM D7589 procedure. Under this method a reference is always run after the candidate, ensuring there is no reference drift in the testing which could result in a misleading candidate fuel consumption result.

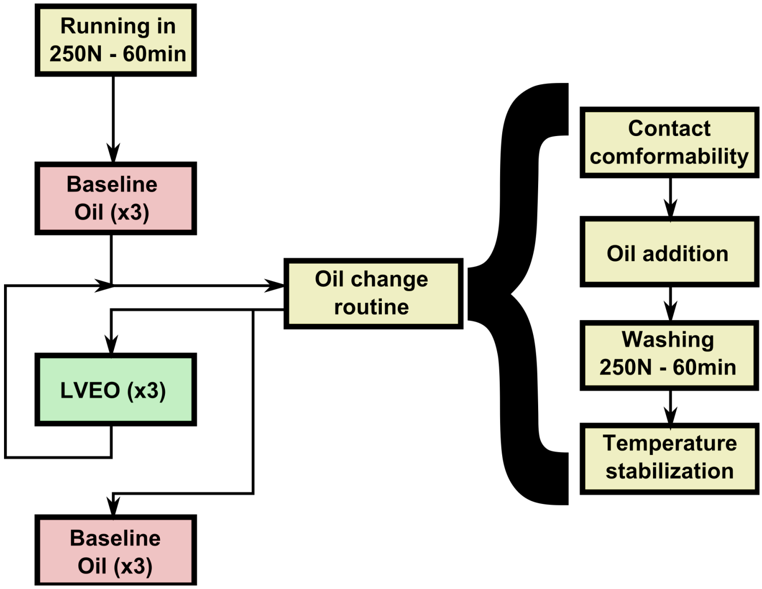

Figure 3.9: Bracketing methodology to measure the friction differences in the Cameron-Plint machine TE77

Test specimens

The specimens tested in the reciprocating rig were taken from real Heavy Duty spare parts. This engine corresponded to the reference used in the CNG buses with a nominal bore diameter of 128 mm. Table 3.9 shows the geometric characteristics of the ring and cylinder liner specimens used during the test. As the parts used as matrix of the specimens were new, a Running-in process in the test rig took place for each specimen before the measurements; the running in process consisted in 60 minutes under 250 N of load at 1 Hz.

	Length [mm]	Width/Land width [mm]
Compression Ring	80	3.5
Scrapper Ring	80	3
Oil Control Ring	80	0.8
Liner	50	8

Table 3.9: TE77 specimens characteristics

[1]Given by normal contact pressure value P_o

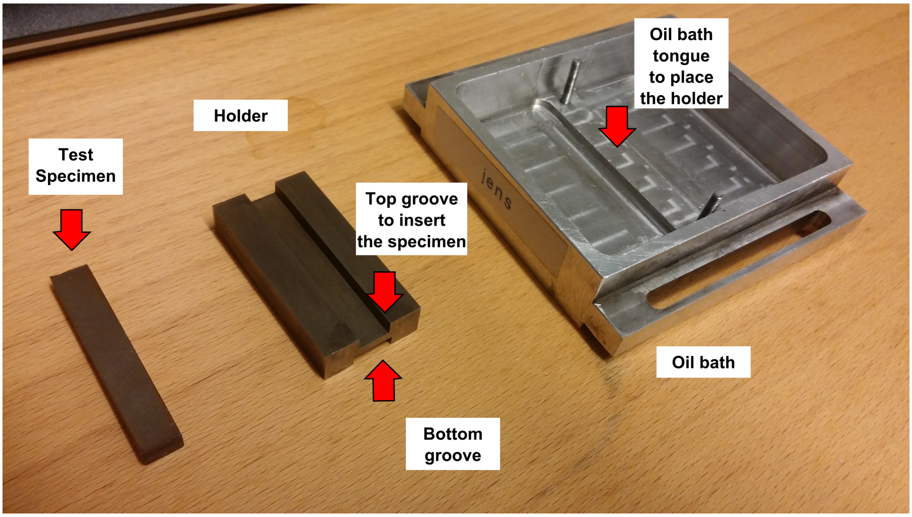

Figure 3.10: TE77 specimen, specimen holder and oil bath

Test engine oils

With the aim of successfully reproduce the conditions on the actual cylinder liner and piston rings interface of the test fleet, fresh lubricant oils of the exact formulation were used as candidate and reference during these laboratory tests. The engine oils characteristics can be seen in table 3.10.

Oil	5W30	10W40
Base Oil	API G-III/G-IV	API G-III
kv@40° C [cSt]	68	96
kv@100° C [cSt]	11.7	14.4
HTHS@150° C [cP]	3.577	3.853
VI [-]	<169	>145

Table 3.10: Oils' characteristics

Nominal contact pressure of the piston ring pack

The tension for the compression and scraper ring was taken from other rings with the same bore diameter. From these values, the nominal contact pressure was derived from the expression 3.1.

$$P_o = \frac{2F_t}{d_n h_c} \tag{3.1}$$

Where F_t is the ring tension, d_n is the bore nominal diameter and h_c is the piston ring land width. In the case of the oil control ring (OCR) the nominal contact pressure value was taken from the SAE J2003-1998 [61].

Ring	OCR	SCR	CRR
d_n [mm]	128	128	128
h_c [mm]	0.8	3	3.5
P_o [N/mm]	1.22^{II}	0.133	0.167
F_t [N]	62.5	25.6	37.3
Contact area [mm^2]	6.4	24	28
F [N]	7.8	3.2	4.7
F_{test} [N]	10	10	10

Table 3.11: Ring pack characteristics, nominal contact pressure, specimen characteristics and normal force to be applied in the Cameron Plint machine.

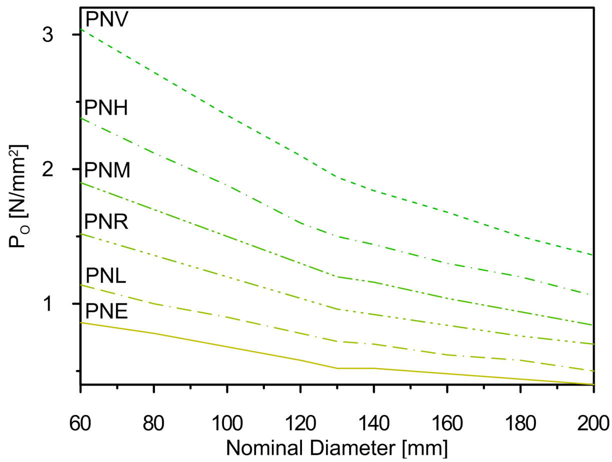

Figure 3.11: Oil control ring nominal contact pressure P_o for different bore diameters and scraping classes. (Adapted from SAE J2003-1998).

Test methodology and friction calculations

The friction coefficient results obtained in the screening test where load, speed and oil viscosity are independent variables which will be studied by means of a multivariate analysis ANOVA with interactions at level two. For the results of the second test, the friction coefficient for each oil and speed will be plotted against the Sommerfeld number defined in terms of the speed, the load held and the lubricant viscosity as can be seen in equation 3.2. In the case of piston rings, the load is changed by the contact pressure value.

$$Hersey = \frac{\eta U}{P} \tag{3.2}$$

It is important to mention that the η value is taken directly from the extrapolation made by the HTHS capillary viscometer software, based on the kinematic viscosity at 40° C and 100° C and the density curve of the tested oils as it was described in sections 3.6.1 and 3.6.2.

3.5.2 WAM Machine

The ball-on-disc friction measurements were conducted in a Wedeven Associates Machine (WAM). As described in [62] this device use a ball loaded against a solid disc resulting in a circular EHL contact. The tribometer has a constant oil supply from the center of the disc and the rotation of both, ball and disc, drags the lubricant into the contact where a lubricant layer is formed. The ball and the disc rotatory movements are driven by two independent electric motors, the former to a speed up to 25000 min^{-1} and the latter up to 12000 min^{-1} (see figure 3.12).

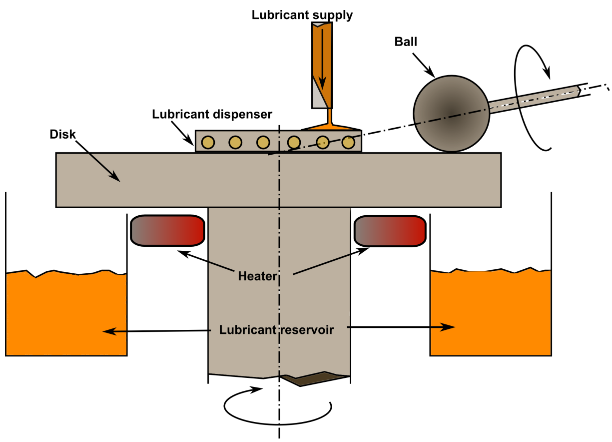

Figure 3.12: WAM machine, ball-on-disc test device.

From the test geometric configuration and rotational speeds, the ball linear speed U_b and the disc linear speed at the contact U_d can be calculated. From those speeds the lubricant entrainment speed is given by the equation 3.3:

[II] Given by normal contact pressure value P_o taken from SAE J2003 (see figure 3.11)

$$U_e = \frac{U_b + U_d}{2} \tag{3.3}$$

As the rotational speed of ball and disc are independent, different linear speeds at the ball and disc contact can be achieved, resulting in a sliding contact. The Slide to Roll Ratio is defined by the equation 3.4

$$SRR = \frac{U_b - U_d}{U_e} \tag{3.4}$$

Load cells are used to measure the force on the three principal axes and to calculate the contact friction coefficient.

Test points

A test plan including different entrainment speeds, Slide to Roll Ratios (SRR) and working temperatures was set for the two engine oils used in the CNG buses described in section 3.4.2. The load used was 300 N, equivalent to 1.94 Gpa in maximum Hertzian pressure, a usual value found at the valvetrain of Heavy Duty engines. The complete description of the test conditions is shown in the table 3.12.

Entrainment speed [m/s]	1, 2.5, 4
Slide to roll ratio (SRR)	0.0002 to 1.05
Pressure (GPa)	1.94
Temperature [° C]	40, 80
Oils	Low and High viscosity

Table 3.12: Test points for the WAM - Machine test

3.6 Viscosity, density and other rheologic measurements of fuels and oils

As it was exposed in section 2.5.1, the viscosity is the oil's physical property which has shown a sound correlation with the capability to reduce friction in ICE at any tribocontact working under the hydrodynamic regime. During this thesis development, kinematic viscosity, HTHS viscosity where controlled by an oil sampling program during the fleet test. In the same way, these values where used to find the dynamic viscosity at low temperature values in order to calculate the Stribeck number to compare the performance of those engine oils in the tribometers tests. The instruments and methodologies used for undertake this task are describe in this section.

3.6.1 Hydrometer

A hydrometer is an instrument that measures the specific gravity (relative density) of engine oils, this is, the ratio of the density of the oil to the density of water. Operation of the hydrometer is based on Archimedes' principle that a solid suspended in a fluid is buoyed by a force equal to the weight of the fluid displaced by the submerged part of the suspended solid. Thus, the lower the density of the substance, the farther the hydrometer sinks. Thus, it is based on the principle of flotation.

A hydrometer is usually made of glass, and consists of a cylindrical stem and a bulb weighted with mercury or lead shot to make it float upright. The liquid to test is poured into a tall container, often a graduated cylinder, and the hydrometer is gently lowered into the liquid until it floats freely. The point at which the surface of the liquid touches the stem of the hydrometer correlates to specific gravity. Hydrometers usually contain a scale inside the stem, so that the person using it can read specific gravity . A variety of scales exist for different oils and an extend of the methodology can be seen in the ASTM D1298 standard[63].

3.6.2 Viscometers

As seen in the the section 2.4.3 the viscosity of Newtonian fluids does not depend on the shear rate for a given temperature. However, this is not the case for non-Newtonian fluids, as multigrade engine oils. Each type of viscosity describes the oil behavior for specific engine operation conditions (*i.e.* temperature, shear rate, pressure, so on.) and engine subsystems (*i.e.* piston assembly, bearings, valvetrain, etc). The procedures and instruments used to measure engine oil's viscosity under the different operation conditions and engine subsystems described on section 2.4.2 are made in the following sections.

Capillary viscometer

This type of viscometer is based on the principle that a specific volume of fluid will flow through the capillary following the ASTM D 445[64] (see figure 3.13). The "kinematic viscosity" is given by the time required for this fluid volume to flow through the capillary. The flow must be laminar and the deductions are based on Poiseuille's law for steady viscous flow in a pipe.

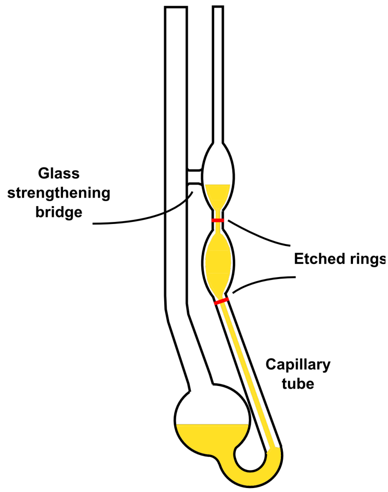

Figure 3.13: Cannon-Fenske Capillary viscometer.

Assuming that the fluid is Newtonian, the kinematic viscosity is given by the equation 3.5, where v is the kinematic viscosity, r is the capillary radius, g is the force of gravity, l is the mean hydro-static head, L is the capillary length, V is the volume of the fluid flowing, Δt is the flow time through the capillary and k is the capillary constant. k has to be determined experimentally by measuring with a reference oil which viscosity is known, and has a different value for each viscometer[65]. This type of viscometer is used at temperatures of 40° C and 100° C, the temperatures from which the viscosity index (V.I.) can be read from the tables of ASTM D 2270[66, 67]

$$v = \frac{\pi r^4 g l t}{8LV} = k(t_2 - t_1) = k\Delta t \tag{3.5}$$

HTHS Capillary viscometer

This instrument has been designed to meet the ASTM D5481 standard[68] which reproduces shear rate and temperature conditions representative of the bearings of automotive engines in severe service. This instrument (see figure 3.14) allows the laboratory determination of (HTHS) viscosity of engine oils at a temperature of 150° C using a multicell capillary viscometer containing pressure, temperature, and timing instrumentation.

Figure 3.14: HTHS Capillary viscometer.

The shear rate for this test method corresponds to an apparent shear rate at the wall of 1.4 million reciprocal seconds (1.4 x 10^6 s^{-1}). This shear rate has been found to decrease the discrepancy between this test method and other high-temperature high-shear test methods ASTM D4683 and ASTM D4741 used for engine oil specifications[69, 70]. Viscosities are determined directly from calibrations that have been established with Newtonian oils with nominal viscosities from 1.4 mPas to 5.0 mPas at 150° C.

This viscometer is designed to determine the viscosity of engine oils, typically at 150° C. Oil samples are first introduced into the viscometric cells at the top of the HTHS. The oils then flow through small glass capillaries under pressure to achieve the desired shear rate. The five viscometric cells in the instrument may be operated in rapid succession. A digital stop-clock measures flow (efflux) time within 0.01 s. Flow times, temperature, and pressure are all displayed digitally.

Data can be analyzed with the computer software and test results displayed and printed. [71]

$$S = \frac{4V}{\pi r^3 t} \tag{3.6}$$

Where V is the capillary volume in mm^3, r the capillary radius in mm and t is the time in s.

The HTHS viscometer comprises 5 measure cells placed in a metal block which temperature can be controlled The HTHS viscometer is supplied with a digital temperature control system (variable from 30° C to 150° C). Pressure and time controller are also present. Each measure cell contains a precision glass capillary in order to adjust the sample volume to the preset value. The oil viscosity is obtained from the determination of the required pressure to reach a flow correspondent to a shear rate of $1.4x10^6 s^{-1}$ in the wall. Each cell calibration is used to determine the viscosity at the measured pressure, by comparing these values to known pressure and flow correlation of Newtonian calibration oils.

This test repeatability, defined as the difference of the results of consecutive tests performed in the equipment by a single operator at constant operation conditions, was around 1.6%. The reproducibility, defined as the difference in the outcome of different tests performed by independent operators in different laboratories, only reached 5.4%.

Capillary diameter [mm]	0.15
Capillary length [mm]	$15-18$
Temperature control	$\pm 1°C$
Precision [kPa]	$350-3500$
Pressure control	±1%
Sample volume [ml]	7 ± 1
Viscometer	HTHS Series II Viscometer
Dimensions [mm]	$521x387x686$
Weight [kg]	40.5
Operation conditions	$15°C-30°C$, $10\%-90\%RH$

Table 3.13: Specifications of HTHS capillary viscometer

The test procedure for this viscometer can be seen in the diagram 3.15 and it is described in the following paragraph:

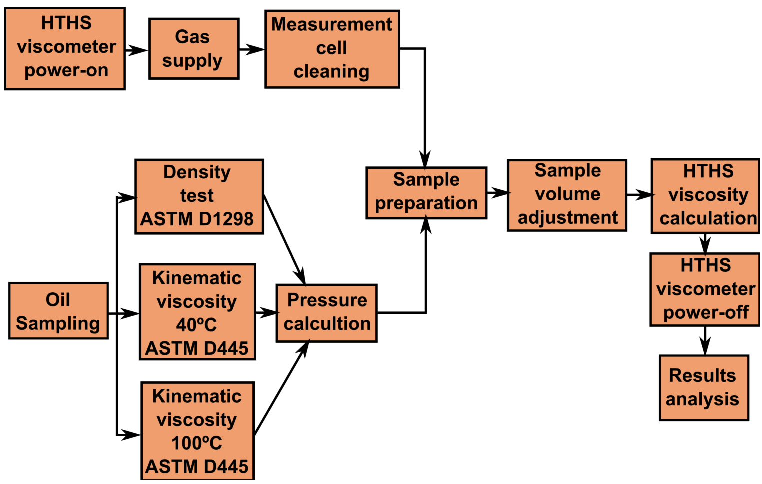

Figure 3.15: HTHS Capillary viscometer measurement methodology.

- HTHS Power-on: The HTHS capillary viscometer is switched on with the re button. The temperature is set to 150° C. The instrument will reach this temperature in 30 minutes.

- Gas supply: The valve of the nitrogen bottle is opened until the pressure reaches 400 psi. At the measurements outlets rubber stoppers should be placed. In the same way a plastic reservoir should be placed downstream the cells.

- Measurement cell cleaning: The pressure knob has to be in the PRESSURE position. A syringe has to be placed in the measurement cell where the sample will be placed. Then the syringe must be filled with 5 ml of sample oil, in order to drag remainders of previous tests.

  Once the syringe nozzle has been connected to the plastic tube, the green valve should be opened (vertical position) and the the pump switch should be turned ON. Once the syringe content has been dragged completely the green valve should be closed again (horizontal position), and the pump switch should be turned OFF.

  The pressure is then set between 100 psi - 150 psi with the INCREASE/DECREASE buttons of the control panel. The stopper correspondent to the capillary where the syringe is placed should be taken off and the START/STOP button should be pressed allowing the oil to flow through the measurement cell downstream towards the plastic reservoir. Then the stopper should be put on again.

