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Experimental assessment of engine charge air cooling by a refrigeration unit

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Abstract

Following the increasing awareness on the global warming, international governments have set up severe targets on CO2 emission in transportation sector: the overcoming of these limits produces a fine which directly influences the market value of the vehicle. Moreover, concerning the traditional pollutants, they still remain targeted by future EURO6(b-c-d) limits.
Charge air cooling, in turbocharged diesel engines, is widely used to increase air density, improve cylinder filling and, definitively, engine volumetric efficiency. This is usually done through a heat exchanger fed by environmental air positioned after the charge air compression: the cooling is strictly related to vehicle speed, dedicated radiator positioning, engine operating point and environmental temperature. All these factors lead to an in-cylinder intake air temperature in the 40-70°C range.
A refrigerating unit, featuring an evaporator, suitably placed inside the intake manifold, could provide an additional cooling. Such an option is not so difficult to be implemented considering that a conditioning unit is already present on board for cabin comfort. This unit often is over-designed and frequently under-employed.
In this paper an evaporator was placed on the intake line of a turbocharged diesel engine, available on a test bench, and the effects of the under-cooling of the charge air have been experimentally assessed. The evaporator is fed by an air refrigeration unit present on vehicle board for cabin conditioning. Fuel consumption saving has been observed as well as a sensible pollutants reduction, taking obviously into account the mechanical power required by the compressor.

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Keywords: engine, intake air, efficiency

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1. Introduction

The successful implementation of CO_2 reduction measures in internal combustion engines calls for an improvement of the complete vehicle system's efficiency, by either implementing new on-board components or developing technologies based on innovative engine set-up.

Intake air cooling has been receiving growing attention in the last decades, as one of the most effective ways to enhance cylinder filling and to favor a regular combustion process within the cylinder, with obvious advantages, among others, on fuel use and emissions reduction. Traditional Engine performances (torque, power, fun to drive, etc.) increase as well.

Nowadays, the most used technology to cool charge air is thanks an air cooled heat exchanger placed in the intake line just after the compressor (in turbocharged engine). A commonly accepted and proven technology is to use water to cool the intake air, with benefits in engine efficiency [1] and also coolant warm-up time reduction [2]. More recently research focuses on the waste heat recovery on exhaust gases to feed cooling-dedicated components, e.g. air cooling below ambient temperature using the heat from exhaust gases to feed a jet ejector cooling device [3, 4]. Other approaches address the potential improvement achievable through the charge cooling via a heat pump [5], whereas Guhr [6] analyzed the potential of a mechanically driven compression chiller, whose feasibility should be checked against current trend towards engine down-sizing. Among alternative concepts for an additional charge cooling, some studies present an engine layout featuring an exhaust-driven cooling system, based on sorption [7, 8, 9, 10], with a multi-stage air cooling as well as heat pumps driven by the engine coolant water [11, 12].

Major issues to address with reference to these solutions are system scalability (i.e. components packaging, down-sizing and down-weighting) and integration with present engines configuration: the benefit associated with the waste heat recovery should be compared to the detrimental effect of a higher engine backpressure (the heat exchanger on the exhaust tail induces, [13]) as well as the additional fuel consumption to drive a compression chiller, which is not necessarily compensated by the benefit of charge-cooling, especially in presence of unsteady operating regimes for the engine. Sorption systems suffer from layout complexity, as they are characterized by low power densities and typically require many stages to achieve the expected cooling benefit on the air. Moreover, in presence of cooling media other than water or air, piping and pressure induced leakages represent a major criticality both in a design phase and during engine operation. Weight increase due to these technologies definitively represents an adverse result which increases propulsion power with respect to the original equipment.

As known, the on board conditioning unit for cabin heating and cooling in present vehicles is often over-sized and under-employed which suggests the opportunity to use it for intake air cooling. Moreover, it can be integrated in a double circuit cooling system [14], achieving better cooling performances if a water cooled condenser is used [15]. This eventuality would lead to an enhanced engine operation with no additional on-board components, i.e. no drawbacks in terms of engine layout complexity and weight increase.

