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# MAPPING OPTIMIZATION FOR PARTIAL LOADS OF COMMON RAIL DIESEL PISTON ENGINES

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**ABSTRACT**

Theoretically, from a control design point of view, modern diesel engines are dynamic, nonlinear, MIMO (multiple-input and multiple-output) systems. This paper demonstrates that this assumption is not correct and a suitable model for predictive control (MPC) of power (torque), NOx and soot emissions based on temperature feedback is perfectly possible on SCR (Selective Catalytic Reduction) CRDITDs (Common Rail Direct Injection Turbocharged Diesel). The method optimizes the temperature at a selected point of the engine exhaust. This reference point is the turbocharger intake for Euro 0 (aircraft). For Euro 6+/US Tier 3a+ SCR diesels, the reference point is the intake of the "emission control system" usually at the outlet of turbocharging system. The traditional five-inputs are only theoretically independent. In fact, fuel injection duration depends on torque (load) and efficiency. Fuel advance is retarded to obtain the required reference temperature. HPEGR (high pressure exhaust gas recirculation) is adopted only when the emissions cannot be controlled by the fuel advance. VGT (variable geometry turbo) valve positions and low pressure LPEGR maximize the air flow (efficiency) at the engine intake. On the outputs, peak pressure and peak pressure derivative should be kept within structural limits. Soot and NOx are two faces of the same problems. In fact, high NOx means low soot and good combustion efficiency. Temperature and air flow are the keys to obtain optimum engine performance. Air flow is controlled by the turbocharger, while temperature depends on injection. This paper demonstrates that CRDITDs mapping is much easier when the fundamentals of diesel combustion and SCR are simplified to basic concepts. The strategy to retard the injection advance increases efficiency of 30% over traditional LPP (Optimal Location of Peak Pressure)- mapping at low loads.

**Keywords:** diesel, electronic, control unit, mapping, optimization, emission.

## INTRODUCTION

The automotive engine market has seen the growth of the compression-ignition diesel engines, due to the high energy conversion efficiency and the good performances of CRDITDs (Common Rail Direct Injection Turbocharged Diesel). Legislators have partially hampered this success by introducing severe emission regulations. In fact, this CRDITD efficiency has come at the price of high levels of nitrogen oxides (NOx) and soot particle emissions. In the last decade, the progressively tightened emissions legislation has pushed for a shift in technology, either through introducing an after-treatment system for NOx reduction or through in-cylinder techniques. In the latter case, which is typical of Euro 5 engines, the combustion process is cooled down using lower combustion peak pressure, high levels of high pressure exhaust gas recirculation (HPEGR), and delayed injection. This suitably chosen fuel injection timings resulting in so called Low Temperature Combustion (LTC).

Also LPEGR (low pressure exhaust gas recirculation) is used to maximize compressor(s) performance and air flow. Unfortunately, LTC increases Soot production. Therefore, Soot traps or filters were introduced in the exhaust system. The post-injection tended to pollute the lubricant with diesel fuel reducing engine life. The EGR tended to encrust the manifold reducing engine efficiency. With the Euro 6/Tier 4 it was not possible to follow this approach and SCR (Selective Catalytic Reduction) became common.

### The modern engine power plant

CRDITDs are in fact composed by three distinct "machines". The first one is the Diesel engine that works with the air provided by the TC(s) and power the TC through the exhaust. At the TC exhaust a "cleaner" or exhaust control system uses the energy (temperature) and claims a small pressure drop from the TC exhaust and, consequently, to the Diesel exhaust. The turbine of the TC is linked to the Diesel engine through the exhaust. This extremely loose connection does not synchronize the TC with the engine. Therefore, the TC system should be matched to the engine. This matching is obtained by multiple TC arrangements either serial or/and parallel or/and through VGT (Variable Geometry Turbo) systems. Turbo-matching can be also improved through HP-EGR (High Pressure Exhaust Gas Recirculation) and LP-EGR (Low Pressure Exhaust Gas Recirculation). The diesel fuel combustion is commanded by the autoignition point (Point of Start of Combustion PSOC). This point is the when the apparent heat release shows a minimum. In this crank position the energy release due to combustion exothermic reactions begins to exceed the energy losses due to the fuel evaporation. This first stage of combustion emits high quantities of OH, CH and HCO, due the thermal decomposition of the hydrocarbon molecules preceding true ignition. This is the "flameless" or low temperature ignition at very low thermal loads. This high intense energetic chemical activity is found into the air entrapped into the spray around the jet. near exhaust valves, where the temperature is higher. Then the OH evolves following