- Sample preparation: All the steps described in the cleaning procedure should be followed until the closing of the green valve. There, the filling of the syringe should be made with 10 ml of sample, always being sure that temperature is at 150° C. The time must be set to 15 minutes in the software chronometer. This will be the necessary time for the sample to reach the 150° C temperature. The pressure should be set 2 psi above the goal value calculated with the software from the density and kinematic viscosity values (normally it will fall during the test).

- Sample volume adjustment: The green valve should be opened (vertical position). The pressure switch should be placed in VACUUM. The syringe must be covered. The pump switch should be pressed in ON. Bubbles appearance after extracting 0.5 ml is signal of a good adjustment. The pump should be turned OFF again and the green valve closed. The syringe should be uncovered.

- Sample analysis: With 150° C and a pressure value 2 psi above the target value the stopper should be removed. The START/STOP button should be pressed and then the oil will flow through the measurement cell towards the outlet and the plastic reservoir downstream. The pressure value at 10 seconds after pressing the START/STOP button should be recorded. Once the test has finished the time should be recorded as well.

- HTHS calculation: The pressure values at 10 seconds and the total time of the test should be introduced in the the software and the CALCULATE VISCOSITY option should be chosen. The value in the screen is the HTHS viscosity at 150° C value. If this value is somehow out the limits, the software will calculate another test pressure.

- Viscometer cleaning: The system will be cleaned by letting flow 5 ml to 10 m of cleaning dilution. If the syringe remains dirty, it should be removed from the equipment and cleaned with solvent.

- Turning off the viscometer: The nitrogen valve should be closed. The red button should be pressed to switch off the viscometer.

3.6.3 Compensated jacket calorimeter

This instrument was used to determine the variability of the diesel fuel heat value during the fleet test. The compensated jacket calorimeter can be seen in figure 3.16. It consists mainly of a cup to contain the fuel sample, a stainless steel combustion vessel to contain the fuel sample and the O_2, a stirrer, a thermometer, an ignition circuit to induce the combustion process and jacket vessel to be filled with water where the combustion vessel will be placed. To measure the heat release given from the combustion, a certain amount of fuel is placed inside the combustion vessel which is ignite by means of an electrical current. As the fuel burns, it heats up pressurized O_2 used before to fill the combustion vessel, as consequence, the volume of water in the surroundings of the vessel will heat up as well. The change of waters temperature allows to calculate the calorie content of the fuel. The details of this procedure can be found in ASTM D240 [72]. It has to be mentioned also, that every measurement done with this equipment has been repeated three times, being the reported value the average of those measurements.

Figure 3.16: Calorimeter used to evaluate the Q_{HV} of the diesel fuel during the fleet test.

3.6.4 Karl Fischer tritration for moisture determination

The Karl Fischer method has been used to determine the water content of the diesel fuel samples during the fleet test. It is a chemical analysis procedure which is based on the oxidation of sulfur dioxide by iodine in a methanolic hydroxide solution. The instrument showed in figure 3.17 performs the tritration coulometrically, this means that the iodine participating in the reaction is generated in the tritration cell by an electrochemical oxidation. Further information of this technique can be found in ASTM D6304 [73].

Figure 3.17: Karl fischer tritator.

3.6.5 Fourier transform infrared spectroscopy (FTIR)

A Fourier transform infrared spectroscope is an instrument which obtains an infrared spectrum of absorption or emission of a given sample, in this case, Diesel fuel (see figure 3.18. To obtain the spectrum the FTIR uses the principles of the Michelson interferometer which are widely explained in Gomez 2013[74]. In ASTM E2412 [75] the description of oil analysis through FTIR is established.

Figure 3.18: FTIR instrument.

Moreover, to measure Biodiesel content, as described in Bradley 2007 [76]. The measurement principle is to find in the spectra peaks of absorbance of ester bonds, present only in biodesel at 1750 cm^{-1} (C=O vibration) and around 1170 - 1200 cm^{-1} (C-O vibration) as shown in figure 3.19.

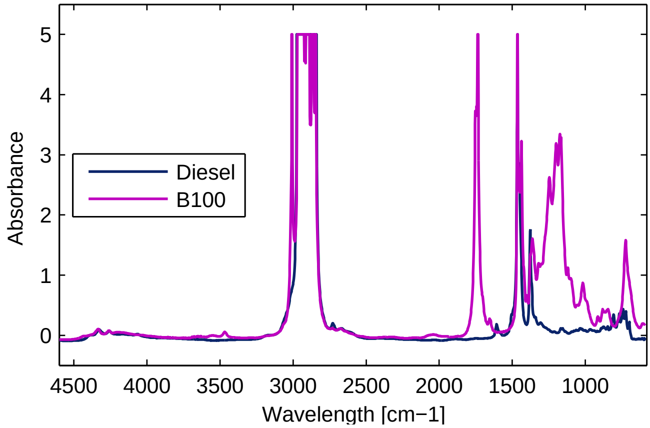

Figure 3.19: Infrared spectra of several petrochemicals where the differences between diesel and biodiesel can be observed. (Adapted from [77])

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3.6.6 Flash point

The flash point is a measure of how volatile an hydrocarbon is, defined as the minimum temperature at the fuel ignites in presence of a flame. For this thesis studies a closed cup tester as the one shown in figure 3.20 was used. More details about this procedure can be found in ASTM D93 [78].

3.6.7 Liquid petrol fuels distillations and cetane number calculation

The cetane index can be calculated using a correlation established with the distillations at recovery temperatures. Using a distillation equipment as the one shown in figure 3.21, this recovery temperatures are found. A regular distillator comprises a distillation balloon where the fuel sample is placed. A bottom heater, makes the sample temperature rises as the lighter fractions of fuel start to evaporate. Then the evaporated fuel is condensed and re directed to a test tube where the liquid fuel is recovered. Temperature is registered at different 10%, 50% and 90% recovery values.

The correlation formula given by the ASTM D4737 is shown in equation 3.7, where CCI is the cetane index, D is the density at 15° C (g/ml), T_{10} is 10% recovery

Figure 3.20: Pensky-Martens closed cup flash point tester.

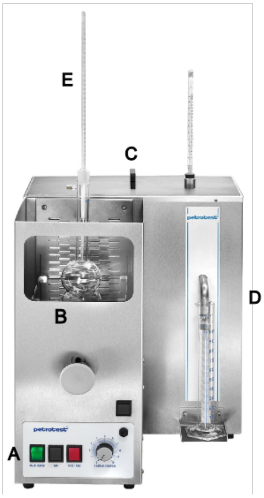

Figure 3.21: ASTM D86 distillation equipment. (a) ON/OFF switch, (b) distillation balloon, (c) condenser, (d) test tube, (e) thermometer.

temperature in Celsius, T_{50} is 50% recovery temperature, T_{90} is 90% recovery temperature. More details can be found in ASTM D4737 [79].

$$CCI = -386.26(D) + 0.1740(T_{10}) + 0.1215(T_{50}) + 0.01850(T_{90}) + 297.42 \quad (3.7)$$

References

[1] MONTGOMERY, D. *Design and Analysis of Experiments*. Ed. by W. ANDERSON, K. HEPBURN, S. RUSSELL, K. MURPHY, and E. AIELLO. Fifth Edition. New York: John Wiley & Sons, INC., 2001.

[2] MURTONEN, T. and M. SUTTON. "New Crankcase Lubricants for Heavy-duty Diesel Engines: Effect on Fuel Consumption and Exhaust Emissions". *SAE Technical Paper 2005-01-3717* (2005).

[3] DAM, W. VAN, P. KLEIJWEGT, M. TORREMAN, and G. PARSONS. "The Lubricant Contribution to Improved Fuel Economy". *SAE Technical Paper 2009-01-2856* (2009).

[4] PARK, S., Y. CHO, K. SUNG, and N. HAN. "The Effect of Viscosity and Friction Modifier on Fuel Economy and the Relationship Between Fuel Economy and Friction". *SAE Technical Paper* 2009-01-2662 (2009).

[5] KANEKO, T., M. YAMASHITA, M. OHORI, and M. MURAKAMI. "Development of Low Sulfated Ash and Fuel Economy Diesel Engine Oil". *SAE Techinical Paper* 2009-01-1845 (2009).

[6] FONTARAS, G., E. VOUITSIS, and Z. SAMARAS. "Experimental Evaluation of the Fuel Consumption and Emissions Reduction Potential of Low Viscosity Lubricants". *SAE Technical Paper 2009-01-1803* (2009).

[7] DAM, W. VAN et al. "The Impact of Lubricant Viscosity and Additive Chemistry on Fuel Economy in Heavy Duty Diesel Engines". *SAE Technical Paper 2011-01-2124* (2011).

[8] SINGH, D., M. SHUKLA, S. SINGH, Y. SINGH SR, and N. SINGH. "Fuel Economy of SI Vehicle Fueled by Ethanol/Gasoline Blends - Role of Engine Lubricant Viscosity". *SAE Technical Paper 2013-01-0051* (2013).

[9] CARDEN, P., C. PISANI, J. ANDERSSON, and I. FIELD. "The Effect of Low Viscosity Oil on the Wear, Friction and Fuel Consumption of a Heavy Duty Truck Engine". *SAE Technical Paper* 2013-01-0331 (2013).

[10] MANNI, M. and S. FLORIO. "An Experimental Evaluation of the Impact of Ultra Low Viscosity Engine Oils on Fuel Economy and CO2 Emissions". *SAE Technical Paper 2013-01-2566* (2013).

[11] MACIAN, V., B. TORMOS, S. RUIZ, L. RAMÍREZ, and J. DE DIEGO. "In-Use Comparison Test to Evaluate the Effect of Low Viscosity Oils on Fuel Consumption of Diesel and CNG Public Buses". *SAE Technical Paper* 2014-01-2794 (2014).

[12] BROWNE, D. et al. "Novel Fuel Efficiency Engine Lubricants for Urban Transit Applications". *SAE Technical Paper* 2014-01-2793 (2014).

[13] TAYLOR, O. P., R. PEARSON, R. STONE, P. CARDEN, and H. BALLARD. "Tribological Behavior of Low Viscosity Lubricants in the Piston to Bore Zone of a Modern Spark Ignition Engine". *SAE Technical Paper* 2014-01-2859 (2014).

[14] THE COORDINATING EUROPEAN COUNCIL. *Fuel Economy Effects of Engine Lubricants (MB M111 E20). CEC L-54-96.*

[15] ASTM INTERNATIONAL. *ASTM D7589-16, Standard Test Method for Measurement of Effects of Automotive Engine Oils on Fuel Economy of Passenger Cars and Light-Duty Trucks in Sequence VID Spark Ignition Engine.* Tech. rep. West Conshohocken, PA: ASTM International, 2016.

[16] ILSAC. *Consortium to Develop a New for Engine Oils - Final Report -*. Tech. rep. ILSAC, 2008.

[17] MCFALL, D. "Road Map for Engine Oils. 'Six D' Fuel Economy Test Points the Way". *Lubes 'n' Greases* (2006), pp. 2–4.

[18] INTERNATIONAL LUBRICANT STANDARDIZATION AND APPROVAL COMMITTEE. *ILSAC GF-5 Standard for Passenger Car Engine Oils*. Tech. rep. 2009.

[19] FUENTES, E. and K. KANASE. "Relevance of Lubrication Regimes in the Study of Bearing Wear". *Critical Component in Heavy Duty Engines*. Ed. by P. LAKSHMINARAYANAN and N. S. NAYAK. First Edition. Singapore: John Wiley & Sons, INC., 2011. Chap. 12, pp. 199–252.

[20] ENGELJEHRINGER, K. *Emission : Heavy Duty and Off-Road. Emission Test Systems.* Tech. rep. AVL, 2014.

[21] DEMUYNCK, J. et al. "Recommendations for the new WLTP cycle based on an analysis of vehicle emission measurements on NEDC and CADC". *Energy Policy* 49 (2012), pp. 234–242.

[22] SILEGHEM, L., D. BOSTEELS, J. MAY, C. FAVRE, and S. VERHELST. "Analysis of vehicle emission measurements on the new WLTC, the NEDC and the CADC". *Transportation Research Part D: Transport and Environment* 32 (2014), pp. 70–85.

[23] TIETGE, U. et al. *From Laboratory To Road. a 2014 Update of Official and Real-World Fuel Consumption and CO2 Values for Passenger Cars*. Tech. rep. September. Berlin: International Council on Clean Transportation Europe, 2015.

[24] SAE INTERNATIONAL. *SAE J1321. Fuel Consumption Test Procedure-Type II*. Tech. rep. SAE International, 2012, p. 29.

[25] U.S. ENVIRONMENTAL PROTECTION AGENCY. *Interim Test Method for Verifying Fuel-Saving Components for SmartWay: Modifications to SAE J1321*. 2011.

[26] U.S. ENVIRONMENTAL PROTECTION AGENCY. *Track Testing for SmartWay Verification*. 2014.

[27] NATIONAL RESEARCH COUNCIL. *Technologies and Approaches to Reducing the Fuel Consumption of Medium- and Heavy-Duty Vehicles*. Tech. rep. Washington D.C.: National Academy of Sciences, 2010.

[28] HOLMBERG, K., P. ANDERSSON, and A. ERDEMIR. "Global energy consumption due to friction in passenger cars". *Tribology International* 47 (Mar. 2012), pp. 221–234.

[29] MAHLE GMBH. *Pistons and engine testing*. Ed. by E. SCHMITT and E. LANGE. First Edition. Stuttgart: AZ Druck und Datentechnik, 2012.

[30] CUI, J., S. OBEROI, S. BRIGGS, I. GOLDMINTS, and D. BAXTER. "Measuring Fuel Efficiency in Various Driving Cycles : How to Get Maximum Fuel Economy Improvement from the Lubricant". *SAE Technical Paper* 2015-01-2042 (2015).

[31] HASSON, V. et al. "Exhaust Toxicological Profiles from Direct Injection Engine With and Without Diesel Particulate Filter Regeneration During NEDC Cycling". *SAE Technical Paper* 2009-01-1090 (2009).

[32] RUNDO, M. and P. PUMP. "Experimental Procedure for Measuring the Energy Consumption of IC Engine Lubricating Pumps during a NEDC Driving Cycle". *SAE Technical Paper 2009-01-1919* 2.1 (2009), pp. 1690–1700.

[33] GIAKOUMIS, E. G. "Lubricating oil effects on the transient performance of a turbocharged diesel engine". *Energy* 35.2 (2010), pp. 864–873.

[34] KOFFLER, C. and K. ROHDE-BRANDENBURGER. "On the calculation of fuel savings through lightweight design in automotive life cycle assessments". *International Journal of Life Cycle Assessment* 15.1 (2010), pp. 128–135.

[35] ALTENSTRASSER, H., Y. KATO, N. KEUTH, and T. WINSEL. "MiL-Based Calibration and Validation of Diesel-ECU Models Using Emission and Fuel Consumption Prediction during Dynamic Warm-Up Tests (NEDC)". *SAE Technical Paper 2012-01-0432* (2012), pp. 3–10.

[36] KIM, J., E. YIM, C. JEON, C. JUNG, and B. HAN. "Cold performance of various biodiesel fuel blends at low temperature". *International Journal of Automotive Technology* 13.2 (2012), pp. 293–300.

[37] KIM, Y. et al. “Fuel Effect on Particle Emissions of a Direct Injection Engine”. *SAE Technical Paper* 2013-01-1559 (2013).

[38] SHIM, B. J., K. S. PARK, J. M. KOO, and S. H. JIN. “Work and speed based engine operation condition analysis for new European driving cycle (NEDC)”. *Journal of Mechanical Science and Technology* 28.2 (2014), pp. 755–761.

[39] MOHAMMADPOUR, M., R. RAHMANI, and H. RAHNEJAT. “The Effect of Cylinder De-Activation on Thermo-Friction Characteristics of the Connecting Rod Bearing in the New European Drive Cycle (NEDC)”. *SAE Technical Papers* 2014-June (2014).

[40] CIPOLLONE, R., D. DI BATTISTA, and M. MAURIELLO. “Effects of Oil Warm up Acceleration on the Fuel Consumption of Reciprocating Internal Combustion Engines”. *Energy Procedia* 82 (2015), pp. 1–8.

[41] DI BATTISTA, D. and R. CIPOLLONE. “Experimental and numerical assessment of methods to reduce warm up time of engine lubricant oil”. *Applied Energy* 162 (2016), pp. 570–580.

[42] MEHROTRA, A., S. JUTTU, S. S. RAVISHANKAR, G. P. RAMBHAJI, and J. G. SURYAQANSHI. “Simultaneous Reduction of NOx and PM Emissions through Low Temperature EGR Cooling in Diesel Engines”. *SAE Technical Paper 2014-01-2803* (2016).

[43] BIELACZYC, P., J. WOODBURN, and A. SZCZOTKA. “A Comparison of Carbon Dioxide Exhaust Emissions and Fuel Consumption for Vehicles Tested over the NEDC , FTP-75 and WLTC Chassis Dynamometer Test Cycles”. *SAE Technical Paper 2015-01-1065* (2016).

[44] PAYRI, F. and J. M. DESANTES. *Motores de Combustión Interna Alternativos*. Barcelona: Reverté, 2005.

[45] EUROPEAN COMMISSION. *Transport and Enviroment: Road Vehicles.* 2015.

[46] MACIÁN, V., B. TORMOS, S. RUIZ, and G. MIRÓ. “Low viscosity engine oils: Study of wear effects and oil key parameters in a heavy duty engine fleet test”. *Tribology International* 94 (2016), pp. 240–248.

[47] MALACZYNSKI, G. W., A. H. HAMDI, A. A. ELMOURSI, and X. QIU. “Diamond-like carbon coating for aluminum 390 alloy - automotive applications”. *Surface and Coatings Technology* 93.2-3 (1997), pp. 280–286.

[48] GALLIGAN, J., A. A. TORRANCE, and G. LIRAUT. “A scuffing test for piston ring/bore combinations: Pt. II. Formulated motor lubrication”. *Wear* 236.1-2 (1999), pp. 210–220.

[49] WANG, Y., K. BROGAN, and S. C. TUNG. "Wear and scuffing characteristics of composite polymer and nickel/ceramic composite coated piston skirts against aluminum and cast iron cylinder bores". *Wear* 250-251.PART 1 (2001), pp. 706–717.

[50] TRUHAN, J. J., J. QU, and P. J. BLAU. "The effect of lubricating oil condition on the friction and wear of piston ring and cylinder liner materials in a reciprocating bench test". *Wear* 259.7-12 (2005), pp. 1048–1055.

[51] TRUHAN, J. J., J. QU, and P. J. BLAU. "A rig test to measure friction and wear of heavy duty diesel engine piston rings and cylinder liners using realistic lubricants". *Tribology International* 38.3 (Mar. 2005), pp. 211–218.

