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TURBOCHARGING AND TURBOCOMPOUNDING OPTIMIZATION IN AUTOMOTIVE RACING

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ABSTRACT

Turbocharged spark ignition engine for automotive racing have a long and controversial history. From the times of high torque at all cost, to the actual F1 era of maximum efficiency. However turbocharging and turbocoumpounding basic concepts have not changed. It is surprising that, through the years, the same identical errors are repeated. Turbocharger (TC) unit design is a highly optimized task, that requires good concepts, good mathematical models, lots of experimental data and a very good optimization. Performances vary completely with design choices, with big differences between even close solutions. Present software for mathematical modeling of performances are far from accurate and should be corrected with experimental data to obtain effective results. Genetic Algorithms are to be used as optimization method to evaluate the best design solution. Even minor errors in design choices result in large penalties on performance.

Keywords: Turbocharger, automotive racing.

INTRODUCTION

The turbocharging for racing car is still a technique to amplify engine performance. The application of the turbo in F1 was initially pioneered during the late '70s. In that day, with primitive electronics and very large TCs, turbolag was controlled in many ways. Enormous technical advance was made in those few years, with the definition of concept that still holds, since the materials and the devices have been greatly improved, but the principles remain the same. In this paper two different subjects will be treated separately: the turbocharging and the turbocompounding. Both concepts were exploited starting from the '70s and the research have been carried out by a few companies to present days. In spark ignition engines turbocharging is done for torque and for emissions, for reasons that will be explained later, efficiency can be brilliant, but optimization possibilities are limited. Turbocompounding makes it possible to obtain higher efficiencies but the results are highly dependent on design choices. This choice is linked to the use of the engine. In the case of car racing, circuit geometry should be considered and vehicle performance should be simulated to obtain acceptable result. Simulation accuracy is a problem, since present software may give misleading results.

The racing engine: a torque producer

During the 90', in Italy we had fantastic naturally aspirated racing engines. This V12 - 5 valves machines, with extremely high crankcase speed outputted on the brake fantastic power outputs, well above the V10-4 valves of the other foreigner adversaries. However, results fatigued to arrive. Lap times were not in line with output results. Frame, aerodynamic and tires were blamed for this lack of satisfaction. However it was the torque concept that lacked. We didn't understand that was torque in the right position that was to be sought and not only sheer power. From the same know-how and obviously the same physics, the more the displacement is fractioned the more the diameter of the valve ducts are small. 3 intake valves instead of 2 make the situation even worse. In a very simplified and approximated way, the highest torque CS depends on the Reynolds number in natural aspirated engines:

$$\mathrm{Re} = \frac{\rho c D}{\mu} \tag{1}$$

If D (intake valve duct equivalent hydraulic diameter) is reduced, in order to preserve the same Re of maximum efficiency (maximum torque), the speed c should be increased. This is a simplified explanation why larger unitary displacement is usually characterized, in Naturally Aspirated engines (NA), by maximum torque at lower Crankshaft Speed (CS). The maximum torque position is then a design variable that should be chosen from the type of circuit and vehicle you will use

Supercharging

In natural aspirated engines, volumetric efficiency and tumble are the main variables of a good head design. Resonant conditions may be sought, but it is a technique extremely difficult to master efficiently. In supercharged engines it is a completely different world. Just to make an example, sometimes ago, we were comparing the swirl map of the 2V common rail diesel with another "new" one. An expert of natural aspirated engine immediately detected that the new one had a better volumetric efficiency. This concept proved to be irrelevant on the performance of this new Common Rail Direct Injection Diesel (CRDID), since it had an offset in the "centre of swirl" and the centre of the injection bowl where the injector is placed. The volumetric efficiency "deficiency" of the old 2V CRDID was largely compensated by an overboost of 0.1bar, while the swirl offset gave large problems on the brake. So the volumetric efficiency does not depend only on geometrical considerations but also on the difference between intake

