New engine concept development-Integrate engine design methodology

Antonio Turturro – Supervisors: Prof. C. Delprete and L. Magro

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PART II
# INTEGRATE ENGINE DESIGN METHODOLOGY

In the first part of this thesis, one point up engine architecture used in sedan segment and in particular IC engine with $1 - 3$ liter displacement, because this kind of IC engine with proper hybrid strategy achieves best fuel economy. Now, in this section, one would like to design IC engine, starting from engine concept and manufacturing constraints and arriving to product ready to pass to pre-execution group.

II-I Target definition

Nowadays, engine target is made by customer: this is the key point that lead engine target definition. As this thesis show in first part, marketing department trough market researches and benchmarketing, identifies customer requirements: the most important are vehicle shape and performance. Vehicle shape is strictly related to engine compartment dimension while vehicle performance is strongly related to engine performance. In the market, in particular, the future market so today engine concept will arrive on the market around 2019, there are some external forces as pollutant engine emission, and pedestrian accident regulation. These are the most important external forces related directly to vehicle purchasing trend. The first one is heavily related to engine design because forces after treatment system which use a lot of space of the engine compartment. Actually, this is severe problem for all engine manufacturers because treatment systems for exhaust engine gases uses engine compartment portion not negligible. Many solutions to this problem is to move the after treatment far away to IC engine, puts under floor for instance, but this increases the light time of converter defined as time to reaches significant efficiency of three-way pollutant converter as increases harmful emission in

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cold start phase. Pedestrian accident regulation is one of guide line to define the upper shape of engine compartment because this one is heavily related to hood shape and then engine compartment is limited on the top from pedestrian accident regulation requirements and on the bottom from engine support frame linked to wheel hubs.

In according with Ulrich and Eppinger [1], as shown in Figure 1, concept generation and development process presents how customer needs lead product development as well as in IC engine product.

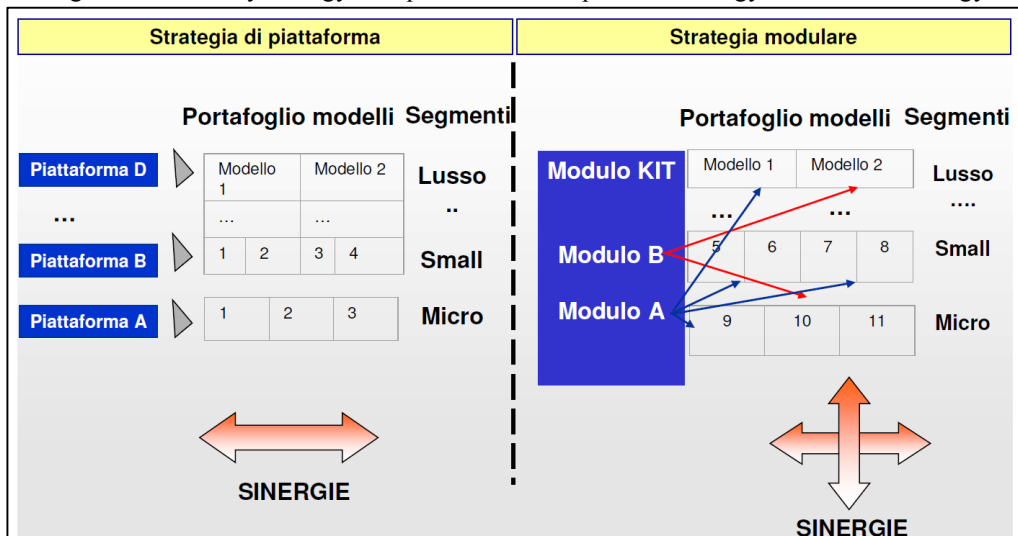Fig. 1: concept generation and development by Ulrich and Eppinger [1]
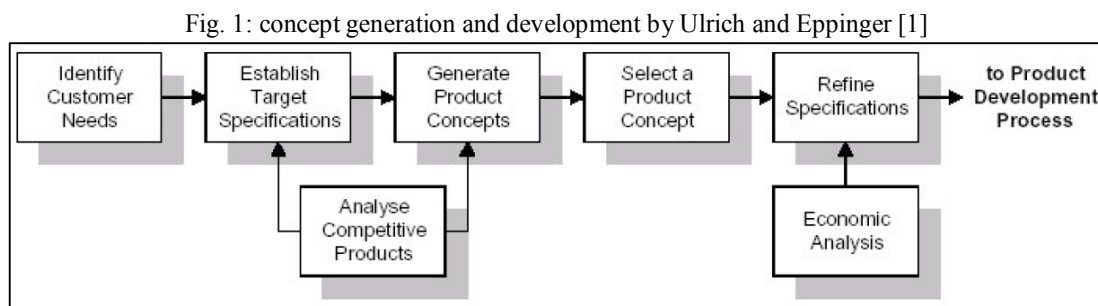
Vehicle performance as engine performance are customer requirements through quality function deployment they become target specifications for new IC engines family. As explain later, there are different engine architectures that satisfies target specifications. Driven by cost, product complexity, materials, technologies, a proper engine architecture is chosen as product concept. Once product concept is chosen, first CAD modeling is done and after certain level of CAD model reliability, firsts numerical analysis are carried on and their results provide information about right guide line of product and its feasibility. These analysis are:

- crank balance analysis: general benchmarketing, piston pin and connection rod small end, crankshaft, flywheel, balancer shaft, and engine mount vibration analysis. These analysis are addressed to understand whole crank mechanism integrity and in particular mechanical parts involved in active, from piston to flywheel and torsion damper, and passive, from piston to chassis, flow energy transmission.

- crankcase ventilation analysis to understand engine ventilation concept validity and feasibility.

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- valvetrain analysis and cam profile design: geometry, kinematic and dynamic motion, system durability, spring analysis, system performance and parametric study. These analysis are focused to understand valvetrain layout feasibility and performance, or in other words to verify valvetrain concept layout.

These analysis as whole are addressed to verify engine concept robustness. Concept development consists in research of not easy compromises among concept requirements, components selection, target cost, and concept architecture. In engine design are identified three particularly critical activities: engine performance definition, engine architecture definition, finally components selection. Engine performance definition comes directly from customer needs in term of vehicle performance. Engine architecture definition is most critical activity that involve target market, engine cost estimation, specific torque and power target, material, technologies, pollutant emission, weight, engine efficiency as whole. About components selection, there are two fundamentals decisional areas influence product competitiveness through trade-off approach: using new or existing components known as carry over approach; in-house or supplier parts development known as make-or-buy approach. Carry over approach reduces design and manufacturing new tool cost, these could be cost sustained from supplier so buyer could achieve better price, as well as risk of parts reliability problem. On the other hand, excessive use of common parts could have negative effects because part not tailored design for application under development leads with its some compromises. Use of external design resources could have positive effects from quality components and planning – coordination internal effort points of view. On the other hand, external design resources could cause deterioration of technical internal know-how. Engine is complex product that means there are several variants manage in the product development. Product complexity leads two opposite effects: diversity and proliferation. Diversity is product complexity justified from customer needs or from value on cost rate. While proliferation is over product complexity that is not visible to customer or that induce additional cost without any benefit from market. Proliferation reduction could be done through communality strategy and in particular platform strategy and modular strategy as shown in Figure 2.

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Fig. 2: commonality strategy: comparison between platform strategy and modular strategy
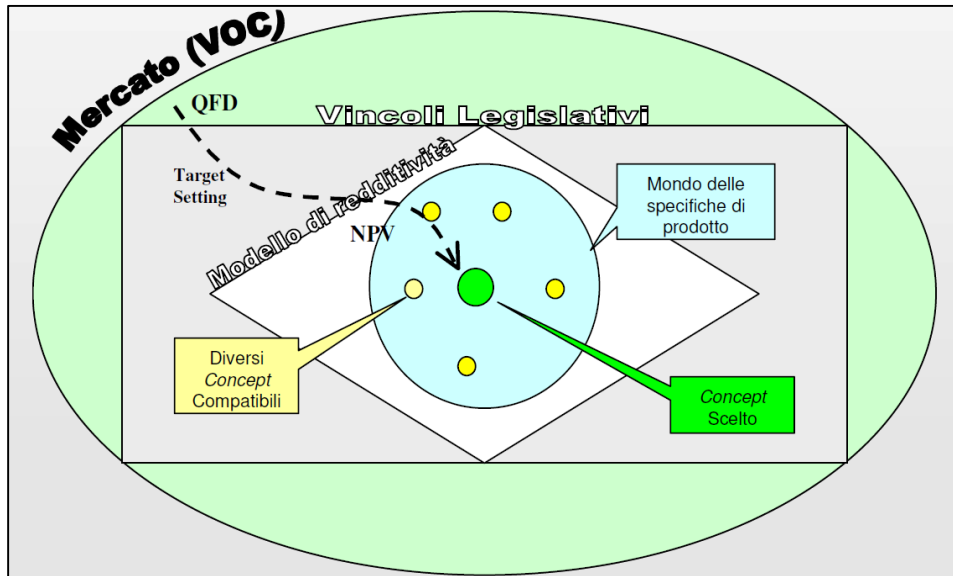

Platform strategy means use same engine into same platform design to match requirements of certain segment in this way same engine could be used by different car models. About modular strategy means same engine is used on different car models from different segment: engine is black box with detailed system characteristic and they have to mach both vehicle and customer requirements. To sum up, factors contribute to concept definition are:

- market: targets definition consistent with customer needs;
- regulation: different national legislative constraints as pollutant emission, pedestrian accident, transport approval;
- profitability: model analysis of Net Present Value;
- product requirements: declined from customer requirements and defined through target setting activities and refined using trend-off analysis among cost and performance.

As shown in Figure 3, last choice of dominant concept occurs after convergent process selection with successive interaction.

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Fig. 3: development and selection of product concept scenario

In case under discussion, as shown in Figure 4, customer needs are changed in numerical and univocal requirements that form Vehicle Technical Specification.

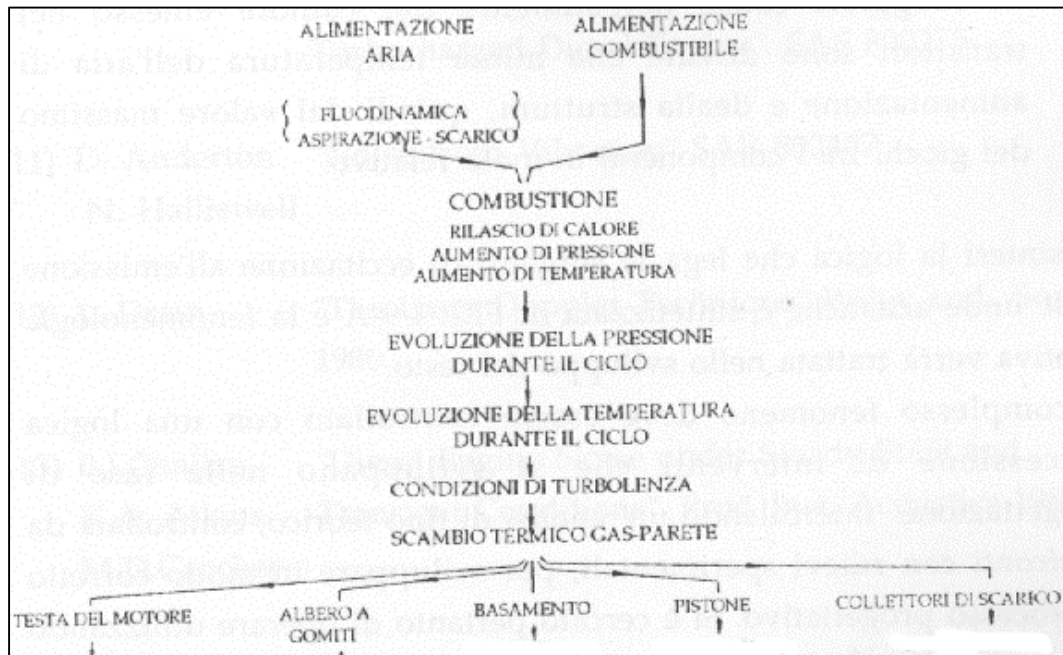Fig. 4: generation, development and selection of IC engine concept scenario
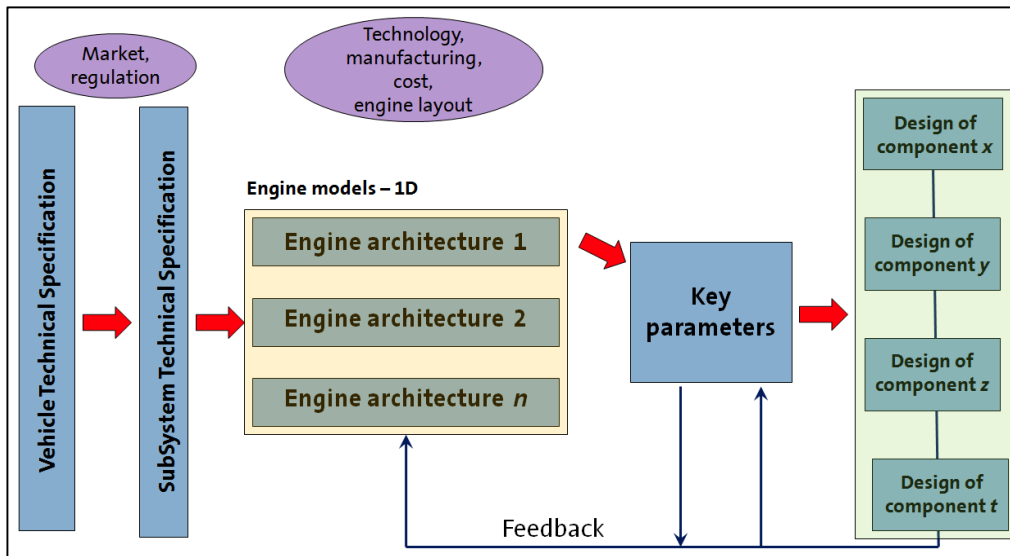
VTS are number that show vehicle main system performance. So each VTS could be analyzed in detail through SubSystem Technical Specification that precisely describe performance of different function of vehicle main systems. In case under discussion,

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focus is IC engine and then vehicle customer needs as brilliancy, fuel consumption, vibration, noise, drivability, from qualitatively judgments through quality function deployment activities become quantitative specifications as vehicle Performance Index, Brake Fuel Specific Consumption, maximum displacement at engine mounts, dB level prescription, so a group of these ones become SSTS for IC engine. SSTS for IC engine could achieve through different engine architecture based on type, displacement, valvetrain layout, injection system, cost, material, manufacture and technology. So from group of engine architectures satisfy IC engine SSTS only few of those go on and in the only one is chosen as dominant concept. This concept is modeled through mono-dimension thermodynamic model to have engine performance prediction which highlights if engine concept achieves targets or which is distance from target. In this way, designer could realize and have first feedback about validity of engine concept chosen. If engine concept selected is proper one, from 1-D thermodynamic model are extracted some key parameters as Peak Firing Pressure, turbine intake temperature, and exhaust compressor temperature that will lead components design.

Figure 5 shows first part of IC engine design process [3] published in 1992.

Fig. 5: structural design of internal combustion engine

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It highlights combustion process as core of engine design that means engine design is led from thermal-structural point of view approach. Customer needs are out of development of product. This is mean different from Figures 4 and 5. Then Figure 5 does not include concept development and selection: it is one-flow chart that respects only structural and NVH requirements. Nowadays, the Nuvolari's time, when in the race car there were two seats: one for the driver, and one for the engineer due to the extremely low reliability of components, are far far away from our scenario. In other words, the components reliability as well as the whole car reliability is the first point to go on the market.

In engine concept development, modification design management is done through basket management as shown in Figure 6.

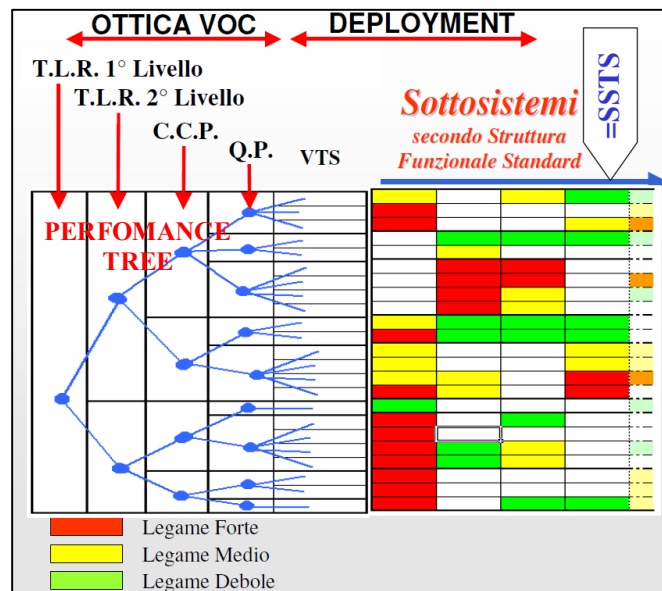Fig. 6: modification design management through basket approach
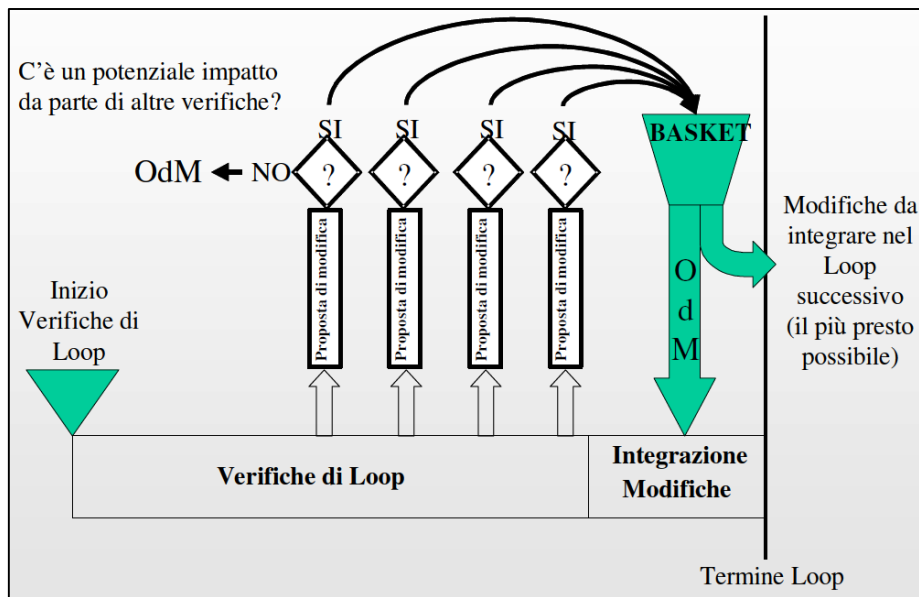
In scope of verification loop, modification proposals cause from cross – functional team are collected and after valuation process, if accept, they become Modification Orders. MO flow in unique basket that decide which one has to execute before loop closure and which one could be participated to next phase.

During engine concept development, target definition and managing process is articulated in two main phases: target setting, and target deployment and achieving. During concept development and product definition, target setting establishes product targets from

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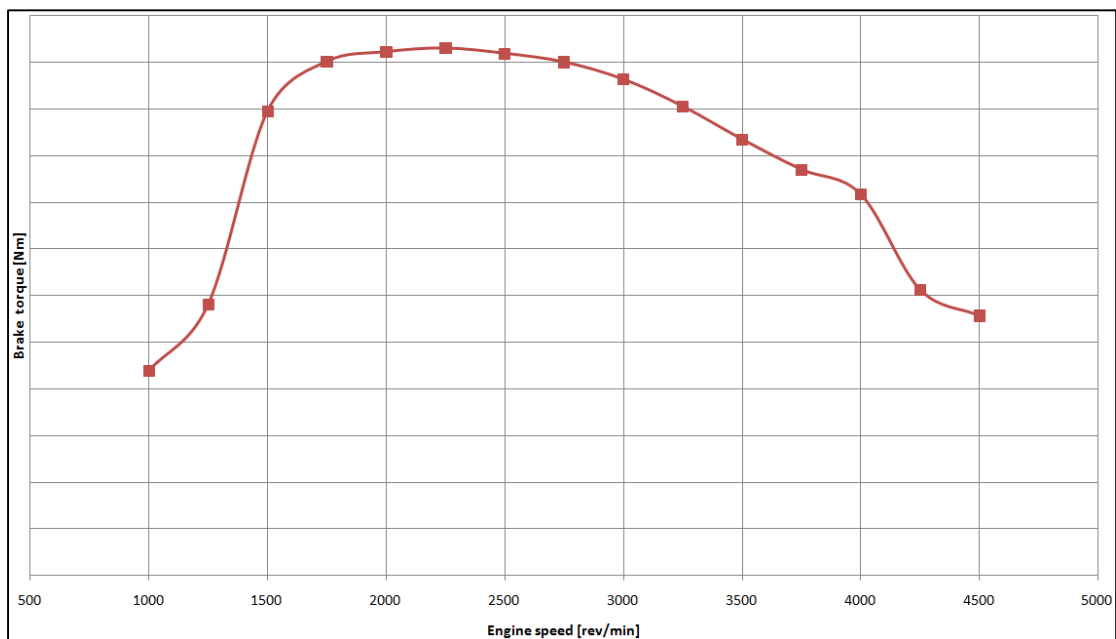Antonio Turturro – Supervisors: Prof. C. Delprete and L. Magro

customer point of view at vehicle level and translate them to technical targets expressed in engineering parameters. During engine development and definition, targets defined at product level are translated to engineering objectives per each sub systems and components. To collect data for target setting phase, once Customer Car Profile test is done, from customer in free way, and Quality Profile test, from tester follows standard way, is done as well, on reference vehicle group, a correlation is matched between two evaluation to verify grade alignment among customers and tester. After, QP's subjective evaluation has to be translated in engineering objectives, through correlation model that bond in two-way an evaluation in SAE rating of determinate QP with certain values of measured engineering parameter. Target deployment works per performance, and arrives to define contribution for each sub system to achieve VTS targets, through determination of objectives values for SSTS. Achieving activity is another in-depth examination of target deployment and works per physical elements based on design requirements. Figure 7 shows practically how customer needs arrive to define engine targets and visualize whole processes of target setting, deployment and achievement.

Fig. 7: processes of target setting, deployment and achievement

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Finally, follow the process described above, one of the most important IC engine targets are Air-to-Fuel ratio and BSFC per each engine speed, torque curve shown in Figure 8 which could be verified as described in vehicle performance paragraph. Anyway, curve in Figure 8 is obtained through vehicle torque requirements and, for each speeds, evaluate exuberant torque for vehicle acceleration. While AF and BSFC are strongly related with pollutant emission and are limited from emission limits in particular NO_x and CO_2 in g/km. This curve has same characteristic lines as follow: quasi constant torque line between 1500 and 3250 rpm; introduction line amid 1000 – 1500 rpm; defluxion line among 3250 – 4000 rpm; last line, amid 4000 – 4500 rpm, is connection between torque at 4000 and an end torque value at 4500 rpm.

Fig. 8: torque target curve

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II-II Engine breath capacity and ports design

First step in engine design is bore dimension definition. It depends on manufacturing constrains and in particular fasteners position, and centre bore distance. Bore dimension mainly depends on injection strategy and is strongly related with whole engine length. On the other hand, inter – bore distance depends on engine block architecture, material, presence of cross – drill or kind of liner. Injection strategy is defined as interaction of sprays, in-cylinder motion and combustion chamber and for this reason bore dimension is a key parameter.

Once defined bore dimension, it could start to think about breath engine capacity β or valve capacity and ports: since air breathing capacity and utilization determine the output of the diesel engine, the flow characteristics of the intake and exhaust systems are crucial in the achievement of good performance. Breath engine capacity β is defined as:

$$(1) \qquad \beta = \frac{A_d - A_s}{A_b}$$

where A_d is area of latest constant part of port; A_s is steam area of valve; and A_b is cylinder area. In the same way, but slightly different is defined valve capacity [4] as:

$$(2) \qquad valve\ capacity = \frac{d}{b}$$

where d is diameter of latest constant part of port; while b is bore dimension. Both β and valve capacity, if one keeps constant all the other conditions, are strongly related to volumetric efficiency and in other words with amount of air that engine is able to trap, because much air is introduced into engine, more fuels are able to burn that mean higher performance. In this point of engine design, one has only bore dimension, inter – bore distance and fasteners location and it is impossible evaluate volumetric efficiency but with equation (1) and (2), designer could have idea about volumetric efficiency. This idea

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is truer if they try to keep a valid injection strategy. Now, one could start to think about valves number and dimension. There are collection of structural constraint around valve positioning and dimension. First of all, number of valves depend on engine performance, injection strategy and in particular on in-cylinder motion or depend on engine costs. In case under discussion, designer tries to keep a valid injection strategy but change number of valves, so create similar in-cylinder motion field. Anyway, structural constraint to take account are: seat insert dimension minimum 2.5 mm width and 5 mm height; valves distance around 12 mm; injector location; valve axis slope; wall thickness between injector and valves minimum 4 mm; glow position; and finally distance between poppet valve trace and bore trace, on flame deck plane, has to be minimum 0.8 mm to take account of both thermo-structural material behavior and a safety area. About injector location, the best point for spray tip is on centre bore but to obtain bigger valve dimension, it could be located in another position. Best practice injector location suggests locating it inside a circle area centered on bore axis with diameter as up 9 % of bore dimension. The reason behind this guide lines are combustion efficiency and in particular fluid-dynamic plunge behavior in term of wall interaction and homogeneous air-fuel mixture formation. About valve axis slope, the best position is vertical. In extremis, it could be accepted maximum slope of 3° to have more space amid two valve stem but it forces to have doomed combustion chamber and so valve sack on piston. This solution creates many edges on piston and combustion chamber which become hot points and failure points in operation condition especially at high loads.   Then, valve dimension could be evaluated.

