

The Effect of Engine, Axle and Transmission Lubricant, and Operating Conditions on Heavy Duty Diesel Fuel Economy: Part 2: Predictions

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ABSTRACT

A predictive model for estimating the fuel saving of “top tier” engine, axle and transmission lubricants (compared to “mainstream” lubricants), in a heavy duty truck, operating on a realistic driving cycle, is described. Simulations have been performed for different truck weights (10, 20 and 40 tonnes) and it was found that the model predicts percentage fuel economy benefits that are of a similar magnitude to those measured in well controlled field trials¹.

The model predicts the percentage fuel saving from the engine oil should decrease as the vehicle load increases (which is in agreement with field trial results). The percentage fuel saving from the axle and gearbox oils initially decreases with load and then stays more or less constant. This behaviour is due to the detailed way in which axle and gearbox efficiency varies with speed/load and lubricant type. A customer that uses fuel economy engine, axle and gearbox lubricants will achieve higher fuel savings compared to a customer that just uses a fuel economy engine oil.

Predicted fuel savings amount to just over 0.6 kg/hour for a 40 tonne truck. This estimate is for an SAE 5W-30 synthetic engine oil (compared to a mineral based SAE 15W-40), a synthetic SAE 75-80 gearbox oil (compared to a mineral SAE 80W gearbox oil) and a synthetic SAE 75W-90 axle oil (compared to a mineral SAE 90). This equates to a percentage fuel consumption saving of approximately 2.3%. Larger savings are predicted if lower viscosity engine lubricants were to be used (and example calculations have been performed for an SAE 0W-20 engine oil).

INTRODUCTION

Transport of freight and goods using heavy duty diesel trucks is recognized as a major source of fuel consumption. In a previous SAE paper², it was pointed out that fuel costs account for at least 30% of the total operating costs of a typical fleet truck operation.

There have been many useful reports that have studied ways in which truck fuel consumption can be reduced^{3,4}. The use of low friction, synthetic-based lubricants, in the engine, axle and transmission, have been identified³ as one option to help reduce truck fuel consumption and CO₂ emissions. Compared to other options for improving truck fuel economy (such as the use of low rolling resistance tyres, improvements to truck aerodynamics, cylinder de-activation, etc.), Miller⁵ has pointed out that the use of low friction, synthetic lubricants, is one of the most cost effective ways of improving truck fuel economy, since no hardware changes are needed on the truck. The %change in fuel economy, divided by the cost (in US dollars) of using low friction lubricants was estimated by Miller⁵ at 0.16%/ \$, which is at least four times higher than other improved fuel economy options based on truck hardware changes.

There have been many studies of the effect of the lubricant on heavy-duty diesel engine fuel economy⁶⁻¹⁹. Many of these studies simply focused on the engine oil, and many studies are experimental, generating results on test bed engines or in well controlled field trials. There has been less focus on the potential fuel savings that can be achieved if engine, axle, and gearbox lubricants can be optimized together. In addition, there have not been many reports that claim

to be able to predict the benefits that are seen experimentally. In this paper, a model is proposed that attempts to develop a predictive capability, which captures the main features of the truck operation and driving cycle, which also takes into account the realistic changes in engine friction mean effective pressure (FMEP) when the engine lubricant is changed, and also includes measured axle and gearbox efficiency data (from Shell's Driveline rig in Hamburg, Germany).

The model has been used to predict truck fuel consumption, for truck weights of 10, 20 and 40 tonnes, running on the European Transient Cycle²⁰. Simulations were carried out for "mainstream" engine, axle and gearbox lubricants, and also for "top tier", synthetic based engine, axle and gearbox lubricants. Simulations were also carried out to predict the truck fuel consumption in the absence of engine friction (the FMEP was set to zero) and assuming the efficiency of the axle and gearbox was 100%.