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and exhaust pressure, which is critical in turbocharged engines and varies during the lap. Given combustion chamber, intake and exhaust geometry, a certain fuel, a temperature map....the maximum pressure is limited by the fuel performance and the peak pressure pattern. It is not a mystery that the turbocharged F1 Ferrari engine had a 30% increase of performance, just by switching from ordinary fuel to a special fuel that had been prepared for aircraft racing for a tuned WWII piston engine. This later part may be a myth, but the 30% power and torque increase is true. Given a fuel, the maximum pressure depends on the geometric pressure ratio, so the lower the better for performance. The maximum pressure should be given by boost, because this increases the Air Charge (AC), the Fuel Charge (FC) and the torque. However, in the following paragraph, it will be seen that also the geometric pressure ratio is a design variable. A main improvement factor in supercharging technology is the aftercooler. In F1 the continuous struggle between aerodynamic and engine people have led to elevated temperatures for coolant and lubricant. This choice is justified by the Meredith effect additional thrust that is higher with high temperatures. However, as it was seen in WWII fighters, for example the P51H, aftercooler output temperature is critical and should be kept as low as possible, being the benefits in terms of fuel charge and performance extremely relevant. In fact temperature affects engine cooling, since fresh air is a main cooling agent for pistons, detonation and output torque. Being piston cooling and detonation main problems in supercharged spark ignition engines. Cooler AC means higher combustion temperatures, being temperature the main factor for efficiency. In fact detonation is outside the combustion volume. It should be clear that spark ignition engines have a hot volume of exhausts that is delimited the igniting surface by the piston and by the head interfaces with the combustion chamber. So, more fuel is injected and more heat is transferred to the piston to the head and to the cylinder walls. While the head and the cylinder wall are efficiently cooled by the liquid coolant, the piston is cooled by the oil spray and by the fresh AC that comes from the intake valves. Since the hot gases transfer the heat as an heat flux, the longer the time of exposure to the hot gases the more the amount of heat transferred to the surrounding surfaces. This is the reason why, in racing engines, it is convenient to have high CS. In fact, the engine is more adiabatic and the percent quantity of heat in the exhaust is higher. This means, that given a technology of piston manufacturing, the cooler the piston the higher the allowable temperature of the hot gases and the higher the torque output. It should be pointed out that the specific fuel consumption or engine efficiency can be written in the following way (2):

$$\eta_{tot} = \eta_{Carnot}\eta_{therm}\eta_{mec} = \frac{T_{\max} - T_{outside}}{T_{\max}}\eta_{therm}\eta_{mec} \tag{2}$$

So fuel consumption directly depends on T_{max}, the higher during the lap the better. Aftercooler efficiency is then fundamental for efficiency and fuel consumption. On the other side, higher exhaust temperature are favorable for energy recovery especially in turbocompounding.

Turbocharging

If the supercharging is obtained by using a turbine connected to a compressor it becomes turbocharging. The main advantage of turbocharging is that exhaust temperature is partially recovered as useful work by the engine. In fact for turbines it is possible to write (3) (4) (5) (6) (7):

TPR = pt5 / pt4 <= 1.0 (3)

Tt5 / Tt4 = (pt5 / pt4) ^ ((gam -1) / gam) (4)

TW = ht4 - ht5 (5)

TW = cp * (Tt4 - Tt5) (6)

TW = (nt * cp * Tt4)* [1 - TPR ^ ((gam -1) / gam)] (7)

From equation (3) it can be seen that the available turbine energy TW depends on the enthalpy difference between exhaust and external air, which is proportional to exhaust temperature Tt4. The higher the exhaust temperature the better for energy recovery. For this reason more energy is available in spark ignition engines than in diesel ones, in fact air to fuel is higher in diesel than in spark ignition engines. The price to be paid is overpressure in the exhaust pt4. If the intake pressure is higher or equal to the exhaust pressure the engine will have a larger of equal air volume available for combustion. Thanks to the compressor the air density is higher and the output power higher.

