

A New Method to Warm Up Lubricating Oil to Improve the Fuel Efficiency During Cold Start

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ABSTRACT

Cold start driving cycles exhibit an increase in friction losses due to the low temperatures of metal and media compared to normal operating engine conditions. These friction losses are responsible for up to 10% penalty in fuel economy over the official drive cycles like the New European Drive Cycle (NEDC), where the temperature of the oil even at the end of the 1180 s of the drive cycle is below the fully warmed up values of between 100°C and 120°C. At engine oil temperatures below 100°C the water from the blow by condensates and dilutes the engine oil in the oil pan which negatively affects engine wear. Therefore engine oil temperatures above 100°C are desirable to minimize engine wear through blow by condensate. The paper presents a new technique to warm up the engine oil that significantly reduces the friction losses and therefore also reduces the fuel economy penalty during a 22°C cold start NEDC. Chassis dynamometer experiments demonstrated fuel economy improvements of over 7% as well as significant emission reductions by rapidly increasing the oil temperature. Oil temperatures were increased by up to 60°C during certain parts of the NEDC.

It is shown how a very simple sensitivity analysis can be used to assess the relative size or efficiency of different heat transfer passes and the resulting fuel economy improvement potential of different heat recovery systems system. Due to its simplicity the method is very fast to use and therefore also very cost effective. The method demonstrated a very good correlation for the fuel consumption within $\pm 1\%$ compared to measurements on a vehicle chassis roll.

INTRODUCTION

Maximum efficiencies of up to 35% are possible for gasoline engines but during typical day to day driving in passenger cars only between 10% and 20% of the used fuel is converted into mechanical energy to move a car [1]. The majority of the energy is wasted either through the exhaust mass flow into the ambient air or via direct heat transfer from the engine and radiator into the environment. Engine friction is another large contributor of efficiency losses. For a fully warmed up engine the amount of energy wasted through engine friction can be anywhere between 10% at wide open throttle (WOT) up to 40% during part load, for example during city driving or cruising at constant speeds of up to 120km/h on the highway [2]. At lower temperatures, for example between 20°C and 30°C, the start temperature of the NEDC, the engine friction can be even 2.5 times higher [2, 3]. This strong increase in friction at low temperature is caused by logarithmic increase of oil viscosity with decreasing temperature [4]. The effects of ambient temperature and vehicle soak temperature were investigated by Ford at a vehicle level where three vehicles of different size and manufacturer with different engine concepts, for example direct injection gasoline and port fuel injection, were soaked and tested at different temperatures. Fuel economy improvements of between 5% and 10% were demonstrated for a temperature increase of only 10°C as shown in [figure 1](#) [5, 6].

As friction levels are reduced with increasing temperatures and on the other hand a lot of heat is wasted through the exhaust and the cooling system it seems to be logical to use some of the wasted heat to reduce friction. This idea has been investigated to some extend by warming up the coolant with exhaust heat through an exhaust gas heat exchanger. Such

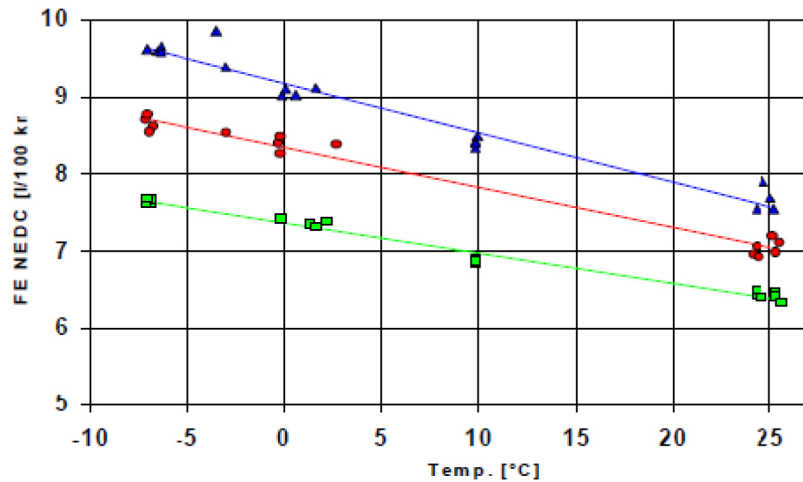


Figure 1. Fuel consumption as a function of test temperatures [6]

