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Fuel conservation and emission reduction through novel waste heat recovery for internal combustion engines

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HIGHLIGHTS

▸ Variable oil pumps reduce the heat transfer between engine and oil by 50%.
▸ Exhaust gas/oil heat exchangers can reduce fuel usage by over 7%.
▸ NOx and CO emissions are also reduced by such systems.
▸ Water condensation contributes up to 50% of usable exhaust heating power.
▸ A simple external engine oil bypass can achieve similar benefits.

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ABSTRACT

Lubrication systems of combustion engines offer a large potential for energy conservation and reduction of emissions. Different approaches include variable oil pumps to adjust oil pressure and flow rates to the engines requirements or thermal management to reduce the viscosity of the engine oil. For both of these systems the fuel conservation during physical tests is typically much smaller than the predictions through computations. The root cause of these differences between simulations and test results are analysed in this paper with specific focus on the heat transfer from the engine to the lubrication oil and the effects of water condensation in the exhaust. The analysis resulted in different waste heat recovery system configurations that are presented. Vehicle test results for one system with a gasoline engine demonstrate a fuel conservation potential of over 7% together with two digit reductions of several emission components. For another more effective but also more simple system configuration a similar improvement potential is shown. Risks and benefits of such novel waste heat recovery systems are discussed. Further benefits are the positive effects on performance, reduction of wear and the potential of extended oil change intervals.

1. Introduction

Historically the major focus during the development of engine lubrication systems was to reduce the wear and to increase the durability of an engine. With the introduction of emission regulations the reduction of oil emissions like unburned hydro-carbons (HC) became another priority. With increases in fuel prices the focus has been shifted to the reduction of friction to reduce fuel consumption.

Loss through friction is the second largest factor – after wasted heat – that reduces the efficiency of internal combustion engines. For fully warmed up operating temperatures above 90 °C, the frictional losses within the engine have been reported to be in the range of between 10% and 40%, depending on the load of the engine [1–5]. However, during most normal driving conditions of a car the oil temperatures are much lower. The legal drive cycles to determine fuel consumption and exhaust emissions, for example the New European Drive Cycle (NEDC), start at a much lower test temperature between 20 °C and 30 °C [6] and at such a temperature of 20 °C the engine friction can be 2.5 times as high compared to a warm engine operation [4,8]. Another study performed in Germany revealed that the yearly average ambient temperature is even lower with only 9.8 °C [7] which means even higher friction contribution. For the average oil temperature of 55 °C during the NEDC test the friction is still 50% higher compared to a hot engine with 100 °C oil temperature [4]. That means that under normal driving conditions the frictional losses of a passenger car's combustion engine can account to up to 60% of the total fuel energy!

Historically the main driver to reduce fuel consumption by reducing friction was to reduce the operating cost of a vehicle together with the positive effects on engine performance. With

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rising fuel prices the reduction of friction has become a much higher priority [9]. Over a typical lifetime of a vehicle of 150,000 km between Euro 1500 and Euro 2100 can be saved at current fuel prices.

Even though the end customers could save substantial operating expenses with most fuel saving technologies, only those fuel saving technologies with a very attractive cost/fuel consumption reduction ratio have been introduced into production, depending on the individual strategies of the different car companies. To reduce the friction of the lubrication system the biggest focus was dedicated towards developing new low friction material combinations – for example for piston skirt, cylinder liner, bearing shells or camshafts – low friction oils, or to reduce the tangential tension of piston rings [10,11]. A study performed by Bezdek and Wendling from the US National Research Council [12] showed the best cost/value ratio of just below an average of US$10 (ca. Euro 7.8) for low friction lubricants compared to the highest value of US$140 (ca. Euro 109) for six speed automatic transmissions. In the mean time most of the technologies from this study have been introduced into production on a large scale. For the low friction oils this value related only to the initial fill in the factory, for subsequent oil changes similar or more extra costs apply so the lifetime cost/value ratio is much less attractive.

Just recently this boundary condition has been changed dramatically with the introduction of CO_2 penalty taxation particularly in Europe. In December 2008 "the Council of Ministers and the European Parliament agreed to reduce fleet-average CO_2 emissions of cars sold in Europe to (nominally) 130 g/km by 2015, and to 95 g/km by 2020" [13]. For car manufacturers that fail to meet this target they need to pay penalty taxes of up to Euro 95 for each car per 1 g CO_2/km exceeding the target. To convert fuel consumption into CO_2 emissions (in g/km) through the carbon content the fuel consumption of a Gasoline engine in l/100 km needs to be multiplied by 23.7 [14]. That means for a fuel consumption excess of 1 l/100 km this could lead to a penalty tax of up to Euro 2251.5 per car! This is more than twice the actual savings for the same fuel consumption reduction. This highlights the urgency for the automotive industry and in particular the car manufacturers to dramatically reduce the fuel consumption as soon as possible.

2. Friction effect of lubrication systems on vehicle fuel economy

Increasing the variability of oil pumps is another effective way to reduce engine friction and fuel consumption. A very attractive ratio of only Euro 5 per 1% fuel consumption reduction has been quoted [15]. With non-variable oil pumps the oil pump power increases quadratically with the engine speed, but the required oil pressure is almost independent of the engine speed. The maximum required oil pump performance is determined by the condition of very hot oil in combination with idle speed to ensure sufficient lubrication for all bearings and the function of the variable valve train systems. So at low temperatures or at high speeds or a combination of both the oil pump delivers much more pressure compared to what is required. This resulting excessive flow into the oil pan also results in additional foaming losses and higher aeration levels that also have a negative impact on friction. To avoid any leakages due to over-pressure the oil pumps have a pressure relieve valve (or panic valve). A typical oil pump flow characteristic including the pressure relieve valve is depicted in Fig. 1. The example shows that the discharge volume of the pump reduces dramatically once the pressure relieve valve is open. At discharge pressures over 430 kPa virtually no oil is discharged at an engine speed of 2000 RPM in this case.