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the fuel jet traces. Therefore, the combustion begins asymmetrically. The fuel jets proceed radially inside the combustion chamber, slowing down toward the "cool" combustion bowl walls. The high swirl ratio bends the the combustion along the jet axis. In fact, the fuel burns quicker on the surface that is more exposed to the incoming air. The combustion in CRDITD is completely different than in stratified combustion spark ignition engines. In spark ignition engines the flame volume is full of hot burning gas, while the front surface is burning toward the fresh, oxygen rich air. If burning surface reaches all the air before detonation the combustion is correct. In CRDITD combustion timing is extremely critical. A +/-5-degree error in timing means that your engine is breaking down in an uncontrolled fuzzy way [1][2]. Combustion pressure transducers are installed into the most modern CRDITD injectors. Autoignition takes place as air inside the cylinder is compressed it heats. In the 180-degree compression stroke, the final 30 degrees of rotation adds the same amount of heat as in the first 150 degrees of rotation. Air temperatures over 600°C are commonly reached. Pressure rise inside the combustion chamber is associated with this temperature increase. Therefore, increasing boost has enormous consequences on ignition timing. Once air is very hot somewhere near the Top Dead Centre (TDC), the ECU commands a time-defined precisely-measured shot of micron-sized diesel jet under the mapped rail pressure. The time took by the tiny droplet to vaporize and mix with the air up to autoignition is called delay time. This delay $t_d$ [ms] can be evaluated by the "historical" equation from Wolfer (1938) (1).

$$t_d = aP^b e^{\frac{c}{T}} \qquad (1)$$

Wolfer suggested that values for a=0.44, b=1.19 and c=4650 are valid. While coefficients a,b and c of equation 1 can easily change from different Authors, it is extremely clear that delay time is approximately proportional to chamber pressure p [MPa], but it is exponentially influenced by camber temperature T [K]. Therefore, it is extremely important to reach extremely high temperatures to enhance combustion performance. On the other side extremely high temperature would not be tolerated by the engine piston that is cooled by the incoming air on the top and by the lubricant on the bottom surface. The intake air is also intercooled to increase the air charge. Typically, at the intercooler outlet the air is at 50 DEG C.

**Optimal Location of Peak Pressure (LPP) at full load**

The maximum torque (maximum efficiency) at maximum load is obtained when the location of peak cylinder pressure, LPP, occurs between 10 and 16 degrees ATDC (After TDC). The reference full-load optimum LPP for CRDITCs is 12 DEG ATDC (After Top Dead Centre). In this case BTDC (Before Top Dead Centre) injection timing advance is adjusted accordingly [3].

**Exhaust emission control systems**

Any change in diesel combustion to reduce NOx emissions results in an increase in particle emissions. In fact, higher combustion temperatures increase the oxidation level of the fuel, thus less soot, but also cause more formation of NOx. However, the use of ultrafine injector orifice diameters in conjunction with lower excess oxygen content in the fuel mixture reduces combustion temperature with a more homogeneous distribution of the charge, thus reducing both NOx and Soot. The DOC (Diesel Oxidation Catalyst) positioned immediately after the TC(s) (TurboCharger) heats up rapidly and therefore immediately oxidizes the CO and the hydrocarbons during cold-start. The heat generated by oxidation in the DOC speeds up the heating of the downstream catalysts, such lean NOx catalysts and SCR catalysts. DOCs reduce total PM (Particulate Matter) up to 50 percent by mass depending on the composition of the PM being emitted. Diesel oxidation catalysts also eliminate the obnoxious odors associated with diesel exhaust. DOCs reduce more than 90 percent of the CO and HC emissions. DOCs have operated trouble free for hundreds of thousands of kilometers on billions of vehicles for over 30 years.