VEHICLE FUEL CONSUMPTION MODEL

In order to estimate truck fuel consumption, it is necessary to calculate the power required from the fuel. This is done using equation (1).

$$\eta P_{\text{fuel}} = \frac{P_{\text{wheels}}}{\varepsilon} + P_{\text{access}} + P_{\text{friction}} \quad \dots(1)$$

Where:

P_{fuel} = power (Watts) required from the fuel
 P_{wheels} = power (Watts) required at the wheels
 P_{access} = power (Watts) required for the accessories (air conditioning, etc.)
 P_{friction} = power (Watts) due to engine friction losses
 η = engine thermal efficiency (approx. 40%)
 ε = efficiency of the whole transmission system

The power required at the wheels, for a truck travelling on a flat road, is:

$$P_{\text{wheels}} = C_R M g v + \frac{1}{2} C_D \rho A v^3 + M a v \quad \dots(2)$$

Where:

C_R = tyre rolling resistance coefficient (dimensionless)
 M = mass of truck (kg)
 g = acceleration due to gravity (=9.81 m/s²)
 v = speed of truck (m/s)
 C_D = truck drag coefficient (dimensionless)
 ρ = density of air (kg/m³)
 A = frontal area of truck (m²)
 a = acceleration of truck (m/s²)

The expression for the engine friction loss, P_{friction} (in Watts) is:

$$P_{\text{friction}} = \frac{\text{FMEP(kPa)} \cdot V \cdot N}{2} \quad \dots(3)$$

Where FMEP (kPa) is the engine friction mean effective pressure, V is the engine displacement (in litres) and N is the engine speed (in revs/second).

In general, we will have a vehicle driving cycle, in which the truck speed (in miles/hour or km/hour) is specified as a function of time (seconds). At each second, the truck speed can be converted into engine speed (revs/minute) if the gear (and gear ratio) at that point is known. Engine speed (rpm) is needed in order to estimate engine FMEP. For some driving cycles, BOTH the vehicle speed versus time data, AND engine speed/engine torque data are provided, which is clearly convenient for carrying out the fuel consumption calculations. An example of such a driving cycle, which is used in this paper, is the European Transient Cycle (ETC)²⁰.

At a specified time in the driving cycle, given all the data above, the various power losses can be determined, and an estimate of the power required from the fuel (in Watts) can be obtained.

The instantaneous fuel consumption rate (in grams of fuel per second) can then be calculated by dividing the predicted P_{fuel} by the Lower Heating Value (LHV) of the fuel – if P_{fuel} is in kW, we can divide by 44 kJ/g, which is a typical fuel LHV.

Once we have the instantaneous fuel consumption rate (in grams of fuel per second), at each point in the cycle, a simple integration over the whole driving cycle enables an estimate of the total fuel used (in grams).

ESTIMATION OF HDDEO FUEL CONSUMPTION FOR THE EUROPEAN TRANSIENT CYCLE

DETAILS OF EUROPEAN TRANSIENT CYCLE (ETC)

Figure 1 shows the vehicle/speed driving profile for a heavy duty truck running on the European Transient Cycle (ETC)²⁰.

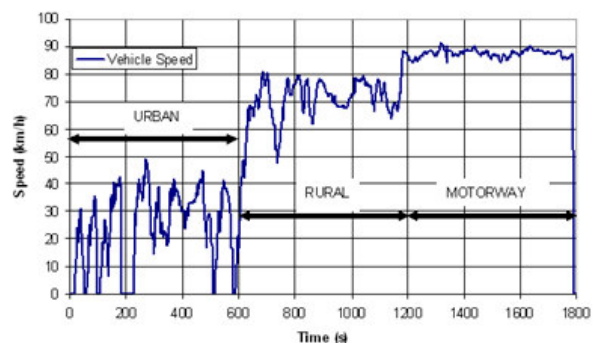


Figure 1: Vehicle speed versus time for European Transient Cycle (ETC)²⁰

In addition to the vehicle/speed driving profile, the ETC can also be run in an engine on a test stand, using normalized engine speed/engine torque versus time data, as shown in Figure 2.

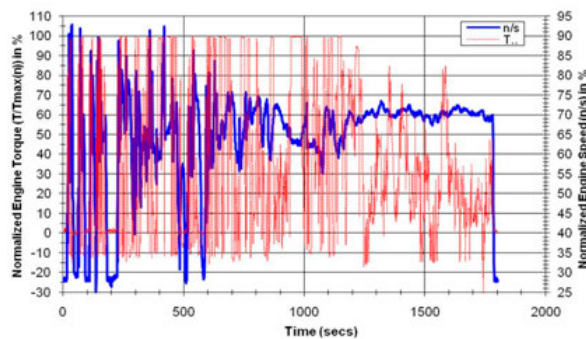


Figure 2: Normalised engine speed & torque versus time for ETC²⁰

Other driving cycles, such as the World Harmonized Transient Cycle (WHTC)²¹ were considered, but although there is a test bed engine cycle, there is no equivalent vehicle speed versus time data.