[52] JOHANSSON, S., P. H. NILSSON, R. OHLSSON, and B. G. ROSÉN. "Experimental friction evaluation of cylinder liner/piston ring contact". *Wear* 271.3-4 (June 2011), pp. 625–633.

[53] KAPSIZ, M., M. DURAT, and F. FICICI. "Friction and wear studies between cylinder liner and piston ring pair using Taguchi design method". *Advances in Engineering Software* 42.8 (Aug. 2011), pp. 595–603.

[54] SPENCER, A., E. Y. AVAN, A. ALMQVIST, R. S. DWYER-JOYCE, and R. LARSSON. "An experimental and numerical investigation of frictional losses and film thickness for four cylinder liner variants for a heavy duty diesel engine". *Proceedings of the Institution of Mechanical Engineers, Part J: Journal of Engineering Tribology* 227.12 (2013), pp. 1319–1333.

[55] DARUT, G., H. LIAO, C. CODDET, J. M. BORDES, and M. DIABY. "Steel coating application for engine block bores by Plasma Transferred Wire Arc spraying process". *Surface and Coatings Technology* 268 (2015), pp. 115–122.

[56] BAKUNIN, V. N., G. N. KUZMINA, M. KASRAI, O. P. PARENAGO, and G. M. BANCROFT. "Tribological behavior and tribofilm composition in lubricated systems containing surface-capped molybdenum sulfide nanoparticles". *Tribology Letters* 22.3 (2006), pp. 289–296.

[57] BJÖRLING, M., P. ISAKSSON, P. MARKLUND, and R. LARSSON. "The influence of DLC coating on EHL friction coefficient". *Tribology Letters* 47.2 (2012), pp. 285–294.

[58] BJÖRLING, M., W. HABCHI, S. BAIR, R. LARSSON, and P. MARKLUND. "Friction reduction in elastohydrodynamic contacts by thin-layer thermal insulation". *Tribology Letters* 53.2 (2014), pp. 477–486.

[59] BJÖRLING, M., R. LARSSON, and P. MARKLUND. "The effect of DLC coating thickness on elstohydrodynamic friction". *Tribology Letters* 55.2 (2014), pp. 353–362.

[60] Björling, M., J. Miettinen, P. Marklund, A. Lehtovaara, and R. Larsson. "The correlation between gear contact friction and ball on disc friction measurements". *Tribology International* 83 (2015), pp. 114–119.

[61] SAE International. *Internal Combustion Engines: Piston Rings Coil Spring Loaded Oil Control Rings*. Tech. rep. 724. SAE International, 2008, pp. 776–790.

[62] Björling, M., R. Larsson, P. Marklund, and E. Kassfeldt. "EHL friction mapping - The influence of lubricant, roughness, speed and slide to roll." *Proceedings of the Institution of Mechanical Engineers, Part J: Journal of Engineering Tribology.* 225.7 (2011), pp. 671–681.

[63] ASTM International. *ASTM D1298-12b, Standard Test Method for Density, Relative Density, or API Gravity of Crude Petroleum and Liquid Petroleum Products by Hydrometer Method*. Tech. rep. C. West Conshohocken, PA: ASTM International, 2012, pp. 1–8.

[64] ASTM International. *ASTM D445-09: Standard Test Method for Kinematic Viscosity of Transparent and Opaque Liquids (and Calculation of Dynamic Viscosity)*. Tech. rep. West Conshohocken, PA: ASTM International, 2015.

[65] Stachowiak, G. W. and A. W. Batchelor. *Engineering Tribology*. Third Edition. Oxford: Elsevier Butterworth - Heinemann, 2005.

[66] Haycock, R. F., A. J. Caines, and J. Hillie. *Automotive lubricants reference book*. First Edition. Warrendale: Society of Automotive Engineers, Inc., 2004.

[67] ASTM International. *ASTM D2270-10e1, Standard Practice for Calculating Viscosity Index From Kinematic Viscosity at 40 and 100° C*. Tech. rep. West Conshohocken, PA: ASTM International, 2010.

[68] ASTM International. *ASTM D5481, Standard Test Method for Measuring Apparent Viscosity at High-Temperature and High-Shear Rate by Multicell Capillary Viscometer*. Tech. rep. West Conshohocken, PA: ASTM International, 2013.

[69] ASTM International. *ASTM D4741, Standard Test Method for Measuring Viscosity at High Temperature and High Shear Rate by Tapered-Plug Viscometer*. Tech. rep. West Conshohocken, PA: ASTM International, 2013.

[70] ASTM International. *ASTM D4683, Standard Test Method for Measuring Viscosity at High Shear Rate and High Temperature by Tapered Bearing Simulator*. Tech. rep. West Conshohocken, PA: ASTM International, 2013.

[71] Cannon. *Cannon HTHS Series II*.

[72] ASTM INTERNATIONAL. *ASTM D240-14, Standard Test Method for Heat of Combustion of Liquid Hydrocarbon Fuels by Bomb Calorimeter*. Tech. rep. West Conshohocken, PA: ASTM International, 2014, pp. 1–9.

[73] ASTM INTERNATIONAL. *D 6304: Test method for determination of water in petroleum products, lubricating oils, and additives by coulometric karl fisher titration*. Tech. rep. West Conshohocken, PA: ASTM International, 2007.

[74] GÓMEZ, Y. "Contribución al desarrollo y mejora para la cuantificación de la degradación en aceites lubricantes usados de MCIA a través de la técnica de espectometria infrarroja por transformada de Fourier (FT-IR)". Doctoral Thesis. Universitat Politècnica de València, 2013.

[75] ASTM INTERNATIONAL. *ASTM E2412-10, Standard Practice for Condition Monitoring of Used Lubricants by Trend Analysis Using Fourier Transform Infrared (FT-IR) Spectrometry*. Tech. rep. West Conshohocken, PA: ASTM International, 2010.

[76] BRADLEY, M. "Biodiesel (FAME) Analysis by FT-IR". *Thermoscietific* X (2007), pp. 2–4.

[77] HEISE, H. M. et al. "Recent advances in mid- and near-infrared spectroscopy with applications for research and teaching, focusing on petrochemistry and biotechnology relevant products". *European Journal of Physics* 34.6 (2013), S139–S159.

[78] ASTM INTERNATIONAL. *ASTM D93-15a, Standard Test Methods for Flash Point by Pensky-Martens Closed Cup Tester*. Tech. rep. West Conshohocken, PA: ASTM International, 2015.

[79] ASTM INTERNATIONAL. *ASTM D4737, Standard Test Method for Calculated Cetane Index by Four Variable Equation*. Tech. rep. West Conshohocken, PA: ASTM International, 2010, pp. 1–5.

Chapter 4

Results and Discussion

The aim of this chapter is to present the results of the different tests performed using the methodologies exposed during chapter 3, all of them focused on finding the variation of either the friction coefficient, friction torque or fuel consumption. The results will be shown following the order presented in chapter 3 starting with the three different engine bed tests which used a Light Duty diesel engine and were focused on variations in engine friction power and fuel consumption; a motored test, a fired stationary test and a transient cycle test. Then the fleet test results with their respective statistical analysis of fuel consumption and CO_2 emissions reduction through the use of LVEO is portrayed to finish the chapter with the tribometer tests centered over friction coefficient differences of some tribo-contacts of the engine when LVEO are used.

4.1 Introduction

This chapter comprises the results obtained experimentally during this thesis development followed by the analysis of the research findings in order to answer the research questions exposed in chapter 1. Data were analyzed to identify, describe and explore the relationship between the use of LVEO and friction coefficient reduction and its consequent fuel consumption and CO_2 emissions decrease.

It has to be remembered that three different types of tests were conducted in order to establish and measure this relationship; engine bed tests (motored, fired under stationary conditions and fired under transient conditions), a fleet test, were different bus models used baseline and candidate oils in a long term test controlling daily fuel consumption and finally, laboratory tests using two different tribometers, the Cameron Plint TE77 machine and the ball-on-disc WAM machine.

4.2 Engine bed test results

4.2.1 Engine motored test results

As stated in section 3.3.4 the procedure in this test consisted in measuring the torque consumed by the dynamometer to drive the engine at a given engine speed using different commercial engine oils. Following the materials and method described in the section mentioned above the results obtained will be shown in the following section.

As it can be seen in figure 4.1 and table 4.1, the use of a less viscous oil led to significant lower motor torque values. The friction data from this test presented an increasing trend paired with engine speed, with a local peak at 1500 min^{-1}. The unusual shape of the torque curves for both oils could be explained by an irregular behavior of pumping losses detected on these engine speeds which lead to indicated pressure increase.

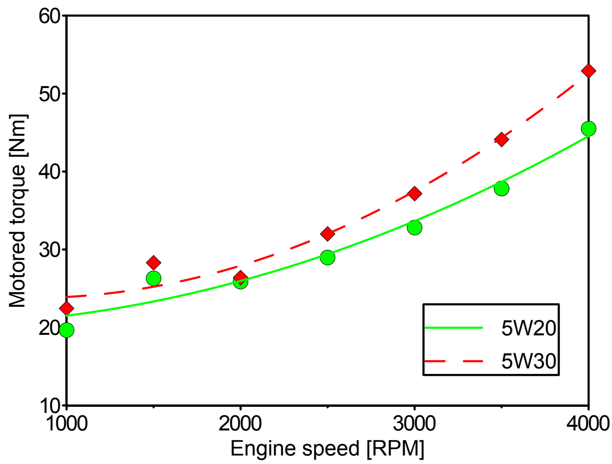

Figure 4.1: Motored test results for 5W20 and 5W30 SAE engine oils.

From there to 4000 min^{-1} the data show the typical friction behavior of the hydrodynamic regime, with rising values of required torque as the engine speed was increasing. In the same way, the difference of required torque in percentage between 5W20 and 5W30 oil increased as the engine was reaching higher velocities as it can be seen in figure 4.2.

	Engine speed [min^{-1}]						
Oil	1000	1500	2000	2500	3000	3500	4000
5W30	22.5	28.3	26.4	32.0	37.2	44.1	52.9
5W20	19.7	26.3	25.9	29.0	32.8	37.8	45.5
5W30-5W20 [%]	12.4	7.1	1.9	9.5	11.7	14.3	14.0

Table 4.1: Torque [Nm] required to reach different engine speeds [min^{-1}] during the motored test

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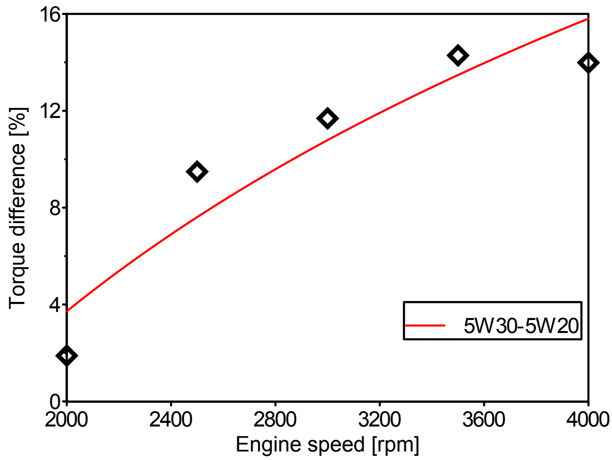

Figure 4.2: Torque differences for 5W20 and 5W30 SAE engine oils at different engine speeds.

4.2.2 Engine motored test results' discussion

The effect of the oil viscosity on torque required is clearly observed. Unfortunately, further tests were not performed in order to analyze the behavior of the different engine subsystems contribution to engine friction. However, from previous studies it is assumable that as engine speed increases, the Stribeck curve of the engine tribo-contacts (see section 2.4.2) moves towards the right. The friction coefficient of engine tribo-contacts already under hydrodynamic regimes will rise due to viscous friction increase. This is the case for piston rings-cylinder liner contact, crankshaft and camshaft journal bearings and piston skirt. On the other hand, the FMEP of those tribo-contacts under boundary and mixed regimes like the valvetrain (cam-tappet), tend to remain quasi-steady in absolute values as demonstrated by Sandoval *et.al.*, and James[1, 2]. Actually, there are studies where the increasing engine speed results in lower friction torque[3–5]. The magnitude of FMEP of each engine tribo-contact is different and even when valvetrain FMEP keeps constant or decrease slightly with increasing engine speed, the total FMEP increases since piston assembly and journal bearings is increasing as showed by Sorrentino *et.al.* through their Willans motored experiments and mathematical models [6]. In addition, the absence of combustion and the prevalence of the hydrodynamic lubrication regime allows the less viscous oil to show its lower inner shear resistance as the engine speed increases as the required torque rises for both SAE 5W20 and SAE 5W30 oil. Despite of the load differences which could be found against a fired test, the motored test confirms the LVEO potential when the engine works at medium or high speeds and low loads.

In the table 4.2 the temperatures at the engine bed cell are portrayed. Even when the engine bed test was provided with an oil cooling circuit, the high ambient temperatures during the SAE 5W20 test confounded the data when engine speed was higher than 2500 min^{-1}. It is assumable that the oil high temperatures had reduced the oil film thickness when SAE 5W20 measurements took place.

Engine speed [min^{-1}]	5W20	5W30
1000	60.7	60.8
1500	65.4	65.1
2000	65.0	64.9
2500	67.7	65.0
3000	71.7	65.0
3500	76.3	68.6
4000	81.1	73.2

Table 4.2: Average oil temperatures in Celsius [° C] at given engine speeds for oils 5W20 and 5W30 during motored test.

4.2.3 Engine fired stationary test results

It has to be remembered that accordingly with the description made in section 3.3.5 for this test, a different baseline oil was used: 15W40 SAE grade engine oil, which properties can be seen in table 3.2. Figure 4.3 and table 4.3 show the results of the screening test for four different engine speeds. An average reduction of 1.64% in the break specific fuel consumption BSFC over the 12 points test when the engine is using SAE 5W30 oil compared against the BSFC when the engine was using SAE 15W40 was found. The results of this test revealed a high correlation between the oil viscosity and the BSFC when the engine works at speeds under 2000 min^{-1} and low loads. Decreases as high as 4% in BSFC when SAE 15W40 is compared to SAE 5W30 as baseline can be found at low engine speeds and low load. Nonetheless, as the load increases, the effect of LVEO changes, leading even to an increase on fuel consumption especially at low engine speeds (e.g. as it can be seen in the table 4.3. BSFC can rise when LVEO are used at low speeds and high loads). These results are consistent with previous studies where the influence of load on the tribological conditions where studied[7]. Apparently this trend is mitigated as the engine speed reach values over 2000 min^{-1}, as it could be expected from the lubrication theory.

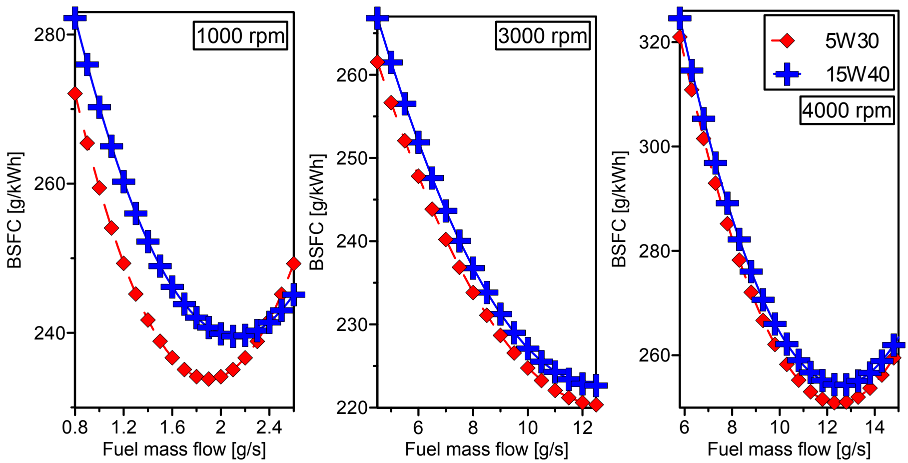

Figure 4.3: Screening test results for 15W40 and 5W30 oils under fired stationary conditions.

	Load [%]		
Engine speed [min^{-1}]	25	50	75
1000	4.44	3.71	-1.02
2000	3.42	1.77	0.24
3000	2.06	1.24	1.07
4000	0.27	1.41	1.05

Table 4.3: Differences of fuel consumption between 15W40 and 5W30 SAE grade engine oils for different engine speeds.

A loss of the effect of LVEO over fuel consumption when the engine load was over the 50% was observed. It could be explained by using the Stribeck theory [8], where an increase in load could lead to a possible increase in the friction coefficient, due the change of the lubrication regime from hydrodynamic to mixed or boundary. It also has to be taken into account that the load increment of 25% of the maximum torque the engine can achieve will not be the same at 1000 min^{-1} than at 4000 min^{-1}. The weight of the load increase over the friction coefficient in a particular engine will vary depending on its constructive and functional design and it will be probably different from one to another engine (i.e. the results concerning this study may not be quantitatively directly extrapolated to other engines).

4.2.4 Engine fired stationary test results' discussion

As it can be seen the lower the speed, greater the influence that load has over the BSFC improvement. As the engine speed increases the relative effect of load variation on the BSFC improvement decreases. It can be said that at lower engine speeds a greater effect of LVEO on fuel consumption could be seen at low loads but this effect rapidly disappears as the load increases. In the other hand, with higher engine speed the effect of LVEO on fuel consumption may could not achieve greater values but it can be maintained over the all range of engine loads.

From the results showed in table 4.3 have been used to find a BSFC reduction surface when SAE 5W30 oil is used. The result can be seen in figure 4.4.

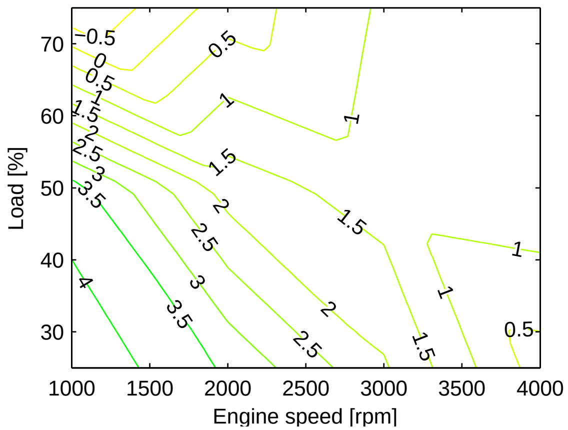

Figure 4.4: Contour map of BSFC Improvement of 5W30 using 15W40 oil as baseline.

Taking into account the results of the motored test (see section 4.2.2) where the use of LVEO reduce the torque required to move the engine at certain speed without combustion and given the fact that this reduction was more noticeable as engine speed was increasing, the results of the screening tests may seem to be contradictory. However, the combustion during the second test makes the difference since other engine losses affect the overall effect of LVEO on engine fuel consumption.