The paper goes deep inside the experimental activity carried out on charge-air under-cooling, through an evaporator, featuring R134a as cooling medium, placed on the intake line of a turbocharged diesel engine. The novelty of the application called for a preliminary assessment of the best components and engine lay-out, as the one providing the least penalty to engine operation, in terms of additional fuel consumption and emissions and the highest gain in terms of intake air cooling. Once completed, such an analysis pointed out main limitations for each plant set-up and gave a clear indication on how to maximize the temperature reduction on the inlet air, with no or little detrimental effect on other parameters controlling engine operation, such as air mass flowrate, manifold temperature and intake pressure.

2. Experimental setup and preliminary testing

The experimental activity has been carried out on a IVECO F1C test bed engine. This is a 3.0L turbocharged diesel engine, with a charge air cooler between the compressor outlet and the intake manifold. In test bench, this intercooler is fed by tap water, which reproduces the cooling effect of the external air in the vehicle. The under-cooling (below the environmental temperature) of the charge air is realized thanks to a chiller: an evaporator fed by refrigerant working fluid (R134a) has been inserted in the intake air-line. The chiller is also composed by a swashplate compressor, a water cooled plate type condenser and a thermal expansion valve. Ten engine operating steady points have been performed, varying the engine speed from 1200 to 3600 rpm, repeated at 100 and 200 Nm.

In order to feed the cylinders with a under-cooled charge air, two configurations are possible, dependently on the position the evaporator has on the intake tail, i.e. upstream the intercooler or downstream it (Figure 1). The first assessment of the effects that the air temperature reduction induces on engine fuel consumption and emissions was performed with a cooling set-up featuring a single-pass flat plate heat exchanger, in which the air flows through the plate pack and the coolant R134a within the tubes. In the upstream configuration, the air flows through the heat exchanger before entering the filter. Due to all following passages through components on the intake line – markedly the filter and the compressor, with its non-adiabatic compression – the air temperature experiences an increase with respect to its value at the evaporator outlet. When the no-cooling condition is the reference, the cooling effect can be assessed on the temperature difference at the engine intake manifold: with a 100 Nm load, it ranges between 1 K and 4.5 K, whereas at 200 Nm load, between 0.5 K and 4.7 K (Figure 2).

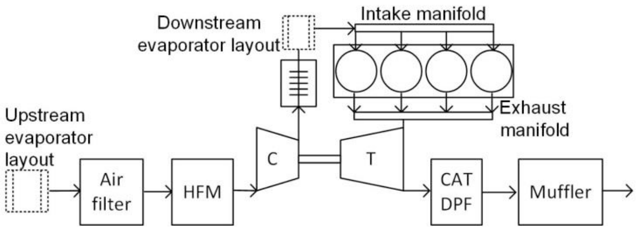

Figure 1. Upstream vs. downstream undercooling layout

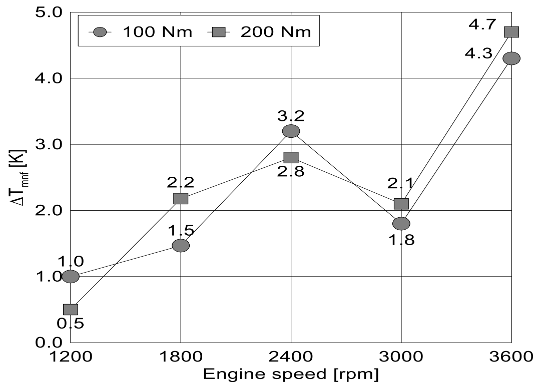

Figure 2. Air temperature – Manifold inlet (upstream undercooling layout)

As a matter of fact, in terms of mass flowrate, the whole benefit achieved at the evaporator is entirely counterbalanced difference in air mass flow rate between cooled and no-cooled one are not significant (Figure 3): already at 1800 rpm, 200 Nm, air flowrate with no cooling exceeds the one with cooling of about 2.0 kg/h. Same goes at higher rpms: at 2400 rpm, the air mass flowrate with no cooling is higher than the one with cooling, with 3.0 kg/h and 1.0 kg/h differences at 100 Nm and 200 Nm load, respectively. With a 100 Nm torque, 3000 rpm air flow rate without cooling is about 3.0 kg/h higher than the one with cooling. Hence, negligible benefits occurs on the fuel side and the need to overcome pressure losses at intake leads in some operating conditions to an increased fuel consumption: a maximum 0.07 kg/h and 0.14 kg/h reduction is at 1800 rpm and 3600 rpm for 100 Nm and 200 Nm, respectively (Figure 4), that are not worthy of investigation.

