Available online at www.sciencedirect.com

ScienceDirect

Energy Procedia 82 (2015) 1 – 8

Energy Procedia

ATI 2015 - 70th Conference of the ATI Engineering Association

Effects of oil warm up acceleration on the fuel consumption of reciprocating internal combustion engines

Roberto Cipollone[a], Davide Di Battista[a,]*, Marco Mauriello[a]

[a]*Department of Industrial and Information Engineering andEconomics - University of L'Aquila – Italy,*

Abstract

The homologation cycle of vehicles for private passenger transportation or for light duty applications considers a cold start from ambient temperature. The most part of harmful substances (≈ 60-65 %) are produced during the thermal engine stabilization which occurs in the very of the driving cycle. This strongly influences also engine efficiency, i.e. fuel consumption. The more recent commitments on CO_2, therefore, reinforce the concept of reducing warm up time encountering it in the low carbon engine technologies. Due to this importance, engine thermal management has been the subject of a huge interest opening the way to new components, technologies and control strategies. This regards not only the coolant fluid, which undoubtedly influences engine warm up, but also the lubricant:an its heating acceleration produces much faster benefits.. The purpose of this paper is to assess the effect of a faster oil heating during the homologation cycle on the fuel consumption. An experimental campaign has been done on an 3L Iveco F1C engine mounted on a dynamometer test bench operated in order to reproduce the NEDC. The engine OEM has been characterized and the effect of the oil temperature has been studied according to: (a) an external heat source which brings the oil at its stabilized temperature value before engine start, (b) an internal heat source represented by the exhaust gases which almost immediately reach a temperature value able to heat-up the oil. The effects on CO_2 emissions during the cycle have been evaluated. The benefits are noteworthy and justify some oil circuit modifications.

Keywords: internal combustion engine, oil thermal management, emissions, fuel consumption

1. Introduction

In response to actual global sensibleness on environmental issues, internal combustion engines lives a technological revolution in order to to achieve more efficient energy conversion and reduce pollutants emissions.In this regards, European Community and others international governments set important

---

* Corresponding author. Tel.:+39-0862434319.
E-mail address: davide.dibattista@univaq.it

doi:10.1016/j.egypro.2015.11.870

targets on pollutants (HC, CO, NOx and PM) and CO_2 emissions from passenger cars and light duty vehicles. European manufacturer are on-the-road to achieve the 95 g/km target (which is an average fleet value, based on the vehicle mass) on the carbon dioxide emissions: they reached a strong reduction in recent years, touching an average value of about 130 g/km in 2013 (Figure 1), but the yearly reduction can not keep this trend without a technological breakthrough. In fact, before EC first proposal of a regulation on the transportation emissions (2007) the yearly reduction was about 1%, and only after this milestone the average reduction reaches 3-4% per year (Figure 2).

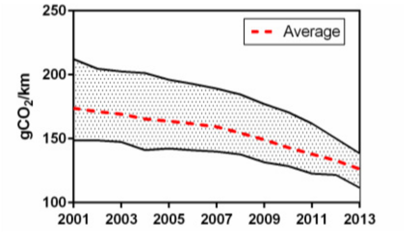

Figure 1: CO_2 emissions trend of European carmakers [1]

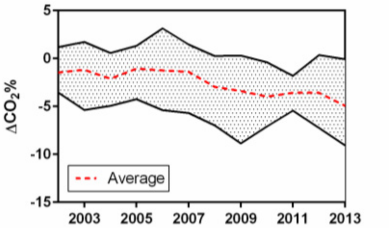

Figure 2: year reduction of CO_2 emissions of EU[1]

This CO_2 evaluation passes through homologation procedure according to specific driving cycles, which considers the cold start of the engine. Unfortunately, in a typical homologation cycle the engine reaches its thermal steady state very close to the end of the cycle and lubrication oil does not even reach a regime temperature. When these fluids and engine components are below their steady temperature, the thermal efficiency of an ICE is significantly reduced as it happens during the cold-start phase: combustion quality is poor and mechanical losses are high. A fast engine warm up decreases the cold phase in the homologation cycle and produces sensible benefits in terms of fuel saving and pollutant reduction [2, 3].