As exposed in section 2.2.5, vehicles and engines overall efficiency depend on several factors like exhaust, cooling, air drag, rolling resistance among others. For ICE, thermal losses are the larger contributors for overall inefficiency. Some interesting studies regarding this phenomena have been made by Fernandez Riberio *et.al.* where a mapping of engine energy losses was presented. Using three different engines the study found that mechanical losses slighty increases with engine speed, however, thermal losses (exhaust and cooling) rises steeply, besides at high engine speeds, the absolute energy of thermal losses is far larger than the energy consumed by mechanical friction losses (420 MJ/h the former and 40 MJ/h the latter at 5000 min^{-1} in a 1.6l SI engine at whole open throttle WOT conditions).[9].

In a similar way, Trattner *et.al.* studied how these losses behave under different engine speeds. The study measured BMEP of different engine losses. The study shows that engines FMEP increases with engine speed, similar to results of the motored test. However, at higher engine speeds exhaust losses and cooling losses increase at a higher rate, reducing the FMEP weight over the engine efficiency[10]. Similar results are shown by Kogo *et.al.* [11]. It can be said that even when LVEO could reduce the mechanical friction losses as the engine speed increases the overall effect over fuel consumption is masked by the effect of thermal losses as plotted in figure 4.4.

In figure 4.5, shows the trend of BSFC improvement when load is increasing for different engine speeds. The graph shows the linear regression for each data set, all of them with good R^2 values (over 0.8) except for 4000 min^{-1} where the correlation was very low (0.45) due an estrange value at 4000 min^{-1} and 25% of load, which could be a possible measurement error.

The ambient temperature, pressure and humidity in the test cell during the fired stationary test performed for the two oils can be seen on table 4.4.

Oil tested	Temperature [° C]	Pressure [mbar]	Humidity [%]
5W30	32.8	1023	31.8
15W40	33.8	1022	30.9

Table 4.4: Test cell ambient conditions when tests took place with 5W30 and 15W40 SAE grade oils.

4.2.5 Engine transient cycle results

Test results indicate that 5W20 SAE grade oil can reduce the fuel consumption around 1.7% compared with a 5W30 SAE grade as it can be seen in table 4.5.

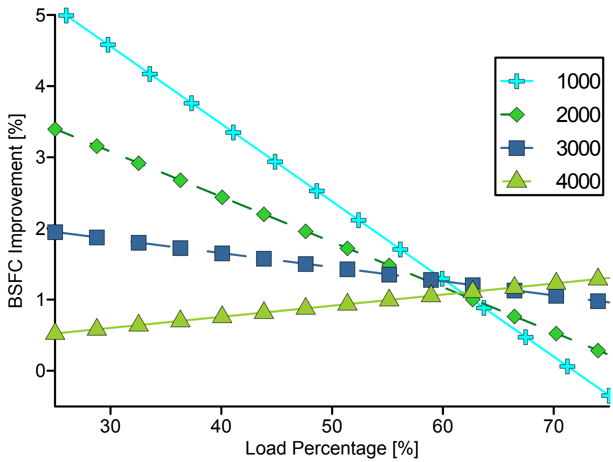

Figure 4.5: BSFC Improvement vs. engine load using 5W30 engine oil as baseline for different engine speeds.

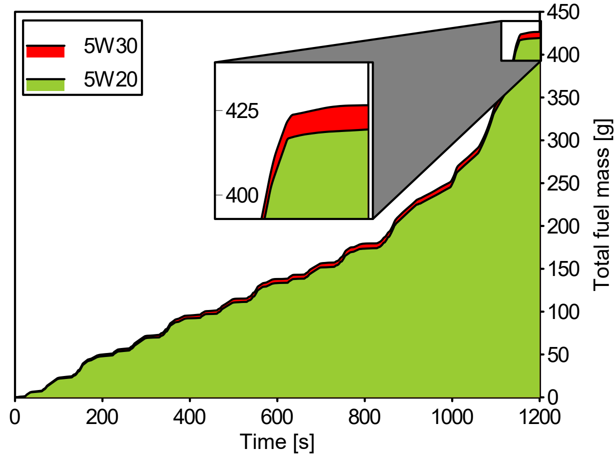

Figure 4.6: Accrued fuel consumption during the NEDC cycle for oils 5W30 and 5W20.

In accordance, figure 4.6 shows the accumulate fuel consumption for each oil during the NEDC cycle period. It can be clearly seen that a major portion of fuel consumption has taken place during the last 400 s of the cycle when the EUDC takes place. However during this cycle, the decrease of fuel consumption is the lowest compared to the other sub-cycles as can be seen in table 4.6.

It is noticeable that fuel consumption reduction takes place mainly in the UDC cycles and then the improvement tends to decline, especially during the EUDC.

Oil	Repetitions	Average F.C. [g]	Std. deviation	F.C. decrease [%]
5W30	4	426.55	5.61	
5W20	4	419.38	9.68	1.67

Table 4.5: NEDC fuel consumption test results.

Sub-Cycle	5W30 [g]	5W20 [g]	Difference [%]
UDC1	49.99	48.50	2.98
UDC2	45.57	44.18	3.0
UDC3	42.54	41.50	2.45
UDC4	41.34	40.42	2.23
EUDC	247.11	245.83	0.52

Table 4.6: Fuel consumption differences in percentage for the NEDC sub-cycles.

It has to be taken into account that NEDC cycle simulates the so called cold start (between 20° C and 30° C), this implies that the most viscous lubricant will give more resistance leading to higher fuel consumption values. In addition, it has to be stated that other engine variables increase the effect of higher fuel consumption while the engine is reaching the optimal operational temperatures. An approximation of oil's temperature trend (measured at the engine sump) during the NEDC performance can be seen in figure 4.7, where the instant temperature for every UDC cycle is plotted both for oil A and oil B. It can be clearly seen that oil temperature increase is slightly minor for 5W20 engine oil which could be interpreted as an indicator of less friction when this less viscous oil is used.

4.2.6 Engine transient cycles results' discussion

Temperature effect over fuel consumption of LVEO

Additional analysis on NEDC average fuel consumption data confirms the correlation between the oil temperature and fuel consumption. In figure 4.8 a Box-Whisker diagram show the trend of fuel consumption during the NEDC cycle in a clear way. In order to relate these variables better, the average temperature and the fuel consumption average of UDC 1 were taken as a baseline to calculate the relative increment or decrease of these variables during the subsequent UDC cycles. These differentials exhibit a good linear correlation $R^2 = 96.13$ as can be seen in figure 4.9.

This results are similar to those reported by Sandoval *et.al.* which studied the effect of oil temperature on the different components of engine friction. The

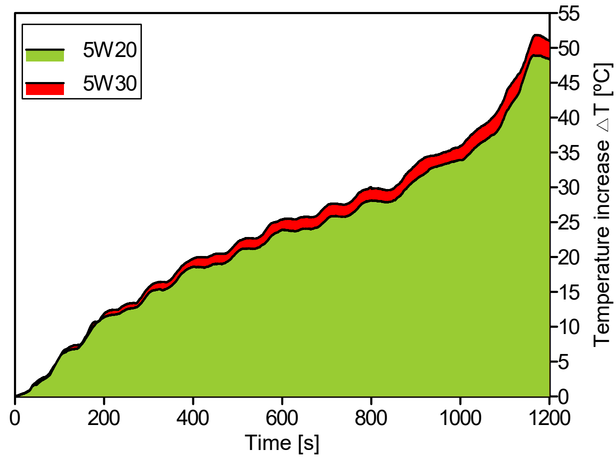

Figure 4.7: Oil temperatures during the NEDC cycles during.

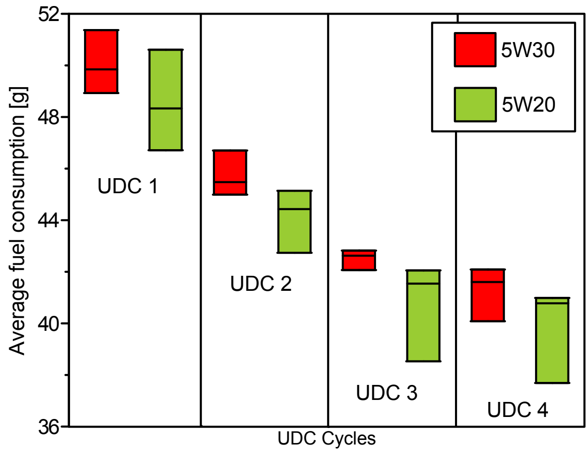

Figure 4.8: Box-Whiskers diagrams for each of the UDC cycles by oil type.

studies showed that friction power loss at engine start up is around two times higher than the warmed-up engine friction power loss. The comparisson was made on a model using 20° C as start-up temperature and 90° C as operative temperature[1]. Another clear example of this was shown by Macek *et.al.* using a Skoda engine at 2500 min^{-1} WOT, the FMEP was reduced from 140 kPa to 85 kPa increasing the oil temperature from 30° C to 80° C[12]. One study presented by Singh *et.al.* showed how the BSFC of a single point of the UDC cycle could be improved just by warming up the engine from 40° C to 80° C by 18%[13]. Going one step further, Usman *et.al.* have calculated the FMEP of the compression ring

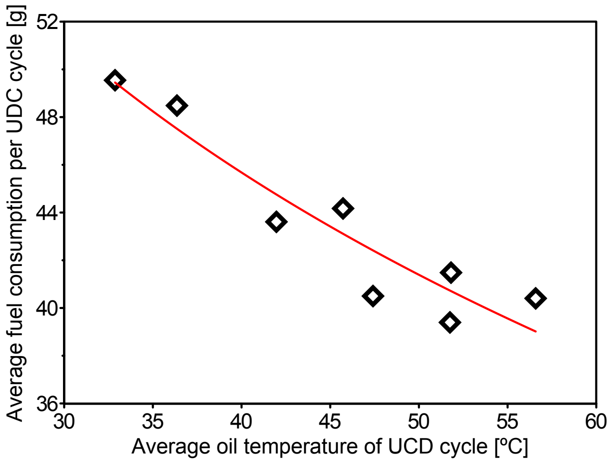

Figure 4.9: Correlation between oil temperature and average fuel consumption for the UDC cycles during NEDC cycles.

for cold (20° C) and fully warmed-up (90° C) conditions for several engine oil formulations. The FMEP of the piston ring - liner interface increased by 4.70 to 13.49 times depending on oil formulation [14].

Fuel consumption differences between urban cycles and extra-urban cycles

Taking into account the results from the stationary fired test, and comparing them to the NEDC tests results it can be clearly seen that engine loads and speed of the typical urban driving (see figure 4.4 and figure 4.11) are those where the stationary fired test reported the greatest fuel consumption reduction when the low viscosity lubricants were used, being this reduction as high as 4% when 15W40 and 5W30 are compared in low speeds and low loads.

Another good concordance between stationary fired and the NEDC tests results can be observed in the figure 4.10, where the cumulative fuel consumption difference in grams between 5W30 cycles and 5W20 cycles is plotted for the complete duration of the NEDC. Alongside the engine speed over the NEDC cycle is plotted as well in order to make visible the relation between the engine speed with the effect of oil viscosity on fuel consumption.

As it can be observed, the cumulative difference increases in periods where engine speed and torque (not plotted) are low, mainly during the urban segment of the cycle, whereas during the extraurban segment this cumulative difference tends to diminish (being this an indicator of higher fuel consumption with the

5W20 oil than with the 5W30 oil), especially when medium to high loads are being reached.

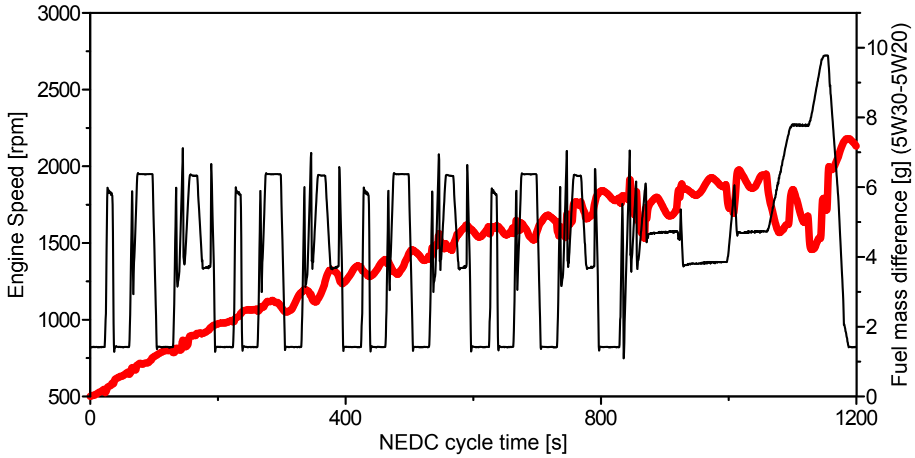

Figure 4.10: Fuel mass accrued difference between oils 5W30 and 5W20 and engine speed during NEDC cycle.

Similar results have been shown by Calwell *et.al.* both by NEDC transient cycles and in real world conditions, using a SAE 10W40 as baseline and a SAE 5W30 as candidate. In the former, the fuel consumption differences during the UDC 1 and 2 cycles was around 7%, 4% during the UDC 3 and 4 cycles and 2% during EUDC cycle. As the results presented in section 4.2.5, the difference between urban and extra-urban operation is confounded with the oil engine temperature effect. However in the latter, fuel consumption in real world operation were affected only by the duty cycle: during urban operation, the difference was around 5% and in *autobahn* operation, the difference was 2.2%. The main reason for this behavior is the fact that under urban operation, engine normally does not reach high load operation points, not being this the case when extra urban and highway conditions are reached. These aspects are relevant when the following European driving patterns are taken into account[15]:

- The average trips during the day are around 2.5.
- Nearly 40 % of the trips take place before noon.
- Average commuting distance is around 18 km.
- Average duration of trips takes between 20 and 30 minutes.

- Car is used around 1.5 hours per day. The active parking[I] is around 6.5 hours per day.
- The inactive parking is around 16 hours per day.

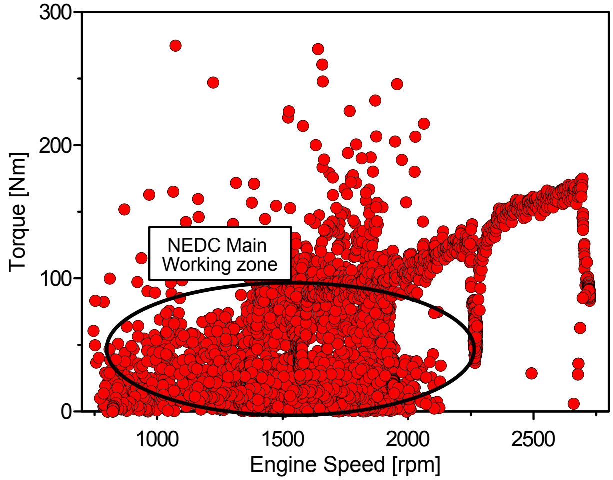

Figure 4.11: Engine speed and torque measured during NEDC cycle.

[I]'Active and inactive parking refers to the time the car is parked between trips during the day and the time the car spends parked until the next day use respectively.'

4.3 LVEO effect over fuel consumption during the fleet test

As buses fuel consumption is the magnitude that can be directly quantified with the proposed methodology, the results are going to be referred in fuel consumption units (l/100 km for Diesel buses and m^3/100 km for CNG buses).

4.3.1 Variables included in the ANOVA analysis

In order to assess the effect of LVEO over the fuel consumption of the buses, the complete data set was subject to Analysis of Variance (ANOVA) technique to quantify the significance of the experimental variables considered. From the facts exposed in chapter 3, it is clear that the experiment was performed under severe limitations: *i.e.*, not all buses used all the oils involved in the test and not all buses worked in every possible route. Taking into account this situation the ANOVA analysis was performed by bus model, blocking the variability in fuel consumption due differences among buses model and routes. These sort of inconvenience could not be handled due to fleet operation requirements. Variables used to perform the ANOVA analysis were:

Daily temperature

This factor makes reference to the ambient temperature during the test registered in Celcius. This value was introduced as a factor for two reasons: firstly, the inverse relationship between temperature and oil viscosity, and secondly, the use of air conditioning system (AC) during summer suppose an abnormal power consumption compared with other year seasons. Even though both assumptions are correct, it is clear that the oil temperature during engine operation should tend to be the same once transient operation has finished, hence, air conditioning will have more specific weight and correlation with the vehicle fuel consumption.

The seasonal variation of fuel consumption found for all the buses corresponds to the use of the air conditioning (AC) system during the hottest months of the year. This seasonality has been described by means of polynomial regressions of 5^{th} level. This approximation is applicable since the intention is to describe an historical behavior and not to extrapolate future fuel consumption values [16]. As it can be seen in figure 4.12 the fuel consumption presented a strong seasonality during the test period.

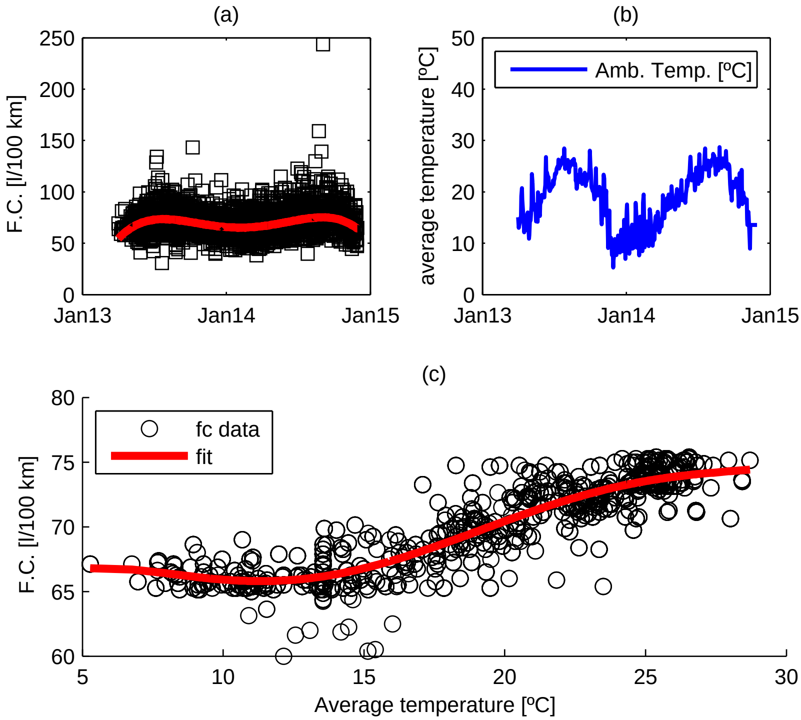

Figure 4.12: (a) Fuel consumption for the Diesel I buses with a polynomial fit model (5^{th} degree), (b) average daily temperature variation across the test period, it is remarkably similar to the fuel consumption trend and, (c) scatter diagram with the daily temperature and the fuel consumption fit.

Oil mileage

It is well known that depending on oil formulation and the engine operating parameters, the values of viscosity could change over the ODI. If viscosity tends to be higher at the end of ODI fuel consumption would increase given the extra effort that moving parts must do to overcome lubricants inner friction, if the opposite case happens, that is, viscosity decreases over the ODI, less power, hence less fuel consumption would be required to reach one operation point.