DPF (Passive Diesel Particulate Filter) uses normal exhaust temperatures and nitrogen dioxide (NO2) as the catalyst to oxidizes PM in the DPF. In contrast to higher temperature active regeneration DPF that uses fuel for the same purpose (regeneration cycle). Operators benefit from the simplicity of the passive regeneration process, and fewer components make it a lightweight and easily installed solution.

EU Stage IV/U.S. EPA Tier 4 emission standards require a reduction of the PM or soot entering the atmosphere by ì 98 percent when compared with Euro 0 engines. Ultra-fine particulates pass through and are captured by the DPF. Over time, these particulates build up. Therefore, passive regeneration is used to oxidize the PM within the filter. The continuous regeneration process of passive DPF keeps the filter efficiency without the need to raise the temperature typical of active DPF. On the contrary active or high-temperature DPF removes the PM by burning or oxidizing it on the filter when exhaust temperatures are adequate. By burning off trapped material, the filter is cleaned or "regenerated." The problem of passive DPF is insufficient exhaust gas temperatures associated with the operation of some types of diesel engines. Closed loop SCR (Selective Catalytic Reduction) using urea reduces NOx emissions up to 97 percent while simultaneously reducing HC emissions by 90 percent. A final CUC (Clean-Up Catalyst) is often necessary to complete the oxidation process. Therefore, a typical EU Stage IV/U.S. EPA Tier 4 includes a DOC, a DPF, a metering module for AdBlue, an AdBlue mixer, a SCR and a CUC. The Euro VI emission limits can be reached through the use of this SCR based system with or without the combined use of the HP-EGR (High Pressure Exhaust Gas Recirculation) system. The use of the HP-EGR system allows for a reduction of NOx emissions in the combustion chamber via recirculation of exhaust gases, with the consequent increase in the production of

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particulates and reduction of combustion efficiency. The increase in particulate emissions from the engine needs forced regeneration of the DPF. Therefore, EGR should be avoided in passive DPF systems. The key effects of EGR are lowering the flame temperature and the oxygen concentration of the working fluid in the combustion chamber There are two types of EGR, high pressure loop HP-EGR and low pressure loop LP-EGR (Low Pressure EGR). The HP loop takes the gases before the TC(s), while the LP loop operates after the TC(s). Table1 shows advantages and drawbacks of these type of EGR.

**Table-1.** Comparison between high pressure and low pressure EGR.

| | Advantages | Drawbacks |
|---|---|---|
| HP-EGR | Lower HC-CO emissions<br>Fast response time | Cooler fouling<br>Unstable cylinder-by-cylinder EGR distribution |
| LP-EGR | High cooled EGR<br>Clean EGR (no fouling)<br>Stable cylinder-by-cylinder EGR distribution | Corrosion of compressor due condensation water<br>Slow response time<br>HC/CO increase |

The HPL EGR has fast response, especially at lower speed and load. Therefore, it is widely used in automotive applications. However, it is only applicable when the turbine upstream pressure is sufficiently higher than the boost pressure. On the contrary the LP EGR has slow response, especially at the low loads or speeds typical of the automotive application. Therefore, it is more suited for aircraft/helicopter/heavy-duty applications.

In any case the emission control system requires a defined temperature interval to work properly. A secondary necessity is oxygen and oxidizers content in the exhaust. This secondary condition can be easily met with proper TC(s) controls. The temperature condition remains very stringent especially at the very low loads typical of modern CRDITDs cars that are typically overpowered for marketing reasons.

**A few basic considerations on emissions**

In an automotive application emission depends on powertrain (engine+gearbox+wheels) and on vehicle (aero) dynamic. Therefore, it is the vehicle that satisfies the EU Stage IV/U.S. EPA Tier 4 emission standards, not only the engine assembly. Standard tests are performed to check the compliance to the standards. The first way to improve emissions is to reduce gearbox speeds. In this way the amount of air ingested by the engine is increased. So vehicle tuning is always a compromise between drivability/performance and emissions. Generally, targets are given to obtain acceptable results. Normally, fuel consumption in 'combined', 'urban' and 'extra-urban' are the target, since they are easily understandable by the consumer. 0-100 km/h time is another typical parameter. However, to comply emissions it is necessary to keep the exhaust temperature within strict limits. In aircraft, no emission control is required for engines under 26.7kN thrust.