The technics to reduce warm up phase falls into the terms of “thermal management”: this subjectaims to lead as faster as possible metallic components to their stabilized temperatures, instead of waiting that the engine does it “naturally”, according to traditional technological solutions. Engine cooling strategies have recently evolved with the use of electronic components in the cooling system. Electric pumps, thermostats, and fans play major roles in decreasing the losses [4]. Use of switchable pumps reduces the parasitic losses from auxiliary components and allows better control of the engine temperature for the best engine efficiency [5, 6]. In the same way, new cooling layouts [7-9] and advanced thermostat [10] were considered. Light metal alloys were also extensively adopted with higher thermal conductivity [11].

Lubricating oil plays an important role in overall engine efficiency: a faster oil warm up would improve mechanical efficiency which is particularly low during homologation cycle when engine is cold. A decrease in lubricant warm up time, however, is more critical to be reached: oil warm up rate is about three times lower than the coolant one [12] and it does not reach its optimal temperature during a traditional homologation drive cycle [13]. During the warm up phase, oil viscosity is much higher than at regime condition and, so the engine FMEP can be till 25% higher during cold phase with respect to the hot stabilized phase [14, 15]. Only recently, the possibility to reduce the warm-up time of the transmission and engine oils have been introduced [16, 17]: using exhaust heat to increase their temperature, the fuel economy improvement is up to 4.4% on regulatory cycles.

In this paper, the Authors present an experimental campaign aimed to reduce oil warm up time. Tests were done on an Iveco F1C engine test bench, in steady working points and during a NEDC cycle. The engine OEM has been characterized and the effect of the oil temperature has been studied according to: (a) an external heat source which brings the oil at its stabilized temperature value before engine start, (b) an internal heat source represented by the exhaust gases which almost immediately reach a temperature able to speed up oil temperature. The effects on CO_2 emissions and fuel consumption during the cycle have been evaluated.

2. Description of test bench, measuring equipment and engine lubrication system

The experimental activity has been done on a dynamic engine test-bench AVL APA100. The engine tested is the IVECO F1C 3.0 L. It is a turbocharged direct-injection diesel engine equipped to comply the regulatory constraints EURO IV-V. Fuel consumption has been measured with fuel gravimetric balance (AVL 733s) with an uncertainty of 0.12%; BOSCH HFM 5 has been used to measure air mass flow rate with an uncertainty of 3%. The specifications of the engine were listed in Table 1. It was equipped by a series of sensors adapted to detect relevant variables, in order to monitor the behavior of the engine during the experimental tests.

Table 1. Specifications of the diesel engine used (IVECO F1C)

Parameter	Value
Displaced volume	2998 cc
Stroke	104 mm
Bore	95.8 mm
Connecting Rod	255 mm
Compression ratio	19:1
Number of Valves	16
Number of cylinders	4 in line
Maximum power	130 kW @ 3250 RPM
Maximum torque	400 Nm @ 2000 RPM

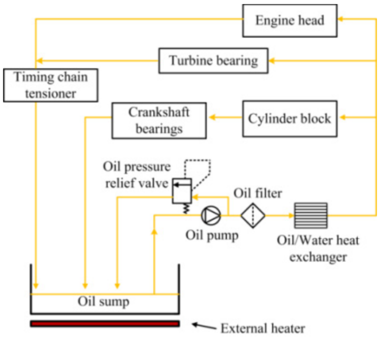

Fig. 3 disposition scheme of the heated oil test bed

The lubrication of the diesel engine Iveco F1C tested is characterized by forced circulation and it is composed by an oil gear pump with apressure regulation valve incorporated in it, a water/oil heat exchanger and a filter with double filtration and a security valve (Fig. 3). During operation, the lubricating oil is sucked up from the sump through the oil gear pump and it crosses the heat exchanger where it is continuously refrigerated. The oil, then, prosecutes through the oil filter and goes towards the engine through three main branches: one reaches the engine head and lubricates the camshafts, the rockers and the valves and feeds the hydraulic tappets; the second branch lubricates the shaft of the turbocompressor and the third branch lubricates the engine block and feeds the crankshaft bearings. The oil, after the lubrication cycle, returns to the sump thanks to piping and galleries. Moreover, the lubricating oil feed the tensioners of the timing chain. A sketch of the lubrication circuit is reported in Fig. 3. Oil pump is able to give a flow rate from 13.5 to 55 L/min, according to its revolution speed (from 860 to 4480 RPM); the relief valve guarantees a maximum pressure of 6 bar. Eventual new other devices on the oil circuit must not lead to overcome this value, in order to keep the right lubrication of the engine and its components. Hence, an external heat source (1500 W)was used to bring the oil at its stabilized temperature value (Fig. 3). This is an electric flexible resistance (a pad heater with power density of 12.5 W/cm^2) attached to the outside of the oil pan.The pad is constituted by a foil engraved (1.4 mm) and has a special adhesive on one side which ensures maximum adherence to the oil pan and correct heat exchanged.

