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Charge Air Subcooling in a Diesel Engine via Refrigeration Unit – Effects on the Turbocharger Equilibrium

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Abstract

The stringent regulations on fuel saving and emissions reduction in the transportation sector have become game-raisers in the development of present internal combustion engines for road applications, even if under-the-hood space constraints, downsizing and down-weighting prevent from adopting radical changes in the engine layout. Charge air cooling is the standard in present turbocharged diesel engines, to the point that a dedicated heat exchanger, fed by environmental air, is located downstream the compressor. The paper proves the option of an additional cooling through the cabin-heating unit - usually over-sized with respect to normal operation - very effective to increase charge air density and improve cylinder filling. The intercooler downstream the compressor would be provided with a lower thermal load, hence calling for smaller heat exchange surfaces, leading to reduced weight, space saving and no increased layout complexity. By pushing this idea forward, a properly sized cabin-heating unit could even supply enough air cooling to replace the intercooler instead of just assisting it, further raising the weight/space/layout advantages. In presence of an additional heat exchanger, the cooling efficiency would be no longer related to the vehicle speed and the benefit in terms of cylinder filling could be kept to the desired value on a wider operating range for the engine. Plus, the lower combustion temperatures associated with both a colder air and a more diluted charge approaching the chamber would also result in a more regular combustion process, in spite of a moderate penalty on the thermodynamic efficiency. The additional fuel consumption to compress the cooling fluid is always offset by the fuel saving with respect to normal operation and a beneficial effect is appreciated on emissions. Nonetheless, major variables to account for when evaluating the feasibility of such a layout are (i) the impact it has on the equilibrium of the turbocharger, i.e. on the efficiency at each operating point, (ii) to what extent the presence of a colder air affects the turbine/compressor matching and (iii) the rack position the ECU fixes at the VGT, to face both pressure losses at the additional evaporator and the different enthalpy content for both the air at the compressor outlet and the exhaust gases at the turbine inlet. A comprehensive experimental activity supported by a detailed 1D model of the engine unit, aimed at assessing the benefits of the air under-cooling, allowed to select the most appropriate cooling layout and, once validated based on the experimental evidence, to investigate the equilibrium at the turbocharger.

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1. Introduction

In light of the always lower caps on both admissible fuel consumption and pollutants emissions for manufacturers to stay on the market, the vehicle efficiency is continuously addressed at both R&D and deployment level. The attempts to increase it mostly consist in either the integration of new on-board components in standard layouts or the development of innovative engine configurations, rather than a radical change in the engine set-up, with the additional complexity of meeting market requirements in terms of down-sizing and down-weighting. Either way, a more complex engine architecture comes out and results in a calibration method harder to implement, due to the higher amount of components and degrees of freedom to be tuned.

As a consequence, numerical modeling is becoming the standard in ICE development, as it allows a leaner development process, plus time and costs savings at once. This is particularly true for 1D codes, whose versatility makes them applicable in the concept phase as well as in the powertrain control and calibration stages. Moreover, the extensive literature available proves that the 1D modeling approach is a perfect fit for the investigation of a variety of operation-related aspects, with main areas of concern in the modeling of combustion [1, 2], in-cylinder charge motion [3, 4], pollutants formation [5, 6], ducts dynamics [7, 8] and heat transfer [9, 10]. 1D models effectiveness is even higher when they integrate experimentally-supported empirical relations, hence preventing the lack in predictive capability, assuring results consistency with experimental evidence and providing a safe platform for engine control during operation (e.g. peak power at full load and minimum fuel consumption at part load). Moreover, the 1D codes accuracy is a key factor, since they often represent the early stages of a more detailed engine characterization, carried out on 3D simulation platforms for which they provide the numerical input [11, 12]. In addition to the areas of application mentioned above, the 1D model approach is highly performing for the investigation of the engine performance enhancement through charge air cooling, an already consolidated technique in present turbocharged diesel engines, in which a dedicated heat exchanger allows to partly recover the issues associated with the air density reduction downstream the compressor (i.e. not proper cylinder filling and low volumetric efficiency) and to restore acceptable system efficiencies. Nonetheless, some operating and plant-layout-related issues – namely, the cooling capability/vehicle speed dependence, the radiator size in comparison to the under-the-hood space availability, weight increase and piping complexity – limit the advantages of such a layout and result in a 40–70°C intake air temperature at cylinders [13]. Previous studies by the Authors proved the feasibility of an additional air cooling through an evaporator at the intake manifold [13, 14]. Main strength of such a configuration has been proved to be in the fact that the conditioning unit already present on board for cabin heating/cooling purposes, oversized with respect to normal employment, can safely be in charge for the sub-cooling. Plus, the possibility to count on a dedicated evaporator for air cooling provides a better control of the engine performance over the whole operating regime for the engine, with no dependence on the relative speed between vehicle and outside air. Hence, no penalty would insist on the system, from either additional weight or unavailable room for the addition of a new component [13–16]. Both factors, indeed, along with system scalability, components packaging issues, low power density and the backpressure from an additional heat exchanger, limit the feasibility of the waste heat recovery through other techniques, such as exhaust-fed jet ejectors [17, 18], heat pumps [19], mechanically driven compression chillers [20], sorption-based cooling systems [21, 22, 23, 24], multi-stage air cooling [25] and cooling water-fed heat pumps [26, 27].

