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Thermal Management Opportunity on Lubricant Oil to Reduce Fuel Consumption and Emissions of a Light-Duty Diesel Engine

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Abstract. The high viscosity of the lubricant oil in internal combustion engines at cold starts is responsible for poor friction reduction and inadequate thermal stabilization of metallic masses and represents a major bottleneck in the efforts to reduce specific fuel consumption and pollutant emissions. Consequently, the possibility of integrating techniques for proper thermal management of the lubricant oil on internal combustion engines is of utmost importance to both homologation and daily on-road operation. Main options for reducing the warm-up time for the engine lubricant are the upgrade of the engine cooling and lubricating circuits, dedicated heating, different flow management of the oil/coolant heat exchanger, a renewed design of the oil sump or a thermal storage section to increase the oil temperature in the early phases of the warm up. The paper presents a new opportunity, using a hot storage medium to heat up the oil in the early phase of a driving cycle. A certain quantity of hot water, so, is stored in a tank, which can be used to warm up the lubricating oil when the engine is started up. The heating of this service water can be done by using exhaust gas heat, which is always wasted in the atmosphere. The activity is realized on an IVECO 3.0 L light-duty diesel engine, during a transient cycle (NEDC) on a dynamometric test bench. The benefits in terms of both fuel consumption and CO_2 emissions reduction. The characterization of the backpressure associated with an eventual additional heat exchangers and the more complex layout of the oil circuit is assessed, as well as the transient effects produced by the faster oil warm-up and oil-coolant interaction on the engine thermal stabilization.

1 Introduction

Air quality protection, primarily via the reduction in greenhouse gases emissions, is a priority for legislation worldwide. Emissions regulations currently in place for the transportation sector drive present and future technological development and conjugate the objectives of fuel saving to those of a higher vehicle – namely, engine – efficiency and lower overall pollutant emissions [1].

The expectation on highly performing vehicles rebounds on the efforts to develop and implement improved testing procedures on a global basis, which led, among others, to the introduction of the World Harmonized Light Vehicle Test Procedure – WLTP for engine homologation and the setting of a new, more demanding RDE (Real Driving Emission) cycle for major pollutants (i.e., HC, CO, NOx, and PM) assessment. Globally, the transportation sector accounts for 30-35 % of the overall CO_2 emissions. Regarding the EU scenario, car and van fleets and heavy-duty vehicles (e.g., buses and trucks) are responsible for an average of 15% and 5% of the total CO_2 emissions, respectively [1, 2].

Consequently, binding emissions limitations apply to light-duty passenger cars and commercial vehicles, resulting in the requirement for Euro 6 vehicles to comply with the 95 gCO_2/km threshold. In addition, ambitious goals have been set by EU policymakers, for the short run, on both new cars and vans, with an expected 37,5% and 31% emissions reduction, with respect to the 2021 value, to be achieved by 2030 [2, 3].

The complexity and multi-faceted regulatory scenario on the one hand, and the increasingly more stringent requirements on transportation technology, still largely dependent on the traditional internal combustion engine, whose improvement is already at the asymptote, suggest the importance that even the slightest gain on fuel consumption could have on the market success of both diesel and gasoline engines in the years to come.

An extensive literature proves that the largest share of CO_2 emissions from internal combustion engines takes place at cold starts and in operating conditions similar to those in the first three quarters of engine homologation cycles, i.e., when both the power unit and the aftertreatment are inefficiently operated, due to torque and engine RPM variation from the steady state [4, 5]. In such conditions, the fast warm-up of lube oil up to its optimum operating temperature is addressed as an effective option for reducing the friction mean effective pressure (FMEP), particularly between the rings and cylinder liner and in the crankshaft bearings [6, 7, 8]

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Since the engine parts the oil gets in contact with, are still not thermally stabilized, the oil heating by direct contact (conduction and convection with the metallic masses) is not possible. Moreover, any attempt to harvest thermal energy for lubricant heating purposes in such phases is limited by the occurrence of misfiring and incomplete combustions, plus the inefficient charge motion inside the cylinder [9, 10, 11] that results in a heat release rate below expected theoretical values. As a matter of fact, in these phases, the available thermal power from the combustion process must be employed entirely to raise the temperature of the combustion chamber and the piston and to set more favorable conditions to subsequent combustions and the only chance to warm the oil up is by feeding it with thermal energy in a dedicated on-board heating circuit, that would necessarily integrate the serial ones for lubricant circulation and cooling.