The aircraft emission standards over 26.7kN have been updated in the last 30 years. Over the years, emission standards have been set for different aspects of aircraft engines: from 1974 engine smoke standards have been revised several times. In 1984 fuel venting and hydrocarbon emissions were regulated. From 1997 restrictions have been introduced on NOx and carbon monoxide. However, costumers and manufactures are increasingly concerned about emissions even for smaller engines. In the case of thrusters, engines (propeller/fan+engine) and not aerial vehicles should comply to the emission standards [5].

**A few considerations about CRDITD injection**

The main problem of common rail injection system is the random and systematic differences in the spray pattern between nozzles even in the same injector. The random differences are due to turbulence and cavitation and systematic ones depends on the extremely critical nozzle tolerances. Furthermore, needle velocity is extremely difficult to control, especially in the initial opening phase. In fact, the solenoid controlled common-rail fuel injector has a relatively slow needle opening velocity, and the needle remains in partial lift for a relatively long time. Therefore, the proportioning of small fuel quantities is difficult and inefficient. The minimum fuel charge injectable depends on true plays inside injector and fuel kinematic viscosity. Both these quantities are affected by temperature. If the injection time is very short, the initial (opening) and final (closing) phases become critical and varies from injector to injector even in the same production batch. Calibration corrections are insufficient for a proper control of the combustion and its pressure gradients. The volume for a single injection cycle depends non-linearly on common rail pressure and current pulse time. However, over a certain threshold of the rail pressure, after the opening phase, the injection quantity is proportional to time. Under a fixed pulse time, the injection volume dos not increase in a proportional way due to the variable opening time. The most modern type of injector without the hydraulic amplifier partially corrects the problem. This is due to an improved opening dynamic. Another typical problematic of the common rail injectors is that there is a limit in minimum pulse duration. The best injectors need a pulse duration is 0.4ms to inject fuel. The quantity of fuel effectively injected depends on many parameters; a very important one is the fuel temperature and viscosity. Equation (2) evaluates the volumetric injection quantity per-cycle.

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$$q = \int_{t_1}^{t_2} \cdot \frac{10^{12} F_L (p - p_0)}{\sqrt{1 + \frac{2E}{E_0}\left(\frac{D^2 + d_L^2}{D^2 - d_L^2} + \mu\right) \cdot \frac{2L}{t_w} \rho}} dt \quad (2)$$

Up to about 2,000 rpm it is possible to use the pre-injection technique. This extremely tiny portion of fuel makes in main injection to burn better with reduced delay time. This process it is also highly in controlling the emission and in improving the noise levels. In fact, at low loads temperatures and pressures are often sub-optimal.

**Efficiency and load**

An optimized indicated cycle for a CRDITD at full load is shown in Figure-1:

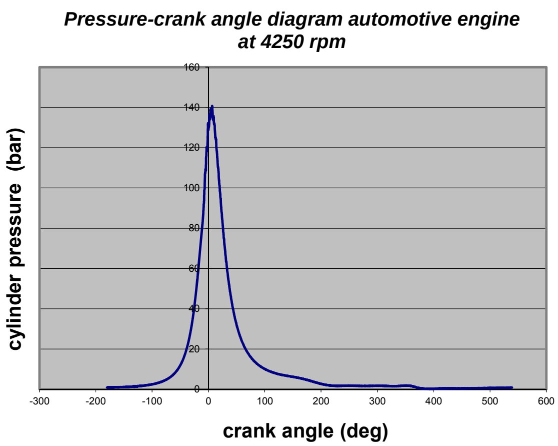


**Figure-1.** Optimum full load indicator diagram CRDITD.

Figure-1 shows that the "true" CRDITD cycle is closer to the Otto cycle (combustion at constant volume) than to the theoretical Diesel cycle (combustion at constant pressure). For this reason, the classical efficiency equation of the Otto cycle (3) is used for general considerations.

$$\eta = 1 - \frac{1}{r^{\gamma-1}} \quad (3)$$

From equation 3 it is evident that the Otto cycle efficiency depends directly upon the compression ratio r. Since the heat capacity ratio $\gamma$ is always greater of equal to 1, an increase in r will produce an increase in efficiency $\eta$.