Several different options are possible to increase the variability of an oil pump to reduce pumping losses and most of them have already been introduced into production to some extend. The simplest form is the control of the oil pressure with an additional by-pass valve that already opens at a lower pressure than the panic valve. The complexity of such variable lubrication systems increase if the flow rate is also adjustable. The most variable solution is a full electric oil pump where the flow rate and pressure can be adjusted independent from the engine speed. Other advantages of these systems are reduced aeration and reduced slashing losses. Woeckel conducted a study that demonstrated a reduction of gas content in the oil by 90% for a reduction of oil pressure by only 50% [16]. In the same study also a significant noise reduction was demonstrated with reduced oil pressure, but more importantly also the noise quality was improved because the distance from the most dominant noise order to the overall noise level increased, which means that the most significance of the most dominant order also was reduced.

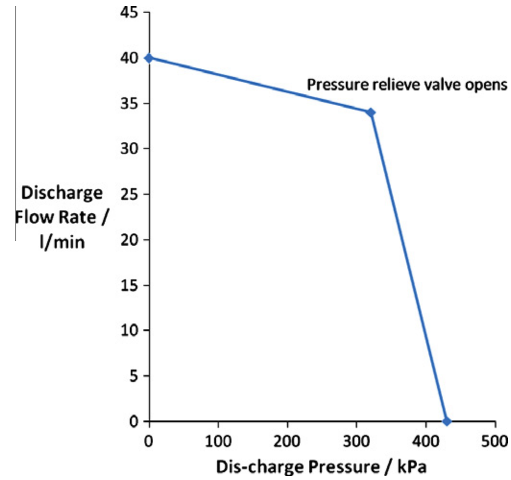

Fig. 1. Typical oil pump flow characteristic for a speed of 2000 RPM.

However, there is a significant difference between the fuel consumption reduction predicted for variable lubrication systems with the help of simulation tools and the actual measured test results. Simulations predict fuel consumption reductions of up to 4% over the NEDC test and up to 15% in specific test conditions [15,16]. In test results on the other hand maximum fuel consumption reductions of only between 1% and 2% have been published [17–20]. There are several reasons for this discrepancy:

The most important difference between the simulations and the physical tests is that during cold start tests with reduced oil pressure and reduced oil flow rates the heat transfer between the oil and the engine metal structure, especially the hot cylinder head, is reduced significantly. The result is that the engine oil warms up slower which actually increases the friction again and compensates some of the friction reduction potential of the lower oil pressure and flow rates. The worst case scenario for the heat transfer between cylinder head and engine oil is when the pressure relieve valve is fully open and virtually no oil flows through the cylinder head oil galleries. Some engines also have pressure reduction valves between cylinder block and -head to avoid damage of sensible hydraulic valve lash adjusters, these valves reduce the heat transfer from the cylinder head to the engine oil even further.

Secondly the simulations are relatively inaccurate because most of them are based on quasi-static models that do not consider any warm up effects. The friction reductions are calculated for hot engine operating temperatures for various steady state conditions according to the distribution of operating points in the drive cycle and the effect of the warm up is adjusted simply by multiplying a cold/hot factor which is typically between around 10% and 15% [21–23]. The definition of the cold hot factor is the difference between the fuel consumption of a cold test and a hot test that is followed directly after the cold test divided through the hot test fuel consumption.

Thirdly the simulations are also based on friction analyses that use the strip down method: the engine is motored by a dynamometer and the required motoring torque is measured. Oil temperatures are typically maintained at a constant level by an external heat management. At the beginning the motoring torque is measured for the complete engine, than one component after the other is removed and the torque measurements repeated, starting with the removal of the generator, water pump, oil pump, fuel pump (only for Diesel and direct injection gasoline), cam shafts, valves (to evaluate the pumping losses), and the piston group including connecting rods. There are several issues with this approach: the results vary quite significantly between tests without load and with load. The friction at higher loads is typically much higher than at lower loads due to the higher combustion forces, especially at low engine speeds where the compensating mass forces are small [24]. This increase of friction with increase of load is most relevant to the crankshaft main bearings and the con-rod bearings. But for the piston friction this is not as relevant due to the piston pin offset from the centre-line. The friction of the piston group is often quoted to be in the range of 40%, However this includes the con-rod big end bearings and the piston pin bearings. For a four-cylinder engine with five main bearings and four con-rod big end bearings the friction of the connection rod big end bearings are almost in the range as the main bearing friction which amounts to around 20% of the total engine friction [24]. The additional friction due to the bending of the crankshaft under load is also not included in strip down friction measurements.

