ISSN: 2277-3754
ISO 9001:2008 Certified
International Journal of Engineering and Innovative Technology (IJEIT)
Volume 3, Issue 6, December 2013

# Turbo compound systems to recover energy in ICE

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***Abstract***— ***The design of the powertrains for transportation on the road is even more oriented to the emission reduction. Recently, green-house gases commitments added new technological challenges.***

***Energy recovery from exhaust gases has a great potential considering the amount of mechanical or electrical work which could be generated on board. The paper considers the recovery which could be obtained from the exhaust gases expanding them in an additional turbine (turbo compounding). An engine model has been developed and validated thanks to an extensive experimental activity which concerned the F1C Iveco engine equipped with a Variable Geometry Turbine (VGT). Two potential technologies are presented and the recovery has been calculated by the model which behaves as a virtual engine platform. Energy recoverable has been estimated referring to engine operating points which reproduce the NEDC and the ESC13 approval cycle.***

*Index Terms*—**Energy, engine, ice, recovery**

## I. INTRODUCTION

Since many years the road transportation sector is facing a period of strong technological transformations aimed at reducing harmful emissions and, more recently, $CO_2$ emissions i.e. fuel consumption. The real technological surprise was that emission reduction has been reached without losing the expected mechanical engine performances (torque, speed, fun to drive, etc…). The results obtained were excellent: in the last two decades specific power (per unit of swept volume) increased of a factor of 1.5 while emission level of a factor of 10. The EURO limits cut emission levels progressively (and significantly) and technology followed step by step, producing cleaner and powerful engines.

Recently, a new driver appeared; this is related to the $CO_2$ emission which is directly associated to the fuel consumption reduction. Car manufacturers introduced immediately many new technologies to limit $CO_2$ emissions "to the root": downsizing, turbocharging, start and stop systems, tire pressure monitoring are just few of the measures which could achieve significant fuel savings. They are eroding the distance between actual $CO_2$ emissions per km (for passenger cars) and emission targets. Figure 1 shows the international commitments on $CO_2$ emissions for light duty vehicles and all the geographical contexts seems to converge to a common limit by 2025 close to 90 $gCO_2$/km.

One of the biggest challenge today to reduce $co_2$ emissions is energy recovery, a key sector because of the quantitative importance: exhaust gases of an internal combustion engine have an energy content equal to about one-third of the chemical energy of the fuel. Even a limited recovery of this energy into mechanical or electrical form would represent a significant contribution.[2-3]

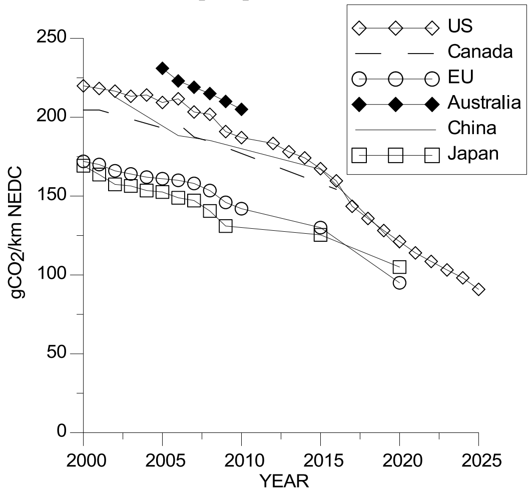


**Fig 1 $CO_2$ emissions in different regions [1].**

Energy recovery from exhaust gas has an additional advantage particularly important in the engine sector: Figure 2 reports the cost increase related to different technologies (which reduces CO2 emissions) per unit of CO2 not emitted: intervention on friction losses (oil improvement), tire rolling resistance, engine cooling and energy recovery from the exhaust gas are among the cheapest technologies, so expected to enter quickly into the market [4]. Hybridization technologies have the greatest CO2 potential reduction but at very high costs and they require a strong revision of the power train.

Exhaust gas leaving cylinders has two mains energy contributions: the first is related to its pressure, the second to its temperature. Therefore, work could be recovered:

- by means of a direct expansion, considering that gas pressure leaving the cylinders is greater than ambient pressure;
- By means of a heat recovery, being the gas temperature higher than ambient temperature. Heat recovered produces the vaporization of a high pressure organic fluid which expands and produces work inside a proper expander. The fluid is, then, condensed and pressurized, realizing a thermodynamic cycle (ORC, Organic Rankine Cycle) [5-6].