The implementation of enhanced techniques for proper thermal management and fast warm-up of the lube oil can definitely rely on the definition of lumped parameters models of the engine, providing preliminary results in terms of the relative advantage of an electrical pump over a mechanical one for coolant circulation [12, 13], pressure drops and overall performance of the modified lubricant circuit [14, 15], optimum oil sump design for fast thermal stabilization [16], on-board thermal storage for lubricant heating purposes [17, 18]. An increasing interest is currently drawn by the option of integrating on-board thermal storage units to standard layouts [19, 20], to feed thermal energy to the lubricant at cold starts, when no other heat source is ready and immediately exploitable.

Aside from components down-sizing and down-weighting requirements to meet, the idea of harvesting thermal energy for optimum oil thermal management calls for the proper plant set-up, whose characteristics mostly depend on (i) the heat transfer fluid [21, 22], (ii) the option of direct electrical heating of the storage unit [23] and the room offered by exhaust gases [24], (iii) the pressure losses in the oil heating circuit and (iv) power absorption for circulation, based on the selected pump technology [25]. An in-depth experimental activity is then crucial to properly account for the complex phenomenology controlling the engine operation during initial transients [26, 27] and to allow the assessment of both the actual thermal availability to oil and the benefit in terms of friction reduction, fuel saving and enhanced emissions performance.

The present paper deals with the experimental assessment of the option of oil heating via on-board stored thermal energy by water. The heat is harvested in a shell and tube heat exchanger from exhaust gases during previous driving cycles, provided to the water, and stored inside a thermally insulated unit, until the next engine start. The experimental validation is carried out on a light-duty diesel engine on a dynamometric test bench, to fully reproduce the NEDC homologation cycle and close-to-normal driving conditions. The experimental dataset supports the definition of a proper control strategy for fuel saving and emissions reduction.

2 Materials and Methods

An IVECO F1C 3.0L turbocharged diesel engine is operated according to the NEDC homologation cycle on a dynamic test bench. In standard setups, the lubricant temperature is controlled by phasing the oil adduction to a flat plate heat exchanger, right downstream the pump and upstream the engine (Figure 1): the cooling fluid draws thermal energy from the lube oil during the whole driving cycle, independently on the actual need for oil cooling (i.e., on the oil actual operating temperature).

Table 1. reference engine and vehicle specs.

Displaced volume	2998 cc		Number of cylinders	4 in line
Stroke	104 mm		Maximum power	130 kW @ 3250 RPM
Bore	95.8 mm		Maximum torque	400 Nm @ 2000 RPM
Connecting rod	255 mm		Supercharging	Variable Geometry Turbine
Compression ratio	19:1		CO_2 rated emissions	260 g/km
Number of valves	16			

Since the cooling system is designed for proper oil thermal stabilization during normal and continuous engine operation, it is oversized for oil thermal management at cold starts. As a matter of fact, it slows down the reaching of the designed value of oil viscosity, hence increasing frictional losses. More in detail, the lubricant circuit features a chain-driven gear-type oil pump - linear characteristic with RPM and a 55 L/min maximum volumetric flowrate – that circulates oil from a 7 L sump. A thermostatic valve regulates the coolant flow to the radiator and allows to by-pass it when no heat removal on the coolant is needed.