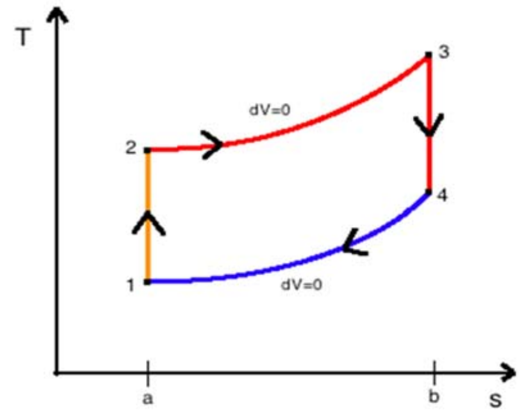


**Figure-2.** Otto cycle from Austin Tutor - Own work, CC0, https://commons.wikimedia.org/w/index.php?curid=14916376.

The compression ratio r is defined from equation (4) Figure-2.

$$r = \frac{v_2}{v_1} = \left(\frac{p_2}{p_1}\right)^{\gamma} \quad (4)$$

The compression ratio r depends also on the boost pressure in CRDITDs. For this reason, it is convenient to begin the combustion at the highest pressure possible. The air is compressed into the intake manifold by the TC compressor. This air is also cooled by the intercooler to maximize its density. In fact, power goes with fuel mass that depends on air density. As the fresh air is forced into the cylinder through the intake valve(s), it cools down the piston that is very hot from the combustion and exhaust cycle (see figure 3 for maximum piston temperature in a CRDITD). Therefore, air temperature increases not only for the compression, but also for the heat exchange mainly with the hot piston.

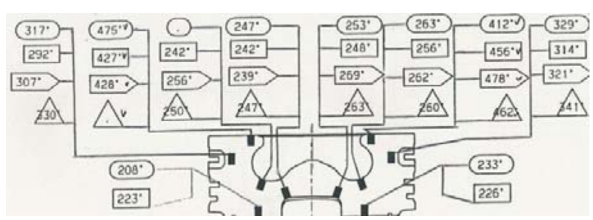

**Figure-3.** Maximum temperatures on a piston for a 4 cylinder CRDITD. The different symbols indicate the different cylinders.

For this reason, in a cycle without combustion, the air charge temperature increases also after the TDBC point. High air temperature not only increases efficiency of the cycle (equation 3) but also reduces the combustion delay $T_d$ and increases combustion efficiency. The reduction in combustion delay increases the pressure peak, while the better combustion efficiency reduces the amount of unburnt fuel. Both these facts are beneficial to the

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overall engine efficiency. In fact, the amount of unburnt fuel in terms of energy may easily reach 10% at full load. In diesel engines, combustion is activated by the injection. After a defined delay time the jets begin to burn inside the combustions chamber. The combustion rate and the delay time depends on air charge temperature with an exponential law. Therefore, in order to maximize efficiency, fuel injection should be delayed as much as possible.

**Fuel charge and load**

In CRDITD the fuel charge depends on load and fuel energy conversion. At high loads it is relatively easy to obtain very good efficiencies. At lower loads friction and accessory loss tend to reduce the overall efficiency of the engine. Therefore, partial loads are much more critical than full load for engine mapping and fuel consumption. The efficiency of Diesel Engine when measured against load is mainly dependent on its 'air handling capacity'. The excess air and higher compression ratio also leads to increase in its efficiency. Theoretically, the fuel injection is ended sooner and the diesel droplets have enough time to complete the combustion process resulting in complete conversion of chemical energy of fuel to mechanical energy leading to higher efficiency. Practically, during the jet build up phase, the combustion is not efficient, with a relatively high percentage of unburnt. Therefore, very partial load and very short injection require the maximum air temperature possible. As the load increases, at “Higher Partial Load', the fuel injection is more optimized. The spray built up time is a smaller percentage of the total injection time and the diesel droplets have stillt enough time to complete the combustion process resulting in better conversion of chemical energy of fuel to mechanical energy leading to higher efficiency. As the load reaches the full value, the fuel injection time and rate has to increase so as to supply the required amount of energy, which reduces the time of combustion for diesel droplets, as such the complete conversion of fuel chemical energy into mechanical energy doesn't take place, leading to lower efficiency, soot and CO and HC in the exhaust.

