Tribology International 110 (2017) 23–34

Contents lists available at ScienceDirect

Tribology International

journal homepage: www.elsevier.com/locate/triboint

Fuel consumption and friction benefits of low viscosity engine oils for heavy duty applications

Bernardo Tormos[a], Leonardo Ramírez[a,*], Jens Johansson[b], Marcus Björling[b], Roland Larsson[b]

[a] CMT-Motores Térmicos, Universitat Politècnica de València, Avenida Naranjos S/N, 46022 Valencia, Spain
[b] Division of Machine Elements, Luleå University of Technology, SE97187 Luleå, Sweden

ARTICLE INFO

Keywords:
Low viscosity engine oils
Engine friction losses
Piston-assembly
Journal bearings
Valve-train

ABSTRACT

One of the most attractive ways to tackle vehicle engine's inefficiencies is the use of Low Viscosity Engine Oils (LVEO). Adopted some decades ago for their use in the Light Duty segment, LVEO are now reaching the Heavy Duty segment.

In this study, a comparative fuel consumption test, where a LVEO performance is evaluated on an urban compressed natural gas buses fleet is portrayed. Then the friction performance of the same oils are studied on a Cameron-Plint tribometer, on an adapted twin disc tribometer to simulate journal bearing friction and on a Ball-on-Disc rig, using real engine parts in the former and the same set of engine oils used during the fleet test.

Results show a fuel consumption reduction in the fleet test and corresponding friction reduction in the tribometers when LVEO are used.

1. Introduction

The CO_2 emissions and fuel consumption reduction has arisen as a key driver in the automotive industry R & D, linked to a general public concern over Global Warming and the Green House Effect caused partially by the Green House Gases emitted by the vehicles which use Internal Combustion Engines as powertrain.

This concern has led to more restrictive CO_2 emissions standards in a vast number of industrialized countries. Although these regulations have been set for light duty passenger cars initially, the oncoming trend is to embrace Heavy Duty Vehicles (HDV) as well. It has to be mentioned that research in the HDV segment during the last years has been dedicated to reduce pollutant emissions, especially HC, CO, NO_x and particulate matter; this trend is evident when the progression limits of the Euro emission standards is analyzed [1].

From the cycle energy break down of a HDV, it is evident that most of the energy that comes from the fuel is used to overcome the different losses in the vehicle. Several energy distributions for HDV have been proposed by different authors where the type of vehicle and its duty cycle are the main factors defining those distributions. Holmberg et al, have proposed the energy break down for urban buses where 51% of energy is lost in exhaust and cooling, 18.5% in engine and transmission friction and 5.5% in auxiliary loads leaving just 25% of the initial energy contained in the fuel to move the vehicle [2].

An obvious approach to reduce the CO_2 emissions is to tackle the different sources of vehicle losses. One proven cost-effective way to increase engine efficiency is the use of Low Viscosity Engine Oils (LVEO) in order to reduce the friction losses in engine tribo-contacts which represent nearly 10% of the total losses, making them a good target in order to enhance engine efficiency, hence reducing CO_2 emissions. To understand how the use of LVEO could enhance engine efficiency it is crucial to understand engine friction and lubrication. In every pair of elements sliding against each other with relative motion exists a force acting against this movement, that force is friction, which depending on the lubricated pair characteristics will require more or less work to be overcome. The relationship between the lubricated pair and the friction coefficient is described by the Stribeck curve [3]; the curve shows the friction coefficient behavior for all the lubrication conditions, depending mainly on the lubricant rheology (specifically on lubricant viscosity η), the relative speed between the moving parts (U) and the normal force held by the parts (F). From the Stribeck curve three main lubrication regimes can be distinguished: the first one, where the lubricant layer between the parts in relative motion does not hold any load by hydrodynamic effects, allowing direct contact between the parts, which is called Boundary Lubrication Regime. The second one where the lubricant film layer is fully developed and the main resistance is given by the lubricant inner friction is known as the Hydrodynamic Lubrication Regime. A mixture of the previous two with miscellaneous characteristics of boundary and hydrodynamic regimes along the contact interface is called mixed lubrication. Specifically for

* Corresponding author.
E-mail address: leoraro@mot.upv.es (L. Ramírez).

http://dx.doi.org/10.1016/j.triboint.2017.02.007
Received 8 November 2016; Received in revised form 31 January 2017; Accepted 5 February 2017
Available online 07 February 2017

Table 1
Distribution of the engine friction losses by lubrication regimes for a bus (year 2000, bus @20 km/h).