3. Heat transfer in engine oil galleries

The accurate modelling of the heat transfer between the gas in the combustion chamber and the lubrication oil for a dynamic vehicle test such as the NEDC is very difficult. Many parameters are changing rapidly, particularly the pressure, temperature and material properties of the gases in the combustion chamber but also to a less extreme extent the vehicle speed and the engine speed. Alone the modelling of the heat transfer between combustion gas and the walls of the combustion chamber requires the solution of various complex differential equations. Although a solution to these differential equations was published some time ago [25,26] this solution has not been introduced in any commercially available combustion simulation tool, maybe due to the complexity of this new equation and the resulting programming difficulties. Instead, these combustion simulation tools utilise relatively basic empirical formulas in the form of “heat transfer coefficient” times “temperature difference between chamber wall and gas” [27], for instance the equations from Woschni [28], Bargende [29], Hohenberg [30], or Annand [31]. Therefore a similar simplification is justified for this analysis, so it is based on the assumption of drive cycle average values for engine speed and material properties. A very simple equation was used to calculate the heat transfer coefficient between engine galleries and oil [32]:

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

$$h_O = \frac{Nu\, k}{D} \tag{2}$$

The engine dimensions were taken from a 4.0 l Ford Falcon Turbo In-line 6 cylinder engine. This vehicle was also used for experimental tests that will be described later. The material properties were taken from [33]. A resulting heat transfer coefficient h_O = 36.2 W/(m^2 °C) was calculated. If the oil flow rate is reduced by 50% to 15 l/min this results in a reduction of the heat transfer coefficient by 43%. This explains the difference between experimental tests and simulations for variable oil pump systems. An ideal method to minimise fuel consumption would be to increase the heat transfer without increasing the oil pump power requirements by raising the oil flow rate.

4. Oil warm up system – test results and benefits

The experiments were conducted with an Australian Ford Falcon vehicle. The engine is a Barra 245 T, a turbo charged In-line six cylinder engine with intercooler, the displacement is 3984 ccm. The engine has four valves per cylinder and two overhead cam shafts, each of them with an independent variable cam phaser and knock control, similar to most modern down sized engines but with port fuel injection. The vehicle is a rear wheel drive with automatic transmission.

4.1. Fuel consumption

Experimental tests that were completed to evaluate the maximum fuel saving potential of warming up the engine oil often started with the assumption that the 10–15% fuel consumption difference between a cold- and a hot NEDC test is the maximum fuel saving potential through any warm up measures and even though the warm up potential of the engine is a major contributor towards the cold/hot-factor there are also some other factors that contribute towards the cold–hot factor. The most important of these factors are the warm up of the transmission (particularly automatic transmissions that have lower efficiencies than manual transmissions), calibration effects required to warm up the catalyst to meet emission legislations (e.g. increased idle speed or cold start spark retard), tyre temperature, the friction of the remaining drive shafts and wheel bearings as well as differences in the battery charge mainly caused by electric consumers during the 6–32 h soak period. That often leads to the general assumption that warming up the oil can be specifically relevant for short drive cycles that include a cold start but that it would not be very relevant for warmer and longer drive cycles like the EUDC. Farrant [34] predicted a maximum fuel economy improvement potential of only 2% for the EUDC if the complete test would be conducted with 94 °C engine oil temperature. So most approaches focus on increasing the engine temperature during the first part of the drive cycle but not so much on the second part, also because the coolant – different to the oil – reaches the operating temperature early in the EUDC.

To investigate the general potential of increasing the lubrication temperature by utilising wasted exhaust heat vehicle tests with an exhaust gas heat exchanger have been performed. Initial results have been presented in [35,36]. Five NEDC vehicle tests were conducted in two different configurations, with and without an active exhaust gas heat exchanger. The vehicle was tested on a chassis dynamometer in an emission lab that was accredited according to ISO9000 and ISO17025, details of the testing equipment were described in detail in various publications [21,42]. The results of five NEDC tests were averaged for each system configuration and compared for the two different configurations.

The fuel consumption was reduced by 7.3% for the combined drive cycle, and for the urban part a fuel consumption reduction of even 7.8% was achieved (Fig. 2). For the urban part the result was much lower than Farrant’s simulations because even though the engine inlet temperature with the exhaust gas heat exchanger (HE) was up to 60 °C higher than without HE, for most parts of the drive cycle the oil temperature was still much lower than Farrant’s fully warmed up temperature of 94 °C. Of particular interest were the high fuel economy improvements of 6.9% for the EUDC. This result demonstrates that the maximum fuel economy potential through increasing engine oil temperature is much larger than just

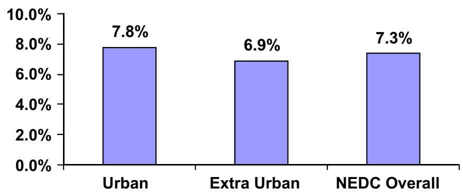

Fig. 2. Fuel economy improvements with oil/exhaust gas heat exchanger (vehicle measurements averaged over five tests in each configuration) [36].

the cold/hot factor if the oil temperature can be increased to higher levels, for instance to up to 120 °C as shown in this example. The higher oil temperatures in the EUDC are possible because the exhaust flow rates are much higher compared to the urban part of the drive cycle due to much higher vehicle speeds. Due to the quadratic effect of the vehicle speed on the vehicle road load the engine speeds and -loads are also much higher. The higher engine speed causes the friction to increase in absolute terms but also as a percentage of the total energy usage and, therefore, offers more efficiency optimisation potential with higher oil temperatures.