The limited benefits of this upstream location is due also to the fact that the original intercooler is surely overdesigned and it vanishes the cooling previously done, bringing the intake manifold temperature at a temperature related to the external source of the intercooler itself, which is very close to the original equipment configuration one. Anyway, such a datum clearly states the need to relocate the evaporator on the intake tail, in order to achieve the maximum possible effect in terms of cooling, i.e. in terms of reduced air temperature at the manifold.

The most effective way to achieve this goal is by placing the additional heat exchanger right after the intercooler, just before the intake manifold. First issue to address is the need to minimize pressure losses, i.e. to select a heat exchanger type that can be easily integrated to the intake tail and a proper flow arrangement. Such a need represents a major constraint here, with respect to the previously investigated configuration, where no size, shape and, generally speaking, geometry limitations applied to the heat exchange section: in the upstream configuration, the evaporator could be considered an external device to the intake line, whose only function was to blow cooled air through the filter. Since the evaporator integrates now the intake tail and does not only provide it with air from the environment, its geometry needs to be optimized so that the resulting pressure drop should be minimized. A flat plate heat exchanger, as the one employed in the upstream configuration, does not seem to be the best technological solution in a downstream set-up, since, even in presence of an optimized geometry for the air passages between plates, it introduces high air leakages and important intake line geometry modifications.

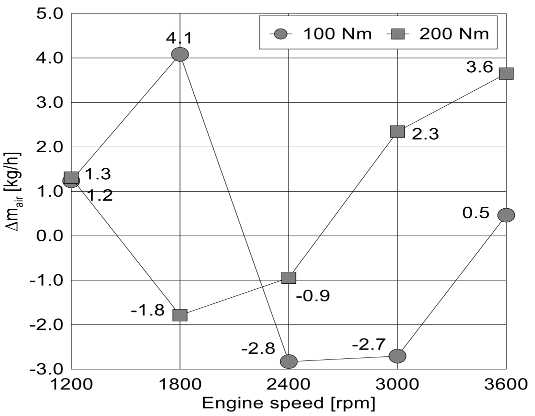

Figure 3. Air mass flowrate (upstream layout)

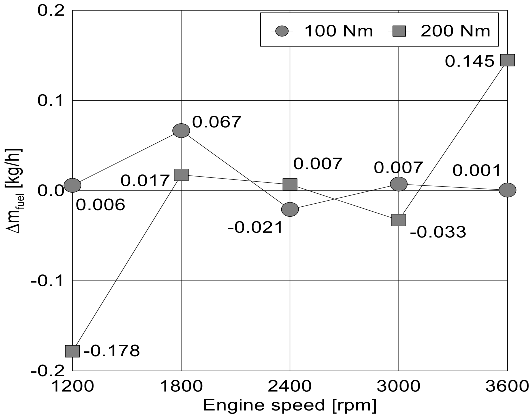

Figure 4. Fuel consumption (upstream layout)

A tubular configuration seems to be the best option and specifically, the shell (coolant) and tube (air) heat exchanger type is by far more suitable for the set-up at hand, due to the more compact geometry and the higher available heat exchange surface. Nonetheless, pressure losses are still present and increase with the length of the tubes and decreases with the diameter of the shell. Experimental evidence suggests that if the new evaporator is too long, the losses prevent the pressure from achieving the required level in the combustion chamber, even if the potential of the variable geometry turbine in the turbocharging group is fully exploited, with the throat area for stator vanes at its minimum. The layout in the analysis accounts for features a single shell and tube heat exchanger (single pass, counter-flow), with reduced air passages and consequently a lower pressure drop (Figure 5).

In conclusion, the position of the evaporator downstream the air cooling unit was recommended by the present engine configuration and temperature of the cooling medium. The fulfillment of other needs (e.g. the implementation of a dual temperature cooling system) could call for a different manifold layout, which would lead to a further upgrade of the final set up.