3. Results and benefits

The engine OEM has been characterized and the effect of the oil temperature on the fuel consumption has been studied following to approaches: (a) oil is heated by an external heat source before engine start; (b) oil is heated by the exhaust gases which represent an effective heat source almost immediately available. Two preliminary trials with stationary operating points (1800 RPM -60 Nm and 2500 RPM - 100 Nm) were considered: fuel consumption, water and oil temperatures were monitored. Three tests were done and specified as “cold engine and cold oil”, “cold engine and hot oil” and “hot engine and hot oil”. Table 2 summarizes main testing data and results. The tests were performed in different days with a maximum day-by-day variation of 3°C on temperatures. They were executed till to the stabilization of the oil and water temperature. For the engine considered, it is almost 900 s.

Table 2. Comparison on stationary operating points (duration 900 s)

	1800 RPM – 60 Nm			2500 RPM - 100 Nm		
	Cold engine and cold oil	Cold engine and hot oil	Hot engine and hot oil	Cold engine and cold oil	Cold engine and hot oil	Hot engine and hot oil
Water starting temperature [°C]	20,9	24,2	64,6	21,5	24,1	73,1
Sump oil temperature [°C]	24,4	77	68,3	24	80	78,4
Average fuel consumption [kg/h]	3,95	3,89	3,65	7,7	7,5	7,4

Data in Table 2 show that heating the oil before engine start produces an interesting reduction of the fuel consumption rate (averaged on the 900 s); at 60 Nm@1800 RPM this reduction is 1.5%, while it is 2.6% at 100 Nm@2500 RPM. This was due to the faster oil warm up when the engine is run at higher load.Engine efficiency and CO_2 emissions follow similarly. Making reference to the first engine operating point, Figure 4A shows the instantaneous fuel consumption reduction for the three cases under study. The grey area represents the differences and how it became nil when engine is warmed. The behaviour when the engine is started already hot shows a fuel consumption almost constant during time. A similar result is showed in Figure 5A referred at 100 Nm @ 2500 RPM. Figure 4B reports oil and cooling fluid temperatures during time.

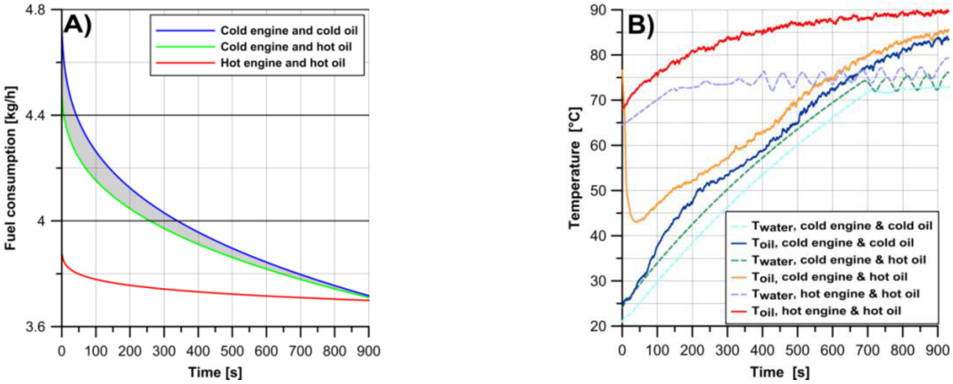

Fig. 4. Operating point 1800 rpm-60 Nm: A) fuel consumption analysis, B) temperature trends