systems are already in production for various vehicles with very efficient drive trains even though these systems delivered only very minor fuel economy improvements up to a maximum of 1%, in some instances they even lead to a reduction of fuel economy [7, 8]. So the main reason why such exhaust gas/coolant heat exchangers were implemented is that they improve the heater performance which means that the passenger compartment is warmed up faster at very low temperatures that are typical during winter in regions with cold climates like for example Scandinavia.

ANALYSIS OF HEAT TRANSFER FROM EXHAUST GAS TO ENGINE OIL

A relatively simple sensitivity analysis helps to explain why only marginal fuel economy improvements can be realized by transferring exhaust heat to the engine coolant. To reduce engine friction the ultimate aim is to transfer as much heat into the engine oil as soon as possible. The analysis considers typical efficiencies for the various steps of the heat transfer process that are involved to warm up the engine lubrication oil with the exhaust gas. The advantage of using the approach with typical efficiencies is that it does not require a detailed and complicated model for the components involved. The underlying heat transfer equations are based on empirical investigations anyway with relatively large uncertainties. The main aim of this analysis is to demonstrate the order of magnitude of exhaust heat transferred into the oil. The efficiencies for the different heat transfer steps are displayed in [figure 2](#) where two different systems are compared, firstly a standard production configuration that uses a heat exchanger to transfer heat from the exhaust gas to the coolant, and secondly a new approach that transfers the exhaust heat directly to the engine oil with the same heat exchanger.

EXHAUST GAS/COOLANT HEAT EXCHANGER

As mentioned before the main purpose of such systems is to warm up the cabin faster at very cold temperatures. An exhaust gas heat exchanger is installed in the exhaust behind the catalyst. To avoid overheating of the coolant the heat exchanger can be bypassed on the exhaust side, controlled by a flap valve. On the coolant side the heat exchanger is in the “bypass” circuit that warms up the passenger compartment when required. Such an arrangement is displayed in figure 17 of [7]. The first step in this heat transfer process is the transfer of heat from the exhaust gas to the engine coolant. Here a typical heat exchanger efficiency η_{HE} of 80% is assumed. This is an arbitrary value that depends on the specific heat exchanger design and installation as well as on the inlet temperatures of the two fluids.

In a second step the remaining heat is transferred from the coolant into the metal structure of the engine that is also in contact with the engine oil, mainly the cylinder head and the cylinder block. So the cylinder head and -block could be considered as a typical heat exchanger to transfer heat from the coolant to the engine metal structure. However, this heat transfer path normally acts in the opposite direction: the combustion heat is transferred into the cylinder head and -block from where it is transferred into the coolant to avoid local overheating. With an increase in coolant temperature the heat loss from the combustion chamber to the coolant is actually reduced. The effects on the combustion process are neglected in this example for further simplification. Because the heat transfer function of cylinder head and a cylinder block is only a side function, they are not designed like typical heat exchangers. For example the wall thickness of the metal is a multiple of a typical heat exchanger and the surface area is relatively small due to the absence of fins that are difficult to incorporate in the manufacturing process of a

cylinder block and cylinder head. For dynamic warm up considerations the wall thicknesses of the cylinder head and -block with its specific heat capacities would be also important which in fact further reduces the warm up of the oil, but again these effects are not discussed here. The efficiency number should be smaller compared to the exhaust gas heat exchanger. Even though in most areas of the cylinder head and -block the heat is transferred from the metal to the coolant we can introduce an efficiency factor η_{CM} of 50% that represents the percentage of heat from the coolant that is remaining in the metal structure as a result of the reduction of heat loss from the metal to the coolant due to higher coolant temperatures. Again this is an arbitrary assumption based on a best engineering guess.