Before starting to fit larger valve diameter in our archetype, it is better evaluate breath engine capacity of engine as much as possible similar to one under discussion. Figures 9 and 10 show β values for several engines.

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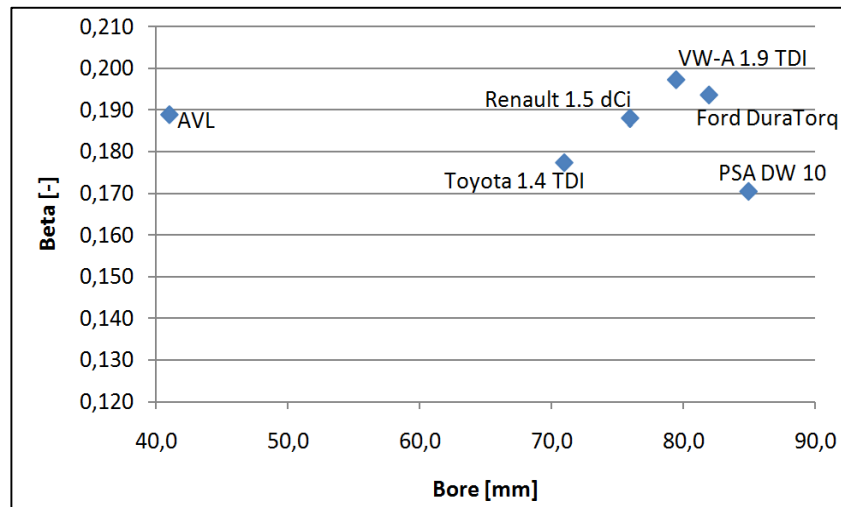
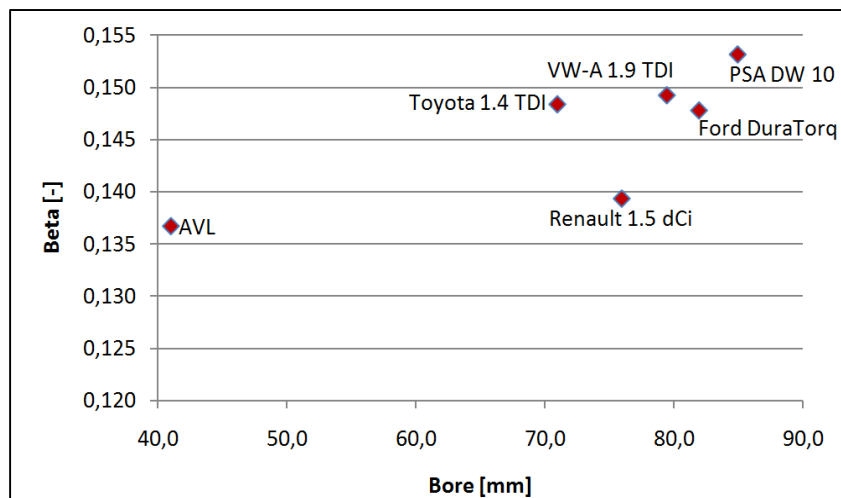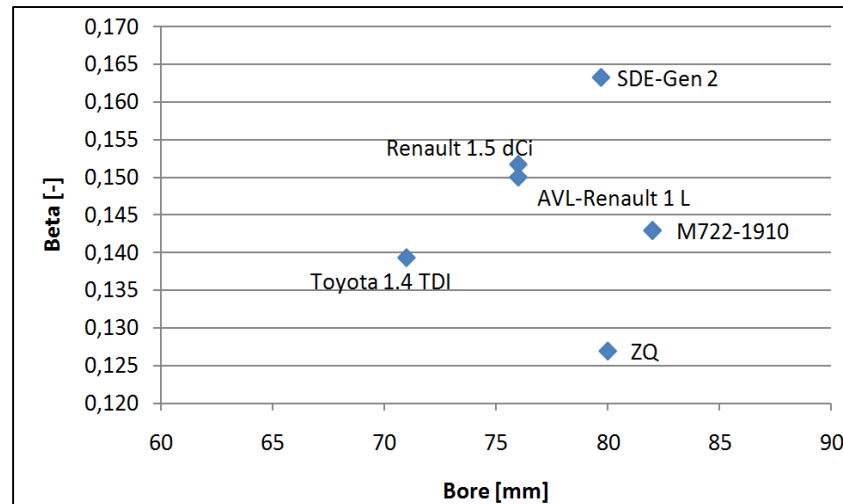Fig. 9: examples β intake for several engines


Fig. 10: examples β exhaust for several engines

From Figures 9 and 10, seems to figure out a V trend with vertex around 80 mm, VW engine, for intake chart and around 75 mm, Renault engine, for exhaust chart. However, Figure 9 and 10 would give an example of benchmark to explain engine design methodology. For example, Figures 11 and 12 show breath engine capacity for two valves engine that is case under discussion.

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Fig. 11: β intake for two valves engines

Every engine try to achieve the higher β intake value because means low flow losses, low pumping losses and much air but disadvantages are on valvetrain performance because bigger valves mean higher inertia. AVL – Renault 1 liter engine [16] is concept engine for aggressive downsizing. M722, 1910 and ZQ engines belong to Fiat Group. One could suppose β intake for new General Motors engine around 0,163 as first attempt to understand it is possible go over Renault's engine value. Or in alternative way, check β intake value of Renault's engine and then try to go over. The last way to use this chart is suppose first attempt of valve dimension on archetype, then evaluates β intake value on chart and modify valve dimension according to comparison on Figure 11. The best way in according with Challen and Baranescu [17] is impose β intake and try to achieve the biggest valve dimension, so designers could improve team work and obtain best result. Figure 12 shows β exhaust situation.

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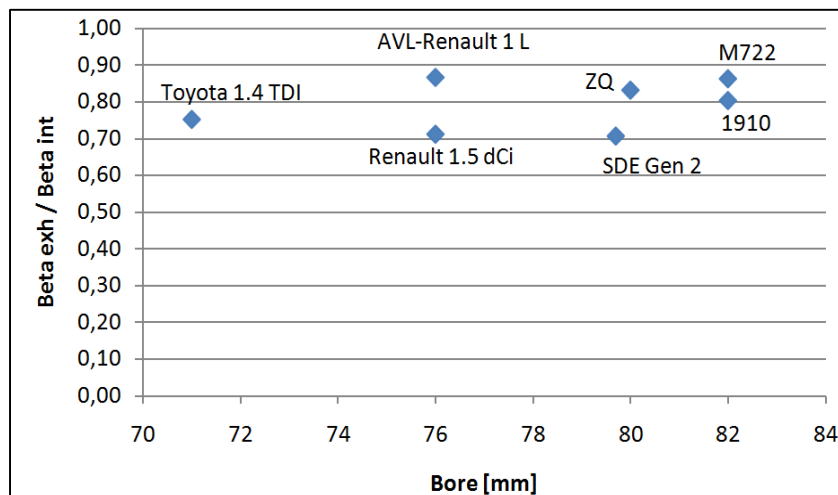Antonio Turturro – Supervisors: Prof. C. Delprete and L. Magro

After rough evaluation β intake value of 0,163 lead to huge valve dimension, so for β exhaust value, first attempt is in the middle of range, around 0,117. Advantages for higher exhaust valve dimension are: low pumping losses, low fluid losses, low back pressure; but disadvantage is higher thermal stress.

Fig. 12: β exhaust for two valves engines
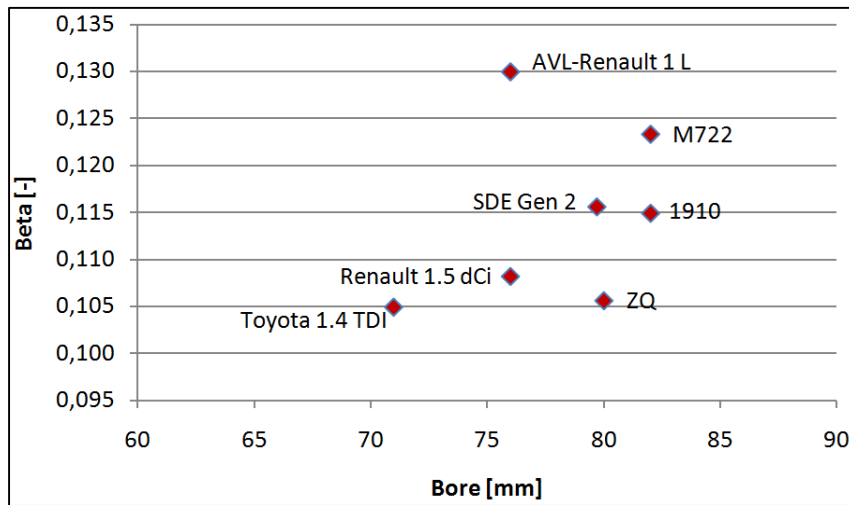
β exhaust and β intake present respectively ratio between exhaust - intake flow area and cylinder area. It is useful compare intake flow area with exhaust flow area and one could do it through ratio among β

Fig. 13: β ratio for two valves engines

exhaust and β intake as shown in Figure 13.

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β ratio helps to understand if it uses right proportion amid exhaust and intake flow area. About β intake of 0,163, after rough analysis it leads to huge valve dimension. So in according with Challen and Baranescu [17], β intake of 0,148 is imposed and valve dimension is evaluated through equation (3) derived from (1):

$$(3) \qquad\qquad d = \sqrt{b^2 \beta_{int} + d_s^2}$$

where b = 79,7 mm is bore dimension and d_s = 5 mm is stem diameter. With these data valve diameter d is 31,07 mm. Next, valve diameter is drawn on archetype and in case under discussion there is possibility to enlarge it up 32 mm to get β_{int} of 0,157.

Next runner and port design are presented. Design of these components is lead by fluid losses and permeability. First one relates to air velocity because fluid losses are proportional to fluid velocity at power of three. While second one is strongly related to type and dimension of port, and amount of quantum that it has to give to intake air. Before focus is moved on port design, is extremely useful designs runner and in particular quantifies transversal area, to have an idea of port dimension and air velocity. First step to calculate runner transversal area is imposed equality volumetric flow rate condition between air volume travels in the runner and air volume enters in cylinder:

$$(4) \qquad\qquad \frac{\pi b^2}{4} u = \frac{\pi d^2}{4} w$$

where u is average piston velocity in m/s and w is air velocity in m/s. In embryonic stage of engine design, runner diameter could be approximate to poppet diameter; and bore dimension could be approximate to twice times valve diameter. Runner diameter is half of bore diameter so flow velocity is four times average piston velocity. For high speed diesel engine, average piston velocity is around 12 m/s while rule-of-thumb to have

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reasonable fluid losses is that *w* has to be in the range of 50 – 60 m/s. In this way, with previous assumption, guide line value for flow velocity is 48 m/s. It is slight under inferior limit that means one has enough margin to cover all simplification errors and if further analysis show too low air velocity, there will be possibility to decrease runner dimension. From equation (4), it could evaluate runner dimension *d*:

(5)
$$d = \sqrt{\frac{d^2 u}{w}}$$

In this example *d* is about 40 mm and leads to transversal circle area of around 1247 mm^2. Using equivalent diameter concept from fluid-dynamic, rectangular shape dimension of runner is evaluated through impose same perimeter length as follow:

(6)
$$\pi d = 2(B + H)$$

where *B* and *H* are base and height of rectangular shape runner. These are only the firsts steps to start runner and port design. They are useful just to have an idea of above components dimensions.

The intake ports of a diesel engine may have to generate air motion, swirl, in the cylinder so that the air charge retains significant angular motion when the fuel is later injected, improving air-fuel mixing. Most engines rely on some swirl but modern high-speed direct injection engines require only low levels of swirl, much of the air-fuel mixing energy coming from the fuel spray. In intake runner for two valves per cylinder engine, swirl motion defined as in-cylinder air motion around cylinder axis, is generated from helical port. Actually, valve location respects to cylinder axis adds no negligible contribute. Main characteristics for swirl generation are helix wrapping, as tread, and rise of portion runner that wrap in coil way, or in other words, aspect ratio of section where helix begins. These characteristics have effect to increase swirl motion. In this way, decrease of permeability is limited thanks to dynamic effects introduced through aspect ratio, and so runner efficiency increases. Other parameters are trimmed radius among helix beginning

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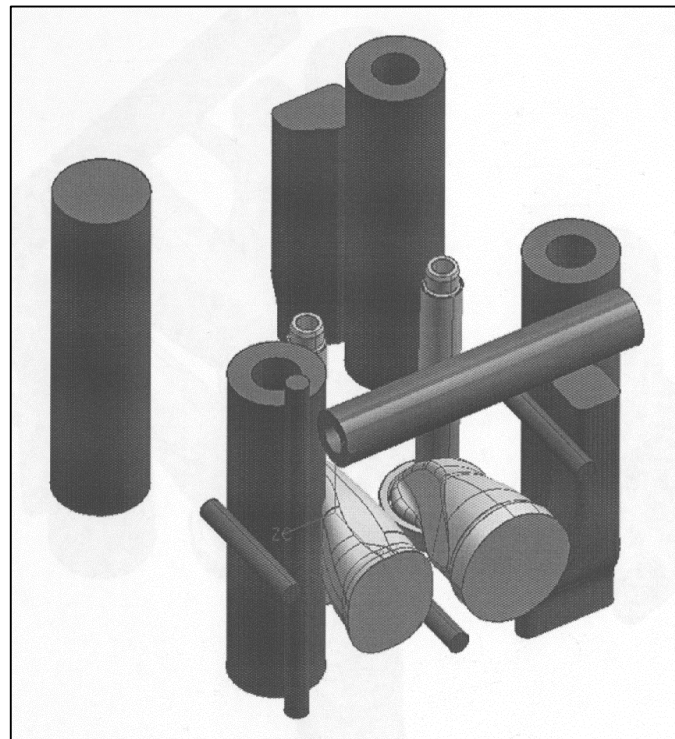
section and intake runner, and minimum section. The result of intake port design is shown in   Figure 14.

Fig. 14: helical port and intake runner for two valves engine
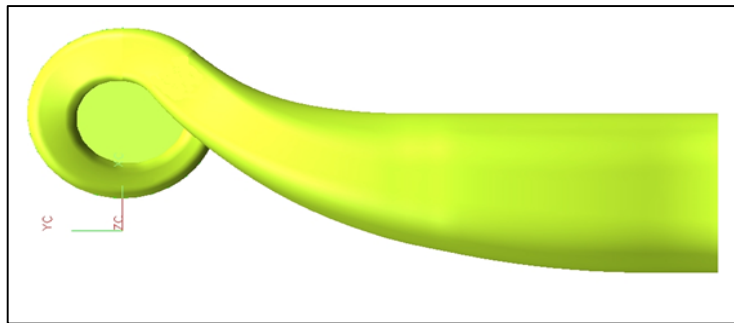
Finally, archetype to start engine design is complete and shown in Figure 15, where all elements presented above are included.

Fig. 15: archetype for 4 valves engine which shows intake ports, fasteners and valve guides

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The archetype with other constraints as valve guide dimension around seven times stem and spring work length, are the base to cylinder head design.

II-III Engine performance baseline

At present, base engine parameters are defined and 1D thermo-dynamic model is been able to build up through use the most similar injection strategy. So, follow this concept, stroke is defined into narrow range as well as con rod length and then crank mechanism is ready to be modeled. A new engine project that have to face the future challenges, it has to account several fundamental aspects as CO_2 reduction and Euro 6 and over requirements in whole; hybridization support, attractive cost for emerging markets; use of learn lessons of previous projects. The new engines series is developed as follow: first of all, engine core is designed as combustion chamber, crank mechanism, injection strategy; and then all others features are added to build matrix in which engines are developed towards main markets and applications requirements. Some of features are: number of cylinders; number of valves; different performances; different injection systems; different materials; different hybrid functions; different engine mounted strategies. So, 1D thermo-dynamic engine model of two cylinders, four valves, turbocharged engine is built up starting from GT-Power model of GM 2 litre Euro 5 – GM 2l E5 - diesel engine, which shares injection strategy and engine core with new engine concept. Figure 16 shows GT-Power model of new engine to evaluate if it could achieve targets in terms of torque and power, and respects all others requirements as outlet temperature at compressor and inlet temperature of turbine.

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Fig. 16: GT-Power model of new two cylinders, four valves, turbocharged concept engine
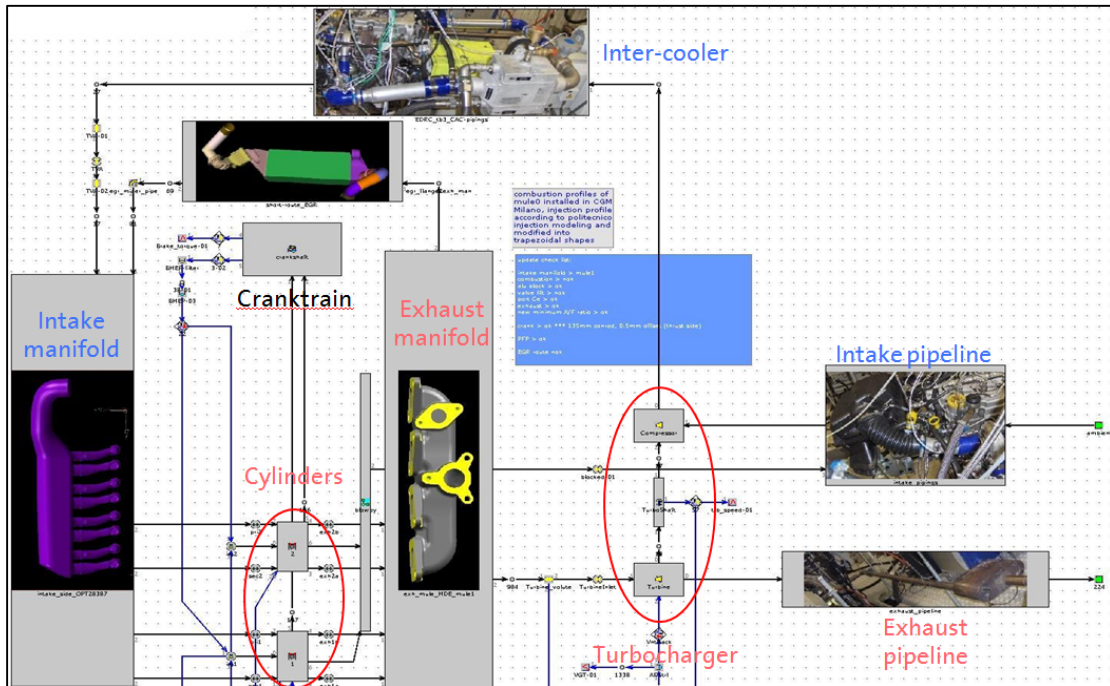
The concept idea is to introduce less variations as possible to maintain constant the cylinder unit performance, so caps are introduced on both intake and exhaust runner for erased cylinders. The main variations from GM 2l E5 engine are: two cylinders left, as shown in Figure 16; four inlet runner and port left, as shown in Figure 17; four exhaust runner and port left, as shown in Figure 18.

Fig. 17: GT-Power model of intake manifold for new two cylinders, four valves, turbocharged concept

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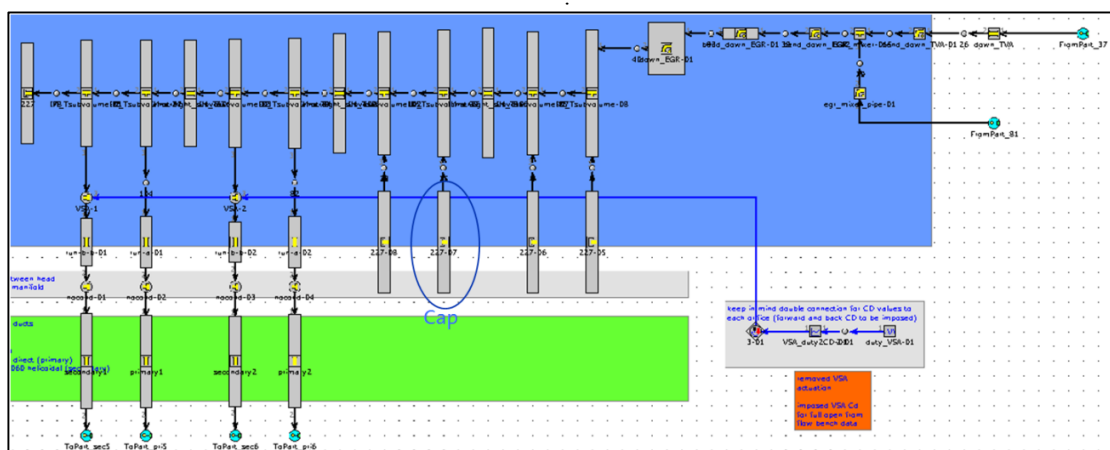
Fig. 18: GT-Power model of exhaust manifold for new two cylinders, four valves, turbocharged concept engine
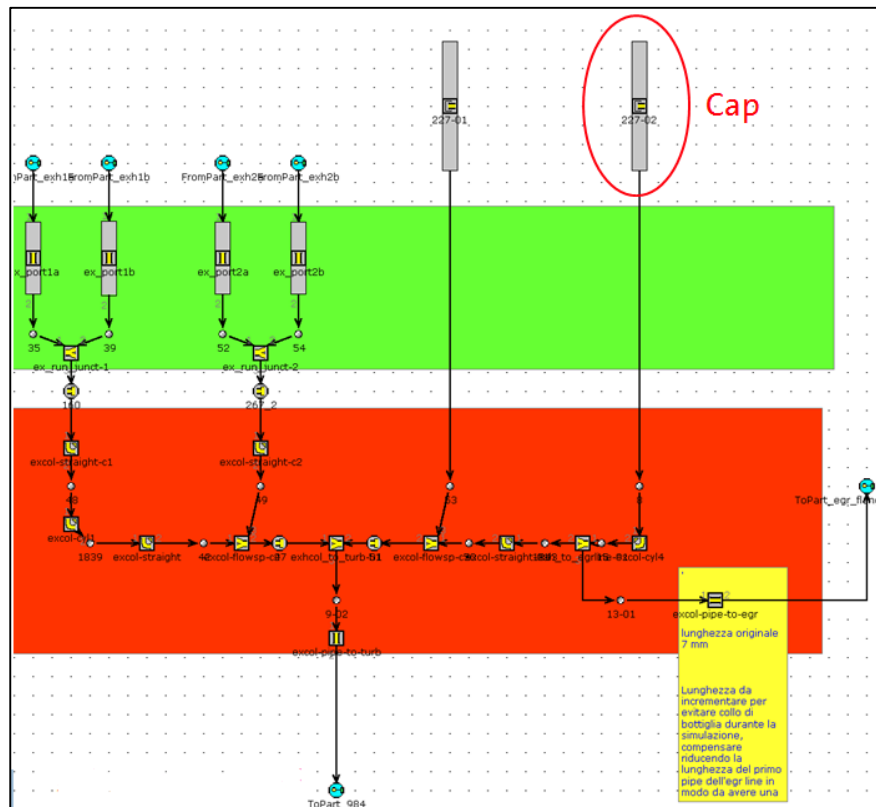
About engine friction, main variations from GM 2l E5 engine is evaluated algebraic sum of the friction source; two cylinders engine respects to four cylinders engine has: two main journals left; four cam journals left; two piston groups left; two countershaft journals more. It is estimated an average friction contribution on total engine friction by strip-down tests for four cylinders engine, as follow: crankshaft 20 % that means 4 % for each main journal; piston groups 40 % that means 10% for each piston group; valvetrain 10 % that means 1 % for each cam journal; oil pump 15 %; water pump 15 %. To sum up, the engine friction of new two cylinders, four valves, turbocharged concept engine, in comparison with that of GM 2l E5 engine, is reduced of –30 % owing to –8 % crankshaft contribution; -20 % piston group contribution; -4 % valve train contribution; +2 % countershaft contribution. But, unfortunately the contribution of engine accessories on the friction grows up with the reduction of engine displacement. Practically, reduction components consequences from engine friction point of view, is increase Friction Mean

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Effective Pressure and decrease engine motoring torque, because same amount of accessories have to be driven by smaller engine displacement than GM 2l E5 engine, so amount of energy losses to do it increase. On the other hand, smaller and simpler engine has less parts to be driven and then motoring torque decrease. So, it is consistently not conservative hypothesis to consider constant accessories contribution on total engine friction but it disappears for comparison purposes and to screen different engine concept solutions with same boundary conditions. Obviously, the fire interval sets up on 0° and 360°. About turbocharger, main variations from GM 2l E5 engine are compressor and turbine mass multiplier. In particular, it is used the same turbine and compressor of GM 2l E5 engine but with different mass multiplier that vary from 1 to 0.5, in this way it has an ideal turbocharger suitable for this application. Now, GT-Power read the same values of efficiency, pressure ratio, turbocharger speed but with a reduced mass flow rate. Figures 19 and 20 show respectively turbine and compressor efficiency maps; efficiency is depicted as color contour map. This concept is merged on GM technology strategic plan and all data are under industrial confidential restriction.