Turbines

Centrifugal turbines are used due to their better off-design performance than axial turbine that are more efficient. However, the energy available for an the turbine is so high, that in turbocharging the efficiency is usually very low, in fact a lot of energy is still wasted through the exhaust. Just to have an idea,if the exhaust temperature is Tt4=950°C, the energy available at the exhaust for every kg of intake air will be (8):

$$TW = ht4 - ht5 = 1323.85 - 287.96 \approx 1000 \tag{8}$$

While the energy necessary for the ideal adiabatic compression with a compression ratio of 3 is only 320 kJ/kg. Even by considering compressor and turbine efficiency, the energy "wasted" through the exhaust is still very high.

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Compressors

If we take a F1 2014 engine and we look at the Garrett Catalogue a reasonable TC may be the GT3017R of Figure-1.

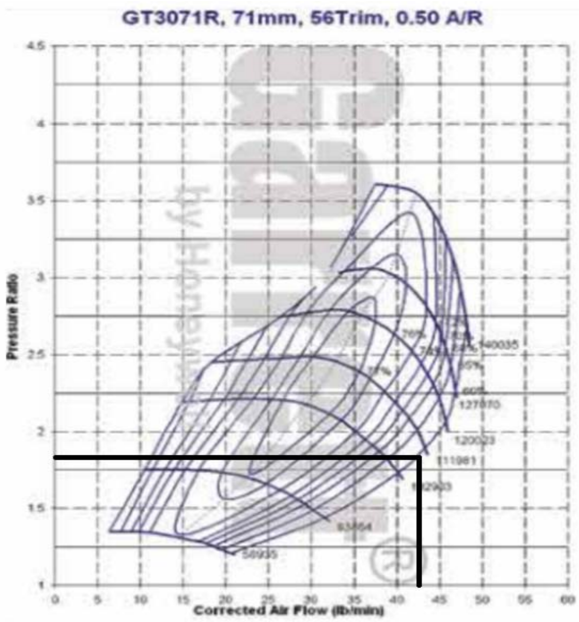

Figure-1. A compressor map, just an example, far from the ideal choice for a F1 racing car.

The engine will have the best efficiency at 25 lb/min that means, theoretically, the higher torque at 6800 rpm. This value and a curve may be added to the mathematical 1D model of the engine to be virtually tested on the car to evaluate car racing performance on the different circuit. However compressors are tricky machines. The first consideration is that the blade-to-housing gap is important. In racing compressors the gap is dug directly by the compressor during its work, as the temperatures vary during the races, the gap widens and the efficiency varies. A second consideration is that this map is interpolated by a few points effectively measured. Holes and variations present in the real TC are not visible in the diagram. Even in steady state tests and with an extremely refined 1D model you will not replicate the true engine performance in every point. The third and most important consideration is that it is a static map. As racing cars are always in transient, this map will never be reached. As the pilot commands full throttle at "low" rpm, the TC may be at very low rpm, below the map limit or in a very border area. In this condition the exhaust gases act on the turbine blades with a large speed/pressure difference, this will produce cavitation in the unloaded part of the blade and hot gases will flow from the high pressure side to the low pressure side of the turbine that has a significant play from its housing. Even it is not a true surge situation it will somehow resemble it even if you are not outside the "static" maps. The net result is a very inefficient situation for the turbine and an increase in back-pressure on the exhaust; this will reduce the air mass flow and reduce exhaust gas speed. The turbine will start to rotate faster, the back pressure will be reduced and the compressor will began to create suction in the intake duct. After a while the air mass will be accelerated toward the compressor inlet. This process is reapeted with damped oscillation until the stable "static" condition is reached. However, normally, due to continue variations of throttle and load, this stable condition is normally never reached, so transient is the normal working condition for a turbocharged racing car. This transient phenomenon is more marked when the increasing of: the difference of ideal-to -true TC speed, engine load and TC inertia. That is the most important factor. The net effect is that your compressor map will be moved to right, with the surge line being translated and the choke line remaining where it is. So your TC will be always too large and the maximum torque will be at higher rpm than calculated. When you need torque, at the exit of bend on a steep climb, the dynamic compressor map will move further left, effectively increasing the turbolag in the time domain. These dynamic maps are not available and it is not easy to calculate it. Experimental racing data are needed to find an acceptable match. In other words, your TC will lazily follow your engine, with a retard called turbolag. Turbolag cannot be eliminated; it can be reduced and someway controlled, but not eliminated. The best way to improve the situation is to keep TC and engine related inertias as low as possible. Fundamentally your TC is never too small and your ducts are always too long. The best thing to do it is to try to increase TC rotational speed. It should be noted that, usually, TC manufacturers take huge safety margins in their design, as it was discovered in the explosion containment test during the certification of the Dieseljet 1900jtd. Even in case of not high safety margin, it is possible to rework both the compressor and the turbine wheels in order to contain the inertia and increase the rotational speed. However, this will worsen the static fluid dynamic and the efficiency of the compressor and of the turbine. Care should be taken to the surface of compressor and turbine, since etching is possible to improve dynamic behavior. Journal bearings are not a limit and they can be upgraded through tolerance control and surface treatments. Low viscosity lubricants are also beneficial as they are already used in racing engines for low friction.