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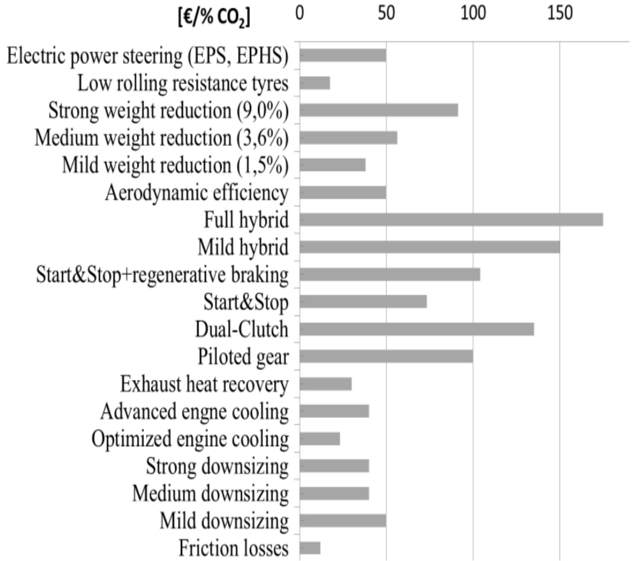


**Fig 2 Expected cost increase for a large diesel vehicle (from TNO Science & Industry - Support for the revision of Regulation (EC) No 443/2009 on $CO_2$ emissions from cars)**

The first type of recovery appears to be simpler from a technological point of view with respect to the second, which introduces an evident complexity when managed on board. First recovery, which has the nature of a direct recovery, is addressed in literature as turbo compounding, and it has been widely treated.

Lots of manufacturers (John Deere, Volvo, Caterpillar…), especially concerning heavy duty engines, had already applied turbo compounding, but greatly oriented to the possibility of having extra-mechanical power. Recently, in literature the possibility of having an energy recovery, thanks to the turbo compounding, in the form of electrical energy, has gained interest. Patterson et al. [7] made a review of what have been done both in mechanical and electrical turbo compounding, reporting all the concerns and the advantages of using such technology on board, and also examining the possibility of using this technology for generating electricity on "Power Gens" (stationary plants for energy production based on ICE) .

Hountalas et al. in [8] examined the possibility of both having mechanical and electrical turbo compounding, using an auxiliary turbine which recovers power downstream the turbocharging turbine, solution which can be seen as a different utilization of a two stage turbocharging. For an electric recovery, a significant difference is related to the positioning of the generator: when only one turbine is considered, it is inserted between turbine ad compressors with the additional benefit of speeding up the compressor to face turbo lag. Turbine efficiency can't be optimized for energy recovery. When it is done using an additional in series turbine, this could be designed inside the maximum efficiency range. New generator technologies have been presented in Michon et al. [9], making reference to the switched reluctance type and to a naturally aspirated gasoline engine. A similar application has been developed in Hyboost project [10], with a 3% of energy recovered on a urban driving cycle.

The presence of a direct energy recovery system influences Engine performances, [11], producing, as resulting situation, a backpressure at the exhaust.

In this paper the authors further explore the issue of energy recovery from an engine Iveco F1C Diesel 3.0L Euro 4 tested at the "Carmelo Caputo" Engine Laboratory at University of L'Aquila, Italy. In order to evaluate the potentiality of this recovery, a 1D model of the engine and the turbocharging group has been developed and calibrated on the cited engine, equipped both as passenger-LDV and HDV. Engine performances, as well as relevant data concerning the behavior of the turbo-charging system (flow rates, pressure drops, temperatures, VGT rack positions, etc…) have been measured and used to calibrate the model. Once it has been deeply calibrated, it has been used as virtual platform to calculate the energy recovery according to a sequence of engine operating conditions which reproduces NEDC and ESC13 approval cycles.

As in modern diesel engines, the Iveco F1C has a VGT, which allows the equilibrium of the turbocharger for a specified engine boost pressure. This is done acting on the closure degree of a fixed stage upstream the turbine.

The paper presents two technologies, named as DHR-1 and DHR-2, which consider:

a) An auxiliary turbine operated in parallel with the existing one, which is managed keeping always the stator stage upstream the turbine rotor at a high degree of closure. In this way, at higher engine speed, a part of the exhaust gas can be bypassed towards the parallel turbine;
b) An auxiliary turbine operated in parallel with a new proper designed fixed geometry turbine (which drives the compressor). This solution allows a preliminary estimation of the losses introduced by the existing turbine when it is operated in a fixed rack position (as in DHR-1).

Finally, the paper evaluates the $CO_2$ saving due to the recovery for the DHR-2 technology, when it is done into mechanical or electrical form. In the first case an iterative procedure is required because the recovery allows to reduce engine load for a given mechanical power delivered to the powertrain.

## II. MODEL DESCRIPTION AND VALIDATION

For the evaluation of the potential of the DHR, a mathematical model of the engine (1D/0D) and of the two technologies described above have been developed.

The model subdivides main processes as:

- Engine intake and exhaust;
- In cylinders transformations;
- Turbocharging system;

and reproduce the behavior of components usually present on board (pump, heat exchangers, ECU and its control laws, piping, etc…). A particular attention has been given to the

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ECU whose function has been simulated with all the look-up tables and control actions.