3. Results and discussion

The experimental activity accounted for ten operating conditions (stationary points, 120 s duration) for the engine. The adoption of the single pass, counter-flow shell and tube heat exchanger at the intercooler outlet allowed an air temperature reduction at the evaporator always above 10 K, leading to a more favorable temperature regime for the air within the manifold: differences range between 10.7 K and 15.7 K and 10.6 K and 20.0 K at 100 Nm and 200 Nm torque, respectively (Figure 6).

As in Figure 7, with a 100 Nm load, the heat removal at the evaporator has its peak at 3600 rpm, with 1.74 kW thermal power removed, whereas with a 200 Nm load, the maximum is detected at 3000 rpm (1.61 kW). Lower heat removal is realized at lower engine speed, due to the lower air flow rate requested by the engine itself. In Figure 8, the experimental pressure drops on the air side across the evaporator are showed: they are not negligible, reaching a value close to 100 mbar at about 320 kg/h, corresponding to an engine revolution speed of 3000 rpm.

It is worth observing that, since the highest thermal removal occurs at higher engine revolution speed, the maximum benefit in terms of cylinder filling and consequently fuel consumption and emissions reduction must be expected at higher rpms. Figure 9 summons these experimental evidences with up to 5% increase in the air mass flow rate within the cylinder at 200 Nm load at higher engine revolution speeds. Lower percentages occur at lower rpms, in the 2-to-3.5% range, dependently on rpms and load.

The pressure drops introduced by the evaporator in the air side have, however, detrimental effects on the intake manifold air pressure. Although the desired boost pressure is imposed by the Electronic Control Unit (ECU) through variable geometry turbine regulation [16], this turned out to be slightly decreased in the undercooling configuration

(Figure 10), probably because evaporator pressure drops are too higher to be compensated by the actual turbocharger.

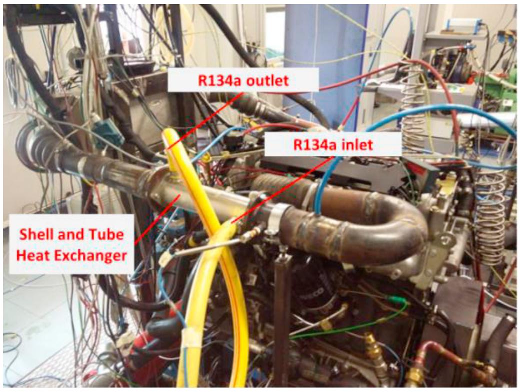

Figure 5. Shell and tube engine set-up

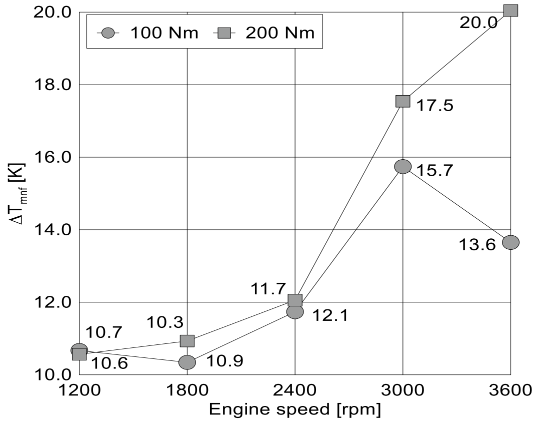

Figure 6. Manifold temperature (downstream layout)

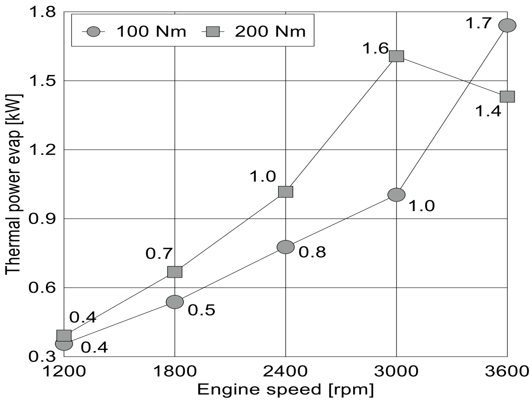

Figure 7. Thermal power exchanged in the evaporator (downstream layout)