Diesel engines in automotive configuration are normally optimized for emissions. Low load operations of diesel engines are defined as engine operations below 40% of maximum continuous rating for a certain engine speed. Low load operations cause lower cylinder pressure and thus lower temperature. Low temperature leads to less efficient combustion which causes increased soot formation and aggregation of unburned fuel in the cylinder. Low cylinder pressure, soot and unburned fuel deteriorate the piston ring sealing efficiency allowing hot combustion gases, soot particles and unburned fuel to leak past the piston rings. This results in increased lubricant consumption and fuel dilution. Fuel dilution reduces oil viscosity which causes premature wear of pistons, rings, liners and bearings. Therefore, low load lead to a cycle of degradation which means that diesel engines that run at low loads for longer periods of time can have shorter TBO (Time Between Overhaul) [6] [7] [8] [9]. This fact is aggravated by the fact that most modern diesel engines operate at lower cylinder pressure and thus lower temperatures to comply with stringent NOX emission requirements. Therefore, modern NOX optimized engines are far more exposed to the damages induced by low load operations than their predecessors due to lower design cylinder pressures and temperatures. To improve TBO the engines must be loaded to at least 50% of rated power regularly during low load operations.

**Low load operations, efficiency and emissions**

It was already demonstrated in the previous paragraphs that delayed injection is advantageous for higher efficiency at low load. Injection mapping and advance at low load should not be aimed to optimize the LPP (Location of peak pressure), but to obtain the right temperature at the exhaust. In the case of Euro 0 engines the target temperature is the TC turbine intake temperature. At low loads the TC is struggling with low volumetric flow rate both at the compressor and the turbine. High exhaust temperatures reduce this problem at the turbine, thus increasing TC rpm and reducing the probability of surge of the compressor and the turbolag. In EU Stage VI+/U.S. EPA Tier 3a+ engines it is essential to keep the exhaust cleaning system at the right temperature interval, which is between two well defined values. Especially with SCR system, relatively high combustion temperature is accepted to reduce soot and to increase efficiency. However, in these new engines, it is essential to achieve a very high NOX reduction in the exhaust with efficiency that, in a few cases, should reach 97%. For this reason, injection timing at low loads should be delayed as much as possible to keep exhaust temperature high enough to remain in the optimum range.

**Traditional racing mapping**

The combustion of DID (Direct Injection Diesel) relies on auto-ignition, a far less precise mechanism of ignition than spark. As air inside the cylinder is compressed, it heats up to 600°C before combustion even begins. After the combustion temperatures can very briefly reach $T_3$=3000K.This means that the Carnot Cycle reference efficiency $\eta_c$ is extremely high (1). In equation (5) the room temperature $T_2$=288.15K is ISA+0°C (International Standard Atmosphere) sea level.

$$\eta_c = \frac{T_3 - T_1}{T_3} = \frac{3000 - 288.15}{3000} = 0.9 \tag{5}$$

This is the reason of the efficiency of CRDITD that can easily exceed 50%. For high efficiency, the maximum temperature should be kept as high as possible. Since engine durability depends on average temperatures, while efficiency depends on maximum temperature and high temperature peak is highly desired.

The Ideal Gas Law (6) for Vcc (Volume of Combustion Chamber) links $p_3$ and $T_3$.

$$p_3 V_{cc} = RT_3 \tag{6}$$

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Therefore, the higher is the pressure the better. Maximum cranking efficiency is obtained with an LPP of 12 DEG ATDC. For this reason, full load mapping is based on maximum allowed (structural) pressure at 12 DEG ATDC. If problems arise with combustion the LPP angle shifts up to 16 DEG ATDC remaining into the best LPP interval. Combustion problems or delays are extremely common in DIDs due to several reasons. The most important is fuel quality. For this reason, tests should be made with the "worst" fuel available that should be purchased ad hoc. After full load mapping is performed, the injection advance and timing is reduced to allow for lower loads.