Fig. 19: turbine efficiency map uses on GM 2l E5 engine

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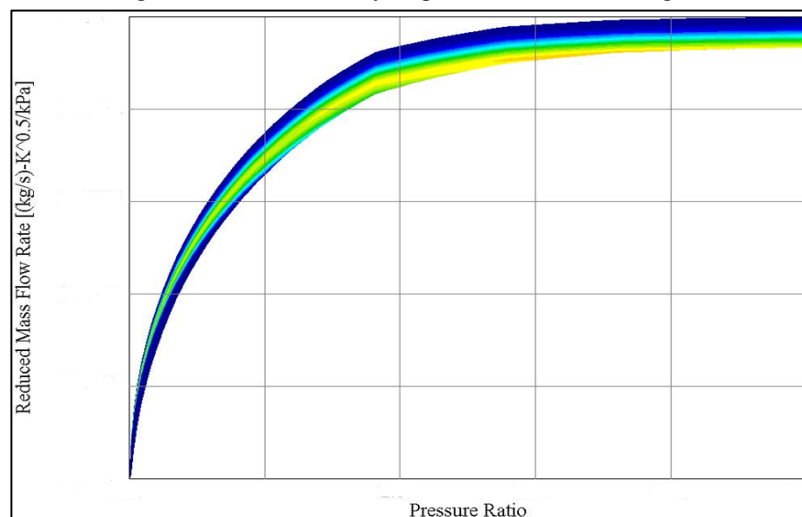
Fig. 20: compressor efficiency map uses on GM 2l E5 engine
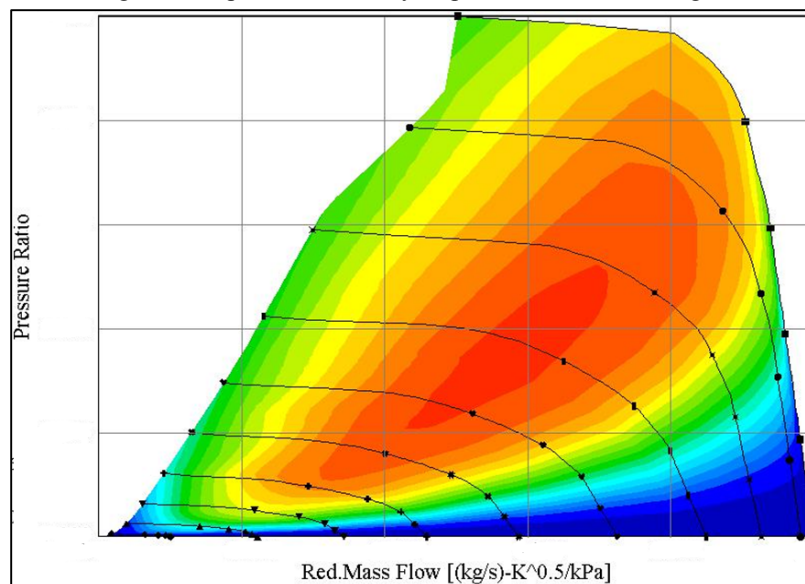
The performance target for new engine are strongly linked to the vehicle where the engine will mount. In this phase, it looks for the maximum performance that new engine can achieve to have a competitiveness advantage to use in several ways. The present case does not have data from marketing and quality department and then base reference input for engine concept development are the engine torque that come from GM 2l E5 engine multiply for a coefficient derived from both engine displacement and brake torque, and the Air-to-Fuel rate that is strongly linked to emissions, because both engines use similar injection strategy. The multiplicative coefficient is for 1000 – 4000 engine speed range the displacement rate among GM 2l E5 engine and the new engine concept, two cylinders four valves; and for 4250 - 4500 the torque rate among the previous engines. In this section it will be discussed the results about two couple of simulations. The first one compare the new engine concept, two cylinders four valves, with two different FMEP, Friction Mean Effective Pressure, values and a constant BMEP, Break Mean Effective Pressure, values. The second one compare the same engine model with two different FMEP values and nearly same values of outlet compressor temperature and inlet turbine temperature. The reason of this choice is the difficult estimation of engine friction, so to understand the advantage of a simpler architecture and to show the effects of it on the

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engine performance. The first comparison is among the model of new engine concept, two cylinders four valves, with the friction model of GM 2l E5 engine and the new engine concept, two cylinders four valves, with a friction model that consider the light architecture of two cylinders engine. How it explain before, the friction reduction is nearly -30% respect to the four cylinders engine. In the follow charts there will be this label:

- STEP 1 = new engine concept, two cylinders four valves, friction model of GM 2l E5 engine, turbocharger of GM 2l E5 engine with mass multiplayer 0.5;
- STEP 2 = new engine concept, 2cylinders 4 valves, friction model of GM 2l E5 engine reduced of 30%, turbocharger of GM 2l E5 with mass multiplier 0.5; equal BMEP values to STEP 1
- "xxx" reference new engine concept, 2cylinder 4 valves = reference parameter of new engine concept, 2cylinder 4 valves
- "xxx" GM 2l E5 = parameter of GM 2l E5 diesel engine.

In Figure 21 brake torque of new engine concept is depicted.

Fig. 21: brake torque comparison among new engine concept STEP 1, STEP 2 and reference
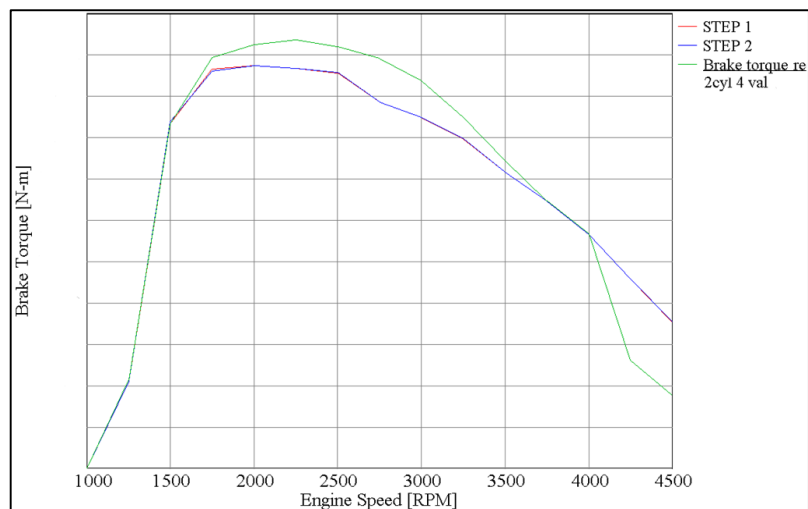
The torque performance shows a good trend also if there is a loss of performance between 1750 – 3750 rpm. The difference between STEP 1, STEP 2 and the reference is attributed at maximum temperature allowed for outlet compressor and inlet turbine.

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About torque curve over 4000 rpm will be reduced because not useful, in particular torque trend over 4000 rpm uses to harmonize engine switch off at 6000 rpm. However, there is no difference among STEP 1 and STEP 2.

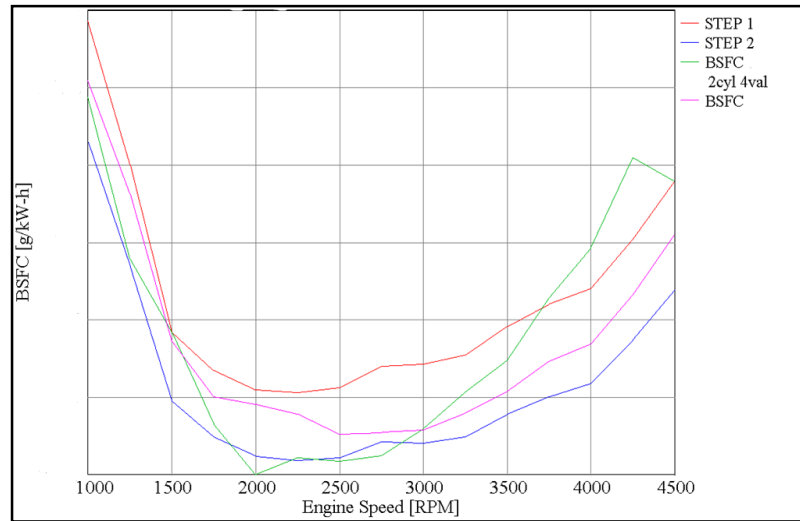Figure 22 shows brake power curves for STEP 1, STEP 2 and reference.

Fig. 22: brake power comparison among new engine concept STEP 1, STEP 2 and reference
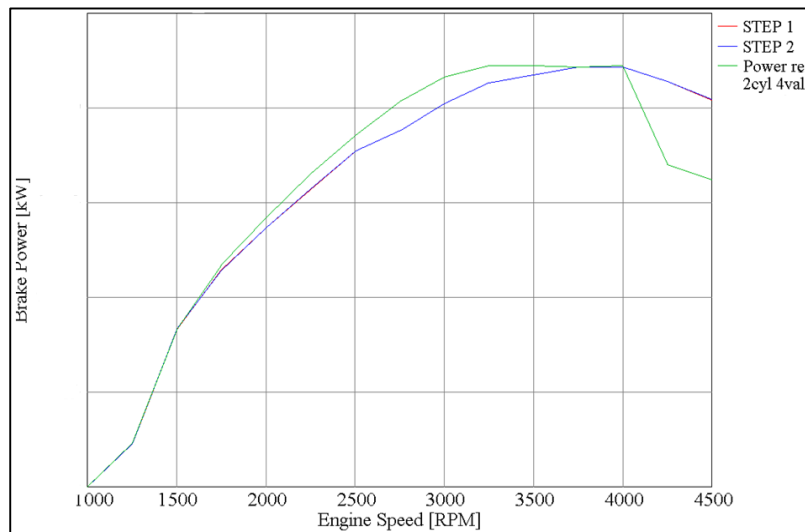
The difference between them is attributed at maximum temperature allowed for outlet compressor and inlet turbine. The torque performance shows a good trend also if there is a loss of performance between 1750 – 3750 rpm. However, there is no difference among STEP 1 and STEP 2.

Figure 23 shows Brake Specific Fuel Consumption of STEP 1, STEP 2, reference and GM 2l E5 engine for comparison purpose.

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Fig. 23: BSFC comparison among new engine concept STEP 1, STEP 2
and reference

In Figure 23 there is GM 2l E5 engine's BSFC because this parameter is concentrated index not related with engine displacement and further because new engine concept shares injection strategy of GM 2l E5 engine. STEP 1's BSFC curve follows GM 2l E5 engine trend pretty well with variable gap accentuated amid 2500 and 4500 rpm, and at 1000 rpm. In other area, BSFC values are similar. Anyway BSFC rises because STEP 1 uses same friction model of GM 2l E5 engine with reduced engine displacement; reduced turbocharger mass multiplier controls increase of BSFC. Choice of STEP 1 definition is done to highlight effects of engine friction reduction. STEP 2 uses engine friction model reduced by 30%. Practically, GT-Power [18] uses Chen – Flyn friction model described in equation (7) which models it with polynomial empirical law.

(7) $$FMEP = A + B * PFP + C * u + D * u^2$$

Where A is constant value; B is value related to Peak Firing Pressure; C and D are parameters related respectively with average piston velocity and with average piston velocity at power of two. BSFC of STEP 2 follows strictly GM 2l E5 trend but unfortunately engine friction is under estimated because engine accessories contribution are kept constant. However, up to 3000 rpm, new engine concept BSFC follows pretty

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well BSFC reference but over 3000 rpm is far away from reference. Figures 24 and 25 shows Indicated Mean Effective Pressure respectively evaluated on compression and expansion strokes, IMEP360, and on all strokes, IMEP720.

Fig. 24: IMEP360 comparison among new engine concept STEP 1 and STEP 2
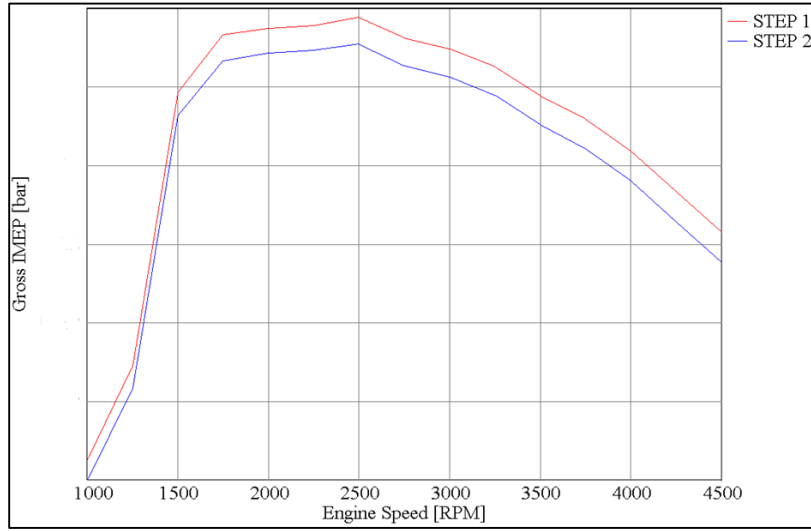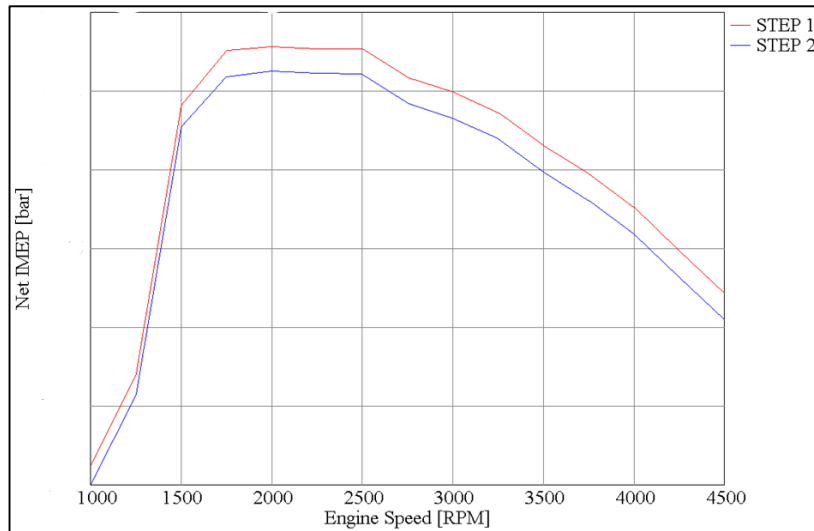

Fig. 25: IMEP720 comparison among new engine concept STEP 1 and STEP 2

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Differences among STEP 1 and STEP 2 is owing to constant BMEP and different FMEP. In particular, IMEP, FMEP and BMEP are connected as follow equation (8):

(8) $\qquad$ *BMEP = IMEP720 - FMEP*

The fall of engine friction with the same BMEP decrease the IMEP values, because of (8). In the other hand, a minor values of IMEP720 means a less stress on engine requirements, how it will see on the charts of both outlet compressor temperature and inlet turbine temperature in Figures 29 and 30. Figure 26 shows FMEP between new engine concept STEP 1 and STEP 2. Difference is related to Chen – Flyn engine friction model reduction. Unfortunately, experimental data from literature show contrary.

Fig. 26: FMEP comparison among new engine concept STEP 1 and STEP 2
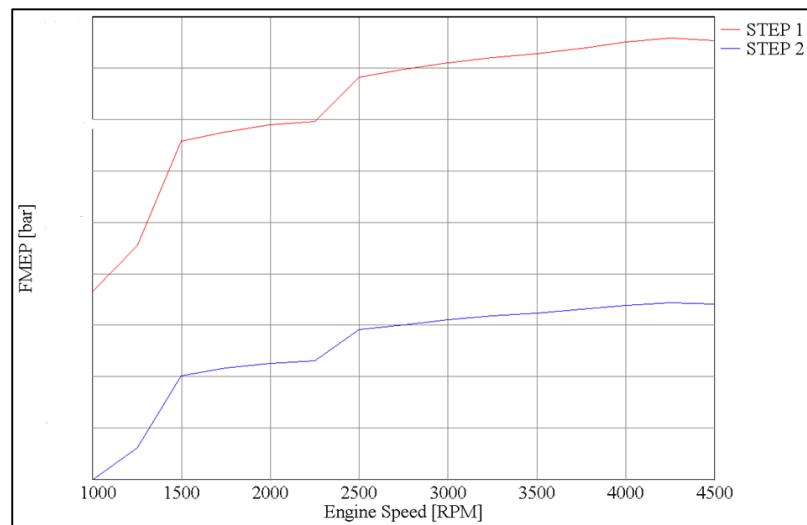
Figure 27 depicts Brake Mean Effective Pressure between new engine concept STEP 1, STEP 2 and engine reference. BMEP is strongly related to brake torque shown in Figure 21 and so they have same trend. However, there is no difference among STEP 1 and STEP 2.

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Fig. 27: BMEP comparison among new engine concept STEP 1, STEP 2
and reference
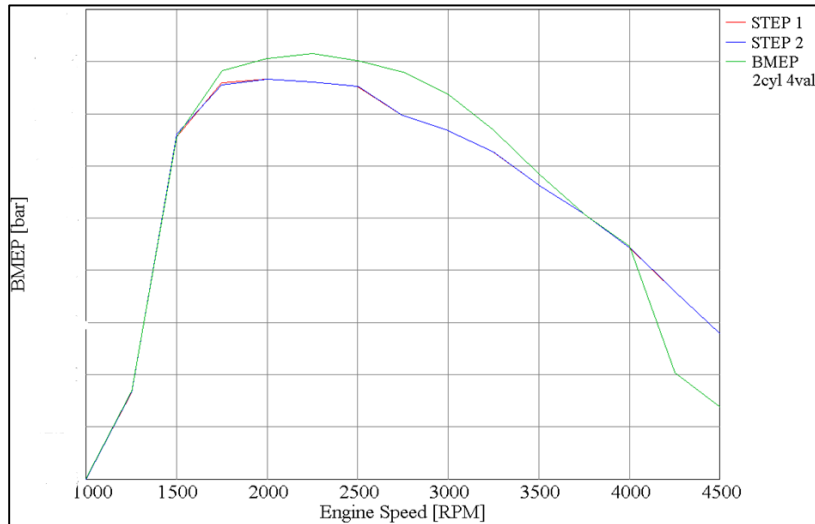
Figure 28 shows volumetric efficiency and has same trend for STEP 1 and STEP 2. However, there is no difference among STEP 1 and STEP 2.

Fig. 28: volumetric efficiency comparison among new engine concept
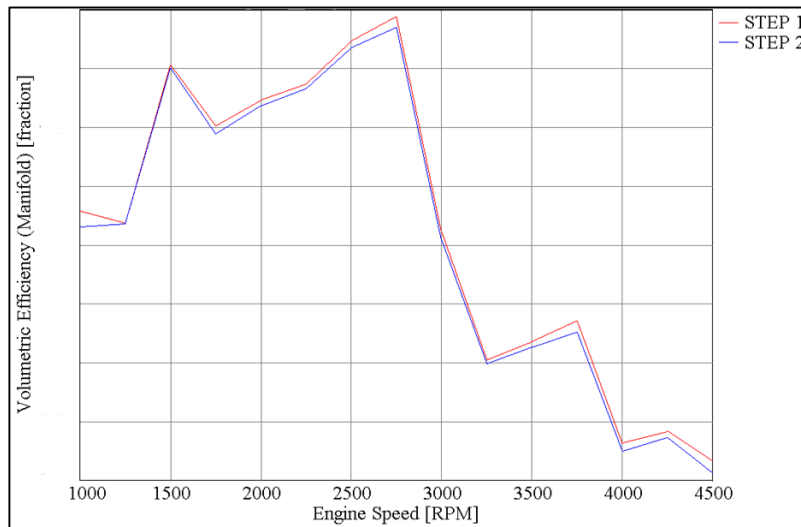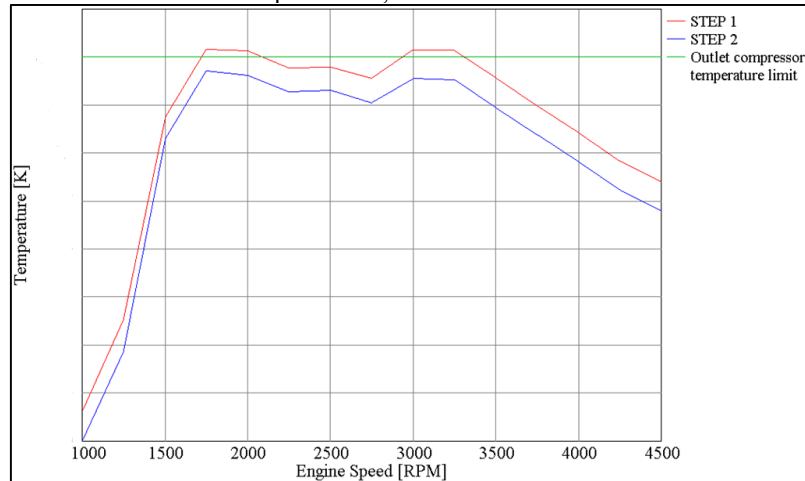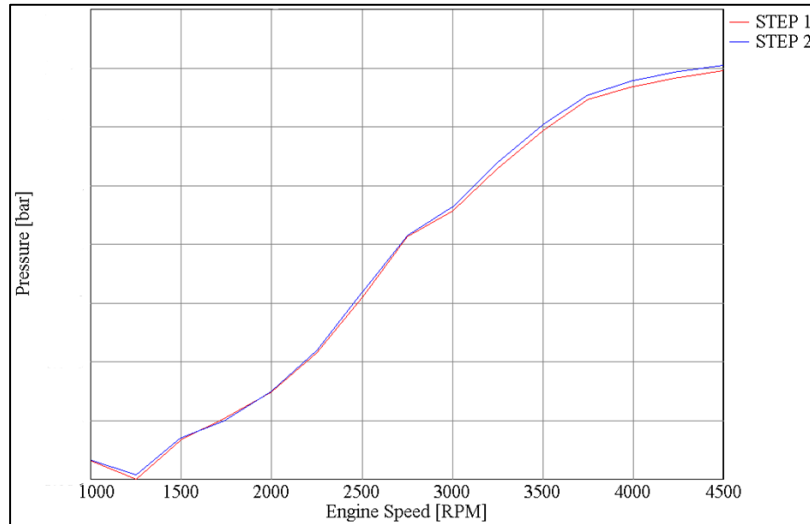STEP 1 and STEP 2

Figure 29 depicts outlet compressor temperature for new engine concept at STEP 1,

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STEP 2 and its limit.

Fig. 29: outlet compressor temperature comparison among new engine concept STEP 1, STEP 2 and its limit

Difference between STEP 1 and STEP 2 is owing to boundary condition of simulation that means constant BMEP values. In STEP 2 same BMEP level is generated with less friction so engine is more efficient and pressure ratio on compressor decrease, then outlet compressor temperature decrease. Not last IMEP720 decreases. STEP 2 is always under outlet compressor temperature limit while STEP 1 is over in two narrow engine range: 1750 – 2000 rpm; and 3000 – 3250 rpm. However, temperature extension is not consistent.

Figure 30 depicts inlet turbine temperature between new engine concept at STEP 1, STEP 2 and its limit.

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Fig. 30: inlet turbine temperature comparison among new engine concept
STEP 1, STEP 2 and its limit
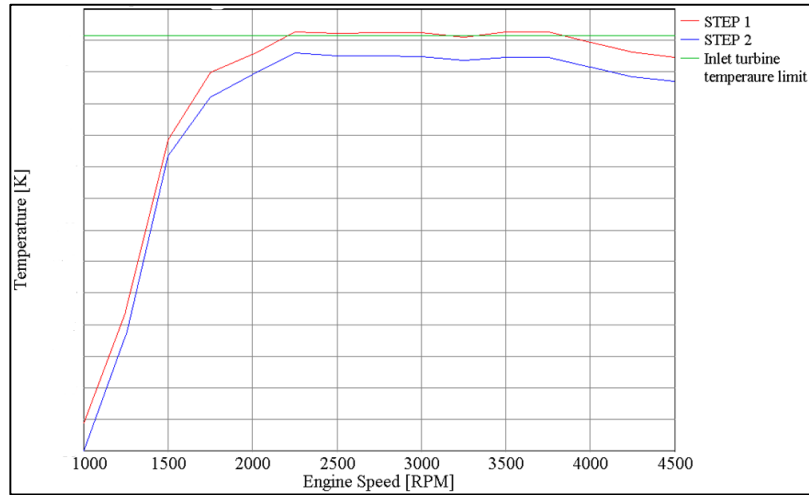
Reason for trend-off among STEP 1 and STEP 2 is same for outlet compressor temperature. Outlet compressor temperature and inlet turbine temperature are related to structural withstand of pipe for compressor and of house for turbine. STEP 2 is always under inlet turbine temperature limit while STEP 1 slightly over in engine range between 2250 – 3750 rpm.

Figure 31 shows engine back pressure related to exhaust system losses. New engine concept at STEP 1 and STEP 2 have same back pressure. However, there is no difference among STEP 1 and STEP 2.

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Fig. 31: engine back pressure comparison among new engine concept
STEP 1 and STEP 2

Figure 32 depicts turbine speed for new engine concept at STEP 1, STEP 2 and its limit related to maximum centrifugal force that turbine wheel is able to withstand.

Fig. 32: turbine speed comparison among new engine concept STEP 1,
STEP 2 and its limit
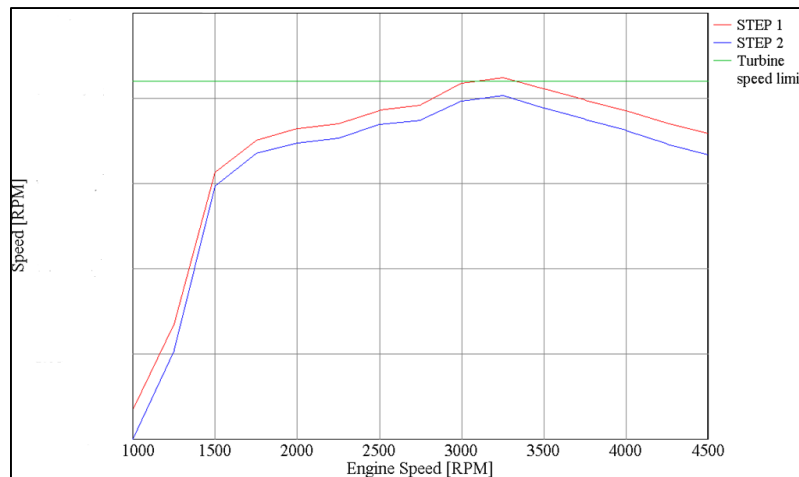
STEP 2 turbine speed is lower than STEP 1 because with same BMEP and low FMEP, IMEP decrease then engine needs less boost and Variable Geometry Turbine control decreases wheel speed. STEP 2 is always under turbine speed limit while STEP 1 is slightly over at 3250 rpm.