Coolant follows a dynamic which is close to a first order model, a discontinuity happens when the thermostat opens towards the radiator, producing temperatures oscillations. Oil temperature seems to have a faster warming up at the beginning of the test, but the continuously cooling by the water delays the reaching higher temperatures. A simply bypass of the cooling fluid across the oil heat exchanger would significantly reduce the oil warm up time. The negative effect of the cooling fluids more clear observing the sudden decrease of the oil temperature when the engine is started in the “hot oil” mode (Figures 4B and 5B): oil temperature approaches coolant temperature and, then, it is governed by the cooling fluid. Figure 5A shows also that 900 s are almost enough to lead the two fluid at their stabilized states. Figure 5B reinforces the concept previously discussed: the higher engine load consents a faster warm up of both oil and coolant. The cooling fluid governs the warm up of the lubricant oil, which would be faster if the oil/water heat exchanger would be by-pass in warming up phase.

Summarizing, in both steady cases presented in Fig. 4 and Fig. 5 the benefits related to a hotter oil are evident: fuel consumption is significant lower when the oil is hotter, in particular at higher engine load; the temperature behaviour shows, however, the effect of the oil-coolant exchanger after the very start of the engine: oil temperature falls down and follows coolant temperature. This is a detrimental aspect, because it nullifies the effect of the heating. This issue can be solved bypassing with a thermostatic valve the oil/water heat exchanger.

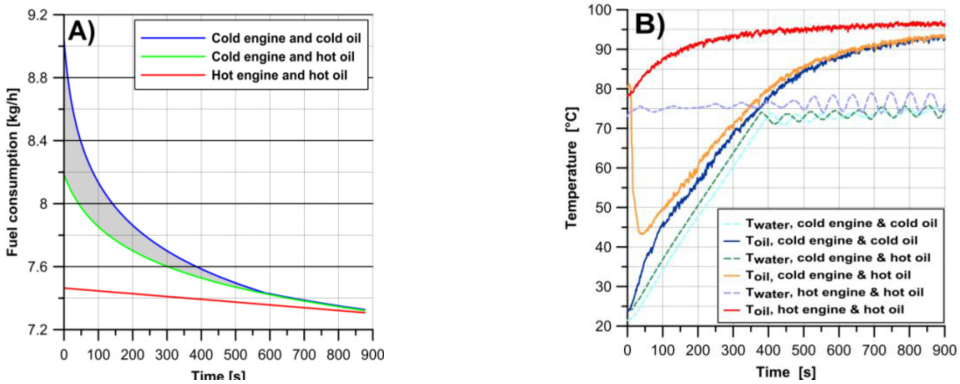

Fig. 5. Operating point 2500 rpm-100 Nm: A) fuel consumption analysis, B) temperature trends

In order to asses real benefits of a warmer oil, the F1C engine was considered when it propels a passenger car or a light duty vehicle. A NEDC was completed on the dynamometer, adapting the vehicle data in order to determinate engine load and speed which would compete to the given vehicle. Table 3 shows a comparison between the three different engine-oil modes, during a traditional NEDC homologation cycle: obviously, the better performances are reached when engine and oil are fully warmed, but, similarly, simply heating the oil, the fuel consumption reduction is equal to 2.8%. From the instantaneous values of the fuel consumption and the engine efficiency, the fuel economy improvements during the “urban” and “extra-urban” parts of the NEDC have been evaluated. Making reference to a comparison between “cold engine-cold oil” and “cold engine-hot oil” modes, An interesting reduction of 3% of fuel consumption is achieved in urban part and 2.3 % in extra-urban cycle.

Fig. 6 shows temperature trends of water and oil in the different engine and oil modes. Most interesting results are obtained considering the “cold engine and cold oil” mode:

1) thermostat opens at almost 1000 s, very close to the end of the NEDC. Oil, during NEDC, does not reach the stabilized thermal state. Oil warm up is delayed by the slower dynamic of the coolant that refrigerates it;
2) the coolant temperature is not affected by the thermal state of the oil (Fig. 6);
3) in the case of “hot engine and hot oil” cooling fluid stays close to the steady value; when the extra-urban sub-cycle starts, the thermostat opens and produces an oscillating temperature; oil temperature, on the other hand, has increases till to a steady value very close to the value reached in the case of “cold engine and cold oil (temperature regime is not dependent on initial state).