Concerning the engine processes, intake and exhaust phases were modeled reproducing the real situation (plenums, tubes, fittings, intake ad exhaust manifolds, distributed and concentrated pressure losses, valves, etc…). Quasi-steady processes have considered for the cylinder filling and emptying. The combustion process considers a multi – Wiebe model, which assumes that the fuel burned has a premixed, diffusive, oxidative phases. Woschni correlation describes convective heat transfer. A more detailed description of the model is given in [12].

Particular attention has been given to simulate the turbocharging system: real maps of the existing turbo-machines have been introduced, provided by the manufacturers, and the equilibrium point (revolution speed, compression and expansion ratio) has been predicted once an operating engine condition is specified.

Particular care has been used to model the variable geometry turbine, for which a complex extrapolation of experimental data provided by the manufacturer was necessary; the model makes use of a map, derived by the ECU, which defines a specific closing degree of a stator stage at the impeller inlet ("rack position"). The reconstruction of the turbine maps for the different rack positions was a concern and quality of the predictions (when compared with measured values) depends on turbine data when it is managed at different rack positions.

The model was calibrated using a wide experimental activity done on a 3L F1C IVECO Engine done having the engine mounted on a AVL EMCON 300 Dyno control system.

Fig 3a shows the prediction of the compression ratio (boost pressure) as a function of the engine speed, at 75% of the full load; the same figure reports the air inducted by the engine. The model is very precise at low speed: maximum deviation occurs towards high speeds (worst case close to 10%). Fig 3b reports the overall expansion ratio in the turbine: predicted values differ from the one controlled by the ECU of about 10% in the worst case, which is a good result considering the complexity of reproducing a variable geometry turbine.

As it is known a variable geometry turbine (VGT) provides greater turbine efficiency, reduced response times, good control possibilities and a greater flexibility of operation. A VGT is characterized by the possibility to change the opening degree of a stator stage just upstream the rotor, when the gas already crossed the scroll volute. For a fixed engine load, at higher speed the exhaust gas flow rate is high and requires an opened stage to avoid excessive kinetic power at the rotor inlet. When the engine speed decreases, exhaust flow rate is lower and the stator stage must be closed in order to ensure the right kinetic power at the rotor inlet. So, a part of the exhaust gas can be bypassed at higher engine speeds instead of a full opening of the stator stage. This bypass could allow an energy recovery inside a turbine which operates in parallel to main one driving the compressor.

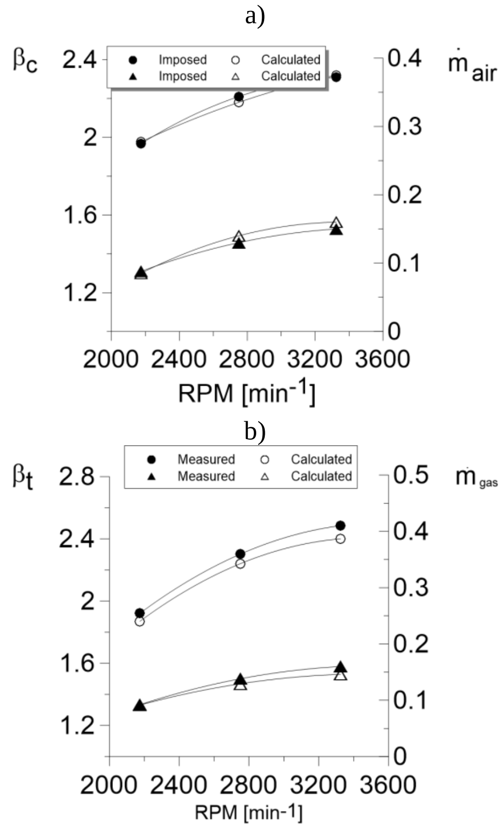


**Fig 3 a) Error between calculated data and data imposed by the ECU on board for compression ratio and air mass flow rate; b) Error between calculated data and measured data for expansion ratio and exhaust gas flow rate**

Two direct heat recovery technologies were designed in order to make use of the available exhaust gas flow rate, after have satisfied the power required by the compressor:

• DHR-1: semi-conventional solution performed using the variable turbine geometry installed on the engine with an additional turbine operating in parallel with the previous one. The existing turbine is operated with the stator stage kept at the maximum closing degree and the excess of the gas flow rate (mainly at higher engine speeds) is bypassed on the additional turbine (Fig 4). This solution requires the development of a map for the bypass valve opening, which has been done by the model. Engine boost pressure has been kept constant in order to have same engine load.