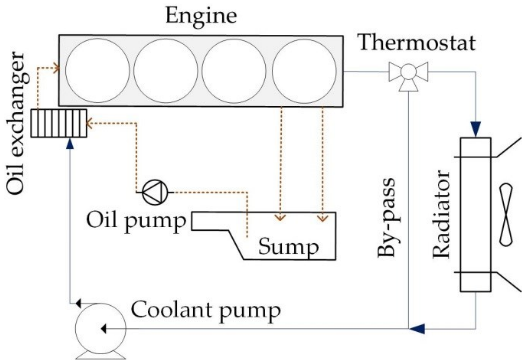
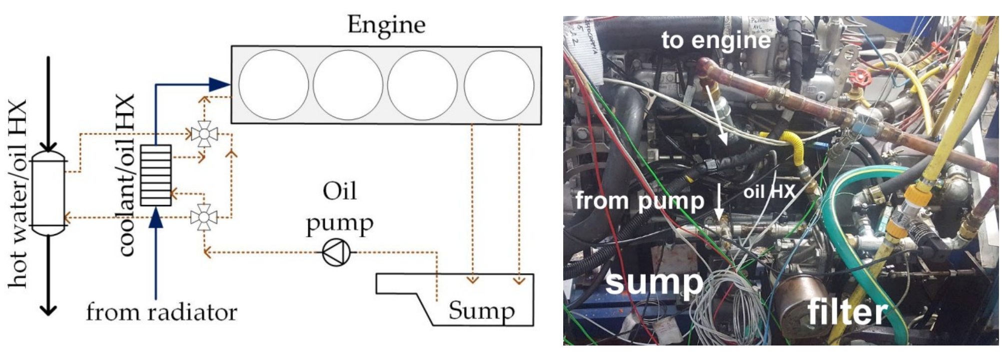


Fig. 1. Standard oil cooling circuit of the reference engine [28]

In order to accelerate the oil warm up at cold starts, an additional hot source is needed. It has been identified in a thermal storage section (5 L of hot water in the tank), which can be accumulated from a previous utilization of the engine. The work aims to use this hot source in the very early phase of the starting of the engine, in order to suddenly give the thermal energy owned by the storage to the lubricant oil. Therefore, the oil path was modified, with the addition of an oil/water heat exchanger and an exhaust heat recovery section: as the water (5 L) temperature decreases due to the cycle-by-cycle heat release to oil, the thermal power recovered from exhaust gases allows the restoring of water temperatures suitable for oil heating (Figure 2). After being filtered, the lubricant can either (i) reach or (ii) bypass the coolant heat exchanger, and (iii) engage the heat exchange with water, based on whether the oil is already at the optimum 80 °C temperature or not.

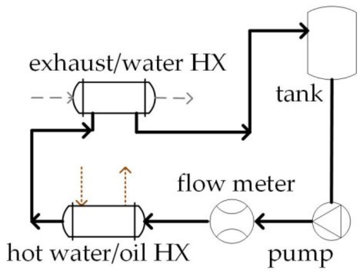


(a) (b)
Fig. 2. Modified oil circuit – Schematics (a) and picture of the bench built (b)

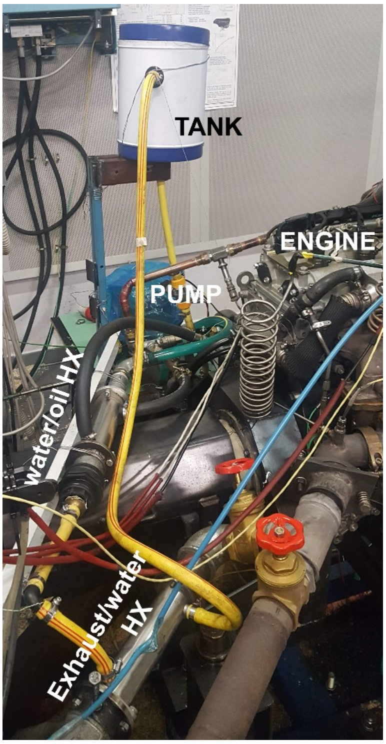
The detailed water circuit has been sketched in Figure 3 a) and built in the engine test bench as represented in Figure 3 b). The water circuit is composed of an insulated tank, which is the main thermal storage device, a circulating pump, the heat exchanger between hot water and lubricating oil, and the second heat exchanger between the exhaust gases and the service water, which ensures an upper thermal source to the water, in order to enhance the warming up of the oil and, mainly, to reheat the water when the thermal regimentation is reached and, so, making available the hot thermal storage for next starting. Temperatures are measured in each node of the circuit, while a turbine flow meter is used to assess the water flow rate. A thorough thermal and hydraulic characterization of the oil circuit was possible via the monitoring of both temperatures and pressures, according to the schematics in Figure 2 and specifications – types of sensors and uncertainties - in Table 2.