4.2. Heat transfer

To support that theory, further analysis was conducted. Fig. 3 shows the heat flow rate that was transferred by the exhaust gas into the oil over the drive cycle. The heat flow rate was calculated with the following equation:

$$\dot{Q} = \dot{m}_E C_{PE}(T_{E1} - T_{E2}) \tag{3}$$

The average heat flow rate transferred over the NEDC is 3.8 kW but during the extra urban part 2.3 times as much heat was transferred compared to the urban part. Another interesting effect is the potential of utilising the water condensation. The lower solid line in Fig. 3 represents the heat flow rate calculated only through the difference in exhaust temperature alone, refer to Fig. 5. The dashed line indicates the additional heat flow rate available if all water contained in the exhaust would condense. For petrol it can be calculated with the fuel's heat value h_F and the Air Fuel Ratio (AFR)

$$\dot{H} = 0.08 \frac{\dot{m}_E}{(AFR + 1)} h_F \tag{4}$$

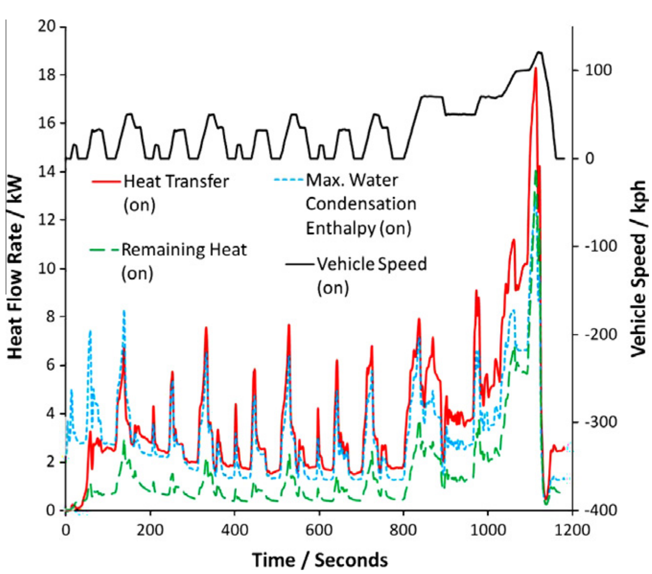

Fig. 3. Heat flow rates for a NEDC vehicle test.

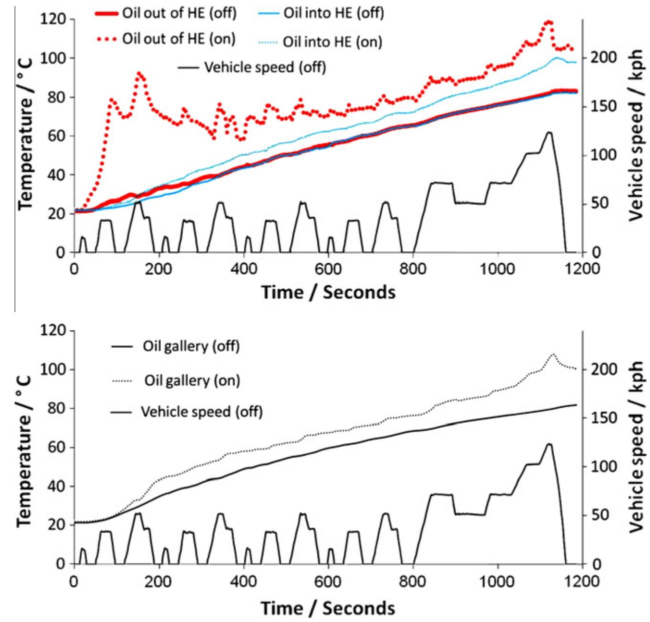

Fig. 4. Oil temperatures with and without new exhaust gas/oil heat exchanger [35].

The condensation enthalpy represents an average of 3.2 kW, which is almost as much as the heat flow rate through the exhaust temperature reduction itself. That means by utilising the water condensation enthalpy in such a heat exchanger the effectiveness of that system could almost be doubled in this case. For more efficient Diesel engines with much lower exhaust temperatures this effect is even more important.

Without HE the oil temperatures show an almost liner behaviour (Fig. 4). With the HE the situation is quite different as most of the heat is transferred to the oil through the HE instead of the engine itself. The operation of the pressure relieve valve causes a different characteristic. After the HE the temperature firstly rises fast without a significant change in the oil gallery, as the oil mass flow is small so the gallery temperature is mainly influenced by the block temperature. When the valve closes after 150 s, the flow through HE and gallery increases causing a drop of temperature after HE but at the same time the gallery temperature increases.

Further analysis of the heat exchanger effectiveness indicates that even in the tested configuration some significant amounts of water condensed. Specifically during the first part of the drive cycle

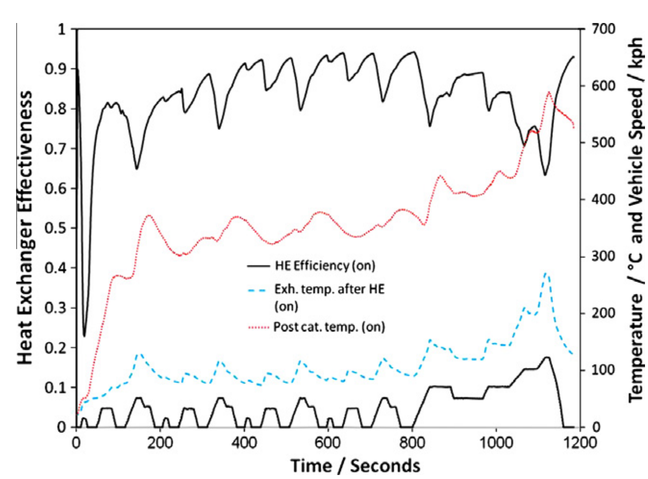

Fig. 5. Exhaust temperatures and heat exchanger effectiveness.