The idea of employing the cabin heating/cooling unit to provide charge air cooling represents a major breakthrough with respect to the research standard in the automotive sector [13, 14, 28] and the novelty of such an application calls for an extensive modeling activity on the engine performances.

The model, the Authors developed in-house on the 1D GT-Suite™ platform, moves from experimental data provided by an extensive experimental campaign on an turbocharged 3L IVECO F1C diesel engine, featuring an evaporator in the intake line. Based on the power, torque and EGR maps for normal operation, directly accessed from the ECU, the data allowed the proper model definition (e.g. heat exchange surfaces sizing, heat exchange and performance coefficient selection) over the whole NEDC cycle. Based on the experimental data, the best option in terms of heat exchanger type and layout turned out to be the one featuring a shell (R134a) and tube (air) heat exchanger in the intake manifold, as it maximizes the air sub-cooling at the cylinder intake. The 1D model confirms the effectiveness of such a layout and supports a proper heat exchanger sizing, dependently on engine type, suitability with the engine operation as defined by the NEDC homologation cycle and minimum power absorption by the compressor. Once validated based on the experimental evidence, the model allows to characterize the equilibrium of the turbocharging unit, in both presence and absence of the additional cooling, in terms of efficiency for both turbine and compressor and operating speed for the engine (i.e. for the turbocharger itself). The combined effect of increased pressure drops and modified thermodynamic coordinates for the group operation are investigated as well and the ECU-controlled adjustment on the VGT is assessed, by monitoring the rack position in every operating point of interest.

2. Preliminary experimental activity

Ten engine operating steady points were performed for a 120 s duration, varying the engine speed from 1200 to 3600 rpm, repeated at 100 and 200 Nm. The IVECO F1C diesel engine air cooling system, already featuring a tap water-fed intercooler, is integrated through a R134a single pass counter-flow shell and tube heat exchanger, responsible for the air undercooling downstream the intercooler (Figure 1). Such a layout is selected among all those possible as the most effective one, i.e. the one assuring a proper utilization for the intercooler, the lowest air temperature at the manifold intake and the lowest pressure drops in the intake runner [13]. The uncertainty affecting the measures is in Table 1. A temperature reduction in the 15-30 K range is appreciated with the air cooling unit on (A/C ON), with little differences between the 100 Nm and 200 Nm torque, corresponding to a heat removal at the evaporator in the 0.3-2.5 kW, depending on the air mass flow rate (Figure 2). Generally speaking, the lower the air mass flowing in the runner and eventually in the heat exchanger (i.e. the lower the engine load), the lower the heat exchange and hence, the lower the gain with respect to normal operation with no air cooling (A/C OFF). The mechanical power absorption by the compressor ranges between 212 W and 520 W, when the air mass flowrate shifts from 100 kg/h to 480 kg/h and increases with the air mass flow rate being processed, in accordance with expectations.