• DHR-2: the solution considers a new turbocharger equipped with fixed geometry turbine, which drives the compressor, and a further parallel fixed geometry turbine for energy recovery (Fig 4). The required boost pressure is obtained by controlling the mass flow which passes through the turbine which drives the compressor, while the remaining part of the flow goes to the recovery turbine. Even in this case, the map has to be implemented on the engine, which defines the rate of the main flow, leaving the residual flow on other turbine. As for the other case, this map has been determined in a virtual way. DHR-2 technology uses two fixed geometry turbine, for which efficiencies have been hypothesized in absence of real data for such application. In particular an isentropic efficiency

ISSN: 2277-3754
ISO 9001:2008 Certified
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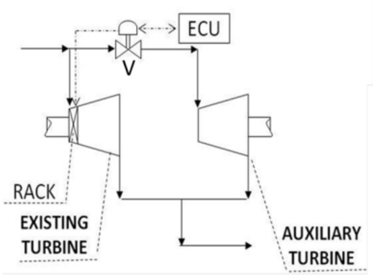
of 65% and 60% have been hypothesized respectively for the turbocharging turbine and the auxiliary turbine.

a)

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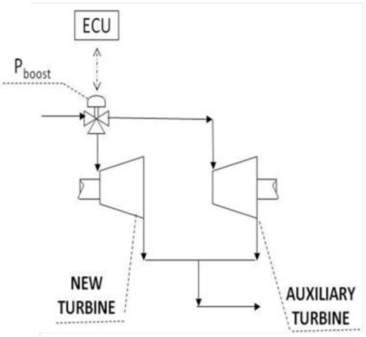


**Fig 4 Scheme of DHR-1 (a) and DHR-2 (b) technologies**

DHR-1 as a real experimental basis, thanks to the possibility to have, for the VGT, the real values as measured on the test bench. The same thing cannot be considered for DHR-2, which analysis can be seen as a completely theoretical first approach. The aim of the study of this arrangement is, however, to analyze if the recovery technology presents a considerable retrofitting aspect.

The potential of the two recovery systems were evaluated when the engine operating points are the ones characteristic of the NEDC cycle and ESC13 cycle, but in a more particular way in the last case, being the possibilities of the recovery much more promising.

NEDC cycle has been considered in this analysis only as a sequence of engine operating points, considered as a sequence of steady states. When considering transient condition, as it must be done when simulating NEDC cycle, further improvements to the study have to be done, mainly referred to the thermal transients in the exhaust line. Nevertheless it should be observed that the thermal transients of the exhaust gases are enough shorter than the overall engine thermal transient: this allows to consider the results obtained when referred to a NEDC as enough confident, considering that the exhaust line transients are not an easy issue.

In paragraphs that follow the simulation will be labeled as "NEDC" and "ESC13" only for easiness.

## III. SIMULATION OVER NEDC

The NEDC cycle specifies a given sequence of time and vehicle speed. The energy recoverable can be calculated once the engine operating conditions (which propel a given vehicle) are known in terms of engine speed and torque. When these two values are specified, the use of the engine virtual platform gives all the parameters which specify the DHR. So, a preliminary activity has to be done in order to evaluate engine speed and torque which will allow, for a given vehicle, the time-speed sequence specified by the NEDC cycle.

Fig 5 shows the engine load required to propel the considered vehicle through a NEDC. The Y-data are expressed in % of the maximum engine load, which comes from engine data once the engine speed is known.

The torque and speed derived previously were entered as input to the virtual model on which the two DHR technologies have been implemented.

For DHR-1, for simplicity, the auxiliary turbine has been considered as a fixed geometry turbine with an assigned adiabatic isentropic efficiency (60 %); the real efficiency values of the existing installed VGT turbine were used as they are.

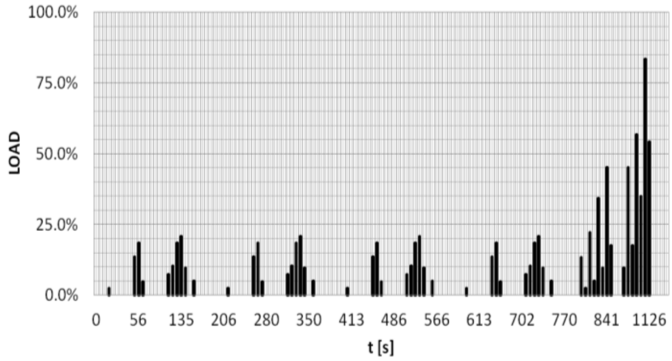


**Fig 5 Engine load on NEDC cycle**

The results of this procedure are shown in Fig 6 and in Table I.

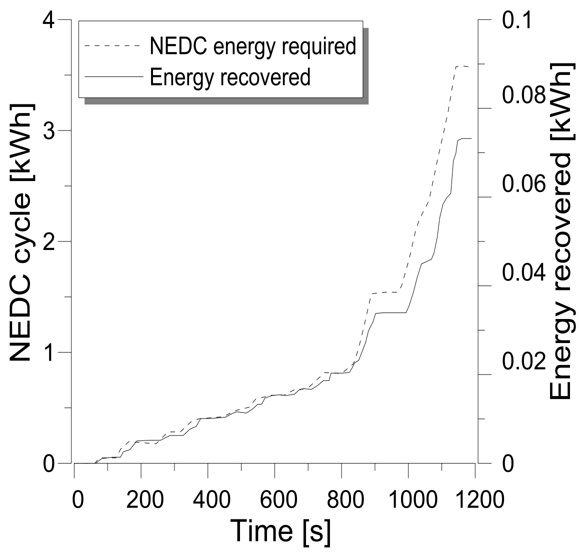


**Fig 6 Comparison between energy required for a NEDC cycle and energy recovered in different scales for DHR-1 technology**