Turbolag correction

From the first turbo F1 era back in '70s several tricks have been adopted to correct this situation. In that day the TC available were extremely large, the small TCs became to be available when the turbo era ended. For this reason brilliant solutions were adopted. At low engine rpm, since the TC is still at low rpm, the wastegate was kept open and a bypass was introduced in the intake to make the engine work as a NA engine. On braking the pop off valve was opened at the intake (in a few cases even a bypass was opened from the intake to the TC inlet) and additional fuel was added to the exhaust into the turbine to accelerate the TC, this strategy reduces the turbolag, but the pilot should anticipate the acceleration in order to have the best response from the engine. At that age the electronic began to be truly important. Ing. Naldini of Magneti Marelli manufactured for Ing. Forghieri's Ferrari, the first electronic system to actuate the waste gate. Intake and exhaust were keep as small as possible since gas inertia is a destabilizing factor in the system and they contribute to turbolag. Pressure pulse is also used to improve the pressure build up. A common error is to increase the turbine diameter. This is possible since the turbine has a lot of energy available and theoretically a larger turbine with output larger torque to the compressor and will improve the dynamic behavior. This is a false answer, since the larger turbine not also will increase inertia, but will also increase the instability region of transients. This will increase turbolag and worsen the situation. Another important point to be considered is that turbocharger dynamic is affected by turbine dynamic, so turbine exhaust should be tuned to the application. If there is no room In particular very short and straight piping are convenient. If it is possible, long and divergent exhaust have also been successfully tested (see Figures 2 and 3), with the trumpet shape showing the most favorable behavior. The advantage of these solutions is that kinetic exhaust energy can be partially recovered, so reducing the engine exhaust backpressure and improving VE.

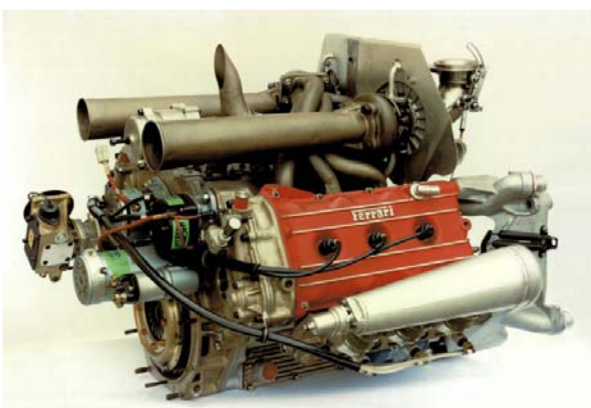

Figure-2. Ferrari F1 1981. Basic concepts are still valid, straight divergent exhausts are clearly visible.

As it can be seen in Figure-2, the best F1 turbocharged engine of the 1981 season had all the basic concepts: straight divergent exhausts, the small and straight exhaust ducts from heads to the enormous turbochargers available at that time. Look at the funny “monster” wastegate on the left hand side and the “primitive” electronic actuator. The pop-off valves on the intake are also visible. Figure-3 shows an engine that lived its glory due to a right choice of fuel; however its large unitary displacement with cooling and detonation problems limited performances. Again, see the straight exhaust and the enormous twin scroll TC of that age.