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Figure 33 shows air flow consumption between new engine concept at STEP 1 and STEP 2.

Fig. 33: turbine speed comparison among new engine concept STEP 1, STEP 2 and its limit
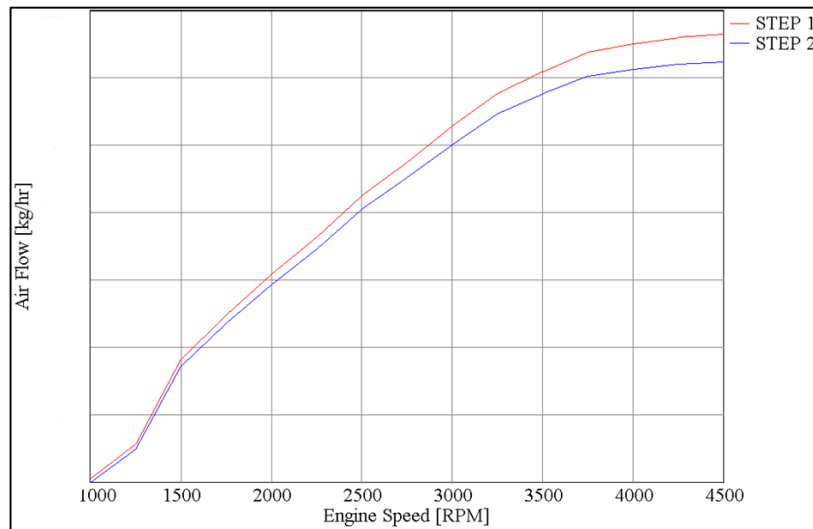
Difference is related at lower FMEP of STEP 2 which with same BMEP and same AF ratio needs less fuel and so less air. In fact, STEP 2's BSFC values are lower than STEP 1.

Figure 34 depicts AF ratio for new engine concept at STEP 1, STEP 2 and its reference.

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Fig. 34: AF ratio comparison among new engine concept STEP 1, STEP 2
and its limit
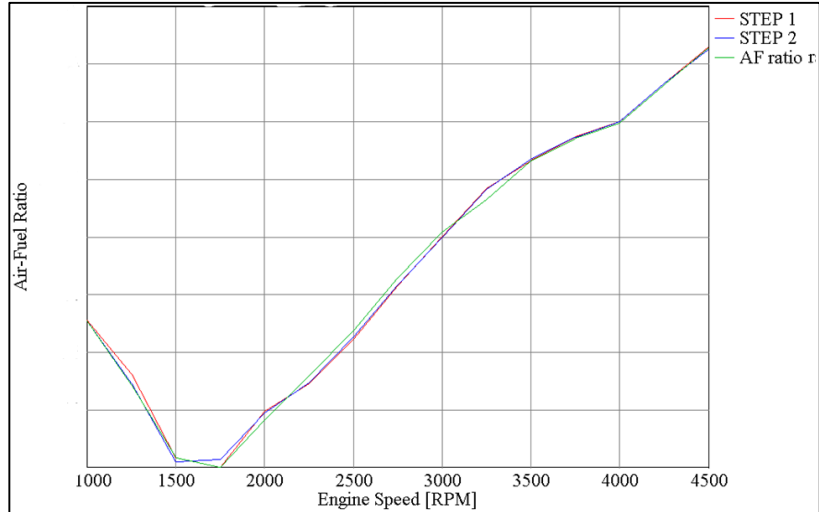
AF ratio is an extremely important parameter because is strongly related to fuel consumption or in other words BSFC value and pollutant emission in particular CO_2 and NO_x emissions. However, there is no difference among STEP 1 and STEP 2.

Figure 35 depicts inlet turbine pressure for new engine concept at STEP 1, STEP 2 and its limit.

Fig. 35: inlet turbine pressure comparison among new engine concept
STEP 1, STEP 2 and its limit
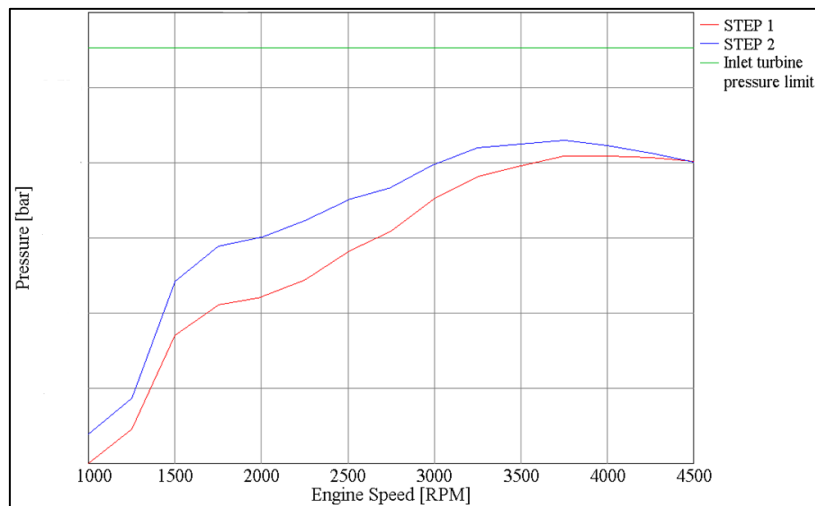
Both STEP 1 and STEP 2 are under inlet turbine pressure limit. STEP 2 pressure increases because air flow rate decreases.

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Figure 36 depicts Peak Firing Pressure for new engine concept at STEP 1, STEP 2 and its reference. There is no difference among STEP 1 and STEP 2.

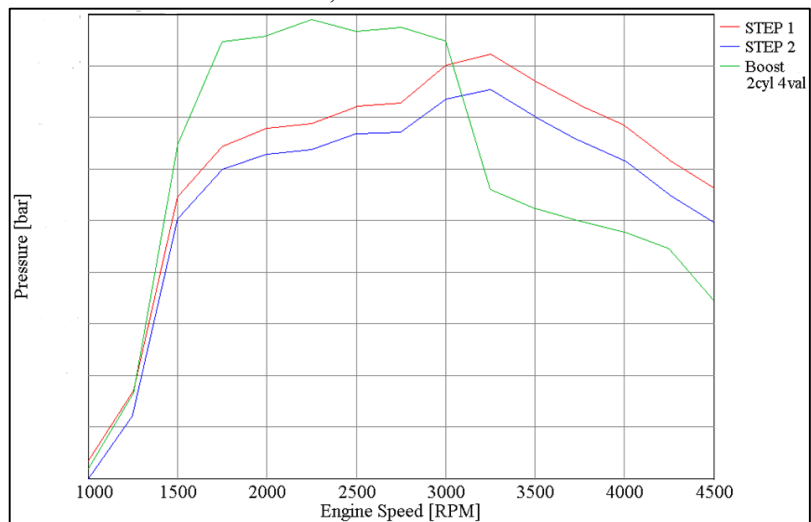Fig. 36: peak firing pressure comparison among new engine concept STEP 1, STEP 2 and its limit
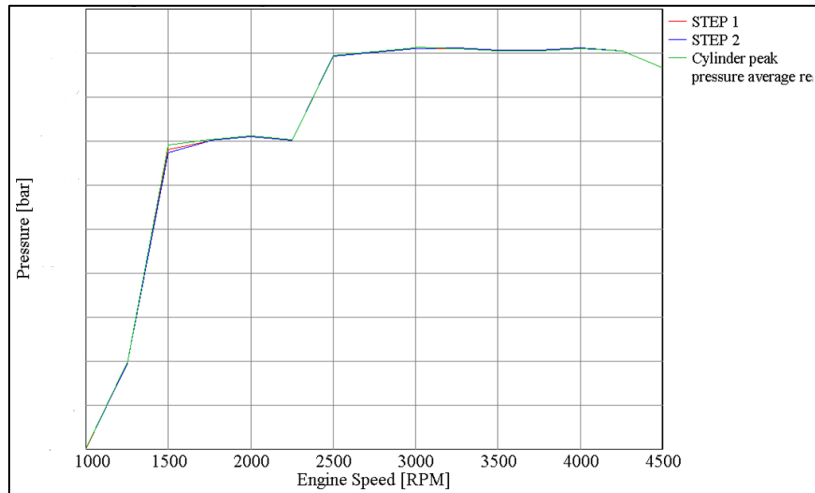
Figure 37 shows boost pressure for new engine concept at STEP 1, STEP 2 and its reference.

Fig. 37: peak firing pressure comparison among new engine concept STEP 1, STEP 2 and its limit

First of all, both STEPs do not follow boost pressure reference because boost estimation tool is not calibrated for small engine. Anyway, difference among STEP 1 and STEP 2 are related to

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strategy for engine friction reduction effects. In particular, with same BMEP and lower FMEP, engine needs lower IMEP and then lower boost pressure.

Figure 38 depicts compressor characteristic for new engine concept at STEP 1 and STEP 2.

Fig. 38: compressor characteristic comparison among new engine concept STEP 1 and STEP 2
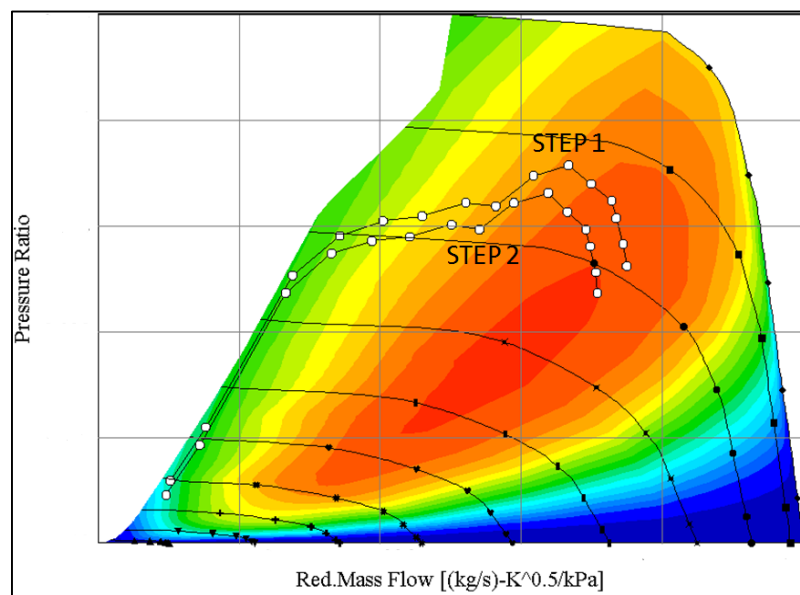
Difference are lower pressure ratio due to lower boost; lower mass flow owing to less air consumption, then lower IMEP with lower FMEP at same BMEP; obviously lower mass flow means lower compressor wheel speed. From STEP 1 to STEP 2, compressor works at higher efficiency points described through color contour map. About compressor characteristic could identify three phases. First one between 1000 and 1500 rpm, compressor works with almost constant mass flow and increase sharply pressure ratio amid inlet and outlet. In first phase, compressor efficiency is at last 50 %. In second phase among 1500 and 3000 rpm, compressor works with almost constant pressure ratio but increase sharply mass flow and so efficiency up to 70 %. In third phase, amid 3000 and 4500 rpm, compressor works almost at constant mass flow and sharply decrease pressure ratio in this way compressor efficiency increases up to 75 %.

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After the comparison among the STEP 1 engine and STEP 2 engine, it is possible conclude that with less friction, the engine becomes more efficient and all the requirements on the temperature are less stressed. The reason for this behavior is that the engine works with the same BMEP request. One of the main goal in this phase is know the performance limits of the model of new engine concept 2 cylinders 4 valves. For this reason it is analyzed the case that is increased the BMEP values and maintain the same temperature limits. In the case of STEP 3 engine is decided to set up the inlet turbine temperature limit to 1075.15 K that means 802 °C. In this way, the whole gap between the inlet turbine temperature and his structural limit have not used to increase the engine performance but only nearly half of it. To sum up, with the STEP 3 it is tried both to increase the BMEP that means an increased of engine performance and to maintain the BSFC limit that of STEP 2 or obviously less of it. In particular STEP 3 means new engine concept two cylinders 4 valves, friction model of GM 2l E5 reduce of 30%, turbocharger of GM 2l E5 with mass multiplier 0.5.

Figure 39 shows brake torque between new engine concept at STEP 2, STEP 3 and its reference.

Fig. 39: brake torque comparison among new engine concept STEP 2, STEP 3 and its reference
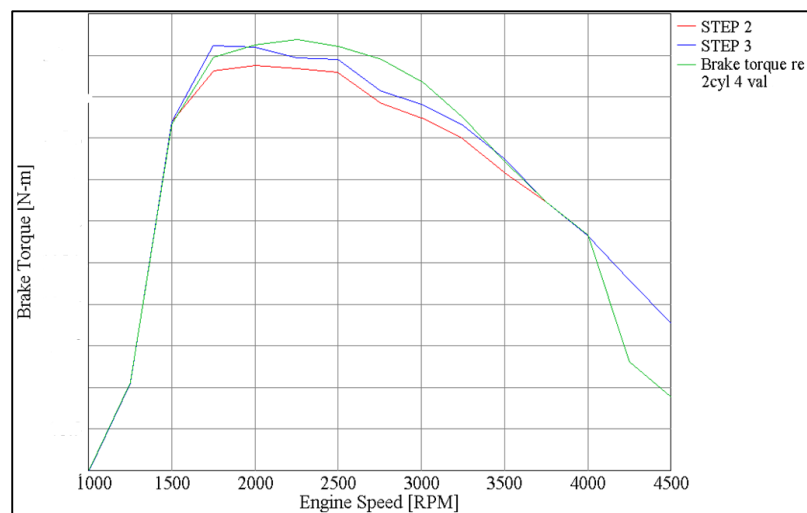
The difference between STEP 2 and STEP 3 is concerning the reduced engine friction that allow to use the whole IMEP that the engine generates. In this way BMEP increases

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and then brake torque curve is higher than STEP 2. In particular, in range 1500 - 1750 rpm, STEP 3's brake torque is over reference; in range 1000 – 1500 rpm and 3250 – 4500 rpm, STEP'3 torque curve follows strictly reference. Finally, brake torque curve improves in engine speed range of 1500 – 3750 rpm.

Figure 40 shows brake power for new engine concept at STEP 2, STEP 3 and reference.

Fig. 40: brake power comparison among new engine concept STEP 2, STEP 3 and its reference
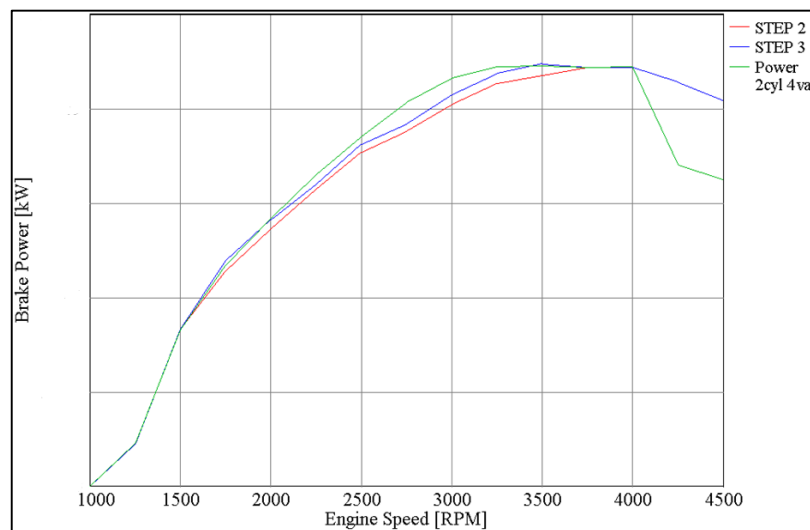
Brake power curve at STEP 3 improves in the same way of brake torque curve for same reason and also because torque and power are strongly related. At moment, there is a loss of power respects to reference only in engine range 2500 – 3250 rpm, but in general STEP 3's brake power curve achieves reference.

About BSFC of new engine concept at STEP 3, is the same of STEP 2. It follows reference up to 3000 rpm, see Figure 23, but over there is not negligible gap.

Figures 41 and 42 depict respectively IMEP360, gross that means only compression and expansion strokes, and IMEP720, net that consider all engine strokes, between STEP 2 and STEP 3 of new engine concept.

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Fig. 41: IMEP360 comparison among new engine concept STEP 2 and STEP 3
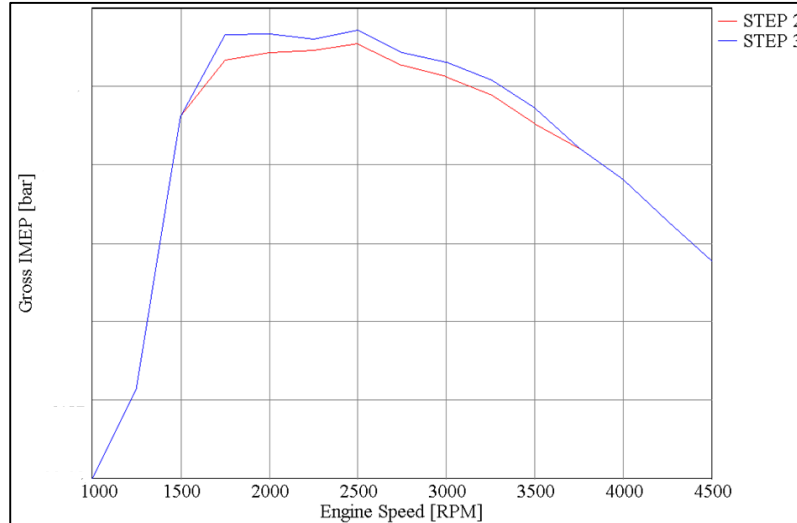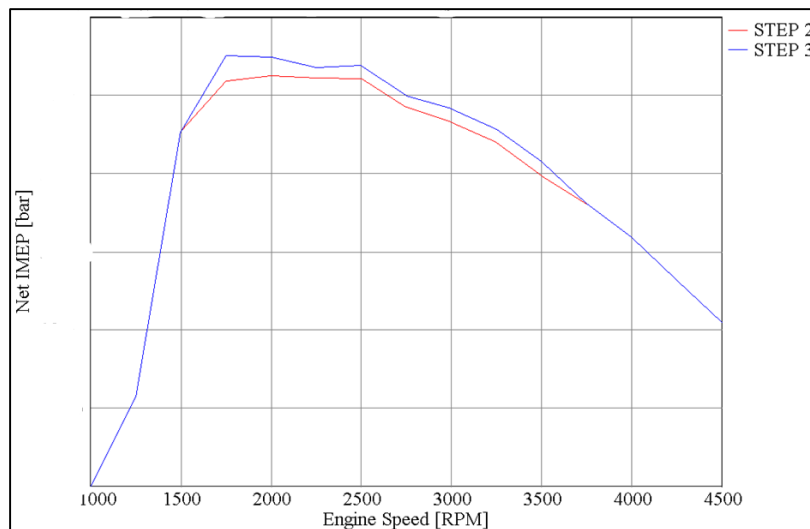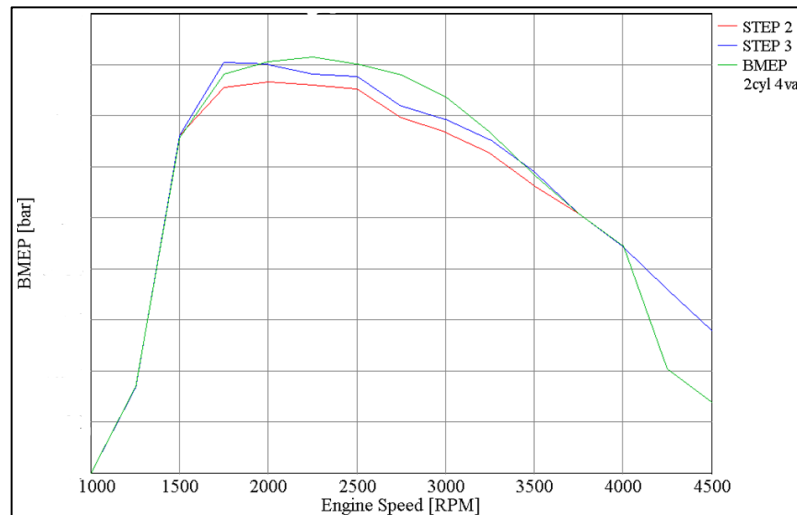

Fig. 42: IMEP720 comparison among new engine concept STEP 2 and STEP 3

Without BMEP limitation, engine could generate maximum IMEP and with same FMEP level leads improvements of torque and power. In fact, engine speed range of torque and power improvements match to higher IMEP.

About engine friction for new engine concept at STEP 3, is exactly the same of STEP 2.

Figure 43 depicts BMEP for new engine concept at STEP 2, STEP 3 and its reference.

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Fig. 43: BMEP comparison among new engine concept STEP 2, STEP 3 and its reference

Difference between STEP 2 and STEP 3 is owing to no limitation on BMEP level that allow to produce more IMEP and with same level of STEP 2's FMEP level means higher BMEP, higher torque and higher power. In general, new engine concept at STEP 3 follows strictly BMEP reference but unfortunately there is engine speed range among 2250 and 3000 rpm where is small gap. Between 1750 and 2250 rpm it works increasing both the inlet turbine temperature and the compressor outlet temperature, so the BMEP peak moves to 1750 as the brake torque peak. To sum up, an increase of BMEP with the same engine friction model leads to higher values of IMEP that means higher engine performance or in other words the engine has competitiveness advantage that it could use in several ways.

About volumetric efficiency of new engine concept at STEP 3, is almost the same of STEP 2. There is only slightly improvement in engine speed range of 1750 – 2750 rpm but situation is the same on whole.

Figure 44 depicts outlet compressor temperature for new engine concept at STEP 2, STEP 3 and its limit.

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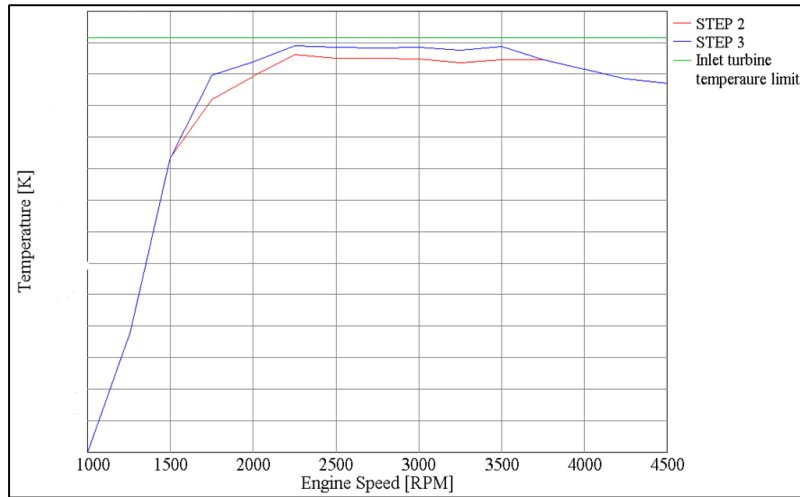
Fig. 44: outlet compressor temperature comparison among new engine concept STEP 2, STEP 3 and its limit
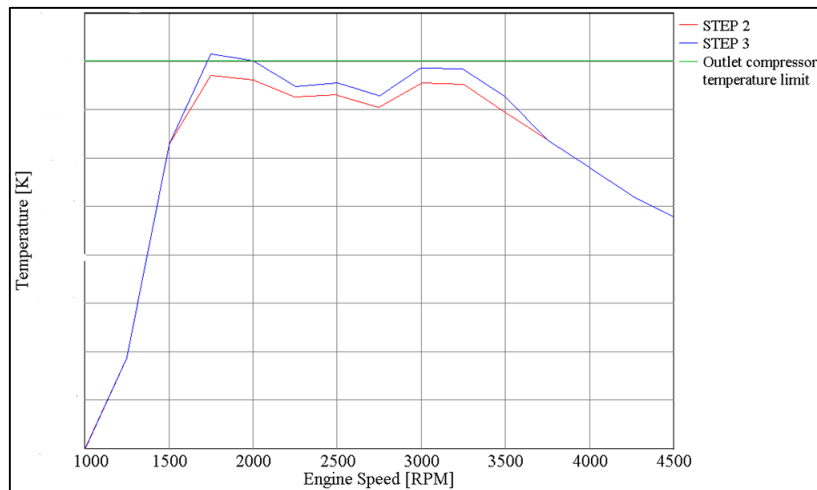
This limit is related to structural withstand of compressor outlet pipe and to avoid engine fault it has to be strictly controlled. In new engine concept at STEP 3, maximum BMEP level is achieved and for this reason more air is needed and then higher pressure ratio that mean higher outlet compressor temperature. It increases only in engine speed range 1500 – 3750 rpm because STEP 2 performance are far away from reference in this range. However, outlet compressor temperature is over limit of few degrees only at 1750 rpm where there is IMEP, BMEP and brake torque peak.

Figure 45 shows inlet turbine temperature between new engine concept at STEP 2, STEP 3 and its limit.

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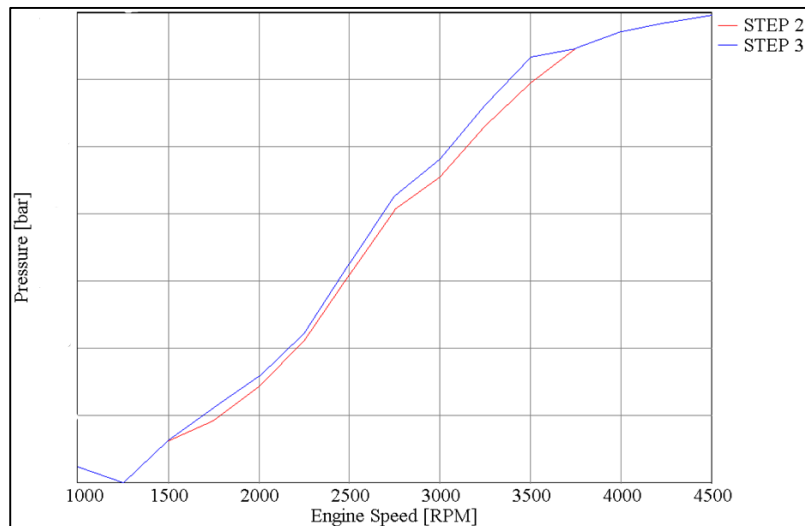Antonio Turturro – Supervisors: Prof. C. Delprete and L. Magro

Fig. 45: inlet turbine temperature comparison among new engine concept STEP 2, STEP 3 and its limit

Difference is related to higher IMEP values but anyway temperature limit is respected. How outlet compressor limit, inlet turbine limit is related to structural withstand of turbine house mainly. Between 1750 rpm and 3500 rpm there is an increase of inlet turbine temperature that means more air, more fuel, more BMEP, more torque.