Main results in Fig 6 are summarized in Table I:

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**Table I Summary of the performances of DHR-1 technology**

| PARAMETER | RESULTS |
|---|---|
| **NEDC Cycle Energy [kWh]** | 3.580 |
| **Mean Propulsive Power [kW]** | 10.922 |
| **Mech. Recovered Energy [kWh]** | 0.074 |
| **Recovered/Required [%]** | 2.07% |

The most interesting datum is the amount of energy recovery respect to the energy required to drive the NEDC: the estimation gives 2.07% which reflects on fuel consumption depending on how this energy is used. It should be noted that this DHR-1 technology does not require changing the actual (variable geometry) turbine.

The evaluation of the recovery using DHR-2 technology can be done only if a procedure based on a energy balance is set up. This procedure is organized as follows:

- Calculation of the specific work of compression, ($W_c$) knowing the boost pressure required by the engine (from its specific map, $\beta_c$). The values of compressor efficiency $\eta_c$ are derived from the actual maps:

$$W_c = \frac{\frac{k}{k-1} RT_1 \left[ \beta_c^{\frac{k-1}{k}} - 1 \right]}{\eta_c} \qquad (1)$$

- Calculation of the air flow rate entering the compressor and, therefore, the power needed by it according to:

$$\dot{m}_{air} = \frac{\dot{m}_{fuel} \cdot RPM \cdot \alpha}{60 \cdot \varepsilon} \qquad (2)$$

$$P_c = \dot{m}_{air} \cdot W_c \qquad (3)$$

- Calculation of the specific work required by the new turbine (without any rack control):

$$W_t = \frac{k}{k-1} RT_3 \left[ 1 - \beta_t^{\frac{1-k}{k}} \right] \eta_t \qquad (4)$$

- Calculation of the gas mass flow rate strictly required to balance the turbo:

$$\dot{m}_{\min} = \frac{P_c}{W_t \cdot \eta_m} \qquad (5)$$

The difference between the mass flow rate of exhaust gases of the engine (as shown by the simulation of the engine point) and $\dot{m}_{\min}$ (Eq. 5) is the mass flow rate that expands in the auxiliary turbine and recover energy.

The results of this procedure are shown in Fig 7 and in Table II.

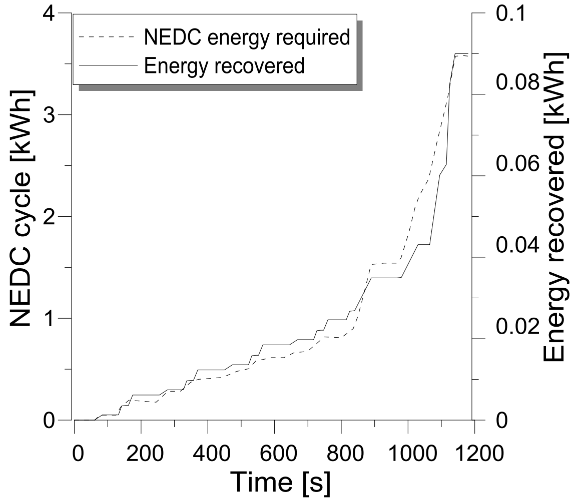


**Fig 7 Comparison between energy required for a NEDC cycle and energy recovered in different scales for DHR-2 technology**

Main results in Fig 7 are summarized in Table II:

**Table II Summary of the performances of DHR-2 technology**

| PARAMETER | RESULTS |
|---|---|
| **NEDC Cycle Energy [kWh]** | 3.580 |
| **Mean Propulsive Power [kW]** | 10.922 |
| **Mech. Recovered Energy [kWh]** | 0.090 |
| **Recovered/Required [%]** | 2.51% |

It is observed that the potential recovery rises to 2.51%, a value of great interest considering the simplicity of the technology. Furthermore the hypothesized efficiencies are lower than the values which can be obtained from technologies on the market. This recovery could reflect itself almost directly in fuel saving and a $CO_2$ saving too, and the amount of this reflections is related to the efficiency of the conversion of such energy in mechanical or electrical form. Even if some conversion efficiencies have to be considered, the potential of this recovery still remains very interesting.

## IV. SIMULATION OVER ESC-13

In previous paragraph the possibilities of the direct heat recovery technologies have been evaluated over the NEDC cycle, and all the results obtained have been reported.

This kind of applications shows their importance in HD applications, where the potential of the recovery arises just like the importance of the energy recovery in such automotive field.

As a matter of fact F1C engine is used also to propel heavy duty vehicles: for completeness of analysis additional simulations were carried out to define the extent of the energy recovery in some characteristic points of the ESC13 cycle. The results reported in this article are only relative to the

ISSN: 2277-3754
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simulations of the DHR-2 technology on the entire ESC-13 cycle, being this application the most interesting one.