Figure-3. Tturbocharged BMW F1 engine 1981? See the dynamic air intake.

Turbocompound

With the massive arrival of the electronics in the automotive racing fields, the idea to "move" the map at choice by adding a "small" electric motor to the TC became attractive. This solution was initially installed on truck turbines where rotational speeds are smaller. The highly boosted diesel engine has even larger exhaust energy at disposal. This means that it is possible to add an additional electric turbocompound system that outputs energy when working at fixed point on motorways and reduces turbolag when needed. Caterpillar introduced this concept some years ago in a presentation (Figure-4) [1]. Many other manufacturers worked on it more or less secretly. Finally, Volvo an Scania introduced a mechanical turbocompounding systems with a clutch. In the case of Volvo (Figure-5) the TC shaft was mechanically coupled to the crankcase, while the Scania (Figure-6) solution had a fluid dynamic coupling with an additional turbine. Power levels are not far from a modern F1, with the additional fact that exhaust energy for diesel is limited by exhaust temperature that are usually around 750°C, while spark ignition racing engines arrive easily at 1050°C and over with special alloys or with turbine cooling. The big advantage of large truck diesels is that the TCs work in the area of 75,000-150,000 rpm. In this region the solutions of Figure 2, 3 and 4 are possible. Racing cars with their requirement of brilliant dynamics should use smaller TCs with speeds up to 400,000 rpm. In this range only electrical or fluid coupled turbocompounding are possible. An electric solution (ETC) is depicted in Figure-4.

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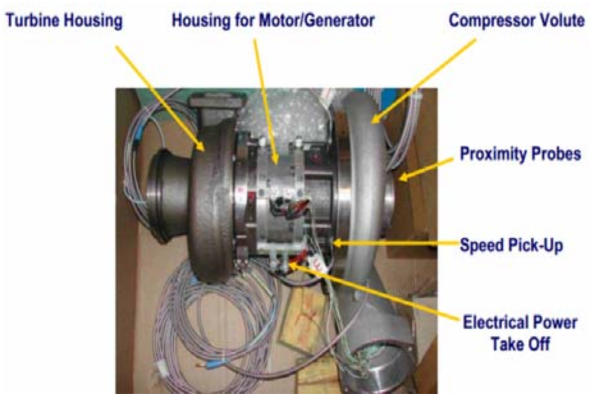

Figure-4. Electric TC (ETC) from Caterpillar, 2004 [1].

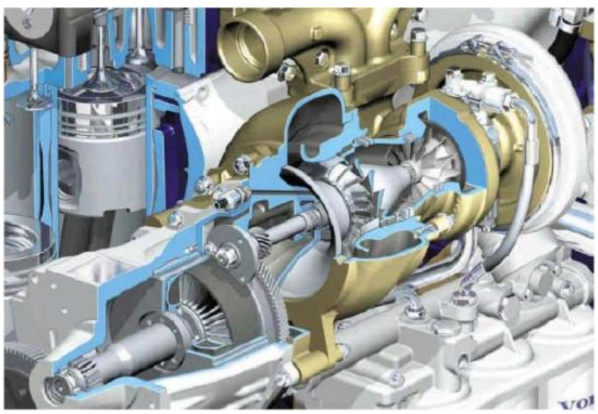

Figure-5. Volvo D12D 500HP Euro3 turbocompound, the TC is mechanically coupled to the output shaft through a clutch.

The lack of success of ETC and in general of turbocompounding in the truck field is due to several factors linked to emissions, costs, reliability and weight. ETC have been recently forcibly introduced in the F1 rules with a lot of work and expenses by the leading F1 engine manufacturers.