Table 2. Main sensors specifications

Variable	Sensor type	Accuracy
Temperature	K-type thermocouple	±2,2 °C
Pressure	GEMS2200	±0,1 bar
Volumetric flowrate	Turbine-type sensor	± 0.5%
Fuel consumption	AVL 733s fuel balance	±0,05 % measured value



(a)



(b)

Fig. 3: auxiliary water circuit for hot thermal storage exploitation: (a) layout sketch and (b) experimental arrangement

3 Experimental Characterization

The reconstruction of the thermal profiles was performed, during a whole NEDC homologation cycle, on the water stream (for lubricant heating), the engine coolant, and the oil itself. The analysis of the temperature profiles for water in each side of the devices which compose the water circuit (Figure 4) allows the assessment of the actual feasibility of oil heating at cold starts: the initial temperature of the water is close to 70°C, which is reasonable considering the thermal dispersion between the previous run of the engine and the actual start, although the tank is insulated.

Exhaust gases flowing through the heat exchanger seem to give additional aid to fulfill the purpose of warming up the water, prior it flows back into the storage unit and eventually is employed for oil heating, already after 50 s. The water starts at 68-70 °C, loses 10 °C during the first 60 s, when exhaust gases do not provide any thermal back up to the water itself and metallic masses within the heat exchanger are cold. As the engine warms up and the combustion temperature inside the cylinders increases, exhaust gases start re-integrating the water thermal power, lost to the surroundings and lubricant oil, and a continuously growing trend is appreciated until the water pump is switched off, little after 800 s, when oil temperature and water one are similar.

The temperature fluctuations at the tank inlet are dampened, due to the intrinsic larger thermal inertia of the water inside the storage unit: the temperature of water at the tank outlet is always 10-15 °C lower than the one at the tank inlet and never exceeds 90 °C. Due to the initial transient, a temperature difference-driven heat exchange between the water and the oil starts at 100 s and continues until the water is fed by the pump.

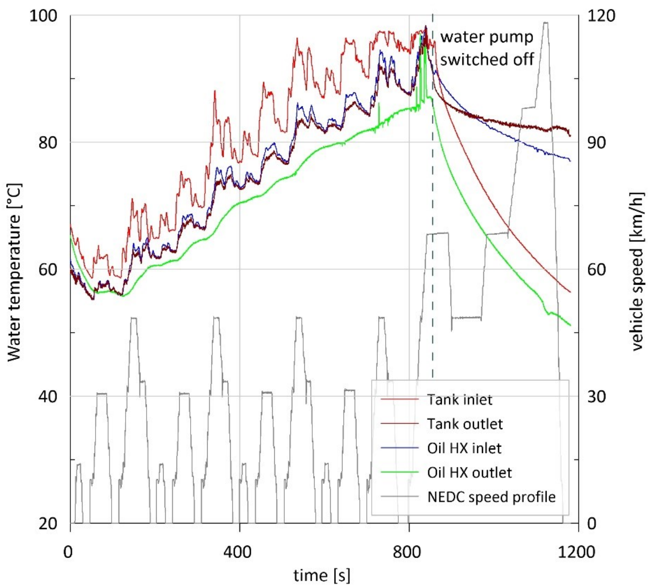


Fig.4. Water thermal profiles.

The engine is always presented with a warmer lubricant (Figure 5): a mean 5 °C difference is appreciated at the engine inlet, between the situation where the oil is warmed up in the external circuit, with respect to the baseline, where no initial heating takes place. It is worth observing that the temperature difference tends to stabilize after 100 s, suggesting that the advantage in terms of increased oil temperature is appreciated well before the first urban cycle is over.

The same trend is detected on the temperature values of the lubricant oil in the sump, in spite of a lower temperature offset, mostly due to the mixing of the oil with the large mass stored inside the sump: here the temperature shift does not exceed 3 °C during the whole test run.