The ESC13 cycle is represented by as a sequence of points whose engine load and duration are known. It is therefore easy to calculate the propulsive power, which was conducted as in the cases previously analyzed.

Fig 8 summarizes exhaust gas temperature and pressure at the turbine inlet, during this cycle. Each point has duration of 120 sec, but only the idle operating point has duration of 240 sec.

These modes were simulated on the engine virtual model described above so calculating all the data required to evaluate the DHR-2 potential. Results are shown in Fig 9 and in Table III.

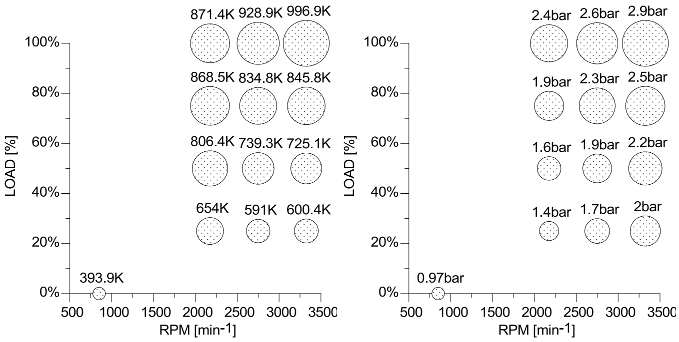


**Fig 8 Temperature and pressure at turbine inlet in ESC-13 cycle**

In Fig 9, the solid line represents the propulsive energy of the ESC13 and the energy available from the auxiliary turbine. Table III summarizes the most relevant data. They derive from the simulated 13-modes cycle: for each mode the recoverable energy using DHR-2 technology was calculated, multiplying the recoverable power for the time interval of each mode. The sum of these recovered energies, compared to the energy needed to propel the whole ESC13 cycle, offers an energy recovery of about 9%. The values of efficiency of gas turbines and compressor are the same kept for the previous DHR-2 case.

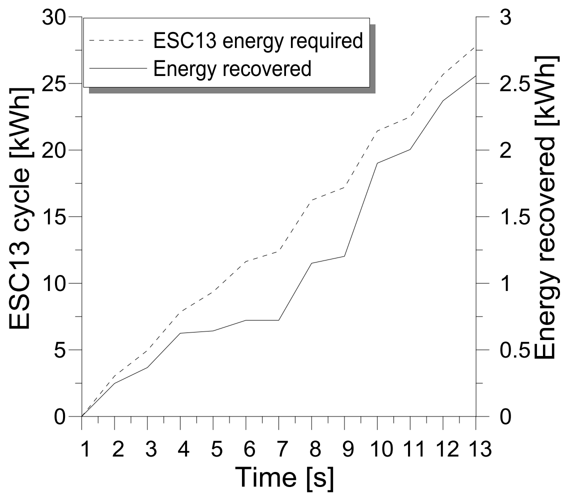


**Fig 9 Comparison between energy required for a ESC13 cycle and energy recovered in different scales for DHR-2 technology**

**Table III Summary of the performances of DHR-2 technology on ESC13 cycle**

| PARAMETER | RESULTS |
|---|---|
| **ESC13 Cycle Energy [kWh]** | 27.792 |
| **Mean Propulsive Power [kW]** | 59.554 |
| **Mech. Recovered Energy [kWh]** | 2.557 |
| **Recovered/Required [%]** | 9.20% |

This result demonstrates how the exhaust heat recovery is important in this kind of application. In particular this result is much more important than the corresponding approval for passenger cars, this because the ESC13 has working points which are higher than the NEDC ones, so exhaust available energy is higher too; another advantage is that the ESC13 represents a duty cycle closer to the reality than the other one. Furthermore the cost reduction due to fuel saving represents a very interesting prospective when moving freights, and from an environmental point of view $CO_2$ reduction are also very promising considering others technologies proposed for energy recovery which today are considered needed and market ready, and that have not such balance between energy savings and changes to be done on board.

All these aspects invited the Authors toward a further analysis which considered a possible utilization of this energy recovered. The procedure of this analysis is not reported here but it is presented in previous works [13-15], and here the results are reported. Concerning these aspects two possibilities were considered: (a) use of the energy recovered to propel the vehicle instead of using mechanical energy produced by the ICE; (b) use the recovered energy to produce electrical energy stored inside batteries and re-use on board as electrical energy. The intention was to evaluate with a deeper degree of precision the real net energy and $CO_2$ saved.

The first evaluation has been inspired by the long tradition that turbo compounding has in engine study, and it is also the most naturally way to re-use the mechanical energy recovered from the exhaust. Turbo compound allows the substitution of some mechanical energy from the engine with energy that comes from the recovery, and it means that the same power could be obtained with less fuel burnt.

The Table IV shows what is the difference in torque when the recovered energy is transferred via an hypothetic gear to the engine shaft: the engine torque is changed by a quantity that represents what is recovered in terms of torque, and this gives a difference in torque that can be converted in fuel saving and less $CO_2$ emissions.