Basics of ETC turbocompounding

The main problems of ETC are dynamic performances and efficiency. For dynamic performances nothing changes from normal turbocharging. For efficiency it should be kept in consideration the fact that is not the ETC-battery-motor/generator system the efficient one. A TC with its centrifugal compressor and turbine has, from the thermodynamic point of view a very limited performance, with true thermodynamic cycle efficiency below 10%. The situation gets even worse when you consider the generator efficiency and the battery charging efficiency. The nimble motor/generator installed on the turbine has so many design constraints that its efficiency rarely reaches the 90%. The high C level charging of the battery has a very low efficiency too bringing the total ETC system total Figure to very low values. However, it should be taken in mind, that the ETC takes energy “out of the dustbin”. Turbocompounding partially recovers the thermal energy of exhaust. The low ETC efficiency means that the ETC should improve at first the spark-ignition engine efficiency and, when possible, without affecting the dynamic response of the main engine, it should recover energy. In fact the ETC may improve the dynamic behavior of the turbocharged engine performance by pulling the dynamic TC compressor maps to the left and keeping the working point of the engine in the high efficiency area.

Optimization parameters for the ETC turbocompound

With the modern simulation software it is mandatory to run the most accurate possible simulations of the vehicle engine system. Only in this way it is possible to obtain good performance results. Since efficiency is crucial, for high efficiency it is necessary to keep the compressor in the high efficiency zone as long as possible during the lap. The electric motor of the TC is to be used for this purpose. It is not so important in the straight part of the circuit as in the more tormented part of it. In fact low speeds means low aftercooler efficiency, aftercooler efficiency is critical for various aspects, from engine cooling, to FC (Fuel Charge), to detonation. This factor affects directly the maximum cycle temperatures and, as it can be seen from (2), the spark ignition engine efficiency. For this reason also aftercooler efficiency should be optimized. The turbomatching of the TC is of primary importance, being the smaller the better. Another important factor is backpressure on the engine exhaust that affects both VE and the dynamic behavior of the system, with a direct influence on turbolag and overall efficiency. Pressure loss should be carefully reduced, narrow bends should be avoided and the exhaust from the turbine should be as straight as possible and, if it is possible divergent and trumpet shaped. Not so much can be done in the intake part of the TC. A dynamic inlet is obviously advisable and a compact high efficient aftercooler should be adopted. These component are more or less of-the-shelf since a lot of work have been done for the optimization of both. However, high stress should be given to the air ducts to and from the aftercooler. Differently from lubricant and coolant temperature, that can even output and additional thrust to the vehicle, the aftercooler temperature directly affects the efficiency and the fuel consumption in the most adverse way. Exhaust gas temperature should be optimized for the spark-ignition engine and not for the TC. Remember that the Turbine can survive up to 1400°C if properly manufactured and cooled, so is the spark ignition engine that should work at its optimum efficiency, the ETC, being a low-efficient-waste-recovery-system, should follow. The optimization is critical and an optimization algorithm and a good mathematical model of the vehicle-engine system on various circuits should be performed. It is highly advisable to use the Genetic Algorithm for this purpose. Multiple variable optimizations should be performed, being the fuel consumption the parameter to be optimized. Finally a low inertia alternative to the truck-derived Caterpillar solution is proposed in Figure-6. In this

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Figure a turbocompound for high altitude aircraft application is introduced. In this case at low altitude an external burner is used instead of the exhaust. The motor/generator has its own shaft, with a magnetic coupling that connects the two shafts. This solution is ideal, since the electric motor can have a very low inertia and the dynamic behavior of the ETC can be improved. Also motor/generator cooling is improved since it is installed inside the air intake. The shorter shaft of the TC reduces torsional vibration problems and shaft diameter improving the TC bearing efficiency. Finally, the absence of mechanical coupling between motor generator and TC improves dynamic performance, allowing a damped asynchronous behavior of the two machines.

CONCLUSIONS

The TC/ETC follows the spark ignition engine with turbolag. Low inertia, low friction, straight ducts (especially on the exhaust) are fundamental to obtain good performances. The optimization is aimed to best spark ignition efficiency. This is obtained through optimum efficiency of compressor and aftercooler. The dynamic compressor maps should be taken into account and a multiple variable Genetic Algorithm optimization should be carried out to tune the design variables. The aim to obtain a high efficiency of the ETC system is not so important, since ETC efficiency is in any case far lower that the spark ignition engine one.