It was realized during the course of the current work that the engine test bed cycle shown in Figure 2, which is supposed to be representative of the vehicle driving cycle shown in Figure 1, must have been generated for a specific truck weight. In the normalized data of Figure 2, the maximum engine speed and maximum engine torque need specifying. When the truck weight is altered, the power required at the wheels will also be altered. A rough estimate of the power at the wheels was obtained for trucks of weight 10, 20 and 40 tonnes, and it was decided to scale the maximum engine torque (used as input for the data in Figure 2) according to vehicle weight. However, for the 40 tonne truck, the scaling actually gave a maximum engine torque which was greater than that available from the engine, so in that case we simply used the maximum engine torque as specified by the engine manufacturer.

Therefore, for a 10 tonne truck, it was assumed that the maximum engine torque was 750 Nm. For a 20 tonne truck, it was assumed that the maximum engine torque was 1500 Nm, and for a 40 tonne truck, it was assumed that the maximum engine torque was 2000 Nm. The engine was assumed to be a Mercedes Benz OM 460 LA diesel engine (a 12.8 litre six cylinder diesel engine with peak torque of approx 2000 Nm and peak speed of 2200 rpm).

VEHICLE DATA

M	Mass of truck (tonnes)	10: 20 or 40
A	Frontal area of truck (m ²)	9.0
N _{max}	Max engine speed (rpm)	2200
T _{max}	Max engine torque (Nm)	750:1500:2000
C _D	Drag coefficient	0.8
C _R	Rolling resistance coefficient	0.009
N _{idle}	Assumed idle speed (rpm)	550
P _{access}	Accessories power (kW)	5.0
V	Engine displacement (litres)	12.8
η	Engine thermal efficiency (%)	40

Table One: Data required for fuel consumption model

TEST LUBRICANTS

Table Two shows details of the “mainstream” lubricants and Table Three shows details of the “top-tier” lubricants.

	Engine oil	Gear oil	Axle oil
CCS (mPa.s)	6,600 @ -20°C	82,284 @ -30°C	702,560 @ -30°C
Vk40 (cSt)	105.1	66	145
Vk100 (cSt)	14.3	9.2	14.3
HTHS 150°C (mPa.s)	4.06	N/A	N/A

Table Two: Details of “mainstream” lubricants

	Engine oil	Gear oil	Axle oil
CCS (mPa.s)	5,638 @ -30°C	36,500 @ -40°C	13,500 @ -40°C
Vk40 (cSt)	66.9	56	115
Vk100 (cSt)	12.13	9.1	15.2
HTHS 150°C (mPa.s)	3.37	N/A	N/A

Table Three: Details of “top-tier” lubricants

The “mainstream” lubricants were:

- SAE 15W-40 mineral based engine oil
- SAE 80W mineral based gear oil
- SAE 90 mineral based axle oil

The “top-tier” lubricants were:

- SAE 5W-30 synthetic engine oil
- SAE 75-80 synthetic gear oil
- SAE 75W-90 synthetic mineral axle oil

In addition to differences in lubricant viscosity grade, there will be differences in base oil type and additive packages between these various oils, although neither of the engine oils contained friction modifiers. It is assumed that for the engine oil, changes in friction are entirely due to changes in viscosity only. For the gear and axle oils, measured efficiency data is used in the model described below, and the difference in efficiency between the different oils is likely to be due to a combination of (1) different viscosity grade, (2) different pressure viscosity coefficient due to different base oils used, and (3) different additive packages used.

ENGINE FRICTION AND LUBRICANT DEPENDENCE

The friction mean effective pressure, FMEP, in kPa, for the OM 460 LA diesel engine was estimated to be:

$$\text{FMEP(kPa)} = a + bN + c \cdot \text{BMEP(kPa)} \quad \dots(4)$$

Where: a = 1.0223, b = 0.0984 and c = 0.0540.

The above relationship between FMEP and engine speed/engine BMEP was for an engine lubricated with