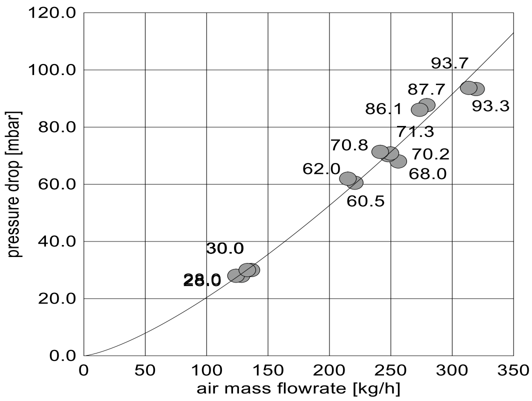

Figure 8. Shell and tube evaporator air side pressure drops

Maximum decreasing of 20 hPa is experienced, which, however, seems to have no significant effects on the cylinder filling and the turbocharger equilibrium.

Table 1 reports data coming from the experimental campaign on steady engine working points: the reduced intake manifold temperature (and consequently better cylinder filling) produces a significant average fuel consumption reduction of about 2% (gross value), with peak reduction at higher engine load of -6%. When the impact of the refrigerant compressor was accounted on fuel consumption, slightly lower values are obtained. Its contribution in terms of additional power absorption from the engine was evaluated measuring the COP (Coefficient of Performance) of the refrigeration unit: mechanical power absorption is in the range of 200-500 W and can be translated in fuel rate by calculating the global engine efficiency in each tested point. Fuel consumption reduction is also confirmed by the lower percentage of acceleration pedal showed in Table 1: having tested the engine at constant torque, thanks to the lower intake temperature, an inferior percentage of acceleration pedal is needed to reach the desired engine load. This average reduction is 1.8% and, in ECU control, this is translated in lower cycle-by-cycle fuel injection.

The effect in terms of primary pollutant emissions (CO, NO_x, HC and particulate matter) and CO_2 were monitored. Mean CO_2 reduction of 5.8% have been resulted, which is in accord with the fuel saving. Great reduction of CO and particulate matter (Soot) have been achieved. Average values are 8% for CO emission and 14% for Soot emission saving, which are very significant values, and they are above 10% for CO and 20% for Soot in many engine operating points. This is mainly due to the higher mass flow rate and, therefore, the higher air/fuel ratio, which allows better combustion performances.

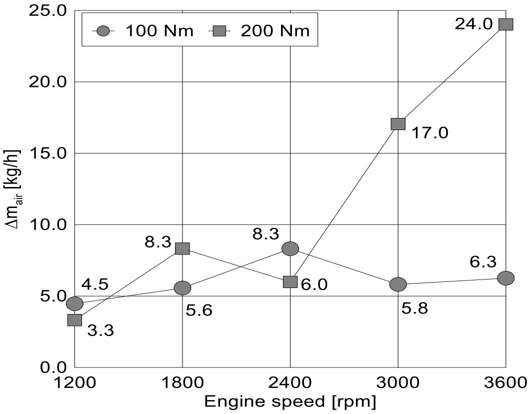

Figure 9: Air mass flowrate increase (downstream layout)