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**Table IV "Engine torque saved" for each ESC-13 mode when a mechanical recovery is applied**

| Mode | Original torque [Nm] | Torque at convergence [Nm] | Δ |
|---|---|---|---|
| **1** | 0 | 0 | 0.0% |
| **2** | 400 | 372 | -7.0% |
| **3** | 200 | 189 | -5.5% |
| **4** | 300 | 278 | -7.3% |
| **5** | 200 | 198 | -1.0% |
| **6** | 300 | 291 | -3.0% |
| **7** | 100 | 100 | 0.0% |
| **8** | 400 | 363 | -9.3% |
| **9** | 100 | 95 | -5.0% |
| **10** | 365 | 322 | -11.8% |
| **11** | 91 | 83 | -8.8% |
| **12** | 274 | 249 | -9.1% |
| **13** | 183 | 168 | -8.2% |

The $CO_2$ emissions reduction needs the evaluation the fuel consumption for each working point, both after and before the energy recovery. The calculation of the fuel consumption requires the knowledge of the engine efficiency (Fig 10).

The regulation about the cycle ESC13 does not report any evaluation about $CO_2$ emissions, so an equivalent procedure has been used for such evaluation.

$CO_2$ emissions, so, were calculated adopting the methodology suggested by the ESC-13 normative for pollutant emissions, waiting for a specific normative.

$$CO_2 = C\frac{\sum(fuel \cdot weight)}{\sum(power \cdot weight)} \qquad (6)$$

In the equation 6 the C factor is a conversion factor, which is different as the fuel changes. In this case a $C_{13.5}H_{23.6}$ is considered, that means a C factor of 3.2 $g_{co_2}/g_{fuel}$. The initial value of $CO_2$ emissions is 696.5 g/kWh, while the "new" emissions are 642.8 g/kWh. The important reduction of 7.7% is reached when using this technology "directly" linked (via gears) to the crankshaft.

The second way to re-use energy recovered considers its transformation in electrical energy, its storage inside the batteries and its re-use in an electrical form. This availability would push toward a massive electrification of the auxiliaries (cooling fluid, oil, brakes, compressed air service, etc…).

Electricity on board is provided by the engine, and so it costs in terms of fuel: the energy recovered in electrical form means, therefore, a fuel and $CO_2$ saving.

For the electrical generation on board the "state-of-art" in the field of generation has been considered, and this consists in technologies like Permanent Magnets Generators or Switched Reluctance, that from our point of view are considered as an increase in efficiency. The saving estimated is equal to 7.8% in electrical form, starting from the mechanical recovery.

The $CO_2$ and fuel saving advantages when electricity is produced have been evaluated with respect to the same electrical energy produced in a traditional way by the engine itself. This calls for the knowledge of engine efficiency ($\eta_{eng}$) on each ESC-13 working point.

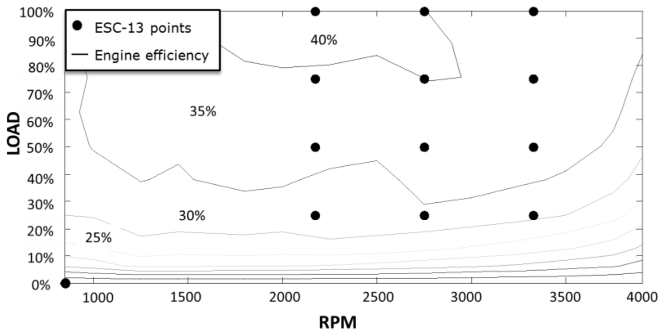


**Fig 10 ESC-13 modes on engine efficiency map**

Table V shows also the fuel required by the engine to produce this energy ($\eta_{alt}$ = 0.75, $H_i$ = 43.25 MJ/kg). If we consider that the same electrical energy will be produced using the auxiliary turbine, and not the engine, this fuel can be seen as equivalent to saved fuel, having in output the same electrical energy but in input no burned fuel for this production.

**Table V Fuel request scenario for electrical generation in ESC-13 cycle**

| Mode | $\eta_{eng}$ [%] | $E_{rec\text{-}el,i,}$ [kWh] | $E_{rec\text{-}mec,i,}$ [kWh] | Fuel required [g] |
|---|---|---|---|---|
| **1** | 0% | 0 | 0 | 0 |
| **2** | 42% | 0.25 | 0.33 | 66 |
| **3** | 38% | 0.12 | 0.16 | 34 |
| **4** | 40% | 0.26 | 0.34 | 71 |
| **5** | 37% | 0.02 | 0.02 | 5 |
| **6** | 39% | 0.08 | 0.11 | 23 |
| **7** | 34% | 0.00 | 0.00 | 0 |
| **8** | 40% | 0.43 | 0.57 | 119 |
| **9** | 34% | 0.05 | 0.07 | 17 |
| **10** | 37% | 0.70 | 0.93 | 209 |
| **11** | 31% | 0.10 | 0.14 | 37 |
| **12** | 38% | 0.37 | 0.49 | 106 |
| **13** | 37% | 0.19 | 0.25 | 56 |

The engine without energy recovery requires 6.1 kg of fuel to cover ESC-13 and 0.7 kg of fuel to produce an equivalent electrical energy (recovered by the DHR-2). This means that a DHR-2 is responsible of a saving up to 12%. A similar saving can be obtained referring to $CO_2$.

