

# The Effect of Engine, Axle and Transmission Lubricant, and Operating Conditions on Heavy Duty Diesel Fuel Economy. Part 1: Measurements

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## ABSTRACT

It is expected that the world's energy demand will double by 2050, which requires energy efficient technologies to be readily available. With the increasing number of vehicles on our roads the demand for energy is increasing rapidly, and with this there is an associated increase in CO<sub>2</sub> emissions. Through the careful use of optimized lubricants it is possible to significantly reduce vehicle fuel consumption and hence CO<sub>2</sub>.

This paper evaluates the effects on fuel economy of high quality, low viscosity heavy-duty diesel engine type lubricants against mainstream type products for all elements of the vehicle driveline. Testing was performed on Shell's driveline test facility for the evaluation of fuel consumption effects due to engine, gearbox and axle oils and the variation with engine operating conditions.

To complement the rig based testing, a field test protocol has been developed to better understand the linkage between operating conditions and fuel economy changes when driveline lubricants are changed. Two standard, delivery-type, 18 tonne trucks have been modified with fuel-flow, engine operation and GPS (global positioning system) measurement equipment. Following a fixed on-road test route in the UK containing city, local, highway and hill driving conditions, the fuel consumption data was separated out for each situation to demonstrate the relative differences. To minimise variability, tight controls were placed on the test set-up, measurements taken and the operation of the vehicles.

Using a Mercedes Benz OM 460LA heavy-duty diesel engine run under the World Harmonised Transient Cycle (WHTC) and World Harmonised Stationary Cycle (WHSC), a combination of a SAE 5W-30 engine oil, SAE 75W-80 gearbox oil and SAE 75W-90 axle oil gave average fuel consumption improvements of 1.8% and 1.1%, respectively, relative to a SAE 15W-40 engine oil, SAE 80W gearbox and a SAE 90 axle oil. Using the WHSC cycle, significant variations in the individual lubricant contribution under different speed /load conditions within the cycle were identified. Additionally, an average fuel consumption improvement of 1.8% was observed using medium-duty trucks under a range of typical European driving conditions in a controlled field trial.

## INTRODUCTION

The US EPA (Environmental Protection Agency) estimates that heavy-duty trucks are the second largest source of GHG (Green House Gas) emissions in the transportation sector after light-duty vehicles. In the US, this sector represents approximately 22% of all transportation related GHG emissions. Moreover, the heavy-duty truck sector is also the fastest growing source of GHG emissions; from 1990-2008 GHG emissions increased over 72 %, while GHG emissions from passenger cars grew approximately 20% over the same period<sup>1</sup>. Past reports have identified technologies which could be implemented to reduce GHG<sup>2,3</sup>. The use of low friction lubricants in the engine, axle and transmission have been identified as a pathway which could reduce truck fuel consumption, and GHG emissions cost effectively<sup>4</sup>.

In recent years there have been multiple studies reporting the effect of the lubricant on heavy-duty diesel engine fuel economy<sup>4-8</sup>. These experimental studies focus on test bed engines or well controlled field trials, but seldom are these combined or reported together.

The objective of this paper is to assess fuel consumption improvements of high quality, low viscosity lubricants against standard, mainstream products for all elements of the vehicle driveline gathered from test bed tests and highly controlled field trial tests.

## DRIVELINE LABORATORY TESTING

In order to understand the relative efficiencies and improvements of driveline lubricants (engine, gearbox and axle oils), testing was performed on Shell's Driveline facility. A schematic of the facility is shown in Figure 1.

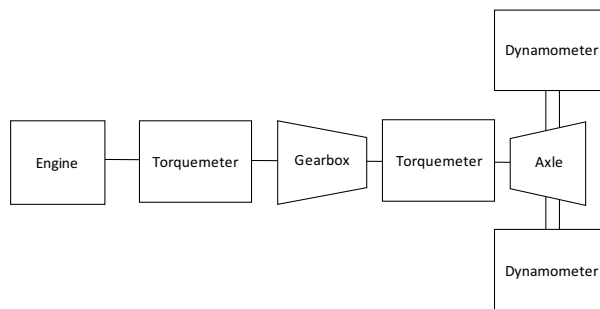


Figure 1. A schematic of the driveline test facility.

The driveline set-up is able to measure independently the energy losses within the engine, gearbox and axles by placing torquemeters between each element and at the end of the axle, the dynamometers having the capacity of functioning as torquemeters. Details of the engine, gearbox and axle are described in Table 1.