Figure 46 shows engine back pressure among new engine concept at STEP 2 and STEP 3.

Fig. 46: engine back pressure comparison among new engine concept STEP 2 and STEP 3

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There is increase of engine back pressure due to exhaust system that means increase of outlet turbine pressure owing to higher air consumption. Unfortunately, all turbine and compressor parameter relate to pressure, temperature and speed rise.

Figure 47 depicts turbine speed between new engine concept at STEP 2, STEP 3 and its limit related to maximum centrifugal force that turbine wheel withstand.

Fig. 47: turbine speed comparison among new engine concept STEP 2, STEP 3 and its limit
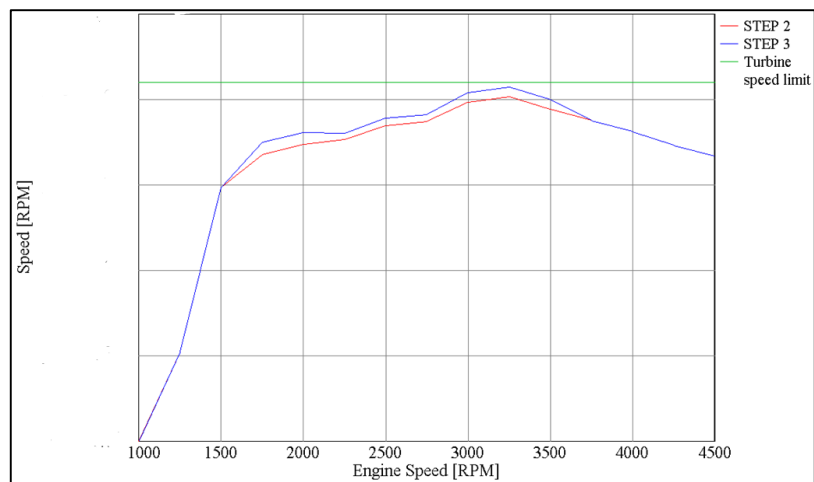
Higher engine performance need higher turbine performance to allow higher compressor performance and so turbine speed increase.

Figure 48 shows air consumption of new engine concept at STEP 2 and STEP 3.

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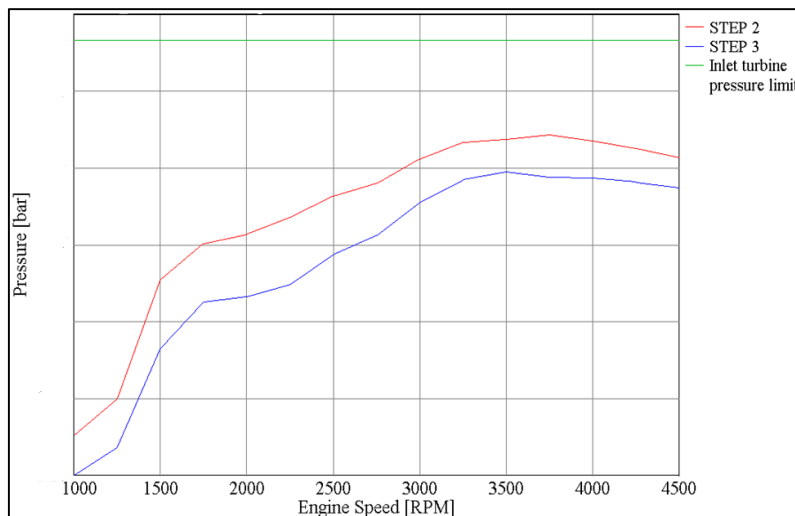Antonio Turturro – Supervisors: Prof. C. Delprete and L. Magro

Fig. 48: air consumption comparison among new engine concept STEP 2 and STEP 3
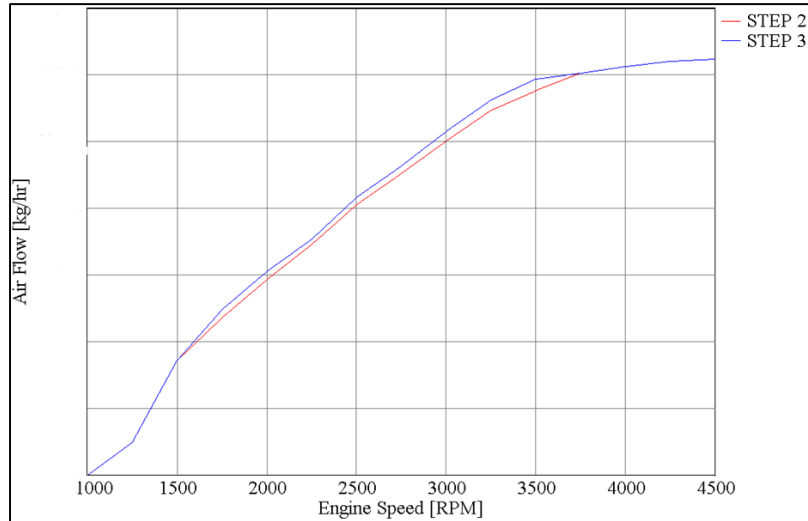
STEP 2 engine does not strictly follow performance reference in central part of engine regimes then is tried to increase engine performance through higher IMEP than means, with same FMEP level, higher BMEP. So more air is needed to burn more fuel with same pollutant limit that means at same AF ratio. For this reason air consumption of new engine concept at STEP 3 is higher.

Figure 49 depicts inlet turbine pressure for new engine concept at STEP 2, STEP 3 and its limit.

Fig. 49: inlet turbine pressure comparison among new engine concept STEP 2, STEP 3 and its limit

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STEP 2 as well as STEP 3 is under pressure limit. STEP 3 works with higher air flow rate and then turbine control opens much more by-pass and so pressure decrease.

About Peak Firing Pressure, new engine concept at STEP 3 follows pretty well PFP reference trend shown in Figure 36.

Figure 50 depicts boost pressure level of new engine concept at STEP 2, STEP 3 and its reference.

Fig. 50: boost pressure comparison among new engine concept STEP 2, STEP 3 and its reference
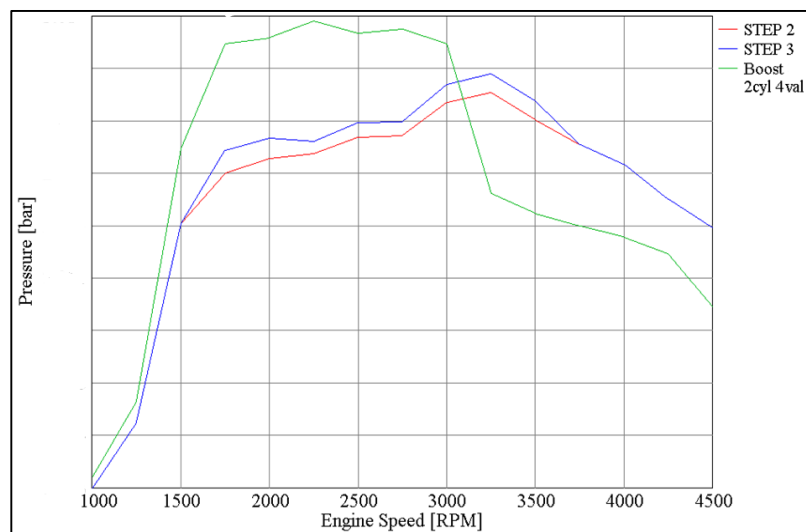
Boost estimator tool overvalues the boost request, however the STEP 3 model need a higher boost level to generate maximum IMEP values in according with all others charts.

Figure 51 depicts compressor characteristic for new engine concept at STEP 2 and STEP 3.

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Fig. 51: compressor characteristic comparison among new engine concept STEP 2 and STEP 3
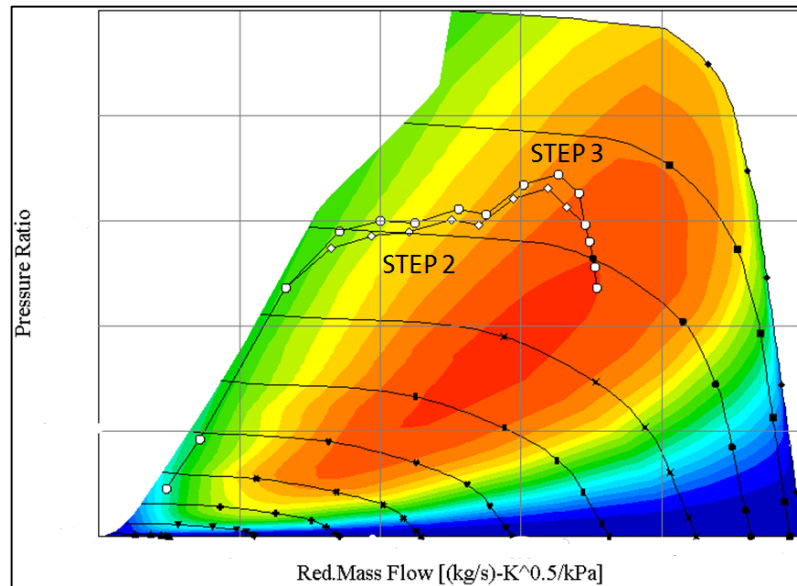
The compressor characteristic curve for STEP 3 model shows an increase of all compressor parameters in engine speed range where increase of engine performance is achieved. In particular, with higher IMEP levels that mean higher air flow rate and higher boost levels, compressor works in much efficiency way as shown efficiency map, color contour map, in Figure 51.

The comparison between STEP 1 and STEP 2 shows that a simplified architecture leads to less engine friction and increase the engine efficiency. In this way, the engine can achieve the performance request with a lower outlet compressor and inlet turbine temperature. On the other hand, in the case of STEP 3 it is used the gap between the previous temperatures and their limits to increase the engine performance but maintain the brake specific fuel consumption lower than STEP 2. Figures 52 and 53 sum up the main engine parameters of STEP 3 model.

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Fig. 52: brake torque and power curves of new engine concept at STEP 3
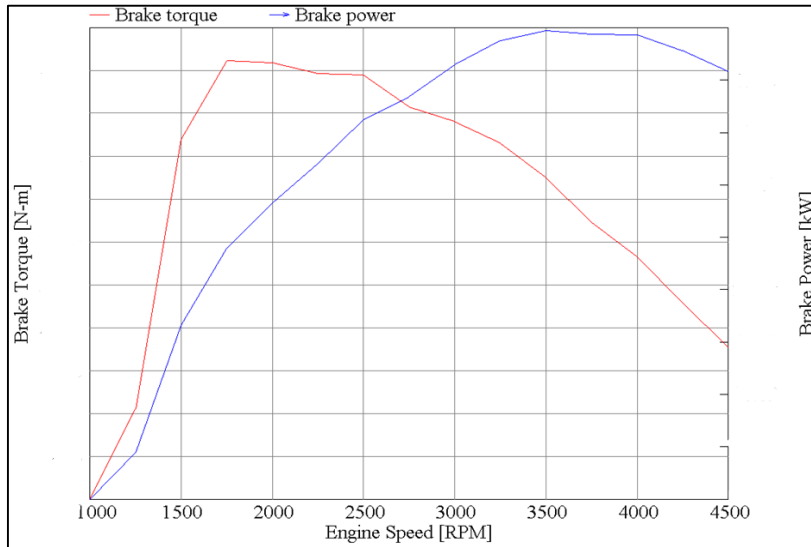
Fig. 53: BMEP and BSFC curves of new engine concept at STEP 3

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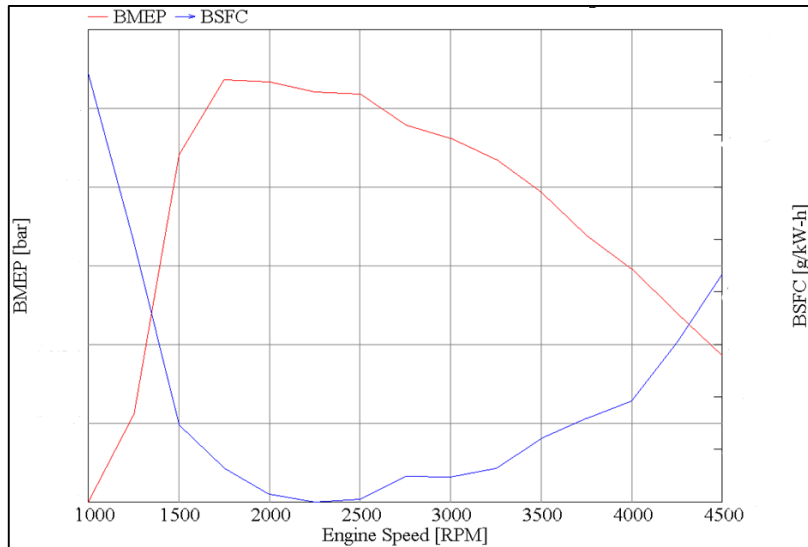
II-IV Effects of exhaust engine breath capacity on engine performance

The exhaust valve diameter could affect different aspects on engine performance as the residual gas fraction and the blowdown phase of the exhaust process. Moreover, smaller exhaust valves diameter allows to locate bigger intake valve diameter and then increases volumetric efficiency. Further, small valves give to designer same degrees of freedom to locate in better way all components that face on combustion chamber with important effects on cylinders head design. The residual gas fraction is strongly linked to the combustion that means engine performance and the volumetric efficiency. While the blowdown phase of the exhaust process in bonded to the organic efficiency and the turbocharger performance. For this reason the exhaust valve reference diameter is reduced by 0.5 mm and 1 mm. The starting point is an exhaust valve diameter of 20.1 mm then the model has run with a diameter of 19.6 mm and 19.1 mm. In this analysis, the forward and backward discharge coefficients are the same of the starting point that means that the model use the same ducts.

In legends of next charts there will be follow simulations:

- STEP 3 = same meaning of previous section and in particular new engine concept 2cylinders 4 valves, friction model of GM 2l E5 reduce of 30%, turbocharger of GM 2l E5 with mass multiplier 0.5;
- STEP 3_Dia_Ref_Exh-05 = as STEP 3 engine model but with exhaust valve reference diameter reduces of 0.5 mm;
- STEP 3_Dia_Ref_Exh-1 = as STEP 3 engine model but with exhaust valve reference diameter reduces of 1 mm.

Brake torque and power as well as BSFC do not have any change because exhaust valve diameter does not relate with these engine parameters. In particular, increase of engine speed, the BSFC slightly increase too because with a reduction of exhaust diameter the blowdown phase decrease and increase the displacement exhaust process.

IMEP360, IMEP720 and FMEP are insensible at exhaust valve diameter.

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On the other hand, Pumping Mean Effective Pressure is affected for exhaust valve diameter especially at high engine speed as shown in Figure 54.

Fig. 54: PMEP comparison for new engine concept with all exhaust diameter variations
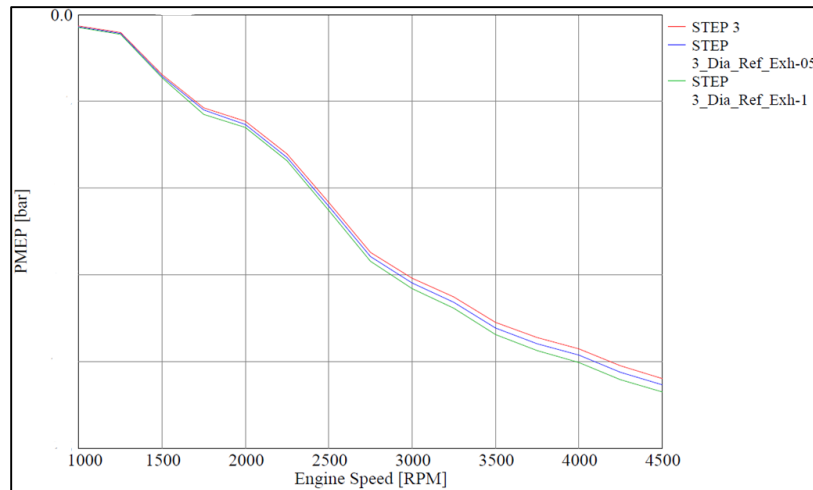
As it is clear from this graph the pumping losses increase with the engine speed that means the phenomena that occurs in reduction of exhaust valve diameter is fundamentally fluid dynamic.

BMEP keeps constant at same level of new engine concept at STEP 3 because exhaust valve diameter variations do not affect this parameter.

About volumetric efficiency, as shown in Figure 55, there is slightly deterioration of this parameter over 3250 rpm.

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Fig. 55: volumetric efficiency comparison for new engine concept with all exhaust diameter variations
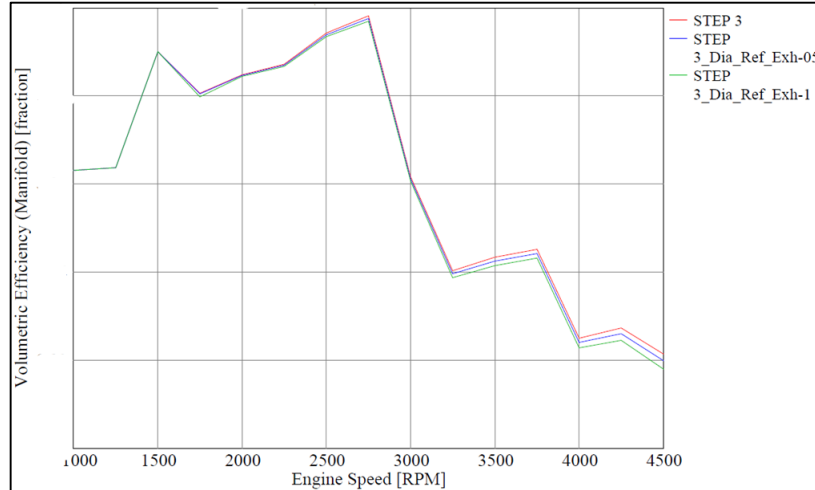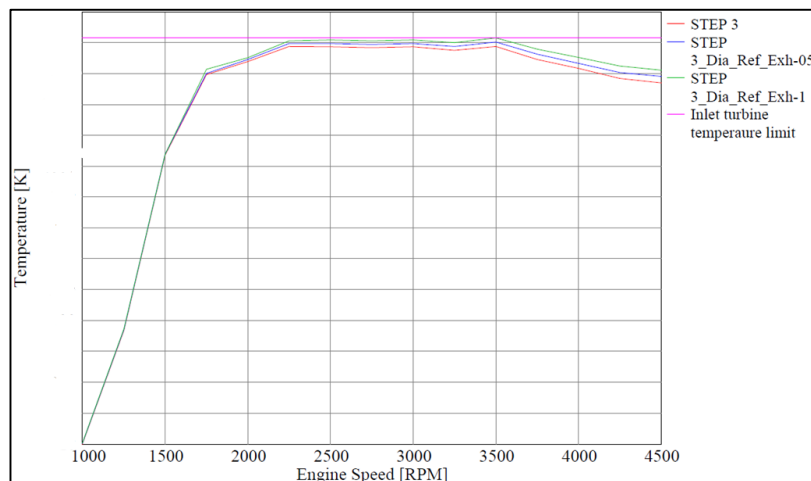

With a smaller exhaust valve diameter, the residual gas fraction increase so the volumetric efficiency decrease, as shown in previous graph.

Outlet compressor temperature maintains same values of new engine concept at STEP 3.

Inlet turbine temperature rises especially for new engine concept at STEP 3 with exhaust valve diameter reduction of 1 mm as shown in Figure 56.

Fig. 56: inlet turbine temperature comparison for new engine concept with all exhaust diameter variations

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Exhaust valve diameter reduction affects fluid-dynamic behavior and it highlights at high engine speed in particular over 3500 rpm. However all outputs are under inlet turbine temperature limit. With the reduced exhaust valve diameter, there is a little increase of the inlet turbine temperature because exhaust gases have few more energy than reference diameter.

Engine back pressure, turbine speed, air consumption, AF ratio, rack position, inlet turbine pressure, PFP, boost pressure, compressor characteristic are not affected of exhaust valve diameter reduction.

The results of the exhaust valve diameter reduction are a generalize increase of temperature and pressure level of the engine with a reduction of volumetric efficiency and an increase of Break Specific Fuel Consumption. There is an increase of pumping mean effective pressure if also the whole friction of the engine are pretty the same. To sum up, exhaust valves diameter reduction over 20.1 mm does not have any positive effects. This diameter is optimum one.

II-V Turbochargers screening and its effect on engine performance

Until now, for the new engine concept, 2 cylinders 4 valves, it has used an ideal turbocharger came from GM 2l E5 engine. Now, four turbochargers will be matched with the engine model. The main characteristics of the four volumetric machines are presented on Table 1. The data are only indicative.

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Tab. 1: turbochargers parameters overview

Model	VGT	WGT	Turb dia [mm]	Comp dia [mm]	Max speed [krpm]	Trim turb – comp [%]	AR turb – comp [mm]
G	X		35	35	275	70 - 50	0.75–0.40
T		X	30	35	300	65 - 45	0.20-0.40
1	X		35	40	275	70 - 45	0.75-0.25
0		X	30	40	300	65 - 45	0.20-0.25

First column is for turbochargers name; they are only indicative. Second and third column is about turbine control technologies: Variable Geometry Turbine and Waste-Gate Turbine. Fourth and fifth column represent respectively turbine and compressor wheel diameter. Sixth one is for maximum turbine speed. Seventh one is trim parameter defined as ratio between minimum and maximum wheel diameter. Eighth column is AR ratio defined as ratio between pipe area at beginning of tongue and distance of its centre from volute's centre. In a pre-screening phase, three turbochargers is deleted because their dimensions are too big for new engine concept as shown in Table 2. Data are only indicative.

Tab. 2: discard turbochargers parameters overview

Model	VGT	WGT	Turb. Dia. [mm]	Comp. Dia. [mm]	Max Speed [krpm]
K	X		30	40	260
2	X		35	40	260
P		X	30	30	265

Figures 57 – 60 show four compressor characteristics select for screening activity. On these charts, pre-surge line is presented. It needs to respect noise comfort requirement, otherwise compressor will be too noisy for customers.