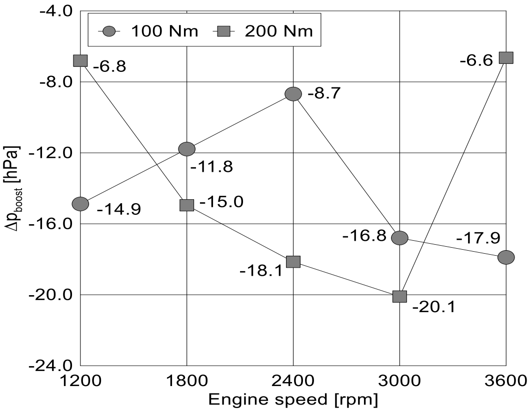

Figure 10. Intake air pressure difference between undercooled on and off cases

Also NOx resulted in an important average reduction of 9%. No significant (Exhaust Gas Recirculation) EGR differences have resulted, so, NOx reduction is obtained thanks to the lower combustion temperatures: this is demonstrated by the exhaust manifold temperature that decreases significantly in the range of 15-40 °C less than the value without air undercooling and it causes lower NOx formation. But the lower global temperatures are also responsible for the higher HC emissions, which ranges from 2 to 23% (Table 1). Concerning the primary pollutants variations, while the NOx reduction was predictable, the CO and Soot decrease and HC increase deserve some discussions. NOx formation is mainly due to the peak temperature inside the combustion chamber: the first stage which brings N to NO is extremely influenced by local reaction temperature (any reaction temperature increase of two hundred degrees doubles the NO equilibrium concentration and reduced significantly the reaction time) and this leads to a final NO_2 concentration (second final stage) also strongly reduced. CO and HC, on the contrary, follow other directions of reaction. CO represents the first oxidation stage of C and when it is available (already "produced" by a complex thermal cracking of the original fuel in the hottest combustion regions), the lower local reaction temperature decreases the chemical kinetics during its formation (its kinetics is much faster than that of the CO_2) producing an overall lower CO (and also CO_2) concentration. The final oxidation to CO_2 of a reduced CO is guaranteed by the high O_2 concentration due to the high A/F and by the reduced NOx concentration too, which definitively catches the available O_2 around the reaction zone. Concerning HC increase, the question is more complex and some further research is needed. Anyway, HC are the result of a combustion which in a certain sense "failed" due to many reasons, mainly for a not sufficient oxidation temperature and O_2 lack. HCs have not a clear kinetic chemistry during formation, being in reality a stage of transition from the original fuel to H_2O and CO_2 and characterized by a chemical composition which is not specified in terms of known molecular species. According to this, it could be expected that overall resulting effects on HC will be dominated by the lower temperature inside the combustion chamber (independently from specific reaction chemistry) which enhances the main reason of their presence. The effect on Soot should be more investigated as well: the resultant higher A/F appears to be main reasons of this reduction. A beneficial effect could be also due to an eventual entrainment of some water condensed thanks to the resulting lower air temperature after the sub-cooling.

The steady working points analysis suggests that the air cooling at cylinder intake assures a benefit in terms of fuel saving and emissions reduction – namely CO, NOx and Soot. Based on these findings, the next stage of the analysis focused on a preliminary evaluation of the effects the air cooling has on the New European Driving Cycle (NEDC). Similar results to those expected, based on the preliminary findings, came out. The air mass flowrate feeding the engine is higher when the additional air cooling is on during the whole cycle (Figure 11) and an up-shift of the profile for measured intake air can be appreciated in Figure 12. Fuel consumption saving on NEDC and emission variations are showed in Table 2 and are in accord with the steady state analysis.

Table 1. Engine operating parameters and emissions: comparison between the undercooled air mode on and without undercooling

Regime	Temperature		Fuel cons.		A/F		Acc.	EGR	Emissions				
	Intake manifold	Exhaust manifold	gross	net	off	on			CO_2	CO	NO_x	Soot	HC
rpm-Nm	°C	°C	%	%	-	-	%	%	%				
1200-100	-10.7	-14.4	-1.0	-0.7	37.4	39.4	-1.1	0.99	-4.3	-2.4	-4.0	-8.4	2.1
1200-200	-10.6	-20.0	-1.7	-0.7	25.2	26.2	-0.8	-0.15	-4.2	-7.0	-6.0	-18.0	2.5
1800-100	-10.3	-19.0	-2.1	-0.7	41.5	43.2	-2.0	-0.35	-3.6	-3.0	-10.2	-9.6	3.5
1800-200	-10.9	-25.0	-2.8	-1.6	31.0	32.7	-1.6	-0.14	-4.3	-5.5	-10.0	-19.0	3.1
2400-100	-11.7	-21.5	-1.6	-0.4	32.6	34.1	-1.5	0.27	-4.8	-3.1	-7.7	-23.0	7.3
2400-200	-12.1	-27.5	-2.1	-1.6	28.3	29.5	-1.6	-0.31	-6.2	-12.3	-5.9	-18.0	6.9
3000-100	-15.7	-29.5	-1.9	-1.1	27.2	28.4	-0.9	-0.95	-6.9	-6.5	-14.8	-21.9	12.0
3000-200	-17.5	-40.0	-4.3	-4.1	23.5	25.7	-3.8	-1.60	-8.7	-12.0	-14.5	-10.6	22.8
3600-100	-13.6	-17.0	-1.6	-1.0	30.9	32.1	-0.8	-0.17	-7.4	-14.6	-9.0	-8.0	15.0
3600-200	-20.0	-27.7	-6.0	-5.3	29.9	33.4	-2.4	-2.67	-8.0	-12.1	-4.5	-5.6	21.9