The final step is the transfer of heat from the cylinder block and -head metal structure to the lubrication oil. The additional heat from the coolant that is remaining in the metal structure as a result of the reduction of heat loss from the metal to the coolant is transferred in two directions: Firstly the target direction is to transfer this excessive heat into the engine oil and secondly the majority of the heat is lost into the ambient air with its much lower temperature. The heat rate transferred into the ambient air can be described simply by the convective heat transfer equation

$$Q_A = h_A A_A (T_M - T_A) \quad (1)$$

where h_A is the convective heat transfer coefficient to the air, A_A is the surface area of the engine in contact with the ambient air, T_M the average metal temperature of cylinder head and -block and T_A the ambient air temperature. In the same way the heat rate transferred from the engine metal to the lubrication oil can be described as

$$Q_O = h_O A_O (T_M - T_O) \quad (2)$$

Here h_O is the convective heat transfer coefficient to the oil, A_O is the surface area of the engine in contact with the lubrication oil and T_O the average oil temperature. The combined heat transfer rate from the metal to the air and oil is

$$Q_C = Q_A + Q_O \quad (3)$$

So the resulting efficiency of the heat transfer rate into the oil related to the combined heat transfer rate is

$$\eta_R = \frac{Q_O}{Q_C} \quad (4)$$

By dividing equation (2) through equation (1) the heat rate to the oil Q_O is described in relation to the heat rate to the air Q_A

$$\frac{Q_O}{Q_A} = \frac{h_O A_O (T_M - T_O)}{h_A A_A (T_M - T_A)} \quad (5)$$

With the introduction of three oil to air ratio factors Q_O can be described as a function of Q_A . The ratio factors are

$$\eta_h = \frac{h_O}{h_A} ; \quad \eta_A = \frac{A_O}{A_A} ; \quad \eta_T = \frac{T_M - T_O}{T_M - T_A} \quad (6)$$

The combined oil to air ratio factor is

$$\eta_C = \eta_h \eta_A \eta_T \quad (7)$$

So the resulting efficiency of the heat transfer rate into the oil related to the combined heat transfer rate is

$$\eta_R = \frac{Q_O}{Q_C} = \frac{\eta_C}{(1 + \eta_C)} \quad (8)$$

Some of these ratio factors were determined through measurements of the vehicle that was tested as described later. The test object was a Ford Falcon Turbo In-line 6 cylinder engine with a displacement of 4.0 liter. At the start of a NEDC test all temperatures are the same at 22°C. At the end of the test the maximum metal temperature of 96°C was recorded and the maximum oil temperature was 82°C, as shown in [figure 6](#) for the baseline configuration (heat exchanger off). The resulting average temperatures are $T_M = 59^\circ\text{C}$ and $T_O = 52^\circ\text{C}$, so the resulting 'temperature ratio' η_T is only 0.19. For the surface ratio a value of $\eta_A = 0.1$ was calculated based on the outside dimensions of the engine (0.7m long, 0.2m wide and 0.6m high) and oil gallery (total length 3.3m, average diameter $D = 13\text{mm}$), for more accurate calculations a CAD model of the block and head would be required.

The heat transfer processes during an NEDC cold start test is quite dynamic due to the constantly changing temperatures, material properties, vehicle speeds and engine speeds. Describing that process would require the solution of various complex differential equations which is not the purpose of this study. To simplify the process average values over the drive cycle were assumed for vehicle speed and material properties. This is a common practice even for complex combustion simulations with modern simulation tools. Most of these combustion simulation tools use empirical equations for the heat transfer in the combustion chamber in the form of heat transfer times the temperature difference, for example the equations from Woschni [9], Bargende [10], Hohenberg [11], or Annand [12]. None of these equations include solutions for the fundamental differential equations, even

though an heat transfer equation based on the solutions of these differential equations has been developed by Kleinschmidt [13, 14]. So the heat transfer coefficient for the engine oil was calculated using the following equations

$$Nu = 0.036 Re^{0.8} Pr^{1/3} (d/L)^{0.055} \quad (9)$$

$$h_o = \frac{Nu k}{D} \quad (10)$$

Material properties and dimensions that were used for that calculation are: effective length of oil gallery $L=2m$, engine width $d=0.2m$, oil flow rate 30l/min at an average engine speed of 1500RPM, and the oil properties at $50^{\circ}C$: kinematic viscosity $\nu=161mm^2/s$, conductivity $k=0.142W/(m^{\circ}C)$, density $\rho=870kg/m^3$, specific heat capacity $c_p=2.0055 kJ/(kg^{\circ}C)$ [15]. For the convection of the oil in the galleries that lead to a heat transfer coefficient $h_o=36.2W/(m^2^{\circ}C)$.