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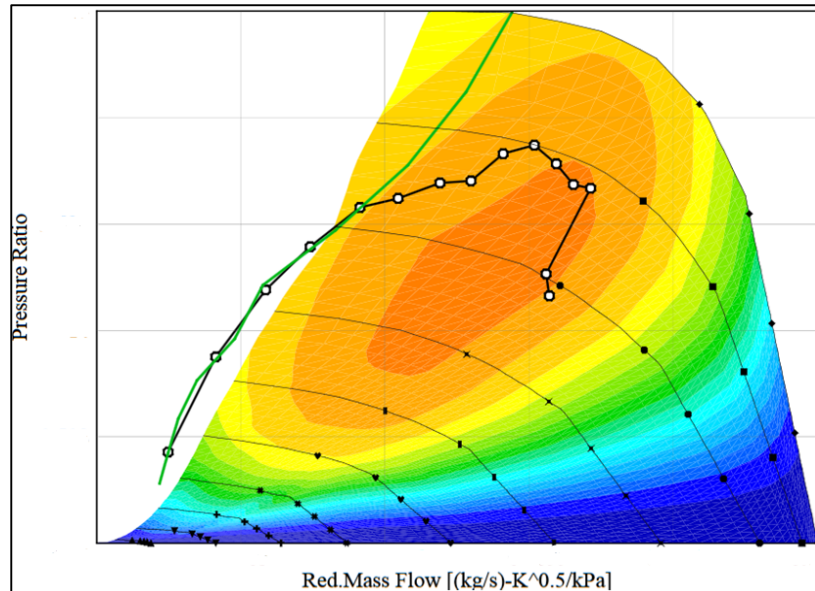Antonio Turturro – Supervisors: Prof. C. Delprete and L. Magro

Fig. 57: "1" compressor characteristic and pre-surge line; color contour map is efficiency

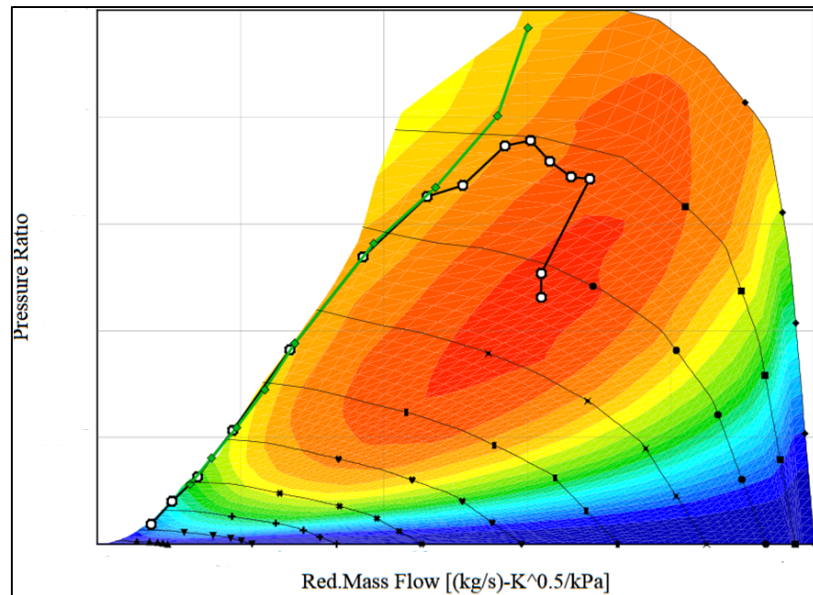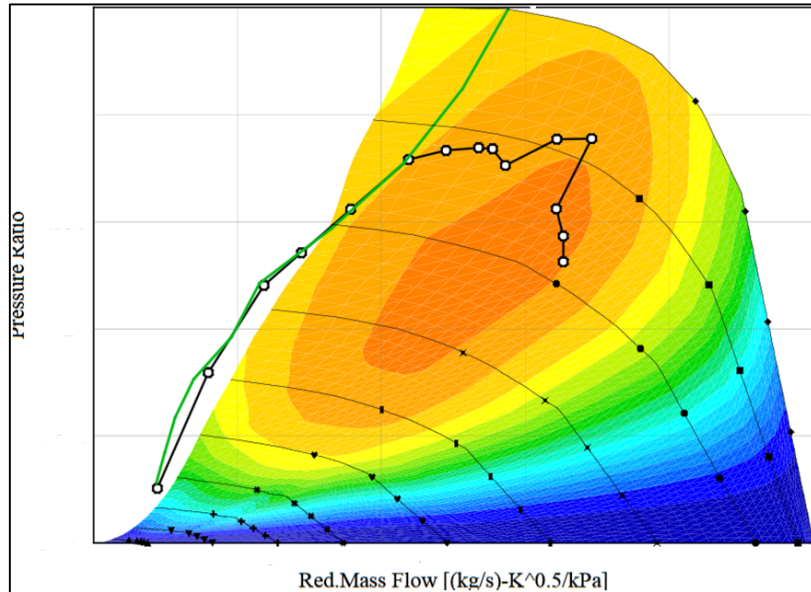Fig. 58: "G" compressor characteristic and pre-surge line; color contour map is efficiency

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Fig. 59: "0" compressor characteristic and pre-surge line; color contour map is efficiency

Fig. 60: "T" compressor characteristic and pre-surge line; color contour map is efficiency
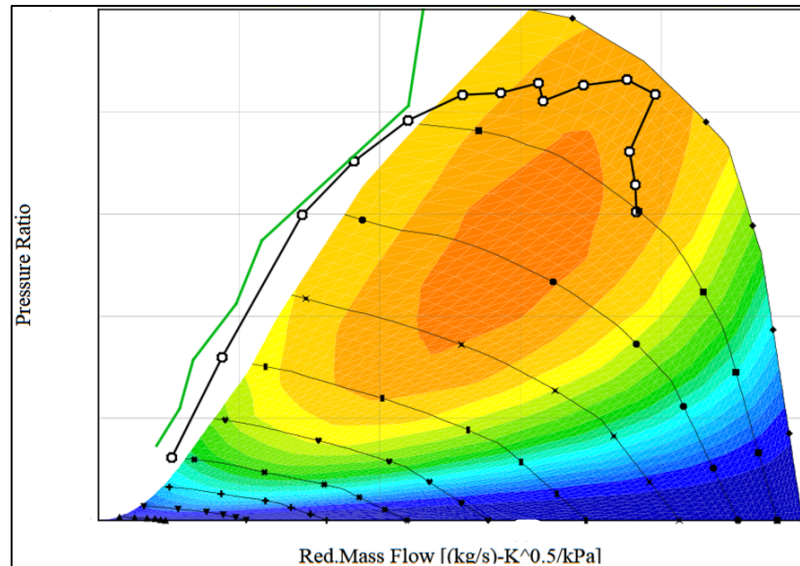
First of all, compressor "1", "0", "T" have pre-surge line over map, while compressor "G" has pre-surge line on surge edge of map. The "1" has the low-end performance far away from the target if the operation line of the compressor is inside the map. So, the

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operation line is forced until the pre-surge line. And in the same way, both the WGT, "1" and "0", have a performance far away from the target if the operation line of the compressor is inside the map. So, the operation line is forced until the pre-surge line. Unfortunately, in case of "G", has to accept very low performance up 2000 rpm because operation points of compressor correspond at low pressure ratio and low mass flow rate. In fact, as shown in Figure 61, compressor "G" is not able to partialize enough to generate a proper boost level remaining inside the allowable compressor map.

Fig. 61: rack position and waste-gate diameter of four turbine
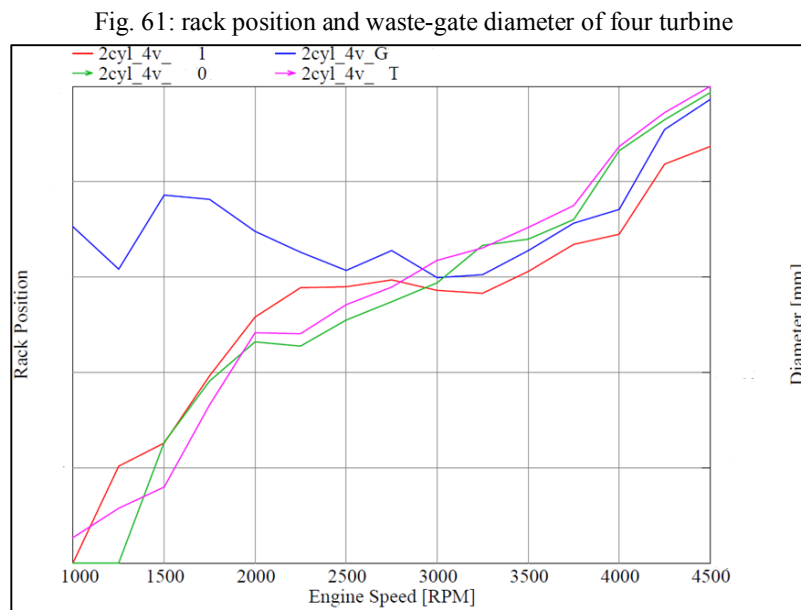
Apart the "G" that is out of discussion for compressor characteristic not suitable for new engine concept application, the trend of the others turbocharger are very good. Turbocharger "G" to respect compressor characteristic and pre-surge line cannot close rack and then engine does not have right amount of air with proper boost level as shown in Figures 62 and 63.

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Fig. 62: air consumption for four turbochargers
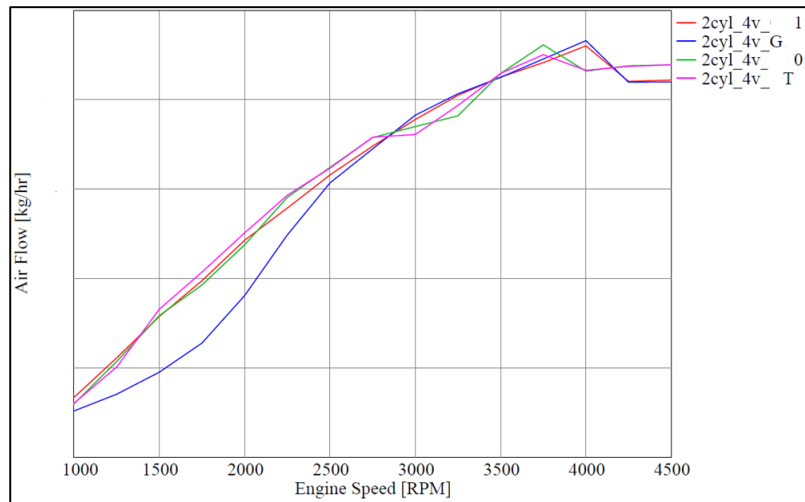
The air consumption are pretty the same for turbochargers "1", "0" and "T" that means right air flow values but in case of "G", air consumption values are much lower because compressor "G" is not able to manage higher mass flow rate. In particular, compressor "G" is not able to generate proper air flow rate up 2500 rpm.

Fig. 63: boost level for four turbochargers and reference
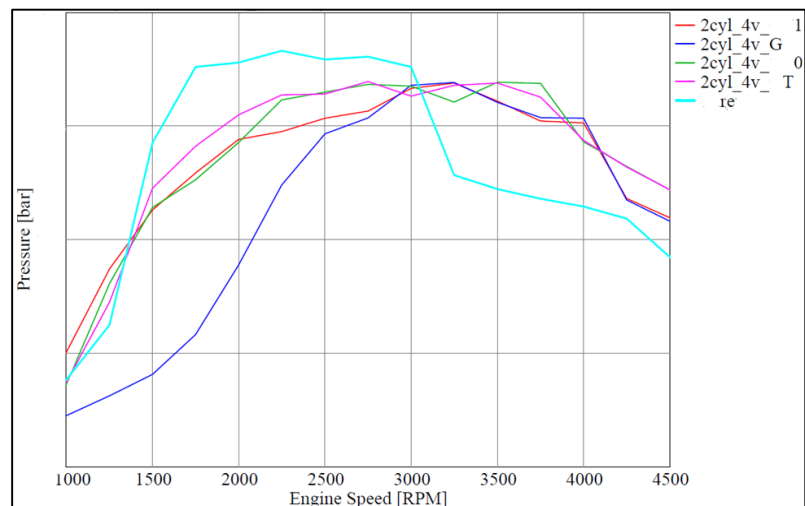
Apart the "G", the boost pressure trend does not follow the target but is sufficiently good. Compressor "G" is not able to generate adequate boost level up 2500 rpm. "G"

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characteristic has to low pressure ratio for small mass flow rate and for this reason turbocharger "G" is not suitable for new engine concept application.

Next charts study pressure and temperature parameters for four turbochargers at all. Figure 64 depicts outlet compressor temperature for four cases.
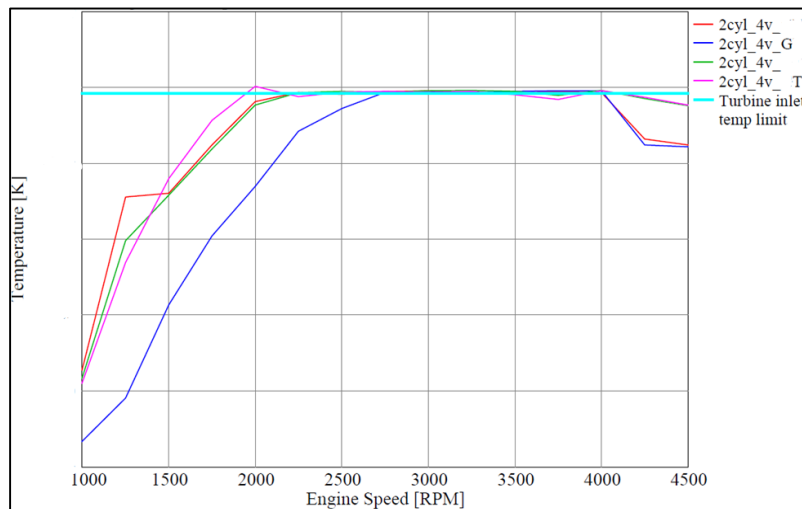
Fig. 64: outlet compressor temperature curves for four turbochargers
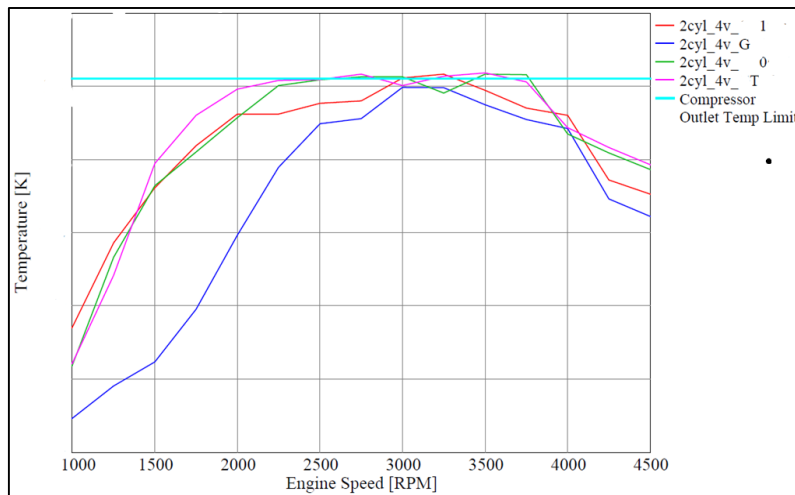
Obviously, the "G" – blue line – has a bad behavior up 2500 rpm after that the operation line of compressor separates the pre-surge line. It does not compress enough and then air temperature does not rise.

Figure 65 depicts inlet turbine temperature for four turbines at all.

Fig. 65: turbine inlet temperature curves for four turbochargers

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Apart "G", other three turbines work in same temperature range per every engine speed. "T" goes over inlet turbine temperature for few degrees at 2000 rpm but it is still acceptable. "G" starts to work over 2500 rpm. For the same reason of the previous chart, the "G" has a bad behavior too.
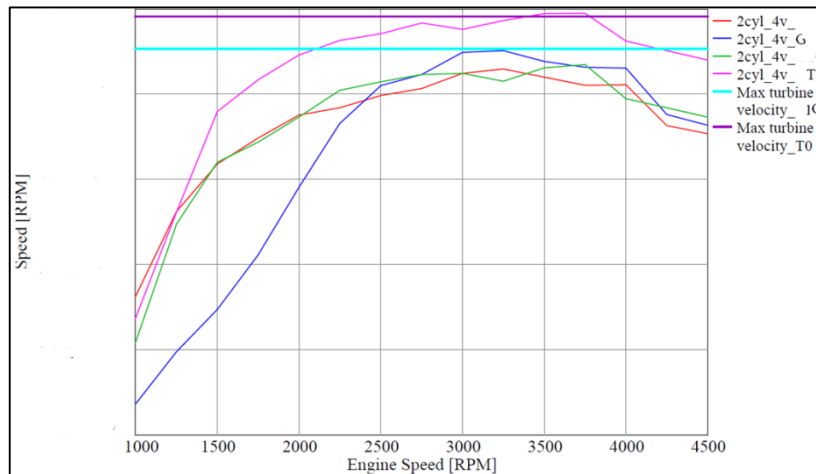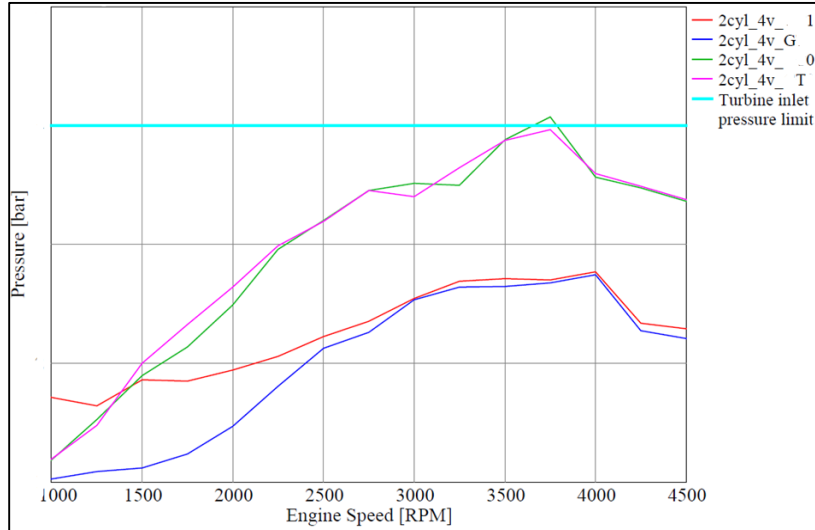
Fig. 66: turbine speed curves for four turbochargers

Figure 66 shows four turbine speed for new engine concept. "G" and "1" use same turbine with two different compressor; over 2500 rpm, turbine speed difference is owing to different compression dimension. There is same situation for "T" and "0". While the "T" arrives at his limit, the "0" not; maybe because a bigger compressor needs to run slowly than a small one. This opinion could be apply also at "G" and "1".

Figure 67 shows inlet turbine pressure for four cases at all.

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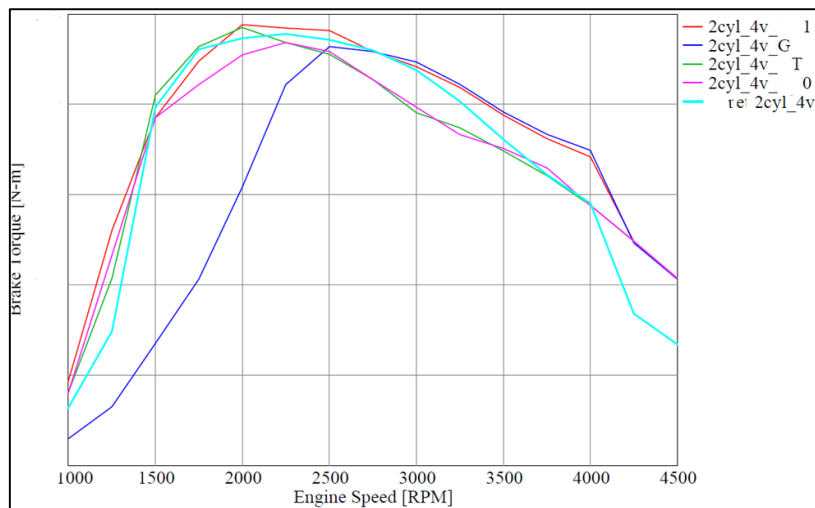Antonio Turturro – Supervisors: Prof. C. Delprete and L. Magro

Fig. 67: inlet turbine pressure curves for four turbochargers

The dimension of "T" and "0" turbines are too small and they reach inlet pressure limit. Turbine "1" has best inlet pressure trend while turbine "G" has extremely low inlet pressure up 2500 rpm.

Next charts present new engine concept performances obtained with previous four turbochargers. Figure 68 presents brake torque of new engine concept.

Fig. 68: brake torque curves for four turbochargers

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First of all, the "G" – blue line – has a low performance owing to both the compressor map and the pre-surge line. Then it is possible to analysis the 4 turbochargers in 2 groups: the WGTs and the VGTs. Until 1500 rpm, the performances are pretty the same and over the reference, so there is an increase of low end performances. After that the turbocharger have different behavior. The "T" goes over the target up 2000 rpm after, it goes down up to 3750 rpm. The "0" has nearly the same performance of the "T" but it never goes over the reference. The "1" is the best turbocharger for brake torque performance and is always over the reference.
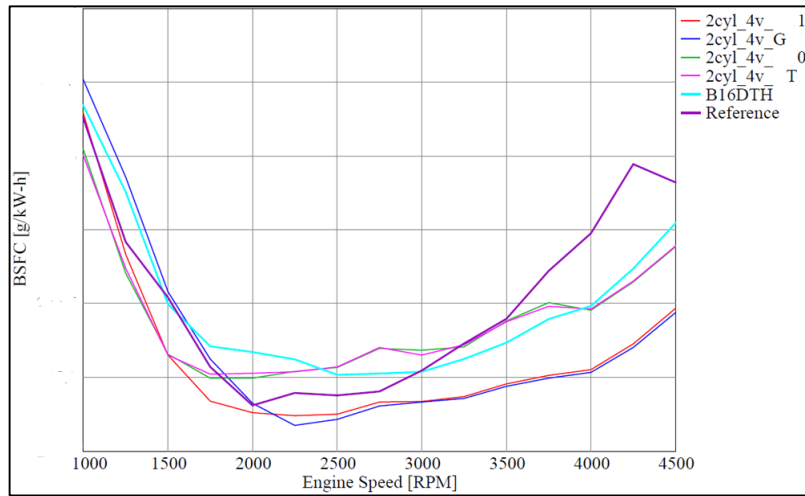
Figure 69 shows brake power for new engine concept matched with four turbochargers.

Fig. 69: brake power curves for four turbochargers
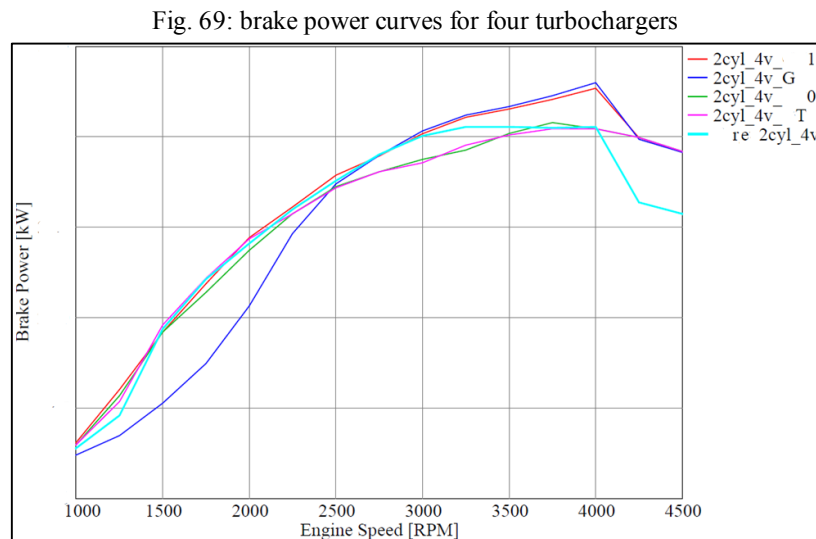
Apart the "G", the others turbocharger follow pretty well the brake power reference trend. There is an increase of low end performance and over 3250 rpm in particular for VGT's turbochargers as "G" and "1". The "T" and "0" have a power gap between 2750 and 3500 rpm.

Figure 70 depicts Brake Specific Fuel Consumption for new engine concept equipped with four turbochargers.

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Fig. 70: BSFC curves for four turbochargers

Substantially the four turbochargers have a good performance in term of BSFC and all of those respect the reference. The "G" and "1" have a better BSFC performance in comparison with the "T" and "0", so the lower turbocharger performances. Obviously, under 2500 rpm, "G" has higher BSFC because it generates lower performance in terms of torque and power.

Figures 71 and 72 depict gross, 360, and net, 720, Indicated Mean Effective Pressure for new engine concept with four turbochargers.

Fig. 71: IMEP360 curves for four turbochargers

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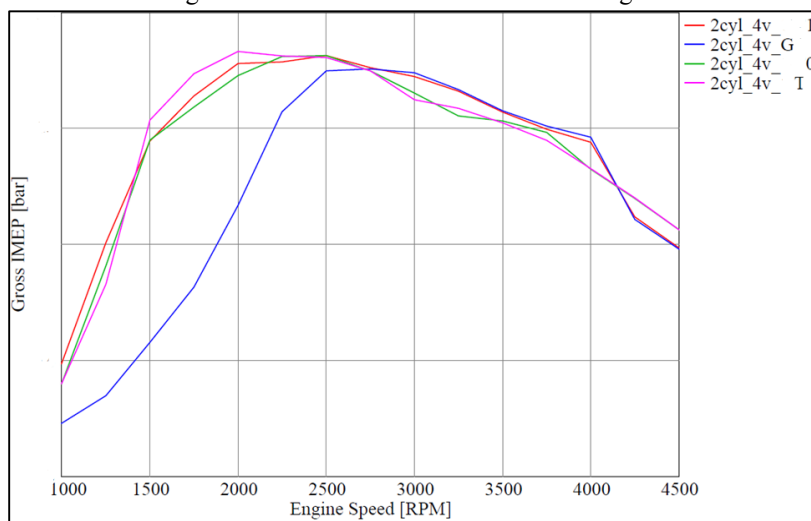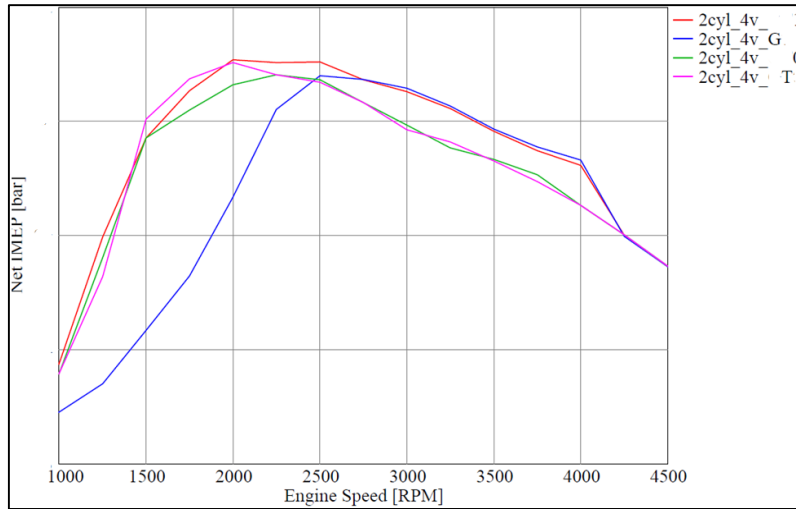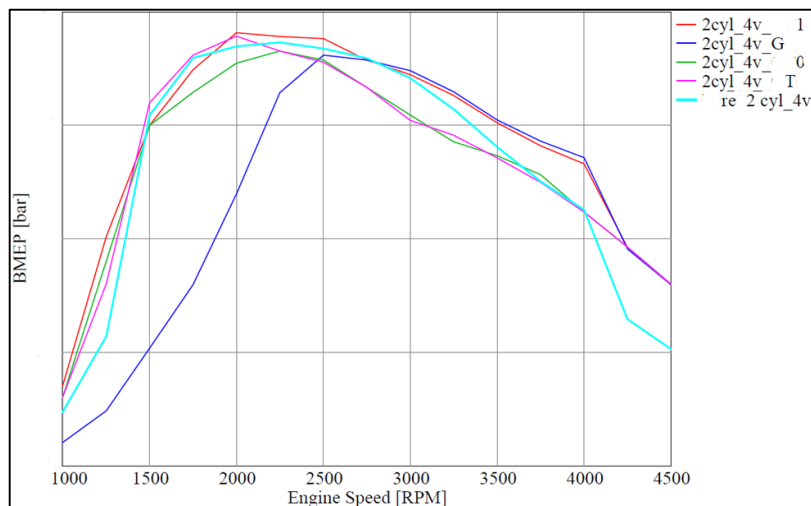Antonio Turturro – Supervisors: Prof. C. Delprete and L. Magro

Fig. 72: IMEP720 curves for four turbochargers

Both IMEP360 and IMEP720 highlight importance of right turbocharger in terms of characteristic and dimensions. Turbocharger "G" is not able to generate good combustion pressure up to 2250 rpm. This behavior is strongly negative for new engine concept performance.