Fuel consumption seasonality

Transportation demand varies across the year (*e.g.* some places like the beaches often have more visitors during summer than winter, and routes passing near Universities or school would present more demand over class periods). These changes

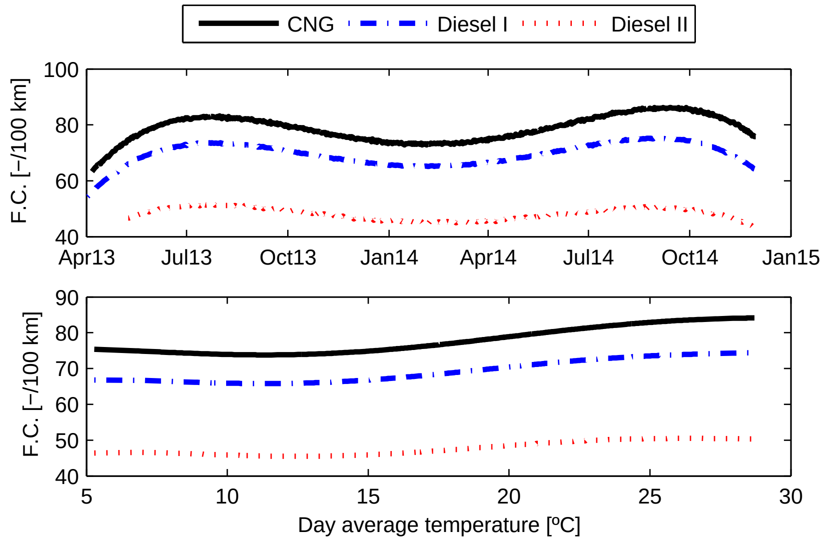

Figure 4.13: Fuel consumption behavior for the three bus models and its relation with the average temperature of the sampling day.

would represent a significant variation in fuel consumption given the load differences as it can be seen in figure 4.14 where is clear that august presents a slight decline in fuel consumption even when daily mean temperature has similar values than the adjacent months. This corresponds to the holidays period in Valencia when a large number of inhabitants leave the city and it is reflected in buses passengers and fuel consumption.

Engine oil viscosity

The main factor to be considered during this test. It has been considered as a nominal variable with two defined levels depending on the type of oil used; candidate (LVEO) or baseline oil.

Differential oil viscosity

In similar fashion as engine oil viscosity, differential oil has been considered as a nominal value with two levels depending on oil viscosity. However, this factor will only be included in the ANOVA analysis for Diesel I buses.

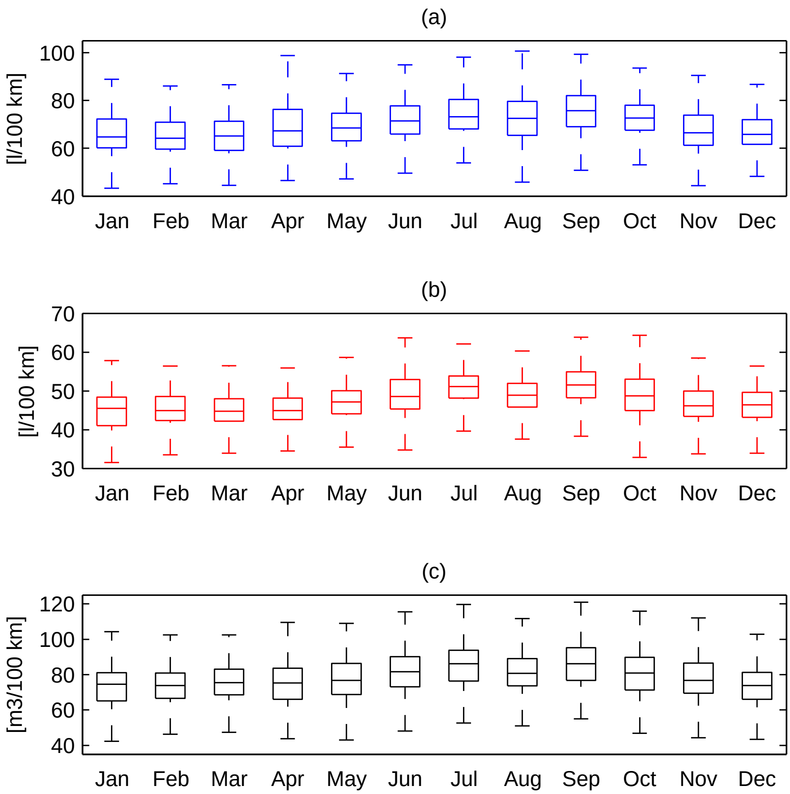

Figure 4.14: Boxplot diagrams of fuel consumption by month of the year for (a) Diesel I buses, (b) Diesel II buses and (c) CNG buses.

Engine and differential oil interaction

This factor evaluates the interaction between engine and differential oil viscosity regarding vehicle fuel consumption. Since differential oil will be analyzed only for Diesel I buses, this interaction will only be suitable for this buses model.

4.3.2 Variables controlled during the fleet test

Some variables which could influence the fuel consumption of the vehicles involved in the test have been controlled during the test period in order to assure they did not influence the final result.

Fuels characterization

Among other factors, fuel heat value Q_{HV} is one of the most important variables that must be taken into account when fuel consumption measurements. These values among other properties were measured for different fuel samples during the test period and they can be seen on table 4.7.

Property	Method	S1	S2	S3	Average	S.D.
Density@15° C [kg/m^3]	ASTM D1298	838.8	835.9	838.3	837.7	1.55
Water content [ppm]	ASTM D6304	48.89	45.62	35.03	43.18	7.24
kV@40° C [cSt]	ASTM D445	3.181	2.967	3.028	3.059	0.11
Q_{HV} [MJ/kg]	ASTM D240	45.63	46.01	46.01	45.88	0.21
Biodiesel content [%]	ASTM E2412	6.47	4.22	5.45	5.38	1.12
Flash point [° C]	ASTM D93	77	74	69	73.3	4.04
Cetane number	ASTM D4737	50.05	50.30	51.26	50.54	0.64

Table 4.7: Fuel samples characterization during the fleet test period

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Tires pressure and general condition

Several studies have shown the influence of tire pressure and tire condition effect over vehicles fuel consumption, during the test this variable was not taken into account since the fleet technical service fill the tires to the adequate operation value of 9 bar before the buses leave the garage on a daily basis.

Driver effect

It has been widely reported how the driving style (DS) is directly related with a vehicle fuel consumption. Different drivers tend to exhibit different speed profiles, acceleration and deceleration rates which are reflected on different fuel consumption. It has been estimated that driving style could induce variability up to 5% in CO_2 tailpipe emissions[17]. Other studies with passenger cars claims differences as high as 20%[18]. In the case of urban buses, Ma *et.al.* found a difference of 12% in urban conditions, 19% in extra-urban conditions and 16.9% in high way conditions. However, the buses used in during the test had manual transmissions, which have been in the center of debate about their benefits in fuel consumption terms against automatic transmissions. Either way, what has been proven is that automatic transmissions dampers the differences of different drivers acceleration patters where most of the differences in fuel consumption can be found during urban operation[19].

As the buses models involved in the fleet test are provided with automatic transmissions, the only precaution the fleet manager did was to assign a group of drivers to a single type of bus model, making them shifting among different vehicles regardless of the type of oil they were filled with. In such a way the variability among drives should be randomized per bus model. In addition, the driver were unaware of the ongoing test in order to prevent unusual driving styles and patterns.

4.3.3 ANOVA analysis results

Table 4.8 summarizes the effect of LVEO on fuel consumption for each bus model after the vehicles completed a 60000 km mileage. The table also indicates if the resulting fuel consumption benefits are either statistically significant or not, with a confidence level of 95%. In the same way, the limits for confidence interval are included in the table. It has to be noted that for Diesel I buses the effect of LVEO on differential over fuel consumption was calculated as well.

	Ref	Cand	Ref-Cand [%]	Ref-Cand [fuel/100 km]	+/- Limits
Diesel I	15W40	5W30	1.83	1.3	0.98
	80W90	75W90	0.58 N.S.	0.4	0.91
Diesel II	10W40 Low SAPS	5W30	0.98 N.S.	0.46	0.53
CNG	10W40 Low SAPS	5W30 Low SAPS	3.7	3.27	0.99

Table 4.8: LVEO fuel consumption benefits per bus model. Values with confidence level of 95%. N.S indicates the absence of statistically significant differences.

However, to address the results and particularities for each bus model and their respective test oils, the results are analyzed separately as follows.

Diesel I

After completing two ODI and performing the ANOVA analysis, it was proven that engine oil viscosity had an effect over Diesel I buses fuel consumption: the buses using 15W40 SAE grade engine oil showed a fuel consumption of 70.9 l/100 km which represents a difference of 1.3% with respect to buses using SAE 5W30 which consumed an average of 69.69 l/100 km as it can be seen on figure 4.15. This difference is statistically significant with 95% of confidence level. In the same way the effect of differential oil viscosity was proven through ANOVA(see table 4.9 and figure 4.16). As in the case of engine oil, the less viscous oil lead to lower fuel consumption (70.54 l/100 km for SAE 80W90 in contrast to 70.13 l/100 km for SAE 75W90), yet this difference was not statistically significant so even when results seem to be logic it is not possible to completely claim favorable fuel consumption results for LVEO.

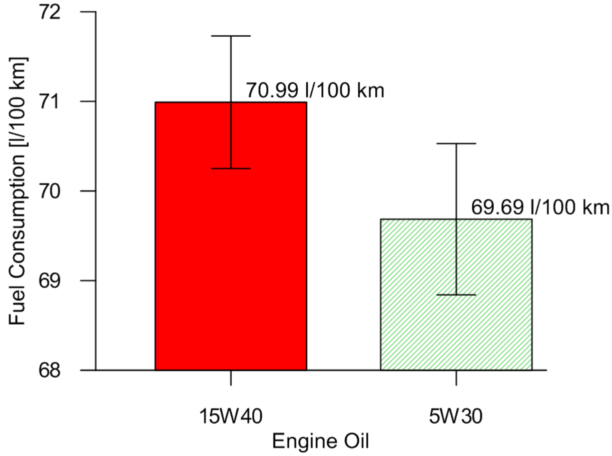

Figure 4.15: Fuel consumption differences for Diesel I buses.

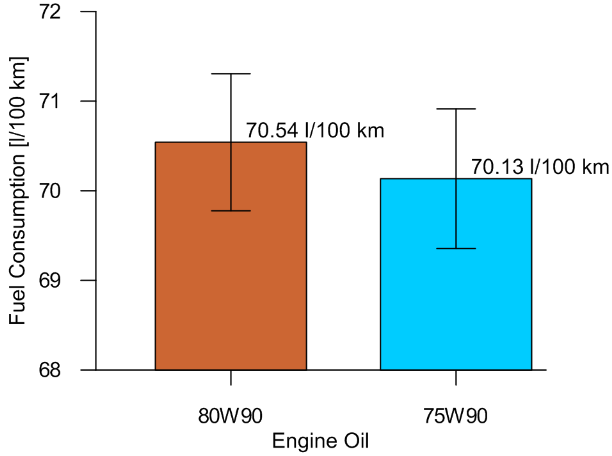

Figure 4.16: Fuel consumption differences for Diesel I buses regarding Low viscosity differential oils.

Factor	SS	DoF	P-Value
Daily Temp [° C]	3662.38	1	0.0
Oil mileage [km]	1895.72	1	0.0004
Engine Oil	1038.29	1	0.0092
Month	4850.19	12	0.0002
Differential Oil	117.48	1	0.3812
Interaction (Engine-Differential)	1620.72	11	0.85

Table 4.9: ANOVA results for Diesel I buses.

Diesel II

From ANOVA results (see table 4.10), fuel consumption difference between the buses using reference SAE 10W40 and the buses using candidate SAE 5W30 was 0.98% as it can be seen on figure 4.17. However, these differences could not be proven as statistically significant.

Factor	SS	DoF	P-Value
Daily Temperatue [° C]	3662.38	1	0.0003
Oil mileage [km]	1895.72	1	0.0447
Engine Oil	1038.29	1	0.0814
Month	4850.19	11	0.0000

Table 4.10: ANOVA results for Diesel II buses.

Compressed Natural Gas (CNG)

After carrying out the 60000 km mileage, the buses that used SAE 5W30 Low SAPS gave a fuel consumption of 85.1 Nm^3/100 km, considerably lower than the 88.37 Nm^3/100 km of fuel consumption given by the buses using SAE 10W40 Low SAPS. For CNG buses this difference of 3.7% is statistically significant, demonstrating again the benefits of using LVEO in terms of fuel consumption. The complete results can be seen on the table 4.11 and figure 4.18.

4.3.4 Discussion on fleet test results

Mean Effective Pressure (mep), thermal load, and LVEO effects over fuel consumption

From the previous results is easy to note the variation of the fuel consumption benefit of LVEO among different bus models. Being engine friction losses the

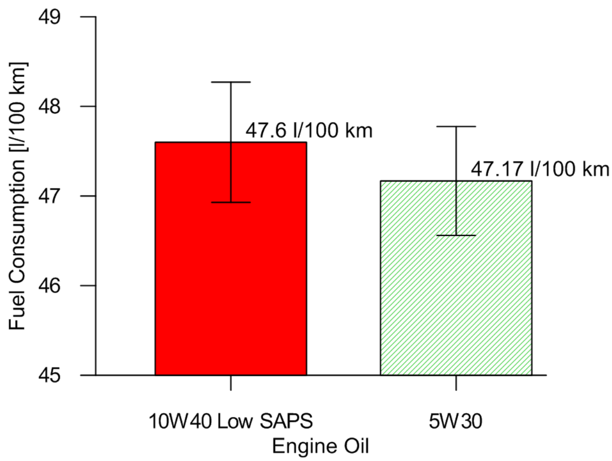

Figure 4.17: Fuel consumption differences for Diesel II buses.

Factor	SS	DoF	P-Value
Daily Temp [° C]	670.4	1	0.048
Oil mileage [km]	13561.0	1	0.006
Engine Oil	16733.1	1	0.004
Route	375386.0	1	0.000
Month	4850.19	11	0.0125

Table 4.11: ANOVA results for CNG buses.

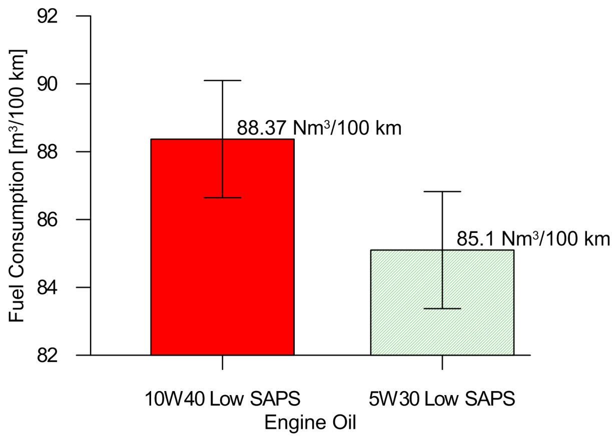

Figure 4.18: Fuel consumption differences for CNG buses.

main parameter affected when LVEO are used, it would be desirable to establish a correlation of those benefits depending on the type of engine. Examining at engine thermal loads but specially the break mean effective pressure in table 3.4 and the fuel consumption differences in table 4.8, one possible approximation would be like the one plotted in figure 4.19

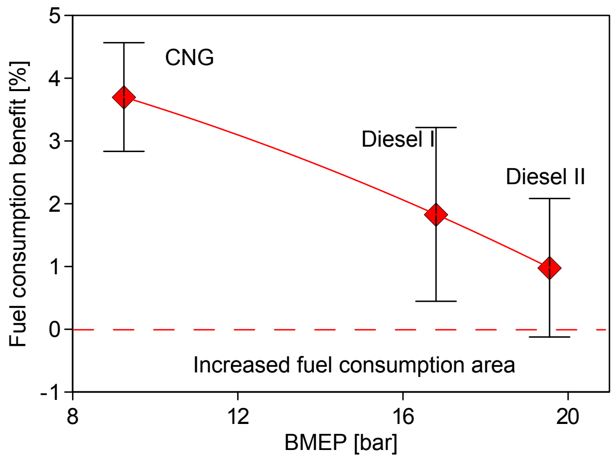

Figure 4.19: Break mean effective pressure (BMEP) effects of the three engines used during the test over the fuel consumption benefit of using LVEO.

HTHS viscosity behavior of baseline and LVEO during the test

The influence of the engine oil viscosity over the friction coefficient of tribo-contacts working under hydrodynamic regimes has been linked to a specific rheological property: the High Temperature High Shear viscosity (HTHS). For this mean reason only HTHS viscosity behavior was studied in detail to analyze fuel consumption differences between the buses using baseline oils and LVEO. As described in section 3.4.5 an extensive sampling plan to control the HTHS behavior of the different oils used during the fuel consumption test was made. The HTHS viscosity was measured under the ASTM D5481 as shown in section 3.6.2. The trends followed by the oils can be seen in figure 4.20.