the exhaust temperature after the HE was lower than 100 °C when the vehicle speed was lower than 35 kph. This correlates very well with increases in HE effectiveness, and respectively the effectiveness dropped when the exhaust temperature after HE was exceeding 100 °C. The heat exchanger effectiveness was calculated with

$$\eta_{HE} = \frac{T_{O2} - T_{O1}}{T_{E1} - T_{O1}} \tag{5}$$

T_{O2} is the oil temperature after the heat exchanger, T_{O1} describes the temperature of the oil entering the heat exchanger and the exhaust temperature into the heat exchanger is T_{E1}. The rapid effectiveness drop immediately after the start correlates with the opening of the pressure relieve valve that causes the flow through the HE to decrease significantly, together with low exhaust gas temperatures. Until the end of the urban cycle at 800 s the peak effectiveness continues to increase through rising oil flow rates. For the EUDC the peak effectiveness drops again due to the increase in the exhaust gas flow rate.

4.3. Exhaust emissions

Other interesting findings from these vehicle level tests were the significant reductions of all regulated exhaust emissions that were measured in g/km (Fig. 6). The reduction in fuel consumption resulted in an equivalent reduction of exhaust mass flow by 7% that should lead to a similar reduction of all exhaust emissions. Carbon Monoxide (CO) were reduced even further by 27% which can be explained with reduced flame quenching because of the warmer oil – the engine oil in the combustion chamber is the first substance in contact with the combustion gas before the heat is transferred further into the walls of cylinder head, -liner and piston. The reduction in engine load helped to reduce Nitrogen Oxides (NO_x) due to lower maximum combustion temperatures, a total reduction of 19% was achieved. Surprisingly most of the emission mass flow reductions were compensated in the case of the Hydro Carbon (HC) emissions.

The oil balance in the combustion chamber is depicted in Fig. 7 and helps to explain the HC compensation. For the oil emissions seven different mechanisms are shown. The higher oil temperatures can have a negative effect on most of these mechanisms. A reduced viscosity increases the flow rates for scraping, dashing, leakages and blow by in both directions. Higher oil temperatures are also causing more oil to vaporise so it can be exhausted in gaseous form and more oil burns or cokes within the combustion chamber and forms HC emission components that are more difficult to be converted within the catalyst. The HC emissions caused through oil emissions can actually account to up to 30% of the HC emissions [37]. Dilution is expected to be reduced with higher oil temperatures, particularly related to liquids like fuel, water and coolant as they are more likely to vaporise, especially if the oil temperature is higher than their boiling temperature. The effect of reduced oil dilution on the emissions is expected to be minor; there might be a slightly positive effect if more water can be re-circulated through the crankcase ventilation system as it would help to reduce combustion temperatures and thus NO_x emissions.

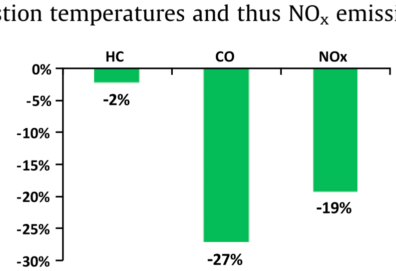

Fig. 6. Emission reductions with oil/exhaust gas heat exchanger (vehicle measurements averaged over five tests in each configuration) [36].

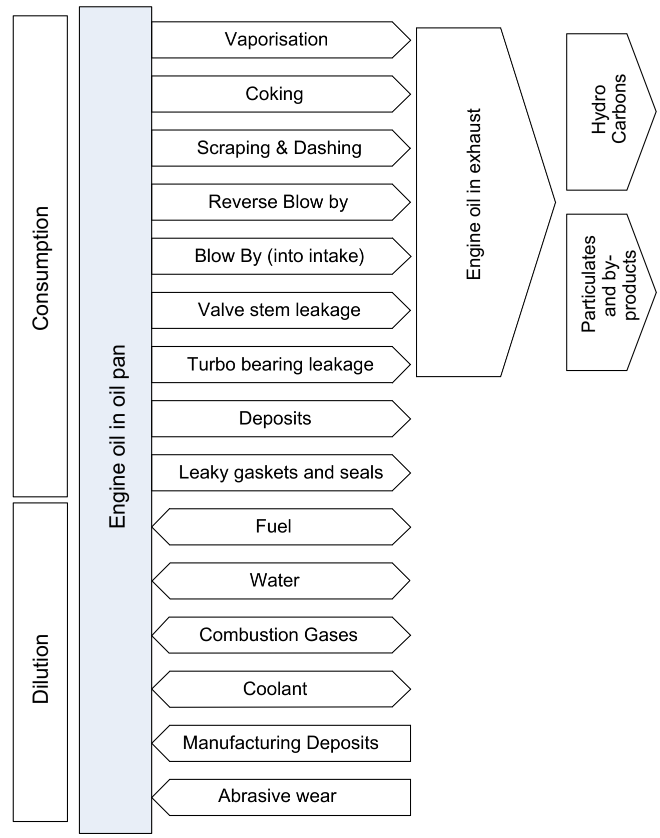

Fig. 7. Oil balance (modified from [37]).

5. New system configuration

The previous analysis in combination with the test results with the exhaust gas heat exchanger led to a new system configuration so that the heat transfer to the engine oil can be adjusted and controlled depending on certain parameters [38]. It can be maximised during operating conditions where the oil viscosity is too low to reduce friction and it can be reduced for high engine loads and speeds when the oil temperature needs to be limited to prevent coking, so that the need for an engine oil cooler can be eliminated.