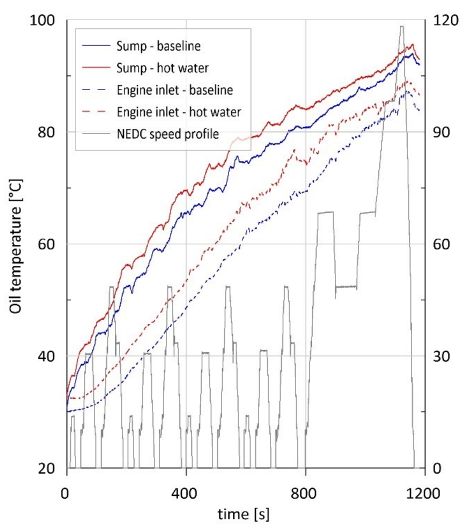


Fig.5. Lubricant oil thermal profile.

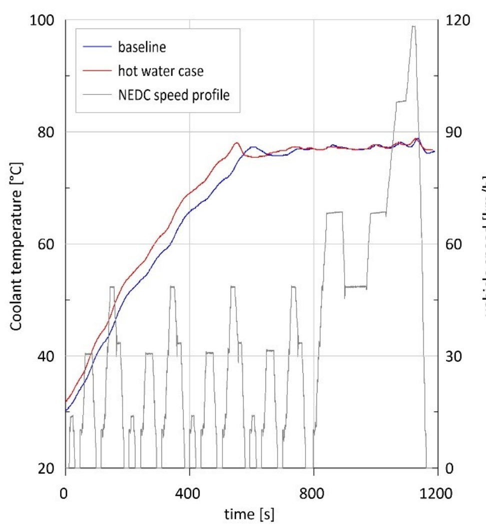


Fig.6. Coolant temperature.

The analysis of the coolant temperature (Figure 6) suggests that, as long as the water warms up the oil, the temperature profile shifts upward (+ 4 °C) with respect to the baseline. At about 600 s, the coolant stabilizes around temperature values typical of normal operation (75-78 °C) and no difference is appreciated between the baseline situation and the case where hot water is employed. An important assessment is that related to the oil pressure along the external circuit. In fact, when the additional heat exchanger between oil and hot water is introduced, the pressure of the oil at the engine inlet should be guaranteed in order to feed the valve, the hydraulic rockers, the chains, and their hydraulic tensioners. Fig.7a shows the pressures of the oil at the engine inlet in both cases, demonstrating only a little suffering (about 0.2 bar lower, Fig. 7b) in the case of "long circuit", i.e. hot water case, with the additional heat exchanger. Higher differences are experienced in the pump outlet (but with positive values, Fig. 7b), which must guarantee the right pressure at engine inlet and, so, is high enough to overcome the pressure drop along the longer external circuit. When the hot water circuit is switched off and heat exchangers bypassed, the pressure levels became similar and the difference between "hot water case" and "baseline" approach a value close to zero (Fig. 7b).

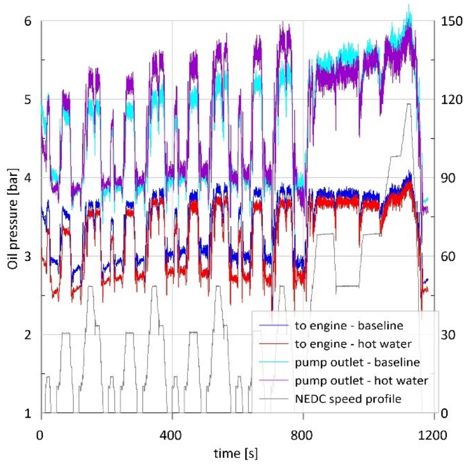


Fig. 7a: oil pressures in the lubricating circuits

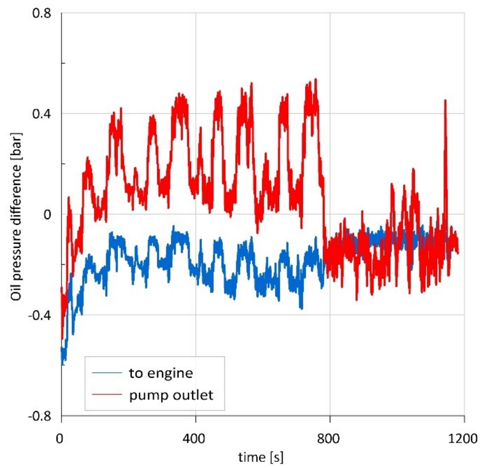


Fig. 7b: oil pressures differences between "hot water" and "baseline" cases

4 Results and Discussion

Figure 8 reports the thermal power (P_{th}) profiles at the heat exchangers – both water, exhaust, and oil – and the pump of the water circuit, calculated as in eq.1 for each i-th component.