In general, as can be seen in table 4.12 the most noticeable trend is the gain of HTHS viscosity by both oils used in the CNG model. This supports the findings showed previously in the table 4.11, where the oil mileage displayed a *p-value* greater than 0.05. On the other hand, engine oil used in Diesel models exhibit more discrete changes in HTHS values being this especially true for the candidate oil, which HTHS viscosity increment is almost imperceptible. However, it is clear

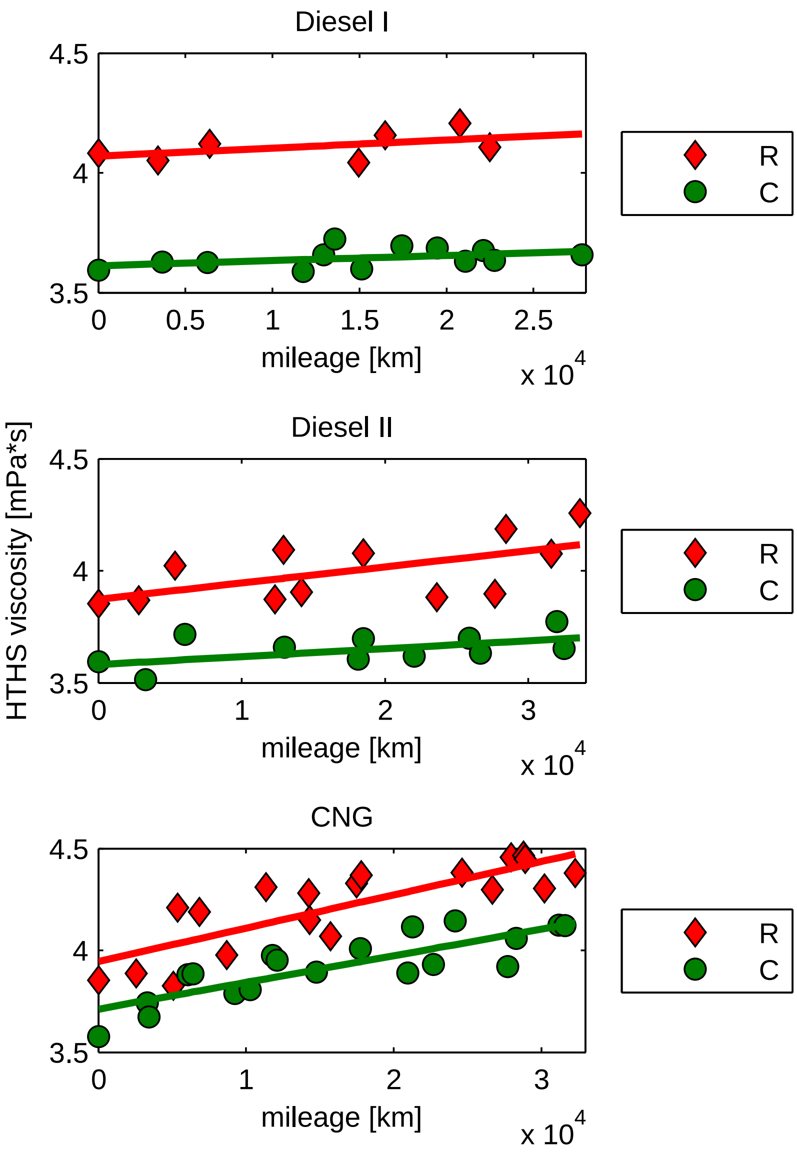

Figure 4.20: HTHS dynamic viscosity behavior during the ODI for the different bus models.

that for Diesel I and CNG buses which exhibited fuel consumption benefits when using LVEO, the HTHS viscosity gap between the candidate and the baseline oil lasted during the test. This means that the possible increment in fuel consumption due the oil viscosity increase was applicable to both oils in such a way it did not suppose a fuel consumption benefit variation throughout the test. In the case of Diesel II buses there was a significantly different behavior; baseline oil showed a steepest HTHS increase than the one exhibited by the LVEO. Anyhow, this differences in HTHS viscosity performance should have increased the differences in fuel consumption between the oils as the test continued and that was not the case as exposed in section 4.3.3.

Bus model	Oil	HTHS$_{fresh}$ [cP]	HTHS$_{ODI}$ [cP]	variation [%]
Diesel I	5W30	3.594	3.672	2.2
	15W40	4.082	4.163	2.0
Diesel II	5W30	3.594	3.701	2.9
	10W40 Low Saps	3.853	4.116	6.8
CNG	5W30 Low Saps	3.577	4.135	15.6
	10W40 Low Saps	3.853	4.473	16.1

Table 4.12: HTHS viscosity variation for the oils involved in the test after one ODI.

Differential oil and engine oil interaction effect over fuel consumption

For this type of analysis sometimes it is important to find if there is any level of interaction between certain variables. In this case, it was important to know how engine LVEO oils and differential LVEO oils interact, it means, if the reduction of fuel consumption presented by engine LVEO was maintained, decreased or increased when a differential LVEO was used. To figure out how was this interaction and if it has an impact on fuel consumption, it was included in the model, resulting into a positive but not statistically significant interaction between the two levels of the oils as it can be seen in figure 4.21, where despite the lack of significance, it is clear that engine LVEO combined with differential LVEO give the lowest fuel consumption value in comparison with other combinations. As expected the highest fuel consumption occurs if both oils correspond to reference viscosity. The complete values of all combinations can be seen on table 4.13.

Route effects over fuel consumption for CNG buses

As shown in table 4.11, route effect on fuel consumption had statistical significance ($p-value$<0.05). Also in table 3.6, the 10 and 62 routes characteristics

Engine Oil	Differential Oil	Fuel consumption [l/100 km]
5W30	75W90	62.52
5W30	80W90	69.84
15W40	75W90	70.74
15W40	80W90	71.23

Table 4.13: Fuel consumption values for the interactions between Engine and Differential oils at two levels.

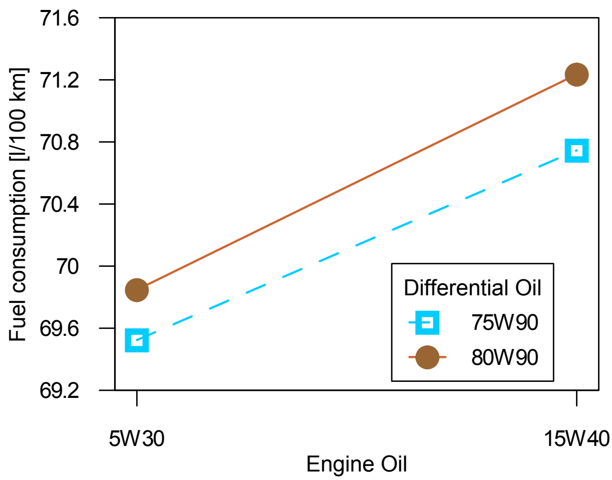

Figure 4.21: Combined effects of engine and differential low viscosity oils over fuel consumption taken from the ANOVA results of Diesel I buses.

were depicted. I has to be remembered that the main differences between these routes is the average speed since route 62 has fewer stops and has a major length of which more than a third runs over a highway. The fuel consumption results can be seen in figure 4.22.

CO_2 emissions equivalence of fuel consumption benefits found

The fuel consumption reductions achieved by means of LVEO use presented in section 4.3.3 can be easily translated into CO_2 emissions reduction terms, since the latter is a direct product of fuel combustion in the engine. Taking into account only the bus models which presented statistically significant differences, the next step was simply to find the CO_2 emission reduction benefits for the test period of 60000 km. The procedure used to calculate the equivalence involves knowing the elementary composition of fuel to find the amount of carbon contained,

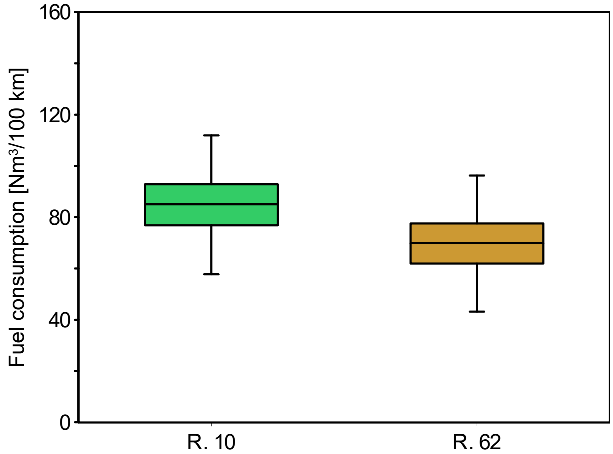

Figure 4.22: Route effects over the fuel consumption of CNG buses.

then supposing a stoichiometric combustion, a carbon balance is made in order to estimate the CO_2 produced in the reaction.

As B5 and CNG were the two fuels used during the test, the elementary composition of these fuels would be required to perform the calculation. The following compositions were supposed:

- Diesel: $C_{12}H_{22}$
- Biodiesel B100: $C_{19}H_{35}O_2$
- CNG: CH_4

The combustion reactions of these fuels are
Diesel:

$$2C_{12}H_{22} + 35O_2 -> 24CO_2 + 22H_2O \tag{4.1}$$

B100:

$$C_{19}H_{35}O_2 + 26.75O_2 -> 19CO_2 + 17.5H_2O \tag{4.2}$$

CNG:

$$CH_4 + 2O_2 -> CO_2 + 2H_2O \tag{4.3}$$

If carbon molar mass is 12 g/mol, oxygen is 16 g/mol and Hydrogen is 1 g/mol, the molar mass for each fuel and combustion product are shown in table 4.14 and the consequent CO_2 emissions per g of fuel are those shown in table 4.15.

Compound	Molar mass [g/mol]
CO_2	44
CH_4	16
$C_{12}H_{22}$	166
$C_{19}H_{35}O_2$	295

Table 4.14: Molar mass of the different compounds involved in fossil fuels' combustion.

Fuel	k (CO_2 /fuel [g/g])
Diesel	3.18
B100	2.83
CNG	2.75

Table 4.15: Grams of CO_2 emissions per gram of fuel.

Given the fuels densities:

Fuel	Density
Diesel	837.7 g/l
B100	880 g/l
CNG	788 g/Nm^3

Table 4.16: Density values for different fuels.

With the given values the equivalent CO_2 emissions for a given fuel consumption could be calculated by equation 4.4.

$$CO_2[\tfrac{g}{km}] = \text{F.C. } [\tfrac{l}{100km}] \text{ x } \rho_{fuel} \; [\tfrac{g}{l}] \text{ x k x } [\tfrac{1}{100}] \tag{4.4}$$

For Diesel I buses and CNG buses which obtained statistically significant differences, the CO_2 emissions reductions per kilometer were 34.29 g/km and 70.14 g/km respectively. It is worth remembering that each of the test buses covered an average 60000 km mileage during the test hence, the total amount of CO_2 emissions reduction per Diesel I and CNG bus using LVEO is easy to plot, being this values 2.05 CO_2 Tons and 4.2 CO_2 Tons respectively.

4.3.5 Costs and benefits of integrating LVEO into urban buses fleets

Once the fuel consumption effects of using LVEO in an urban buses fleet were determined, a cost and benefits balance was made in order to be used as a guide to fleet owners and managers to make a decision to integrate these type of oils to the fleet.

Contrary to some cost-benefits studies of emerging technologies that normally underestimates the problems which emerge as the technology matures to commercial application, the use of commercial LVEO and the implied costs come straightforward.

To calculate the costs of using LVEO in a fleet the following factors were included:

- *Price of LVEO:* Normally LVEO are formulated with superior base stocks of groups III and IV (as seen in table 3.5). As explained before, the obtaining these base stock entails different refining processes to offer better performance owing to their consistent molecular structure and purity.
- *Oil Drain Interval:* This parameter was included considering the possibility change the ODI if wear problems would have emerged during the test. However, as demonstrated in Macián *et.al.* this was not the case and buses using LVEO could finish the ODI uneventfully[20].
- *Variations in oil consumption due lower viscosities:* oils with lower viscosities tend to boost "blow-by" phenomena resulting this in higher oil transportation from crankcase to the combustion chamber, hence oil consumption.

All this items are computed with the fuel consumption benefits of using the LVEO and an estimate of the fuel price based on the trends prices have shown during the last months.

Diesel I	15W40	5W30
Sump volume [l]	30	
Annual mileage [km/year]	45000	
ODI [km]	30000	
Annual Oil changes [-/year]	1.5	
Oil consumption [l/1000 km]	0.35	0.32
Annual oil consumption [l/year]	60.75	59.4
Oil aprox. cost [€ /l]	1	2
Fuel consumption [l/100 km]	75	73.5
LVEO F.C. savings [%]	1.98	
Annual fuel consumption [l/year]	33750	33075
Fuel price [€ /l]	1.1	
Fuel costs [€ /year]	37125	36382.5
Annual savings by fuel [€ /year]	742.5	
LVEO annually extra cost [€ /year]	-58.05	
Total annual savings [€ /year]	684.45	
Annual savings by bus model [€ /year]	6160	

Table 4.17: Cost-benefit studies for Diesel I buses. The last row indicates the total savings if all buses of Diesel I model had used LVEO.

4.4 Friction coefficient measurements in different tribometers

4.4.1 Cameron-Plint machine TE77

As stated in section 3.5.1 two different tests were performed with the Cameron-Plint machine: one screening test varying the oil, load and stroke frequency (average speed)using only the Oil Control Ring (OCR) and another with the three piston rings, compression, scrapper and OCR working at a load similar to those found during the piston reverse stroke and varying the stroke frequency. To plot the resulting Stribeck curves from the test, the Sommerfeld number was used as reference for the different friction coefficient values. The Sommerfeld number or Hersey parameter is given by the expression 3.2 where η is the oil dynamic viscosity, U is the relative speed and P is the contact pressure.

Given the fact that friction coefficient measurements were made at low temperatures in order to avoid the consequent engine oil viscosity drop, the actual viscosity value was calculated with the mean temperature at each test point and the Vogel equation 4.5 which is the most used approximation in engineering cal-

CNG	10W40 Low Saps	5W30 Low Saps
Sump volume [l]	33	
Annual mileage [km/year]	45000	
ODI [km]	30000	
Annual Oil changes [-/year]	1.5	
Oil consumption [l/1000 km]	0.675	0.89
Annual oil consumption [l/year]	79.875	89.55
Oil aprox. cost [€ /l]	1	2
Fuel consumption [Nm^3/100 km]	80	77.04
LVEO F.C. savings [%]	3.7	
Annual fuel consumption [l/year]	36000	34668
Fuel price [€ /Nm^3]	0.819	
Fuel costs [€ /year]	29484	28393.09
Annual savings by fuel [€ /year]	1090.91	
LVEO annually extra cost [€ /year]	-99.23	
Total annual savings [€ /year]	991.68	
Annual savings by bus model [€ /year]	19833.5	

Table 4.18: Cost-benefit studies for CNG buses. The last row indicates the total savings if all buses of CNG model had used LVEO.

culations[21]. The equation coefficients were found based on previous viscosity measurements at 40° C 100° C and 150° C following ASTM D5481[22, 23].

$$\eta = ae^{\frac{b}{(T-c)}} \tag{4.5}$$

Screening test

The results of the screening test can be seen in table 4.19 and in figure 4.23. The ANOVA shows that the three main effects under study have a significant effect over the friction coefficient since the *p-value* is less than 0.05.

From the results of ANOVA it is possible to state that the use of SAE 5W30 engine oil instead of SAE 10W40 in this tribo-contact reduced the friction coefficient by 4.24%. In the same way, the increase of normal load had the greatest impact over the friction coefficient: an increase from 20N to 150N produced a 33.51% decrease of the friction coefficient. Lastly, the variation of entrainment speed showed to have a significant effect on friction coefficient as well having a difference of 15.13% between the slowest and the fastest entrainment speed. It has to be stated that the role of load in the Cameron Plint test seems to be predominant (as can be seen in table 4.19). However, compared to the real situation

Main effects	SS	FD	p-value
A: Oil	0.000076	1	0.0385
B: Load	0.004365	2	0.0001
C: Frequency	0.000729	2	0.0019
Interactions			
CB	0.000058	4	0.2949
AC	0.000009	2	0.6162
BA	0.000026	2	0.3134
Residues	0.000033	4	-
Total	0.005295	17	-

Table 4.19: ANOVA results for the screening test

in the engine, the contact pressure values of the rings (directly related to normal forces in the Cameron Plint) and the entrainment speed exhibit opposite scenarios: 20N to 150N over the contact area represent nominal contact pressures of 3.125 N/mm_2 and 23.44 N/mm_2 respectively. Despite the fact that this screening test was done over the OCR, these contact pressure values could be present in compression and scraper rings during engine operation. *Per contra*, the relative speeds reached in the Cameron Plint are distant from the actual engine speeds, and from the engine point of view all the three values used as input in the rig are relatively low and close to the speeds found at top and bottom dead centers (TDC and BDC).

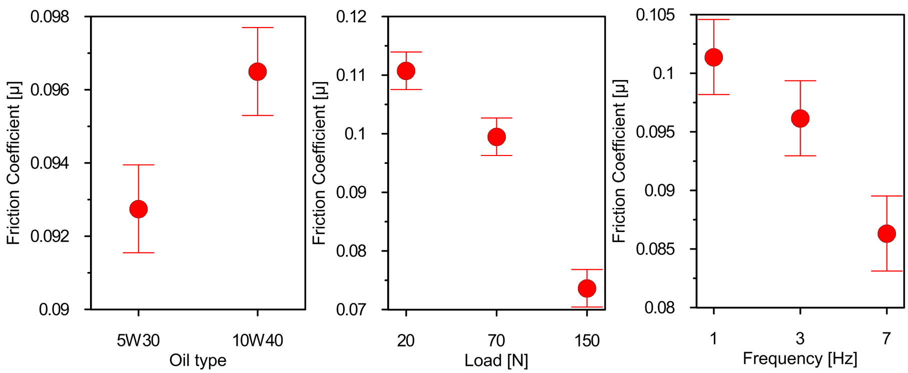

Figure 4.23: ANOVA results for the Oil Control Ring.

Piston rings under reverse stroke conditions

As it was described in section 3.5.1, the aim of this test was to mimic the loading conditions of the rings at reverse conditions (that is some crank angle grades before and after BDC and TDC in absence of combustion). Figure 4.24 describes the friction coefficient behavior against the Sommerfeld number for the three piston rings and the two oils. It should be noticed that the scale on x-axis decrease for each piston ring, following the contact pressure values given in table 3.11 and the Sommerfeld number in equation 3.2. The higher contact pressure value of the OCR correlates precisely with the higher values of friction coefficient which decrease slightly as the speed increases (as part of the Sommerfeld number the load and viscosity are fixed for this test) in contrast with the notable decline of friction coefficient as the entrainment speed raises of the compression and scrapper ring. As a general trend the friction coefficient curves have moved towards the left. This outcome can be interpreted in two ways: in most of cases for a given Hersey number (that is the relation between lubricant viscosity, relative speed and contact pressure) the friction coefficient value declines, hence the friction force is going to decrease using the less viscous oil. On the other hand, it is not possible to say that for every fixed values of load and relative speed, the friction coefficient will drop by the use of an oil with lower viscosity. Having the Stribeck curve moved to the left the boundary and mixed regimes could be found easier if the oil layer could not hold the applied load (as at high speeds in the case of piston rings of figure 4.24).

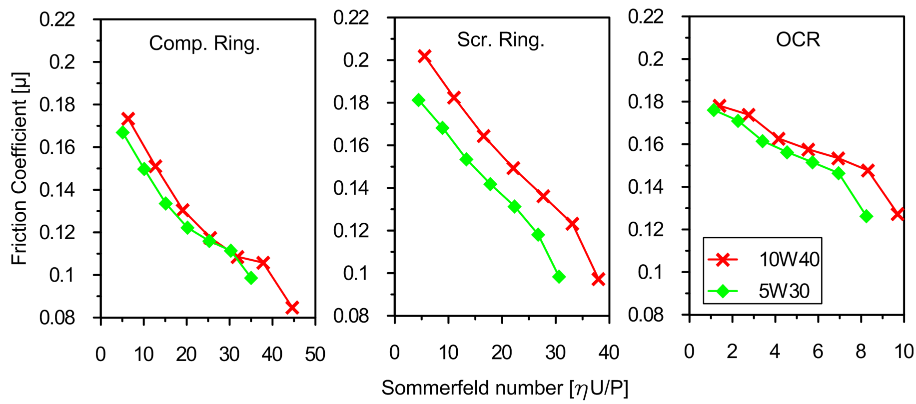

Figure 4.24: Test results for the different rings.