Figure 73 shows Brake Mean Effective Pressure of new engine concept equipped with four turbochargers.

Fig. 73: BMEP curves for four turbochargers

BMEP is strongly related to brake torque and it strictly follows this parameter.

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Figures 74 - 75 sum up new engine concept equipped respectively with turbocharger "1" for VGT technology and turbocharger "0" for WGT technologies, respectively in terms of brake torque and power, and BSFC and BMEP.

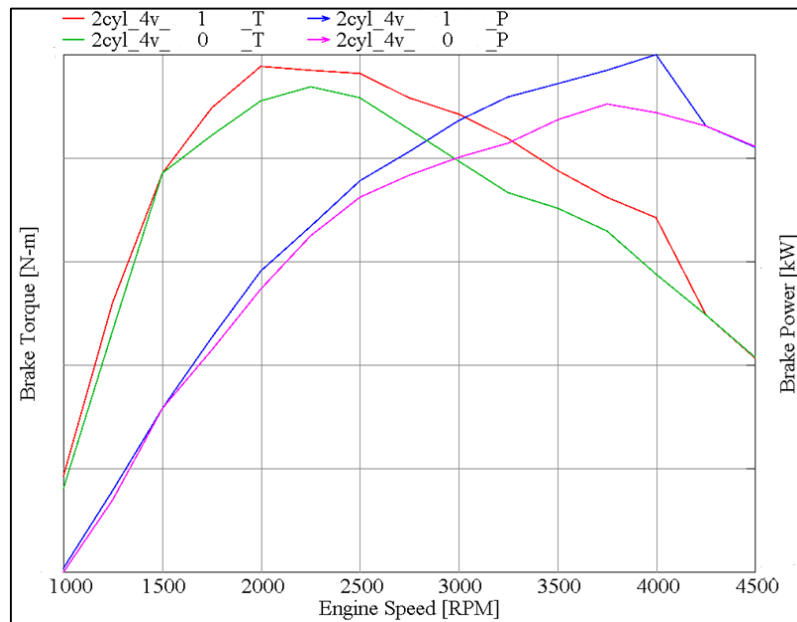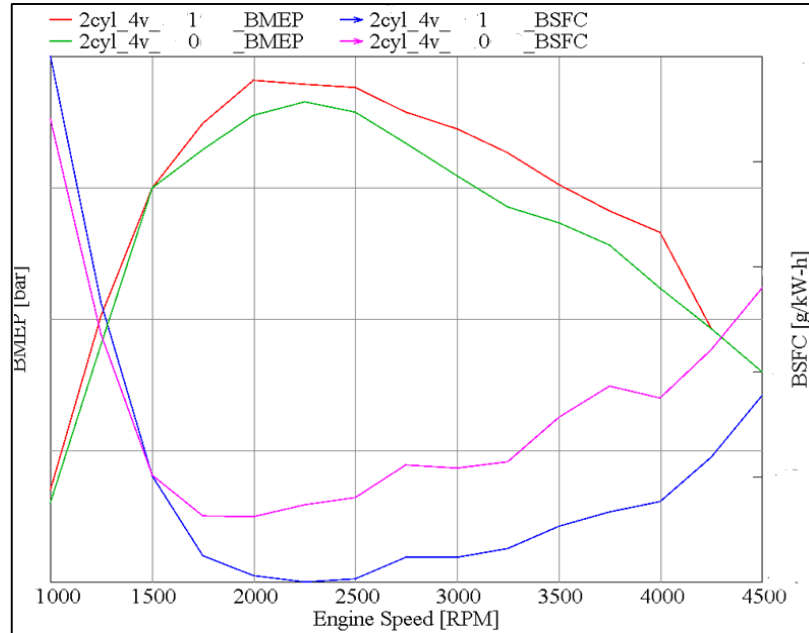Fig. 74: brake torque and power curves for new engine concept equipped with turbochargers "1" and "0"

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Fig. 75: BMEP and BSFC curves for new engine concept equipped with turbochargers "1" and "0"

II-VI Crankshaft design & Peak Firing Pressure

Crankshaft is one of most important structural component of Internal Combustion engine and transforms reciprocating motion of piston assembly to rotating motion through connecting rod. Crankshaft is formed by several parts as main journals, pin journals, webs, counterweights, nose and rear flange to joint respectively engine accessories pulley and flywheel. Main journals are supported bearings and together with bore and inter-bore distance define engine block length and so total engine length. On the other hand, longer and bigger crankshaft increases engine weight and consequently increases main journal supports dimensions. In details, starting to pin journal dimensions which are strongly related to actions exchange with big end con rod. Pin journal diameter is structural key parameter of crankshaft and is affected by maximum inertia force produced through piston assembly and maximum pressure on combustion chamber. About inertia force, two

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most important parameters are piston velocity related to engine type and piston assembly mass related to maximum combustion pressure. So, who mainly guides pin journal diameter are reciprocating acceleration and piston assembly mass. About pin journal length, is driven by oil film pressure at big end bearing generated from actions transmitted by con rod and in particular big end con rod structure. Again, who leads bearing length are maximum combustion pressure and inertia force through big end structure resistance and oil film pressure of big end bearing. Once defines pin journal dimensions, they strongly affect balance behavior of crankshaft and in particular contribute to centrifugal force exchanges with main journal supports. Consequently, crankshaft webs have to design accordingly with pin journals design and finally main journals are defined. To sum up maximum combustion pressure or Peak Firing Pressure leads crankshaft design and basically defines general stress level for components faced on combustion chamber. At moment, focus is about PFP which is driven through engine performance. Nowadays, light motive of automotive market is sustainable mobility that means small cars with small engine mainly oriented to pollutant emission reduction as CO_2 or NO_x rather than high performance. Moreover, lightweight design of vehicle and engine comes directly towards pollutant emission reduction. In this scenario, where dimension and weight are more important than performance, key role is played by PFP because its reduction could decrease, first of all, stress level on whole for components faced on combustion chamber, and further it could decrease components dimensions related to power transmission as crankshaft. In this way, lower PFP decreases pin journal dimensions and in cascade all crankshaft dimensions and so engine block dimensions.

For instance, crankshaft of new engine concept, 3 cylinders 4 valves are analyzed from strength point of view. Crankshaft is divided in parts starting from nose, so the worst part in the front is web 2 under quasi-static and torsional vibration loads at full load as shown in Figure 76.

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Fig. 76: fillet and oil hole stress for new engine concept three cylinders four valves;
every dashed vertical line is 500 rpm engine
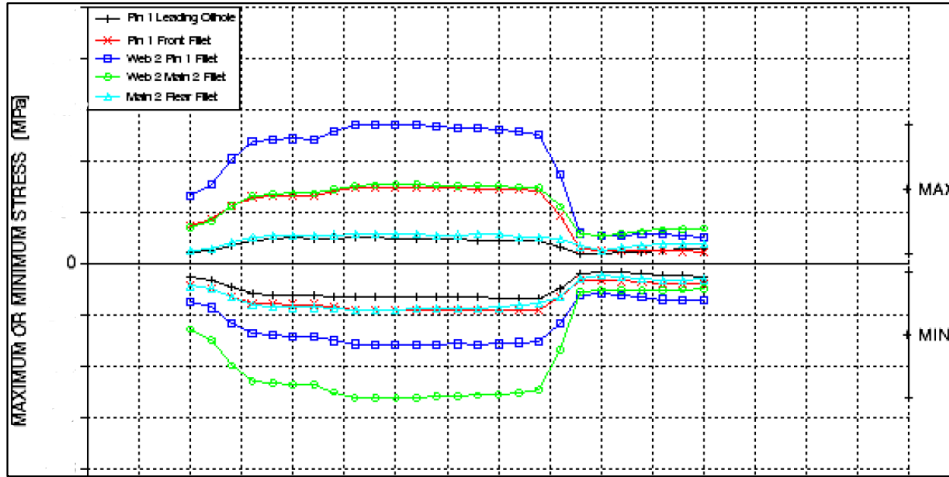
Safety factors are calculated using Goodman criterion and quasi-static loads are calculated using statically determinate method as shown in Figure 77.
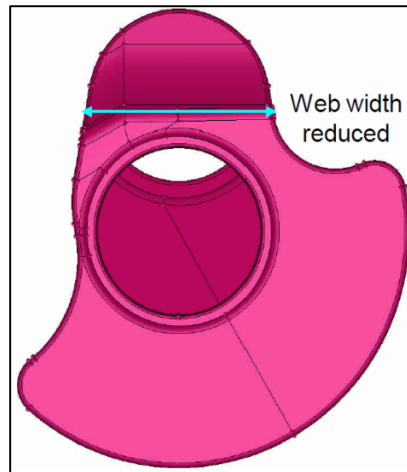
Fig. 77: safety factors for fillet and oil hole stress of new engine concept three cylinders four valves; use legend of previous chart
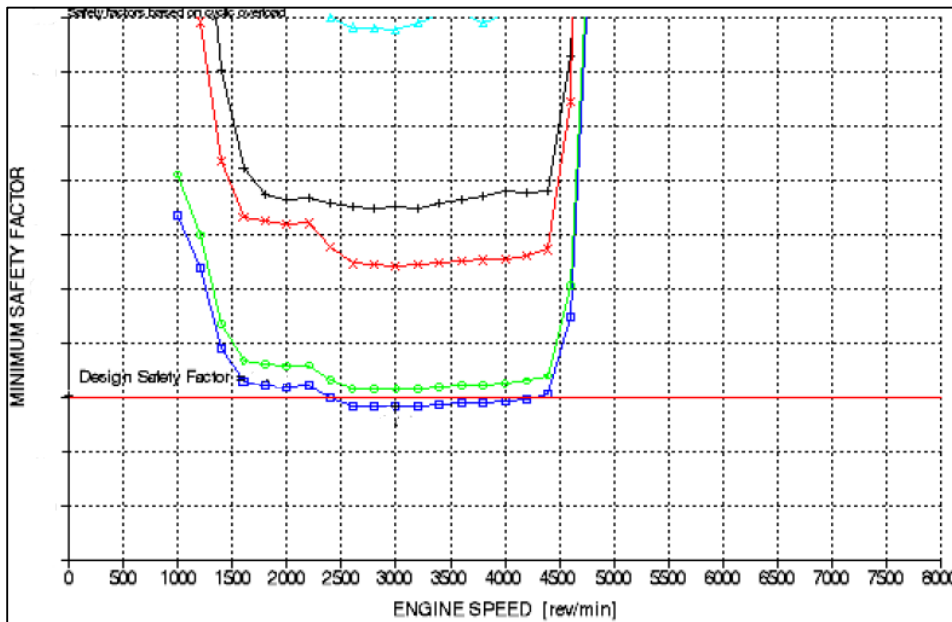
Considering Figures 76 and 77, crankshaft does not withstand engine operation loads. There are a number of reasons for the reduction in cover factor from previous crankshaft

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design. The most important ones are owing to torque slightly higher and section modulus slightly smaller: inertia increased with larger counterweights and stiffness decreased due to smaller crankshaft web section in overlap region as shown in Figure 78.

Fig. 78: web 2 of crankshaft for
new engine concept

First solution to increase safety factor and lightweight crankshaft is to decrease PFP of 5%. Figure 79 shows result of this proposal.

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Fig. 79: fillet and oil hole stress, and safety factors of crankshaft for
new engine concept three cylinders four valves with 95 % PFP
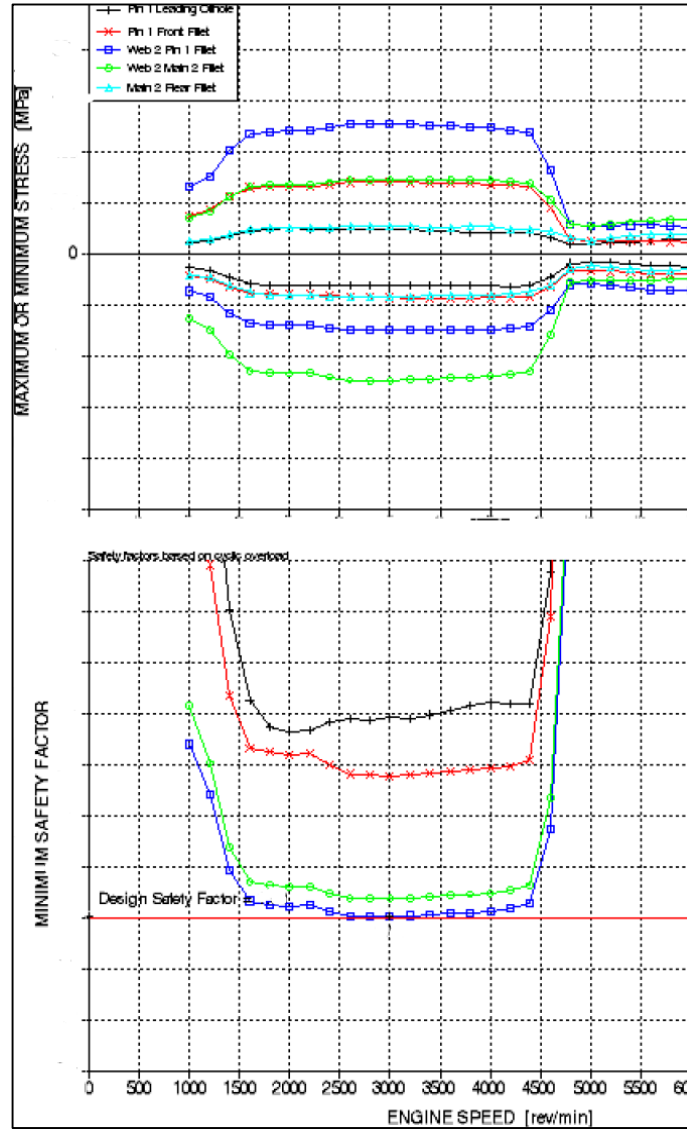
Reduction of PFP about 5 % increases safety factor in weak point of 6 % and safety factor of web 2 pin 1 fillet is slightly over guide line. Whit this result PFP is decreased of about 10 % starting from reference. Figure 80 shows results of last proposal.

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Fig. 80: fillet and oil hole stress, and safety factors of crankshaft for new engine concept three cylinders four valves with 90 % PFP
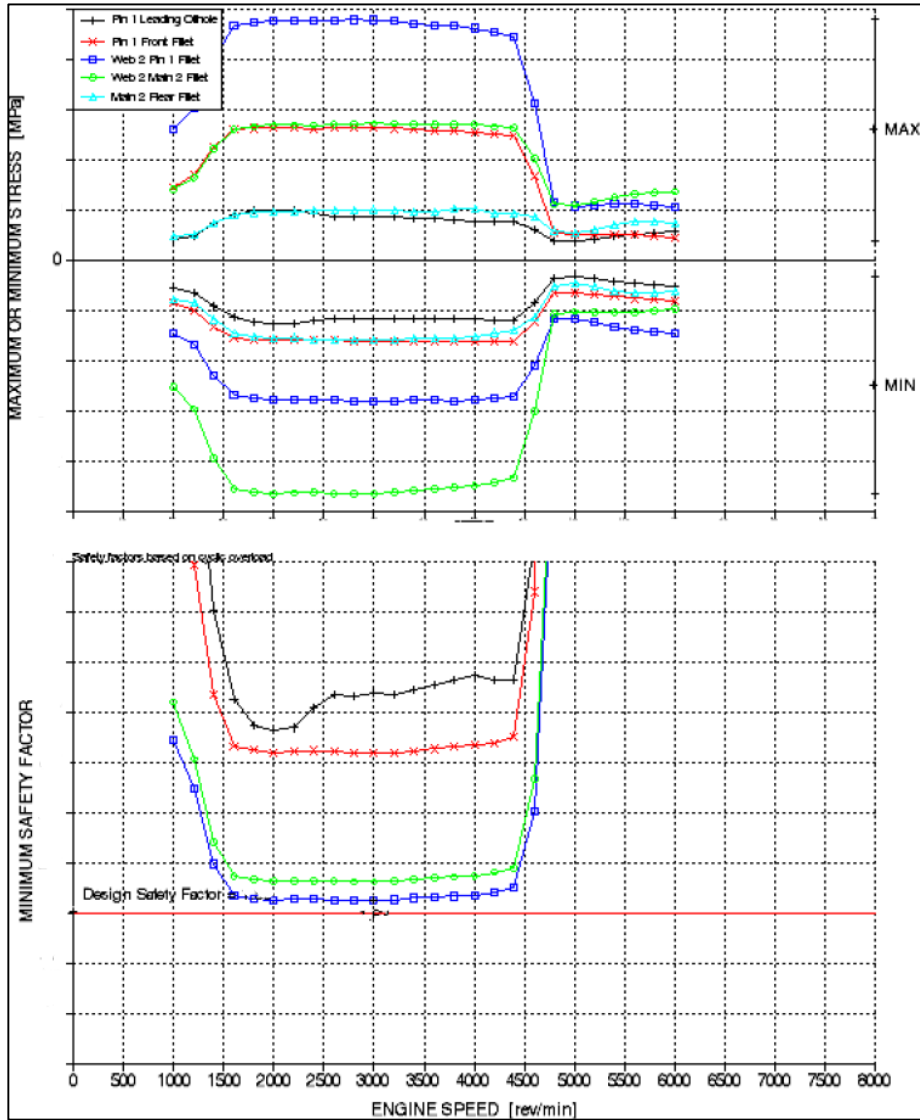
With PFP reduction of about 10 %, safety factor of weak crankshaft point and in particular web 2 pin 1 fillet increases of 15 %. Now, when cover factor is about 9 % over guide line, it could evaluate engine performance using reduced PFP of about 10 %. If engine performance will be good enough, designer could decide to reduce crankshaft dimension and so weight and keeps cover factor over 5 % guide line. Same speech is possible for new engine concept two cylinders four valves, and for this reason, first 1-D

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thermo-dynamic model uses reduced 10 % PFP is this one. Finally, next charts will compare engine performance for new engine concept two cylinders four valves using full PFP and 90 % PFP.

The PFP is one of the most important key structural parameter because it determines the stress level of the components around the combustion chamber. At now full PFP is used in basic stage of the model to assure the target performance. On the other hand, this high level of PFP produces a high level of stress that means big dimensions components. The purpose of PFP analyze is to achieve the engine performance target with 90 % PFP, so the dimension of engine components may be reduced. Apart of the difference of PFP value, there is another difference on engine back pressure. In the 90 % PFP model the back pressure is set to 1290 mbar in comparison with 1220 of full PFP engine model. Higher engine back pressure takes account of more realistic exhaust system effect on engine. Figures 81 and 82 shows brake torque and brake power for new engine concept two cylinders four valves.

Fig. 81: brake torque comparison between new engine concept full PFP – red- , 90 % PFP – blue - and reference

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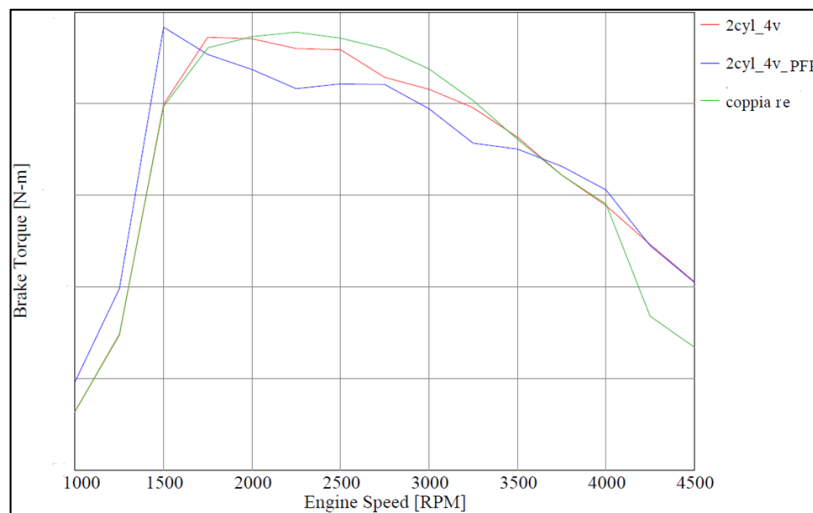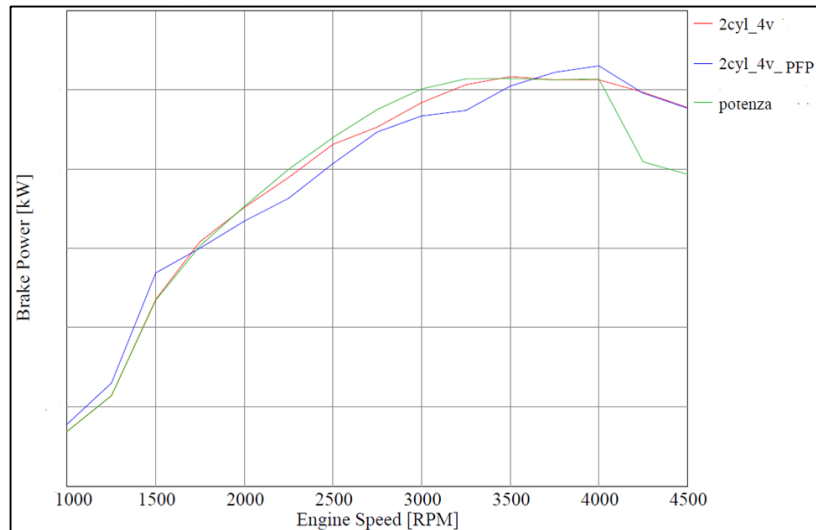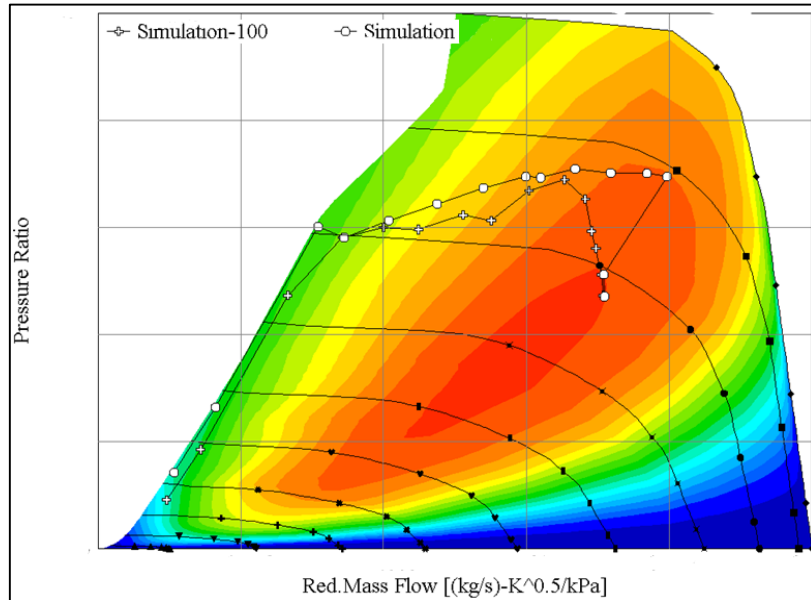Antonio Turturro – Supervisors: Prof. C. Delprete and L. Magro

Fig. 82: brake power comparison between new engine concept full PFP –
red -, 90 % PFP – blue - and its reference

The reduced PFP engine model does not achieve the brake torque target but it is enough for the concept. From 1000 – 1500 and 3750 – 4000 rpm the 90 % PFP engine is more performed and vice-versa from 1750 – 3500 rpm. There is the possibility to increase the trend of 90 % PFP model engine reducing his engine back pressure. Increase of brake torque and power are owing to different compressor operational point as shown in Figure 83.

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Fig. 83: compressor characteristic comparison between new engine
concept full PFP and 90 % PFP

The 90 % PFP engine model compressor map shows that it works with a higher pressure ratio, slightly higher mass flow rate and for these reasons reduced PFP model engine has slightly higher performance.

Figures 84 – 86 depict respectively turbine rack position, boost level and air consumption of new engine concept PFP comparison.