Another advantage of this configuration is that the thermal masses of different oil sections are partially separated which helps to warm up the oil in the oil galleries much faster. A system schematic is presented in Fig. 8. Similar to some variable oil pumps the system has an oil bypass controlled by a valve. The difference is that it is an external bypass that is not integrated in the oil pump. The bypass connects the cylinder head oil gallery to the inlet side of the oil pump. When the valve is opened, the oil pressure is reduced and the oil flow rate through the cylinder head is increased. The configuration in Fig. 8 also shows an exhaust gas/oil heat exchanger similar to the one that was tested in [35,36], even though the system delivers benefits without such a heat exchanger. The components that are added to a standard configuration as described before are shown in red[1].

The fuel economy benefits are expected to be similar to the results discussed in [35,36]. The theoretical background is discussed in the following case study. The total heat required to warm up the oil during a NEDC test is:

$$Q_t = c_p \cdot m_o(T_{2o} - T_1) \tag{6}$$

[1] For interpretation of color in Figs. 1–11, the reader is referred to the web version of this article.

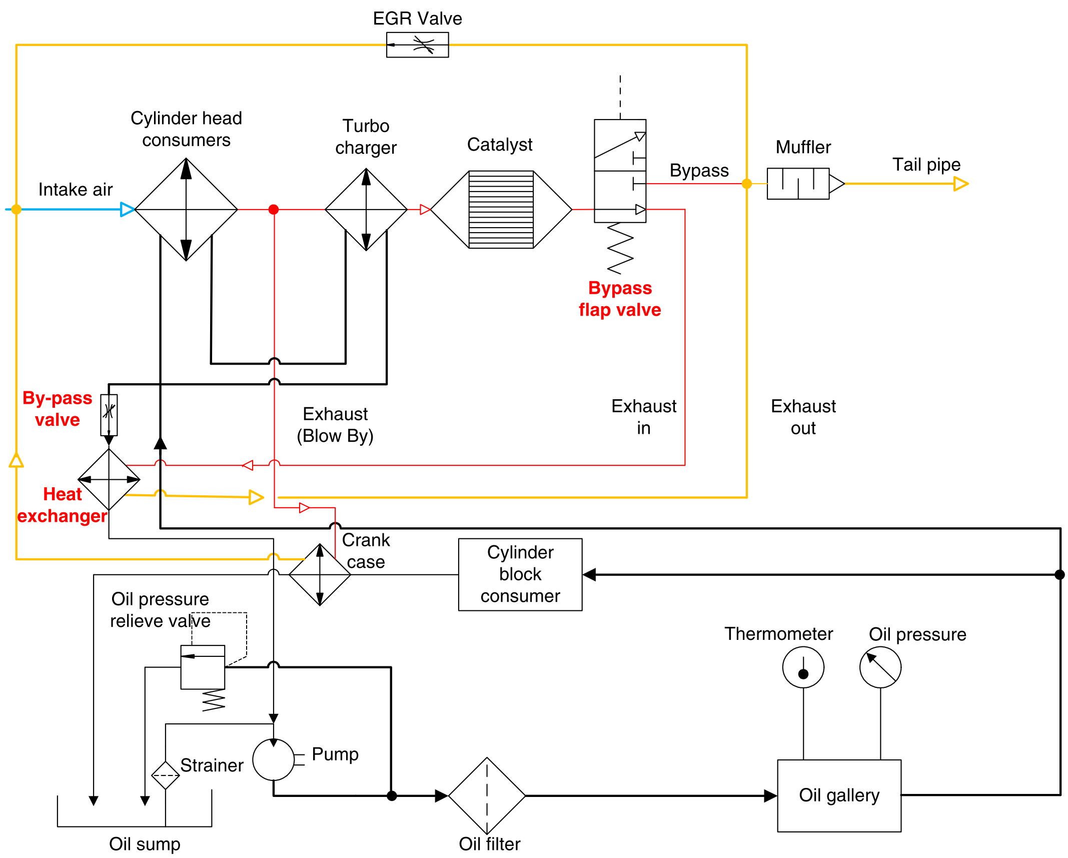

Fig. 8. New lubrication system configuration – cylinder head oil bypass with exhaust gas/oil heat exchanger.

with c_p the specific heat of oil (2.09 kJ/kg K), m_o the total oil mass in the engine, T_1 the oil temperature at test start (22 °C) and T_{2o} the original oil temperature at the end of drive cycle (82 °C). The heat rate transferred into the oil $\dot{Q} = Q_t/\Delta t$ is considered to be constant based on the oil temperature graph from Fig. 7. The ratio of the bypass mass flow divided by the total mass flow is defined as x so with the oil mass of the bypass m_B the bypass oil temperature at the end of the drive cycle can be calculated as

$$T_{2B} = T_1 + \frac{x \cdot Q \cdot \Delta t}{c_p \cdot m_B} \quad (7)$$

For the oil sump mass m_S the temperature at the end of the drive cycle is

$$T_{2S} = T_1 + \frac{(1-x) \cdot Q\Delta t}{c_p \cdot m_s} \quad (8)$$

So for the mixed oil that enters the oil pump the temperature at the end of the test is

$$T_{2m} = x \cdot T_{2B} + (1-x) \cdot T_{2S} \quad (9)$$

and with $y = m_B/m_S$ being the ratio of oil mass in engine circuit (including bypass) divided through the oil mass in sump, in this case $y = 0.1$ was determined, so

$$T_{2m} = T_1 + \frac{Q \cdot \delta t}{c_p \cdot m} \cdot \left(\left(\frac{1 - 2x + x^2}{1-y} \right) + \frac{x^2}{y} \right) \quad (10)$$