$$P_{th,i} = m \cdot c \cdot (T_{out,i} - T_{in,i}) \qquad \text{eq.1}$$

Where m is the measured mass flow rate, c is the specific heat of the water (4.187 kJ/kg°C) and T represents the temperature of the fluid. The water (red line) feeds thermal energy to the oil – 3 kW mean value – with peaks up to 6 kW around 400 s and 700 s, corresponding to the acceleration of the vehicle during urban phases. Negative values in Figure 6 demonstrate that the auxiliary water is giving thermal power to the oil, heating it. The exhaust-to-water heat exchange (blue line) is responsible for a higher thermal power transferred to water, and it plays a crucial role in the enhancement of the warming up of the lubricant [29].

As expected, as the thermal stabilization goes on and the engine temperature rises, the available thermal energy to water increases: during the first 400 s, the average thermal power provided to water never exceeds 7 kW. As the exhaust gases reach higher characteristic temperatures, an average 10 kW thermal power is available to water, with peaks above 15 kW, around 700 s. The pump (green line) contributes to the water heating, with characteristic values of thermal power lower than 1 kW, up until it is switched off. The balance closes with thermal dispersion through the environment, which could be reduced by a higher level of insulation of each component and piping.

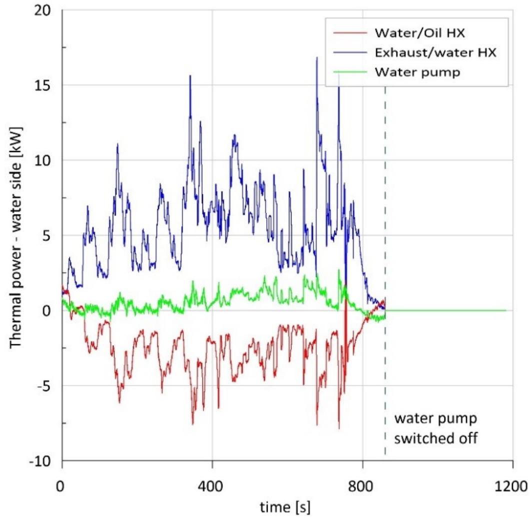


Fig. 8. Thermal power on the water side

The measured fuel consumption is in Figure 9: some advantage in the hundredth liter range is appreciated, despite the additional backpressure for the engine to overcome. The benefits resulted is approximatively 1.7% of fuel saved and so similar value for CO_2 emission reduction. Considering the engine and vehicle considered, this accounts for about 4.4 gCO_2/km. The lower frictional losses, associated with a high performing oil, due to the faster thermal stabilization to characteristic design operating temperatures, seem to overcome the drawbacks – in terms of additional work required to the engine - of having an external circuit for oil heating.