## V. CONCLUSIONS

The paper makes reference to an existing engine (IVECO F1C, 3L) equipped both in LD and in HD mode and with a VGT turbocharger, extensively tested in order to validate a comprehensive mathematical model. The use of this virtual platform allowed the knowledge of all the variables which were needed in order to estimate the potentiality of the recovery.

The paper discusses two recovery configurations having different potential but characterized by greater engine modifications. The first keeps the same turbine driving the

ISSN: 2277-3754
ISO 9001:2008 Certified
International Journal of Engineering and Innovative Technology (IJEIT)
Volume 3, Issue 6, December 2013

compressor with its VGT. The second replaces the actual turbine with a new one without any inlet.

It is clear that a VGT has some advantages in terms of engine performance with respect to the one equipped with a FG turbo (higher PMI for each RPM), but this advantages strictly depends on the engine design. Analyzing DHR-2 configuration shows the advantages and the possibilities of a retrofit, investigating the advantages of these modifications on not-so-advanced engines. Applications on VGT, instead, can be seen as further possibilities; in fact, often VGT has good results on engine low points, but in high points advantages are slight with respect to the FG turbo. In these points, perhaps, an utilization as energy recovery technology for DHR-1 could be convenient, so in each point a good compromise between engine performances and energy recovery has to be found.

The extent of the recovery was evaluated using a full simulation of the engine and recovery technologies, over points that reproduce the NEDC cycle (for passenger and light duty vehicles) and the ESC13 cycle (for heavy duty applications) operating points.

The first technology has a limited degree of recovery close to 2% evaluated on the energy required to drive the NEDC cycle: this issue is reached keeping the same actual VGT technology, so it is characterized by a very limited engine (and vehicle) changes. This value of the recovery grows till values next to the 2.5% using the DHR-2 technology over the same cycle, showing results of great interest in this research field.

More interesting is the analysis of the same technological solutions for the ESC-13 cycle that is an approval cycle with advantages over the NEDC:

- ESC-13 presents working points higher than the NEDC;
- ESC-13 has a duty cycle close to the real use of the vehicle

For these reasons he simulations were conducted deeply over this approval, showing the result that, using a DHR-2 tech, mean energy saving is next to 9 % of the overall propulsive energy. DHR-2, however, needs a more detailed study, particularly in evaluating the overall efficiency of the system, but results seems to be promising and calls for a real experimental activity focused on. According to a different use of the mechanical energy recovered, it followed:

a) if the energy is used for propulsion, the $CO_2$ saving is about 8% of the original emissions (without any recovery), a similar saving is achievable on fuel consumption;
b) if the energy is transformed in an electrical form a fuel saving of 12% is reached with respect to the fuel consumption required for propulsion and electrical energy (in the traditional engine). Same result applies to $CO_2$.

Considering the simplicity of the turbine control and also the technology which is proven and characterized by low cost (increase) and high reliability, the DHR recovery has the entire characteristic to be ready to market.

## VI. NOMENCLATURE

| | |
|---|---|
| $W_c$ | Compressor specific work [kj/kg] |
| $k$ | Adiabatic coefficient |
| $R$ | Gas constant [J/kgK] |
| $T_1$ | Compressor inlet temperature [K] |
| $R$ | Gas constant |
| $\beta_c$ | Compression ratio |
| $\eta_c$ | Compressor efficiency |
| $\dot{m}_{air}$ | Air mass flow rate [kg/s] |
| $\dot{m}_{fuel}$ | Fuel mass flow rate [kg/s] |
| RPM | Engine rotational speed [$min^{-1}$] |
| $\alpha$ | Air fuel ratio |
| $\varepsilon$ | Crankshaft throw |
| $P_c$ | Compressor power [kW] |
| $W_t$ | Turbine specific work [kj/kg] |
| $T_3$ | Turbine inlet temperature [K] |
| $\beta_t$ | Expansion ratio |
| $\dot{m}_{min}$ | Equilibrium mass flow rate [kg/s] |
| $\eta_m$ | Mechanical efficiency |
| $\eta_t$ | Turbine efficiency |

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He was responsible of many research projects with private national and international companies, National Public Authorities, European Community, Private Research Centers concerning energy planning, environmental opportunities related to the energy production, energy conversion systems, etc… Since 10 years, he is responsible of a relevant national Research Program concerning pollutant reduction in combustion engines. Prof Roberto Cipollone published about two hundred scientific papers, chapter of books, books, etc…

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