Factor interactions on the screening test

In addition to the fact that results on figure 4.23 are useful to determine the sensibility of friction coefficient, it is of special interest to determine how the load and the relative speed affects the low viscosity oil capacity to reduce fuel consumption. In figure 4.25, the combined effects of load and relative speed over the friction coefficient are depicted. It is clear that for every measured entrainment, speed friction coefficient drops sharply as the load increases from 20 N to 150 N. This trend is somehow unexpected if interpreted by the theory enclosed in the equation 2.12: higher loads, in this case contact pressures, should give higher friction coefficients; however, it is possible that a severe load variation as the one proposed for the screening test led to strong deformations making the contact to have independent Stribeck curves as is plotted on figure 4.27.

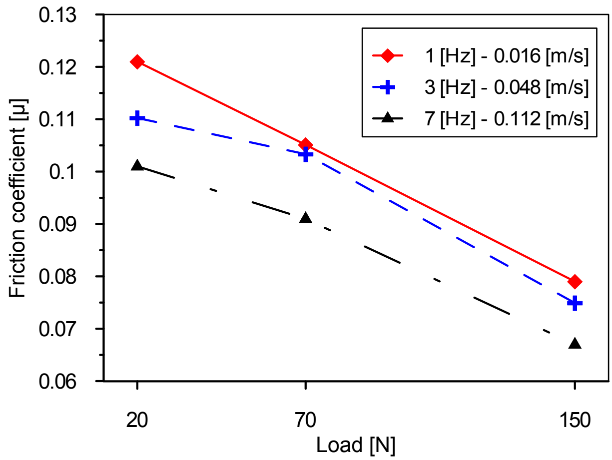

Figure 4.25: Load and frequency effects over friction coefficient of Oil Control Ring.

Figure 4.28 depicts the friction coefficient dependence on engine oil viscosity and relative speed. It is clear that for the three measured speeds, the SAE 5W30 engine oil gives a lower friction coefficient value. However, for the lowest speeds the friction coefficient has fallen marginally in contrast to the behavior at high speed that shows a substantial decrease around to 25%. This trend is somehow expected: high relative speeds favor the hydrodynamic lubrication regime precisely where the less viscous oil has a greater potential to reduce friction.

A similar situation can be seen in figure 4.26 where the combined effect of oil viscosity and load over friction coefficient are showed. As expected the less viscous oil presented lower friction coefficients for the 20 N and 70 N levels of load. However, at the highest load the friction coefficient remained stable, that is,

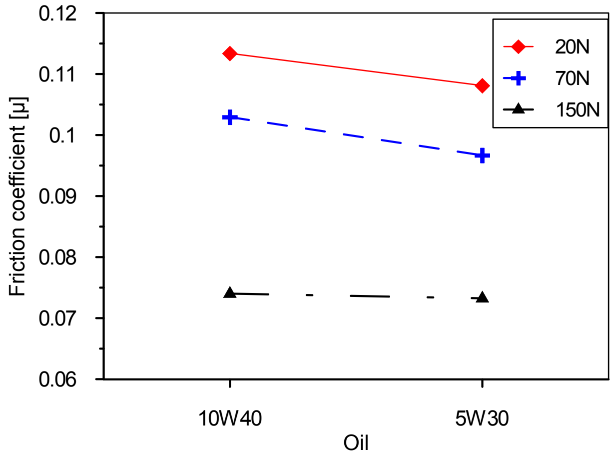

Figure 4.26: Oil and load effects over friction coefficient of Oil Control Ring.

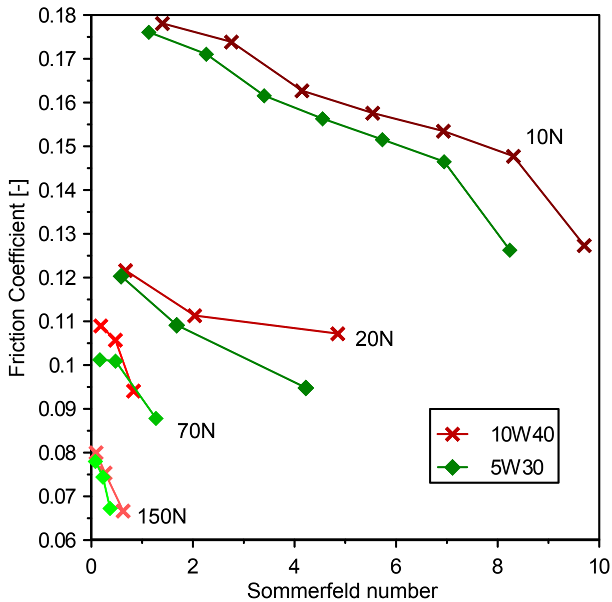

Figure 4.27: Stribeck curves of the OCR for different loads.

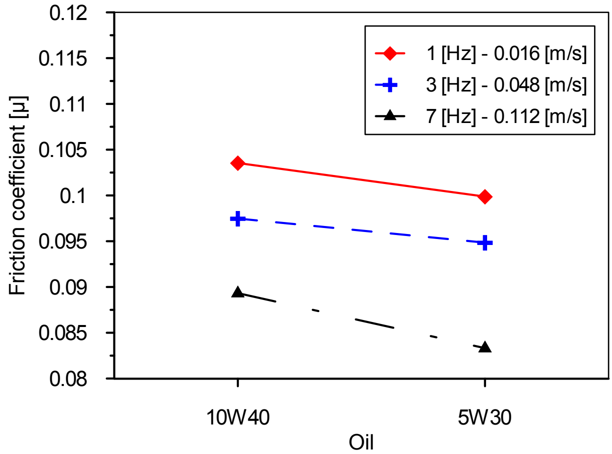

Figure 4.28: Oil and frequency effects over friction coefficient of Oil Control Ring.

the oil viscosity could not offer any upturn with the given conditions. Probably at this point the contact at the Cameron-Plint machine is working under the boundary lubrication regime and the reduction of the viscosity of the oil could lead to even higher friction coefficients. In fact, the friction coefficient at 150 N and 7 Hz is higher for the 5W30 oil as the Stribeck curve moves towards the left due the decrease of engine oil viscosity, behavior that was clear during the "reverse stroke" like test.

Friction coefficient and lubrication regime of tests

As it was observed in figure 4.24, the friction coefficient value for all the conformed contacts of liner and the correspondent piston ring showed values near 0.1 that are typically associated with mixed and boundary lubrication regimes. This behavior is especially evident in the OCR that presents high friction coefficient values for all ranges of speed probably due the greater value of nominal contact pressure. In the other hand the scraper ring, (which is the one with less nominal contact pressure) displays a clear trend towards low friction coefficients as the speed increases, typical of the mixed lubrication regime. Complementarily, it is in this ring where the greatest difference of friction coefficient between the two oil formulations can be seen. These results simply show that the relative speed was too low to ensure enough pressure in the lubricant film to separate the surfaces. That fact should be kept in mind when analyzing the values of friction coefficient reduction; the Cameron-Plint results are not showing the engine mid-stroke friction coefficient but the reverse points where speed is low and the pressure in the combustion chamber does not correspond to the values near the top dead center when combustion takes place. However, it is remarkable that even with the test rig limitations, the differences in friction coefficient at mixed and boundary regimes caused by the difference in oil viscosity can be detected.

4.4.2 WAM machine results

After carrying out the proposed test plan described in section 3.5.2, the friction coefficient for the two oils under different entraintment speeds, SRRs, and temperatures can be seen in figure 4.29.

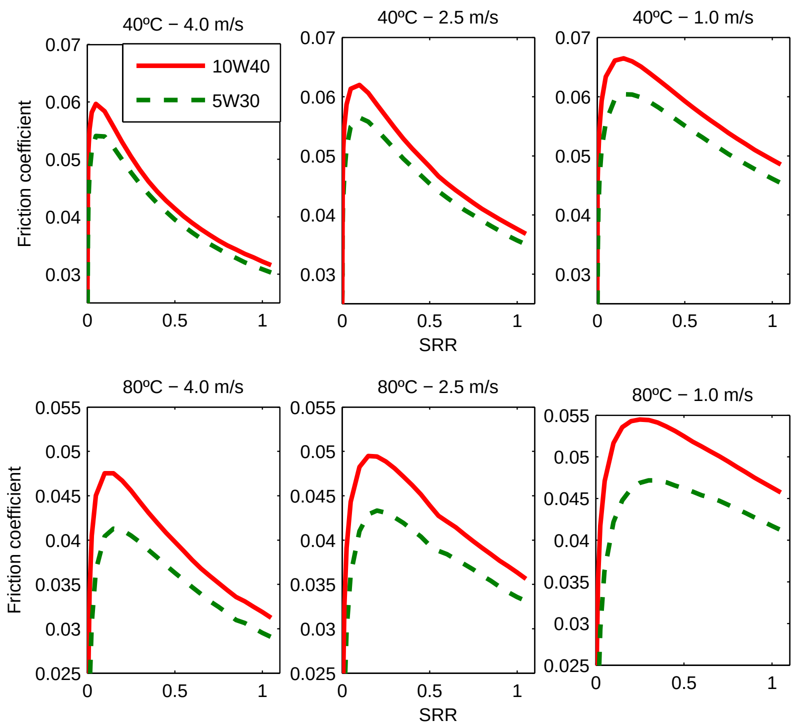

Figure 4.29: Friction coefficient results for WAM machine tests.

All the plots have shown the expected behavior of these "μ-slip" curves, with a linear increase in friction coefficient with SRR, followed by a non-linear region and then a maximum value due to the limiting shear stress of the oil. Then the friction decreases as the SRR increases mostly due to thermal softening of the lubricant. From the plots it is clear that the friction coefficient decreases when:

the 5W30 engine oil is used, the entrainment speeds increases and, once the local maximum of friction coefficient is reached and the curve enters into the thermal zone, when the SRR increases.

4.4.3 The effect of LVEO in friction coefficient

As it was mentioned previously, the friction coefficient for the 5W30 oil has been lower in all test scenarios. However, the magnitude of these friction differences due to oil viscosity oscillate depending on the other parameters: temperature, SRR and entrainment speed. According to the behavior seen in figure 4.29, it is obvious that the friction reduction with the SAE 5W30 oil is greater at 80 ° C than 40 °

WAM machine results discussion

As seen in figure 4.29, oil viscosity played a key role on the friction coefficient during the test and as it can be seen in figure 4.30 the difference in percentage tends to remain steady after certain SRR is reached, this is when the thermal region has been reached and the oil thinning due to thermal effects is evident, being this behavior more prominent for the 40° C test.

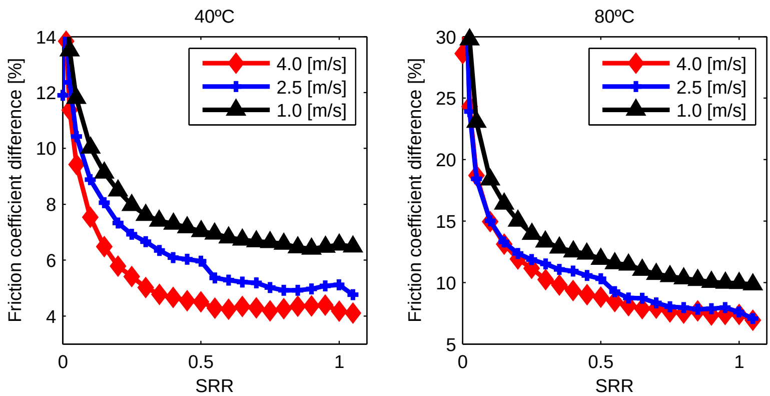

Figure 4.30: Friction coefficient differences between 10W40 and 5W30 oils at 40° C and 80° C and different entrainment speeds.

Temperature effect over friction coefficient differences

Friction coefficients measured at 40° C reached higher values than those measured at 80° C, both for 5W30 and 10W40 oils. In the graphs it is marked the sharp friction coefficient descent once the maximum value has been reached, and how this changes even more rapidly in the case of 40° test.

On the other hand, and taking into account the viscosity and temperature relationship, one could expect to see strong differences between "μ-slip" curves of the same oil at different temperatures. This is partially true as it can be seen in figure 4.31: firstly, at low SRR the difference is noticeable, however, as SRR increases, the differences tend to decrease asymptotically towards zero.

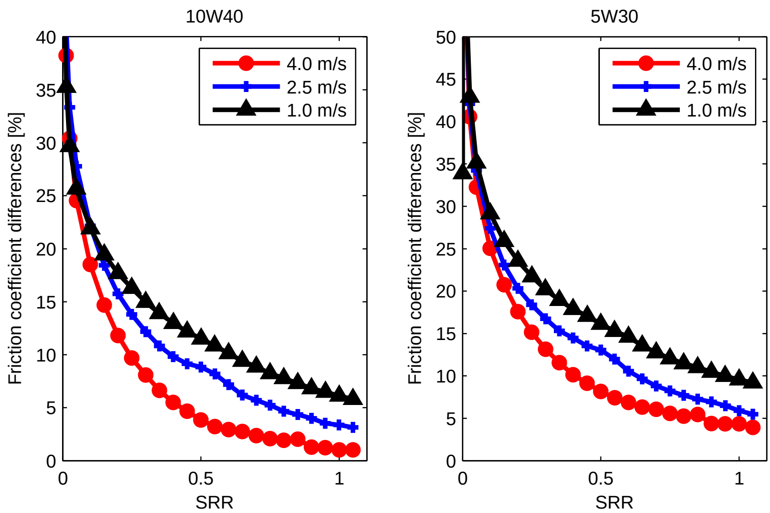

Figure 4.31: Friction coefficient differences by oil operation temperature (40° C and 80° C using as baseline the 40° C friction coefficient) of 10W40 and 5W30 oils different entrainment speeds.

Entrainment speed over friction coefficient differences

As stated by Taylor *et.al.*, proper values of entrainment speed in the cam-follower interface are fundamental to prevent wear, and must be carefully calculated when the cam profile is being designed[24]. In like a manner the behavior of friction coefficient when a less viscous oil is used should be studied. The entrainment speed in the valvetrain is dependent on engine speed and cam profile which determines the contact point speed; at the cam circle base entraintment speed has a low value which increases rapidly after the curvature radius reaches the cam flank, over the flanks the combination of entrainment speed and radius of curvature of the cam assure a good oil film thickness. The entrainment speed in these regions values varies depending on the cam size and engine speed, but normally does not exceed

4-6 m/s [24–29] given the limitations of the push rod design. Once the cam nose is reached, entrainment speed falls being this counterproductive to film formation. The ball-on-disc test results have shown how 5W30 oil reduced the friction coefficient at low and high entrainment speed, enhancing the mechanical losses related both for flank and nose cam regions.

Slide to roll ratio (SRR) over friction coefficient differences

SRR is a measure of slip, which normally occurs once the contact point in the cam-follower contact is leaving the nose region. As it can be seen on figure 4.30, as the slip increases, the friction coefficient values decrease, however, and similarly to the case of entrainment speed, the differences between 5W30 and 10W40 tend to decline as well being this more appreciable at 80° C, a temperature closer to real engine temperature.

References

[1] SANDOVAL, D. and J. B. HEYWOOD. “An Improved Friction Model for Spark-Ignition Engines”. *SAE Technical Paper* 2003-01-0725 (2003).

[2] JAMES, C. J. “Analysis of Parasitic Losses in Heavy Duty Diesel Engines by”. Master Thesis. Massachusetts Institute of Technology, 2012.

[3] RAKOPOULOS, C., D. HOUNTALAS, A. KOUTROUBOUSIS, and T. ZANNIS. “Application and Evaluation of a Detailed Friction Model on a DI Diesel Engine with Extremely High Peak Combustion Pressures”. *SAE Technical Paper* 724 (2002), SAE 2002–01–0068.

[4] OGAWA, T., T. SUZUKI, S. EZAKI, and T. SUZUKI. “Reduction of Friction Losses in Crankcase at High Engine Speeds”. *SAE Technical Paper* 2006-01-3350 (2006).

[5] CALABRETTA, M., D. CACCIATORE, and P. CARDEN. “Valvetrain Friction - Modeling, Analysis and Measurement of a High Performance Engine Valvetrain System”. *SAE Technical Paper* 2010-01-1492 (2010).

[6] SORRENTINO, M., F. MAURAMATI, I. ARSIE, A. CRICCHIO, and C. PIANESE. “Application of Willans Line Method for Internal Combustion Engines Scalability towards the Design and Optimization of Eco-Innovation Solutions”. *SAE Technical Paper* 2015-24-2397 (2015).

[7] TAMMINEN, J., C. E. SANDSTRÖM, and P. ANDERSSON. “Influence of load on the tribological conditions in piston ring and cylinder liner contacts in a medium-speed diesel engine”. *Tribology International* 39.12 (2006), pp. 1643–1652.

[8] TAYLOR, C. M. “Automobile engine tribology-design considerations for efficiency and durability”. *Wear* 221.1 (1998), pp. 1–8.

[9] RIBERIO FERNANDEZ, M. M., R. ZAMPA CARLINI, and E. M. TOMANIK. “Comparison of the potential to reduce fuel consumption by engine”. *SAE Technical Paper* (2013), pp. 2013–36–0221.

[10] TRATTNER, A., F. KUPELWIESER, P. PERTL, F. WINKLER, and R. KIRCHBERGER. “Thermodynamic Limits of Efficiency Enhancement of Small Displacement Single - Cylinder Engines”. *SAE Technical Paper* 2015-32-0817 (2015).

[11] KOGO, T. et al. “High Efficiency Diesel Engine with Low Heat Loss Combustion Concept - Toyota ’ s Inline 4-Cylinder”. *SAE International* 2016-01-0658 (2016).

[12] MACEK, J., D. FUENTE, and M. EMRICH. “A Simple Physical Model of ICE Mechanical Losses.” *SAE Technical Paper* 2011-01-0610 (2011).

[13] SINGH, D., M. SHUKLA, S. SINGH, Y. SINGH SR, and N. SINGH. "Fuel Economy of SI Vehicle Fueled by Ethanol/Gasoline Blends - Role of Engine Lubricant Viscosity". *SAE Technical Paper 2013-01-0051* (2013).

[14] USMAN, A. and C. W. PARK. "Transient lubrication of piston compression ring during cold start-up of SI engine". *International Journal of Precision Engineering and Manufacturing-Green Technology* 3.1 (2016), pp. 81–90.

[15] PASAOGLU, G. et al. *Driving and parking patterns of European car drivers - a mobility survey*. Tech. rep. European Commission, 2012.

[16] HORNIK, K. *Introductory Time Series with R*. Ed. by R. GENTLEMAN and K. HORNIK. First Edition. Auckland: Springer, 2009, p. 254.

[17] FREY, H. C., K. ZHANG, and N. M. ROUPHAIL. "Fuel use and emissions comparisons for alternative routes, time of day, road grade, and vehicles based on in-use measurements". *Environmental Science and Technology* 42.7 (2008), pp. 2483–2489.

[18] LENAERS, G. "Real Life CO2 Emission and Consumption of Four Car Powertrain Technologies Related to Driving Behavior and Road Type." *SAE Technical Paper* 2009-24-0127 (2009).

[19] MA, H., H. XIE, D. HUANG, and S. XIONG. "Effects of driving style on the fuel consumption of city buses under different road conditions and vehicle masses". *Transportation Research Part D: Transport and Environment* 41 (2015), pp. 205–216.