Table 1. Details of hardware tested in Shell's driveline test facility.

| Element  | Maker/Model        | Description                                                                                                                                 |
|----------|--------------------|---------------------------------------------------------------------------------------------------------------------------------------------|
| Engine   | Daimler OM460LA    | 12.8 litre, 6 cylinder, in-line configuration, Euro 4 emissions standard (Power: 315 kW @ 1800 1/min & Torque: 2050 Nm @ 1150 - 1450 1/min) |
| Gear Box | ZF Astronic        | Manual gearbox - 16 speed                                                                                                                   |
| Axle     | Mercedes-Benz HL 8 |                                                                                                                                             |

For all of the testing, the 11<sup>th</sup> gear was selected in the gearbox, providing a gear ratio of 2.5 combined with a final drive (axle) ratio of 2.9. This set-up was applied for all engine speed and load conditions.

Consequently, the engine speed and load were always fixed against the cycle demand, but the wheel speed for some gears differed from on-road expectation. The shortcomings of this approach were judged tolerable based on set goals. Future studies will consider the effect of gear changes on fuel consumption.

Measurements of engine, gearbox and axle efficiencies were made operating the engine using the World Harmonised Transient Cycle (WHTC)<sup>9</sup> and the World Harmonised Stationary Cycle (WHSC) including additional test points as described in Table 2. The WHTC test (Figure 2) is a transient cycle based on world-wide pattern of real heavy commercial vehicle use.<sup>9</sup> This cycle was run under both cold and hot start conditions – where a cold start was at a starting sump lubricant temperature of 25°C, and the hot start was at a starting sump lubricant temperature of 90°C. Similarly, the transmission and axle oils reach 50 and 45°C, respectively. The WHSC cycle was run only under hot conditions at a sump lubricant temperature of 90°C.

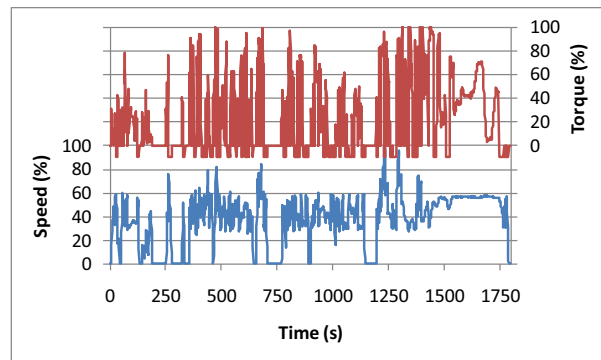


Figure 2. World Harmonised Transient Cycle (WHTC).

Table 2. World Harmonised Stationary Cycle: Details of engine test conditions.

| Mode                       | Speed (% of max.) | Load (% of max.) |
|----------------------------|-------------------|------------------|
| 0                          | Motoring          | -                |
| 1                          | 0                 | 0                |
| 2                          | 55                | 100              |
| 3                          | 55                | 25               |
| 4                          | 55                | 70               |
| 5                          | 35                | 100              |
| 6                          | 25                | 25               |
| 7                          | 45                | 70               |
| 8                          | 45                | 25               |
| 9                          | 55                | 50               |
| 10                         | 75                | 100              |
| 11                         | 35                | 50               |
| 12                         | 35                | 25               |
| 13                         | 0                 | 0                |
| Additional test conditions |                   |                  |
| 14                         | 25                | 100              |
| 15                         | 25                | 70               |
| 16                         | 75                | 25               |
| 17                         | 75                | 50               |
| 18                         | 75                | 70               |
| 19                         | 45                | 50               |

The test procedure was: (1) start the engine; (2) idle until oil sump is 25 °C; (3) WHTC-1: 'cold' stage (the oil sump starts at 25 °C and ends at about 90 °C); (4) WHTC-2: 'hot stage'; (5) WHTC-3: 'hot stage'; (6) WHSC. The oil sump temperature for steps (4), (5) and (6) is around 90 °C.

## TEST LUBRICANTS

Two driveline sets of lubricants were used in this testing representative of mainstream and top-tier oil formulations. Mainstream oils are identified as Oil Set A and the top tier type oils as Oil Set B. Lubricant performance levels are provided in Table 3.a, engine oil elemental composition in Table 3.b and viscometric properties are provided in Table 4.