Table 3. Comparison on NEDC fuel consumption and CO_2 emissions

	Cold engine and cold oil	Cold engine and hot oil	Hot engine and hot oil
Water initial temperature [°C]	26,1	25,9	71,3
Sump oil initial temperature [°C]	27,4	75	73,1
Average fuel consumption [kg/h]	2,51	2,44	2,21
Average engine efficiency [%]	16,84	17,32	19,12
Emissions [gCO_2/km]	241,5	234,7	212,6
Fuel consumption reduction [%]	-	2,8	12
CO_2 Emissionreduction [gCO_2/km]	-	6,8	28,9

Fig. 7 shows the cumulative values of CO_2 emissions, directly calculated from the fuel consumption: starting the drive cycle with a heated oil produces a CO_2 saving equal to 76.8 g, which could be compared to the OEM emissions equal to 2.73 kg (2.8% reduction). This means a saving of 6.8 gCO_2/km.

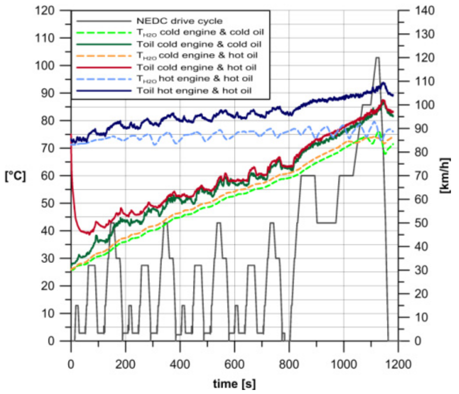

Fig. 6: Temperature profiles during NEDC in the different starting conditions considered

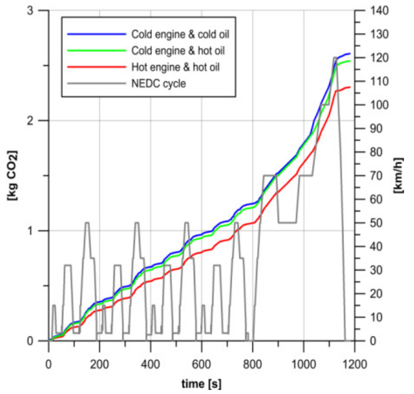

Fig. 7: CO_2 cumulative emissions during NEDC

The positive effects obtained by heating the oil with an external hot source invited the Authors to proceed with an internal heat source represented by the exhaust gases [16]. Fig. 8 shows the exhaust gas temperature during NEDC: it is evident how fast the gas increase its temperature with respect to oil and water. The thermal power available to speed up oil temperature has been fully characterized when gas flow rate is known (Fig. 8).

In order to improve testing flexibility, an engine oil circuit has been built according to Fig. 9. An oil receiver (representing engine oil sump) feeds a fully controlled pump which operates under the same conditions of the F1C engine. An electric heater is controlled in order to exchange the same heating rate produced by the F1C engine: this is clearly demonstrated by Fig. 10, where the oil heating rate follows the same trend of green curve in Fig. 6. Of course, this behaviour embed the real effects of the oil/water heat exchanger inside the F1C. A shell and tube heat exchanger has been mounted in the engine oil reproduction test bench (Fig. 9) in order to assess the effects of the oil heating due to the exhaust gases. For sake of simplicity, the circuit was operated at averaged values (oil flow rate and pressure drops are 25 L/min and 2 bar; exhaust gas flow rate and temperature are 170 kg/h and 120 °C, Fig. 8). The benefits on the oil warm up are reported in Fig. 10. If 80 °C is considered as set point temperature, the warm up is shortened of a factor of about 60 %. Oil dynamics so fast that a feedback control is needed in order to avoid temperature swing above the desired value.