**High efficiency mapping and single variable optimization**

The LPP-pressure aimed mapping has several shortcomings at partial loads. With SCR and Euro 0 engines the objective is to burn at the highest temperature possible, leaving to the exhaust control system the task to reduce NOx. Soot should be minimized at any cost, since EGR and active DPF should be avoided. For this reason, Euro 0 and EU Stage VI+/U.S. EPA Tier 3a+ engines can be mapped with the same technique. The problem of LPP-pressure aimed mapping is that combustion efficiency depends on injection temperature. At low rpm it is possible to adopt pre-injection to increase chamber temperature and to reduce noise and engine stress. The low efficiency combustion of pre-injection is corrected by the main injection. It is not convenient to adopt multiple injections since transients in injection should be avoided at any costs. If the rail is not able to keep pressure within tolerances it should be increased in volume. Modern high temperature (1050 DEG C) glow plugs solve easily the problem of "long starting time" due to low rail pressure in high volume rails. Modern injectors with short transient time and pressure feedback should be preferred. Temperature rises even ATDC when pressure began to fade. This is due to heat exchange. In fact, equation (6) holds only when heat exchange rate is low. Therefore, main injection should be aimed to maximize the temperature at the critical point. This point is usually at the turbine TC intake or at a selected point of the emission control system on exhaust. Injection should be delayed as much as possible to allow to air charge temperature to reach the maximum value possible. This reduces ignition delay and increases combustion efficiency, that can be, in the worst cases as low as 0.7.

The temperature maximization increases the Carnot reference cycle (equation (5)) and the combustion efficiency. This strategy may reduce fuel consumption at partial load up to 30% when compared with LPP-pressure traditional mapping strategy. Since fuel mass and volume to be injected can be easily calculated, the injection pulse duration is known. Therefore, the injection becomes a single variable optimization. The optimization variable is injection advance. A software loop can be implemented into the development ECU to optimize injection advance in order to keep temperature at the reference value.

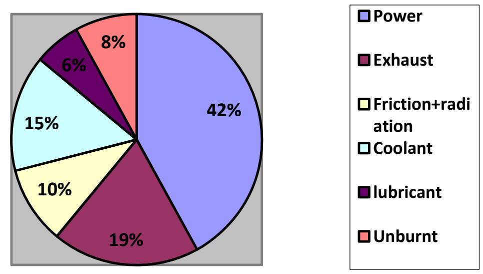


**Figure-4.** Thermal efficiency of an automotive CRDITD.

**CONCLUSIONS**

Theoretically, from a control design point of view, modern diesel engines are dynamic, nonlinear, MIMO (multiple-input and multiple-output) systems. This paper demonstrates that this assumption is not correct and a suitable model for predictive control (MPC) of power (torque), NOx and soot emissions based on on-line temperature and emission measurement is perfectly possible on SCR (Selective Catalytic Reduction) CRDITDs (Common Rail Direct Injection Turbocharged Diesel). The method optimizes the temperature at a selected point of the engine exhaust. This reference point is the turbocharger intake for Euro 0 (aircraft). For Euro 6+/US Tier 3a+ SCR diesels, the reference point is the intake of the "emission control system" usually at the outlet of turbocharging system. The traditional five-inputs are only theoretically independent. In fact, fuel injection duration depends on torque (load) and efficiency. Fuel advance is retarded to obtain the required reference temperature. HPEGR (high pressure exhaust gas recirculation) is adopted only when the emissions cannot be controlled by the fuel advance. VGT (variable geometry turbo) valve positions and low pressure LPEGR maximize the air flow (efficiency) at the engine intake. On the outputs, peak pressure and peak pressure derivative should be kept within structural limits. Soot and NOx are two faces of the same problems. In fact, high NOx means low soot and good combustion efficiency. Temperature and air flow are the keys to obtain optimum engine performance. Air flow is controlled by the turbocharger, while temperature depends on injection. This paper demonstrates that CRDITDs mapping is much easier when the fundamentals of diesel combustion and SCR are simplified to basic concepts. The strategy to retard the injection advance increases efficiency of 30% over traditional LPP mapping.