Fig. 84: VGT's rack position comparison between new engine concept
full PFP – red - and 90 % PFP – blue -

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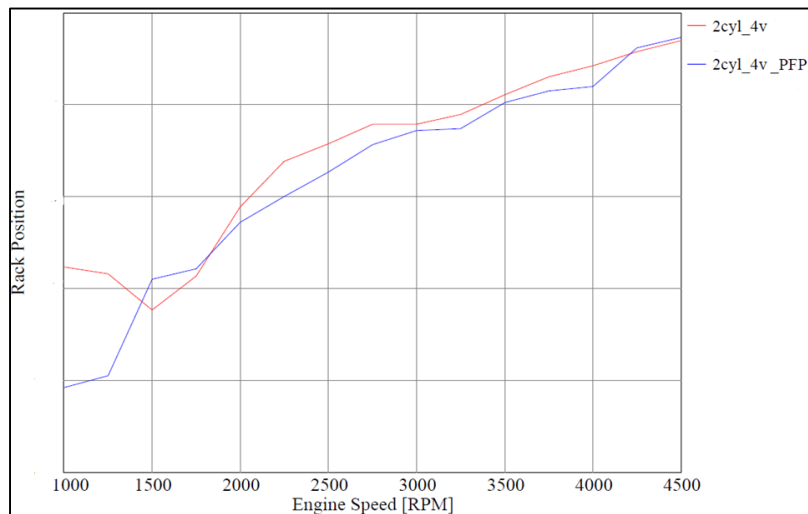
Fig. 85: boost level comparison between new engine concept full PFP –
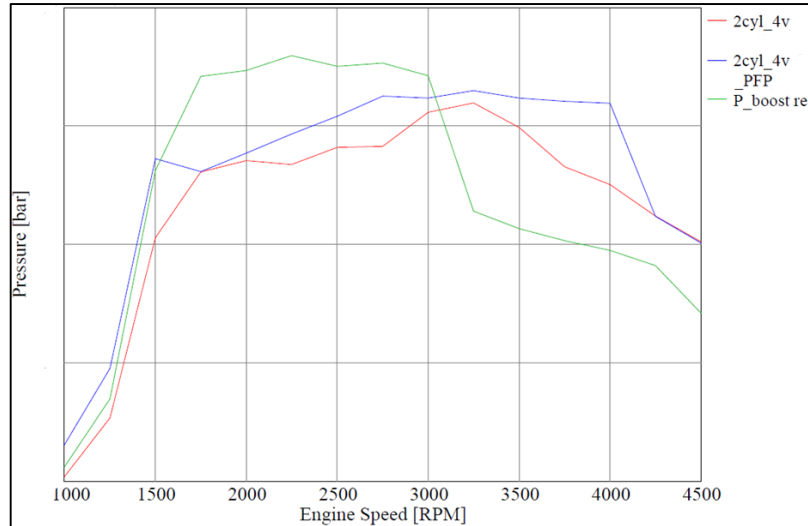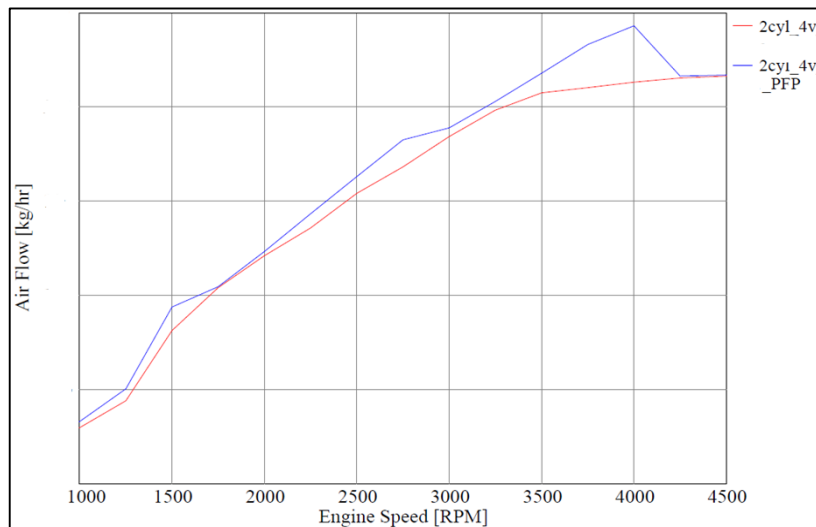red -, 90 % PFP – blue - and reference

Fig. 86: air consumption comparison between new engine concept full
PFP – red - and 90 % PFP – blue -

Rack position for reduced PFP engine model, as shown in Figure 84, is much lower up to
1500 rpm and is generally lower on all engine speed regimes that means turbine transmits
more energy towards compressor and to intake air. As shown in Figure 85 boost level of
reduced PFP new engine concept model works with higher boost level on whole that
close gap for reduced PFP and increase low end engine performance. Finally, as shown in

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Figure 86, higher compressor pressure ratio, high boost level leads to higher air consumption which means more fuel with same AF ratio.

Unfortunately, BSFC trend of new engine concept with reduced PFP increase in comparison to full PFP one, as shown in Figure 87.

Fig. 87: BSFC curves comparison between new engine concept full PFP – red -, 90 % PFP – blue -, GM 2l E5, and reference
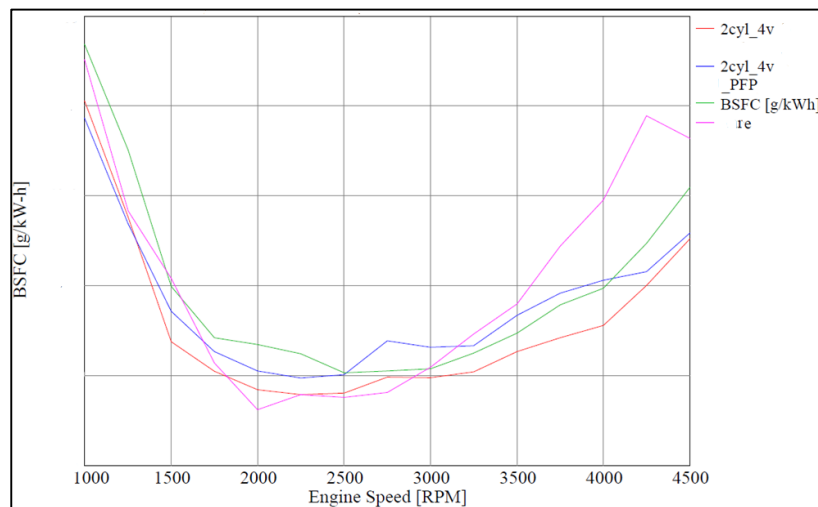
Main reasons for this behavior are lower PFP values that means lower combustion efficiency and increase of engine performance which needs more fuel to achieve it.

Next charts show in details new engine concept indicative index as IMEP360, IMEP 720, FMEP and BMEP, respectively in Figures 88 - 91.

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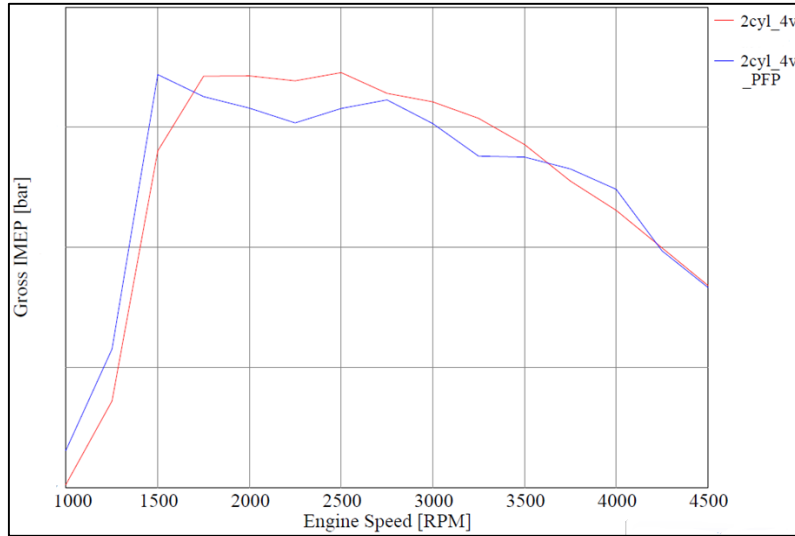Antonio Turturro – Supervisors: Prof. C. Delprete and L. Magro

Fig. 88: IMEP360 curves comparison between new engine concept full
PFP – red - and 90 % PFP – blue -

Fig. 89: IMEP360 curves comparison between new engine concept full
PFP - red - and 90 % PFP – blue -

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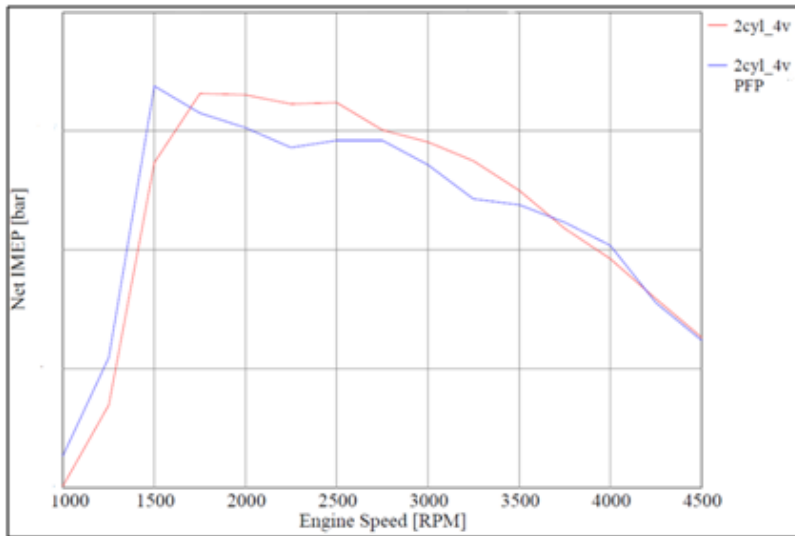Antonio Turturro – Supervisors: Prof. C. Delprete and L. Magro

Fig. 90: FMEP curves comparison between new engine concept full PFP –
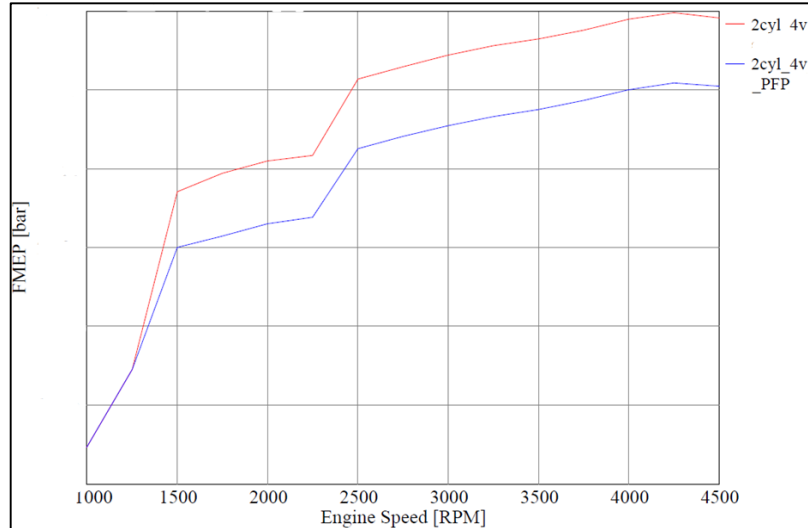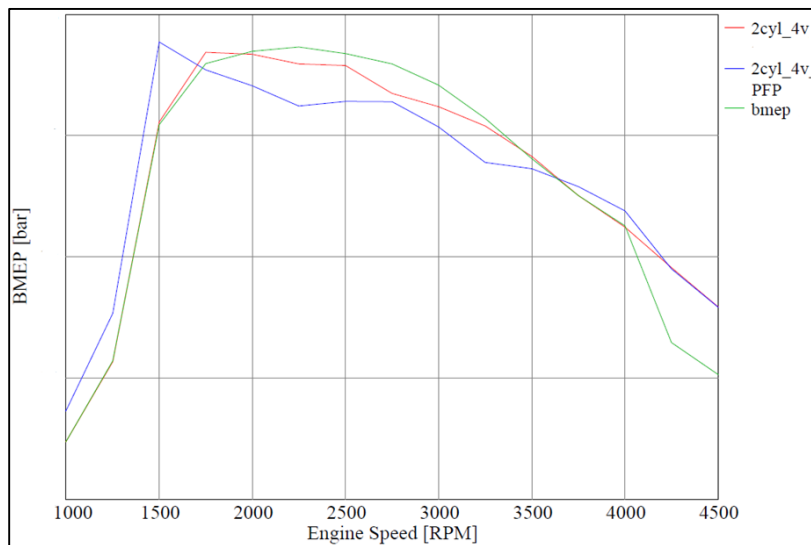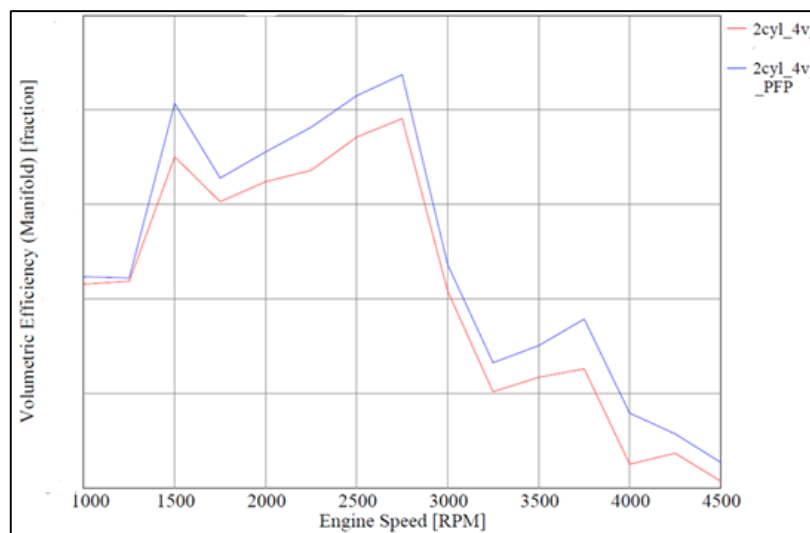red - and 90 % PFP – blue -

Fig. 91: BMEP curves comparison between new engine concept full PFP
– red -, 90 % PFP – blue - and reference

Indicated Mean Effective Pressure, gross and net, are shown in Figures 88 and 89 respectively and highlight how new engine concept with reduced PFP and increase turbocharger performance generates more power up to 1500 rpm then full one. After, IMEPs decrease and they bake up full PFP engine at 3500 rpm. IMEPs trend do not affect

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in drastic way engine performance because PFP reduction decreases Friction Mean Effective Pressure in the same time, as shown in Figure 90. So difference between IMEP and FMEP as described in (8) produces good trend of Brake Mean Effective Pressure, see Figure 90, which is strongly related to brake torque profile as shown in Figure 91.

Figure 92 depicts effect of higher pressure ratio and higher boost level for new engine concept with reduced PFP on volumetric efficiency.

Fig. 92: volumetric efficiency comparison between new engine concept
full PFP – red - and 90 % PFP – blue -

Next Figures 93 – 96 show main compressor and turbine parameters as outlet compressor temperature, inlet turbine pressure, turbine speed, and engine back pressure that means pressure due mainly to exhaust system.

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Fig. 93: outlet compressor temperature comparison between new engine concept full PFP – red -, 90 % PFP – blue - and reference

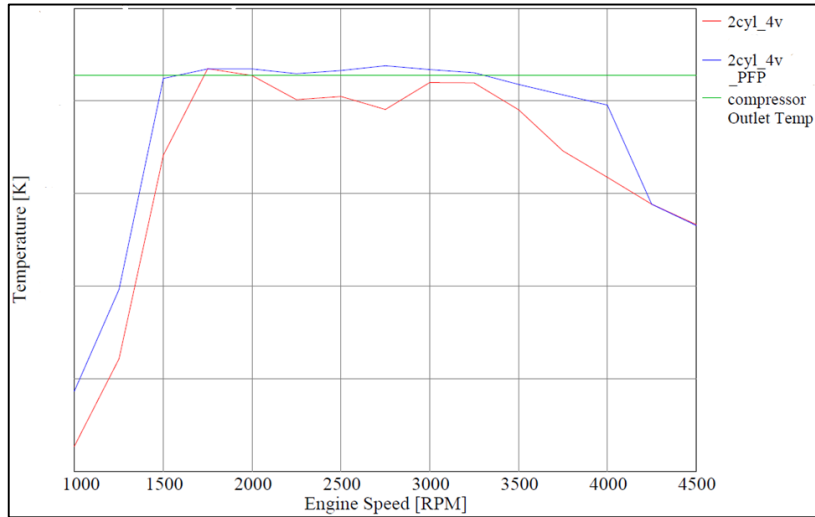Fig. 94: inlet turbine pressure comparison between new engine concept full PFP – red -, 90 % PFP – blue - and reference

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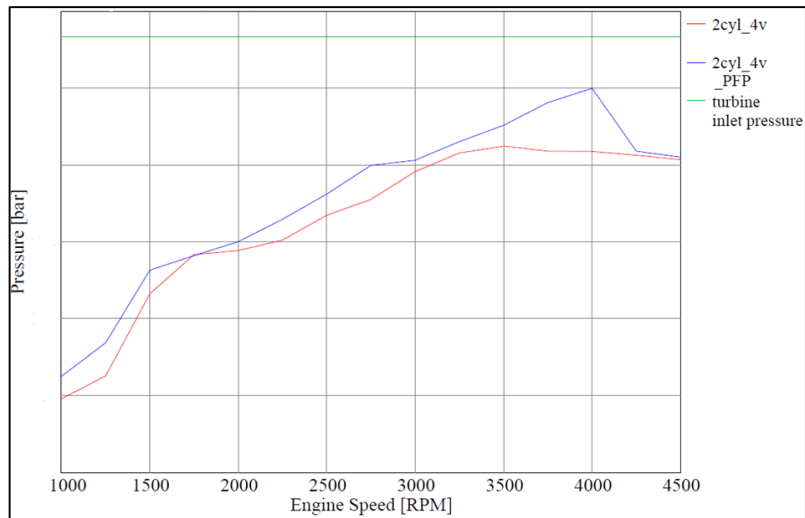Antonio Turturro – Supervisors: Prof. C. Delprete and L. Magro

Fig. 95: engine back pressure comparison between new engine concept
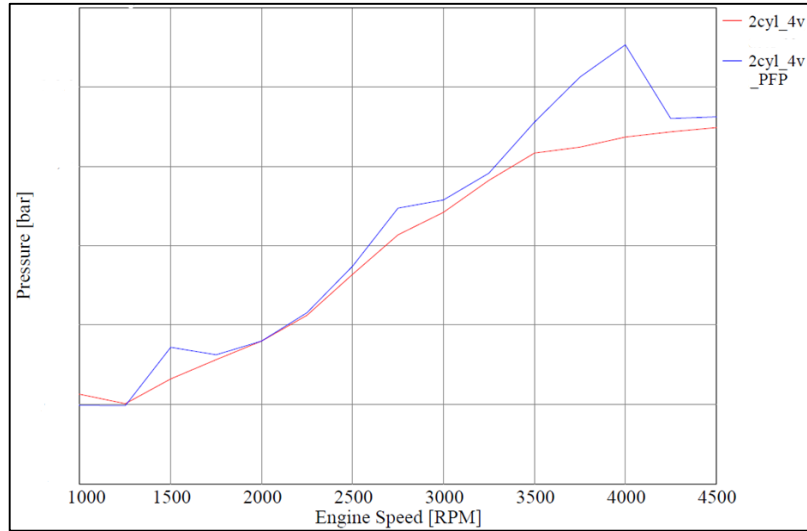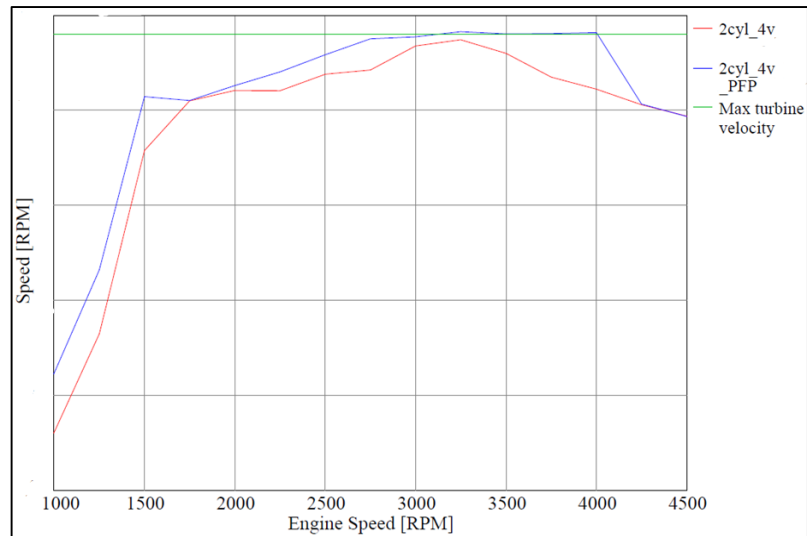full PFP – red - and 90 % PFP – blue -

Fig. 96: turbocharger speed comparison between new engine concept full
PFP – red -, 90 % PFP – blue - and limit

As shown in Figure 93, outlet compressor temperature for new engine concept two cylinders 4 valves with 90 % PFP is always higher than full PFP one. This compressor behavior is not owing to PFP reduction but to higher pressure ratio, see Figure 83. In particular, outlet compressor temperature of new engine concept with reduced PFP is same degrees over limit between 1750 – 3250 rpm but is still acceptable. Figure 94

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depicts inlet turbine pressure comparison between new engine concept with full PFP and reduced PFP. The reduced one trend is slightly higher on whole but there is peak due to engine back pressure around 4000 rpm to better simulate pressure load of exhaust system as shown in Figure 95. Figure 96 shows turbocharger velocity for new engine concept with full PFP and reduce PFP. In particular, reduced PFP engine model works with higher turbocharger speed due to high performance demand, in fact, it works at maximum speed for structural integrity between 2750 – 4000 rpm.

The model using a reduce PFP does not achieve the same engine performance of full PFP value but it obtains enough performance for the application. The reduced PFP engine model have several advantages:

- torque peak at lower engine speed;
- satisfaction performance curve with a higher engine back pressure value;
- to use a lower PFP leads to a smaller components dimension.

To sum up, a lower value of Peak Firing Pressure is possible to use with a satisfied engine performance and a reduced components dimension. The use of a reduced PFP value leads many advantages because it is possible achieve a satisfied engine performance with a strong reduction of the stress level of the engine.

II-VII Cam profile & engine performance

New engine concept needs to be complain with weight, durability, reliability and emission regulation as Euro 7 and over. For these reasons new cylinder head concept is been developed. So, new cylinder head architecture needs new valvetrain concept layout. First of all, valvetrain kinematic and dynamic are been investigated and then to test new valvetrain layout on new engine concept, 1-D thermodynamic model is been built up containing cam profile of new valvetrain layout. This section present results of new engine concept two cylinders four valves equipped with new cam profile to evaluate if it is able to achieve target performance. The outputs are compared with new engine concept at STEP 3.

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Figures 97 – 98 show brake torque and brake power of new engine concept two cylinders four valves equipped with new cam profile.

Fig. 97: brake torque comparison between new engine concept at STEP 3, STEP 3 equipped new cam profile and reference
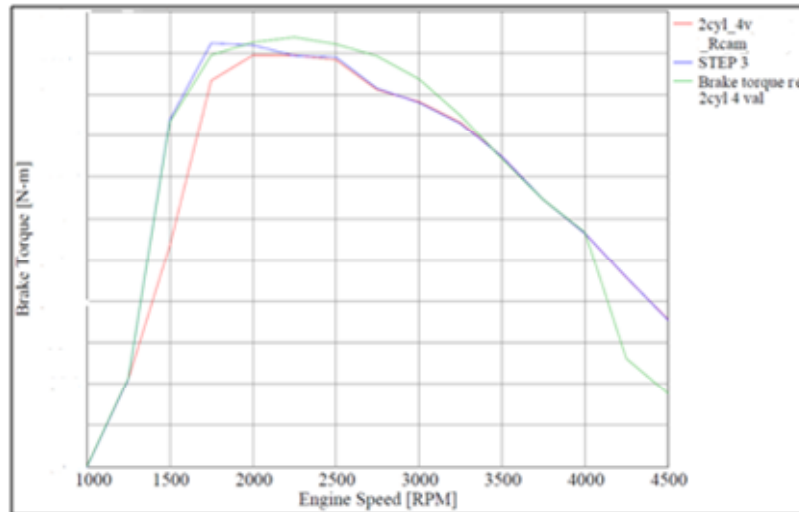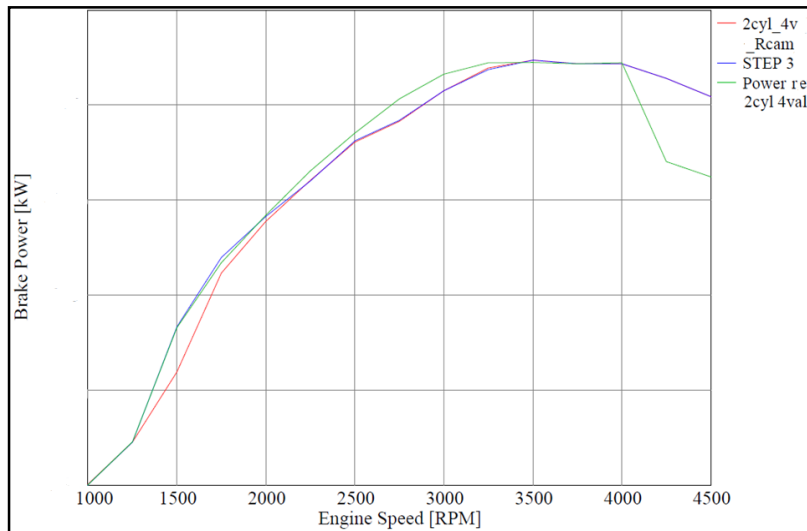

Fig. 98: brake power comparison between new engine concept at STEP 3, STEP 3 equipped new cam profile and reference

The brake torque with the new cam has a loose of performance between 1750 – 2250 rpm in comparison with the old one. The new cam profile has the follow disadvantages a

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loose of performance at 1500 – 1750 rpm; and a lower volumetric efficiency. To sum up, the new cam profile does not achieve the whole engine performance target.

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PART III

MULTI-BODY DYNAMICS OF VALVETRAIN SYSTEM AND
FRICTION ASSESSMENT FOR CAM-ROLLER FOLLOWER
VALVETRAIN SYSTEM SUBJECTED TO NON-NEWTONIAN
REGIME OF LUBRICATION

III-I   Introduction   to   elasto-hydrodynamic   lubrication   for   cam tappet/roller contact

In the last 20 years, a challenge towards an extreme increase of the specific engine power has done. On the other hand, the engine layout and the emission regulation has became more and more stringent. So, severe engine component loading occurs because increase bulk temperature, contact pressures and inertia imbalances. These account for a sizeable proportion of heat and frictional losses in internal combustion (IC) engine as shown by Taylor [1,2].

In order to target the sources of energy losses in an IC engine, Anderson [3] analyzed the power distribution in a specific vehicle during an urban cycle – since the major of part of a automobile mission is done during the city drive -. Anderson's conclusion showed that out the total energy introduced in the vehicle system, about the 70 % of the energy was lost by the cooling system to remove the heat from the engine surfaces. Out of remaining 30 %, approximately 15 % of the energy introduced in the system was used to win the mechanical losses, or in other words, to win the mechanical friction by all the articulating