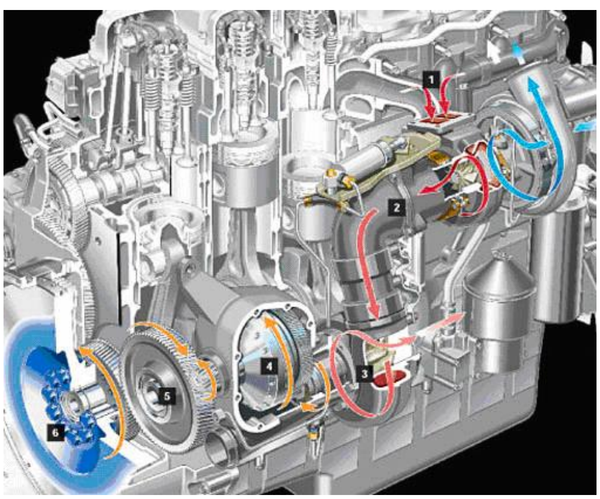

Figure-6. Scania turbocompound, the power turbine is gas coupled to the TC.

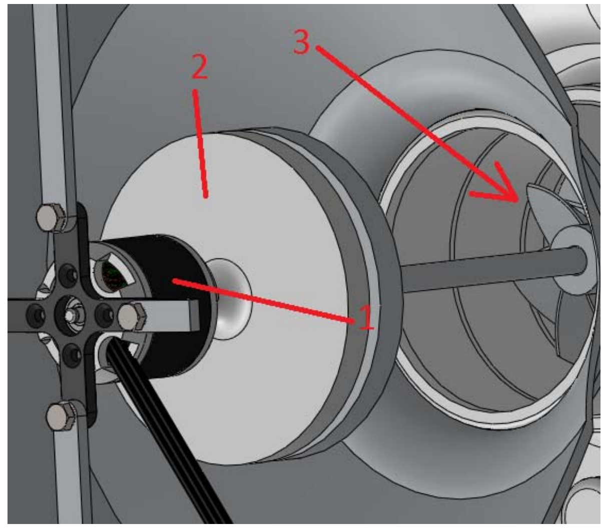

Figure-7. Schema of double shafts ETC.
1-Motor/Generator. 2-Magnetic Joint. 3-Compressor intake.

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Symbols

Symbol	Description	Unit
τ_{ign}	Ignition delay	Ms
p_o	Combustion chamber pressure	MPa
T_o	Combustion chamber temperature	K
Δp	Pressure difference between rail and combustion chamber pressure	Bar
ρ	Fuel density	Kg/m^3
ξ	Concentrated loss factor in injector	-
Q	Volumetric flow	m^3/s
A	Injector nozzle area	m^2
FC	Fuel charge for single injection	m^3
η_{Carnot}	Carnot cycle efficiency	-
η_{therm}	Thermal efficiency of the true engine cycle referred to the Carnot one	-
η_{mech}	Mechanical efficiency of the engine	-
η_{engine}	Efficiency of the engine	-[gr/HPh]
T_{max}	Maximum Temperature in combustion chamber	K
$T_{outside}$	Outside air temperature	K
V_{cc}	Combustion chamber volume	m^3
P_{peak}	Peak chamber pressure	Pa
TPR	Total pressure ratio	-
Pt4	Turbine inlet pressure	Pa
Pt5	Turbine outlet pressure	Pa
Tt4	Turbine inlet temperature	K
Tt5	Turbine outlet temperature	K
ht4	Turbine inlet specific stagnation enthalpy	kJ/kg
ht5	Turbine outlet specific stagnation enthalpy	kJ/kg
TW	Specific turbine work	kJ/kg
cp	Specific Heat	kJ/kgxK
nt	Turbine adiabatic efficiency	-
gam	Specific heat ratio	-
τ_{ign}	Ignition delay	ms
p_o	Combustion chamber pressure	MPa
T_o	Combustion chamber temperature	K