For the convection of air around the engine a heat transfer coefficient is calculated through [4]

$$h_A = 6.4 \left(\frac{\nu}{\left[\frac{m}{s} \right]} \right)^{0.75} \text{ in } [W/(m^2^{\circ}C)] \quad (11)$$

With an average drive cycle speed of $\nu=9.3m/s$ that results in an heat transfer coefficient $h_A=34.1 W/(m^2^{\circ}C)$ which leads to a heat transfer coefficient ratio η_h of 1.06.

Therefore the combined ratio factor η_C is only 0.0202 leading to a resulting efficiency for the convective heat transfer from the engine structure to the lubrication oil η_R of only 0.0198.

Now the total heat transfer from the exhaust gas to the engine oil is a multiplication of the individual efficiencies of all three heat transfer steps

$$\eta_{Total} = \eta_{HE} \eta_{CM} \eta_R \quad (12)$$

So the total efficiency is 0.8% and therefore very insignificant.

That means that only 0.8% of the available exhaust heat is transferred into the engine oil to reduce friction. If the average engine friction during a NEDC test is considered as 25% of the fuel energy as stated before in [2] and [3] this results in a fuel economy improvement of 0.2%. That does not include negative effects of increased power requirements of the water pump to overcome the additional resistance of the additional heat exchanger and the relevant hoses and the

additional thermal masses of the heat exchanger itself and particularly the additional cooling fluid in the heat exchanger which will compensate at least some of this theoretical fuel economy improvement potential.

So the results of this very simple sensitivity analysis are very well in line with results from actual measurements [7, 8] even though such a small amount is very difficult to verify in a standard fuel economy test cycle like the NEDC with typical test to test variability in the range of $\pm 1\%$ under very controlled conditions.

To get an idea how potential variations to the single value assumptions would affect the results of this analysis, a best- and worst case scenario are added. This also reflects the problem that most convective heat transfer calculations are based on empirical equations and it is not unusual to see variations of up to $\pm 25\%$ depending on the type of equation that is used [15]. Therefore in the best case scenario all three efficiencies are increased by 25% and in the worst case scenario reduced by 25%. The result is that the total efficiency will be multiplied by 0.42 in the worst case scenario or with 1.95 for best case. For the worst case that leads to a new heat transfer efficiency of only 0.3% and a fuel consumption reduction of 0.08%. In the best case scenario the heat transfer efficiency increases to 1.5% leading to a fuel consumption reduction of 0.4 %. Both of these results are very small and therefore still in line with the measurements of such a system [7, 8].

DIRECT HEAT TRANSFER FROM EXHAUST GAS TO ENGINE OIL

The above analysis demonstrated that using coolant to warm up the engine oil involves a series of heat transfer processes that are not very efficient, in particular the heat transfer from the coolant to the engine oil by using the engine as a heat exchanger. The reasons are significant heat losses from the metal structure of the cylinder block and head to the ambient air. Even though one primary function of the cylinder head and block is to transfer heat, the aim is to transfer the heat to the coolant and the ambient air to avoid local overheating of various components. But the engine is not really designed to transfer heat quickly to the engine oil. So if the major loss factors in [equation \(12\)](#) could be eliminated, particularly η_R , that would dramatically improve the efficiency of such an exhaust gas recovery system. Therefore a new approach is to warm up the engine oil directly with the exhaust gas. A standard production exhaust gas heat exchanger that is designed to warm up the coolant was installed in the lubrication system instead of the coolant system. An adaptor between engine block and oil filter was used to pick up the oil from behind the oil pump. From the exhaust gas heat exchanger the oil was returned back into the oil filter and from there back into the engine oil gallery. A system diagram