[20] MACIÁN, V., B. TORMOS, S. RUIZ, and G. MIRÓ. "Low viscosity engine oils: Study of wear effects and oil key parameters in a heavy duty engine fleet test". *Tribology International* 94 (2016), pp. 240–248.

[21] STACHOWIAK, G. W. and A. W. BATCHELOR. *Engineering Tribology*. Third Edition. Oxford: Elsevier Butterworth - Heinemann, 2005.

[22] FUENTES, E. and K. KANASE. "Relevance of Lubrication Regimes in the Study of Bearing Wear". *Critical Component in Heavy Duty Engines*. Ed. by P. LAKSHMINARAYANAN and N. S. NAYAK. First Edition. Singapore: John Wiley & Sons, INC., 2011. Chap. 12, pp. 199–252.

[23] CANNON. *Cannon HTHS Series II*.

[24] TAYLOR, C. M. "Valve Train - Cam and Follower: Background and Lubrication Analysis". *Tribology Series*. Ed. by C. TAYLOR. First Edition. Vol. 26. C. Amsterdam: Elsevier, 1993, pp. 159–181.

[25] BALL, A. D., D. DOWSON, and C. M. TAYLOR. "Cam and follower design". *Tribology Series* 14.C (1989), pp. 111–130.

[26] ZHU, G. “Valve trains - Design Studies, Wider Aspects and Future Developments”. *Engine Tribology*. Ed. by C. TAYLOR. First Edition. Amsterdam: Elsevier, 1993, pp. 183–211.

[27] CHANG, Q., P. YANG, J. WANG, and Q. CHEN. “Influence of temperature on cam-tappet lubrication in an internal combustion engine”. *Frontiers of Mechanical Engineering in China* 2.4 (2007), pp. 489–492.

[28] KUSHWAHU, M. “Tribological issues in cam-tappet contacts”. *Tribology and Dynamics of Engine and Powertrain*. Ed. by H. RAHNEJAT. First Edition. Oxford: Woodhead Publishing, 2010, pp. 545–566.

[29] GUO, J., W. ZHANG, and D. ZOU. “Investigation of dynamic characteristics of a valve train system”. *Mechanism and Machine Theory* 46.12 (2011), pp. 1950–1969.

Chapter 5

Conclusions y future work

In this chapter the final conclusions of the work described during this thesis are presented. General conclusions have been organized following the sequence of the objectives presented in the chapter 1. In the same way, specific tests conclusions follow the sequence of the tests descriptions in chapters 3 and 4. After the conclusions are portrayed a set of recommendations and future work are made based on the scientific gaps found through the literature review and the experience acquired during the comparative testing. Research such as that presented in this document is helpful to final users to understand the final outcome of using LVEO to reduce fuel consumption and CO_2 emissions. The originality of this thesis comes from the performance of testing at all levels; from the friction measurements in the tribo-contacts, through the BSFC measurements with engine at stationary operation and absolute fuel consumption in homologation cycles in the engine dynamometer, to real results in a large fleet of HDV.

5.1 Conclusions

The first objective of the thesis was to find the order of magnitude of the fuel consumption savings when LVEO were used. As seen in chapter 3 two tests addressed specifically this issue, the NEDC cycle test and the fleet test. From the results of these tests it is possible to conclude that:

- As it has been observed, low viscosity oils (LVEO) can be considered a key player in the fuel consumption reduction goal of the automotive sector. Both tests demonstrated that fuel consumption of LDV and HDV engines can be diminished using oils with lower values of HTHS viscosity.

- The order of magnitude of the fuel savings was in the limits of found in the literature, however, it has been seen that this value depends strongly on the vehicle and engine characteristics and the duty cycle.

- In the NEDC cycles, the diesel engine using a SAE 5W20 oil with HTHS viscosity of 2.8 cP consumed 1.67% less compared to itself when running with a SAE 5W30 with HTHS visocisity of 2.9 cP.

- During the fleet test, 2 of the 3 bus models showed less fuel consumption with statistical significance. After completing around 60000 km, the group of Diesel I buses using a SAE 5W30 with HTHS 3.594 cP consumed 1.83% less than similar buses using SAE 15W40 with HTHS 4.082 cP under similar ambient conditions, duty cycle and period of time. CNG buses using candidate SAE 5W30 Low Saps engine oil with HTHS viscosity 3.577 exhibited 3.7% less fuel consumption than their counterparts filled with SAE 10W40 Low Saps oil with HTHS viscosity 3.853 cP.

- The bus model which fuel consumption difference could not be proven statistically still exhibited the fuel consumption reduction effect of the LVEO. Diesel II buses using SAE 5W30 with HTHS viscosity 3.594 cP consumed 0.98% less than buses of the same model using SAE 10W40 Low Saps with HTHS viscosity 3.853 cP.

It was also an objective of this thesis to find the engine working zones where the effect of LVEO over fuel consumption could be enhanced. In order to accomplish this objective, the motored and the screening test done under fired stationary conditions were design and performed. In addition, other tests as the motored and NEDC tests run with the same engine were useful to find correlations and to extend the results to real applications in detail:

- From the motored test, the mechanical losses relation with engine speed was determined by the behavior of the torque required to move the engine. As shown in figure 4.1 and table 4.1 the mechanical losses increase as the engine speed increase.

- Regarding the effect of LVEO over the measured torque, it was clear that the use of LVEO reduce the required torque and the difference follow the same pattern as the torque, that is, as engine speed increases the effect of LVEO over mechanical losses was more noticeable, being as high as 14% between the SAE 5W20 and SAE 5W30 oil at 4000 min^{-1}.

- From the screening test where the performance of a SAE 15W40 engine oil and a SAE 5W30 where studied the fuel consumption reduction due the use of the former compared to the latter resulted to be significant, especially when the engine worked at low loads and low engine speeds. At 1000 min^{-1} and 2000 min^{-1} and 25% and 50% of WOT the engine exhibited the greatest BSFC reduction (4.44% and 3.7% respectively).

- Load has proven to have significant counter-back effect to the fuel consumption benefits given by the engine oil viscosity reduction. During the screening test the fuel consumption benefit decreased as the load increased, being most noticeable with engine running at low speed (at 1000 min^{-1} and 75% WOT the baseline oil showed lower BSFC).

- The counter intuitive behavior of the BSFC differences between LVEO and baseline oil during the screening test compared to the results of the motored test where the mechanical losses reduction given by the less viscous oil increased with the engine speed can be explained by the fact that during fired test other losses influence the BSFC behavior as well. As seen in section 4.2, even when mechanical losses rise with engine speed, their relevance within the overall engine losses decrease.

- An average of 1.6% of BSFC reduction was found for the 12 screening points test.

- From the analysis of NEDC cycles, it was possible to find good correlations with the previous screening test results: if the total fuel consumption reduction was divided by type of cycle, that is, urban and extra-urban, the most prominent fact was the very small benefit on LVEO during the extra-urban part of the cycle compared to the urban part (fuel consumption reduction of 0.5% for the former and 2.7% in average for the latter). Even when the cold-start condition of the test highlighted the contrast, the cycle effect is clear and it can be seen when comparing the fourth urban cycle (fuel consumption reduction of 2.23%) and the extra-urban cycle, which oil temperatures do not differ as drastically as the first couple of urban cycles (see figure 4.7.

- From the previous conclusions it can be said then that the effect of using LVEO to reduce fuel consumption will be noticeable for duty cycles where the low engine speeds but specially low loads are frequent. Extra-urban, aggressive drive style will make the engine to work in zones with higher loads, furthermore, other engine and vehicle losses will mask the mechanical losses reduction achieved by the use of LVEO.

As it was observed during the NEDC transient cycle, the effect of LVEO over fuel consumption has to be considered as well during the cold start process given the mentioned relation between lubricant viscosity and lubricant temperature. From the results on NEDC it can be concluded that:

- The benefit of LVEO over fuel consumption is more pronounced at low engine temperatures. At this point, the viscosity gap between regular oils and LVEO will be higher, as the engine reaches optimum temperatures the viscosity gap will be tighter and the viscous losses differences will not count as much as before. Comparing the values of the fourth urban cycles, it was found that BSFC reduction given by the SAE 5W20 oil compared to the SAE 5W30 baseline was 2.9% during the first two urban cycles and 2.3% during the last two of them.

- Taking into account the typical daily Light Duty Vehicles use in Europe (less than 80 km/day, an average of 6,5 hours parked during the day, and 16 hours during the night) the previous conclusions have major importance in terms of fuel consumption diminish just by using LVEO.

As mentioned in the literature review, CO_2 emissions come from the combustion in the ICE. As the main driver to reduce fuel consumption is tackling this adverse emissions, and an approximated level of this type of emissions can be obtain from carbon balance knowing the fuel composition the use of LVEO contributed to reduce CO_2.

- Each Diesel I bus using SAE 5W30 engine oil emitted 2.05 CO_2 Tons less than their counterparts using SAE 15W40 engine oil for the 60000 km mileage.

- Each CNG bus using SAE 5W30 Low SAPS engine oil emitted 4.2 CO2 Tons less than their counterparts using SAE 10W40 Low SAPS engine for the 60000 km mileage.

- In the case of Diesel I buses, during the second ODI the possible effect of lower viscosity differential oil over buses fuel consumption was included in the test. Despite the fact that candidate buses showed lower fuel consumption, it was not possible to statistically state that SAE 75W90 lead to lower fuel consumption compared to SAE 80W90.

The final phase of experimentation was done using tribometers to simulate some engine tribo-contacts using the same engine oils involved in the fleet test.

Two tribometers were used: a high frequency reciprocating rig, to study the piston ring - cylinder liner interface and a ball on disc tribometer to simulate the EHL contact of the cam-follower interface. In the former, specimens cut from a real liner and piston rings of the CNG bus engine were use to perform the test. Based on test results it is possible to conclude that:

- A significant difference in friction coefficient was detected for the less viscous oil using in the Cameron-Plint reciprocating machine.
- Given the Cameron-Plint limitations, only reverse operation points could be measured. However, the results supports the results of the fleet tests over the fuel consumption diminishing effect of LVEO.
- The friction coefficient reduction due the use of 5W30 oil is more sensitive at 20N of load and higher entrainment speeds. Hence, Heavy-Duty vehicles with working cycles with these kind of low load and high speed operating points are more susceptible to offer fuel consumption reductions.
- The high speed was the factor that maximize the effect of friction reduction of the 5W30 engine oil (6.727 % reduction).
- Extreme loads could prevent the benefits of low viscosity engine oils over fuel consumption as it was demonstrated in the ANOVA analysis. At 150N the difference between 5W30 and 15W40 is almost negligible.
- The 5W30 oil proved to give lower friction coefficient values at entrainment speeds and slip values similar to those found in a Heavy Duty engine valvetrain.

The final objective of the thesis was to determine if the cost-benefit balance of using LVEO results to be convenient for the final user. On chapter 4 taking the results from the fleet test and working with approximate cost values, the balance result beneficial for the final user.

- Taking into account the extra cost of the LVEO, the variation in engine oil consumption and the fuel savings and the fuel price, the use of LVEO in Diesel I buses could save 684€ per bus in a year. For the CNG the use of LVEO could save 991€ in a year. The total savings are remarkable taking into account that oil consumption was considerably higher in the case of CNG buses.

5.2 Future work

As it has been described in this document, the development of this thesis has been focused on extending the knowledge on how friction coefficient, fuel consumption and CO_2 emissions of Internal Combustion Engines (ICE) change when Low Viscosity Engine Oils (LVEO) are used. A special effort was made perform a real-world fleet test given the three following reasons: firstly, the inaccuracies of the outcomes of engine and chassis dynamometer testing when it comes to reflect the reality, secondly, the recent emissions scandals affecting some of the most important OEMs in the automotive sector and lastly, the fact that fleet test results can be easily understood by the final user. In the following paragraphs some future works are suggested in order to complete and extend the studies presented in this thesis:

- Perform steady and transient engine bed tests and the tribometers testing using a large number of oils of which only difference is the base stock viscosity index. The aim of this test would be to reduce the confounded effects of the different additives which could be found in an already formulated commercial oil, as those used during this thesis. It would be desirable to include oils with HTHS values below the 2.6 cP, which is the limit of SAE 20 grade oil. It has to be remembered that SAE 20 was the lowest viscosity grade approved by SAE J300 oil classification until 2013 when engine oils with HTHS viscosities as low as 1.7 cP were included. At the moment there are few studies which address the capabilities and counter-backs of using these ultra low viscosity engine oils (ULVEO).

- Study the influence of different surface finishes and/or surface coatings on the friction coefficient in different tribo-contacts in order to find the optimum combination of ultra low viscosity engine oils (ULVEO) and surface finishing and/or surface coatings. Some similar studies can be found for the last generation of engine oils, however, as indicated above, literature is still scarce when it comes to oils below SAE 20.

- Study the effects of using LVEO and ULVEO on engine pollutant emissions. As reported in the literature review, one of the most important roles of the engine oil is to seal the combustion chamber in order to prevent the "blow-by" effect, which leads to oil thinning by fuel dilution and pollutant emissions as HC and PM increase in the engine tailpipe.

- Perform long lasting engine test under very demanding conditions (high loads and engine speeds) alongside an oil sampling routine in order to study the HTHS viscosity behavior of ULVEO during engine operation. This would show possible wear of the engine by oil extra thinning.

- The fleet test presented in this thesis was very interesting since it reflects real fuel consumption and CO_2 emissions reduction of using LVEO. However, given the uncontrolled nature of the test it resulted very costly in time and resources. Besides given the experiment blocking due operation limitations the results showed in chapter 4 are not easy to extrapolate to other engines, or duty cycles. For all these reasons, a new fleet test where the effect of LVEO and ULVEO over buses fuel consumption and CO_2 emissions reduction can be found regardless engine design, bus model or route would be a good next step. This would be a controlled short test, using a small number of buses.

- As seen in fleet test results, it was likely that the fuel consumption benefit of LVEO depends strongly on engines thermal load, however, this could not be proved given the fact that bus engines where different in other characteristics besides this parameter and all of them where using baseline and candidate oils with different formulations. This problem could be solved by an engine bed test with BMEP as the only independent variable.

- In the fleet test, during the second ODI the effect of low viscosity differential oils was included. However, it was not possible to prove statistically the fuel consumption benefit exhibit in the data. As in the case of Diesel II buses, this seems to reflect a lack of data more than a negligible effect of the oils on vehicles efficiency. In a future test the differential oil could be included.

- The CO_2 emissions variation due the use of LVEO was found by carbon balance equations from the fuel consumption savings of the fleet test. However, in upcoming test this value could be directly measured with gas exhaust analyzers either in the test cell or under real world conditions with Portable Emissions Measurement systems (PEMS).

Appendix: Relevant original papers

Paper A: Assessment of the effect of low viscosity oils usage on a light duty diesel engine fuel consumption in stationary and transient conditions.

Tribology International. 79 (2014) 132-139.

DOI:10.1016/j.triboint.2014.06.003

Authors: Macián V., Tormos, B., Ruíz, S., Ramírez, L.

Abstract

Regarding the global warming due to CO_2 emissions, the crude oil depletion and its corresponding rising prices, OEMs are exploring different solutions to increase the internal combustion engine efficiency, among which, the use of Low Viscosity Oils (LVO) represents one attractive cost-effective way to accomplish this goal. Reported in terms of fuel consumption, the effect of LVO is round 2%, depending on the test conditions, especially if the test has taken place in laboratory or "on road" conditions. This study presents the fuel consumption benefits of a commercial 5W20, compared against higher SAE grade oils, on a light duty diesel engine, when it is running under motored test, stationary fired test and the New European Driving Cycle (NEDC). 2014 Elsevier Ltd.

Paper B: Potential of low viscosity oils to reduce CO_2 emissions and fuel consumption of urban buses fleets.

Transportation Research Part D: Transport and Environment. 39 (2015) 76-88.

DOI:10.1016/j.trd.2015.06.006

Authors: Macián V., Tormos, B., Ruíz, S., Ramírez, L.

Abstract

This paper shows the results of a comparative fleet test the main objective of which was to measure the influence of Low Viscosity Oils (LVO) over the fuel consumption and CO_2 emissions of urban buses. To perform this test, 39 urban buses, classified into candidate and reference groups depending on the engine oil viscosity, covered a 60000km mileage corresponding to two rounds of standard Oil Drain Interval (ODI). In the same way, for 9 buses of the 39 buses, the effect of differential LVO over fuel consumption and their interaction with engine LVO was assessed during the second ODI.Test results confirm that the use of LVO could reduce fuel consumption, hence CO_2 emissions. However, special attention should be taken prior to its implementation in a fleet, particularly if the vehicles are powered by engines with high mechanical and thermal stresses during vehicle operation because this could lead to friction loss increase, loss of the potential fuel consumption reduction of LVO and, in the worst scenario, higher rates of engine wear.

Conference participation: Evaluation of the Fuel Economy Improvement due to Low Viscosity Lubricants in a Light Duty Diesel Engine Running under the New European Driving Cycle (NEDC).

TAE 19th International Colloquium Tribology: Industrial and Automotive Lubrication, 21 - 23 January 2014 in Stuttgart, Ostfildern, Germany

Authors: Macián V., Tormos, B., Ramírez, L. Pérez, T.

Abstract

Low Viscosity Lubricants (LVL) have been proposed as an effective method to reduce engine's mechanical losses and consequently fuel consumption. Several

studies have been taken place in test bed bench, showing benefits never greater than 5%. However, the real effect of LVL is observable only at hydrodynamic lubrication regime, and special attention must be taken when using them since its effectiveness relies not only in viscosity reduction, but includes other factors as friction modifiers additives, engine design, engine's parts materials, aging of the lubricant oil, so on. This study present the fuel economy benefits of a commercial LVL, 5W20, compared against a 5W30, on a light duty diesel engine widely used in the European market, when this is run under the New European Driving Cycle (NEDC). Stationary fired and motored tests took place to support the analysis.

Conference participation: In-Use Comparison Test to Evaluate the Effect of Low Viscosity Oils on Fuel Consumption of Diesel and CNG Public Buses.

SAE Powertrains, Fuels & Lubricants 2014

DOI: 10.4271/2014-01-2794

Authors: Macián V., Tormos, B., Ramírez, L. de Diego, J.

Abstract

This paper shows the results of a fuel consumption in-use comparison test where the effect of Low Viscosity Oils (LVO) was evaluated over a sample of 39 urban buses powered by Diesel and CNG engines. The aim of the test was to verify the fuel consumption benefits of LVO in Heavy Duty Vehicles (HDV) found in previous works, which were obtained mainly in engine test bench, when engines are working on "on-road" conditions. In order to achieve this goal, a sample of 39 urban buses was studied over an Oil Drain Interval or 30.000 km (approximately an 11 month period), measuring daily mileage and fuel consumed to calculate each bus fuel consumption. Mileage was measured by GPS and fuel consumed was measured from refueling system.