Table 1. Measurements uncertainty

Temperature [°C]	±2.2	Air mass flowrate [-]	±3%
Pressure [mbar]	±0.8	Fuel mass flowrate [-]	±1%

The benefit achieved on the air mass flowrate entering the cylinder is in Figure 3, along with the increase in the engine global efficiency: a 5.7% increase in the air mass approaching the cylinder is appreciated at 1800 rpm, followed by 3.9% increase at 3000 rpm. A slightly different trend is appreciated on the engine global efficiency, for which a higher spread occurs as the engine speed varies: the efficiency increase tops 4.6% at 1200 rpm, with the lowest at 2400 rpm (1.5%). The in between values range between 2% and 3%, with negligible differences among engine speed values. Pressure drops monitored on the air side are not negligible, though, and reach 100 mbar at 450 kg/h, 200 Nm engine load (Figure 4): dependently on the operating regime, such a penalty offsets the pressure boost achieved through the ECU-controlled variable geometry turbine and reflects on the manifold intake pressure, where a 20 hPa maximum penalty occurs.

Figure 1. Downstream undercooling layout

Figure 2. Clima unit power

Figure 3. Air mass flow rate and engine global efficiency

Figure 4. Subcooling unit air side pressure drops

3. Model validation and equilibrium assessment for the turbocharger

Specifics for all engine subsystems the GT Suite™ model accounts for are in Table 2. Look-up tables and PID controllers allow to reproduce the ECU control over the engine at a model level, with main control variables in the boost pressure and the amount of fuel being injected - as a function of engine speed and engine load - and the rack position via ECU actuation [29]. Intake and exhaust manifolds were modeled as elementary ducts – rectilinear or curvilinear – along with fittings and connecting elements. Geometry and operation-related coefficients (e.g. friction, discharge, heat exchange) were specified at the model initialization [30]. Filling/emptying processes have been included, accounting for the mass trapping inside the cylinder. Same goes for the combustion process, the gas/wall heat transfer and the work availability at the shaft. Conservation equations for mass, momentum and energy are solved for steady states, according to specified boundary conditions. For instance, the target intake pressure is fixed by the ECU actuation on the VGT.

A three-Wiebe model describes the burned fuel fraction as a function of the crankshaft angle and the combustion phase (i.e. premixed, diffusive, oxidation-controlled): the injection timing and the diffusive combustion duration are assigned [16] and the power at the shaft calculated accordingly, under the condition that the cumulative burned fuel fraction adds up to 1. Woschni correlation regulates the heat transfer in the cylinder. Real maps for both turbine and compressor of the turbocharging system are introduced as well.

Table 2. IVECO F1C 3.0L data and model parameters

Bore [mm]	95.8	Peak power [kW]	130
Stroke [mm]	104	Peak torque [Nm]	402
Connecting rod length [mm]	220	Ignition sequence	1-3-4-2
Displacement per cylinder [L]	0.75	IVC [CSA]	-160
Total displacement [L]	2.998	EVO [CSA]	183
Cylinders number	4	IVO [CSA]	340
Compression ratio	19:1	EVC [CSA]	366

Experimental data referring to 100 Nm engine load allow the model validation (Figure 5): when the boost pressure is assumed as the control variable, a maximum 8% difference is appreciated, with the engine speed varying between 1200 rpm and 3600 rpm. The model underestimates the torque, with a 10% maximum error, occurring at 2400 rpm, which confirms about the good fit between experimental and simulated data, especially considered the relevance of the torque datum in the engine characterization. This assures about the fact that a negligible shift has to be expected between predicted and experimental data and that the only source of bias are (i) the extrapolation procedure performed on the characteristic curves for the turbine - to properly account for the choke on the air mass flowrate admitted to the rotor stage - and (ii) the shell and tube heat exchanger modeling, particularly sensitive in 1D models to the correlation adopted to describe thermal interactions both within the heat exchanger (e.g. flow splitting, air/refrigerant heat exchange coefficients) and with the surroundings (e.g. wall heat transfer coefficients). As far as the heat exchanger modeling is concerned, a Colburn-Klimenko relation for heat transfer seems to be particularly suitable for a detailed description of refrigerants evaporation at the temperature and pressure levels at hand [28, 31].