Table 3.a Engine, gearbox and axle oil performance levels.

| Lubricant  | A-Mainstream type                 | B-Top Tier type                    |
|------------|-----------------------------------|------------------------------------|
| Engine oil | SAE 15W-40<br>(API CH-4, ACEA E5) | SAE 5W-30<br>(ACEA E6, E7)         |
| Gear oil   | SAE 80W<br>(API GL-4)             | SAE 75W-80<br>(API GL-4, API MT-1) |
| Axle oil   | SAE 90<br>(API GL-5)              | SAE 75W-90<br>(API GL-5, API MT-1) |

Table 3.b Engine oil composition

| Lubricant | A-Mainstream type | B-Top Tier type |
|-----------|-------------------|-----------------|
| Ca (ppm)  | 3600              | 2400            |
| P (ppm)   | 1170              | 750             |
| Zn (ppm)  | 1250              | 800             |

Table 4. Test lubricant details.

| Identifier             | A- Mainstream type |                  |                   | B- Top Tier type |                  |                  |
|------------------------|--------------------|------------------|-------------------|------------------|------------------|------------------|
| Lubricant              | Engine Oil         | Gear box oil     | Axle oil          | Engine Oil       | Gear box oil     | Axle oil         |
| Viscometric Properties |                    |                  |                   |                  |                  |                  |
| SAE Visc. Grade        | 15W-40             | 80W              | 90                | 5W-30            | 75W-80           | 75W-90           |
| CCS (mPas)             | 6600<br>@ -20°C    | 82284<br>@ -30°C | 702560<br>@ -30°C | 5638<br>@ -30°C  | 36500<br>@ -40°C | 13500<br>@ -40°C |
| Vk40 (cSt)             | 105.1              | 66               | 145               | 66.9             | 56               | 115              |
| Vk100 (cSt)            | 14.3               | 9.2              | 14.3              | 12.1             | 9.1              | 15.2             |
| HTHS @150 °C (cSt)     | 4.06               | N/A              | N/A               | 3.37             | N/A              | N/A              |

CCS= Cold crankcase viscosity (ASTM D5293), Vk40,100°C = Kinematic viscosity (ASTM D445), HTHS=High temperature, high shear viscosity (CEC-L-036-90).

## TEST DESIGN

A total of seven tests were completed with the entire driveline suite of lubricants A and B being alternated as shown in Table 5; such an arrangement ensures statistically balanced design with sufficient data to

provide statistically significant differences between oil sets.

Table 5. Test matrix.

| Test Order | Oil Set |
|------------|---------|
| 1          | A       |
| 2          | B       |
| 3          | A       |
| 4          | B       |
| 5          | A       |
| 6          | B       |
| 7          | A       |

## RESULTS

The speed and torque measurements were averaged over the individual time periods within the test cycle and across repeat runs. Efficiencies were obtained for each of the elements in the driveline by the following methods:

### Engine

The overall thermal efficiency of the engine ( $\eta_{\text{engine}}$ ) is defined as the ratio of the brake (or useful) power delivered by the engine to the fuel energy input

$$\eta_{\text{engine}} = \left( \frac{T * \omega}{\dot{m} * CV_{\text{fuel}}} \right) = \left( \frac{1}{CV_{\text{fuel}}} \right) \left( \frac{1}{BSFC} \right)$$

Where the T is the torque at the engine (N\*m),  $\omega$  is the engine speed (rad/s),  $\dot{m}$  the fuel flow (kg/s) and  $CV_{\text{fuel}}$  the calorific value of diesel fuel (43.03 MJ/kg).

### Gearbox

The gearbox mechanical efficiency ( $\eta_{\text{gb}}$ ) is defined as the ratio of the power delivered by the gearbox ( $P_{\text{gb}}$ ) to the power delivered by the engine ( $P_{\text{e}}$ ).

$$\eta_{\text{gb}} = \frac{P_{\text{gb}}}{P_{\text{e}}}$$

### Axle

The axle mechanical efficiency ( $\eta_{\text{axle}}$ ) is defined as the ratio of the power delivered by the axle ( $P_{\text{a}}$ ) to the power delivered by the gearbox ( $P_{\text{gb}}$ ).  $P_{\text{a}}$  is calculated as the sum of the power delivered to each output ( $P_{\text{a}} = P_{\text{a1}} + P_{\text{a2}}$ ).

$$\eta_{\text{Axle}} = \frac{P_{\text{a}}}{P_{\text{gb}}}$$

### World Harmonised Stationary Cycle

The energy efficiency changes moving from Oil A to Oil B for a single driveline component is calculated from the actual element efficiency values as the percentage increase in efficiency using the following equation. The results are summarized in Table 6.

$$\% \text{ Improvement in Energy Efficiency of Oil A vs. Oil B} = \left( \frac{\eta_{\text{B}} - \eta_{\text{A}}}{\eta_{\text{A}}} \right) * 100$$