Interface	Hydrodynamic	EHD	Mixed	Boundary
Piston assembly (5.5%)	2.2%	2.1%	0.6%	0.6%
Journal bearings (3%)	3%	–	–	–
Valve train (1.5%)	–	–	1.5%	–

ICE, several authors [4–6] have studied the friction distribution among the most important lubricated engine pairs: the piston-cylinder liner, followed by the bearings and finally the engine distribution system. Holmberg et al. have proposed a distribution of lubrication regimes for these three lubricated pairs, this time focused on the urban buses, the type of vehicle which is interesting for this study (see Table 1).

1.1. Piston ring pack interface

As it can be seen, nearly a 6% of total vehicle losses are present at the piston ring pack interface, most of it under hydrodynamic lubrication regime. This fact opens the possibility to reduce friction coefficient only by reducing oil viscosity. This effect has been measured by several authors in terms of fuel consumption reduction particularly for the passenger cars segment [7–13], however this focus has been changing and some studies have addressed the effect of LVEO on HDV efficiency improvement [5,14–18].

Some of these studies have used reciprocating rigs to simulate the piston ring dynamics and loads over the cylinder in order to find the friction coefficient behavior, both using laboratory specimens or real engine parts [19–30].

1.2. Journal bearings

This tribo-contact is the second source of engine friction as seen in Table 1. Although Journal Bearings could work under boundary and mixed regimes owing to changes in loads, speeds, and temperature [31], during engine operation this friction occurs under hydrodynamic lubrication regime. As for piston assembly, the use of LVEO could reduce losses in this interface, however, as these losses are decreased by reducing lubricant viscosity, the appearance of metal-metal contact becomes more likely, hence in recent years the study of bearing materials, coatings, transient loads and their respective wear performance have been widely studied [31–34].

1.3. Valvetrain

There are several cam-follower configurations where push-rod cam-followers is the most used for large HDV engines. The main frictional losses in the valve train occur between the cam and the tappet, the tappet and its bore, the rocker arm bearing, the valve stem and the valve guide and in the camshaft bearings. However, in terms of total energy, the energy dissipated in the cam and the tappet interface usually rises up to the 85% of the total energy dissipated in the valvetrain [35,36], hence the importance to study how the LVEO behaves in this interface. The valve train works normally under the hydrodynamic and elastohydrodynamic (EHD) lubrication regimes [37]; the former on the base of the cam circle and the latter case when the contact point is in the vicinity of the cam nose.

1.4. Use of LVEO on Heavy Duty Vehicles

The adoption of LVEO in the Heavy Duty Vehicles segment has lagged behind the passenger cars segment, due to a concern about their capability to withstand the loads associated with heavy duty cycles. However, in the recent years and following the general trend to reduce fuel consumption and CO_2 emissions of the automotive industry, new Heavy Duty Engine Oil categories were proposed by API (the American Petroleum Institute) in order to reach gains in fuel consumption benefits. On December 2016, API introduced two engine oil categories, CK-4 and FA-4, the former having a backward compatible role with previous category CJ-4 and the latter dedicated to increase vehicles fuel economy, surpassing the historic High Temperature High Shear viscosity (HTHS) limit of 3.5 cP. In Europe however, the recent ACEA engine oil specifications played it safe keeping the HTHS value in 3.5 cP [38].

2. Experimental methodology

For this study, fuel consumption data from a previous fleet experiment have been taken to be complemented with friction coefficient variation data from laboratory test rigs. The methodology used for laboratory tests was simple: to compare the friction coefficient in the tribo-contacts using the same engine oils used during the fleet test. The specific procedures are explained below.

2.1. Fleet test

The fleet test data have been taken from a fuel consumption study where CNG buses of the same model working under real conditions were divided in two groups; one using a SAE 10W40 Low SAPS engine oils as a baseline and another using a SAE 5W30 Low SAPS acting as Low Viscosity Engine Oil (LVEO) [39]. All buses worked during two Oil Drain Intervals (ODI) of 30,000 km each, and fuel consumption data were calculated daily from mileage and consumed fuel. Buses characteristics can be seen in Table 2.

2.1.1. Baseline and low viscosity engine oils

The oils used during this test as LVEO and baseline oil can be seen in Table 3. Both oils were commercial available.

2.2. 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 (see Fig. 1). 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 piezo-

Table 2
CNG buses characteristics.

Characteristic	CNG Vehicle
Year	2007
Length/width/height [m]	12/2.5/3.3
Engine displacement [cm^3]	11,967
Cylinders	6
Max. effect power [kW]	180 @ 2200 [1/min]
Max. effect torque [N m]	880 @ 1000 [1/min]
Crankcase volume [l]	33
BMEP [bar]	9.24 @1000 [1/min]
Thermal load [W/mm^2]	2,33
Valve train config.	OHV Push-rod Cam Follower