5.1. Oil pressure map

The oil pressure for different temperatures was measured to evaluate how much excessive pressure – and therefore also flow – is available during a cold start (Fig. 9). Therefore such an oil pressure map was an essential requirement for the following simulations. The oil pressures were recorded with a Setra 206 250PSIG pressure transducer. Compared to the results that were presented in Section 4, these measurements were recorded in dynamic acceleration tests with the transmission in neutral, but with the same test vehicle. In this way the required oil pressure map could be established in a very quick way without the need for expensive engine dynamometer installations and testing. The tests were started with the lowest oil temperature. Three accelerations from idle across the speed range were performed at each oil temperature. After each of these three tests the engine was left idling until the next oil temperature level was recorded and subsequent accelerations were performed.

The difference between the minimum pressure that is required for a reliable engine function, in this case 100 kPa, and the

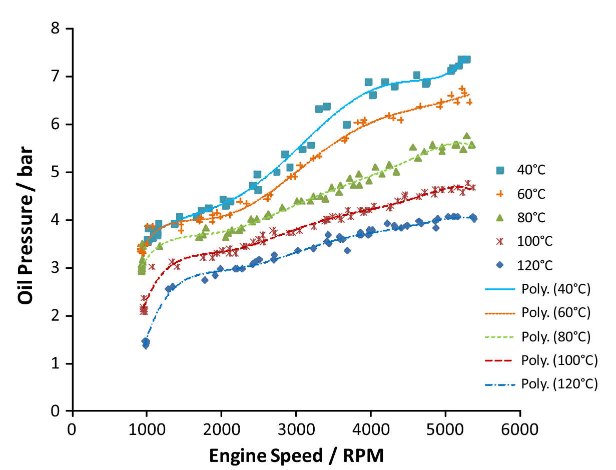

Fig. 9. Engine oil pressure map for different temperatures and engine speeds.

theoretical maximum pressure (without opening of pressure relieve valve) could be used to increase the flow rate through the cylinder head.

It can be seen that at the lowest engine speed the difference between the minimum functional pressure of 100 kPa and the maximum pressure at 40 °C is 300 kPa. This represents excessive flow availability of 75% through the cylinder head by-pass. At the maximum engine speed of 5300 RPM the pressure is 740 kPa, representing an excessive flow availability of 86%. The actual excessive flow availability at low temperatures and low engine revolutions will be even larger because at temperatures below 80 °C the oil pressure is already higher than the opening pressure of the pressure relieve valve so that the relieve valve releases a significant amount of oil flow (Fig. 1). The pressure curves at 100 °C and 120 °C initially show a steep increase of pressure but after 1500 RPM the curve flattens out and the slope of the pressure increase over engine speed is reduced by over 50% due to the opening of the pressure relieve valve. At temperatures below 60 °C this significant reduction of the slope is missing because the pressure relieve valve is already open at 1000 RPM.

5.2. Warm up simulation results

A very simple simulation was conducted for a new system configuration that only includes the bypass through the cylinder head but without the heat exchanger. For different by-pass flow ratios x the mixed oil temperature after 300 s was calculated and compared with the calculated temperature for the original configuration (Fig. 10). At a bypass flow ratio of 70% the oil temperature increased by 64 °C and for a ratio of 0.5 the temperature gain is still 29 °C. At a lower ratio of 0.2 however, the temperature improvement is less than 2 °C. That means to achieve a good warm up benefit, the bypass flow ratio needs to be as large as possible. If the ratio is too small, no measureable fuel economy benefits can be realised.

In Fig. 11 a similar simulation result is shown for the duration of the complete NEDC test for a constant by-pass flow ratio of 0.5. After 700 s an oil temperature of 130 °C is achieved. After that no further temperature increase is desired so the bypass is closed and the maximum oil temperature is capped. The temperature increase is much larger than with the tested HE so the resulting fuel consumption reductions are expected to be at least similar or potentially larger than with the tested heat exchanger.

An important assumption in the above simulation is that the heat rate transferred into the oil is independent of the location of the flow. For example if $x = 0.2$, 20% of the total heat rate transferred into the total oil mass is transferred through the bypass.

The accuracy of the simulation will depend on the detailed design of the bypass. The bypass should be integrated directly into the cylinder head and cylinder block, as close as possible to the combustion chamber. If the bypass is further away from the combustion chamber than the normal oil galleries, for example implemented through an external pipe, the warm up benefits will be lower than predicted. In reality, a percentage of the heat flow into

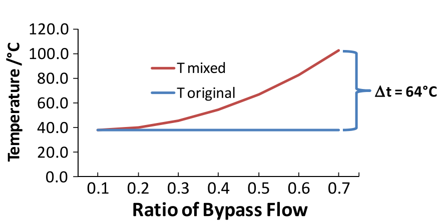

Fig. 10. Mixed oil temperatures after 300 s with new oil bypass as a function of by-pass flow ratio x compared with baseline configuration without bypass.