**REFERENCES**

[1] L. Piancastelli, L. Frizziero, E. Pezzuti. 2014. Aircraft diesel engines controlled by fuzzy logic. Asian Research Publishing Network (ARPN). Journal of Engineering and Applied Sciences. ISSN 1819-6608,

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9(1): 30-34, EBSCO Publishing, 10 Estes Street, P.O. Box 682, Ipswich, MA 01938, USA.

[2] L. Piancastelli, L. Frizziero, E. Morganti, A. Canaparo. 2012. Fuzzy control system for aircraft diesel engines, International Journal of Heat and Technology. ISSN 0392-8764, 30(1): 131-135.

[3] L. Piancastelli, L. Frizziero and I. Rocchi. 2012. Feasible optimum design of a turbocompound Diesel Brayton cycle for diesel-turbo-fan aircraft propulsion. International Journal of Heat and Technology. 30(2): 121-126.

[4] L. Piancastelli, L. Frizziero, N.E. Daidzic, I. Rocchi. 20136. Analysis of automotive diesel conversions with KERS for future aerospace applications. International Journal of Heat and Technology. ISSN 0392-8764, 31(1).

[5] L. Piancastelli, L. Frizziero, E. Pezzuti. 2014. Kers applications to aerospace diesel propulsion. Asian Research Publishing Network (ARPN). Journal of Engineering and Applied Sciences. ISSN: 1819-6608, 9(5): 807-818, EBSCO Publishing, 10 Estes Street, P.O. Box 682, Ipswich, MA 01938, USA.

[6] L. Piancastelli, L. Frizziero, I. Rocchi. 2012. An innovative method to speed up the finite element analysis of critical engines components, International Journal of Heat and Technology. ISSN 0392-8764, 30(2): 127-132.

[7] L. Piancastelli, L. Frizziero, G. Zanuccoli, N.E. Daidzic, I. Rocchi. 2013. A comparison between CFRP and 2195-FSW for aircraft structural designs. International Journal of Heat and Technology. ISSN 0392-8764, 31(1): 17-24.

[8] L. Piancastelli, L. Frizziero, E. Morganti, E. Pezzuti. 2012. Method for evaluating the durability of aircraft piston engines. Walailak Journal of Science and Technology. ISSN 1686-3933, 9(4): 425-431.

[9] L. Piancastelli, L. Frizziero, G. Zanuccoli, N.E. Daidzic, I. Rocchi. 2013. The "C-Triplex" approach to design of CFRP transport-category airplane structures. International Journal of Heat and Technology. ISSN 0392-8764, 31(2): 51-59.

**Symbols**

| Symbol | Description | Unit |
|---|---|---|
| $K$ | modifying coefficient of sound velocity | - |
| $E$ | flexibility modulus of diesel oil | Pa |
| $tw$ | time interval between pressure waves in long tube | s |
| D | external diameter of long tube | m |
| $d_L$ | internal diameter of long tube | m |
| $\mu$ | Poisson ratio of tube material | - |
| $a$ | sound velocity at measurement point | m/s |
| $L$ | length of long tube | m |
| $F_L$ | area of tube's internal cross section | m2 |
| $\rho$ | density of diesel oil | kg/m2 |
| $p$ | injection instantaneous pressure at the measurement point | MPa |
| $p_0$ | stable pressure in long tube before injection | MPa |
| $E_0$ | flexibility modulus of tube material | Pa |
| $t$ | injection duration | ms |
| $t_1$ | injection start time | ms |
| $t_2$ | injection end time | ms |
| r | Compression ratio | - |
| $\gamma$ | Heat capacity ratio | - |
| p | Pressure | Pa |
| T | Temperature | K |
| V | Volume | $m^3$ |
| R | Ideal Gas Constant | $m^3PaK^{-1}$ |
| $\eta_c$ | Carnot efficiency | - |