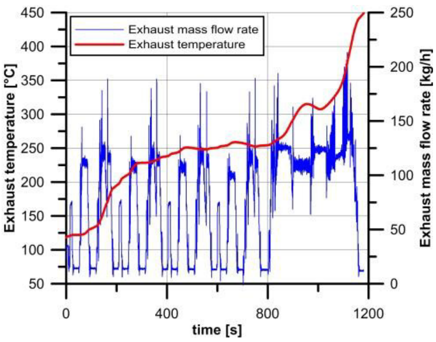

Fig. 8: Exhaust temperature and mass flow rate of tested engine during NEDC

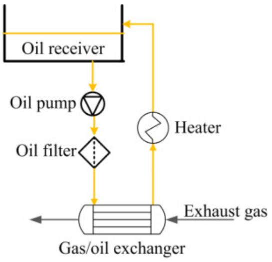

Fig. 9: test bench for the reproduction of the engine oil circuit

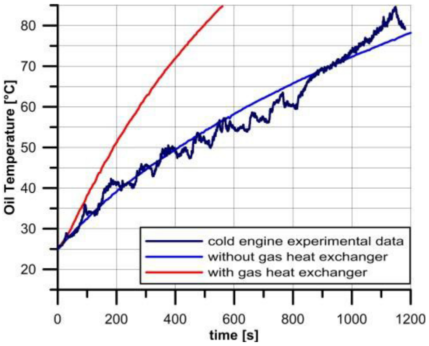

Fig. 10: oil warm up temperatures: experimental engine data comparedwith ones of the circuit on Fig.9.

A final remark concerning the overall energy balance produced by the oil heating: oil amount in the engine tested is about 7 kg and the energy to bring it from 25°C to 80°C is about 750 kJ. In terms of fuel energy (LHV), this datum corresponds to about 17 grams, which, on their turn, correspond to a direct emission of 53 grams of CO_2.

The fuel saving realized by heating the lubricant oil thanks to an engine internal source, however, is not straightforward to be assessed. If external electrical energy (or specifically produced by the engine) is used, additional CO_2 emissions should be accounted: the energy conversion chain almost doubles the emission value. This invites to consider the oil warm up acceleration only supported by an energy recovery in the engine. Exhaust gases are the most promising energy source and requires components widely used and assessed. The use of electrical energy (stored in the battery) is possible only if this energy is the result of a recovery: regenerative breaking or electricity produced by ORC based units fed by exhaust gases, turbo-compounding, etc. are useful technologies to do this. In all these cases, the benefits that have been measured are net and no other accounting is needed. In any cases (using internal or external heat source), the by pass of the oil heat exchanger is mandatory considering that it has a detrimental effect on the oil warming up.

4. Conclusions

In this paper, the effect of the acceleration of the engine oil warm-up has been treated in terms of fuel saving and, consequently, CO_2 emissions. An experimental campaign has been realized on a IVECO F1C 3.0L engine to evaluate all relevant quantities required to measure the oil and cooling fluid dynamics and the effects on fuel consumption. Two steady engine operating points have been set (60 Nm@1800 RPM and 100Nm@2500 RPM). The NEDC has ben also implemented as reference homologation procedure.

In order to subdivide the effects due to the oil warm up from that due to the cooling fluid, different testing modes have been conceived. In the engine tested, if a NEDC is started having the oil already hot, a reduction in fuel consumption of about 2.8 % and CO_2 emissions of 6.8 g/km have been measured. This result is limited by the presence of the oil heat exchanger which produces a detrimental effects being the cooling fluid colder than the oil. If the NEDC is run with the cold oil (as in real situations), the oil temperature dynamics is faster than that of the oil and the presence of the heat exchanger slows down the oil temperature rise.

An oil layout modification has been tested in order to deal about the effect of an oil heating realized by the exhaust gases: they demonstrate to have almost immediately the enthalpy required to support the oil warm up. Oil is heated in a shell and tube heat exchanger before it enters inside the engine when it was running a NEDC. This has been done reproducing the engine real oil dynamics (during a NEDC) in a more flexible test bench. A significant oil warm up time reduction has been measured: in normal conditions, oil warm up ends almost at the end of the NEDC. If the exhaust gases are used to support oil warm up, a reduction of about 60% in the warm up time is reached anticipating significantly the hot oil engine mode.

References

[1] International Council on Clean Transportation Europe - European Vehicle Market Statistics Pocketbook 2014

[2] Will F., Boretti A., A new method to warm up lubricating oil to improve the fuel efficiency during cold start, SAE International J. Engines Paper, 2011-01-0318, Published 04/12/2011, do1 10 427112011-01-031.