Table 2. NEDC – Emissions cumulative datum

parameter	air undercooling off	air undercooling on	Δ [%]
fuel consumption [l/100 km]	10,0	9,2	-7,96%
CO_2 [g/km]	287,0	263,7	-8,13%
NOx [g/km]	0,3	0,3	-9,93%
CO [g/km]	6,6	5,4	-18,46%
Soot [mg/km]	132,2	113,9	-13,84%
HC [g/km]	0,8	0,9	16,19%

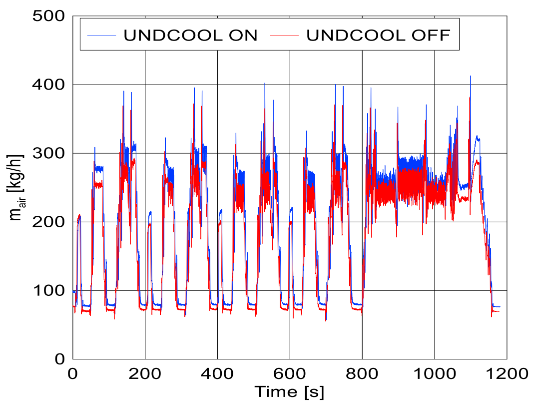

Figure 11: NEDC air mass flowrate

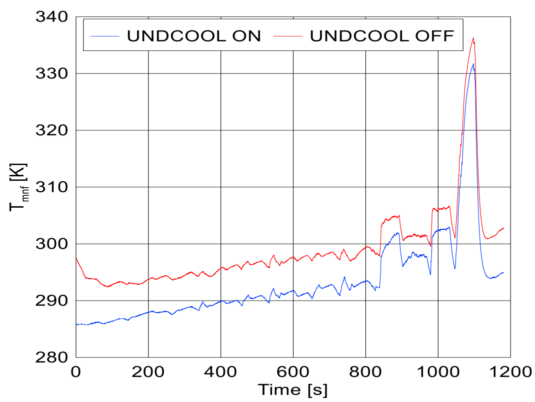

Figure 12: NEDC intake temperature regimes

4. Conclusions

The present work presents the experimental results on an improved engine layout, featuring an enhanced charge air cooling. A F1C 3L turbocharged diesel engine has been tested coupled with a chiller unit, to cool the intake air below the ambient temperature. The experimental activity, done on a dynamic test bench, accounted for two layouts, dependently on the undercooling heat exchanger (evaporator) position.

In the first configuration the evaporator is at the intake line inlet, i.e. before the air filter, whereas in the second one, it is right after the actual intercooler, i.e. at the intake manifold inlet. In the first case, no significant performance variations occur on the engine, whereas, in the second case, great intake manifold air temperature decrease came out, in the 10-20°C range. This have undiscussed benefits on the cylinder filling and a net fuel consumption reduction up to 6 % has been demonstrated, which definitely brings to a 8 % saving on CO_2 emissions. However, when compressor work is taken into account, the fuel consumption percentage reduction decreases at the value of 5.3% and emissions rearrange accordingly.

At the same time, thanks to the higher air mass within the cylinder, also primary pollutant emissions have been reduced, with a 4.5%, 12.1% and a 5.6% reduction on NOx, CO and Soot emissions, respectively. HC increase is produced (average +9.7%)) due to lower chamber temperatures.

In conclusion, the undercooling of the engine charge air at the intake manifold seemed very beneficial in terms of fuel saving and emissions reduction. Care must be take on a proper evaporator selection: pressure drops along the intake line should be limited in order to prevent the turbocharger from overloading and, at the same time, shape, compactness and size should fit the existing intake line. The 1-2 kW range of air cooling power represents about the 50% of the rated cabin cooling power of an average passenger car. In real use, however, a dual evaporator layout should be used: one for the cabin cooling and the second one for the charge air. An eventual oversizing of A/C unit , in any case, is justified by the benefits demonstrated on the engine efficiency.

Acknowledgements

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