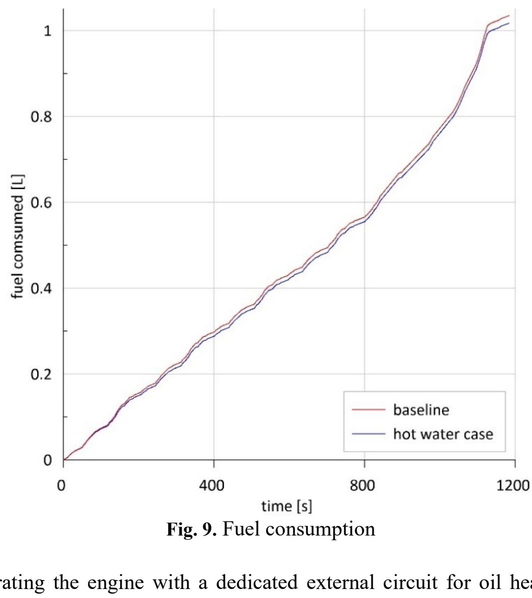


Fig. 9. Fuel consumption

The downside of integrating the engine with a dedicated external circuit for oil heating, plus accounting for an intermediate fluid – water in the present case – is the backpressure the engine needs to overcome. Particularly the heat exchanger on the exhaust path induces highly concentrated pressure drops [30]: in absence of the shell and tube heat exchanger, the engine backpressure stays always between 7 mbar and 25 mbar (Figure 10), between engine start and 800 s. The heat exchanger-induced pressure losses result in a minimum 20 mbar engine backpressure, with peaks around and above 100 mbar. Once the exhaust line no longer feeds the heat exchanging section, the pressure profile matches the regular one, according to the control strategy, which bypasses the exhaust heat exchanger on the gas side when hot water pump is switched off. Hence, extra backpressure resulted is in the maximum range of approximatively 80 mbar, which is still manageable by the engine without an excessive pumping losses increase and, so, with a negligible induced overconsumption (<1%, [31])

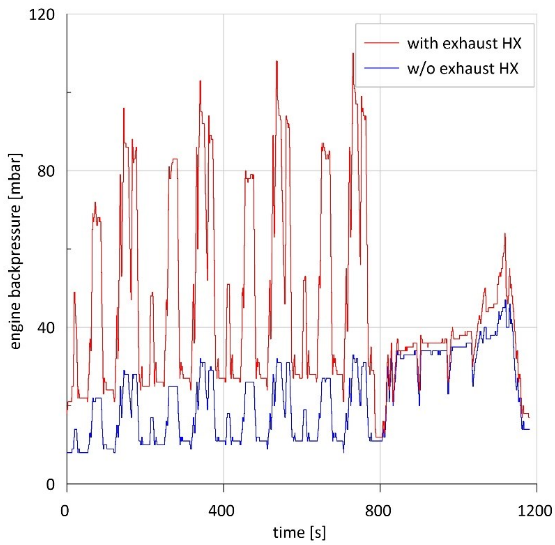


Fig. 10. engine backpressure at turbine outlet, in presence and absence of exhaust heat exchanger

5 Conclusions

In this paper, a new technological way to improve engine oil warm up time has been conceived, realized, and studied. A hot thermal storage was considered: it should be sufficiently hot at the starting up of the engine and the vehicle, in order to transfer its thermal energy to the lubricating oil in the very early phase of a driving cycle.

The hot thermal storage considered is a 5 L tank of water, which can be warmed by a previous utilization of the engine and kept at a certain temperature level by correct insulation of the tank itself. Therefore, an auxiliary hot water circuit is built, in which a heat exchanger between the hot water and the engine lubricant oil is used to heat up the oil, and a second heat exchanger is placed in the exhaust line, in order to favor the re-heating of the water, when the thermal regimentation of the engine is achieved, and to sustain the temperature level of the water during the warming up of the oil.

Thanks to this secondary hot water circuit, an average 3-4 kW of thermal power has been given to the oil during the four urban phases of the NEDC homologation cycle. Then the hot water circuit is deactivated, since thermal stabilization is close to being reached and oil temperatures approaches to the hot water ones. A reduction of the oil warm up time is achieved in about 100 s, which leads to a faster warm up also of the engine itself and, in particular, of the coolant (-60s). The final results, in terms of fuel consumption reduction and emissions saving are noteworthy: -1.7% of fuel used during an NEDC homologation cycle and a corresponding -4.4 g/km of CO_2 emissions reduction.

Finally, the technology proposed is simple enough to be introduced on a vehicle. The layout of the hot water circuit can be designed to fit in the engine bay and the initial temperature of the tank can be furtherly optimized by proper insulation. The mechanical power absorbed by the circulating pump can be negligible, since the circuit is at atmospheric pressure and the hydraulic head is, so, very limited. The backpressure at the exhaust can be controlled with suitable heat exchangers technology and flows management.

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