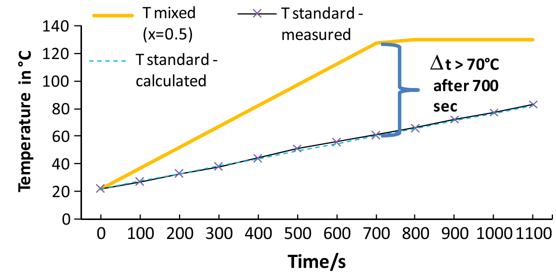

Fig. 11. Mixed oil temperature with new bypass.

the oil is generated by the friction in the bearings themselves and through contact with the bottom side of piston. On the other hand much of that heat flow is also lost again through the heat transfer from the oil pan to the ambient air.

The maximum oil temperature of 130 °C can only be achieved, if the bypass temperature is not determined mainly by the coolant temperature. If the bypass temperature is determined mainly by the coolant temperature, only a maximum oil temperature similar to the coolant temperature can be achieved.

6. Discussion of further benefits and risks

The benefits of the new bypass configuration are very low complexity to achieve a relatively large reduction of fuel consumption. The only additional parts required are a bypass tube – that could also be integrated in the cylinder head and block if a new engine is designed – and the by-pass valve including controller and control strategy. The maximum benefits are possible if an exhaust gas heat exchanger is also installed in the new cylinder head oil bypass. Such a configuration has the advantage that the colder exhaust gases after the HE could be used as Exhaust Gas Recirculation (EGR) gas to further reduce NO_x emissions. However, if the water condensation should be utilised to a larger extend, other effects like potential corrosion and heat exchanger fouling need to be considered carefully within the design process. Because the heat exchanger is located behind the catalyst such fouling effects are expected to be smaller than for EGR coolers that extract the exhaust gases upstream of the catalyst and potential particulate filters and NO_x traps where the exhaust contains a multiple of the tailpipe emissions.

The higher oil temperature also reduces the amount of blow by gases – in particular water and fuel – that condenses in the crankcase and dilutes the engine oil. For oil temperatures above the water boiling point of 100 °C the water- and fuel condensates are directly vaporised again so they don't dilute the oil. The reduction of water content in the oil can enable longer oil change intervals and reduces wear [39]. This is particularly beneficial for vehicles with a hybrid power train that often require a reduction of oil change intervals [40] because these engines warm up less fast as they are often switched off during driving.

A potential risk associated with higher oil temperatures is oil oxidation and potentially coking. That happens during local overheating of the oil, for example in the combustion chamber where the oil in the honing cavities is in direct contact with the very hot combustion gas. Coking can also happen in turbo chargers if the oil flow is too small. For the bypass system with HE appropriate valve operating strategies should avoid potential overheating of oil.

A more serious problem could be oil leaks into or onto the hot exhaust system. For leaks into the exhaust system emissions would be affected and also components from the exhaust system like the muffler could be damaged. Leaks onto the exhaust could cause fires. Both need to be prevented carefully by applying best practice design principles like Failure Mode and Effect Analysis similar as for other oil pipes that are close to the exhaust like for the turbo charger lubrication.

On the positive side the engine performance at cold temperatures and cold start drive away will be improved a lot, sluggish cold drive is a problematic area of modern down-sized engines. The effect on the cabin warm up will depend on the system configuration. Without exhaust/oil HE there could be a negative effect as less wasted heat is available due to reduced engine load. With HE the heater performance is expected to improve.

Engine noise can also be affected depending on the detailed design solution. Some articles suggest a reduction of engine noise and particularly a reduction of the dominating orders compared to the overall noise level with the reduction of oil pressures [41], the same is expected with an external by-pass that also reduces oil pressures to a similar extend.

7. Conclusions

(1) Tests with variable oil pumps confirm only 50% of the fuel economy improvements that were predicted with computer simulations. The discrepancy can be explained because variable oil pumps reduce the heat transfer to the oil and thus reduce the oil temperature during warm up. Increasing the heat transfer to the oil therefore offers further potential to improve fuel economy.
(2) Vehicle tests showed fuel consumption reductions of over 7% if the engine oil is heated directly with an exhaust gas heat exchanger. The percentage improvements for the cold extra urban drive cycle and urban drive cycle were of similar magnitude indicating a similar real world benefit.
(3) The cold/hot factor does not describe the maximum fuel economy improvement due to warm up because the oil temperature can be elevated well above the oil temperatures that are relevant for the determination of the cold/hot factor.
(4) A maximum heat exchanger heating power of 18 kW was measured during a NEDC test. Water condensation further increases the heat exchanger effectiveness. The condensation of exhaust water offers a benefit that is in the same range as reducing the exhaust gas temperature. A maximum available condensation enthalpy of 14 kW has been determined for the vehicle tested.
(5) Emission reduction is another important benefit of warming up the oil faster, particularly for CO and NO_x. For HC emissions the benefits are partially offset by increased oil vaporisation in the combustion chamber.
(6) A new system with an external engine oil bypass has been presented that is less complex compared to an exhaust gas heat exchanger. Simulations have demonstrated that such a system is capable of large increases in oil temperature and therefore similar fuel consumption reductions as for the heat exchanger system that was tested. For a standard engine the percentage of oil mass flow that can be directed into such a new bypass can account to up to 80% of the total oil flow at NEDC test start temperature.
(7) Potential risks of such a system are oil degradation or leaks. These risks are already mastered in engine applications with turbo chargers.
(8) Warming up the engine oil offers further advantages like better oil quality leading to extended oil change intervals or reduced wear, improved engine performance, faster cabin warm up and reduced noise. The extend of these benefits will depend on specific configurations of the new system and the engine to be modified which offers a broad range of opportunities for further research.

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