[3] F. Chiara, M Canova, A review of energy consumption, management, and recovery in automotive systems, with considerations of future trends, Proceedings of the Institution of Mechanical Engineers, Part D: Journal of Automobile Engineering June 2013 vol. 227 no. 6 914-936

[4] Pang H. H., Brace C. J., Review of engine cooling technologies for modern engines, Proceedings of the Institution of Mechanical Engineers, Part D: Journal of Automobile Engineering November 1, 2004 vol. 218 no. 11 1209-1215,

[5] Andrew Roberts, Richard Brooks, Philip Shipway, Internal combustion engine cold-start efficiency: A review of the problem, causes and potential solutions, Energy Conversion and Management, Volume 82, June 2014, Pages 327-350.

[6] Roberto Cipollone, Davide Di Battista, Sliding vane rotary pump in engine cooling system for automotive sector, Applied Thermal Engineering, Volume 76, 5 February 2015, Pages 157-166,

[7] R. Cipollone, D. Di Battista, A. Gualtieri - Head and Block Split Cooling in ICE – IFAC Workshop on Engine and Powertrain Control Simulation and Modeling (ECOSM12), Les Rencontres Scientifiques d'IFP Energies nouvelles, Rueil-Malmaison, France 23 - 25 Oct. 2012.

[8] Hyungmook Kang, HyunchulAhn, Kyoungdoug Min, Smart cooling system of the double loop coolant structure with engine thermal management modeling, Applied Thermal Engineering, Volume 79, 25 March 2015, Pages 124-131, ISSN 1359-4311,

[9] R.Cipollone, D. Di Battista - Performances and opportunities of an engine cooling system with a double circuit at two temperature levels – SAE 2012 World Congress – SAE tecnichal paper 2012-01-638.

[10] Bing Zhou, XuDong Lan, XiangHua Xu, XinGang Liang, Numerical model and control strategies for the advanced thermal management system of diesel engine, Applied Thermal Engineering, Volume 82, 5 May 2015, Pages 368-379, ISSN 1359-4311,

[11] MuhammetCerit, Mehmet Coban, Temperature and thermal stress analyses of a ceramic-coated aluminum alloy piston used in a diesel engine, International Journal of Thermal Sciences, Volume 77, March 2014, Pages 11-18, ISSN 1290-0729,

[12] Andrews G, Ounzain A, Li H, Bell M, Tate J, Ropkins K. The use of a water/lube oil heat exchanger and enhanced cooling water heating to increase water and lube oil heating rates in passenger cars for reduced fuel consumption and CO2 emissions during cold start. SAE technical paper 2007-01-2067; 2007.

[13] Trapy JD, Damiral P. An investigation of lubricating system warm-up for the improvement of cold start efficiency and emissions of SI automotive engines. SAE technical paper 902089; 1990.

[14] Burke RD, Brace CJ, Hawley JG, Pegg I. Review of the systems analysis of interactions between the thermal, lubricant, and combustion processes of diesel engines. Proc Inst MechEng Part D: J AutomobEng 2010;224(5):681–704.

[15] Shayler PJ, Allen AJ, Leong DKW, Pegg I, Brown AJ, Dumenil J-C. Characterising lubricating oil viscosity to describe effects on engine friction. SAE technical paper 2007-01-1984; 2007.

[16] Will F. Comparison of advanced waste heat recovery systems with a novel oil heating system, International Journal of Automotive Engineering 4 (2013) 33-40, Presented at the JSAE Annual congress on May 25, 2012.

[17] Laboe K and Canova M. Powertrain waste heat recovery – a systems approach to maximize drivetrain efficiency. In: ASME Internal Combustion Engine Division spring technical conference, Torino, Italy, 6–9 May 2012. ASME. ICES2012-81160

Biography

Davide Di Battista graduated as Mechanical Engineering in 2010 (Italy). He took his PhD in March, 2014 at the Department of Mechanical Engineering, Energy and Management of the University of L'Aquila working on engine thermal management, turbocharging, hybrid and electric vehicles, energy recovery, energetic territorial planning. He is also involved in several technical projects (European projects, national projects and manufactures collaborations). At the moment, he is a contract researcher and has published about 25 scientific papers.