As far as the extrapolation on the manufacturer data for the Garrett Turbocharger T202H(50)T74 VGT Turbine Compressor C224(60)T50 is concerned, [32] provides the best option for data correction, in presence of a variable geometry admission section to the rotor stage. The equilibrium point for every mass flow rate of interest remains completely defined (i.e. rpm, compression and expansion ratio), once the ECU provides the boost pressure at given engine speed and load, as in Figure 6-7. Iso-lines for efficiency and rack opening are reported for the turbine, to allow a complete characterization of the operating condition for the unit. Same goes for the compressor, with the efficiency and the revolution speed as parameters. For sake of clarity, Figure 6-7 only report experimental data for the 100 Nm engine load and suggests that the group equilibrium is not strongly affected by the additional cooling performed on the air. Each point on the charts corresponds to an assigned engine speed and engine torque: they are close each other and little variations are appreciated on the reduced mass flowrate, independently on the regime of additional cooling that applies. Same goes for the turbine and compressor efficiency. The temperature experiences a 15-30 K reduction at the intake manifold, dependently on the engine speed, and such a variation is mirrored by the one at the turbine inlet. As the engine speed increases, the rack always closes up a bit, to compensate pressure losses at the evaporator and results in a group equilibrium almost unaffected by the changes in the thermodynamic coordinates of the turbocharging unit:

- point A (1200 rpm engine, 40000 rpm turbo): the low engine speed leads to a low mass flowrate and the increase in air density from the additional cooling does not affect the group equilibrium, with a 50% compressor efficiency and a 39% turbine efficiency. 27.8 mbar pressure losses occur at the evaporator;
- point B (1800 rpm engine, 70000 rpm turbo): a negligible (2%) reduction on the speed of the group is appreciated. Both the compressor and the turbine efficiency are slightly higher with the subcooling (1.5% and 2%, respectively). No additional pressure losses occur at the under-cooling section (56.5 mbar) and consequently, the rack position stays the same;
- point C (2400 rpm engine, 75000 rpm turbo): the greatest shift is on the turbine section, with a 3.2% efficiency reduction, entirely due to the lower enthalpy content (i.e. the lower temperature) of the exhaust gases at the turbine inlet, due to the more diluted charge in the cylinders. The compressor efficiency tops 70% in both cases. These data suggest that the ECU controlled VGT-rack opens a bit and effectively compensates the increasing pressure losses (71.8 mbar) at the evaporator;

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- A scatter plot showing boost pressure [bar] on the y-axis (ranging from 1.0 to 1.5) versus engine speed [RPM] on the x-axis (ranging from 1200 to 3600). The plot compares model validation results (circles) with experimental data (squares). The data points are labeled with their respective boost pressure values.
Engine Speed [RPM]	Model Boost Pressure [bar]	Experimental Boost Pressure [bar]
1200	1.16	1.08
1800	1.18	1.25
2400	1.18	1.29
3000	1.45	1.35
3600	1.45	1.39

Model validation - Torque

engine speed [RPM]	model torque [Nm]	experimental torque [Nm]
1200	100	100
1800	99	100
2400	110	100
3000	98	100
3600	105	100

Figure 6.Compressor map

4. Conclusions

A comprehensive analysis of a turbocharged diesel engine for heavy duty applications was performed, through both an extensive experimental characterization and a detailed virtual modeling activity, to assess the effectiveness of an additional charge air cooling. A sheet and tube evaporator was integrated to the existing layout and placed downstream the compressor, at the intake manifold and fed with R134a, with the intention to reproduce operating conditions and cooling capabilities of a standard cabin heating unit. The potential of such a unit to replace the already on-board intercooler is also evaluated, due to its reflections on the size/weight benefits it assures. A heat removal in the range 0.3–2.5 kW is possible, dependently on the operating conditions (engine speed and engine load) for the system at hand, with an average 15–30 K temperature reduction at both the intake manifold and turbine inlet, i.e. a lower temperature combustion within the cylinder. Along with the obvious benefits such an engine set-up has, in terms of down-sizing, down-weighting, fuel consumption and emissions reduction, the presence of an additional air cooling unit leads to modified thermodynamic coordinates for the turbocharging group, as the compressor faces higher pressure losses and the turbine is fed with a lower enthalpy content, due to the more diluted charge within the cylinder. Based on experimental data and the findings of a validated 1D model of both components and connections in the engine set-up at hand, various operating conditions were swept and the equilibrium of the turbocharging unit was assessed. The turbocharger does not suffer the presence of the additional heat exchanger: for a 100 Nm engine load, the turbocharger speed experiences 2% maximum variations, dependently on the operating points being considered; even smaller variations occur on the rack position. The efficiency value for both turbine and compressor stays the same with and without the air subcooling unit and the effect of the pressure losses is negligible, mostly due to the VGT compensation of both the increased resistance to the fluid motion within the pipes and the modification of the thermodynamic state for the air being compressed and the exhausts approaching the turbine.

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